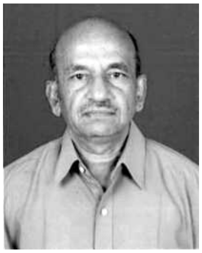
MACHINE DESIGN DATA BOOK

About the Author



V. B. Bhandari retired as Professor and Head, Department of Mechanical Engineering at Vishwakarma Institute of Technology, Pune. He has obtained his B.E. and M.E. degrees in Mechanical Engineering from Pune University in First Class. He has received five prizes of Pune University during his academic career besides being National Scholar. His teaching experience spans over 38 years in Government College of Engineering at Pune, Karad and Aurangabad. He was Post-Graduate Teacher of Pune University, Shivaji University and Marathwada University. He was a Member of 'Board of Studies in Mechanical Engineering' and Member of 'Faculty of Engineering' of Pune University. He is a Fellow of Institution of Engineers (India) and Fellow of Institution of Mechanical Engineers (India) as well as Senior Member of Computer Society of India. He was a Fellow of Institution of Production Engineers (India) and a Member of American Society of Mechanical Engineers (U.S.A.). He has published twenty technical papers in National and International Conferences and Journals. He is also the author of the textbook *Design of Machine Elements* and *Introduction to Machine Design* published by McGraw Hill Education (India) Private Limited.

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Contents

Preface

Machine Design occupies an important place in the syllabus of Mechanical Engineering course. It is an interdisciplinary subject and draws its subject matter from different subjects like Thermodynamics, Heat transfer, Fluid mechanics, Theory of Machines, Statics and Dynamics, Theory of elasticity, Metallurgy, Production engineering and Mathematics. Machine design acquires and assimilates more information from these subjects than any other engineering subject.

This book is intended to serve as a databook for two to three subjects in Machine Design of BE (Mechanical) course. It covers the syllabi of all universities, technical boards and professional examining bodies such as Institute of Engineers in the country. The students of this subject have to complete Design Projects as part of their termwork. For design projects, extensive data is required from various books, national and international standards, catalogues of various companies and technical papers from professional journals. This data is not readily available and students have very limited scope to collect this information.

The objective of *Machine Design Databook* is to provide the aforementioned data in one single book. It provides the latest topics like statistical considerations, international standards on load capacities of spur, helical and bevel gears, exponential relationships for stress concentration factors, mechanical property classes for threaded fasteners and abutment and fillet dimensions for rolling contact bearings.

Design of machine elements usually involves use of a number of formulae and equations. These equations are tabulated in topic-wise manner in this book. This will eliminate the task of remembering the formulae and equations and help students to solve the design problems in lesser time.

SALIENT FEATURES

- Data provided in an exhaustive and structured manner on almost all machine elements
- Well-segregated chapters for ease of use
- SI units are used throughout
- Updated information on Bearings, Gears and Clutches
- Similarity in flow of content between design data book and the textbook of the author
- ISO and DIN standards are used throughout the book for tolerances, threads and threaded fasteners, springs, roller chains, bearings and gears
- The data required for selection of machine components from manufacturer's catalogue is provided with a particular reference to Indian product like SKF bearings, Goodyear belt and Diamond chain
- International procedure is adapted for calculating the load capacities of spur, helical and bevel gears
- Complete set of Raimondi and Boyd charts and Tables (40 charts and 5 tables) are included for the design of hydrodynamic bearings
- Sectional views are provided for different types of gearboxes showing constructional details of housing and its attachments

ORGANIZATION OF THE BOOK

The book is divided into 29 chapters.

Chapter 1 contains basic tables required by design students that include SI units, conversion factors, and mathematical formulae from algebra, trigonometry, analytical geometry, differentiation and integration. **Chapter 2** provides mechanical properties of materials for machine elements such as tensile strength, yield stress and hardness. It also includes classification and designation of materials like cast iron, cast steel, carbon and alloy steels, aluminium alloys, bronze and plastics.

Chapter 3 deals with manufacturing aspect of design namely casting design and tolerances. Limit deviations for shafts and holes as per ISO recommendations are tabulated and the concept of geometrical tolerances is illustrated in this chapter. **Chapter 4** explains formulae for design against static load. It covers the topics of curved beams, levers, torsion of non-circular bars, buckling of column, theories of failure and factor of safety.

Chapter 5 explains formulae for design against fluctuating load and contact stresses. This chapter contains charts for stress concentrations factors and notch sensitivity and estimation of endurance limit of machine components. **Chapter 6** deals with power screws and provides data for square, trapezoidal and saw tooth threads and their torque capacities. **Chapter 7** provides design data for ISO metric screw threads. It also covers dimensions for standard hexagon and square head bolts, screws and studs, nuts, washers, split pins and their mechanical property classes.

Chapter 8 deals with two types of joints namely welded and riveted joints. The data for welded joints include welding symbols, line properties of welds and strength equations for different types of welded joints. Riveted joints contain dimensions of various types of rivets and design of rivets for boiler and structural applications. **Chapter 9** explains basic elements of transmission system namely shafts, keys and couplings. Design equations for strength and rigidity of transmission shafts, dimensions of various types of keys, couplings, splines and serrations, lateral rigidity of shafts, critical speeds and flexible shafts are included in this chapter. **Chapter 10** covers springs namely, properties of spring wires and design equations for helical, torsion, spiral, and leaf springs.

Chapters 11 and **12** contain equations for theoretical design of different types of clutches and brakes respectively. **Chapter 13** covers selection of flat and V-belts from manufacturer's catalogues and dimensions of respective pulleys for the belt drive. Ribbed V-belts are also covered in this chapter. **Chapter 14** deals with selection of roller chains and sprocket wheels for the chain drive.

Chapter 15 covers selection of different types of rolling contact bearings. The selection factors of these bearings, their dimensions, abutment and fillet dimensions and tolerances are tabulated in this chapter. It also covers locknuts, lock washers, oilseal units, circlips, set collars and plumber blocks for bearing mountings. **Chapter 16** provides Raimondi and Boyd charts and tables for hydrodynamic journal bearings. Charts and design equations for hydrostatic bearings are also provided in this chapter. **Chapters 17**, **18** and **19** cover Lewis and Buckingham's equations for academic design of spur, helical and bevel gears respectively. **Chapter 20** contains international method for calculation of load capacities of spur and helical gears.

Chapter 21 explains design procedure for gears using present data books, which are often used by students during examination. Design twisting moment, design bending stress, design contact stress and design check for plastic deformation or brittle fracture are the important topics of this chapter. **Chapter 22** covers the international method of calculating load capacity of bevel gears. **Chapter 23** contains design of worm gears based on strength rating, wear rating and thermal rating.

Chapter 24 is based on assembly drawings of different types of gearboxes showing constructional details of housing and its attachment. It also explains ray diagrams and kinematic layout of multi-speed gearboxes. **Chapter 25** provides design equations for solid disk and rimmed flywheels. It also deals with cams and followers namely, different types of follower motions, specific counters for cams and polynomial cams.

Chapter 26 covers design of unfired pressure vessels including gasketed joints in cylinder covers and thick and thin cylinders. **Chapter 27** contains materials handling equipment that include wire ropes, rope sheaves, welded lifting chains, hooks, slings, arresting gears and Geneva mechanism.

Chapter 28 provides data for statistical consideration in design and includes curve fitting, different types of distributions and probabilistic approach to margin of safety. **Chapter 29** deals with design of engine components namely, cylinder, piston, connecting rod, crankshaft and valve gear mechanism.

ACKNOWLEDGEMENT

For this databook, information has been collected from various sources such as textbooks, references books, manufacturer's catalogue and articles from technical journals and proceedings. I would like to express my gratitude to authors and editors of these books, publishers and manufacturers of machine components for this information.

My special thanks to Judy Enblom from the Society of Tribologists and Lubrication Engineers for permission to reproduce Raimondi and Boyd charts and Tables from ASLE Transactions.

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V B Bhandari

FEEDBACK

Any difficulties related to this data book, suggestion and comments for improvement will be appreciated. Readers can write to me at *bhandariprof@gmail.com*

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Publisher's Note

We look forward to receiving valuable views, comments and suggestions for improvements from teachers and students, all of which can be sent to *tmh.mechfeedback@gmail.com*, mentioning the title and author's name on the subject line.

Piracy related issues can also be reported.

1 Basic Tables

1.1 CONVERSION FACTORS

Table 1.1 *Greek Alphabets*

Greek name	Greek letter	
	Lower case	Capital
Alpha	α	A
Beta	β	B
Gamma	γ	Γ
Delta	δ	Δ
Epsilon	ε	E
Zeta	ζ	Z
Eta	η	H
Theta	θ	Θ
Iota	ι	I
Kappa	κ	K
Lambda	λ	Λ
Mu	μ	M
Nu	ν	N
Xi	ξ	Ξ
Omicron	o	O
Pi	π	Π
Rho	ρ	P

(*Contd.*)

(*Contd.*)

Sigma	σ	Σ
Tau	τ	T
Upsilon	υ	Y
Phi	ϕ	Φ
Chi	χ	X
Psi	ψ	Ψ
Omega	ω	Ω

Table 1.2 *Standard prefixes in SI units*

Name of Prefix	Symbol	Factor
tera	T	10^{12}
giga	G	10^{9}
mega	M	10^{6}
kilo	k	10^{3}
hecto*	h	10^{2}
deka*	da	10^{1}
deci*	d	10^{-1}
centi*	c	10^{-2}
milli	m	10^{-3}
micro	μ	10^{-6}
nano	n	10^{-9}
pico	p	10^{-12}
femto	f	10^{-15}
atto	a	10^{-18}

Note: *These prefixes are not recommended but sometimes used in practice

Table 1.3 *Conversion factors*

Quantity	Conversion Factor
Length	1 foot (ft) = 0.3048 metre (m)
	1 metre (m) = 3.28 foot (ft)
	1 inch (in) = 25.4 millimetre (mm)
	1 metre (m) = 39.37 inches (in)
	1 mile (mi) = 1.609 kilometre (km)
	1 kilometre (km) = 0.6214 mile (mi)
	1 micron (μ) = 10^{-6} metre (m)
	1 micron (μ) = 0.001 millimetre (mm)
	1 angstrom (A) = 10^{-10} metre (m)
Area	1 square metre (m^2) = 10.76 square feet (ft^2)
	1 square foot (ft^2) = 0.0929 square metre (m^2)
	1 square mile (mi^2) = 640 acres
	1 acre = 43 560 square feet (ft^2)
Volume	1 litre (l) = 1000 cubic centimetres (cm^3)
	1 litre (l) = 61.02 cubic inches (in^3)
	1 litre (l) = 0.03532 cubic feet (ft^3)
	1 cubic metre (m^3) = 1000 litre (l)
	1 cubic metre (m^3) = 35.32 cubic feet (ft^3)
	1 cubic foot (ft^3) = 28.32 litres (l)
	1 cubic foot (ft^3) = 0.2832 cubic metre (m^3)
	1 US gallon (gal) = 3.785 litres (l)
	1 British gallon = 4.546 litres (l)
Mass	1 kilogram (kg) = 2.2046 pounds (lb)
	1 kilogram (kg) = 0.06852 slug
	1 pound (lb) = 0.4536 kilogram (kg)
	1 slug = 32.174 pound (lb)
	1 slug = 14.59 kilogram (kg)
Speed	1 kilometre per hour (km/hr) = 0.2778 metre per second (m/s)
	1 kilometre per hour (km/hr) = 0.6214 mile per hour (mi/hr)
	1 kilometre per hour (km/hr) = 0.9113 foot per second (ft/s)
	1 mile per hour (mi/hr) = 1.609 kilometre per hour (km/hr)
	1 mile per hour (mi/hr) = 0.4470 metre per second (m/s)

(*Contd.*)

(*Contd.*)

Density	1 gram per cubic centimetre (gm/cm^3) = 10^3 kilogram per cubic metre (kg/m^3)
	1 gram per cubic centimetre (gm/cm^3) = 62.43 pounds per cubic feet (lb/ft^3)
	1 pound per cubic foot (lb/ft^3) = 0.01602 gram per cubic centimetres (gm/cm^3)
Force	1 Newton (N) = 10^5 dynes
	1 Newton (N) = 0.1020 kilogram force (kgf)
	1 Newton (N) = 0.2248 pound force (lbf)
	1 pound force (lbf) = 4.448 Newton (N)
	1 pound force (lbf) = 0.4536 kilogram force (kgf)
	1 kilogram force (kgf) = 9.807 Newton (N)
	1 kilogram force (kgf) = 2.205 pound force (lbf)
	1 US short ton = 2000 pound force (lbf)
	1 long ton = 2240 pound force (lbf)
	1 metric ton = 1000 kilogram force (kgf)
	1 metric ton = 2205 pound force (lbf)
Energy	1 Joule (J) = 1 Newton metre (N-m)
	1 J = 1 N-m = 0.102 kgf-m
	1 Joule (J) = 10^7 ergs
	1 Joule (J) = 0.7376 foot pound force (ft-lbf)
	1 Joule (J) = 0.2389 calorie (cal)
	1 Joule (J) = 9.481×10^{-4} Btu
	1 foot pound force (ft-lbf) = 1.356 Joules (J)
	1 British thermal unit (Btu) = 1055 Joules (J)
	1 kilowatt hour (kW-hr) = 3.6×10^6 Joules (J)
Power	1 Watt (W) = 1 Joule per second (J/s)
	1 Watt (W) = 0.2389 calories per second (cal/s)
	1 horsepower (hp) = 0.7457 kilowatt (kW)
	1 kilowatt (kW) = 1.341 horsepower (hp)
Stress/Pressure	1 Newton per square metre (N/m^2) = 10 dynes per square centimetre (dyne/cm^2)
	1 Newton per square metre (N/m^2) = 2.089×10^{-2} pound force per square foot (lbf/ft^2)

(*Contd.*)

(*Contd.*)

	1 Pascal (Pa) = 1 N/m^2
	1 MPa = 1 MN/m^2 = 1 N/mm^2 = 0.102 kgf/mm^2
	1 Pound force per square inch (psi) = 6895 Newton per square metre (N/m^2)
	1 psi = 6895 Pascals (Pa)
	1 atmosphere (atm) = 14.7 pound force per square inch (psi)
	1 atmosphere (atm) = 1.013×10^5 N/m^2
	1 atmosphere (atm) = 76 cm of mercury
	1 atmosphere (atm) = 406.8 in of water
Angle	(2π) radians = 360 degrees (º)
	1 radian = 57.296º
	1º = 0.01745 radians
Other quantities	1 W/m ºC = 2.383×10^{-3} cal/cm ºC
	1 J/g ºC = 0.2388 cal/g ºC

Note: (g = 32.174 or 32.2 ft/s^2) (g = 9.806 or 9.8 m/s^2)

1.2 PREFERRED NUMBERS

Table 1.4 *Series factors for preferred numbers*

Series	Series factor
R5 Series	$\sqrt[5]{10} = 1.58$ (1.1)
R10 Series	$\sqrt[10]{10} = 1.26$ (1.2)
R20 Series	$\sqrt[20]{10} = 1.12$ (1.3)
R40 Series	$\sqrt[40]{10} = 1.06$ (1.4)
R80 Series	$\sqrt[80]{10} = 1.03$ (1.5)

(ISO: 3)

Note: The series is established by taking the first number and multiplying it by a series factor, to get the second number. The second number is again multiplied by the series factor to get the third number. This procedure is continued until the complete series is built up.

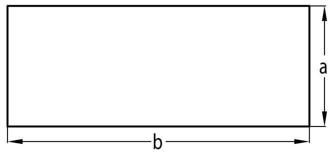
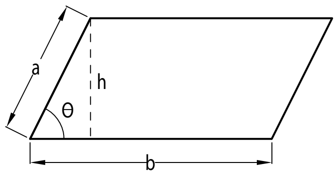
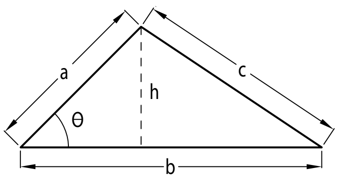
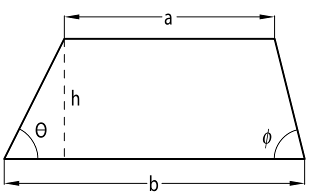
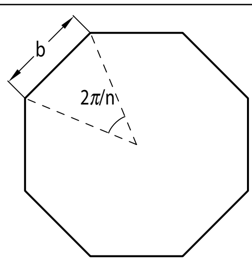
Table 1.5 *Preferred numbers*

R5	R10	R20	R40
1.00	1.00	1.00	1.00
			1.06
		1.12	1.12
			1.18
	1.25	1.25	1.25
			1.32
		1.40	1.40
			1.50
1.60	1.60	1.60	1.60
			1.70
		1.80	1.80
			1.90
	2.00	2.00	2.00
			2.12
		2.24	2.24
			2.36
2.50	2.50	2.50	2.50
			2.65
		2.80	2.80
			3.00
	3.15	3.15	3.15
			3.35
		3.55	3.55
			3.75
4.00	4.00	4.00	4.00
			4.25
		4.50	4.50
			4.75
	5.00	5.00	5.00
			5.30
		5.60	5.60
			6.00
6.30	6.30	6.30	6.30
			6.70
		7.10	7.10
			7.50
	8.00	8.00	8.00
			8.50
		9.00	9.00
			9.50
10.00	10.00	10.00	10.00

(ISO: 3)

1.3 FORMULAE—AREAS AND VOLUMES

Table 1.6 *Area and perimeter of geometric shapes*

Shape	**Area and Perimeter**
Rectangle of length b and width a	
	Area = ab (1.6) Perimeter = $2a + 2b$ (1.7)
Parallelogram of altitude h and base b	
	Area = $bh = ab \sin \theta$ (1.8) Perimeter = $2a + 2b$ (1.9)
Triangle of altitude h and base b	
	Area = $\frac{1}{2}bh = \frac{1}{2}ab \sin \theta$ $= \sqrt{s(s-a)(s-b)(s-c)}$ (1.10) where $s = \frac{1}{2}(a+b+c)$ (1.11) Perimeter = $a + b + c$ (1.12)
Trapezoid of altitude h and parallel sides a and b	
	Area = $\frac{1}{2}h(a+b)$ (1.13) Perimeter = $a + b + h\left(\frac{1}{\sin \theta} + \frac{1}{\sin \phi}\right)$ (1.14)
Regular polygon of n sides each of length b	
	Area = $\frac{1}{4}nb^2 \cot\left(\frac{\pi}{n}\right)$ $= \frac{1}{4}nb^2 \frac{\cos(\pi/n)}{\sin(\pi/n)}$ (1.15) Perimeter = nb (1.16)

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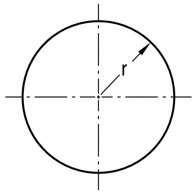
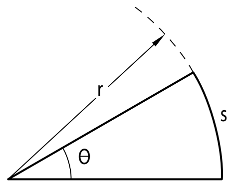
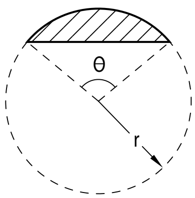
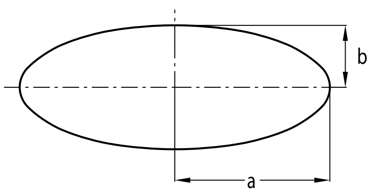
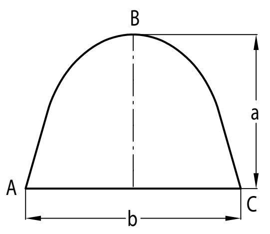
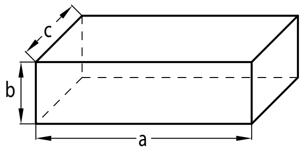
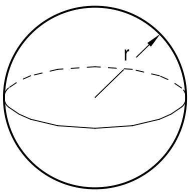
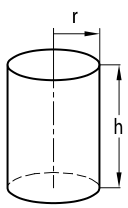
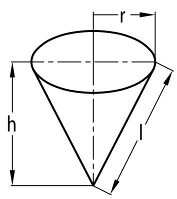
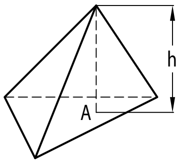
Circle of radius r	
	Area = πr^2 (1.17) Perimeter = $2\pi r$ (1.18)
Sector of circle of radius r	
	Area = $\frac{1}{2} r^2 \theta$ (θ in radians) (1.19) Arc length s = $r\theta$ (1.20)
Segment of circle of radius r	
	Area of shaded part = $\frac{1}{2} r^2 (\theta - \sin\theta)$ (θ in radians) (1.21)
Ellipse of semi-major axis a and semi-minor axis b	
	Area = πab (1.22) Perimeter = $2\pi \sqrt{\frac{1}{2}(a^2 + b^2)}$ (1.23) (approximately)
Segment of parabola	
	Area = $\frac{2}{3} ab$ (1.24) Arc length ABC = $= \frac{1}{2}\sqrt{b^2 + 16a^2} + \left(\frac{b^2}{8a}\right) \ln\left(\frac{4a + \sqrt{b^2 + 16a^2}}{b}\right)$ (1.25)

(*Contd.*)

Table 1.7 *Volume and surface area of solids*

Shape	**Volume and Surface area**
Rectangular box of length a, height b and width c	
	Volume = abc (1.26) Surface area = $2(ab + ac + bc)$ (1.27)
Sphere of radius r	
	Volume = $\frac{4}{3}\pi r^3$ (1.28) Surface area = $4\pi r^2$ (1.29)
Right circular cylinder of radius r and height h	
	Volume = $\pi r^2 h$ (1.30) Lateral surface area = $2\pi rh$ (1.31)
Right circular cone of radius r and height h	
	Volume = $\frac{1}{3}\pi r^2 h$ (1.32) Lateral surface area = $\pi r\sqrt{r^2 + h^2}$ $= \pi rl$ (1.33)
Pyramid of base area A and height h	
	Volume = $\frac{1}{3}Ah$ (1.34)

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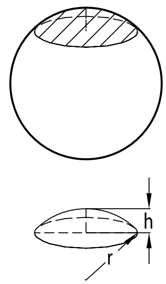
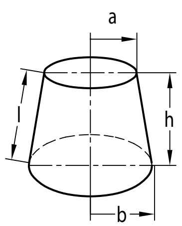
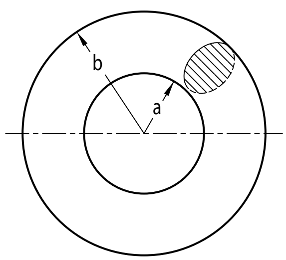
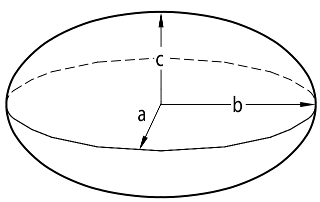
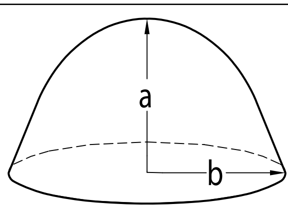
(*Contd.*)

<table>
<tr><td colspan="2">Spherical cap of radius r and height h</td></tr>
<tr><td>
</td><td>Volume of shaded part = $\frac{1}{3}\pi h^2(3r - h)$ (1.35)
Surface area = $2\pi rh$ (1.36)</td></tr>
<tr><td colspan="2">Frustum of right circular cone of radii a, b and height h</td></tr>
<tr><td>
</td><td>Volume = $\frac{1}{3}\pi h(a^2 + ab + b^2)$ (1.37)
Lateral surface area =
$= \pi(a+b)\sqrt{h^2 + (b-a)^2}$
$= \pi(a+b)l$ (1.38)</td></tr>
<tr><td colspan="2">Torus of inner radius a and outer radius b</td></tr>
<tr><td>
</td><td>Volume = $\frac{1}{4}\pi^2(a+b)(b-a)^2$ (1.39)
Surface area = $\pi^2(b^2 - a^2)$ (1.40)</td></tr>
<tr><td colspan="2">Ellipsoid of semi-axes a, b, c</td></tr>
<tr><td>
</td><td>Volume = $\frac{4}{3}\pi abc$ (1.41)</td></tr>
<tr><td colspan="2">Paraboloid of revolution</td></tr>
<tr><td>
</td><td>Volume = $\frac{1}{2}\pi b^2 a$ (1.42)</td></tr>
</table>

1.4 FORMULAE—ALGEBRA AND TRIGONOMETRY

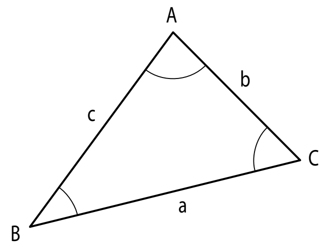
Table 1.8 *Algebraic and trigonometric formulae*

Basic relationships			
$(x+y)^2 = x^2 + 2xy + y^2$	(1.43)	$(x-y)^2 = x^2 - 2xy + y^2$	(1.44)
$(x+y)^3 = x^3 + 3x^2y + 3xy^2 + y^3$	(1.45)	$(x-y)^3 = x^3 - 3x^2y + 3xy^2 - y^3$	(1.46)
$x^2 - y^2 = (x+y)(x-y)$	(1.47)	$x^3 - y^3 = (x-y)(x^2 + xy + y^2)$	(1.48)
sin (0) = 0 cos (0) = 1 tan (0) = 0	(1.49)	sin (90) = 1 cos (90) = 0 tan (90) = ∞	(1.50)
sin (180) = 0 cos (180) = –1 tan (180) = 0	(1.51)	sin (270) = –1 cos (270) = 0 tan (270) = ∞	(1.52)
sin (360) = 0 cos (360) = 1 tan (360) = 0 $\sin(45) = \cos(45) = \dfrac{\sqrt{2}}{2}$ tan (45) = 1	(1.53) (1.54) (1.55)	$\sin(30) = \dfrac{1}{2}$ $\cos(30) = \dfrac{\sqrt{3}}{2}$ $\tan(30) = \dfrac{1}{\sqrt{3}}$	(1.56)
Addition formulae			
$\sin^2 A + \cos^2 A = 1$	(1.57)		
$\sin(A+B) = \sin A\cos B + \cos A\sin B$ $\sin(A-B) = \sin A\cos B - \cos A\sin B$	(1.58) (1.59)	$\cos(A+B) = \cos A\cos B - \sin A\sin B$ $\cos(A-B) = \cos A\cos B + \sin A\sin B$	(1.60) (1.61)
$\tan(A+B) = \dfrac{\tan A + \tan B}{1 - \tan A\tan B}$ $\tan(A-B) = \dfrac{\tan A - \tan B}{1 + \tan A\tan B}$	(1.62) (1.63)	$\sin 2A = 2\sin A\cos A$ $\cos 2A = \cos^2 A - \sin^2 A = 1 - 2\sin^2 A = 2\cos^2 A - 1$ $\tan 2A = \dfrac{2\tan A}{1 - \tan^2 A}$	(1.64) (1.65) (1.66)
$\sin 3A = 3\sin A - 4\sin^3 A$ $\cos 3A = 4\cos^3 A - 3\cos A$ $\tan 3A = \dfrac{3\tan A - \tan^3 A}{1 - 3\tan^2 A}$	(1.67) (1.68) (1.69)	$\sin A + \sin B = 2\sin\left[\frac{1}{2}(A+B)\right]\cos\left[\frac{1}{2}(A-B)\right]$ $\sin A - \sin B = 2\cos\left[\frac{1}{2}(A+B)\right]\sin\left[\frac{1}{2}(A-B)\right]$	(1.70) (1.71)
$\cos A + \cos B = 2\cos\left[\frac{1}{2}(A+B)\right]\cos\left[\frac{1}{2}(A-B)\right]$ $\cos A - \cos B = 2\sin\left[\frac{1}{2}(A+B)\right]\sin\left[\frac{1}{2}(B-A)\right]$	(1.72) (1.73)	$\sin A\sin B = \frac{1}{2}[\cos(A-B) - \cos(A+B)]$ $\cos A\cos B = \frac{1}{2}[\cos(A-B) + \cos(A+B)]$ $\sin A\cos B = \frac{1}{2}[\sin(A-B) + \sin(A+B)]$	(1.74) (1.75) (1.76)

(*Contd.*)

(*Contd.*)

Laws of triangle



Law of sines

$$\frac{a}{\sin A} = \frac{b}{\sin B} = \frac{c}{\sin C} \qquad (1.77)$$

Law of cosines

$$c^2 = a^2 + b^2 - 2ab\cos C \qquad (1.78)$$

Law of tangents

$$\frac{a+b}{a-b} = \frac{\tan\left[\frac{1}{2}(A+B)\right]}{\tan\left[\frac{1}{2}(A-B)\right]} \qquad (1.79)$$

Complex numbers

Equality of complex numbers

$$a + bi = c + di$$

if and only if

$$a = c \quad \text{and} \quad b = d \qquad (1.80)$$

Addition of complex numbers

$$(a + bi) + (c + di) = (a + c) + (b + d)i \qquad (1.81)$$

Subtraction of complex numbers

$$(a + bi) - (c + di) = (a - c) + (b - d)i \qquad (1.82)$$

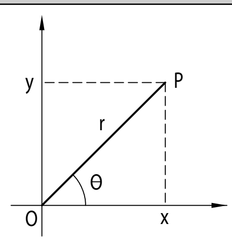
Multiplication of complex numbers

$$(a + bi) \times (c + di) = (ac - bd) + (ad + bc)i \qquad (1.83)$$

Division of complex numbers

$$\frac{a+bi}{c+di} = \frac{a+bi}{c+di} \times \frac{c-di}{c-di} = \frac{ac+bd}{c^2+d^2} + \left(\frac{bc-ad}{c^2+d^2}\right)i \qquad (1.84)$$

Polar form of complex numbers



Coordinates of point P are (x, y) or (r, θ)

$$x + iy = r(\cos\theta + i\sin\theta) \qquad (1.85)$$

where

$$r = \sqrt{x^2 + y^2} \quad \text{and} \quad \tan\theta = \frac{y}{x} \qquad (1.86)$$

Laws of exponents

$$a^p \times a^q = a^{p+q} \qquad (1.87)$$

$$a^p / a^q = a^{p-q} \qquad (1.88)$$

$$(a^p)^q = a^{pq} \qquad (1.89)$$

$$\sqrt[n]{a} = a^{1/n} \qquad (1.90)$$

$$a^0 = 1 \quad \text{if} \quad (a \neq 0) \qquad (1.91)$$

$$a^{-p} = 1/a^p \qquad (1.92)$$

$$(ab)^p = a^p b^p \qquad (1.93)$$

$$\sqrt[n]{a^m} = a^{m/n} \qquad (1.94)$$

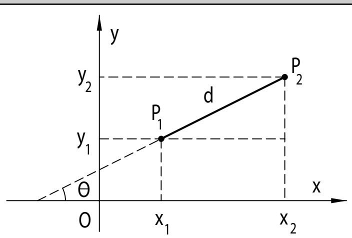
(*Contd.*)

(*Contd.*)

Laws of logarithms	
$\log_a (MN) = \log_a M + \log_a N$ (1.95) $\log_a (M^p) = p \log_a M$ (1.96) $\log_a N = \dfrac{\log_b N}{\log_b a}$ (1.97)	$\log_a \left(\dfrac{M}{N}\right) = \log_a M - \log_a N$ (1.98)
$\log_e N = \ln N = 2.3 \log_{10} N$ (1.99)	$\log_{10} N = \log N = 0.434 \log_e N$ (1.100)
Exponential and trigonometric relationships	
$e^{i\theta} = \cos\theta + i\sin\theta$ (1.101) $e^{-i\theta} = \cos\theta - i\sin\theta$ (1.102) $(x + iy) = r(\cos\theta + i\sin\theta) = re^{i\theta}$ (1.103)	$\sin\theta = \dfrac{e^{i\theta} - e^{-i\theta}}{2i}$ (1.104) $\cos\theta = \dfrac{e^{i\theta} + e^{-i\theta}}{2}$ (1.105) $\tan\theta = \dfrac{(e^{i\theta} - e^{-i\theta})}{i(e^{i\theta} + e^{-i\theta})}$ (1.106)
Solution of quadratic equation	
Quadratic equation $ax^2 + bx + c = 0$ (1.107)	Solution $x = \dfrac{-b \pm \sqrt{b^2 - 4ac}}{2a}$ (1.108)

1.5 FORMULAE—ANALYTICAL GEOMETRY

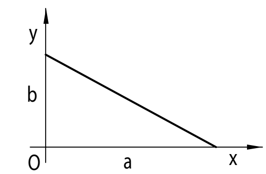
Table 1.9 *Formulae of plane analytical geometry*

Line between two points $P_1(x_1, y_1)$ and $P_2(x_2, y_2)$	
	(i) Distance d $d = \sqrt{(x_2 - x_1)^2 + (y_2 - y_1)^2}$ (1.109) (ii) Slope m $m = \tan\theta = \dfrac{(y_2 - y_1)}{(x_2 - x_1)}$ (1.110) (iii) Equation of line P_1P_2 $\dfrac{y - y_1}{x - x_1} = m$ (1.111) or $y = m\,x + b$ (1.112) where $b = (y_1 - m\,x_1)$ (intercept on y-axis) (1.113)

(*Contd.*)

(*Contd.*)

Line with intercepts on x and y axes

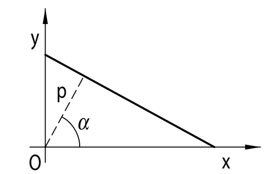


Equation of line

$$\frac{x}{a} + \frac{y}{b} = 1 \qquad (1.114)$$

a = intercept on x-axis
b = intercept on y-axis

Normal form of equation of line



Equation of line

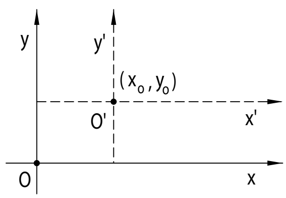
$$x \cos \alpha + y \sin \alpha = p \qquad (1.115)$$

p = perpendicular distance from origin to line
α = angle of inclination of perpendicular with positive x-axis

General equation of line

$$A\,x + B\,y + C = 0 \qquad (1.116)$$

Transformation of coordinates involving pure translation



$$x = x' + x_o$$
$$y = y' + y_o \qquad (1.117)$$

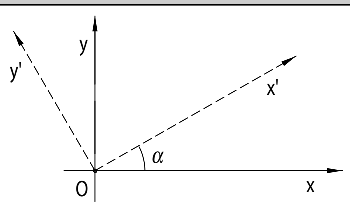
or

$$x' = x - x_o$$
$$y' = y - y_o \qquad (1.118)$$

where

x, y = old coordinates
x', y' = new coordinates
x_o, y_o = coordinates of new origin O' relative to old xy coordinate system

Transformation of coordinates involving pure rotation



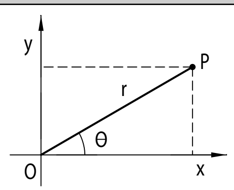
$$x = x' \cos \alpha - y' \sin \alpha$$
$$y = x' \sin \alpha + y' \cos \alpha \qquad (1.119)$$

or

$$x' = x \cos \alpha + y \sin \alpha$$
$$y' = y \cos \alpha - x \sin \alpha \qquad (1.120)$$

The origin of old (x, y) and new (x', y') coordinate systems is same, but x' axis makes an angle α with positive x-axis.

Transformation of rectangular and polar coordinates



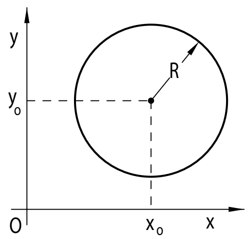
$$x = r \cos \theta \quad \text{and} \quad y = r \sin \theta \qquad (1.121)$$

$$r = \sqrt{x^2 + y^2} \quad \text{and} \quad \theta = \tan^{-1}\left(\frac{y}{x}\right) \qquad (1.122)$$

(*Contd.*)

(*Contd.*)

Equation of circle



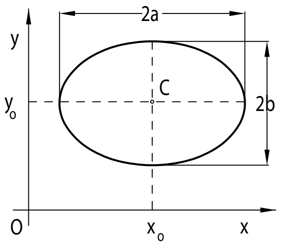
Equation

$$(x - x_o)^2 + (y - y_o)^2 = R^2 \qquad (1.123)$$

where

x_o, y_o = coordinates of centre of the circle
R = radius of circle

Equation of ellipse with major axis parallel to *x*-axis



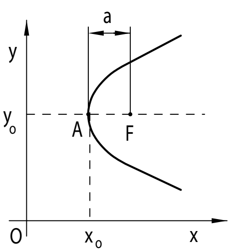
Equation

$$\frac{(x - x_o)^2}{a^2} + \frac{(y - y_o)^2}{b^2} = 1 \qquad (1.124)$$

where

$2a$ = major axis
$2b$ = minor axis
x_o, y_o = coordinates of centre of the ellipse

Equation of parabola with axis parallel to *x*-axis



Equations

$$(y - y_o)^2 = 4a(x - x_o)$$

(if parabola opens to right side) (1.125)

$$(y - y_o)^2 = -4a(x - x_o)$$

(if parabola opens to left side) (1.126)

where

x_o, y_o = coordinates of vertex A
a = distance of focus F from vertex A

1.6 FORMULAE—DIFFERENTIATION AND INTEGRATION

Table 1.10 *Formulae of differentiation*

Basic Formulae			
Note: (i) u, v. w are functions of x and (ii) c, n, a, are constants			
$\frac{d}{dx}(x) = 1$	(1.127)	$\frac{d}{dx}(u + v) = \frac{du}{dx} + \frac{dv}{dx}$	(1.132)
$\frac{d}{dx}(c) = 0$	(1.128)	$\frac{d}{dx}(u - v) = \frac{du}{dx} - \frac{dv}{dx}$	(1.133)
$\frac{d}{dx}(cx) = c$	(1.129)	$\frac{d}{dx}(uv) = u\frac{dv}{dx} + v\frac{du}{dx}$	(1.134)
$\frac{d}{dx}(x^n) = nx^{n-1}$	(1.130)	$\frac{d}{dx}\left(\frac{u}{v}\right) = \frac{v\left(\frac{du}{dx}\right) - u\left(\frac{dv}{dx}\right)}{v^2}$	(1.135)
$\frac{dy}{dx} = \left(\frac{dy}{du}\right)\left(\frac{du}{dx}\right)$	(1.131)		

(*Contd.*)

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Derivatives of trigonometric functions	
$\frac{d}{dx}(\sin x) = \cos x$ (1.136) $\frac{d}{dx}(\cos x) = -\sin x$ (1.137) $\frac{d}{dx}(\tan x) = \sec^2 x$ (1.138)	$\frac{d}{dx}(\sin^{-1} x) = \frac{1}{\sqrt{1-x^2}}$ provided $\left[-\frac{\pi}{2} < \sin^{-1} x < \frac{\pi}{2}\right]$ (1.139) $\frac{d}{dx}(\cos^{-1} x) = \frac{-1}{\sqrt{1-x^2}}$ provided $[0 < \cos^{-1} x < \pi]$ (1.140) $\frac{d}{dx}(\tan^{-1} x) = \frac{1}{1+x^2}$ provided $\left[-\frac{\pi}{2} < \tan^{-1} x < \frac{\pi}{2}\right]$ (1.141)
Derivatives of exponential and logarithmic functions	
$\frac{d}{dx}(\log_e x) = \frac{1}{x}$ (1.142)	$\frac{d}{dx}(a^x) = a^x \log_e a$ (1.143) $\frac{d}{dx}(e^x) = e^x$ (1.144)
Derivatives of hyperbolic functions	
$\frac{d}{dx}(\sinh x) = \cosh x$ (1.145) $\frac{d}{dx}(\cosh x) = \sinh x$ (1.146) $\frac{d}{dx}(\tanh x) = \text{sech}^2 x$ (1.147)	$\frac{d}{dx}(\sinh^{-1} x) = \frac{1}{\sqrt{x^2+1}}$ (1.148) $\frac{d}{dx}(\cosh^{-1} x) = \frac{1}{\sqrt{x^2-1}}$ (1.149) $\frac{d}{dx}(\tanh^{-1} x) = \frac{1}{1-x^2}$ (1.150)
Higher derivatives	
Second Derivative $\frac{d}{dx}\left(\frac{dy}{dx}\right) = \frac{d^2 y}{dx^2} = y''$ (1.151)	Third Derivative $\frac{d}{dx}\left(\frac{d^2 y}{dx^2}\right) = \frac{d^3 y}{dx^3} = y'''$ (1.152)

Table 1.11 *Formulae of indefinite integral*

Note: (i) u, v are functions of x and (ii) c, n, a, b are constants	
$\int dx = x$ (1.153) $\int a\,dx = ax$ (1.154) $\int (u+v)dx = \int u\,dx + \int v\,dx$ (1.155) $\int (u-v)dx = \int u\,dx - \int v\,dx$ (1.156) $\int \frac{dx}{x} = \log_e x$ (1.157) $\int e^x dx = e^x$ (1.158) $\int e^{ax} dx = \frac{e^{ax}}{a}$ (1.159)	$\int u\,dv = uv - \int v\,du$ (1.160) (Integration by parts) $\int f(ax)dx = \frac{1}{a}\int f(x)dx$ (1.161) $\int x^n dx = \frac{x^{n+1}}{n+1}$ $(n \neq -1)$ (1.162) $\int a^x dx = \frac{a^x}{\log_e a}$ (1.163) $\int \log_e x\,dx = x\log_e x - x$ (1.164)
$\int \sin x\,dx = -\cos x$ (1.165) $\int \cos x\,dx = \sin x$ (1.166) $\int \tan x\,dx = \log_e \sec x$ (1.167) $\int \sin ax\,dx = -\frac{\cos ax}{a}$ (1.168) $\int \cos ax\,dx = \frac{\sin ax}{a}$ (1.169) $\int \tan ax\,dx = -\left(\frac{1}{a}\right)\log_e \cos ax$ (1.170)	$\int \sin^2 x\,dx = \left[\frac{x}{2} - \frac{\sin 2x}{4}\right]$ (1.171) $\int \cos^2 x\,dx = \left[\frac{x}{2} + \frac{\sin 2x}{4}\right]$ (1.172) $\int \tan^2 x\,dx = \tan x - x$ (1.173) $\int \sin ax \cos ax\,dx = \frac{\sin^2 ax}{2a}$ (1.174) $\int \frac{dx}{\sin ax \cos ax} = \left(\frac{1}{a}\right)\log_e \tan ax$ (1.175)
$\int \sinh x\,dx = \cosh x$ (1.176) $\int \cosh x\,dx = \sinh x$ (1.177) $\int \tanh x\,dx = \log_e \cosh x$ (1.178)	$\int \frac{dx}{x^2+a^2} = \frac{1}{a}\tan^{-1}\left(\frac{x}{a}\right)$ (1.179) $\int \frac{dx}{x^2-a^2} = \frac{1}{2a}\log_e\left(\frac{x-a}{x+a}\right)$ (1.180) $\int \frac{dx}{a^2-x^2} = \frac{1}{2a}\log_e\left(\frac{a+x}{a-x}\right)$ (1.181)
$\int \frac{dx}{\sqrt{x^2+a^2}} = \log_e[x+\sqrt{x^2+a^2}]$ (1.182) $\int \frac{dx}{\sqrt{x^2-a^2}} = \log_e[x+\sqrt{x^2-a^2}]$ (1.183) $\int \frac{dx}{\sqrt{a^2-x^2}} = \sin^{-1}\left(\frac{x}{a}\right)$ (1.184)	$\int \frac{dx}{(ax+b)} = \left(\frac{1}{a}\right)\log_e(ax+b)$ (1.185) $\int \frac{x\,dx}{(ax+b)} = \left(\frac{x}{a}\right) - \left(\frac{b}{a^2}\right)\log_e(ax+b)$ (1.186) $\int \frac{dx}{\sqrt{(ax+b)}} = \frac{2\sqrt{(ax+b)}}{a}$ (1.187) $\int \frac{x\,dx}{\sqrt{(ax+b)}} = \frac{2(ax-2b)}{3a^2}\sqrt{(ax+b)}$ (1.188) $\int \sqrt{(ax+b)}\,dx = \frac{2\sqrt{(ax+b)^3}}{3a}$ (1.189) $\int x\sqrt{(ax+b)}\,dx = \frac{2(3ax-2b)}{15a^2}\sqrt{(ax+b)^3}$ (1.190)

(*Contd.*)

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$\int \frac{dx}{(x^2+a^2)} = \left(\frac{1}{a}\right)\tan^{-1}\left(\frac{x}{a}\right)$ (1.191)	$\int \frac{dx}{(x^2-a^2)} = \left(\frac{1}{2a}\right)\log_e\left(\frac{x-a}{x+a}\right)$ (1.194)
$\int \frac{x\,dx}{(x^2+a^2)} = \left(\frac{1}{2}\right)\log_e(x^2+a^2)$ (1.192)	$\int \frac{x\,dx}{(x^2-a^2)} = \left(\frac{1}{2}\right)\log_e(x^2-a^2)$ (1.195)
$\int \frac{x^2dx}{(x^2+a^2)} = x - a\tan^{-1}\left(\frac{x}{a}\right)$ (1.193)	$\int \frac{x^2dx}{(x^2-a^2)} = x + \left(\frac{a}{2}\right)\log_e\left(\frac{x-a}{x+a}\right)$ (1.196)

Table 1.12 *Formulae of definite integral*

Note: a, b are constants	
$\int_0^\infty \frac{dx}{x^2+a^2} = \frac{\pi}{2a}$ (1.197)	$\int_0^\pi \sin mx \sin nx\,dx = 0$ (1.207) [if m, n integers and $(m \neq n)$]
$\int_0^a \frac{dx}{\sqrt{a^2-x^2}} = \frac{\pi}{2}$ (1.198)	$\int_0^\pi \sin mx \sin nx\,dx = \frac{\pi}{2}$ (1.208) [if m, n integers and $(m = n)$]
$\int_0^a \sqrt{a^2-x^2}\,dx = \frac{\pi a^2}{4}$ (1.199)	$\int_0^\pi \cos mx \cos nx\,dx = 0$ (1.209) [if m, n integers and $(m \neq n)$]
$\int_0^{2\pi} \frac{dx}{a+b\sin x} = \frac{2\pi}{\sqrt{a^2-b^2}}$ (1.200)	$\int_0^\pi \cos mx \cos nx\,dx = \frac{\pi}{2}$ (1.210) [if m, n integers and $(m = n)$]
$\int_0^{2\pi} \frac{dx}{a+b\cos x} = \frac{2\pi}{\sqrt{a^2-b^2}}$ (1.201)	$\int_0^\pi \sin mx \cos nx\,dx = 0$ (1.211) [if m, n integers and $(m + n)$ even]
$\int_0^\infty e^{-ax}\cos bx\,dx = \frac{a}{a^2+b^2}$ (1.202)	$\int_0^\pi \sin mx \cos nx\,dx = \frac{2m}{(m^2-n^2)}$ (1.212) [if m, n integers and $(m + n)$ odd
$\int_0^\infty e^{-ax}\sin bx\,dx = \frac{b}{a^2+b^2}$ (1.203)	$\int_0^{\pi/2} \sin^2 x\,dx = \int_0^{\pi/2} \cos^2 x\,dx = \frac{\pi}{4}$ (1.213)
$\int_0^\infty \frac{x\,dx}{e^x-1} = \frac{\pi^2}{6}$ (1.204)	
$\int_0^1 \frac{\log_e x}{1+x}dx = -\frac{\pi^2}{12}$ (1.205)	
$\int_0^1 \frac{\log_e x}{1-x}dx = -\frac{\pi^2}{6}$ (1.206)	

Properties of Engineering Materials

2

2.1 PHYSICAL PROPERTIES

Table 2.1 *Physical properties of common metals*

Metal	Modulus of elasticity (E) (MPa or N/mm^2)	Modulus of rigidity (G) (MPa or N/mm^2)	Poisson's ratio (ν)	Mass density (kg/m^3)	Specific gravity (ρ)
Aluminium alloys	72×10^3	27×10^3	0.32	2800	2.8
Brass, Bronze	110×10^3	41×10^3	0.33	8600	8.6
Grey cast iron	103×10^3	41×10^3	0.26	7200	7.2
Carbon steel	207×10^3	79×10^3	0.3	7800	7.8
Alloy steel	207×10^3	79×10^3	0.3	7800	7.8
Stainless steel	190×10^3	73×10^3	0.3	7800	7.8
Copper	121×10^3	46×10^3	0.33	8900	8.9
Magnesium alloy	45×10^3	17×10^3	0.35	1800	1.8
Titanium alloy	114×10^3	43×10^3	0.33	4400	4.4
Zinc alloy	83×10^3	31×10^3	0.33	6600	6.6

Note: (i) The values given in the above table are for representative purposes. (ii) Exact values depend upon composition and processing and may sometimes vary considerably.

Table 2.2 *Thermal properties of common metals*

Metal	Coefficient of thermal expansion (α) (per $^\circ$C) $\times$ (10^{-6})	Thermal conductivity (k) (W/m-$^\circ$C)	Specific heat (C_p) (J/kg-$^\circ$C)
Aluminium alloys	22	173	920
Brass, Bronze	19	78	420
Grey cast iron	12	50	540
Carbon steel	12	47	460
Alloy steel	11	38	460

(*Contd.*)

(*Contd.*)

Stainless steel	14	21	460
Copper	17	381	420
Magnesium alloy	26	95	1170
Titanium alloy	9	12	500
Zinc alloy	27	111	460

Note: (i) The values given in above table are for representative purposes. (ii) Exact values depend upon composition and processing and may sometimes vary considerably.

2.2 CONVERSION OF HARDNESS NUMBERS

Table 2.3 *Conversion of hardness numbers (Approximate equivalent numbers)*

Brinell Hardness Number (BHN)	Rockwell Hardness Number Scale B	Rockwell Hardness Number Scale C
739	—	65
722	—	64
705	—	63
688	—	62
670	—	61
654	—	60
634	—	59
615	—	58
595	—	57
577	—	56
560	—	55
543	—	54
525	—	53
512	—	52
496	—	51
481	—	50
469	—	49
455	—	48
443	—	47
432	—	46
421	—	45
409	—	44

(*Contd.*)

(*Contd.*)

400	—	43
390	—	42
381	—	41
371	—	40
362	—	39
353	—	38
344	—	37
336	109.0	36
327	108.5	35
319	108.0	34
311	107.5	33
301	107.0	32
294	106.0	31
286	105.5	30
279	104.5	29
271	104.0	28
264	103.0	27
258	102.5	26
253	101.5	25
247	101.0	24
243	100.0	23
237	99.0	22
231	98.5	21
226	97.8	20
219	96.7	18
212	95.5	16
203	93.9	14
194	92.3	12
187	90.7	10
179	89.5	8
171	87.1	6
165	85.5	4
158	83.5	2
152	81.7	0

Note: The approximate relationship between tensile strength (S_{ut}) of carbon and low alloy steels and BHN is given by,

$$S_{ut} \cong 3.45\ (\text{BHN}) \pm 0.2\ (\text{BHN}) \qquad (\text{in MPa or N/mm}^2)$$

2.3 GREY CAST IRON

Table 2.4 *Symbols for designation of ferrous castings*

Type of casting	B.I.S. symbol for designation
Grey cast iron	FG
Black-heart malleable cast iron	BM
White-heart malleable cast iron	WM
Pearlitic malleable cast iron	PM
Spheroidal or nodular graphite cast iron	SG
Austenitic flake graphite cast iron	AFG
Austenitic spheroidal or nodular graphite cast iron	ASG
Abrasion resistant cast iron	ABR
High-tensile steel castings	CS
Heat-resistant steel casting	CSH
Corrosion-resistant steel casting	CSC

Table 2.5 *Mechanical properties of grey cast iron*

Grade	Tensile strength (minimum) (MPa or N/mm^2)	Brinell Hardness (HB)
FG 150	150	130–180
FG 200	200	160–220
FG 220	220	180–220
FG 260	260	180–230
FG 300	300	180–230
FG 350	350	207–241
FG 400	400	207–270

Note: (i) The properties in above table are based on tests conducted on a 30 mm diameter bar. (ii) Grey cast iron is specified by the symbol FG followed by tensile strength in MPa or N/mm^2 for a 30 mm section. FG 220 means grey cast iron with an ultimate tensile strength of 220 N/mm^2.

Table 2.6 *Tensile strength of different section thickness for grey cast iron*

Grade	Section thickness of casting (mm)	Tensile strength (MPa or N/mm^2)
FG 150	2.5–10	155
	10–20	130
	20–30	115
	30–50	105

(*Contd.*)

(*Contd.*)

FG 200	2.5–10	205
	10–20	180
	20–30	160
	30–50	145
FG 260	4–10	260
	10–20	235
	20–30	215
	30–50	195
FG 300	10–20	270
	20–30	245
	30–50	225
FG 350	10–20	315
	20–30	290
	30–50	270

Table 2.7 *Properties of grey cast iron*

Properties	**Grade**						
	FG 150	**FG 200**	**FG 220**	**FG 260**	**FG 300**	**FG 350**	**FG 400**
Tensile strength (MPa or N/mm^2)	150	200	220	260	300	350	400
0.01 percent Proof stress (MPa or N/mm^2)	42	56	62	73	84	98	112
0.1 percent Proof stress (MPa or N/mm^2)	98	130	143	169	195	228	260
Notched Tensile strength (Circumferential 45º V-notch, root radius 0.25 mm or notch depth 2.5 mm, notch dia.20 mm or notch depth 3.3 mm, notch diameter 7.6 mm) (MPa or N/mm^2)	120	160	176	208	240	280	320
Notched Tensile strength (Circumferential notch, radius 9.5 mm, notch depth 2.5 mm, notch dia.20 mm) (MPa or N/mm^2)	150	200	220	260	300	350	400
Compressive strength (MPa or N/mm^2)	600	720	768	864	960	1080	1200

(*Contd.*)

(*Contd.*)

Shear strength (MPa or N/mm^2)	173	230	253	299	345	403	460
Modulus of elasticity (MPa or N/mm^2)	100×10^3	114×10^3	120×10^3	128×10^3	135×10^3	140×10^3	145×10^3
Fatigue limit (Wohler) -Un-notched (8.4 mm dia) (MPa or N/mm^2)	63	90	99	117	135	149	152
Fatigue limit (Wohler)-Notched Circumferential 45º V-notch (0.25 mm root radius, diameter at notch 8.4 mm, depth of notch 3.4 mm) (MPa or N/mm^2)	68	87	94	108	122	129	127
Specific gravity	7.05	7.10	7.15	7.20	7.25	7.30	7.30

Note: Poisson's ratio for all grades of grey cast iron is 0.26.

Table 2.8 *Applications of grey cast iron*

Grade	Applications
FG 150	Miscellaneous soft iron castings, in which strength is not of primary consideration, exhaust manifolds.
FG 200	Small engine cylinder blocks, cylinder heads, air cooled cylinders, pistons, clutch plates, oil pump bodies, transmission cases, gear boxes, clutch housings, and light duty brake drums.
FG 220	Brake drums, clutch plates, flywheels.
FG 260	Automobile and diesel engine cylinder blocks, cylinder heads, flywheels, cylinder liners, pistons, medium-duty brake drum, and clutch plates.
FG 300	Diesel engine blocks, truck and tractor cylinder blocks and heads, heavy flywheels, tractor transmission cases, differential carrier castings, and heavy duty gear boxes.
FG 350	Diesel engine castings, liners, cylinders and pistons and heavy parts in general.
FG 400	Special high-strength castings.

Table 2.9 *ASTM and DIN designations of grey cast iron*

ASTM designations
The American Society for Testing Materials (ASTM) has classified grey cast iron by means of a number. This class number gives minimum tensile strength in kpsi. For example, ASTM Class No. 20 has minimum ultimate tensile strength of 20000 psi. Similarly, a cast iron with minimum ultimate tensile strength of 50000 psi is designated as ASTM Class No. 50. Commonly used ASTM classes of cast iron are 20, 25, 30, 35, 40, 50 and 60.
DIN designations
In Germany, *Deutches Institut Fuer Normung* (DIN) has specified grey cast iron by minimum ultimate tensile strength in kgf/mm^2. For example, GG-12 indicates grey cast iron with minimum ultimate tensile strength of 12 kgf/mm^2. The common varieties of grey cast iron according to DIN standard are GG-12, GG-14, GG-18, GG-22, GG-26 and GG-30.

2.4 MALLEABLE CAST IRON

Table 2.10 *Mechanical properties of malleable cast iron*

Grade	Section thickness (mm)	Diameter of test bar (mm)	Tensile strength Min. (MPa or N/mm^2)	0.2 percent Proof stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Hardness (HB)
WM 350	0–8	9	340	—	5	230 max.
	8–13	12	350	—	4	
	over 13	15	360	—	3	
WM 400	0–8	9	360	200	8	220 max.
	8–13	12	400	220	5	
	over 13	15	420	230	4	
BM 300	All sizes	15	300	–	6	150 max.
BM 320	All sizes	15	320	190	12	150 max.
BM 350	All sizes	15	350	200	10	150 max.
PM 450	All sizes	15	450	270	6	150–200
PM 500	All sizes	15	500	300	5	160–200
PM 550	All sizes	15	550	340	4	180–230
PM 600	All sizes	15	600	390	3	200–250
PM 700	All sizes	15	700	530	2	240–290

Note: (i) There are three basic types of malleable cast iron – whiteheart, blackheart and pearlitic, which are designated by symbols WM, BM and PM, respectively and followed by minimum tensile strength in MPa or N/mm^2.
(ii) For all grades: (a) The shear strength is approximately 0.9 times the tensile strength. (b) The unnotched fatigue limit (Wohler) is about 0.45 times the tensile strength. The notched fatigue limit is about 0.6 times the unnotched fatigue limit.

Table 2.11 *Physical properties of malleable cast iron*

Type	Modulus of elasticity (E) (MPa or N/mm^2)	Modulus of rigidity (G) (MPa or N/mm^2)	Poisson's ratio (ν)	Mass density (kg/m^3)	Specific gravity (ρ)
Whiteheart Malleable cast iron (all grades)	175.8×10^3	70.3×10^3	0.26	7400	7.4
Blackheart Malleable cast iron (all grades)	168.9×10^3	67.6×10^3	0.26	7350	7.35
Pearlitic Malleable cast iron (all grades)	172.4×10^3	68.9×10^3	0.26	7300	7.3

Note: (i) The values given in above table are representative. (ii) Exact values depend upon composition and processing and may vary.

Table 2.12 *Applications of malleable and spheroidal graphite cast iron*

Type	Applications
Malleable cast iron	Pipe flanges, valve parts, housings for ball bearings, pulleys, and substitute castings for ordinary forgings.
Spheroidal graphite cast iron	Valves and fittings for steam and chemical equipments, crankshafts, gears, rollers, slides, hubs, forming dies.

Note: The above applications are tentative for the use of students. Detail analysis should be carried out for correct selection of materials for any application.

2.5 SPHEROIDAL GRAPHITE CAST IRON

Table 2.13 *Mechanical properties of spheroidal or nodular graphite cast iron*

Grade	Tensile strength Min. (MPa or N/mm^2)	0.2 percent Proof stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Hardness (HB)
SG 900/2	900	600	2	280–360
SG 800/2	800	480	2	245–335
SG 700/2	700	420	2	225–305
SG 600/3	600	370	3	190–270
SG 500/7	500	320	7	160–240
SG 450/10	450	310	10	160–210
SG 400/15	400	250	15	130–180
SG 400/18	400	250	18	130–180
SG 350/22	350	220	22	$\leq$ 150

Note: (i) Spheroidal or nodular graphite cast iron is commonly known as ductile cast iron. (ii) Spheroidal or nodular graphite cast iron is specified by the symbol SG followed by the minimum tensile strength in MPa or N/mm^2 and minimum elongation in percent.

2.6 SPHEROIDAL GRAPHITE AUSTENITIC CAST IRON

Table 2.14 *Mechanical properties of spheroidal graphite austenitic cast iron*

Grade	Tensile strength Min. (MPa or N/mm^2)	0.2 percent Proof stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Brinell Hardness Max. (HB)
ASG Ni 13 Mn 7	390	210	15	170
ASG Ni 20 Cr 2	370	210	7	200
ASG Ni 20 Cr 3	390	210	7	255
ASG Ni 20 Si 5 Cr 2	370	210	10	230
ASG Ni 22	370	170	20	170
ASG Ni 23 Mn 4	440	210	25	180
ASG Ni 30 Cr 1	370	210	13	190

(*Contd.*)

(*Contd.*)

ASG Ni 30 Cr 3	370	210	7	200
ASG Ni 30 Si 5 Cr 5	390	240	–	250
ASG Ni 35	370	210	20	180
ASG Ni 35 Cr 3	370	210	7	190

Note: (i) Spheroidal (or nodular) graphite austenitic cast iron is designated by initial letter 'ASG' followed by chemical symbols and figures indicating the alloying elements and their approximate mean levels.
(ii) For all grades: (a) Specific gravity = 7.3 to 7.6 (b) Thermal conductivity = 12.6 W/(m ºC).

Table 2.15 *Applications of spheroidal graphite austenitic cast iron*

Grade	Applications
ASG Ni 13 Mn 7	Pressure covers for turbines, housings for switchgear insulator flanges, terminals and ducts.
ASG Ni 20 Cr 2	Pumps, valves, compressors, bushings, turbo supercharger housings, exhaust gas manifolds.
ASG Ni 20 Cr 3	Pumps, valves, compressors, bushings, turbo supercharger housings, exhaust gas manifolds.
ASG Ni 20 Si 5 Cr 2	Pump components, valves, castings for industrial furnaces.
ASG Ni 22	Pumps, valves, compressors, bushings, turbo supercharger housings, exhaust gas manifolds.
ASG Ni 23 Mn 4	Castings for refrigerating equipments.
ASG Ni 30 Cr 1	Pumps, boilers, filter parts, exhaust gas manifolds, valves, turbo supercharger housings.
ASG Ni 30 Cr 3	Pumps, boilers, filter parts, exhaust gas manifolds, valves, turbo supercharger housings.
ASG Ni 30 Si 5 Cr 5	Pump components and valve castings for industrial furnaces.
ASG Ni 35	Parts with dimensional stability (for example, machine tools), scientific instruments.
ASG Ni 35 Cr 3	Parts of gas turbine housings, glass moulds.

2.7 FLAKE GRAPHITE AUSTENITIC CAST IRON

Table 2.16 *Mechanical properties of flake graphite austenitic cast iron*

Grade	Tensile strength (MPa or N/mm^2)	Compressive strength (MPa or N/mm^2)	Elongation (Percent)	Brinell Hardness (HB)
AFG Ni 13 Mn 7	140–220	630–840	—	120–150
AFG Ni 15 Cu 6 Cr 2	170–210	700–840	2	140–200
AFG Ni 15 Cu 6 Cr 3	190–240	860–1100	1–2	150–250
AFG Ni 20 Cr 2	170–210	700–840	2–3	120–215
AFG Ni 20 Cr 3	190–240	860–1100	1–2	160–250
AFG Ni 20 Si 5 Cr 3	190–280	860–1100	2–3	140–250
AFG Ni 30 Cr 3	190–240	700–910	1–3	120–215
AFG Ni 30 Si 5 Cr 5	170–240	560	—	150–210
AFG Ni 35	120–180	560–700	1–3	120–140

Note: (i) Flake graphite austenitic cast iron is designated by initial letter 'AFG' followed by chemical symbols and figures indicating the alloying elements and their approximate mean levels.
(ii) For all grades: (a) Specific gravity = 7.3 (b) Thermal conductivity = 37.7 to 41.9 W/(m ºC) (c) Specific heat = 0.46 to 0.50 J/(g ºC).

Table 2.17 *Applications of flake graphite austenitic cast iron*

Grade	Applications
AFG Ni 13 Mn 7	Pressure covers for turbine generator sets, housings for switchgear, insulator flanges, terminals and ducts.
AFG Ni 15 Cu 6 Cr 2	Pumps, valves, furnace components, bushings, piston ring carriers for light alloy metal pistons.
AFG Ni 15 Cu 6 Cr 3	Pumps, valves, furnace components, bushings, piston ring carriers for light alloy metal pistons.
AFG Ni 20 Cr 2	Same as for AFG Ni 15 Cu 6 Cr 2, but preferable for pumps handling alkalis, used in the soap, food, artificial silk and plastics industries. Suitable where copper-free material is required.
AFG Ni 20 Cr 3	Same as for AFG Ni 15 Cu 6 Cr 2, but preferred for high temperature applications.
AFG Ni 20 Si 5 Cr 3	Pump components, valve castings for industrial furnaces.
AFG Ni 30 Cr 3	Pumps, pressure vessels, valves, filter parts, exhaust gas manifolds, turbo charger housings.
AFG Ni 30 Si 5 Cr 5	Pump components, valve castings for industrial furnaces.
AFG Ni 35	Parts with dimensional stability (for example, machine tools), scientific instruments.

2.8 CAST STEELS

Table 2.18 *Mechanical properties of carbon steel castings for general engineering purposes*

Grade	Tensile strength Min. (MPa or N/mm^2)	Yield stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Reduction of area Min. (Percent)	Impact strength Min. (J or N-m)
200–400 N	400	200	25	40	30
200–400 W	400	200	25	40	45
230–450 N	450	230	22	31	25
230–450 W	450	230	22	31	45
280–520 N	520	280	18	25	22
280–520 W	520	280	18	25	22
340–570 N	570	340	15	21	20
340–570 W	570	340	15	21	20

Note: (i) Carbon steel castings are designated by two numbers indicating minimum yield stress and minimum tensile strength respectively.
(ii) Suffix N indicates normal grades.
(iii) Suffix W indicates welding grades with chemical composition restricted to ensure ease of welding.

Table 2.19 *Mechanical properties of high tensile steel castings for general engineering and structural purposes*

Grade	Tensile strength Min. (MPa or N/mm^2)	Yield stress (0.5 percent Proof stress) Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Reduction in area Min. (Percent)	Charpy, V-notch Impact value Min. (J or N-m)	Brinell Hardness Min. (HB)
CS 640	640	390	15	35	25	190
CS 700	700	580	14	30	25	207
CS 840	840	700	12	28	20	248
CS 1030	1030	850	8	20	15	305
CS 1230	1230	1000	5	12	—	355

Note: High tensile steel casting is designated by letters CS followed by minimum tensile strength in MPa or N/mm^2.

Table 2.20 *Mechanical properties of steel castings for case carburising (core properties)*

Grade	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Charpy, V-notch Impact value Min. (J or N-m)
Grade 1	490	12	25
Grade 2	1000	7	20

Table 2.21 *Mechanical properties of carbon steel castings for surface hardening*

Grade	Tensile strength Min. (MPa or N/mm^2)	Yield stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)
Grade 1	620	320	12
Grade 2	690	490	12
Grade 3	700	370	8

2.9 PLAIN CARBON STEELS

Table 2.22 *Mechanical properties of steels specified by tensile or yield properties (without detailed chemical composition)*

Steel designation	Tensile strength Min. (MPa or N/mm^2)	Yield strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)
Fe 290	290	170	27
FeE 220	290	220	27
Fe 310	310	180	26
FeE 230	310	230	26
Fe 330	330	200	26

(*Contd.*)

(*Contd.*)

FeE 250	330	250	26
Fe 360	360	220	25
FeE 270	360	270	25
Fe 410	410	250	23
FeE 310	410	310	23
Fe 490	490	290	21
FeE 370	490	370	21
Fe 540	540	320	20
FeE 400	540	400	20
Fe 620	620	380	15
FeE 460	620	460	15
Fe 690	690	410	12
FeE 520	690	520	12
Fe 770	770	460	10
FeE 580	770	580	10
Fe 870	870	520	8
FeE 650	870	650	8

Note: The steels in above table are designated by two ways: the symbol 'Fe' followed by the minimum tensile strength in MPa (or N/mm^2) or the symbol 'FeE' followed by minimum yield strength in MPa (or N/mm^2).

Table 2.23 *Applications of steels specified by tensile or yield properties (without detailed chemical composition)*

Steel designation	**Applications**
Fe 290 and FeE 220	Structural steel sheets for plain drawn or enameled parts, tubes for oil well casing, steam, water and air passage, cycle, motor-cycle and automobile tubes, rivet bars and wire.
Fe 310 and FeE 230 Fe 230 and FeE 250	Steels for locomotive, carriage and car structures, screw stock, and other general engineering purposes.
Fe 360 and FeE 270	Structural steel for chemical pressure vessels and other general engineering purposes.
Fe 410 and FeE 310	Structural steel for bridges and building construction, railway rolling stock, screw spikes, oil well casing, tube piles, and other general engineering purposes.
Fe 490 and FeE 370	Structural steel for mines, forgings for marine engines, sheet piling and machine parts.
Fe 540 and FeE 400	High tensile steel for locomotive, carriage, wagon and tramway axles, arches for mines, bolts, and seamless and welded tubes.
Fe 620 and FeE 460	High tensile steel for tramway axles and seamless tubes.
Fe 770 and FeE 580	High tensile steel for locomotive, carriage, and wagon wheels and tyres, and machine parts for heavy loading.
Fe 870 and FeE 650	High tensile steel for locomotive, carriage, and wagon wheels and tyres.

Table 2.24 *Mechanical properties of carbon steels (unalloyed steels)*
(For plates, sections, bars, billets and forgings in hot-rolled or normalised condition)

Steel designation	Tensile strength (MPa or N/mm^2)	Elongation Min. (Percent)
7C4	320–400	27
10C4	340–420	26
14C6	370–450	26
15C4	370–490	25
15C8	420–500	25
20C8	440–520	24
25C4	440–540	23
25C8	470–570	22
30C8	500–600	21
35C4	520–620	20
35C8	550–650	20
40C8	580–680	18
45C8	630–710	15
50C4	660–780	13
50C13	720 Min	11
55C8	720 Min	13
60C4	720 Min	11
65C6	720 Min	10

The designation of carbon steel consists of following three quantities in the following order:
(i) Number indicating 100 times the average percentage of carbon content.
(ii) Letter C: and
(iii) Number indicating 10 times the average percentage of manganese content.

Note: The minimum yield stress is 55% of the minimum tensile strength for all the grades.

Table 2.25 *Mechanical properties of carbon steels (unalloyed steels) (For cold-drawn bars)*

Designation	Bars up to 20 mm		Bars 20-40 mm		Bars 40-63 mm		Bars over 63 mm	
	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)
10C4	490	11	450	13	410	15	360	18
15C8	540	11	510	13	470	15	430	18
20C8	540	10	510	12	470	15	430	18
30C8	610	9	570	10	530	12	490	15
40C8	640	8	610	9	570	10	540	12
50C4	670	7	630	8	610	9	590	10
55C8	730	7	690	8	670	9	630	10

Table 2.26 *Mechanical properties of carbon steels (unalloyed steels) (For bars and forgings in hardened and tempered condition)*

Designation	Tensile strength (MPa or N/mm^2)	Yield stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Izod Impact value Min. (J or N-m)	Limiting ruling section (mm)
30C8	600–750	400	18	55	30
35C8	600–750	400	18	55	63
40C8	600–750 700–850	380 480	18 17	41 35	100 30
45C8	600–750 700–850	380 480	17 15	41 35	100 30
50C4	700–850 800–950	460 540	15 13	— —	63 30
55C8	700–850 800–950	460 540	15 13	— —	63 30

Table 2.27 *Mechanical properties of carbon steels (unalloyed steels)*
(For case-hardening steels in refined and quenched condition) (core properties)

Designation	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Izod Impact value Min. (J or N-m)	Limiting ruling section (mm)
10C4	500	17	55	15
14C6	500	17	55	15–30
15C8	500	17	55	30
20C8	500	16	55	30

Table 2.28 *Applications of carbon steels (unalloyed steels)*

Steel designation	Applications
7C4, 10C4	Used where cold formability is the primary requirement. They are used as sheet, strip, rod and wire especially where excellent surface finish or good drawing qualities are required, such as automobile body, hoods, lamps, and a multiple of deep drawn and formed products. They are also used for cold heading wire and rivets.
10C4, 14C6	Case hardening steel used for making camshafts, cams, light duty gears, worms, gudgeon pins, selector forks, spindles, pawls, ratchets, chain wheels, and tappets.
15C4	Used for lightly stressed parts.
15C8, 20C8, 25C4, 25C8	General purpose steels for low stressed components.
30C8	Cold formed parts such as brake levers. After suitable case hardening or hardening and tempering, this steel is also used in making parts, such as socket, tie-rod, shaft fork and rear hub, two wheeler and three wheeler parts such as sprocket, levers, cams, rocker arms, and bushes. Tubes for aircraft, automobile, bicycle and furniture are made of this steel.
35C4	Low stressed parts, automobile tubes and fasteners.
35C8	Low stressed parts in machine structures, cycle and motorcycle chassis tubes, fishplates for rails and fasteners.
40C8	Crankshafts, shafts, spindles, automobile axle beams, push rods, connecting rods, studs, bolts, lightly stressed gears, chain parts, and washers.
45C8	Spindles of machine tools, gears, bolts, lead screws, and shafts.
50C4	Keys, shafts, cylinders and machine components requiring moderate wear resistance. In surface hardened condition, it is suitable for large pitch worms and gears.
50C8	Rail steel. Also used for making spike bolts, gear shafts, rocking levers and cylinder liners.
55C4, 55C8	Gears, cylinders, cams, keys, crankshafts, sprockets, machine parts requiring moderate wear resistance, industrial chains, and springs.
60C4	Spindles for machine tools, hardened screws and nuts, couplings, crankshafts, axles and pinions.
65C6	High tensile structural steel for making locomotive carriage and wagon tyres, valve springs, small washers and stamped parts.

2.10 FREE CUTTING STEELS

Table 2.29 *Mechanical properties of carbon and carbon-manganese free cutting steels (for bars, billets and forgings in hot-rolled or normalised condition)*

Steel designation	Tensile strength (MPa or N/mm^2)	Elongation Min. (Percent)
10C8S10	370–490	24
14C14S14	440–540	22
25C12S14	500–600	20
40C10S18	550–650	17
11C10S25	370–490	22
40C15S12	600–700	15

The designation of free cutting steel consists of following quantities in the given order:
(i) Number indicating 100 times the average percentage of carbon content.
(ii) Letter C
(iii) Number indicating 10 times the average percentage of manganese content.
(iv) Symbol S, Se, Te or Pb depending upon the element that is present and which makes the steel free cutting.
(v) Number indicating 100 times the average percentage of the above element that makes the steel free cutting.

Note: The minimum yield stress is 55% of the minimum tensile strength for all the grades.

Table 2.30 *Mechanical properties of carbon and carbon-manganese free cutting steels (for cold-drawn bars)*

Size	Bars up to 20 mm		Bars 20-40 mm		Bars 40-63 mm		Bars over 63 mm	
Designation	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)
10C8S10	500	10	460	10	420	13	370	17
14C14S14	550	10	520	11	480	12	440	15
25C12S14	620	8	560	10	520	11	500	13
40C10S18	640	8	660	10	560	11	550	11
11C10S25	500	8	440	11	400	13	370	13
40C15S12	680	7	640	8	620	10	600	11

Table 2.31 *Mechanical properties of carbon and carbon-manganese free cutting steels (for bars and forgings in hardened and tempered condition)*

Designation	Tensile strength (MPa or N/mm^2)	Yield stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Izod Impact value Min. (J or N-m)	Limiting ruling section (mm)
40C10S18	600–750 700–850	380 480	18 17	41 35	60 30
40C15S12	600–750 700–850 800–950	420 500 560	18 18 16	48 48 41	100 60 30

Table 2.32 *Applications carbon and carbon-manganese free cutting steels*

Designation	Applications
10C8S10	Small parts to be cyanided or carbonitrided.
14C14S14	Parts where good machinability and surface finish are important and where higher sulphur content is undesirable.
25C12S14	Bolts, studs and other heat-treated parts of small section. The steel is suitable in either cold drawn, normalised or heat-treated condition for moderately stressed parts requiring more strength than mild steel.
40C10S18	Heat-treated bolts, engine shafts, connecting rods, gun carriage, small parts not subjected to high stresses and severe wear.
11C10S25	Lightly stressed parts not subjected to shock like nuts and studs. This steel is not recommended for case hardened parts.
40C15S12	Heat-treated axles, shafts, and small crankshafts. This steel is not recommended for forgings in which transverse properties are important.

2.11 ALLOY STEELS

Table 2.33 *BIS system of designation for alloy and high-alloy steels*

Alloy steels

The term 'alloy' steel is used for low and medium alloy steels containing total alloying elements not exceeding 10%. The designation of alloy steels consists of following quantities in the given order:

(i) Number indicating 100 times the average percentage of carbon; and

(ii) Chemical symbols for alloying elements each followed by the number for its average percentage content multiplied by a factor. The multiplying factor depends upon the alloying element. The values of this factor are as follows,

Element	Multiplying factor
Cr, Co, Ni, Mn, Si and W	4
Al, Be, V, Pb, Cu, Nb, Ti, Ta, Zr and Mo	10
P, S, N	100

(*Contd.*)

(*Contd.*)

In alloy steels, symbol 'Mn' for manganese is included only if the content of manganese is equal to or greater than 1%. The chemical symbols and their numbers are arranged in descending order of their percentage content. As an example, 25Cr4Mo2 is an alloy steel having average 0.25% carbon, 1% chromium and 0.2% molybdenum. Consider an alloy steel with following composition: Carbon = 0.12 to 0.18% Chromium = 0.50 to 0.80% Silicon = 0.15 to 0.35% Manganese = 0.40 to 0.60% The average percentage of carbon is 0.15%, which is denoted by the number (0.15×100) or 15. Percentage content of silicon and manganese is negligible and as such they are deleted from the designation. The significant element is chromium and its average percentage is 0.65. The multiplying factor for chromium is 4 and (0.65×4) is 2.6, which is rounded to 3. Therefore, the complete designation of steel is 15Cr3.
High alloy steels
The term 'high alloy steels' is used for alloy steels containing more than 10% of alloying elements. The designation of high alloy steels consists of following quantities in the given order: (i) Letter 'X'; (ii) Number indicating 100 times the average percentage of carbon; (iii) Chemical symbol for alloying elements each followed by the number for its average percentage content rounded off to nearest integer, and (iv) Chemical symbol to indicate specifically added element to attain desired properties, if any. For example, X15Cr25Ni12 is high alloy steel with 0.15% carbon, 25% chromium and 12% nickel. As a second example, consider a steel with the following chemical composition: Carbon = 0.15 to 0.25% Silicon = 0.10 to 0.50% Manganese = 0.30 to 0.50% Nickel = 1.5 to 2.5% Chromium = 16 to 20% The average content of carbon is 0.20% which is denoted by a number (0.20×100) or 20. The major alloying elements are chromium (average 18%) and nickel (average 2%). Hence, the designation of steel is X20Cr18Ni2.

Table 2.34 *Chemical composition of alloy steels (in percent)*

Designation	C	Si	Mn	Ni	Cr	Mo
20C15	0.16–0.24	0.10–0.35	1.3–1.7	—	—	—
27C15	0.22–0.32	0.10–0.35	1.3–1.7	—	—	—
37C15	0.32–0.42	0.10–0.35	1.3–1.7	—	—	—
35Mn6Mo3	0.3–0.4	0.10–0.35	1.3–1.8	—	—	0.20–0.35
35Mn6Mo4	0.3–0.4	0.10–0.35	1.3–1.8	—	—	0.35–0.55
40Cr4	0.35–0.45	0.10–0.35	0.6–0.9	—	0.90–1.2	—
42Cr4Mo2	0.38–0.45	0.10–0.35	0.6–0.9	—	0.90–1.2	0.15–0.30
15Cr13Mo6	0.1–0.2	0.15–0.35	0.4–0.7	0.3 max.	2.9–3.4	0.45–0.65

(*Contd.*)

(*Contd.*)

25Cr13Mo6	0.2–0.3	0.10–0.35	0.4–0.7	0.3 max.	2.9–3.4	0.45–0.65
40Cr13Mo10V2(*)	0.35–0.45	0.10–0.35	0.4–0.7	0.3 max.	3.0–3.5	0.90–1.1
42Cr6V1(**)	0.38–0.46	0.10–0.35	0.5–0.8	—	1.4–1.7	—
40Cr7A110Mo2 (***)	0.35–0.45	0.10–0.45	0.4–0.7	0.30 max.	1.5–1.8	0.10–0.25
40Ni14	0.35–0.45	0.10–0.35	0.5–0.8	3.2–3.6	0.30 max.	—
35Ni5Cr2	0.3–0.4	0.10–0.35	0.6–0.9	1.0–1.5	0.45–0.75	—
30Ni16Cr5	0.26–0.34	0.10–0.35	0.4–0.7	3.9–4.3	1.1–1.4	—
40Ni6Cr4Mo2	0.35–0.45	0.10–0.35	0.4–0.7	1.2–1.6	0.90–1.3	0.1–0.2
40Ni6Cr4Mo3	0.35–0.45	0.10–0.35	0.4–0.7	1.25–1.75	0.90–1.3	0.20–0.35
31Ni10Cr3Mo6	0.27–0.35	0.10–0.35	0.4–0.7	2.25–2.75	0.5–0.8	0.4–0.7
40Ni10Cr3Mo6	0.36–0.44	0.10–0.35	0.4–0.7	2.25–2.75	0.5–0.8	0.4–0.7

Note: (*) Vanadium = 0.15–0.25 Percent; (**) Vanadium = 0.07–0.12 Percent;
(***) Aluminium = 0.90–1.30 Percent.

Table 2.35 *Mechanical properties of alloy steels (for plates, sections, bars, billets and forgings in hot-rolled or normalised condition)*

Designation	Tensile strength (MPa or N/mm^2)	0.2 percent Proof stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Limiting ruling section (mm)
11C15	460–560 430–530	270 250	26 26	100 100–150
20C15	540–640 540–640 510–610 510–610 490–590	350 320 310 290 280	20 20 20 20 20	15 30 63 100 100–150
27C15	570–670 570–670 570–670 540–640 540–640	350 340 320 300 290	20 20 20 20 20	30 45 63 100 100–150

Table 2.36 *Mechanical properties of alloy steels (for cold drawn 1.5% manganese steel bars)*

Designation	Tensile strength (MPa or N/mm^2)	Elongation Min. (Percent)	Limiting ruling section (mm)
20C15	790 740 690	8 10 12	20 20–40 40–63

Table 2.37 *Mechanical properties of alloy steels (for bars and forgings in hardened and tempered condition – oil hardened)*

Designation	Tensile strength (MPa or N/mm^2)	0.2 percent Proof stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Izod Impact value Min. (J or N-m)	Brinell Hardness (HB)	Limiting ruling section (mm)
20C15	590–740 690–840	390 450	18 16	48 48	170–217 201–248	63 30
27C15	590–740 690–840	390 450	18 16	48 48	170–217 201–248	100 63
37C15	590–740 690–840 790–940 890–1040	390 490 550 650	18 18 16 15	48 48 48 41	170–217 201–248 229–277 255–311	150 100 30 15
35Mn6Mo3	690–840 790–940 890–1040 990–1140	490 550 650 750	14 12 12 10	55 50 50 48	201–248 229–277 255–311 285–341	150 100 63 30
35Mn6Mo4	790–940 890–1040 990–1140	550 650 750	16 15 13	55 55 48	229–277 255–311 285–341	150 100 63
40Cr4	690–840 790–940 890–1040	490 550 650	14 12 11	55 50 50	201–248 229–277 255–311	100 63 30
40Cr4Mo2	700–850 800–950 900–1050 1000–1150	490 550 650 750	13 12 11 10	55 50 50 48	201–248 229–277 255–311 285–341	150 100 63 30
15Cr13Mo6 and 25Cr13Mo6	690–840 790–940 890–1040 990–1140 1090–1240 1540 min.	490 550 650 750 830 1240	14 12 11 10 9 8	55 50 50 48 41 14	201–248 229–277 225–311 285–341 311–363 444 min.	150 150 150 150 100 63
40Cr13Mo10V2	1340 min. 1540 min.	1050 1240	8 8	21 14	363 min. 444 min.	63 30
42Cr6V1	880–1030 980–1180 1080–1280	690 780 880	12 11 10	68 58 49	265–310 295–350 320–380	100 30 15
40Cr7A110Mo2	690–840 790–940 890–1040	490 550 650	18 16 15	55 55 48	201–248 229–277 255–311	150 100 63
40Ni14	790–940 890–1040	550 650	16 15	55 55	229–277 255–311	100 63

(*Contd.*)

(*Contd.*)

35Ni5Cr2	690–840 790–940 890–1040	490 550 650	14 12 10	55 50 50	201–248 229–277 255–311	150 100 63(*)
30Ni16Cr5	1540 min.	1240	8	14	444 min.	150
40Ni6Cr4Mo2	790–940 890–1040 990–1140 1090–1240	550 650 750 830	16 15 13 11	55 55 48 41	229–277 255–311 285–341 311–363	150 100 63 30
40Ni6Cr4Mo3	790–940 890–1040 990–1140 1090–1240 1190–1340 1540 min.	550 650 750 830 930 1240	16 15 13 11 10 6	55 55 48 41 30 11	229–277 255–311 285–341 311–363 341–401 444 min.	150 150 100 63 30 30
31Ni10Cr3Mo6	890–1040 990–1140 1090–1240 1190–1340 1540 min.	650 750 830 930 1240	15 12 11 10 8	55 48 41 35 14	255–311 285–341 311–363 341–401 444 min.	150 150 100 63 63
40Ni10Cr3Mo6	990–1140 1090–1240 1190–1340 1540 min.	750 830 930 1240	12 11 10 8	48 41 35 14	285–341 311–363 341–401 444 min.	150 150 150 100

Note: (*) air hardening.

Table 2.38 *Mechanical properties of alloy steels (for wear resisting steel in hardened and tempered condition)*

Designation	Tensile strength (MPa or N/mm^2)	0.2 percent Proof stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)	Izod Impact value Min. (J or N-m)	Brinell Hardness (HB)	Limiting ruling section (mm)
55Cr3	890–1040 990–1140	650 730	12 10	35 17	255–311 285–341	63 63

Table 2.39 *Applications of alloy steels*

Designation	Applications
20C15 and 27C15	General-purpose steel for low stressed components.
37C15	Low stressed parts, automobile tubes and fasteners.
35Mn6Mo3 and 35Mn6Mo4	General engineering components such as crankshafts, bolts, wheel studs, axle shafts, levers, and connecting rods.

(*Contd.*)

(*Contd.*)

40Cr4	Gears, connecting rods, stub axles, steering arms, wear-resistant plates for earth moving and concrete handling machinery.
40Cr4Mo2	Axle shafts, crankshafts, connecting rods, gears, high tensile bolts and propeller shaft joints.
15Cr13Mo6 and 25Cr13Mo6	Components subjected to medium to high tensile stresses. In nitrided condition, the steel is used for crankshafts, cylinder liners for aero and automobile engines and machine parts requiring high surface hardness and wear resistance.
40Cr13Mo10V2	Highly stressed components. In nitrided condition, the steel is used for components subjected to heavy stress and severe wear.
42Cr6V1	Laminated, coil and volute springs.
40Cr7A110Mo2	Used for components requiring maximum surface hardness of a nitrided case combined with a fairly high core strength.
40Ni14	Components requiring excessively high toughness, heavy forgings, turbine blades, severely stressed screws, bolts and nuts. Used as 'cold-tough' steel for components working at low temperatures in refrigerators and compressors.
35Ni5Cr2	Crankshafts, connecting rods, gear shafts, chain parts, clutches, flexible shafts and camshafts.
30Ni16Cr5	Highly stressed gears subjected to stresses of the order of 1600 MPa and where minimum distortion in heat treatment is required.
40Ni6Cr4Mo2	High strength machine parts, collets, spindles, screws, high tensile bolts and studs, gears, pinions, axle shafts, tappets, crankshafts, connecting rods, boring bars and arbours.
40Ni6Cr4Mo3	Heavy duty gears and transmission components for heavy vehicles.
31Ni10Cr3Mo6	Highly stressed bolts and studs, shafts, crankshafts, connecting rods, gears, axles and mandrel bars.
40Ni10Cr3Mo6	Highly stressed gears, connecting rods, crankshafts, axles, die blocks and mandrel bars. Also used for components working at low temperature.

Table 2.40 *American and British systems of designation of carbon and alloy steels*

SAE and AISI systems
The numbering system for carbon and alloy steels is prescribed by Society of Automotive Engineers (SAE) of USA and American Iron and Steel Institute (AISI). It is based on chemical composition of the steel. The number is composed of four or five digits. The first two digits indicate the type or alloy classification. The last two or three digits give the carbon content. Since carbon is the most important element in steel affecting the strength and hardness, it is given proper weightage in this numbering system. The basic numbers for various types of steel are given in Table 2.41. For example, plain carbon steel has 1 and 0 as its first two digits. Thus, steel designated as 1045 indicates plain carbon steel with 0.45% carbon. Similarly, a nickel-chromium steel with 1.25% Ni, 0.60% Cr and 0.40% carbon is specified as SAE 3140.
The AISI number for steel is same as SAE number. In addition, there is a capital letter A, B, C, D or E that is prefixed to the number. These capital letters indicate the manufacturing process of steel. The meaning of these letters is as follows: A — Basic open-hearth alloy steel B — Acid Bessemer carbon steel C — Basic open-hearth carbon steel D — Acid open-hearth carbon steel E — Electric furnace alloy steel

(*Contd.*)

(*Contd.*)

British system
The British system designates steel in a series of numbers known as 'En' series. The En number of steel has no correlation either with the chemical composition such as carbon content and types of alloying element or mechanical properties such as ultimate tensile strength. For example, the number 3 in En3 steel has no relationship with carbon content, alloying element or strength of steel.

Table 2.41 *Basic numbering system of SAE and AISI steels*

Material	**SAE or AISI Number**
Carbon steels • plain carbon • free-cutting, screw stock	1xxx 10xx 11xx
Chromium steels • low chromium • medium chromium • corrosion and heat resisting	5xxx 51xx 52xxx 51xxx
Chromium-nickel-molybdenum steels	86xx
Chromium-nickel-molybdenum steels	87xx
Chromium-vanadium steels 1.00% Cr	6xxx 61xx
Manganese steels	13xx
Molybdenum steels • carbon-molybdenum • chromium-molybdenum • chromium-nickel-molybdenum • nickel-molybdenum; 1.75% Ni • nickel-molybdenum; 3.50% Ni	4xxx 40xx 41xx 43xx 46xx 48xx
Nickel-chromium steels • 1.25% Ni, 0.60% Cr • 1.75% Ni, 1.00% Cr • 3.50% Ni, 1.50% Cr	3xxx 31xx 32xx 33xx
Silicon-manganese steels 2.00% Si	9xxx 92xx
Nickel steels • 3.5% Ni • 5.0% Ni	2xxx 23xx 25xx

Table 2.42 *Overseas equivalent designations of steels*

BIS designation	En Number	SAE	AISI	DIN
Plain carbon steels				
7C4	2A	1010	C 1010	17210
10C4	32A	1012	C 1012	17155
30C8	5	1030	C 1030	—
45C8	43B	1045	C 1045	17200
50C4	43A	1049, 1050	C 1049, C 1050	—
55C8	43J, 9K	1055	C 1055	—
60C4	43D	1060	C 1060	17200
65C6	42B	1064	C 1064	17222
Free cutting steels				
10C8S10	—	1109	C 1109	—
14C14S14	7A, 202	1117, 1118	C 1117, C 1118	—
25C12S14	7	1126	C 1126	—
40C10S18	8M	1140	C 1140	—
40C15S12	15AM	1137	C 1137	—
Alloy steels				
40Cr4	18	5135	5135	—
40Ni14	22	2340	2340	—
35Ni5Cr2	111	3140	3140	1662
30Ni16Cr5	30A	—	—	—
40Ni6Cr4Mo2	110	4340	4340	17200
27C15	14B	1036	C 1036	17200
37C15	15, 15A	1041, 1036	C 1041, C 1036	17200
50Cr4V2	47	6150	6150	17221

Note: The above equivalents are tentative for the use of students. Detail comparison based on chemical composition should be carried out for correct equivalence.

2.12 STAINLESS AND HEAT-RESISTING STEELS

Table 2.43 *Mechanical properties of stainless and heat-resisting steels (for bars and flats in annealed, quenched or solution treated condition)*

Designation	Tensile strength Min. (MPa or N/mm^2)	0.2 percent Proof stress Min. (MPa or N/mm^2)	Elongation in 50 mm Min. (Percent)	Reduction of area Min. (Percent)	Brinell Hardness Max. (HB)
Chromium-Nickel steels					
X02Cr19Ni10	483	172	40	50	—
X04Cr19Ni9	517	207	40	50	—
X10Cr18Ni9	517	207	40	50	—
X04Cr18Ni10Ti	517	207	40	50	—
X04Cr18Ni10Nb	517	207	40	50	—
X04Cr17Ni12Mo2	517	207	40	50	—
X02Cr17Ni12Mo2	483	172	40	50	—
X04Cr17Ni12Mo2Ti	517	207	40	50	—
X04Cr19Ni13Mo3	517	207	40	50	—
X15Cr24Ni13	490	210	40	50	—
X20Cr25Ni20	490	210	40	50	—
X04Cr25Ni20	517	207	40	50	—
X10Cr17Mn6Ni4	515	275	40	45	217
X40Ni14Cr14W3Si2	785	345	35	40	269
Chromium steels					
X04Cr12	445	276	20	45	—
X12Cr12	483	276	20	45	—
X07Cr17	483	276	20	45	—
X20Cr13	—	—	—	—	241
X30Cr13	—	—	—	—	241
X40Cr13	600-750	—	—	—	225
X15Cr16Ni2	—	—	—	—	285
X108Cr17Mo	—	—	—	—	269
X15Cr25N	490	280	16	45	212

Table 2.44 *Mechanical properties of stainless and heat-resisting steels (for sheets, plates, strips, bars and flats in hardened and tempered condition)*

Designation	0.2 percent Proof stress Min. (MPa or N/mm^2)	Tensile strength (MPa or N/mm^2)	Elongation Min. (Percent)
X12Cr12	410	590–780	16
X20Cr13	490	690–880	14
X30Cr13	590	780–980	11
X15Cr16Ni2	640	830–1030	10

Table 2.45 *Applications of stainless and heat-resisting steels*

Designation	Applications
X20Cr13	Machine parts and springs.
X30Cr13	Structural parts with high strength and kitchen utensils.
X07Cr17	Decorative trim, oil-burner rings, cold headed fasteners, nitric acid storage tanks, nitrogen fixation equipment and annealing boxes.
X04Cr19Ni9	Chemical handling equipment, textile dyeing equipment, door bars, kitchen ware and coffee urns.
X04Cr18Ni10Ti and X04Cr18Ni10Nb	Aircraft engine exhaust manifolds, boiler shells, expansion joints, and high temperature chemical equipment.
X04Cr17Ni12Mo2 and X04Cr17Ni12Mo2Ti	High temperature chemical equipment for rayon, rubber and marine industries, photographic developing equipment, pulp handling equipment, steam jacketed kettles, coke plant equipment, food processing equipment, and edible oil-storage tanks.
X20Cr25Ni20	Heat exchangers, furnace doors, retorts, annealing boxes, gas turbine and aircraft engine exhaust systems, furnace conveyor belts, and hydrogenation equipment.
X40Ni14Cr14W3Si2	Inlet and exhaust valves of aeroengines.

2.13 CAST ALUMINIUM ALLOYS

Table 2.46 *System of designation of aluminium alloys*

Aluminium alloys are designated by a particular numbering system. The numbers given to alloying elements are as follows:

Aluminium	—	1	Magnesium	—	5
Copper	—	2	Magnesium silicide	—	6
Manganese	—	3	Zinc	—	7
Silicon	—	4	Other elements	—	8

(*Contd.*)

(*Contd.*)

Cast aluminium alloys are specified by a 'four digit' system while wrought alloys by a 'five digit' system. The meaning of these digits is as follows: First digit: It identifies the major alloying element. Second digit: It identifies the average percentage of the major alloying element halved and rounded off. Third, fourth and fifth digit: They identify the minor alloying elements in order of their decreasing percentage. For example, consider an aluminium alloy casting with 9.8% Cu, 1.0% Fe and 0.25% Mg. First digit : identification of copper : 2 Second digit: (9.8/2 = 4.9 or 5) : 5 Third digit: identification of iron : 8 Fourth digit: identification of magnesium : 5 Complete designation = 2585

Table 2.47 *Temper designations of aluminium and its alloys*

Designation	Meaning of temper designation
M	As manufactured (applicable to cast products only).
F	As fabricated (applicable to wrought products only).
O	Annealed (applicable to wrought products only).
H (*)	Strain hardened (applicable to wrought products only).
H1	Strain-hardened.
H2	Strain-hardened and partially annealed.
H3	Strain-hardened and stabilized.
T (**)	Thermally treated.
T1	Cooled from an elevated temperature shaping process and naturally aged to a substantially stable condition.
T2	Cooled from an elevated temperature shaping process, cold worked and naturally aged to a substantially stable condition.
T3	Solution heat-treated, cold worked and naturally aged to a substantially stable condition.
T4	Solution heat-treated and naturally aged to a substantially stable condition.
T5	Cooled from an elevated temperature shaping process and then artificially aged.
T6	Solution heat-treated and then artificially aged.
T7	Solution heat-treated and stabilised.
T8	Solution heat-treated, cold worked and then artificially aged.
T9	Solution heat-treated, artificially aged and then cold worked.
T10	Cooled from an elevated temperature shaping process, cold worked and artificially aged.

Note: (*) Symbol H is always followed by one or more digits, according to basic operation and final degree of strain hardening.
(**) Symbol T is always followed by one or more digits. Numerals 1 through 10 indicate specific sequence of basic treatments.

Table 2.48 *Chemical composition of cast aluminium and its alloys*

Desig- nation	Copper	Silicon	Magne- sium	Iron	Manga- nese	Nickel	Zinc	Lead	Tin	Titanium	Alumin- ium
1900	0.2	0.5	0.05	0.6	0.2	0.1	0.1	0.05	0.05	–	99 min.
1950	0.03	0.3	0.03	0.4	0.03	0.03	0.07	0.03	0.03	–	99.5 min.
2280	4–5	0.25	0.1	0.25	0.1	0.1	0.1	0.05	0.05	0.2–0.3	R
2285*	3.5–4.5	0.7	1.2–1.8	0.7	0.6	1.7–2.3	0.1	0.05	0.05	0.2	R
2550	9–11	2.5	0.2–0.4	1	0.6	0.5	0.8	0.1	0.1	0.2	R
4223	2–4	4–6	0.15	0.8	0.2–0.6	0.3	0.5	0.1	0.1	0.2	R
4223A	2.8–3.8	4–6	0.05	0.6	0.2–0.6	0.2	40.15	0.1	0.05	0.2	R
4225	1–1.5	4.5-6	0.3–6	0.8	0.5	0.3	0.5	0.2	0.1	0.2	R
4300	0.1	4.5–6	0.1	0.6	0.5	0.1	0.1	0.1	0.05	0.2	R
4323	3–5	5–7	0.1–0.3	1	0.2–0.6	0.3	2	0.2	0.1	0.2	R
4420	3–4	7.5–9.5	0.3	1.3	0.5	0.5	3	0.3	0.2	0.2	R
4423	1.5–2.5	6–8	0.3	0.8	0.2–0.6	0.3	1	0.2	0.1	0.2	R
4450	0.1	6.5–7.5	0.2–0.45	0.5	0.3	0.1	0.1	0.1	0.05	0.2	R
4520	0.7–2.5	9–11.5	0.3	1	0.5	0.5	2	0.3	0.2	0.2	R
4525	2–4	8.5–10.5	0.5-1.5	1.2	0.5	1	1	0.2	0.1	0.2	R
4528	1.75–2.5	8.5–9.5	0.15	0.4–0.6	0.8	0.8	0.5	0.1	0.1	0.2	R
4600	0.1	10–13	0.1	0.6	0.5	0.1	0.1	0.1	0.05	0.2	R
4600A	0.4	10–13	0.2	1	0.5	0.1	0.2	0.1	0.1	0.2	R
4628	1.75–2.5	11–12.5	0.3	0.7-1.1	0.5	0.3	1.5	0.05	0.1	0.2	R
4635	0.1	10–13	0.2–0.6	0.6	0.3–0.7	0.1	0.1	0.1	0.05	0.2	R
4652	0.7–1.5	10–12	0.8–1.5	1	0.5	0.7–1.5	0.5	0.1	0.1	0.2	R
5230	0.1	0.3	3–6	0.6	0.3–0.7	0.1	0.1	0.05	0.05	0.2	R
5500	0.1	0.25	9.5–11	0.4	0.1	0.1	0.1	0.05	0.05	0.2	R

Note: R = remainder, * Chromium content of this alloy should not exceed 2%.

Table 2.49 *Mechanical properties of cast aluminium and its alloys*

Designation	Condition	Tensile strength Min. (MPa or N/mm^2)		Elongation Min. (Percent)	
		Sand cast	Chill cast	Sand cast	Chill cast
1900	M	—	—	—	—
1950	M	—	—	—	—
2280	T4 T6	215 275	265 310	7 4	13 9
2285	T6	215	280	—	—
2550	M	—	170	—	—
4223	M T6	140 225	160 280	2 —	2 —
4223A	T4	—	245	—	8
4225	T4 T6	175 230	230 280	2 —	3 —
4300	M	120	140	3	4
4323	M	160	175	1	1
4420	M	—	180	—	1.5
4423	M	140	160	1	2
4450	M T5 T7 T6	135 160 160 225	160 190 225 275	2 1 2.5 —	3 2 5 —
4520	M	125	150	—	—
4525	T5	—	210	—	—
4528	M T5 T6	150 140 —	220 200 320	1 1.5 —	1.5 3 2
4600	M	165	190	5	7
4600A	M	165	190	5	5
4628	M	—	270	—	1.5
4635	M T5 T6	— 170 240	190 230 295	— 1.5 —	3 2 —
4652	T5 T6 T7	— 140 175	210 200 280	— — —	— — —
5230	M	140	170	3	5
5500	T4	275	310	8	12

Note: Sand castings are made in dry sand moulds while chill castings are produced from metallic moulds or dies.

Table 2.50 *Applications of cast aluminium and its alloys*

Designation of alloy	Applications
Sand, gravity die and investment casting alloys	
Alloy 4223 and 4423	These are the most versatile alloys with medium strength and good casting characteristics, suitable for general engineering applications, household fittings, office equipment, automobile parts, electrical tools and switchgear. These alloys can be cast in thin or thick sections. They are suitable for pressure tight castings.
Alloy 4450	This alloy is used for components requiring high mechanical strength and high resistance to wear such as cylinder blocks and cylinder heads in internal combustion engines. High proof strength, greater hardness, pressure tightness and dimensional stability on temperature variation are the important characteristics of this alloy. It is used for valve bodies, large fan blades and pneumatic tools.
Alloy 4528	This is a new alloy used in automobile industries because of good castability and low shrinkage. It can be used as cast as well as heat-treated condition.
Alloy 4600	Excellent castability and fluidity permits this alloy to be used in case of intricate and thin walled castings, water cooled manifolds, instrument cases, switch boxes, motor housings, and castings for marine applications, pumps, chemical and dye industries.
Alloy 4600A	The applications of this alloy are similar to those of alloy 4600. However, this alloy should not be used where high resistance to corrosion is required.
Pressure die casting alloys	
Alloy 4420	This alloy has a high mechanical strength, greater hardness and better machinability than alloys 4520 and 4600. The casting characteristics are very good.
Alloy 4520	Most of the pressure die castings requiring medium strength are produced from this alloy. This alloy is preferred for very thin walled castings. It has greater mechanical strength, hardness and machinability than alloy 4600.
Alloy 4600	This alloy is recommended for those castings for which the service operating conditions require a resistance to corrosion better than that offered by alloy 4420 or 4520.This alloy is most suitable for applications like direct contact with chemicals, foodstuffs and sea water. The ideal fluidity and freedom from hot-tearing of this alloy facilitates the production of complex castings of large surface area and their walls.
Alloy 4600A	The applications of this alloy are similar to those of alloy 4600. However, this alloy should not be used where high resistance to corrosion is required.
Alloy 4628	This alloy is widely used in automobile industries where high volume production die castings having thin walls are required. It has good castability, corrosion resistance, pressure tightness, low shrinkage and weldability. However, it has poor machinability.
Special application alloys	
Alloys 1950 and 1900	These alloys have high electrical conductivity. They are used in fittings required for electrical transmission. They are also suitable for food and chemical industries. However, they have poor castability due to high shrinkage and hot tearing.
Alloy 2280	This alloy possesses high strength and good ductility in fully heat-treated condition. It is used for reciprocating parts of engines, flywheel housing and propellers, artificial limbs and moulding boxes.
Alloy 2285	This alloy is known as 'Y' alloy and possesses good mechanical strength at elevated temperatures. It is used for pistons and cylinder heads.
Alloy 2550	This alloy is specifically suitable for castings in hydraulic equipment.

(*Contd.*)

(*Contd.*)

Alloy 4223A	This alloy possesses high strength and shock resistance. It is used for structural components and castings for heavy duty service in road transport vehicles.
Alloy 4225	This alloy has greater strength and hardness, which is maintained at temperatures up to 200ºC. It is good for pressure tightness.
Alloy 4300	It has good resistance to chemical and atmospheric corrosion. It is widely used for electrical household appliances such as steam irons, waffle irons, electric floor polishers and vacuum cleaners.
Alloy 4323	This alloy is specifically suitable for castings required to have a combination of high proof strength and hardness, good machinability and castability. It gets aged at room temperature. It is used for heavy duty automotive parts.
Alloy 4525	This alloy possesses good strength at elevated temperatures and low coefficient of expansion. It is used for low pressure die-castings such as scooter components.
Alloy 4635	This alloy has fluidity and corrosion resistance of alloy 4300 with higher mechanical strength and hardness. It is used for low pressure die castings such as scooter components.
Alloy 4652	This alloy has low thermal expansion and good strength at elevated temperatures. It is used for pistons of internal combustion engines.
Alloy 5230	This alloy has excellent corrosion resistance and surface finish. It is used for castings of marine, food processing and decorative applications. It is most suitable for decorative anodising. It is used for window and ornamental hardware.
Alloy 5500	This alloy has highest tensile strength, good shock resistance, excellent corrosion resistance and machinability. It is used for highly stressed castings in marine and mining equipments. However, this alloy is susceptible to stress corrosion at high temperatures and requires special foundry techniques.

2.14 WROUGHT ALUMINIUM ALLOYS

Table 2.51 *Chemical composition of wrought aluminium and its alloys for general engineering purposes (for bars, rods and extruded sections)*

Designation	Aluminium	Copper	Magnesium	Silicon	Iron	Manganese	Zinc	Titanium	Chromium
19000	99 min.	0.1	—	0.5	0.6	0.1	—	—	—
19500	99.5 min.	0.05	—	0.3	0.4	0.05	—	—	—
19600	99.6 min.	0.05	—	0.25	0.35	0.03	—	—	—
24345	R	3.8–5	0.2–0.8	0.5–1.2	0.7	0.3–1.2	0.2	0.3	0.3
24534	R	3.5–4.7	0.4–1.2	0.2–0.7	0.7	0.4–1.2	0.2	0.3	—
43000	R	0.1	0.2	4.5–6	0.6	0.5	0.2	—	—
46000	R	0.1	0.2	10–13	0.6	0.5	0.2	—	—
52000	R	0.1	1.7–2.6	0.6	0.5	0.5	0.2	0.2	0.25
53000	R	0.1	2.8–4	0.6	0.5	0.5	0.2	0.2	0.25
54300	R	0.1	4–4.9	0.4	0.7	0.5—1	0.2	0.2	0.25
63400	R	0.1	0.4–0.9	0.3–0.7	0.6	0.3	0.2	0.2	0.1

(*Contd.*)

(*Contd.*)

64423	R	0.5–1	0.5–1.3	0.7–1.3	0.8	1	—	—	—
64430	R	0.1	0.4–1.2	0.6–1.3	0.6	0.4–1	0.1	0.2	0.25
65032	R	0.15–0.4	0.7–1.2	0.4–0.8	0.7	0.2–0.8	0.2	0.2	0.15–0.35
74530	R	0.2	1–1.5	0.4	0.7	0.2–0.7	4–5	0.2	0.2
76528	R	1.2–2	2.1–2.9	0.5	0.7	0.3	5.1–6.1	0.2	0.2–0.28

(R = remainder)

Table 2.52 *Mechanical properties of wrought aluminium and its alloys for general engineering purposes (for bars, rods and extruded sections)*

Designation	Condition	Size (diameter or minor cross-sectional dimension) (mm)		0.2 percent Proof stress (MPa or N/mm^2)		Tensile strength (MPa or N/mm^2)		Elongation Min. (Percent)
		over	up to	Min.	Max.	Min.	Max.	
19000	M (*)	—	—	20	—	65	—	18
	O	—	—	—	—	—	110	25
19500	M	—	—	18	—	65	—	23
	O	—	—	—	—	—	100	25
19600	M	—	—	17	—	65	—	23
	O	—	—	—	—	—	95	25
24345	M	—	—	90	—	150	—	12
	O	—	—	—	175	—	240	12
	W	—	10	225	—	375	—	10
		10	75	235	—	385	—	10
		75	150	235	—	385	—	8
		150	200	225	—	375	—	8
	WP	—	10	375	—	430	—	6
		10	25	400	—	460	—	6
		25	75	420	—	480	—	6
		75	150	405	—	460	—	6
		150	200	380	—	430	—	6
24534	M	—	—	90	—	150	—	12
	O	—	—	—	175	—	240	12
	W	—	10	220	—	375	—	10
		10	75	235	—	385	—	10
		75	150	235	—	385	—	8
		150	200	225	—	375	—	8
43000	M	—	15	—	—	90	—	18
	O	—	15	—	—	—	130	18
46000	M	—	15	—	—	100	—	10
	O	—	15	—	—	—	150	12
52000	M	—	150	70	—	160	—	14
	O	—	150	—	—		240	18

(*Contd.*)

(*Contd.*)

53000	M	—	50	100	—	215	—	14
		50	150	100	—	200	—	14
	O	—	150	—	—	—	260	16
54300	M	—	150	130	—	265	—	11
	O	—	150	125	—	—	350	13
63400	M	—	—	—	—	110	—	13
	O	—	—	—	—	—	130	18
	W	—	150	80	—	140	—	14
		150	200	80	—	125	—	13
	P	—	25	110	—	150	—	7
	WP	—	150	150	—	135	—	7
		150	200	130	—	150	—	6
64423	M	—	—	—	—	120	—	10
	O	—	—	—	125	—	215	15
	W	—	—	155	—	266	—	13
	WP	—	—	265	—	330	—	7
64430	M	—	—	80	—	110	—	12
	O	—	—	—	—	—	150	16
	W	—	150	120	—	185	—	14
		150	200	100	—	170	—	12
	WP	—	5	255	—	295	—	7
		5	75	270	—	310	—	7
		75	150	270	—	295	—	7
		150	200	240	—	280	—	6
65032	M	—	—	50	—	110	—	12
	O	—	—	—	115	—	150	16
	W	—	150	115	—	185	—	14
		150	200	100	—	170	—	12
	WP	—	150	235	—	280	—	7
		150	200	200	—	245	—	6
74530	W (**)	—	6	220	—	255	—	9
		6	75	230	—	275	—	9
		75	150	220	—	265	—	9
	WP	—	6	245	—	285	—	7
		6	75	260	—	310	—	7
		75	150	245	—	290	—	7
76528	O	All	Sizes	—	—	—	290	10
	WP	—	6	430	—	500	—	6
		6	75	455	—	530	—	6
		75	150	430	—	500	—	6

Note: (i) Symbols for condition: M: As manufactured; O: Annealed;
W: Solution treated; WP: Solution and precipitation treated. (ii) (*) Properties of 'M' temper are only typical values and are for information only. (iii) (**) Naturally aged for 30 days.

Table 2.53 *Applications of wrought aluminium and its alloys*

Designation	Applications
19000	This alloy is used for paneling and moulding, refrigeration tubing, equipment for food, chemical and brewing industrial packaging, cooking utensils, deep drawn parts and electrical appliances.
19500 and 19600	These alloys are used for corrosion resistant cladding on stronger alloys, food, chemical and brewing and processing equipment, tanks and pipes, marine fittings, reflectors, pressed and anodised utility items and jewellery.
24345	This alloy is used for heavy duty forgings and high strength structures.
24534	This alloy is used for stressed parts in aircrafts and other structures.
43000 and 46000	These alloys are used for filler wire for welding.
52000	This alloy is used for paneling and structures, sheet metal work, domestic appliances and lining for bottom of boats.
53000	This alloy is used for ship building, rivets, pressure vessels, processing tanks, cryogenic applications and welded structures.
54300	This alloy is used for welded structures, cryogenic applications, marine structures, rail and road tank cars, rivets and missile components.
63400	This alloy is used for architectural applications such as window and door frames, wall facings, partitions and hand rails.
63401	This alloy is used for bus bar applications.
64401	This alloy is used for conductor applications.
64423	This alloy is used where very high strength and good machinability are the main requirements such as missile components.
64430 and 65032	These alloys are used for structural applications in vehicles and cargo containers, milk containers, bridges, cranes, roof trusses, rivets, deep drawn containers and flooring.
74530	This alloy is used for structural applications requiring welding such as bridges, pressure vessels and rail coaches.
76528	This alloy is used for stressed structural applications working at low temperatures.

Table 2.54 *Mechanical properties of wrought aluminium and aluminium alloy rivet stock for general engineering purposes*

Designation	Condition	Diameter of rod (mm)	Tensile strength Min. (MPa or N/mm^2)
19000	H2	0–12	110
24345	W	0–12	390
53000	O or M	0–25	215
	O	0–25	245
55000	O or M	0–25	245
	O	0–25	280

(*Contd.*)

(*Contd.*)

64430	W	0–25	200
65032	W	0–25	185

Note: Symbols for Condition = M: As manufactured; O: Annealed; W: Solution treated; H2: Strain-hardened and annealed

Table 2.55 *Applications of wrought aluminium and aluminium alloy rivet stock*

Designation	Applications
19000	Rivets used in equipments for food, chemical, brewing and processing, cooking utensils, architectural and builder's hardware and in aircraft manufacture.
24345	Rivets used in highly stressed structures and aircraft structures.
53000	Rivets used in ship building, pressure vessels and other processing tanks and aircraft manufacture.
55000	Rivets used in ship building, aircraft manufacture and other applications requiring moderately high strength and good corrosion resistance.
64430	Rivets used in structures of all kinds, such as road and rail transport vehicles, bridges, cranes, roof trusses, cargo container and flooring.
65032	Rivets used in structures of all kinds, such as road and rail transport vehicles, bridges, cranes, roof trusses, cargo container, milk containers, deep drawn containers and flooring.

2.15 ALUMINIUM ALLOYS FOR BEARINGS

Table 2.56 *Properties of aluminium alloy castings for bearings (at 27°C)*

Designation	Condition	Rockwell Hardness Min. (H)
8482	Aged	102
8328	Aged	85

Note: (i) Alloy 8482 is aged at 210°C for 10 h.
(ii) Alloy 8328 is aged at 227°C for 7 h.

Table 2.57 *Properties of wrought aluminium alloy strip for bearings*

Designation	Condition	Rockwell Hardness Min. (H)	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)
83428	Cold reduction (*)	99	220	7
	As rolled and Annealed	67	145	29
89200	As Manufactured	—	190	10

Note: (*) Typical cold reduction of 28% is given to attain the properties.

2.16 FREE CUTTING LEADED BRASS

Table 2.58 *Chemical composition of free-cutting leaded Brass bars, rods and sections*

Constituent	Percent		
	Grade 1	Grade 2	Grade 3
Copper	56–59	60–63	60–63
Lead	2–3.5	2.5–3.7	0.5–1.5
Iron (max.)	0.35	0.35	0.2
Impurities (max.)	0.7	0.5	0.5
Zinc	Remainder	Remainder	Remainder

Table 2.59 *Mechanical properties of free-cutting leaded brass bars, rods and sections*

Condition	Size	Tensile strength Min. (MPa or N/mm^2)			Elongation Min. (Percent)		
		Grade 1	Grade 2	Grade 3	Grade 1	Grade 2	Grade 3
Annealed (O)	6–25	345	355	315	12	15	22
	25–50	315	305	285	17	20	27
	over 50	285	275	255	22	25	32
Half Hard (HB)	6–12	405	395	355	4	7	8
	12–25	395	385	345	6	10	12
	25–50	355	345	305	12	15	22
	over 50	325	315	285	17	20	27
Hard (HD)	6–12	550	500	460	–	–	–
	12–25	490	485	400	4	4	4

Table 2.60 *Chemical composition of leaded brass ingots and castings*

Constituent	Percent			
	Grade LCB 1		Grade LCB 2	
	Ingots	Castings	Ingots	Castings
Copper plus incidental nickel	70–77	70–80	63–67	63–79
Lead	2–5	2–5	1–3	1–3
Tin	1–3	1–3	1.5 max.	1.5 max.
Iron (max.)	0.5	0.75	0.5	0.75
Aluminium (max.)	0.01	0.01	0.01	0.01
Zinc	Remainder	Remainder	Remainder	Remainder

Note: (i) The castings of grades LCB 1 and LCB 2 have tensile strength of about 180 MPa and about 12 percent elongation.
(ii) Sand castings of grades LCB 1 and LCB 2 are used for small size valves, cocks, plumbing fittings and other general engineering applications.

2.17 ALUMINIUM BRONZE

Table 2.61 *Chemical composition of aluminium bronze rods and bars*

Constituent	Grade		
	CuAl8Fe3	CuAl10Fe3	CuAl10Fe5Ni5
Aluminium	6.5–8	8.5–10	8.5–11
Iron Nickel	2–3.5 0.5 max.	4.0 max. (iron + nickel)	4–6 4–6
Manganese (max.)	0.5	0.5	0.05
Copper	Remainder	Remainder	Remainder
Tin (max.)	0.1	0.1	0.1
Lead (max.)	0.05	0.05	0.05
Zinc (max.)	0.4	0.4	0.4
Silicon (max.)	0.15	0.1	0.1
Magnesium (max.)	0.05	0.05	0.05
Total impurities (max.)	0.5	0.5	0.5

Table 2.62 *Mechanical properties of aluminium bronze rods and bars*

Designation	Section-thickness (mm)		Condition	Tensile strength Min. (MPa or N/mm^2)	0.2 percent Proof stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)
	Over	Up to and including				
CuAl8Fe3	—	6	Annealed	460	190	30
	6	70	As manufactured	520	220	30
	70	—	” ”	460	190	30
CuAl10Fe3	—	6	” ”	520	215	22
CuAl10Fe5Ni5	6	70	” ”	700	400	12
	70	120	” ”	700	350	14
	120	—	” ”	650	320	12

Table 2.63 *Applications of aluminium bronze rods and bars*

Designation	Applications
CuAl8Fe3	The material of this grade has good strength combined with corrosion resistance. Iron content increases strength and refines grain. It is used for structural components, machine parts, nuts, bolts and threaded components.
CuAl10Fe3	The material of this grade is used for corrosion resistant parts, marine hardware and heat application.
CuAl10Fe5Ni5	The material of this grade is especially adopted to hot rolled rods, bars and forgings. It is used in marine shipping applications.

Table 2.64 *Chemical composition of aluminium bronze ingots and castings*

Constituent	AB 1	AB 2
Aluminium	8.5–10.5	8.8–10.0
Iron	1.5–3.5	4.0–5.5
Manganese	1 max.	1.5 max.
Nickel	1 max.	4.0-5.5
Zinc	0.5 max.	0.5 max.
Tin	0.1 max.	0.1 max.
Lead	0.05 max.	0.05 max.
Silicon	0.25 max.	0.10 max.
Magnesium	0.05 max.	0.05 max.
Copper	Remainder	Remainder

Table 2.65 *Mechanical properties of aluminium bronze ingots and castings*

Method of casting	Property	AB 1	AB 2
Sand cast	Ultimate tensile strength min. (MPa or N/mm^2)	500	640
	0.2 percent permanent set stress Min. (MPa or N/mm^2)	170	250
	Elongation min. (Percent)	18	13
Chill cast	Ultimate tensile strength min. (MPa or N/mm^2)	540	650
	0.2 percent Permanent set stress min. (MPa or N/mm^2)	200	250
	Elongation min. (Percent)	18	13

2.18 ALUMINIUM SILICON BRONZE

Table 2.66 *Mechanical properties of aluminium silicon bronze rods and bars*

Diameter of cross-section (mm)	Condition	Yield strength Min. (MPa or N/mm^2)	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)
6–10	As manufactured	310	620	9
10–25	” ”	310	585	12
25–50	” ”	290	550	12
50–75	” ”	240	515	15

2.19 PHOSPHOR BRONZE

Table 2.67 *Mechanical properties of phosphor bronze rods and bars*

Thickness of Cross section (mm)	Condition	0.2 percent Proof stress Min. (MPa or N/mm^2)	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)
6–18	As manufactured	410	500	12
18–38	” ”	380	460	12
38–70	” ”	315	380	16
70–100	” ”	235	315	20
100–120	” ”	118	275	22
Over 120	” ”	80	255	25

Table 2.68 *Chemical composition of phosphor bronze ingots and castings*

Constituent	Percent			
	Grade 1	Grade 2	Grade 3	Grade 4
Tin	6–8	10 min.	6.5–8.5	9–11
Phosphorus	0.3–0.5	0.5 min.	0.3 min.	0.15 min.
Lead	0.25 max.	0.25 max.	2–5	0.25 max.
Zinc	0.5 max.	0.05 max.	2 max.	0.05 max.
Silicon	0.02 max.	0.02 max.	—	—
Iron	0.3 max.	0.1 max.	—	—
Aluminium	0.01 max.	0.01 max.	—	—
Antimony	0.1 max.	—	—	—
Nickel	0.7 max.	0.1 max.	1.0 max.	0.25 max.
Total impurities	1.2 max.	0.6 max.	0.5 max.	0.8 max.
Copper	Remainder	Remainder	Remainder	Remainder

Table 2.69 *Mechanical properties of phosphor bronze ingots and castings*

Grade	Method of casting	Tensile strength Min. (MPa or N/mm^2)	Elongation Min. (Percent)
Grade 1	Sand cast	190	3
	Chill cast	205	5
	Continuously cast	275	8
Grade 2	Sand cast	220	3
	Chill cast	310	2
	Continuously cast	360	7
Grade 3	Sand cast	190	3
	Chill cast	220	2
	Continuously cast	270	5

(*Contd.*)

(*Contd.*)

Grade 4	Sand cast	230	6
	Chill cast	270	5
	Continuously cast	310	9

2.20 TIN BRONZE

Table 2.70 *Mechanical properties of tin bronze ingots and castings*

Method of casting	Tensile strength Min. (MPa or N/mm^2)	0.2 percent Proof stress Min. (MPa or N/mm^2)	Elongation Min. (Percent)
Sand casting	260	120	13
Chill casting	210	120	3

2.21 SILICON BRONZE

Table 2.71 *Mechanical properties of silicon bronze ingots and castings*

Tensile strength min. (MPa or N/mm^2)	310
Elongation min. (Percent)	20

Note: Silicon bronze castings are used in sewage disposal equipment, chemical process equipment, marine hardware and anti-corrosive pipe fittings such as valves and pumps. It provides a good substitute for tin and tin-zinc bronzes.

2.22 ZINC DIE-CASTINGS

Table 2.72 *Chemical composition of zinc base alloy die-castings*

Constituent	Percent	
	Alloy Zn Al 4	Alloy Zn Al 4 Cu1
Aluminium	3.8–4.3	3.8–4.3
Copper	—	0.75–1.25
Magnesium	0.03–0.06	0.03–0.06
Zinc	Remainder	Remainder

Table 2.73 *Mechanical properties of zinc base alloy die-castings*

Property	Alloy	Original value (5 weeks after casting)
Tensile strength (MPa or N/mm^2) (strain rates 6.3 mm/minute cross head speed)	Zn Al 4	286
	Zn Al 4 (S)	273
	Zn Al 4 Cu1	335
	Zn Al 4 Cu1 (S)	312
Elongation (percent) (on 50.8 mm × 6.35 mm diameter)	Zn Al 4	15
	Zn Al 4 (S)	17
	Zn Al 4 Cu1	9
	Zn Al 4 Cu1 (S)	10
Impact strength (J or N-m) (on un-notched Charpy test piece of 6.35 × 6.35 mm section)	Zn Al 4	57
	Zn Al 4 (S)	61
	Zn Al 4 Cu1	58
	Zn Al 4 Cu1 (S)	60
Brinell Hardness (HB)	Zn Al 4	83
	Zn Al 4 (S)	69
	Zn Al 4 Cu1	92
	Zn Al 4 Cu1 (S)	83

Note: (i) (S) = stabilised condition.
(ii) Alloy Zn Al 4 is usually selected for engineering applications. It has greater dimensional stability, better response to stabilizing treatment, and better corrosion resistance than Zn Al 4 Cu1. It is also more ductile and retains its impact strength during prolonged service at 100ºC. (iii) Alloy Zn Al 4 Cu1 is sometimes used on account of its slightly greater strength and hardness.

Table 2.74 *Mechanical and physical properties of zinc base alloy die-castings (Other than those given in Table 2.73)*

Property	Zn Al 4	Zn Al 4 Cu1
Compressive strength (MPa or N/mm^2)	415	600
Modulus of rupture (MPa or N/mm^2)	650	720
Shearing strength (MPa or N/mm^2)	215	250
Melting point (ºC)	387	388
Solidification point (ºC)	382	379
Casting contraction accepted mean value (mm/mm)	0.006	0.006

(*Contd.*)

(*Contd.*)

Specific heat capacity (J/kg ºC)	420	420
Thermal conductivity (W/m ºC) at 18ºC	113	109
Thermal expansion (μm/m/ºC) (20ºC to 100ºC)	27	27
Specific gravity	6.7	6.7

2.23 PLASTICS

Table 2.75 *Mechanical properties of plastics*

Material	Specific gravity	Tensile strength (MPa or N/mm^2)	Compressive strength (MPa or N/mm^2)
Polyamide	1.04–1.14	70	50–90
Low density polythene	0.92–0.94	7–20	—
Acetal	1.41–1.42	55–70	—
Polyurethane	1.21–1.26	35–60	25–80
Teflon	2.14–2.20	10–25	10–12
Phenolic	1.30–1.90	30–70	—

Table 2.76 *Applications of plastics*

Material	Application
Polyamide (Nylon, Capron Nylon, Zytel, Fosta)	Polyamide is a thermoplastic material. It has excellent toughness and wear resistance. The coefficient of friction is low. It is used for gears, bearings, conveyor rollers and automotive cooling fans.
Low density Polyethylene (Polythene)	It is a thermoplastic material. It is flexible and tough, light in weight, easy to process and a low-cost material. It is used for gaskets, washers and pipes.
Acetal (Delrin)	It is a thermoplastic material and a strong engineering material with exceptional dimensional stability. It has low coefficient of friction and high wear resistance. It is used for self-lubricating bearings, cams and running gears.
Polyurethane (Duthane, Texin)	It is a thermoplastic material and a tough, abrasion-resistant and impact-resistant material. It has good dimensional properties and self-lubricating characteristics. It is used for bearings, gears, gaskets and seals.
Polytetrafluoroethylene (Teflon)	It is thermoplastic material. It has low coefficient of friction and self-lubricating characteristics. It can withstand a wide range of temperature from –260 to +250°C. It is ideally suitable for self-lubricating bearings.
Phenolic	It is thermosetting plastic material. It is low cost with good balance of mechanical and thermal properties. It is used in clutch and brake linings as filler material. Glass reinforced phenolic is used for pulleys and sheaves.

Manufacturing Considerations in Design

3

3.1 DESIGN OF CASTINGS

Table 3.1 *Design of grey iron castings*

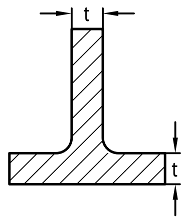
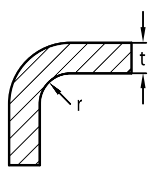
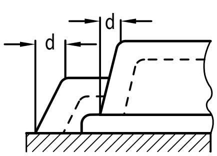
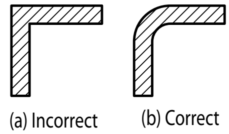
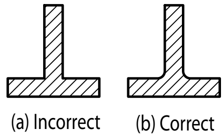
<table>
<tr><td rowspan="1"></td><td>Minimum section thickness:
The minimum section thickness (t) depends upon the process of casting such as sand casting, permanent mould casting or die casting. The minimum section thickness for grey cast iron component is about 7 mm for parts up to 500 mm long, which gradually increases to 20 mm for large and heavy castings.</td></tr>
<tr><td></td><td>Corner radius:
The values of corner radii (r) for different section thickness are as follows:
<table>
<tr><th>Wall thickness (t) (mm)</th><th>Inside corner radius (r) (Min)(mm)</th></tr>
<tr><td>0–30</td><td>10</td></tr>
<tr><td>30–50</td><td>15</td></tr>
<tr><td>50–80</td><td>20</td></tr>
<tr><td>80–120</td><td>30</td></tr>
</table></td></tr>
<tr><td></td><td>Provision of draft:
Patterns without draft make the mould difficult and costly. The minimum draft (d) of 3º should be provided.</td></tr>
</table>

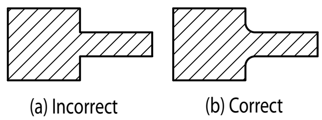
Table 3.2 *Do's and Don'ts of casting design*



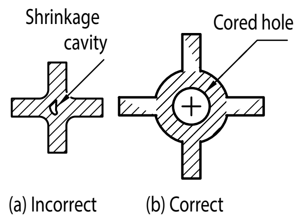
Adequate corner radius reduces stress concentration.



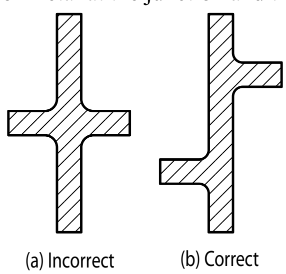
Adequate fillet radius reduces stress concentration.



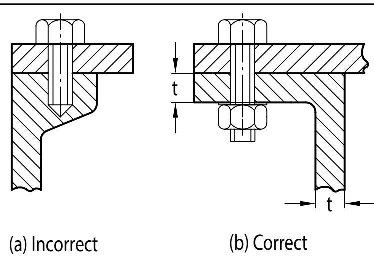
Gradual change in section thickness reduces stress concentration.



Cored opening in ribs reduce concentration of metal at the junction and the resulting formation of shrinkage cavities.



Staggered ribs reduce concentration of metal at the junction and the resulting formation of shrinkage cavities.



The wall adjacent to drilled hole should have thickness equal to thickness of main casting in order to improve the strength.

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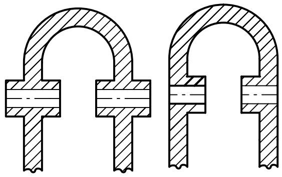
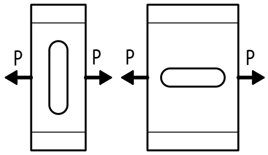


Outside bosses should be avoided to facilitate straight pattern draft.



Oval shaped holes are preferred with larger dimension along the direction of force.

Table 3.3 *Recommended wall thickness for grey iron castings*

Equivalent overall size N (m)	Thickness of outer walls (mm)	Thickness of inner walls (mm)
0.4	6	5
0.75	8	6
1.0	10	8
1.5	12	10
1.8	14	12
2.0	16	12
2.5	18	14
3.0	20	16
3.5	22	18
4.5	25	20

Note: (i) The equivalent overall size 'N' is given by,

$$N = \left(\frac{2l + b + h}{4}\right) \tag{3.1}$$

where l, b and h are length, width and height of a box like casting which is considered to be equivalent to the actual casting with respect to foundry conditions. (ii) Inner walls cool more slowly than outer walls and therefore, to ensure simultaneous cooling, the thickness of inner walls is taken as 0.8 times of the thickness of outer walls.

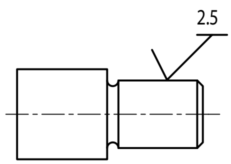
Table 3.4 *Linear shrinkage values of casting alloys*

Casting alloy	Linear shrinkage (Percent)
Grey cast iron	1.0–1.2
High tensile cast iron	1.5–1.8
Carbon steel casting	1.8–2.0
Alloy steel casting	1.8–2.5
Phosphor bronze	0.6–0.8
Tin bronze	1.3–1.6
Aluminium bronze	2.0–2.2
Aluminium silicon bronze	1.0–1.2
Magnesium alloys	1.5–1.7

Note: The shrinkage of casting is taken into account of by increasing the dimensions of mould using Patternmaker's Rule.

3.2 SURFACE ROUGHNESS

Table 3.5 *Surface roughness*

	The symbol for surface roughness consists of two legs of unequal length inclined at approximately 60º to the line representing the surface and the number indicating the surface roughness (rms) in microns.
Typical surface roughness values of common machining methods	
Machining Method	**Typical Surface Roughness (microns or μm)**
Turning, shaping and milling	12.5–1.0
Boring	6.5–0.5
Drilling	6.25–2.5
Reaming	2.5–0.5
Surface grinding	6.25–0.5
Cylindrical grinding	2.5–0.25
Honing and lapping	0.5–0.05
Polishing and buffing	0.5–0.05
Typical surface roughness values of common machine elements	
Machine Part	**Typical Surface Roughness (Microns or μm)**
Gear shafting and bores	1.5
Bronze bearings	0.75
Splined shafts, O-ring grooves, gear teeth and ball bearings	0.40
Cylinder bores and pistons	0.30
Crankshafts, connecting rods, cams and hydraulic cylinders	0.20

(ISO: 1302) (1 micron = 0.001 mm)

Table 3.6 *Surface roughness grades*

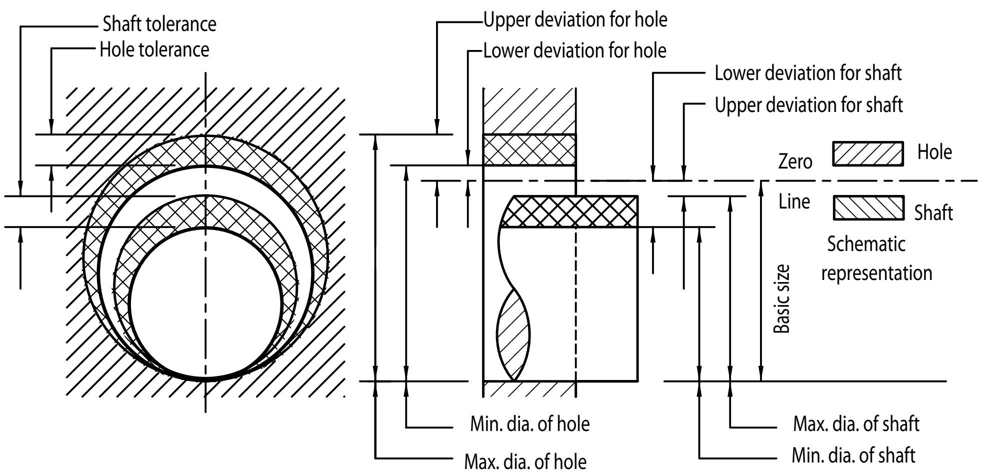
Surface roughness (microns or μm)	Grade number
50	N 12
25	N 11
12.5	N 10
6.3	N 9
3.2	N 8
1.6	N 7
0.8	N 6
0.4	N 5
0.2	N 4
0.1	N 3
0.05	N 2
0.025	N 1

(ISO: 1302)

3.3 TOLERANCES

Table 3.7 *Tolerances—Basic concepts*

Basic Nomenclature



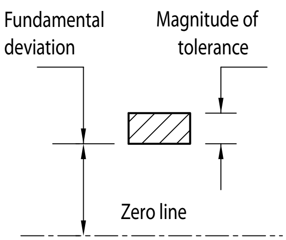
Notations

Upper deviation of hole = ES (*Ecart Superieur*)
Lower deviation of hole = EI (*Ecart Interieur*)
Upper deviation of shaft = es
Lower deviation of shaft = ei

(*Contd.*)

(*Contd.*)

Basic Systems of Giving Tolerance	
Unilateral system: In unilateral system, one tolerance is zero, while the other takes care of all permissible variation in basic size. For example, $100^{+0.05}_{+0.00}$ or $100^{+0.00}_{-0.05}$	*Bilateral system*: In bilateral system, the variations are given in both directions from basic size. For example, $100^{\pm 0.025}$ or $100^{+0.05}_{-0.05}$
Hole-basis system: In hole-basis system, the different clearances or interferences are obtained by associating various shaft sizes with single hole size, whose lower deviation is zero. In this case, the size of the hole is the basic size, and the clearance or interference is applied to the shaft dimension. This system is denoted by the symbol 'H'. Hole-basis system is widely used in practice because the holes can be machined by standard drills or reamers having fixed dimensions, while the shafts can be machined and ground to any given dimension.	*Shaft-basis system*: In shaft-basis system, the different clearances or interferences are obtained by associating various hole sizes with single shaft size, whose upper deviation is zero. In this case, the size of the shaft is basic size, while the clearance or interference is applied to the hole dimension. This system is denoted by the symbol 'h'. Shaft-basis system is popular in industries using semi-finished or finished shafting, such as bright bars as raw material.

Description of Tolerance	
	According to the system recommended by the Bureau of Indian Standards, the tolerance is specified by an alphabet, capital or small, followed by a number, e.g., H7 or g6. The description of tolerance consists of two parts—'fundamental deviation' and 'magnitude of tolerance'. The fundamental deviation gives the location of the tolerance zone with respect to the zero line. It is indicated by an alphabet—capital letters for holes and small letters for shafts. There are 25 fundamental deviations that are used in practice and are designated as A, B, C, etc., for the internal features (for holes) and a, b, c, etc., for the external features (for shafts). The magnitude of tolerance is designated by a number, called the grade. The grade of tolerance is defined as a group of tolerances, which are considered to have the same level of accuracy for all basic sizes. There are eighteen grades of tolerances with designations—IT1, IT2,………IT17, and IT18. The letters of symbol IT stand for 'International Tolerance' grade. The tolerance for a shaft of 50 mm diameter as the basic size, with the fundamental deviation denoted by the letter 'g' and the tolerance of grade 7 is written as 50 g7.

(ISO: 286-1)

Note: Refer to Tables 3.8 to 3.20 for tolerances of holes and shafts.

3.4 TOLERANCES FOR HOLES

Table 3.8 *Limit deviations for holes of sizes up to 500 mm (Holes A, B and C)*
(Values of deviations in microns or mm) (1 micron = 0.001 mm)

Basic size (mm)		A		B		C			
Above	Up to and including	9	10	9	10	8	9	10	11
–	3	+295 +270	+310 +270	+165 +140	+180 +140	+74 +60	+85 +60	+100 +60	+120 +60
3	6	+300 +270	+318 +270	+170 +140	+188 +140	+88 +70	+100 +70	+118 +70	+145 +70
6	10	+316 +280	+338 +280	+186 +150	+208 +150	+102 +80	+116 +80	+138 +80	+170 +80
10	18	+333 +290	+360 +290	+193 +150	+220 +150	+122 +95	+138 +95	+165 +95	+205 +95
18	30	+352 +300	+384 +300	+212 +160	+244 +160	+143 +110	+162 +110	+194 +110	+240 +110
30	40	+372 +310	+410 +310	+232 +170	+270 +170	+159 +120	+182 +120	+220 +120	+280 +120
40	50	+382 +320	+420 +320	+242 +180	+280 +180	+169 +130	+192 +130	+230 +130	+290 +130
50	65	+414 +340	+460 +340	+264 +190	+310 +190	+186 +140	+214 +140	+260 +140	+330 +140
65	80	+434 +360	+480 +360	+274 +200	+320 +200	+196 +150	+224 +150	+270 +150	+340 +150
80	100	+467 +380	+520 +380	+307 +220	+360 +220	+224 +170	+257 +170	+310 +170	+390 +170
100	120	+497 +410	+550 +410	+327 +240	+380 +240	+234 +180	+267 +180	+320 +180	+400 +180
120	140	+560 +460	+620 +460	+360 +260	+420 +260	+263 +200	+300 +200	+360 +200	+450 +200
140	160	+620 +520	+680 +520	+380 +280	+440 +280	+273 +210	+310 +210	+370 +210	+460 +210
160	180	+680 +580	+740 +580	+410 +310	+470 +310	+293 +230	+330 +230	+390 +230	+480 +230
180	200	+775 +660	+845 +660	+455 +340	+525 +340	+312 +240	+355 +240	+425 +240	+530 +240
200	225	+855 +740	+925 +740	+495 +380	+565 +380	+332 +260	+375 +260	+445 +260	+550 +260
225	250	+935 +820	+1005 +820	+535 +420	+605 +420	+352 +280	+395 +280	+465 +280	+570 +280
250	280	+1050 +920	+1130 +920	+610 +480	+690 +480	+381 +300	+430 +300	+510 +300	+620 +300

(*Contd.*)

(*Contd.*)

280	315	+1180 +1050	+1260 +1050	+670 +540	+750 +540	+411 +330	+460 +330	+540 +330	+650 +330
315	355	+1340 +1200	+1430 +1200	+740 +600	+830 +600	+449 +360	+500 +360	+590 +360	+720 +360
355	400	+1490 +1350	+1580 +1350	+820 +680	+910 +680	+489 +400	+540 +400	+630 +400	+760 +400
400	450	+1655 +1500	+1750 +1500	+915 +760	+1010 +760	+537 +440	+595 +440	+690 +440	+840 +440
450	500	+1805 +1650	+1900 +1650	+995 +840	+1090 +840	+577 +480	+635 +480	+730 +480	+880 +480

(ISO: 286-2)

Table 3.9 *Limit deviations for holes of sizes up to 500 mm (Holes D, E and F)*
(Values of deviations in microns or μm) (1 micron = 0.001 mm)

Basic size (mm)		**D**				**E**			**F**		
Above	**Up to and including**	**8**	**9**	**10**	**11**	**6**	**7**	**8**	**6**	**7**	**8**
-	3	+34 +20	+45 +20	+60 +20	+80 +20	+20 +14	+24 +14	+28 +14	+12 +6	+16 +6	+20 +6
3	6	+48 +30	+60 +30	+78 +30	+105 +30	+28 +20	+32 +20	+38 +20	+18 +10	+22 +10	+28 +10
6	10	+62 +40	+76 +40	+98 +40	+130 +40	+34 +25	+40 +25	+47 +25	+22 +13	+28 +13	+35 +13
10	18	+77 +50	+93 +50	+120 +50	+160 +50	+43 +32	+50 +32	+59 +32	+27 +16	+34 +16	+43 +16
18	30	+98 +65	+117 +65	+149 +65	+195 +65	+53 +40	+61 +40	+73 +40	+33 +20	+41 +20	+53 +20
30	50	+119 +80	+142 +80	+180 +80	+240 +80	+66 +50	+75 +50	+89 +50	+41 +25	+50 +25	+64 +25
50	80	+146 +100	+174 +100	+220 +100	+290 +100	+79 +60	+90 +60	+106 +60	+49 +30	+60 +30	+76 +30
80	120	+174 +120	+207 +120	+260 +120	+340 +120	+94 +72	+107 +72	+125 +72	+58 +36	+71 +36	+90 +36
120	180	+208 +145	+245 +145	+305 +145	+395 +145	+110 +85	+125 +85	+148 +85	+68 +43	+83 +43	+106 +43
180	250	+242 +170	+285 +170	+355 +170	+460 +170	+129 +100	+146 +100	+172 +100	+79 +50	+96 +50	+122 +50
250	315	+271 +190	+320 +190	+400 +190	+510 +190	+142 +110	+162 +110	+191 +110	+88 +56	+108 +56	+137 +56
315	400	+299 +210	+350 +210	+440 +210	+570 +210	+161 +125	+182 +125	+214 +125	+98 +62	+119 +62	+151 +62
400	500	+327 +230	+385 +230	+480 +230	+630 +230	+175 +135	+198 +135	+232 +135	+108 +68	+131 +68	+165 +68

(ISO: 286-2)

Table 3.10 *Limit deviations for holes of sizes up to 500 mm (Holes H)*
(Values of deviations in microns or μm) (1 micron = 0.001 mm)

Basic size (mm)		H									
Above	**Up to and including**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**
–	3	+3 0	+4 0	+6 0	+10 0	+14 0	+25 0	+40 0	+60 0	+100 0	+140 0
3	6	+4 0	+5 0	+8 0	+12 0	+18 0	+30 0	+48 0	+75 0	+120 0	+180 0
6	10	+4 0	+6 0	+9 0	+15 0	+22 0	+36 0	+58 0	+90 0	+150 0	+220 0
10	18	+5 0	+8 0	+11 0	+18 0	+27 0	+43 0	+70 0	+110 0	+180 0	+270 0
18	30	+6 0	+9 0	+13 0	+21 0	+33 0	+52 0	+84 0	+130 0	+210 0	+330 0
30	50	+7 0	+11 0	+16 0	+25 0	+39 0	+62 0	+100 0	+160 0	+250 0	+390 0
50	80	+8 0	+13 0	+19 0	+30 0	+46 0	+74 0	+120 0	+190 0	+300 0	+460 0
80	120	+10 0	+15 0	+22 0	+35 0	+54 0	+87 0	+140 0	+220 0	+350 0	+540 0
120	180	+12 0	+18 0	+25 0	+40 0	+63 0	+100 0	+160 0	+250 0	+400 0	+630 0
180	250	+14 0	+20 0	+29 0	+46 0	+72 0	+115 0	+185 0	+290 0	+460 0	+720 0
250	315	+16 0	+23 0	+32 0	+52 0	+81 0	+130 0	+210 0	+320 0	+520 0	+810 0
315	400	+18 0	+25 0	+36 0	+57 0	+89 0	+140 0	+230 0	+360 0	+570 0	+890 0
400	500	+20 0	+27 0	+40 0	+63 0	+97 0	+155 0	+250 0	+400 0	+630 0	+970 0

(ISO: 286-2)

Table 3.11 *Limit deviations for holes of sizes up to 500 mm (Holes G, J and K)*
(Values of deviations in microns or μm) (1 micron = 0.001 mm)

Basic size (mm)		G			J			K		
Above	**Up to and including**	**6**	**7**	**8**	**6**	**7**	**8**	**6**	**7**	**8**
–	3	+8 +2	+12 +2	+16 +2	+2 −4	+4 −6	+6 −8	0 −6	0 −10	0 −14
3	6	+12 +4	+16 +4	+22 +4	+5 −3	±6	+10 −8	+2 −6	+3 −9	+5 −13

(*Contd.*)

(*Contd.*)

6	10	+14 +5	+20 +5	+27 +5	+5 −4	+8 −7	+12 −10	+2 −7	+5 −10	+6 −16
10	18	+17 +6	+24 +6	+33 +6	+6 −5	+10 −8	+15 −12	+2 −9	+6 −12	+8 −19
18	30	+20 +7	+28 +7	+40 +7	+8 −5	+12 −9	+20 −13	+2 −11	+6 −15	+10 −23
30	50	+25 +9	+34 +9	+48 +9	+10 −6	+14 −11	+24 −15	+3 −13	+7 −18	+12 −27
50	80	+29 +10	+40 +10	+56 +10	+13 −6	+18 −12	+28 −18	+4 −15	+9 −21	+14 −32
80	120	+34 +12	+47 +12	+66 +12	+16 −6	+22 −13	+34 −20	+4 −18	+10 −25	+16 −38
120	180	+39 +14	+54 +14	+77 +14	+18 −7	+26 −14	+41 −22	+4 −21	+12 −28	+20 −43
180	250	+44 +15	+61 +15	+87 +15	+22 −7	+30 −16	+47 −25	+5 −24	+13 −33	+22 −50
250	315	+49 +17	+69 +17	+98 +17	+25 −7	+36 −16	+55 −26	+5 −27	+16 −36	+25 −56
315	400	+54 +18	+75 +18	+107 +18	+29 −7	+39 −18	+60 −29	+7 −29	+17 −40	+28 −61
400	500	+60 +20	+83 +20	+117 +20	+33 −7	+43 −20	+66 −31	+8 −32	+18 −45	+29 −68

(ISO: 286-2)

Table 3.12 *Limit deviations for holes of sizes up to 500 mm (Holes M, N and P) (Values of deviations in microns or μm) (1 micron = 0.001 mm)*

Basic size (mm)		**M**			**N**			**P**		
Above	**Up to and including**	**6**	**7**	**8**	**6**	**7**	**8**	**6**	**7**	**8**
–	3	−2 −8	−2 −12	−2 −16	−4 −10	−4 −14	−4 −18	−6 −12	−6 −16	−6 −20
3	6	−1 −9	0 −12	+2 −16	−5 −13	−4 −16	−2 −20	−9 −17	−8 −20	−12 −30
6	10	−3 −12	0 −15	+1 −21	−7 −16	−4 −19	−3 −25	−12 −21	−9 −24	−15 −37
10	18	−4 −15	0 −18	+2 −25	−9 −20	−5 −23	−3 −30	−15 −26	−11 −29	−18 −45
18	30	−4 −17	0 −21	+4 −29	−11 −24	−7 −28	−3 −36	−18 −31	−14 −35	−22 −55
30	50	−4 −20	0 −25	+5 −34	−12 −28	−8 −33	−3 −42	−21 −37	−17 −42	−26 −65

(*Contd.*)

(*Contd.*)

50	80	−5 −24	0 −30	+5 −41	−14 −33	−9 −39	−4 −50	−26 −45	−21 −51	−32 −78
80	120	−6 −28	0 −35	+6 −48	−16 −38	−10 −45	−4 −58	−30 −52	−24 −59	−37 −91
120	180	−8 −33	0 −40	+8 −55	−20 −45	−12 −52	−4 −67	−36 −61	−28 −68	−43 −106
180	250	−8 −37	0 −46	+9 −63	−22 −51	−14 −60	−5 −77	−41 −70	−33 −79	−50 −122
250	315	−9 −41	0 −52	+9 −72	−25 −57	−14 −66	−5 −86	−47 −79	−36 −88	−56 −137
315	400	−10 −46	0 −57	+11 −78	−26 −62	−16 −73	−5 −94	−51 −87	−41 −98	−62 −151
400	500	−10 −50	0 −63	+11 −86	−27 −67	−17 −80	−6 −103	−55 −95	−45 −108	−68 −165

(ISO: 286-2)

Table 3.13 *Limit deviations for holes of sizes up to 500 mm (Holes R, S and T) (Values of deviations in microns or μm) (1 micron = 0.001 mm)*

Basic size (mm)		**R**			**S**			**T**		
Above	**Up to and including**	**6**	**7**	**8**	**6**	**7**	**8**	**6**	**7**	**8**
–	3	−10 −16	−10 −20	−10 −24	−14 −20	−14 −24	−14 −28			
3	6	−12 −20	−11 −23	−15 −33	−16 −24	−15 −27	−19 −37			
6	10	−16 −25	−13 −28	−19 −41	−20 −29	−17 −32	−23 −45			
10	18	−20 −31	−16 −34	−23 −50	−25 −36	−21 −39	−28 −55			-
18	30	−24 −37	−20 −41	−28 −61	−31 −44	−27 −48	−35 −68			
30	50	−29 −45	−25 −50	−34 −73	−38 −54	−34 −59	−43 −82			
50	65	−35 −54	−30 −60	−41 −87	−47 −66	−42 −72	−53 −99	−60 −79	−55 −85	−66 −112
65	80	−37 −56	−32 −62	−43 −89	−53 −72	−48 −78	−59 −105	−69 −88	−64 −94	−75 −121
80	100	−44 −66	−38 −73	−51 −105	−64 −86	−58 −93	−71 −125	−84 −106	−78 −113	−91 −145
100	120	−47 −69	−41 −76	−54 −108	−72 −94	−66 −101	−79 −133	−97 −119	−91 −126	−104 −158

(*Contd.*)

(*Contd.*)

120	140	−56 −81	−48 −88	−63 −126	−85 −110	−77 −117	−92 −155	−115 −140	−107 −147	−122 −185
140	160	−58 −83	−50 −90	−65 −128	−93 −118	−85 −125	−100 −163	−127 −152	−119 −159	−134 −197
160	180	−61 −86	−53 −93	−68 −131	−101 −126	−93 −133	−108 −171	−139 −164	−131 −171	−146 −209
180	200	−68 −97	−60 −106	−77 −149	−113 −142	−105 −151	−122 −194	−157 −186	−149 −195	−166 −238
200	225	−71 −100	−63 −109	−80 −152	−121 −150	−113 −159	−130 −202	−171 −200	−163 −209	−180 −252
225	250	−75 −104	−67 −113	−84 −156	−131 −160	−123 −169	−140 −212	−187 −216	−179 −225	−196 −268
250	280	−85 −117	−74 −126	−94 −175	−149 −181	−138 −190	−158 −239	−209 −241	−198 −250	−218 −299
280	315	−89 −121	−78 −130	−98 −179	−161 −193	−150 −202	−170 −251	−231 −263	−220 −272	−240 −321
315	355	−97 −133	−87 −144	−108 −197	−179 −215	−169 −226	−190 −279	−257 −293	−247 −304	−268 −357
355	400	−103 −139	−93 −150	−114 −203	−197 −233	−187 −244	−208 −297	−283 −319	−273 −330	−294 −383
400	450	−113 −153	−103 −166	−126 −223	−219 −259	−209 −272	−232 −329	−317 −357	−307 −370	−330 −427
450	500	−119 −159	−109 −172	−132 −229	−239 −279	−229 −292	−252 −349	−347 −387	−337 −400	−360 −457

(ISO: 286-2)

3.5 TOLERANCES FOR SHAFTS

Table 3.14 *Limit deviations for shafts of sizes up to 500 mm (Shafts a, b and c)*
(Values of deviations in microns or μm) (1 micron = 0.001 mm)

Basic size (mm)		a			b			c		
Above	**Up to and including**	**9**	**10**	**11**	**9**	**10**	**11**	**9**	**10**	**11**
–	3	−270 −295	−270 −310	−270 −330	−140 −165	−140 −180	−140 −200	−60 −85	−60 −100	−60 −120
3	6	−270 −300	−270 −318	−270 −345	−140 −170	−140 −188	−140 −215	−70 −100	−70 −118	−70 −145
6	10	−280 −316	−280 −338	−280 −370	−150 −186	−150 −208	−150 −240	−80 −116	−80 −138	−80 −170

(*Contd.*)

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10	18	−290 −333	−290 −360	−290 −400	−150 −193	−150 −220	−150 −260	−95 −138	−95 −165	−95 −205
18	30	−300 −352	−300 −384	−300 −430	−160 −212	−160 −244	−160 −290	−110 −162	−110 −194	−110 −240
30	40	−310 −372	−310 −410	−310 −470	−170 −232	−170 −270	−170 −330	−120 −182	−120 −220	−120 −280
40	50	−320 −382	−320 −420	−320 −480	−180 −242	−180 −280	−180 −340	−130 −192	−130 −230	−130 −290
50	65	−340 −414	−340 −460	−340 −530	−190 −264	−190 −310	−190 −380	−140 −214	−140 −260	−140 −330
65	80	−360 −434	−360 −480	−360 −550	−200 −274	−200 −320	−200 −390	−150 −224	−150 −270	−150 −340
80	100	−380 −467	−380 −520	−380 −600	−220 −307	−220 −360	−220 −440	−170 −257	−170 −310	−170 −390
100	120	−410 −497	−410 −550	−410 −630	−240 −327	−240 −380	−240 −460	−180 −267	−180 −320	−180 −400
120	140	−460 −560	−460 −620	−460 −710	−260 −360	−260 −420	−260 −510	−200 −300	−200 −360	−200 −450
140	160	−520 −620	−520 −680	−520 −770	−280 −380	−280 −440	−280 −530	−210 −310	−210 −370	−210 −460
160	180	−580 −680	−580 −740	−580 −830	−310 −410	−310 −470	−310 −560	−230 −330	−230 −390	−230 −480
180	200	−660 −775	−660 −845	−660 −950	−340 −455	−340 −525	−340 −630	−240 −355	−240 −425	−240 −530
200	225	−740 −855	−740 −925	−740 −1030	−380 −495	−380 −565	−380 −670	−260 −375	−260 −445	−260 −550
225	250	−820 −935	−820 −1005	−820 −1110	−420 −535	−420 −605	−420 −710	−280 −395	−280 −465	−280 −570
250	280	−920 −1050	−920 −1130	−920 −1240	−480 −610	−480 −690	−480 −800	−300 −430	−300 −510	−300 −620
280	315	−1050 −1180	−1050 −1260	−1050 −1370	−540 −670	−540 −750	−540 −860	−330 −460	−330 −540	−330 −650
315	355	−1200 −1340	−1200 −1430	−1200 −1560	−600 −740	−600 −830	−600 −960	−360 −500	−360 −590	−360 −720
355	400	−1350 −1490	−1350 −1580	−1350 −1710	−680 −820	−680 −910	−680 −1040	−400 −540	−400 −630	−400 −760
400	450	−1500 −1655	−1500 −1750	−1500 −1900	−760 −915	−760 −1010	−760 −1160	−440 −595	−440 −690	−440 −840
450	500	−1650 −1805	−1650 −1900	−1650 −2050	−840 −995	−840 −1090	−840 −1240	−480 −635	−480 −730	−480 −880

(ISO: 286-2)

Table 3.15 *Limit deviations for shafts of sizes up to 500 mm (Shafts d and e)*
(Values of deviations in microns or μm) (1 micron = 0.001 mm)

Basic size (mm)		d					e				
Above	**Up to and including**	**6**	**7**	**8**	**9**	**10**	**6**	**7**	**8**	**9**	**10**
–	3	−20 −26	−20 −30	−20 −34	−20 −45	−20 −60	−14 −20	−14 −24	−14 −28	−14 −39	−14 −54
3	6	−30 −38	−30 −42	−30 −48	−30 −60	−30 −78	−20 −28	−20 −32	−20 −38	−20 −50	−20 −68
6	10	−40 −49	−40 −55	−40 −62	−40 −76	−40 −98	−25 −34	−25 −40	−25 −47	−25 −61	−25 −83
10	18	−50 −61	−50 −68	−50 −77	−50 −93	−50 −120	−32 −43	−32 −50	−32 −59	−32 −75	−32 −102
18	30	−65 −78	−65 −86	−65 −98	−65 −117	−65 −149	−40 −53	−40 −61	−40 −73	−40 −92	−40 −124
30	50	−80 −96	−80 −105	−80 −119	−80 −142	−80 −180	−50 −66	−50 −75	−50 −89	−50 −112	−50 −150
50	80	−100 −119	−100 −130	−100 −146	−100 −174	−100 −220	−60 −79	−60 −90	−60 −106	−60 −134	−60 −180
80	120	−120 −142	−120 −155	−120 −174	−120 −207	−120 −260	−72 −94	−72 −107	−72 −126	−72 −159	−72 −212
120	180	−145 −170	−145 −185	−145 −208	−145 −245	−145 −305	−85 −110	−85 −125	−85 −148	−85 −185	−85 −245
180	250	−170 −199	−170 −216	−170 −242	−170 −285	−170 −355	−100 −129	−100 −146	−100 −172	−100 −215	−100 −285
250	315	−190 −222	−190 −242	−190 −271	−190 −320	−190 −400	−110 −142	−110 −162	−110 −191	−110 −240	−110 −320
315	400	−210 −246	−210 −267	−210 −299	−210 −350	−210 −440	−125 −161	−125 −182	−125 −214	−125 −265	−125 −355
400	500	−230 −270	−230 −293	−230 −327	−230 −385	−230 −480	−135 −175	−135 −198	−135 −232	−135 −290	−135 −385

(ISO: 286-2)

Table 3.16 *Limit deviations for shafts of sizes up to 500 mm (Shafts f and g)*
(Values of deviations in microns or μm) (1 micron = 0.001 mm)

Basic size (mm)		f					g				
Above	**Up to and including**	**5**	**6**	**7**	**8**	**9**	**4**	**5**	**6**	**7**	**8**
–	3	−6 −10	−6 −12	−6 −16	−6 −20	−6 −31	−2 −5	−2 −6	−2 −8	−2 −12	−2 −16
3	6	−10 −15	−10 −18	−10 −22	−10 −28	−10 −40	−4 −8	−4 −9	−4 −12	−4 −16	−4 −22

(*Contd.*)

(*Contd.*)

6	10	−13 −19	−13 −22	−13 −28	−13 −35	−13 −49	−5 −9	−5 −11	−5 −14	−5 −20	−5 −27
10	18	−16 −24	−16 −27	−16 −34	−16 −43	−16 −59	−6 −11	−6 −14	−6 −17	−6 −24	−6 −33
18	30	−20 −29	−20 −33	−20 −41	−20 −53	−20 −72	−7 −13	−7 −16	−7 −20	−7 −28	−7 −40
30	50	−25 −36	−25 −41	−25 −50	−25 −64	−25 −87	−9 −16	−9 −20	−9 −25	−9 −34	−9 −48
50	80	−30 −43	−30 −49	−30 −60	−30 −76	−30 −104	−10 −18	−10 −23	−10 −29	−10 −40	−10 −56
80	120	−36 −51	−36 −58	−36 −71	−36 −90	−36 −123	−12 −22	−12 −27	−12 −34	−12 −47	−12 −66
120	180	−43 −61	−43 −68	−43 −83	−43 −106	−43 −143	−14 −26	−14 −32	−14 −39	−14 −54	−14 −77
180	250	−50 −70	−50 −79	−50 −96	−50 −122	−50 −165	−15 −29	−15 −35	−15 −44	−15 −61	−15 −87
250	315	−56 −79	−56 −88	−56 −108	−56 −137	−56 −185	−17 −33	−17 −40	−17 −49	−17 −69	−17 −98
315	400	−62 −87	−62 −98	−62 −119	−62 −151	−62 −202	−18 −36	−18 −43	−18 −54	−18 −75	−18 −107
400	500	−68 −95	−68 −108	−68 −131	−68 −165	−68 −223	−20 −40	−20 −47	−20 −60	−20 −83	−20 −117

(ISO: 286-2)

Table 3.17 *Limit deviations for shafts of sizes up to 500 mm (Shafts h and j) (Values of deviations in microns or µm) (1 micron = 0.001 mm)*

Basic size (mm)		**h**							**J**		
Above	**Up to and including**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**5**	**6**	**7**
–	3	0 −4	0 −6	0 −10	0 −14	0 −25	0 −40	0 −60	±2	+4 −2	+6 −4
3	6	0 −5	0 −8	0 −12	0 −18	0 −30	0 −48	0 −75	+3 −2	+6 −2	+8 −4
6	10	0 −6	0 −9	0 −15	0 −22	0 −36	0 −58	0 −90	+4 −2	+7 −2	+10 −5
10	18	0 −8	0 −11	0 −18	0 −27	0 −43	0 −70	0 −110	+5 −3	+8 −3	+12 −6
18	30	0 −9	0 −13	0 −21	0 −33	0 −52	0 −84	0 −130	+5 −4	+9 −4	+13 −8
30	50	0 −11	0 −16	0 −25	0 −39	0 −62	0 −100	0 −160	+6 −5	+11 −5	+15 −10

(*Contd.*)

(*Contd.*)

50	80	0 −13	0 −19	0 −30	0 −46	0 −74	0 −120	0 −190	+6 −7	+12 −7	+18 −12
80	120	0 −15	0 −22	0 −35	0 −54	0 −87	0 −140	0 −220	+6 −9	+13 −9	+20 −15
120	180	0 −18	0 −25	0 −40	0 −63	0 −100	0 −160	0 −250	+7 −11	+14 −11	+22 −18
180	250	0 −20	0 −29	0 −46	0 −72	0 −115	0 −185	0 −290	+7 −13	+16 −13	+25 −21
250	315	0 −23	0 −32	0 −52	0 −81	0 −130	0 −210	0 −320	+7 −16	±16	±26
315	400	0 −25	0 −36	0 −57	0 −89	0 −140	0 −230	0 −360	+7 −18	±18	+29 −28
400	500	0 −27	0 −40	0 −63	0 −97	0 −155	0 −250	0 −400	+7 −20	±20	+31 −32

(ISO: 286-2)

Table 3.18 *Limit deviations for shafts of sizes up to 500 mm (Shafts k and m) (Values of deviations in microns or μm) (1 micron = 0.001 mm)*

Basic size (mm)		k				m			
Above	**Up to and including**	**5**	**6**	**7**	**8**	**5**	**6**	**7**	**8**
–	3	+4 0	+6 0	+10 0	+14 0	+6 +2	+8 +2	+12 +2	+16 +2
3	6	+6 +1	+9 +1	+13 +1	+18 0	+9 +4	+12 +4	+16 +4	+22 +4
6	10	+7 +1	+10 +1	+16 +1	+22 0	+12 +6	+15 +6	+21 +6	+28 +6
10	18	+9 +1	+12 +1	+19 +1	+27 0	+15 +7	+18 +7	+25 +7	+34 +7
18	30	+11 +2	+15 +2	+23 +2	+33 0	+17 +8	+21 +8	+29 +8	+41 +8
30	50	+13 +2	+18 +2	+27 +2	+39 0	+20 +9	+25 +9	+34 +9	+48 +9
50	80	+15 +2	+21 +2	+32 +2	+46 0	+24 +11	+30 +11	+41 +11	
80	120	+18 +3	+25 +3	+38 +3	+54 0	+28 +13	+35 +13	+48 +13	
120	180	+21 +3	+28 +3	+43 +3	+63 0	+33 +15	+40 +15	+55 +15	
180	250	+24 +4	+33 +4	+50 +4	+72 0	+37 +17	+46 +17	+63 +17	

(*Contd.*)

(*Contd.*)

250	315	+27 +4	+36 +4	+56 +4	+81 0	+43 +20	+52 +20	+72 +20	
315	400	+29 +4	+40 +4	+61 +4	+89 0	+46 +21	+57 +21	+78 +21	
400	500	+32 +5	+45 +5	+68 +5	+97 0	+50 +23	+63 +23	+86 +23	

(ISO: 286-2)

Table 3.19 *Limit deviations for shafts of sizes up to 500 mm (Shafts n and p)*
(Values of deviations in microns or µm) (1 micron = 0.001 mm)

Basic size (mm)		**n**				**p**			
Above	**Up to and including**	**5**	**6**	**7**	**8**	**5**	**6**	**7**	**8**
–	3	+8 +4	+10 +4	+14 +4	+18 +4	+10 +6	+12 +6	+16 +6	+20 +6
3	6	+13 +8	+16 +8	+20 +8	+26 +8	+17 +12	+20 +12	+24 +12	+30 +12
6	10	+16 +10	+19 +10	+25 +10	+32 +10	+21 +15	+24 +15	+30 +15	+37 +15
10	18	+20 +12	+23 +12	+30 +12	+39 +12	+26 +18	+29 +18	+36 +18	+45 +18
18	30	+24 +15	+28 +15	+36 +15	+48 +15	+31 +22	+35 +22	+43 +22	+55 +22
30	50	+28 +17	+33 +17	+42 +17	+56 +17	+37 +26	+42 +26	+51 +26	+65 +26
50	80	+33 +20	+39 +20	+50 +20		+45 +32	+51 +32	+62 +32	+78 +32
80	120	+38 +23	+45 +23	+58 +23		+52 +37	+59 +37	+72 +37	+91 +37
120	180	+45 +27	+52 +27	+67 +27		+61 +43	+68 +43	+83 +43	+106 +43
180	250	+51 +31	+60 +31	+77 +31		+70 +50	+79 +50	+96 +50	+122 +50
250	315	+57 +34	+66 +34	+86 +34		+79 +56	+88 +56	+108 +56	+137 +56
315	400	+62 +37	+73 +37	+94 +37		+87 +62	+98 +62	+119 +62	+151 +62
400	500	+67 +40	+80 +40	+103 +40		+95 +68	+108 +68	+131 +68	+165 +68

(ISO: 286-2)

Table 3.20 *Limit deviations for shafts of sizes up to 500 mm (Shafts r, s and t)*
(Values of deviations in microns or μm) (1 micron = 0.001 mm)

Basic size (mm)		r				s				t	
Above	**Up to and including**	**5**	**6**	**7**	**8**	**5**	**6**	**7**	**8**	**6**	**7**
–	3	+14 +10	+16 +10	+20 +10	+24 +10	+18 +14	+20 +14	+24 +14	+28 +14		
3	6	+20 +15	+23 +15	+27 +15	+33 +15	+24 +19	+27 +19	+31 +19	+37 +19		
6	10	+25 +19	+28 +19	+34 +19	+41 +19	+29 +23	+32 +23	+38 +23	+45 +23		
10	18	+31 +23	+34 +23	+41 +23	+50 +23	+36 +28	+39 +28	+46 +28	+55 +28		
18	30	+37 +28	+41 +28	+49 +28	+61 +28	+44 +35	+48 +35	+56 +35	+68 +35		
30	50	+45 +34	+50 +34	+59 +34	+73 +34	+54 +43	+59 +43	+68 +43	+82 +43		
50	65	+54 +41	+60 +41	+71 +41	+87 +41	+66 +53	+72 +53	+83 +53	+99 +53	+85 +66	+96 +66
65	80	+56 +43	+62 +43	+73 +43	+89 +43	+72 +59	+78 +59	+89 +59	+105 +59	+94 +75	+105 +75
80	100	+66 +51	+73 +51	+86 +51	+105 +51	+86 +71	+93 +71	+106 +71	+125 +71	+113 +91	+126 +91
100	120	+69 +54	+76 +54	+89 +54	+108 +54	+94 +79	+101 +79	+114 +79	+133 +79	+126 +104	+139 +104
120	140	+81 +63	+88 +63	+103 +63	+126 +63	+110 +92	+117 +92	+132 +92	+155 +92	+147 +122	+162 +122
140	160	+83 +65	+90 +65	+105 +65	+128 +65	+118 +100	+125 +100	+140 +100	+163 +100	+159 +134	+174 +134
160	180	+86 +68	+93 +68	+108 +68	+131 +68	+126 +108	+133 +108	+148 +108	+171 +108	+171 +146	+186 +146
180	200	+97 +77	+106 +77	+123 +77	+149 +77	+142 +122	+151 +122	+168 +122	+194 +122	+195 +166	+212 +166
200	225	+100 +80	+109 +80	+126 +80	+152 +80	+150 +130	+159 +130	+176 +130	+202 +130	+209 +180	+226 +180
225	250	+104 +84	+113 +84	+130 +84	+156 +84	+160 +140	+169 +140	+186 +140	+212 +140	+225 +196	+242 +196
250	280	+117 +94	+126 +94	+146 +94	+175 +94	+181 +158	+190 +158	+210 +158	+239 +158	+250 +218	+270 +218
280	315	+121 +98	+130 +98	+150 +98	+179 +98	+193 +170	+202 +170	+222 +170	+251 +170	+272 +240	+292 +240
315	355	+133 +108	+144 +108	+165 +108	+197 +108	+215 +190	+226 +190	+247 +190	+279 +190	+304 +268	+325 +268
355	400	+139 +114	+150 +114	+171 +114	+203 +114	+233 +208	+244 +208	+265 +208	+297 +208	+330 +294	+351 +294
400	450	+153 +126	+166 +126	+189 +126	+223 +126	+259 +232	+272 +232	+295 +232	+329 +232	+370 +330	+393 +330
450	500	+159 +132	+172 +132	+195 +132	+229 +132	+279 +252	+292 +252	+315 +252	+349 +252	+400 +360	+423 +360

(ISO: 286-2)

Table 3.21 *Typical values of tolerance for different manufacturing processes*

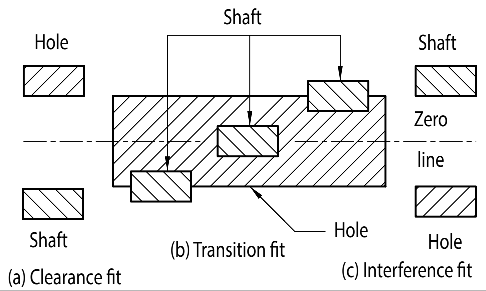
Manufacturing Process	Typical Tolerance (±) (mm)
Sand casting	0.5–2.0
Investment casting	0.2–0.8
Die casting	0.1–0.5
Forging	0.2–1.0
Hot rolling	0.2–0.8
Cold rolling	0.05–0.2
Turning and boring	0.025–0.05
Drilling	0.05–0.25
Shaping and planing	0.025–0.125
Milling	0.01–0.02
Reaming	0.01–0.05
Broaching	0.01–0.05
Grinding	0.01–0.02
Honing	0.005–0.01
Polishing	0.005–0.01
Lapping	0.004–0.01

Table 3.22 *Relationship between tolerance grades and manufacturing processes*

Grades and Manufacturing Processes	
Grade	**Manufacturing Methods**
Grade 16	Sand casting and flame cutting
Grade 15	Stamping
Grade 14	Die casting
Grade 11	Drilling, rough turning and boring
Grade 10	Milling, slotting, planing, rolling and extrusion
Grade 9	Horizontal and vertical boring and turning on automatic lathes
Grade 8	Turning, boring and reaming on capstan and turret lathes
Grade 7	High precision turning, broaching and honing
Grade 6	Grinding and fine honing
Grade 5	Lapping, fine grinding and diamond boring
Grade 4	Lapping
Tolerance and Manufacturing Processes for Holes	
H5	Precision boring, fine internal grinding and honing
H6	Precision boring, honing and hand reaming
H7	Grinding, broaching and precision reaming
H8	Boring and machine reaming

3.6 SELECTION OF FITS

Table 3.23 *Types of fits*



When two parts are to be assembled, the relationship resulting from the difference between their sizes before assembly is called a fit. Depending upon the limits of the shaft and the hole, fits are broadly classified into three groups – clearance fit, transition fit and interference fit. Clearance fit is a fit, which always provides a positive clearance between the hole and the shaft over the entire range of tolerances. In this case, the tolerance zone of the hole is entirely above that of the shaft. Interference fit is a fit, which always provides a positive interference over the whole range of tolerances. In this case, the tolerance zone of the hole is completely below that of the shaft. Transition fit is a fit, which may provide either a clearance or interference, depending upon the actual values of the individual tolerances of the mating components. In this case, the tolerance zones of the hole and the shaft overlap.

Table 3.24 *Selection of fits*

Clearance Fits
The guidelines for the selection of clearance fits are as follows: (i) The fits H7-d8, H8-d9 and H11-d11 are loose running fits, and are used for plummer-block bearings and loose pulleys. (ii) The fits H6-e7, H7-e8 and H8-e8 are loose clearance fits, and are used for properly lubricated bearings, requiring appreciable clearances. The finer grades are used for heavy-duty, high-speed bearings and large electric motors. (iii) The fits H6-f6, H7-f7 and H8-f8 are normal running fits, widely used for grease or oil lubricated bearings having low temperature rise. They are also used for shafts of gearboxes, small electric motors and pumps. (iv) The fits H6-g5, H7-g6 and H8-g7 are expensive from manufacturing considerations. They are used in precision equipment, pistons, slide valves and bearings of accurate link mechanisms.
Transition Fits
The typical types of transition fits are H6-j5, H7-j6 and H8-j7. They are used in applications where slight interference is permissible. Some of their applications are spigot and recess of the rigid coupling and the composite gear blank, where steel rim is fitted on ordinary steel hub.
Interference Fits
The general guidelines for the selection of interference fits are as follows: (i) The fit H7-p6 or H7-p7 results in interference, which is not excessive but sufficient to give non-ferrous parts a light press fit. Such parts can be dismantled easily as and when required, e.g., fitting a brass bush in the gear. (ii) The fit H6-r5 or H7-r6 is a medium tight fit on ferrous parts, which can be easily dismantled. (iii) The fits H6-s5, H7-s6 and H8-s7 are used for permanent and semi-permanent assemblies of steel and cast-iron parts. The amount of interference in these fits is sufficiently large to provide a considerable gripping force. They are used in valve seats and shaft collars. The selection of interference fit depends upon a number of factors, such as materials, diameters, surface finish and machining methods. It is necessary to calculate the maximum and minimum interference in each case. The torque transmitting capacity is calculated for minimum interference, while the force required to assemble the parts is decided by the maximum value of interference.

3.7 GEOMETRICAL TOLERANCES

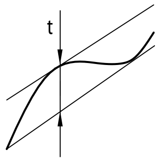
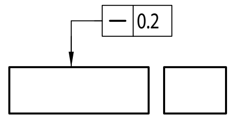
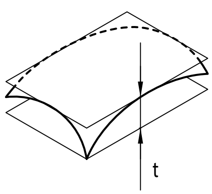
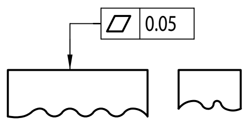
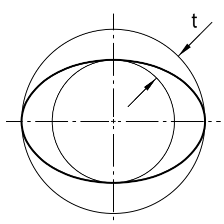
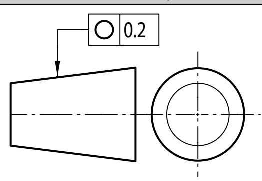
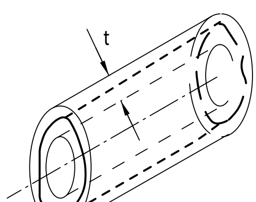
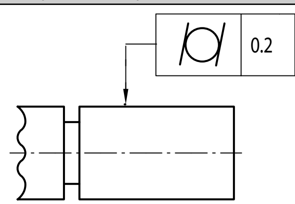
Table 3.25 *Symbols for toleranced characteristics*

Features and Tolerances		**Toleranced Characteristics**	**Symbol**
Single Features	Form Tolerances	Straightness	⏤
		Flatness	⏥
		Circularity	○
		Cylindricity	⌭
Single or Related Features		Profile of any line	⌒
		Profile of any surface	⌓
Related Features	Orientation Tolerances	Parallelism	∥
		Perpendicularity	⊥
		Angularity	∠
	Location Tolerances	Position	⌖
		Concentricity and Coaxiality	◎
		Symmetry	⌯
	Run-out Tolerances	Circular run-out	↗
		Total run-out	⌰

(ISO: 1101)

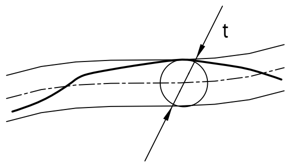
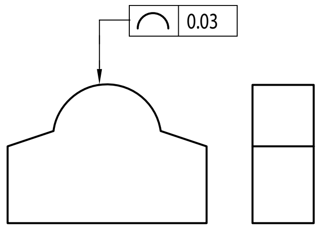
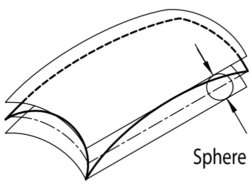
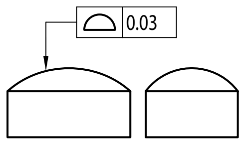
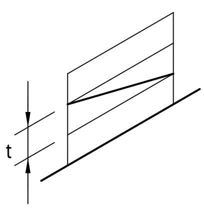
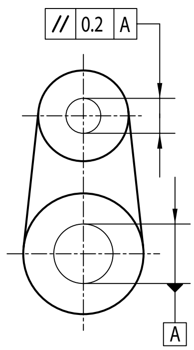
Note: (i) A geometrical tolerance applied to a feature defines the tolerance zone within which the feature (surface, axis, or median plane) is to be contained. (ii) Geometrical tolerances are to be specified only where they are essential, that is, in the light of functional requirements, interchangeability and probable manufacturing circumstances.

Table 3.26 *Indication and interpretation of geometrical tolerances*

Definition of tolerance zone	Indication on drawing	Interpretation of tolerance
Straightness Tolerance		
Definition: The tolerance zone, when projected in a plane, is limited by two parallel straight lines at a distance *t* apart. 		Any line on the upper surface parallel to the plane of projection in which the indication is shown shall be contained between two parallel straight lines 0.2 apart.
Flatness Tolerance		
Definition: The tolerance zone is limited by two parallel planes at a distance *t* apart 		The surface shall be contained between two parallel planes 0.05 apart.
Circularity Tolerance		
Definition: The tolerance zone in the considered plane is limited by two concentric circles at a distance *t* apart. 		The circumference of each cross-section shall be contained between two co-planar concentric circles 0.2 apart.
Cylindricity Tolerance		
Definition: The tolerance zone is limited by two coaxial cylinders at a distance *t* apart. 		The considered surface shall be contained between two coaxial cylinders 0.2 apart.

(*Contd.*)

(*Contd.*)

<table>
<tr><th colspan="3">Profile Tolerance of any Line</th></tr>
<tr>
<td>Definition: The tolerance zone is limited by two lines enveloping circles of diameter t, the centres of which are situated on a line having the true geometrical form.
</td>
<td></td>
<td>In each section parallel to the plane of projection, the considered profile shall be contained between two lines that are enveloping circles of diameter 0.03, the centres of which are situated on a line having the true geometrical profile.</td>
</tr>
<tr><th colspan="3">Profile Tolerance of any Surface</th></tr>
<tr>
<td>Definition: The tolerance zone is limited by two surfaces enveloping spheres of diameter t, the centres of which are situated on a surface having the true geometrical form.
</td>
<td></td>
<td>The considered surface shall be contained between two surfaces that are enveloping spheres of diameter 0.03, the centres of which are situated on a surface having true geometrical form.</td>
</tr>
<tr><th colspan="3">Parallelism Tolerance</th></tr>
<tr>
<td>Definition: The tolerance zone, when projected in a plane, is limited by two parallel straight lines at a distance t apart and parallel to the datum line, if the tolerance is specified only in one direction.
</td>
<td></td>
<td>The toleranced axis shall be contained between two straight lines 0.2 apart, which are parallel to datum axis A and lie in the vertical direction.</td>
</tr>
</table>

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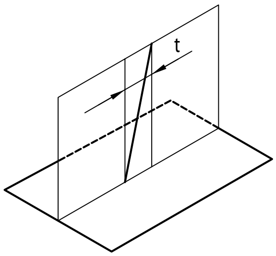
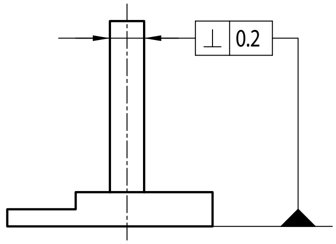
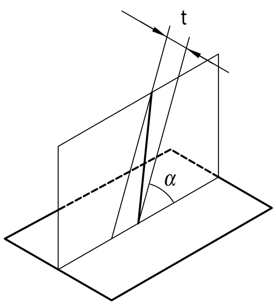
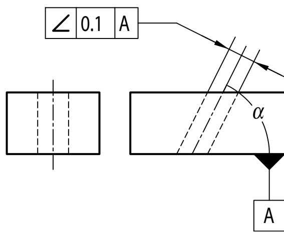
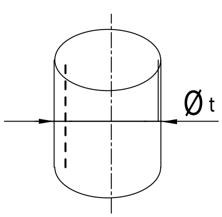
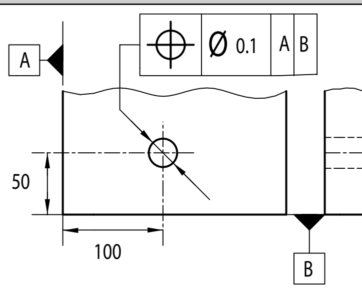
Perpendicularity Tolerance		
Definition: The tolerance zone, when projected in a plane, is limited by two parallel straight lines at a distance *t* apart and perpendicular to the datum plane, if the tolerance is specified only in one direction. 		The axis of the cylinder, to which the tolerance frame is connected, shall be contained between two parallel planes 0.2 apart, perpendicular to the datum surface.
Angularity Tolerance		
Definition: The tolerance zone, when projected in a plane, is limited by two parallel straight lines at a distance *t* apart and inclined at the specified angle to the datum surface. 		The axis of the hole shall be contained between two parallel planes 0.1 apart which are inclined at an angle α to the surface **A** (datum surface)
Positional Tolerance		
Definition: The tolerance zone is limited by a cylinder of diameter *t*, the axis of which is in the theoretically exact position of the considered line if the tolerance value is preceded by the sign ϕ. 		The axis of the hole shall be contained within a cylindrical zone of diameter 0.1, the axis of which is in the theoretically exact position of the considered line, with reference to surfaces **A** and **B** (datum planes)

(*Contd.*)

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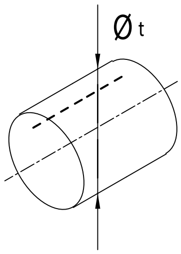
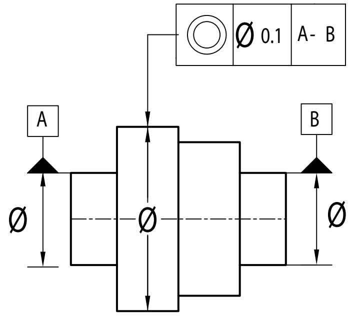
<table>
<tr><th colspan="3">Coaxiality Tolerance</th></tr>
<tr>
<td>

Definition: The tolerance zone is limited by a cylinder of diameter *t*, the axis of which coincides with the datum axis, if the tolerance value is preceded by the sign ϕ.

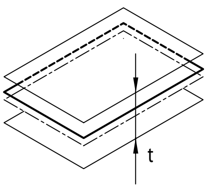


</td>
<td>

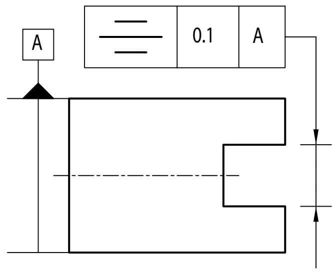


</td>
<td>The axis of the cylinder, to which the tolerance frame is connected, shall be contained in a cylindrical zone of diameter 0.1 coaxial with datum axis A-B.</td>
</tr>
<tr><th colspan="3">Symmetry Tolerance</th></tr>
<tr>
<td>

Definition: The tolerance zone is limited by two parallel planes at a distance *t* apart and disposed symmetrically to the median plane with respect to the datum axis or datum plane.

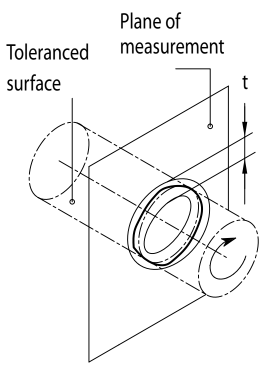


</td>
<td>

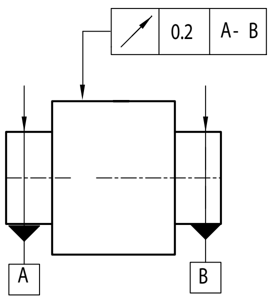


</td>
<td>The median plane of the slot shall be contained between two parallel planes, which are 0.1 apart and symmetrically disposed about the median plane with respect to the datum feature A.</td>
</tr>
<tr><th colspan="3">Circular run-out Tolerance</th></tr>
<tr>
<td>

Definition: The tolerance zone is limited within any plane of measurement perpendicular to the axis by two concentric circles at a distance **t** apart, the centre of which coincides with the datum axis.



</td>
<td>



</td>
<td>The radial run-out shall not be greater than 0.2 in any plane of measurement during one revolution about the datum axis A–B.</td>
</tr>
</table>

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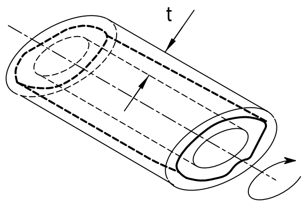
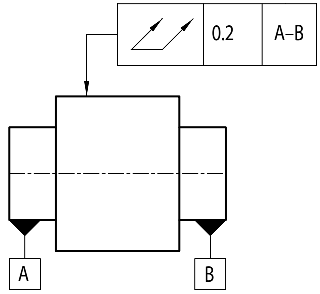
Total Radial Run-out Tolerance		
Definition: The tolerance zone is limited by two coaxial cylinders at a distance *t* apart, the axes of which coincide with the datum axis. 		The total radial run-out shall not be greater than 0.2 at any point on the specified surface during several revolutions about the datum axis **A–B,** and with relative axial movement between part and measuring instrument. With relative movement, the measuring instrument or the work piece shall be guided along a line having the theoretically perfect form of the counter and being in correct position to the datum axis.

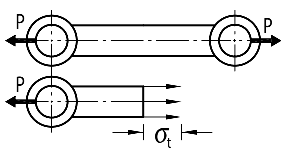
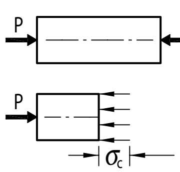
(ISO: 1101)

Static Stresses

4

4.1 STRESS-STRAIN RELATIONSHIPS

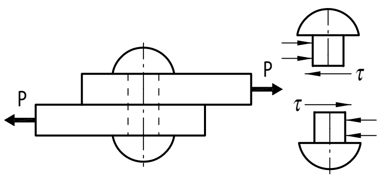
Table 4.1 *Static stresses*

Tensile stress	
	$$\sigma_t = \frac{P}{A} \quad (4.1)$$ σ_t = tensile stress (MPa or N/mm^2) P = external axial force (N) A = cross-sectional area (mm^2) $$\varepsilon = \frac{\delta}{l} \quad (4.2)$$ ε = strain (mm/mm) δ = elongation of tension rod (mm) l = original length of the rod (mm) $$\sigma_t = E\,\varepsilon \quad (4.3)$$ E = Young's modulus or Modulus of elasticity (MPa or N/mm^2). For Carbon steels, E = 207 000 MPa For Grey cast iron, E = 120 000 MPa $$\delta = \frac{Pl}{AE} \quad (4.4)$$
Compressive stress	
	$$\sigma_c = \frac{P}{A} \quad (4.5)$$ σ_c = compressive stress (MPa or N/mm^2) P = external axial force (N) A = cross-sectional area (mm^2)

(*Contd.*)

(*Contd.*)

Direct shear stress



$$\tau = \frac{P}{A} \qquad (4.6)$$

τ = shear stress (MPa or N/mm^2)

A = cross-sectional area of the rivet (mm^2)

$$\tau = G\gamma \qquad (4.7)$$

γ = shear strain (radians)

G = shear modulus or Modulus of rigidity (MPa or N/mm^2)

For Carbon steels G = 79 000 N/mm^2

For Grey cast iron G = 41 000 N/mm^2

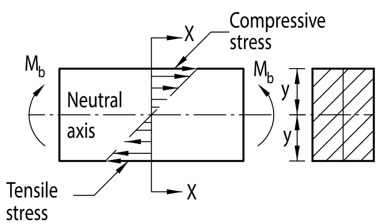
$$E = 2G\,(1 + \mu) \qquad (4.8)$$

μ = Poisson's ratio

For Carbon steels μ = 0.30

For Grey cast iron μ = 0.26

Bending stresses



$$\sigma_b = \frac{M_b\,y}{I} \qquad (4.9)$$

σ_b = bending stress at a distance of y from the neutral axis (MPa or N/mm^2)

M_b = applied bending moment (N-mm)

y = distance of the fibre from neutral axis (mm)

I = moment of inertia of the cross section about the neutral axis (mm^4)

For a rectangular cross-section,

$$I = \frac{bh^3}{12} \qquad (4.10)$$

b = distance parallel to the neutral axis (mm)

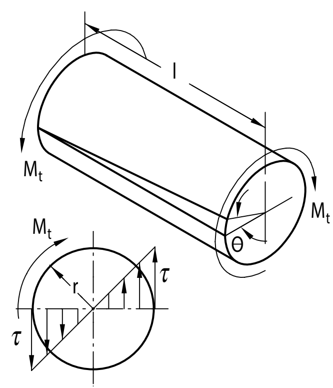
h = distance perpendicular to the neutral axis (mm)

For a circular cross sections,

$$I = \frac{\pi d^4}{64} \qquad (4.11)$$

d = diameter of the cross section (mm)

Torsional shear stress



$$\tau = \frac{M_t\,r}{J} \qquad (4.12)$$

τ = torsional shear stress at the fibre (MPa or N/mm^2)

M_t = applied torque (N-mm)

r = radial distance of the fibre from the axis of rotation (mm)

J = polar moment of inertia of the cross section about the axis of rotation (mm^4)

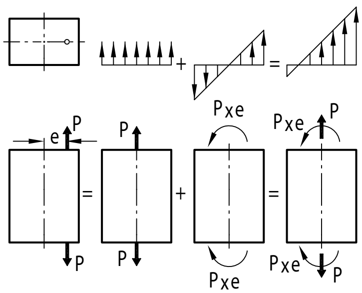
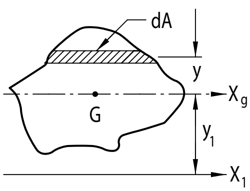
$$\theta = \frac{M_t\,l}{JG} \qquad (4.13)$$

θ = angle of twist (radians)

l = length of the shaft (mm)

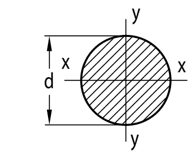
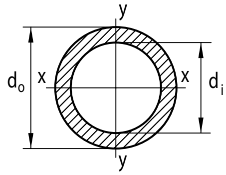
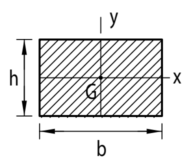
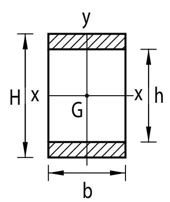
(*Contd.*)

(*Contd.*)

	The polar moment of inertia of a solid circular shaft of diameter d is given by $$J = \frac{\pi d^4}{32} \quad (4.14)$$ For a hollow circular cross section, $$J = \frac{\pi(d_o^4 - d_i^4)}{32} \quad (4.15)$$
Stresses due to eccentric axial load	
	$$\sigma = \frac{P}{A} \pm \frac{Pey}{I} \quad (4.16)$$ σ = resultant tensile stress (MPa or N/mm^2) P = eccentric axial force (N) e = eccentricity of force from centroidal axis (mm) A = cross-sectional area (mm^2) y = distance of the fibre from neutral axis (mm) I = moment of inertia of the cross section about the neutral axis (mm^4)
Parallel-axis theorem	
	$$I_{xg} = \int y^2\, dA \quad (4.17)$$ The parallel-axis theorem is given by, $$I_{x1} = I_{xg} + Ay_1^2 \quad (4.18)$$ I_{x1} = moment of inertia of the area about X_1 axis, which is parallel to axis X_g, and located at a distance y_1 from X_g I_{xg} = moment of inertia of the area about its own centroidal axis A = area of the cross section.

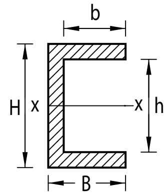
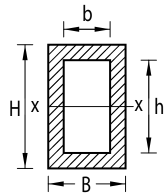
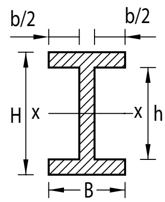
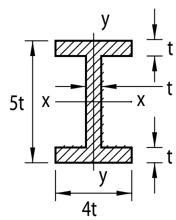
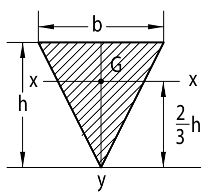
4.2 PROPERTIES OF CROSS SECTIONS

Table 4.2 *Properties of cross sections*

Form of cross section	Properties
	$I_{xx} = I_{yy} = \dfrac{\pi d^4}{64}$ $k_{xx} = k_{yy} = \dfrac{d}{4}$
	$I_{xx} = I_{yy} = \dfrac{\pi(d_o^4 - d_i^4)}{64}$ $k_{xx} = k_{yy} = \dfrac{\sqrt{d_i^2 + d_o^2}}{4}$
	$I_{xx} = \dfrac{bh^3}{12}$ $k_{xx} = \dfrac{h}{\sqrt{12}} = 0.289h$ $I_{yy} = \dfrac{hb^3}{12}$ $k_{yy} = \dfrac{b}{\sqrt{12}} = 0.289b$
	$I_{xx} = \dfrac{b(H^3 - h^3)}{12}$ $k_{xx} = \sqrt{\dfrac{(H^3 - h^3)}{12(H - h)}}$ $I_{yy} = \dfrac{(H - h)b^3}{12}$ $k_{yy} = \dfrac{b}{\sqrt{12}} = 0.289b$

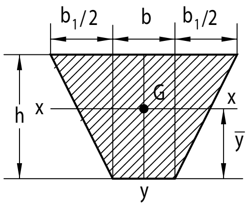
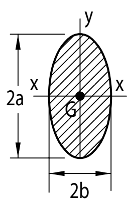
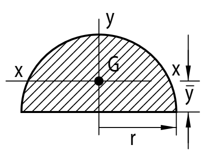
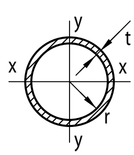
(*Contd.*)

(*Contd.*)

	$I_{xx} = \dfrac{BH^3 - bh^3}{12}$ $k_{xx} = \sqrt{\dfrac{BH^3 - bh^3}{12(BH - bh)}}$
	$I_{xx} = \dfrac{BH^3 - bh^3}{12}$ $k_{xx} = \sqrt{\dfrac{BH^3 - bh^3}{12(BH - bh)}}$
	$I_{xx} = \dfrac{BH^3 - bh^3}{12}$ $k_{xx} = \sqrt{\dfrac{BH^3 - bh^3}{12(BH - bh)}}$
	$I_{xx} = \left(\dfrac{419}{12}\right)t^4$ $k_{xx} = \sqrt{3.17}\,t = 1.78t$ $I_{yy} = \left(\dfrac{131}{12}\right)t^4$ $k_{xx} = \sqrt{0.992}\,t = 0.996t$
	$I_{xx} = \dfrac{bh^3}{36}$ $k_{xx} = \dfrac{h}{\sqrt{18}} = 0.235h$

(*Contd.*)

(*Contd.*)

	$\bar{y} = \left(\frac{h}{3}\right)\frac{(3b+2b_1)}{(2b+b_1)}$ $I_{xx} = h^3\left[\frac{(6b^2+6bb_1+b_1^2)}{36(2b+b_1)}\right]$ $k_{xx} = \frac{h}{3(2b+b_1)}\sqrt{\frac{6b^2+6bb_1+b_1^2}{2}}$
	$I_{xx} = \left(\frac{\pi}{4}\right)a^3b$ $k_{xx} = \frac{a}{2}$ $I_{yy} = \left(\frac{\pi}{4}\right)ab^3$ $k_{yy} = \frac{b}{2}$
	$\bar{y} = \frac{4r}{3\pi} = 0.4244r$ $I_{xx} = r^4\left(\frac{\pi}{8} - \frac{8}{9\pi}\right) = 0.1093r^4$ $k_{xx} = 0.2643r$ $I_{yy} = \frac{\pi r^4}{8}$ $k_{yy} = \frac{r}{2}$
	$I_{xx} = I_{yy} = \pi r^3 t$ $k_{xx} = k_{yy} = \frac{r}{\sqrt{2}} = 0.707r$ (*t* is too small)

Notations: –I = moment of inertia of cross section; k = radius of gyration; $k = \sqrt{\frac{I}{A}}$.

4.3 PROPERTIES OF ROLLED STEEL SECTIONS

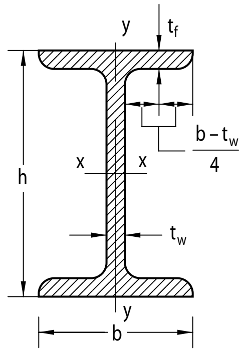


Table 4.3 *Properties of rolled steel beams*

Designation	Depth of Beam (h) (mm)	Width of Flange (b) (mm)	Thickness of Web (t_w) (mm)	Thickness of Flange (t_f) (mm)	Weight per metre (kgf)	Sectional area (cm^2)	Moment of Inertia		Radius of Gyration	
							I_{xx} (cm^4)	I_{yy} (cm^4)	k_{xx} (cm)	k_{yy} (cm)
ISLB 75	75	50	3.7	5.0	6.1	7.71	72.7	10.0	3.07	1.14
ISLB 100	100	50	4.0	6.4	8.0	10.21	168.0	12.7	4.06	1.12
ISLB 125	125	75	4.4	6.5	11.9	15.12	406.8	43.4	5.19	1.69
ISLB 150	150	80	4.8	6.8	14.2	18.08	688.2	55.2	6.17	1.75
ISLB 175	175	90	5.1	6.9	16.7	21.30	1096.2	79.6	7.17	1.93
ISLB 200	200	100	5.4	7.3	19.8	25.27	1696.6	115.4	8.19	2.13
ISLB 225	225	100	5.8	8.6	23.5	29.92	2501.9	112.7	9.15	1.94
ISLB 250	250	125	6.1	8.2	27.9	35.53	3717.8	193.4	10.23	2.33
ISLB 275	275	140	6.4	8.8	33.0	42.02	5375.3	287.0	11.31	2.61
ISLB 300	300	150	6.7	9.4	37.7	48.08	7332.9	376.2	12.35	2.80
ISLB 325	325	165	7.0	9.8	43.1	54.90	9874.6	510.8	13.41	3.05
ISLB 350	350	165	7.4	11.4	49.5	63.01	13158.3	631.9	14.45	3.17
ISLB 400	400	165	8.0	12.5	56.9	72.43	19306.3	716.4	16.33	3.15
ISLB 450	450	170	8.6	13.4	65.3	83.14	27536.1	853.0	18.20	3.20
ISLB 500	500	180	9.2	14.1	75.0	95.50	38579.0	1063.9	20.10	3.34
ISLB 550	550	190	9.9	15.0	86.3	109.97	53161.6	1335.1	21.99	3.48
ISLB 600	600	210	10.5	15.5	99.5	126.69	72867.6	1821.9	23.98	3.79
ISMB 100	100	75	4.0	7.2	11.5	14.60	257.5	40.8	4.20	1.67
ISMB 125	125	75	4.4	7.6	13.0	16.60	449.0	43.7	5.20	1.62
ISMB 150	150	80	4.8	7.6	14.9	19.00	726.4	52.6	6.18	1.66
ISMB 175	175	90	5.5	8.6	19.3	24.62	1272.0	85.0	7.19	1.86

(*Contd.*)

(*Contd.*)

ISMB 200	200	100	5.7	10.8	25.4	32.33	2235.4	150.0	8.32	2.15
ISMB 225	225	110	6.5	11.8	31.2	39.72	3441.8	218.3	9.31	2.34
ISMB 250	250	125	6.9	12.5	37.3	47.55	5131.6	334.5	10.39	2.65
ISMB 300	300	140	7.5	12.4	44.2	56.26	8603.6	453.9	12.37	2.84
ISMB 350	350	140	8.1	14.2	52.4	66.71	13630.3	537.7	14.29	2.84
ISMB 400	400	140	8.9	16.0	61.6	78.46	20458.4	622.1	16.15	2.82
ISMB 450	450	150	9.4	17.4	72.4	92.27	30390.8	834.0	18.15	3.01
ISMB 500	500	180	10.2	17.2	86.9	110.74	45218.3	1369.8	20.21	3.52
ISMB 550	550	190	11.2	19.3	103.7	132.11	64893.6	1833.8	22.16	3.73
ISMB 600	600	210	12.0	20.8	122.6	156.21	91813.0	2651.0	24.24	4.12

Note: ISLB = Indian Standard Light Beam; ISMB = Indian Standard Medium weight Beam; (1 kgf = 9.807 N)

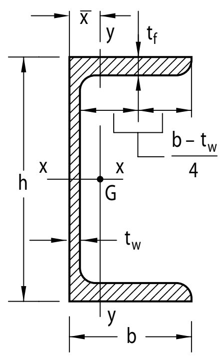


Table 4.4 *Properties of rolled steel channels*

Designation	Depth of Channel (h) (mm)	Width of Flange (b) (mm)	Thickness of Web (t_w) (mm)	Thickness of Flange (t_f) (mm)	Centre of Gravity $\bar{x}$ (cm)	Weight per metre (kgf)	Sectional area (cm^2)	Moment of Inertia		Radius of Gyration	
								I_{xx} (cm^4)	I_{yy} (cm^4)	k_{xx} (cm)	k_{yy} (cm)
ISJC 100	100	45	3.0	5.1	1.40	5.8	7.41	123.8	14.9	4.09	1.42
ISJC 125	125	50	3.0	6.6	1.64	7.9	10.08	270.0	25.7	5.18	1.60
ISJC 150	150	55	3.6	6.9	1.66	9.9	12.67	471.1	37.9	6.10	1.73
ISJC 175	175	60	3.6	6.9	1.75	11.2	14.24	719.9	50.5	7.11	1.88
ISJC 200	200	70	4.1	7.1	1.97	13.9	17.77	1161.2	84.2	8.08	2.18
ISLC 75	75	40	3.7	6.0	1.35	5.7	7.26	66.1	11.5	3.02	1.26
ISLC 100	100	50	4.0	6.4	1.62	7.9	10.02	164.7	24.8	4.06	1.57
ISLC 125	125	65	4.4	6.6	2.04	10.7	13.67	356.8	57.2	5.11	2.05
ISLC 150	150	75	4.8	7.8	2.38	14.4	18.39	697.2	103.2	6.16	2.37

(*Contd.*)

(*Contd.*)

ISLC 175	175	75	5.1	9.5	2.40	17.6	22.42	1148.4	126.5	7.16	2.38
ISLC 200	200	75	5.5	10.8	2.35	20.6	26.26	1725.5	146.9	8.11	2.37
ISLC 225	225	90	5.8	10.2	2.46	24.0	30.60	2547.9	209.5	9.14	2.62
ISLC 250	250	100	6.1	10.7	2.70	28.0	45.72	3687.9	298.4	10.17	2.89
ISLC 300	300	100	6.7	11.6	2.55	33.1	42.19	6047.9	346.0	11.98	2.87
ISLC 350	350	100	7.4	12.5	2.41	38.8	49.54	9312.6	394.6	13.72	2.82
ISLC 400	400	100	8.0	14.0	2.36	45.7	58.34	13989.5	460.4	15.5	2.81
ISMC 75	75	40	4.4	7.3	1.31	6.8	8.72	76.0	12.6	2.96	1.21
ISMC 100	100	50	4.7	7.5	1.53	9.2	11.75	186.7	25.9	4.00	1.49
ISMC 125	125	65	5.0	8.1	1.94	12.7	16.25	416.4	59.9	5.07	1.92
ISMC 150	150	75	5.4	9.0	2.22	16.4	20.94	779.4	102.3	6.11	2.21
ISMC 175	175	75	5.7	10.2	2.20	19.1	24.45	1223.3	121.0	7.08	2.23
ISMC 200	200	75	6.1	11.4	2.17	22.1	28.28	1819.3	140.4	8.03	2.23
ISMC 225	225	80	6.4	12.4	2.30	25.9	33.10	2694.6	187.2	9.03	2.38
ISMC 250	250	80	7.1	14.1	2.30	30.4	38.76	3816.8	219.1	9.94	2.38
ISMC 300	300	90	7.6	13.6	2.36	35.8	45.74	6362.6	310.8	11.81	2.61
ISMC 350	350	100	8.1	13.5	2.44	42.1	53.74	10008.0	430.6	13.66	2.83
ISMC 400	400	100	8.6	15.3	2.42	49.4	63.04	15082.8	504.8	15.48	2.83

Note: ISJC = Indian Standard Junior Channel; ISLC = Indian Standard Light Channel; ISMC = Indian Standard Medium weight Channel; (1 kgf = 9.807 N)

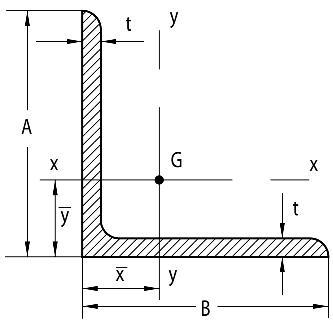


Table 4.5 *Properties of rolled steel equal angles*

Designation	Size $A \times B$ (mm × mm)	Thickness t (mm)	Sectional area (cm^2)	Weight per metre (kgf)	Moment of Inertia $I_{xx} = I_{yy}$ (cm^4)	Radius of Gyration $k_{xx} = k_{yy}$ (cm)	Centre of Gravity $\bar{x} = \bar{y}$ (cm)
ISA 2020	20 × 20	3.0	1.12	0.9	0.4	0.58	0.59
		4.0	1.45	1.1	0.5	0.58	0.63

(*Contd.*)

(*Contd.*)

ISA 2525	25 × 25	3.0	1.41	1.1	0.8	0.73	0.71
		4.0	1.84	1.4	1.0	0.73	0.75
		5.0	2.25	1.8	1.2	0.72	0.79
ISA 3030	30 × 30	3.0	1.73	1.4	1.4	0.89	0.83
		4.0	2.26	1.8	1.8	0.89	0.87
		5.0	2.77	2.2	2.1	0.88	0.92
ISA 3535	35 × 35	3.0	2.03	1.6	2.3	1.05	0.95
		4.0	2.66	2.1	2.9	1.05	1.00
		5.0	3.27	2.6	3.5	1.04	1.04
		6.0	3.86	3.0	4.1	1.03	1.08
ISA 4040	40 × 40	3.0	2.34	1.8	3.4	1.21	1.08
		4.0	3.07	2.4	4.5	1.21	1.12
		5.0	3.78	3.0	5.4	1.20	1.16
		6.0	4.47	3.5	6.3	1.19	1.20
ISA 4545	45 × 45	3.0	2.64	2.1	5.0	1.38	1.20
		4.0	3.47	2.7	6.5	1.37	1.25
		5.0	4.28	3.4	7.9	1.36	1.29
		6.0	5.07	4.0	9.2	1.35	1.33
ISA 5050	50 × 50	3.0	2.95	2.3	6.9	1.53	1.32
		4.0	3.88	3.0	9.1	1.53	1.37
		5.0	4.79	3.8	11.0	1.52	1.41
		6.0	5.68	4.5	12.9	1.51	1.45
ISA 5555	55 × 55	5.0	5.27	4.1	14.7	1.67	1.53
		6.0	6.26	4.9	17.3	1.66	1.57
		8.0	8.18	6.4	22.0	1.64	1.65
		10.0	10.02	7.9	26.3	1.62	1.72
ISA 6060	60 × 60	5.0	5.75	4.5	19.2	1.82	1.65
		6.0	6.84	5.4	22.6	1.82	1.69
		8.0	8.96	7.0	29.0	1.80	1.77
		10.0	11.00	8.6	34.8	1.78	1.85
ISA 6565	65 × 65	5.0	6.25	4.9	24.7	1.99	1.77
		6.0	7.44	5.8	29.1	1.98	1.81
		8.0	9.76	7.7	37.4	1.96	1.89
		10.0	12.00	9.4	45.0	1.94	1.97
ISA 7070	70 × 70	5.0	6.77	5.3	31.1	2.15	1.89
		6.0	8.06	6.3	36.8	2.14	1.94
		8.0	10.58	8.3	47.4	2.12	2.02
		10.0	13.02	10.2	57.2	2.10	2.10

(*Contd.*)

(*Contd.*)

ISA 7575	75 × 75	5.0	7.27	5.7	38.7	2.31	2.02
		6.0	8.66	6.8	45.7	2.30	2.06
		8.0	11.38	8.9	59.0	2.28	2.14
		10.0	14.02	11.0	71.4	2.26	2.22
ISA 8080	80 × 80	6.0	9.29	7.3	56.0	2.46	2.18
		8.0	12.21	9.6	72.5	2.44	2.27
		10.0	15.05	11.8	87.7	2.41	2.34
		12.0	17.81	14.0	101.9	2.39	2.42
ISA 9090	90 × 90	6.0	10.47	8.2	80.1	2.77	2.42
		8.0	13.79	10.8	104.2	2.75	2.51
		10.0	17.03	13.4	126.7	2.73	2.59
		12.0	20.19	15.8	147.9	2.71	2.66
ISA 100100	100 × 100	6.0	11.67	9.2	111.3	3.09	2.67
		8.0	15.39	12.1	145.1	3.07	2.76
		10.0	19.03	14.9	177.0	3.05	2.84
		12.0	22.59	17.7	207.0	3.03	2.92
ISA 110110	110 × 110	8.0	17.02	13.4	195.0	3.38	3.00
		10.0	21.06	16.5	238.4	3.36	3.08
		12.0	25.02	19.6	279.6	3.34	3.16
		15.0	30.81	24.2	337.4	3.31	3.27
ISA 130130	130 × 130	8.0	20.22	15.9	328.3	4.03	3.50
		10.0	25.06	19.7	402.7	4.01	3.58
		12.0	29.82	23.4	473.8	3.99	3.66
		15.0	36.81	28.9	574.6	3.95	3.78
ISA 150150	150 × 150	10.0	29.03	22.8	622.4	4.63	4.06
		12.0	34.59	27.2	735.4	4.61	4.14
		15.0	42.78	33.6	896.8	4.58	4.26
		18.0	50.79	39.9	1048.9	4.54	4.38
ISA 200200	200 × 200	12.0	46.61	36.6	1788.9	6.20	5.36
		15.0	57.80	45.4	2197.7	6.17	5.49
		18.0	68.81	54.0	2588.7	6.13	5.61
		25.0	93.80	73.6	3436.3	6.05	5.88

Note: ISA = Indian Standard Angle

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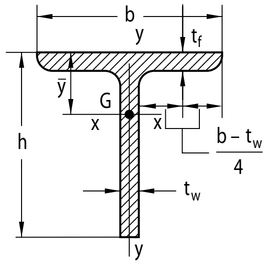


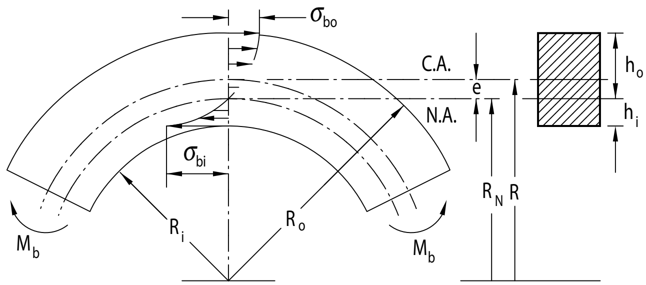
Table 4.6 *Properties of rolled steel bars (Tee section)*

Designation	Weight per metre (kgf)	Sectional area (cm^2)	Depth of Section (*h*) (mm)	Width of Flange (*b*) (mm)	Thickness of Flange (t_f) (mm)	Thickness of Web (t_w) (mm)	Centre of Gravity $\bar{y}$ (cm)	Moment of Inertia		Radius of Gyration	
								I_{xx} (cm^4)	I_{yy} (cm^4)	k_{xx} (cm)	k_{yy} (cm)
ISNT 20	1.1	1.45	20	20	4.0	4.0	0.60	0.5	0.2	0.58	0.41
ISNT 30	1.8	2.26	30	30	4.0	4.0	0.82	1.8	0.8	0.89	0.59
ISNT 40	3.5	4.45	40	40	6.0	6.0	1.14	6.1	2.9	1.18	0.81
ISNT 50	4.4	5.66	50	50	6.0	6.0	1.35	12.3	5.7	1.47	1.01
ISNT 60	5.4	6.85	60	60	6.0	6.0	1.56	21.4	9.7	1.77	1.19
ISNT 75	10.0	6.90	75	75	9.0	9.0	2.04	62.0	29.2	2.21	1.52
ISNT 100	14.9	18.97	100	100	10.0	10.0	2.62	163.9	76.8	2.94	2.01
ISNT 150	22.7	28.88	150	150	10.0	10.0	3.61	541.1	250.3	4.33	2.94
ISDT 100	8.1	10.37	100	50	10.0	5.8	3.03	99.0	9.6	3.09	0.96
ISDT 150	15.7	19.96	150	75	11.6	8.0	4.75	450.2	37.0	4.75	1.36
ISLT 200	28.4	36.22	200	165	12.5	8.0	4.78	1267.8	358.2	5.92	3.15
ISLT 250	37.5	47.75	250	180	14.1	9.2	6.40	2774.4	532.0	7.62	3.34
ISMT 50	5.7	7.30	50	75	7.2	4.0	0.96	9.7	20.4	1.15	1.67
ISMT 62.5	6.5	8.30	62.5	75	7.6	4.4	1.30	21.3	21.9	1.60	1.62
ISMT 75	7.5	9.50	75	80	7.6	4.8	1.67	40.1	26.3	2.05	1.66
ISMT 87.5	9.7	12.31	87.5	90	8.6	5.5	1.98	72.6	42.5	2.43	1.86
ISMT 100	12.7	16.16	100	100	10.8	5.7	2.13	115.8	75.0	2.68	2.15
ISHT 75	15.3	19.49	75	150	9.0	8.4	1.62	96.2	230.2	2.22	3.44
ISHT 100	20.0	25.47	100	200	9.0	7.8	1.91	193.8	497.3	2.76	4.42
ISHT 125	27.4	34.85	125	250	9.7	8.8	2.37	415.4	1005.8	3.45	5.37
ISHT 150	29.4	37.42	150	250	10.6	7.6	2.66	573.7	1096.8	3.92	5.41

Note: ISNT = Indian Standard Normal Tee bars; ISDT = Indian Standard Deep legged Tee bars; ISLT = Indian Standard slit Light weight Tee bars; ISMT = Indian Standard slit Medium weight Tee bars; ISHT = Indian Standard slit bars from H-section.

4.4 CURVED BEAMS

Table 4.7 *Bending stresses in curved beams*



Notations

R_o = radius of outer fibre (mm)
R_i = radius of inner fibre (mm)
R = radius of centroidal axis (mm)
R_N = radius of neutral axis (mm)
h_i = distance of inner fibre from neutral axis (mm)
h_o = distance of outer fibre from neutral axis (mm)
M_b = bending moment with respect to centroidal axis (N-mm)
A = area of the cross section (mm^2)
C.A. = centroidal axis
N.A.= neutral axis
e = eccentricity between centroidal and neutral axes

$$e = R - R_N \tag{4.19}$$

The bending stress (σ_b) at a fibre, which is at a distance of 'y' from the neutral axis is given by,

$$\sigma_b = \frac{M_b y}{Ae(R_N - y)} \tag{4.20}$$

The equation indicates the hyperbolic distribution of (σ_b) with respect to 'y' The bending stress at the inner fibre is given by,

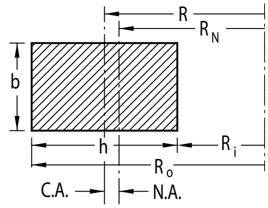
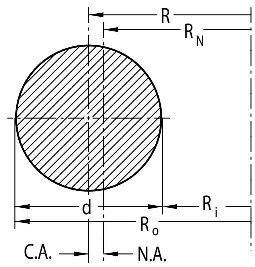
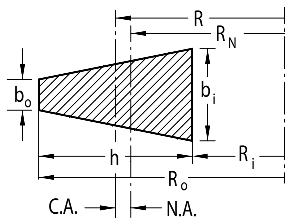
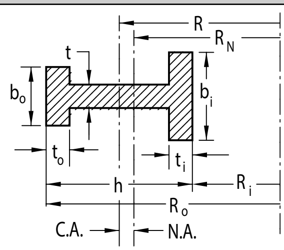
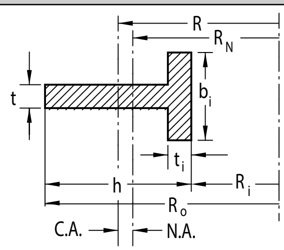
$$\sigma_{bi} = \frac{M_b h_i}{AeR_i} \tag{4.21}$$

The bending stress at the outer fibre is given by,

$$\sigma_{bo} = \frac{M_b h_o}{AeR_o} \tag{4.22}$$

Note: In symmetrical cross sections, such as circular or rectangular, the maximum bending stress always occurs at the inner fibre. In unsymmetrical cross sections, it is necessary to calculate the stresses at the inner as well as outer fibres to determine the maximum stress. In most of the engineering problems, the magnitude of 'e' is very small and it should be calculated precisely to avoid large percentage error in the final results.

Table 4.8 *Location of centroidal and neutral axes in curved beams*

Rectangular cross section	
	$R = R_i + \frac{h}{2}$ $R_N = \frac{h}{\log_e\left(\frac{R_o}{R_i}\right)}$ $e = R - R_N$
Circular cross section	
	$R = R_i + \frac{d}{2}$ $R_N = \frac{(\sqrt{R_o} + \sqrt{R_i})^2}{4}$ $e = R - R_N$
Trapezoidal cross section	
	$R = R_i + \frac{h(b_i + 2b_o)}{3(b_i + b_o)}$ $R_N = \frac{\left(\frac{b_i + b_o}{2}\right)h}{\left(\frac{b_i R_o - b_o R_i}{h}\right)\log_e\left(\frac{R_o}{R_i}\right) - (b_i - b_o)}$ $e = R - R_N$
I –section	
	$R = R_i + \frac{\frac{1}{2}th^2 + \frac{1}{2}t_i^2(b_i - t) + t_o(b_o - t)(h - t_o/2)}{t_i(b_i - t) + t_o(b_o - t) + th}$ $R_N = \frac{t_i(b_i - t) + t_o(b_o - t) + th}{b_i \log_e\left(\frac{R_i + t_i}{R_i}\right) + t \log_e\left(\frac{R_o - t_o}{R_i + t_i}\right) + b_o \log_e\left(\frac{R_o}{R_o - t_o}\right)}$ $e = R - R_N$
Tee section	
	$R = R_i + \frac{\frac{1}{2}th^2 + \frac{1}{2}t_i^2(b_i - t)}{th + t_i(b_i - t)}$ $R_N = \frac{t_i(b_i - t) + th}{(b_i - t)\log_e\left(\frac{R_i + t_i}{R_i}\right) + t \log_e\left(\frac{R_o}{R_i}\right)}$ $e = R - R_N$

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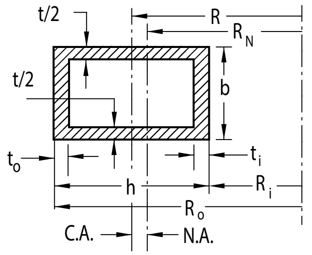
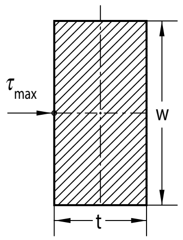
Box section	
	$R = R_i + \dfrac{\frac{1}{2}h^2 t + \frac{1}{2}t_i^2 (b-t) + (b-t)(t_o)\left(h - \frac{1}{2}t_o\right)}{ht + (b-t)(t_i + t_o)}$ $R_N = \dfrac{(b-t)(t_i + t_o) + th}{b\left[\log_e\left(\dfrac{R_i + t_i}{R_i}\right) + \log_e\left(\dfrac{R_o}{R_o - t_o}\right)\right] + t\log_e\left(\dfrac{R_o - t_o}{R_i + t_i}\right)}$ $e = R - R_N$

4.5 TORSION OF NON-CIRCULAR BARS

Table 4.9 *Torsion of rectangular-section bar*

Shape	Equations
	Maximum shear stress $$\tau_{max} = \frac{M_t}{\alpha w t^2} \cong \frac{M_t}{wt^2}\left[3 + \frac{1.8}{(w/t)}\right] \quad (4.23)$$ Angle of twist $$\theta = \frac{M_t l}{\beta w t^3 G} \quad (4.24)$$ Notations: τ_{max} = maximum torsional shear stress (MPa or N/mm^2) θ = angle of twist (radians) M_t = applied torque (N-mm) w = longer side of rectangle (mm) t = shorter side of rectangle (mm) l = length of bar (mm) α = factor depending upon the ratio (w/t) (Table 4.10) β = factor depending upon the ratio (w/t) (Table 4.10) G = Modulus of rigidity (MPa or N/mm^2)

Note: (i) The maximum shear stress occurs at in the middle of longest side '*w*'.

(ii) *w* (long side) and *t* (short side) are non interchangeable, and '*t*' is always shortest dimension.

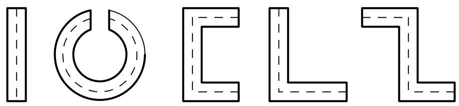
(iii) For thin plates, (t/w) is small and the second term of stress equation, namely, $\left[\dfrac{1.8}{(w/t)}\right]$ may be neglected. [Assumption, $(w/t) = \infty$ or $(t/w) = 0$]

(iv) The stress equation is approximately valid for equal-sided angle-sections. In this case, angle-section is considered as two rectangles, each is transmitting half the torque.

Table 4.10 *Values of α and β factors*

w/t	1.00	1.50	1.75	2.00	2.50	3.00	4.00	6.00	8.00	10	∞
α	0.208	0.231	0.239	0.246	0.258	0.267	0.282	0.299	0.307	0.313	0.333
β	0.141	0.196	0.214	0.228	0.249	0.263	0.281	0.299	0.307	0.313	0.333

Table 4.11 *Torsion of open thin-walled sections*



Equations:	Notations:
$$\tau_{\max} = \frac{3M_t}{Lt^2} \quad (4.25)$$ $$\theta = \frac{3M_t l}{Lt^3 G} \quad (4.26)$$	$\tau_{\max}$ = maximum torsional shear stress (MPa or N/mm^2) θ = angle of twist (radians) M_t = applied torque (N-mm) L = length of median line (mm) t = wall thickness (mm) l = length of section (mm) G = Modulus of rigidity (MPa or N/mm^2)

4.6 BUCKLING OF COLUMNS

Table 4.12 *Critical buckling load*

Euler's equation	
$$P_{\text{cr}} = \frac{n\pi^2 EA}{(l/k)^2} \quad (4.27)$$ **Note:** Euler's Equation is used for long columns.	Notations: P_{cr} = critical load (at which buckling starts) (N) n = end fixity coefficient (Table 4.13) E = modulus of elasticity (MPa or N/mm^2) A = area of cross section (mm^2) (l/k) = slenderness ratio l = length of column (mm) k = least radius of gyration of cross section about its axis (mm) $k = \sqrt{\frac{I}{A}}$ I = least moment of inertia of cross-section (mm^4)
Johnson's equation	
$$P_{cr} = S_{yc} A \left[1 - \frac{S_{yc}}{4n\pi^2 E}\left(\frac{l}{k}\right)^2\right] \quad (4.28)$$ **Note:** Johnson's Equation is used for short columns.	Notations: S_{yc} = compressive yield strength of material (MPa or N/mm^2)

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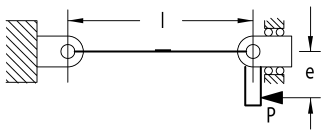
Rankine's equation	
$$P_{cr} = \frac{S_{yc} A}{1 + a\left(\frac{l_e}{k}\right)^2} \quad (4.29)$$	S_{yc} = yield stress in compression (MPa or N/mm^2) (Table 4.14) a = Rankine's Constant (Table 4.14) $a = \frac{S_{yc}}{\pi^2 E}$ l_e = equivalent length of column (mm) $(l_e)^2 = \left(\frac{l^2}{n}\right)$
Secant formula	
	
$$P_{cr} = \frac{S_{yc} A}{1 + \left(\frac{ec}{k^2}\right)\sec\left[\left(\frac{l}{2k}\right)\sqrt{\frac{P}{AE}}\right]} \quad (4.30)$$ **Note:** Secant formula is used for columns subjected to eccentric axial load	e = eccentricity of load (or distance between centroidal axis of cross-section and line of action of force (mm) c = distance of extreme fibre on outer surface from neutral axis of the cross-section of column (mm) eccentricity ratio = $\left(\frac{ec}{k^2}\right)$

Table 4.13 *Values of end fixity coefficient (n)*

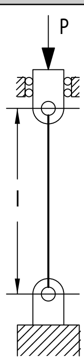
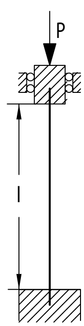
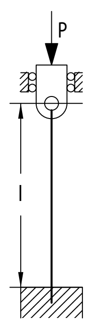
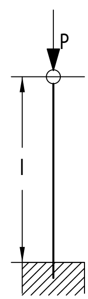
Both ends hinged (or rounded)	Both ends fixed	One end fixed and other end hinged (or rounded)	One end fixed and other end free
			
Theoretical value $n = 1$	Theoretical value $n = 4$	Theoretical value $n = 2$	Theoretical value $n = ¼$
Conservative value $n = 1$	Conservative value $n = 1$	Conservative value $n = 1$	Conservative value $n = ¼$
Recommended value $n = 1$	Recommended value $n = 1.2$	Recommended value $n = 1.2$	Recommended value $n = 1/4$

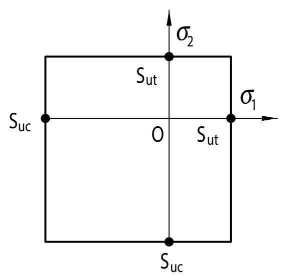
Table 4.14 *Values of (S_{yc}) and (a) in Rankine's equation*

Material	S_{yc} (MPa or N/mm^2)	Rankine's constant 'a' $a = \dfrac{S_{yc}}{\pi^2 E}$
Wrought iron	250	$\dfrac{1}{9000}$
Cast iron	550	$\dfrac{1}{1600}$
Mild steel	320	$\dfrac{1}{7500}$

4.7 THEORIES OF FAILURE

Table 4.15 *Theories of failure*

Maximum principal stress theory



Statement–The theory states that the failure of the mechanical component subjected to bi-axial or tri-axial stresses occurs when the maximum principal stress reaches the yield or ultimate strength of the material.

Note: (i) This theory is also known as Rankine's theory.

(ii) This theory is mainly applicable to components made of brittle materials like cast iron.

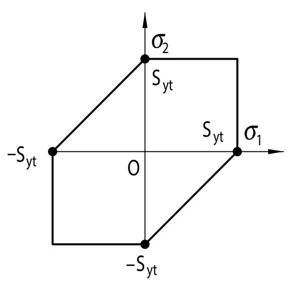
Design Equations

For tensile stresses, $\sigma_t = \dfrac{S_{ut}}{(fs)}$ or $\sigma_t = \dfrac{S_{yt}}{(fs)}$ (4.31)

For compressive stresses,

$$\sigma_c = \frac{S_{uc}}{(fs)} \text{ or } \sigma_c = \frac{S_{yc}}{(fs)} \tag{4.32}$$

Maximum shear stress theory



Statement–The theory states that the failure of a mechanical component subjected to bi-axial or tri-axial stresses occurs when the maximum shear stress at any point in the component becomes equal to the maximum shear stress in the standard specimen of the tension test, when yielding starts.

Note: (i) This theory is also known as Coulomb's theory.

(ii) This theory is mainly applicable to components made of ductile materials like steel.

(iii) According to this theory, the yield strength in shear is half of the yield strength in tension.

$$S_{sy} = 0.5\, S_{yt} \tag{4.33}$$

Design Equation

$$\tau_{\max} = \frac{S_{sy}}{(fs)} = \frac{0.5\, S_{yt}}{(fs)} \tag{4.34}$$

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<table>
<tr><th colspan="2">Distortion-energy theory</th></tr>
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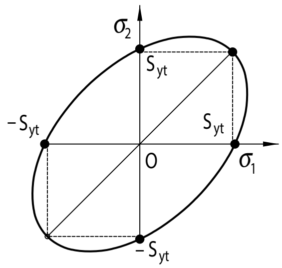

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Statement–The theory states that the failure of the mechanical component subjected to bi-axial or tri-axial stresses occurs, when the strain energy of distortion per unit volume at any point in the component, becomes equal to the strain energy of distortion per unit volume in a standard specimen of tension test, when yielding starts.

Note: (i) This theory is also known as Huber von Mises and Hencky's theory.

(ii) This theory is mainly applicable to components made of ductile materials like carbon steel.

(iii) According to this theory, the yield strength in shear is 0.577 times of the yield strength in tension.

$$S_{sy} = 0.577\, S_{yt} \tag{4.35}$$

Design Equation

$$\tau_{\max} = \frac{S_{sy}}{(fs)} = \frac{0.577\, S_{yt}}{(fs)} \tag{4.36}$$

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Note: The figures in left column of the table show safety regions for components subjected to bi-axial stresses.

4.8 FACTOR OF SAFETY

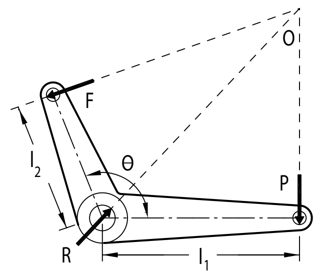
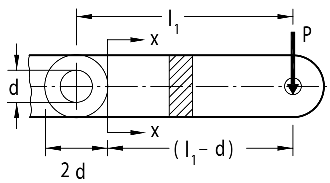
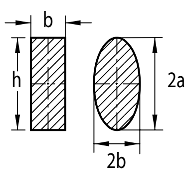
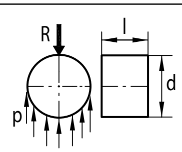
Table 4.16 *Guidelines for selection of factor of safety*

Component	Factor of safety
Components made of brittle material like cast iron	3 to 5 based on ultimate tensile strength of material
Components made of ductile material like steel and subjected to external static forces	1.5 to 2 based on yield strength of material
Components made of ductile material like steel and subjected to external fluctuating forces	1.3 to 1.5 based on endurance limit of component
Components made of ductile material like steel and subjected to contact stresses such as gears, cams, rolling contact bearings or rail and wheel	1.8 to 2.5 based on surface endurance limit of component
Components subjected to buckling such as piston rod, power screws or studs	3 to 5 based on critical buckling load (P_{cr}) of component

Note: The above values of factor of safety are given for guidance of students only.

4.9 DESIGN OF LEVERS

Table 4.17 *Design of levers*

	Force analysis $$F \times l_2 = P \times l_1 \quad (4.37)$$ F = load (N) P = effort (N) l_1= effort arm (mm) l_2 = load arm (mm) $$R = \sqrt{F^2 + P^2 - 2FP\cos\theta} \quad (4.38)$$ θ = angle included between two arms of bell-crank lever (deg) When arms of bell-crank lever are at right angles, $$\cos\theta = 0 \text{ and } R = \sqrt{F^2 + P^2} \quad (4.39)$$
	Design of lever arm $M_b = P(l_1 - d)$ M_b= maximum bending moment at section xx (N-mm) d = diameter of fulcrum pin (mm) $\sigma_b = \dfrac{M_b y}{I}$ σ_b = maximum bending stress in lever (MPa or N/mm^2) $\sigma_b = \dfrac{S_{yt}}{(fs)}$ or $\sigma_b = \dfrac{S_{ut}}{(fs)}$
	For rectangular cross section, $I = \dfrac{bh^3}{12}$ and $y = \dfrac{h}{2}$ For elliptical cross section, $I = \left(\dfrac{\pi}{4}\right)a^3 b$ and $y = a$
	Design of fulcrum pin $$R = p(dl) \quad (4.40)$$ p = permissible bearing pressure between bush and pin (5 to 10 MPa or N/mm^2) d = diameter of pin (mm) l = length of pin (mm) $\dfrac{l}{d} = 1$ to 2

Fluctuating Stresses

5

5.1 STRESS CONCENTRATION FACTORS

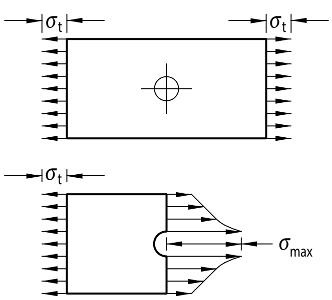
Table 5.1 *Stress concentration factor*

<table>
<tr><td colspan="2">

$$K_t = \frac{\text{Highest value of actual stress near discontinuity}}{\text{Nominal stress obtained by elementary equations}}$$

$$K_t = \frac{\sigma_{\max}}{\sigma_0} = \frac{\tau_{\max}}{\tau_0} \tag{5.1}$$

</td></tr>
<tr><td>

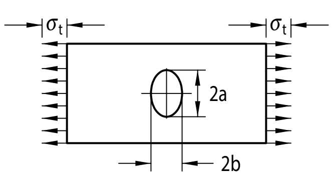


</td><td>

K_t = theoretical stress concentration factor
$\sigma_{\max}$, $\tau_{\max}$ = maximum localized stresses at the discontinuities (MPa or N/mm^2)
σ_o, τ_o = stresses determined by elementary equations for minimum cross-section (MPa or N/mm^2)
Elementary Equations

$$\sigma_o = \frac{P}{A} \qquad \sigma_o = \frac{M_b y}{I} \qquad \tau_o = \frac{M_t r}{J}$$

</td></tr>
<tr><td>



</td><td>

Stress concentration factor for flat plate with elliptical hole:

$$K_t = 1 + 2\left(\frac{a}{b}\right) \tag{5.2}$$

a = half width (or semi-axis) of ellipse perpendicular to the direction of load
b = half width (or semi-axis) of ellipse in the direction of load

As 'b' approaches zero, the ellipse becomes sharper and sharper. A very sharp crack is indicated and the stress at the edge of crack becomes very large. It is observed from above equation,

$K_t = \infty$ when $b = 0$

Therefore, as the width of elliptical hole in the direction of load approaches zero, stress concentration factor becomes infinity.

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</table>

(*Contd.*)

(*Contd.*)

	The ellipse becomes circle when ($a = b$). $$K_t = 1 + 2\left(\frac{a}{b}\right) = 1 + 2 = 3 \quad (5.3)$$ Therefore, theoretical stress concentration factor due to a small circular hole in a flat plat, which is subjected to tensile force, is 3.

Note: The values of stress concentration factor (K_t) obtained from charts or calculated by polynomial or exponential expressions from Tables 5.2 to 5.29 are approximate.

Table 5.2 *Rectangular plate with transverse hole in axial tension—Stress concentration chart*

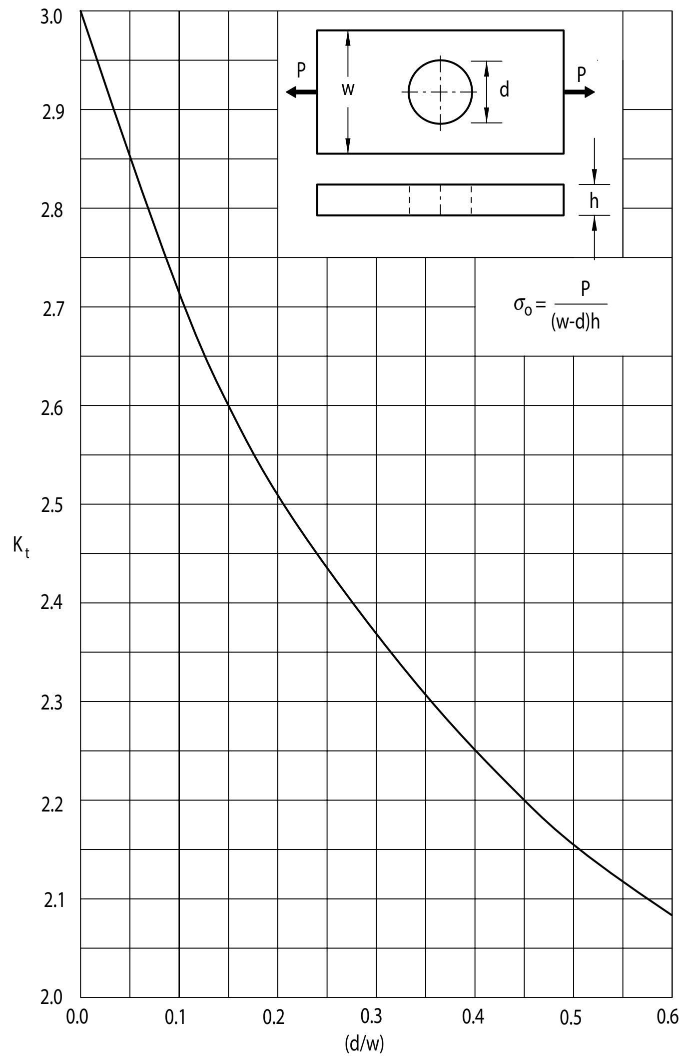


Table 5.3 *Rectangular plate with transverse hole in axial tension – Polynomial relationship for chart*

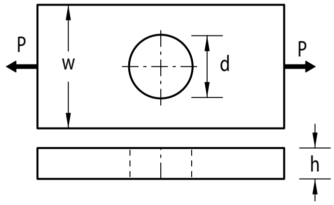
	Polynomial relationship
	For $\left(\dfrac{d}{w}\right) \leq 0.65$, $K_t \approx \left[3.00 - 3.75\left(\dfrac{d}{w}\right) + 7.97\left(\dfrac{d}{w}\right)^2 - 9.27\left(\dfrac{d}{w}\right)^3 + 1.81\left(\dfrac{d}{w}\right)^4 + 2.97\left(\dfrac{d}{w}\right)^5\right]$

Table 5.4 *Rectangular plate with transverse hole in bending—Stress concentration charts*

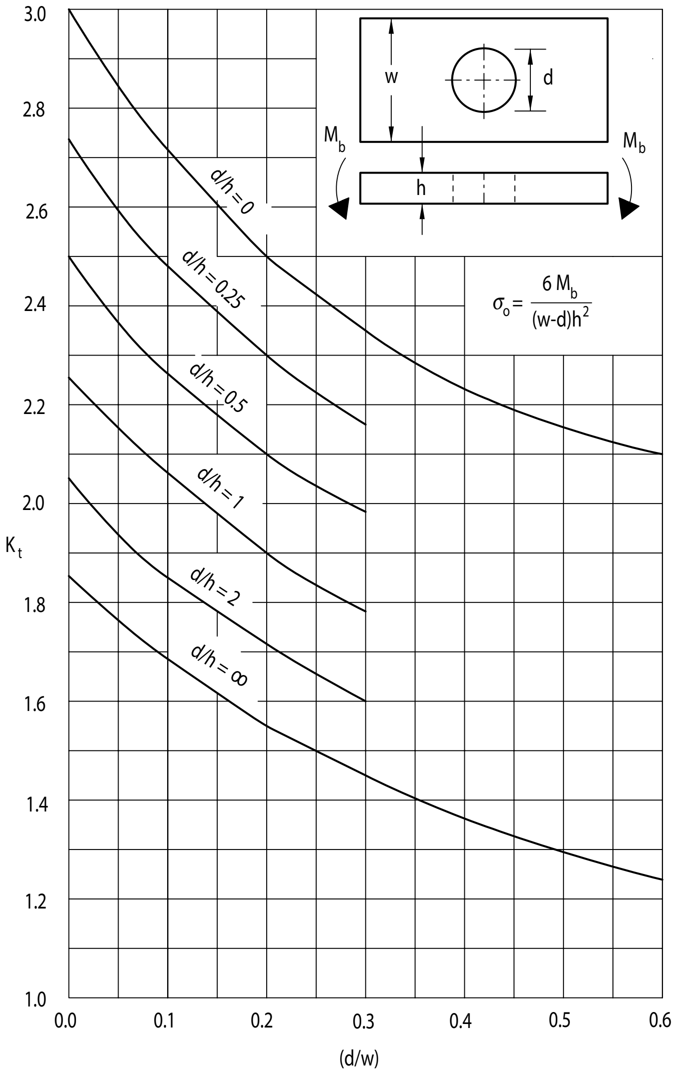
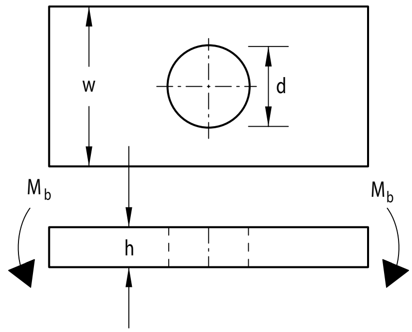


Table 5.5 *Rectangular plate with transverse hole in bending – Polynomial relationships for chart*



Polynomial relationships

For $\left(\frac{d}{w}\right) \le 0.65$,

$$K_t \approx \left[3.00 - 3.48\left(\frac{d}{w}\right) + 5.83\left(\frac{d}{w}\right)^2 - 4.20\left(\frac{d}{w}\right)^3\right]$$

For $\left(\frac{d}{h}\right) \ge 0.25$,

$$K_t \approx A e^{[b(d/w)]}$$

where,

(d/h)	**A**	**b**
0.25	2.69	–0.75
0.50	2.47	–0.77
1.00	2.24	–0.79
1.50	2.02	–0.81
2.00	2.11	–0.80
∞	1.81	–0.67

Table 5.6 *Flat plate with shoulder fillet in axial tension—Stress concentration charts*

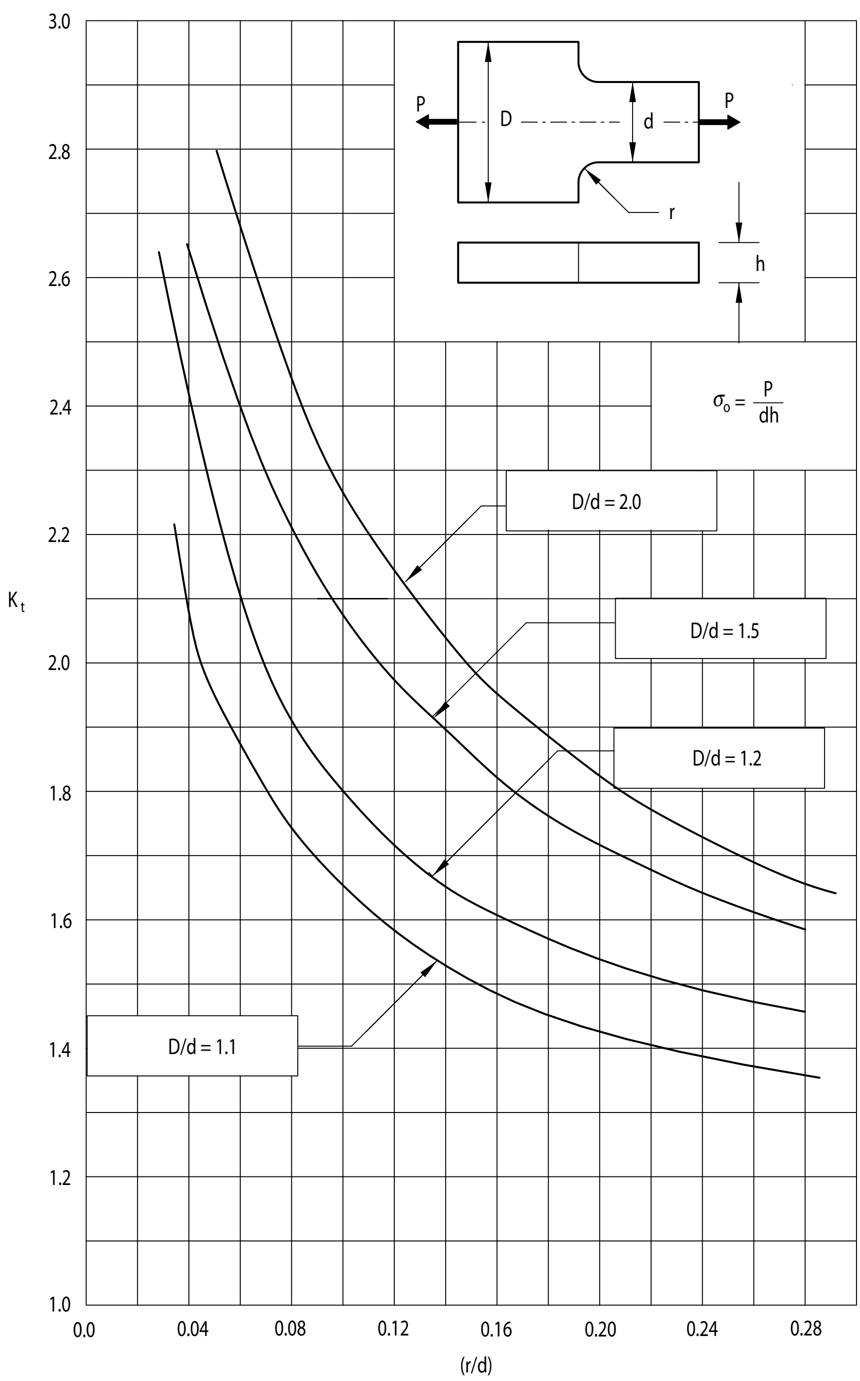
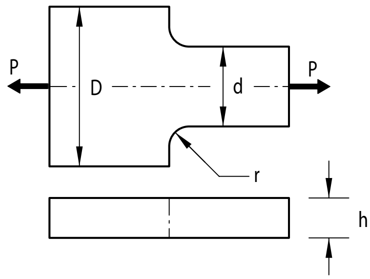


Table 5.7 *Flat plate with shoulder fillet in axial tension – Exponential relationship for charts*



Exponential relationship

$$K_t \approx A\left(\frac{r}{d}\right)^b$$

where

(*D*/*d*)	**A**	**b**
2.00	1.10	–0.32
1.50	1.08	–0.30
1.30	1.05	–0.27
1.20	1.04	–0.25
1.15	1.01	–0.24
1.10	1.01	–0.22
1.07	1.01	–0.19
1.05	0.99	–0.14
1.02	1.03	–0.17
1.01	0.98	–0.11

Table 5.8 *Flat plate with shoulder fillet in bending—Stress concentration charts*

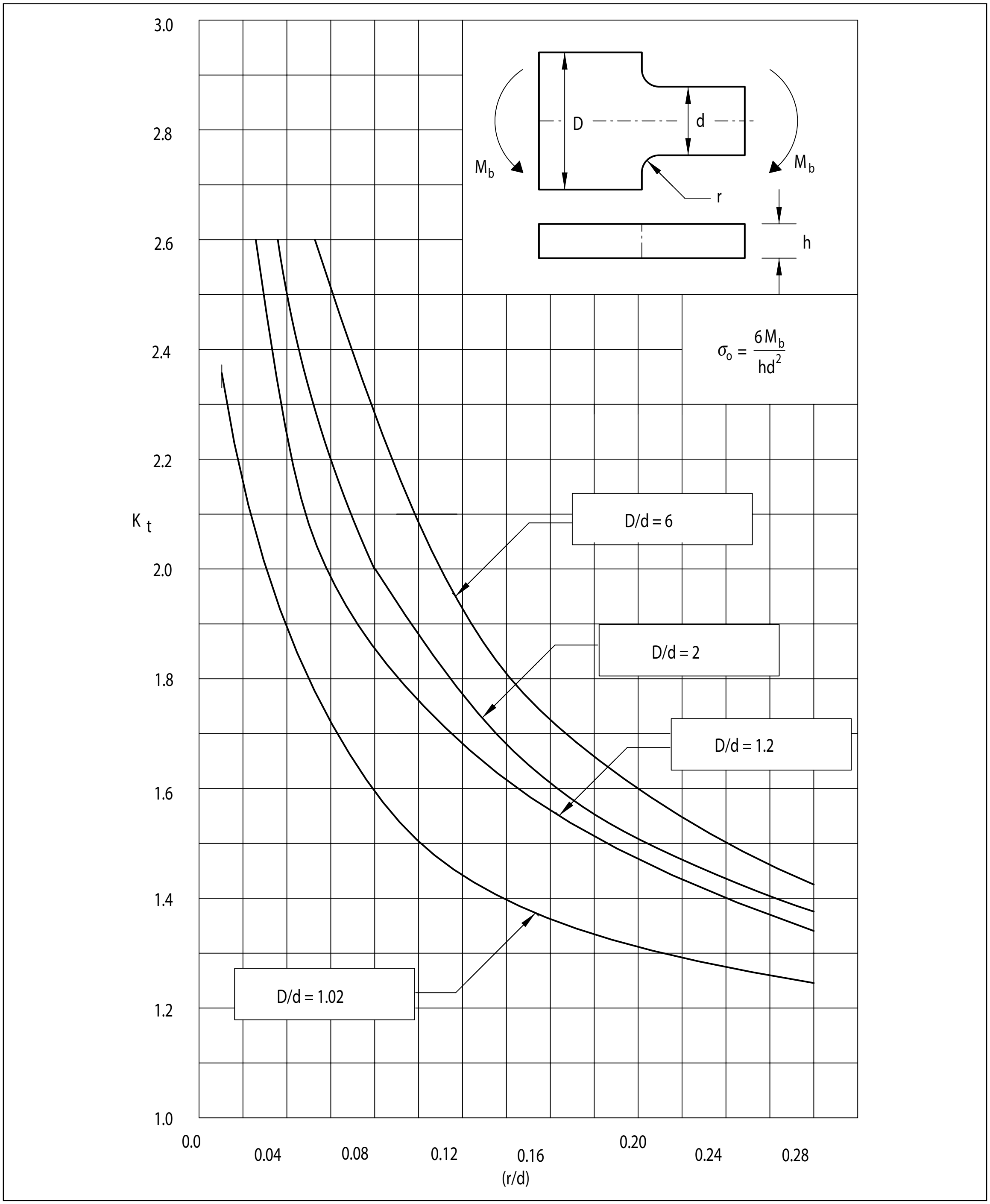
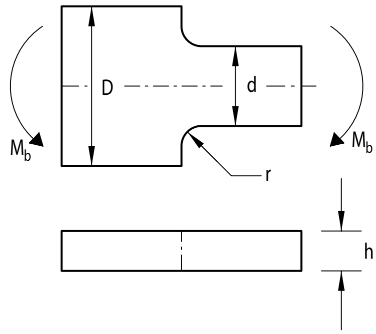


Table 5.9 *Flat plate with shoulder fillet in bending – Exponential relationship for charts*



Exponential relationship

$$K_t \approx A\left(\frac{r}{d}\right)^b$$

where

(*D*/*d*)	**A**	**b**
6.00	0.90	–0.36
3.00	0.91	–0.33
2.00	0.93	–0.30
1.30	0.96	–0.27
1.20	1.00	–0.24
1.10	1.02	–0.22
1.07	1.02	–0.20
1.05	1.02	–0.19
1.03	1.02	–0.18
1.02	1.00	–0.17
1.01	0.97	–0.15

Table 5.10 *Flat plate with semi-circular notch in axial tension —Stress concentration charts*

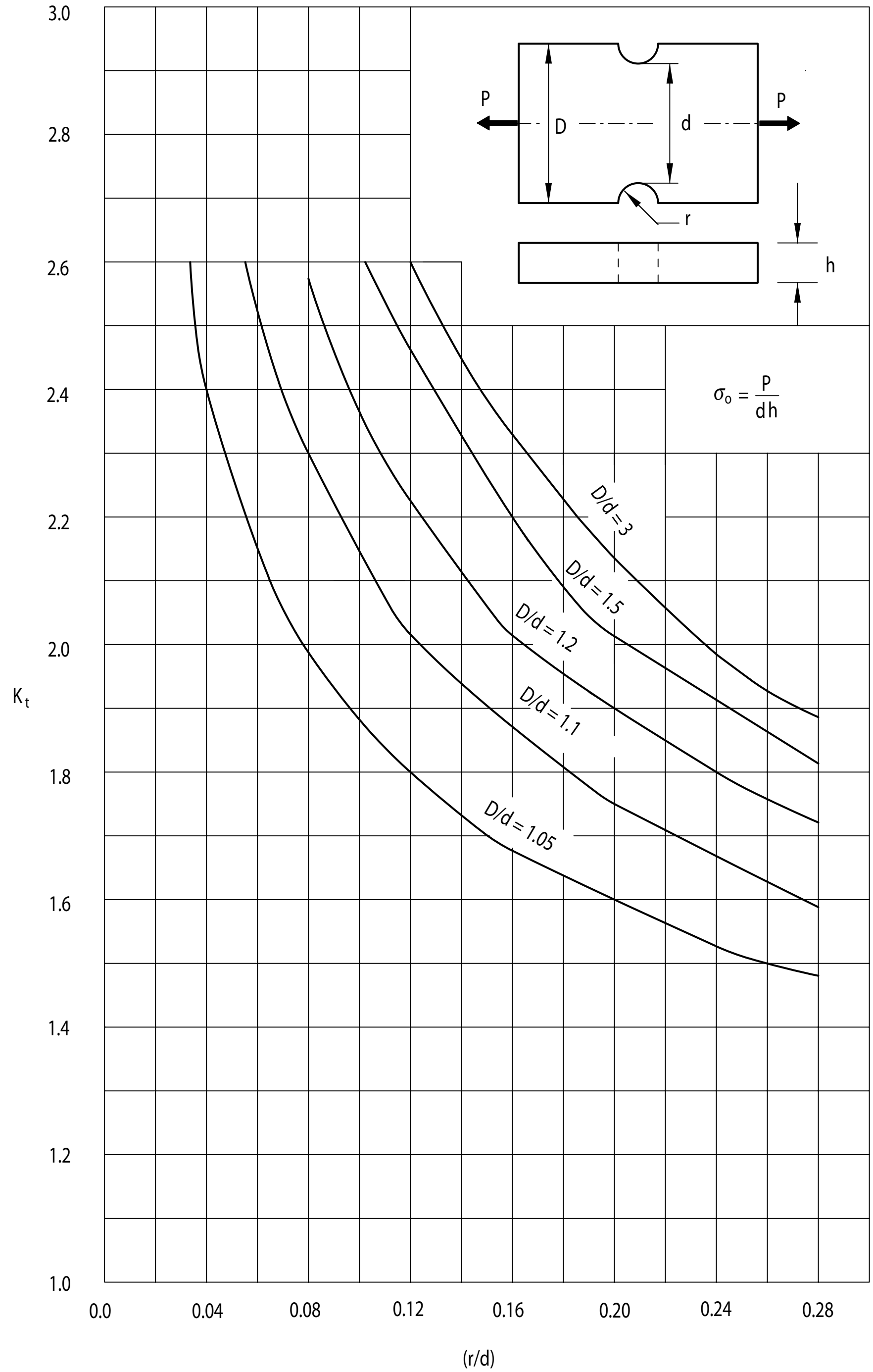
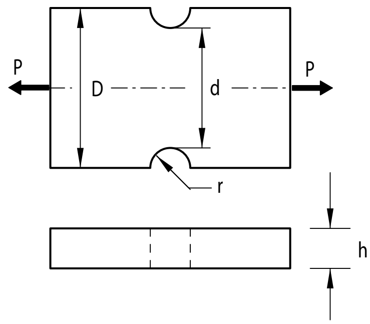


Table 5.11 *Flat plate with semi-circular notch in axial tension—Exponential relationship for charts*



Exponential relationship

$$K_t \approx A\left(\frac{r}{d}\right)^b$$

where

(*D*/*d*)	**A**	**b**
∞	1.11	–0.42
3.00	1.11	–0.41
2.00	1.13	–0.39
1.50	1.13	–0.37
1.30	1.16	–0.33
1.20	1.15	–0.32
1.15	1.10	–0.33
1.10	1.09	–0.30
1.07	1.09	–0.27
1.05	1.09	–0.24
1.03	1.05	–0.22
1.02	1.05	–0.19
1.01	1.04	–0.14

Table 5.12 *Flat plate with semi-circular notch in bending—Stress concentration charts*

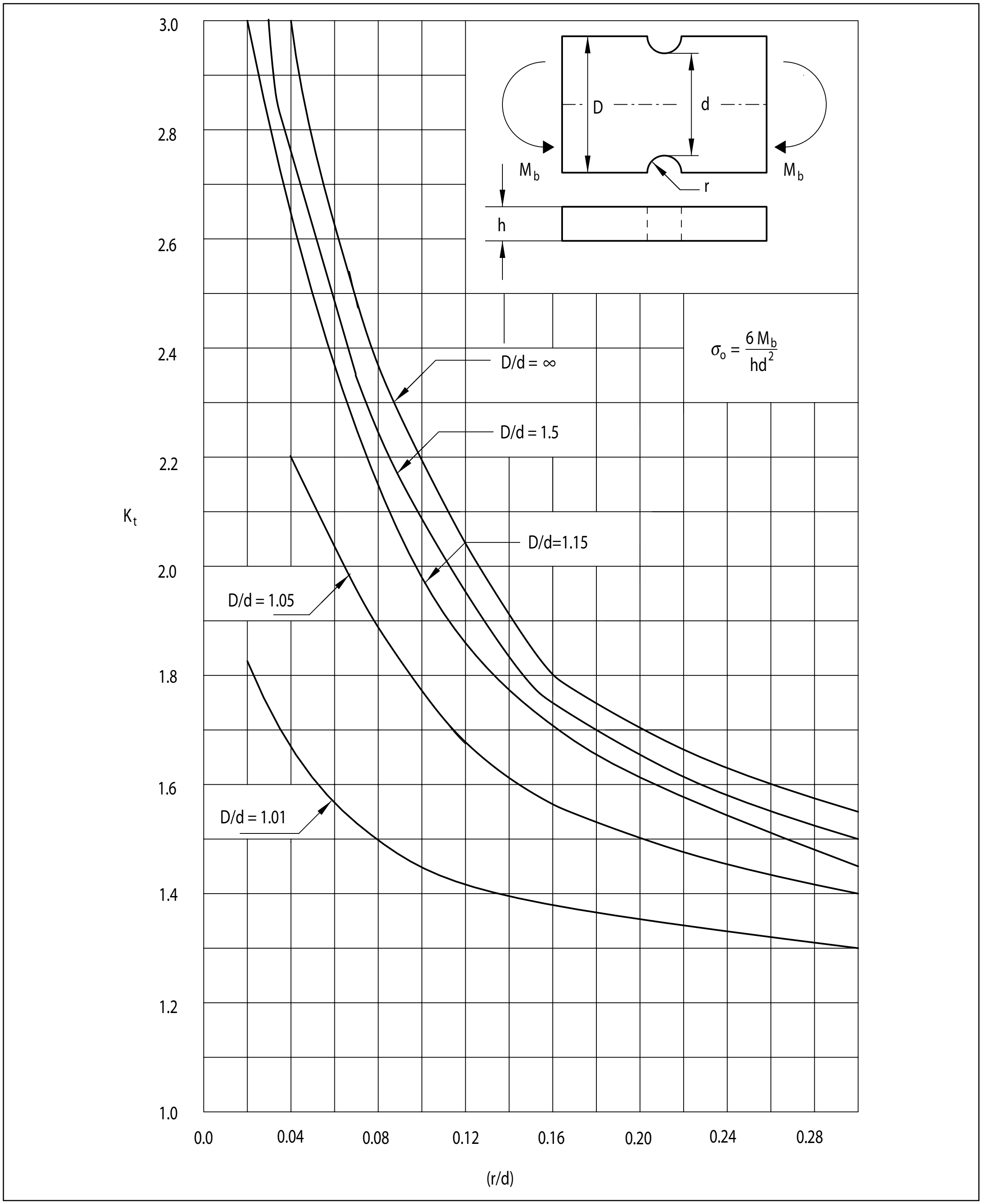
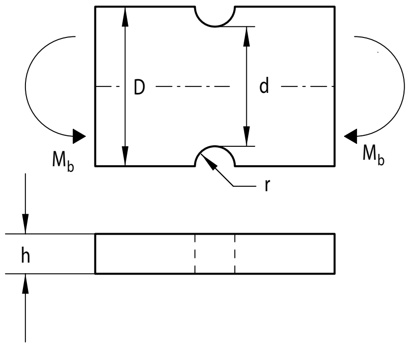


Table 5.13 *Flat plate with semi-circular notch in bending—Exponential relationship for charts*



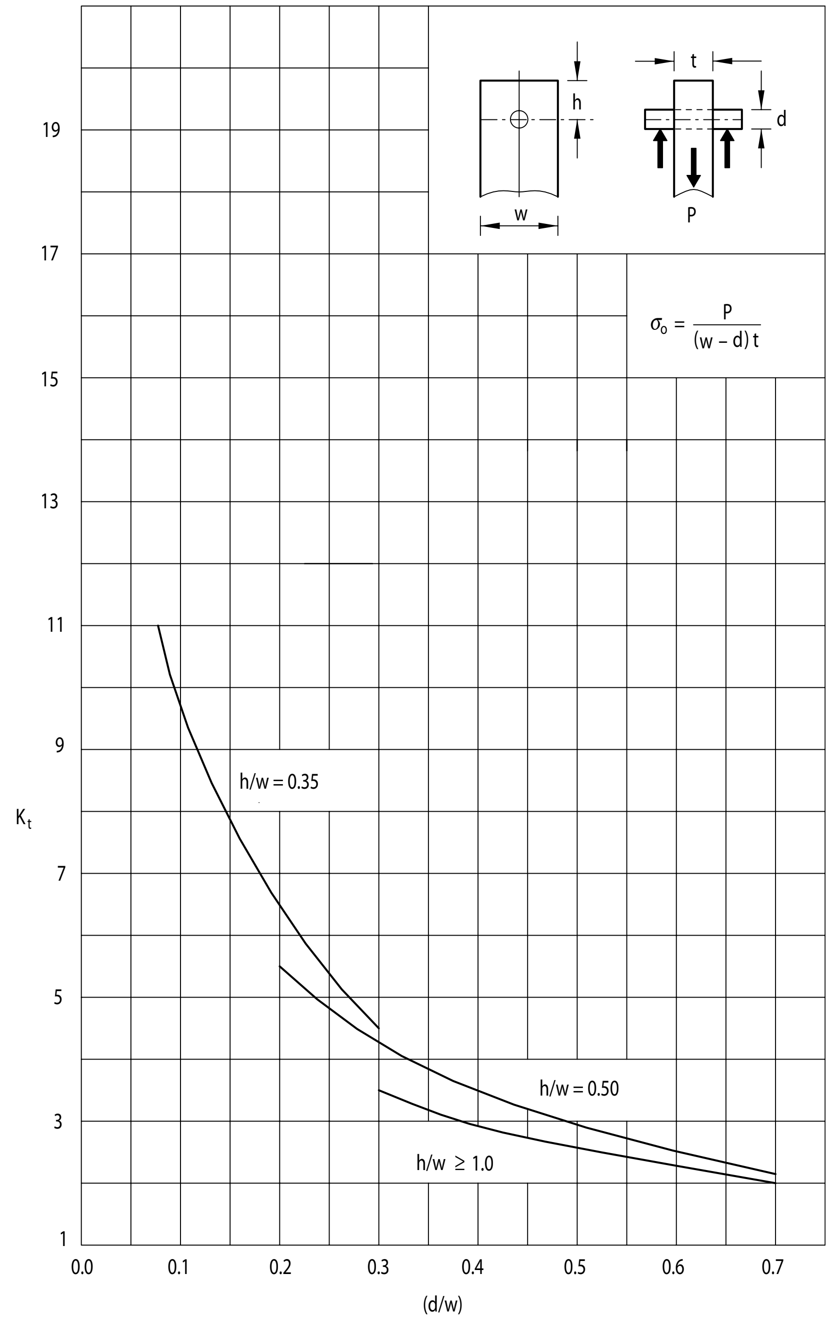
Exponential relationship

$$K_t \approx A\left(\frac{r}{d}\right)^b$$

where

(*D*/*d*)	**A**	**b**
∞	0.97	–0.36
3.00	0.97	–0.35
2.00	0.97	–0.35
1.50	0.98	–0.33
1.30	0.98	–0.33
1.20	0.99	–0.31
1.15	0.99	–0.30
1.10	1.01	–0.28
1.07	1.01	–0.26
1.05	1.03	–0.24
1.03	1.03	–0.21
1.02	1.04	–0.18
1.01	1.06	–0.13

Table 5.14 *Flat plate loaded in tension by a pin through a hole—Stress concentration charts*



Note: When clearance exists, the stress concentration factor is increased by 35 to 50 percent.

Table 5.15 *Round shaft with shoulder fillet in axial tension—Stress concentration charts*

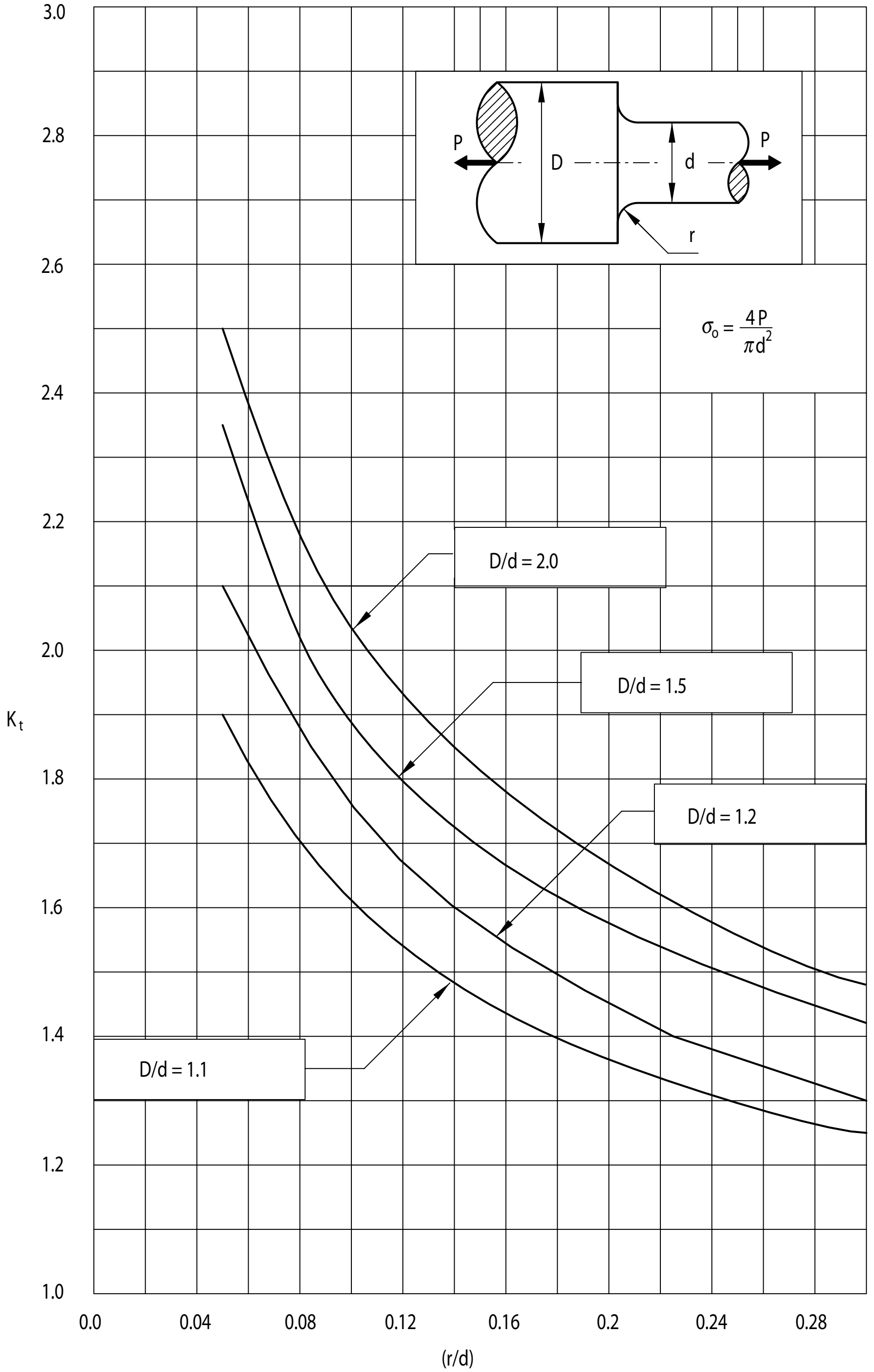
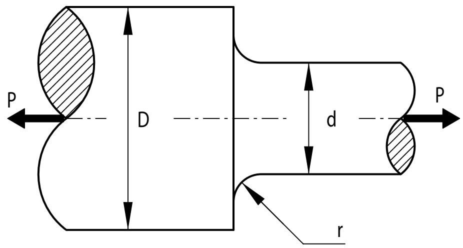


Table 5.16 *Round shaft with shoulder fillet in axial tension—Exponential relationship for charts*



Exponential relationship

$$K_t \approx A\left(\frac{r}{d}\right)^b$$

where

(D/d)	A	b
2.00	1.01	–0.30
1.50	1.00	–0.28
1.30	1.00	–0.26
1.20	0.96	–0.26
1.15	0.98	–0.22
1.10	0.98	–0.21
1.07	0.98	–0.20
1.05	1.00	–0.17
1.02	1.01	–0.12
1.01	0.98	–0.10

Table 5.17 *Round shaft with shoulder fillet in bending—Stress concentration charts*

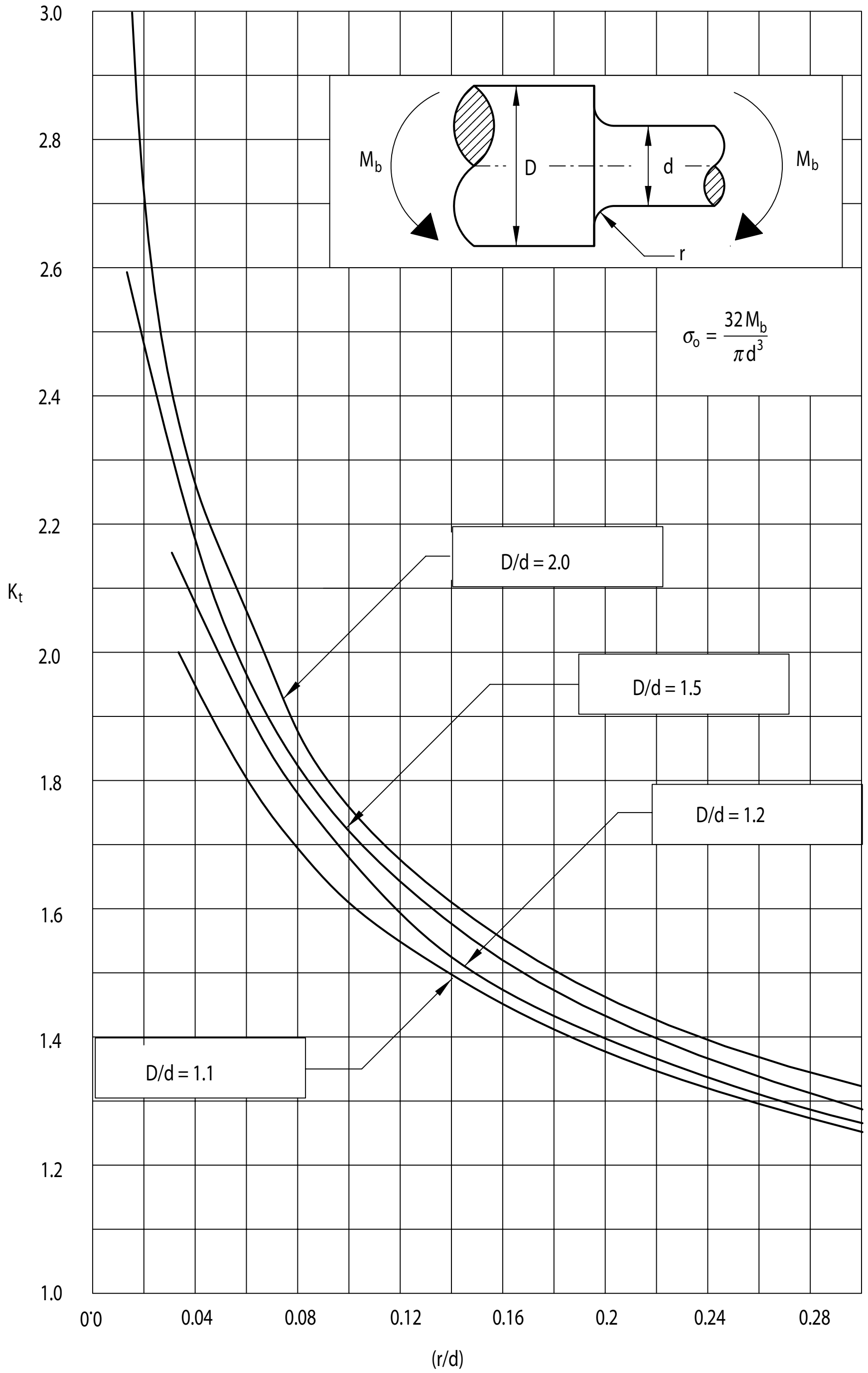
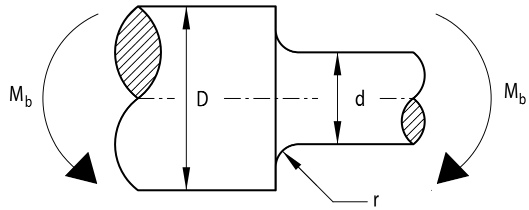


Table 5.18 *Round shaft with shoulder fillet in bending—Exponential relationship for charts*



Exponential relationship

$$K_t \approx A\left(\frac{r}{d}\right)^b$$

where

(*D*/*d*)	**A**	**b**
6.00	0.88	–0.33
3.00	0.89	–0.31
2.00	0.91	–0.29
1.50	0.94	–0.26
1.20	0.97	–0.22
1.10	0.95	–0.24
1.07	0.98	–0.21
1.05	0.98	–0.20
1.03	0.98	–0.18
1.02	0.96	–0.18
1.01	0.92	–0.17

Table 5.19 *Round shaft with shoulder fillet in torsion—Stress concentration charts*

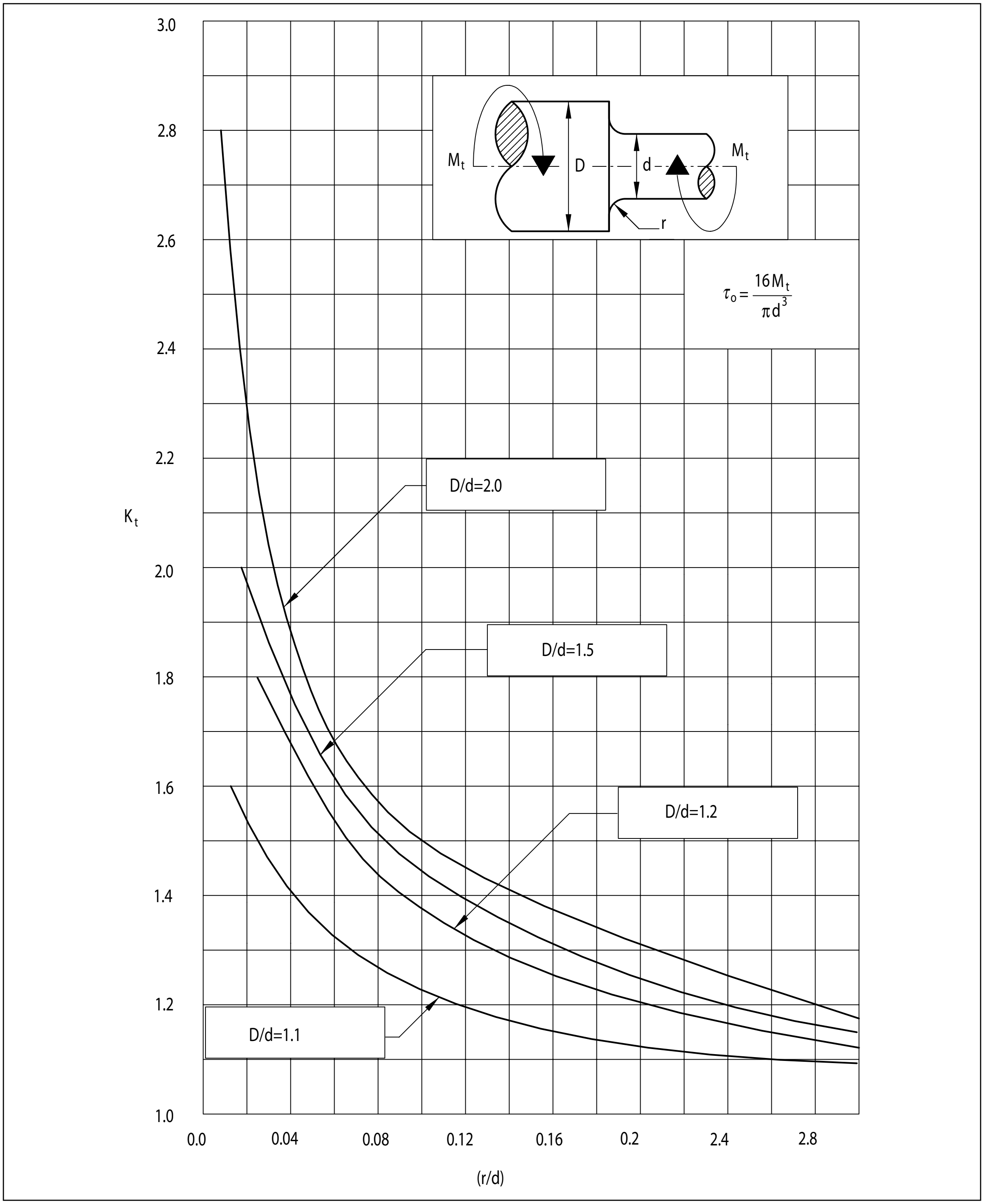
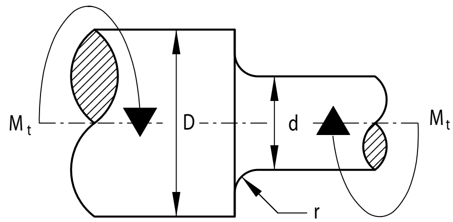


Table 5.20 *Round shaft with shoulder fillet in torsion—Exponential relationship for charts*



Exponential relationship

$$K_t \approx A\left(\frac{r}{d}\right)^b$$

where

(*D*/*d*)	**A**	**b**
2.00	0.86	–0.24
1.33	0.85	–0.23
1.20	0.83	–0.22
1.09	0.90	–0.13

Table 5.21 *Round bar with semi-circular notch in axial tension—Stress concentration charts*

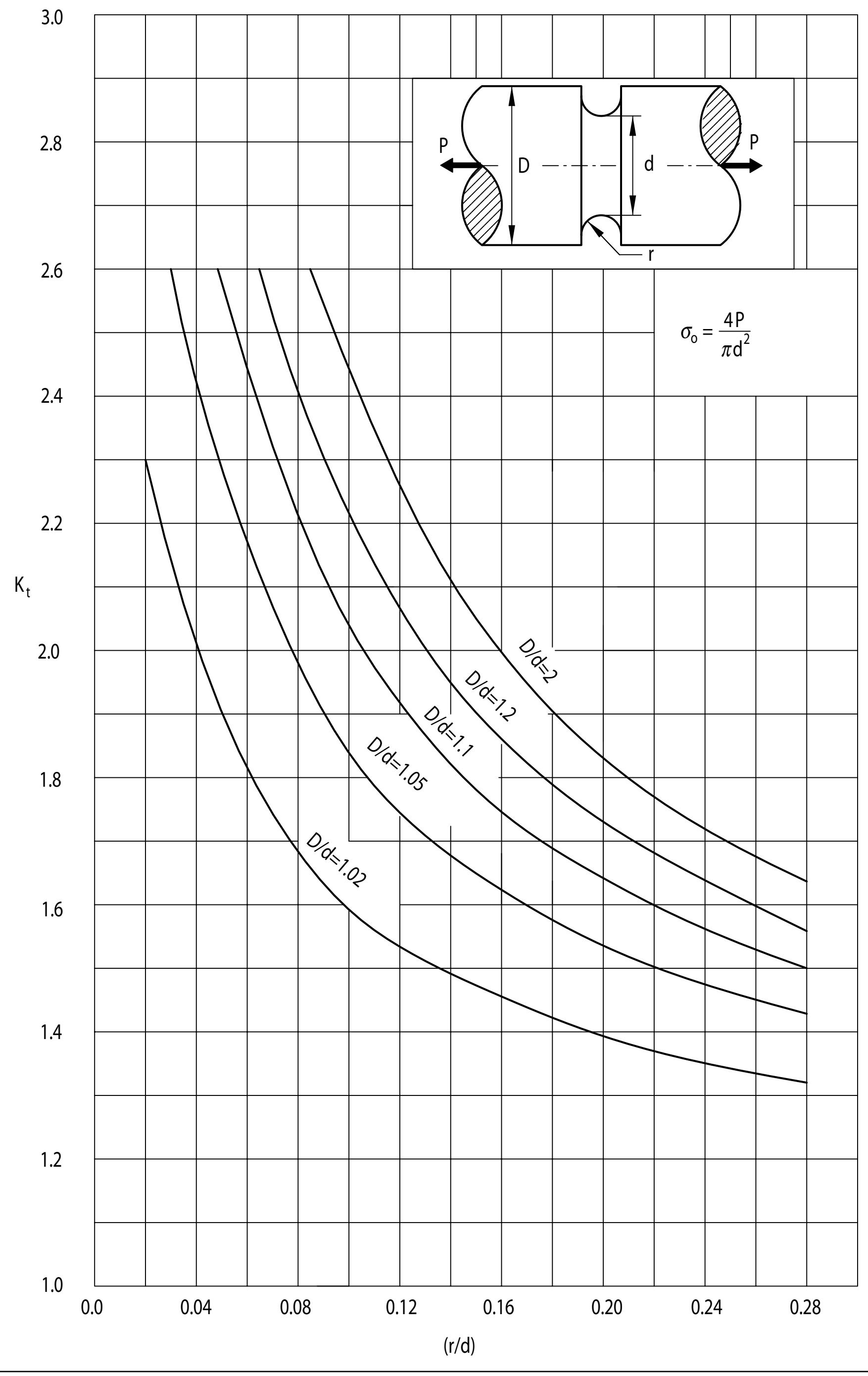
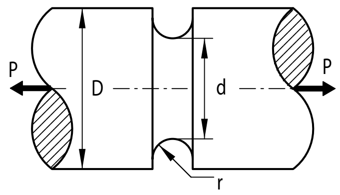


Table 5.22 *Round bar with semi-circular notch in axial tension—Exponential relationship for charts*



Exponential relationship

$$K_t \approx A\left(\frac{r}{d}\right)^b$$

where

(*D*/*d*)	**A**	**b**
∞	0.99	–0.39
2.00	0.99	–0.38
1.50	1.00	–0.37
1.30	1.00	–0.36
1.20	1.01	–0.34
1.15	1.03	–0.32
1.10	1.03	–0.29
1.07	1.02	–0.28
1.05	1.03	–0.25
1.03	1.04	–0.22
1.02	1.04	–0.19
1.01	1.00	–0.16

Table 5.23 *Round bar with semi-circular notch in bending—Stress concentration charts*

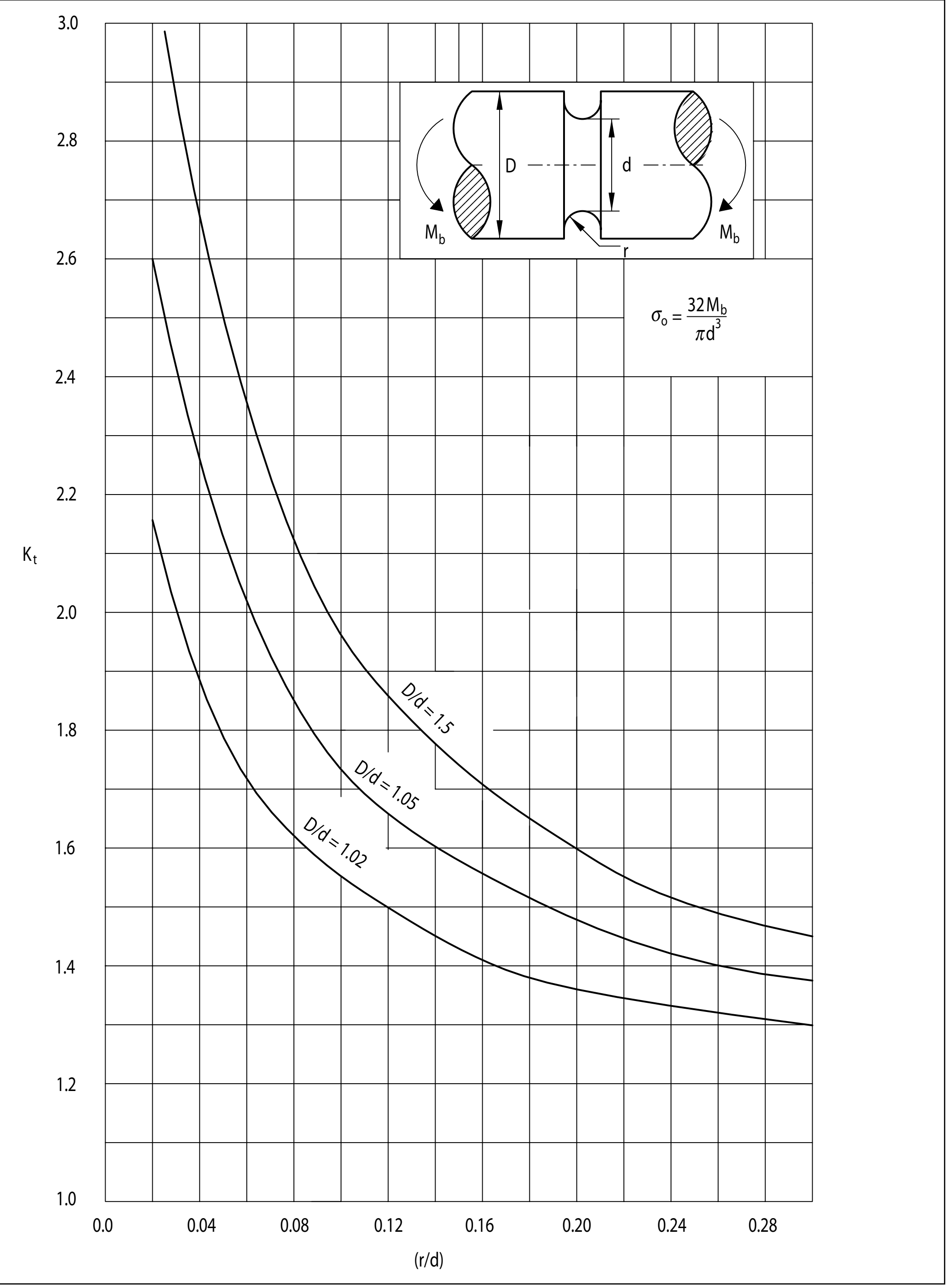
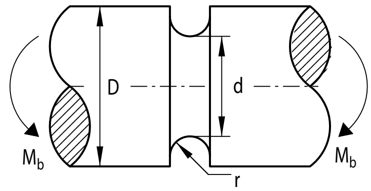


Table 5.24 *Round bar with semi-circular notch in bending—Exponential relationship for charts*



Exponential relationship		
$$K_t \approx A\left(\frac{r}{d}\right)^b$$ where		
(*D*/*d*)	**A**	**b**
∞	0.95	–0.33
2.00	0.94	–0.33
1.50	0.94	–0.32
1.30	0.94	–0.32
1.20	0.95	–0.31
1.15	0.95	–0.30
1.12	0.96	–0.29
1.10	0.95	–0.28
1.07	0.97	–0.26
1.05	0.99	–0.24
1.03	0.99	–0.22
1.02	0.98	–0.20
1.01	0.99	–0.15

Table 5.25 *Round bar with semi-circular notch in torsion—Stress concentration charts*

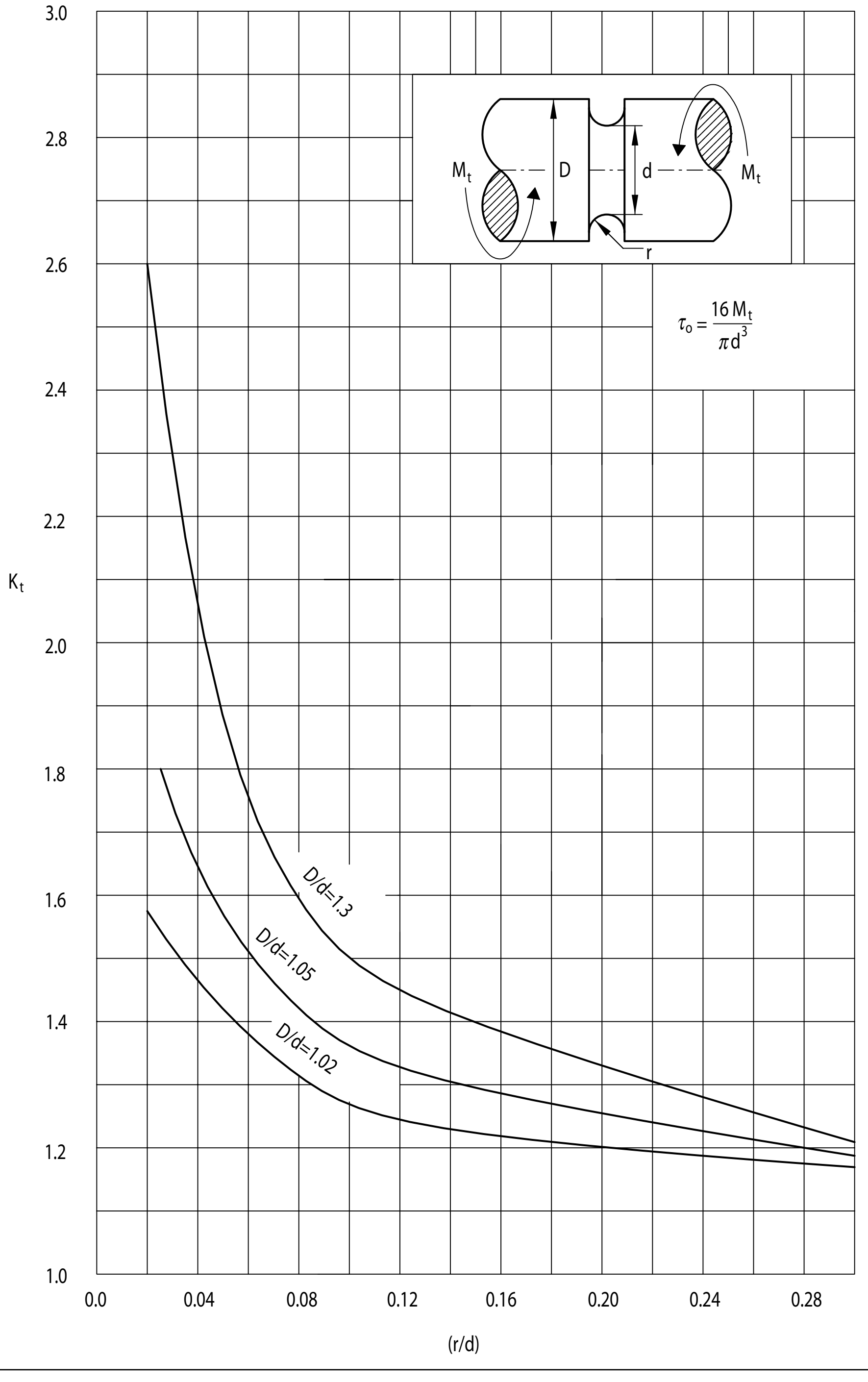
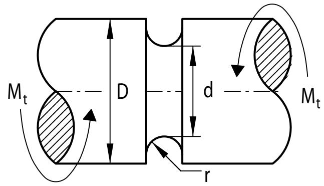


Table 5.26 *Round bar with semi-circular notch in torsion—Exponential relationship for charts*



Exponential relationship

$$K_t \approx A\left(\frac{r}{d}\right)^b$$

where

(*D*/*d*)	**A**	**b**
∞	0.88	–0.25
2.00	0.89	–0.24
1.30	0.89	–0.23
1.20	0.90	–0.22
1.10	0.92	–0.20
1.05	0.94	–0.17
1.02	0.97	–0.13
1.01	0.97	–0.10

Table 5.27 *Round shaft with transverse hole in bending—Stress concentration chart*

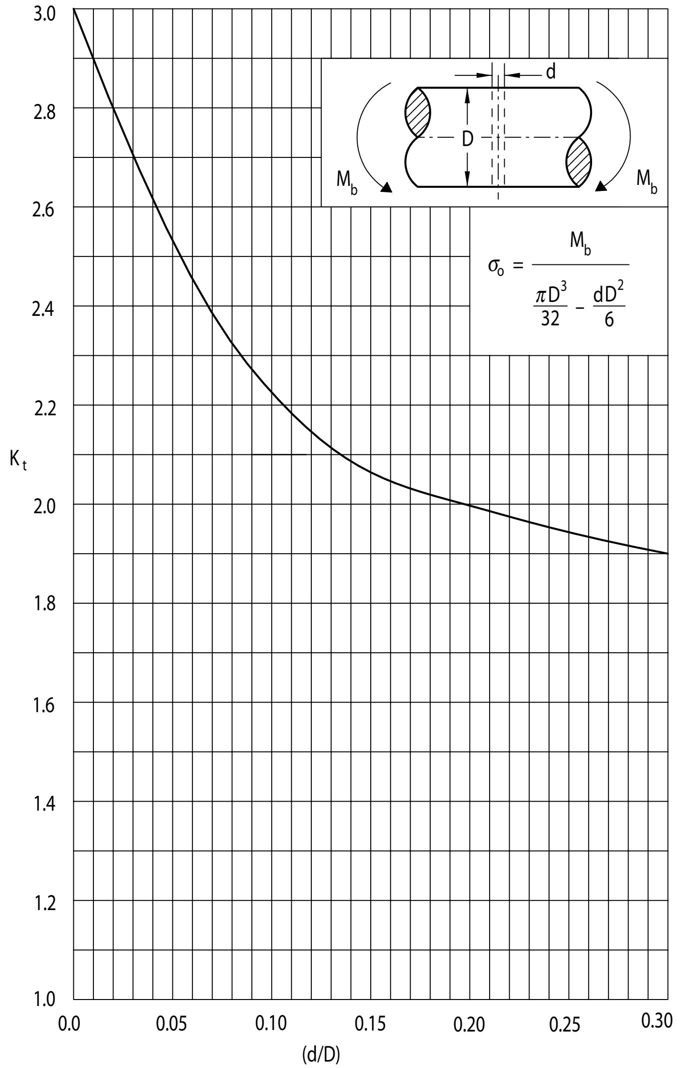
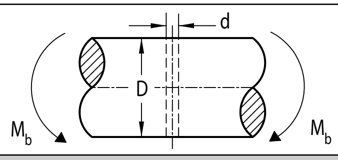
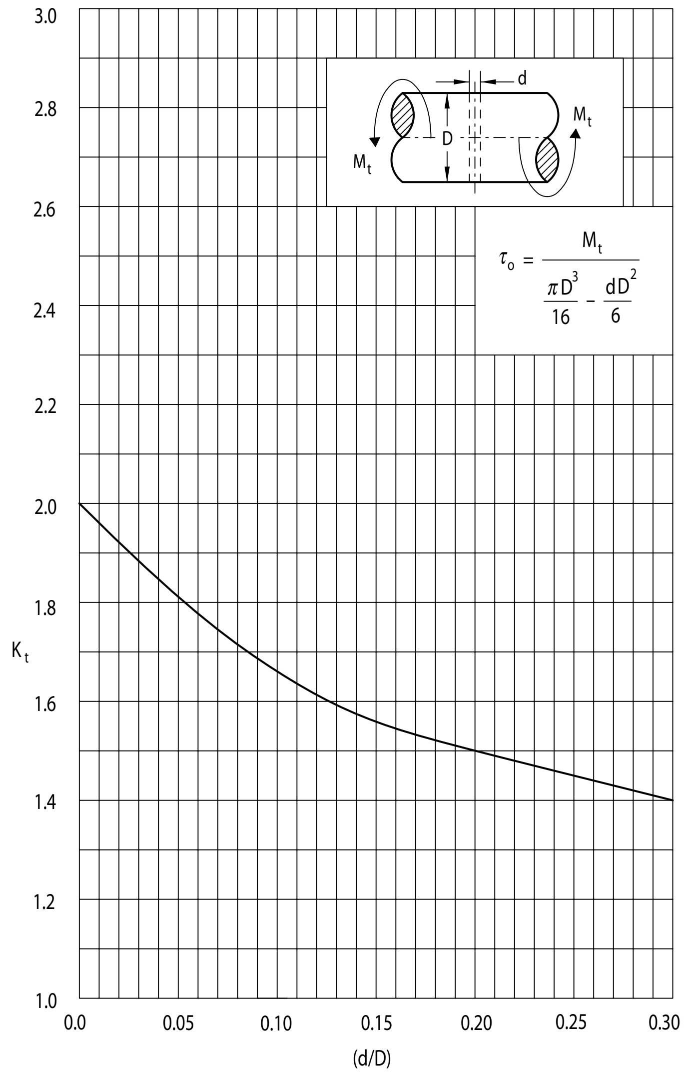


Table 5.28 *Round shaft with transverse hole in bending—Logarithmic relationship for chart*



Logarithmic relationship
$K_t \approx 1.59 - 0.64 \log\left(\dfrac{d}{D}\right)$

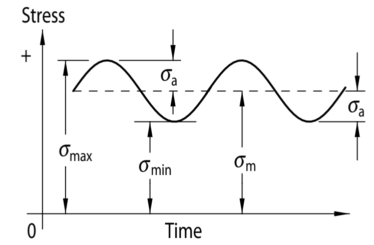
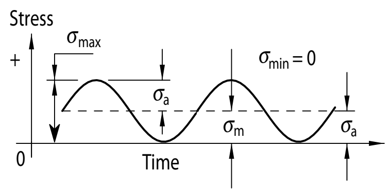
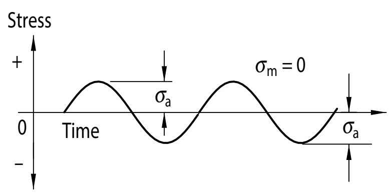
Table 5.29 *Round shaft with transverse hole in torsion—Stress concentration chart*



Note: The stress concentration chart is for stresses below shaft surface in hole.

5.2 CYCLIC STRESSES

Table 5.30 *Cyclic stresses*

Fluctuating stress	
	The fluctuating or alternating stress varies in a sinusoidal manner with respect to time. It has some mean value as well as amplitude value. It fluctuates between two limits–maximum and minimum stress. The stress can be tensile or compressive or partly tensile and partly compressive.
Repeated stress	
	The repeated stress varies in a sinusoidal manner with respect to time, but the variation is from zero to some maximum value. The minimum stress is zero in this case and therefore, amplitude stress and mean stress are equal.
Reversed stress	
	The reversed stress varies in a sinusoidal manner with respect to time, but it has zero mean stress. In this case, half portion of the cycle consists of tensile stress and remaining half of compressive stress. There is complete reversal from tensile stress to compressive stress between these two halves and therefore, mean stress is zero.

$$\sigma_m = \frac{1}{2}(\sigma_{max} + \sigma_{min}) \tag{5.4}$$

$$\sigma_a = \frac{1}{2}(\sigma_{max} - \sigma_{min}) \tag{5.5}$$

σ_{max} = Maximum stress (MPa or N/mm^2)
σ_{min} = Minimum stress (MPa or N/mm^2)
σ_m = Mean stress (MPa or N/mm^2)
σ_a = Stress amplitude (MPa or N/mm^2)

In analysis of fluctuating stresses, tensile stress is considered as positive, while compressive stress as negative. It can be observed that repeated stress and reversed stress are special cases of fluctuating stress with ($\sigma_{min} = 0$) and ($\sigma_m = 0$) respectively.

5.3 NOTCH SENSITIVITY

Table 5.31 *Notch sensitivity*

Fatigue stress concentration factor (K_f)	
$K_f = \dfrac{\text{Endurance limit of the notch free specimen}}{\text{Endurance limit of the notched specimen}}$	
Notch sensitivity factor (q)	
$q = \dfrac{\text{Increase of actual stress over nominal stress}}{\text{Increase of theoretical stress over nominal stress}}$	
Relationship between factors	
$q = \dfrac{(K_f - 1)}{(K_t - 1)}$ (5.6)	$K_f = 1 + q(K_t - 1)$ (5.7) K_t = Theoretical stress concentration factor

Note: The values of notch sensitivity factor (q) can be obtained from Tables 5.32 and 5.33.

Table 5.32 *Notch sensitivity charts for reversed bending and reversed axial stresses*

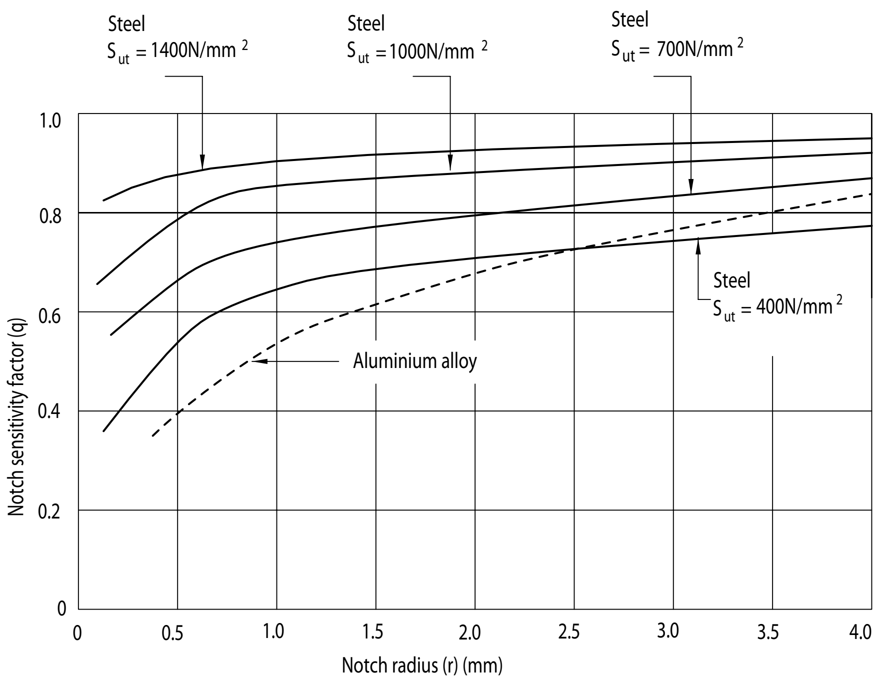
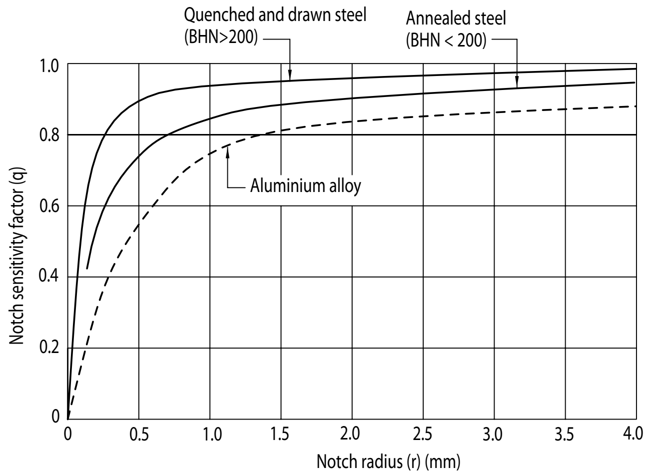


Table 5.33 *Notch sensitivity charts for reversed torsional shear stress*



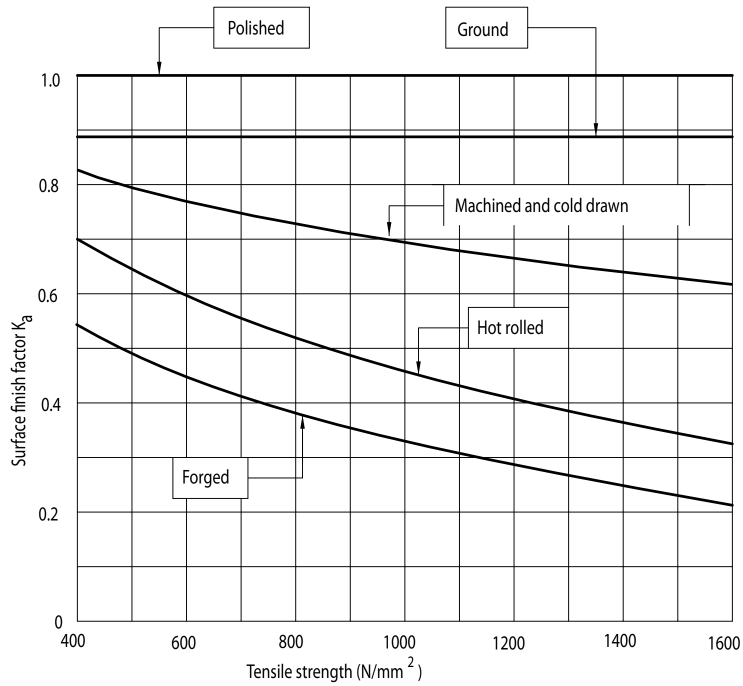
5.4 ESTIMATION OF ENDURANCE LIMIT

Table 5.34 *Estimation of endurance limit*

Basic relationships	
For steels, $S'_e = 0.5\, S_{ut}$ (5.8) For cast iron and cast steels, $S'_e = 0.4\, S_{ut}$ (5.9) For wrought aluminium alloys, $S'_e = 0.4 S_{ut}$ (5.10) For cast aluminium alloys, $S'_e = 0.3\, S_{ut}$ (5.11) These relationships are based on 50% reliability.	S'_e = endurance limit stress of a rotating beam specimen subjected to reversed bending stress (MPa or N/mm^2) S_{ut} = ultimate tensile strength of material (MPa or N/mm^2)
$S_e = K_a\, K_b\, K_c\, K_d\, S'_e$ (5.12)	S_e = endurance limit stress of a particular mechanical component subjected to reversed bending stress (MPa or N/mm^2) K_a = surface finish factor (Table 5.35) K_b = size factor (Table 5.36) K_c = reliability factor (Table 5.37) K_d = modifying factor to account for stress concentration. $K_d = \frac{1}{K_f}$ (5.13)
According to the maximum shear-stress theory, $S_{se} = 0.5 S_e$ (5.14) According to distortion-energy theory, $S_{se} = 0.577\, S_e$ (5.15)	S_{se} = endurance limit of a component subjected to fluctuating torsional shear stresses (MPa or N/mm^2)
$(S_e)_a = 0.8 S_e$ (5.16)	$(S_e)_a$ = endurance limit of a component subjected to axial load. (MPa or N/mm^2)

Table 5.35 *Values of surface finish factor (K_a)*

Experimental curves



Exponential relationship

$K_a = a(S_{ut})^b$ [if K_a>1, set K_a=1]

Values of 'a' and 'b'

Surface finish	a	b
Ground	1.58	–0.085
Machined and Cold drawn	4.51	–0.265
Hot rolled	57.7	–0.718
As forged	272	–0.995

Cast iron components

The value of surface finish factor for cast iron parts is taken as 1 irrespective of machining

Table 5.36 *Values of size factor (K_b)*

Values of size factor (K_b) for cylindrical components

Diameter (d) (mm)	K_b
$d \leq 7.5$ mm	1.00
$7.5 < d \leq 50$ mm	0.85
$d > 50$ mm	0.75

Exponential relationships of size factor (K_b) for cylindrical components

For bending and torsion, the equation is in the following form:

For 2.79 mm $\leq d \leq$ 51 mm	$K_b = 1.24^{-0.107}$
For 51 mm $< d \leq$ 254 mm	$K_b = 0.859 - 0.000\,873\,d$
For axial loading	$K_b = 1$

(*Contd.*)

(*Contd.*)

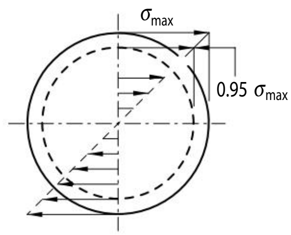
Values of size factor (K_b) for non-cylindrical components
(Kuguel's equality)

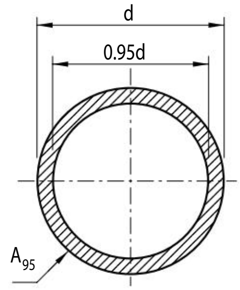
The 'effective' diameter of any non-circular cross section is given by,

$$d_e = \sqrt{\frac{A_{95}}{0.0766}} \tag{5.17}$$

A_{95} = portion of cross sectional area of the non-cylindrical part that is stressed between 95% and 100% of the maximum stress

d_e = effective diameter of non-cylindrical part (for calculating size factor)





Values of **(A_{95})**

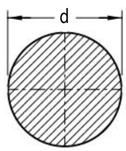
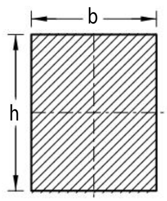
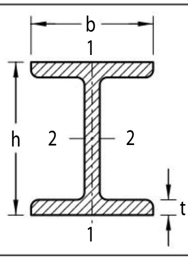
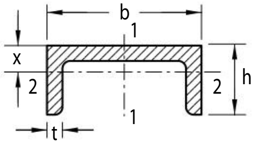
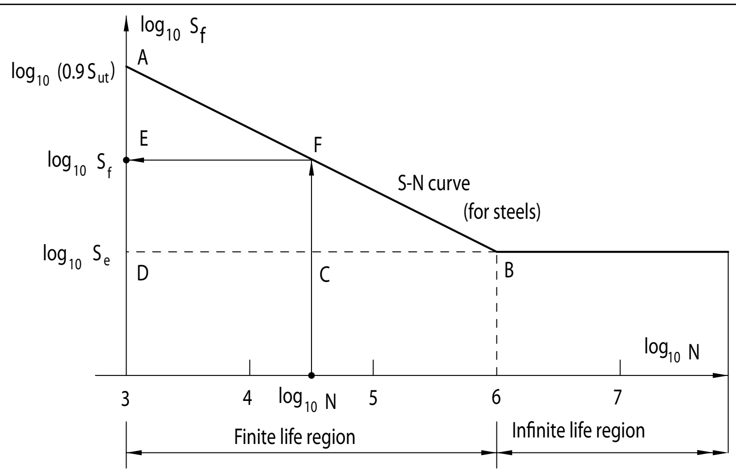
Cross section	(A_{95})
	For rotating solid circular shaft, $A_{95} = 0.0766\ d^2$ (5.18) For non-rotating solid circular bar, $A_{95} = 0.0105\ d^2$ (5.19)
	For non-rotating rectangular section bar, $A_{95} = 0.05\ bh$ (5.20)
	For non-rotating I-section bar, $(A_{95})_{1-1} = 0.10\ bt$ (5.21-a) $(A_{95})_{2-2} = 0.05\ bh\ (t > 0.025\ \text{b})$ (5.21-b)
	For channel section bar, $(A_{95})_{1-1} = 0.05\ bh$ (5.22-a) $(A_{95})_{2-2} = 0.052\ xb + 0.1t\ (h - x)$ (5.22-b)

Table 5.37 *Values of reliability factor (K_c)*

Reliability R (%)	K_c
50	1.000
90	0.897
95	0.868
99	0.814
99.9	0.753
99.99	0.702
99.999	0.659

5.5 FATIGUE DESIGN UNDER REVERSED STRESSES

Table 5.38 *S-N curve*



Component with infinite life	Component with finite life
When the component has infinite life, the endurance limit becomes the criterion of failure. The amplitude stress induced in such components should be lower than the endurance limit in order to withstand the infinite number of cycles. The design equations for such components are: $$\sigma_a = \frac{S_e}{(fs)} \quad (5.23)$$ $$\tau_a = \frac{S_{se}}{(fs)} \quad (5.24)$$	When the component has finite life, the S-N curve for steels consists of a straight line AB drawn from (0.9 S_{ut}) at 10^3 cycles to (S_e) at 10^6 cycles on a log-log paper. (i) The coordinates of point A are [3 $\log_{10}$ (0.9 S_{ut})] since $\log_{10}(10^3) = 3$ (ii) The coordinates of point B are [6, $\log_{10}$ (S_e)] since $\log_{10}(10^6) = 6$ (iii) Line $\overline{AB}$ is used as a criterion of failure for finite-life components.

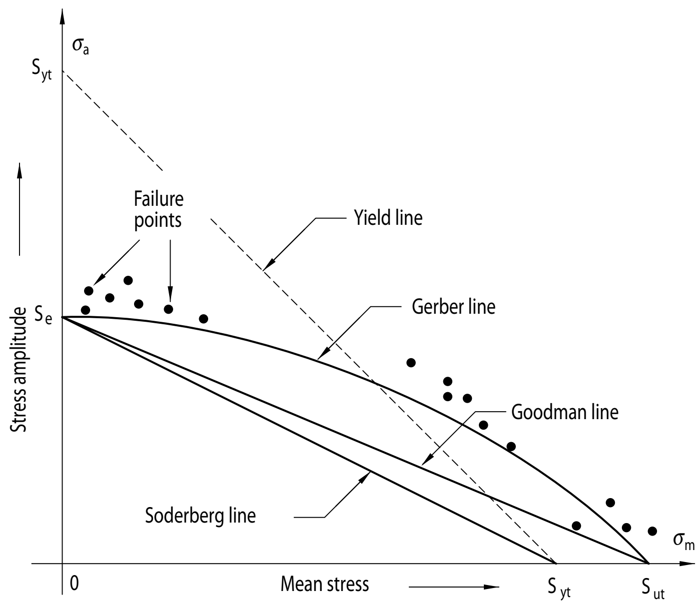
(*Contd.*)

(*Contd.*)

where (σ_a) and (τ_a) are permissible stress amplitudes for calculating the dimensions of the component and S_e and S_{se} are corrected endurance limits in reversed bending and torsion respectively.	(iv) N is the life of component. A vertical line passing through $\log_{10}(N)$ on the abscissa intersects $\overline{AB}$ at point F. (v) Line $\overline{FE}$ is parallel to the abscissa. The ordinate at point E, i.e., $\log_{10}(S_f)$, gives the fatigue strength corresponding to N cycles.

5.6 FATIGUE DESIGN UNDER FLUCTUATING STRESSES

Table 5.39 *Fluctuating stresses—Failure criteria*



Soderberg line	
A straight line joining S_e on the ordinate to S_{yt} on the abscissa is called the Soderberg line.	The equation of Soderberg line is given by, $$\frac{S_m}{S_{yt}} + \frac{S_a}{S_e} = 1 \qquad (5.25)$$
Goodman line	
A straight line joining S_e on the ordinate to S_{ut} on the abscissa is called the Goodman line.	The equation of Goodman line is given by, $$\frac{S_m}{S_{ut}} + \frac{S_a}{S_e} = 1 \qquad (5.26)$$

(*Contd.*)

(*Contd.*)

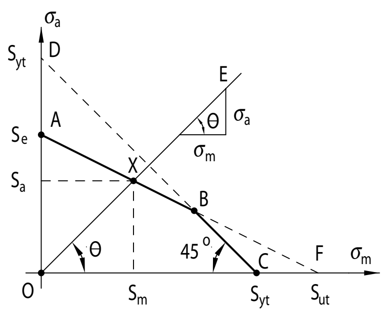
Gerber line	
A parabolic curve joining S_e on the ordinate to S_{ut} on the abscissa is called the Gerber line.	The equation for Gerber curve is as follows, $$\frac{S_a}{S_e} + \left(\frac{S_m}{S_{ut}}\right)^2 = 1 \qquad (5.27)$$ It can be also written in the following form: $$S_a = S_e\left[1 - \left(\frac{S_m}{S_{ut}}\right)^2\right] \qquad (5.28)$$
Yield line	
A straight line joining S_{yt} on the ordinate to S_{yt} on the abscissa is called the Yield line.	The equation of yield line is given by, $$\frac{S_m}{S_{yt}} + \frac{S_a}{S_{yt}} = 1 \qquad (5.29)$$ It can be also written in the following form: $$S_m + S_a = S_{yt} \qquad (5.30)$$
Design equations	
$$\sigma_a = \frac{S_a}{(fs)} \qquad (5.31)$$	$$\sigma_m = \frac{S_m}{(fs)} \qquad (5.32)$$

Notations:

S_{ut} = ultimate tensile strength (MPa or N/mm^2)
S_{yt} = yield strength (MPa or N/mm^2)
S_e = endurance limit stress of component subjected to reversed bending (MPa or N/mm^2)
S_a = limiting value of stress amplitude (MPa or N/mm^2)
S_m= limiting value of mean stress (MPa or N/mm^2)
σ_a= permissible stress amplitude (MPa or N/mm^2)
σ_m = permissible mean stress (MPa or N/mm^2)

Table 5.40 *Modified Goodman diagrams*

Modified Goodman diagram for axial and bending stresses



(i) $\overline{CD}$ is a yield line which joins S_{yt} on the ordinate to S_{yt} on the abscissa. Obviously the line $\overline{CD}$ is inclined at 45° to the abscissa.

(ii) $\overline{AF}$ is a Goodman line which joins S_e on the ordinate with S_{ut} on the abscissa.

(iii) The point of intersection of these two lines is *B*.

(iv) The area *OABC* represents the region of safety for components subjected to fluctuating stresses. The region *OABC* is called modified Goodman diagram. All the points inside the modified Goodman diagram should cause neither fatigue failure nor yielding.

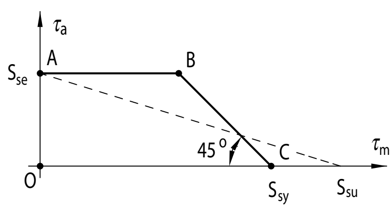
(v) The slope of line $\overline{OE}$ is tan θ.

$$\tan\theta = \frac{\sigma_a}{\sigma_m} = \frac{P_a}{P_m} = \frac{(M_b)_a}{(M_b)_m} \tag{5.33}$$

(vi) The point of intersection of lines $\overline{AB}$ and $\overline{OE}$ is *X*. The point *X* indicates the dividing line between the safe region and the region of failure. The coordinates of point *X* (S_m, S_a) represent the limiting values of stresses, The permissible stresses are as follows:

$$\sigma_a = \frac{S_a}{(fs)} \text{ and } \sigma_m = \frac{S_m}{(fs)} \tag{5.34}$$

Modified Goodman diagram for torsional shear stresses



(i) In this diagram, torsional mean stress is plotted on abscissa while the torsional stress amplitude on ordinate.

(ii) $\overline{CB}$ is a yield line which is inclined at 45° to the abscissa and passes through S_{sy} on abscissa.

(iii) The line $\overline{AB}$ is passes through S_{se} on the ordinate and parallel to the abscissa.

(iv) The point of intersection of these two lines is *B*. The area *OABC* represents the region of safety in this case. It is not necessary to construct fatigue diagram for fluctuating torsional shear stresses because $\overline{AB}$ is parallel to X-axis. Instead a fatigue failure is indicated if,

$$\tau_a = S_{se} \tag{5.35}$$

and a static failure is indicated if,

$$.\ \tau_{max} = \tau_a + \tau_m = S_{sy} \tag{5.36}$$

The permissible stresses are as follows:

$$\tau_a = \frac{S_{se}}{(fs)} \text{ and } \tau_{max} = \frac{S_{sy}}{(fs)} \tag{5.37}$$

(*Contd.*)

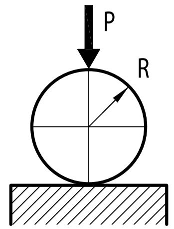
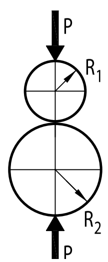
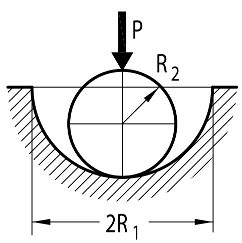
5.7 FATIGUE DESIGN UNDER COMBINED STRESSES

Table 5.41 *Fatigue design under combined stresses*

Two-dimensional stresses
In case of two-dimensional stresses, the component is subjected to stresses σ_x and σ_y in X- and Y-directions. The mean and alternating components of σ_x are σ_{xm} and σ_{xa} respectively. Similarly, the mean and alternating components of σ_y are σ_{ym} and σ_{ya} respectively. In this analysis, the mean and alternating components are separately combined.
$$\sigma_m = \sqrt{(\sigma_{xm}^2 - \sigma_{xm}\sigma_{ym} + \sigma_{ym}^2)} \quad (5.38)$$
$$\sigma_a = \sqrt{(\sigma_{xa}^2 - \sigma_{xa}\sigma_{ya} + \sigma_{ya}^2)} \quad (5.39)$$
The two stresses σ_m are σ_a obtained by the above equations are used in the modified Goodman diagram for design of the component.
Combined bending and torsional shear stresses
In case of combined bending and torsional moments, there is a normal stress σ_x accompanied by the torsional shear stress τ_{xy}. The mean and alternating components of σ_x are σ_{xm} and σ_{xa} respectively. Similarly, the mean and alternating components of τ_{xy} are τ_{xym} and τ_{xya} respectively. Combining these components separately,
$$\sigma_m = \sqrt{\sigma_{xm}^2 + 3\tau_{xym}^2} \quad (5.40)$$
$$\sigma_a = \sqrt{\sigma_{xa}^2 + 3\tau_{xya}^2} \quad (5.41)$$
The two stresses σ_m are σ_a obtained by the above equations are used in the modified Goodman diagram for design of the component.
Miner's equation
The component is subjected to completely reversed stress (σ_1) for (n_1) cycles, (σ_2) for (n_2) cycles and so on. Let N_1 be the number of stress cycles before fatigue failure, if only the alternating stress (σ_1) is acting and N_2 be the number of stress cycles before fatigue failure, if only the alternating stress (σ_2) is acting.
$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \ldots\ldots + \frac{n_x}{N_x} = 1 \quad \text{(Miner's equation)} \quad (5.42)$$
Suppose that α_1, α_2,……….. are proportions of the total life that will be consumed by the stress levels σ_1, σ_2,…….. etc. Let N be the total life of the component. Then, $n_1 = \alpha_1 N$ and $n_2 = \alpha_2 N$ Miner's equation can be written as,
$$\frac{\alpha_1}{N_1} + \frac{\alpha_2}{N_2} + \ldots\ldots + \frac{\alpha_x}{N_x} = \frac{1}{N} \quad (5.43)$$
$$\alpha_1 + \alpha_2 + \alpha_3 + \ldots\ldots + \alpha_x = 1 \quad (5.44)$$

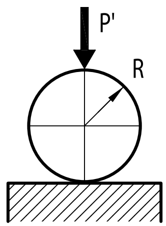
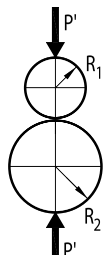
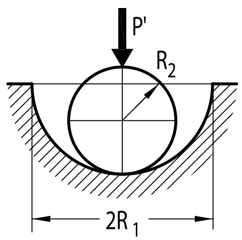
5.8 CONTACT STRESSES

Table 5.42 *Surface contact stresses*

Figure	Shape	Surface compressive stress for dissimilar metals $(\sigma_c)_{max}$	Surface compressive stress for similar metals $(\sigma_c)_{max}$	Maximum shear stress τ_{max} (*)	Depth at τ_{max} (*)
	Sphere and flat plate	$0.578\left[\dfrac{P}{R^2\left(\dfrac{1-\nu_1^2}{E_1}+\dfrac{1-\nu_2^2}{E_2}\right)^2}\right]^{\frac{1}{3}}$ (5.45-a)	$0.388\left[\dfrac{PE^2}{R^2}\right]^{\frac{1}{3}}$ (5.45-b)	$0.34\,(\sigma_c)_{max}$	0.63 a
	Two spheres	$0.578\left[\dfrac{P\left(\dfrac{1}{R_1}+\dfrac{1}{R_2}\right)^2}{\left(\dfrac{1-\nu_1^2}{E_1}+\dfrac{1-\nu_2^2}{E_2}\right)^2}\right]^{\frac{1}{3}}$ (5.46-a)	$0.388\left[PE^2\left(\dfrac{1}{R_1}+\dfrac{1}{R_2}\right)^2\right]^{\frac{1}{3}}$ (5.46-b)	$0.34\,(\sigma_c)_{max}$	0.63 a
	Sphere and spherical Socket	$0.578\left[\dfrac{P\left(\dfrac{1}{R_2}-\dfrac{1}{R_1}\right)^2}{\left(\dfrac{1-\nu_1^2}{E_1}+\dfrac{1-\nu_2^2}{E_2}\right)^2}\right]^{\frac{1}{3}}$ (5.47-a)	$0.388\left[PE^2\left(\dfrac{1}{R_2}-\dfrac{1}{R_1}\right)^2\right]^{\frac{1}{3}}$ (5.47-b)	$0.34\,(\sigma_c)_{max}$	0.63 a

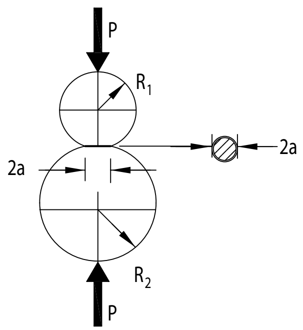
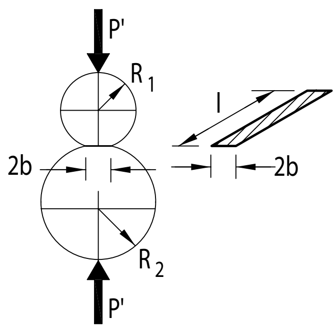
(*Contd.*)

(*Contd.*)

	Cylinder and flat plate	$0.564\left[\dfrac{P'}{R\left(\dfrac{1-\nu_1^2}{E_1}+\dfrac{1-\nu_2^2}{E_2}\right)}\right]^{\frac{1}{2}}$ (5.48-a)	$0.418\left[\dfrac{P'E}{R}\right]^{\frac{1}{2}}$ (5.48-b)	$0.304\ (\sigma_c)_{\max}$	0.786 b
	Two parallel cylinders	$0.564\left[\dfrac{P'\left(\dfrac{1}{R_1}+\dfrac{1}{R_2}\right)}{\left(\dfrac{1-\nu_1^2}{E_1}+\dfrac{1-\nu_2^2}{E_2}\right)}\right]^{\frac{1}{2}}$ (5.49-a)	$0.418\left[P'E\left(\dfrac{1}{R_1}+\dfrac{1}{R_2}\right)\right]^{\frac{1}{2}}$ (5.49-b)	$0.304\ (\sigma_c)_{\max}$	0.786 b
	Cylinder and cy-lindrical groove	$0.564\left[\dfrac{P'\left(\dfrac{1}{R_2}-\dfrac{1}{R_1}\right)}{\left(\dfrac{1-\nu_1^2}{E_1}+\dfrac{1-\nu_2^2}{E_2}\right)}\right]^{\frac{1}{2}}$ (5.50-a)	$0.418\left[P'E\left(\dfrac{1}{R_2}-\dfrac{1}{R_1}\right)\right]^{\frac{1}{2}}$ (5.50-b)	$0.304\ (\sigma_c)_{\max}$	0.786 b

Notations:
P = force pressing the two spheres or cylinders together
l = axial length of contacting cylinders
P' = (P/l) = force per unit length of parallel cylinders
a = radius of circular contact between two spheres (Table 5.43)
b = half width of rectangular contact between two cylinders (Table 5.43)
For similar metals, ($\nu_1 = \nu_2 = 0.3$) and $E_1 = E_2 = E$
(*) References – (i) Burr and Cheatham (ii) Nortan

Table 5.43 *Radius of deformation for contacting spheres (a) and half width of deformation for cylinders (b)*

Figure	Shape	Values of a and b	Other combinations
	Two Spheres	$$a = 0.908\left[\frac{P\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)}{\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}\right]^{\frac{1}{3}}$$ (5.51)	(i) For sphere and flat surface, $R_2 = \infty$ and $R_1 = R$ (ii) For sphere and spherical socket, R_2= negative
	Two Parallel Cylinders	$$b = 1.13\left[\frac{P'\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)}{\left(\frac{1}{R_1}+\frac{1}{R_2}\right)}\right]^{\frac{1}{2}}$$ (5.52)	(i) For cylinder and flat surface, $R_2 = \infty$ and $R_1 = R$ (ii) For cylinder and cylindrical groove, R_2= negative

Notations:

P = force pressing the two spheres or cylinders together

l = axial length of contacting cylinders

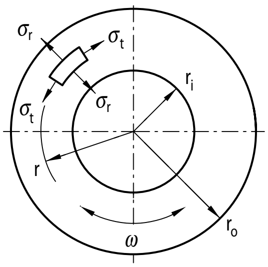
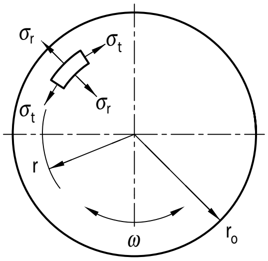
P' = (P/l) = force per unit length of parallel cylinders

a = radius of circular contact between two spheres

b = half width of rectangular contact between two cylinders

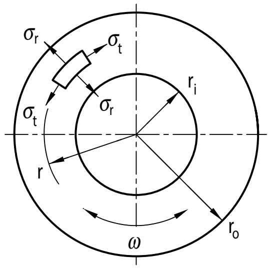
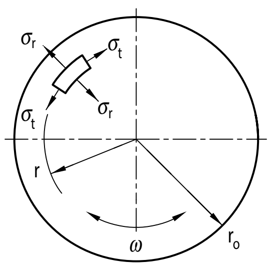
5.9 ROTATING DISKS AND CYLINDERS

Table 5.44 *Stresses in rotating disks of uniform width*

Figure	Shape	Equations for stresses
	Rotating disk with central hole	Radial stress at any radius r is given by, $\sigma_r = \rho v^2 \left(\frac{3+\nu}{8}\right)\left[1+\left(\frac{r_i}{r_o}\right)^2 - \left(\frac{r}{r_o}\right)^2 - \left(\frac{r_i}{r}\right)^2\right]$ (5.53) Tangential stress at any radius r is given by, $\sigma_t = \rho v^2 \left(\frac{3+\nu}{8}\right)\left[1+\left(\frac{r_i}{r_o}\right)^2 - \left(\frac{1+3\nu}{3+\nu}\right)\left(\frac{r}{r_o}\right)^2 + \left(\frac{r_i}{r}\right)^2\right]$ (5.54) The maximum radial stress occurs at ($r = \sqrt{r_i r_o}$). It is given by, $(\sigma_r)_{\max} = \rho v^2 \left(\frac{3+\nu}{8}\right)\left(1-\frac{r_i}{r_o}\right)^2$ (5.55) The maximum tangential stress occurs at ($r = r_i$). It is given by, $(\sigma_t)_{\max} = \rho v^2 \left(\frac{3+\nu}{8}\right)\left[1+\left(\frac{1-\nu}{3+\nu}\right)\left(\frac{r_i}{r_o}\right)^2\right]$ (5.56)
	Rotating solid disk	Radial stress at any radius r is given by, $\sigma_r = \rho v^2 \left(\frac{3+\nu}{8}\right)\left[1-\left(\frac{r}{r_o}\right)^2\right]$ (5.57) Tangential stress at any radius r is given by, $\sigma_t = \rho v^2 \left(\frac{3+\nu}{8}\right)\left[1 - \left(\frac{1+3\nu}{3+\nu}\right)\left(\frac{r}{r_o}\right)^2\right]$ (5.58) The maximum radial stress and maximum tangential stress are equal and both occur at ($r = 0$). They are given by, $(\sigma_r)_{\max} = (\sigma_t)_{\max} = \rho v^2 \left(\frac{3+\nu}{8}\right)$ (5.59)

Notations:
σ_r and σ_r are in (N/m^2)
$(\sigma_r)_{\max}$ = Maximum radial stress (N/m^2)
$(\sigma_t)_{\max}$ = Maximum tangential stress (N/m^2)
ρ = mass density of disk material (kg/m^3)
ν = Poisson's ratio (0.30 for steel and 0.27 for cast iron)
$v = r_o\omega$ = peripheral velocity (m/s)
r_i = inside radius of plate (m)
r_o = outside radius of plate (m)
r = variable radius (m)

Table 5.45 *Stresses in rotating long cylinders*

Figure	Shape	Equations for stresses
	Rotating hollow cylinder	Radial stress at any radius r is given by, $\sigma_r = \frac{\rho v^2}{8}\left(\frac{3-2\nu}{1-\nu}\right)\left[1+\left(\frac{r_i}{r_o}\right)^2-\left(\frac{r}{r_o}\right)^2-\left(\frac{r_i}{r}\right)^2\right]$ (5.60) Tangential stress at any radius r is given by, $\sigma_t = \frac{\rho v^2}{8}\left(\frac{3-2\nu}{1-\nu}\right)\left[1+\left(\frac{r_i}{r_o}\right)^2-\left(\frac{1+2\nu}{3-2\nu}\right)\left(\frac{r}{r_o}\right)^2+\left(\frac{r_i}{r}\right)^2\right]$ (5.61) Axial stress at any radius r is given by, $\sigma_l = \frac{\rho v^2}{4}\left(\frac{\nu}{1-\nu}\right)\left[1+\left(\frac{r_i}{r_o}\right)^2-2\left(\frac{r}{r_o}\right)^2\right]$ (for free ends) (5.62)
	Rotating solid cylinder	Radial stress at any radius r is given by, $\sigma_r = \frac{\rho v^2}{8}\left(\frac{3-2\nu}{1-\nu}\right)\left[1-\left(\frac{r}{r_o}\right)^2\right]$ (5.63) Tangential stress at any radius r is given by, $\sigma_t = \frac{\rho v^2}{8}\left(\frac{3-2\nu}{1-\nu}\right)\left[1-\left(\frac{1+2\nu}{3-2\nu}\right)\left(\frac{r}{r_o}\right)^2\right]$ (5.64) Axial stress at any radius r is given by, $\sigma_l = \frac{\rho v^2}{4}\left(\frac{\nu}{1-\nu}\right)\left[1-2\left(\frac{r}{r_o}\right)^2\right]$ (for free ends) (5.65)

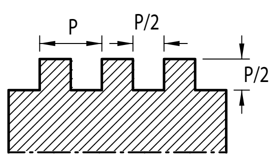
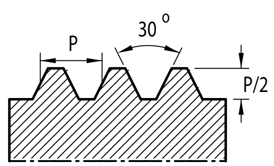
Notations:
σ_r, σ_r and σ_1 are in (N/m^2)
ρ = mass density of disk material (kg/m^3)
ν = Poisson's ratio (0.30 for steel and 0.27 for cast iron)
$v = r_o\omega$ = peripheral velocity (m/s)
r_i = inside radius of cylinder (m)
r_o = outside radius of cylinder (m)
r = variable radius (m)

Power Screws

6

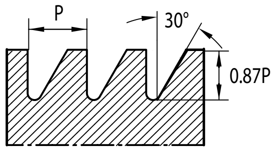
6.1 FORMS OF THREADS

Table 6.1 *Forms of threads for power screws*

Square threads	
 	Designation— A power screw with single-start square threads is designated by the letters 'Sq' followed by the nominal diameter and the pitch expressed in millimeters and separated by the sign ' × '. For example, Sq 30 × 6 It indicates single-start square threads with 30 mm nominal diameter and 6 mm pitch. Applications – Screw jacks, presses and clamping devices
ISO metric trapezoidal threads	
 	Designation— A single-start I.S.O. metric trapezoidal thread is designated by letters 'Tr' followed by the nominal diameter and the pitch expressed in millimetres and separated by the sign ' × '. For example, Tr 40 × 7 It indicates single-start trapezoidal threads with 40 mm nominal diameter and 7 mm pitch. Multiple-start trapezoidal threads are designated by letters 'Tr' followed by the nominal diameter and the lead, separated by sign ' × ' and in brackets the letter 'P' followed by the pitch expressed in millimetres. For example, Tr 40 × 14 (P 7) In above designation, lead = 14 mm pitch = 7 mm $\therefore$ No. of starts = 14/7 = 2 Therefore, the above designation indicates two-start trapezoidal thread with 40 mm nominal diameter and 7 mm pitch. In case of left hand threads, the letter 'LH' is added to thread designation. For example, Tr 40 × 14 (P 7) LH

(*Contd.*)

(*Contd.*)

	Applications – Lead screw and other power transmitting screws in machine tools **Note:** There is a special type of trapezoidal thread called acme thread. Trapezoidal and acme threads are identical in all respects except the thread angle. In acme thread, the thread angle is 29º instead of 30º.
Saw tooth threads	
	Designation— A single-start saw tooth thread is designated by letters 'ST' followed by the nominal diameter and the pitch expressed in millimetres and separated by the sign '×'. For example, ST 40 × 7 It indicates single-start saw tooth threads with 40 mm nominal diameter and 7 mm pitch. Multiple-start saw tooth threads are designated by letters 'ST' followed by the nominal diameter and the lead, separated by sign '×' and in brackets the letter 'P' followed by the pitch expressed in millimetres. For example, ST 40 × 14 (P 7) In above designation, lead = 14 mm pitch = 7 mm ∴ No. of starts = 14/7 = 2 Therefore, the above designation indicates two-start saw tooth thread with 40 mm nominal diameter and 7 mm pitch. Applications – Vices where force is applied only in one direction and for connecting tubular components that must carry large force such as connecting the barrel to the housing in anti-aircraft guns **Note:** There is a special type of saw tooth thread called buttress thread. Saw tooth and buttress threads are similar in all respects except the thread angle and height of external thread. In buttress thread, the thread angle is 45º instead of 30º and height is 0.75 P instead of 0.87 P.

6.2 DIMENSIONS OF SQUARE THREADS

Table 6.2 *Proportions for square threads*

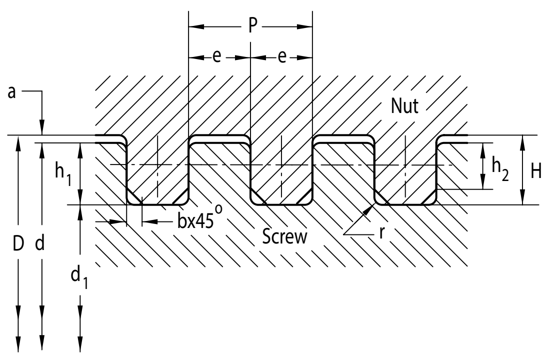
	Proportions $h_1 = 0.5\ P$ $H = 0.5\ P + a$ $h_2 = 0.5P - b$ $e = 0.5P$ $D = d + 2a$ $d_1 = d - 2\ h_1$ Area of core $= \frac{\pi}{4} d_1^2$

Table 6.3 *Basic dimensions of square threads – Fine series*

Nominal Diameter	Major Diameter Screw d	Major Diameter Nut D	Minor Diameter d_1	Pitch P	e	r	h_2	b	h_1	a	H	Area of Core (mm^2)
10	10	10.5	8	2	1	0.12	0.75	0.25	1	0.25	1.25	50.3
12	12	12.5	10									78.5
14	14	14.5	12									113
16	16	16.5	14									154
18	18	18.5	16									201
20	20	20.5	18									254
22	22	22.5	19	3	1.5	0.12	1.25	0.25	1.5	0.25	1.75	284
24	24	24.5	21									346
26	26	26.5	23									415
28	28	28.5	25									491
30	30	30.5	27									573
32	32	32.5	29									661
36	36	36.5	33									855
40	40	40.5	37									1075
42	42	42.5	39									1195
44	44	44.5	41									1320
48	48	48.5	45									1590
50	50	50.5	47									1735
52	52	52.5	49									1886
55	55	55.5	52									2124
60	60	60.5	57									2552
65	65	65.5	61	4	2	0.12	1.75	0.25	2	0.25	2.25	2922
70	70	70.5	66									3421
75	75	75.5	71									3959
80	80	80.5	76									4536
85	85	85.5	81									5153
90	90	90.5	86									5809
95	95	95.5	91									6504
100	100	100.5	96									7238
110	110	110.5	106									8825
120	120	120.5	114	6	3	0.25	2.5	0.5	3	0.25	3.25	10207
130	130	130.5	124									12076
140	140	140.5	134									14103
150	150	150.5	144									16286
160	160	160.5	154									18627
170	170	170.5	164									21124

(*Contd.*)

(*Contd.*)

180	180	180.5	172									23235
190	190	190.5	182									26016
200	200	200.5	192									28953
210	210	210.5	202	8	4	0.25	3.5	0.5	4	0.25	4.25	32047
220	220	220.5	212									35299
230	230	230.5	222									38708
240	240	240.5	232									42273
250	250	250.5	238									44488
260	260	260.5	248									48305
270	270	270.5	258	12	6	0.25	5.5	0.5	6	0.25	6.25	52279
280	280	280.5	268									56410
290	290	290.5	278									60699
300	300	300.5	288									65144

Note: All dimensions in mm.

Table 6.4 *Basic dimensions for square threads – Normal series*

Nominal Diameter	Major Diameter		Minor Diameter d_1	Pitch P	e	r	h_2	b	h_1	a	H	Area of Core (mm^2)
	Screw d	Nut D										
22	22	22.5	17									227
24	24	24.5	19	5	2.5	0.25	2	0.5	2.5	0.25	2.75	284
26	26	26.5	21									346
28	28	28.5	23									415
30	30	30.5	24									452
32	32	32.5	26	6	3	0.25	2.5	0.5	3	0.25	3.25	531
36	36	36.5	30									707
40	40	40.5	33	7	3.5	0.25	3	0.5	3.5	0.25	3.75	855
44	44	44.5	37									1075
48	48	48.5	40									1257
50	50	50.5	42	8	4	0.25	3.5	0.5	4	0.25	4.25	1385
52	52	52.5	44									1521
55	55	55.5	46	9	4.5	0.25	4	0.5	4.5	0.25	4.75	1662
60	60	60.5	51									2043
65	65	65.5	55									2376
70	70	70.5	60	10	5	0.25	4.5	0.5	5	0.25	5.25	2827
75	75	75.5	65									3318
80	80	80.5	70									3848

(*Contd.*)

(*Contd.*)

85	85	85.5	73	12	6	0.25	5.5	0.5	6	0.25	6.25	4185
90	90	90.5	78									4778
95	95	95.5	83									5411
100	100	100.5	88									6082
110	110	110.5	98									7543
120	120	121	106	14	7	0.5	6	1	7	0.5	7.5	8825
125	125	126	111									9677
130	130	131	116									10568
140	140	141	126									12469
150	150	151	134	16	8	0.5	7	1	8	0.5	8.5	14103
160	160	161	144									16286
170	170	171	154									18627
180	180	181	162	18	9	0.5	8	1	9	0.5	9.5	20612
190	190	191	172									23235
200	200	201	182									26016
210	210	211	190	20	10	0.5	9	1	10	0.5	10.5	28353
220	220	221	200									31416
230	230	231	210									34636
240	240	241	218	22	11	0.5	10	1	11	0.5	11.5	37325
250	250	251	228									40828
260	260	261	238									44488
270	270	271	246	24	12	0.5	11	1	12	0.5	12.5	47529
280	280	281	256									51472
290	290	291	266									55572
300	300	301	274	26	13	0.5	12	1	13	0.5	13.5	58965

Note: *All dimensions in mm.*

Table 6.5 *Basic dimensions for square threads – Coarse series*

Nominal Diameter	Major Diameter		Minor Diameter d_1	Pitch P	e	r	h_2	b	h_1	a	H	Area of Core (mm²)
	Screw d	Nut D										
22	22	22.5	14	8	4	0.25	3.5	0.5	4	0.25	4.25	164
24	24	24.5	16									201
26	26	26.5	18									254
28	28	28.5	20									314
30	30	30.5	20	10	5	0.25	4.5	0.5	5	0.25	5.25	314
32	32	32.5	22									380
36	36	36.5	26									531

(*Contd.*)

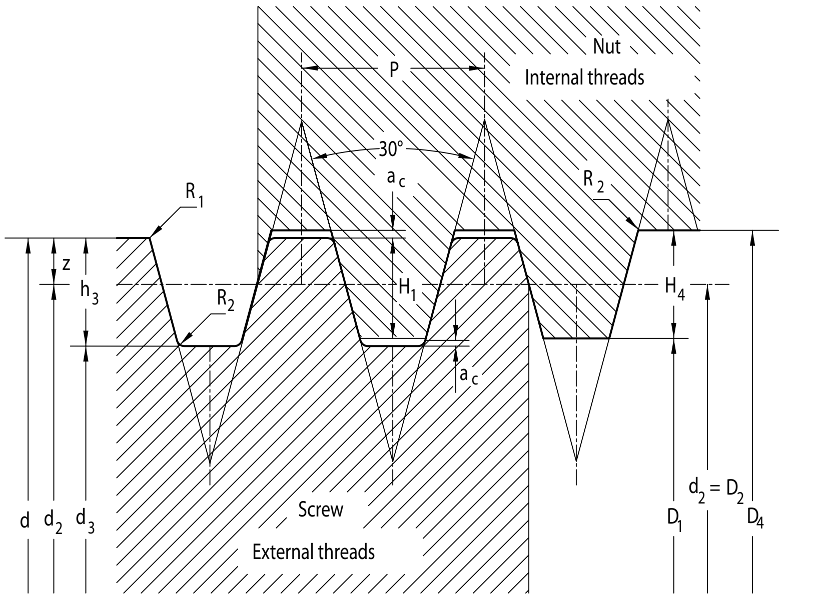
(*Contd.*)

40	40	40.5	28	12	6	0.25	5.5	0.5	6	0.25	6.25	616
44	44	44.5	32									804
48	48	48.5	36									1018
50	50	50.5	38									1134
52	52	52.5	40									1257
55	55	56	41	14	7	0.5	6	1	7	0.5	7.5	1320
60	60	61	46									1662
65	65	66	49	16	8	0.5	7	1	8	0.5	8.5	1886
70	70	71	54									2290
75	75	76	59									2734
80	80	81	64									3217
85	85	86	67	18	9	0.5	8	1	9	0.5	9.5	3526
90	90	91	72									4072
95	95	96	77									4657
100	100	101	80	20	10	0.5	9	1	10	0.5	10.5	5027
110	110	111	90									6362
120	120	121	98	22	11	0.5	10	1	11	0.5	11.5	7543
130	130	131	108									9161
140	140	141	116	24	12	0.5	11	1	12	0.5	12.5	10568
150	150	151	126									12469
160	160	161	132	28	14	0.5	13	1	14	0.5	14.5	13685
170	170	171	142									15837
180	180	181	152									18146
190	190	191	158	32	16	0.5	15	1	16	0.5	16.5	19607
200	200	201	168									22167
210	210	211	174	36	18	0.5	17	1	18	0.5	18.5	23779
220	220	221	184									26590
230	230	231	194									29559
240	240	241	204									32685
250	250	251	210	40	20	0.5	19	1	20	0.5	20.5	34636
260	260	261	220									38013
270	270	271	230									41548
280	280	281	240									45239
290	290	291	246	44	22	0.5	21	1	22	0.5	22.5	47529
300	300	301	256									51472

Note: All dimensions in mm.

6.3 DIMENSIONS OF TRAPEZOIDAL THREADS

Table 6.6 *Proportions for ISO metric trapezoidal threads*



Proportions:
$H_1 = 0.5\,P$
$H_4 = H_1 + a_c = 0.5\,P + a_c$
$h_3 = H_1 + a_c = 0.5\,P + a_c$
$z = H_1/2 = 0.25\,P$
$D_1 = d - 2\,H_1$
$D_4 = d + 2\,a_c$
$d_3 = d - 2\,h_3$
$d_2 = D_2 = d - 2z = d - 0.5\,P$
$(R_1)_{\max} = 0.5\,a_c$
$(R_2)_{\max} = a_c$

Notations:
a_c = clearance on crest
D_4 = major diameter of internal threads
D_2 = pitch diameter of internal threads
D_1= minor diameter of internal threads
d = major diameter of external threads (nominal diameter)
d_2= pitch diameter of external threads
d_3 = minor diameter of external threads

H_1 = height of the overlapping
H_4 = height of internal threads
h_3 = height of external threads
P = pitch

(ISO: 2901)

Table 6.7 *Basic dimensions for ISO metric trapezoidal thread (Thread profile)*

P	a_c	$H_4 = h_3$	H_1	$(R_1)_{\max}$	$(R_2)_{\max}$
1.5	0.15	0.9	0.75	0.08	0.15
2	0.25	1.25	1	0.13	0.25
3	0.25	1.75	1.5	0.13	0.25
4	0.25	2.25	2	0.13	0.25
5	0.25	2.75	2.5	0.13	0.25
6	0.5	3.5	3	0.25	0.5
7	0.5	4	3.5	0.25	0.5
8	0.5	4.5	4	0.25	0.5
9	0.5	5	4.5	0.25	0.5
10	0.5	5.5	5	0.25	0.5

(*Contd.*)

(*Contd.*)

12	0.5	6.5	6	0.25	0.5
14	1	8	7	0.5	1
16	1	9	8	0.5	1
18	1	10	9	0.5	1
20	1	11	10	0.5	1
22	1	12	11	0.5	1
24	1	13	12	0.5	1
28	1	15	14	0.5	1
32	1	17	16	0.5	1
36	1	19	18	0.5	1
40	1	21	20	0.5	1
44	1	23	22	0.5	1

(ISO: 2901) **Note:** All dimensions in mm.

Table 6.8 *Basic dimensions for ISO metric trapezoidal thread*

Nominal diameter d	Pitch P	Pitch diameter $d_2 = D_2$	Major diameter D_4	Minor diameter	
				d_3	D_1
8	1.5	7.25	8.3	6.2	6.5
10	2	9.0	10.5	7.5	8.0
12	3	10.5	12.5	8.5	9.0
16	4	14.0	16.5	11.5	12.0
20	4	18.0	20.5	15.5	16.0
24	5	21.5	24.5	18.5	19.0
28	5	25.5	28.5	22.5	23.0
32	6	29.0	33.0	25.0	26.0
36	6	33.0	37.0	29.0	30.0
40	7	36.5	41.0	32.0	33.0
44	7	40.5	45.0	36.0	37.0
48	8	44.0	49.0	39.0	40.0
52	8	48.0	53.0	43.0	44.0
60	9	55.5	61.0	50.0	51.0
70	10	65.0	71.0	59.0	60.0
80	10	75.0	81.0	69.0	70.0
90	12	84.0	91.0	77.0	78.0
100	12	94.0	101.0	87.0	88.0
120	14	113.0	122.0	104.0	106.0
140	14	133.0	142.0	124.0	126.0
160	16	152.0	162.0	142.0	144.0

(*Contd.*)

(*Contd.*)

180	18	171.0	182.0	160.0	162.0
200	18	191.0	202.0	180.0	182.0
220	20	210.0	222.0	198.0	200.0
240	22	229.0	242.0	216.0	218.0
260	22	249.0	262.0	236.0	238.0
280	24	268.0	282.0	254.0	256.0
300	24	288.0	302.0	274.0	276.0

(ISO: 2904) **Note:** All dimensions in mm.

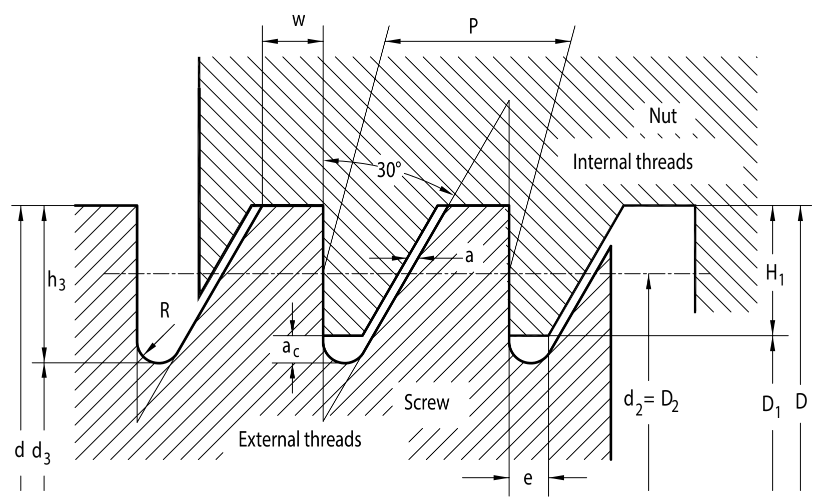
Table 6.9 *Tolerance classes for trapezoidal screw threads*

Tolerance class for Pitch diameter	Thread engagement group			
	Internal thread		External thread	
	N	L	N	L
Medium	7H	8H	7e	8e
Coarse	8H	9H	8c	9c

(ISO: 2903) **Note:** (i) Medium class – for general use; (ii) Coarse class – for cases when manufacturing difficulties can arise; (iii) Group N is recommended if the actual length of thread engagement is unknown.

6.4 DIMENSIONS OF SAW TOOTH THREADS

Table 6.10 *Proportions for saw tooth threads*



Proportions:

$H_1 = 0.75\ P$

$a_c = 0.11777\ P$

$a = 0.1\sqrt{P}$ (axial play)

$h_3 = H_1 + a_c = 0.86777\ P$

$w = 0.26384\ P$

$e = 0.26384\ P - 0.1\sqrt{P} = w - a$

$R = 0.12427\ P$

$D_1 = d - 2\ H_1 = d - 1.5\ P$

$d_3 = d - 2\ h_3$

$d_2 = D_2 = d - 0.75\ P$

$A = \dfrac{\pi}{4} d_3^2$

(*Contd.*)

(*Contd.*)

Notations: D = major diameter of internal thread d = major diameter of external thread D_2 = pitch diameter of internal thread d_2 = pitch diameter of external thread R = radius at root of external thread a_c = clearance at minor diameter of external thread	D_1= minor diameter of internal thread d_3= minor diameter of external thread P = pitch H_1 = height of internal thread profile h_3 = height of external thread profile w = profile width at major diameter

(DIN 513-1)

Table 6.11 *Basic dimensions for thread profile of saw tooth threads*

P	a_c	a	e	h_3	w
2	0.236	0.1414	0.386	1.736	0.528
3	0.353	0.1732	0.618	2.603	0.792
4	0.471	0.2	0.855	3.471	1.005
5	0.589	0.2236	1.096	4.339	1.319
6	0.707	0.2449	1.338	5.207	1.583
7	0.824	0.2646	1.582	6.074	1.847
8	0.942	0.2828	1.828	6.942	2.111
9	1.060	0.3	2.075	7.810	2.374
10	1.178	0.3162	2.322	8.678	2.638
12	1.413	0.3464	2.820	10.413	3.166
14	1.649	0.3742	3.320	12.149	3.694
16	1.884	0.4	3.821	13.884	4.221
18	2.120	0.4243	4.325	15.620	4.749
20	2.355	0.4472	4.830	17.355	5.276
22	2.591	0.4690	5.335	19.091	5.804
24	2.826	0.4899	5.842	20.826	6.332
28	3.298	0.5292	6.858	24.298	7.388
32	3.769	0.5657	7.877	27.769	8.443
36	4.240	0.6	8.898	31.240	9.498
40	4.711	0.6325	9.921	34.711	10.554
44	5.182	0.6633	10.946	38.182	11.609

(DIN 513-2) **Note:** All dimensions in mm.

Table 6.12 *Basic dimensions for saw tooth threads*

Nominal Diameter d	Pitch P	Pitch Diameter $d_2 = D_2$	Minor Diameter		h_3	H_1	R	Area of core (mm^2)
			d_3	D_1				
10	2	8.50	6.528	7.0	1.736	1.50	0.249	34
12	3	9.75	6.794	7.5	2.603	2.25	0.373	36

(*Contd.*)

(*Contd.*)

16	4	13.00	9.058	10.0	3.471	3.00	0.497	64
20	4	17.00	13.058	14.0	3.471	3.00	0.497	133
24	5	20.25	15.322	16.5	4.339	3.75	0.621	184
28	5	24.25	19.322	20.5	4.339	3.75	0.621	293
32	6	27.50	21.586	23.0	5.207	4.50	0.746	370
36	6	31.50	25.586	27.0	5.207	4.50	0.746	514
40	7	34.75	27.852	29.5	6.074	5.25	0.870	606
44	7	38.75	31.852	33.5	6.074	5.25	0.870	794
48	8	42.00	34.116	36.0	6.942	6.00	0.994	914
52	8	46.00	38.116	40.0	6.942	6.00	0.994	1141
60	9	53.25	44.380	46.5	7.810	6.75	0.994	1550
70	10	62.50	52.644	55.0	8.678	7.50	1.243	2177
80	10	72.50	62.644	65.0	8.678	7.50	1.243	3082
90	12	81.00	69.174	72.0	10.415	9.00	1.491	3758
100	12	91.00	79.174	82.0	10.415	9.00	1.491	4923
120	14	109.50	95.702	99.0	12.149	10.50	1.740	7210
140	14	129.50	125.702	119.0	12.149	10.50	1.740	12380
160	16	148.00	132.232	136.0	13.884	12.00	1.988	13733
180	18	166.50	148.760	153.0	15.620	13.50	2.237	17340
200	18	186.50	168.760	173.0	15.620	13.50	2.237	22340
220	20	205.00	185.290	190.0	17.355	15.00	2.485	26965
240	22	223.50	201.818	207.0	19.091	16.50	2.734	31780
260	22	243.50	221.818	227.0	10.091	16.50	2.734	38900
280	24	262.00	238.348	244.0	20.826	18.00	2.982	44618
300	24	282.00	258.348	264.0	20.826	18.00	2.982	52450

(DIN 513-2) **Note:** All dimensions in mm.

Table 6.13 *Tolerance classes for saw tooth threads*

Tolerance class for Pitch diameter	Thread engagement group			
	Internal thread		External thread	
	N	L	N	L
Medium	7H	8H	7e	8e
Coarse	8H	9H	8c	9c

(DIN 513-3) **Note:** (i) Medium class – for general use; (ii) Coarse class – for cases when manufacturing difficulties can arise; (iii) Group N is recommended if the actual length of thread engagement is unknown.

6.5 TORQUE EQUATIONS

Table 6.14 *Basic equations for power screw with square threads*

Equation		Notations
	Core diameter	
$d_c = (d - P)$	(6.1)	d_c = core or minor diameter of screw (mm) d = nominal or major diameter of screw (mm) P = pitch of thread (mm)
	Mean diameter	
$d_m = (d - 0.5P)$	(6.2)	d_m = mean diameter of screw (mm)
	Helix angle of thread	
$\tan\alpha = \dfrac{l}{\pi d_m}$	(6.3)	α = helix angle of the thread l = lead of screw
	Friction angle	
$\mu = \tan\phi$	(6.4)	ϕ = friction angle μ = coefficient of friction between screw and nut threads
	Effort required to lift load	
$E = W\tan(\phi + \alpha)$	(6.5)	E = effort required to lift load W (N) (effort E is perpendicular to load W) W = load (N)
	Torque required to lift load	
$M_t = \dfrac{Wd_m}{2}\tan(\phi + \alpha)$	(6.6)	M_t = torque required to lift load (N-mm)
	Effort required to lower load	
$E = W\tan(\phi - \alpha)$	(6.7)	E = effort required to lower load W (N) (effort E is perpendicular to load W)
	Torque required to lower load	
$M_t = \dfrac{Wd_m}{2}\tan(\phi - \alpha)$	(6.8)	M_t = torque required to lower load (N-mm)
	Self-locking screw	
For self-locking screw, $\phi > \alpha$ $\tan\phi > \tan\alpha$ or $\mu > \tan\alpha$ $\mu > \dfrac{l}{\pi d_m}$	(6.9)	"A screw will be self-locking if the coefficient of friction is equal to or greater than the tangent of the helix angle".

(*Contd.*)

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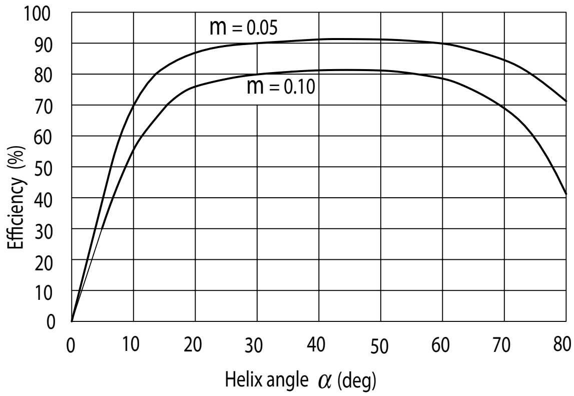
Efficiency of screw	
$$\eta = \frac{\tan \alpha}{\tan(\phi + \alpha)} \quad (6.10)$$	η = efficiency of the screw



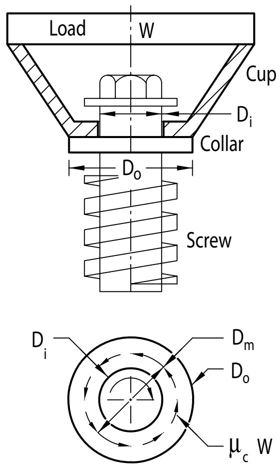
Note: An average value of 0.15 can be taken for the coefficient of friction between screw and nut threads, when the screw is lubricated with mineral oil.

Table 6.15 *Basic equations for power screw with trapezoidal threads*

Effort required to lift load	
$$E = \frac{W(\mu \sec \theta + \tan \alpha)}{(1 - \mu \sec \theta \tan \alpha)}$$	(6.11)
Torque required to lift load	
$$M_t = \frac{Wd_m}{2} \frac{(\mu \sec \theta + \tan \alpha)}{(1 - \mu \sec \theta \tan \alpha)}$$	(6.12)
Effort required to lower load	
$$E = \frac{W(\mu \sec \theta - \tan \alpha)}{(1 + \mu \sec \theta \tan \alpha)}$$	(6.13)
Torque required to lower load	
$$M_t = \frac{Wd_m}{2} \frac{(\mu \sec \theta - \tan \alpha)}{(1 + \mu \sec \theta \tan \alpha)}$$	(6.14)
Efficiency of screw	
$$\eta = \frac{\tan \alpha(1 - \mu \sec \theta \tan \alpha)}{(\mu \sec \theta + \tan \alpha)}$$	(6.15)

Note: θ = thread angle
For I.S.O. metric trapezoidal thread, $2\theta = 30^\circ$
For acme thread, $2\theta = 29^\circ$

Table 6.16 *Collar friction torque*



According to the uniform pressure theory,

$$(M_t)_c = \frac{\mu_c W}{3} \times \frac{D_0^3 - D_i^3}{D_0^2 - D_i^2} \tag{6.16}$$

According to the uniform wear theory,

$$(M_t)_c = \frac{\mu_c W}{4} \times (D_0 + D_i) \tag{6.17}$$

$$(M_t)_c = \mu_c W R_m \tag{6.18}$$

μ_c = coefficient of friction at the collar
D_0 = outer diameter of the collar (mm)
D_i = inner diameter of the collar (mm)
R_m = mean radius of collar (mm)
$(M_t)_c$ = collar friction torque (N-mm)

Table 6.17 *Coefficient of friction for thrust collars*

Material combination	μ_c	
	Starting	Running
Soft-steel – cast iron	0.17	0.12
Hardened steel – cast iron	0.15	0.09
Soft steel – bronze	0.10	0.08
Hardened steel – bronze	0.08	0.06

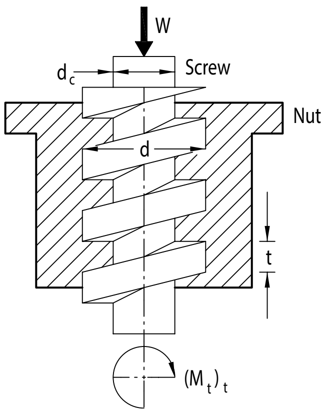
Note: When thrust ball bearing is used at the collar surface, its coefficient of friction is about 1/10th of plain sliding surface. It varies from 0.01 to 0.02.

Table 6.18 *Unit bearing pressure (S_b) for power screws*

Type of application	Material		S_b (MPa or N/mm^2)	Rubbing speed
	Screw	Nut		
Hand press	Steel	Bronze	18–24	Low speed
Screw-jack	Steel	Cast iron	13–17	Low speed < 2.5 m/min
Screw-jack	Steel	Bronze	11–17	Low speed < 3 m/min
Hoisting screw	Steel	Cast iron	4–7	Medium speed 6–12 m/min
Hoisting screw	Steel	Bronze	5–10	Medium speed 6–12 m/min
Lead screw	Steel	Bronze	1–1.5	High speed > 15 m/min

6.6 DESIGN EQUATIONS OF POWER SCREWS

Table 6.19 *Power screws —Stresses, bearing pressure and materials*



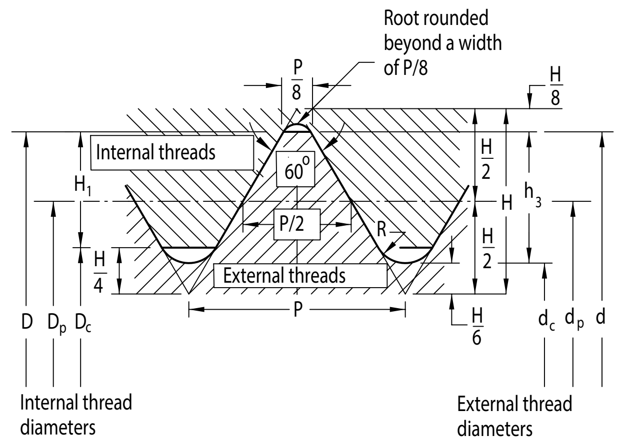
	Stresses in screw body
	$$\sigma_c = \frac{W}{\left(\frac{\pi}{4}d_c^2\right)} \quad (6.19)$$ W = axial load (N) σ_c = direct compressive stress (MPa or N/mm^2) d_c = core diameter of screw (mm) $$\tau = \frac{16(M_t)_t}{\pi d_c^3} \quad (6.20)$$ τ = torsional shear stress (MPa or N/mm^2) $(M_t)_t$ = total torque (N-mm) $$\tau_{max} = \sqrt{\left(\frac{\sigma_c}{2}\right)^2 + (\tau)^2} \quad (6.21)$$ τ_{max} = principal shear stress (MPa or N/mm^2) $$(M_t)_t = M_t + (M_t)_c \quad (6.22)$$ $(M_t)_t$ = external torque required to raise the load (N-mm) M_t = torque required to overcome friction at the thread surface (N-mm) $(M_t)_c$ = collar friction torque (N-mm)
	Stresses in threads of screw
	$$\tau_s = \frac{W}{\pi d_c t z} \quad (6.23)$$ τ_s = transverse shear stress at the root of the screw (MPa or N/mm^2) t = thread thickness at the core diameter (mm) z = number of threads in engagement with the nut
	Stresses in threads of nut
	$$\tau_n = \frac{W}{\pi d t z} \quad (6.24)$$ τ_n = transverse shear stress at the root of the nut (MPa or N/mm^2) t = thread thickness at the root of the nut (mm)
	Bearing pressure
	$$S_b = \frac{4W}{\pi z(d^2 - d_c^2)} \quad (6.25)$$ S_b = unit bearing pressure (MPa or N/mm^2) (Table 6.18)
Materials for Screw Plain carbon steels 30C8, 40C8 and 45C8 – alloy steel 40Cr1	Materials for Nut Phosphor bronze – tin bronze – cast iron

Threaded Fasteners

7

7.1 ISO METRIC SCREW THREADS

Table 7.1 *Profiles of internal and external threads*



Proportions	Notations
$D_p = D - 2\left(\frac{3}{8}H\right) = D - 0.650\,P$	D = major diameter of internal thread
$d_p = d - 2\left(\frac{3}{8}H\right) = d - 0.650\,P$	D_c = minor diameter of internal thread
$D_c = D - 2\left(\frac{5}{8}H\right) = D - 1.083\,P$	D_p = pitch diameter of internal thread
$d_c = d - 2\left(\frac{17}{24}H\right) = D - 1.227\,P$	d = major diameter of external thread
Tensile stress area = $\frac{\pi}{4}\left(\frac{d_p + d_c}{2}\right)^2$	d_c = minor diameter of external thread
	d_p = pitch diameter of external thread
	H = height of fundamental triangle
	H_1 = maximum depth of engagement
	h_3 = basic depth of external thread
	P = pitch

Table 7.2 *Basic dimensions of ISO metric screw threads (Coarse threads)*

Designation	Nominal or Major diameter ($d = D$) (mm)	Pitch (P) (mm)	Pitch diameter ($d_p = D_p$) (mm)	Minor diameter (mm)		Tensile stress area (mm^2)
				d_c	D_c	
M 2.5	2.5	0.45	2.208	1.948	2.013	3.39
M 3	3	0.5	2.675	2.387	2.459	5.03
M 4	4	0.7	3.545	3.141	3.242	8.78
M 5	5	0.8	4.480	4.019	4.134	14.2
M 6	6	1	5.350	4.773	4.917	20.1
M 8	8	1.25	7.188	6.466	6.647	36.6
M 10	10	1.5	9.026	8.160	8.376	58.0
M 12	12	1.75	10.863	9.853	10.106	84.3
M 16	16	2	14.701	13.546	13.835	157
M 20	20	2.5	18.376	16.933	17.294	245
M 24	24	3	22.051	20.319	20.752	353
M 30	30	3.5	27.727	25.706	26.211	561
M 36	36	4	33.402	31.093	31.670	817
M 42	42	4.5	39.077	36.479	37.129	1120
M 48	48	5	44.752	41.866	42.587	1470
M 56	56	5.5	52.428	49.252	50.046	2030
M 64	64	6	60.103	56.639	57.505	2680

(ISO: 724)

Note: A screw thread of coarse series is designated by the letter 'M' followed by the value of the nominal diameter in mm. For example, M 12.

Table 7.3 *Basic dimensions of ISO metric screw threads (Fine threads)*

Designation	Nominal or Major diameter ($d = D$) (mm)	Pitch (P) (mm)	Pitch diameter ($d_p = D_p$) (mm)	Minor diameter (mm)		Tensile stress area (mm^2)
				d_c	D_c	
M 2.5 × 0.45	2.5	0.45	2.208	1.948	2.013	3.39
M 2.5 × 0.35	2.5	0.35	2.273	2.071	2.121	3.70
M 3 × 0.5	3	0.5	2.675	2.387	2.459	5.03
M 3 × 0.35	3	0.35	2.773	2.571	2.621	5.61
M 4 × 0.7	4	0.7	3.545	3.141	3.242	8.78
M 4 × 0.5	4	0.5	3.675	3.387	3.459	9.79

(*Contd.*)

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M 5 × 0.8	5	0.8	4.480	4.019	4.134	14.2
M 5 × 0.5	5	0.5	4.675	4.387	4.459	16.1
M 6 × 1	6	1	5.350	4.773	4.917	20.1
M 6 × 0.75	6	0.75	5.513	5.080	5.188	22.0
M 8 × 1.25	8	1.25	7.188	6.466	6.647	36.6
M 8 × 1	8	1	7.350	6.773	6.917	39.2
M 8 × 0.75	8	0.75	7.513	7.080	7.188	41.8
M 10 × 1.5	10	1.5	9.026	8.160	8.376	58.0
M 10 × 1.25	10	1.25	9.188	8.466	8.647	61.2
M 10 × 1	10	1	9.350	8.773	8.917	64.5
M 10 × 0.75	10	0.75	9.513	9.080	9.188	67.9
M 12 × 1.75	12	1.75	10.863	9.853	10.106	84.3
M 12 × 1.5	12	1.5	11.026	10.160	10.376	88.1
M 12 × 1.25	12	1.25	11.188	10.466	10.647	92.1
M 12 × 1	12	1	11.350	10.773	10.917	96.1
M 16 × 2	16	2	14.701	13.546	13.835	157
M 16 × 1.5	16	1.5	15.026	14.160	14.376	167
M 16 × 1	16	1	15.350	14.773	14.917	178
M 20 × 2.5	20	2.5	18.376	16.933	17.294	245
M 20 × 2	20	2	18.701	17.546	17.835	258
M 20 × 1.5	20	1.5	19.026	18.160	18.376	272
M 20 × 1	20	1	19.350	18.773	18.917	285
M 24 × 3	24	3	22.051	20.319	20.752	353
M 24 × 2	24	2	22.701	21.546	21.835	384
M 24 × 1.5	24	1.5	23.026	22.160	22.376	401
M 24 × 1	24	1	23.350	22.773	22.917	418
M 30 × 3.5	30	3.5	27.727	25.706	26.211	561
M 30 × 3	30	3	28.051	26.319	26.752	581
M 30 × 2	30	2	28.701	27.546	27.835	621
M 30 × 1.5	30	1.5	29.026	28.160	28.376	642
M 30 × 1	30	1	29.350	28.773	28.917	663
M 36 × 4	36	4	33.402	31.093	31.670	817
M 36 × 3	36	3	34.051	32.319	32.752	865

(*Contd.*)

(*Contd.*)

M 36 × 2	36	2	34.701	33.546	33.835	915
M 36 × 1.5	36	1.5	35.026	34.160	34.376	940
M 42 × 4.5	42	4.5	39.077	36.479	37.129	1120
M 42 × 4	42	4	39.402	37.093	37.670	1150
M 42 × 3	42	3	40.051	38.319	38.752	1210
M 42 × 2	42	2	40.701	39.546	39.835	1260
M 42 × 1.5	42	1.5	41.026	40.160	40.376	1290
M 48 × 5	48	5	44.752	41.866	42.587	1470
M 48 × 4	48	4	45.402	43.093	43.670	1540
M 48 × 3	48	3	46.051	44.319	44.752	1600
M 48 × 2	48	2	46.701	45.546	45.835	1670
M 48 × 1.5	48	1.5	47.026	46.160	46.376	1710
M 56 × 5.5	56	5.5	52.428	49.252	50.046	2030
M 56 × 4	56	4	53.402	51.093	51.670	2140
M 56 × 3	56	3	54.051	52.319	52.752	2220
M 56 × 2	56	2	54.701	53.546	53.835	2300
M 56 × 1.5	56	1.5	55.026	54.160	54.376	2340
M 64 × 6	64	6	60.103	56.639	57.505	2680
M 64 × 4	64	4	61.402	59.093	59.670	2850
M 64 × 3	64	3	62.051	60.319	60.752	2940
M 64 × 2	64	2	62.701	61.546	61.835	3030
M 64 × 1.5	64	1.5	63.026	62.160	62.376	3080
M 72 × 6	72	6	68.103	64.639	65.505	3460
M 72 ×4	72	4	69.402	67.093	67.670	3660
M 72 × 3	72	3	70.051	68.319	68.752	3760
M 72 × 2	72	2	70.701	69.546	69.835	3860
M 72 × 1.5	72	1.5	71.026	70.160	70.376	3910
M 80 × 6	80	6	76.103	72.639	73.505	4340
M 80 × 4	80	4	77.402	75.093	75.670	4570
M 80 × 3	80	3	78.051	76.319	76.752	4680
M 80 × 2	80	2	78.701	77.546	77.835	4790
M 80 × 1.5	80	1.5	79.026	78.160	78.376	4850
M 90 × 6	90	6	86.103	82.639	83.505	5590

(*Contd.*)

(*Contd.*)

M 90 × 4	90	4	87.402	85.093	85.670	5840
M 90 × 3	90	3	88.051	86.319	86.752	5970
M 90 × 2	90	2	88.701	87.546	87.835	6100
M 100 × 6	100	6	96.103	92.639	93.505	7000
M 100 × 4	100	4	97.402	95.093	95.670	7280
M 100 × 3	100	3	98.051	96.319	96.752	7420
M 100 × 2	100	2	98.701	97.546	97.835	7560

(ISO: 724)

Note: A screw thread of fine series is specified by the letter 'M', followed by the values of the nominal diameter and the pitch in mm and separated by symbol '×'. For example, M 12 × 1.25.

Table 7.4 *Tolerance classes for commercial bolts and nuts*

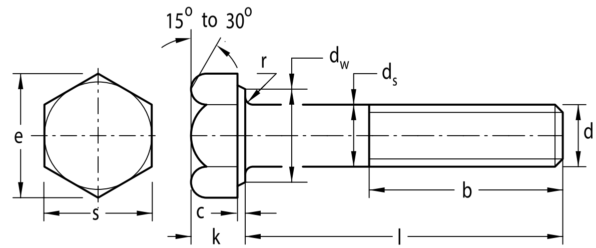
Tolerance class		Nut	Bolt
Medium	For threads of 1 to 1.4 mm dia	5H	6h
	For threads over 1.4 mm dia	6H	6g
Coarse		7H	8g

(ISO: 965-1)

Note: (i) Tolerance class medium is used for general use. (ii) Tolerance class coarse is used when no special requirements are specified with regard to accuracy and in cases where production difficulties may arise, for example, in case of threads on hot-rolled rods or when threads are cut in deep blind holes.

7.2 HEXAGON AND SQUARE BOLTS, SCREWS AND NUTS

Table 7.5(a) *Dimensions for hexagon head bolts of product grades A and B*



Thread size d			M3	M4	M5	M6	M8	M10	M12	M16	M20	M24	M30	M36
P (pitch)			0.5	0.7	0.8	1	1.25	1.5	1.75	2	2.5	3	3.5	4
b ref		*	12	14	16	18	22	26	30	38	46	54	66	78
		**	–	–	–	–	28	32	36	44	52	60	72	84
		***	–	–	–	–	–	–	–	57	65	73	85	97
c		Min	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.2	0.2	0.2	0.2	0.2
		Max	0.4	0.4	0.5	0.5	0.6	0.6	0.6	0.8	0.8	0.8	0.8	0.8
d_s		Max	3	4	5	6	8	10	12	16	20	24	30	36
		Min	2.86	3.82	4.82	5.82	7.78	9.78	11.73	15.73	19.67	23.67	29.67	35.61
d_w	1 ≤ 10d or 150 mm	Min	4.6	5.9	6.9	8.9	11.6	14.6	16.6	22.5	28.2	33.6	42.7	51.1
	1 > 10d or 150 mm		–	–	6.7	8.7	11.14	14.4	16.4	22	27.7	33.2		
e	1 ≤ 10d or 150 mm	Min	6.07	7.66	8.79	11.05	14.38	17.77	20.03	26.75	33.53	39.98	50.85	60.79
	1 > 10d or 150 mm		–	–	8.63	10.89	14.20	17.59	19.85	26.17	32.95	39.55		
k	1 ≤ 10d or 150 mm	Nom	2	2.8	3.5	4	5.3	6.4	7.5	10	12.5	15	18.7	22.5
		Min	1.88	2.68	3.35	3.85	5.15	6.22	7.32	9.82	12.28	14.78	–	–
		Max	2.12	2.92	3.65	4.15	5.45	6.58	7.68	10.18	12.72	15.22	–	–
	1 > 10d or 150 mm	Min	–	–	3.26	3.76	5.06	6.11	7.21	9.71	12.15	14.65	18.28	22.08
		Max	–	–	3.74	4.24	5.54	6.69	7.79	10.29	12.85	15.35	19.12	22.92
r		Min	0.1	0.2	0.2	0.25	0.4	0.4	0.6	0.6	0.8	0.8	1	1

(*Contd.*)

(*Contd.*)

s		Max	5.5	7	8	10	13	16	18	24	30	36	46	55
	1 ≤ 10d or 150 mm	Min	5.32	6.78	7.78	9.78	12.73	15.73	17.73	23.67	29.67	35.38	45	53.8
	1 > 10d or 150 mm		–	–	7.64	9.64	12.57	15.57	17.57	23.16	29.16	35		

(ISO: 4014)

Note: (*) For nominal lengths ≤ 125 mm
(**) For nominal lengths >125 mm and ≤ 200 mm
(***) For nominal lengths > 200 mm (All dimensions are in mm)
Product Grades – (i) Grade A for products with d ≤ M24 and l ≤ 10d or 150 mm, whichever is shorter
(ii) Grade B for products with d > M24 and l > 10d or 150 mm, whichever is shorter
Threads – (i) Pitch – Coarse (ii) Tolerance – 6 g (iii) Property Class – 4.6, 4.8, 5.6, 6.8, 8.8, 9.8 and 10.9.
Designation – A hexagon head bolt of thread size M20, nominal length 100 mm and property class 8.8 is designated as,
Hexagon head bolt M20 × 100 – 8.8

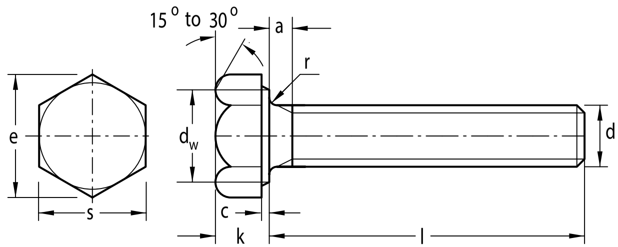
Table 7.5(b) *Preferred length-size combinations for hexagon head bolts of product grades A and B*

l (Nom)	Size d											
	M3	**M4**	**M5**	**M6**	**M8**	**M10**	**M12**	**M16**	**M20**	**M24**	**M30**	**M36**
20	X											
25	X	X	X									
30	X	X	X	X								
35		X	X	X	X							
40		X	X	X	X	X						
45			X	X	X	X	X					
50			X	X	X	X	X					
55				X	X	X	X	X				
60				X	X	X	X	X				
65					X	X	X	X	X			
70					X	X	X	X	X			
80					X	X	X	X	X	X		
90						X	X	X	X	X	X	
100						X	X	X	X	X	X	
110							X	X	X	X	X	X
120							X	X	X	X	X	X
130								X	X	X	X	X
140								X	X	X	X	X
150								X	X	X	X	X

(ISO: 4014)

Note: Preferred lengths are indicated by ‘X’.

Table 7.6 *(a) Dimensions for hexagon head screws of product grades A and B*



Thread size d			M3	M4	M5	M6	M8	M10	M12	M16	M20	M24	M30	M36
P (pitch)			0.5	0.7	0.8	1	1.25	1.5	1.75	2	2.5	3	3.5	4
a		Max.	1.5	2.1	2.4	3	3.75	4.5	5.25	6	7.5	9	10.5	12
c		Min.	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.2	0.2	0.2	0.2	0.2
		Max.	0.4	0.4	0.5	0.5	0.6	0.6	0.6	0.8	0.8	0.8	0.8	0.8
d_w	1 ≤ 10d or 150 mm	Min.	4.6	5.9	6.9	8.9	11.6	14.6	16.6	22.5	28.2	33.6	42.7	51.1
	1 > 10d or 150 mm		–	–	6.7	8.7	11.4	14.4	16.4	22	27.7	33.2		
e	1 ≤ 10d or 150 mm	Min.	6.01	7.66	8.79	11.05	14.38	17.77	20.03	26.75	33.53	39.98	50.85	60.79
	1 > 10d or 150 mm		–	–	8.63	10.89	14.20	17.59	19.85	26.17	32.95	39.55		
k		Nom.	2	2.8	3.5	4	5.3	6.4	7.5	10	12.5	15	18.7	22.5
	1 ≤ 10d or 150 mm	Min.	1.88	2.68	3.35	3.85	5.15	6.22	7.32	9.82	12.28	14.78	–	–
		Max.	2.12	2.92	3.65	4.15	5.45	6.58	7.68	10.18	12.72	15.22	–	–
	1 > 10d or 150 mm	Min.	–	–	3.26	3.76	5.06	6.11	7.21	9.71	12.15	14.65	18.28	22.08
		Max.	–	–	3.74	4.24	5.54	6.69	7.79	10.29	12.85	15.35	19.12	22.92
r		Min.	0.1	0.2	0.2	0.25	0.4	0.4	0.6	0.6	0.8	0.8	1	1

(*Contd.*)

(*Contd.*)

s		Max.	5.5	7	8	10	13	16	18	24	30	36	46	55
	l ≤ 10d or 150 mm	Min.	5.32	6.78	7.78	9.78	12.73	15.73	17.73	23.67	29.67	35.38	45	53.8
	l > 10d or 150 mm		–	–	7.64	9.64	12.57	15.57	17.57	23.16	29.16	35		

(ISO: 4017)

Note: All dimensions are in mm.
Product Grades – (i) Grade A for products with d ≤ M24 and l ≤ 10d or 150 mm, whichever is shorter
(ii) Grade B for products with d > M24 and l > 10d or 150 mm, whichever is shorter
Threads – (i) Pitch – Coarse (ii) Tolerance – 6 g (iii) Property Class – 4.6, 4.8, 5.6, 5.8, 6.8, 8.8, 9.8 and 10.9.
Designation – A hexagon head screw of thread size M20, nominal length 60 mm and property class 8.8 is designated as,
Hexagon Head Screw M20 × 60 – 8.8

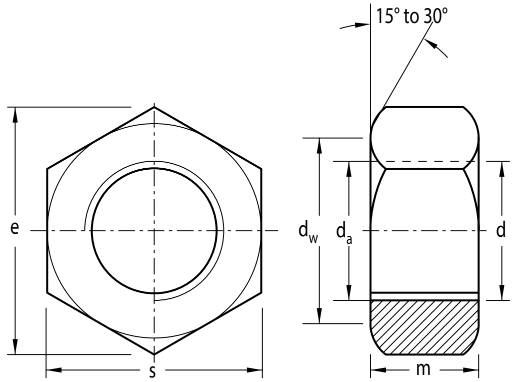
Table 7.6 *(b) Preferred length-size combinations for hexagon head screws of product grades A and B*

l (Nom)	Thread size d											
	M3	M4	M5	M6	M8	M10	M12	M16	M20	M24	M30	M36
6	X											
8	X	X										
10	X	X	X									
12	X	X	X	X								
16	X	X	X	X	X							
20	X	X	X	X	X	X						
25	X	X	X	X	X	X	X					
30	X	X	X	X	X	X	X					
35		X	X	X	X	X	X	X				
40		X	X	X	X	X	X	X	X	X	X	X
45			X	X	X	X	X	X	X	X	X	X
50			X	X	X	X	X	X	X	X	X	X
55				X	X	X	X	X	X	X	X	X
60				X	X	X	X	X	X	X	X	X
65					X	X	X	X	X	X	X	X
70					X	X	X	X	X	X	X	X
80					X	X	X	X	X	X	X	X
90						X	X	X	X	X	X	X
100						X	X	X	X	X	X	X

(ISO: 4017)

Note: Preferred lengths are indicated by 'X'.

Table 7.7 *Dimensions for hexagon nut of product grades A and B*

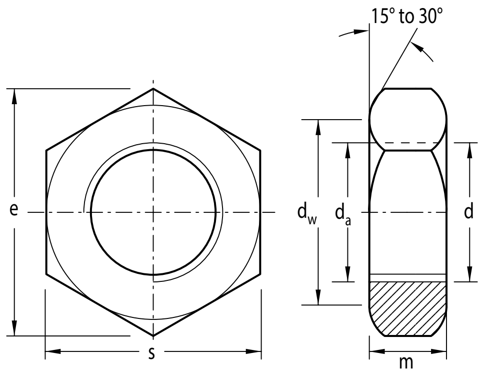


Thread size d		M3	M4	M5	M6	M8	M10	M12	M16	M20	M24	M30	M36
P (pitch)		0.5	0.7	0.8	1	1.25	1.5	1.75	2	2.5	3	3.5	4
d_a	Min.	3	4	5	6	8	10	12	16	20	24	30	36
	Max.	3.45	4.6	5.75	6.75	8.75	10.8	13	17.3	21.6	25.9	32.4	38.9
d_w	Min.	4.6	5.9	6.9	8.9	11.6	14.6	16.6	22.5	27.7	33.2	42.7	51.1
e	Min.	6.01	7.66	8.79	11.05	14.38	17.77	20.03	26.75	32.95	39.55	50.85	60.79
m	Max.	2.4	3.2	4.7	5.2	6.8	8.4	10.8	14.8	18	21.5	25.6	31
	Min.	2.15	2.9	4.4	4.9	6.44	8.04	10.37	14.1	16.9	20.2	24.3	29.4
s	Max.	5.5	7	8	10	13	16	18	24	30	36	46	55
	Min.	5.32	6.78	7.78	9.78	12.73	15.73	17.73	23.67	29.16	35	45	53.8

(ISO: 4032)

Note: All dimensions are in mm.
Product Grades – (i) Grade A for products with d ≤ M16
(ii) Grade B for products with d > M16
Threads – (i) Pitch – Coarse (ii) Tolerance – 6 H (iii) Property Class – For size < M3 – 6, for size ≥ M3 – 6, 8 or 10
Designation – A hexagon nut of thread size M10 and property class 8 is designated as, Hexagon Nut M10 – 8

Table 7.8 *Dimensions for hexagon thin nut (chamfered) of product grades A and B*

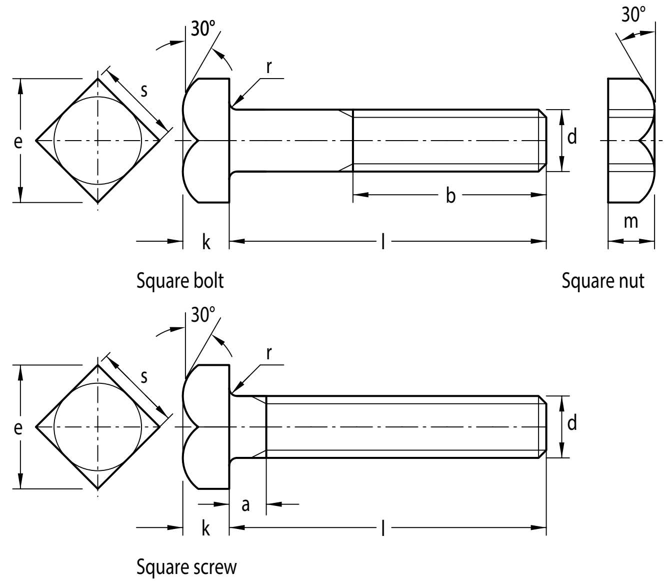


Thread size d		M3	M4	M5	M6	M8	M10	M12	M16	M20	M24	M30	M36
P (pitch)		0.5	0.7	0.8	1	1.25	1.5	1.75	2	2.5	3	3.5	4
d_a	Min.	3	4	5	6	8	10	12	16	20	24	30	36
	Max.	3.45	4.6	5.75	6.75	8.75	10.8	13	17.3	21.6	25.9	32.4	38.9
d_w	Min.	4.6	5.9	6.9	8.9	11.6	14.6	16.6	22.5	27.7	33.2	42.7	51.1
e	Min.	6.01	7.66	8.79	11.05	14.38	17.77	20.03	26.75	32.95	39.55	50.85	60.79
m	Max.	1.8	2.2	2.7	3.2	4	5	6	8	10	12	15	18
	Min.	1.55	1.95	2.45	2.9	3.7	4.7	5.7	7.42	9.1	10.9	13.9	16.9
s	Max.	5.5	7	8	10	13	16	18	24	30	36	46	55
	Min.	5.32	6.78	7.78	9.78	12.73	15.73	17.73	23.67	29.16	35	45	53.8

(ISO: 4035)

Note: All dimensions are in mm.
Product Grades – (i) Grade A for products with d ≤ M16
(ii) Grade B for products with d > M16
Threads – (i) Pitch – Coarse (ii) Tolerance – 6 H (iii) Property Class – 04, 05
Designation – A hexagon thin nut (chamfered) of thread size M8 and property class 05 is designated as,
Hexagon Thin Nut (Chamfered) M8 – 05

Table 7.9(a) *Dimensions for black square head bolts, screws and nuts*



Thread size d		M6	M8	M10	M12	M16	M20	M24	M30	M36
d	Nom.	6	8	10	12	16	20	24	30	36
	Max.	6.48	8.90	10.90	13.10	17.10	21.50	25.30	31.30	37.60
	Min.	5.70	7.64	9.64	11.57	15.57	19.48	23.48	29.48	35.38
s	Nom.	10	13	17	19	24	30	36	46	55
[$s \le 19$ – h 14]	Max.	10	13	17	19	24	30	36	46	55
[s > 19 – h 15]	Min.	9.64	12.57	16.57	18.48	23.16	29.16	35	45	53.80
e	Min.	12.53	16.34	21.54	24.02	30.11	37.91	45.50	58.50	69.94
k (*js*16)	Nom.	4	5.5	7	8	10	13	15	19	23
	Max.	4.38	5.88	7.45	8.45	10.45	13.55	15.55	19.65	23.65
	Min.	3.62	5.12	6.55	7.55	9	12.45	14.45	18.35	22.35
r	Min.	0.25	0.4	0.4	0.6	0.6	0.8	0.8	1.0	1.0
a	Max.	4.0	4.5	5.0	6.0	7.5	9.0	11	–	–
b	b_1 Min.	18	22	26	30	38	46	54	66	78
	b_2 Min.	–	28	32	36	44	52	60	72	84
	b_3 Min.	–	–	–	49	57	65	73	85	97
m (*js*16)	Nom.	5	6.5	8	10	13	16	19	24	29
	Max.	5.38	6.95	8.45	10.45	13.55	16.55	19.65	24.65	29.65
	Min.	4.62	6.05	7.55	9.55	12.45	15.45	18.35	23.35	28.35

Note: (i) Use b_1 for $l \le 130$, b_2 for $130 < l \le 200$ and b_3 for $l > 200$.
(ii) All dimensions are in mm.
(iii) In the above table, only first preference thread sizes are given. For second preference thread sizes, refer to standard.

(*Contd.*)

(*Contd.*)

Grade – The bolts, screws and nuts conform to black grade B.
Property Class – For bolts and screws, property class 4.6 and for nuts, property class 4.
Designations – (i) A square bolt of thread size M10 and length 30 mm is designated as Square Bolt M10 × 30.
(ii) A square screw of thread size M10 and length 30 mm is designated as square screw M10 × 30.
(iii) A square nut of thread size M10 is designated as Square Nut M10.

Table 7.9(b) *Preferred length-size combinations for black square head bolts*

l	M6	M8	M10	M12	M16	M20	M24	M30	M36
25	X								
30	X	X							
35	X	X	X						
40	X	X	X	X					
45	X	X	X	X					
50	X	X	X	X	X				
55	X	X	X	X	X				
60	X	X	X	X	X	X			
65	X	X	X	X	X	X			
70	X	X	X	X	X	X	X		
75	X	X	X	X	X	X	X		
80	X	X	X	X	X	X	X		
90	X	X	X	X	X	X	X	X	
100	X	X	X	X	X	X	X	X	
110		X	X	X	X	X	X	X	X
120		X	X	X	X	X	X	X	X
130			X	X	X	X	X	X	X
140			X	X	X	X	X	X	X
150			X	X	X	X	X	X	X

Note: Preferred lengths are indicated by 'X'.

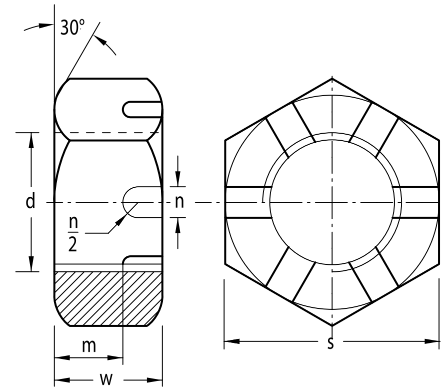
Table 7.9(c) *Preferred length-size combinations for black square head screws*

l	M6	M8	M10	M12	M16	M20	M24
15	X	X	X				
20	X	X	X	X			
25	X	X	X	X	X		
30	X	X	X	X	X		
35	X	X	X	X	X		
40	X	X	X	X	X		
45		X	X	X	X	X	
50		X	X	X	X	X	
55			X	X	X	X	X
60			X	X	X	X	X
65				X	X	X	X
70				X	X	X	X
75				X	X	X	X
80				X	X	X	X

Note: Preferred lengths are indicated by 'X'.

7.3 DIMENSIONS OF NUTS

Table 7.10 *Dimensions for slotted nuts – Precision grade*

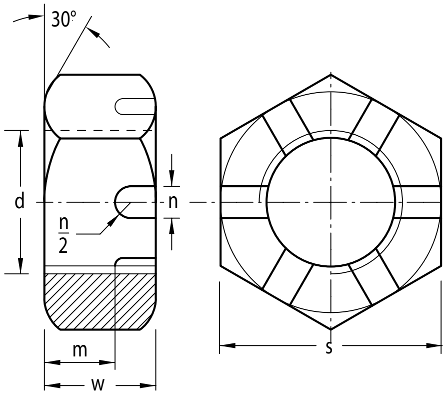


Size d		M4	M5	M6	M8 M8 × 1	M10 M10 × 1.25	M12 M12 × 1.25	M16 M16 × 1.5	M20 M20 × 1.5	M30 M30 × 2	M36 M36 × 3
w (h14)	Nom.	5	6	7.5	9.5	12	15	19	22	33	38
	Max.	5	6	7.5	9.5	12	15	19	22	33	38
	Min.	4.7	5.7	7.14	9.14	11.57	14.57	18.48	21.48	32.38	37.38
m (*h*15)	Nom.	3.2	4	5	6.5	8	10	13	16	24	29
	Max.	3.2	4	5	6.5	8	10	13	16	24	29
	Min.	2.72	3.52	4.52	5.92	8	9.42	12.30	15.30	23.16	28.16
n (*H*14)	Nom.	1.2	1.4	2	2.5	2.8	3.5	4.5	4.5	7	7
	Max.	1.45	1.65	2.25	2.75	3.05	3.8	4.8	4.8	7.36	7.36
	Min.	1.20	1.40	2	2.5	2.8	3.5	4.5	4.5	7	7
s (≤8 – *h*12) (>8 – *h*13)	Nom.	7	8	10	13	17	19	24	30	46	55
	Max.	7	8	10	13	17	19	24	30	46	55
	Min.	6.85	7.85	9.78	12.73	16.73	18.67	23.67	29.67	45.38	54.26
Size of split cotter pin		1	1.2	1.6	2	2.5	3.2	4	4	6.3	6.3

(ISO: 288-1)

Note: (i) All dimensions are in mm. (ii) Property Class – 4.6 or 8.
Designations –A slotted nut of nominal size M10 of precision grade and property class 8 is designated as Slotted Nut M10-P-8.

Table 7.11 *Dimensions for slotted nuts – Black grade*

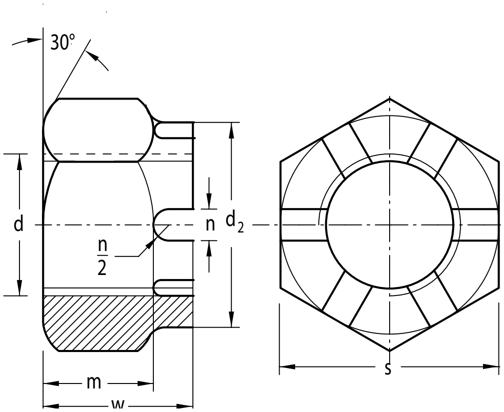


Size d		**M12**	**M16**	**M20**	**M24**	**M30**
w (*js*16)	Nom.	15	19	22	27	33
	Max.	15.55	19.65	22.65	27.65	33.80
	Min.	14.55	18.35	21.35	26.35	32.20
m (*h*16)	Nom.	10	13	16	19	24
	Max.	10	13	16	19	24
	Min.	9.10	11.90	14.90	17.70	22.70
n (*H*14)	Nom.	3.5	4.5	4.5	5.5	7
	Max.	3.8	4.8	4.8	5.8	7.36
	Min.	3.5	4.5	4.5	5.5	7
s (≤19 – *h*14) (>19 – *h*15)	Nom.	19	24	30	36	46
	Max.	19	24	30	36	46
	Min.	18.48	23.16	29.16	35	45
Size of split cotter pin	–	3.2	4	4	5	6.3

(ISO: 288-1)

Note: (i) All dimensions are in mm. (iii) Property Class – 4.6 or 8.
Designations –A slotted nut of nominal size M10 of black grade and property class 8 is designated as Slotted Nut M10-B-8.

Table 7.12(a) *Dimensions for castle nuts – Precision grade (Diameter range – M12 to M42)*



Size d		M12 M12 × 1.25	M16 M16 × 1.5	M20 M20 × 1.5	M24 M24 × 2	M30 M30 × 2	M36 M36 × 3	M42
d_2 (h14)	Nom.	17	22	28	34	42	50	58
	Max.	17	22	28	34	42	50	58
	Min.	16.57	21.48	27.48	33.38	41.38	49.38	57.26
w (h14)	Nom.	15	19	22	27	33	38	46
	Max.	15	19	22	27	33	38	46
	Min.	14.57	18.48	21.48	26.48	32.38	37.38	45.38
m (h15)	Nom.	10	13	16	19	24	29	34
	Max.	10	13	16	19	24	29	34
	Min.	9.42	12.30	15.30	18.16	23.16	28.16	33
n (H14)	Nom.	3.5	4.5	4.5	5.5	7	7	9
	Max.	3.8	4.8	4.8	5.8	7.36	7.36	9.36
	Min.	3.5	4.5	4.5	5.5	7	7	9
s (h14)	Nom.	19	24	30	36	46	55	65
	Max.	19	24	30	36	46	55	65
	Min.	18.67	23.67	29.67	35.38	45.38	54.26	64.26
Size of split cotter pin		3.2	4	4	5	6.3	6.3	8

(ISO: 288-1)

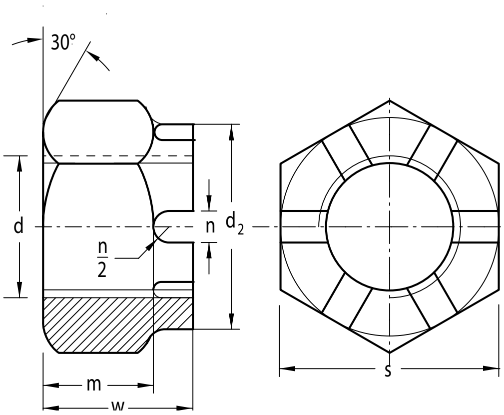
Table 7.12(b) *Dimensions for castle nuts – Precision grade (Diameter range – M48 to M100)*

Size d		M48	M56	M64	M72 × 6	M80 × 6	M90 × 6	M100 × 6
d_2 ($h14$)	Nom.	65	75	85	95	105	120	135
	Max.	65	75	85	95	105	120	135
	Min.	64.26	74.26	84.13	94.13	104.13	119.13	134
w ($h14$)	Nom.	50	57	66	73	79	92	100
	Max.	50	57	66	73	79	92	100
	Min.	49.38	56.26	65.26	72.26	78.26	91.13	99.13
m ($h15$)	Nom.	38	45	51	58	64	72	80
	Max.	38	45	51	58	64	72	80
	Min.	37	44	49.80	56.80	62.80	70.80	78.80
n ($H14$)	Nom.	9	9	11	11	11	14	14
	Max.	9.36	9.36	11.43	11.43	11.43	14.43	14.43
	Min.	9	9	11	11	11	14	14
s ($h14$)	Nom.	75	85	95	105	115	130	145
	Max.	75	85	95	105	115	130	145
	Min.	74.26	84.13	94.13	104.13	114.13	129.00	144
Size of split cotter pin		8	8	10	10	10	13	13

(ISO: 288-1)

Note: (i) All dimensions are in mm. (ii) Property Class – 4.6 or 8.
Designations –A castle nut of nominal size M10 of precision grade and property class 8 is designated as
Castle Nut M10-P-8.

Table 7.13(a) *Dimensions for castle nuts – Black grade (Diameter range – M12 to M42)*



Size d		M12	M16	M20	M24	M30	M36	M42
d_2 (h16)	Nom.	17	22	28	34	42	50	58
	Max.	17	22	28	34	42	50	58
	Min.	15.90	20.70	26.70	32.40	40.40	48.40	56.10
w (js16)	Nom.	15	19	22	27	33	38	46
	Max.	15.55	19.65	22.65	27.65	33.80	38.80	46.80
	Min.	14.45	18.35	21.35	26.35	32.20	37.20	45.20
m (h16)	Nom.	10	13	16	19	24	29	34
	Max.	10	13	16	19	24	29	34
	Min.	9.10	11.90	14.90	17.70	22.70	27.70	32.40
n (H14)	Nom.	3.5	4.5	4.5	5.5	7	7	9
	Max.	3.8	4.8	4.8	5.8	7.36	7.36	9.36
	Min.	3.5	4.5	4.5	5.5	7	7	9
s (≤19 – h14) (>19 – h15)	Nom.	19	24	30	36	46	55	65
	Max.	19	24	30	36	46	55	65
	Min.	18.48	23.16	29.16	35.00	45.00	53.80	63.80
Size of split cotter pin		3.2	4	4	5	6.3	6.3	8

(ISO: 288-1)

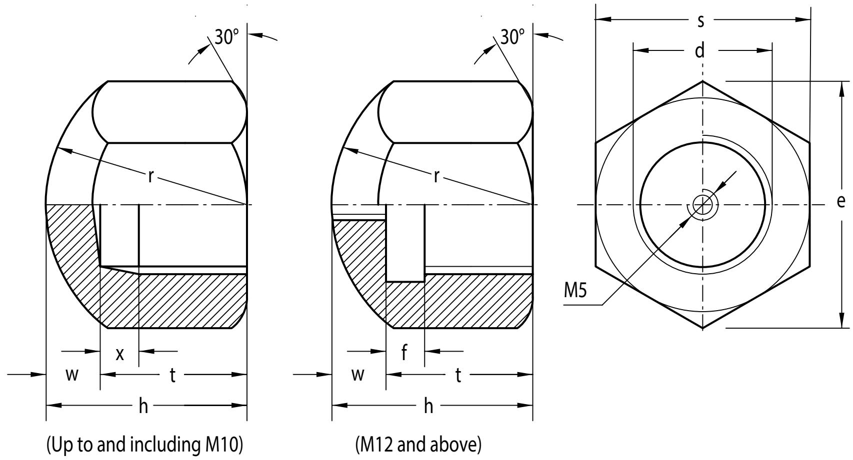
Table 7.13(b) *Dimensions for castle nuts – Black grade (Diameter range – M48 to M100)*

Size d		M48	M56	M64	M72 × 6	M80 × 6	M90 × 6	M100 × 6
d_2 ($h16$)	Nom.	65	75	85	95	105	120	135
	Max.	65	75	85	95	105	120	135
	Min.	63.10	73.10	82.80	92.80	102.80	117.80	132.50
w ($js16$)	Nom.	50	57	66	73	79	92	100
	Max.	50.80	57.95	66.95	73.95	79.95	93.10	101.10
	Min.	49.20	56.05	65.05	72.05	78.05	90.90	98.90
m ($h16$)	Nom.	38	45	51	58	64	72	80
	Max.	38	45	51	58	64	72	80
	Min.	36.40	43.40	49.10	56.10	62.10	70.10	78.10
n ($H14$)	Nom.	9	9	11	11	11	14	14
	Max.	9.36	9.36	11.43	11.43	11.43	14.43	14.43
	Min.	9	9	11	11	11	14	14
s ($h15$)	Nom.	75	85	95	105	115	130	145
	Max.	75	85	95	105	115	130	145
	Min.	73.80	83.60	93.60	103.60	113.60	128.40	143.40
Size of split cotter pin		8	8	10	10	10	13	13

(ISO: 288-1)

Note: (i) All dimensions are in mm. (ii) Property class – 4.6 or 8.
Designations –A castle nut of nominal size M10 of black grade and property class 8 is designated as
Castle Nut M10-B-8.

Table 7.14 *Dimensions for cap nuts*

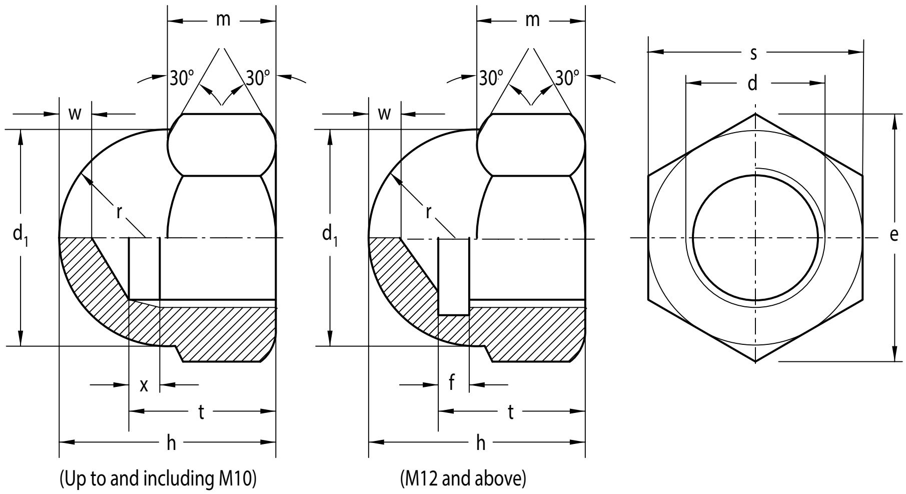


d		M6	M8	M10	M12	M16	M20	M24	M30	M34
s (h13)	Nom.	10	13	17	19	24	30	36	46	55
	Max.	10	13	17	19	24	30	36	46	55
	Min.	9.78	12.73	16.73	18.67	23.67	29.67	35.38	45.38	54.38
e	Min.	11.05	14.38	18.90	21.10	26.75	33.53	39.98	51.28	61.31
h (*h*14)	Nom.	9	12	14	16	20	25	30	34	44
	Max.	9	12	14	16	20	25	30	34	44
	Min.	8.64	11.57	13.57	15.57	19.48	24.48	29.48	33.38	43.38
r	Nom.	12	15	20	25	30	35	40	60	70
w	Min.	1.5	2	2	2	2	2.5	3	3	4
t (*js*15)	Nom.	7	9.5	11	13.5	17	21	24	28	36
	Max.	7.29	9.79	11.35	13.85	17.35	21.42	24.42	28.42	36.50
	Min.	6.71	9.21	10.65	13.15	16.65	20.58	23.58	27.58	35.50

(DIN: 917)

Note:
(i) Cap nuts are used for providing sealing for projected thread parts or sometimes merely for decorative purposes.
(ii) All dimensions are in mm.
(iii) Grade – Precision (P)
(iv) Use Slotted Grub screw of size ‘A’ M 5 × 6, where the hole is required to be sealed. (For sizes above M24 only)
(v) Property Class – 6 for sizes up to M30 and 4 for sizes above M30.
Designations –A cap nut of nominal size M10 is designated as Cap Nut M10.

Table 7.15 *Dimensions for domed cap nuts*



d		M6	M8	M10	M12	M16	M20	M24
s *s*≤32, *h*13 *s*>32 *h*14	Nom.	10	13	17	19	24	30	36
	Max.	10	13	17	19	24	30	36
	Min.	9.78	12.75	16.73	18.67	23.67	29.67	35.38
e	Min.	11.05	14.38	18.90	21.10	26.75	33.53	39.98
h (*h*13)	Nom.	12	15	18	22	28	34	42
	Max.	12	15	18	22	28	34	42
	Min.	11.73	14.73	17.73	21.67	27.67	33.61	41.61
r	Nom.	4.75	6.25	8	9	11.5	14	17
w	Min.	2	2	2	3	3	3	3
t (*js*15)	Nom.	8	11	13	16	21	26	31
	Max.	8.29	11.35	13.35	16.35	21.42	26.42	31.5
	Min.	7.71	10.65	12.65	15.65	20.58	25.58	30.5
m (*js*14)	Nom.	5	6.5	8	10	13	16	19
	Max.	5.15	6.68	8.18	10.68	13.22	16.22	19.22
	Min.	4.85	6.32	7.82	9.32	12.78	15.78	17.78
d_1	Nom.	9.5	12.5	16	18	23	28	34

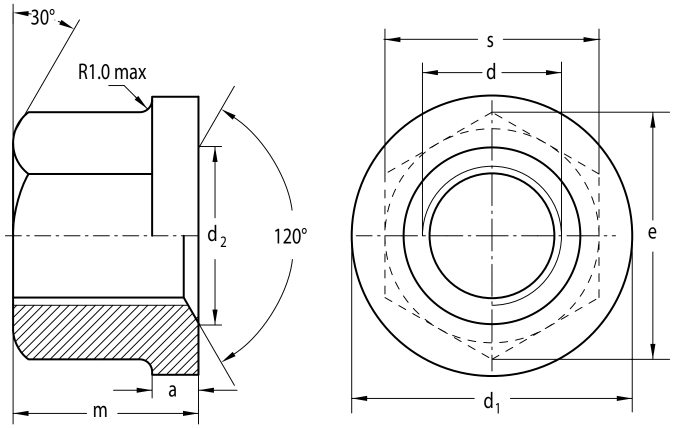
(DIN: 1587)

Note:

(i) Domed cap nuts are used for providing sealing for projected thread parts or sometimes merely for decorative purposes.
(ii) All dimensions are in mm.
(iii) Grade – Precision (P)
(iv) Property Class – 6

Designations –A domed cap nut of nominal size M10 is designated as Domed Cap Nut M10.

Table 7.16 *Dimensions for hexagon nut with collar*



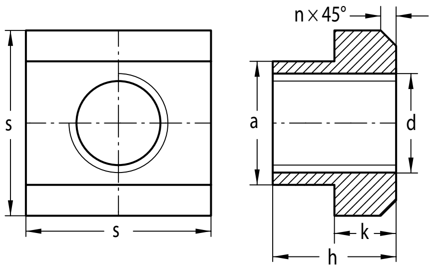
Size d		M8	M10	M12	M16	M20	M24	M30	M36
d_1 (h13)	Nom.	18	22	25	31	37	45	58	68
	Max.	18	22	25	31	37	45	58	68
	Min.	17.73	21.67	24.67	30.61	36.61	44.61	57.54	67.54
a (js14)	Nom.	3.5	4	4	5	6	6	8	10
	Max.	3.65	4.15	4.15	5.15	6.15	6.15	8.18	10.18
	Min.	3.35	3.85	3.85	4.85	5.85	5.85	7.82	9.82
d_2		10	12	14	18	22	26	32	38
m (js15)	Nom.	12	15	18	24	30	36	45	54
	Max.	12.35	15.35	18.35	24.42	30.42	36.50	45.50	54.60
	Min.	11.65	14.65	17.65	23.58	29.58	35.50	44.50	53.40
e	Min.	14.38	18.90	20.88	26.75	33.53	39.98	51.28	61.31
s	Nom.	13	17	19	24	30	36	46	55
$s \le 32$, h13	Max.	13	17	19	24	30	36	46	55
$s > 32$, h14	Min.	12.73	16.73	18.67	23.67	29.67	35.38	45.38	54.26

(DIN: 6331)

Note:
(i) Hexagon nut with collar are used where frequent tightening and loosening of nut is required. The collar acts like a washer.
(ii) All dimensions are in mm.
(iii) Grade – Precision (P)
(iv) Property Class – 8 and 10

Designations –A hexagon nut with collar of nominal size M10 is designated as Hexagon Nut with Collar M10.

Table 7.17 *Dimensions of nuts for T-slot*



<table>
<tr><th rowspan="2">Nominal size of T-slot</th><th colspan="2">Width of T- nut</th><th rowspan="2">d</th><th rowspan="2">s</th><th rowspan="2">Tol. on s</th><th rowspan="2">n</th><th rowspan="2">h</th><th rowspan="2">k</th><th rowspan="2">Tol. on k</th></tr>
<tr><th>a</th><th>Tol. on a</th></tr>
<tr><td>5</td><td>5</td><td rowspan="4">–0.3
–0.5</td><td>M4</td><td>9</td><td rowspan="8">0.0
–0.5</td><td>1.0</td><td>6.5</td><td>2.5</td><td>0.0
–0.3</td></tr>
<tr><td>6</td><td>6</td><td>M5</td><td>10</td><td rowspan="3">1.6</td><td>8</td><td>4</td><td rowspan="7">0.0
–0.5</td></tr>
<tr><td>8</td><td>8</td><td>M6</td><td>13</td><td>10</td><td rowspan="2">6</td></tr>
<tr><td>10</td><td>10</td><td>M8</td><td>15</td><td>12</td></tr>
<tr><td>12</td><td>12</td><td rowspan="5">–0.3
–0.6</td><td>M10</td><td>18</td><td rowspan="4">2.5</td><td>14</td><td>7</td></tr>
<tr><td>14</td><td>14</td><td>M12</td><td>22</td><td>16</td><td>8</td></tr>
<tr><td>18</td><td>18</td><td>M16</td><td>28</td><td>20</td><td>10</td></tr>
<tr><td>22</td><td>22</td><td>M20</td><td>35</td><td>28</td><td>14</td></tr>
<tr><td>28</td><td>28</td><td>M24</td><td>44</td><td rowspan="5">0.0
–1.0</td><td>4.0</td><td>36</td><td>18</td><td rowspan="5">0.0
–1.0</td></tr>
<tr><td>36</td><td>36</td><td rowspan="3">–0.4
–0.7</td><td>M30</td><td>54</td><td rowspan="4">6.0</td><td>44</td><td>22</td></tr>
<tr><td>42</td><td>42</td><td>M36</td><td>65</td><td>52</td><td>26</td></tr>
<tr><td>48</td><td>48</td><td>M42</td><td>75</td><td>60</td><td>30</td></tr>
<tr><td>54</td><td>54</td><td>–0.4
–0.8</td><td>M48</td><td>85</td><td>70</td><td>34</td></tr>
</table>

(DIN: 508)

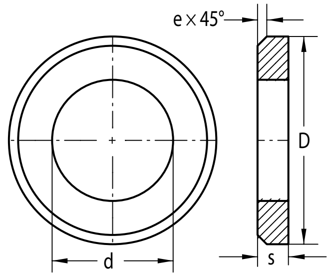
Note:

(i) T - nuts are used with studs on machine tools and other applications wherein T- slots are provided.
(ii) All dimensions are in mm.
(iii) Property Class – 6

Designations –A T-nut of width a = 22 mm is designated as T- Nut 22.

7.4 DIMENSIONS OF WASHERS

Table 7.18 *Dimensions of machined washers*



Size d (H12)	D		s		e Nom	For Bolt or Screw size
	Basic	Tol	Basic	Tol		
4.3	9	+0.0 –0.3	0.8	±0.1	0.3	M4
5.3	10	+0.0 –0.3	1.0	±0.1	0.4	M5
6.4	12.5	+0.0 –0.4	1.6	±0.2	0.6	M6
8.4	17	+0.0 –0.4	1.6	±0.2	0.6	M8
10.5	21	+0.0 –0.5	2.0	±0.2	0.6	M10
13	24	+0.0 –0.5	2.5	±0.3	0.6	M12
17	30	+0.0 –0.5	3	±0.3	0.6	M16
21	37	+0.0 –0.8	3	±0.3	1.0	M20
25	44	+0.0 –0.8	4	±0.3	1.0	M24
31	56	+0.0 –1.0	4	±0.3	1.0	M30
37	66	+0.0 –1.0	5	±0.6	1.6	M36

(ISO: 7090)

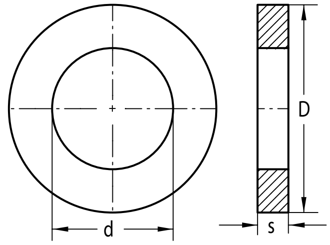
Note:

(i) Machined washers are used for precision and semi-precision grade of general-purpose bolts and screws.

(ii) All dimensions are in mm.

Designations –A machined washer of size 10.5 mm is designated as Machined Washer 10.5.

Table 7.19 *Dimensions for Type-A punched washers for hexagon head bolts and screws*



Size *d*	*D* Nominal	*s* Nominal	For bolt or screw size
4.5	9	0.8	M4
5.5	10	1	M5
6.6	12.5	1.6	M6
9	17	1.6	M8
11	21	2	M10
14	24	2.5	M12
18	30	3.15	M16
22	37	3.15	M20
26	44	4	M24
33	56	4	M30
39	66	5	M36

(ISO: 7089)

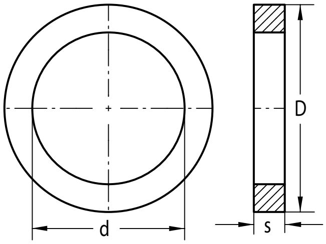
Note:

(i) Punched washers, Type A, are used for black grade of general-purpose bolts and screws.

(ii) All dimensions are in mm.

Designations –A punched washer, Type A, of size 14 mm is designated as Punched Washer A 14.

Table 7.20 *Dimensions for Type-B punched washers for round and cheese head screws*



Size *d*	*D* Nominal	*s* Nominal	For screw size
4.5	8	0.8	M4
5.5	9.5	1	M5
6.6	11	1.6	M6
9	14	1.6	M8
11	18	2	M10
14	20	2.5	M12
18	27	3.15	M16
22	33	3.15	M20

(ISO: 7089)

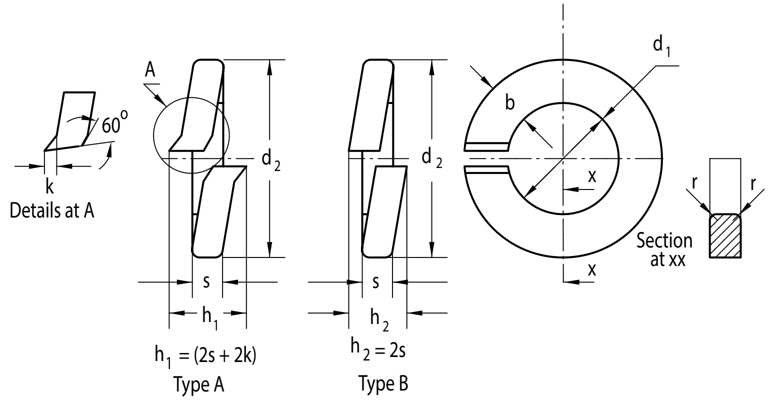
Note:

(i) Punched washers, Type B, are used for slotted head screws.

(ii) All dimensions are in mm.

Designations –A punched washer, Type B, of size 14 mm is designated as Punched Washer B 14.

Table 7.21 *Dimensions for spring washers*



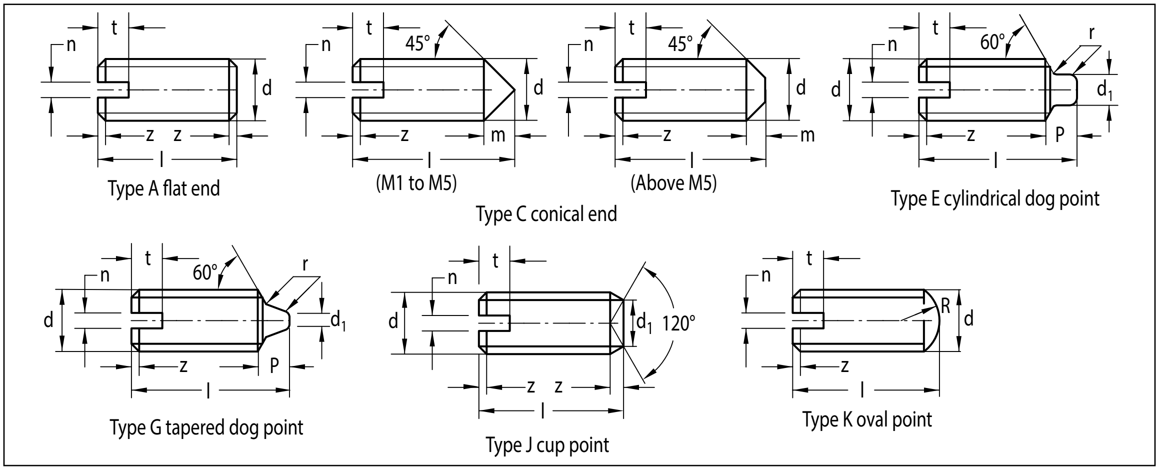
Nominal Size	d_1		d_2 Max	b		s		r Nominal	k	For Bolt, Nut or Screw Size
	Basic	Tol +		Basic	Tol ±	Basic	Tol ±			
4	4.1	0.3	7.6	1.5	0.1	0.9	0.1	0.2	0.15	M4
5	5.1	0.3	9.2	1.8	0.1	1.2	0.1	0.2	0.15	M5
6	6.1	0.4	11.8	2.5	0.15	1.6	0.1	0.3	0.2	M6
8	8.2	0.4	14.8	3	0.15	2	0.1	0.5	0.3	M8
10	10.2	0.6	18.1	3.5	0.2	2.2	0.15	0.5	0.3	M10
12	12.2	0.8	21.1	4	0.2	2.5	0.15	1.0	0.4	M12
16	16.2	0.8	27.4	5	0.2	3.5	0.2	1.0	0.4	M16
20	20.2	1	33.6	6	0.2	4	0.2	1.0	0.4	M20
24	24.5	1	40	7	0.25	5	0.2	1.6	0.5	M24
30	30.5	1.2	48.2	8	0.25	6	0.2	1.6	0.8	M30
36	36.5	1.2	58.2	10	0.25	6	0.2	1.6	0.8	M36

(DIN: 127)

Note:
(i) Type A – Spring washers with bent ends.
(ii) Type B – Spring washers with flat ends.
(iii) All dimensions are in mm.
(iv) Spring washers are heat treated after coiling, to obtain a hardness of 43 to 50 HRC.
Designations –A single coil washer having a nominal size 10 mm of Type A is designated as Spring Washer A 10.

7.5 DIMENSIONS OF SCREWS AND STUDS

Table 7.22(a) *Dimensions of slotted grub screws (M1 to M6)*



Type	Size d		M1	M1.2	M1.6	M2	M2.5	M3	M4	M5	M6
A, C, E, G, J, and K	*n* (H14)	Nom.	0.25	0.25	0.3	0.3	0.4	0.5	0.6	0.8	1
		Max.	0.35	0.35	0.45	0.45	0.55	0.65	0.75	0.95	1.15
		Min.	0.25	0.25	0.30	0.30	0.40	0.50	0.60	0.80	1.00
	t (js15)	Nom.	0.6	0.6	0.8	0.8	1	1.2	1.4	1.8	2.0
		Max.	0.70	0.70	0.90	0.90	1.10	1.40	1.60	2.00	2.20
		Min.	0.50	0.50	0.70	0.70	0.90	1.00	1.20	1.60	1.80
C	m	Nom.	0.5	0.6	0.8	1.0	1.3	1.5	2.0	2.5	2.5
E	d_1 (h13)	Nom.	–	–	–	1.3	1.6	2.0	2.5	3.5	4.0
		Max.	–	–	–	1.30	1.60	2.00	2.50	3.50	4.00
		Min.	–	–	–	1.16	1.46	1.86	2.36	3.32	3.82
	P (+IT14)	Nom.	–	–	–	1.5	2.0	2.5	3.0	3.0	3.5
		Max.	–	–	–	1.75	2.25	2.75	3.25	3.25	3.80
		Min.	–	–	–	1.50	2.00	2.50	3.00	3.00	3.50
	r	Nom.	–	–	–	0.2	0.2	0.3	0.3	0.3	0.4
G	d_1 (h13)	Nom.	–	–	–	–	–	–	–	1.6	2.4
		Max.	–	–	–	–	–	–	–	1.60	2.40
		Min.	–	–	–	–	–	–	–	1.46	2.26
	P (+IT14)	Nom.	–	–	–	–	–	–	–	4.0	4.5
		Max.	–	–	–	–	–	–	–	4.30	4.80
		Min.	–	–	–	–	–	–	–	4.00	4.50
	r	Nom.	–	–	–	–	–	–	–	0.4	0.4
J	d_1 (h14)	Nom.	–	–	–	–	–	1.4	2.0	2.5	3.0
		Max.	–	–	–	–	–	1.40	2.00	2.50	3.00
		Min.	–	–	–	–	–	1.15	1.75	2.25	2.75
K	*R*	Nom.	–	–	–	–	–	2.25	3.00	3.75	4.50
		Max.	–	–	–	–	–	2.50	3.25	4.00	4.75
		Min.	–	–	–	–	–	2.00	2.75	3.50	4.25

Table 7.22(b) *Dimensions of slotted grub screws (M8 to M24)*

Type	Size d		M8	M10	M12	(M14)	M16	(M18)	M20	(M22)	M24
A, C, E, G, J, and K	*n* (H14)	Nom.	1.2	1.6	2.0	2.0	2.5	3.0	3.0	4.0	4.0
		Max.	1.45	1.85	2.25	2.25	2.75	3.25	3.25	4.30	4.30
		Min.	1.20	1.60	2.00	2.00	2.50	3.00	3.00	4.00	4.00
	t (*js*15)	Nom.	2.5	3.0	4.0	4.0	4.5	5.0	5.0	6.0	6.0
		Max.	2.70	3.20	4.24	4.24	4.74	5.24	5.24	6.24	6.24
		Min.	2.30	2.80	3.76	3.76	4.26	4.76	4.76	5.76	5.76
C	*m*	Nom.	3.0	4.0	5.0	5.0	6.0	6.5	7.0	8.0	8.0
E	d_1 (*h*13)	Nom.	6.0	7.0	9.0	10.0	12.0	13.0	15.0	–	–
		Max.	6.00	7.00	9.00	10.00	12.00	13.00	15.00	–	–
		Min.	5.82	6.78	8.78	9.78	11.73	12.73	14.73	–	–
	P (+IT14)	Nom.	5.0	5.5	7.0	7.0	9.0	9.0	9.0	–	–
		Max.	5.30	5.80	7.36	7.36	9.36	9.36	9.36	–	–
		Min.	5.00	5.50	7.00	7.00	9.00	9.00	9.00	–	–
	r	Nom.	0.4	0.5	0.6	0.8	0.8	0.8	1.0	–	–
G	d_1 (*h*13)	Nom.	3.7	4.4	5.1	–	7.4	–	9.7	–	–
		Max.	3.70	4.40	5.10	–	7.40	–	9.70	–	–
		Min.	3.52	4.22	4.92	–	7.18	–	9.48	–	–
	P (+IT14)	Nom.	5.0	6.0	8.0	–	10.0	–	12.0	–	–
		Max.	5.30	6.30	8.36	–	10.36	–	12.43	–	–
		Min.	5.0	6.0	8.0	–	10.0	–	12.0	–	–
	r	Nom.	0.4	0.5	0.6	–	0.8	–	0.8	–	–
J	d_1 (h14)	Nom.	5.0	6.0	8.0	8.0	10.0	12.0	14.0	16.0	16.0
		Max.	5.00	6.00	8.00	8.00	10.00	12.00	14.00	16.00	16.00
		Min.	4.70	5.70	7.64	7.64	9.64	11.53	13.53	15.53	15.53
K	*R*	Nom.	6.0	7.5	9.0	10.5	12.0	13.5	15.0	16.5	18.0
		Max.	6.25	7.75	9.25	10.75	12.25	13.75	15.25	16.75	18.25
		Min.	5.75	7.25	8.75	10.25	11.75	13.25	14.75	16.25	17.75

Note:
(i) All dimensions are in mm.
(ii) Grade – Precision (P)
(iii) Property Class – 6.6 (for sizes up to and including M10), 4.8 (for sizes M12 and above)
(iv) The sizes given in brackets are of second preference.

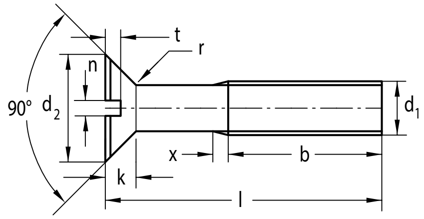
Designations –A slotted grub screw of Type A, nominal size M12, nominal length 50 mm, and property class 4.8 is designated as Grub Screw A M12 × 50 – 4.8

Table 7.22(c) *Length – Diameter combinations for slotted grub screws*

l (js15)	M1	M1.2	M1.6	M2	M2.5	M3	M4	M5	M6	M8	M10	M12	M14	M16	M18	M20	M22	M24
2	X	X	X															
3	X	X	X	X														
4	X	X	X	X	X	X												
5			X	X	X	X	X	X										
6			X	X	X	X	X	X	X									
8				X	X	X	X	X	X	X								
10					X	X	X	X	X	X	X							
12						X	X	X	X	X	X	X						
14							X	X	X	X	X	X	X					
16								X	X	X	X	X	X	X				
20									X	X	X	X	X	X	X	X		
25										X	X	X	X	X	X	X	X	X
30											X	X	X	X	X	X	X	X
35											X	X	X	X	X	X	X	X
40												X	X	X	X	X	X	X
45												X	X	X	X	X	X	X
50														X	X	X	X	X
55															X	X	X	X
60															X	X	X	X
65																X	X	X
70																	X	X

Note: Preferred lengths are indicated by 'X'

Table 7.23(a) *Dimensions for slotted countersunk head screws*



Nominal size d_1		**M1.6**	**M2**	**M2.5**	**M3**	**M4**	**M5**	**M6**	**M8**	**M10**	**M12**	**M16**	**M20**
d_2 ($h13$ if $d_2 < 3$) ($h14$ if $d_2 \geq 3$)	Nom.	3	3.8	4.7	5.6	7.5	9.2	11	14.5	18	22	29	36
	Max.	3.00	3.80	4.70	5.60	7.50	9.20	11.00	14.50	18.00	22.00	29.00	36.00
	Min.	2.75	3.50	4.40	5.30	7.19	8.34	10.57	14.07	17.57	21.48	28.48	35.38
k	Max.	0.96	1.2	1.5	1.65	2.2	2.5	3	4	5	6	8	10
n (c13 if $n \leq 1$) (c14 if $n > 1$)	Nom.	0.4	0.5	0.6	0.8	1	1.2	1.6	2	2.5	3	4	5
	Max.	0.6	0.7	0.8	1	1.2	1.51	1.91	2.31	2.31	3.31	4.37	5.37
	Min.	0.46	0.56	0.66	0.86	1.06	1.26	1.66	2.06	2.56	3.06	4.07	5.07
t	Max.	0.5	0.6	0.73	0.85	1.1	1.35	1.6	2.1	2.6	3	4	5
	Min.	0.32	0.4	0.5	0.6	0.8	1	1.2	1.6	2	2.4	3.2	4
$r\approx$		0.16	0.2	0.25	0.3	0.4	0.5	0.6	0.8	1	1.2	1.6	2
b	+2P	15	16	18	19	22	25	28	34	40	46	58	70

(ISO: 2009)

Note:

(i) All dimensions are in mm.
(ii) Grade – Precision (P)
(iii) Property Class – 4.8 (Property Class 4.6 is also permitted)

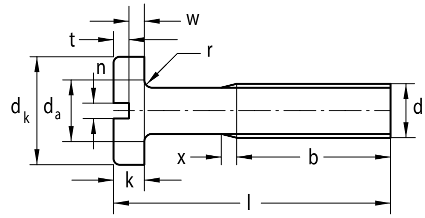
Designations –A slotted countersunk head screw of nominal size M4, nominal length 10 mm, and property class 4.8 is designated as Countersunk Screw M4 × 10 – 4.8

Table 7.23(b) *Preferred length size combinations for slotted countersunk head screws*

l (js15)	M1.6	M2	M2.5	M3	M4	M5	M6	M8	M10	M12	M16	M20
2.5	X											
3	X	X										
4	X	X	X									
5	X	X	X	X								
6	X	X	X	X	X							
8	X	X	X	X	X	X	X					
10	X	X	X	X	X	X	X	X				
12	X	X	X	X	X	X	X	X	X			
14	X	X	X	X	X	X	X	X	X			
16	X	X	X	X	X	X	X	X	X	X		
20		X	X	X	X	X	X	X	X	X		
22			X	X	X	X	X	X	X	X		
25			X	X	X	X	X	X	X	X	X	
30				X	X	X	X	X	X	X	X	X
35					X	X	X	X	X	X	X	X
40					X	X	X	X	X	X	X	X
45						X	X	X	X	X	X	X
50						X	X	X	X	X	X	X
55							X	X	X	X	X	X
60							X	X	X	X	X	X
70								X	X	X	X	X
80								X	X	X	X	X
90									X	X	X	X
100									X	X	X	X
110										X	X	X
120										X	X	X
130											X	X
140											X	X
150											X	X
160											X	X
170												X
180												X
190												X
200												X

Note: Preferred lengths are indicated by 'X'

Table 7.24 *Dimensions for slotted cheese head screws*



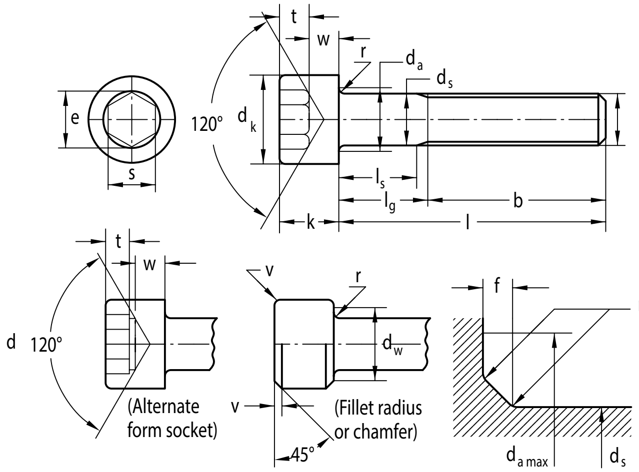
Size d		M1.6	M2	M2.5	M3	M4	M5	M6	M8	M10
Pitch P		0.35	0.4	0.45	0.5	0.7	0.8	1	1.25	1.5
b	Max.	25.7	25.8	25.9	26	39.4	39.6	40	40.5	41
	Min.	25	25	25	25	38	38	38	38	38
d_k	Max.	3	3.8	4.5	5.5	7	8.5	10	13	16
	Min.	2.86	3.62	4.32	5.32	6.78	8.28	9.78	12.73	15.73
d_a	Max.	2	2.6	3.1	3.6	4.7	5.7	6.8	9.2	11.2
k	Max.	1	1.3	1.6	2	2.6	3.3	3.9	5	6
	Min.	0.86	1.16	1.46	1.86	2.46	3.12	3.6	4.7	5.7
n	Nom.	0.4	0.5	0.6	0.8	1.2	1.2	1.6	2	2.5
	Max.	0.6	0.7	0.8	1	1.51	1.51	1.91	2.31	2.81
	Min.	0.46	0.56	0.66	0.86	1.26	1.26	1.66	2.06	2.56
r	Min.	0.1	0.1	0.1	0.1	0.2	0.2	0.25	0.4	0.4
t	Min.	0.4	0.55	0.7	0.8	1.1	1.3	1.6	2	2.4
w	Min.	0.35	0.45	0.6	0.8	1.1	1.3	1.6	2	2.4

(ISO: 1207)

Note:
(i) All dimensions are in mm.
(ii) Grade – A
(iii) Property Class – 4.8
(iv) The minimum tensile strength of screw material should be 300 MPa.

Designations –A slotted cheese head screw of nominal size M4, length 10 mm and property class 4.8 is designated as Cheese Head Screw M4 × 10 – 4.8.

Table 7.25(a) *Dimensions of hexagon socket head cap screws*



Nominal size *d*		M4	M5	M6	M8	M10	M12	M16	M20	M24	M30	M36
Pitch *P*		0.7	0.8	1	1.25	1.5	1.75	2	2.5	3	3.5	4
b		20	22	24	28	32	36	44	52	60	72	84
d_k	Nom.	7	8.5	10	13	16	18	24	30	36	45	54
	Max.	7.22	8.72	10.22	13.27	16.27	18.27	24.33	30.33	36.39	45.39	54.46
	Min.	6.78	8.28	9.78	12.73	15.73	17.73	23.67	29.67	35.61	44.61	53.54
d_a	Max.	4.7	5.7	6.8	9.2	11.2	14.2	18.2	22.4	26.4	33.4	39.4
d_s	Max.	4	5	6	8	10	12	16	20	24	30	36
*h*13	Min.	3.82	4.82	5.82	7.78	9.78	11.73	15.73	19.67	23.67	29.67	35.61
e	Min.	3.44	4.58	5.72	6.86	9.15	11.43	16.00	19.44	21.73	25.15	30.85
f	Max.	0.6	0.6	0.68	1.02	1.02	1.87	1.87	2.04	2.04	2.89	2.89
k	Max.	4	5	6	8	10	12	16	20	24	30	36
*h*13,*h*14	Min.	3.82	4.82	5.70	7.64	9.64	11.57	15.57	19.48	23.48	29.48	35.38
r	Min.	0.2	0.2	0.25	0.4	0.4	0.6	0.6	0.8	0.8	1	1
s	Nom.	3	4	5	6	8	10	14	17	19	22	27
*h*13	Min.	3.02	4.02	5.02	6.02	8.025	10.025	14.032	17.05	19.065	22.065	27.065
	Max.	3.071	4.084	5.084	6.14	8.175	10.127	14.159	17.216	19.275	22.275	27.275
t	Min.	2	2.5	3	4	5	6	8	10	12	15.5	19
v	Max.	0.4	0.5	0.6	0.8	1	1.2	1.6	2	2.4	3	3.6
d_w	Min.	6.53	8.03	9.38	12.33	15.53	17.23	23.17	28.87	34.81	43.61	52.54
w	Min.	1.4	1.9	2.3	3.3	4	4.8	6.8	8.6	10.4	13.1	15.3

(ISO: 4762)

Note:

(i) All dimensions are in mm. (ii) Grade – A (iii) Property Class – 8.8 and 12.9

$b = (2d + 12)$ $t = 0.5\text{k}$ $e_{\text{Min}} = (1.14\, s_{\text{Min}})$ $f_{\text{Max}} = 1.7\, r_{\text{Max}}$ $r_{\text{Max}} = \left(\dfrac{d_a - d_s}{2}\right)$

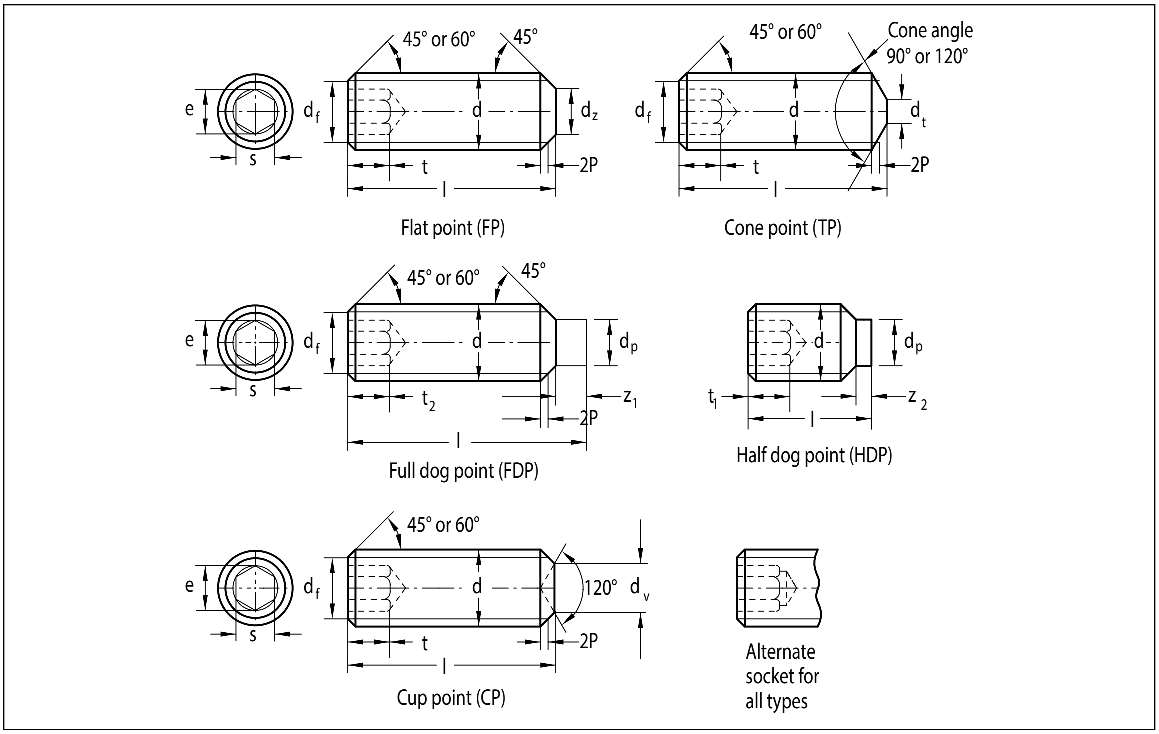
Designations –A hexagon socket head cap screw of nominal size M10, nominal length 60 mm, and property class 12.9 is designated as Hexagon Socket Head Cap Screw M10 × 60 – 12.9.

Table 7.25(b) *Preferred length-size combinations for hexagon socket head cap screws*

Size d			M4	M5	M6	M8	M10	M12	M16	M20	M24	M30	M36
l													
Nom.	Min.	Max.											
6	5.76	6.24	X										
8	7.71	8.29	X	X									
10	9.71	10.29	X	X	X								
12	11.65	12.35	X	X	X	X							
16	15.65	16.35	X	X	X	X	X						
20	19.58	20.42	X	X	X	X	X	X					
25	24.58	25.42	X	X	X	X	X	X	X				
30	29.58	30.42	X	X	X	X	X	X	X	X			
35	34.5	35.5	X	X	X	X	X	X	X	X			
40	39.5	40.5	X	X	X	X	X	X	X	X	X		
45	44.5	45.5		X	X	X	X	X	X	X	X	X	
50	49.5	50.5		X	X	X	X	X	X	X	X	X	
55	54.4	55.6			X	X	X	X	X	X	X	X	X
60	59.4	60.6			X	X	X	X	X	X	X	X	X
65	64.4	65.6				X	X	X	X	X	X	X	X
70	69.4	70.6				X	X	X	X	X	X	X	X
80	79.4	80.6				X	X	X	X	X	X	X	X
90	89.3	90.7					X	X	X	X	X	X	X
100	99.3	100.7					X	X	X	X	X	X	X
110	109.3	110.7						X	X	X	X	X	X
120	119.3	120.7						X	X	X	X	X	X
130	129.2	130.8							X	X	X	X	X
140	139.2	140.8							X	X	X	X	X
150	149.2	150.8							X	X	X	X	X
160	159.2	160.8							X	X	X	X	X
180	179.2	180.8								X	X	X	X
200	199.075	200.925								X	X	X	X

Note: (i) $l_{g\,Max} = (l_{nom} - b)$ (ii) $l_{s\,Min} = (l_{g\,Max} - 0.5P)$.
(ii) Preferred lengths are indicated by 'X'.

Table 7.26(a) *Dimensions of hexagon socket set screws*



Type	Size d		M3	M4	M5	M6	M8	M10	M12	M16	M20	M24
All	e	Min.	1.73	2.30	2.87	3.44	4.58	5.72	6.86	9.15	11.43	13.72
	s	Nom.	1.5	2.0	2.5	3.0	4.0	5.0	6.0	8.0	10.0	12.0
		Min.	1.52	2.02	2.52	3.02	4.02	5.02	6.02	8.025	10.025	12.032
		Max.	1.545	2.045	2.56	3.08	4.095	5.095	6.095	8.115	10.115	12.042
	$t\ t_1$	Min.	1.2	1.5	2.0	2.0	3.0	4.0	4.8	6.4	8.0	10.0
	t_2	Min.	2.0	2.5	3.0	3.5	5.0	6.0	8.0	10.0	12.0	15.0
FP	d_z	Max.	2.0	2.5	3.5	4.0	5.5	7.0	8.5	12.0	15.0	18
		Min.	1.75	2.25	3.2	3.7	5.2	6.64	8.14	11.57	14.57	17.57
TP	d_t	Max.	0	0	0	1.5	2.0	2.5	3.0	4.0	5.0	6.0
FDP and HDP	d_p	Max.	2.0	2.5	3.5	4.0	5.5	7.0	8.5	12.0	15.0	18.0
		Min.	1.75	2.25	3.2	3.7	5.2	6.64	8.14	11.57	14.5	17.57
	z_2	Min.	0.75	1.0	1.25	1.5	2.0	2.5	3.0	4.0	5.0	6.0
		Max.	1.0	1.25	1.5	1.75	2.25	2.75	3.25	4.3	5.3	6.3
	z_1	Min.	1.5	2.0	2.5	3.0	4.0	5.0	6.0	8.0	10.0	12.0
		Max.	1.75	2.25	2.75	3.25	4.30	5.30	6.30	8.36	10.36	12.43

(*Contd.*)

(*Contd.*)

CP	d_v	Max.	1.4	2.0	2.5	3.0	5.0	6.0	8.0	10.0	14.0	16.0
		Min.	1.15	1.75	2.25	2.75	4.7	5.7	7.64	8.64	13.57	15.57

(ISO: 4026, 4027, 4028 and 4029)

Note:
(i) All dimensions are in mm.
(ii) Grade – A
(iii) Property Class – 45H
(iv) Other dimensions of half dog point (not shown in figure) are same as for full dog point.
(v) $e_{Min} = 1.14\ s_{Min}$

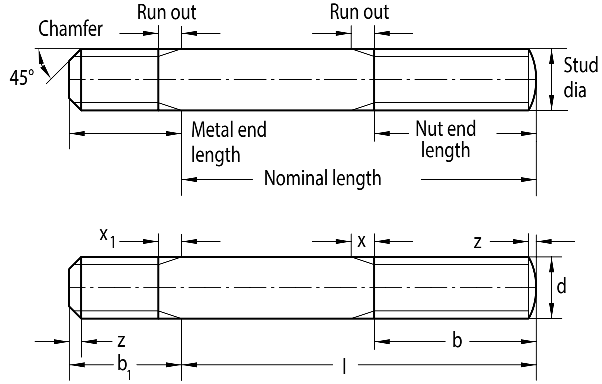
Designations –A hexagon socket set screw of nominal size M12, nominal length 50 mm and with a flat point (FP) is designated as Hexagon Socket Set Screw M12 × 50 – FP.

Table 7.26(b) *Preferred length–size combinations for hexagon socket set screws*

l (js15)	Size d									
	M3	M4	M5	M6	M8	M10	M12	M16	M20	M24
2	X									
2.5	X	X								
3	X	X	X							
4	X	X	X	X						
5	X	X	X	X	X					
6	X	X	X	X	X	X				
8	X	X	X	X	X	X	X			
10	X	X	X	X	X	X	X	X		
12	X	X	X	X	X	X	X	X	X	
16	X	X	X	X	X	X	X	X	X	X
20		X	X	X	X	X	X	X	X	X
25			X	X	X	X	X	X	X	X
30				X	X	X	X	X	X	X
35					X	X	X	X	X	X
40					X	X	X	X	X	X
45						X	X	X	X	X
50						X	X	X	X	X
55							X	X	X	X
60							X	X	X	X

Note: Preferred lengths are indicated by 'X'.

Table 7.27(a) *Dimensions of studs (Size range M3 to M14)*



Nominal size d (h13)			M3	(M3.5)	M4	(M4.5)	M5	M6	M8 M8 × 1	M10 M10 × 1.25	M12 M12 × 1.25	(M14) (M14 × 1.5)
$b + 2P - 0.0$	For (1 ≤ 125)		12	13	14	15	16	18	22	26	30	34
	125 < 1 < 200		–	–	–	–	–	–	–	32	36	40
b_1 ($js16$)	Type A	Nom.	3	3.5	4	4.5	5	6	8	10	12	14
		Max.	3.3	3.88	4.38	4.88	5.38	6.38	8.45	10.45	12.55	14.55
		Min.	2.7	3.12	3.62	4.12	4.62	5.62	7.55	9.55	11.45	13.45
	Type B	Nom.	4.5	5	6	7	7.5	9	12	15	18	21
		Max.	4.88	5.38	6.38	7.45	7.95	9.45	12.55	15.55	18.55	21.65
		Min.	4.12	4.62	5.62	6.55	7.05	8.55	11.45	14.45	17.45	20.35
	Type C	Nom.	–	–	8	–	–	–	16	–	24	–
		Max.	–	–	8.45	–	–	–	16.55	–	24.65	–
		Min.	–	–	7.55	–	–	–	15.45	–	23.35	–

Table 7.27(b) *Dimensions of studs (Size range M16 to M39)*

<table>
<tr><th colspan="3">Nominal size d
(h13)</th><th>M16
M16 × 1.5</th><th>(M18)
(M18 × 1.5)</th><th>M20
M20 × 1.5</th><th>(M22)
(M22 × 1.5)</th><th>M24
M24 × 2</th><th>(M27)
(M27 × 2)</th><th>M30
M30 × 2</th><th>(M33)
(M33 × 2)</th><th>M36
M36 × 3</th><th>(M39)
(M39 × 3)</th></tr>
<tr><td rowspan="3">$b + 2P - 0$</td><td colspan="2">For (l ≤ 125)</td><td>38</td><td>42</td><td>46</td><td>50</td><td>54</td><td>60</td><td>66</td><td>72</td><td>78</td><td>84</td></tr>
<tr><td colspan="2">125 < l ≤ 200</td><td>44</td><td>48</td><td>52</td><td>56</td><td>60</td><td>66</td><td>72</td><td>78</td><td>84</td><td>90</td></tr>
<tr><td colspan="2">For l > 200</td><td>57</td><td>61</td><td>65</td><td>69</td><td>73</td><td>79</td><td>85</td><td>91</td><td>97</td><td>103</td></tr>
<tr><td rowspan="9">b_1
($js16$)</td><td rowspan="3">Type A</td><td>Nom.</td><td>16</td><td>18</td><td>20</td><td>22</td><td>24</td><td>27</td><td>30</td><td>33</td><td>36</td><td>39</td></tr>
<tr><td>Max.</td><td>16.55</td><td>18.55</td><td>20.65</td><td>22.65</td><td>24.65</td><td>27.65</td><td>30.65</td><td>33.8</td><td>36.8</td><td>39.8</td></tr>
<tr><td>Min.</td><td>15.45</td><td>17.45</td><td>19.35</td><td>21.35</td><td>23.35</td><td>26.35</td><td>29.35</td><td>32.2</td><td>35.2</td><td>38.2</td></tr>
<tr><td rowspan="3">Type B</td><td>Nom.</td><td>24</td><td>27</td><td>30</td><td>33</td><td>36</td><td>40.5</td><td>45</td><td>49.5</td><td>54</td><td>57</td></tr>
<tr><td>Max.</td><td>24.65</td><td>27.65</td><td>30.65</td><td>33.8</td><td>36.8</td><td>41.3</td><td>45.8</td><td>50.3</td><td>54.95</td><td>57.95</td></tr>
<tr><td>Min.</td><td>23.35</td><td>26.35</td><td>29.35</td><td>32.2</td><td>35.2</td><td>39.7</td><td>44.2</td><td>48.7</td><td>53.05</td><td>56.05</td></tr>
<tr><td rowspan="3">Type C</td><td>Nom.</td><td>32</td><td>–</td><td>40</td><td>–</td><td>–</td><td>–</td><td>–</td><td>–</td><td>–</td><td>–</td></tr>
<tr><td>Max.</td><td>32.8</td><td>–</td><td>40.8</td><td>–</td><td>–</td><td>–</td><td>–</td><td>–</td><td>–</td><td>–</td></tr>
<tr><td>Min.</td><td>31.2</td><td>–</td><td>39.2</td><td>–</td><td>–</td><td>–</td><td>–</td><td>–</td><td>–</td><td>–</td></tr>
</table>

Types:

(i) Type A – recommended for use in steel (Property classes – 4.6, 6.6, 8.8 and 10.9)
(ii) Type B – recommended for use in cast iron (Property classes – 4.6, 6.6, 8.8 and 10.9)
(iii) Type C – recommended for use in aluminium alloys (Property classes – 4.6, 6.6, 8.8)

Note:

(i) Metal end of stud –The end of the stud, which is screwed into the component.
(ii) Nut end – The end of the stud, which is not screwed into the component.
(iii) All dimensions are in mm.
(iv) Grade – Precision (P) or Black (B).
(v) The sizes given in brackets are of second preference.

Designations –A stud of Type A, of nominal size M5, nominal length 25 mm, precision grade and property class 8.8 is designated as Stud A M5 × 25 – P – 8.8.

Table 7.27(c) *Preferred length-size combinations for Type A and Type B studs*

l	M3	M4	M5	M6	M8 M8 × 1	M10 M10 × 1.25	M12 M12 × 1.25	M16 M16 × 1.5	M20 M20 × 1.5	M24 M24 × 2	M30 M30 × 2	M36 M36 × 3
14	X											
16	X	X										
20	X	X	X									
25	X	X	X	X								
30		X	X	X	X	X						
35		X	X	X	X	X	X					
40		X	X	X	X	X	X					
45		X	X	X	X	X	X	X				
50		X	X	X	X	X	X	X				
55		X	X	X	X	X	X	X	X			
60		X	X	X	X	X	X	X	X			
65		X	X	X	X	X	X	X	X	X		
70		X	X	X	X	X	X	X	X	X		
75			X	X	X	X	X	X	X	X	X	
80			X	X	X	X	X	X	X	X	X	
85				X	X	X	X	X	X	X	X	
90					X	X	X	X	X	X	X	X
100					X	X	X	X	X	X	X	X
110					X	X	X	X	X	X	X	X
120						X	X	X	X	X	X	X
130						X	X	X	X	X	X	X
140						X	X	X	X	X	X	X
150						X	X	X	X	X	X	X
160							X	X	X	X	X	X
170							X	X	X	X	X	X
180								X	X	X	X	X
190								X	X	X	X	X
200								X	X	X	X	X
225								X	X	X	X	X
250								X	X	X	X	X
275								X	X	X	X	X
300								X	X	X	X	X

Note: (i) Preferred lengths are indicated by 'X' (ii) Tolerance on length $l = js15$ for precision grade studs and js17 for black grade studs.

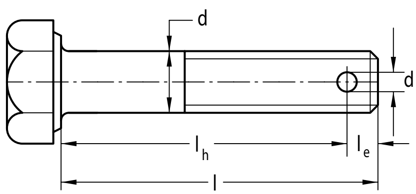
Table 7.27(d) *Preferred length-size combinations for Type C studs*

l	M4	M8 M8 × 1	M12 M12 × 1.25	M16 M16 × 1.5	M20 M20 × 1.5
25	X				
30	X				
40	X	X			
60	X	X			
80		X	X		
100		X	X	X	
120			X	X	X
140			X	X	X
160				X	X
200					X

Note: Preferred lengths are indicated by 'X'.

7.6 DIMENSIONS OF SPLIT PINS

Table 7.28 *Dimensions for location and diameter of split pin holes*

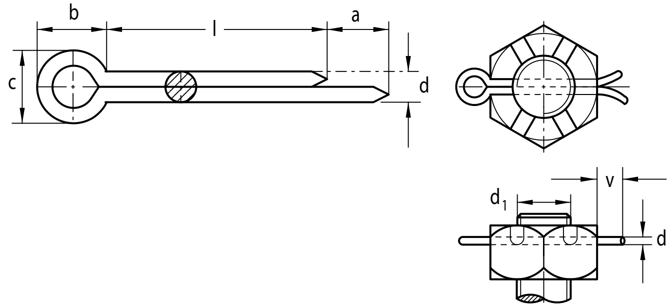


Nominal size d		M4	M5	M6	M8	M10	M12	M14	M16	M18	M20
d_1 (H14)		1	1.2	1.6	2	2.5	3.2	3.2	4	4	4
l_e	Min.	2.3	2.6	3.3	3.9	4.9	5.9	6.5	7.7	7.7	7.7

Nominal size d		M22	M24	M26	M30	M33	M36
d_1 (H14)		5	5	5	6.3	6.3	6.3
l_e	Min.	8.7	10	10	11.2	11.2	12.5

Note: All dimensions are in mm.

Table 7.29(a) *Dimensions for split pins*



Dimension		Nominal Size										
		0.6	**0.8**	**1**	**1.2**	**1.6**	**2**	**2.5**	**3.2**	**4**	**5**	**6.3**
d	Max.	0.5	0.7	0.9	1	1.4	1.8	2.3	2.9	3.7	4.6	5.9
	Min.	0.4	0.6	0.8	0.9	1.3	1.7	2.1	2.7	3.5	4.4	5.7
a	Max.	1.6	1.6	1.6	2.5	2.5	2.5	2.5	3.2	4	4	4
	Min.	0.8	0.8	0.8	1.2	1.2	1.2	1.2	1.6	2	2	2
b	≈	2	2.4	3	3	3.2	4	5	6.4	8	10	12.6
c	Max.	1	1.4	1.8	2	2.8	3.6	4.6	5.8	7.4	9.2	11.8
	Min.	0.9	1.2	1.6	1.7	2.4	3.2	4	5.1	6.5	8	10.3
Bolt Dia d_1	Over	–	2.5	3.5	4.5	5.5	7	9	11	14	20	27
	Up to	2.5	3.5	4.5	5.5	7	9	11	14	20	27	39

(ISO: 1234)

Note:

(i) Nominal size (*d*) – The nominal size is the diameter of the hole for receiving the split pin.
(ii) Nominal length (*l*) – The nominal length is the distance from underside of eye to the extreme end of the short leg.
(iii) The corresponding size of bolt is d_1.
(iv) The split pin is made from half round mild steel wire.

Designation – A split pin of nominal size 5 mm, nominal length 50 mm is designated as Split Pin 5 × 50.

Table 7.29(b) *Preferred length-size combinations for split pins*

l Nom.	Nominal Size										
	0.6	**0.8**	**1**	**1.2**	**1.6**	**2**	**2.5**	**3.2**	**4**	**5**	**6.3**
4	X										
5	X	X									
6	X	X	X								
8	X	X	X	X	X						
10	X	X	X	X	X	X					
12	X	X	X	X	X	X	X				
14		X	X	X	X	X	X	X			
16		X	X	X	X	X	X	X			
18			X	X	X	X	X	X	X		
20			X	X	X	X	X	X	X		
22				X	X	X	X	X	X	X	
25				X	X	X	X	X	X	X	
28					X	X	X	X	X	X	
32					X	X	X	X	X	X	X
36						X	X	X	X	X	X
40						X	X	X	X	X	X
45							X	X	X	X	X
50							X	X	X	X	X
56								X	X	X	X
63								X	X	X	X
71									X	X	X
80									X	X	X
90										X	X
100										X	X
112											X
125											X

(ISO: 1234)

Note: Preferred lengths are indicated by 'X'.

Table 7.29(c) *Dimensions of wires for split pins*

Nominal Size	Width		Thickness	
	Max.	Min.	Max.	Min.
0.6	0.5	0.4	0.25	0.20
0.8	0.7	0.6	0.35	0.30
1.0	0.9	0.8	0.45	0.40
1.2	1.0	0.9	0.50	0.45
1.6	1.4	1.3	0.70	0.65
2.0	1.8	1.7	0.90	0.85
2.5	2.3	2.1	1.15	1.05
3.2	2.9	2.7	1.45	1.35
4	3.7	3.5	1.85	1.75
5	4.6	4.4	2.30	2.20
6.3	5.9	5.7	2.95	2.85

Note: All dimensions are in mm.

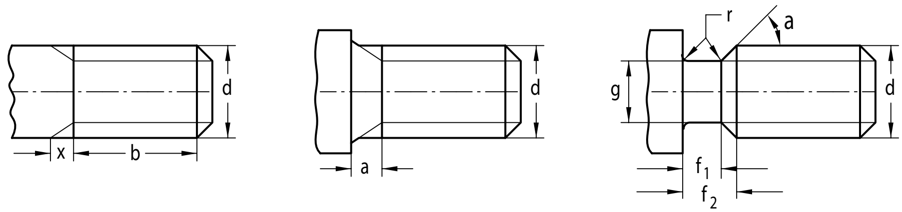
Table 7.29(d) *Tensile properties of wires for split pins*

Nominal size (mm)	Gauge length (mm)	Tensile strength (MPa or N/mm^2)	Elongation (%)
Up to 1.6	125	450	10
Over 1.6 and up to 4	250	450	10
Over 4	8 times nominal size	450	20

Note: The chemical composition of wire is as follows:
(i) Carbon (Max) – 0.1% (ii) Manganese – 0.3 to 0.5% (iii) Sulphur (Max) – 0.04% (iv) Phosphorus (Max) – 0.04%

7.7 THREAD RUNOUTS, UNDERCUTS AND CLEARANCE HOLES

Table 7.30 *Dimensions for thread runouts and undercuts for external threads*



Thread runout

Thread undercut

Pitch P	Thread runout $x_{(Max)}$		Runout distance $a_{(Max)}$			Thread undercut					
						g (h13) (Note 6)	$f_{1(Min.)}$		$f_{2(Max.)}$		*r* ≈
	Normal (Note 1)	Short (Note 2)	Normal (Note 3)	Short (Note 4)	Long (Note 5)		Normal	Short (Note 7)	Normal	Short (Note 7)	
0.2	0.5	0.25	0.6	0.4	0.8	d-0.3	0.45	0.25	0.7	0.5	0.1
0.25	0.6	0.3	0.75	0.5	1	d-0.4	0.55	0.25	0.9	0.6	0.12
0.3	0.75	0.4	0.9	0.6	1.2	d-0.5	0.6	0.3	1.05	0.75	0.15
0.35	0.9	0.45	1.05	0.7	1.4	d-0.6	0.7	0.4	1.2	0.9	0.17
0.4	1	0.5	1.2	0.8	1.6	d-0.7	0.8	0.5	1.4	1	0.2
0.45	1.1	0.6	1.35	0.9	1.8	d-0.7	1	0.5	1.6	1.1	0.22
0.5	1.25	0.7	1.5	1	2	d-0.8	1.1	0.5	1.75	1.25	0.25
0.6	1.5	0.75	1.8	1.2	2.4	d-1	1.2	0.6	2.1	1.5	0.3
0.7	1.75	0.9	2.1	1.4	2.8	d-1.1	1.5	0.8	2.45	1.75	0.35
0.75	1.9	1	2.25	1.5	3	d-1.2	1.6	0.9	2.6	1.9	0.4
0.8	2	1	2.4	1.6	3.2	d-1.3	1.7	0.9	2.8	2	0.4
1	2.5	1.25	3	2	4	d-1.6	2.1	1.1	3.5	2.5	0.5
1.25	3.2	1.6	4	2.5	5	d-2	2.7	1.5	4.4	3.2	0.6
1.5	3.8	1.9	4.5	3	6	d-2.3	3.2	1.8	5.2	3.8	0.75
1.75	4.3	2.2	5.3	3.5	7	d-2.6	3.9	2.1	6.1	4.3	0.9
2	5	2.5	6	4	8	d-3	4.5	2.5	7	5	1
2.5	6.3	3.2	7.5	5	10	d-3.6	5.6	3.2	8.7	6.3	1.25
3	7.5	3.8	9	6	12	d-4.4	6.7	3.7	10.5	7.5	1.5
3.5	9	4.5	10.5	7	14	d-5	7.7	4.7	12	9	1.75
4	10	5	12	8	16	d-5.7	9	5	14	10	2
4.5	11	5.5	13.5	9	18	d-6.4	10.5	5.5	16	11	2.25
5	12.5	6.3	15	10	20	d-7	11.5	6.5	17.5	12.5	2.5
5.5	14	7	16.5	11	22	d-7.7	12.5	7.5	19	14	2.75

(*Contd.*)

(*Contd.*)

6	15	7.5	18	12	24	d-8.3	14	8	21	15	3
Indicated dimensions approx. correspond to	2.5P	1.25P	3P	2P	4P	–	–	–	3.5P	2.5P	2.5P

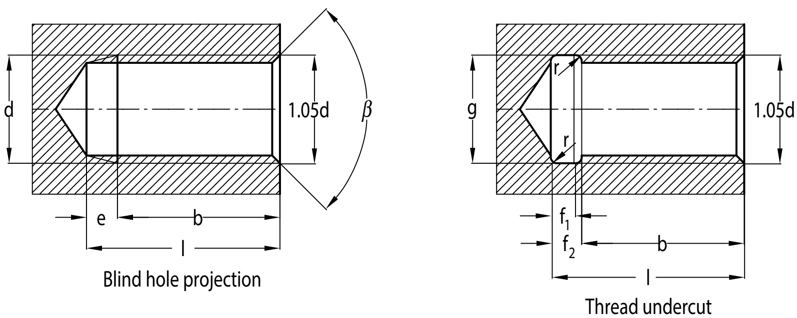
(ISO: 3508)

Note:

(i) Runout x normal for all types of screws of product grades A, B and C.
(ii) Runout x short only in cases where a short runout is required for technical reasons.
(iii) Distance a normal for all types of screws of product grade A.
(iv) Distance a short for special cases in which for technical reasons a short distance is necessary.
(v) Distance a long for all types of screws of product grades B and C.
(vi) Tolerance zone h12 for g for threads up to 3 mm nominal diameter.
(vii) Undercut short is for special cases.
(viii) The transition angle α is 30º to 60º.
(ix) All dimensions are in mm.

Notations – a = Distance from last full thread to contact face.
b = Length of full thread
x = Thread run out

Table 7.31 *Dimensions for blind hole projection and undercuts for internal threads*



Pitch P	Blind hole projection e Normal	Short	Long	Thread undercut g (h13)	$f_{1(Min.)}$ Normal	$f_{1(Min.)}$ Short (See Note 1)	$f_{2(Max.)}$ Normal	$f_{2(Max.)}$ Short (See Note1)	r ≈
0.2	1.3	0.8	2	$d + 0.1$	0.8	0.5	1.2	0.9	0.1
0.25	1.5	1	2.4	$d + 0.1$	1	0.6	1.4	1	0.12
0.3	1.8	1.2	2.9	$d + 0.1$	1.2	0.75	1.6	1.25	0.15
0.35	2.1	1.3	3.3	$d + 0.2$	1.4	0.9	1.9	1.4	0.17

(*Contd.*)

(*Contd.*)

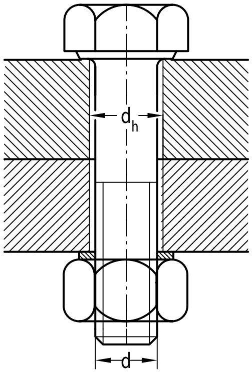
0.4	2.3	1.5	3.7	$d + 0.2$	1.6	1	2.2	1.6	0.2
0.45	2.6	1.6	4.1	$d + 0.2$	1.8	1.1	2.4	1.7	0.22
0.5		1.8	4.5	$d + 0.3$	2	1.25	2.7	2	0.25
0.6		2.1	5.4	$d + 0.3$	2.4	1.5	3.3	2.4	0.3
0.7		2.4	6.1	$d + 0.3$	2.8	1.75	3.8	2.75	0.35
0.75	4	2.5	6.4	$d + 0.3$	3	1.9	4	2.9	0.4
0.8	4.2	2.7	6.8	$d + 0.3$	3.2	2	4.2	3	0.4
1	5.1	3.2	8.2	$d + 0.5$	4	2.5	5.2	3.7	0.5
1.25	6.2	3.9	10	$d + 0.5$	5	3.2	6.7	4.9	0.6
1.5	7.3	4.6	11.6	$d + 0.5$	6	3.8	7.8	5.6	0.75
1.75	8.3	5.2	13.3	$d + 0.5$	7	4.3	9.1	6.4	0.9
2	9.3	5.8	14.8	$d + 0.5$	8	5	10.3	7.3	1
2.5	11.2	7	17.9	$d + 0.5$	10	6.3	13	9.3	1.25
3	13.1	8.2	21	$d + 0.5$	12	7.5	15.2	10.7	1.5
3.5	15.2	9.5	24	$d + 0.5$	14	9	17.7	12.7	1.75
4	16.8	10.5	26.9	$d + 0.5$	16	10	20	14	2
4.5	18.4	11.5	29.4	$d + 0.5$	18	11	23	16	2.25
5	20.8	13	33.3	$d + 0.5$	20	12.5	26	18.5	2.5
5.5	22.4	14	35.8	$d + 0.5$	22	14	28	20	2.75
6	24	15	38.4	$d + 0.5$	24	15	30	21	3
Indicated dimensions approx. correspond to	6.3P to 4P	4P to 2.5P	10P to 6.3P	–	4P	2.5P	–	–	0.5P

(ISO: 3508)

Note:

(i) Undercut short is for special cases.
(ii) The countersunk angle β is normally 120º and in special cases 90º.
(iii) All dimensions are in mm.

Table 7.32 *Dimensions for clearance holes for bolts and screws*



Thread diameter d	Clearance hole (d_h)			Thread diameter d	Clearance hole (d_h)		
	Series				Series		
	Fine	Medium	Coarse		Fine	Medium	Coarse
1	1.1	1.2	1.3	42	43	45	48
1.2	1.3	1.4	1.5	45	46	48	52
1.4	1.5	1.6	1.8	48	50	52	56
1.6	1.7	1.8	2	52	54	56	62
1.8	2	2.1	2.2	56	58	62	66
2	2.2	2.4	2.6	60	62	66	70
2.5	2.7	2.9	3.1	64	66	70	74
3	3.2	3.4	3.6	68	70	74	78
3.5	3.7	3.9	4.2	72	74	78	82
4	4.3	4.5	4.8	76	78	82	86
4.5	4.8	5	5.3	80	82	86	91
5	5.3	5.5	5.8	85	87	91	96
6	6.4	6.6	7	90	93	96	101
7	7.4	7.6	8	95	98	101	107
8	8.4	9	10	100	104	107	112
10	10.5	11	12	105	109	112	117
12	13	13.5	14.5	110	114	117	122
14	15	15.5	16.5	115	119	122	127
16	17	17.5	18.5	120	124	127	132
18	19	20	21	125	129	132	137
20	21	22	24	130	134	137	144

(*Contd.*)

(*Contd.*)

22	23	24	26	140	144	147	155
24	25	26	28	150	155	158	165
27	28	30	32				
30	31	33	35				
33	34	36	38				
36	37	39	42				
39	40	42	45				

(ISO: 273)

Note:
(i) The tolerance fields are as follows: (a) Fine series – H12 (b) Medium series – H13 (c) Coarse series – H14
(ii) All dimensions are in mm.

7.8 MECHANICAL PROPERTY CLASSES

Table 7.33 *Mechanical property classes for steel bolts, screws and studs*

Property class	Tensile strength (MPa or N/mm^2)		Yield stress (MPa or N/mm^2)		Proof stress ($R_{p0.2}$) (MPa or N/mm^2)		Material
	Nom.	Min.	Nom.	Min.	Nom.	Min.	
4.6	400	400	240	240	–	–	Low or medium carbon steel
4.8	400	420	320	340	–	–	
5.6	500	500	300	300	–	–	
5.8	500	520	400	420	–	–	
6.8	600	600	480	480	–	–	
8.8	800	800	–	–	640	640	Medium carbon steel-quenched and tempered
9.8	900	900	–	–	720	720	
10.9	1000	1040	–	–	900	940	Low or Medium carbon steel with boron, or Mn or Cr - quenched and tempered
12.9	1200	1220	–	–	1080	1100	Alloy steel

(ISO: 898-1)

Designation system – The designation of property class consists of two figures: (a) the first figure indicates 1/100 of nominal tensile strength in MPa; and (b) the second figure indicates 1/10 of the ratio, expressed as a percentage, between nominal yield stress and nominal tensile strength. The multiplication of these two figures will give 1/10 of nominal yield stress in MPa.

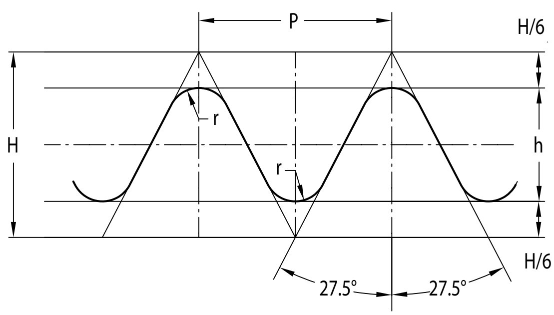
Table 7.34 *Mechanical property classes for steel nuts*

Nominal size	Up to M4	M4 to M7	M7 to M10	M10 to M16	M19 to M39
Property class	Proof stress (MPa or N/mm^2)				
04	380				
05	500				
4	510				
5	520	580	590	610	630
6	600	670	680	700	720
8	800	810	830	840	920
9	900	915	940	950	920
10	1040	1040	1040	1050	1060
12	1150	1150	1160	1190	1200

Note: The nuts are made of carbon steel or low alloy steel.

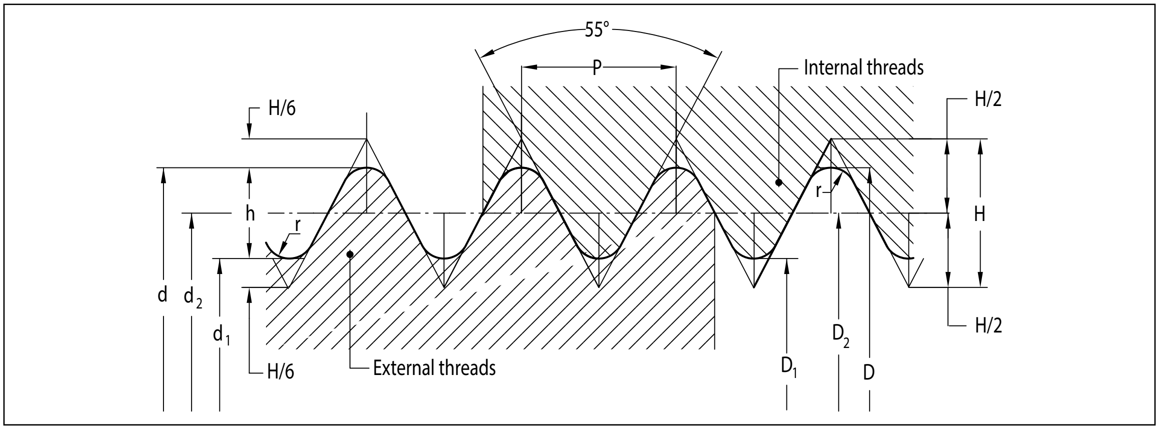
7.9 PIPE THREADS

Table 7.35 *Basic profile for pipe threads*



Proportions	Notations
$H = 0.960\ 491$ P	P = Pitch of threads
$h = 0.640\ 327$ P	H = Height of fundamental triangle
$r = 0.137\ 329$ P	h = Height of basic profile
	r = Radius of root

Table 7.36 *Dimensions for pipe threads for fastening purposes*



Size	Pitch P	Major diameter d = D	Pitch diameter $d_2 = D_2$	Minor diameter $d_1 = D_1$	Depth of thread h	Radius r ≈
$\frac{1}{16}$	0.907	7.723	7.142	6.561	0.581	0.125
$\frac{1}{8}$	0.907	9.728	9.147	8.566	0.581	0.125
$\frac{1}{4}$	1.337	13.157	12.301	11.445	0.856	0.184
$\frac{3}{8}$	1.337	16.662	15.806	14.950	0.856	0.184
$\frac{1}{2}$	1.814	20.955	19.793	18.631	1.162	0.249
$\frac{5}{8}$	1.814	22.911	21.749	20.587	1.162	0.249
$\frac{3}{4}$	1.814	26.441	25.279	24.117	1.162	0.249
$\frac{7}{8}$	1.814	30.201	29.039	27.877	1.162	0.249
1	2.309	33.249	31.770	30.291	1.479	0.317
$1\frac{1}{8}$	2.309	37.897	36.418	34.939	1.479	0.317

(*Contd.*)

(*Contd.*)

$1\frac{1}{4}$	2.309	41.910	40.431	38.952	1.479	0.317
$1\frac{1}{2}$	2.309	47.803	46.324	44.845	1.479	0.317
$1\frac{3}{4}$	2.309	53.746	52.267	50.788	1.479	0.317
2	2.309	59.614	58.135	56.656	1.479	0.317
$2\frac{1}{4}$	2.309	65.710	64.231	62.752	1.479	0.317
$2\frac{1}{2}$	2.309	75.184	73.705	72.226	1.479	0.317
$2\frac{3}{4}$	2.309	31.534	80.055	78.576	1.479	0.317
3	2.309	87.884	86.405	84.926	1.479	0.317
$3\frac{1}{2}$	2.309	100.330	98.851	97.372	1.479	0.317
4	2.309	113.030	111.551	110.072	1.479	0.317
$4\frac{1}{2}$	2.309	125.730	124.251	122.772	1.479	0.317
5	2.309	138.430	136.951	135.472	1.479	0.317
$5\frac{1}{2}$	2.309	151.130	149.651	148.172	1.479	0.317
6	2.309	163.830	162.351	160.872	1.479	0.317

(ISO: 228)

Tolerance Classes – There are two classes of tolerances on pitch diameter of external threads – Class A and Class B.

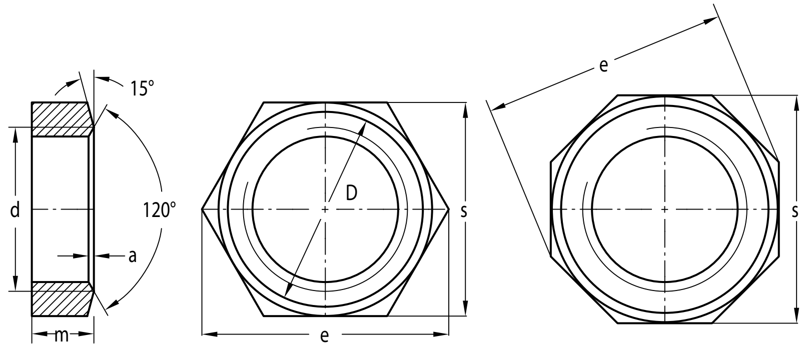
Class A – In this case, negative tolerance on pitch diameter of external thread is numerically equal to positive tolerance on pitch diameter of internal thread. Class A applies to screw threads requiring a snug fit where accuracy of thread form is essential.

Class B – The value of this tolerance is twice that of Class A, the tolerance being also negative. Class B applies to threads of ordinary commercial quality.

Designations – (i) An external pipe thread of size 2 with Class B tolerance is designated as G2 B

(ii) An internal pipe thread of size 2 is designated as G2

Table 7.37 *Dimensions for pipe nuts*



Size d	D Nom	e (Min) For Hexagonal	e (Min) For Octagonal	a Nom	m (h15) Nom.	m (h15) Max.	m (h15) Min.	s (≤19, *h*14) and (>19, h15) Nom.	s Max.	s Min.
$G\frac{1}{8}$	18	20.88	–	1	6	6	5.52	19	19	18.48
$G\frac{1}{4}$	20	23.91	–	1	6	6	5.52	22	22	21.16
$G\frac{3}{8}$	25	29.56	–	1	6	6	5.52	27	27	26.16
$G\frac{1}{2}$	30	35.03	–	1.5	8	8	7.42	32	32	31
$\left(G\frac{5}{8}\right)$	30	35.03	–	1.5	8	8	7.42	32	32	31
$G\frac{3}{4}$	33	39.55	–	1.5	9	9	8.42	36	36	35
$\left(G\frac{7}{8}\right)$	38	45.20	–	1.5	9	9	8.42	41	41	40
G1	43	50.85	–	1.5	10	10	9.42	46	46	45
$\left(G1\frac{1}{8}\right)$	47	55.37	–	1.5	10	10	9.42	50	50	48.80

(*Contd.*)

(*Contd.*)

G1 $\frac{1}{4}$	52	60.79	–	2	11	11	10.30	55	55	53.80
G1 $\frac{1}{2}$	56	66.44	–	2	12	12	11.30	60	60	58.80
$\left(G1\frac{3}{4}\right)$	65	77.74	–	2	13	13	12.30	70	70	68.80
G2	70	83.39	–	2	13	13	12.30	75	75	73.80
$\left(G2\frac{1}{4}\right)$	80	94.47	–	2	16	16	15.30	85	85	83.60
G2 $\frac{1}{2}$	90	105.77	–	2	16	16	15.30	95	95	93.60
G3	100	117.07	–	2	19	19	18.16	105	105	103.60
$\left(G3\frac{1}{2}\right)$	115	139	–	2	19	19	18.16	120	120	118.60
G4	128	150.74	–	2	22	22	21.16	135	135	133.40
$\left(G4\frac{1}{2}\right)$	143	–	162.5	2	22	22	21.16	150	150	148.40
G5	158	–	179	2	22	22	21.16	165	165	163.40
$\left(G5\frac{1}{2}\right)$	172	–	194.5	2	25	25	24.16	180	180	178.40
G6	182	–	206	2	25	25	24.16	190	190	188.15

(DIN: 431)

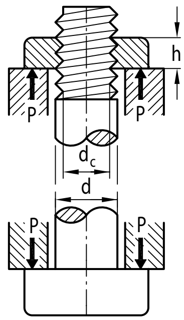
Note:

(i) Grade – Semi-precision (S)
(ii) Property class – 6
(iii) Minimum tensile strength of pipe material = 300 MPa.
(iv) All dimensions are in mm.

Designations – (i) Hexagonal pipe nut of nominal size G6 is designated as Hex Pipe Nut G6
(ii) Octagonal Pipe Nut of Nominal size G6 is designated as Oct Pipe Nut G6

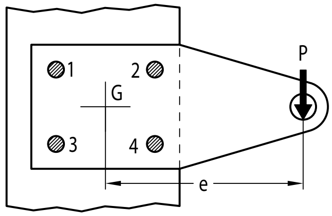
7.10 DESIGN OF BOLTED JOINTS

Table 7.38 *Simple analysis of bolted joint*



Equations	Notations
Tensile stress in core cross section	
$$\sigma_t = \frac{P}{\left(\frac{\pi}{4} d_c^2\right)} \quad (7.1)$$	σ_1 = tensile stress in core cross-section of bolt (MPa or N/mm^2) P = tensile force acting on bolt (N) d_c = core diameter of bolt (mm)
$$\sigma_t = \frac{S_{yt}}{(fs)} \quad (7.2)$$	S_{yt} = tensile yield strength (MPa or N/mm^2) (fs) = factor of safety
Direct shear stress at core diameter	
$$\tau = \frac{P}{\pi d_c h} \quad (7.3)$$	τ = shear stress at core diameter (MPa or N/mm^2) h = height of nut (mm)
$$\tau = \frac{S_{sy}}{(fs)} = \frac{S_{yt}}{2(fs)} \quad (7.4)$$	S_{sy} = yield strength in shear (MPa or N/mm^2) ($S_{sy} = 0.5\ S_{yt}$)
Core diameter—Correct relationship	
$$d_c = d - 1.226\ 87\ \mathrm{p} \quad (7.5)$$	d = nominal diameter of bolt (mm) p = pitch of thread (mm)
Core diameter —Approximate relationship	
$$d_c = 0.8\ d \quad (7.6)$$	

Table 7.39 *Eccentrically loaded bolted joint in shear*



Equations	Notations
Primary shear forces	
$$P_1' = P_2' = P_3' = P_4' = \frac{P}{(\text{No.of bolts})} \qquad (7.7)$$	P = external force (N) P_1', P_2', P_3', P_4' = primary shear forces acting on bolts 1,2,3 and 4 respectively (N)
Secondary shear forces	
$$P_1'' = \frac{Per_1}{(r_1^2 + r_2^2 + r_3^2 + r_4^2)}$$ $$P_2'' = \frac{Per_2}{(r_1^2 + r_2^2 + r_3^2 + r_4^2)} \qquad (7.8)$$ and so on	e = eccentricity of force P from centre of gravity of bolts (mm) $P_1'', P_2'', P_3'', P_4''$ = secondary shear forces acting on bolts 1, 2, 3 and 4 respectively (N) r_1, r_2, r_3, r_4 = distances of bolt centre lines form common centre of gravity (mm)
Resultant shear forces	

The primary and secondary shear forces are added by vector addition method to get the resultant shear forces P_1, P_2, P_3, and P_4. The bolt is designed on the basis of maximum out of these forces.

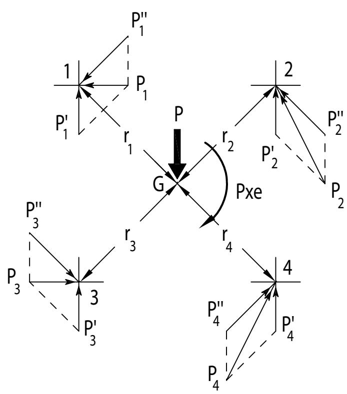
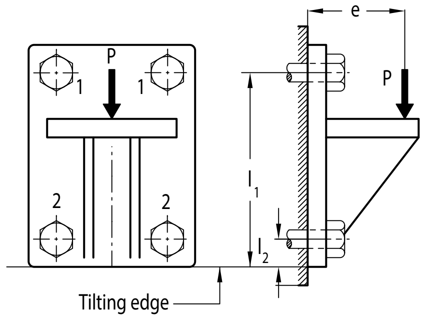
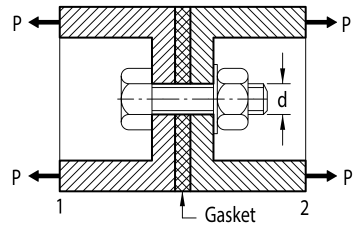


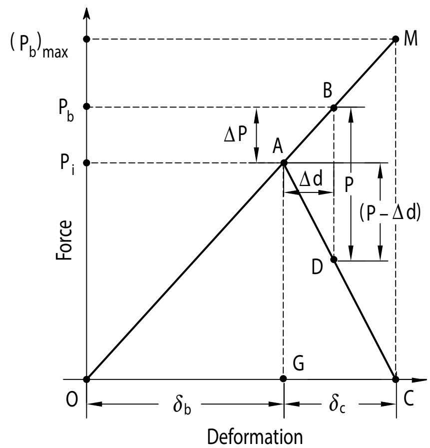
Table 7.40 *Bolts subjected to eccentric load perpendicular to its axis*



Equations	Notations
Direct shear force on bolts	
$P_1' = P_2' = \dfrac{P}{(\text{No. of bolts})}$ (7.9)	P = external force (N) P_1' and P_2' = direct shear forces acting on each bolt at 1 and 2 respectively (N)
Direct shear stress in bolts	
$\tau = \dfrac{P_1'}{A}$ (7.10)	τ = direct shear stress in bolt (MPa or N/mm^2) A = cross-sectional area of bolt (mm^2)
Axial tensile force on bolts	
$P_1'' = \dfrac{Pel_1}{2(l_1^2 + l_2^2)}$ $P_2'' = \dfrac{Pel_2}{2(l_1^2 + l_2^2)}$ (7.11)	P_1' and P_2' = resisting tensile forces induced in each bolt at 1 and 2 respectively (N) l_1 and l_2 = distances of bolts at 1 and 2 from tilting edge (mm) e = eccentricity of force P (mm)
Axial tensile stress in bolts	
$\sigma_t = \dfrac{P_1''}{A}$ (7.12)	σ_1 = axial tensile stress in bolt (MPa or N/mm^2)
Principle stress	
$\sigma_1 = \dfrac{\sigma_t}{2} + \sqrt{\left(\dfrac{\sigma_t}{2}\right)^2 + \tau^2}$ (7.13)	σ_1 = principle stress (MPa or N/mm^2)
Principle shear stress	
$\tau_{\max} = \sqrt{\left(\dfrac{\sigma_t}{2}\right)^2 + \tau^2}$ (7.14)	$\tau_{\max}$ = principle shear stress (MPa or N/mm^2)
Permissible stress	
$\sigma_1 = \dfrac{S_{yt}}{(fs)}$ and $\tau_{\max} = \dfrac{S_{sy}}{(fs)} = \dfrac{0.5 S_{yt}}{(fs)}$ (7.15)	

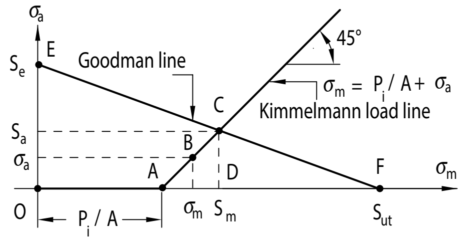
Table 7.41 *Bolted joint with preloaded gasket*





Resultant load on bolt	
$P_b = P_i + \Delta p$ (7.16) $\Delta P = P\left(\dfrac{k_b'}{k_b' + k_c'}\right)$ (7.17)	P_b = resultant load in bolt (N) P_i = initial preload in bolt due to tightening of nut (N) P = external load acting on bolted assembly (N) k_b' = stiffness of the bolt (N/mm) k_c' = combined stiffness of two parts and gasket (N/mm)
Maximum bolt load when joint opens	
$(P_b)_{\max} = P_i\left(\dfrac{k_b' + k_c'}{k_c'}\right)$ (7.18)	$(P_b)_{\max}$ = limiting bolt load when the joint will open and the parts can no longer maintain tight joint (N)

Table 7.42 *Bolted joint under fluctuating load*



Endurance limit stress for bolt

$$S'_e = 0.5\, S_{ut} \quad (7.19)$$

$$S_e = K_b K_c K_d S'_e \quad (7.20)$$

$$K_d = \frac{1}{K_f} \quad (7.21)$$

S'_e = endurance limit stress of rotating beam specimen subjected to reversed bending stress (MPa or N/mm^2)

S_e = endurance limit stress of bolt subjected to reversed bending stress (MPa or N/mm^2)

K_b = size factor = 1 (for axial load)

K_c = reliability factor = 0.897 (for 90% reliability)

K_d = modifying factor to account for stress concentration.

K_f = fatigue stress concentration factor (Table 7.43)

Note: It is not required to consider surface finish factor (K_a) separately. It is incorporated in fatigue stress concentration factor

Maximum and minimum forces on bolt

$$C = \left(\frac{k'_b}{k'_b + k'_c}\right) \quad (7.22)$$

$$(P_b)_{\text{max}} = P_i + CP \quad (7.23)$$

$$(P_b)_{\text{min}} = P_i \quad (7.24)$$

$(P_b)_{\text{max}}$ = maximum force on bolt (N)

$(P_b)_{\text{min}}$ = minimum force on bolt (N)

P_i = initial preload in bolt due to tightening of nut (N)

P = external load acting on bolted assembly (N)

Mean and alternating components of force

$$P_m = \frac{1}{2}[P_{\text{max}} + P_{\text{min}}] \quad (7.25)$$

$$P_a = \frac{1}{2}[P_{\text{max}} - P_{\text{min}}] \quad (7.26)$$

Mean and alternating components of stress

$$\sigma_a = \frac{CP}{2A} \quad (7.27)$$

$$\sigma_m = \frac{P_i}{A} + \frac{CP}{2A} = \frac{P_i}{A} + \sigma_a \quad (7.28)$$

Permissible stress amplitude

$$S_a = \frac{S_{ut} - (P_i/A)}{1 + (S_{ut}/S_e)} \quad (7.29)$$

$$\sigma_a = \frac{S_a}{(fs)} = \frac{P_a}{A} \quad (7.30)$$

Table 7.43 *Fatigue stress concentration factors (K_f) for threaded parts*

SAE grade	Metric grade	Rolled threads	Cut threads	Fillet
0 to 2	3.6 to 5.8	2.2	2.8	2.1
4 to 8	6.6 to 10.9	3.0	3.8	2.3

Welded and Riveted Joints

8

8.1 WELDABILITY

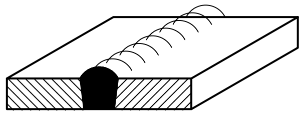
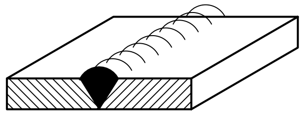
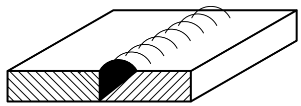
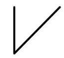
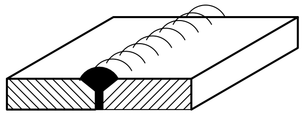
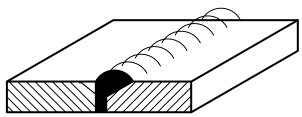
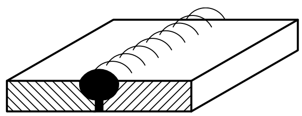
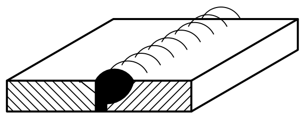
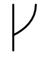
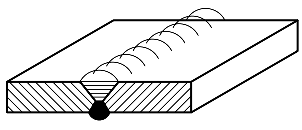
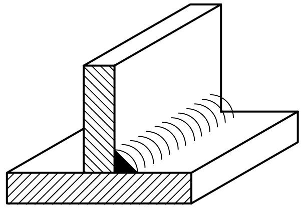
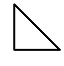
Table 8.1 *Weldability of common metals*

Metal	Arc welding	Gas welding
Carbon steel		
(1) Low and medium carbon steel	G	G
(2) High carbon steel	G	F
(3) Tool steel	F	F
Cast steel- Carbon steel castings	G	G
Grey and alloy cast iron	F	F
Malleable iron	F	F
Low –alloy, high-strength steel		
(1) Ni-Cr- Mo and Ni-Mo	F	F
(2) All other usual compositions	G	G
Stainless steel		
(1) Chromium steel	G	F
(2) Chromium-nickel steel	G	G
Aluminium and its alloys		
(1) Commercially pure aluminium	G	G
(2) Al-Mn alloy	G	G
(3) Al-Mg- Mn and Al-Si-Mg alloy	G	F
(4) Al-Cu-Mg- Mn alloy	F	X
Magnesium alloy	X	G
Copper and copper alloys		
(1) Deoxidized copper	F	G
(2) Pitch, electrolytic and lake	G	F
(3) Commercial bronze, red brass and low brass	F	G
(4) Spring, admiralty, yellow and commercial brass	F	G
(5) Muntz metal, naval brass and magnesium bronze	F	G
(6) Phosphor bronze, bearing bronze and bell metal	G	G
(7) Aluminium bronze	G	F
(8) Beryllium copper	G	—
Nickel and nickel alloys	G	G
Lead	X	G

Notations: G = Good, commonly used; F = Fair, occasionally used under favorable conditions; X = Not used.

8.2 SYMBOLS FOR WELDING

Table 8.2 *Elementary symbols*

Designation	Illustration	Symbol
Square butt weld		
Single-V butt weld		
Single-bevel butt weld		
Single-V butt weld with broad root face		
Single-bevel butt weld with broad root face		
Single-U butt weld (parallel or sloping sides)		
Single-J butt weld		
Backing run, back or backing weld		
Fillet weld		

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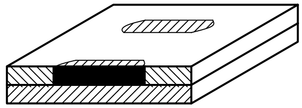
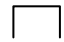
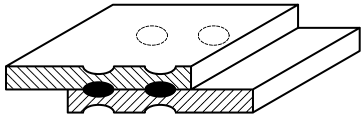
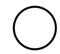
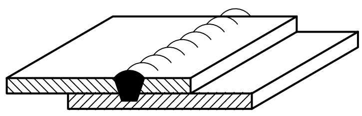
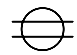
Plug or slot weld		
Spot weld		
Seam weld		

(ISO: 2553)

Note: Plates up to 4 mm thickness are generally welded without bevelling the edges (square butt weld), those of 5 to 15 mm thickness need single V-butt weld and plates of 10 to 30 mm thickness require double V- butt weld, while for higher thickness single U-butt weld are used.

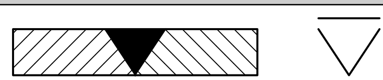
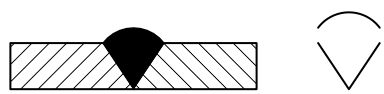
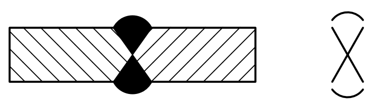
Table 8.3 *Supplementary symbols*

Shape of weld surface	Symbol
(a) Flat (usually finished flush)	
(b) Convex	
(c) Concave	

(ISO: 2553)

Note: The supplementary symbols characterise the shape of external surface of the weld.

Table 8.4 *Examples of application of supplementary symbols*

Designation	Illustration and symbol
Flat (flush) single-V butt weld	
Convex single-V butt weld	
Convex double-V butt weld	

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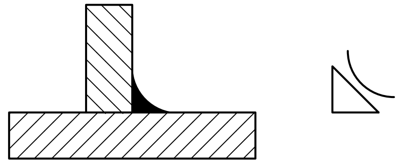
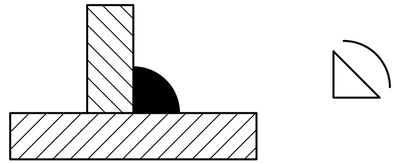
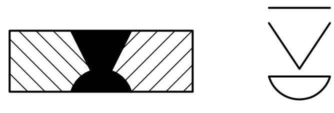
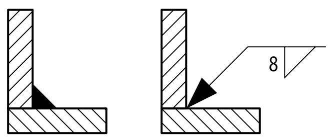
Concave fillet weld	
Convex fillet weld	
Flat (flush) single-V butt weld with flat (flush) backing run	

Table 8.5 *Location of welding symbol*

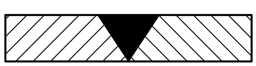
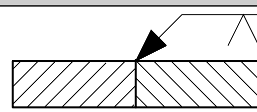
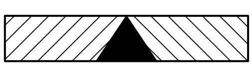
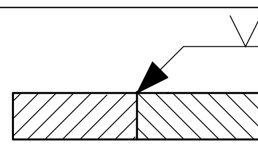


The complete information about the welded joint is conveyed by the designer to the welding operator by placing suitable welding symbol on the drawing. The information includes the type of welded joint, the size of weld, the location of weld and certain special instructions. The complete weld symbol consists of following elements:

(i) An elementary symbol to specify the type of weld.
(ii) An arrow and a reference line to indicate the location of the weld.
(iii) Supplementary symbols to indicate the shape of external surface of weld.
(iv) Dimensions of the weld in cross section and length.

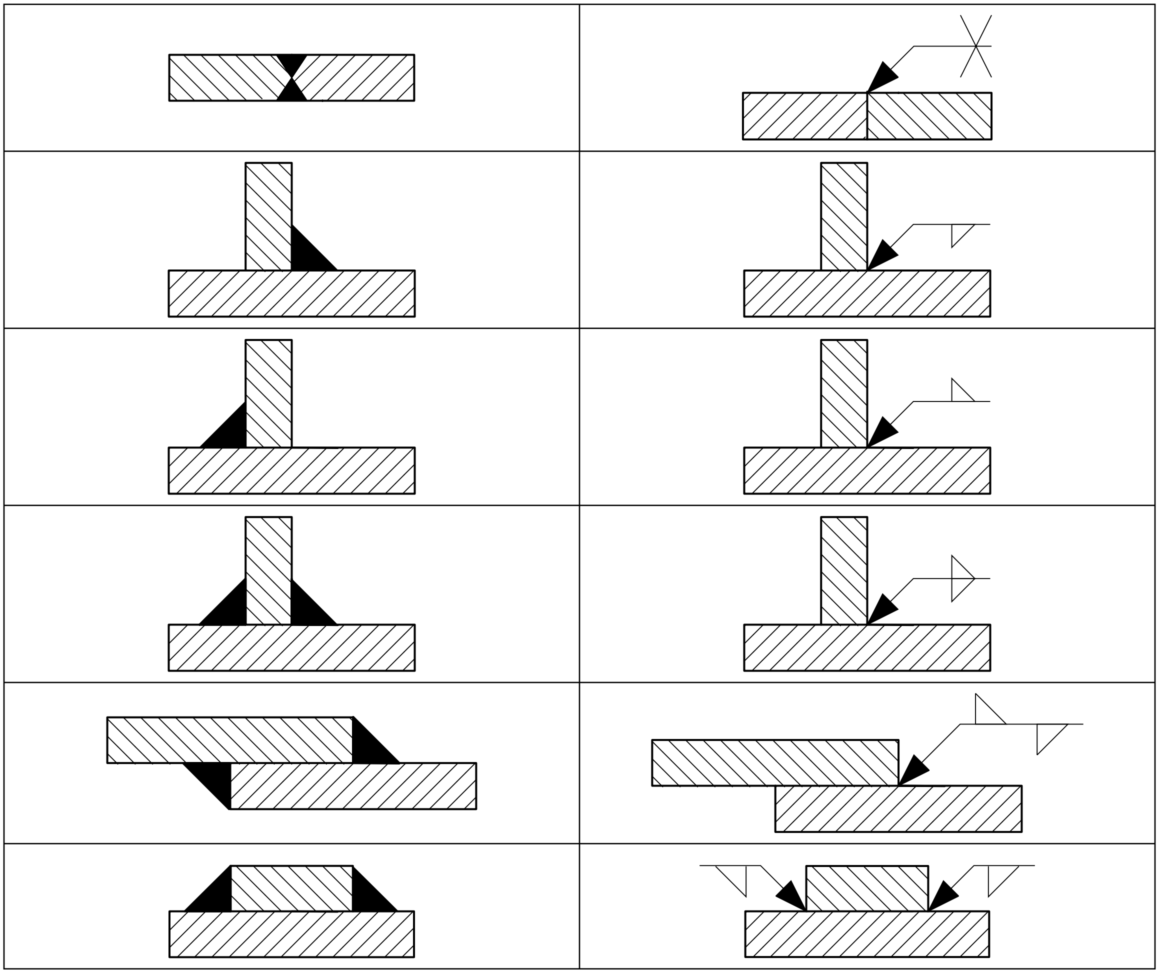
The location of weld is indicated by an arrow and a reference line. The head of the arrow indicates the reference side of the joint. When the weld symbol is below the reference line, the weld is made on the same side of the joint as the arrowhead. When the weld symbol is above the reference line, the weld is made on the other side of the joint opposite the arrowhead.

Table 8.6 *Examples of use of welding symbol*

Illustration	Symbolisation
	
	

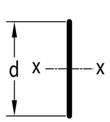
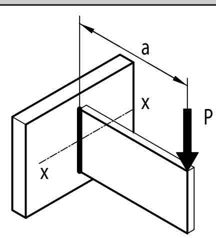
(*Contd.*)

(*Contd.*)



8.3 LINE PROPERTIES OF WELDS

Table 8.7 *Bending properties of weld treated as a line*

Outline of welded joint	Illustration	Bending properties
		$Z_w = \dfrac{d^2}{6}$

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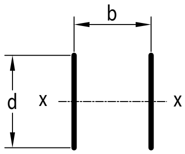
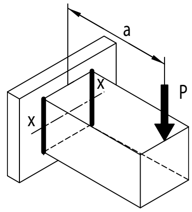
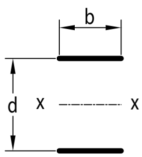
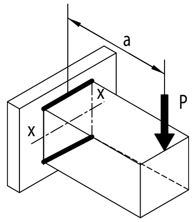
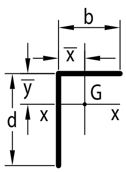
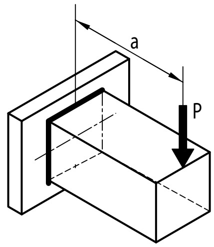
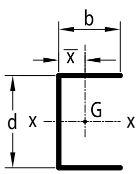
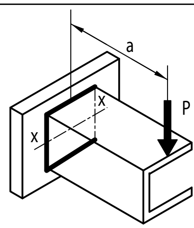
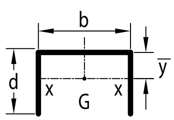
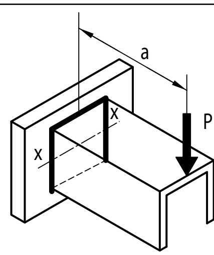
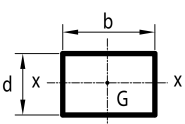
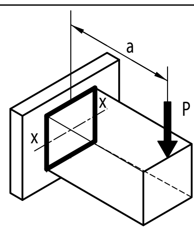
(*Contd.*)

		$Z_w = \frac{d^2}{3}$
		$Z_w = bd$
		$\bar{x} = \frac{b^2}{2(b+d)}$ $\bar{y} = \frac{d^2}{2(b+d)}$ At top fibre $Z_w = \frac{4bd + d^2}{6}$ At bottom fibre $Z_w = \frac{d^2(4b+d)}{6(2b+d)}$
		$\bar{x} = \frac{b^2}{(2b+d)}$ $Z_w = bd + \frac{d^2}{6}$
		$\bar{y} = \frac{d^2}{b+2d}$ At top fibre $Z_w = \frac{2bd + d^2}{3}$ At bottom fibre $Z_w = \frac{d^2(2b+d)}{3(b+d)}$
		$Z_w = bd + \frac{d^2}{3}$

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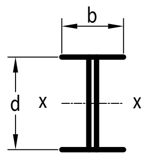
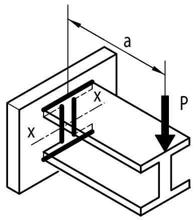
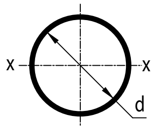
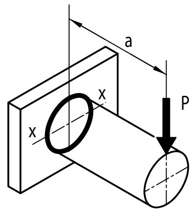
(*Contd.*)

		$Z_w = bd + \dfrac{d^2}{3}$
		$Z_w = \pi\left(\dfrac{d^2}{4}\right)$

Note: For parallel loading, $s = \dfrac{M}{0.707 Z_w h}$

For transverse loading, $s = \dfrac{M}{0.828 Z_w h}$

(Z_w) = section modulus of weld treated as a line about x-axis (mm^2)

M = bending moment (N-mm) ($M = P \times a$)

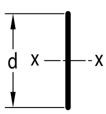
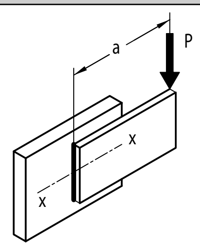
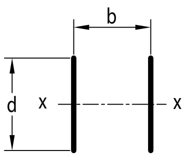
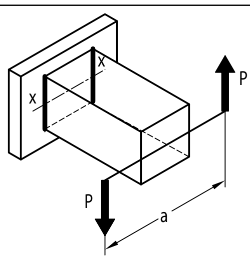
h = leg dimension of weld (mm)

s = stress in standard design formula (N/mm^2)

$\left(\dfrac{M}{Z_w}\right)$ = force per unit length of weld (N/mm)

For unsymmetrical connections, the maximum bending stress is at the bottom.

Table 8.8 *Torsional properties of weld treated as a line*

Outline of welded joint	Illustration	Torsional properties
		$J_w = \dfrac{d^3}{12}$
		$J_w = \dfrac{d(3b^2 + d^2)}{6}$

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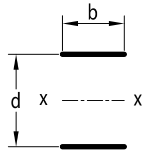
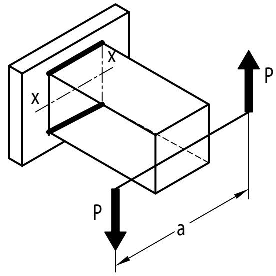
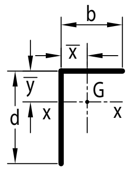
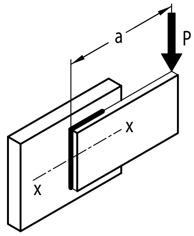
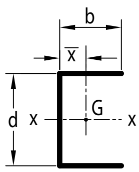
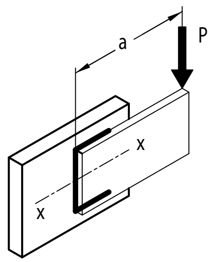
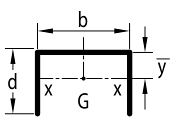
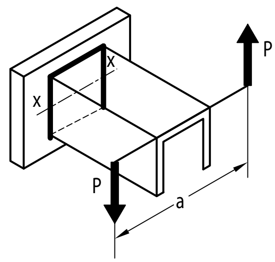
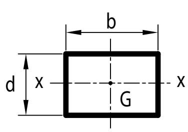
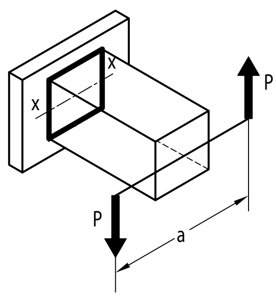
(*Contd.*)

		$J_w = \dfrac{b^3 + 3bd^2}{6}$
		$\overline{x} = \dfrac{b^2}{2(b+d)}$ $\overline{y} = \dfrac{d^2}{2(b+d)}$ $J_w = \dfrac{(b+d)^4 - 6b^2d^2}{12(b+d)}$
		$\overline{x} = \dfrac{b^2}{(2b+d)}$ $J_w = \dfrac{(2b+d)^3}{12} - \dfrac{b^2(b+d)^2}{(2b+d)}$
		$\overline{y} = \dfrac{d^2}{b+2d}$ $J_w = \dfrac{(b+2d)^3}{12} - \dfrac{d^2(b+d)^2}{(b+2d)}$
		$J_w = \dfrac{(b+d)^3}{6}$

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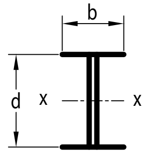
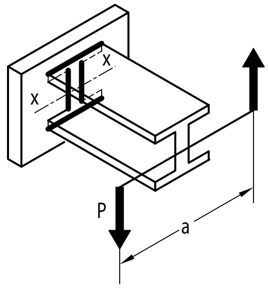
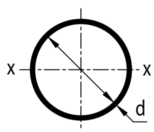
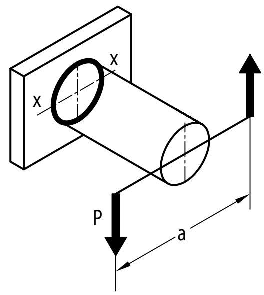
(*Contd.*)

		$J_w = \frac{b^3 + 3bd^2 + d^3}{6}$
		$J_w = \pi\left(\frac{d^3}{4}\right)$

Note: J_w = polar moment of inertia of weld treated as a line (mm^3)
M_t = torsional moment (N-mm) ($M_t = P \times a$)

8.4 STRENGTH OF WELDED JOINTS

Table 8.9 *Strength properties of AWS electrodes*

AWS electrode number	Ultimate tensile strength (Min.) (MPa)	Yield strength (Min.) (MPa)	Percentage elongation (%)
E60XX	425	345	17-25
E70XX	480	390	22
E80XX	550	460	19
E90XX	620	530	14–17
E100XX	690	600	13–16
E120XX	820	730	14

Note: (i) AWS = American Welding Society (ii) The numbering system for electrodes uses letter E followed by four digits. The first two digits indicate the approximate tensile strength. The last digit indicates variables in welding technique, such as current supply. The next to last digit indicate welding position such as horizontal, vertical, flat or overhead.

Table 8.10 *Permissible weld stresses for weld metal (AISC code)*

Type of load	Type of joint	Permissible stress	Factor of safety
Tension	Butt	0.60 S_{yt}	1.67
Bearing	Butt	0.90 S_{yt}	1.11
Bending	Butt	0.60 to 0.66 S_{yt}	1.52 to 1.67
Simple Compression	Butt	0.60 S_{yt}	1.67
Shear	Butt or Fillet	0.40 S_{yt}	1.44

Note: (i) AISC = American Institute of Steel Construction, (ii) For shear load, the factor of safety is calculated by distortion energy theory. (iii) S_{yt} = yield strength of material.

Table 8.11 *Applications of electrodes*

Designation of electrode	Penetration	Application
E6010	Deep	It has good mechanical properties. It is used where multi-pass welds are employed such as welds in buildings, bridges, pressure vessels and pipes.
E6011		
E6012	Medium	It is used for single- pass high- speed horizontal fillet welds.
E6013	Medium	It is used for obtaining good quality weld within metal.
E6020	Medium deep	It is used for high-production horizontal fillet welds in heavy sections.
E6027	Medium	It is iron powder electrode that is fast and easy to handle.

Table 8.12 *Weld design stress by C H Jennings*

Types of weld and stress	Permissible stresses (MPa or N/mm^2)	
	Static load	Reversed load
Butt welds		
Tension	110	55
Compression	125	55
Shear	70	35
Fillet welds	95	35

Table 8.13 *Mechanical properties of weld deposits*

Electrode number	Tensile strength (MPa or N/mm^2)	Yield strength (minimum) (MPa or N/mm^2)
E XXX - 41	410–510	330
E XXX - 51	510–610	360

Table 8.14 *Fatigue stress concentration factor for welded joints*

Type of weld	K_f
Reinforced butt weld	1.2
Toe of transverse fillet weld	1.5
End of parallel fillet weld	2.7
T- butt joint with sharp corners	2.0

Table 8.15 *Minimum size of fillet welds*

Thickness of thicker part to be joined (mm)	Weld Size (mm)
Up to and including 6.4	3.2
Over 6.4 and up to 12.5	4.8
Over 12.5 and up to18	6.4
Over 18 and up to 38	8.0
Over 38 and up to 57	9.5
Over 57 and up to 150	12
Over 150	15

Table 8.16 *Fatigue strength of fillet welds (AWS recommendations)*

Number of stress cycles	Fatigue strength (S_f) (MPa or N/mm^2)
20×10^5	$S_f = \left[\dfrac{50}{1 - \frac{1}{2}K}\right]$ or 84 MPa (whichever is minimum)
6×10^5	$S_f = \left[\dfrac{70}{1 - \frac{1}{2}K}\right]$ or 84 MPa (whichever is minimum)
1×10^5	$S_f = \left[\dfrac{80}{1 - \frac{1}{2}K}\right]$ or 84 MPa (whichever is minimum)

Note: (i) $K = \dfrac{\text{minimum load}}{\text{maximum load}} = \dfrac{\text{minimum stress}}{\text{maximum stress}}$

(ii) $K = +1$ (for steady load) (iii) $K = 0$ (if load varies in one direction)

(iv) $K = -1$ (for completely reversed load)

(v) $(S_f)_A = (S_f)_B \left(\dfrac{N_B}{N_A}\right)^c$

$(S_f)_A$ = fatigue strength for N_A cycles

$(S_f)_B$ = fatigue strength for N_B cycles

$c = 0.13$ for butt weld

$c = 0.18$ for plates in axial loading- tension or compression

(vi) The above expressions are recommended by AWS in bridge design and based on more conservative design.

8.5 STRENGTH EQUATIONS FOR WELDED JOINTS

Table 8.17 *Basic strength equations*

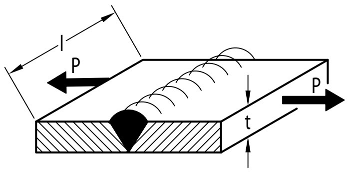
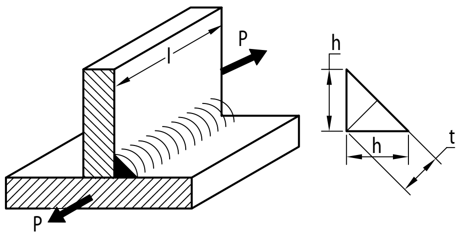
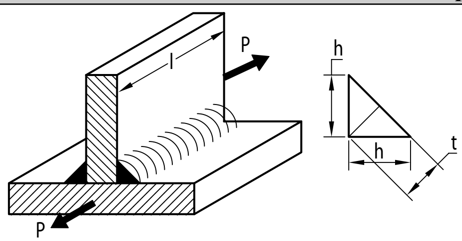
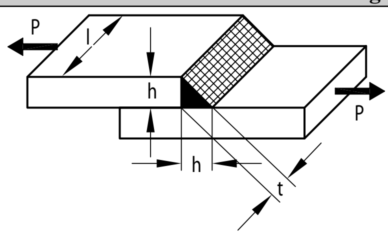
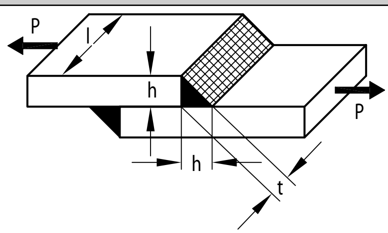
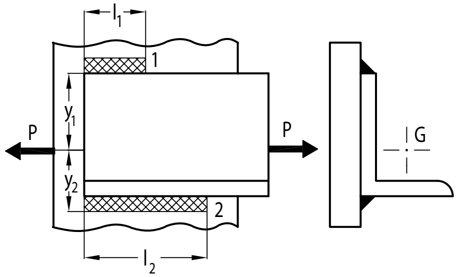
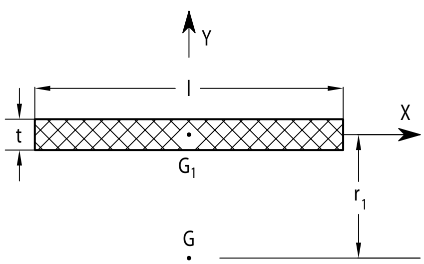
<table>
<tr><th colspan="2">Butt weld</th></tr>
<tr><td>
</td><td>$P = \sigma_t tl$ (8.1)
$P = \sigma_t tl\eta$ (8.2)
σ_t = permissible tensile stress for weld (MPa or N/mm^2)
P = tensile force on plates (N)
t = thickness of plates (mm)
l = length of weld (mm)
η = efficiency of welded joint (in fraction)</td></tr>
<tr><th colspan="2">Single parallel fillet weld</th></tr>
<tr><td>
</td><td>$t = h \cos(45°)$ or $t = 0.707h$ (8.3)
$P = 0.707hl\tau$ (8.4)
P = tensile force on plates (N)
h = leg of weld (mm)
t = throat of weld (mm)
l = length of weld (mm)
τ = permissible shear stress for weld (MPa or N/mm^2)</td></tr>
<tr><th colspan="2">Double parallel fillet welds</th></tr>
<tr><td>
</td><td>$P = 1.414hl\tau$ (8.5)</td></tr>
<tr><th colspan="2">Single transverse fillet weld</th></tr>
<tr><td>
</td><td>$P = 0.707hl\sigma_t$ (8.6)
σ_t = permissible tensile stress for weld (MPa or N/mm^2)</td></tr>
<tr><th colspan="2">Double transverse fillet welds</th></tr>
<tr><td>
</td><td>$P = 1.414hl\sigma_t$ (8.7)</td></tr>
</table>

Table 8.18 *Axially loaded unsymmetrical welded joints*



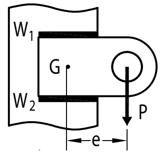
Equations		Notations
$P_1 = 0.707hl_1\tau$	(8.8)	P = external force acting on joint and passing through centre of gravity G (N)
$P_2 = 0.707hl_2\tau$	(8.9)	P_1 and P_2 = resisting forces set up in the welds 1 and 2 respectively (N)
$P = P_1 + P_2$	(8.10)	l_1 and l_2 = lengths of welds 1 and 2 respectively (mm)
$l_1y_1 = l_2y_2$	(8.11)	y_1 and y_2 = distances of welds 1 and 2 respectively from centre of gravity G (mm)
$l_1 + l_2 = l$	(8.12)	l = total length of all welds (mm)
		h = leg of weld (mm)
		τ = permissible shear stress for weld (MPa or N/mm^2)

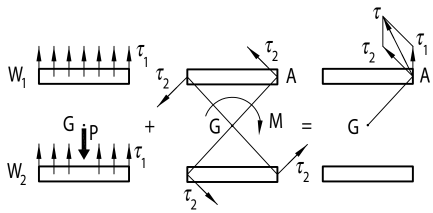
Table 8.19 *Polar moment of inertia for group of welds*



Equations		Notations
$J_{G_1} = \dfrac{Al^2}{12}$	(8.13)	J_{G_1} = polar moment of inertia of the weld about its centre of gravity G_1 (mm^4) A = throat area of weld (mm^2) l = length of weld (mm)
$J_G = J_{G_1} + A\,r_1^2$ $J_G = A\left[\dfrac{l^2}{12} + r_1^2\right]$	(8.14) (8.15)	J_G = polar moment of inertia of weld about axis passing through G (mm^4) r_1 = distance between G and G_1 (mm)
$J = J_1 + J_2 + J_3 + \ldots$	(8.16)	J_1, J_2, J_3,.= polar moment of inertia of welds 1,2,3…respectively about centre of gravity G (mm^4) J = resultant polar moment of inertia of all welds about G (mm^4)

Table 8.20 *Eccentric load in plane of welds*

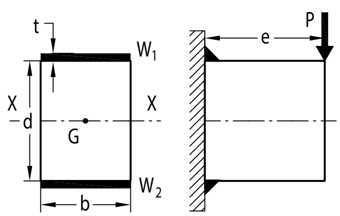




Equations		Notations
$\tau_1 = \dfrac{P}{A}$	(8.17)	τ_1 = primary shear stress (MPa or N/mm^2) P = eccentric force (N) A = throat area of all welds (mm^2)
$\tau_2 = \dfrac{Mr}{J}$ $M = P \times e$	(8.18) (8.19)	τ_2 = secondary shear stress (MPa or N/mm^2) M = couple (N-mm) e = eccentricity of force P (mm) r = distance of a point in weld from G (mm) J = polar moment of inertia of all welds about G (mm^4)

Note: The secondary shear stress at any point in the weld is proportional to its distance from the centre of gravity. Obviously, it is maximum at the farthest point such as *A*. The resultant shear stress at this point is obtained by vector addition of primary and secondary shear stresses.

Table 8.21 *Welded joint subjected to bending moment*



Equations		Notations
$\tau_1 = \dfrac{P}{A}$	(8.20)	τ_1 = primary shear stress (MPa or N/mm^2) P = eccentric force (N) A = throat area of all welds (mm^2)
$\sigma_b = \dfrac{M_b y}{I}$ $M_b = P \times e$	(8.21) (8.22)	M_b = bending moment (N-mm) I = moment of inertia of all welds based on throat area (mm^4) y = distance of the point in weld from neutral-axis (mm) e = eccentricity of force P (mm) σ_b = bending stress (MPa or N/mm^2)
$I = 2I_{xx} = t\left(\dfrac{bd^2}{2}\right)$	(8.23)	I = moment of inertia of two welds about XX axis passing through G (mm^4) b = length of each weld (mm) d = distance between two welds (mm)
$\tau = \sqrt{\left(\dfrac{\sigma_b}{2}\right)^2 + (\tau_1)^2}$	(8.24)	τ = resultant shear stress (MPa or N/mm^2)

8.6 BOILER RIVETS

Table 8.22 *Preferred length and diameter combinations for boiler rivets*

Length (mm)	**Diameter (mm)**													
	12	**14**	**16**	**18**	**20**	**22**	**24**	**27**	**30**	**33**	**36**	**39**	**42**	**48**
28	X													
31.5	X	X												
35.5	X	X	X											
40	X	X	X	X										
45	X	X	X	X	X									
50	X	X	X	X	X	X								
56	X	X	X	X	X	X	X							
63	X	X	X	X	X	X	X	X						
71	X	X	X	X	X	X	X	X	X					
80	X	X	X	X	X	X	X	X	X					
85		X	X	X	X	X	X	X	X	X				
90		X	X	X	X	X	X	X	X	X				
95		X	X	X	X	X	X	X	X	X	X			
100			X	X	X	X	X	X	X	X	X			
106			X	X	X	X	X	X	X	X	X	X		
112			X	X	X	X	X	X	X	X	X	X		
118				X	X	X	X	X	X	X	X	X	X	
125					X	X	X	X	X	X	X	X	X	X
132						X	X	X	X	X	X	X	X	X
140						X	X	X	X	X	X	X	X	X
150							X	X	X	X	X	X	X	X
160							X	X	X	X	X	X	X	X
180								X	X	X	X	X	X	X
200									X	X	X	X	X	X
224										X	X	X	X	X
250													X	X

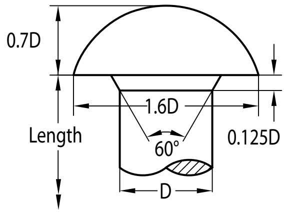
Note: Preferred combinations are indicated by X.

Table 8.23 *Rivet hole diameters and tolerances on boiler rivet head*

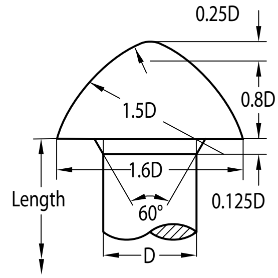
Basic size of rivet (D) (mm)	Rivet hole diameter Min (mm)	Tolerances (in mm) on			
		Diameter of head		Height of head	Permissible eccentricity of head
		Countersunk head* (both types)	All other types of heads*		
12	13.0	±0.35	±1.2	+0.8 –0.0	0.7
14	15.0	±0.4	±1.2	+0.8 –0.0	0.7
16	17.0	±0.5	±1.2	+0.8 –0.0	0.8
18	19.0	±0.5	±1.2	+0.8 –0.0	0.8
20	21.0	±0.8	±1.2	+1.2 –0.0	0.9
22	23.0	±0.8	±1.2	+1.2 –0.0	1.1
24	25.0	±0.8	±1.2	+1.2 –0.0	1.3
27	28.5	±0.8	±1.2	+1.2 –0.0	1.5
30	31.5	±0.8	±1.2	+1.6 –0.0	1.6
33	34.5	±0.8	±1.2	+1.6 –0.0	1.6
36	37.5	±1.6	±1.2	+1.6 –0.0	1.8
39	41.0	±1.6	±1.2	+1.6 –0.0	1.8
42	44.0	±1.6	±1.2	+1.6 –0.0	2.0
48	50.0	±1.6	±1.2	+1.6 –0.0	2.0

* Refer to Table 8.24 for different types of rivet heads.

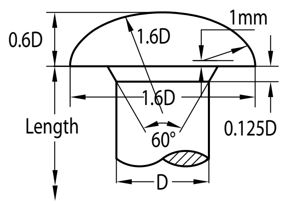
Table 8.24 *Proportions of boiler rivet heads*



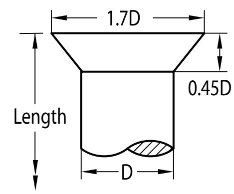
Snap head



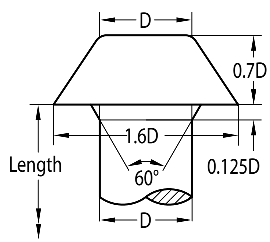
Conical head



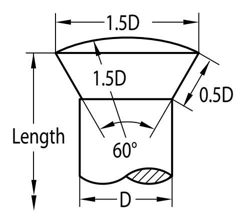
Ellipsoid head



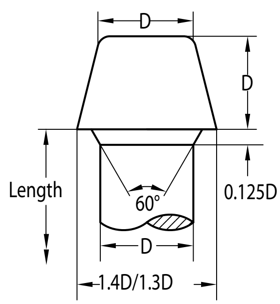
Countersunk head



Pan head (Type 1)

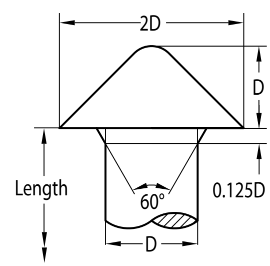


Rounded countersunk head



Pan head (Type 2)

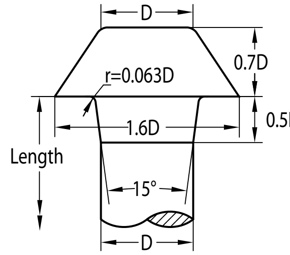
1.4D = for rivets under 24 mm
1.3D = for rivets 24 mm and over



Steeple head

(*Contd.*)

(*Contd.*)



Pan head with tapered neck

(Steeple means a tall tower of church)

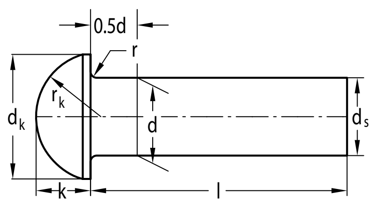
Table 8.25 *Mechanical properties of steel rivet bars for boilers*

Property	Grade	
	Grade St 37 BR	Grade St 42 BR
Tensile strength (MPa or N/mm^2)	360–440	410–500
Yield stress Min. (MPa or N/mm^2) (a) Up to and including 20 mm (b) Over 20 mm	 220 200	 250 240
Elongation (percent) Min.	26	23

Note: (i) Grade St 37 means minimum tensile strength 37 kgf/mm^2.
(ii) Grade St 42 means minimum tensile strength 42 kgf/mm^2.

8.7 COLD FORGED RIVETS (GENERAL PURPOSE)

Table 8.26 *Dimensions of cold forged steel rivets with snap head*



d	Nom.	6	8	10	12	(14)	16
	Max.	6.15	8.15	10.3	12.3	14.3	16.3
	Min.	5.85	7.85	9.7	11.7	13.7	15.7

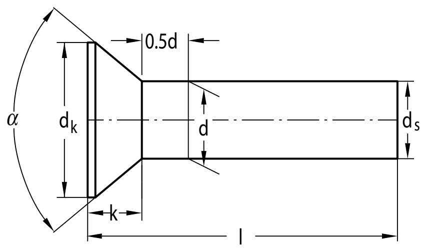
(*Contd.*)

(*Contd.*)

d_s	Min.	5.82	7.76	9.4	11.3	13.2	15.2
d_k	Nom.	9.6	12.8	16.0	19.2	22.4	25.6
	Max.	9.9	13.1	16.36	19.6	22.8	26.1
	Min.	9.3	12.5	15.64	18.8	22	25.1
k	Nom.	4.2	5.6	7	8.4	9.8	11.2
	Max.	4.44	5.84	7.29	8.69	10.09	11.55
	Min.	3.96	5.36	6.71	8.11	9.51	10.85
r	Max.	0.3	0.4	0.5	0.6	0.7	0.8
r_k	≈	5.7	7.5	8	9.5	11	13

Note: The nominal diameter *d* in parenthesis is of second preference.
Designation – A snap head rivet of 6 mm diameter having a length 30 mm is designated as 'Snap Head Rivet 6 × 30'.

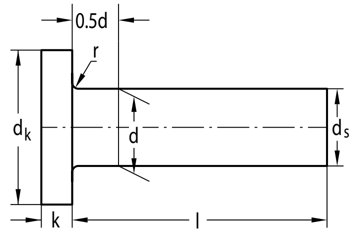
Table 8.27 *Dimensions of cold forged steel rivets with flat countersunk head*



d	Nom.	6	8	10	12	(14)	16
	Max.	6.15	8.15	10.3	12.3	14.3	16.3
	Min	5.85	7.85	9.7	11.7	13.7	15.7
d_s	Min.	5.82	7.76	9.4	11.3	13.2	15.2
d_k	Nom.	12	16	20	24	21	24
	Max.	12	16	20	24	21	24
	Min.	11.3	15.3	19.16	23.16	20.16	23.16
k	≈	3	4	5	6	6	6.9
α		90°				60°	

Note: The nominal diameter *d* in parenthesis is of second preference.
Designation – A flat countersunk head rivet of 6 mm diameter having a length 30 mm is designated as 'Flat countersunk head rivet 6 × 30'.

Table 8.28 *Dimensions of cold forged steel rivets with flat head*



d	Nom.	6	8	10	12	(14)	16
	Max.	6.15	8.15	10.3	12.3	14.3	16.3
	Min.	5.85	7.85	9.7	11.7	13.7	15.7
d_s	Min.	5.82	7.76	9.4	11.3	13.2	15.2
d_k	Nom.	12	16	20	24	28	32
	Max.	12	16	20	24	28	32
	Min.	11.3	15.3	19.16	23.16	27.16	31
k	Nom.	1.5	2	2.5	3	3.5	4
	Max.	1.9	2.5	3	3.6	4.1	4.6
	Min.	1.5	2	2.5	3	3.5	4
r	Max.	0.3	0.4	0.5	0.6	0.7	0.8

Note: The nominal diameter *d* in parenthesis is of second preference.
Designation – A flat head rivet of 6 mm diameter having a length 30 mm is designated as 'Flat head rivet 6 × 30'.

Table 8.29 *Diameter-length combinations for cold forged rivets for hot closing*

Length *l* (mm)	**Nominal diameter *d* (mm)**					
	6	**8**	**10**	**12**	**(14)**	**16**
12	X					
14	X	X				
16	X	X				
18	X	X	X			
20	X	X	X	X		
22	X	X	X	X	X	
24	X	X	X	X	X	X
26	X	X	X	X	X	X
28	X	X	X	X	X	X
30	X	X	X	X	X	X
32	X	X	X	X	X	X
35	X	X	X	X	X	X

(*Contd.*)

(*Contd.*)

38	X	X	X	X	X	X
40	X	X	X	X	X	X
42	X	X	X	X	X	X
45	X	X	X	X	X	X
48	X	X	X	X	X	X
50	X	X	X	X	X	X
55	X	X	X	X	X	X
60		X	X	X	X	X
65		X	X	X	X	X
70		X	X	X	X	X
75			X	X	X	X
80			X	X	X	X
85			X	X	X	X
90				X	X	X
95				X	X	X
100				X	X	X
105					X	X
110					X	X
Rivet hole diameter (mm)	6.3	8.4	10.5	13	15	17

Note: (i) The nominal diameter *d* in parenthesis is of second preference.
(ii) Preferred combinations are indicated by *X*.
(iii) The tolerances on length are as follows:

(a) For ($l \leq 10$ mm) Tolerance = $\frac{+0.5}{0.0}$ mm

(b) For (10 mm $< l \leq 20$ mm) Tolerance = $\frac{+1.0}{0.0}$ mm

(c) For ($l > 20$ mm) Tolerance = $\frac{+1.5}{0.0}$ mm

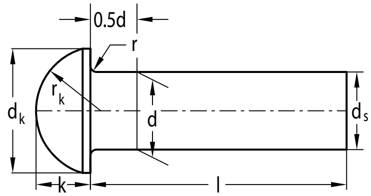
(iv) Basic tolerance on rivet hole = H12

Table 8.30 *Mechanical properties of steel for cold forged rivets*

Property	**Grade**	
	Annealed condition	**As-drawn condition**
Tensile strength (MPa or N/mm^2)	330–410	410–490
Yield stress Min. (MPa or N/mm^2)	160	190
Shear strength Min. (MPa or N/mm^2)	270	300
Elongation (percent) Min.	30	20

8.8 HOT FORGED RIVETS (STRUCTURAL PURPOSE)

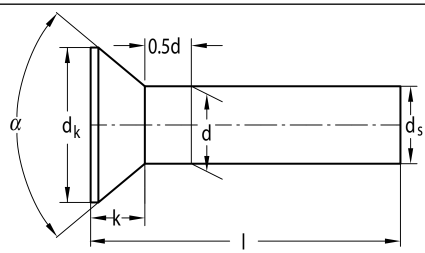
Table 8.31 *Dimensions of hot forged steel rivets with snap head*



d	Nom.	12	(14)	16	(18)	20	(22)	24	(27)	30	(33)	36
	Max.	12.8	14.8	16.8	18.8	20.8	22.8	24.8	27.8	30.8	33.8	36.8
	Min.	12	14	16	18	20	22	24	27	30	33	36
d_s	Min.	11.3	13.2	15.2	17.1	19.1	20.9	22.9	25.8	28.6	31.6	34.6
d_k	Nom.	19.2	22.4	25.6	28.8	32	35.2	38.4	43.2	48	52.8	57.6
	Max.	19.55	22.8	26.1	29.3	33.2	36.4	39.6	44.4	49.2	54	59
	Min.	18.85	22	25.1	28.3	30.8	34	37.2	42	46.8	51.6	56.2
k	Nom.	8.4	9.8	11.2	12.6	14	15.4	16.8	18.9	21	23.1	25.2
	Max.	9.3	10.7	12.3	13.7	15.1	16.5	17.9	20.2	22.3	24.4	26.5
	Min.	8.4	9.8	11.2	12.6	14	15.4	16.8	18.9	21	23.1	25.2
r	Max.	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.4	1.5	1.7	1.8
r_k	≈	9.5	11	13	14.5	16.5	18.5	20.5	22	24.5	27	30

Note: The nominal diameters *d* shown in parenthesis are of second preference.

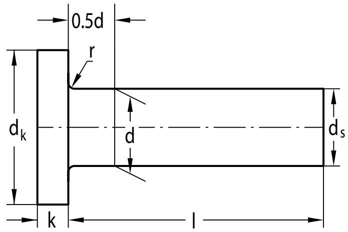
Table 8.32 *Dimensions of hot forged steel rivets with flat countersunk head*



d	Nom.	12	(14)	16	(18)	20	(22)	24	(27)	30	(33)	36
	Max.	12.8	14.8	16.8	18.8	20.8	22.8	24.8	27.8	30.8	33.8	36.8
	Min.	12	14	16	18	20	22	24	27	30	33	36
d_s	Min.	11.3	13.2	15.2	17.1	19.1	20.9	22.9	25.8	28.6	31.6	34.6
d_k	Nom.	18	21	24	27	30	33	35	40.5	45	49.5	54
	Max.	18	21	24	27	30	33	35	40.5	45	49.5	54
	Min.	16.9	19.7	22.7	25.7	28.7	31.4	34.4	38.9	43.4	47.9	52.1
k	≈	5.2	6.1	6.9	7.8	8.7	9.5	10.4	11.7	13	14.3	15.6
α		60°										

Note: The nominal diameters *d* shown in parenthesis are of second preference.

Table 8.33 *Dimensions of hot forged steel rivets with flat head*



d	Nom.	12	(14)	16	(18)	20	(22)	24	(27)	30	(33)	36
	Max.	12.8	14.8	16.8	18.8	20.8	22.8	24.8	27.8	30.8	33.8	36.8
	Min.	12	14	16	18	20	22	24	27	30	33	36
d_s	Min.	11.3	13.2	15.2	17.1	19.1	20.9	22.9	25.8	28.6	31.6	34.6
d_k	Nom.	24	28	32	36	40	44	48	54	60	66	72
	Max.	24	28	32	36	40	44	48	54	60	66	72
	Min.	22.7	26.7	30.4	34.4	38.4	42.4	46.4	52.1	58.1	64.1	70.1
k	Nom.	3	3.5	4	4.5	5	5.5	6	6.75	7.5	8.25	9
	Max.	3.6	4.25	4.75	5.25	5.75	6.25	6.75	7.65	8.40	9.15	9.9
	Min.	3	3.5	4	4.5	5	5.5	6	6.75	7.50	8.25	9.0
r	Max.	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.4	1.5	1.7	1.8

Note: The nominal diameters *d* shown in parenthesis are of second preference.

Table 8.34 *Diameter-length combinations for hot forged rivets for hot closing*

Length *l* (mm)	**Nominal diameter *d* (mm)**										
	12	**(14)**	**16**	**(18)**	**20**	**(22)**	**24**	**(27)**	**30**	**(33)**	**36**
28	X										
30	X										
32	X	X									
35	X	X	X								
38	X	X	X								
40	X	X	X	X							
45	X	X	X	X	X						
50	X	X	X	X	X	X					
55	X	X	X	X	X	X	X				
60	X	X	X	X	X	X	X				
65	X	X	X	X	X	X	X	X			
70	X	X	X	X	X	X	X	X	X		
75	X	X	X	X	X	X	X	X	X		
80	X	X	X	X	X	X	X	X	X		
85		X	X	X	X	X	X	X	X	X	

(*Contd.*)

(*Contd.*)

90		X	X	X	X	X	X	X	X	X	
95		X	X	X	X	X	X	X	X	X	X
100			X	X	X	X	X	X	X	X	X
105			X	X	X	X	X	X	X	X	X
110			X	X	X	X	X	X	X	X	X
115				X	X	X	X	X	X	X	X
120				X	X	X	X	X	X	X	X
125					X	X	X	X	X	X	X
130						X	X	X	X	X	X
135						X	X	X	X	X	X
140						X	X	X	X	X	X
145							X	X	X	X	X
150							X	X	X	X	X
155							X	X	X	X	X
160							X	X	X	X	X
165								X	X	X	X
170								X	X	X	X
175								X	X	X	X
180								X	X	X	X
185									X	X	X
190									X	X	X
195									X	X	X
200									X	X	X
205										X	X
210										X	X
215										X	X
220										X	X
225										X	X
Rivet hole diameter (mm)	13.5	15.5	17.5	19.5	21.5	23.5	25.5	29.0	32.0	35.0	38.0

Note: (i) The nominal diameters *d* shown in parenthesis are of second preference.
(ii) Preferred combinations are indicated by X.
(iii) The tolerances on length are as follows:

(a) For ($d \leq 16$ mm) Tolerance = $\dfrac{+1.5}{0.0}$ mm

(b) For ($d > 16$ mm) Tolerance = $\dfrac{+3.0}{0.0}$ mm

(iv) Basic tolerance on rivet hole = H12

Table 8.35 *Mechanical properties of hot-rolled steel rivet bars for structural purposes*

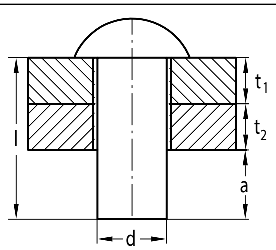
Property	Requirement
Tensile strength (MPa or N/mm^2)	410–530
Yield stress Min (MPa or N/mm^2)	
(a) 6 mm up to including 12mm	260
(b) over 12 mm up to and including 20 mm	250
(c) over 20 mm up to and including 40 mm	240
Elongation (percent) Min	23
Ultimate shear strength Min (MPa or N/mm^2)	330

Table 8.36 *Mechanical properties of high tensile steel rivet bars for structural purposes*

Property	Requirement
Tensile strength Min (MPa or N/mm^2)	460
Yield stress Min (MPa or N/mm^2)	
(a) 6 mm up to including 12 mm	310
(b) over 12 mm up to and including 20 mm	300
(c) over 20 mm up to and including 40 mm	280
Elongation (percent) Min	22
Ultimate shear strength Min (MPa or N/mm^2)	370

8.9 RIVETED JOINTS

Table 8.37 *Length of rivet*



$$l = (t_1 + t_2) + a \qquad (8.25)$$

l = length of rivet shank (mm)

t_1, t_2 = thickness of plates (mm)

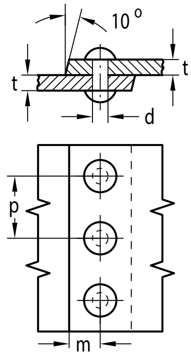
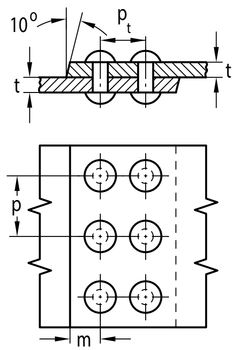
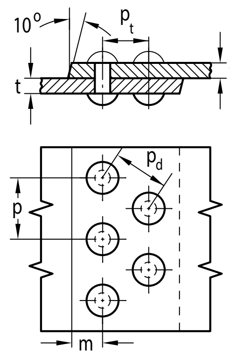
a = length of shank portion necessary to form the closing head (mm)

Note: Depending upon the shape of the head, the magnitude of 'a' varies from 0.7d to 1.3d

$$a = 0.7d \text{ to } 1.3d$$

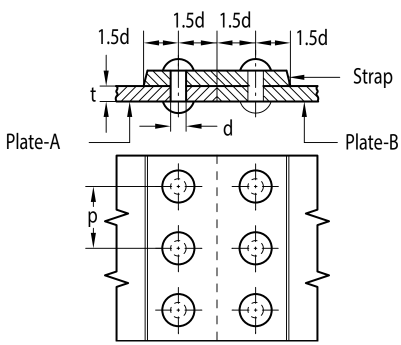
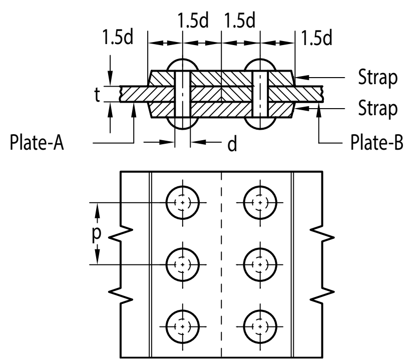
d = diameter of the shank of rivet (mm)

Table 8.38 *Types of lap joints*

Single-riveted Lap joint	Double-riveted Lap joint (Chain riveted joint)	Double-riveted Lap joint (Zigzag riveted joint)
		

Proportions	
For single-riveted Lap joint, $p = 3d$ $m = 1.5d$	p = pitch of rivet (mm) d = shank diameter of rivet (mm) m = margin from edge (mm)
For double-riveted Lap joint, $p = 3d$ $m = 1.5d$ $p_t = 0.8p$ (for chain riveting) $p_t = 0.6p$ (for zigzag riveting)	p_t = transverse pitch (mm) p_d = diagonal pitch (mm)

Table 8.39 *Types of butt joints*

Single-riveted butt joints	
Single-strap butt joint	*Double-strap butt joint*
	

(*Contd.*)

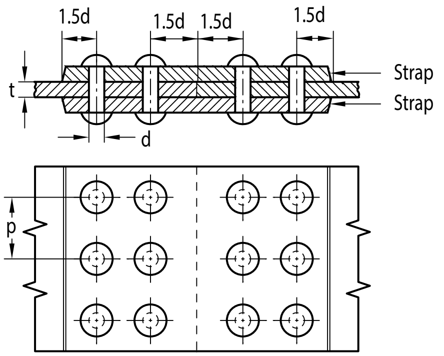
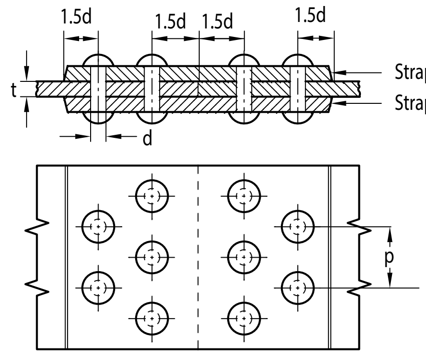
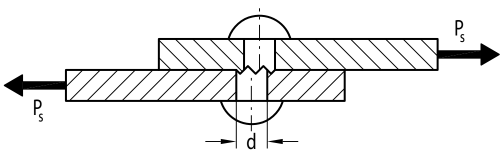
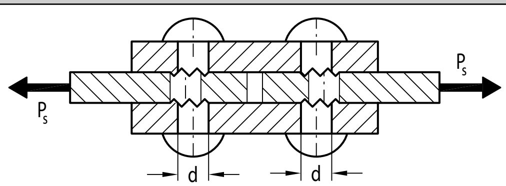
Double-riveted double-strap butt joints	
Chain pattern	*Zigzag pattern*
	

Table 8.40 *Strength equations of riveted joints*

Shear strength of single-riveted lap joint	
	For single-riveted lap joint, $$P_s = \frac{\pi}{4} d^2 \tau \qquad \text{(for single shear)} \qquad (8.26)$$ P_s = shear resistance of rivet per pitch length (N) d = shank diameter of rivet (mm) τ = permissible shear stress for rivet material (MPa or N/mm^2) For double or triple-riveted lap joint, $$P_s = \frac{\pi}{4} d^2 \tau n \qquad \text{(for single shear)} \qquad (8.27)$$ n = number of rivets per pitch length For double-riveted joint, $n = 2$ For triple-riveted joint, $n = 3$
Shear strength—Double-strap single-riveted butt joint	
	$$P_s = 2\left[\frac{\pi}{4} d^2 \tau n\right] \qquad \text{(for double shear)} \qquad (8.28)$$

(*Contd.*)

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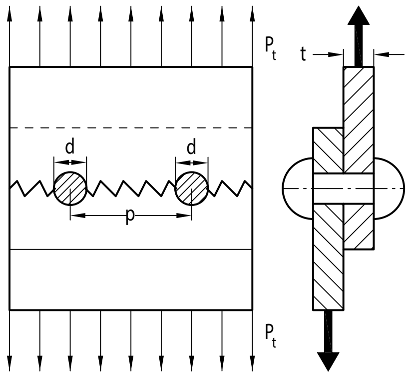
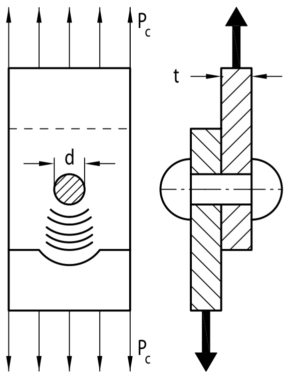
Tensile strength of plate between rivets	
	$$P_t = (p - d)t\sigma_t \quad (8.29)$$ P_t = tensile resistance of plate per pitch length (N) p = pitch of rivets (mm) t = thickness of plate (mm) σ_t = permissible tensile stress of plate material (MPa or N/mm^2)
Crushing strength of plate	
	$$P_c = dt\sigma_c n \quad (8.30)$$ P_c = crushing resistance of plate per pitch length (N) σ_c = permissible compressive stress of plate material (MPa or N/mm^2)
Efficiency of riveted joint	
$$\eta = \frac{\text{Lowest of } P_s, P_t, \text{and } P_c}{P}$$	$$P = pt\sigma_t \quad (8.31)$$

Note: (i) The strength of riveted joint is defined as the force that the joint can withstand without causing the failure.
(ii) The efficiency of riveted joint is defined as the ratio of the strength of riveted joint to the strength of un-riveted solid plate.

8.10 RIVETED JOINTS FOR BOILER SHELL

Table 8.41 *Efficiencies of riveted joints for boiler*

Type of joint	Efficiency (percent)
Lap joint	
Single-riveted	45–60
Double-riveted	63–70
Triple-riveted	72–80

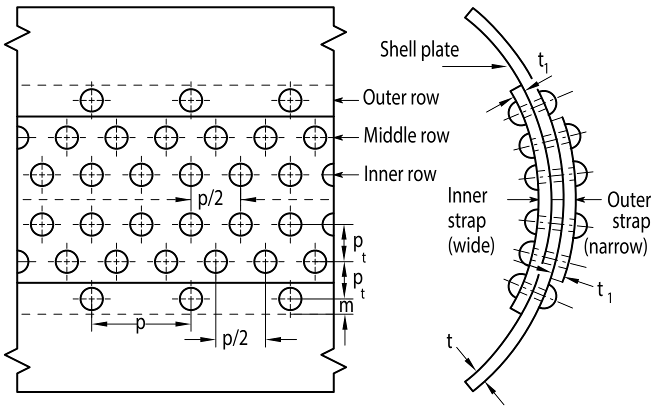
(*Contd.*)

(*Contd.*)

Double -strap butt joint	
Single-riveted	55–60
Double-riveted	70–83
Triple-riveted	80–90
Quadruple-riveted	85–94

Table 8.42 *Longitudinal butt joint for boiler shell*

Layout of longitudinal butt joint



Thickness of cylindrical boiler shell

$$t = \frac{P_i D_i}{2\sigma_t \eta} + CA \qquad (8.32)$$

t = thickness of cylinder wall (mm)
P_i = internal pressure (MPa or N/mm^2)
D_i = inner diameter of the cylinder (mm)
σ_t = permissible tensile stress for the cylinder material (MPa or N/mm^2)
η = efficiency of the riveted joint (in fraction) (Table 8.41)
CA = corrosion allowance (mm)
Note: The minimum corrosion allowance is 1.5 to 2 mm.

Permissible tensile stress

$$\sigma_t = \frac{S_{ut}}{(fs)} \qquad (8.33)$$

S_{ut} = ultimate tensile strength of the plate material (MPa or N/mm^2)
(fs) = factor of safety
Note: The factor of safety in boiler application varies from 4.5 to 4.75. It is safe practice to assume the factor of safety as 5.

Ultimate tensile strength of plate material

Grade	S_{ut} (MPa or N/mm^2)
St 37 BR	360–440
St 42 BR	410–500

(**Note:** There are two popular grades of steel used for boiler shell and boiler rivets. They are designated as Grade-St 37 BR and Grade-St 42 BR)

Diameter of rivet

(a) When the thickness of plate is more than 8 mm, the rivet diameter is calculated by following Unwin's formula,

$$d = 6\sqrt{t} \qquad (8.34)$$

where,
d = diameter of the rivet (mm)
t = thickness of the cylinder wall (mm)

(*Contd.*)

(*Contd.*)

(b) When the thickness of plate is less than 8 mm, the diameter of rivet is obtained by equating shear resistance of rivets to crushing resistance. (c) In no case, the diameter of rivet should be less than the plate thickness.	
Pitch of rivets	
The pitch of the rivets in outer row is maximum. The pitch of the rivets in middle and inner rows is one half of the pitch in outer row, that is, ($p/2$). The pitch of the rivets in outer row is obtained by equating the shear strength of the rivets to the tensile strength of the plate. $p = \frac{(n_1 + 1.875 n_2)\pi d^2 \tau}{4t\sigma_t} + d$	(8.35)
where, n_1 = number of rivets subjected to single shear per pitch length n_2 = number of rivets subjected to double shear per pitch length The pitch obtained by above equation has maximum and minimum limits. According to Boiler Regulations, (a) The pitch of the rivets should not be less than ($2d$) to enable the forming of the rivet head. $p_{min} = 2d$	(8.36)
(b) In order to provide leak proof joint, the maximum pitch is given by, $p_{max} = Ct + 41.28$	(8.37)
Refer to Table 8.44 for values of 'C'	
Transverse pitch (p_t)	
Case I In a lap or butt joint, in which there are more than one row of rivets and in which there is an equal number of rivets in each row, the minimum distance between the rows of rivets is given by (For zigzag riveting) $p_t = 0.33p + 0.67d$	(8.38)
(For chain riveting) $p_t = 2d$	(8.39)
Case II In joints, in which the number of rivets in outer row is one-half of the number of rivets in each of the inner rows and in which the inner rows are zigzag riveted, the minimum distance between the outer row and next row is given by $p_t = 0.2p + 1.15d$	(8.40)
The minimum distance between the rows in which there are full number of rivets is given by $p_t = 0.165p + 0.67d$	(8.41)
where p is the pitch of rivets in outer row.	
Margin (m)	
The distance between the center of rivet hole from the edge of the plate is called margin. The minimum margin is given by $m = 1.5d$	(8.42)
Thickness of straps (t_1)	
Case I When the straps are of unequal width and in which every alternate rivet in outer row is omitted, $t_1 = 0.75t$ (for wide strap)	(8.43)
$t_1 = 0.625t$ (for narrow strap)	(8.44)
Case II When the straps are of equal width and in which every alternate rivet in outer row is omitted, $t_1 = 0.625t \left[\frac{p-d}{p-2d}\right]$	(8.45)
where t_1 is the thickness of straps. The thickness of strap, in no case, shall be less than 10 mm.	

(*Contd.*)

(*Contd.*)

Permissible stresses
According to Clause 5 of Indian Boiler Regulations, the ultimate tensile strength and shear strength of steel plates and rivets are 26 and 21 tons per square inch respectively. Therefore, $$S_{ut} = \frac{26 \times 2240 \times 6890}{10^6} = 401.27 \text{ N/mm}^2$$ $$S_{us} = \frac{21 \times 2240 \times 6890}{10^6} = 324.11 \text{ N/mm}^2$$ Assuming a factor of safety of 5, the permissible tensile and shear stresses are given by, $$\sigma_t = \frac{S_{ut}}{(fs)} = \frac{401.27}{5} = 80.25 \text{ N/mm}^2$$ $$\tau = \frac{S_{us}}{(fs)} = \frac{324.11}{5} = 64.82 \text{ N/mm}^2$$ There is no provision for calculating permissible compressive stress in Boiler Regulations. Assuming, $$\sigma_c = 1.5\sigma_t$$ we get, $$\sigma_c = 1.5(80.25) = 120.38 \text{ N/mm}^2$$ Therefore, the permissible tensile, shear and compressive stresses are assumed as 80, 60 and 120 N/mm^2 respectively

Note: The procedure and equations given in above table are for the academic exercise of students. The practicing engineers should strictly follow Boiler Regulations and other Codes.

Table 8.43 *Relationship between rivet diameter and diameter of rivet hole for boiler*

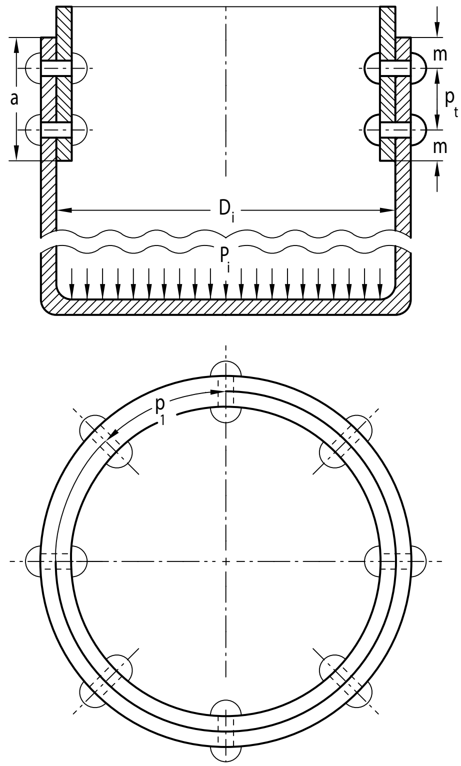
d (mm)	12	14	16	18	20	22	24	27	30	33	36	39	42	48
d' (mm)	13	15	17	19	21	23	25	28.5	31.5	34.5	37.5	41	44	50

d' = diameter of rivet hole (mm)

Table 8.44 *Values of C*

Number of rivets per pitch length	Values of C		
	Lap joint	Single-strap butt joint	Double-strap butt joint
1	1.31	1.53	1.75
2	2.62	3.06	3.5
3	3.47	4.05	4.63
4	4.17	—	5.52
5	—	—	6.0

Table 8.45 *Circumferential lap joint for boiler shell*



Thickness of cylindrical shell (t)	
$t = \dfrac{P_i D_i}{2\sigma_t \eta} + CA$	(8.46)
Diameter of rivets (d)	
(a) When the thickness of plate is more than 8 mm, the rivet diameter is calculated by Unwin's formula.	
$d = 6\sqrt{t}$	(8.47)
(b) When the thickness of plate is less than 8 mm, the diameter of rivet is obtained by equating shear resistance of rivets to crushing resistance.	
(c) In no case, the diameter of rivet should be less than the plate thickness.	
Number of rivets (n)	
$n = \left(\dfrac{D_i}{d}\right)^2 \dfrac{P_i}{\tau}$	(8.48)

(*Contd.*)

(*Contd.*)

Circumferential pitch of rivets (p_1)	
$$\eta_1 = \frac{p_1 - d}{p_1}$$	(8.49)
η_1 = efficiency of circumferential joint (in fraction) p_1 = pitch of circumferential joint (mm) The guidelines for the values of efficiency are as follows: (i) When there are number of circumferential joints in the shell, the efficiency of intermediate circumferential seams is taken as 62%. (ii) The efficiency of end circumferential joint is assumed to be 50% of that of the longitudinal joint, but in no case less than 42%. Therefore, For intermediate joints $\eta_1 = 0.62$ For end joints $\eta_1 = 0.5\eta$ or 0.42 (whichever is less)	
Number of rows	
$$n_1 = \frac{\pi(D_i + t)}{p_1}$$	(8.50)
n_1 = number of rivets in one row	
$$\text{number of rows} = \frac{n}{n_1}$$	(8.51)
Limiting values of pitch	
After determining the number of rows, the type of joint such as single-riveted lap joint or double-riveted lap joint is decided. The pitch is again readjusted. The pitch p_1 obtained by above procedure has minimum and maximum limits. They are as follows:	
$$p_{\min} = 2d$$	(8.52)
$$p_{\max} = Ct + 41.28$$	(8.53)
Refer to Table 8.44 for values of C.	
Transverse pitch (p_t)	
The overlap of the plate, denoted by a, is given by	
$$a = p_t + 2m$$	(8.54)
m = margin The number of rivets in each row is equal. (For zigzag riveting)	
$$p_t = 0.33p + 0.67d$$	(8.55)
(For chain riveting)	
$$p_t = 2d$$	(8.56)
The margin m is given by	
$$m = 1.5d$$	(8.57)

Note: (i) The notations used in above table are same as those given in Table 8.42. (ii) The procedure and equations given in the above table are for the academic exercise of students. The practicing engineers should strictly follow Boiler Regulations and other Codes.

8.11 RIVETS FOR STRUCTURAL APPLICATIONS

Table 8.46 *Permissible stresses for rivets*

Type of stresses and method of manufacture of riveted joints	Permissible stresses (MPa or N/mm^2)
Axial tensile stress on gross area of rivet:	
(i) power-driven rivets	100
(ii) hand-driven rivets	80
Shear stress on gross area of rivet:	
(i) power-driven rivets	100
(ii) hand-driven rivets	80
Bearing stress on gross area of rivet:	
(i) power-driven rivets	300
(ii) hand-driven rivets	250

Table 8.47 *Minimum edge distance of holes (Margin)*

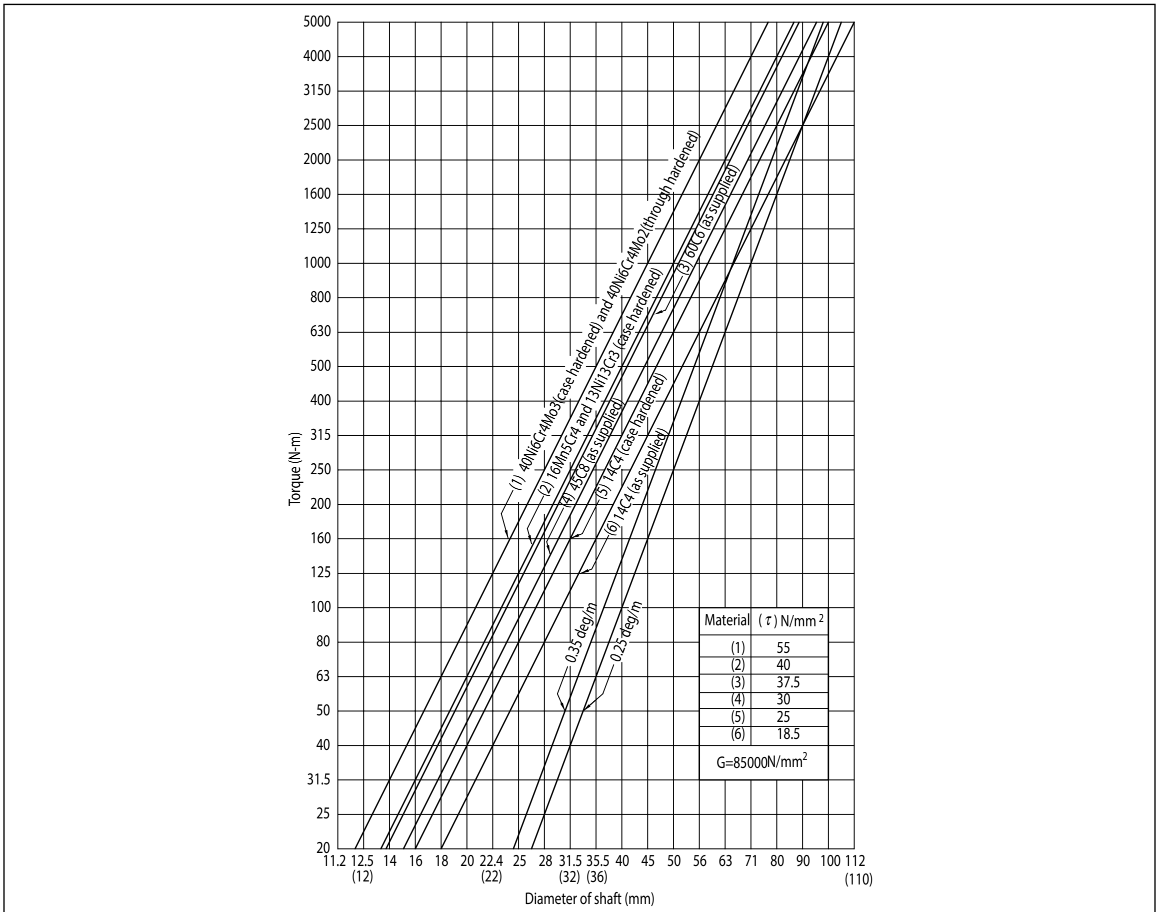
Gross diameter of rivet (mm)	Edge distance of holes (mm)	
	For sheared or hand flange cut edges	For machine flange cut or planed edges
13.5 and below	19	17
15.5	25	22
17.5	29	25
19.5	32	29
21.5	32	29
23.5	38	32
25.5	44	38
29	51	44
32	57	51
35	57	51

Shafts, Keys and Couplings

9

9.1 DIMENSIONS OF TRANSMISSION SHAFTS

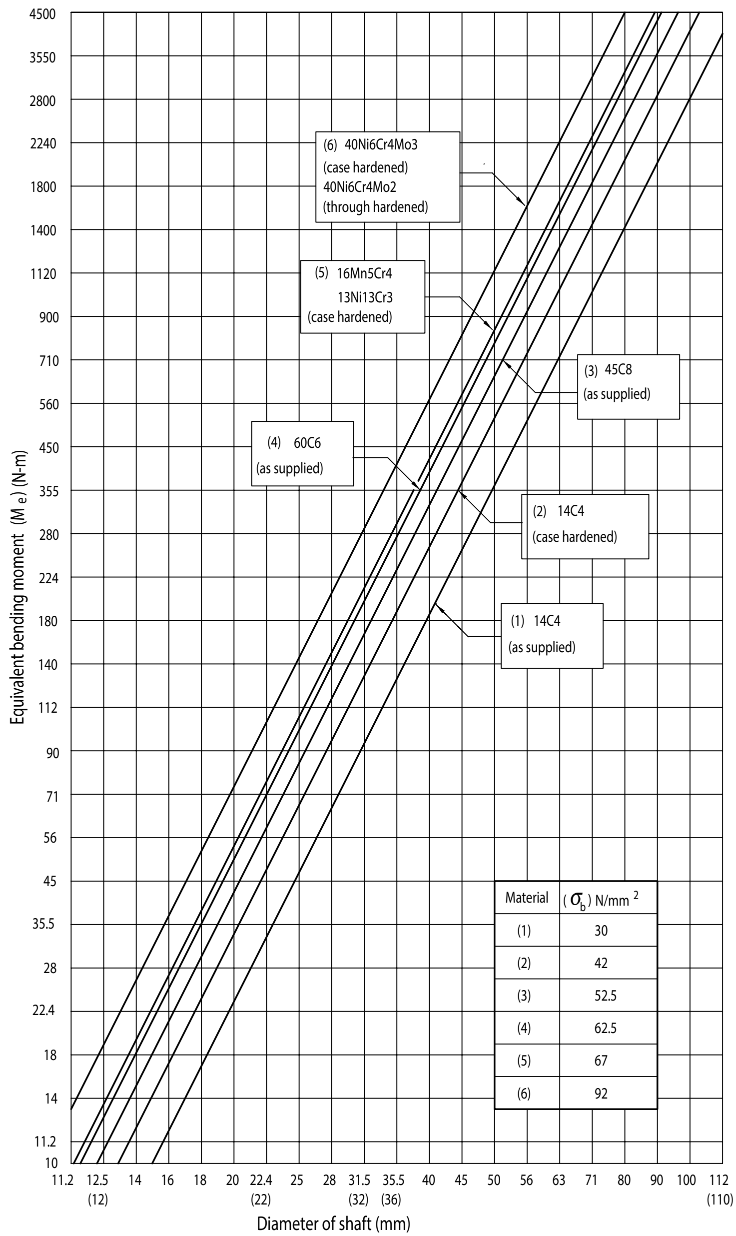
Table 9.1 *Diameter of shaft subjected to torsional moment*



The design of shaft is based on following relationship:

$$d^3 = \frac{16M_t}{\pi(\tau)} \text{ and } d^4 = \frac{584M_t l}{G\theta}$$

Table 9.2 *Diameter of shaft subjected to bending and torsional moments*



The design of shaft is based on following relationship:

$$d^3 = \frac{32M_e}{\pi(\sigma_b)} \quad \text{and} \quad M_e = [0.35M_b + 0.65\sqrt{(M_b)^2 + (M_t)^2}]$$

Table 9.3 *Dimensions of round steel bars for structural and general engineering purposes (hot-rolled)*

Designation	Diameter (mm)	Sectional area (cm^2)	Mass per metre length (kg)
ISRO 5	5	0.196	0.154
ISRO 6	6	0.283	0.222
ISRO 8	8	0.503	0.395
ISRO 10	10	0.785	0.617
ISRO 12	12	1.13	0.888
ISRO 14	14	1.54	1.21
ISRO 16	16	2.01	1.58
ISRO 18	18	2.54	2.00
ISRO 20	20	3.14	2.47
ISRO 22	22	3.80	2.98
ISRO 25	25	4.91	3.85
ISRO 28	28	6.16	4.83
ISRO 30	30	7.07	5.55
ISRO 32	32	8.04	6.31
ISRO 35	35	9.62	7.55
ISRO 40	40	12.6	9.85
ISRO 45	45	15.9	12.5
ISRO 50	50	19.6	15.4
ISRO 55	55	23.8	18.7
ISRO 60	60	28.3	22.2
ISRO 65	65	33.2	26.0
ISRO 70	70	38.5	30.2
ISRO 75	75	44.2	34.7
ISRO 80	80	50.3	39.5
ISRO 90	90	63.6	49.9
ISRO 100	100	78.5	61.7
ISRO 110	110	95.0	74.6
ISRO 120	120	113	88.8
ISRO 140	140	154	121
ISRO 160	160	201	158
ISRO 180	180	254	200
ISRO 200	200	314	247

Table 9.4 *Dimensions of square steel bars for structural and general engineering purposesa (hot-rolled)*

Designation	Side width (mm)	Sectional area (cm^2)	Mass per metre length (kg)
ISSQ 5	5	0.25	0.196
ISSQ 6	6	0.36	0.283
ISSQ 8	8	0.64	0.502

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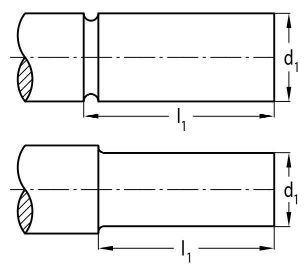
ISSQ 10	10	1.00	0.785
ISSQ 12	12	1.44	1.13
ISSQ 14	14	1.96	1.54
ISSQ 16	16	2.56	2.01
ISSQ 18	18	3.24	2.54
ISSQ 20	20	4.00	3.14
ISSQ 22	22	4.84	3.80
ISSQ 25	25	6.25	4.91
ISSQ 30	30	9.00	7.06
ISSQ 35	35	12.2	9.58
ISSQ 40	40	16.0	12.6
ISSQ 50	50	25.0	19.6
ISSQ 60	60	36.0	28.3
ISSQ 70	70	49.0	38.5
ISSQ 80	80	64.0	50.2
ISSQ 100	100	100	78.5
ISSQ 120	120	144	113

Table 9.5 *Dimensions of cylindrical shaft ends*

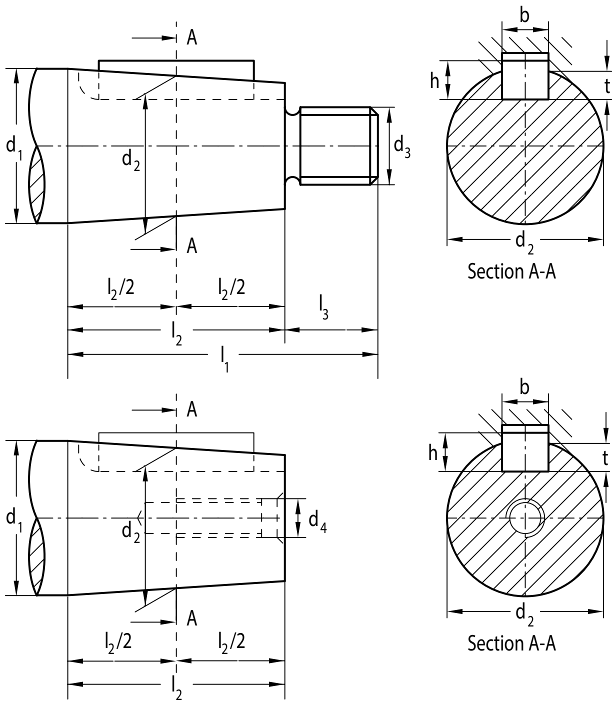


Diameter (d_1) (mm)	**Length (l_1) (mm)**		**Diameter (d_1) (mm)**	**Length (l_1) (mm)**	
	Long series	**Short series**		**Long series**	**Short series**
6, 7	16	–	80, 85, 90, 95	170	130
8, 9	20	–	100, 110, 120, 125	210	165
10, 11	23	20	130, 140, 150	250	200
12, 14	30	25	160, 170, 180	300	240
16, 18, 19	40	28	190, 200, 220	350	280
20, 22, 24	50	36	240, 250, 260	410	330
25, 28	60	42	280, 300, 320	470	380
30, 32, 35, 38	80	58	340, 360, 380	550	450
40, 42, 45, 48, 50, 55, 56	110	82	400, 420, 440, 450, 460, 480. 500	650	540
60, 63, 65, 70, 71, 75	140	105	530, 560, 600, 630	800	680

Table 9.6 *Tolerances for cylindrical shaft ends*

Diameter (d_1) (mm)	Tolerance
6, 7, 8, 9, 10, 11, 12, 14, 16, 18, 19, 20, 22, 24, 25, 28, 30	j6
32, 35, 38, 40, 42, 45, 48, 50	k6
55, 56, 60, 63, 65, 70, 71, 75, 80, 85, 90, 95, 100, 110, 120, 125, 130, 140, 150, 160, 170, 180, 190, 200, 220, 240, 250, 260, 280, 300, 320, 340, 360, 380, 400, 420, 440, 450, 460, 480, 500, 530, 560, 600, 630	m6

Table 9.7 *Dimensions of 1/10 conical shaft ends–Short series*



Diameter d_1	Length			Key and keyway			External thread d_3	Internal thread d_4
	l_1	l_2	l_3	d_2	$b \times h$	t		
16	28	16	12	15.2	3 × 3	1.8	M10 × 1.25	M4
18	28	16	12	17.2	4 × 4	2.5	M10 × 1.25	M5
19	28	16	12	18.2	4 × 4	2.5	M10 × 1.25	M5
20	36	22	14	18.9	4 × 4	2.5	M12 × 1.25	M6
22	36	22	14	20.9	4 × 4	2.5	M12 × 1.25	M6
24	36	22	14	22.9	5 × 5	3	M12 × 1.25	M6

(*Contd.*)

(*Contd.*)

25	42	24	18	23.8	5 × 5	3	M16 ×1.25	M8
28	42	24	18	26.8	5 × 5	3	M16 × 1.25	M8
30	58	36	22	28.2	5 × 5	3	M20 × 1.5	M10
32	58	36	22	30.2	6 × 6	3.5	M20 × 1.5	M10
35	58	36	22	33.2	6 × 6	3.5	M20 × 1.5	M10
38	58	36	22	36.2	6 × 6	3.5	M24 × 2	M12
40	82	54	28	37.3	10 × 8	5	M24 × 2	M12
42	82	54	28	39.3	10 × 8	5	M24 × 2	M12
45	82	54	28	42.3	12 × 8	5	M30 × 2	M16
48	82	54	28	45.3	12 × 8	5	M30 × 2	M16
50	82	54	28	47.3	12 × 8	5	M36 × 3	M16
55	82	54	28	52.3	14 × 9	5.5	M36 × 3	M20
56	82	54	28	53.3	14 × 9	5.5	M36 × 3	M20
60	105	70	35	56.5	16 × 10	6	M42 × 3	M20
63	105	70	35	59.5	16 × 10	6	M42 × 3	M20
65	105	70	35	61.5	16 × 10	6	M42 × 3	M20
70	105	70	35	66.5	18 × 11	7	M48 × 3	M24
71	105	70	35	67.5	18 × 11	7	M48 × 3	M24
75	105	70	35	71.5	18 ×11	7	M48 × 3	M24
80	130	90	40	75.5	20 × 12	7.5	M56 × 4	M30
85	130	90	40	80.5	20 × 12	7.5	M56 × 4	M30
90	130	90	40	85.5	22 × 14	9	M64 × 4	M30
95	130	90	40	90.5	22 × 14	9	M64 × 4	M36
100	165	120	45	94	25 × 14	9	M72 × 4	M36
110	165	120	45	104	25 × 14	9	M80 × 4	M42
120	165	120	45	114	28 × 16	10	M90 × 4	M42
125	165	120	45	119	28 × 16	10	M90 × 4	M48
130	200	150	50	122.5	28 × 16	10	M100 × 4	–
140	200	150	50	132.5	32 × 18	11	M100 × 4	–
150	200	150	50	142.5	32 × 18	11	M110 × 4	–
160	240	180	60	151	36 × 20	12	M125 × 4	–
170	240	180	60	161	36 × 20	12	M125 × 4	–
180	240	180	60	171	40 × 22	13	M140 × 6	–
190	280	210	70	179.5	40 × 22	13	M140 × 6	–
200	280	210	70	189.5	40 ×22	13	M160 × 6	–
220	280	210	70	209.5	45 × 25	15	M160 × 6	–

Note: All dimensions are in mm.

9.2 STRESS EQUATIONS OF TRANSMISSION SHAFTS

Table 9.8 *Stresses and angle of twist for solid shafts*

Maximum principal stress theory of failure	
$\sigma_1 = \dfrac{16}{\pi d^3}\,[M_b + \sqrt{(M_b)^2 + (M_t)^2}\,]$ (9.1) $\sigma_1 = \dfrac{S_{yt}}{(fs)}$ (9.2)	σ_1 = maximum principal stress (MPa or N/mm^2) d = diameter of shaft (mm) M_b = bending moment acting on shaft (N-mm) M_t = torsional moment (torque) acting on shaft (N-mm) S_{yt} = yield strength of shaft material (MPa or N/mm^2) (fs) = factor of safety
Maximum shear stress theory of failure	
$\tau_{max} = \dfrac{16}{\pi d^3}\sqrt{(M_b)^2 + (M_t)^2}$ (9.3) $S_{sy} = 0.5\, S_{yt}$ (9.4) $\tau_{max} = \dfrac{S_{sy}}{(fs)} = \dfrac{0.5\,S_{yt}}{(fs)}$ (9.5)	τ_{max} = maximum shear stress (MPa or N/mm^2) S_{sy} = yield strength of shaft material in shear (MPa or N/mm^2)
Equivalent torsional moment (torque)	
$(M_t)_e = \sqrt{(M_b)^2 + (M_t)^2}$ (9.6)	$(M_t)_e$ = equivalent torsional moment (torque) (N-mm)
Equivalent bending moment	
$(M_b)_e = [M_b + \sqrt{(M_b)^2 + (M_t)^2}\,]$ (9.7)	$(M_b)_e$ = equivalent bending moment (N-mm)
Angle of twist	
$\theta = \dfrac{584 M_t l}{G d^4}$ (9.8)	θ = angle of twist (°) l = length of shaft subjected to twisting moment (mm) G = modulus of rigidity (N/mm^2) **Note:** The permissible angle of twist is 0.25° per metre length for machine tool applications and 3° per metre length for line shafts. For steel shafts, G = 79300 N/mm^2.
ASME code	
$\tau_{max} = \dfrac{16}{\pi d^3}\sqrt{(k_b M_b)^2 + (k_t M_t)^2}$ (9.9)	or $\tau_{max} = 0.30\, S_{yt}$ $\tau_{max} = 0.18\, S_{ut}$ (9.10) (whichever is minimum) If keyways are present, the above values are to be reduced by 25 percent.
k_b = combined shock and fatigue factor applied to bending moment	k_t = combined shock and fatigue factor applied to torsional moment (torque)

(*Contd.*)

(*Contd.*)

Values of k_b and k_t		
Application	**k_b**	**k_t**
(i) load gradually applied	1.5	1.0
(ii) load suddenly applied (minor shock)	1.5 – 2.0	1.0 – 1.5
(iii) load suddenly applied (heavy shock)	2.0 – 3.0	1.5 – 3.0

Table 9.9 *Stresses and angle of twist for hollow shafts*

Maximum principal stress theory of failure

$$\sigma_1 = \frac{16}{\pi d_o^3(1-C^4)}\left[M_b + \sqrt{(M_b)^2 + (M_t)^2}\right] \tag{9.11}$$

$$\sigma_1 = \frac{S_{yt}}{(fs)} \quad \text{and} \quad \frac{d_i}{d_o} = C \tag{9.12}$$

d_o = outside diameter of hollow shaft (mm) d_i = inside diameter of hollow shaft (mm) C = ratio of inside diameter to outside diameter	σ_1 = maximum principal stress (MPa or N/mm^2) M_b = bending moment acting on hollow shaft (N-mm) M_t = torsional moment (torque) acting on hollow shaft (N-mm) S_{yt} = yield strength of shaft material (MPa or N/mm^2) (fs) = factor of safety

Maximum shear stress theory of failure

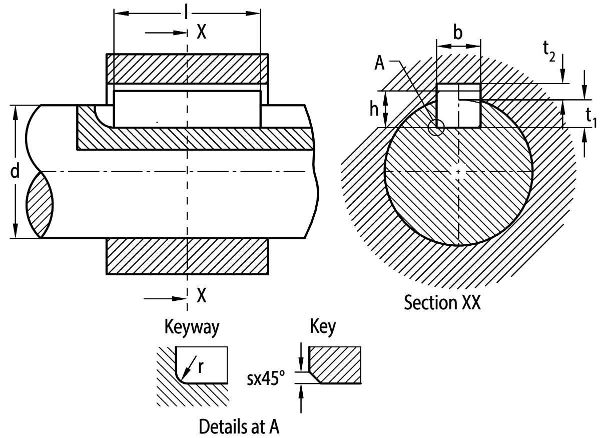
$\tau_{max} = \frac{16}{\pi d_o^3(1-C^4)}\sqrt{[(M_b)^2 + (M_t)^2]}$ (9.13) $\tau_{max} = \frac{S_{sy}}{(fs)} = \frac{0.5 S_{yt}}{(fs)}$ (9.14)	τ_{max} = maximum shear stress (MPa or N/mm^2) S_{sy} = yield strength of shaft material in shear (MPa or N/mm^2)

Angle of twist

$\theta = \frac{584 M_t l}{G d_o^4(1-C^4)}$ (9.15)	θ = angle of twist (°) l = length of hollow shaft subjected to twisting moment (torque) (mm) G = modulus of rigidity (N/mm^2)

9.3 KEYS AND KEYWAYS

Table 9.10 *Dimensions of parallel keys and keyways*



Shaft diameter		Key dimensions		Keyway depth		Keyway radius (r)		Chamfer of key (s)		Key length	
Above	Up to	b	h	In shaft (t_1)	In hub (t_2)	Min.	Max.	Min.	Max.	Min.	Max.
6	8	2	2	1.2	1.0	0.08	0.16	0.16	0.25	6	20
8	10	3	3	1.8	1.4	0.08	0.16	0.16	0.25	6	36
10	12	4	4	2.5	1.8	0.08	0.16	0.16	0.25	8	45
12	17	5	5	3.0	2.3	0.16	0.25	0.25	0.40	10	56
17	22	6	6	3.5	2.8	0.16	0.25	0.25	0.40	14	70
22	30	8	7	4.0	3.3	0.16	0.25	0.25	0.40	18	90
30	38	10	8	5.0	3.3	0.25	0.40	0.40	0.60	22	110
38	44	12	8	5.0	3.3	0.25	0.40	0.40	0.60	28	140
44	50	14	9	5.5	3.8	0.25	0.40	0.40	0.60	36	160
50	58	16	10	6.0	4.3	0.25	0.40	0.40	0.60	45	180
58	65	18	11	7.0	4.4	0.25	0.40	0.40	0.60	50	200
65	75	20	12	7.5	4.9	0.40	0.60	0.60	0.80	56	220
75	85	22	14	9.0	5.4	0.40	0.60	0.60	0.80	63	250
85	95	25	14	9.0	5.4	0.40	0.60	0.60	0.80	70	280
95	110	28	16	10.0	6.4	0.40	0.60	0.60	0.80	80	320
110	130	32	18	11.0	7.4	0.40	0.60	0.60	0.80	90	360
130	150	36	20	12.0	8.4	0.70	1.00	1.00	1.20	100	400
150	170	40	22	13.0	9.4	0.70	1.00	1.00	1.20	110	400
170	200	45	25	15.0	10.4	0.70	1.00	1.00	1.20	125	400
200	230	50	28	17.0	11.4	0.70	1.00	1.00	1.20	140	400
230	260	56	32	20.0	12.4	1.20	1.60	1.60	2.00	160	400
260	290	63	32	20.0	12.4	1.20	1.60	1.60	2.00	180	400

(*Contd.*)

(*Contd.*)

290	330	70	36	22.0	14.4	1.20	1.60	1.60	2.00	200	400
330	380	80	40	25.0	15.4	2.00	2.50	2.50	3.00	220	400
380	440	90	45	28.0	17.4	2.00	2.50	2.50	3.00	250	400
440	500	100	50	31.0	19.5	2.00	2.50	2.50	3.00	280	400

Note: (i) All dimensions are in mm.
(ii) The material for the key is steel with minimum tensile strength of 600 MPa.
Designation – A parallel key of width 12 mm, height 8 mm and length 50 mm is designated as Parallel key 12 × 8 × 50.

Table 9.11 *Preferred lengths for parallel keys (2 × 2 to 20 × 12 mm)*

Preferred length (*l*) (mm)	Key dimensions ($b \times h$)											
	2 × 2	3 × 3	4 × 4	5 × 5	6 × 6	8 × 7	10 × 8	12 × 8	14 × 9	16 × 10	18 × 11	20 × 12
6	X	X										
8	X	X	X									
10	X	X	X									
12	X	X	X	X								
14	X	X	X	X	X							
16	X	X	X	X	X							
18	X	X	X	X	X	X						
20	X	X	X	X	X	X						
22		X	X	X	X	X	X					
25		X	X	X	X	X	X					
28		X	X	X	X	X	X	X				
32		X	X	X	X	X	X	X				
36		X	X	X	X	X	X	X	X			
40			X	X	X	X	X	X	X			
45			X	X	X	X	X	X	X	X		
50				X	X	X	X	X	X	X	X	
56				X	X	X	X	X	X	X	X	X
63					X	X	X	X	X	X	X	X
70					X	X	X	X	X	X	X	X
80						X	X	X	X	X	X	X
90						X	X	X	X	X	X	X
100							X	X	X	X	X	X
110							X	X	X	X	X	X
125								X	X	X	X	X
140								X	X	X	X	X
160									X	X	X	X
180										X	X	X
200											X	X
220												X

Note: (i) The preferred lengths are indicated by X.
(ii) Assume density of steel as 7.85 g/cm^3 for calculating the weight of key.

Table 9.12 *Preferred lengths for parallel keys (22 × 14 to 56 × 32 mm)*

Preferred length (l) (mm)	Key dimensions ($b \times h$)								
	22 × 14	25 × 14	28 × 16	32 × 18	36 × 20	40 × 22	45 × 25	50 × 28	56 × 32
63	X								
70	X	X							
80	X	X	X						
90	X	X	X	X					
100	X	X	X	X	X				
110	X	X	X	X	X	X			
125	X	X	X	X	X	X	X		
140	X	X	X	X	X	X	X	X	
160	X	X	X	X	X	X	X	X	X
180	X	X	X	X	X	X	X	X	X
200	X	X	X	X	X	X	X	X	X
220	X	X	X	X	X	X	X	X	X
250	X	X	X	X	X	X	X	X	X
280		X	X	X	X	X	X	X	X
320			X	X	X	X	X	X	X
360				X	X	X	X	X	X
400					X	X	X	X	X

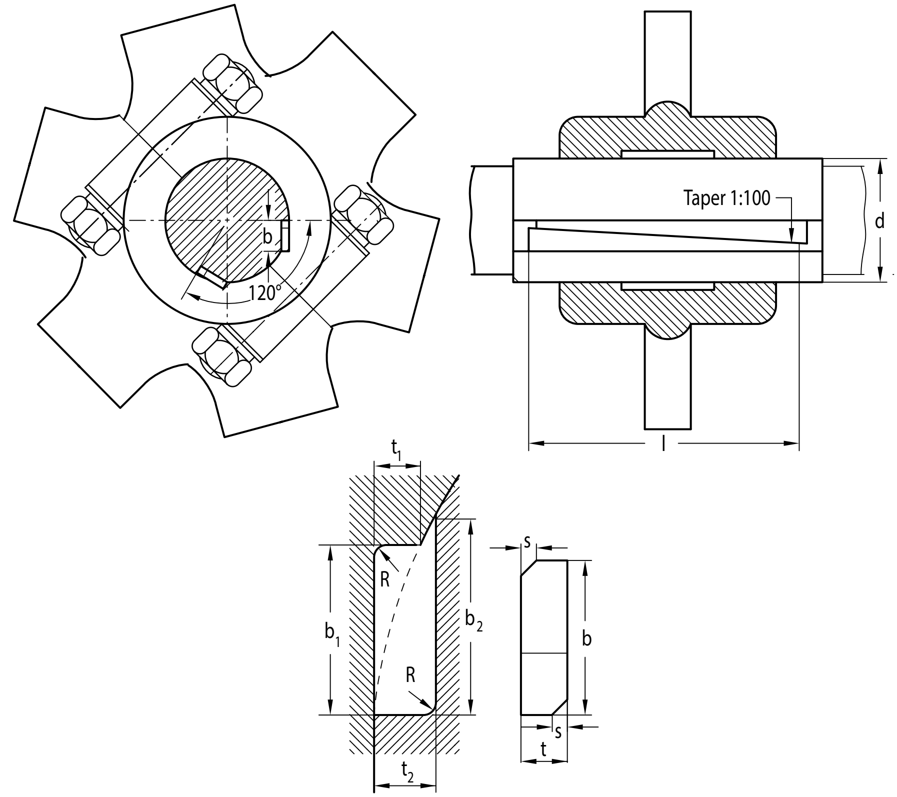
Note: The preferred lengths are indicated by X.

Table 9.13 *Preferred lengths for parallel keys (63 × 32 to 100 × 50 mm)*

Preferred length (l) (mm)	Key dimensions ($b \times h$)				
	63 × 32	70 × 36	80 × 40	90 × 45	100 × 50
180	X				
200	X	X			
220	X	X	X		
250	X	X	X	X	
280	X	X	X	X	X
320	X	X	X	X	X
360	X	X	X	X	X
400	X	X	X	X	X

Note: The preferred lengths are indicated by X.

Table 9.14 *Dimensions of tangential keys and keyways*



Shaft dia. (d)	Key				Keyway					
	Thickness (t)	Calcu-lated width (b)	Chamfer (s)		Depth		Calculated width		Radius (R)	
			Min.	Max.	In hub (t_1)	In shaft (t_2)	In hub (b_1)	In shaft (b_2)	Max.	Min.
60	7	19.3	0.6	0.8	7	7.3	19.3	19.6	0.6	0.4
63	7	19.8	0.6	0.8	7	7.3	19.8	20.2	0.6	0.4
65	7	20.1	0.6	0.8	7	7.3	21.1	20.5	0.6	0.4
70	7	21.0	0.6	0.8	7	7.3	21.0	21.4	0.6	0.4
71	8	22.5	0.6	0.8	8	8.3	22.5	22.8	0.6	0.4
75	8	23.2	0.6	0.8	8	8.3	23.2	23.5	0.6	0.4
80	8	24.0	0.6	0.8	8	8.3	24.0	24.4	0.6	0.4
85	8	24.8	0.6	0.8	8	8.3	24.8	25.2	0.6	0.4
90	8	25.6	0.6	0.8	8	8.3	25.6	26.0	0.6	0.4

(*Contd.*)

(*Contd.*)

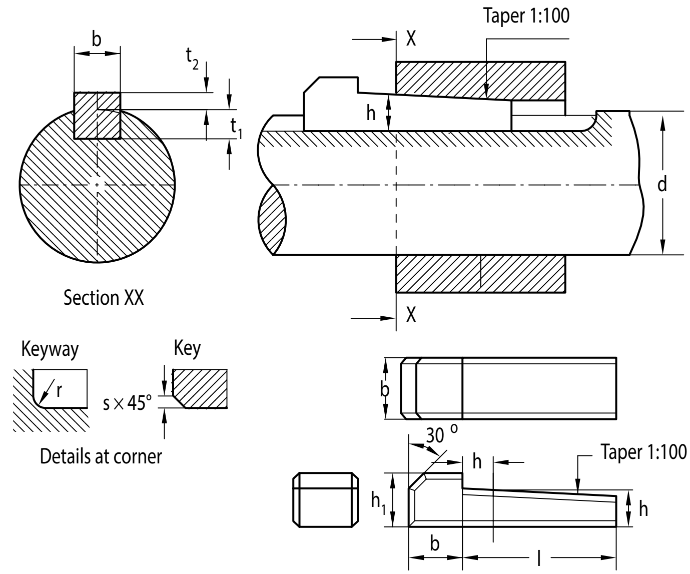
95	9	27.8	0.6	0.8	9	9.3	27.8	28.2	0.6	0.4
100	9	28.6	0.6	0.8	9	9.3	28.6	29.0	0.6	0.4
110	9	30.1	0.6	0.8	9	9.3	30.1	30.6	0.6	0.4
120	10	33.2	1.0	1.2	10	10.3	33.2	33.6	1.0	0.7
125	10	33.9	1.0	1.2	10	10.3	33.9	34.4	1.0	0.7
130	10	34.6	1.0	1.2	10	10.3	34.6	35.1	1.0	0.7
140	11	37.7	1.0	1.2	11	11.4	37.7	38.3	1.0	0.7
150	11	39.1	1.0	1.2	11	11.4	39.1	39.7	1.0	0.7
160	12	42.1	1.0	1.2	12	12.4	42.1	42.8	1.0	0.7
170	12	43.5	1.0	1.2	12	12.4	43.5	44.2	1.0	0.7
180	12	44.9	1.0	1.2	12	12.4	44.9	45.9	1.0	0.7
190	14	49.6	1.0	1.2	14	14.4	49.6	50.3	1.0	0.7
200	14	51.0	1.0	1.2	14	14.4	51.0	51.7	1.0	0.7
220	16	57.1	1.6	2.0	16	16.4	57.1	57.8	1.6	1.2
240	16	59.9	1.6	2.0	16	16.4	59.9	60.6	1.6	1.2
250	18	64.6	1.6	2.0	18	18.4	64.6	65.3	1.6	1.2
260	18	66.0	1.6	2.0	18	18.4	66.0	66.7	1.6	1.2
280	20	72.1	2.5	3.0	20	20.4	72.1	72.8	2.5	2.0
300	20	74.8	2.5	3.0	20	20.4	74.8	75.5	2.5	2.0
320	22	81.0	2.5	3.0	22	22.4	81.0	81.6	2.5	2.0
340	22	83.6	2.5	3.0	22	22.4	83.6	84.3	2.5	2.0
360	26	93.2	2.5	3.0	26	26.4	93.2	93.8	2.5	2.0
380	26	95.9	2.5	3.0	26	26.4	95.9	96.6	2.5	2.0
400	26	98.6	2.5	3.0	26	26.4	98.6	99.3	2.5	2.0
420	30	108.2	3.0	4.0	30	30.4	108.2	108.8	3.0	2.5
440	30	110.9	3.0	4.0	30	30.4	110.9	111.6	3.0	2.5
450	30	112.3	3.0	4.0	30	30.4	112.3	112.9	3.0	2.5
460	30	113.5	3.0	4.0	30	30.4	113.6	114.3	3.0	2.5
480	34	123.1	3.0	4.0	34	34.4	123.1	123.8	3.0	2.5
500	34	125.9	3.0	4.0	34	34.4	125.9	126.6	3.0	2.5

(ISO: 3117)

Note: (i) All dimensions are in mm.
(ii) The length *l* is taken as 10 to 15 per cent greater than the length of the hub.
(iii) The material for the key is steel with minimum tensile strength of 600 MPa.

Designation – A tangential key with thickness 8 mm, width 24 mm and length 100 mm is designated as Tangential key 8 × 24 × 100.

Table 9.15 *Dimensions of gib-head keys and keyways*



Shaft diameter		Key dimensions		Keyway depth		Keyway radius (r)		Key chamfer (s)		Key length		h_1
Above	Up to	b	h	In shaft (t_1)	In hub (t_2)	Min.	Max.	Min.	Max.	Min.	Max.	
10	12	4	4	2.5	1.2	0.08	0.16	0.16	0.25	14	45	7
12	17	5	5	3.0	1.7	0.16	0.25	0.25	0.40	14	56	8
17	22	6	6	3.5	2.2	0.16	0.25	0.25	0.40	16	70	10
22	30	8	7	4.0	2.4	0.16	0.25	0.25	0.40	20	90	11
30	38	10	8	5.0	2.4	0.25	0.40	0.40	0.60	25	110	12
38	44	12	8	5.0	2.4	0.25	0.40	0.40	0.60	32	140	12
44	50	14	9	5.5	2.9	0.25	0.40	0.40	0.60	40	160	14
50	58	16	10	6.0	3.4	0.25	0.40	0.40	0.60	45	180	16
58	65	18	11	7.0	3.4	0.25	0.40	0.40	0.60	50	200	18
65	75	20	12	7.5	3.9	0.40	0.60	0.60	0.80	56	220	20
75	85	22	14	9.0	4.4	0.40	0.60	0.60	0.80	63	250	22
85	95	25	14	9.0	4.4	0.40	0.60	0.60	0.80	70	280	22
95	110	28	16	10.0	5.4	0.40	0.60	0.60	0.80	80	320	25
110	130	32	18	11.0	6.4	0.40	0.60	0.60	0.80	90	360	28
130	150	36	20	12.0	7.1	0.70	1.00	1.00	1.20	100	400	32
150	170	40	22	13.0	8.1	0.70	1.00	1.00	1.20	110	400	36
170	200	45	25	15.0	9.1	0.70	1.00	1.00	1.20	125	400	40
200	230	50	28	17.0	10.1	0.70	1.00	1.00	1.20	140	400	45
230	260	56	32	20.0	11.1	1.20	1.60	1.60	2.00	–	–	50

(*Contd.*)

(*Contd.*)

260	290	63	32	20.0	11.1	1.20	1.60	1.60	2.00	–	–	50
290	330	70	36	22.0	13.1	1.20	1.60	1.60	2.00	–	–	56
330	380	80	40	25.0	14.1	2.00	2.50	2.50	3.00	–	–	63
380	440	90	45	28.0	16.1	2.00	2.50	2.50	3.00	–	–	70
440	500	100	50	31.0	18.1	2.00	2.50	2.50	3.00	–	–	80

Note: (i) All dimensions are in mm.
(ii) The material for the key is steel with minimum tensile strength of 600 MPa.
Designation – A gib-head key of width 12 mm, height 8 mm and length 50 mm is designated as Gib-head key 12 × 8 × 50.

Table 9.16 *Preferred lengths for gib-head keys (4 × 4 to 25 × 14 mm)*

Preferred length (*l*) (mm)	Key dimensions (*b* × *h*)											
	4 × 4	5 × 5	6 × 6	8 × 7	10 × 8	12 × 8	14 × 9	16 × 10	18 × 11	20 × 12	22 × 14	25 × 14
14	X	X										
16	X	X	X									
18	X	X	X									
20	X	X	X	X								
22	X	X	X	X								
25	X	X	X	X	X							
28	X	X	X	X	X							
32	X	X	X	X	X	X						
36	X	X	X	X	X	X						
40	X	X	X	X	X	X	X					
45	X	X	X	X	X	X	X	X				
50		X	X	X	X	X	X	X	X			
56		X	X	X	X	X	X	X	X	X		
63			X	X	X	X	X	X	X	X	X	
70			X	X	X	X	X	X	X	X	X	X
80				X	X	X	X	X	X	X	X	X
90				X	X	X	X	X	X	X	X	X
100					X	X	X	X	X	X	X	X
110					X	X	X	X	X	X	X	X
125						X	X	X	X	X	X	X
140						X	X	X	X	X	X	X
160							X	X	X	X	X	X
180								X	X	X	X	X
200									X	X	X	X
220										X	X	X
250											X	X
280												X

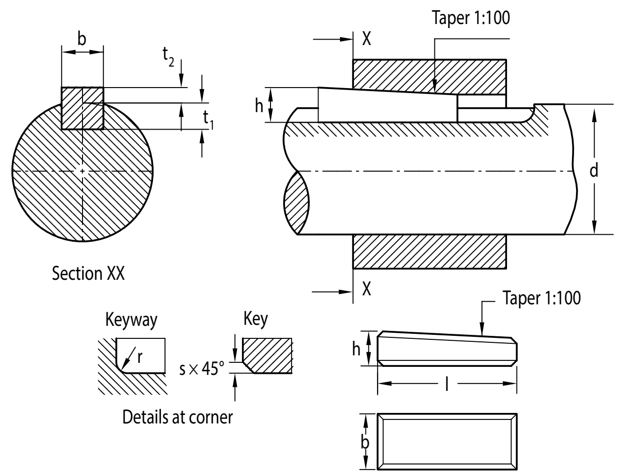
Note: (i) The preferred lengths are indicated by X.
(ii) All dimensions are in mm.

Table 9.17 *Preferred lengths for gib-head keys (28 × 16 to 50 × 28 mm)*

Preferred length (l) (mm)	Key dimensions ($b \times h$)					
	28 × 16	32 × 18	36 × 20	40 × 22	45 × 25	50 × 28
80	X					
90	X	X				
100	X	X	X			
110	X	X	X	X		
125	X	X	X	X		
140	X	X	X	X	X	X
160	X	X	X	X	X	X
180	X	X	X	X	X	X
200	X	X	X	X	X	X
220	X	X	X	X	X	X
250	X	X	X	X	X	X
280	X	X	X	X	X	X
320	X	X	X	X	X	X
360		X	X	X	X	X
400			X	X	X	X

Note: (i) The preferred lengths are indicated by X.
(ii) All dimensions are in mm.

Table 9.18 *Dimensions of taper keys and keyways*



Shaft diameter		Key dimensions		Keyway depth		Keyway radius (r)		Chamfer of key (s)		Key length	
Above	Up to	b	h	In shaft (t_1)	In hub (t_2)	Min.	Max.	Min.	Max.	Min.	Max.
6	8	2	2	1.2	0.5	0.08	0.16	0.16	0.25	6	20
8	10	3	3	1.8	0.9	0.08	0.16	0.16	0.25	8	36

(*Contd.*)

(*Contd.*)

10	12	4	4	2.5	1.2	0.08	0.16	0.16	0.25	10	45
12	17	5	5	3.0	1.7	0.16	0.25	0.25	0.40	12	56
17	22	6	6	3.5	2.2	0.16	0.25	0.25	0.40	16	70
22	30	8	7	4.0	2.4	0.16	0.25	0.25	0.40	20	90
30	38	10	8	5.0	2.4	0.25	0.40	0.40	0.60	25	110
38	44	12	8	5.0	2.4	0.25	0.40	0.40	0.60	32	140
44	50	14	9	5.5	2.9	0.25	0.40	0.40	0.60	40	160
50	58	16	10	6.0	3.4	0.25	0.40	0.40	0.60	45	180
58	65	18	11	7.0	3.4	0.25	0.40	0.40	0.60	50	200
65	75	20	12	7.5	3.9	0.40	0.60	0.60	0.80	56	220
75	85	22	14	9.0	4.4	0.40	0.60	0.60	0.80	63	250
85	95	25	14	9.0	4.4	0.40	0.60	0.60	0.80	70	280
95	110	28	16	10.0	5.4	0.40	0.60	0.60	0.80	80	320
110	130	32	18	11.0	6.4	0.40	0.60	0.60	0.80	90	360
130	150	36	20	12.0	7.1	0.70	1.00	1.00	1.20	100	400
150	170	40	22	13.0	8.1	0.70	1.00	1.00	1.20	110	400
170	200	45	25	15.0	9.1	0.70	1.00	1.00	1.20	125	400
200	230	50	28	17.0	10.1	0.70	1.00	1.00	1.20	140	400
230	260	56	32	20.0	11.1	1.20	1.60	1.60	2.00	–	–
260	290	63	32	20.0	11.1	1.20	1.60	1.60	2.00	–	–
290	330	70	36	22.0	13.1	1.20	1.60	1.60	2.00	–	–
330	380	80	40	25.0	14.1	2.00	2.50	2.50	3.00	–	–
380	440	90	45	28.0	16.1	2.00	2.50	2.50	3.00	–	–
440	500	100	50	31.0	18.1	2.00	2.50	2.50	3.00	–	–

Note: (i) All dimensions are in mm.
(ii) The material for the key is steel with minimum tensile strength of 600 MPa.
Designation – A taper key of width 12 mm, height 8 mm and length 50 mm is designated as Taper key 12 × 8 × 50.

Table 9.19 *Preferred lengths for taper keys (2 × 2 to 20 × 12 mm)*

Preferred length (l) (mm)	Key dimensions ($b \times h$)											
	2 × 2	3 × 3	4 × 4	5 × 5	6 × 6	8 × 7	10 × 8	12 × 8	14 × 9	16 × 10	18 × 11	20 × 12
6	X											
8	X	X										
10	X	X	X									
12	X	X	X	X								
14	X	X	X	X								
16	X	X	X	X	X							
18	X	X	X	X	X							
20	X	X	X	X	X	X						
22		X	X	X	X	X						
25		X	X	X	X	X	X					

(*Contd.*)

(*Contd.*)

28		X	X	X	X	X	X					
32		X	X	X	X	X	X	X				
36		X	X	X	X	X	X	X				
40			X	X	X	X	X	X	X			
45			X	X	X	X	X	X	X	X		
50				X	X	X	X	X	X	X	X	
56				X	X	X	X	X	X	X	X	X
63					X	X	X	X	X	X	X	X
70					X	X	X	X	X	X	X	X
80						X	X	X	X	X	X	X
90						X	X	X	X	X	X	X
100							X	X	X	X	X	X
110							X	X	X	X	X	X
125								X	X	X	X	X
140								X	X	X	X	X
160									X	X	X	X
180										X	X	X
200											X	X
220												X

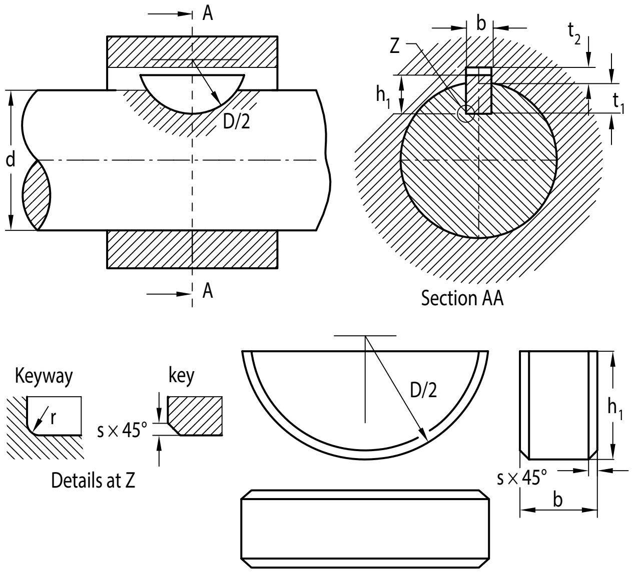
Note: (i) The preferred lengths are indicated by X.
(ii) Assume density of steel as 7.85 g/cm^3 for calculating the weight of key.

Table 9.20 *Preferred lengths for taper keys (22 × 14 to 50 × 28 mm)*

Preferred length (*l*) (mm)	Key dimensions (*b* × *h*)							
	22 × 14	**25 × 14**	**28 × 16**	**32 × 18**	**36 × 20**	**40 × 22**	**45 × 25**	**50 × 28**
63	X							
70	X	X						
80	X	X	X					
90	X	X	X	X				
100	X	X	X	X	X			
110	X	X	X	X	X	X		
125	X	X	X	X	X	X	X	
140	X	X	X	X	X	X	X	X
160	X	X	X	X	X	X	X	X
180	X	X	X	X	X	X	X	X
200	X	X	X	X	X	X	X	X
220	X	X	X	X	X	X	X	X
250	X	X	X	X	X	X	X	X
280		X	X	X	X	X	X	X
320			X	X	X	X	X	X
360				X	X	X	X	X
400					X	X	X	X

Note: The preferred lengths are indicated by X.

Table 9.21 *Dimensions of Woodruff keys and keyways*



Key dimensions			Keyway depth		Keyway radius (*r*)		Chamfer of key (*s*)	
b	***h_1***	***D***	**In shaft (t_1)**	**In hub (t_2)**	**Max.**	**Min.**	**Min.**	**Max.**
1.0	1.4	4.0	1.0	0.6	0.16	0.08	0.16	0.25
1.5	2.6	7.0	2.0	0.8	0.16	0.08	0.16	0.25
2.0	2.6	7.0	1.8	1.0	0.16	0.08	0.16	0.25
2.0	3.7	10.0	2.9	1.0	0.16	0.08	0.16	0.25
2.5	3.7	10.0	2.7	1.2	0.16	0.08	0.16	0.25
3.0	5.0	13.0	3.8	1.4	0.16	0.08	0.16	0.25
3.0	6.5	16.0	5.3	1.4	0.16	0.08	0.16	0.25
4.0	6.5	16.0	5.0	1.8	0.25	0.16	0.25	0.40
4.0	7.5	19.0	6.0	1.8	0.25	0.16	0.25	0.40
5.0	6.5	16.0	4.5	2.3	0.25	0.16	0.25	0.40
5.0	7.5	19.0	5.5	2.3	0.25	0.16	0.25	0.40
5.0	9.0	22.0	7.0	2.3	0.25	0.16	0.25	0.40
6.0	9.0	22.0	6.5	2.8	0.25	0.16	0.25	0.40
6.0	10.0	25.0	7.5	2.8	0.25	0.16	0.25	0.40
8.0	11.0	28.0	8.0	3.3	0.40	0.25	0.40	0.60
10.0	13.0	32.0	10.0	3.3	0.40	0.25	0.40	0.60

(ISO: 3912)

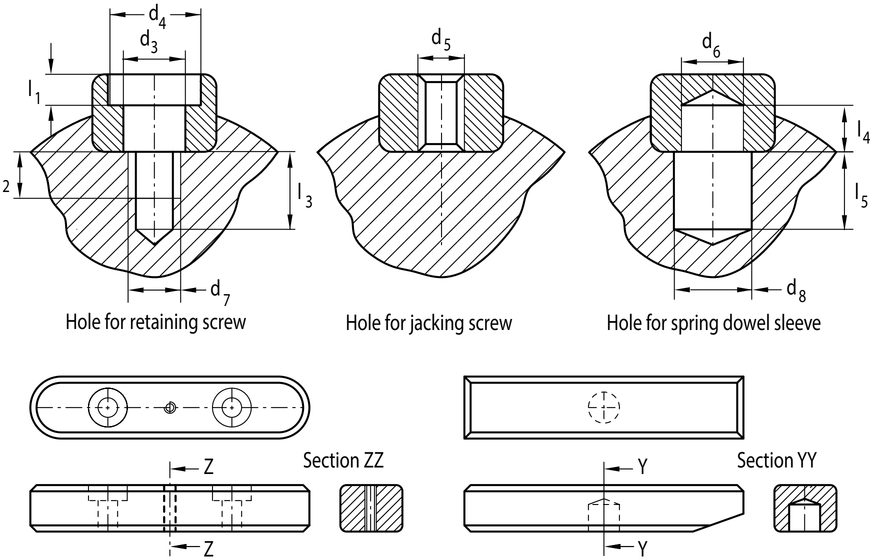
Note: (i) All dimensions are in mm.
(ii) The material for the key is steel with minimum tensile strength of 590 MPa.
Designation – A Woodruff key of width 4 mm and height 6.5 mm is designated as Woodruff key 4 × 6.5.

Table 9.22 *Relationship of shaft diameter to size of Woodruff key*

Shaft diameter (d)				Key size $b \times h_1 \times D$
Series 1		Series 2		
Over	Including	Over	Including	
3	4	3	4	1.0 × 1.4 × 4.0
4	5	4	6	1.5 × 2.6 × 7.0
5	6	6	8	2.0 × 2.6 × 7.0
6	7	8	10	2.0 × 3.7 × 10.0
7	8	10	12	2.5 × 3.7 × 10.0
8	10	12	15	3.0 × 5.0 × 13.0
10	12	15	18	3.0 × 6.5 × 16.0
12	14	18	20	4.0 × 6.5 × 16.0
14	16	20	22	4.0 × 7.5 × 19.0
16	18	22	25	5.0 × 6.5 × 16.0
18	20	25	28	5.0 × 7.5 × 19.0
20	22	28	32	5.0 × 9.0 × 22.0
22	25	32	36	6.0 × 9.0 × 22.0
25	28	36	40	6.0 × 10.0 × 25.0
28	32	40	–	8.0 × 11.0 × 28.0
32	38	–	–	10.0 × 13.0 × 32.0

Note: (i) All dimensions are in mm.
(ii) Series 1 is used for torque applications and Series 2 for positional applications (For example, interference fit where the torque is not transmitted through the key but through the shaft-hub interface.).

Table 9.23 *Dimensions for retaining screws, jacking screws and spring dowel sleeve for parallel keys*



Key $b \times h$	Holes for retaining screws, Jacking screws and Spring dowel sleeves											Retaining screw	Spring dowel sleeve
	Key						Shaft						
	d_3	d_4	d_5	d_6	l_1	l_4	d_7	d_8	l_2	l_3	l_5		
8 × 7	3.4	6	M3	4	2.4	4	M3	4.5	4	7	5	M3 × 8	4 × 8
10 × 8	3.4	6	M3	4	2.4	4	M3	4.5	5	8	5	M3 × 10	4 × 8
12 × 8	4.5	8	M4	5	3.2	5	M4	5.5	6	10	7	M4 × 10	5 × 10
14 × 9	5.5	10	M5	6	4.1	6	M5	6.5	6	10	8	M5 × 10	6 × 12
16 × 10	5.5	10	M5	6	4.1	6	M5	6.5	6	10	8	M5 × 10	6 × 12
18 × 11	6.6	11	M6	8	4.8	7	M6	9	6	11	11	M6 × 12	8 × 16
20 × 12	6.6	11	M6	8	4.8	8	M6	9	6	11	10	M6 × 12	8 × 16
22 × 14	6.6	11	M6	8	4.8	8	M6	9	8	13	10	M6 × 16	8 × 16
25 × 14	9.0	14	M8	10	6.0	10	M8	11	9	15	12	M8 × 16	10 × 20
28 × 16	11	18	M10	12	7.3	10	M10	13	9	16	18	M10 × 16	12 × 24
32 × 18	11	18	M10	12	7.3	12	M10	13	10	17	16	M10 × 20	12 × 24
36 × 20	14	20	M12	16	8.3	14	M12	17	12	20	20	M12 × 25	16 × 30
40 × 22	14	20	M12	16	8.3	16	M12	17	12	20	18	M12 × 25	16×30
45 × 25	14	20	M12	16	8.3	16	M12	17	15	22	18	M12 × 30	16 × 30

(*Contd.*)

(*Contd.*)

50 × 28	14	20	M12	16	8.3	16	M12	17	12	19	18	M12 × 30	16 × 32
56 × 32	14	20	M12	16	8.3	16	M12	17	13	20	18	M12 × 35	16 × 32
63 × 32	14	20	M12	16	8.3	16	M12	17	13	20	18	M12 × 35	16 × 32
70 × 36	18	26	M16	20	11.5	20	M16	21	17	24	24	M16×40	20×40
80 × 40	18	26	M16	20	11.5	20	M16	21	18	25	24	M16 × 45	20 × 40
90 × 45	22	33	M20	25	13.5	25	M20	26	20	28	30	M20 × 50	25 × 40
100 × 50	22	33	M20	25	13.5	25	M20	26	20	28	30	M20 × 55	25 × 50

Note: All dimensions are in mm.

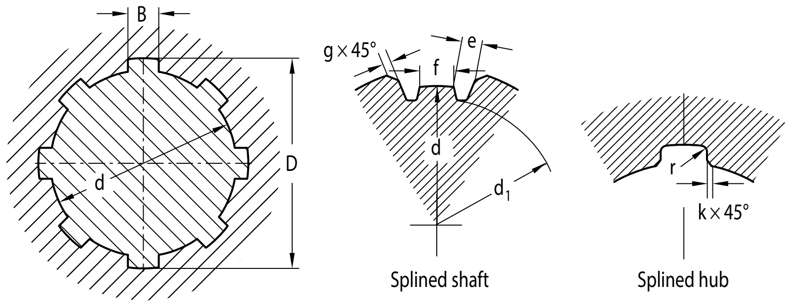
Table 9.24 *Proportions and stress equations of keys*

Empirical proportions of square key	
$b = h = \frac{d}{4}$ (9.16) $l = 1.5d$ (9.17)	b = width of key (mm) h = height or thickness of key (mm) l = length of key (mm) d = diameter of shaft (mm)
Empirical proportions of Flat key	
$b = \frac{d}{4}$ (9.18) $h = \frac{2}{3}\ \ b = \frac{d}{6}$ (9.19) $l = 1.5d$ (9.20)	
Stress equations for Square and Flat keys	
$\tau = \frac{2M_t}{dbl}$ (9.21)	M_t = transmitted torque (N-mm) τ = permissible shear stress (MPa or N/mm^2)
$\sigma_c = \frac{4M_t}{dhl}$ (9.22)	σ_c = permissible compressive stress (MPa or N/mm^2)
Stress equations for Kennedy keys	
$\tau = \frac{M_t}{\sqrt{2}\, dbl}$ (9.23)	$\sigma_c = \frac{\sqrt{2}\, M_t}{dbl}$ (9.24)

Note: Kennedy keys consist of two square keys spaced at an angle of 90° on the circumference of shaft.

9.4 SPLINES

Table 9.25 *Dimensions for straight sided splines (Light duty series)*

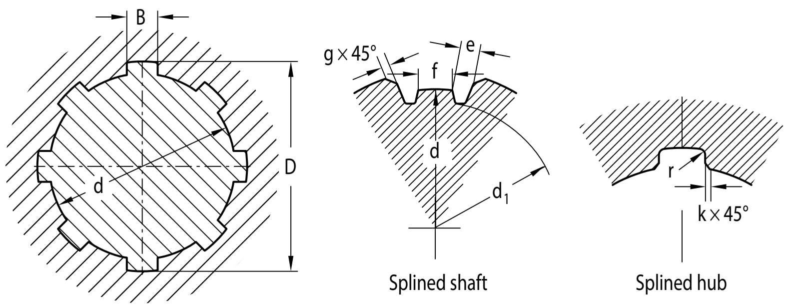


Nominal size $N \times d \times D$	**No. of splines (*N*)**	**Minor dia. (*d*)**	**Major dia. (*D*)**	**Width (*B*)**	d_1 **Min.**	***e*** **Max.**	**f**	***g*** **Max.**	***k*** **Max.**	***r*** **Max.**	**Centreing on**
$6 \times 23 \times 26$	6	23	26	6	22.1	1.25	3.54	0.3	0.3	0.2	Inside diameter
$6 \times 26 \times 30$	6	26	30	6	24.6	1.84	3.85	0.3	0.3	0.2	
$6 \times 28 \times 32$	6	28	32	7	26.7	1.77	4.03	0.3	0.3	0.2	
$8 \times 32 \times 36$	8	32	36	6	30.4	1.89	2.71	0.4	0.4	0.3	Inside diameter or flanks
$8 \times 36 \times 40$	8	36	40	7	34.5	1.78	3.46	0.4	0.4	0.3	
$8 \times 42 \times 46$	8	42	46	8	40.4	1.68	5.03	0.4	0.4	0.3	
$8 \times 46 \times 50$	8	46	50	9	44.6	1.61	5.75	0.4	0.4	0.3	
$8 \times 52 \times 58$	8	52	58	10	49.7	2.72	4.89	0.5	0.5	0.5	
$8 \times 56 \times 62$	8	56	62	10	53.6	2.76	6.38	0.5	0.5	0.5	
$8 \times 62 \times 68$	8	62	68	12	59.8	2.48	7.31	0.5	0.5	0.5	
$10 \times 72 \times 78$	10	72	78	12	69.6	2.54	5.45	0.5	0.5	0.5	
$10 \times 82 \times 88$	10	82	88	12	79.3	2.67	8.62	0.5	0.5	0.5	
$10 \times 92 \times 98$	10	92	98	14	89.4	2.36	10.08	0.5	0.5	0.5	
$10 \times 102 \times 108$	10	102	108	16	99.9	2.23	11.49	0.5	0.5	0.5	
$10 \times 112 \times 120$	10	112	120	18	108.8	3.23	10.72	0.5	0.5	0.5	

(ISO: 14)

Note: All dimensions are in mm.
Designation – A six splined shaft or hub with minor diameter 28 mm and major diameter 32 mm is designated as Spline $6 \times 28 \times 32$.

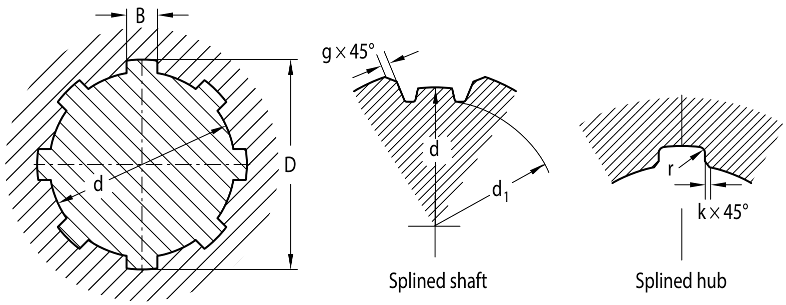
Table 9.26 *Dimensions for straight sided splines (Medium duty series)*



Nominal size $N \times d \times D$	No. of splines (*N*)	Minor dia. (*d*)	Major dia. (*D*)	Width (*B*)	d_1 Min.	*e* Max.	*f*	*g* Max.	*k* Max.	*r* Max	Centreing on
6 × 11 × 14	6	11	14	3	9.9	1.55	–	0.3	0.3	0.2	Inside diameter
6 × 13 × 16	6	13	16	3.5	12.0	1.50	0.32	0.3	0.3	0.2	
6 × 16 × 20	6	16	20	4	14.5	2.10	0.16	0.3	0.3	0.2	
6 × 18 × 22	6	18	22	5	16.7	1.95	0.45	0.3	0.3	0.2	
6 × 21 × 25	6	21	25	5	19.5	1.98	1.95	0.3	0.3	0.2	
6 × 23 × 28	6	23	28	6	21.3	2.30	1.34	0.3	0.3	0.2	
6 × 26 × 32	6	26	32	6	23.4	2.94	1.65	0.4	0.4	0.3	
6 × 28 × 34	6	28	34	7	25.9	2.94	1.70	0.4	0.4	0.3	
8 × 32 × 38	8	32	38	6	29.4	3.30	0.15	0.4	0.4	0.3	Inside diameter or flanks
8 × 36 × 42	8	36	42	7	33.5	3.01	1.02	0.4	0.4	0.3	
8 × 42 × 48	8	42	48	8	39.5	2.91	2.57	0.4	0.4	0.3	
8 × 46 × 54	8	46	54	9	42.7	4.10	0.86	0.5	0.5	0.5	
8 × 52 × 60	8	52	60	10	48.7	4.00	2.44	0.5	0.5	0.5	
8 × 56 × 65	8	56	65	10	52.2	4.74	2.50	0.5	0.5	0.5	
8 × 62 × 72	8	62	72	12	57.8	5.00	2.40	0.5	0.5	0.5	
10 × 72 × 82	10	72	82	12	67.4	5.43	–	0.5	0.5	0.5	
10 × 82 × 92	10	82	92	12	77.1	5.40	3.00	0.5	0.5	0.5	
10 × 92 × 102	10	92	102	14	87.3	5.20	4.50	0.5	0.5	0.5	
10 × 102 × 112	10	102	112	16	97.7	4.90	6.30	0.5	0.5	0.5	
10 × 112 × 125	10	112	125	18	106.3	6.40	4.40	0.5	0.5	0.5	

Note: All dimensions are in mm.

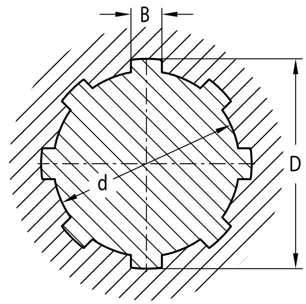
Table 9.27 *Dimensions for straight sided splines (Heavy duty series)*



Nominal size $N \times d \times D$	No. of splines (N)	Minor dia. (d)	Major dia. (D)	Width (B)	d_1 Min.	g Max.	k Max.	r Max.	Centreing on
10 × 16 × 20	10	16	20	2.5	14.0	0.3	0.3	0.15	Inside diameter or flanks
10 × 18 × 23	10	18	23	3	15.6	0.3	0.3	0.15	
10 × 21 × 26	10	21	26	3	18.44	0.3	0.3	0.15	
10 × 23 × 29	10	23	29	4	20.3	0.3	0.3	0.15	
10 × 26 × 32	10	26	32	4	23.0	0.4	0.4	0.15	
10 × 28 × 35	10	28	35	4	24.4	0.4	0.4	0.25	
10 × 32 × 40	10	32	40	5	28.0	0.4	0.4	0.25	
10 × 36 × 45	10	36	45	5	31.3	0.4	0.4	0.25	
10 × 42 × 52	10	42	52	6	36.9	0.5	0.4	0.40	
10 × 46 × 56	10	46	56	7	40.9	0.5	0.5	0.40	
16 × 52 × 60	16	52	60	5	47.0	0.5	0.5	0.40	Flanks
16 × 56 × 65	16	56	65	5	50.6	0.5	0.5	0.40	
16 × 62 × 72	16	62	72	6	56.1	0.5	0.5	0.40	
16 × 72 × 82	16	72	82	7	65.9	0.5	0.5	0.40	
20 × 82 × 92	20	82	92	6	75.6	0.5	0.5	0.40	
20 × 92 × 102	20	92	102	7	85.5	0.5	0.5	0.40	
20 × 102 × 115	20	102	115	8	93.7	0.5	0.5	0.40	
20 × 112 × 125	20	112	125	9	103.7	0.5	0.5	0.40	

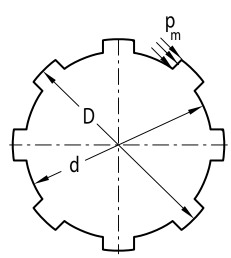
Note: All dimensions are in mm.

Table 9.28 *Tolerances for straight sided splines*



			Tolerance on			
			Width of hub (*B*)		Minor diameter of hub (*d*)	Major diameter of hub (*D*)
			Soft	Hardened	Soft or hardened	Soft or hardened
Splined hub	For centreing on inner diameter or flanks	Shaft sliding or fixed	D9	F10	H7	H11
Splined shaft	For centreing on inner diameter	Shaft sliding inside hub	h8	e8	f7	a11
		Shaft fixed in hub	p6	h6	j6	a11
	For centreing on flanks	Shaft sliding inside hub	h8	e8	–	a11
		Shaft fixed in hub	u6	k6	–	a11

Table 9.29 *Torque capacity of straight sided splines*

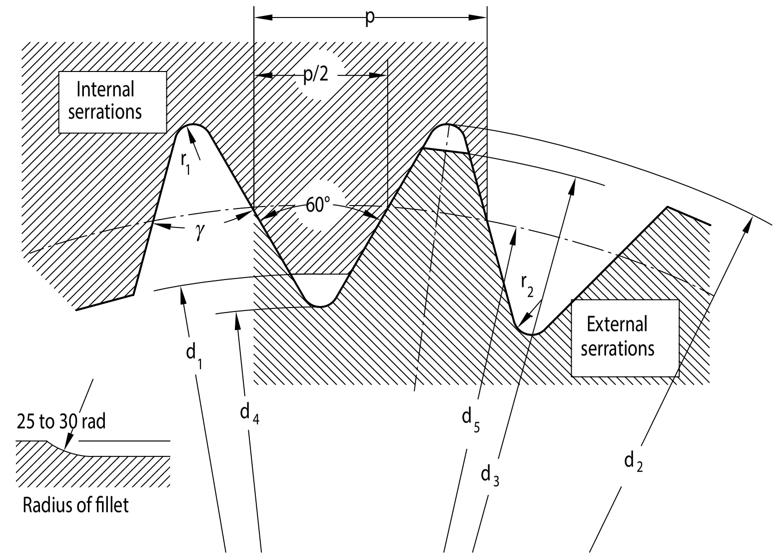


$M_t = \frac{1}{8} p_m l N (D^2 - d^2)$ (9.25)	M_t = transmitted torque (N-mm) p_m = permissible pressure on spline (MPa or N/mm^2) D = major diameter of splines (mm) d = minor diameter of splines (mm) l = length of hub (mm) N = number of splines

Note: The permissible pressure on the splines is limited to 6.5 MPa.

9.5 SERRATIONS

Table 9.30 *Dimensions of serrations with 60° serration angle (for general engineering)*



Nominal size	d_1			d_2 (*)	d_3			d_4 (*)	Pitch Dia. d_5	Allowances for contact diameter in μm			r_1 Approx.	r_2 Approx	Pitch p	γ	Number of teeth
	Nom.	Max.	Min.		Nom.	Max.	Min.			Internal serrations	External serrations						
											Close	Loose					
7 × 8	6.9	7.27	7.18	8.21	8.1	7.82	7.73	6.91	7.5	+63	–63	–126	0.08	0.08	0.842	47°8′35″	28
8 × 10	8.1	8.47	8.38	9.90	10.1	9.81	9.70	8.26	9.0	+65	–65	–130	0.08	0.08	1.010	47°8′35″	28
10 × 12	10.1	10.50	10.39	12.0	12.0	11.71	11.60	10.20	11.0	+68	–68	–136	0.10	0.10	1.152	48°	30
12 × 14	12.0	12.40	12.29	14.18	14.2	13.91	13.80	12.06	13.0	+70	–70	–140	0.10	0.10	1.317	48°23′14″	31
15 × 17	14.9	15.30	15.19	17.28	17.2	16.91	16.80	14.91	16.0	+75	–75	–150	0.15	0.15	1.571	48°45′	32

(*Contd.*)

(*Contd.*)

17 × 20	17.3	17.70	17.59	20.00	20.0	19.70	19.57	17.37	18.5	+80	–80	–160	0.15	0.20	1.761	49°5′27″	33
21 × 24	20.8	21.23	21.10	23.76	23.9	23.60	23.47	20.76	22.0	+85	–85	–170	0.15	0.25	2.033	49°24′42″	34
26 × 30	26.5	26.93	26.80	30.06	30.0	29.70	29.57	26.40	28.0	+95	–95	–190	0.25	0.30	2.513	49°42′52″	35
30 × 34	30.5	30.97	30.81	34.17	34.0	33.69	33.53	30.38	32.0	+100	–100	–200	0.30	0.40	2.792	50°	36
36 × 40	36.0	36.47	36.31	40.16	39.9	39.59	39.43	35.95	38.0	+110	–110	–220	0.50	0.40	3.226	50°16′13″	37
40 × 44	40.0	40.47	40.31	44.42	44.0	43.68	43.52	39.72	42.0	+115	–115	–230	0.50	0.40	3.472	50°31′35″	38
45 × 50	45.0	45.48	45.32	50.20	50.0	49.68	49.52	44.97	47.5	+125	–125	–250	0.50	0.40	3.826	50°46′9″	39
50 × 55	50.0	50.48	50.32	55.25	54.9	54.56	54.37	49.72	52.5	+135	–135	–270	0.60	0.40	4.123	51°	40
55 × 60	55.0	55.53	55.34	60.39	60.0	59.66	59.47	54.76	57.5	+140	–140	–280	0.60	0.50	4.301	51°25′43″	42

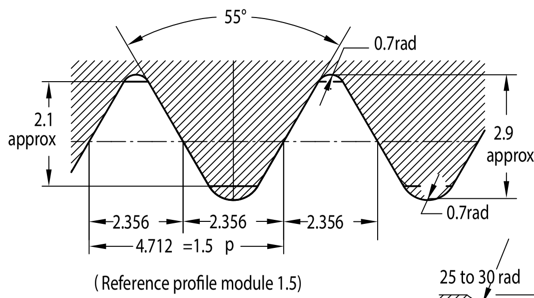
(DIN: 5481)

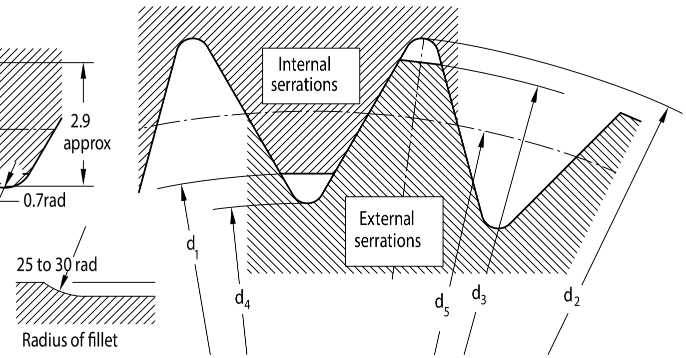
Note: (i) All dimensions are in mm.

(ii) Serrated shafts and hubs are used in automotive, small tools, machine tools and other industries, mostly with close fit and a large number of teeth to allow for many index positions. (*)The values given are obtained by calculations.

Designation – Serrations of nominal size 12 × 14 are designated as Serrations 12 × 14.

Table 9.31 *Dimensions of serrations with 55° serration angle (for general engineering)*





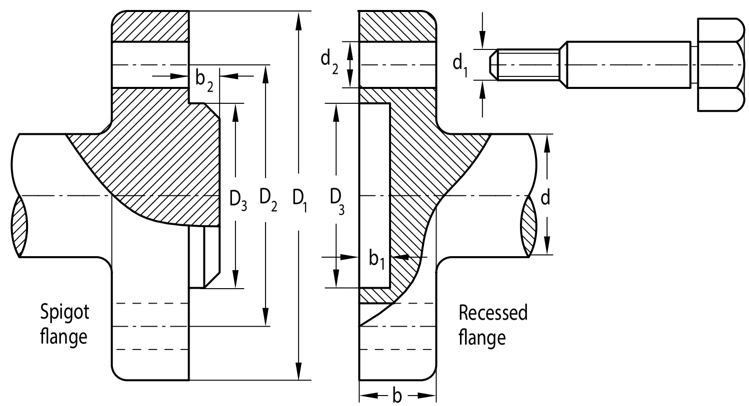
Nominal size	d_1			d_2 (*)	d_3			d_4 (*)	Pitch dia. d_5	Allowances for contact diameter in μm			Number of teeth
	Nom.	Max.	Min.		Nom.	Max.	Min.			Internal serrations	External serrations		
											Close	Loose	
60 × 65	60	60.53	60.34	65.4	65	64.66	64.47	59.6	61.5	+150	–150	–300	41
65 × 70	65	65.53	65.34	70.4	70	69.64	69.45	64.6	67.5	+160	–160	–320	45
70 × 75	70	70.55	70.36	75.4	75	74.64	74.45	69.6	72.0	+165	–165	–330	48
75 × 80	75	75.55	75.36	80.4	80	79.64	79.45	74.6	76.5	+175	–175	–350	51
80 × 85	80	80.55	80.36	85.4	85	84.62	84.40	79.6	82.5	+185	–185	–370	55
85 × 90	85	85.60	85.38	90.4	90	89.62	89.40	84.6	87.0	+190	–190	–380	58
90 × 95	90	90.60	90.38	95.4	95	94.62	94.40	89.6	91.5	+200	–200	–400	61
95 × 100	95	95.60	95.38	100.4	100	99.62	99.40	94.6	97.5	+205	–205	–410	65
100 × 105	100	100.60	100.38	105.4	105	104.59	104.37	99.6	102.0	+215	–215	–430	68
105 × 110	105	105.63	105.41	110.4	110	109.59	109.37	104.6	106.5	+220	–220	–440	71
110 × 115	110	110.63	110.41	115.4	115	114.59	114.37	109.6	112.5	+230	–230	–460	75
115 × 120	115	115.63	115.41	120.4	120	119.59	119.37	114.6	117.0	+240	–240	–480	78
120 × 125	120	120.63	120.41	125.4	125	124.54	124.37	119.6	121.5	+250	–250	–500	81

(DIN: 5481)

Note: (i) All dimensions are in mm.
(*) The values given are obtained by calculations.
Designation – Serrations of nominal size 70 × 75 are designated as Serrations 70 × 75.

9.6 COUPLINGS

Table 9.32 *Dimensions of forged end-type rigid couplings*



Coupling number		for Shaft diameter d		Coupling dimensions						Bolt size d_1	Bolt hole dia d_2 (H8)	Number of bolts
Recessed flange	Spigot flange	Max.	Min.	Flange outside diameter D_1	Flange width b	Locating diameter D_3	Recess depth b_1	Spigot depth b_2	Pitch circle diameter D_2			
R1	S1	35	–	100	17	50	6	4	70	M10	11	4
R2	S2	45	36	120	22	60	6	4	85	M12	13	4
R3	S3	55	46	140	22	75	7	5	100	M14	15	4
R4	S4	70	56	175	27	95	7	5	125	M16	17	6
R5	S5	80	71	195	32	95	7	5	140	M18	19	6
R6	S6	90	81	225	32	125	7	5	160	M20	21	6
R7	S7	110	91	265	36	150	9	7	190	M24	25	6
R8	S8	130	111	300	46	150	9	7	215	M30	32	6

(*Contd.*)

(*Contd.*)

R9	S9	150	131	335	50	195	9	7	240	M33	34	6
R10	S10	170	151	375	55	195	10	8	265	M36	38	8
R11	S11	190	171	400	55	240	10	8	290	M36	38	8
R12	S12	210	191	445	65	240	10	8	315	M42	44	8
R13	S13	230	211	475	70	280	10	8	340	M45	46	8
R14	S14	250	231	500	70	280	10	8	370	M45	46	10
R15	S15	270	251	560	80	330	10	8	400	M52	55	10
R16	S16	300	271	600	85	330	10	8	440	M56	60	10
R17	S17	330	301	650	90	400	10	8	480	M60	65	10
R18	S18	360	331	730	100	400	10	8	520	M68	72	10
R19	S19	390	361	775	105	480	11	9	570	M72	76	10
R20	S20	430	391	875	110	480	11	9	620	M76	80	12
R21	S21	470	431	900	115	560	11	9	670	M80	85	12
R22	S22	520	471	925	120	560	12	10	730	M90	95	12
R23	S23	570	521	1000	125	640	12	10	790	M100	105	12
R24	S24	620	571	1090	130	720	12	10	850	M110	115	12

Note: (i) All dimensions are in mm.
(ii) The spigot diameter has a tolerance of h7.
(iii) The recess diameter has a tolerance of H8.
(iv) The bolt-holes are first drilled and subsequently reamed, the coupling halves being in the same relative angular positions for both operations.
(v) Forged end-type rigid couplings are commonly used in various assemblies for transmission of power from driving to driven shafts.

Designation – (i) A spigot flange of a forged end type coupling with coupling number S16 is designated as Coupling flange S16.
(ii) A recessed flange of a forged end type coupling with coupling number R16 is designated as Coupling flange R16.

Table 9.33 *Types of misalignment between driving and driven shafts*

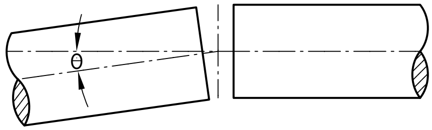
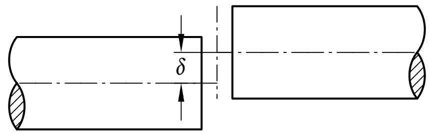
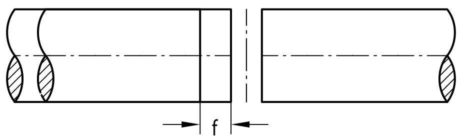
Angular misalignment (θ)

Parallel misalignment (δ)

End float (f)


Table 9.34 *Service factors for bush-type flexible couplings*

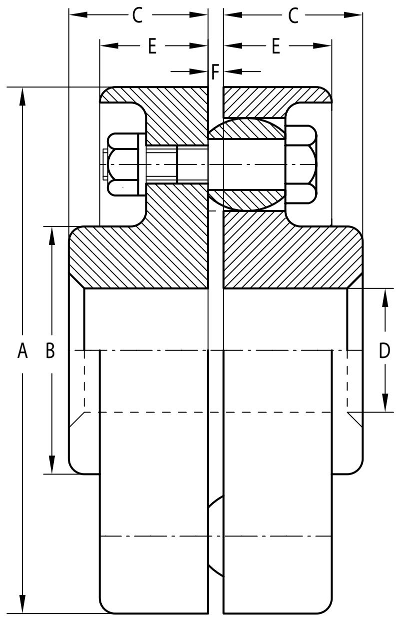
Type of driven machine	Service factors for prime movers					
	Electric motor, steam or water turbine	High speed steam or gas engine	Petrol engine		Diesel engine	
			4 or more cylinders	Less than 4 cylinders	6 or more cylinders	4 or less cylinders
Alternators and generators (excluding welding generators), induced drought fans, printing machinery, rotary pumps, compressors and exhausters, conveyors.	1.5	2.0	2.5	3.0	3.5	5.0
Wood working machinery, machine tools (cutting), excluding planning machine, calendars, mixers and elevators.	2.0	2.5	3.0	3.5	4.0	5.5
Forced draught fans, high speed reciprocating compressors, high speed crushers and pulverises, machine tools (forming)	2.5	3.0	3.5	4.0	4.5	6.0

(*Contd.*)

(*Contd.*)

Rotary screens, rod mills, tube, cable and wire machinery, vacuum pumps.	3.0	3.5	4.0	4.5	5.0	6.5
Low speed reciprocating compressors, haulage gears, metal planning machines, brick and tile machinery, rubber machinery, tube mills, generators (welding)	3.5	4.0	4.5	5.0	5.5	7.0

Table 9.35 *Dimensions for bush-type flexible couplings*



Coupling No.	Load rating per 100 rev/min-kW (*)	Bore diameter *D*		A	B	C	E	F
		Min.	Max.					
FB 1	0.368	15	22	80	40	32	18	2
FB 2	0.441	18	25	90	44	35	25	3
FB 3	0.736	20	30	100	50	40	25	3
FB 4	1.18	25	35	112	62	45	25	3
FB 5	1.84	28	40	125	65	50	28	3
FB 6	2.94	32	45	140	76	55	28	3
FB 7	4.63	35	50	160	85	60	38	3
FB 8	7.36	42	60	180	102	70	38	3
FB 9	10.3	50	70	200	120	80	38	4

(*Contd.*)

(*Contd.*)

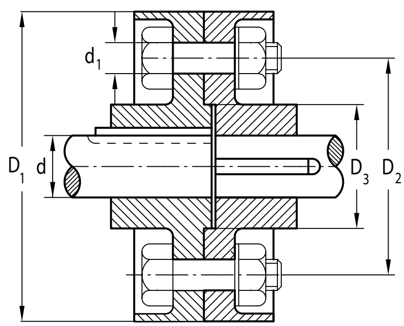
FB 10	14.7	57	80	225	134	90	42	4
FB 11	20.6	64	90	250	154	100	45	4
FB 12	25.7	71	100	280	166	110	55	4
FB 13	38.2	78	110	315	180	125	55	4
FB 14	58.8	85	120	355	200	140	68	5
FB 15	80.9	92	130	400	218	160	80	5
FB 16	110	100	140	450	240	180	80	5
FB 17	235	114	160	560	300	220	122	5
FB 18	331	128	180	630	340	240	122	5
FB 19	456	135	190	710	370	260	145	6
FB 20	618	170	240	800	405	290	145	6
FB 21	883	185	260	900	440	320	165	6
FB 22	1250	200	290	1000	490	350	165	6

Note: (i) All dimensions are in mm.
(ii) The flanges for coupling are made of cast iron FG 200.
(iii) The bushes are made of flexible material such as rubber, having a hardness of 80 IRH.
(iv) For cast iron couplings, allowable maximum peripheral velocity is 30 m/s.
(v) The bore of coupling has a tolerance of H7 and surface finish of N6.
(vi) The maximum permissible shear stress for pins is 35 MPa.
(vii) The maximum allowable compressive stress for rubber bushes is about 2 MPa.

Designation – A bush-type flexible coupling FB 4 for shaft diameter range of 25 to 35 mm is designated as Flexible Coupling FB 4.

(*)Load rating per 100 rev/min kW = $-\dfrac{\text{kW of power application} \times \text{service factor} \times 100}{\text{rev/min of application}}$

Table 9.36 *Rigid flange couplings – Basic relationships*



Case I: Bolts are fitted in reamed and ground holes (Power transmitted by shear resistance of bolts)	
$M_t = P \times \dfrac{D_2}{2} \times N$ (9.26)	M_t = torque transmitted by the coupling (N-mm) P = force acting on each bolt (N) D_2 = pitch circle diameter of bolts (mm) N = number of bolts

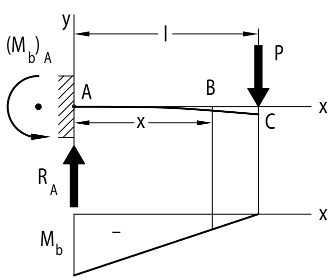
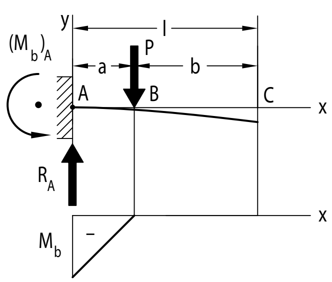
(*Contd.*)

(*Contd.*)

$\tau = \dfrac{P}{\left(\dfrac{\pi}{4} d_1^2\right)}$ or $\tau = \dfrac{8M_t}{\pi D_2 N d_1^2}$ (9.27)	τ = permissible shear stress for bolt (MPa or N/mm^2) d_1 = nominal diameter of the bolt (mm)
Case II: Bolts are fitted in large clearance holes (Power transmitted by friction between two flanges)	
$R_f = \dfrac{2}{3} \dfrac{(R_o^3 - R_i^3)}{(R_o^2 - R_i^2)}$ (9.28)	R_o = outer radius of the flange ($D_1/2$) (mm) R_i = radius of the recess ($D_3/2$) (mm)
$M_t = \mu P_i N R_f$ (9.29)	P_i = initial tension in each bolt (N) μ = coefficient of friction between flanges

9.7 DESIGN FOR LATERAL RIGIDITY

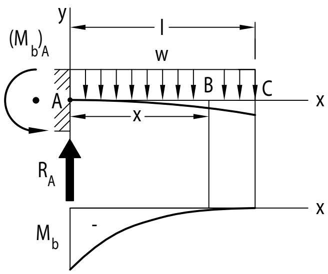
Table 9.37 *Bending moment and deflection of beams*

Case A: Cantilever beam – End load	
	(i) Reactions (R) at $A = P$ (1) (ii) Bending moments (M_b) at $A = -Pl$ (2) (M_b) at $x = P(x - l)$ (3) (iii) Deflections (δ) at $x = \dfrac{Px^2}{6EI}(x - 3l)$ (4) $(\delta)_{max}$ at $(x = l) = -\dfrac{Pl^3}{3EI}$ (5)
Case B: Cantilever beam – Intermediate load	
	(i) Reactions (R) at $A = P$ (6) (ii) Bending moments (M_b) at $A = -Pa$ (7) (M_b) at $x = P(x - a)$ for $(x < a)$ (8) (M_b) at $x = 0$ for $(x > a)$ (9) (iii) Deflections (δ) at $x = \dfrac{Px^2}{6EI}(x - 3a)$ for $(x < a)$ (10) (δ) at $x = \dfrac{Pa^2}{6EI}(a - 3x)$ for $(x > a)$ (11) $(\delta)_{max}$ at $(x = l) = \dfrac{Pa^2}{6EI}(a - 3l)$ (12)

(*Contd.*)

(*Contd.*)

Case C: Cantilever beam – Uniformly distributed load



(i) Reactions

$$(R) \text{ at } A = wl \tag{13}$$

(ii) Bending moments

$$(M_b) \text{ at } A = -\frac{wl^2}{2} \tag{14}$$

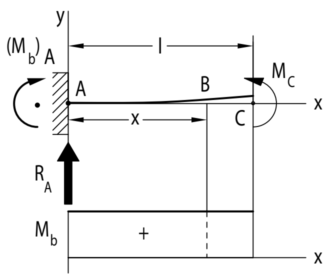
$$(M_b) \text{ at } x = -\frac{w}{2}(l-x)^2 \tag{15}$$

(iii) Deflections

$$(\delta) \text{ at } x = \frac{wx^2}{24EI}(4lx - x^2 - 6l^2) \tag{16}$$

$$(\delta)_{\max} \text{ at } (x = l) = -\frac{wl^4}{8EI} \tag{17}$$

Case D: Cantilever beam – Moment load



(i) Reactions

$$(R) \text{ at } A = 0 \tag{18}$$

(ii) Bending moments

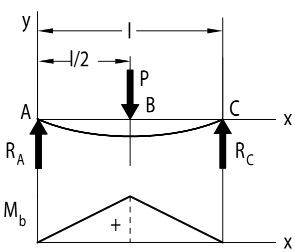
$$(M_b) \text{ at } A = (M_b) \text{ at } x = (M_b) \text{ at } C = M_c \tag{19}$$

(iii) Deflections

$$(\delta) \text{ at } x = \frac{M_c x^2}{2EI} \tag{20}$$

$$(\delta)_{\max} \text{ at } (x = l/2) = \frac{M_c l^2}{2EI} \tag{21}$$

Case E: Simply supported beam –Centre load



(i) Reactions

$$(R) \text{ at } A = (R) \text{ at } C = \frac{P}{2} \tag{22}$$

(ii) Bending moments

$$(M_b) \text{ at } x = \frac{Px}{2} \quad \text{for } (x < l/2) \tag{23}$$

$$(M_b) \text{ at } x = \frac{P(l-x)}{2} \quad \text{for } (x > l/2) \tag{24}$$

$$(M_b)_{\max} \text{ at } (x = l/2) = \frac{Pl}{4} \tag{25}$$

(iii) Deflections

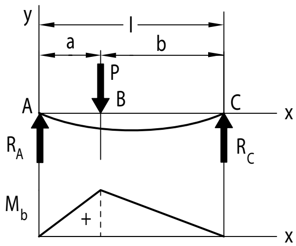
$$(\delta) \text{ at } x = \frac{Px(4x^2 - 3l^2)}{48EI} \quad \text{for } (x < l/2) \tag{26}$$

$$(\delta)_{\max} \text{ at } (x = l/2) = -\frac{Pl^3}{48EI} \tag{27}$$

(*Contd.*)

(*Contd.*)

Case F: Simply supported beam –Intermediate load



(i) Reactions

$$(R) \text{ at } A = \frac{Pb}{l} \tag{28}$$

$$(R) \text{ at } C = \frac{Pa}{l} \tag{29}$$

(ii) Bending moments

$$(M_b) \text{ at } x = \frac{Pbx}{l} \quad \text{for } (x < a) \tag{30}$$

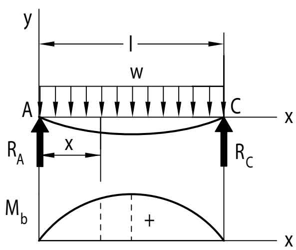
$$(M_b) \text{ at } x = \frac{Pa(l-x)}{l} \quad \text{for } (x > a) \tag{31}$$

(iii) Deflections

$$(\delta) \text{ at } x = \frac{Pbx(x^2 + b^2 - l^2)}{6EIl} \quad \text{for } (x < a) \tag{32}$$

$$(\delta) \text{ at } x = \frac{Pa(l-x)(x^2 + a^2 - 2lx)}{6EIl} \quad \text{for } (x > a) \tag{33}$$

Case G: Simply supported beam – Uniformly distributed load



(i) Reactions

$$(R) \text{ at } A = (R) \text{ at } C = \frac{wl}{2} \tag{34}$$

(ii) Bending moments

$$(M_b) \text{ at } x = \frac{wx(l-x)}{2} \tag{35}$$

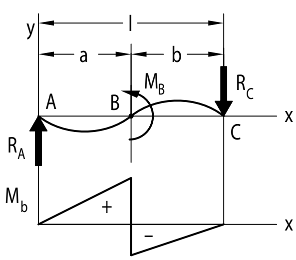
$$(M_b)_{\max} \text{ at } (x = l/2) = \frac{wl^2}{8} \tag{36}$$

(iii) Deflections

$$(\delta) \text{ at } x = \frac{wx(2lx^2 - x^3 - l^3)}{24EI} \tag{37}$$

$$(\delta)_{\max} \text{ at } (x = l/2) = -\frac{5wl^4}{384EI} \tag{38}$$

Case H: Simply supported beam – Moment load



(i) Reactions

$$(R) \text{ at } A = -(R) \text{ at } C = \frac{M_B}{l} \tag{39}$$

(ii) Bending moments

$$(M_b) \text{ at } x = \frac{M_B x}{l} \quad \text{for } (x < a) \tag{40}$$

$$(M_b) \text{ at } x = \frac{M_B(x-l)}{l} \quad \text{for } (x > a) \tag{41}$$

(*Contd.*)

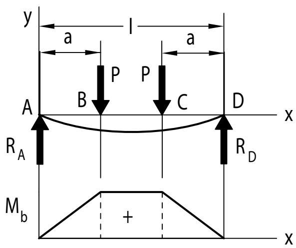
(*Contd.*)

(iii) Deflections

$$(\delta) \text{ at } x = \frac{M_B x(x^2 + 3a^2 - 6al + 2l^2)}{6EIl} \quad \text{for } (x < a) \qquad (42)$$

$$(\delta) \text{ at } x = \frac{M_B[x^3 - 3lx^2 + x(2l^2 + 3a^2) - 3a^2 l]}{6EIl} \quad \text{for } (x > a) \qquad (43)$$

Case J: Simply supported beam –Twin load



(i) Reactions

$$(R) \text{ at } A = (R) \text{ at } D = P \qquad (44)$$

(ii) Bending moments

$$(M_b) \text{ at } x = Px \quad \text{for } (0 < x < a) \qquad (45)$$

$$(M_b) \text{ at } x = Pa \quad \text{for } [a < x < (l - a)] \qquad (46)$$

$$(M_b) \text{ at } x = P(l - x) \quad \text{for } [(l - a) < x < l] \qquad (47)$$

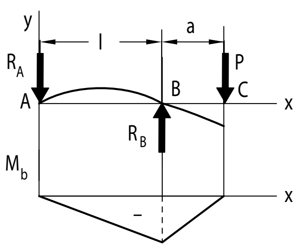
(iii) Deflections

$$(\delta) \text{ at } x = \frac{Px}{6EI}(x^2 + 3a^2 - 3la) \quad \text{for } (0 < x < a) \qquad (48)$$

$$(\delta) \text{ at } x = \frac{Pa}{6EI}(3x^2 + a^2 - 3lx) \quad \text{for } [a < x < (l - a)] \qquad (49)$$

$$(\delta)_{\max} \text{ at } (x = l/2) = \frac{Pa}{24EI}(4a^2 - 3l^2) \qquad (50)$$

Case K: Simply supported beam –Overhanging load



(i) Reactions

$$(R) \text{ at } A = -\frac{Pa}{l} \qquad (51)$$

$$(R) \text{ at } B = \frac{P}{l}(l + a) \qquad (52)$$

(ii) Bending moments

$$(M_b) \text{ at } x = -\frac{Pax}{l} \quad \text{for } (x < l) \qquad (53)$$

$$(M_b) \text{ at } x = P(x - l - a) \quad \text{for } (x > l) \qquad (54)$$

(iii) Deflections

$$(\delta) \text{ at } x = \frac{Pax(l^2 - x^2)}{6EIl} \quad \text{for } (x < l) \qquad (55)$$

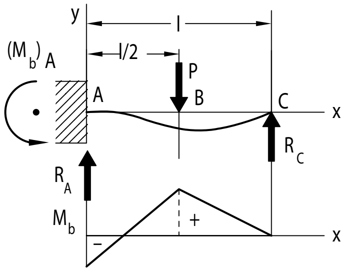
$$(\delta) \text{ at } x = \frac{P(x - l)[(x - l)^2 - a(3x - l)]}{6EI} \quad \text{for } (x > l) \qquad (56)$$

$$(\delta)_{\max} \text{ at } (x = l + a) = -\frac{Pa^2(l + a)}{3EI} \qquad (57)$$

(*Contd.*)

(*Contd.*)

Case L: One end fixed and one end simply supported beam –Center load



(i) Reactions

$$(R) \text{ at } A = \frac{11P}{16} \tag{58}$$

$$(R) \text{ at } C = \frac{5P}{16} \tag{59}$$

(ii) Bending moments

$$(M_b) \text{ at } A = -\frac{3Pl}{16} \tag{60}$$

$$(M_b) \text{ at } x = \frac{P}{16}(11x - 3l) \quad \text{for } (x < l/2) \tag{61}$$

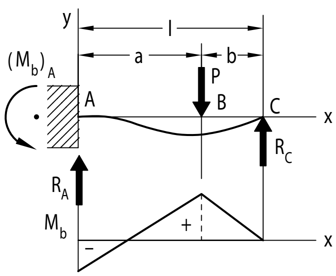
$$(M_b) \text{ at } x = \frac{5P}{16}(l - x) \quad \text{for } (x > l/2) \tag{62}$$

(iii) Deflections

$$(\delta) \text{ at } x = \frac{Px^2}{96EI}(11x - 9l) \quad \text{for } (x < l/2) \tag{63}$$

$$(\delta) \text{ at } x = \frac{P(l-x)}{96EI}(5x^2 + 2l^2 - 10lx) \quad \text{for } (x > l/2) \tag{64}$$

Case M: One end fixed and one end simply supported beam –Intermediate load



(i) Reactions

$$(R) \text{ at } A = \frac{Pb}{2l^3}(3l^2 - b^2) \tag{65}$$

$$(R) \text{ at } C = \frac{Pa^2}{2l^3}(3l - a) \tag{66}$$

(ii) Bending moments

$$(M_b) \text{ at } A = \frac{Pb}{2l^2}(b^2 - l^2) \tag{67}$$

$$(M_b) \text{ at } x = \frac{Pb}{2l^3}[b^2 l - l^3 + x(3l^2 - b^2)] \quad \text{for } (x < a) \tag{68}$$

$$(M_b) \text{ at } x = \frac{Pa^2}{2l^3}(3l^2 - 3lx - al + ax) \quad \text{for } (x > a) \tag{69}$$

(iii) Deflections

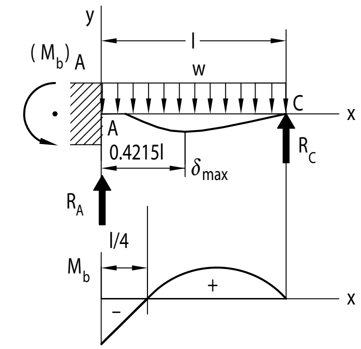
$$(\delta) \text{ at } x = \frac{Pbx^2}{12EIl^3}[3l(b^2 - l^2) + x(3l^2 - b^2)] \quad \text{for } (x < a) \tag{70}$$

$$(\delta) \text{ at } x = \frac{Pbx^2}{12EIl^3}[3l(b^2 - l^2) + x(3l^2 - b^2)] - \frac{P(x-a)^3}{6EI} \quad \text{for } (x > a) \tag{71}$$

(*Contd.*)

(*Contd.*)

Case N: One end fixed and one end simply supported beam – Uniformly distributed load



(i) Reactions

$$(R) \text{ at } A = \frac{5wl}{8} \quad (72)$$

$$(R) \text{ at } C = \frac{3wl}{8} \quad (73)$$

(ii) Bending moments

$$(M_b) \text{ at } A = -\frac{wl^2}{8} \quad (74)$$

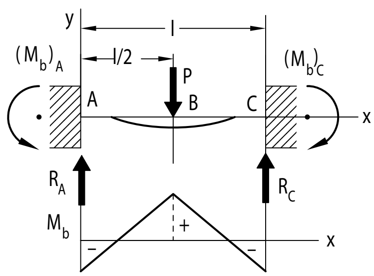
$$(M_b) \text{ at } x = \frac{w}{8}(4x^2 + 5lx - l^2) \quad (75)$$

(iii) Deflections

$$(\delta) \text{ at } x = \frac{wx^2}{48EI}(l - x)(2x - 3l) \quad (76)$$

$$(\delta)_{\max} = -\frac{wl^4}{185EI} \quad (77)$$

Case O: Both ends fixed– Center load



(i) Reactions

$$(R) \text{ at } A = (R) \text{ at } C = \frac{P}{2} \quad (78)$$

(ii) Bending moments

$$(M_b) \text{ at } A = (M_b) \text{ at } C = -\frac{Pl}{8} \quad (79)$$

$$(M_b) \text{ at } x = \frac{P}{8}(4x - l) \;\; \text{for } (x < l/2) \quad (80)$$

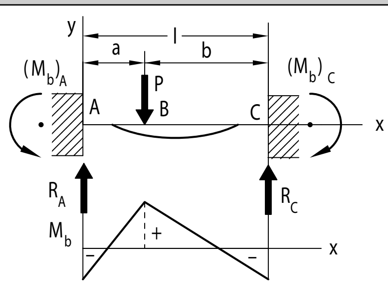
$$(M_b) \text{ at } x = \frac{P}{8}(3l - 4x) \;\; \text{for } (x > l/2) \quad (81)$$

(iii) Deflections

$$(\delta) \text{ at } x = \frac{Px^2}{48EI}(4x - 3l) \;\; \text{for } (x < l/2) \quad (82)$$

$$(\delta)_{\max} \text{ at } (x = l/2) = -\frac{Pl^3}{192EI} \quad (83)$$

Case P: Both ends fixed– Intermediate load



(i) Reactions

$$(R) \text{ at } A = \frac{Pb^2}{l^3}(3a + b) \quad (84)$$

$$(R) \text{ at } C = \frac{Pa^2}{l^3}(3b + a) \quad (85)$$

(ii) Bending moments

$$(M_b) \text{ at } A = -\frac{Pab^2}{l^2} \quad (86)$$

(*Contd.*)

(*Contd.*)

$$(M_b) \text{ at } C = -\frac{Pa^2b}{l^2} \qquad (87)$$

$$(M_b) \text{ at } x = \frac{Pb^2}{l^3}[x(3a+b) - la] \quad \text{for } (x < a) \qquad (88)$$

$$(M_b) \text{ at } x = \frac{Pb^2}{l^3}[x(3a+b) - la] - P(x-a)$$

$$\text{for } (x > a) \qquad (89)$$

(iii) Deflections

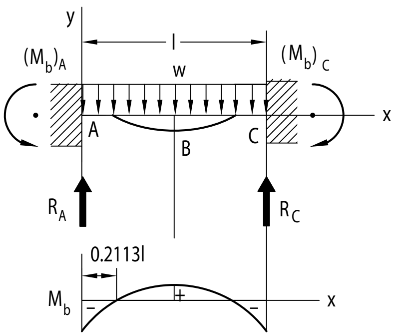
$$(\delta) \text{ at } x = \frac{Pb^2x^2}{6EIl^3}[x(3a+b) - 3al]$$

$$\text{for } (x < a) \qquad (90)$$

$$(\delta) \text{ at } x = \frac{Pa^2(l-x)^2}{6EIl^3}[(l-x)(3b+a) - 3bl]$$

$$\text{for } (x > a) \qquad (91)$$

Case Q: Both ends fixed– Uniformly distributed load



(i) Reactions

$$(R) \text{ at } A = (R) \text{ at } C = \frac{wl}{2} \qquad (92)$$

(ii) Bending moments

$$(M_b) \text{ at } A = (M_b) \text{ at } C = -\frac{wl^2}{12} \qquad (93)$$

$$(M_b) \text{ at } x = \frac{w}{12}(6lx - 6x^2 - l^2) \qquad (94)$$

(iii) Deflections

$$(\delta) \text{ at } x = -\frac{wx^2}{24EI}(l-x)^2 \qquad (95)$$

$$(\delta)_{max} \text{ at } (x = l/2) = -\frac{wl^4}{384EI} \qquad (96)$$

Notations:

(R) = reaction at support (N)

(δ) = deflection (mm)

w = load intensity or external load per unit length (N/mm)

l = span length (mm)

I = moment of inertia of cross section (mm^4)

M_b = bending moment (N-mm)

P = external force (N)

M = external moment (N-mm)

E = modulus of elasticity (MPa or N/mm^2)

Note: (i) The maximum permissible deflection for the transmission shaft is generally taken as

$$(\delta)_{max} = (0.001)l \text{ to } (0.003)l$$

where l is the span length between the two bearings.

(ii) The maximum permissible deflection at the gear is taken as

$$(\delta)_{max} = (0.01)m$$

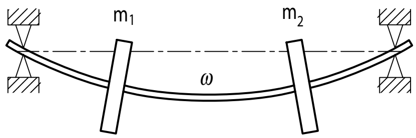
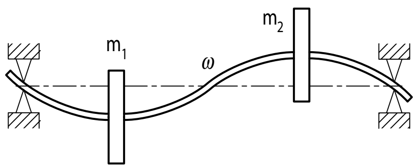
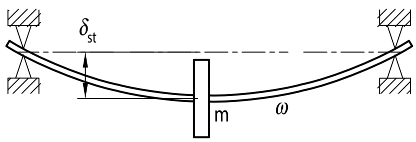
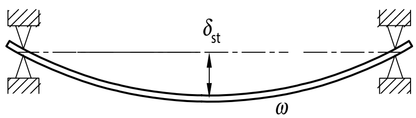
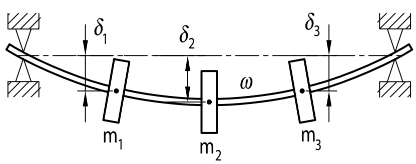
where m is the module of the gear teeth.

(iii) In practice, the lateral rigidity of the structure is improved by the following methods:

(a) Reduce the span length; (b) Increase the number of supports; (c) Reduce the number of joints; (d) Assemble the components with pre-load; (e) Lubricate the contact surfaces with high viscosity oil; and (f) Select a cross section in which the cross-sectional area is away from the neutral axis, such as I or tubular section.

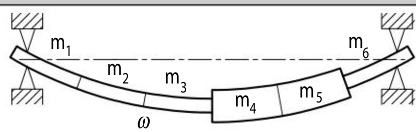
9.8 CRITICAL SPEED OF SHAFTS

Table 9.38 *Critical speed of shafts*

First critical speed of shaft	
	
Second critical speed of shaft	
	
Shaft with negligible mass and attached with a single mass	
	
$\omega_n = \sqrt{\frac{k}{m}} = \sqrt{\frac{kg}{w}} = \sqrt{\frac{g}{\delta_{st}}}$ (9.30)	ω_n = natural frequency (rad/s) k = shaft spring rate (N/m) ($k = \omega/\delta_{st} = \text{mg}/\delta_{st}$) m = attached mass (kg) w = weight of attached mass (N) ($w = m\,g$) g = acceleration of gravity (m/s^2) (g = 9.81 m/s^2) δ_{st} = static deflection (m)
Shaft with uniform cross section	
	
$\omega_n = \sqrt{\frac{5}{4}\frac{g}{\delta_{st}}}$ (9.31)	δ_{st} = maximum static deflection at the centre (m)
Shaft with negligible mass and attached with several masses	
	

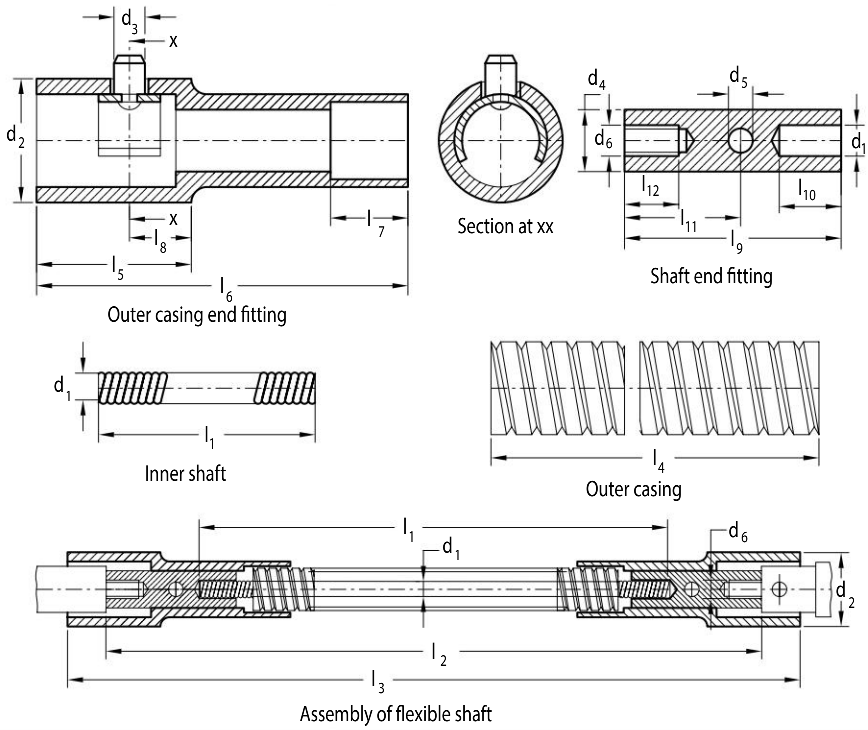
(*Contd.*)

(*Contd.*)

$$\omega_n = \sqrt{\frac{g(w_1\delta_1 + w_2\delta_2 + w_3\delta_3 + \cdots)}{(w_1\delta_1^2 + w_2\delta_2^2 + w_3\delta_3^2 + \cdots)}} \quad (9.32)$$	$w_1 = m_1 g \quad w_2 = m_2 g \quad w_3 = m_3 g$ $\delta_1, \delta_2, \delta_3$, etc. = static deflections at the respective masses viz. m_1, m_2, m_3, …etc. **Note:** This equation is called Rayleigh–Ritz equation.
Shaft with non-uniform cross section	
	
$$\omega_n = \sqrt{\frac{g(w_1\delta_1 + w_2\delta_2 + w_3\delta_3 + \cdots)}{(w_1\delta_1^2 + w_2\delta_2^2 + w_3\delta_3^2 + \cdots)}} \quad (9.33)$$	**Note:** The mass of the entire shaft is broken into number of parts, m_1, m_2, m_3,... etc., and each part is treated as concentrated mass at its centre of gravity.

9.9 FLEXIBLE SHAFTS

Table 9.39 *Dimensions of type-A flexible shaft—Assembly and details*



Designation	d_1	d_2	d_3	d_4	d_5	d_6	l_1	l_2	l_3	l_4	l_5	l_6	l_7	l_8	l_9	l_{10}	l_{11}	l_{12}
7 × 1500	7	20	6	9	3	M6	1500	–	–	1470	40	133	36	13	35	10	19	12
7 × 2000	7	20	6	9	3	M6	2000	–	–	1220	40	133	36	13	35	10	19	12
10 × 1500	10	28	8	12	4.5	M10	1500	1570	1610	1460	40	133	36	13	50	15	27	17

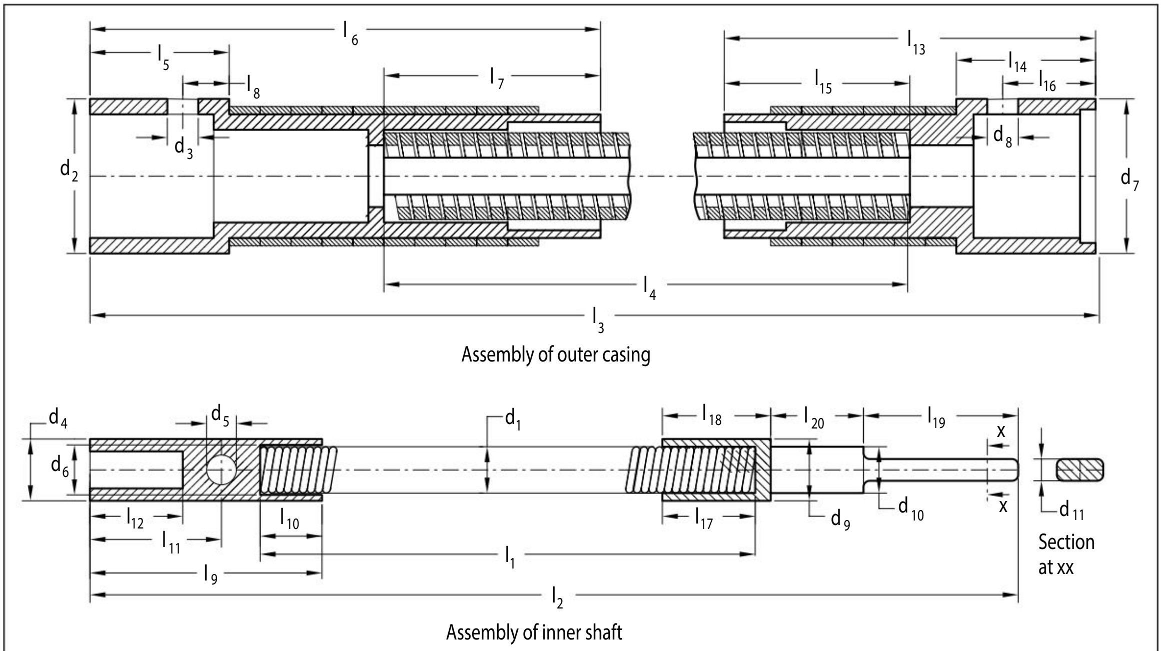
(*Contd.*)

(*Contd.*)

10 × 2000	10	28	8	12	4.5	M10	2000	2070	2110	1960	40	133	36	13	50	15	27	17
12 × 1500	12	28	8	14	4.5	M10	1500	1570	1590	1450	40	133	36	13	55	20	27	17
12 × 2000	12	28	8	14	4.5	M10	2000	2070	2090	1950	40	133	36	13	55	20	27	17
15 × 2000	15	34	8	18	6.5	M10	2000	2090	2114	1930	40	133	36	13	70	25	35	22
15 × 3000	15	34	8	18	6.5	M10	3000	3090	3114	2930	40	133	36	13	70	25	35	22
20 × 2000	20	40	8	23	6.5	M14	2000	2090	2114	1920	40	133	36	13	75	30	35	22
20 × 3000	20	40	8	23	6.5	M14	3000	3090	3114	2920	40	133	36	13	75	30	35	22

Note: (i) All dimensions are in mm.
(ii) Type-A flexible shaft has threaded shaft end fittings on both ends with outer casing sliding at the driving end.
(iii) The inner shaft is made of spring steel.
Designation – A flexible shaft with inner shaft diameter of 12 mm and length 1500 mm is designated as Flexible shaft 12 × 1500.

Table 9.40 *Dimensions of type-B flexible shaft —Assembly and details*



Designa-tion	d_1	d_2	d_3	d_4	d_5	d_6	d_7	d_8	d_9	d_{10}	d_{11}	l_1	l_2	l_3
7 × 1500	7	30	8	12.0	5	M10	22	8	8.5	5.0	3.9	1500	1590	1547
7 × 2000	7	30	8	12.0	5	M10	22	8	8.5	5.0	3.9	2000	2090	2047
10 × 1500	10	30	8	12.0	5	M10	28	8	12.5	7.5	5.9	1500	1605	1548

(*Contd.*)

10 × 2000	10	30	8	12.0	5	M10	28	8	14.0	7.5	5.9	2000	2105	2048
12 × 1500	12	30	8	14.0	5	M10	28	8	14.0	7.5	5.9	1500	1607	1550
12 × 2000	12	30	8	14.0	5	M10	28	8	14.0	7.5	5.9	2000	2107	2050
15 × 2000	15	40	8	17.5	7	M14	35	8	17.5	11.0	8.9	2000	2109	2059
15 × 3000	15	40	8	17.5	7	M14	35	8	17.5	11.0	8.9	3000	3109	3059
20 × 2000	20	40	8	23.0	7	M14	40	8	–	–	–	2000	2086	2110
20 × 3000	20	40	8	23.0	7	M14	40	8	–	–	–	3000	3086	3110

(Refer to figure of Table 9.40)

Designation	l_4	l_5	l_6	l_7	l_8	l_9	l_{10}	l_{11}	l_{12}	l_{13}	l_{14}	l_{15}	l_{16}	l_{17}	l_{18}	l_{19}	l_{20}
7 × 1500	1452	30	86	33	10	40	11	24	18	75	31	33	20	15	20	38	18
7 × 2000	1952	30	96	33	10	40	11	24	18	75	31	33	20	15	20	38	18
10 × 1500	1436	30	102	40	10	47	15	25	18	90	40	40	28	20	23	52	18
10 × 2000	1936	30	102	40	10	47	15	25	18	90	40	40	28	20	23	52	18
12 × 1500	1430	33	105	40	13	50	18	25	18	95	40	40	28	25	30	52	18
12 × 2000	1930	33	105	40	13	50	18	25	18	95	40	40	28	25	30	52	18
15 × 2000	1959	40	95	45	13	64	26	33	23	95	50	45	34	25	28	55	13
15 × 3000	2959	40	95	45	13	64	26	33	23	95	50	45	34	25	28	55	13
20 × 2000	1960	43	110	35	16	73	30	33	23	110	43	35	27	–	–	–	–
20 × 3000	2960	43	110	35	16	73	30	33	23	110	43	35	27	–	–	–	–

Note: (i) All dimensions are in mm.
(ii) Type-B flexible shaft has fixed end fittings at both ends for outer casing and sliding end fitting for the shaft at the driven end.
(iii) The inner shaft is made of spring steel.

Designation – A flexible shaft with inner shaft diameter of 12 mm and length 1500 mm is designated as Flexible shaft 12 × 1500.

Springs 10

10.1 PROPERTIES OF STEEL SPRING WIRES

Table 10.1 *Tensile strength of patented and cold drawn steel wires—unalloyed*

Uncoated wire dia. (Nominal) (mm)	Tensile strength (MPa or N/mm^2)			
	Grade-1	Grade-2	Grade-3	Grade-4
0.10	—	—	2530	—
0.11	—	—	2520	—
0.12	—	—	2520	—
0.14	—	—	2510	—
0.16	—	—	2500	—
0.18	—	—	2500	—
0.20	—	—	2490	2700
0.22	—	—	2480	2680
0.25	—	—	2470	2670
0.28	—	—	2460	2660
0.30	1720	2060	2460	2660
0.32	1710	2050	2450	2650
0.34	1710	2050	2450	2540
0.36	1700	2040	2440	2630
0.38	1700	2040	2430	2620
0.40	1700	2040	2430	2620
0.43	1690	2030	2420	2610
0.45	1680	2020	2410	2600
0.48	1680	2020	2400	2590
0.50	1670	2010	2390	2580
0.53	1660	2000	2380	2570
0.56	1660	2000	2370	2560
0.60	1650	1990	2360	2550
0.63	1640	1980	2340	2540
0.65	1640	1980	2330	2540

(*Contd.*)

(*Contd.*)

0.70	1630	1970	2320	2530
0.75	1620	1960	2300	2500
0.80	1610	1950	2280	2480
0.85	1600	1930	2260	2460
0.90	1590	1920	2250	2440
0.95	1580	1910	2250	2420
1.00	1570	1900	2240	2400
1.05	1560	1890	2210	2380
1.10	1550	1880	2190	2370
1.20	1540	1860	2170	2340
1.25	1530	1850	2140	2320
1.30	1520	1840	2130	2300
1.40	1500	1820	2110	2290
1.50	1490	1800	2100	2260
1.60	1470	1780	2080	2250
1.70	1460	1760	2050	2220
1.80	1440	1750	2030	2190
1.90	1430	1730	2010	2180
2.0	1420	1720	1990	2160
2.1	1410	1700	1960	2130
2.25	1400	1680	1940	2100
2.40	1380	1660	1910	2070
2.5	1370	1640	1890	2050
2.6	1360	1620	1860	2030
2.8	1340	1600	1840	2000
3.0	1320	1570	1830	1980
3.2	1310	1550	1790	1960
3.4	1290	1530	1760	1920
3.6	1270	1510	1750	1890
3.8	1260	1490	1720	1860
4.0	1250	1480	1700	1840
4.25	1250	1460	1680	1820
4.5	1230	1440	1660	1800
4.75	1210	1420	1620	1770
5.0	1190	1390	1600	1750
5.3	1170	1370	1570	1720
5.6	1150	1350	1550	1690
6.0	1130	1320	1530	1670
6.3	1120	1310	1500	1640
6.5	1110	1290	1480	1620
7.0	1090	1260	1460	1610
7.5	1070	1250	1430	1570
8.0	1050	1220	1400	1540

(*Contd.*)

(*Contd.*)

8.5	1020	1200	1370	1500
9.0	1000	1180	1350	1480
9.5	990	1150	1310	—
10.0	980	1130	1290	—
10.5	—	1100	—	—
11.0	—	1080	—	—
12.0	—	1040	—	—
12.5	—	1030	—	—
13.0	—	1020	—	—
14.0	—	990	—	—
15.0	—	970	—	—

(ISO: 8458–2)

Note: The modulus of elasticity for above mentioned wires is 210 790 MPa and the modulus of rigidity is 81370 MPa (or N/mm^2).
Applications – (i) Grade 1 is used for springs subjected to static or low load cycle – low stressed applications. (ii) Grade 2 is used for springs subjected to moderate load cycle – statically stressed applications. (iii) Grade 3 is used for highly stressed static springs, for springs under moderate to high dynamic stresses and in smaller diameters even for impact loaded springs. (iv) Grade 4 is suitable for maximum stressed static springs and moderately stressed dynamic springs.

Table 10.2 *Tensile strength of oil hardened and tempered spring steel wire (SW) and valve spring wire (VW)— unalloyed*

Wire dia. (Nominal) (mm)	Tensile strength (MPa or N/mm^2) (Min)	
	SW	**VW**
1.00	1760	1670
1.05		
1.10		
1.20	1720	1620
1.25		
1.30		
1.40		
1.50	1670	1570
1.60		
1.70		
1.80		
1.90		
2.0	1620	1520
2.1		
2.25		
2.4		
2.5	1570	1470
2.6		
2.8		

(*Contd.*)

(*Contd.*)

3.0	1520	1430
3.2		
3.4		
3.6	1480	1400
3.8		
4.0		
4.25	1440	1370
4.5		
4.75		
5.0		
5.3	1400	1340
5.6		
6.0		
6.3	1360	1300
6.5		
7.0		
7.5		
8.0	1290	—
8.5		
9.0		
9.5		
10.0	1250	—
10.5		
11.0		
12.0		
12.5		
13.0		
14.0		

(ISO: 8458–3)

Note: The modulus of elasticity for above mentioned wires is 205 880 MPa and the modulus of rigidity is 81370 MPa (or N/mm^2).
Applications – (i) Oil hardened and tempered spring steel wire (SW) is used for moderately stressed springs. (ii) Oil hardened and tempered valve spring wire (VW) is used for dynamically stressed and moderate impact loaded springs. (iii) Both types of spring wires are used under elevated temperature conditions.

Table 10.3 *Tolerances for oil hardened and tempered spring steel wire (SW) and valve spring wire (VW)—unalloyed*

Wire diameter (Nominal) (mm)		Tolerance (mm)
From	Up to and including	
1.00	1.4	± 0.015
1.50	1.90	± 0.020

(*Contd.*)

(*Contd.*)

2.0	3.2	± 0.030
3.4	5.6	± 0.040
6.0	8.5	± 0.060
9.0	10.0	± 0.080
10.5	14.0	± 0.12

Table 10.4 *Tensile strength of oil hardened and tempered spring steel wire —alloyed*

Wire dia. (Nominal) (mm)	Tensile strength (MPa or N/mm^2) (Min)			
	1 S	**1 D**	**2 S**	**2 D**
1.00	—	—	2060	2060
1.05	1860	1760	1960	1960
1.10				
1.20				
1.25				
1.30				
1.40				
1.50				
1.60				
1.70				
1.80				
1.90				
2.0				
2.1	1760	1670	1910	1910
2.25				
2.4				
2.5				
2.6				
2.8				
3.0				
3.2				
3.4	1670	1570	1860	1860
3.6				
3.8				
4.0				
4.25	1590	1520	1810	1810
4.5				
4.75				
5.0				
5.3	1520	1470	1760	1760
5.6				
6.0				

(*Contd.*)

(*Contd.*)

6.3	1470	1420	1720	1720
6.5				
7.0				
7.5	—	—	1670	1670
8.0				

Note: The modulus of elasticity for above mentioned wires is 303 920 MPa and the modulus of rigidity is 81370 MPa (or N/mm^2).
Applications – (i) Grade 1 S and 2 S wires are used for static working springs. (ii) Grade 1 D and 2 D wires are used for dynamic working springs. (iii) Both types of spring wires are used under elevated temperature conditions.

Table 10.5 *Tolerances for oil hardened and tempered spring steel wire—alloyed*

Wire diameter (Nominal) (mm)		Tolerance (mm)
From	Up to and including	
1.00	1.4	± 0.015
1.50	1.90	± 0.020
2.0	3.2	± 0.030
3.4	5.6	± 0.040
6.0	8.0	± 0.060

Table 10.6 *Tensile strength of stainless spring steel wire for normal corrosion resistance*

Wire diameter (Nominal) (mm)	Tensile strength (MPa or N/mm^2) (Min)
0.10	2060
0.11	2010
0.125	
0.14	
0.16	
0.18	
0.20	1960
0.22	
0.25	
0.28	
0.30	
0.32	
0.34	1910
0.36	
0.38	
0.40	
0.43	
0.45	
0.48	
0.50	

(*Contd.*)

(*Contd.*)

0.53	1860
0.56	
0.60	
0.63	
0.65	
0.70	
0.75	
0.80	
0.85	1760
0.90	
0.95	
1.00	
1.05	
1.10	
1.15	
1.20	
1.25	
1.30	1670
1.40	
1.50	
1.60	
1.70	
1.80	
1.90	
2.0	
2.1	1570
2.25	
2.40	
2.5	
2.6	
2.8	
3.0	
3.15	
3.2	1470
3.4	
3.6	
3.8	
4.0	
4.25	
4.5	
4.75	
5.0	

(*Contd.*)

(*Contd.*)

5.3	1320
5.6	
6.0	
6.3	
6.5	
7.0	
7.5	
8.0	
8.5	1270
9.0	
9.5	
10.0	

(ISO: 6931–1)

Note: (i) There are two grades of stainless spring steel wires – Grade 1 and Grade 2. The limiting temperatures are 250ºC for Grade 1 and 350ºC for Grade 2.
(ii) The modulus of elasticity and rigidity of the wires are as follows:

Grade	Modulus of elasticity (MPa or N/mm^2)		Modulus of rigidity (MPa or N/mm^2)	
	Tempered	Untempered	Tempered	Untempered
1	190 200 ± 4900	180 390 ± 3920	73530 ± 1960	69610 ± 1960
2	295 100 ± 4900	190 200 ± 3920	78430 ± 1960	73530 ± 1960

Applications– Both grades are used in applications involving corrosive atmosphere. Both grades withstand normal atmospheric as well as steam and other corrosive media.

Table 10.7 *Tolerances for stainless spring steel wire for normal corrosion resistance*

Wire diameter (Nominal) (mm)		Tolerance (mm)
From	Up to and including	
0.10	0.18	± 0.005
0.20	0.80	± 0.010
0.85	1.40	± 0.015
1.50	1.90	± 0.020
2.00	3.20	± 0.030
3.40	5.60	± 0.04
6.00	8.50	± 0.06
9.00	10.00	± 0.08

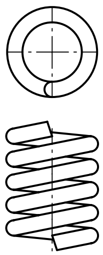
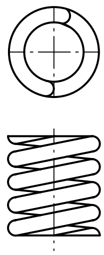
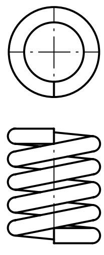
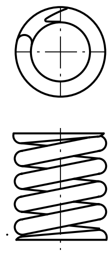
10.2 PROPERTIES OF BARS AND FLATS

Table 10.8 *Mechanical properties of bars and flats for volute, helical and laminated springs for automotive suspension*

Grade of steel	Untreated hardness (HB)	Treatment condition			Elongation (Percent) (Min)
		Soft annealed Hardness (HB)	Quenched and tempered		
			Yield stress (Min.) (MPa or N/mm^2)	Tensile strength (MPa or N/mm^2)	
55Si7	≈ 270	245	1130	1320 to 1570	6
60Si7	≈ 310	255	1130	1320 to 1570	6
65Si7	≈ 310	255	1180	1370 to 1620	6
50Cr4V2	> 310	245	1180	1370 to 1620	6
60Cr4V2	> 310	255	1180	1370 to 1620	6

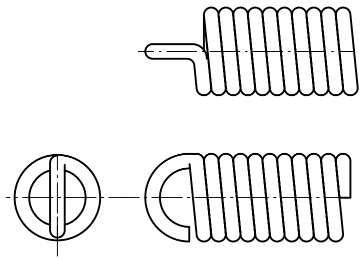
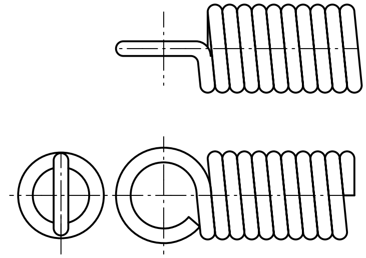
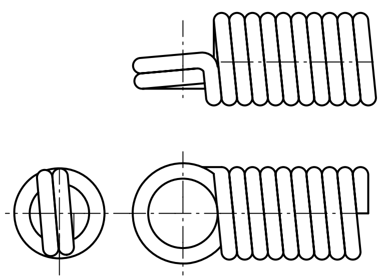
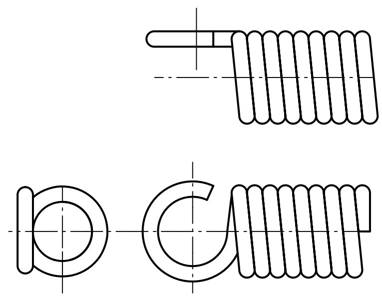
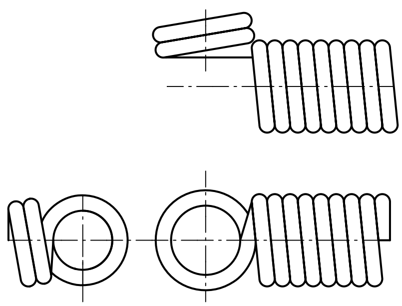
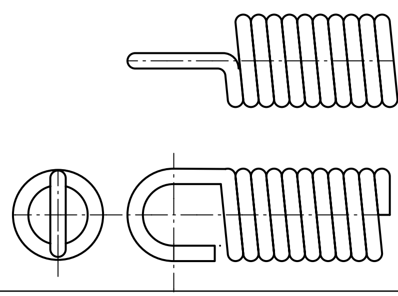
10.3 HELICAL SPRINGS

Table 10.9 *End styles for helical compression springs*

Plain ends	Plain and ground ends	Square ends	Square and ground ends
			
$N = N_t$	$N = (N_t - 1)$	$N = (N_t - 2)$	$N = (N_t - 2)$

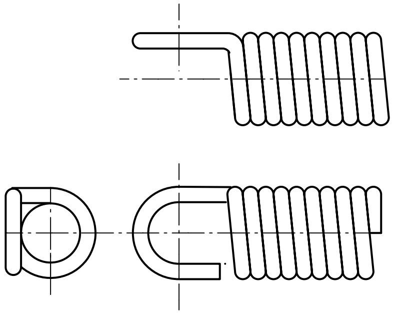
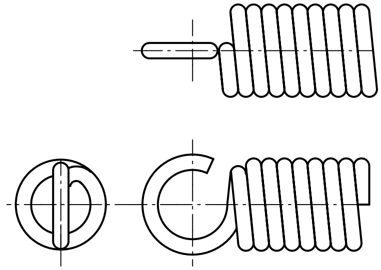
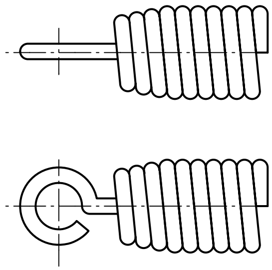
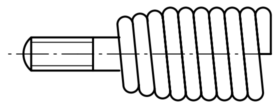
Note: N = number of active coils N_t = total number of coils

Table 10.10 *End styles for helical extension springs*

Form A (Half hook)	**Form B (Full hook)**
	
$L_H = 0.55D_i$ to $0.8\ D_i$	$L_H = 0.8\ D_i$ to $1.1\ D_i$
Form C	**Form D**
	
$L_H = 0.8D_i$ to $1.1D_i$	$L_H \approx D_i$
Form E	**Form F (Raised hook)**
	
$L_H \approx D_i$	—

(*Contd.*)

(*Contd.*)

Form G (Elongated hook)	**Form H**
	
—	$L_H \approx 1.1 D_i$
Form J (Rolled-in hook)	**Form K (Threaded stud rolled –in)**
	
—	—

Notations: L_H = distance of inside edge of hook from body of spring (mm)
D_i = inside diameter of coil (mm)

Note: When the end connections of spring are not considered, number of active coils is equal to total number of coils.

Table 10.11 *Stress and deflection equations for helical springs*

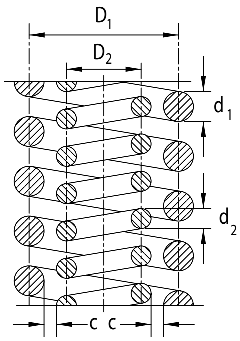
$$D = \frac{D_i + D_o}{2} \qquad (10.1)$$	D_i = inside diameter of spring coil (mm) D_o = outside diameter of spring coil (mm) D = mean coil diameter (mm)
$$C = \frac{D}{d} \qquad (10.2)$$	d = wire diameter of spring (mm) C = spring index
$$k = \frac{P}{\delta} \qquad (10.3)$$	k = stiffness of the spring (N/mm) P = axial spring force (N) δ = axial deflection of the spring corresponding to force P (mm)
$$\tau = K\left(\frac{8PD}{\pi d^3}\right) \qquad (10.4)$$ $$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} \qquad (10.5)$$	τ = permissible shear stress for spring wire (MPa or N/mm^2) K = Wahl factor

(*Contd.*)

(*Contd.*)

$\delta = \dfrac{8PD^3N}{Gd^4}$ (10.6) $k = \dfrac{Gd^4}{8D^3N}$ (10.7)	G = modulus of rigidity of spring wire (MPa or N/mm^2) N = number of active coils
$E = \dfrac{1}{2} P\,\delta$ (10.8)	E = strain energy stored in spring (N-mm)
Two springs in Series $\dfrac{1}{k} = \dfrac{1}{k_1} + \dfrac{1}{k_2}$ or $k = \dfrac{k_1 k_2}{k_1 + k_2}$ (10.9) Two springs in Parallel $k = k_1 + k_2$ (10.10)	k_1 = stiffness of spring 1 (N/mm) k_2 = stiffness of spring 2 (N/mm) k = combined stiffness of two springs (N/mm)
Buckling of helical springs	
A helical compression spring, that is too long compared to the mean coil diameter, acts as a flexible column and may buckle at a comparatively low axial force. The spring should be preferably designed buckle-proof. Compression springs, which cannot be designed buckle-proof, must be guided in a sleeve or over an arbor. The thumb rules for provision of guide are as follows: $\dfrac{\text{free length}}{\text{mean coil diameter}} \leq 2.6$ [Guide not necessary] $\dfrac{\text{free length}}{\text{mean coil diameter}} > 2.6$ [Guide required]	

Table 10.12 *Concentric springs*

A concentric spring consists of two helical compression springs, one inside the other, having the same axis. Assumptions: (i) The springs are made of the same material. (ii) The maximum torsional shear stresses induced in outer and inner springs are equal (iii) They have the same free length. (iv) Both springs are deflected by same amount and therefore, have same solid length.	
	d_1 = wire diameter of outer spring d_2 = wire diameter of inner spring D_1 = mean coil diameter of outer spring D_2 = mean coil diameter of inner spring P_1 = axial force transmitted by outer spring P_2 = axial force transmitted by inner spring P = total axial force δ_1 = deflection of outer spring δ_2 = deflection of inner spring N_1 = number of active coils in outer spring N_2 = number of active coils in inner spring τ_1 = maximum torsional shear stress in outer spring τ_2 = maximum torsional shear stress in inner spring C = spring index c = radial clearance between springs

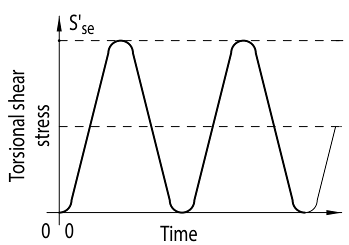
(*Contd.*)

(*Contd.*)

$\tau_1 = \tau_2$ $\delta_1 = \delta_2$	$d_1 N_1 = d_2 N_2$ (10.11)
$\frac{P_1}{P_2} = \frac{d_1^2}{d_2^2}$ (10.12)	$\frac{D_1}{d_1} = \frac{D_2}{d_2} = C$ = spring index (10.13)
$\frac{d_1}{d_2} = \frac{C}{(C-2)}$ (10.14)	$c = \frac{(d_1 - d_2)}{2}$ (10.15)

Applications: Concentric springs are used as valve springs in heavy duty diesel engines, aircraft engines and rail road suspensions.

Table 10.13 *Helical springs subjected to fluctuating load*

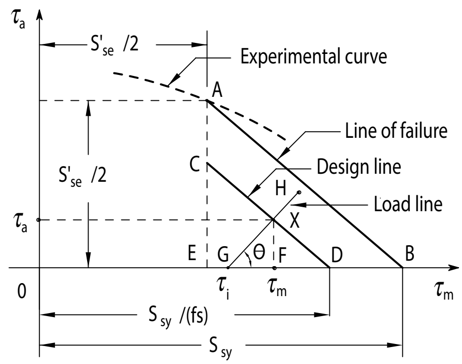
Mean and amplitude shear stresses	
$P_m = \frac{1}{2}(P_{max} + P_{min})$ (10.16) $P_a = \frac{1}{2}(P_{max} - P_{min})$ (10.17)	P_{max} = maximum value of fluctuating external force acting on spring (N) P_{min} = minimum value of fluctuating external force acting on spring (N) P_m = mean force (N) P_a = force amplitude (N)
$\tau_m = K_s\left(\frac{8P_m D}{\pi d^3}\right)$ (10.18) $K_s = \left(1 + \frac{0.5}{C}\right)$ (10.19)	τ_m = mean shear stress (MPa or N/mm^2) K_s = correction factor for direct shear stress
$\tau_a = K\left(\frac{8P_a D}{\pi d^3}\right)$ (10.20) $K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$ (10.21)	τ_a = torsional shear stress amplitude (MPa or N/mm^2) K = Wahl factor
Pulsating shear stress cycle for springs	
	A spring is never subjected to a completely reversed load, changing its magnitude from tension to compression and passing through zero with respect to time. A helical compression spring is subjected to purely compressive force. On the other hand, a helical extension spring is subjected to purely tensile force. In general, the spring wires are subjected to pulsating shear stresses, which vary from zero to (S'_{se}).

(*Contd.*)

(*Contd.*)

H. J. Elmendorf relationships	
For patented and cold-drawn steel wires (Grade 1 to 4), $S'_{se} = 0.21 S_{ut}$ (10.22) $S_{sy} = 0.42 S_{ut}$ (10.23) For oil-hardened and tempered steel wires (SW and VW grade), $S'_{se} = 0.22 S_{ut}$ (10.24) $S_{sy} = 0.45 S_{ut}$ (10.25)	S'_{se} = endurance limit in shear for the stress variation from zero to some maximum value (MPa or N/mm^2) S_{ut} = ultimate tensile strength (MPa or N/mm^2) S_{sy} = yield strength in shear (MPa or N/mm^2)

Fatigue diagram for spring



In this diagram, mean stress (τ_m) is plotted on the abscissa, while stress amplitude (τ_a) on the ordinate. Point *A* with coordinates $\left(\frac{1}{2}S'_{se}, \frac{1}{2}S'_{se}\right)$ indicates the failure-point of the spring wire in fatigue test with pulsating stress cycle. Point *B* on the abscissa indicates the failure under static condition, when the mean stress (τ_m) reaches the torsional yield strength (S_{sy}). Line $\overline{AB}$ is called the line of failure. To consider the effect of the factor of safety, a line $\overline{DC}$ is considered from point *D* on the abscissa in such a way that, $\overline{OD} = \frac{S_{sy}}{(fs)}$

The line $\overline{GH}$ is called load line. It is starts from point *G* on abscissa at a distance τ_i from the origin. The torsional shear stress due to initial pre-load on spring (P_i) is τ_i. The slope of the line $\overline{GH}$ (θ) is given by,

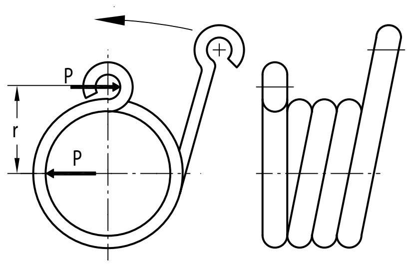
$$\tan\theta = \frac{\tau_a}{\tau_m}$$

The point of intersection between design line $\overline{DC}$ and load line $\overline{GH}$ is *X*. The co-ordinates of point *X* are (τ_m,. τ_a).

$$\frac{\tau_a}{\frac{S_{sy}}{(fs)} - \tau_m} = \frac{\frac{1}{2}S'_{se}}{S_{sy} - \frac{1}{2}S'_{se}} \qquad (10.26)$$

10.4 TORSION SPRINGS

Table 10.14 *Helical torsion springs*

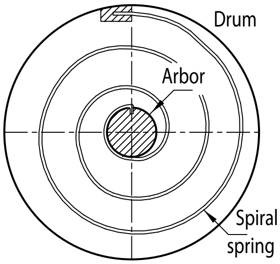
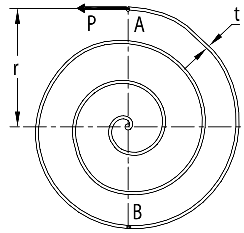


$M_b = Pr$	(10.27)	M_b = bending moment acting on torsion spring (N-mm) P = external force acting on spring (N) r = distance of line of action of external force from central axis of coils (mm)
$\sigma_b = K\left(\dfrac{32M_b}{\pi d^3}\right)$	(10.28)	σ_b = permissible bending stress for spring wire (MPa or N/mm^2) d = wire diameter (mm) K = stress concentration factor due to curvature
$K_i = \dfrac{4C^2 - C - 1}{4C(C-1)}$ $K_o = \dfrac{4C^2 + C - 1}{4C(C+1)}$	(10.29) (10.30)	K_i = stress concentration factors at the inner fibre of coil K_o = stress concentration factors at the outer fibre of coil C = spring index
$\theta = \dfrac{64PrDN}{Ed^4}$ $k = \dfrac{Ed^4}{64DN}$	(10.31) (10.32)	θ = angular deflection of spring (rad) D = mean coil diameter (mm) N = number of active coils k = stiffness of the helical torsion spring, i.e., bending moment required to produce unit angular displacement (N-mm/rad) E = modulus of elasticity (MPa or N/mm^2)

Note: The spring index is generally kept from 5 to 15. When it is less than 5, the strain on the coiling arbor of the torsion winder causes excessive tool breakage. When it is more than 15, the control over the spring pitch is lost.

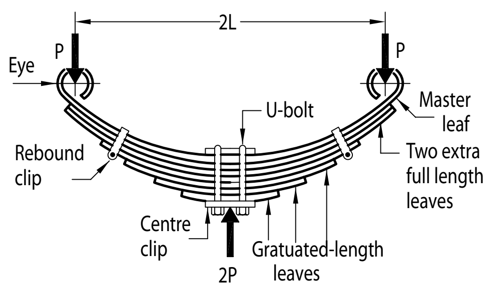
10.5 SPIRAL SPRINGS

Table 10.15 *Spiral springs*

	
$M = Pr$ (10.33)	P = force induced at the outer end A due to winding of the arbor (N) r = distance of centre of gravity of spiral from outer end (mm) M = bending moment due to force P acting at a distance r (N-mm)
$\sigma_b = \dfrac{12M}{bt^2}$ (10.34)	σ_b = permissible bending stress (MPa or N/mm^2) t = thickness of strip (mm) b = width of strip perpendicular to plane of paper (mm)
$\theta = \dfrac{12Ml}{Ebt^3}$ (10.35) $\delta = \dfrac{12Mlr}{Ebt^3}$ (10.36)	θ = rotation of arbor with respect to drum (rad) l = length of strip from outer end to inner end (mm) E = modulus of elasticity (MPa or N/mm^2) δ = deflection of one end of spring with respect to other end (mm)
$U = \dfrac{6M^2 l}{Ebt^3}$ (10.37)	U = strain energy stored in spring (N-mm)

10.6 LEAF SPRINGS

Table 10.16 *Semi-elliptic leaf springs*

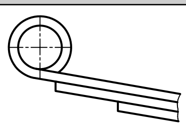
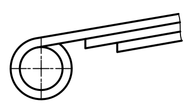
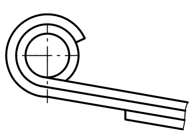
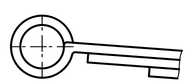
	
	$(\sigma_b)_g$ = bending stress in graduated length leaves (MPa or N/mm^2) $(\sigma_b)_f$ = bending stress in extra full-length leaves (MPa or N/mm^2)

(*Contd.*)

(*Contd.*)

$(\sigma_b)_g = \dfrac{12PL}{(3n_f + 2n_g)bt^2}$ (10.38) $(\sigma_b)_f = \dfrac{18PL}{(3n_f + 2n_g)bt^2}$ (10.39) $\delta = \dfrac{12PL^3}{Ebt^3(3n_f + 2n_g)}$ (10.40)	L = length of the cantilever or half the length of semi-elliptic spring (mm) P = force applied at the end of the spring (N) n_f = number of extra full-length leaves n_g = number of graduated-length leaves including master leaf b = width of each leaf (mm) t = thickness of each leaf (mm) δ = deflection at the end of spring (mm) E = modulus of elasticity (MPa or N/mm^2)
Nipping of leaf springs	
$C = \dfrac{2PL^3}{Enbt^3}$ (10.41) $P_i = \dfrac{2n_g n_f P}{n(3n_f + 2n_g)}$ (10.42) $\sigma_b = \dfrac{6PL}{nbt^2}$ (10.43)	C = 'nip' or initial gap between the extra full-length leaf and the graduated-length leaf before the assembly (mm) n = total number of leaves P_i = initial pre-load required to close the gap C between the extra full-length leaves and graduated-length leaves (N) σ_b = bending stress in all leaves (MPa or N/mm^2)
Nominal thickness of leaves	
t (mm) = 3.2, 4.5, 5, 6, 6.5, 7, 7.5, 8, 9, 10, 11, 12, 14 and 16	
Nominal width of leaves	
b (mm) = 32, 40, 45, 50, 55, 60, 65, 70, 75, 80, 90, 100 and 125	
Hardness of steel leaves	
Material	**Hardness after heat treatment**
50Cr4V2	409 to 485 HB
60Cr4V2	432 to 485 HB
55Si7	378 to 432 HB
60Si7	363 to 435 HB
65Si7	363 to 432 HB

Table 10.17 *Eye styles for semi-elliptic leaf springs*

Up turned eye	**Down turned eye**
	
Reinforced eye	**Berlin eye**
	

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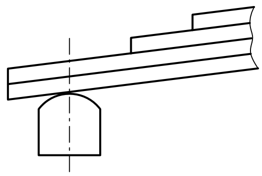
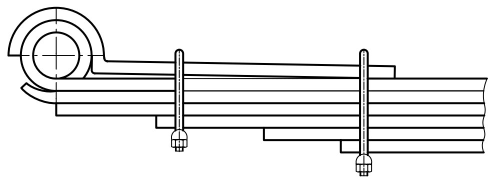
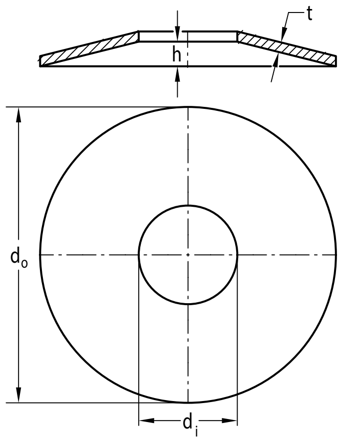
Plain end mounting	**Pretensioned spring eye**
	

10.7 BELLEVILLE SPRINGS

Table 10.18 *Belleville spring washers*



$$P = \frac{E\delta}{(1-\mu^2)M(d_o/2)^2}[(h-\delta/2)(h-\delta)t + t^3] \quad (10.44)$$

$$\sigma = \frac{E\delta}{(1-\mu^2)M(d_o/2)^2}[C_1(h-\delta/2) + C_2 t] \quad (10.45)$$

$$M = \frac{6}{\pi \log_e(d_o/d_i)}\left[\frac{(d_o/d_i)-1}{(d_o/d_i)}\right]^2 \quad (10.46)$$

$$C_1 = \frac{6}{\pi \log_e(d_o/d_i)}\left[\frac{(d_o/d_i)-1}{\log_e(d_o/d_i)} - 1\right] \quad (10.47)$$

$$C_2 = \frac{6}{\pi \log_e(d_o/d_i)}\left[\frac{(d_o/d_i)-1}{2}\right] \quad (10.48)$$

$$(P)_{\text{flat}} = \frac{Eht^3}{(1-\mu^2)M(d_o/2)^2} \qquad \text{when } (\delta = \text{h}) \quad (10.49)$$

P = axial force (N)
$(P)_{\text{flat}}$ = axial force at flat position when ($\delta = h$) (N)
δ = deflection of spring (m)
t = thickness of washer (m)
h = free height minus thickness (m)
E = modulus of elasticity (N/m^2)
σ = stress at the inside circumference (N/m^2)
d_o = outer diameter of washer (m)
d_i = inner diameter of washer (m)
μ = Poisson's ratio (0.3 for steel)

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(*Contd.*)

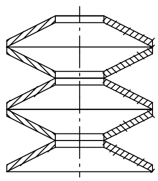
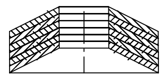
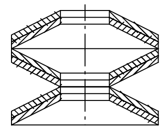
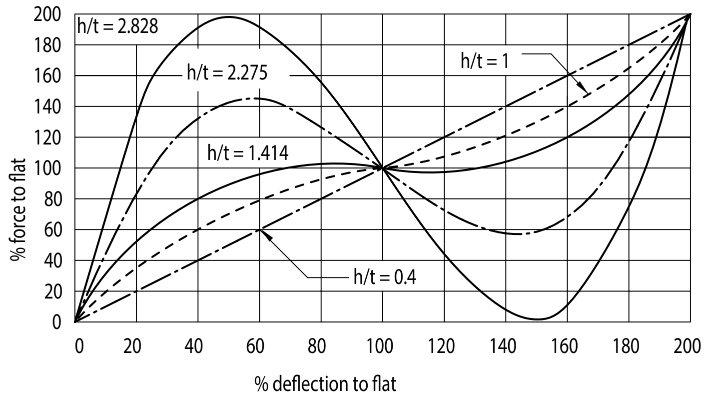
Nesting of Belleville spring washers		
Series combination	**Parallel combination**	**Parallel series combination**
		

Table 10.19 *Load-deflection characteristics of Belleville spring washers*



Significance

(i) The deflection and force of Belleville spring are zero at the free position of spring

(ii) One hundred per cent deflection represents the condition when the spring is completely flat and 100% force represents the axial spring force when the spring is flat. This is the central position of all characteristic curves.

(iii) When [(h/t) = 0.4], the spring rate is almost linear and similar to spring rate of helical spring.

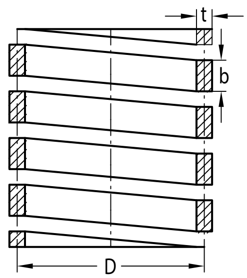
(iv) When [(h/t) > 0.4], the spring rate becomes non-linear.

(v) When [(h/t) = 1.414], the curve has a portion of nearly straight line at the central position where the spring is flat. In this region, the force deviates less than 1% of the force value at 100% deflection.

(vi) When [(h/t) > 1.414], the curves become bimodal. It indicates that for a given force, there is more than one possible deflection. If such a spring is mounted in such a way so as to allow it to go beyond flat condition, it will be bistable, requiring a force in either direction to trip its past centre.

10.8 HELICAL SPRINGS OF RECTANGULAR WIRE

Table 10.20 *Helical spring of rectangular wire*



Load-stress equation $$\tau = \frac{PD}{2\alpha b t^2} \quad (10.50)$$ Load-deflection equation $$\delta = \frac{\pi P D^3 N}{4\beta G b t^3} \quad (10.51)$$	τ = permissible shear stress for spring wire (MPa or N/mm^2) P = axial spring force (N) D = mean coil diameter (mm) b = side of cross section parallel to axis of spring (mm) t = side of cross section perpendicular to axis of spring (mm) δ = axial deflection of spring (mm) N = number of active coils G = modulus of rigidity of spring wire (MPa or N/mm^2) α, β = constants depending upon ratio (b/t) (Table 10.21)

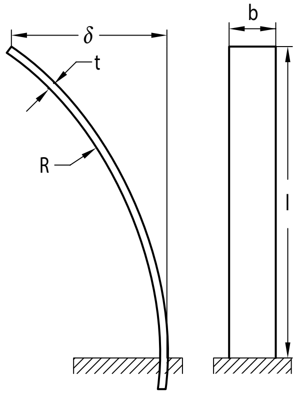
Table 10.21 *Constants α and β for helical spring of rectangular wire*

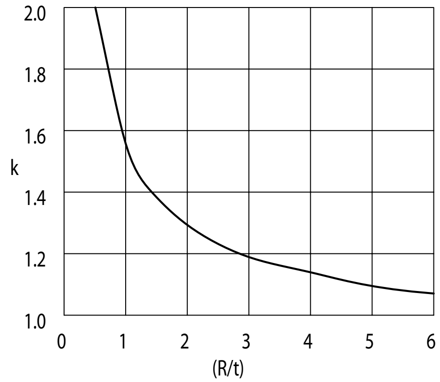
b/t	1.00	1.20	1.50	1.75	2.00	2.50	3.00	4.00	5.00	6.00	8.00	10.0	∞
α	0.208	0.219	0.231	0.239	0.246	0.258	0.267	0.282	0.291	0.299	0.307	0.312	0.333
β	0.1406	0.166	0.196	0.214	0.229	0.249	0.263	0.281	0.291	0.299	0.307	0.312	0.333

(*Contd.*)

10.9 FLAT FORM SPRINGS

Table 10.22 *Flat form springs*



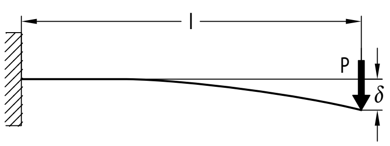


Notations

b = width of spring (mm)	E = modulus of elasticity (MPa or N/mm^2)
t = thickness of spring (mm)	σ_b = bending stress in spring (for static loading) (MPa or N/mm^2)
l = effective length of spring (mm)	σ_{bh} = permissible stroke stress (for dynamic loading) (MPa or N/mm^2)
R = radius assumed by spring on loading (mm)	S_{ut} = ultimate tensile strength of strip material (MPa or N/mm^2)
P = spring force (N)	
δ = deflection of spring (mm)	
k = bending stress correction factor	

Stress and deflection equations

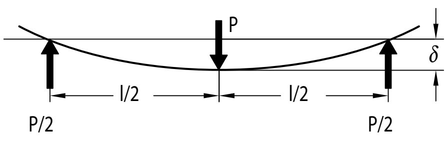
(i) Cantilever spring with end load



$$\sigma_b = k\left(\frac{6Pl}{bt^2}\right) \quad (10.52)$$

$$\delta = \frac{4Pl^3}{Ebt^3} \quad (10.53)$$

(ii) Simply supported spring with centre load



$$\sigma_b = k\left(\frac{3Pl}{2bt^2}\right) \quad (10.54)$$

$$\delta = \frac{Pl^3}{4Ebt^3} \quad (10.55)$$

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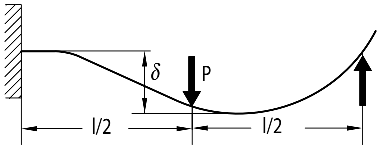
(*Contd.*)

(iii) One end fixed and one end simply supported spring with centre load	
	$$\sigma_b = k\left(\frac{9Pl}{8bt^2}\right) \quad (10.56)$$ $$\delta = \frac{7Pl^3}{64Ebt^3} \quad (10.57)$$
Permissible stresses	
For static loading $\sigma_b = 0.75 S_{ut}$ (10.58)	For dynamic loading $\sigma_{bh} = 0.6(0.75 S_{ut})$ (10.59)
Modulus of elasticity	
Material	E (MPa or N/mm^2)
(i) Cold-rolled steel strip	206 000
(ii) Austenitic stainless steel strip	190 200
(iii) Phosphor bronze strip	105 000
(iv) Copper beryllium strip	127 000

11 Friction Clutches

11.1 SINGLE AND MULTI-PLATE CLUTCHES

Table 11.1 *Service factors for clutches (K_s)*

Type of Prime mover	**Type of Driven equipment**			
	Light steady loads (Starting torque is equal to or slightly greater than running torque)- centrifugal pumps, light-duty fans and blowers, liquid mixers and agitators, centrifugal compressors, gear pumps, textile and wood working machinery	Moderate loads (High starting torque or above average running torque)- machine tools, heavy-duty centrifugal pumps, cooling towers, slurry agitators, boiler feed pumps, hoists and conveyors	Medium load (Starting torque is approximately double running torque) - dredge pumps, light-duty hammermills, lineshafts, paper converting machinery, rotary kilns, rotary or screw-type pumps for high viscosity fluids	Heavy duty loads (High starting torque, shock loading, light torque reversals during drive)- mine ventilating fans, reciprocating pumps or compressors, Paper making machinery, heavy-duty hammermills, ore crushers, pulverising mills
Steam and gas turbines	1.00	1.25	1.50	1.75
AC electric motors	1.25	1.50	1.50	1.75
DC electric motors, hydraulic motors	1.25	1.50	1.75	2.00
Spark ignition engines (Petrol, natural gas or propane)	1.75	1.75	2.00	(*)
Compression ignition engines (Diesel)	2.00	2.50	2.75	(*)

(*) To be consulted with manufacturers

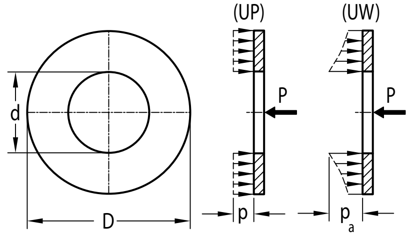
Note: $(M_t)_{des} = K_s\,(M_t)$

where K_s = service factor

$(M_t)_{des}$ = torque capacity of clutch for design purpose

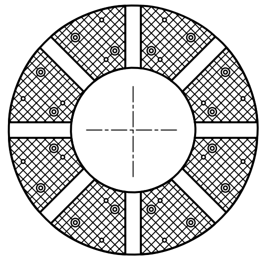
(M_t) = rated torque

Table 11.2 *Single and multi-plate clutches*

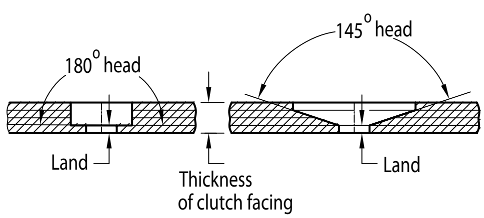


Notations: D = outer diameter of friction disk (mm) d = inner diameter of friction disk (mm) p = intensity of pressure at radius 'r'(MPa or N/mm^2) P = total operating force (N) M_t = torque transmitted by the clutch (N-mm)	μ = coefficient of friction p_a = maximum pressure intensity at inner radius (or at d/2) (MPa or N/mm^2) R_f = friction radius (mm) z = number of pairs of contacting surface z_1 = number of disks on driving shaft z_2 = number of disks on driven shaft
Single-plate clutch	
Uniform pressure theory (UP)	*Uniform wear theory (UW)*
p = constant (11.1)	$p\,r = p_a$ (d/2) = constant (11.7)
$P = \dfrac{\pi p}{4}(D^2 - d^2)$ (11.2)	$P = \dfrac{\pi p_a d}{2}(D - d)$ (11.8)
$M_t = \dfrac{\pi \mu p}{12}(D^3 - d^3)$ (11.3)	$M_t = \dfrac{\pi \mu p_a d}{8}(D^2 - d^2)$ (11.9)
$M_t = \dfrac{\mu P}{3}\dfrac{(D^3 - d^3)}{(D^2 - d^2)}$ (11.4)	$M_t = \dfrac{\mu P}{4}(D + d)$ (11.10)
$M_t = \mu P R_f$ (11.5)	$M_t = \mu P R_f$ (11.11)
$R_f = \dfrac{1}{3}\dfrac{(D^3 - d^3)}{(D^2 - d^2)}$ (11.6)	$R_f = \dfrac{1}{4}(D + d)$ (11.12)
Note: (i) Use uniform wear theory for design of clutches. (ii) The ratio (D/d) is usually taken from1.5 to 2.0.	
Multi-plate clutch	
Uniform pressure theory	*Uniform wear theory*
$M_t = \dfrac{\mu P z}{3}\dfrac{(D^3 - d^3)}{(D^2 - d^3)}$ (11.13)	$M_t = \dfrac{\mu P z}{4}(D + d)$ (11.14) $z = (z_1 + z_2 - 1)$ (11.15)

Table 11.3 *Clutch facings for automotive transmission*



Preferred dimensions for clutch facings	
Preferred outside diameter (mm)	120, 125, 130, 135, 140, 145, 150, 155, 160, 170, 180, 190, 200, 210, 220, 230, 240, 250, 260, 270, 280, 290, 300, 325, 350
Preferred inside diameter (mm)	80, 85, 90, 95, 100, 105, 110, 120, 130, 140, 150, 175, 200
Preferred thickness (mm)	3, 3.5, 4
Permissible deviations for outside diameter	
Outside diameter (mm)	*Permissible deviation (mm)*
120–160	0.0 –0.5
170–300	0.0 –0.8
Over 300	0.0 –1.0
Permissible deviations for inside diameter	
Inside diameter (mm)	*Permissible deviation (mm)*
80–110	+0.5 0.0
120–150	+0.8 0.0
Over 150	+1.0 0.0
Permissible deviations for thickness	
Thickness (mm)	*Permissible deviation (mm)*
3, 3.5, 4	±0.1
Rivet holding land	



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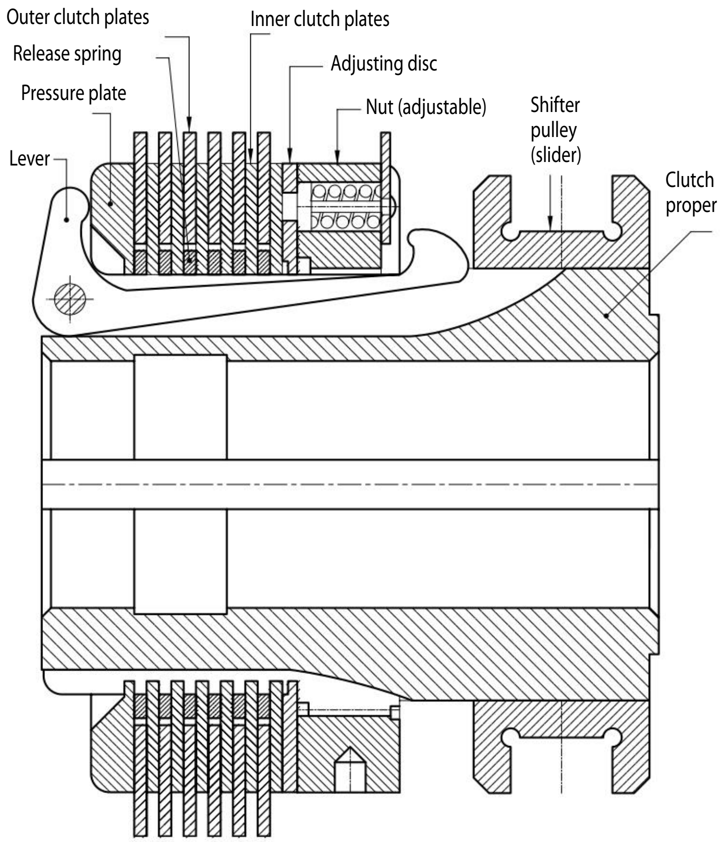
The rivet holding land in drilled facing should not be less than 1.45 mm. Also, it should not be more than half of the thickness of facing.	
Types of clutch facings	
Type A	Solid woven or piled fabric with or without metallic reinforcement
Type B	Moulded or semi-moulded compound
Requirements of clutch facings	
(i) The average values of coefficient of friction should be within the range 0.25 to 0.35 during the friction test. The temperature at the interface during this test should be 150° ± 10°C. (ii) The wear of clutch facing should not be more than 15 cm^3 per 100 horsepower-hour. (iii) The facing should lie flat after riveting to the plate. The difference in thickness from point to point (parallelism) should not exceed 0.08 mm.	

Table 11.4 *Mechanical multi-disc clutches (wet type)*

Single acting multi-disc clutch



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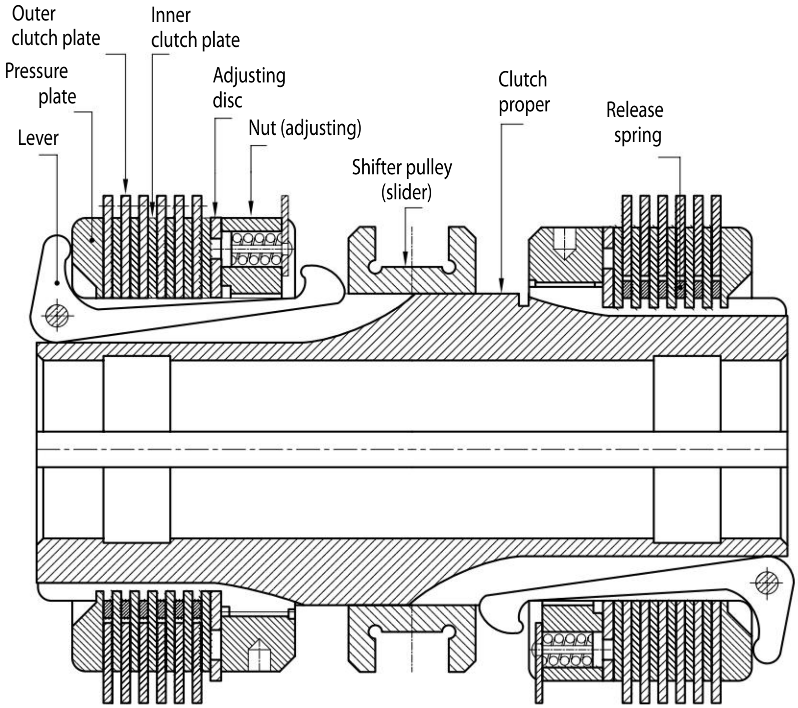
Double acting multi-disc clutch

Construction: (i) The clutch consists of number of pairs of inner and outer plates. The inner plates have internal lugs or splines to engage in hub of the clutch. The outer plates have external lugs or involute teeth to engage with outer housing. One of the pairs of clutch plates may be made from phosphor bronze, while the other shall be made of steel. (ii) Slider is a ring collar that is used to shift and operate the clutch. A pivoted lever is used to transmit force from the slider to the clutch plates. Adjusting nut is a ring nut provided on clutch to adjust normal pressure between plates. Hub is the body of clutch on which inner clutch plates, levers, adjusting nut and sliders are mounted. Hub also receives one of the drive or driven shaft of the machinery. Outer housing engages the outer clutch plates. It also engages one of the drive or driven shaft of the machinery. (iii) The clutch is engaged by shifting the slider which will apply pressure on the clutch plates through the lever. The pressure on the plates creates frictional force to engage the clutch without slip. (iv) Single acting clutch implies a single clutch unit, which is used to engage or disengage only one drive. (v) Double acting clutch implies two clutch units, which is used to engage either one or other drives by a common operation. The double acting clutch may also have a neutral position where both the drives are disengaged simultaneously.

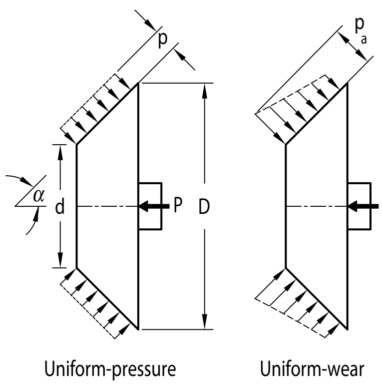
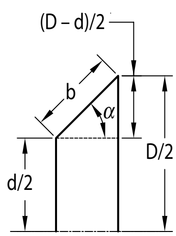
11.2 FRICTION MATERIALS FOR CLUTCHES

Table 11.5 *Coefficient of friction and permissible pressure for clutches*

Contacting surfaces	**Coefficient of friction**		**Permissible pressure (MPa or N/mm^2)**
	Wet	**Dry**	
Cast iron-cast iron	0.05	0.15–0.20	1.00–1.75
Leather-cast iron/steel	0.12	0.3–0.5	0.07–0.28
Cork-cast iron/steel	0.15–0.25	0.3–0.5	0.05–0.10
Woven asbestos-cast iron/steel	0.10–0.20	0.3–0.6	0.35–0.70
Moulded asbestos-cast iron/steel	0.08–0.12	0.2–0.5	0.35–1.00
Sintered metal-cast iron	0.05–0.10	0.1–0.4	1.00
Sintered metal-steel	0.05–0.10	0.1–0.3	2.10

11.3 CONE CLUTCHES

Table 11.6 *Cone clutches*

	
Notations: D = outer diameter of cone clutch (mm) d = inner diameter of cone clutch (mm) α = semi-cone angle of clutch (º) b = face width of friction lining (mm) p = intensity of pressure at radius 'r' from axis of clutch (MPa or N/mm^2)	P = total operating force (N) M_t = torque transmitted by the clutch (N-mm) μ = coefficient of friction p_a = maximum pressure intensity at inner radius (or at d/2) (MPa or N/mm^2)

(*Contd.*)

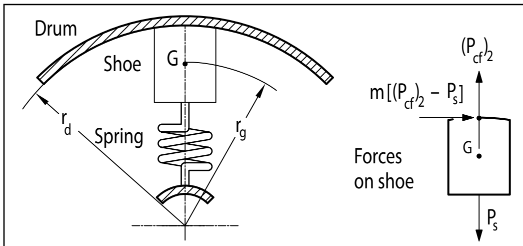
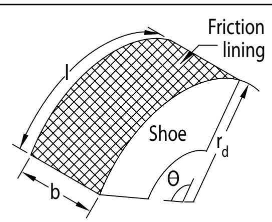
(*Contd.*)

Uniform pressure theory		Uniform wear theory	
$p = \text{Constant}$		$pr = p_a (d/2) = \text{Constant}$	
$P = \dfrac{\pi p}{4}(D^2 - d^2)$	(11.16)	$P = \dfrac{\pi p_a d}{2}(D - d)$	(11.19)
$M_t = \dfrac{\pi \mu p}{12 \sin \alpha}(D^3 - d^3)$	(11.17)	$M_t = \dfrac{\pi \mu p_a d}{8 \sin \alpha}(D^2 - d^2)$	(11.20)
$M_t = \dfrac{\mu P}{3 \sin \alpha} \dfrac{(D^3 - d^3)}{(D^2 - d^2)}$	(11.18)	$M_t = \dfrac{\mu P}{4 \sin \alpha}(D + d)$	(11.21)
$b = \dfrac{D - d}{2 \sin \alpha}$			(11.22)

Note: (i) Use uniform wear theory for design of clutches. (ii) The semi-cone angle of clutch is usually taken as 12.5º.

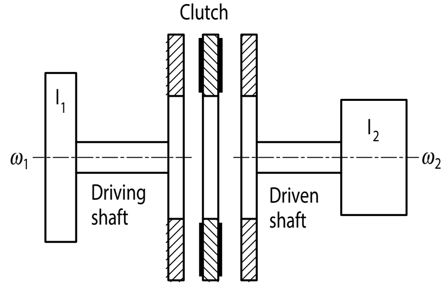
11.4 CENTRIFUGAL CLUTCHES

Table 11.7 *Centrifugal clutches*

		
$P_s = \dfrac{m\omega_1^2 r_g}{1000}$	(11.23)	P_s = spring force (N) P_{cf} = centrifugal force (N) m = mass of each shoe (kg) ω_1 = speed at which engagement starts (rad/s) r_g = radius of the centre of gravity of the shoe in engaged position (mm)
Net force on drum $= \dfrac{mr_g(\omega_2^2 - \omega_1^2)}{1000}$	(11.24)	ω_2 = running speed (rad/s)
Friction force $= \dfrac{\mu mr_g(\omega_2^2 - \omega_1^2)}{1000}$	(11.25)	μ = coefficient of friction between friction lining and drum
$M_t = \dfrac{\mu mr_g r_d z(\omega_2^2 - \omega_1^2)}{1000}$	(11.26)	z = number of shoes r_d = radius of drum (mm) M_t = frictional torque (N-mm)
$l = r_d \theta$	(11.27)	l = contact length of friction lining with drum (mm) θ = contact angle of friction lining or shoe with drum (rad)
$(p_a l b) = \dfrac{mr_g(\omega_2^2 - \omega_1^2)}{1000}$	(11.28)	b = width of friction lining (mm) p_a = permissible pressure on friction lining (MPa or N/mm^2)

11.5 ENERGY EQUATIONS

Table 11.8 *Energy equations for clutches*

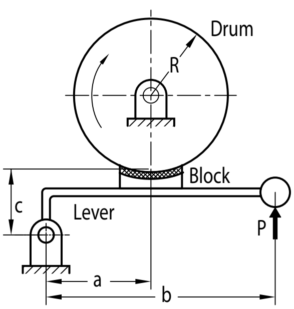
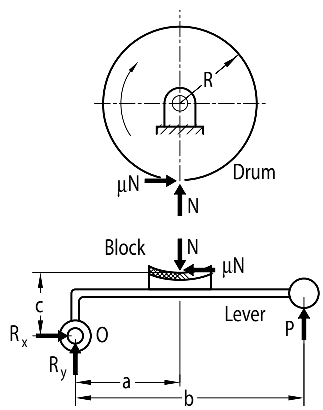


$\omega = (\omega_1 - \omega_2) - M_t\left(\dfrac{I_1 + I_2}{I_1 I_2}\right)t$	(11.29)	ω = difference in velocities of driving and driven shafts or relative velocity at time 't' (rad/s) ω_1 = angular velocity of driving shaft (rad/s) ω_2 = angular velocity of driven shaft (rad/s) M_t = clutch torque (N-m) I_1 = moment of inertia of driving shaft(kg-m^2) I_2 = moment of inertia of driven shaft (kg-m^2) t = time (s)
$t_1 = \dfrac{(\omega_1 - \omega_2) I_1 I_2}{(I_1 + I_2) M_t}$	(11.30)	t_1 = time taken by clutching operation or time at the instance when the angular velocities of two shafts become equal (s)
$u = M_t\left[(\omega_1 - \omega_2) - M_t\left(\dfrac{I_1 + I_2}{I_1 I_2}\right)t\right]$	(11.31)	u = rate of energy dissipation (J/s)
$E = \dfrac{1}{2}\dfrac{(\omega_1 - \omega_2)^2 I_1 I_2}{(I_1 + I_2)}$	(11.32)	E = total energy dissipated or heat generated during clutching operation (J)

Brakes 12

12.1 BLOCK BRAKES

Table 12.1 *Block brake with fixed short shoe*

<table>
<tr>
<td>
</td>
<td>
</td>
</tr>
<tr>
<td colspan="2">Assumptions</td>
</tr>
<tr>
<td colspan="2">(i) The block is rigidly attached to the lever.
(ii) The angle of contact between the block and brake drum is small, resulting in a uniform pressure distribution.
(iii) The brake drum is rotating in a clockwise direction.</td>
</tr>
<tr>
<td>$M_t = \mu N R$ (12.1)</td>
<td>M_t = braking torque (N-mm)
R = radius of the brake drum (mm)
μ = coefficient of friction
N = normal reaction (N)</td>
</tr>
<tr>
<td>$N = p\, l\, w$ (12.2)</td>
<td>p = permissible pressure between the block and the brake drum (MPa or N/mm^2)
l = length of the block (mm)
w = width of the block (mm)</td>
</tr>
</table>

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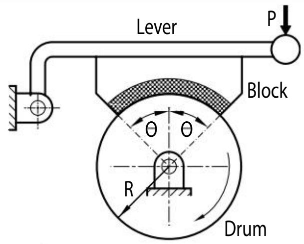
Optimum width of block	
$$\frac{1}{4}(\text{drum diameter}) < w < \frac{1}{2}(\text{drum diameter}) \qquad (12.3)$$	
$R_x = \mu N$ (12.4)	R_x = hinge pin reaction in X direction (N)
$R_y = (N - P)$ (12.5)	R_y = hinge pin reaction in Y direction (N)
Actuating force on lever (P) (Three cases)	
$P = \dfrac{(a - \mu c)}{b} \times N$ (12.6)	P = actuating force on lever (N)
Case I ($a > \mu c$)	
In this case, the friction force (μN) helps to reduce the magnitude of the actuating force P. It is seen from the free-body diagram, that the moment due to braking effort ($P \times b$), and moment due to friction force ($\mu N \times c$) are both anticlockwise. Such a brake is called a partially 'self-energizing' brake. However, the brake is not self-locking, because a small magnitude of positive force P is required for the braking action. This is a very desirable condition.	
Case II ($a = \mu c$)	
In this case, the actuating force P is zero. This indicates that no external force is required for the braking action. Such a brake is called 'self-locking' brake. This is not desirable condition in normal applications. Some positive braking effort (P) should be required to apply the brake, otherwise the brake will be out of control of the operator.	
Case III ($a < \mu c$)	
Under this condition, the actuating force P becomes negative. This is a dangerous operating condition, resulting in uncontrolled braking and grabbing. The brake is out of control of the operator because he cannot apply it.	
Desirable condition	
In designing block brakes, care should be taken to see that the brake is not self-locking and, at the same time, full advantage of the partial self-energizing effect should be taken to reduce the magnitude of the braking effort P. The condition to avoid self-locking is given by, $$a > \mu c \qquad (12.7)$$	

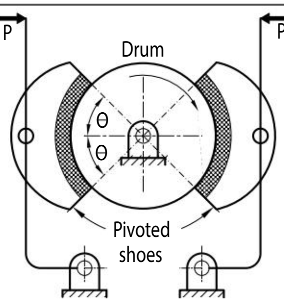
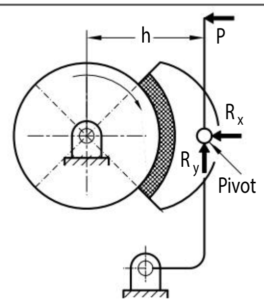
(*Contd.*)

Table 12.2 *Block brake with fixed long shoe*

	
$\mu' = \mu\left[\dfrac{4\sin\theta}{2\theta + \sin 2\theta}\right]$ (12.8) Friction force = $\mu' N$ $M_t = \mu' N R$ (12.9)	M_t = braking torque (N-mm) R = radius of the brake drum (mm) μ = actual coefficient of friction between friction block and drum μ' = equivalent coefficient of friction N = normal reaction (N) θ = semi-block angle (rad)

Note: The examples of block brake with fixed 'long' shoe, can be solved by using the same equations, which are derived for block brake with fixed 'short' shoe, by replacing coefficient of friction (μ) by equivalent coefficient of friction (μ').

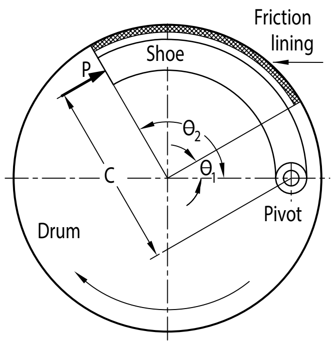
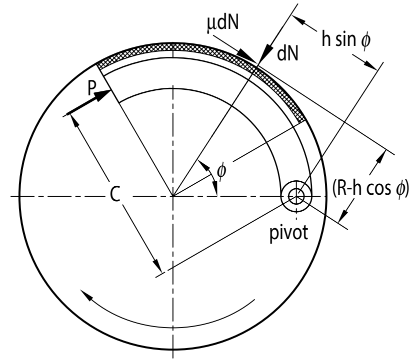
Table 12.3 *Pivoted block brake with long shoe*

	
$h = \dfrac{4R\sin\theta}{2\theta + \sin 2\theta}$ (12.10)	h = distance of pivot from axis of brake drum (mm) R = radius of brake drum (mm) θ = semi-block angle of each shoe (rad)
$M_t = 2\mu R^2 w\, p_{\max} \sin\theta$ (12.11)	M_t = braking torque capacity of each shoe (N-mm) μ = coefficient of friction between friction lining and drum w = width of friction lining (mm) p_{max} = maximum intensity of pressure between friction lining and brake drum (MPa or N/mm^2)
$R_x = \dfrac{1}{2} R w p_{\max}(2\theta + \sin 2\theta)$ (12.12)	R_x = reaction at each pivot in X direction (N)
$R_y = \dfrac{1}{2}\mu R w p_{\max}(2\theta + \sin 2\theta)$ (12.13)	R_y = reaction at each pivot in Y direction (N)

Note: In case of the pivoted shoe brake, the location of the pivot (h) from the axis of brake drum is selected in such a way that the moment of frictional force about the pivot is zero. This is the main advantage of the pivoted shoe brake. On the other hand, when the block is rigidly fixed to the lever, the tendency of the frictional force (μN) is to unseat the block with respect to the lever.

12.2 INTERNAL EXPANDING BRAKES

Table 12.4 *Internal expanding brake*

	
Pressure distribution on friction lining	
$$p = \frac{p_{\max}\sin\phi}{\sin\phi_{\max}} \qquad (12.14)$$ **Note:** $\phi_{\max} = 90^\circ$ when $\theta_2 > 90^\circ$ $\phi_{\max} = \theta_2$ when $\theta_2 < 90^\circ$ (12.15)	p = normal pressure on elemental friction lining located at centre angle ϕ from the shoe pivot and subtending an angle $d\phi$ (MPa or N/mm^2) p_{max} = maximum intensity of pressure for friction lining (MPa or N/mm^2) θ_1 = centre angle from shoe pivot to heel of friction lining (°) θ_2 = centre angle from shoe pivot to toe of friction lining (°)
Moment of frictional force about the axis of shoe pivot (M_f)	
$$dN = pRw\,d\phi \qquad (12.16)$$ $$M_f = \int \mu\, dN (R - h\cos\phi) \qquad (12.17)$$ $$M_f = \frac{\mu p_{\max} Rw}{\sin\phi_{\max}} \int_{\theta_1}^{\theta_2} \sin\phi (R - h\cos\phi) d\phi \qquad (12.18)$$	dN = normal reaction on elemental friction lining located at centre angle ϕ from the shoe pivot and subtending an angle $d\phi$ (N) R = internal radius of brake drum (mm) w = face width of friction lining on shoe (mm) M_f = moment of frictional force about the axis of shoe pivot (N-mm) μ = coefficient of friction h = distance of pivot from axis of brake drum (mm)
$$M_f = \frac{\mu p_{\max} Rw[4R(\cos\theta_1 - \cos\theta_2) - h(\cos 2\theta_1 - \cos 2\theta_2)]}{4\sin\phi_{\max}} \qquad (12.19)$$	
Moment of normal force about the axis of shoe pivot (M_n)	
$$M_n = \int dN (h\sin\phi) \qquad (12.20)$$ $$M_n = \frac{p_{\max} Rwh}{\sin\phi_{\max}} \int_{\theta_1}^{\theta_2} \sin^2\phi\, d\phi \qquad (12.21)$$	M_n = moment of normal force about the axis of shoe pivot (N-mm)
$$M_n = \frac{p_{\max} Rwh[2(\theta_2 - \theta_1) - (\sin 2\theta_2 - \sin 2\theta_1)]}{4\sin\phi_{\max}} \qquad (12.22)$$ (**Note:** θ_1 and θ_2 are in radians)	

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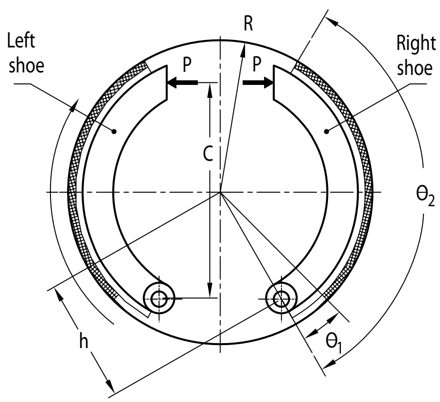
Braking torque (M_t)	
$M_t = \int \mu \, dNR$ (12.23)	$M_t = \dfrac{\mu R^2 p_{max} w}{\sin \phi_{max}} \int_{\theta_1}^{\theta_2} \sin \phi \, d\phi$ (12.24)
$M_t = \dfrac{\mu R^2 p_{max} w(\cos \theta_1 - \cos \theta_2)}{\sin \phi_{max}}$ (12.25)	
Actuating force (P)	
For clockwise rotation of brake drum, $P = \dfrac{M_n - M_f}{C}$ (12.26) For anti-clockwise rotation of brake drum, $P = \dfrac{M_n + M_f}{C}$ (12.27)	P = actuating force (N) C = distance of actuating force from the axis of shoe pivot (mm)

Table 12.5 *Automotive-type double shoe internal expanding brake*



Right-hand shoe	Left-hand shoe
$P = \dfrac{M_n - M_f}{C}$ (12.28)	$P = \dfrac{M'_n + M'_f}{C}$ (12.29)
p_{max} = maximum intensity of pressure for right shoe	p'_{max} = maximum intensity of pressure for left shoe $p'_{max} = p_{max} \left[\dfrac{CP\, p_{max}}{M_n + M_f} \right]$ (12.30)
Total braking torque	
$M_t = (M_t)_R + (M_t)_L$ (12.31) M_t = total torque-absorbing capacity of brake (N-mm) $(M_t)_R$ = torque-absorbing capacity of right-hand shoe (N-mm) $(M_t)_L$ = torque-absorbing capacity of left-hand shoe (N-mm)	
$M_t = \dfrac{\mu R^2 w(\cos \theta_1 - \cos \theta_2)}{\sin \phi_{max}} (p_{max} + p'_{max})$ (12.32)	

(*Contd.*)

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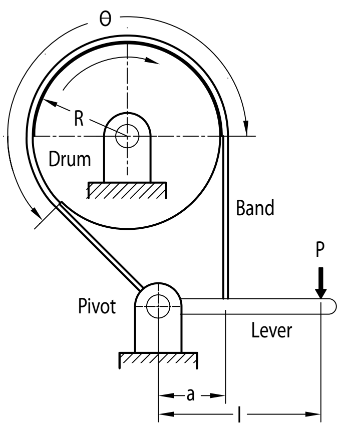
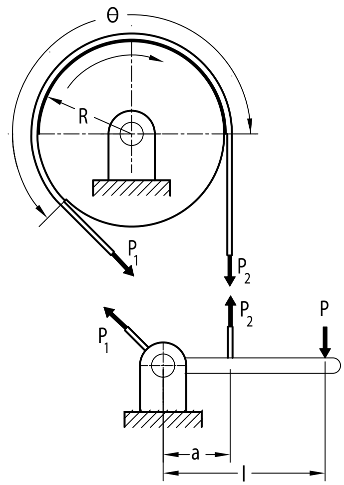
Pedal force and deceleration
(i) The maximum force exerted with right foot for the fifth percentile female is 22 N and for the male approximately 42 N. Therefore from ergonomic considerations, the brake system should be designed for a maximum pedal force of 22 to 25 N. With booster, the pedal force can be as low as 11 to 17 N. (ii) From ergonomic considerations, the pedal travel should not exceed 150 mm. (iii) When no data is available, the brake system should be designed so as to obtain a deceleration of 1g (i.e., 9.81 m/s^2), for a fully loaded vehicle.

12.3 BAND BRAKES

Table 12.6 *Simple band brake*

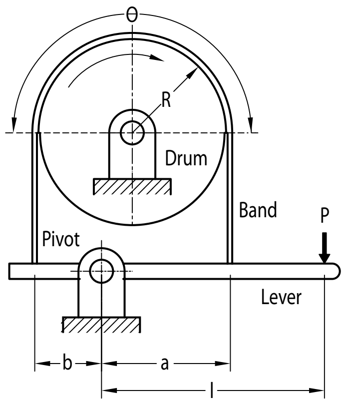
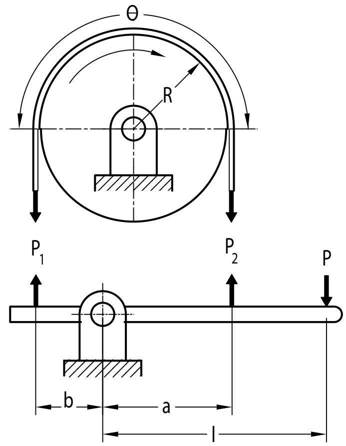




When one end of the band passes through the fulcrum of the actuating lever, the brake is called 'simple' band brake.

$$\frac{P_1}{P_2} = e^{\mu\theta} \qquad (12.33)$$ $$P_1 = \sigma_t w t \qquad (12.34)$$	P_1 = tension on tight side of band (N) P_2 = tension on loose side of band (N) μ = coefficient of friction between friction lining and brake drum θ = angle of wrap (rad) σ_t = permissible tensile stress for band material (MPa or N/mm^2) w = width of steel band parallel to axis of brake drum (mm) t = thickness steel band (mm)
$$M_t = (P_1 - P_2)R \qquad (12.35)$$	M_t = torque capacity of brake (N-mm) R = radius of brake drum (mm)
$$P = \frac{P_2 a}{l} \qquad (12.36)$$ $$p_{max} = \frac{P_1}{Rw} \qquad (12.37)$$	p_{max} = maximum intensity of pressure between friction lining attached to band and brake drum (MPa or N/mm^2) R = radius of brake drum (mm) w = width of friction lining parallel to axis of brake drum (mm)

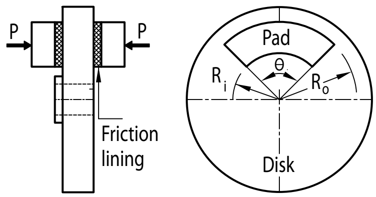
Table 12.7 *Differential band brake*

	
When neither end of the band passes through the fulcrum of the actuating lever, the brake is called 'differential' band brake.	
$\frac{P_1}{P_2} = e^{\mu\theta}$ (12.38) $P_1 = \sigma_t wt$ (12.39)	P_1 = tension on tight side of band (N) P_2 = tension on loose side of band (N) μ = coefficient of friction between friction lining and brake drum θ = angle of wrap (rad) σ_t = permissible tensile stress for band material (MPa or N/mm^2) w = width of steel band parallel to axis of brake drum (mm) t = thickness steel band (mm)
$M_t = (P_1 - P_2)\, R$ (12.40)	M_t = torque capacity of brake (N-mm) R = radius of brake drum (mm)
$P = \frac{P_2 a - P_1 b}{l}$ (12.41) $p_{\max} = \frac{P_1}{Rw}$ (12.42)	$p_{\max}$ = maximum intensity of pressure between friction lining attached to band and brake drum (MPa or N/mm^2) R = radius of brake drum (mm) w = width of friction lining parallel to axis of brake drum (mm)
Condition of self-locking	
$\left(\frac{a}{b}\right) \leq e^{\mu\theta}$ (12.43)	

12.4 DISK BRAKES

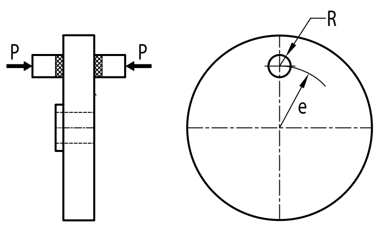
Table 12.8 *Disk brake*

Disk brake with annular pad



$R_f = \dfrac{2}{3}\dfrac{(R_o^3 - R_i^3)}{(R_o^2 - R_i^2)}$ (12.44)	R_o = outer radius of pad (mm) R_i = inner radius of pad (mm) R_f = friction radius (mm) (Uniform Pressure Theory)
$M_t = \mu P R_t$ (12.45)	M_t = torque capacity of pad (N-mm) μ = coefficient of friction P = actuating force (N)
$A = \dfrac{1}{2}\theta(R_o^2 - R_i^2)$ (12.46)	A = area of pad (mm^2) θ = angle subtended by pad with axis of disk (rad)
$P = p_{ave} A$ (12.47)	p_{ave} = average pressure between friction lining and disk (MPa or N/mm^2)

Disk brake with circular pad



$R_f = \delta e$ (12.48)	R_f = friction radius (mm) e = distance of pad centre from axis of disk (mm) R = radius of pad (mm)

Values of δ for circular-pad caliper disk brake

(R/e)	δ
0.0	1.0000
0.1	0.9833
0.2	0.9693
0.3	0.9572
0.4	0.9467
0.5	0.9375

12.5 FRICTION MATERIALS FOR BRAKES

Table 12.9 *Properties of friction materials for brakes*

Material	Coefficient of friction	Permissible temperature (ºC)	Intensity of pressure (MPa or N/mm^2)
Cast iron on cast iron	0.15–0.20	300	1.00
Wood on cast iron	0.25–0.30	60	0.35
Leather on cast iron	0.30–0.50	60	0.25
Woven-asbestos on metal	0.35–0.40	250	0.65
Moulded-asbestos on metal	0.40–0.45	250	1.00
Sintered metal on metal	0.20–0.40	300	2.75

Table 12.10 *Recommended values of product (pv)*

[p = *Intensity of normal pressure* (MPa or N/mm^2), v = *Rubbing speed* (m/min)]

Application	Product (pv)
Intermittent applications, comparatively long period of rest and poor dissipation of heat	115
Continuous application and poor dissipation of heat	58
Continuous application and good dissipation of heat	175
Vehicle-brakes	125

Note: In preliminary design of brakes, product (pv) is considered in place of temperature rise.

12.6 AUTOMOTIVE BRAKE LINING

Table 12.11 *Types and classes of automotive brake lining*

Type I: Rigid moulded sets or flexible moulded rolls or sets		
	Class A	Medium friction
	Class B	High friction
Type II: Rigid woven sets or flexible woven rolls or sets		
	Class A	Medium friction
	Class B	High friction

Table 12.12 *Tolerances on widths and thicknesses for moulded and woven types of brake lining*

	Tolerance on thickness (mm)	Tolerance on width (mm)
Up to and including 5 mm thickness	+0 –0.2	+0 –0.8
Over 5 mm thickness	+0 –0.3	+0 –0.8

Note: The variation in thickness along the width of lining should not be more than 0.2 mm.

Table 12.13 *Coefficient of friction for automotive brake lining*

Type and class of brake lining	Range of coefficient of friction (μ)	Permissible variation for normal test in percent of (μ)
IA	0.28–0.40	+30 –20
IB	0.36–0.45	+30 –20
IIA	0.33–0.43	+20 –30
IIB	0.43–0.53	+20 –30

13 Belt Drives

13.1 FLAT AND V BELTS

Table 13.1 *Construction of flat and V-belts*

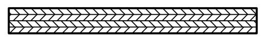
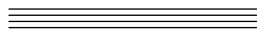
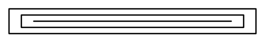
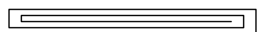
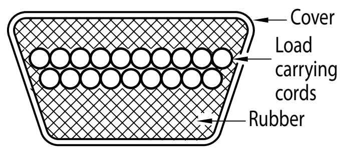
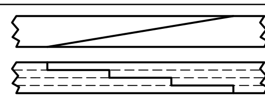
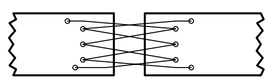
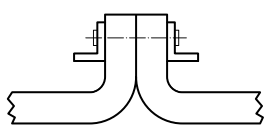
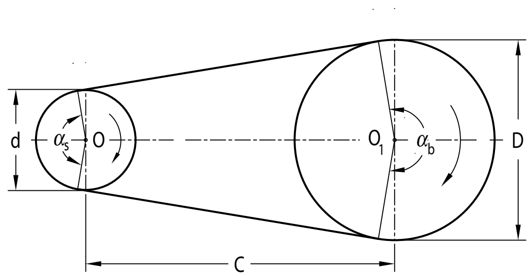
Four-ply leather belt	
Canvas or fabric rubber belt	 (a) Raw-edge belt  (b) Layer by layer folded edge belt  (c) Spirally wrapped folded edge belt
Polyester fabric V belt	
Flat belt joints	 (a) Cemented joint  (b) Laced joint  (c) Hinged joint

Table 13.2 *Geometrical relationships of belt drives*

Notations

α_s = wrap angle for small pulley (degrees)
α_b = wrap angle for big pulley (degrees)
D = diameter of big pulley (mm)
d = diameter of small pulley (mm)
C = center distance (mm)
L = length of belt (mm)

Open belt drive

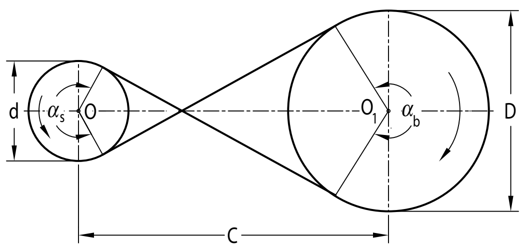


$$\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right) \quad (13.1)$$

$$\alpha_b = 180 + 2\sin^{-1}\left(\frac{D-d}{2C}\right) \quad (13.2)$$

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D-d)^2}{4C} \quad (13.3)$$

Crossed belt drive



$$\alpha_s = \alpha_b = 180 + 2\sin^{-1}\left(\frac{D+d}{2C}\right) \quad (13.4)$$

$$L = 2C + \frac{\pi(D+d)}{2} + \frac{(D+d)^2}{4C} \quad (13.5)$$

Table 13.3 *Power transmitting capacity*

Tensions in flat belt		
$$\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\alpha}$$	(13.6)	P_1 = belt tension in tight side (N) P_2 = belt tension in loose side (N) m = mass of one meter length of belt (kg/m) v = belt velocity (m/s) f = coefficient of friction α = angle of wrap for belt (radians)
Tensions in V-belt		
$$\frac{P_1 - mv^2}{P_2 - mv^2} = e^{f\alpha/\sin(\theta/2)}$$	(13.7)	θ = belt angle (degrees)
Power transmitted by belt		
$$\mathrm{kW} = \frac{(P_1 - P_2)v}{1000}$$	(13.8)	kW = power transmitted by belt

13.2 SELECTION OF FLAT BELTS

Table 13.4 *Load rating of flat Dunlop belt*

Name	Type	Load rating
HI-SPEED	878g Duck belting	0.0118 kW per mm width per ply
FORT	949g Duck belting	0.0147 kW per mm width per ply

(DUNLOP catalogue)

Note: (i) The above values are based on following two assumptions: (a) The arc of contact is 180º. (b) The belt velocity is 5.08 m/s.
(ii) The range of optimum belt velocity for these two belts is 17.8 to 22.9 m/s.
(iii) HI-SPEED duck belting is used in general-purpose applications. FORT duck belting is recommended for heavy duty applications.

Table 13.5 *Standard widths of HI-SPEED and FORT belts*

3 Ply	4 Ply	5 Ply	6 Ply	8 Ply
25(*)	25(!)			
32(!)	32(!)			
40(*)	40(*)			
44(!)	44(*)			
50(*)	50(*)			
63(*)	63(*)			
76(*)	76(*)	76(*)		

(*Contd.*)

(*Contd.*)

90(!)	90(*)	90(!)		
100(!)	100(*)	100(*)	100(!)	
	112(*)	112(*)	112(*)	
	125(*)	125(*)	125(*)	
	140(!)	—`	—	
	152(*)	152(*)	152(*)	
	—	180(#)	180(*)	
	200(!)	200(!)	200(*)	200(#)
		224(!)	—	—
		250(#)	250(#)	250(#)
				305(#)
				355(#)
				400(#)

(DUNLOP catalogue)

Note: The widths are in mm (*) These sizes are available in HI-SPEED and FORT belts. (!)These sizes are available in HI-SPEED belts only. (#)These sizes are available in FORT belts only.

Table 13.6 *Load correction factor (F_a)*

Type of load	**Applications**	**Load correction factor (F_a)**
Normal load	Applications where peak load is known.	1.0
Steady loads	Screens, centrifugal pumps and fans, evaporators, agitators, belt conveyors, laundry machinery, light machine tools, printing and textile machinery.	1.2
Intermittent loads	Heavy duty fans and blowers, brickwork machinery, compressors and reciprocating pumps, saw mill and paper mill machinery, heavy duty machine tools, stokers, elevators and line shafts.	1.3
Shock loads	Vacuum pumps, tube and ball mills, rolling mills, crushing machinery, disintegrators, stamp presses, hammers, grinders, mills and calendars.	1.5

(DUNLOP catalogue)

Note: $(kW)_{max} = F_a \times (kW)$ where
$(kW)_{max}$ = power transmitted by the belt for design purpose
kW = rated power transmitted by the belt in a given application
F_a = load correction factor

Table 13.7 *Arc of contact factor (F_d)*

Arc of contact	90º	120º	130º	140º	150º
Arc of contact factor (F_d)	1.68	1.33	1.26	1.19	1.13
Arc of contact	160º	170º	180º	190º	200º
Arc of contact factor (F_d)	1.08	1.04	1.00	0.97	0.94
Arc of contact	210º	220º	230º	240º	250º
Arc of contact factor (F_d)	0.91	0.88	0.86	0.84	0.82

(DUNLOP catalogue)

Note: (i) The power transmitting capacities of the belt are developed for 180° of arc of contact. The actual arc of contact will be different in different applications. When the arc of contact is less than 180º, there will be additional tension in the belt, to account for which a factor called 'arc of contact factor' (F_d) is used in calculations. (ii) The arc of contact for smaller pulley is considered and it is given by, $\alpha_s = 180 - 2\sin^{-1}\left(\frac{D-d}{2C}\right)$
(iii) $(kW)_{corrected} = (kW)_{max} \times F_d$

Table 13.8 *Dunlop belts*

Optimum belt velocity
The optimum belt velocity (v) of Dunlop belts is from 17.8 to 22.9 m/s. It can be taken as 18 m/s in this range
Pulley diameters
$$d = \frac{60(1000)v}{\pi n} \qquad (13.9)$$ $$D = d\left[\frac{\text{speed of smaller pulley}}{\text{speed of bigger pulley}}\right] = d\left[\frac{\text{input speed}}{\text{output speed}}\right] \qquad (13.10)$$ d = diameter of smaller pulley (mm) D = diameter of bigger pulley (mm) v = optimum belt velocity (m/s) n = input speed of smaller pulley (rpm)
Corrected power
$$(kW)_{max} = F_a \times (kW) \qquad (13.11)$$ $$(kW)_{corrected} = (kW)_{max} \times F_d \qquad (13.12)$$
$(kW)_{max}$ = maximum power for the purpose of belt selection (kW) $(kW)_{corrected}$ = corrected power (kW) F_a = load correction factor (Table 13.6) F_d = arc of contact factor (Table 13.7)

(*Contd.*)

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Corrected power rating of Dunlop belts	
For HI-SPEED belt, corrected kW rating $= \dfrac{0.0118v}{(5.08)}$	(13.13)
For FORT belt, corrected kW rating $= \dfrac{0.0147v}{(5.08)}$	(13.14)
v = correct belt velocity (m/s)	
Product of (width × number of plies)	
$(\text{width} \times \text{number of plies}) = \dfrac{\text{corrected power}}{\text{corrected power rating of belt}}$	
$= \dfrac{(\text{kW})_{\text{corrected}}}{\text{corrected kW rating of belt}}$	(13.15)
Belt width	
The belt width is decided by assuming suitable number of plies.	

(DUNLOP catalogue)

Table 13.9 *Initial belt tension*

Number of plies	Initial belt tension
Belts up to 3 plies	1.5% or 15 mm per metre shorter
Belts of 4, 5 and 6 plies	1.0% or 10 mm per metre shorter
Belts of 7 plies and above	0.5% or 5 mm per metre shorter

(DUNLOP catalogue)

Note: Correct initial tension is obtained by cutting the belt shorter than the actual steel tape measurement around the pulleys.

13.3 PULLEYS FOR FLAT BELTS

Table 13.10 *Types of pulleys for flat belt drives*

Type of pulley	Maximum rim speed (m/min)
Cast iron pulleys	
(a) Solid with flat or crown face	1500
(a) Split with flat or crown face	1000
Mild steel pulleys	
Solid or Split with flat or crown face and mild steel spokes	1500

Table 13.11 *Minimum pulley diameters for given belt speeds and belt plies*

No. of plies	Maximum belt speed (m/s)				
	10	15	20	25	30
	Pulley diameters (mm)				
3	90	100	112	140	180
4	140	160	180	200	250
5	200	224	250	315	355
6	250	315	355	400	450
7	355	400	450	500	560
8	450	500	560	630	710
9	560	630	710	800	900
10	630	710	800	900	1000

Table 13.12 *Recommended diameters and tolerances for cast iron and mild steel flat pulleys*

Nominal diameter (mm)	Tolerance on nominal diameter (mm)
40	± 0.5
45, 50	± 0.6
56, 63	± 0.8
71, 80	± 1.0
90, 100, 112	± 1.2
125, 140	± 1.6
160, 180, 200	± 2.0
224, 250	± 2.5
280, 315, 355	± 3.2
400, 450, 500	± 4.0
560, 630, 710	± 5.0
800, 900, 1000	± 6.3
1120, 1250, 1400	± 8.0
1600, 1800, 2000	± 10.0

Table 13.13 *Relationship between belt and pulley widths*

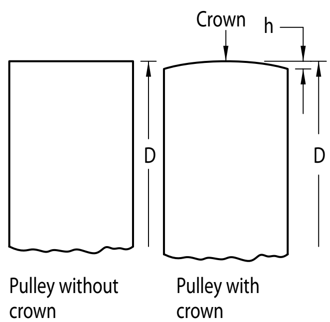
Belt width (mm)	Pulley to be wider than the belt width by (mm)
Up to and including 125	13
From 125 up to and including 250	25
From 250 up to and including 375	38
From 375 up to and including 500	50

Table 13.14 *Recommended widths and tolerances for cast iron and mild steel flat pulleys*

Width (mm)	Tolerance (mm)
20, 25, 32, 40, 50, 63, 71	± 1.0
80, 90, 100, 112, 125, 140	± 1.5
160, 180, 200, 224, 250, 280, 315	± 2.0
355, 400, 450, 500, 560, 630	± 3.0

Table 13.15 *Crown for cast iron and mild steel pulleys*

Crown for pulleys



There is a specific term 'crowning' of pulley in flat belt drive. The thickness of the rim is slightly increased in the mid-plane to give it a convex or conical shape. This is called 'crown' of the pulley. Crown is provided only on one of the two pulleys. The objectives of providing crown are as follows:

(i) The crown on pulley helps to hold the belt on pulley in running condition.

(ii) The crown on pulley prevents the belt from running off the pulley.

(iii) The crown on pulley brings the belt to running equilibrium position near the mid-plane of the pulley.

Crown on pulley is essential particularly when the pulleys are mounted inaccurately or there is a possibility of slip due to non-parallelism between connected shafts.

Note: An approximate relationship between the height of crown (h) and pulley diameter (D) is given by, $h = 0.003D$.

Table 13.16 *Crown of cast iron and mild steel pulleys*

(Pulley diameter from 40 to 355 mm)

Pulley diameter D (mm)	Crown h (mm)
40 – 112	0.3
125 – 140	0.4
160 – 180	0.5
200 – 224	0.6
250 – 280	0.8
315 – 355	1.0

(DUNLOP catalogue)

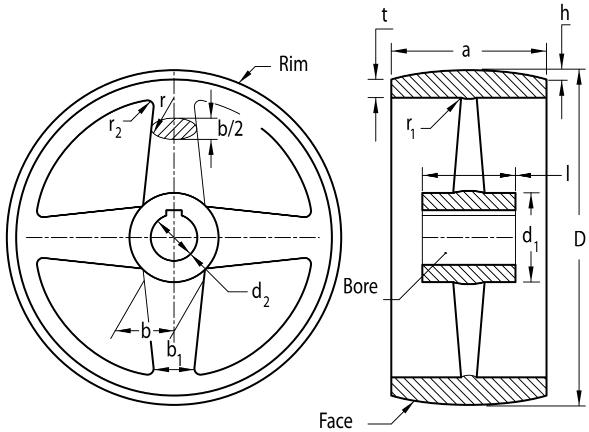
Table 13.17 *Crown of cast iron and mild steel pulleys*

(Pulley diameter from 400 to 2000 mm)

Pulley diameter D (mm)	Width of pulleys (mm)						
	125 and smaller	140 and 160	180 and 200	224 and 250	280 and 315	355	400 and larger
	Crown (mm)						
400	1	1.2	1.2	1.2	1.2	1.2	1.2
450	1	1.2	1.2	1.2	1.2	1.2	1.2
500	1	1.5	1.5	1.5	1.5	1.5	1.5
560	1	1.5	1.5	1.5	1.5	1.5	1.5
630	1	1.5	2.0	2.0	2.0	2.0	2.0
710	1	1.5	2.0	2.0	2.0	2.0	2.0
800	1	1.5	2.0	2.5	2.5	2.5	2.5
900	1	1.5	2.0	2.5	2.5	2.5	2.5
1000	1	1.5	2.0	2.5	3.0	3.0	3.0
1120	1.2	1.5	2.0	2.5	3.0	3.0	3.5
1250	1.2	1.5	2.0	2.5	3.0	3.5	4.0
1400	1.5	2.0	2.5	3.0	3.5	4.0	4.0
1600	1.5	2.0	2.5	3.0	3.5	4.0	5.0
1800	2.0	2.5	3.0	3.5	4.0	5.0	5.0
2000	2.0	2.5	3.0	3.5	4.0	5.0	6.0

(DUNLOP catalogue)

Table 13.18 *Proportions of cast iron pulleys*



(a) Number of arms

(1) For pulleys up to 200 mm diameter, use webs.
(2) For pulleys above 200 mm diameter and up to 450 mm diameter, use 4 arms.
(3) For pulleys above 450 mm diameter, use 6 arms.

(b) Cross section of arms—Elliptical

(c) Thickness of arm b near boss

$$b = 2.94\sqrt[3]{\frac{aD}{4n}} \quad \text{for single belt}$$

$$b = 2.94\sqrt[3]{\frac{aD}{2n}} \quad \text{for double belt}$$

where, a = width of pulley
D = diameter of pulley
n = number of arms in the pulley

(d) Thickness of arm b_1 near rim

Use taper 4 mm per 100 mm (from boss to rim)

(e) Radius of the cross section of arms

$$r = \frac{3}{4}b$$

(f) Minimum length *l* of the bore

$$l = \frac{2}{3}a$$

It may be more for loose pulleys, but in no case it should exceeds *a*.

(g) $$\frac{d_1 - d_2}{2} = 0.412 \times \sqrt[3]{aD} + 6 \text{ mm} \quad \text{for single belt}$$

$$\frac{d_1 - d_2}{2} = 0.529 \times \sqrt[3]{aD} + 6 \text{ mm} \quad \text{for double belt}$$

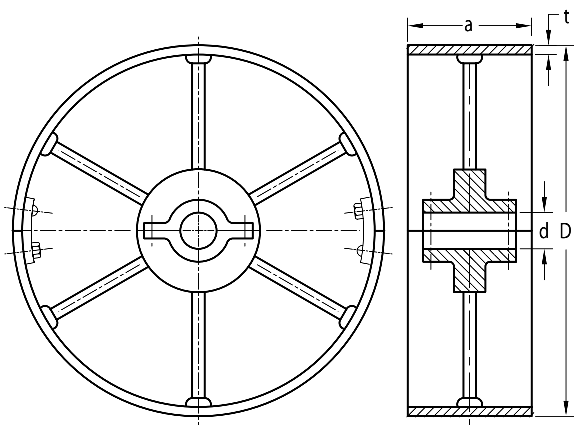
(*Contd.*)

(*Contd.*)

(h) Radius r_1 and r_2 $r_1 = \frac{b}{2}$ (near rim) $r_2 = \frac{b}{2}$ (near rim)
(i) Thickness of rim Rim thickness $= \left(\frac{D}{200} + 3\right)$ mm (for single belt) Rim thickness $= \left(\frac{D}{200} + 6\right)$ mm (for double belt)
(j) Tolerance grades for bore (i) Solid pulley – H7 (ii)Split pulley – U7

Note: The cast iron pulleys are made of grey cast iron FG 200.

Table 13.19 *Proportions of mild steel pulleys*



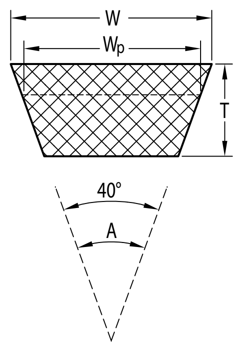
(a) Arrangement of arms Pulleys up to 300 mm width are normally supplied with single row of spokes. Wider pulleys requiring double row of spokes are sometimes used.
(b) Minimum length of boss The length of boss is equal to half width of face, subject to a minimum of 100 mm in case of pulleys with 19 mm diameter spokes and minimum of 138 mm for pulleys with 22 mm diameter spokes. The length of boss is practically greater than the width of the pulleys.
(c) Thickness of rims 5 mm for all pulleys
(d) Tolerance for bore (i) Solid pulley – H7 (ii) Split pulley – U7

Table 13.20 *Spokes for pulley*

Diameter of pulley (mm)	Number of spokes	Diameter of spokes (mm)
280 to 500	6	19
560 to 710	8	19
800 to 1 000	10	22
1 120	12	22
1 250	14	22
1 400	16	22
1 600	18	22
1 800	18	22
2 000	22	22

13.4 V-BELTS

Table 13.21 *Cross sections of V-belt*



Terminology of cross section

(a) Pitch width (W_p) It is the width of the belt at its pitch zone. This is the basic dimension of standardisation for the belt and for the corresponding pulley groove, considered as a whole.
(b) Nominal top width (W) It is the top width of the trapezium outlined on the cross section of the belt.
(c) Nominal height (T) It is the height of the trapezium outlined on the cross section of the belt.
(d) Angle of belt (A) It is the included angle obtained by extending the sides of the belt. The standard value of the belt angle is 40°.
(e) Pitch length (L_p) It is the length of the pitch line of the belt. This is the circumferential length of the belt at the pitch width.

Symbols of cross section

The manufacturers have standardised the dimensions of the cross section. There are six basic symbols – Z, A, B, C, D and E – for the cross section of V-belts. Z- section belts are occasionally used for low power transmission and small pulley diameters, while A, B, C, D and E section belts are widely used as general purpose belts.

(*Contd.*)

(*Contd.*)

Designations of V-belts
V-belts are designated by the symbol of cross section followed by nominal pitch length along with symbol L_p, e.g., a V-belt of cross section B and of nominal pitch length of 4430 mm is designated as B 4430 L_p
Groove angles
The groove angle for the belt is 40º. The groove angle for pulley is from 34º to 38º. This results in wedging action between the belt and the groove, thereby increasing the frictional force, and consequently the transmitted power.

Table 13.22 *Dimensions of V-belt cross sections*

Symbol of cross section	Pitch width W_p (mm)	Nominal top width W (mm)	Nominal height T (mm)	Nominal included angle A (degrees)
Z	8.5	10	6	40
A	11	13	8	40
B	14	17	11	40
C	19	22	14	40
D	27	32	19	40
E	32	38	23	40

(GOOD YEAR V-belt catalogue)

Table 13.23 *Recommended pulley pitch diameters for V-belts*

Symbol of cross section	Recommended minimum pitch diameter of pulley (mm)	Permissible minimum pitch diameter of pulley (mm)
Z	85	50
A	95	75
B	145	125
C	225	200
D	350	315
E	550	500

Table 13.24 *Recommended pitch diameters of pulleys*

Belt section					
Z	**A**	**B**	**C**	**D**	**E**
Pitch diameters of pulleys (mm)					
50	75	125	200	355	500
53	80	132	212	375	530
56	85	140	224	400	560
60	90	150	236	425	600
63	95	160	250	450	630

(*Contd.*)

(*Contd.*)

67	100	170	265	475	670
71	106	180	280	500	710
75	112	190	300	530	750
80	118	200	315	560	800
85	125	224	355	600	900
90	132	250	375	630	1000
95	140	280	400	710	1120
100	150	300	450	750	1250
112	160	315	500	800	1400
125	170	355	530	900	1500
140	180	375	560	1000	1600
160	190	400	600	1060	1800
180	200	450	630	1120	1900
200	224	500	710	1250	2000
250	250	530	750	1400	2240
315	280	560	800	1500	2500
400	300	600	900	1600	--
500	315	630	1000	1800	—
630	350	710	1200	2000	—
800	400	750	1250	—	—
—	450	800	1400	—	—
—	500	900	1600	—	—
—	560	1000	—	—	—
—	630	1120	—	—	—
—	710	—	—	—	—
—	800	—	—	—	—

Note: Tolerance on the pitch diameter is ± 0.8 per cent.

Table 13.25 *Nominal pitch lengths for V-belts*

Belt section					
Z	**A**	**B**	**C**	**D**	**E**
Nominal pitch lengths of belts L_p (mm)					
405	630	930	1560	2740	4660
475	700	1000	1760	3130	5040
530	790	1100	1950	3330	5420
625	890	1210	2190	3730	6100

(*Contd.*)

(*Contd.*)

700	990	1370	2420	4080	6850
780	1100	1560	2720	4620	7650
920	1250	1690	2880	5400	9150
1080	1430	1760	3080	6100	12230
1330	1550	1950	3310	6840	13750
1420	1640	2180	3520	7620	15280
1540	1750	2300	4060	8410	16800
	1940	2500	4600	9140	
	2050	2700	5380	10700	
	2200	2870	6100	12200	
	2300	3200	6820	13700	
	2480	3600	7600	15200	
	2570	4060	9100		
	2700	4430	10700		
	2910	4820			
	3080	5370			
	3290	6070			
	3540				

Table 13.26 *Nominal inside length and nominal pitch lengths*

Nominal inside length (mm)	Belt section					Pitch length limits (mm)
	A	B	C	D	E	
	Pitch lengths of belt L_p (mm)					
610	645	—	—	—	—	+12 –6
660	696	—	—	—	—	
711	747	—	—	—	—	
787	823	—	—	—	—	+12 –8
813	848	—	—	—	—	
889	925	932	—	—	—	
914	950	958	—	—	—	
965	1001	1008	—	—	—	+14 –8
991	1026	1034	—	—	—	
1016	1051	1059	—	—	—	
1067	1102	1110	—	—	—	
1092	1128	1136	—	—	—	
1168	1204	1211	—	—	—	

(*Contd.*)

(*Contd.*)

1219	1255	1262	1275	—	—	+16 –10
1295	1331	1339	1351	—	—	
1372	1408	1415	1428	—	—	
1397	1433	1440	1453	—	—	
1422	1458	1466	1478	—	—	
1473	1509	1516	1529	—	—	
1524	1560	1567	1580	—	—	
1600	1636	1643	1656	—	—	+18 –12
1626	1661	1669	1681	—	—	
1651	1687	1694	1707	—	—	
1727	1763	1770	1783	—	—	
1778	1814	1821	1834	—	—	
1829	1865	1872	1885	—	—	
1905	1941	1948	1961	—	—	+18 –12
1981	2017	2024	2073	—	—	+30 –16
2032	2068	2075	2088	—	—	
2057	2093	2101	2113	—	—	
2159	2195	2202	2215	—	—	
2286	2322	2329	2342	—	—	
2438	2474	2482	2494	—	—	
2464	2500	2507	2520	—	—	+34 –18
2540	—	2583	—	—	—	+36 –18
2667	2703	2710	2723	—	—	
2845	2880	2888	2901	—	—	
3048	3084	3091	3104	3127	—	
3150	—	—	3205	—	—	+38 –20
3251	3285	3294	3307	3330	—	
3404	—	—	3459	—	—	
3658	3693	3701	3713	3736	—	
4013	—	4056	4069	4092	—	+42 –24
4115	—	4158	4171	4194	—	+46 –24
4394	—	4437	4450	4473	—	
4572	—	4615	4628	4651	—	

(*Contd.*)

(*Contd.*)

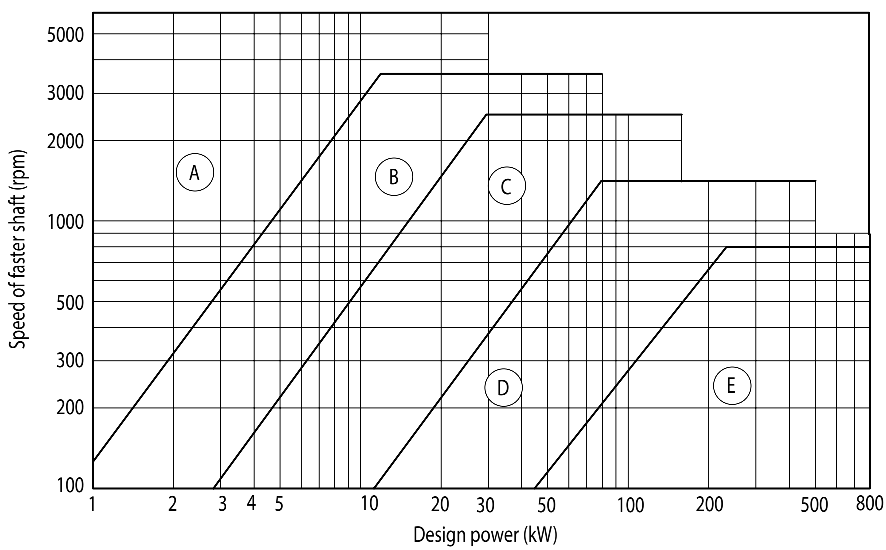
4953	—	4996	5009	5032	—	+48 –28
5334	—	5377	5390	5413	5426	
6045	—	—	6101	6124	6137	
6807	—	—	6863	6886	6899	+56 –32
7569	—	—	7625	7648	7661	+66 –38
8331	—	—	8387	8410	8423	
9093	—	—	9149	9172	9185	+66 –38
9855	—	—	—	9934	9947	
10617	—	—	—	10696	10709	+74 –44
12141	—	—	—	12220	12233	
13665	—	—	—	13744	13757	+90 –50
15189	—	—	—	15268	15281	
16713	—	—	—	16792	16805	+104 –58

(GOOD YEAR V-belt catalogue)

13.5 SELECTION OF V-BELTS

Table 13.27 *Selection of cross section and number of V-belts*

Selection of cross section



The selection of cross section depends upon two factors, namely the power to be transmitted and speed of the faster shaft. Above figure shows the range of speed and power for various cross sections of the belt. Depending upon the power and speed of the faster pulley, a point can be plotted on this diagram and the corresponding cross section selected. In borderline cases, alternative design calculations are made to determine the best cross section.

(*Contd.*)

(*Contd.*)

Number of belts
The number of belts required for a given application is given by, $$\text{Number of belts} = \frac{(\text{transmitted power in kW}) \times (F_a)}{(\text{kW rating of single } V - \text{belt}) \times (F_c) \times (F_d)} = \frac{P \times F_a}{P_r \times F_c \times F_d} \qquad (13.16)$$ where P = drive power to be transmitted (kW) F_a = correction factor for industrial service (Table 13.28) P_r = power rating of single V-belt (from Table 13.29 to Table 13.33) F_c = correction factor for belt pitch length (Table 13.34) F_d = correction factor for arc of contact (Table 13.35)

(GOOD YEAR V-belt catalogue)

Table 13.28 *Correction factor according to service (F_a)*

Type of driven machine		**Type of driving unit**					
Class and service	Examples	'Soft' starts (a) Electric motors A.C. star-delta start, D.C. shunt wound (b) Internal combustion engines with 4 or more cylinders (c) All prime movers fitted with centrifugal clutches, dry or fluid couplings			'Heavy' starts (a) Electric motors A.C.direct on-line start, D.C. series and compound wound (b) Internal combustion engines with less than 4 cylinders		
		Hours of duty per day (hr)			**Hours of duty per day (hr)**		
		0-10	**10–16**	**16–24**	**0–10**	**10–16**	**16–24**
Class 1 (light duty)	(a) Agitators (uniform density) (b) Blower, exhausts and fans (up to 7.5 kW) (c) Centrifugal compressor and pumps (d) Belt conveyor (uniformly loaded)	1.0	1.1	1.2	1.1	1.2	1.3

(*Contd.*)

(*Contd.*)

Class 2 (medium duty)	(a) Agitators and mixers (variable density) (b) Blower, exhausts and fans (over 7.5 kW) (c) Rotary compressors and pumps (other than centrifugal) (d) Belt conveyor (non-uniformly loaded) (e) Generator and exciters (f) Line shafts (g) Machine tools (h) Printing machinery (j) Sawmill and wood working Machinery (k) Screens (rotary)	1.1	1.2	1.3	1.2	1.3	1.4
Class 3 (heavy duty)	(a) Brick machinery (b) Bucket elevators (c) Compressors and pumps (reciprocating) (d) Conveyors (heavy duty) (e) Hoists (f) Hammer mills (g) Pulverizes (h) Punches, presses and shears (j) Quarry plants (k) Rubber machinery (l) Screens (vibrating) (m) Textile machinery	1.2	1.3	1.4	1.4	1.5	1.6
Class 4 (extra-heavy duty)	(a) Crusher (gyratory, jaw and roll) (b) Mills (ball, rod and tube)	1.3	1.4	1.5	1.5	1.6	1.8

Note: (i) For speed-increasing drives of speed ratio 1.00 to 1.24: multiply service factor (F_a) by 1.00.
(ii) For speed-increasing drives of speed ratio 1.25 to 1.74: multiply service factor (F_a) by 1.05.
(iii) For speed-increasing drives of speed ratio 1.75 to 2.49: multiply service factor (F_a) by 1.11.
(iv) For speed-increasing drives of speed ratio 2.5 to 3.49: multiply service factor (F_a) by 1.18.
(v) For speed-increasing drives of speed ratio 3.5 and over: multiply service factor (F_a) by 1.25.

Special conditions
(i) For reversing drives, except where high torque is not present on starting, add 20% to factors
(ii) Idler pulley on slack side (internal) – no addition to the factors
(iii) Idler pulley on tight side (internal) – add 0.1 to the factors.

(GOOD YEAR V-belt catalogue)

Table 13.29 *Power ratings in kW (P_r) for A-section V- belts, 13 mm wide with 180º arc of contact on smaller pulley*

Speed of faster shaft	Power rating for smaller pulley pitch diameter of									Additional power increment per belt for speed ratio of									
	75 mm	80 mm	85 mm	90 mm *	100 mm *	106 mm *	112 mm *	118 mm *	125 mm *	1.00 to 1.01	1.02 to 1.04	1.05 to 1.08	1.09 to 1.12	1.13 to 1.18	1.19 to 1.24	1.25 to 1.34	1.35 to 1.51	1.52 to 1.99	2.00 and over
r.p.m.	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW
720	0.53	0.60	0.68	0.75	0.90	0.99	1.07	1.16	1.26	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
960	0.66	0.76	0.86	0.95	1.14	1.25	1.37	1.49	1.61	0.00	0.01	0.03	0.04	0.05	0.06	0.08	0.09	0.10	0.12
1440	0.91	1.04	1.17	1.31	1.58	1.73	1.90	2.07	2.24	0.00	0.02	0.04	0.06	0.08	0.10	0.12	0.14	0.16	0.17
2880	1.42	1.67	1.91	2.14	2.59	2.76	3.11	3.36	3.63	0.00	0.04	0.08	0.12	0.16	0.20	0.23	0.27	0.31	0.35
100	0.11	0.13	0.12	0.14	0.17	0.18	0.20	0.21	0.23	0.00	0.00	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.01
200	0.19	0.22	0.24	0.26	0.31	0.33	0.36	0.39	0.42	0.00	0.00	0.01	0.01	0.01	0.01	0.02	0.02	0.02	0.03
300	0.26	0.29	0.33	0.37	0.43	0.46	0.51	0.55	0.60	0.00	0.00	0.01	0.01	0.02	0.02	0.02	0.03	0.03	0.04
400	0.33	0.37	0.42	0.46	0.55	0.60	0.66	0.71	0.77	0.00	0.01	0.01	0.02	0.02	0.03	0.03	0.04	0.04	0.05
500	0.39	0.45	0.51	0.56	0.67	0.72	0.79	0.86	0.93	0.00	0.01	0.01	0.02	0.03	0.03	0.04	0.05	0.05	0.06
600	0.46	0.52	0.59	0.65	0.78	0.85	0.93	1.00	1.08	0.00	0.01	0.02	0.02	0.03	0.04	0.05	0.06	0.07	0.07
700	0.52	0.59	0.66	0.74	0.88	0.96	1.05	1.14	1.23	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
800	0.57	0.66	0.74	0.82	0.98	1.08	1.18	1.27	1.38	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.08	0.09	0.10
900	0.63	0.72	0.81	0.90	1.08	1.18	1.30	1.41	1.52	0.00	0.01	0.02	0.04	0.05	0.06	0.07	0.08	0.10	0.11
1000	0.68	0.78	0.88	0.98	1.18	1.29	1.42	1.54	1.66	0.00	0.01	0.03	0.04	0.05	0.07	0.08	0.09	0.11	0.12
1100	0.73	0.84	0.95	1.06	1.28	1.40	1.53	1.66	1.80	0.00	0.02	0.03	0.04	0.06	0.07	0.09	0.10	0.12	0.13
1200	0.78	0.90	1.02	1.13	1.37	1.50	1.64	1.78	1.93	0.00	0.02	0.03	0.05	0.07	0.08	0.10	0.11	0.13	0.15
1300	0.83	0.95	1.10	1.21	1.45	1.60	1.75	1.90	2.06	0.00	0.02	0.04	0.05	0.07	0.09	0.11	0.12	0.14	0.16
1400	0.88	1.01	1.15	1.28	1.54	1.69	1.86	2.02	2.19	0.00	0.02	0.04	0.06	0.08	0.09	0.11	0.13	0.15	0.17
1500	0.92	1.07	1.21	1.35	1.63	1.79	1.96	2.13	2.31	0.00	0.02	0.04	0.06	0.08	0.10	0.12	0.14	0.16	0.18
1600	0.97	1.12	1.27	1.42	1.72	1.89	2.06	2.24	2.43	0.00	0.02	0.04	0.06	0.09	0.11	0.13	0.15	0.17	0.19

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(*Contd.*)

1700	1.01	1.17	1.33	1.48	1.79	1.97	2.16	2.35	2.54	0.00	0.02	0.05	0.07	0.09	0.11	0.14	0.16	0.18	0.21
1800	1.05	1.22	1.39	1.55	1.88	2.07	2.26	2.46	2.66	0.00	0.02	0.05	0.07	0.10	0.12	0.15	0.17	0.19	0.22
1900	1.09	1.27	1.44	1.61	1.95	2.14	2.35	2.56	2.77	0.00	0.03	0.05	0.08	0.10	0.13	0.15	0.18	0.20	0.23
2000	1.13	1.31	1.50	1.68	2.03	2.23	2.44	2.65	2.87	0.00	0.03	0.05	0.08	0.11	0.13	0.16	0.19	0.22	0.24
2100	1.17	1.36	1.55	1.74	2.10	2.31	2.53	2.75	2.98	0.00	0.03	0.06	0.09	0.11	0.14	0.17	0.20	0.23	0.25
2200	1.21	1.40	1.60	1.79	2.17	2.38	2.61	2.84	3.07	0.00	0.03	0.06	0.09	0.12	0.15	0.18	0.21	0.24	0.27
2300	1.24	1.45	1.65	1.85	2.24	2.46	2.69	2.93	3.16	0.00	0.03	0.06	0.09	0.12	0.16	0.18	0.22	0.25	0.28
2400	1.28	1.48	1.70	1.90	2.31	2.54	2.78	3.02	3.26	0.00	0.03	0.07	0.10	0.13	0.16	0.19	0.23	0.26	0.29
2500	1.31	1.53	1.75	1.95	2.37	2.61	2.85	3.09	3.34	0.00	0.03	0.07	0.10	0.14	0.17	0.20	0.24	0.27	0.30
2600	1.34	1.57	1.80	2.01	2.43	2.67	2.92	3.17	3.42	0.00	0.04	0.07	0.11	0.14	0.17	0.21	0.24	0.28	0.31
2700	1.37	1.60	1.84	2.06	2.49	2.74	2.99	3.25	3.51	0.00	0.04	0.07	0.11	0.15	0.18	0.22	0.25	0.29	0.33
2800	1.40	1.64	1.88	2.10	2.55	2.80	3.06	3.31	3.57	0.00	0.04	0.08	0.11	0.15	0.19	0.23	0.26	0.30	0.34
2900	1.43	1.67	1.91	2.15	2.60	2.86	3.12	3.38	3.65	0.00	0.04	0.08	0.12	0.16	0.19	0.23	0.27	0.30	0.35
3000	1.46	1.71	1.95	2.19	2.66	2.92	3.18	3.45	3.72	0.00	0.04	0.08	0.12	0.16	0.20	0.24	0.28	0.32	0.36
3100	1.48	1.74	1.98	2.23	2.71	2.98	3.24	3.50	3.77	0.00	0.04	0.08	0.13	0.17	0.21	0.25	0.29	0.33	0.37
3200	1.51	1.77	2.02	2.28	2.75	3.03	3.29	3.56	3.83	0.00	0.04	0.09	0.13	0.17	0.22	0.26	0.30	0.34	0.39
3300	1.53	1.80	2.06	2.31	2.80	3.07	3.34	3.61	3.88	0.00	0.04	0.09	0.13	0.18	0.22	0.27	0.31	0.36	0.40
3400	1.55	1.82	2.10	2.34	2.83	3.11	3.30	3.65	3.92	0.00	0.05	0.09	0.14	0.18	0.23	0.27	0.32	0.37	0.41
3500	1.57	1.85	2.12	2.38	2.87	3.10	3.43	3.70	3.97	0.00	0.05	0.09	0.14	0.19	0.24	0.28	0.33	0.38	0.42
3600	1.59	1.87	2.14	2.41	2.91	3.19	3.47	3.73	4.00	0.00	0.05	0.10	0.15	0.19	0.24	0.29	0.34	0.39	0.44
3700	1.61	1.89	2.17	2.44	2.95	3.22	3.50	3.77	4.04	0.00	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45
3800	1.62	1.92	2.19	2.46	2.98	3.25	3.53	3.80	4.06	0.00	0.05	0.10	0.15	0.20	0.26	0.31	0.36	0.41	0.46
3900	1.64	1.93	2.22	2.49	2.99	3.28	3.56	3.82	4.08	0.00	0.05	0.11	0.16	0.21	0.26	0.31	0.37	0.42	0.47
4000	1.65	1.95	2.24	2.51	3.03	3.30	3.58	3.84	4.10	0.00	0.05	0.11	0.16	0.22	0.27	0.32	0.38	0.43	0.48
4100	1.67	1.96	2.25	2.53	3.05	3.32	3.60	3.85	4.10	0.00	0.06	0.11	0.17	0.22	0.28	0.33	0.39	0.44	0.50

(*Contd.*)

(*Contd.*)

4200	1.68	1.98	2.27	2.55	3.07	3.35	3.61	3.85	4.10	0.00	0.06	0.11	0.17	0.23	0.28	0.34	0.40	0.45	0.51
4300	1.69	1.99	2.28	2.57	3.08	3.35	3.62	3.85	4.10	0.00	0.06	0.12	0.17	0.23	0.29	0.35	0.40	0.46	0.52
4400	1.69	2.00	2.30	2.57	3.09	3.35	3.63	3.85	4.10	0.00	0.06	0.12	0.18	0.24	0.30	0.35	0.41	0.47	0.53
4500	1.70	2.01	2.31	2.59	3.10	3.35	3.63	3.85	4.07	0.00	0.06	0.12	0.18	0.24	0.30	0.36	0.42	0.48	0.54
4600	1.71	2.01	2.31	2.60	3.10	3.34	3.63	3.84	4.06	0.00	0.06	0.12	0.19	0.25	0.31	0.37	0.43	0.49	0.56
4700	1.71	2.02	2.32	2.60	3.10	3.34	3.62	3.82		0.00	0.06	0.13	0.19	0.25	0.32	0.38	0.44	0.51	0.57
4800	1.71	2.02	2.32	2.60	3.10	3.33	3.60	3.81		0.00	0.06	0.13	0.19	0.26	0.32	0.39	0.45	0.52	0.58
4900	1.71	2.02	2.32	2.60	3.10	3.32	3.58	3.79		0.00	0.07	0.13	0.20	0.26	0.33	0.39	0.46	0.53	0.59
5000	1.71	2.02	2.32	2.60	3.09	3.32	3.56	3.77		0.00	0.07	0.14	0.20	0.27	0.34	0.40	0.47	0.54	0.60

(GOOD YEAR V-belt catalogue)

Note: * Preferred pulley diameters

Table 13.30 *Power ratings in kW (P_r) for B-section V-belts, 17 mm wide with 180º arc of contact on smaller pulley*

Speed of faster shaft	Power rating for smaller pulley pitch diameter of									Additional power increment per belt for speed ratio of									
	125 mm	132 mm	140 mm *	150 mm *	160 mm *	170 mm *	180 mm *	190 mm *	200 mm *	1.00 to 1.01	1.02 to 1.04	1.05 to 1.08	1.09 to 1.12	1.13 to 1.18	1.19 to 1.24	1.25 to 1.34	1.35 to 1.51	1.52 to 1.99	2.00 and over
r.p.m.	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW
720	1.61	1.79	1.99	2.24	2.48	2.73	2.97	3.21	3.45	0.00	0.03	0.05	0.08	0.10	0.13	0.15	0.18	0.20	0.23
960	2.02	2.24	2.50	2.82	3.13	3.44	3.75	4.05	4.35	0.00	0.03	0.07	0.10	0.14	0.17	0.20	0.24	0.27	0.30
1440	2.72	3.03	3.39	3.83	4.26	4.68	5.09	5.50	5.90	0.00	0.05	0.10	0.15	0.20	0.25	0.30	0.36	0.41	0.46
2880	3.96	4.44	4.95	5.55	6.11	6.62	7.08	7.48	-	0.00	0.10	0.20	0.30	0.41	0.50	0.61	0.71	0.81	0.91
100	0.32	0.35	0.38	0.43	0.47	0.51	0.55	0.59	0.63	0.00	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.03	0.03
200	0.57	0.63	0.69	0.77	0.85	0.92	1.00	1.08	1.15	0.00	0.01	0.01	0.02	0.03	0.04	0.04	0.05	0.06	0.06
300	0.80	0.88	0.97	1.08	1.19	1.31	1.42	1.53	1.64	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
400	1.01	1.11	1.23	1.38	1.52	1.67	1.81	1.96	2.10	0.00	0.01	0.03	0.04	0.06	0.07	0.08	0.10	0.11	0.13
500	1.21	1.33	1.48	1.66	1.84	2.01	2.19	2.36	2.54	0.00	0.02	0.04	0.05	0.07	0.09	0.11	0.12	0.14	0.16
600	1.40	1.55	1.72	1.93	2.14	2.35	2.55	2.76	2.96	0.00	0.02	0.04	0.06	0.08	0.11	0.13	0.15	0.17	0.19
700	1.58	1.75	1.94	2.19	2.43	2.66	2.90	3.13	3.37	0.00	0.02	0.05	0.07	0.10	0.12	0.15	0.17	0.20	0.22
800	1.75	1.94	2.16	2.44	2.70	2.97	3.24	3.50	3.78	0.00	0.03	0.06	0.08	0.11	0.14	0.17	0.20	0.23	0.25
900	1.92	2.13	2.37	2.68	2.97	3.27	3.56	3.85	4.13	0.00	0.03	0.06	0.10	0.13	0.16	0.19	0.22	0.25	0.29
1000	2.08	2.31	2.58	2.91	3.23	3.55	3.87	4.18	4.49	0.00	0.04	0.07	0.10	0.14	0.18	0.21	0.25	0.28	0.32
1100	2.23	2.49	2.78	3.13	3.48	3.83	4.17	4.51	4.84	0.00	0.04	0.08	0.12	0.16	0.19	0.23	0.27	0.31	0.35
1200	2.38	2.66	2.96	3.35	3.72	4.09	4.46	4.81	5.17	0.00	0.04	0.08	0.13	0.17	0.21	0.25	0.30	0.34	0.38
1300	2.53	2.82	3.15	3.55	3.95	4.34	4.73	5.11	5.48	0.00	0.05	0.09	0.14	0.18	0.23	0.27	0.32	0.37	0.41
1400	2.66	2.97	3.32	3.75	4.17	4.59	4.98	5.39	5.78	0.00	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.39	0.44
1500	2.79	3.12	3.49	3.94	4.38	4.82	5.24	5.66	6.06	0.00	0.05	0.10	0.16	0.21	0.26	0.32	0.37	0.42	0.48
1600	2.92	3.26	3.65	4.12	4.58	5.04	5.48	5.91	6.33	0.00	0.06	0.11	0.17	0.23	0.28	0.34	0.39	0.45	0.51
1700	3.04	3.40	3.80	4.29	4.77	5.24	5.70	6.14	6.58	0.00	0.06	0.12	0.18	0.24	0.30	0.36	0.42	0.48	0.54
1800	3.15	3.52	3.94	4.45	4.95	5.44	5.90	6.36	6.80	0.00	0.06	0.13	0.19	0.25	0.32	0.38	0.44	0.51	0.57
1900	3.26	3.65	4.08	4.61	5.12	5.62	6.10	6.56	7.00	0.00	0.07	0.13	0.20	0.27	0.33	0.40	0.47	0.54	0.60

(*Contd.*)

(*Contd.*)

2000	3.36	3.76	4.21	4.75	5.28	5.78	6.27	6.74	7.19	0.00	0.07	0.14	0.21	0.28	0.35	0.42	0.49	0.56	0.63
2100	3.45	3.87	4.33	4.88	5.42	5.94	6.43	6.91	7.36	0.00	0.07	0.15	0.22	0.30	0.37	0.44	0.52	0.59	0.67
2200	3.54	4.00	4.44	5.01	5.55	6.07	6.58	7.05	7.50	0.00	0.08	0.16	0.23	0.31	0.39	0.46	0.54	0.62	0.70
2300	3.62	4.06	4.54	5.12	5.68	6.20	6.70	7.18	7.62	0.00	0.08	0.16	0.24	0.32	0.41	0.49	0.57	0.65	0.73
2400	3.70	4.14	4.63	5.22	5.78	6.31	6.81	7.28	7.72	0.00	0.08	0.17	0.25	0.34	0.42	0..51	0.59	0.68	0.76
2500	3.77	4.22	4.64	5.23	5.80	6.33	6.83	7.29	7.79	0.00	0.09	0.18	0.26	0.35	0.44	0.53	0.62	0.70	0.79
2600	3.82	4.28	4.79	5.39	5.95	6.48	6.97	7.42	7.83	0.00	0.09	0.18	0.27	0.37	0.46	0.55	0.64	0.73	0.82
2700	3.88	4.35	4.86	5.46	6.03	6.55	7.03	7.47	7.86	0.00	0.10	0.19	0.29	0.38	0.48	0.57	0.67	0.76	0.86
2800	3.93	4.40	4.91	5.52	6.08	6.60	7.06	7.48	7.85	0.00	0.10	0.20	0.29	0.39	0.49	0.59	0.69	0.79	0.89
2900	3.97	4.44	4.96	5.56	6.12	6.62	7.08	7.48		0.00	0.10	0.20	0.31	0.41	0.51	0.61	0.72	0.82	0.92
3000	4.00	4.48	4.99	5.59	6.14	6.63	7.07	7.44		0.00	0.11	0.21	0.32	0.42	0.53	0.63	0.74	0.85	0.95
3100	4.02	4.50	5.02	5.61	6.15	6.63	7.04			0.00	0.11	0.22	0.33	0.44	0.55	0.65	0.76	0.87	0.98
3200	4.04	4.52	5.03	5.62	6.14	6.60	6.99			0.00	0.11	0.23	0.34	0.45	0.56	0.68	0.79	0.90	1.01
3300	4.05	4.52	5.03	5.61	6.11	6.55				0.00	0.12	0.23	0.35	0.47	0.58	0.70	0.81	0.93	1.05
3400	4.05	4.52	5.02	5.58	6.07	6.48				0.00	0.12	0.24	0.36	0.48	0.60	0.72	0.84	0.96	1.08
3500	4.04	4.50	5.00	5.55	6.01					0.00	0.12	0.25	0.37	0.49	0.62	0.74	0.86	0.99	1.11
3600	4.02	4.48	4.96	5.49						0.00	0.13	0.25	0.38	0.51	0.63	0.76	0.89	1.01	1.14
3700	3.99	4.45	4.92	5.43						0.00	0.13	0.26	0.39	0.52	0.65	0.78	0.91	1.04	1.17
3800	3.95	4.40	4.86	5.34						0.00	0.13	0.27	0.40	0.54	0.67	0.80	0.94	1.07	1.20
3900	3.91	4.34	4.76							0.00	0.14	0.28	0.41	0.55	0.69	0.82	0.96	1.10	1.24
4000	3.85	4.28	4.70							0.00	0.14	0.28	0.42	0.56	0.70	0.84	0.99	1.13	1.27
4100	3.78	4.20								0.00	0.14	0.29	0.43	0.58	0.72	0.87	1.01	1.16	1.30
4200	3.71	4.11								0.00	0.15	0.30	0.44	0.59	0.74	0.89	1.04	1.18	1.33
4300	3.62	4.00								0.00	0.15	0.30	0.45	0.61	0.76	0.91	1.06	1.21	1.36
4400	3.53									0.00	0.15	0.31	0.46	0.62	0.78	0.93	1.08	1.24	1.39
4500	3.42									0.00	0.16	0.32	0.48	0.63	0.79	0.95	1.11	1.27	1.43

(GOOD YEAR V-belt catalogue)

Note: * Preferred pulley diameters

Table 13.31 *Power ratings in kW (P_r) for C-section V- belts, 22 mm wide with 180º arc of contact on smaller pulley*

Speed of faster shaft	Power rating for smaller pulley pitch diameter of										Additional power increment per belt for speed ratio of									
	200 mm *	212 mm *	224 mm *	236 mm *	250 mm *	265 mm *	280 mm *	315 mm *	355 mm *	400 mm *	1.00 to 1.01	1.02 to 1.04	1.05 to 1.08	1.09 to 1.12	1.13 to 1.18	1.19 to 1.24	1.25 to 1.34	1.35 to 1.51	1.52 to 1.99	2.00 and over
r.p.m.	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW
720	4.65	5.18	5.70	6.22	6.81	7.44	8.06	9.49	11.05	12.75	0.00	0.07	0.14	0.21	0.28	0.35	0.42	0.49	0.56	0.63
960	5.76	6.42	7.08	7.72	8.46	9.24	10.00	11.72	13.58	15.51	0.00	0.09	0.19	0.28	0.38	0.47	0.56	0.66	0.75	0.85
1440	7.49	8.36	9.21	10.03	10.95	11.91	12.82	14.76	16.67	–	0.00	0.14	0.28	0.42	0.56	0.71	0.85	0.99	1.13	1.27
100	0.93	1.02	1.11	1.20	1.30	1.42	1.53	1.78	2.07	2.39	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
200	1.66	1.83	2.00	2.16	2.36	2.57	2.77	3.25	3.79	4.39	0.00	0.02	0.04	0.06	0.08	0.10	0.12	0.14	0.16	0.18
300	2.32	2.56	2.80	3.05	3.33	3.62	3.92	4.61	5.38	6.23	0.00	0.03	0.06	0.09	0.12	0.15	0.18	0.21	0.24	0.26
400	2.93	3.24	3.56	3.87	4.23	4.62	5.00	5.88	6.87	7.96	0.00	0.04	0.08	0.12	0.16	0.20	0.23	0.27	0.31	0.35
500	3.50	3.89	4.27	4.65	5.09	5.55	6.02	7.08	8.27	9.58	0.00	0.05	0.10	0.15	0.20	0.24	0.29	0.34	0.39	0.44
600	4.04	4.49	4.94	5.38	5.90	6.44	6.98	8.21	9.59	11.09	0.00	0.06	0.12	0.18	0.24	0.29	0.35	0.41	0.47	0.53
700	4.56	5.07	5.58	6.08	6.66	7.28	7.89	9.28	10.82	12.48	0.00	0.07	0.14	0.21	0.27	0.34	0.41	0.48	0.55	0.62
800	5.04	5.61	6.18	6.74	7.39	8.07	8.75	10.28	11.96	13.75	0.00	0.08	0.16	0.23	0.31	0.39	0.47	0.55	0.63	0.71
900	5.50	6.13	6.75	7.36	8.07	8.82	9.55	11.20	13.00	14.90	0.00	0.09	0.18	0.26	0.35	0.44	0.53	0.62	0.71	0.79
1000	5.93	6.61	7.29	7.95	8.71	9.51	10.29	12.05	13.94	15.90	0.00	0.10	0.20	0.29	0.39	0.49	0.59	0.69	0.78	0.88
1100	6.33	7.06	7.79	8.49	9.30	10.15	10.98	12.82	14.77	16.75	0.00	0.11	0.22	0.32	0.43	0.54	0.65	0.75	0.86	0.97
1200	6.71	7.49	8.25	9.00	9.85	10.74	11.60	13.50	15.48	17.43	0.00	0.12	0.24	0.35	0.47	0.59	0.70	0.82	0.94	1.06
1300	7.05	7.87	8.67	9.46	10.34	11.27	12.15	14.09	16.07	17.95	0.00	0.13	0.26	0.38	0.51	0.64	0.76	0.89	1.02	1.15
1400	7.37	8.23	9.06	9.87	10.79	11.74	12.64	14.59	16.52	18.27	0.00	0.14	0.27	0.41	0.55	0.69	0.82	0.96	1.10	1.23
1500	7.66	8.55	9.41	10.24	11.18	12.14	13.05	14.99	16.84		0.00	0.15	0.29	0.44	0.59	0.73	0.88	1.03	1.18	1.32
1600	7.91	8.83	9.71	10.56	11.52	12.48	13.39	15.28	17.00		0.00	0.16	0.31	0.47	0.63	0.78	0.94	1.10	1.25	1.41
1700	8.14	9.07	9.97	10.83	1179	12.75	13.65	15.46			0.00	0.17	0.33	0.50	0.67	0.83	1.00	1.17	1.33	1.50
1800	8.32	9.28	10.19	11.05	12.00	12.95	13.82	15.52			0.00	0.18	0.35	0.53	0.71	0.88	1.06	1.23	1.41	1.59
1900	8.48	9.44	10.35	11.22	12.15	13.07	13.90				0.00	0.19	0.37	0.56	0.75	0.93	1.12	1.30	1.49	1.67
2000	8.59	9.56	10.47	11.32	12.24	13.12	13.89				0.00	0.20	0.39	0.59	0.78	0.98	1.17	1.37	1.57	1.76

(*Contd.*)

(*Contd.*)

2100	8.67	9.64	10.54	11.37	12.25	13.08					0.00	0.21	0.41	0.62	0.82	1.03	1.23	1.44	1.65	1.85
2200	8.71	9.67	10.55	11.35	12.19						0.00	0.22	0.43	0.65	0.86	1.08	1.29	1.51	1.72	1.94
2300	8.71	9.65	10.50	11.27							0.00	0.23	0.45	0.68	0.90	1.13	1.35	1.58	1.80	2.03
2400	8.67	9.58	10.40	11.12							0.00	0.23	0.47	0.70	0.94	1.18	1.41	1.65	1.88	2.12
2500	8.58	9.47	10.24								0.00	0.24	0.49	0.74	0.98	1.22	1.47	1.71	1.96	2.20
2600	8.45	9.30									0.00	0.25	0.51	0.76	1.02	1.27	1.53	1.78	2.04	2.29
2700	8.28	9.07									0.00	0.26	0.53	0.79	1.06	1.32	1.59	1.85	2.12	2.38
2800	8.05										0.00	0.27	0.55	0.82	1.10	1.37	1.64	1.92	2.19	2.47

(GOOD YEAR V-belt catalogue)

Note: * Preferred pulley diameters

Table 13.32 *Power ratings in kW (P_r) for D-section V- belts, 32 mm wide with 180º arc of contact on smaller pulley*

Speed of faster shaft	Power rating for smaller pulley pitch diameter of										Additional power increment per belt for speed ratio of									
	355 mm *	375 mm *	400 mm *	425 mm *	450 mm *	475 mm *	500 mm *	530 mm	560 mm *	600 mm *	1.00 to 1.01	1.02 to 1.04	1.05 to 1.08	1.09 to 1.12	1.13 to 1.18	1.19 to 1.24	1.25 to 1.34	1.35 to 1.51	1.52 to 1.99	2.00 and over
r.p.m.	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW
720	16.26	17.90	19.90	21.85	23.75	23.59	27.38	29.44	31.42	33.91	0.00	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25
960	19.26	21.16	23.45	25.63	27.70	29.65	31.47	33.50	35.32		0.00	0.33	0.67	1.00	1.34	1.67	2.00	2.33	2.67	3.00
1440	21.22	23.03	–	–	–	–	–	–	–	–	0.00	0.50	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50
100	3.39	3.70	4.08	4.45	4.83	5.20	5.57	6.02	6.46	7.04	0.00	0.03	0.07	0.10	0.14	0.17	0.21	0.24	0.28	0.31
200	6.04	6.61	7.32	8.02	8.72	9.42	10.11	10.93	11.75	12.83	0.00	0.07	0.14	0.21	0.28	0.35	0.42	0.49	0.56	0.63
300	8.41	9.22	10.24	11.24	12.24	13.22	14.20	15.37	16.52	18.04	0.00	0.10	0.21	0.31	0.42	0.52	0.62	0.73	0.83	0.94
400	10.57	11.61	12.91	14.19	15.45	16.70	17.94	19.40	20.85	22.74	0.00	0.14	0.28	0.42	0.56	0.70	0.83	0.97	1.11	1.25
500	12.55	13.80	15.35	16.87	18.38	19.86	21.32	23.03	24.72	26.91	0.00	0.17	0.35	0.52	0.70	0.87	1.04	1.22	1.39	1.56
600	14.34	15.79	17.56	19.30	21.01	22.68	24.32	26.23	28.09	30.49	0.00	0.21	0.42	0.62	0.83	1.04	1.25	1.46	1.67	1.88
700	15.96	17.57	19.54	21.46	23.33	25.15	26.91	28.96	30.92	33.41	0.00	0.24	0.49	0.73	0.97	1.22	1.46	1.70	1.95	2.19
800	17.39	19.14	21.26	23.32	25.31	27.22	29.06	31.17	33.16	35.62	0.00	0.28	0.56	0.83	1.11	1.39	1.67	1.95	2.22	2.50
900	18.62	20.48	22.72	24.87	26.92	28.87	30.73	32.81	34.73	37.02	0.00	0.31	0.63	0.94	1.25	1.56	1.87	2.19	2.50	2.81
1000	19.64	21.57	23.88	26.07	28.14	30.07	31.86	33.82	35.57		0.00	0.35	0.70	1.04	1.39	1.74	2.08	2.43	2.78	3.13
1100	20.43	22.40	24.74	26.91	28.92	30.76	32.42				0.00	0.38	0.77	1.15	1.53	1.91	2.29	2.63	3.06	3.44
1200	20.98	22.96	25.26	27.36	29.25						0.00	0.42	0.84	1.25	1.67	2.09	2.50	2.92	3.34	3.75
1300	21.27	23.21	25.42	27.38							0.00	0.45	0.91	1.35	1.81	2.26	2.71	3.16	3.61	4.06
1400	21.29	23.15	25.21								0.00	0.49	0.98	1.46	1.95	2.43	2.92	3.40	3.89	4.38
1500	21.03	22.76									0.00	0.52	1.05	1.56	2.09	2.61	3.12	3.65	4.17	4.69
1600	20.46										0.00	0.56	1.11	1.67	2.23	2.78	3.33	3.89	4.45	5.00

(GOOD YEAR V-belt catalogue)

Note: * Preferred pulley diameters

Table 13.33 *Power ratings in kW (P_r) for E-section V- belts, 32 mm wide with 180º arc of contact on smaller pulley*

Speed of faster shaft	Power rating for smaller pulley pitch diameter of										Additional power increment per belt for speed ratio of									
	500 mm *	530 mm	560 mm *	600 mm *	630 mm	650 mm	710 mm	750 mm	800 mm	900 mm *	1.00 to 1.01	1.02 to 1.04	1.05 to 1.08	1.09 to 1.12	1.13 to 1.18	1.19 to 1.24	1.25 to 1.34	1.35 to 1.51	1.52 to 1.99	2.00 and over
r.p.m.	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW	kW
585	26.6	28.2	32.1	35.6	37.7	40.1	43.7	47.4	50.8	56.0	0.00	0.44	0.89	1.33	1.77	2.21	3.07	3.11	3.54	3.98
720	30.8	32.0	35.3	39.0	41.1	44.2	47.2	50.1			0.00	0.55	1.09	1.64	2.19	2.73	3.28	3.85	4.37	4.92
960	31.1	33.5	36.5	–	–						0.00	0.73	1.46	2.18	2.91	3.64	4.37	5.08	5.82	6.54
50	3.83	4.13	4.92	5.03	5.33	5.77	6.20	6.80	7.38	8.45	0.00	0.04	0.08	0.08	0.14	0.19	0.23	0.26	0.31	0.34
100	6.85	7.41	8.19	9.11	9.68	10.49	11.3	12.4	13.4	15.3	0.00	0.08	0.14	0.23	0.31	0.37	0.45	0.53	0.61	0.68
150	9.59	10.4	11.4	12.7	13.6	14.7	15.9	17.6	19.1	21.8	0.00	0.11	0.23	0.34	0.45	0.56	0.68	0.79	0.91	1.02
200	12.1	13.1	14.6	16.3	17.3	18.0	20.4	22.3	24.3	27.7	0.00	0.14	0.31	0.45	0.61	0.76	0.91	1.06	1.20	1.36
250	14.4	13.7	17.5	19.5	20.8	22.6	24.3	26.7	29.1	32.9	0.00	0.19	0.37	0.56	0.76	0.95	1.14	1.33	1.52	1.70
300	16.6	18.2	20.1	22.4	24.0	26.1	28.2	30.9	33.7	38.2	0.00	0.23	0.45	0.68	0.91	1.14	1.36	1.59	1.83	2.05
350	18.7	20.5	22.8	25.3	27.0	29.4	31.7	34.8	37.7	42.8	0.00	0.26	0.53	0.79	1.06	1.33	1.59	1.86	2.12	2.43
400	20.7	22.4	25.1	27.9	29.7	31.5	34.9	38.2	41.4	46.6	0.00	0.31	0.61	0.91	1.21	1.52	1.83	2.12	2.43	2.73
450	22.4	24.4	27.3	30.4	32.2	35.0	37.7	41.3	44.7	50.2	0.00	0.34	0.68	1.02	1.36	1.71	2.05	2.39	2.73	3.07
500	24.0	26.3	29.2	32.6	34.5	37.5	40.4	44.0	47.4	50.5	0.00	0.37	0.76	1.13	1.52	1.89	2.28	2.65	3.03	3.41
550	25.6	27.9	31.0	34.4	36.6	39.6	42.5	45.2	49.6	55.0	0.00	0.42	0.84	1.24	1.66	2.08	2.50	2.91	3.33	3.75
600	27.0	29.3	32.6	36.2	38.3	41.5	44.3	48.0	51.4	56.4	0.00	0.45	0.91	1.38	1.83	2.28	2.73	3.18	3.63	4.09
650	28.1	30.5	33.9	37.5	39.7	42.8	45.8	49.3	52.5		0.00	0.50	0.98	1.47	1.97	2.45	2.95	3.44	3.94	4.43
700	29.0	31.6	35.0	38.7	40.8	43.9	46.8	50.2	53.0		0.00	0.53	1.06	1.60	2.12	2.65	3.18	3.70	4.25	4.77
750	29.8	32.5	35.9	39.5	41.7	44.7	47.3	50.4			0.00	0.56	1.14	1.70	2.28	2.84	3.41	3.98	4.54	5.11
800	30.5	33.1	36.4	40.0	42.1	44.9	47.3				0.00	0.61	1.21	1.83	2.43	3.04	3.63	4.25	4.85	5.46

(*Contd.*)

(*Contd.*)

850	30.8	33.4	36.7	40.3	42.1	44.9					0.00	0.65	1.29	1.93	2.57	3.22	3.86	4.50	5.15	5.80
900	31.1	33.7	36.9	40.2	41.8						0.00	0.68	1.36	2.05	2.73	3.41	4.09	4.77	5.46	6.15
950	31.0	33.4	36.6	39.5							0.00	0.73	1.44	2.16	2.88	3.60	4.31	5.04	5.75	6.48
1000	30.8	33.1	36.0								0.00	0.76	1.52	2.28	3.04	3.80	4.54	5.05	6.06	6.82
1050	30.4	32.6									0.00	0.79	1.60	2.39	3.18	3.98	4.77	5.57	6.37	7.16
1100	29.6										0.00	0.84	1.66	2.50	3.33	4.17	4.99	5.83	6.67	7.50

(GOOD YEAR V-belt catalogue)

Note: * Preferred pulley diameters

Table 13.34 *Correction factor for belt pitch length (F_c)*

Correction factor	Belt cross section					
	Z	A	B	C	D	E
	Belt pitch length (mm)					
0.80		630				
0.81			930			
0.82		700		1560	2740	
0.83			1000			
0.84		790		1760		
0.85			1100			
0.86	405	890			3130	
0.87			1210	1950	3330	
0.88		990				
0.89						
0.90	475	1100	1370	2190	3730	4660
0.91				2340		
0.92	530		1560	2490	4080	5040
0.93		1250				
0.94				2720	4620	5420
0.95	625		1760	2800		
0.96		1430		3080		6100
0.97			1950		5400	
0.98	700	1550		3310		
0.99		1640	2180	3520		6850
1.00	780	1750	2300		6100	
1.02		1940	2500	4060		7650
1.03					6840	
1.04	920	2050	2700			
1.05		2200	2850	4600	7620	9150
1.06		2300				
1.07	1080				8410	9950
1.08		2480	3200	5380		
1.09		2570			9140	10710
1.10		2700	3600			
1.11				6100		
1.12		2910			10700	12230
1.13		3080	4060			
1.14		3290		6860		13750
1.15			4430			
1.16		3540	4820	7600	12200	
1.17			5000		13700	15280
1.18			5370			
1.19			6070		15200	16800
1.20				9100		
1.21				10700		

(GOOD YEAR V-belt catalogue)

Table 13.35 *Correction factor for arc of contact (F_d)*

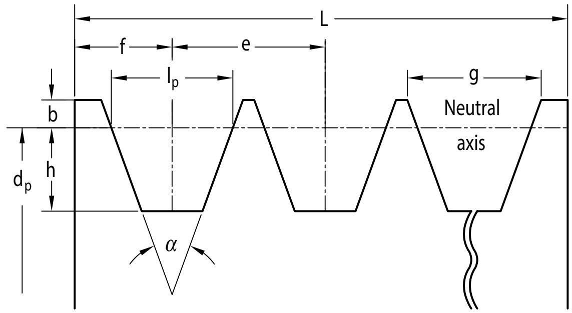
$\frac{D-d}{C}$	**Arc of contact on smaller pulley (°)**	**Correction factor F_d**
0.00	180	1.00
0.05	177	0.99
0.10	174	0.99
0.15	171	0.98
0.20	169	0.97
0.25	166	0.97
0.30	163	0.96
0.35	160	0.95
0.40	157	0.94
0.45	154	0.93
0.50	151	0.93
0.55	148	0.92
0.60	145	0.91
0.65	142	0.90
0.70	139	0.89
0.75	136	0.88
0.80	133	0.87
0.85	130	0.86
0.90	127	0.85
0.95	123	0.83
1.00	120	0.82

(GOOD YEAR V-belt catalogue)

(*Contd.*)

13.6 V-GROOVED PULLEYS

Table 13.36 *V-Grooved pulleys*



Notations	
l_p = pitch width of pulley groove or pitch width of belt. (It is the width of the belt at its neutral axis. The line in the *V*-belt, whose length remains unchanged when the belt is deformed under tension, is called its neutral axis) b = height of groove above the pitch line h = depth of groove below the pitch line e = centre-to-centre distance of adjacent grooves f = distance of the edge of pulley to first groove center α = groove angle d_p = pitch diameter of pulley. (It is diameter of the pulley measured at the pitch width of the groove) g = top width of the groove	
Designation	
A *V*-grooved pulley having 4 grooves in section B and pulley pitch diameter 200 mm is designated as *V*-grooved pulley 4B200.	
Materials	
V-grooved pulleys are usually made of cast iron, carbon steel castings or steel. Closed grained cast iron of Grade FG 200 is used for cast iron pulleys. If steel is used for pulleys, the groove surface should have a surface hardness not less than 130 BHN. The surface roughness value (R_a) is 6.3 microns.	
Tolerances for side wobble and run-out (eccentricity)	
Pulley diameter	**Tolerance**
(i) Up to and including 500 mm	0.01 mm per mm of pulley diameter
(ii) Over 500 mm up to and including 1500 mm	0.0015 mm per mm over 500 mm pulley diameter
(iii) Over 1500 mm	0.002 mm per mm over 1500 mm pulley diameter
Installation and take up allowances	
The limiting values for the adjustment of centres for two transmission pulleys should be as follows: (i) Lower limiting value- [nominal centre distance – 1.5% L_p] (ii) Higher limiting value- [nominal centre distance + 3.0% L_p] where L_p is pitch length of the belt.	

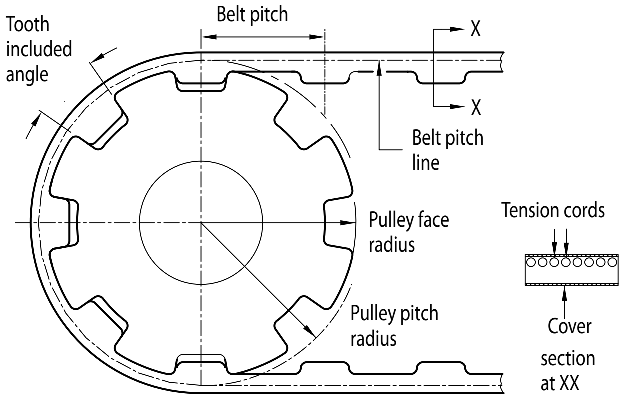
Table 13.37 *Dimensions of V-Grooved pulleys*

Groove section	l_p mm	b (min) mm	h (min) mm	e mm	f mm	α^o	d_p mm	g (min) mm	Outside diameter
Z	8.5	2.00	9	12±0.3	7–9	34	Up to 80	9.7	$d_p + 4.0$
A	11	2.75	11	15±0.3	9–12	34	Up to 118	12.7	$d_p + 5.5$
B	14	3.50	14	19±0.4	11.5–14.5	34	Up to 190	16.1	$d_p + 7.0$
C	19	4.80	19	25.5±0.5	16–19	34	Up to 315	21.9	$d_p + 9.6$
D	27	8.10	19.9	37±0.6	23–27	36	Up to 499	21.9	$d_p + 16.2$
E	32	9.60	23.4	44.5±0.7	28–33	36	Up to 629	38.2	$d_p + 19.2$
$L_{max} = (x - 1)\, e + 2f$ where x is the number of grooves									

(GOOD YEAR V-belt catalogue)

13.7 RIBBED V-BELTS

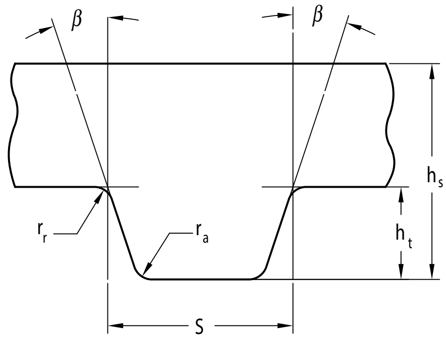
Table 13.38 *Ribbed V-belts*



Standard pitch codes	
Pitch code	**Belt pitch (mm)**
XL	5.080
L	9.525
H	12.700
XH	22.225
XXH	31.750

Note: Ribbed V-Belts are also known as synchronous belt, timing belt or toothed belt.

Table 13.39 *Dimensions of tooth for ribbed V-belts*



Pitch Code	2β (°)	S (mm)	h_t (mm)	r_r (mm)	r_a (mm)
XL	50	2.57	1.27	0.38	0.38
L	40	4.65	1.91	0.51	0.51
H	40	6.12	2.29	1.02	1.02
XH	40	12.57	6.35	1.57	1.19
XXH	40	19.05	9.53	2.29	1.52

Table 13.40 *Pitch lengths and tolerances of ribbed V-belts*

Belt length designation	Pitch length (mm)	Tolerance on pitch length (mm)	Number of Teeth for standard lengths				
			XL	L	H	XH	XXH
60	152.40	±0.41	30				
70	177.80	±0.41	35				
80	203.20	±0.41	40				
90	228.60	±0.41	45				
100	254.00	±0.41	50				
110	279.40	±0.46	55				
120	304.80	±0.46	60				
124	314.33	±0.46		33			
130	330.20	±0.46	65				
140	355.60	±0.46	70				
150	381.00	±0.46	75	40			
160	406.40	±0.51	80				
170	431.80	±0.51	85				
180	457.20	±0.51	90				
187	476.25	±0.51		50			
190	482.60	±0.51	95				
200	508.00	±0.51	100				
210	533.40	±0.61	105	56			
220	558.80	±0.61	110				

(*Contd.*)

(*Contd.*)

225	571.50	±0.61		60			
230	584.20	±0.61	115				
240	609.60	±0.61	120	64	48		
250	635.00	±0.61	125				
255	647.70	±0.61		68			
260	660.40	±0.61	130				
270	685.80	±0.61		72	54		
285	723.90	±0.61		76			
300	762.00	±0.61		80	60		
322	819.15	±0.66		86			
330	838.20	±0.66			66		
345	876.30	±0.66		92			
360	914.40	±0.66			72		
367	933.45	±0.66		98			
390	990.60	±0.66		104	78		
420	1066.80	±0.76		112	84		
450	1143.00	±0.76		120	90		
480	1219.20	±0.76		128	96		
507	1289.05	±0.81				58	
510	1295.40	±0.81		136	102		
540	1371.60	±0.81		144	108		
560	1422.40	±0.81				64	
570	1447.80	±0.81			114		
600	1524.00	±0.81		160	120		
630	1600.20	±0.86			126	72	
660	1676.40	±0.86			132		
700	1778.00	±0.86			140	80	56
750	1905.00	±0.91			150		
770	1955.80	±0.91				88	
800	2032.00	±0.91			160		64
840	2133.60	±0.97				96	
850	2159.00	±0.97			170		
900	2286.00	±0.97			180		72
980	2489.20	±1.02				112	
1000	2540.00	±1.02			200		80
1100	2794.00	±1.07			220		
1120	2844.80	±1.12				128	
1200	3048.00	±1.12					96
1250	3175.00	±1.17			250		
1260	3200.40	±1.17				144	
1400	3556.00	±1.22			280	160	112
1540	3911.60	±1.32				176	

(*Contd.*)

(*Contd.*)

1600	4064.00	±1.32					128
1700	4318.00	±1.37			340		
1750	4445.00	±1.42				200	
1800	4572.00	±1.42					144

Table 13.41 *Widths and heights of ribbed V-belts*

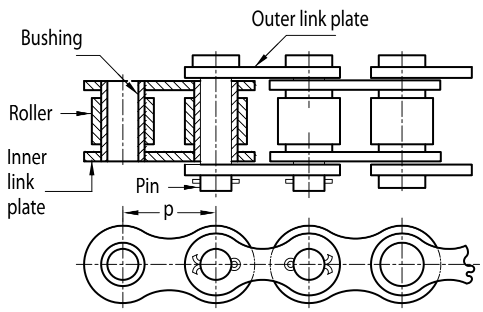
Pitch code	Nominal height h_s (mm) (Refer to figure in Table13.39 for dimension h_s)	Standard Widths		Tolerance on width for belt pitch (mm)		
		Dimensions (mm)	Designation	Up to and including 838.20 mm	Lengths over 838.20 mm up to and including 1676.40 mm	Over 1676.40 mm
XL	2.3	6.4	025	+0.5 –0.8	—	—
		7.9	031			
		9.5	037			
L	3.6	12.7	050	+0.8 –0.8	+0.8 –1.3	—
		19.1	075			
		25.4	100			
H	4.3	19.1	075	+0.8 –0.8	+0.8 –1.3	+0.8 –1.3
		25.4	100			
		38.1	150			
		50.8	200	+0.8 –1.3	+1.3 –1.3	+1.3 –1.5
		76.2	300	+1.3 –1.5	+1.5 –1.5	+1.5 –2.0
XH	11.2	50.8	200	—	+4.8 –4.8	+4.8 –4.8
		76.2	300			
		101.6	400			
XXH	15.7	50.8	200	—	—	+4.8 –4.8
		76.2	300			
		101.6	400			
		127.0	500			

Designation: The belt designation includes length designation, pitch code and width designation. For example, 450H200: (i) pitch length = 1143.00 mm, (ii) belt pitch = 12.70 mm and (iii) width = 50.8 mm.

Chain Drives 14

14.1 DIMENSIONS OF ROLLER CHAINS

Table 14.1 *Roller chains*



Construction
Roller chain consists of five parts: pin, bushing, roller, and inner and outer link plates. The pin is press fitted to two outer link-plates, while the bush is press fitted to inner link-plates. The bush and the pin form a swivel joint and the outer link plates are free to swivel with respect to the inner link plates. The rollers are freely fitted on bushes and, during engagement, turn with the teeth of the sprocket wheel. This results in rolling friction (instead of sliding friction) between the roller and the sprocket teeth. The rolling friction reduces wear and frictional power loss and improves the efficiency of the chain drive. The inner and outer link plates are made of medium carbon steels. These link plates are blanked from cold-rolled sheets and hardened to 50 HRC. The pins, bushes and rollers are made of case carburising alloy steels and hardened to 50 HRC.

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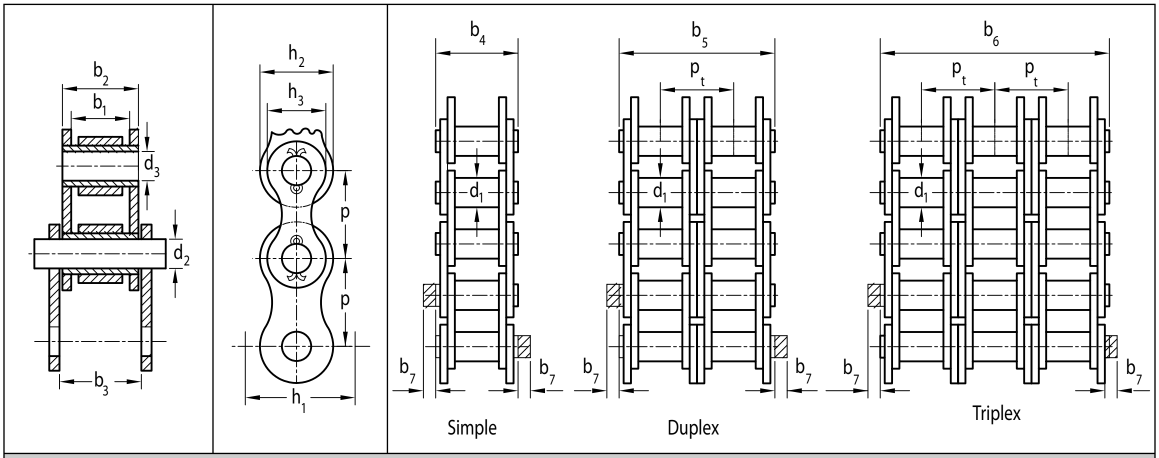
Designation of roller chains
The pitch (p) of the chain is the linear distance between the axes of adjacent rollers. Roller chains are standardised and manufactured on the basis of the pitch. It is designated in the following way, Chain number is given such as 08B or 16A. It consists of two parts – a number followed by a letter. The number in two digits expresses the 'pitch' in sixteenths of an 'inch'. The letter A means American Standard ANSI series and letter B means British Standard series. Most of the chain manufacturers are American and their ANSI series is popular in engineering industries. (i) Let us consider the designation '08B'. The pitch of this chain is (08/16) inch or (08/16) × (25.4) mm i.e., 12.7 mm. The letter B indicates British standard series. (ii) Let us consider the designation '16A'. The pitch of this chain is (16/16) inch or 1 inch i.e., 25.4 mm. The letter A indicates American Standard ANSI series. The chain number is supplemented by a hyphenated suffix 1 for simple chain, 2 for duplex chain, 3 for triplex chain and so on. For example, 08B-2 or 16A-1

Note: Roller chain is also known as power transmission chain.

Table 14.2 *Terminology of roller chains*



Notations			
p = pitch of chain (mm) p_t = transverse pitch (mm) d_1 = roller diameter (mm) b_1 = width between inner plate (mm) d_2 = bearing pin body diameter (mm)	d_3 = bush bore (mm) h_1 = chain path depth (mm) h_2 = inner plate depth (mm) h_3 = outer intermediate plate depth (mm)	b_2 = width over inner link (mm) b_3 = width between outer plate (mm) b_7 = additional width for joint fastener (mm)	b_4 = width over bearing pins (for simple chain) (mm) b_5 = width over bearing pins (for duplex chain) (mm) b_6 = width over bearing pins (for triplex chain) (mm)

Table 14.3 *Dimensions of roller chains*

ISO chain number	p	d_1 max	b_1 min	d_2 max	d_3 min	h_1 min	h_2 max	h_3 max	p_t	b_2 max	b_3 min	b_4 max	b_5 max	b_6 max	b_7 max
05B	8.00	5.00	3.00	2.31	2.36	7.37	7.11	7.11	5.64	4.77	4.90	8.6	14.3	19.9	3.1
06B	9.525	6.35	5.72	3.28	3.33	8.52	8.26	8.26	10.24	8.53	8.66	13.5	23.8	34.0	3.3
08A	12.70	7.95	7.85	3.96	4.01	12.33	12.07	10.41	14.38	11.18	11.23	17.8	32.3	46.7	3.9
08B	12.70	8.51	7.75	4.45	4.50	12.07	11.81	10.92	13.92	11.30	11.43	17.0	31.0	44.9	3.9
10A	15.875	10.16	9.40	5.08	5.13	15.35	15.09	13.03	18.11	13.84	13.89	21.8	39.9	57.9	4.1
10B	15.875	10.16	9.65	5.08	5.13	14.99	14.73	13.72	16.59	13.28	13.41	19.6	36.2	52.8	4.1
12A	19.05	11.91	12.57	5.94	5.99	18.34	18.08	15.62	22.78	17.75	17.81	26.9	49.8	72.6	4.6
12B	19.05	12.07	11.68	5.72	5.77	16.39	16.13	16.13	19.46	15.62	15.75	22.7	42.2	61.7	4.6
16A	25.40	15.88	15.75	7.92	7.97	24.39	24.13	20.83	29.29	22.61	22.66	33.5	62.7	91.9	5.4
16B	25.40	15.88	17.02	8.28	8.33	21.34	21.08	21.08	31.88	25.45	25.58	36.1	68.0	99.9	5.4
20A	31.75	19.05	18.90	9.53	9.58	30.48	30.18	26.04	35.76	27.46	27.51	41.1	77.0	113.0	6.1
20B	31.75	19.05	19.56	10.19	10.24	26.68	26.42	26.42	36.45	29.01	29.14	43.2	79.7	116.1	6.1
24A	38.10	22.23	25.22	11.10	11.15	36.55	36.20	31.24	45.44	35.46	35.51	50.8	96.3	141.7	6.6
24B	38.10	25.40	25.40	14.63	14.68	33.73	33.40	33.40	48.36	37.92	38.05	53.4	101.8	150.2	6.6
28A	44.45	25.40	25.22	12.70	12.75	42.67	42.24	36.45	48.87	37.19	37.24	54.9	103.6	152.4	7.4
28B	44.45	27.94	30.99	15.90	15.95	37.46	37.08	37.08	59.56	46.58	46.71	65.1	124.7	184.3	7.4
32A	50.80	28.58	31.55	14.27	14.32	48.74	48.26	41.66	58.55	45.21	45.26	65.5	124.2	182.9	7.9
32B	50.80	29.21	30.99	17.81	17.86	42.72	42.29	42.29	58.55	45.57	45.70	67.4	126.0	184.5	7.9
40A	63.50	39.68	37.85	19.84	19.89	60.93	60.33	52.07	71.55	54.89	54.94	80.3	151.9	223.5	10.2
40B	63.50	39.37	38.10	22.89	22.94	53.49	52.96	52.96	72.29	55.75	55.88	82.6	154.9	227.2	10.2
48A	76.20	47.63	47.35	23.80	23.85	73.13	72.39	62.48	87.83	67.82	67.87	95.5	183.4	271.3	10.5
48B	76.20	48.26	45.72	29.24	29.29	64.52	63.88	63.88	91.21	70.56	70.69	99.1	190.4	281.6	10.5
56B	88.90	53.98	53.34	34.32	34.37	78.64	77.85	77.85	106.60	81.33	81.46	114.6	221.2	—	11.7
64B	101.60	63.50	60.96	39.40	39.45	91.08	90.17	90.17	119.89	92.02	92.15	130.9	250.8	—	13.0
72B	114.30	72.39	68.58	44.48	44.53	104.67	103.63	103.63	136.27	103.81	103.94	147.4	283.7	—	14.3

(DIAMOND chain catalogue) (ISO: 606)

Note: All dimensions are in mm.

Table 14.4 *Breaking load of roller chains*

Chain number	Breaking load (min) N		
	Simple	Duplex	Triplex
05B	4 400	7 800	11 100
06B	8 900	16 900	24 900
08A	13 800	27 600	41 400
08B	17 800	31 100	44 500
10A	21 800	43 600	65 400
10B	22 200	44 500	66 700
12A	31100	62 300	93 400
12B	28 900	57 800	86 700
16A	55 600	111 200	166 800
16B	42 300	84 500	126 800
20A	86 700	173 500	260 200
20B	64 500	129 000	193 500
24A	124 600	249 100	373 700
24B	97 900	195 700	293 600
28A	169 000	338 100	507 100
28B	129 000	258 000	387 000
32A	222 400	444 800	667 200
32B	169 000	338 100	507 100
40A	347 000	693 900	1040 900
40B	262 400	524 900	787 300
48A	500 400	1000 800	1501 300
48B	400 300	800 700	1201 000
56B	542 700	1085 400	—
64B	711 700	1423 400	—
72B	898 500	1797 100	—

(DIAMOND chain catalogue) (ISO: 606)

Table 14.5 *Dimensions and breaking loads of cycle and moped chains*

ISO chain number	p	d_1 max	b_1 min	d_2 max	d_3 min	h_1 min	h_2 max	h_3 max	b_2 max	b_3 min	b_4 max	b_7 max	Breaking load min (N)
081	12.7	7.75	3.30	3.66	3.71	10.17	9.91	9.91	5.80	5.93	10.2	1.5	8000
082	12.7	7.75	2.38	3.66	3.71	10.17	9.91	9.91	4.60	4.73	8.2	—	9800
083	12.7	7.75	4.88	4.09	4.14	10.56	10.30	10.30	7.90	8.03	12.9	1.5	11 600
084	12.7	7.75	4.88	4.09	4.14	11.41	11.15	11.15	8.80	8.93	14.8	1.5	15 600
085	12.7	7.77	6.38	3.58	3.63	10.17	9.91	9.91	9.07	9.20	14.0	2.0	6700

(DIAMOND chain catalogue) (ISO: 606)

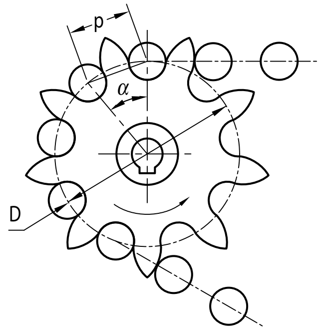
Note: All dimensions are in mm.

Table 14.6 *ISO chain number and ANSI chain number*

ISO chain number	08A	10A	12A	16A	20A	24A	28A	32A	40A
ANSI number	ANSI-40	ANSI-50	ANSI-60	ANSI-80	ANSI-100	ANSI-120	ANSI-140	ANSI-160	ANSI-200

14.2 CHAIN EQUATIONS

Table 14.7 *Geometrical relationships for roller chains*



Pitch angle	
$\alpha = \dfrac{360}{z}$ (14.1)	α = pitch angle (degrees) z = number of teeth on sprocket
Pitch circle diameter of sprocket	
$D = \dfrac{p}{\sin\left(\dfrac{180}{z}\right)}$ (14.2)	D = pitch circle diameter of sprocket (mm) p = pitch of chain (mm)
Velocity ratio	
$i = \dfrac{n_1}{n_2} = \dfrac{z_2}{z_1}$ (14.3)	i = velocity ratio n_1, n_2 = speeds of rotation of driving and driven shafts (rpm) z_1, z_2 = number of teeth on driving and driven sprockets
Average velocity of chain	
$v = \dfrac{\pi D n}{60 \times 10^3} = \dfrac{z p n}{60 \times 10^3}$ (14.4)	v = average velocity of chain (m/s) n = speed of rotation of sprocket (rpm)
Length of chain	
$L = L_n \times p$ (14.5)	L = length of chain (mm) L_n = number of links in chain
Number of links in chain	

$$L_n = 2\left(\frac{a}{p}\right) + \left(\frac{z_1 + z_2}{2}\right) + \left(\frac{z_2 - z_1}{2\pi}\right)^2 \times \left(\frac{p}{a}\right) \quad (14.6)$$

a = centre distance between axes of driving and driven sprockets (mm)
z_1 = number of teeth on the smaller sprocket
z_2 = number of teeth on the larger sprocket

(*Contd.*)

(*Contd.*)

Centre distance	
$$a=\frac{p}{4}\left\{\left[L_n-\left(\frac{z_1+z_2}{2}\right)\right]+\sqrt{\left[L_n-\left(\frac{z_1+z_2}{2}\right)\right]^2-8\left[\frac{z_2-z_1}{2\pi}\right]^2}\right\}$$	(14.7)

Note: The centre distance calculated by above formula does not provide any sag. In practice, a small amount of sag is essential for the links to take the best position on the sprocket wheel. The centre distance is, therefore, reduced by a margin of (0.002a to 0.004a) to account for the sag.

14.3 SELECTION OF CHAINS

Table 14.8 *Power rating of simple roller chains (06B to 16B)*

Pinion speed (rpm)	Power (kW)								
	06 B	**08A**	**08 B**	**10A**	**10 B**	**12A**	**12 B**	**16A**	**16 B**
50	0.14	0.28	0.34	0.53	0.64	0.94	1.07	2.06	2.59
100	0.25	0.53	0.64	0.98	1.18	1.74	2.01	4.03	4.83
200	0.47	0.98	1.18	1.83	2.19	3.40	3.75	7.34	8.94
300	0.61	1.34	1.70	2.68	3.15	4.56	5.43	11.63	13.06
500	1.09	2.24	2.72	4.34	5.01	7.69	8.53	16.99	20.57
700	1.48	2.95	3.66	5.91	6.71	10.73	11.63	23.26	27.73
1000	2.03	3.94	5.09	8.05	8.97	14.32	15.65	28.63	34.89
1400	2.73	5.28	6.81	11.18	11.67	14.32	18.15	18.49	38.47
1800	3.44	6.98	8.10	8.05	13.03	10.44	19.85	—	—
2000	3.80	6.26	8.67	7.16	13.49	8.50	20.57	—	—

Note: (i) The values of power rating given in above table are for simple (single-strand) roller chains and based on the assumption that there are 17 teeth on the driving sprocket wheel. (ii) The above information is compiled from various catalogues of chain manufacturers. The values given in table are approximate. For exact values, the reader should refer to the specific catalogue of chain manufacturers like Diamond and Acme Chains.

Table 14.9 *Power rating of simple roller chains (20A to 40A)*

Pinion speed (rpm)	Power (kW)								
	20A	**20B**	**24A**	**24B**	**28A**	**28B**	**32A**	**32B**	**40A**
25	2.24	2.68	3.75	4.83	5.81	6.80	9.08	10.20	11.88
50	4.29	5.10	7.15	8.95	11.18	12.52	15.65	19.24	22.37
100	7.96	9.66	13.42	16.99	20.57	23.20	30.42	35.79	42.05
200	14.31	17.94	25.05	31.31	38.47	43.84	56.36	67.10	79.63

(*Contd.*)

(*Contd.*)

300	21.47	26.31	35.79	45.03	54.57	63.22	80.52	98.42	116.31
400	26.84	34.27	46.53	58.15	71.57	82.31	111.84	120.78	134.20
500	34.00	42.05	56.36	71.57	89.47	96.63	134.20	129.70	—
600	39.37	47.41	67.10	78.28	107.30	103.78	—	—	—
700	44.73	52.78	76.05	84.99	80.52	109.15	—	—	—
800	49.21	54.87	57.86	89.02	—	—	—	—	—
900	40.26	56.96	48.31	93.05	—	—	—	—	—
1000	34.00	59.05	—	—	—	—	—	—	—

Note: (i) The values of power rating given in above table are for simple (single-strand) roller chains and based on the assumption that there are 17 teeth on the driving sprocket wheel. (ii) The above information is compiled from various catalogues of chain manufacturers. The values given in table are approximate. For exact values, the reader should refer to the specific catalogue of chain manufacturers like Diamond and Acme Chains.

Table 14.10 *Service factor (K_s)*

Type of driven load	Type of input power		
	I.C. Engine with hydraulic drive	**Electric motor or turbine**	**I.C. Engine with mechanical drive**
Agitators for liquid	1.0	1.0	1.2
Beaters	1.2	1.3	1.4
Centrifugal blowers and fans	1.0	1.0	1.2
Boat propellers	1.2	1.3	1.4
Compressors			
(i) Centrifugal and lobe	1.2	1.3	1.4
(ii) Reciprocating with more than 3 cylinders	1.2	1.3	1.4
(iii) Reciprocating with 1 and 2 cylinders	1.4	1.5	1.7
Conveyors			
(i) Belt or chain–smoothly loaded	1.0	1.0	1.2
(ii) Heavy duty–not uniformly loaded	1.2	1.3	1.4
Clay working machinery			
(i) Pug mills	1.2	1.3	1.4
(ii) Brick presses and briquetting machinery	1.4	1.5	1.7
Crushers	1.4	1.5	1.7
Bucket elevators			
(i) Smoothly loaded or fed	1.0	1.0	1.2
(ii) Non-uniformly loaded or fed	1.2	1.3	1.4
Feeders			
(i) Rotary table	1.0	1.0	1.2
(ii) Apron, screw and rotary vane	1.2	1.3	1.4
(iii) Reciprocating	1.4	1.5	1.7

(*Contd.*)

(*Contd.*)

Food processing Slicers, dough mixers and grinders	1.2	1.3	1.4
Kilns and Dryers	1.2	1.3	1.4
Machine tools (i) Drills, grinders and lathes (ii) Boring and milling machines (iii) Punch presses and shears	 1.0 1.2 1.4	 1.0 1.3 1.5	 1.2 1.4 1.7
General machinery (i) Uniform load and non-reversing (ii) Moderate shock load and non-reversing (iii) Severe shock load and reversing	 1.0 1.2 1.4	 1.0 1.3 1.5	 1.2 1.4 1.7
Mills (i) Ball, pebble and tube (ii) Hammer and rolling	 1.2 1.4	 1.3 1.5	 1.4 1.7
Pumps (i) Centrifugal (ii) Reciprocating with more than 3 cylinders (iii) Reciprocating with 1 and 2 cylinders	 1.0 1.2 1.4	 1.0 1.3 1.5	 1.2 1.4 1.7
Paper industry (i) Pulp grinders (ii) Calendars, mixers and sheeters	 1.2 1.4	 1.3 1.5	 1.4 1.7
Printing presses Magazine and newspaper	 1.4	 1.5	 1.7
Textile industry (i) Calendars, mangles and nappers (ii) Carding machinery	 1.2 1.4	 1.3 1.5	 1.4 1.7
Woodworking machinery	1.2	1.3	1.4

(DIAMOND chain catalogue)

Table 14.11 *Multiple strand factor (K_1)*

Number of strands	K_1
1	1.0
2	1.7
3	2.5
4	3.3
5	3.9
6	4.6

(DIAMOND chain catalogue)

Note: The values of the power rating given in Table 14.8 and 14.9 are based on the assumption that the chain has a single strand. There are duplex and triplex chains too. This variation in the number of strands is taken into account by the multiple strand factor K_1.

Table 14.12 *Tooth correction factor (K_2)*

Number of teeth on the driving sprocket	K_2
11	0.53
12	0.62
13	0.70
14	0.78
15	0.85
16	0.92
17	1.00
18	1.05
19	1.11
20	1.18
21	1.26
22	1.29
23	1.35
24	1.41
25	1.46
27	1.57
29	1.68
31	1.77
35	1.95
40	2.15
45	2.37
50	2.51
55	2.66
60	2.80

(DIAMOND chain catalogue)

Note: The power rating given in Tables 14.8 and 14.9 are based on 17 teeth on the driving sprocket wheel. In a given application, the number of teeth on the driving sprocket wheel can be less than or more than 17. The tooth correction factor K_2 accounts for this variation.

Table 14.13 *kW ratings of chain*

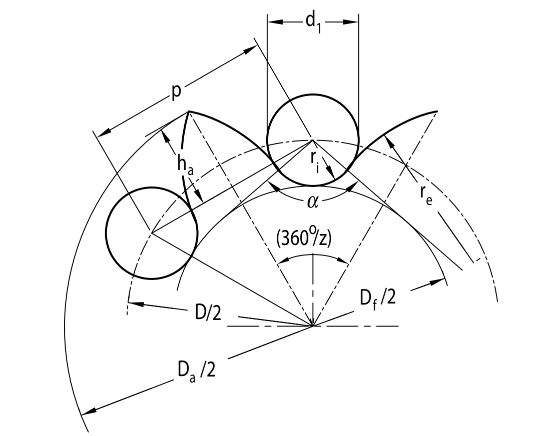
$$\text{kW rating of chain} = \frac{(\text{kW to be transmitted}) \times K_s}{K_1 \times K_2} \tag{14.8}$$

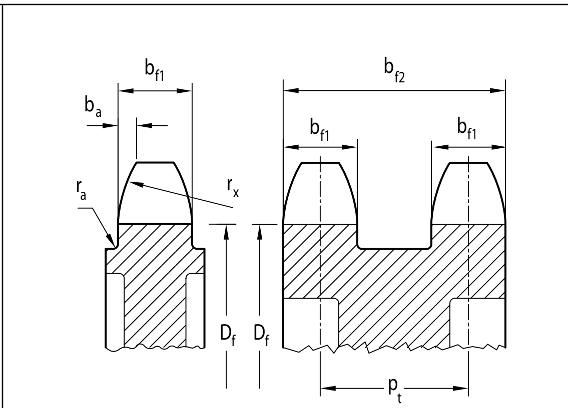
K_s = service factor (Table 14.10)
K_1 = multiple strand factor (Table 14.11)
K_2 = tooth correction factor (Table 14.12)

Note: (i) For satisfactory performance of roller chains, the center distance between the sprockets should provide at least a 120° wrap angle on the smaller sprocket.
(ii) In practice, the recommended centre distance is between 30 to 50 chain pitches. $30\,p < a < 50\,p$ (14.9)
(iii) The expected service life of the chains is 15,000 hr.
(iv) The velocity ratio should be kept below 6:1 to get a satisfactory performance.

14.4 SPROCKET WHEELS

Table 14.14 *Proportions of sprocket wheel*





Dimension	Notation	Equation
1. Chain pitch	p	(Table 14.3)
2. Pitch circle diameter	D	$D = \dfrac{p}{\sin\left(\dfrac{180°}{z}\right)}$
3. Roller diameter	d_1	(Table 14.3)
4. Width between inner plates	b_1	(Table 14.3)
5. Transverse pitch	p_t	(Table 14.3)
6. Top diameter	D_a	$(D_a)_{\max} = D + 1.25\,p - d_1$ $(D_a)_{\min} = D + p\left(1 - \dfrac{1.6}{z}\right) - d_1$
7. Root diameter	D_f	$D_f = D - 2\,r_i$
8. Roller seating radius	r_i	$(r_i)_{\max} = (0.505\,d_1 + 0.069\sqrt[3]{d_1})$ $(r_i)_{\min} = 0.505 d_1$
9. Tooth flank radius	r_e	$(r_e)_{\max} = 0.008 d_1\,(z^2 + 180)$ $(r_e)_{\min} = 0.12 d_1\,(z + 2)$
10. Roller seating angle	α	$\alpha_{\min} = \left[120 - \dfrac{90}{z}\right]$ $\alpha_{\max} = \left[140 - \dfrac{90}{z}\right]$
11. Tooth height above the pitch polygon	h_a	$(h_a)_{\max} = 0.625\,p - 0.5 d_1 + \dfrac{0.8\,p}{z}$ $(h_a)_{\min} = 0.5(p - d_1)$

(*Contd.*)

(*Contd.*)

12. Tooth side radius	r_x	$(r_x)_{nom} = p$
13. Tooth width (for simple chain)	b_{f1}	$b_{f1} = 0.93 b_1$ if $p \leq 12.7$ mm $b_{f1} = 0.95 b_1$ if $p > 12.7$ mm
14. Tooth side relief	b_a	$(b_a)_{nom} = 0.13p$

(ISO: 606)

Note: b_{f2} or b_{f3} = (number of strands – 1) $\times p_t + b_{f1}$.

15

Rolling Contact Bearings

15.1 ROLLING CONTACT BEARINGS

Table 15.1 *Types of rolling contact bearings*

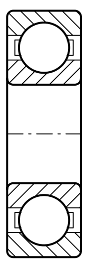
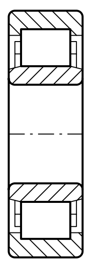
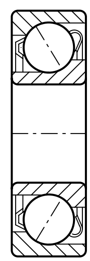
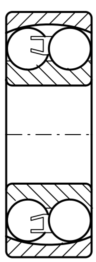
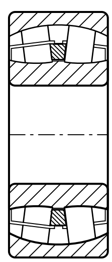
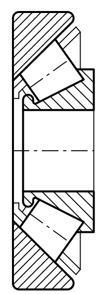
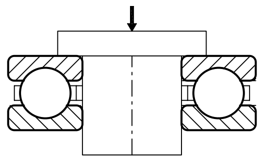
Deep groove ball bearing	**Cylindrical roller bearing**	**Angular contact bearing**
		
Self-aligning ball bearing	**Spherical roller bearing**	**Taper roller bearing**
		
Thrust ball bearing		
		

Table 15.2 *Basic equations of rolling contact bearings*

Equivalent dynamic load – General equation	
$P = X V F_r + Y F_a$ (15.1)	P = equivalent dynamic load (N) F_r = radial load acting on bearing (N) F_a = axial or thrust load acting on bearing (N) V = race-rotation factor X = radial factor Y = thrust factor **Note:** Values of X and Y are given in the manufacturer's catalogues
Note: The race-rotation factor depends upon whether the inner race is rotating or the outer race. The value of V is one, when the inner race rotates while the outer race is held stationary in the housing. The value of V is 1.2, when the outer race rotates with respect to load, while the inner race remains stationary. In most of the applications, the inner race rotates and the outer race is fixed in the housing. Assuming V as unity, the common equation for equivalent dynamic load is written.	
Equivalent dynamic load –Common equation	
$P = X F_r + Y F_a$ (15.2)	**Note:** When the bearing is subjected to pure radial load F_r, $P = F_r$ (15.3) When the bearing is subjected to pure thrust load F_a, $P = F_a$ (15.4)
Load life equation	
$L_{10} = \left(\dfrac{C}{P}\right)^p$ (15.5)	L_{10} = rated bearing life (in million revolutions) C = dynamic load capacity (N) p = 3 (for ball bearings) p = 10/3 (for roller bearings)
For all types of ball bearings, $C = P\,(L_{10})^{1/3}$ (15.6)	For all types of roller bearings, $C = P\,(L_{10})^{0.3}$ (15.7)
Life in hours	
$L_{10} = \dfrac{60 n L_{10h}}{10^6}$ (15.8)	L_{10h} = rated bearing life (hours) n = speed of rotation (r.p.m.)
Cyclic loads and speeds (variation of load in steps)	
$P_e = \sqrt[3]{\left[\dfrac{N_1 P_1^3 + N_2 P_2^3 + \cdots}{N_1 + N_2 + \cdots}\right]}$ (15.9) $P_e = \sqrt[3]{\left[\dfrac{\Sigma N P^3}{\Sigma N}\right]}$ (15.10) $N = N_1 + N_2 + \ldots$ (15.11)	P_e = equivalent dynamic load for complete work cycle (N) $P_1, P_2, \ldots$ = dynamic load during first, second, ... element of work cycle (N) $N_1, N_2, \ldots$ = number of revolutions completed by first, second, ... element of work cycle (rev) N = life of complete work cycle (rev)

(*Contd.*)

(*Contd.*)

Cyclic loads and speeds (continuous variation of load)	
$$P_e = \left[\frac{1}{N}\int P^3 dN\right]^{1/3}$$ (15.12)	
Bearing with probability of survival other than 90%	
$R = e^{-(L/a)^b}$ (15.13)	R = reliability (in fraction) L = corresponding life (in million revolution) a and b = constants $a = 6.84$ and $b = 1.17$ (15.14)
$\left(\frac{L}{L_{10}}\right) = \left[\frac{\log_e\left(\frac{1}{R}\right)}{\log_e\left(\frac{1}{R_{90}}\right)}\right]^{1/b}$ (15.15)	R = reliability (in fraction) L = corresponding life L_{10} = life corresponding to a reliability of 90% or R_{90} $R_{90} = 0.9$ (15.16)
$L_{50} = 5\ L_{10}$ (15.17)	L_{50} = median life or life which 50% of the bearings will complete or exceed before fatigue failure
System reliability	
$R_S = (R)^N$ (15.18)	N = number of bearings in the system, each having the same reliability R R_s = reliability of the complete system **Note:** R_s indicates the probability of one out of N bearings failing during its life time.

Table 15.3 *Bearing life for road and rail vehicles (L_{10s})*

Type of vehicle	L_{10s} (in millions of km)
Wheel hub bearings	
Private cars	0.2
Commercial vehicles and buses	0.4
Axle box bearings for rail vehicles	
Goods wagons	0.8
Suburban stock, tram cars	1.5
Main line passenger carriages	3
Main line motor unit	3 to 4
Main line electric and diesel locomotives	3 to 5
$L_{10} = \left(\frac{1000}{\pi D}\right) L_{10s}$ (15.19)	L_{10s} = nominal life (millions of kilometres) D = wheel diameter (m)

Table 15.4 *Bearing life for wheel applications*

Wheel application	L_{10} (in million rev)
Automobile cars	50
Trucks	100
Trolley cars	500
Railroad cars	1000

Table 15.5 *Bearing life for different classes of machines*

Class of machine	L_{10h} (operating hours)
(i) Domestic machines, agricultural machines, instruments, technical apparatus for medical use	300 to 3000
(ii) Machines used for short periods or intermittently: Electric hand tools, lifting tackles in workshops, construction machines	3000 to 8000
(iii) Machines to work with high operational reliability during short periods or intermittently: Lifts, cranes for packaged goods or slings of drums, bales, etc.	8000 to 12000
(iv) Machines used for eight hours of service per day but not always fully utilised: Gear drives for general purposes, electric motors for industrial use, rotary crushers	10000 to 25000
(v) Machines used for eight hours of service per day and fully utilized: Machine tools, wood working machinery, machines for engineering industry, cranes for bulk materials, ventilator fans, conveyor belts, printing equipment, separators and centrifuges	20 000 to 30000
(vi) Machines used for continuous operation 24 hours per day: Rolling mill gear units, medium sized electrical machinery, compressors, mine hoists, pumps, textile machinery	40000 to 50000
(vii) Water works machinery, rotary furnaces, cable stranding machines, propulsion machinery for ocean-going vessels	60000 to 100000
(viii) Pulp and paper making industry, large electric machinery, power station plants, mine pumps and mine ventilator fans, tunnel shaft bearings for ocean-going vessels	≈ 100000

Table 15.6 *Values of load factor*

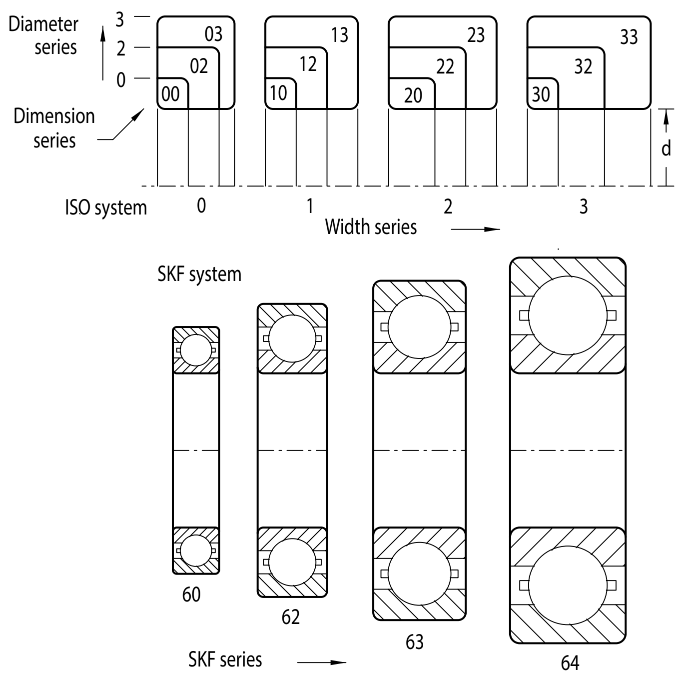
Type of drive	Load factor
(A) Gear drives:	
(i) Rotating machines free from impact like electric motors and turbo-compressors	1.2–1.4
(ii) Reciprocating machines like internal combustion engines and compressors	1.4–1.7
(iii) Impact machines like hammer mills	2.5–3.5
(B) Belt drives:	
(i) V-belts	2.0
(ii) Single-ply leather belt	3.0
(iii) Double-ply leather belt	3.5
(C) Chain drives	1.5

Note: The forces acting on the bearing are calculated by considering the equilibrium of forces in vertical and horizontal planes. These elementary equations do not take into consideration the effect of dynamic load. The forces determined by these equations are multiplied by the load factor to determine the dynamic load carrying capacity of the bearing.

Table 15.7 *Coefficient of friction*

Type of bearing	Coefficient of friction (μ)
(i) Deep groove ball bearing	0.0015
(ii) Self aligning ball bearing	0.0010
(iii) Single-row angular contact bearing	0.0020
(iv) Double-row angular contact bearing	0.0024
(v) Cylindrical roller bearing with cage	0.0011
(vi) Spherical roller bearing	0.0018
(vii) Taper roller bearing	0.0018
(viii) Thrust ball bearing	0.0013
(ix) Cylindrical roller thrust bearing	0.0050
(x) Spherical roller thrust bearing	0.0018
$M = \mu F \left(\dfrac{d}{2} \right)$ (15.20)	M = frictional moment (N-mm) F = bearing load (N) d = bore diameter of bearing (mm)

Table 15.8 *Designation of rolling contact bearings*



The rolling contact bearing is designated by three or four digits. The meaning of these digits is as follows:

(i) The last two digits indicate the bore diameter of the bearing in mm (bore diameter divided by 5). For example, XX07 indicates a bearing of 35 mm bore diameter.

(*Contd.*)

(*Contd.*)

(ii) The third digit from the right indicates the series of the bearing. The numbers used to indicate the series are as follows:

Extra light series –1
Light series – 2
Medium series – 3
Heavy series – 4

For example, X307 indicates a medium series bearing with bore diameter of 35 mm.

(iii) The fourth digit and sometimes fifth digit from right specifies the type of rolling contact bearing. For example digit 6 indicates deep groove ball bearings

Light series bearings permit smallest bearing width and outer diameter for a given bore diameter. They have lowest load carrying capacities. Medium series bearings have 30 to 40 per cent higher dynamic load carrying capacities compared with light series bearings of the same bore diameter. However, they occupy more radial and axial space. Heavy series bearings have 20 to 30 per cent higher dynamic load carrying capacities compared with medium series bearings of the same bore diameter.

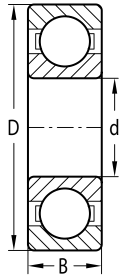
The ISO plan for the dimension series of the bearing consists of two digit numbers. The first number indicates the width series 8, 0, 1, 2, 3, 4, 5, and 6 in order of increasing width. The second number indicates the diameter series 7, 8, 9, 0, 1, 2, 3, and 4 in order of ascending outer diameter of the bearing.

15.2 DEEP GROOVE BALL BEARINGS

Table 15.9 *X and Y factors for single-row deep groove ball bearings (For normal clearance)*

$\left(\frac{F_a}{C_0}\right)$	$\left(\frac{F_a}{F_r}\right) \le e$		$\left(\frac{F_a}{F_r}\right) > e$		e
	X	Y	X	Y	
0.025	1	0	0.56	2.0	0.22
0.040	1	0	0.56	1.8	0.24
0.070	1	0	0.56	1.6	0.27
0.130	1	0	0.56	1.4	0.31
0.250	1	0	0.56	1.2	0.37
0.500	1	0	0.56	1.0	0.44

Table 15.10 *Dimensions and static and dynamic load capacities of single-row deep groove ball bearings*



Principal dimensions (mm)			Basic load ratings (N)		Designation (SKF)
d	*D*	*B*	*C*	C_0	
10	26	8	4620	1960	6000
	30	9	5070	2240	6200
	35	11	8060	3750	6300
20	42	12	9360	4500	6004
	47	14	12700	6200	6204
	52	15	15900	7800	6304
	72	19	30700	16600	6404
30	55	13	13300	6800	6006
	62	16	19500	10000	6206
	72	19	28100	14600	6306
	90	23	43600	24000	6406
40	68	15	16800	9300	6008
	80	18	30700	16600	6208
	90	23	41000	22400	6308
	110	27	63700	36500	6408
50	80	16	21600	13200	6010
	90	20	35100	19600	6210
	110	27	61800	36000	6310
	130	31	87100	52000	6410
60	95	18	29600	18300	6012
	110	22	47500	28000	6212
	130	31	81900	48000	6312
	150	35	108 000	69500	6412

(*Contd.*)

(*Contd.*)

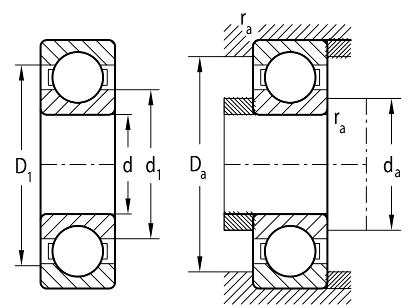
70	110	20	37700	24500	6014
	125	24	61800	37500	6214
	150	35	104 000	63000	6314
	180	42	143 000	104 000	6414
80	125	22	47500	31500	6016
	140	26	70200	45000	6216
	170	39	124 000	80000	6316
	200	48	163 000	125 000	6416
90	140	24	58500	39000	6018
	160	30	95600	62000	6218
	190	43	143 000	98000	6318
	225	54	186 000	146 000	6418
100	150	24	60500	41500	6020
	180	34	124 000	78000	6220
	215	47	174 000	132 000	6320
110	170	28	81900	57000	6022
	200	38	146 000	100 000	6222
	240	50	203 000	166 000	6322
120	180	28	85200	61000	6024
	215	40	146 000	100 000	6224
	260	55	208 000	166 000	6324
130	200	33	106 000	78000	6026
	230	40	156 000	112 000	6226
	280	58	229 000	193 000	6326
140	210	33	111 000	83000	6028
	250	42	165 000	122 000	6228
	300	62	255 000	224 000	6328
150	225	35	125 000	96500	6030
	270	45	174 000	137 000	6230
	320	65	276 000	250 000	6330
160	240	38	143 000	112 000	6032
	290	48	186 000	146 00	6232
	340	68	276 000	250 000	6332

(*Contd.*)

(*Contd.*)

170	260	42	168 000	134 000	6034
	310	52	212 000	180 00	6234
	360	72	312 000	305 000	6334
180	280	46	190 000	156 000	6036
	320	52	229 000	196 000	6236
	380	75	351 000	360 000	6336
190	290	46	195 000	166 000	6038
	340	55	255 000	232 000	6238
	400	78	371 000	380 000	6338
200	310	51	216 000	190 000	6040
	360	58	270 000	250 000	6240

Table 15.11 *Abutment and fillet dimensions of single row deep groove ball bearings*



d	d_1	D_1	d_a	D_a	r_a	Designation (SKF)
	(≈)	(≈)	(Min.)	(Max.)	(Max.)	
10	14.4	21.4	12	24	0.3	6000
	16.7	23.2	14	26	0.6	6200
	17.7	26.8	14	31	0.6	6300
20	27.2	35.1	24	38	0.6	6004
	28.5	38.7	25	42	1	6204
	30.3	42.1	26.5	45.5	1	6304
	37.1	55.6	26.5	65.5	1	6404
30	38.2	47.1	35	50	1	6006
	40.3	52.1	35	57	1	6206
	44.6	59.9	36.5	65.5	1	6306
	50.3	70.7	38	82	1.5	6406

(*Contd.*)

(*Contd.*)

40	49.2	59.1	45	63	1	6008
	52.6	67.9	46.5	73.5	1	6208
	56.1	74.7	48	82	1.5	6308
	62.8	88	49	101	2	6408
50	59.7	70.6	55	75	1	6010
	62.5	78.1	56.5	83.5	1	6210
	68.7	92.1	59	101	2	6310
	75.4	106	61	119	2	6410
60	71.3	84.1	66.5	88.5	1	6012
	75.5	94.2	68	102	1.5	6212
	81.8	109	71	119	2	6312
	88.1	123	71	139	2	6412
70	82.8	97.6	76.5	103.5	1	6014
	87	109	78	117	1.5	6214
	94.9	126	81	139	2	6314
	103	147	83	167	2.5	6414
80	94.4	112	86.5	118.5	1	6016
	101	123	89	131	2	6216
	108	143	91	159	2	6316
	116	164	93	187	2.5	6416
90	105	125	98	132	1.5	6018
	112	139	99	151	2	6218
	121	160	103	177	2.5	6318
	132	182	106	209	3	6418
100	115	135	108	142	1.5	6020
	124	157	111	169	2	6220
	135	181	113	202	2.5	6320
110	129	152	119	161	2	6022
	138	174	121	189	2	6222
	149	201	123	227	2.5	6322
120	139	162	129	171	2	6024
	150	185	131	204	2	6224
	164	216	133	247	2.5	6324

(*Contd.*)

(*Contd.*)

130	152	179	139	191	2	6026
	162	199	143	217	2.5	6226
	177	233	146	264	3	6326
140	162	189	149	201	2	6028
	177	214	153	237	2.5	6228
	190	250	156	284	3	6328
150	174	202	161	214	2	6030
	192	229	163	257	2.5	6230
	205	265	166	304	3	6330
160	185	216	171	229	2	6032
	206	244	173	277	2.5	6232
	217	282	176	324	3	6332
170	198	233	181	249	2	6034
	217	261	186	294	3	6234
	229	300	186	344	3	6334
180	211	250	191	269	2	6036
	225	275	196	304	3	6236
	243	316	196	364	3	6336
190	221	260	201	279	2	6038
	238	291	206	324	3	6238
	258	332	210	380	4	6338
200	235	277	211	299	2	6040
	253	304	216	344	3	6240

Note: All dimensions are in mm.

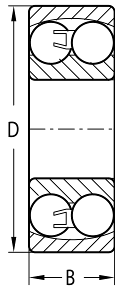
15.3 SELF-ALIGNING BALL BEARINGS

Table 15.12 *Equivalent dynamic load for self-aligning ball bearings (For normal clearance)*

When $\left(\frac{F_a}{F_r}\right) \leq e$	$P = F_r + Y_1 F_a$
When $\left(\frac{F_a}{F_r}\right) > e$	$P = 0.65 F_r + Y_2 F_a$

Note: The values of Y_1, Y_2 and e for each individual bearing are given Table 15.13.

Table 15.13 *Dimensions, static and dynamic load capacities and factors Y_1, Y_2, and e for self-aligning ball bearings (Bearings with cylindrical bore)*



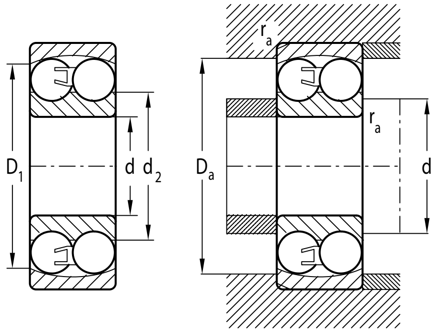
Principal dimensions (mm)			Basic load ratings (N)		Designation (SKF)	*e*	Y_1	Y_2
d	D	B	C	C_0				
10	30	9	5530	1340	1200	0.33	1.9	3
	30	14	7280	1700	2200	0.65	0.97	1.5
20	47	14	9950	3200	1204	0.27	2.3	3.6
	47	18	12500	3900	2204	0.48	1.3	2
	52	15	12400	4000	1304	0.30	2.1	3.3
	52	21	18200	5300	2304	0.52	1.2	1.9
30	62	16	15600	5850	1206	0.24	2.6	4.1
	62	20	15300	5700	2206	0.40	1.6	2.4
	72	19	21200	7800	1306	0.25	2.5	3.9
	72	27	31200	10000	2306	0.44	1.4	2.2
40	80	18	19000	8650	1208	0.22	2.9	4.5
	80	23	22500	9500	2208	0.33	1.9	3
	90	23	29600	12200	1308	0.24	2.6	4.1
	90	33	44900	15600	2308	0.43	1.5	2.3
50	90	20	22900	10800	1210	0.20	3.2	4.9
	90	23	23400	11400	2210	0.28	2.2	3.5
	110	27	43600	17600	1310	0.24	2.6	4.1
	110	40	63700	23600	2310	0.43	1.5	2.3
60	110	22	30200	15600	1212	0.18	3.5	5.4
	110	28	33800	16600	2212	0.28	2.2	3.5
	130	31	57200	26500	1312	0.23	2.7	4.2
	130	46	87100	33500	2312	0.40	1.6	2.4

(*Contd.*)

(*Contd.*)

70	125	24	34500	19000	1214	0.18	3.5	5.4
	125	31	44200	22800	2214	0.27	2.3	3.6
	150	35	74100	35500	1314	0.22	2.9	4.5
	150	51	111 000	45000	2314	0.37	1.7	2.6
80	140	26	39700	23600	1216	0.16	3.9	6.1
	140	33	48800	27000	2216	0.25	2.5	3.9
	170	39	88400	42500	1316	0.22	2.9	4.5
	170	58	135 000	58500	2316	0.37	1.7	2.6
90	160	30	57200	32000	1218	0.17	3.7	5.7
	160	40	70200	38000	2218	0.27	2.3	3.6
	190	43	117 000	56000	1318	0.22	2.9	4.5
	190	64	153 000	69500	2318	0.37	1.7	2.6
100	180	34	68900	40500	1220	0.17	3.7	5.7
	180	46	97500	53000	2220	0.27	2.3	3.6
	215	47	143 000	72000	1320	0.23	2.7	4.2

Table 15.14 *Abutment and fillet dimensions for self-aligning ball bearings (Bearings with cylindrical bore)*



d	d_2	D_1	d_a	D_a	r_a	Designation (SKF)
	(≈)	(≈)	(Min.)	(Max.)	(Max.)	
10	16.7	24.4	14	26	0.6	1200
	15.3	25.2	14	26	0.6	2200
20	28.9	39.1	25	42	1	1204
	28	40.4	25	42	1	2204
	31.3	43.6	26.5	45.5	1	1304
	28.8	43.7	26.5	45.5	1	2304

(*Contd.*)

(*Contd.*)

30	40.1	53.2	35	57	1	1206
	40	53	35	57	1	2206
	44.9	60.9	36.5	65.5	1	1306
	41.7	60.9	36.5	65.5	1	2306
40	53.6	68.8	46.5	73.5	1	1208
	52.4	68.8	46.5	73.5	1	2208
	57.5	76.8	48	82	1.5	1308
	53.5	76.8	48	82	1.5	2308
50	62.3	78.7	56.5	83.5	1	1210
	62.5	79.3	56.5	83.5	1	2210
	70.1	95	59	101	2	1310
	65.8	94.4	59	101	2	2310
60	77.8	97.5	68	102	1.5	1212
	75.5	96.1	68	102	1.5	2212
	87	115	71	119	2	1312
	76.9	112	71	119	2	2312
70	87.4	109	78	117	1.5	1214
	87.5	111	78	117	1.5	2214
	97.7	129	81	139	2	1314
	91.6	130	81	139	2	2314
80	101	125	89	131	2	1216
	98.8	124	89	131	2	2216
	109	147	91	159	2	1316
	104	148	91	159	2	2316
90	112	142	99	151	2	1218
	112	142	99	151	2	2218
	122	165	103	177	2.5	1318
	115	164	103	177	2.5	2318
100	127	159	111	169	2	1220
	125	160	111	169	2	2220
	136	185	113	202	2.5	1320

Note: All dimensions are in mm.

15.4 ANGULAR CONTACT BEARINGS

Table 15.15 *Arrangement of bearing pairs*

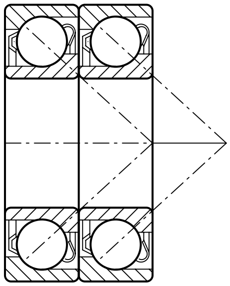
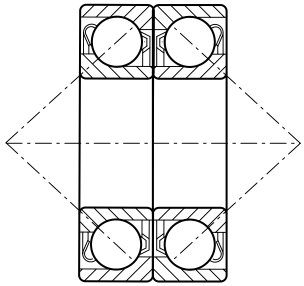
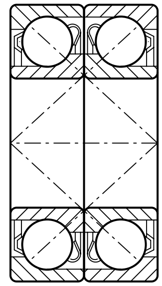
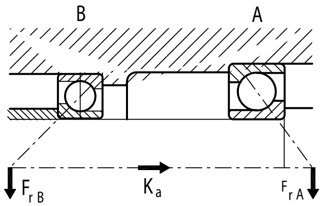
Tandem arrangement	Back-to-back arrangement	Face-to-face arrangement
		

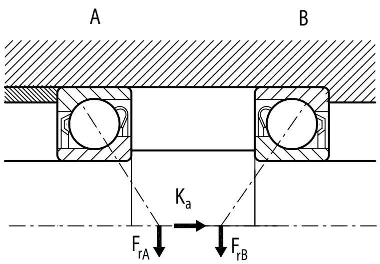
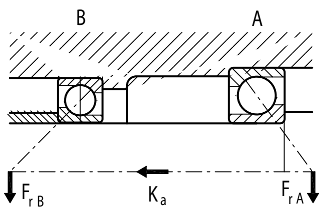
Table 15.16 *Equivalent dynamic load for single row angular contact bearings (B-type)*

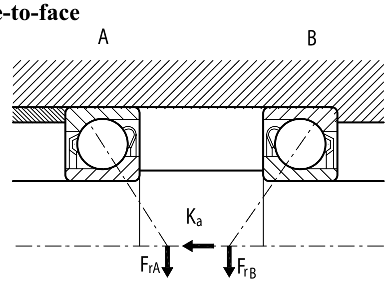
Bearings mounted singly or paired in tandem	
When $\left(\dfrac{F_a}{F_r}\right) \leq 1.14$	$P = F_r$
When $\left(\dfrac{F_a}{F_r}\right) > 1.14$	$P = 0.35F_r + 0.57F_a$
For bearings pairs arranged back-to-back or face-to-face	
When $\left(\dfrac{F_a}{F_r}\right) \leq 1.14$	$P = F_r + 0.55F_a$
When $\left(\dfrac{F_a}{F_r}\right) > 1.14$	$P = 0.57F_r + 0.93F_a$

Note: (i) For paired bearings, F_r and F_a are the loads acting on the pair.
(ii) B-type single row angular contact bearings have contact angle of 40° and have non-separable construction. They are suitable for heavy axial loads and high speeds.

Table 15.17 *Axial loading of single row angular contact bearings (B-type)*

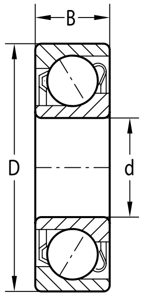
<table>
<tr><th>Arrangement</th><th>Load case</th><th>Axial loads</th></tr>
<tr><td rowspan="3">Back-to-back


Face-to-face

</td>
<td>1(a)
$F_{rA} \geq F_{rB}$
$K_a \geq 0$</td>
<td>$F_{aA} = 1.14 F_{rA}$
$F_{aB} = F_{aA} + K_a$</td></tr>
<tr><td>1 (b)
$F_{rA} < F_{rB}$
$K_a \geq 1.14(F_{rB} - F_{rA})$</td>
<td>$F_{aA} = 1.14 F_{rA}$
$F_{aB} = F_{aA} + K_a$</td></tr>
<tr><td>1 (c)
$F_{rA} < F_{rB}$
$K_a < 1.14(F_{rB} - F_{rA})$</td>
<td>$F_{aA} = F_{aB} - K_a$
$F_{aB} = 1.14 F_{rB}$</td></tr>
<tr><td rowspan="3">Back-to-back


Face-to-face

</td>
<td>2 (a)
$F_{rA} \leq F_{rB}$
$K_a \geq 0$</td>
<td>$F_{aA} = F_{aB} + K_a$
$F_{aB} = 1.14 F_{rB}$</td></tr>
<tr><td>2 (b)
$F_{rA} > F_{rB}$
$K_a \geq 1.14(F_{rA} - F_{rB})$</td>
<td>$F_{aA} = F_{aB} + K_a$
$F_{aB} = 1.14 F_{rB}$</td></tr>
<tr><td>2 (c)
$F_{rA} > F_{rB}$
$K_a < 1.14(F_{rA} - F_{rB})$</td>
<td>$F_{aA} = 1.14 F_{rA}$
$F_{aB} = F_{aA} - K_a$</td></tr>
</table>

Note: B-type single row angular contact bearings have contact angle of 40° and have non-separable construction.

Table 15.18 *Dimensions and static and dynamic load capacities of (SKF) single row angular contact ball bearings*



Principal dimensions (mm)			Basic load ratings (N)		Designation (SKF)
d	*D*	*B*	*C*	C_0	
10	30	9	4940	2120	7200 B
20	47	14	13300	6550	7204 B
	52	15	17400	8500	7304 B
30	62	16	20300	11000	7206 B
	72	19	31200	17000	7306B
40	80	18	31900	18600	7208 B
	90	23	44900	25500	7308 B
50	90	20	37700	22800	7210 B
	110	27	67600	45000	7310 B
60	110	22	55900	36500	7212 B
	130	31	90400	56000	7312 B
70	125	24	68900	45500	7214 B
	150	35	114 000	72000	7314 B
80	140	26	80600	55000	7216 B
	170	39	135 000	91500	7316 B
90	160	30	106 000	73500	7218 B
	190	43	156 000	114 000	7318B
100	180	34	130 000	91500	7220 B
	215	47	190 000	150 000	7320 B
110	200	38	158 000	114 000	7222 B
	240	50	225 000	193 000	7322 B
120	215	40	165 000	129 000	7224 B
	260	55	238 000	208 000	7324 B

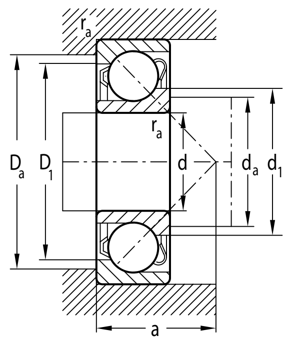
(*Contd.*)

(*Contd.*)

130	230	40	186 000	150 000	7226 B
	280	58	251 000	224 000	7326 B
140	250	42	182 000	150 000	7228 B
	300	62	276 000	255 000	7328 BG
150	270	45	195 000	170 000	7230 BG
	320	65	302 000	300 000	7330 BG

Note: (i) B-type single-row angular contact bearings have contact angle of 40° and have non-separable construction. (ii) G-type bearing is a special design of B-type and intended for paired mountings.

Table 15.19 *Abutment and fillet dimensions of single-row angular contact ball bearings*



d	d_1 (≈)	D_1 (≈)	a	d_a (Min.)	D_a (Max.)	r_a (Max.)	Designation (SKF)
10	18.3	22	13	15	25	0.6	7200 B
20	30.7	37.4	21	26	41	1	7204 B
	32.7	40.7	23	27	45	1	7304 B
30	42.7	50.6	27	36	56	1	7206 B
	47.7	58	31	37	65	1	7306B
40	55.9	65.7	34	47	73	1	7208 B
	59.8	72.3	39	49	81	1.5	7308 B
50	65.6	76.1	39	57	83	1	7210 B
	73.4	89.3	47	60	100	2	7310 B
60	79.5	92.7	47	69	101	1.5	7212 B
	87.4	106	56	72	118	2	7312 B
70	91.5	106	53	79	116	1.5	7214 B
	101	123	64	82	138	2	7314 B
80	103	120	59	90	130	2	7216 B
	115	139	72	92	158	2	7316 B

(*Contd.*)

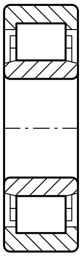
(*Contd.*)

90	117	136	68	100	150	2	7218 B
	129	156	80	104	176	2.5	7318B
100	131	152	76	112	168	2	7220 B
	145	176	90	114	201	2.5	7320 B
110	145	169	64	122	188	2	7222 B
	160	195	99	124	226	2.5	7322 B
120	157	180	90	132	203	2	7224 B
	175	210	107	134	246	2.5	7324 B
130	169	193	96	144	216	2.5	7226 B
	189	227	115	148	262	3	7326 B
140	184	208	103	154	236	2.5	7228 B
	203	243	123	158	282	3	7328 BG
150	199	223	111	164	256	2.5	7230 BG
	218	258	131	168	302	3	7330 BG

Note: (i) All dimensions are in mm.
(ii) B-type single-row angular contact bearings have contact angle of 40° and have non-separable construction.
(iii) G-type bearing is a special design of B-type and intended for paired mountings.

15.5 CYLINDRICAL ROLLER BEARINGS

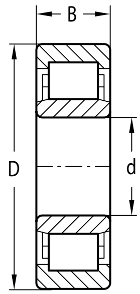
Table 15.20 *Equivalent dynamic load for cylindrical roller bearings (NU-type)*



NU-type bearing
NU type bearings have two integral flanges on the outer ring and a flangeless inner ring. They permit axial displacement of the housing relative to the shaft within certain limits in both directions. They are used as non-locating bearings.
Equivalent dynamic load
NU-type of bearings are subjected to radial loads. The equivalent dynamic load is given by, $$P = F_r$$

Note: There are certain types of cylindrical roller bearings with flanges on both inner and outer rings. They can take axial load in addition to radial loads.

Table 15.21 *Dimensions and static and dynamic load capacities of (SKF) cylindrical roller bearings (NU type)*



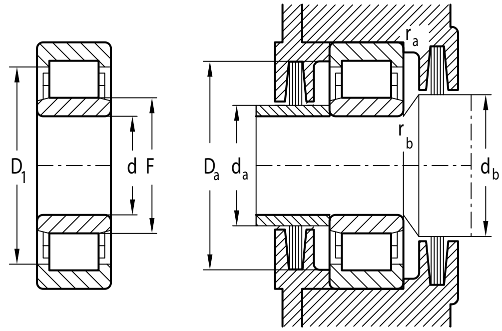
Principal dimensions (mm)			Basic load ratings (N)		Designation (SKF)
d	***D***	***B***	***C***	C_0	
20	47	14	14700	7350	NU 204
	52	15	20500	10400	NU 304
30	62	16	22400	12000	NU 206
	72	19	36900	20000	NU 306
	90	23	60500	34000	NU 406
40	80	18	41800	24000	NU 208
	90	23	56100	32500	NU 308
	110	27	96800	57000	NU 408
50	90	20	45700	27500	NU 210
	110	27	88000	52000	NU 310
	130	31	130 000	78000	NU 410
60	110	22	64400	40000	NU 212
	130	31	123 000	76500	NU 312
	150	35	168 000	106 000	NU 412
70	125	24	79200	51000	NU 214
	150	35	151 000	95000	NU 314
	180	42	229 000	150 000	NU 414
80	140	26	106 000	69500	NU 216
	170	39	190 000	122 000	NU 316
	200	48	303 000	200 000	NU 416

(*Contd.*)

(*Contd.*)

90	160	30	142 000	93000	NU 218
	190	43	242 000	160 000	NU 318
	225	54	385 000	260 000	NU 418
100	180	34	183 000	125 000	NU 220
	215	47	303 000	204 000	NU 320
	250	58	429 000	290 000	NU 420
110	200	38	229 000	156 000	NU 222
	240	50	391 000	270 000	NU 322
	280	65	523 000	355 000	NU 422
120	215	40	260 000	183 000	NU 224
	260	55	457 000	310 000	NU 324
	310	72	644 000	455 000	NU 424
130	230	40	270 000	193 000	NU 226
	280	58	539 000	380 000	NU 326
140	250	42	308 000	224 000	NU 228
	300	62	594 000	425 000	NU 328
150	270	45	358 000	260 000	NU 230

Table 15.22 *Abutment and fillet dimensions for cylindrical roller bearings (NU type)*



d	D_1	F	d_a	d_a	d_b	D_a	r_a	r_b	**Designation (SKF)**
	≈		**(Min.)**	**(Max.)**	**(Min.)**	**(Max.)**	**(Max.)**	**(Max.)**	
20	36.7	27	24	25	29	42	1	0.6	NU 204
	39.8	28.5	24	26	30	45.5	1	0.6	NU 304

(*Contd.*)

(*Contd.*)

30	49.1	38.5	34	37	40	57	1	0.6	NU 206
	56.7	42	36.5	40	44	65.5	1	1	NU 306
	65.8	45	38	43	47	82	1.5	1.5	NU 406
40	64.7	50	46.5	48	52	73.5	1	1	NU 208
	71.2	53.5	48	51	55	82	1.5	1.5	NU 308
	83.3	58	49	56	60	101	2	2	NU 408
50	75.1	60.4	56.5	58	62	83.5	1	1	NU 210
	87.3	65	59	63	67	101	2	2	NU 310
	101	70.8	61	68	73	119	2	2	NU 410
60	91.6	73.5	68	71	76	102	1.5	1.5	NU 212
	104	77	71	74	79	119	2	2	NU 312
	116	83	71	80	85	139	2	2	NU 412
70	104	84.5	78	81	87	117	1.5	1.5	NU 214
	120	90	81	86	92	139	2	2	NU 314
	139	100	83	97	102	167	2.5	2.5	NU 414
80	118	95.3	89	92	97	131	2	2	NU 216
	136	103	91	99	106	159	2	2	NU 316
	156	110	93	106	113	187	2.5	2.5	NU 416
90	134	107	99	104	110	151	2	2	NU 218
	153	115	103	111	118	177	2.5	2.5	NU 318
	175	123.5	106	120	126	209	3	3	NU 418
100	150	120	111	117	123	169	2	2	NU 220
	172	129.5	113	125	132	202	2.5	2.5	NU 320
	194	139	116	135	142	234	3	3	NU 420
110	167	132.5	121	129	135	189	2	2	NU 222
	192	143	123	139	146	227	2.5	2.5	NU 322
	216	155	126	150	158	264	3	3	NU 422

(*Contd.*)

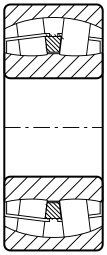
(*Contd.*)

120	180	143.5	131	139	146	204	2	2	NU 224
	209	154	133	150	157	247	2.5	2.5	NU 324
	238	170	140	165	173	290	4	4	NU 424
130	192	156	143	152	159	217	2.5	2.5	NU 226
	225	167	146	163	170	264	3	3	NU 326
140	208	169	153	165	172	237	2.5	2.5	NU 228
	241	180	156	175	183	284	3	3	NU 328
150	225	182	163	178	185	257	2.5	2.5	NU 230

Note: All dimensions are in mm.

15.6 SPHERICAL ROLLER BEARINGS

Table 15.23 *Equivalent dynamic load for spherical roller bearings*



CC-type bearing

Depending upon size and series, spherical roller bearings are classified as CC, CAC and CA. Bearings of CC design have two robust window-type cages of pressed steel or brass. Bearings of CAC and CA designs have a one-piece double-pronged machined cage of steel or brass and retaining flanges on the inner ring.

W33 designation

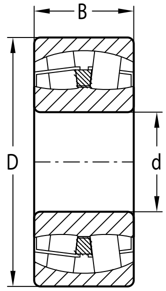
Designation W33 indicates spherical roller bearing with an annular groove and three lubrication holes in the outer ring. This facilitates efficient lubrication of the parts of bearing.

Equivalent dynamic load

When $\left(\frac{F_a}{F_r}\right) \leq e$	$P = F_r + Y_1 F_a$
When $\left(\frac{F_a}{F_r}\right) > e$	$P = 0.67 F_r + Y_2 F_a$

Values of e, Y_1 and Y_2 for each individual bearing are given in Table 15.25.

Table 15.24 *Dimensions and static and dynamic load capacities of (SKF) spherical roller bearings (with cylindrical bore)*



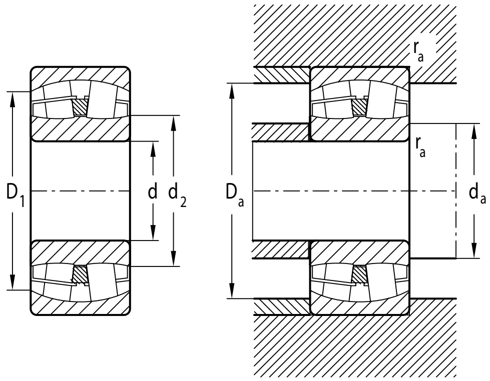
Principal dimensions (mm)			Basic load ratings (N)		Designation (SKF)
d	***D***	***B***	***C***	C_0	
50	90	23	79900	54000	22210 CC
	110	27	120 000	81500	21310 CC
	110	40	176 000	120 000	22310 CC
60	110	28	122 000	83000	22212 CC
	130	31	161 000	114 000	21312 CC
	130	46	235 000	166 000	22312 CC
70	125	31	148 000	104 000	22214 CC
	150	35	207 000	150 000	21314 CC
	150	51	311 000	228 000	22314 CC/W33
80	140	33	176 000	127 000	22216 CC
	170	39	258 000	190 000	21316 CC
	170	58	374 000	275 000	22316 CC/W33
90	160	40	244 000	183 000	22218 CC/W33
	190	43	322 000	240 000	21318 CC
	190	64	477 000	365 000	22318 CC/W33
100	180	46	311 000	236 000	22220 CC/W33
	215	47	385 000	300 000	21320 CC
	215	73	610 000	475 000	22320 CC/W33
110	200	53	408 000	320 000	22222 CC/W33
	240	50	460 000	355 000	21322 CC
	240	80	725 000	570 000	22322 CC/W33
120	215	58	466 000	375 000	22224 CC/W33
	215	76	610 000	520 000	23224 CC/W33
	260	86	845 000	670 000	22324 CC/W33
130	230	64	552 000	450 000	22226 CC/W33
	230	80	690 000	600 000	23226 CC/W33
	280	93	978 000	780 000	22326 CC/W33

(*Contd.*)

(*Contd.*)

140	250	68	621 000	500 000	22228 CC/W33
	250	88	799 000	710 000	23228 CC/W33
	300	102	1130 000	915 000	22328 CC/W33
150	270	73	736 000	620 000	22230 CC/W33
	270	96	937 000	830 000	23230 CC/W33
	320	108	1290 000	1040 000	22330 CC/W33
160	290	80	863 000	735 000	22232 CC/W33
	290	104	1070 000	950 000	23232 CC/W33
	340	114	1380 000	1160 000	22332 CC/W33
170	310	86	978 000	830 000	22234 CC/W33
	310	110	1220 000	1100 000	23234 CC/W33
	360	120	1540 000	1290 000	22334 CC/W33
180	320	86	1010 000	880 000	22236 CC/W33
	320	112	1290 000	1200 000	23236 CC/W33
	380	126	1730 000	1460 000	22336 CC/W33
190	340	92	1110 000	965 000	22238 CC/W33
	340	120	1460 000	1370 000	23238 CC/W33
	400	132	1870 000	1560 000	22338 CC/W33
200	360	98	1270 000	1100 000	22240 CC/W33
	360	128	1610 000	1530 000	23240 CC/W33
	420	138	2020 000	1730 000	22340 CC/W33

Table 15.25 *Abutment and fillet dimensions and factors Y_1, Y_2 and e for spherical roller bearings (with cylindrical bore)*



d	***d*₂**	***D*₁**	***d*ₐ**	***D*ₐ**	***r*ₐ**	***e***	***Y*₁**	***Y*₂**	**Designation (SKF)**
	≈	≈	**(Min.)**	**(Max.)**	**(Max.)**				
50	60	79.2	57	83	1	0.24	2.8	4.2	22210 CC
	66.9	92.4	60	100	2	0.21	3.2	4.8	21310 CC
	63.1	91.7	60	100	2	0.37	1.8	2.7	22310 CC

(*Contd.*)

(*Contd.*)

60	72	96.6	69	101	1.5	0.24	2.8	4.2	22212 CC
	79.7	113	72	118	2	0.20	3.4	5	21312 CC
	74.9	108	72	118	2	0.37	1.8	2.7	22312 CC
70	84	111	79	116	1.5	0.23	2.9	4.4	22214 CC
	92.6	127	82	138	2	0.20	3.4	5	21314 CC
	88	126	82	138	2	0.35	1.9	2.9	22314 CC/W33
80	95	123	90	130	2	0.22	3	4.6	22216 CC
	105	145	92	158	2	0.19	3.6	5.3	21316 CC
	100	144	92	158	2	0.35	1.9	2.9	22316 CC/W33
90	107	140	100	150	2	0.23	2.9	4.4	22218 CC/W33
	118	163	104	176	2.5	0.19	3.6	5.3	21318 CC
	112	160	104	176	2.5	0.35	1.9	2.9	22318 CC/W33
100	119	157	112	168	2	0.24	2.8	4.2	22220 CC/W33
	132	182	114	201	2.5	0.19	3.6	5.3	21320 CC
	125	179	114	201	2.5	0.35	1.9	2.9	22320 CC/W33
110	132	174	122	188	2	0.25	2.7	4	22222 CC/W33
	147	203	124	226	2.5	0.18	3.8	5.6	21322 CC
	140	199	124	226	2.5	0.35	1.9	2.9	22322 CC/W33
120	142	187	132	203	2	0.25	2.7	4	22224 CC/W33
	141	183	132	203	2	0.35	1.9	2.9	23224 CC/W33
	152	217	134	246	2.5	0.35	1.9	2.9	22324 CC/W33
130	153	201	144	216	2.5	0.26	2.6	3.9	22226 CC/W33
	151	197	144	216	2.5	0.33	2	3	23226 CC/W33
	164	234	148	262	3	0.35	1.9	2.9	22326 CC/W33
140	166	217	154	236	2.5	0.26	2.6	3.9	22228 CC/W33
	165	213	154	236	2.5	0.33	2	3	23228 CC/W33
	175	249	158	282	3	0.35	1.9	2.9	22328 CC/W33
150	178	235	164	256	2.5	0.26	2.6	3.9	22230 CC/W33
	175	229	164	256	2.5	0.35	1.9	2.9	23230 CC/W33
	188	268	168	302	3	0.35	1.9	2.9	22330 CC/W33
160	191	251	174	276	2.5	0.26	2.6	3.9	22232 CC/W33
	188	246	174	276	2.5	0.35	1.9	2.9	23232 CC/W33
	201	284	178	322	3	0.35	1.9	2.9	22332 CC/W33
170	203	268	188	292	3	0.27	2.5	3.7	22234 CC/W33
	200	262	188	292	3	0.35	1.9	2.9	23234 CC/W33
	213	302	188	342	3	0.33	2	3	22334 CC/W33

(*Contd.*)

(*Contd.*)

180	213	279	198	302	3	0.26	2.6	3.9	22236 CC/W33
	211	273	198	302	3	0.35	1.9	2.9	23236 CC/W33
	224	319	198	362	3	0.35	1.9	2.9	22336 CC/W33
190	225	296	208	322	3	0.26	2.6	3.9	22238 CC/W33
	222	289	208	322	3	0.35	1.9	2.9	23238 CC/W33
	236	335	212	378	4	0.35	1.9	2.9	22338 CC/W33
200	238	314	218	342	3	0.26	2.6	3.9	22240 CC/W33
	235	305	218	342	3	0.35	1.9	2.9	23240 CC/W33
	249	353	222	398	4	0.33	2	3	22340 CC/W33

Note: All dimensions are in mm.

15.7 TAPER ROLLER BEARINGS

Table 15.26 *Arrangement of bearing pairs*

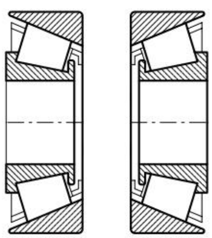
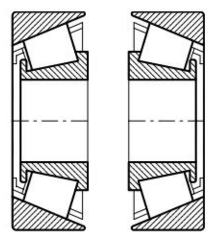
Back-to-back arrangement	**Face-to-face arrangement**
	

Table 15.27 *Equivalent dynamic load for taper roller bearings (For single-row taper roller bearings)*

When $\left(\dfrac{F_a}{F_r}\right) \le e$	$P = F_r$
When $\left(\dfrac{F_a}{F_r}\right) > e$	$P = 0.4F_r + YF_a$

Note: The values of factors e, Y, are given in Table 15.30.

Table 15.28 *Axial loading of taper roller bearings*

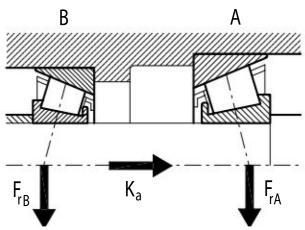
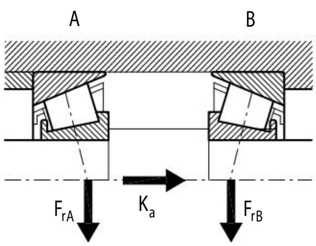
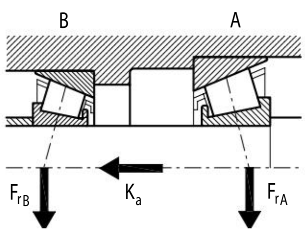
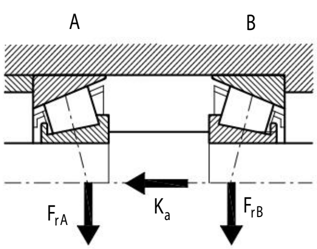
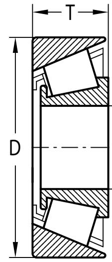
Arrangement	Load case	Axial loads
Back-to-back  **Face-to-face** 	1(a) $\frac{F_{rA}}{Y_A} \geq \frac{F_{rB}}{Y_B}$ $K_a \geq 0$	$F_{aA} = \frac{0.5F_{rA}}{Y_A}$ $F_{aB} = F_{aA} + K_a$
	1 (b) $\frac{F_{rA}}{Y_A} < \frac{F_{rB}}{Y_B}$ $K_a \geq 0.5\left(\frac{F_{rB}}{Y_B} - \frac{F_{rA}}{Y_A}\right)$	$F_{aA} = \frac{0.5F_{rA}}{Y_A}$ $F_{aB} = F_{aA} + K_a$
	1 (c) $\frac{F_{rA}}{Y_A} < \frac{F_{rB}}{Y_B}$ $K_a < 0.5\left(\frac{F_{rB}}{Y_B} - \frac{F_{rA}}{Y_A}\right)$	$F_{aA} = F_{aB} - K_a$ $F_{aB} = \frac{0.5F_{rB}}{Y_B}$
Back-to-back  **Face-to-face** 	2 (a) $\frac{F_{rA}}{Y_A} \leq \frac{F_{rB}}{Y_B}$ $K_a \geq 0$	$F_{aA} = F_{aB} + K_a$ $F_{aB} = \frac{0.5F_{rB}}{Y_B}$
	2 (b) $\frac{F_{rA}}{Y_A} > \frac{F_{rB}}{Y_B}$ $K_a \geq 0.5\left(\frac{F_{rA}}{Y_A} - \frac{F_{rB}}{Y_B}\right)$	$F_{aA} = F_{aB} + K_a$ $F_{aB} = \frac{0.5F_{rB}}{Y_B}$
	2 (c) $\frac{F_{rA}}{Y_A} > \frac{F_{rB}}{Y_B}$ $K_a < 0.5\left(\frac{F_{rA}}{Y_A} - \frac{F_{rB}}{Y_B}\right)$	$F_{aA} = \frac{0.5F_{rA}}{Y_A}$ $F_{aB} = F_{aA} - K_a$

Table 15.29 *Principal dimensions and static and dynamic load capacities of (SKF) taper roller bearings (Single-row)*



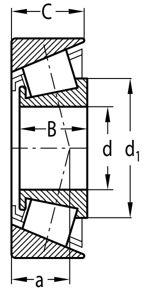
Principal dimensions (mm)			Basic load ratings (N)		Designation (SKF)
d	***D***	***T***	***C***	C_0	
20	47	15.25	26000	16600	30204
	52	16.25	31900	20000	30304
	52	22.25	41300	28000	32304
30	62	17.25	38000	25500	30206
	62	21.25	47300	33500	32206
	62	25	60500	45500	33206
	72	20.75	52800	34500	30306
	72	20.75	44600	29000	31306
	72	28.75	72100	52000	32306
40	80	19.75	58300	40000	30208
	80	24.75	70400	50000	32208
	80	32	96800	78000	33208
	90	25.25	80900	56000	30308
	90	25.25	69300	46500	31308
	90	35.25	110 000	83000	32308
50	90	21.25	70400	52000	30210
	90	24.25	76500	57000	32210
	90	32	108 000	90000	33210
	110	29.25	117 000	83000	30310
	110	29.25	99000	69500	31310
	110	42.25	161 000	127 000	32310
60	110	23.75	91300	65500	30212
	110	29.75	119 000	91500	32212
	110	38	157 000	134 000	33212
	130	33.5	161 000	116 000	30312
	130	33.5	134 000	96500	31312
	130	48.5	216 000	173 000	32312

(*Contd.*)

(*Contd.*)

70	125	26.25	119 000	88000	30214
	125	33.25	147 000	118 000	32214
	125	41	190 000	163 000	33214
	150	38	209 000	153 000	30314
	150	38	176 000	127 000	31314
	150	54	275 000	228 000	32314
80	140	28.5	140 000	104 000	30216
	140	35.25	176 000	137 000	32216
	140	46	233 000	208 000	33216
	170	42.5	255 000	190 000	30316
	170	42.5	212 000	153 000	31316
	170	61.5	358 000	300 000	32316
90	160	32.5	183 000	140 000	30218
	160	42.5	238 000	193 000	32218
	190	46.5	308 000	236 000	30318
	190	46.5	251 000	183 000	31318
	190	67.5	429 000	360 000	32318
100	180	37	233 000	183 000	30220
	180	49	297 000	250 000	32220
	180	63	402 000	375 000	33220
	215	51.5	380 000	290 000	30320
	215	77.5	539 000	465 000	32320
120	215	43.5	319 000	260 000	30224
	215	61.5	440 000	390 000	32224
	260	59.5	528 000	415 000	30324
	260	90.5	748 000	655 000	32324
140	250	45.75	396 000	325 000	30228
	250	71.75	605 000	560 000	32228
	300	67.75	693 000	560 000	30328
160	290	52	495 000	415 000	30232
	290	84	825 000	780 000	32232
	340	75	858 000	695 000	30332
180	250	45	330 000	380 000	32936
	320	57	550 000	465 000	30236
	320	91	935 000	915 000	32236
200	280	51	446 000	500 000	32940
	360	64	737 000	630 000	30240
	360	104	1140 000	1120 000	32240

Table 15.30 *Dimensions and factors e and Y for (SKF) taper roller bearings (Single-row)*



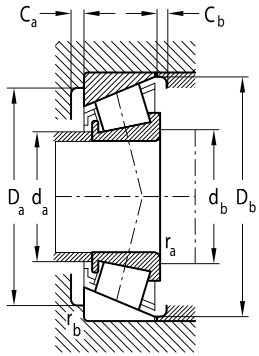
Dimensions (mm)				*a* (mm)	Factors		Designation (SKF)
d	d_1 (≈)	*B*	*C*		*e*	*Y*	
20	33.2	14	12	11	0.35	1.7	30204
	34.3	15	13	11	0.30	2	30304
	34.5	21	18	14	0.30	2	32304
30	44.6	16	14	14	0.37	1.6	30206
	45.2	20	17	15	0.37	1.6	32206
	45.8	25	19.5	16	0.35	1.7	33206
	48.4	19	16	15	0.31	1.9	30306
	52.7	19	14	22	0.83	0.72	31306
	48.7	27	23	18	0.31	1.9	32306
40	57.5	18	16	16	0.37	1.6	30208
	58.4	23	19	19	0.37	1.6	32208
	59.7	32	25	21	0.35	1.7	33208
	62.5	23	20	19	0.35	1.7	30308
	67.1	23	17	28	0.83	0.72	31308
	62.9	33	27	23	0.35	1.7	32308
50	67.9	20	17	19	0.43	1.4	30210
	68.5	23	19	21	0.43	1.4	32210
	70.7	32	24.5	23	0.40	1.5	33210
	77.2	27	23	23	0.35	1.7	30310
	81.5	27	19	34	0.83	0.72	31310
	77.7	40	33	27	0.35	1.7	32310
60	81.5	22	19	22	0.40	1.5	30212
	81.9	28	24	24	0.40	1.5	32212
	85.3	38	29	27	0.40	1.5	33212
	91.9	31	26	26	0.35	1.7	30312
	95.9	31	22	39	0.83	0.72	31312
	91.7	46	37	31	0.35	1.7	32312

(*Contd.*)

(*Contd.*)

70	93.9	24	21	25	0.43	1.4	30214
	95	31	27	28	0.43	1.4	32214
	97.2	41	32	30	0.40	1.5	33214
	105	35	30	29	0.35	1.7	30314
	110	35	25	45	0.83	0.72	31314
	106	51	42	36	0.35	1.7	32314
80	105	26	22	28	0.43	1.4	30216
	106	33	28	30	0.43	1.4	32216
	110	46	35	35	0.43	1.4	33216
	120	39	33	33	0.35	1.7	30316
	124	39	27	52	0.83	0.72	31316
	120	58	48	41	0.35	1.7	32316
90	118	30	26	31	0.43	1.4	30218
	121	40	34	36	0.43	1.4	32218
	132	43	36	36	0.35	1.7	30318
	138	43	30	57	0.83	0.72	31318
	133	64	53	44	0.35	1.7	32318
100	133	34	29	35	0.43	1.4	30220
	135	46	39	41	0.43	1.4	32220
	139	63	48	45	0.40	1.5	33220
	148	47	39	40	0.35	1.7	30320
	151	73	60	51	0.35	1.7	32320
120	161	40	34	43	0.43	1.4	30224
	163	58	50	51	0.43	1.4	32224
	178	55	46	47	0.35	1.7	30324
	181	86	69	60	0.35	1.7	32324
140	186	42	36	47	0.43	1.4	30228
	191	68	58	60	0.43	1.4	32228
	205	62	53	54	0.35	1.7	30328
160	214	48	40	54	0.43	1.4	30232
	221	80	67	70	0.43	1.4	32232
	233	68	58	61	0.35	1.7	30332
180	216	45	34	53	0.48	1.25	32936
	239	52	43	61	0.46	1.3	30236
	247	86	71	78	0.46	1.3	32236
200	239	51	39	53	0.40	1.5	32940
	268	58	48	68	0.43	1.4	30240
	274	98	82	83	0.40	1.5	32240

Table 15.31 *Abutment and fillet dimensions for taper roller bearings (Single-row)*



d	d_a (Max.)	d_b (Min.)	D_a (Min.)	D_a (Max.)	D_b (Min.)	C_a (Min.)	C_b (Min.)	r_a (Max.)	r_b (Max.)	Designation (SKF)
20	27	26	40	41	43	2	3	1	1	30204
	28	27	44	45	47	2	3	1	1	30304
	27	27	43	45	47	3	4	1	1	32304
30	38	36	53	56	57	2	3	1	1	30206
	37	36	52	56	58	3	4	1	1	32206
	36	36	53	56	59	5	5.5	1	1	33206
	41	37	62	65	66	3	4.5	1	1	30306
	40	37	55	65	68	3	6.5	1	1	31306
	39	37	59	65	66	3	5.5	1	1	32306
40	49	47	69	73	74	3	3.5	1	1	30208
	49	47	68	73	75	3	5.5	1	1	32208
	47	47	67	73	76	5	7	1	1	33208
	53	49	77	81	82	3	5	1.5	1.5	30308
	51	49	71	81	86	3	8	1.5	1.5	31308
	51	49	73	81	82	3	8	1.5	1.5	32308
50	58	57	79	83	85	3	4.5	1	1	30210
	58	57	78	83	85	3	5.5	1	1	32210
	57	57	77	83	87	5	7.5	1	1	33210
	65	60	95	100	102	4	6	2	2	30310
	62	60	87	100	104	4	10	2	2	31310
	63	60	90	100	102	5	9	2	2	32310
60	70	69	96	101	103	4	4.5	1.5	1.5	30212
	69	69	95	101	104	4	5.5	1.5	1.5	32212
	69	69	93	101	105	6	9	1.5	1.5	33212
	77	72	112	118	120	5	7.5	2	2	30312
	74	72	103	118	123	5	11.5	2	2	31312
	74	72	107	118	120	6	11.5	2	2	32312

(*Contd.*)

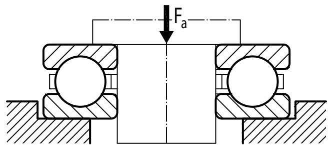
(*Contd.*)

70	82	79	110	116	118	4	5	1.5	1.5	30214
	80	79	108	116	119	4	6	1.5	1.5	32214
	79	79	107	116	120	6	9	1.5	1.5	33214
	90	82	130	138	140	5	8	2	2	30314
	85	82	118	138	141	5	13	2	2	31314
	86	82	125	138	140	6	12	2	2	32314
80	92	90	124	130	132	4	6	2	2	30216
	91	90	122	130	134	5	7	2	2	32216
	89	90	119	130	135	7	11	2	2	33216
	102	92	148	158	159	5	9.5	2	2	30316
	97	92	134	158	159	6	15.5	2	2	31316
	98	92	142	158	159	7	13.5	2	2	32316
90	104	100	140	150	150	5	6.5	2	2	30218
	102	100	138	150	152	5	8.5	2	2	32218
	113	104	165	176	176	6	10.5	2.5	2.5	30318
	109	104	151	176	179	5	16.5	2.5	2.5	31318
	109	104	157	176	177	7	14.5	2.5	2.5	32318
100	116	112	157	168	168	5	8	2	2	30220
	115	112	154	168	171	5	10	2	2	32220
	112	112	151	168	172	10	15	2	2	33220
	127	114	184	201	197	6	12.5	2.5	2.5	30320
	123	114	177	201	200	8	17.5	2.5	2.5	32320
120	141	132	187	203	201	6	9.5	2	2	30224
	137	132	181	203	204	7	11.5	2	2	32224
	153	134	221	246	237	7	13.5	2.5	2.5	30324
	148	134	213	246	239	10	21.5	2.5	2.5	32324
140	164	154	219	236	234	7	9.5	2.5	2.5	30228
	159	154	210	236	238	8	13.5	2.5	2.5	32228
	176	158	255	282	273	8	14.5	3	3	30328
160	189	174	252	276	269	8	12	2.5	2.5	30232
	183	174	242	276	274	10	17	2.5	2.5	32232
	201	178	290	322	310	9	17	3	3	30332
180	194	190	225	240	241	8	11	2	2	32936
	211	198	278	302	297	9	14	3	3	30236
	204	198	267	302	303	10	20	3	3	32236
200	217	212	257	268	271	9	12	2.5	2	32940
	237	218	315	342	336	9	16	3	3	30240
	231	218	302	342	340	11	22	3	3	32240

Note: All dimensions are in mm.

15.8 THRUST BALL BEARINGS

Table 15.32 *Equivalent dynamic load and minimum axial load for thrust ball bearings*



Equivalent dynamic load

For thrust ball bearings that are dynamically loaded in the axial direction, the equivalent dynamic load is given by,

$$P = F_a$$

Minimum axial load

In thrust ball bearings operating at high speeds, sliding can occur between the balls and raceways due to centrifugal forces and gyroscopic couple acting on balls. To prevent the damage due to these factors, the bearings must always be subjected to certain minimum axial load. The minimum axial load is given by,

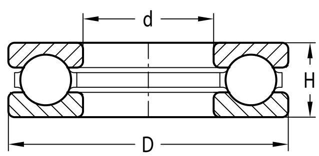
$$F_{am} = A\left(\frac{n}{1000}\right)^2$$

where

F_{am} = minimum axial load (N)
A = minimum load factor (Table 15.33)
n = rotational speed (rpm)

Note: The weight of the components supported by the bearing often exceeds minimum axial load. If not, then the bearings are preloaded by means of springs.

Table 15.33 *Principal dimensions, static and dynamic load capacities and minimum load factor for (SKF) thrust ball bearings (Single direction)*



Principal dimensions (mm)			Basic load ratings (N)		Minimum Load Factor	Designation (SKF)
d	***D***	***H***	***C***	C_0	***A***	
10	24	9	8710	8800	0.63	51100
	26	11	10800	10600	0.92	51200
20	35	10	12700	16600	2.2	51104
	40	14	19900	25000	5.1	51204

(*Contd.*)

(*Contd.*)

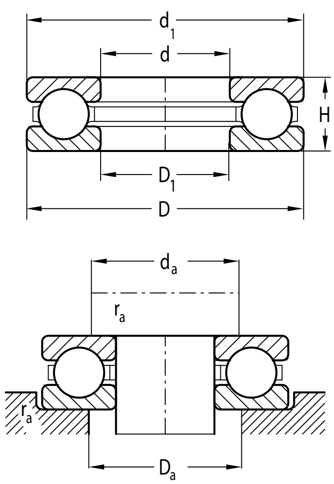
30	47	11	16800	26500	5.7	51106
	52	16	25500	37500	11	51206
	60	21	40300	57000	27	51306
	70	28	67600	90000	66	51406
40	60	13	23400	40000	13	51108
	68	19	39700	64000	33	51208
	78	26	65000	98000	78	51308
	90	36	104 000	146 000	170	51408
50	70	14	25500	50000	20	51110
	78	22	41600	73500	44	51210
	95	31	87100	137 000	150	51310
	110	43	138 000	204 000	340	51410
60	85	17	35800	71000	41	51112
	95	26	65000	118 000	110	51212
	110	35	117 000	196 000	310	51312
	130	51	186 000	285 000	660	51412
70	95	18	37700	81500	54	51114
	105	27	65000	127 000	130	51214
	125	40	133 000	232 000	440	51314
	150	60	234 000	400 000	1300	51414
80	105	19	39700	93000	71	51116
	115	28	68900	143 000	170	51216
	140	44	159 000	290 000	590	51316
	170	68	270 000	490 000	2000	51416
90	120	22	50700	120 000	120	51118
	135	35	106 000	224 000	410	51218
	155	50	199 000	390 000	1200	51318
	190	77	307 000	600 000	2900	51418
100	135	25	74100	173 000	240	51120
	150	38	133 000	280 000	640	51220
	170	55	238 000	475 000	1800	51320
	210	85	371 000	765 000	4800	51420

(*Contd.*)

(*Contd.*)

120	155	25	88400	245 000	490	51124
	170	39	153 000	375 000	1100	51224
	210	70	312 000	695 000	3900	51324
	250	102	423 000	965 000	7600	51424
140	180	31	111 000	320 000	840	51128
	200	46	208 000	520 000	2200	51228
	240	80	351 000	830 000	5600	51328
160	200	31	112 000	340 000	940	51132
	225	51	225 000	570 000	2700	51232
	270	87	423 000	1080 000	9500	51332
180	225	34	135 000	425 000	1500	51136
	250	56	265 000	695 000	3900	51236
	300	95	462 000	1270 000	13000	51336
200	250	37	168 000	520 000	2200	51140
	280	62	312 000	880 000	6300	51240
	340	110	592 000	1760 000	25000	51340

Table 15.34 *Abutment and fillet dimensions for thrust ball bearings (Single direction)*



d	d_1	D_1	d_a (Min.)	D_a (Max.)	r_a (Max.)	Designation (SKF)
10	24	11	19	15	0.3	51100
	26	12	20	16	0.6	51200

(*Contd.*)

(*Contd.*)

20	35	21	29	26	0.3	51104
	40	22	32	28	0.6	51204
30	47	32	40	37	0.6	51106
	52	32	43	39	0.6	51206
	60	32	48	42	1	51306
	70	32	54	46	1	51406
40	60	42	52	48	0.6	51108
	68	42	57	51	1	51208
	78	42	63	55	1	51308
	90	42	70	60	1	51408
50	70	52	62	58	0.6	51110
	78	52	67	61	1	51210
	95	52	77	68	1	51310
	110	52	86	74	1.5	51410
60	85	62	75	70	1	51112
	95	62	81	74	1	51212
	110	62	90	80	1	51312
	130	62	102	88	1.5	51412
70	95	72	85	80	1	51114
	105	72	91	84	1	51214
	125	72	103	92	1	51314
	150	73	118	102	2	51414
80	105	82	95	90	1	51116
	115	82	101	94	1	51216
	140	82	116	104	1.5	51316
	170	83	133	117	2	51416
90	120	92	108	102	1	51118
	135	93	117	108	1	51218
	155	93	129	116	1.5	51318
	187	93	149	131	2	51418
100	135	102	121	114	1	51120
	150	103	130	120	1	51220

(*Contd.*)

(*Contd.*)

	170	103	142	128	1.5	51320
	205	103	165	145	2.5	51420
120	155	122	141	134	1	51124
	170	123	150	140	1	51224
	205	123	173	157	2	51324
	245	123	197	173	3	51424
140	178	142	164	156	1	51128
	197	143	176	164	1.5	51228
	235	144	199	181	2	51328
160	198	162	184	176	1	51132
	222	163	199	186	1.5	51232
	265	164	225	205	2.5	51332
180	222	183	207	198	1	51136
	247	183	222	208	1.5	51236
	295	184	251	229	2.5	51336
200	247	203	230	220	1	51140
	277	204	248	232	2	51240
	335	205	283	257	3	51340

Note: All dimensions are in mm.

15.9 TOLERANCES FOR SHAFTS AND HOUSINGS

Table 15.35 *Tolerances for solid steel shafts (Radial bearings with cylindrical bore)*

Conditions	Examples	Shaft diameter (mm)			Tolerance
		Ball bearings	Cylindrical and Taper roller bearings	Spherical roller bearings	
Stationary inner ring load					
Easy axial displacement of inner ring on shaft desirable	Wheels on non-rotating axles	All diameters	All diameters	All diameters	*g*6
Easy axial displacement of inner ring on shaft unnecessary	Tension pulleys, rope sheaves	All diameters	All diameters	All diameters	*h*6

(*Contd.*)

(*Contd.*)

Rotating inner ring load or direction of loading indeterminate					
Light and variable loads ($P \leq 0.06C$)	Conveyors, lightly loaded gearbox bearings	18 to 100	≤ 40	—	*j*6
		100 to 140	40 to 100	—	*k*6
Normal loads and heavy loads ($P > 0.06C$)	General bearing applications, electric motors, turbines, pumps, internal combustion engines, gear boxes, wood working machinery	≤ 18	—	—	*j*5
		18 to 100	≤ 40	≤ 40	*k*5
		100 to 140	40 to 100	40 to 65	*m*5
		140 to 200	100 to 140	65 to 100	*m*6
		200 to 280	140 to 200	100 to 140	*n*6
		—	200 to 400	140 to 280	*p*6
		—	—	280 to 500	*r*6
		—	—	>500	*r*7
Very heavy loads and shock loads with difficult working conditions ($P > 0.12C$)	Axle boxes for heavy railway vehicles, traction motors, rolling mills	—	50 to 140	50 to 100	*n*6
		—	140 to 200	100 to 140	*p*6
		—	>200	>140	*r*6
High demands on running accuracy with light loads ($P \leq 0.06C$)	Machine tools	≤ 18	—	—	*h*5
		18 to 100	≤ 40	—	*j*5
		100 to 200	40 to 140	—	*k*5
		—	140 to 200	—	*m*5
Axial loads only					
	Bearing applications of all kinds	≤ 250	≤ 250	≤ 250	*j*6
		>250	>250	>250	*js*6

Table 15.36 *Tolerances for solid steel shafts (Thrust bearings)*

Conditions	Shaft diameter (mm)	Tolerance
Axial loads only		
Thrust ball bearings	All diameters	*h*6
Cylindrical roller thrust bearings	All diameters	*h*6
Radial and axial loads on spherical roller thrust bearings		
Stationary load on shaft washer	≤250	*j*6
	>250	*js*6
Rotating load on shaft washer or direction of loading indeterminate	≤200	*k*6
	200 to 400	*m*6
	>400	*n*6

Table 15.37 *ISO limits for shafts (g6 to j6)*

Shaft diameter		Limit deviations for shaft									
		*g*6		*h*5		*h*6		*j*5		*j*6	
over	incl.	high	low	high	low	high	low	high	low	high	low
(mm)	(mm)	μm	μm	μm	μm	μm	μm	μm	μm	μm	μm
3	6	–4	–12	0	–5	0	–8	+3	–2	+6	–2
6	10	–5	–14	0	–6	0	–9	+4	–2	+7	–2
10	18	–6	–17	0	–8	0	–11	+5	–3	+8	–3
18	30	–7	–20	0	–9	0	–13	+5	–4	+9	–4
30	50	–9	–25	0	–11	0	–16	+6	–5	+11	–5
50	80	–10	–29	0	–13	0	–19	+6	–7	+12	–7
80	120	–12	–34	0	–15	0	–22	+6	–9	+13	–9
120	180	–14	–39	0	–18	0	–25	+7	–11	+14	–11
180	250	–15	–44	0	–20	0	–29	+7	–13	+16	–13

Note: 1 μm = 1 micron = 0.001 mm.

Table 15.38 *ISO limits for shafts (js6 to m6)*

Shaft diameter		Limit deviations for shaft									
		*js*6		*k*5		*k*6		*m*5		*m*6	
over	incl.	high	low	high	low	high	low	high	low	high	low
(mm)	(mm)	μm	μm	μm	μm	μm	μm	μm	μm	μm	μm
3	6	+4	–4	+6	+1	+9	+1	+9	+4	+12	+4
6	10	+4.5	–4.5	+7	+1	+10	+1	+12	+6	+15	+6
10	18	+5.5	–5.5	+9	+1	+12	+1	+15	+7	+18	+7
18	30	+6.5	–6.5	+11	+2	+15	+2	+17	+8	+21	+8
30	50	+8	–8	+13	+2	+18	+2	+20	+9	+25	+9
50	80	+9.5	–9.5	+15	+2	+21	+2	+24	+11	+30	+11
80	120	+11	–11	+18	+3	+25	+3	+28	+13	+35	+13
120	180	+12.5	–12.5	+21	+3	+28	+3	+33	+15	+40	+15
180	250	+14.5	–14.5	+24	+4	+33	+4	+37	+17	+46	+17

Table 15.39 *ISO limits for shafts (n6 and p6)*

Shaft diameter		Limit deviations for shaft			
		*n*6		*p*6	
over	incl.	high	low	high	low
(mm)	(mm)	μm	μm	μm	μm
3	6	+16	+8	+20	+12
6	10	+19	+10	+24	+15
10	18	+23	+12	+29	+18
18	30	+28	+15	+35	+22
30	50	+33	+17	+42	+26
50	80	+39	+20	+51	+32
80	120	+45	+23	+59	+37
120	180	+52	+27	+68	+43
180	250	+60	+31	+79	+50

Table 15.40 *ISO limits for shafts (r6 and r7)*

Shaft diameter		Limit deviations for shaft			
		*r*6		*r*7	
over	incl.	high	low	high	low
(mm)	(mm)	μm	μm	μm	μm
120	140	+88	+63	+103	+63
140	160	+90	+65	+105	+65
160	180	+93	+68	+108	+68
180	200	+106	+77	+123	+77
200	225	+109	+80	+126	+80
225	250	+113	+84	+130	+84

Table 15.41 *Tolerances for cast iron and steel housings (Radial bearings-Solid housings)*

Conditions	Examples	Tolerance	Displacement of outer ring
Rotating outer ring load			
Heavy loads on bearings in thin-walled housings, Heavy shock loads ($P > 0.12C$)	Roller bearing wheel hubs, big-end bearings	P7	Cannot be displaced
Normal loads and heavy loads ($P > 0.06C$)	Ball bearing wheel hubs, big-end bearings, crane travelling wheels	N7	Cannot be displaced
Light and variable loads ($P \leq 0.06C$)	Conveyor rollers, rope sheaves, belt tension pulleys	M7	Cannot be displaced

(*Contd.*)

(*Contd.*)

Direction of load indeterminate			
Heavy shock loads	Election traction motors	M7	Cannot be displaced
Normal loads and heavy loads (P > 0.06C), Axial displacement of outer ring unnecessary	Electric motors, pumps, crankshaft bearings	K7	Cannot be displaced as a rule
Accurate or silent running			
	Roller bearings for machine tool spindles	K6	Cannot be displaced as a rule
	Ball bearings for grinding spindles, small electric motors	J6	Cannot be displaced
	Small electric motors	H6	Can easily be displaced

Table 15.42 *Tolerances for cast iron and steel housings (Radial bearings-Split or solid housings)*

Conditions	**Examples**	**Tolerance**	**Displacement of outer ring**
Direction of load indeterminate			
Light loads and normal loads ($P \leq 0.12$C), Axial displacement of outer ring desirable	Medium-sized electrical machines, pumps, crankshaft bearings	J7	Can be normally displaced
Stationary outer ring loads			
Loads of all kinds	Railway axle boxes	H7	Can easily be displaced
Light loads and normal loads ($P \leq 0.12$C) with simple working conditions	General engineering	H8	Can easily be displaced
Heat conduction through shaft	Drying cylinders, large electrical machines with spherical roller bearings	G7	Can easily be displaced

Table 15.43 *Tolerances for cast iron and steel housings (Thrust bearings)*

Conditions	**Tolerance**
Axial loads only	
Thrust ball bearings	H8
Cylindrical roller thrust bearings	H7
Cylindrical roller and cage thrust assemblies	H10
Radial and axial loads on spherical roller thrust bearings	
Stationary load on housing washer	H7
Rotating load on housing washer	M7

Table 15.44 *ISO limits for housings (G7 to H10)*

Housing bore diameter		Limit deviations for housing									
		G7		H6		H7		H8		H10	
Over	Incl.	Low	High	Low	High	Low	High	Low	High	Low	High
(mm)	(mm)	μm	μm	μm	μm	μm	μm	μm	μm	μm	μm
10	18	+6	+24	0	+11	0	+18	0	+27	0	+70
18	30	+7	+28	0	+13	0	+21	0	+33	0	+84
30	50	+9	+34	0	+16	0	+25	0	+39	0	+100
50	80	+10	+40	0	+19	0	+30	0	+46	0	+120
80	120	+12	+47	0	+22	0	+35	0	+54	0	+140
120	150	+14	+54	0	+25	0	+40	0	+63	0	+160
150	180	+14	+54	0	+25	0	+40	0	+63	0	+160
180	250	+15	+61	0	+29	0	+46	0	+72	0	+185
250	315	+17	+69	0	+32	0	+52	0	+81	0	+210
315	400	+18	+75	0	+36	0	+57	0	+89	0	+230
400	500	+20	+83	0	+40	0	+63	0	+97	0	+250

Note: 1 μm = 1 micron = 0.001 mm.

Table 15.45 *ISO limits for housings (J6 to M7)*

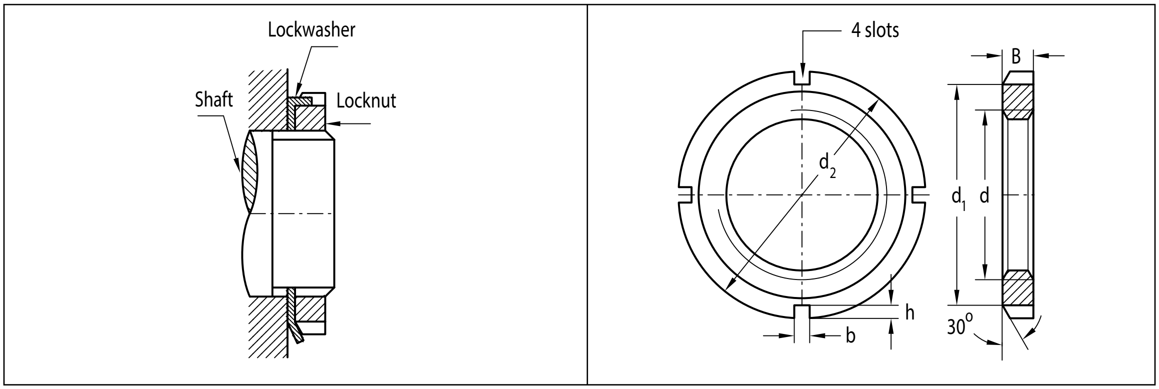
Housing bore diameter		Limit deviations for housing									
		J6		J7		K6		K7		M7	
Over	Incl.	Low	High	Low	High	Low	High	Low	High	Low	High
(mm)	(mm)	μm	μm	μm	μm	μm	μm	μm	μm	μm	μm
10	18	–5	+6	–8	+10	–9	+2	–12	+6	–18	0
18	30	–5	+8	–9	+12	–11	+2	–15	+6	–21	0
30	50	–6	+10	–11	+14	–13	+3	–18	+7	–25	0
50	80	–6	+13	–12	+18	–15	+4	–21	+9	–39	0
80	120	–6	+16	–13	+22	–18	+4	–25	+10	–35	0
120	150	–7	+18	–14	+26	–21	+4	–28	+12	–40	0
150	180	–7	+18	–14	+26	–21	+4	–28	+12	–40	0
180	250	–7	+22	–16	+30	–24	+5	–33	+13	–46	0
250	315	–7	+25	–16	+36	–27	+5	–36	+16	–52	0
315	400	–7	+29	–18	+39	–29	+7	–40	+17	–57	0
400	500	–7	+33	–20	+43	–32	+8	–45	+18	–63	0

Table 15.46 *ISO limits for housings (N7 and P7)*

Housing bore diameter		Limit deviations for housing			
		N7		P7	
Over	Incl.	Low	High	Low	High
(mm)	(mm)	μm	μm	μm	μm
10	18	–23	–5	–29	–11
18	30	–28	–7	–35	–14
30	50	–33	–8	–42	–17
50	80	–39	–9	–51	–21
80	120	–45	–10	–59	–24
120	150	–52	–12	–68	–28
150	180	–52	–12	–68	–28
180	250	–60	–14	–79	–33
250	315	–66	–14	–88	–36
315	400	–73	–16	–98	–41
400	500	–80	–17	–108	–45

15.10 LOCKNUTS AND LOCKWASHERS

Table 15.47 *Dimensions of locknuts*



Nominal size *d*	Screw threads (6H)	d_2 (h11)	d_1	*B* (h14)	b	h
10	M10 × 0.75	18	13.5	4	3	2
20	M20 × 1	32	26	6	4	2
30	M30 × 1.5	45	38	7	5	2
40	M40 × 1.5	58	50	9	6	2.5
50	M50 × 1.5	70	61	11	6	2.5

(*Contd.*)

(*Contd.*)

60	M60 × 2	80	73	11	7	3
70	M70 × 2	92	85	12	8	3.5
80	M80 × 2	105	95	15	8	3.5
90	M90 × 2	120	108	16	10	4
100	M100 × 2	130	120	18	10	4
110	M110 × 2	145	133	19	12	5
120	M120 × 2	155	138	20	12	5
130	M130 × 2	165	149	21	12	5
140	M140 × 2	180	160	22	14	6
150	M150 × 2	195	171	24	14	6
160	M160 × 3	210	182	25	16	7
170	M170 × 3	220	193	26	16	7
180	M180 × 3	230	203	27	18	8
190	M190 × 3	240	214	28	18	8
200	M200 × 3	250	226	29	18	8

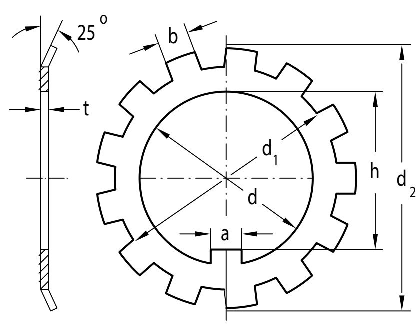
(SKF catalogue)

Note: (i) All dimensions in are mm.
(ii) Material – Carbon steel (Black bars) 20C8 or 30C8 or Spheroidal or Nodular graphite cast iron of grade SG 500/7

Application — Locknuts provide a simple means of axially locating rolling bearings and other machine components on shafts and also facilitate mounting and dismounting small bearings on adapter sleeves.

Designation – Locknut of size M10 and 0.75 mm pitch is designated as,
Locknut M10 × 0.75

Table 15.48 *Dimensions of lockwashers*



Nominal size *d* (C11)	d_1	d_2 (≈)	*a*	*h* (C11)	*b*	*t* (≈)	N
10	13.5	21	3	8.5	3	1	9
20	26	36	4	18.5	4	1	11
30	38	49	5	27.5	5	1.25	13
40	50	62	6	37.5	6	1.25	13

(*Contd.*)

(*Contd.*)

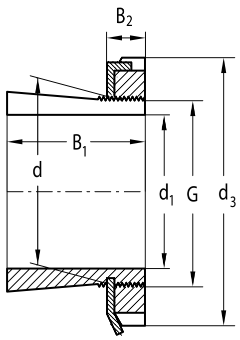
50	61	74	6	47.5	6	1.25	13
60	73	86	8	57.5	7	1.5	17
70	85	98	8	66.5	8	1.5	17
80	95	112	10	76.5	8	1.8	17
90	108	126	10	86.5	10	1.8	17
100	120	142	12	96.5	10	1.8	17
110	133	154	12	105.5	12	1.8	17
120	138	164	14	115	12	2	17
130	149	175	14	125	12	2	17
140	160	192	16	135	14	2	17
150	171	205	16	145	14	2	17
160	182	217	18	154	16	2.5	19
170	193	232	18	164	16	2.5	19
180	203	242	20	174	18	2.5	19
190	214	252	20	184	18	2.5	19
200	226	262	20	194	18	2.5	19

(SKF catalogue)

Note: (i) All dimensions are in mm.
(ii) N = minimum number of outer tabs (since the nut has four slots, N must be an odd number)
(iii) Material – Mild steel 10C4
Designation – Lockwasher of size 100 mm is designated as, Lockwasher 100.

15.11 ADAPTER SLEEVES

Table 15.49 *Dimensions of adapter sleeves*



d_1	d	d_3	B_1	B_2	G	Designation (SKF)
25	30	45	27	8	M30 × 1.5	H 206
	30	45	31	8	M30 × 1.5	H 306
	30	45	38	8	M30 × 1.5	H 2306
35	40	58	31	10	M40 × 1.5	H 208

(*Contd.*)

(*Contd.*)

	40	58	36	10	M40 × 1.5	H 308
	40	58	46	10	M40 × 1.5	H 2308
45	50	70	35	12	M50 × 1.5	H 210
	50	70	42	12	M50 × 1.5	H 310
	50	70	55	12	M50 × 1.5	H 2310
55	60	80	38	13	M60 × 2	H 212
	60	80	47	13	M60 × 2	H 312
	60	80	62	13	M60 × 2	H 2312
60	70	92	52	14	M70 × 2	H 314
	70	92	68	14	M70 × 2	H 2314
70	80	105	46	17	M80 × 2	H 216
	80	105	59	17	M80 × 2	H 316
	80	105	78	17	M80 × 2	H 2316
80	90	120	52	18	M90 × 2	H 218
	90	120	65	18	M90 × 2	H 318
	90	120	86	18	M90 × 2	H 2318
90	100	130	58	20	M100 × 2	H 220
	100	130	71	20	M100 × 2	H 320
	100	130	97	20	M100 × 2	H 2320
100	110	145	63	21	M110 × 2	H 222
	110	145	77	21	M110 × 2	H 322
	110	145	105	21	M110 × 2	H 2322
110	120	145	72	22	M120 × 2	H 3024
	120	155	88	22	M120 × 2	H 3124
	120	155	112	22	M120 × 2	H 2324
115	130	155	80	23	M130 × 2	H 3026
	130	165	92	23	M130 × 2	H 3126
	130	165	121	23	M130 × 2	H 2326
125	140	165	82	24	M140 × 2	H 3028
	140	180	97	24	M140 × 2	H 3128
	140	180	131	24	M140 × 2	H 2328
135	150	180	87	26	M150 × 2	H 3030
	150	195	111	26	M150 × 2	H 3130
	150	195	139	26	M150 × 2	H 2330
140	160	190	93	27.5	M160 × 3	H 3032
	160	210	119	28	M160 × 3	H 3132
	160	210	147	28	M160 × 3	H 2332
150	170	200	101	28.5	M170 × 3	H 3034
	170	220	122	29	M170 × 3	H 3134
	170	220	154	29	M170 × 3	H 2334

(*Contd.*)

(*Contd.*)

160	180	210	109	29.5	M180 × 3	H 3036
	180	230	131	30	M180 × 3	H 3136
	180	230	161	30	M180 × 3	H 2336
170	190	220	112	30.5	M190 × 3	H 3038
	190	240	141	31	M190 × 3	H 3138
	190	240	169	31	M190 × 3	H 2338
180	200	240	120	31.5	M200 × 3	H 3040
	200	250	150	32	M200 × 3	H 3140
	200	250	176	32	M200 × 3	H 2340

(SKF catalogue)

Note: All dimensions are in mm.
Application – Adapter sleeves are primarily used to locate rolling bearings with tapered bore (taper 1:12) on cylindrical seatings.

15.12 SEALING WASHERS

Table 15.50 *Types of sealing washers*

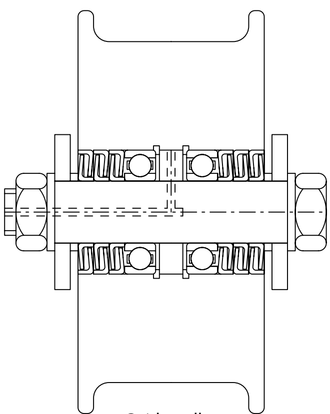
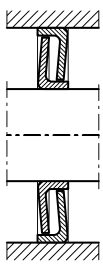
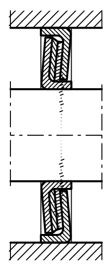
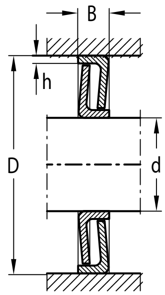
Assembly of sealing washers	**Standard type of sealing washer**	**F- type of sealing washer**
		
Sealing washers provide inexpensive but efficient means of preventing leakage. They are made of pressed steel, which is phosphated and coated with an anti-corrosive preservative. There are two types of sealing washers – standard type and F type with a flocked washer of plastic. The standard type washers can be combined to form single or multiple labyrinth seals. In F-type washers, velvet-like pile of short nylon fibres retain grease applied to the gap between shaft and housing washers. F type washers are particularly efficient in sealing against dry contaminants such as soil, sand or cement dust. The tolerance on width of washers is approximately ±0.1 mm.		

Table 15.51 *Dimensions of sealing washers*

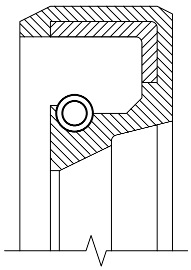
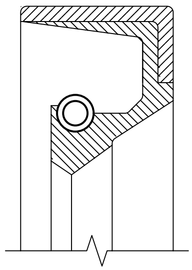
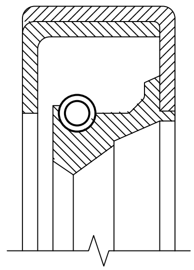
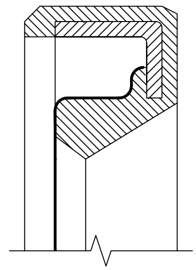
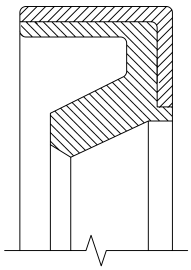
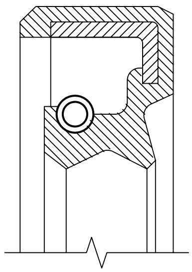
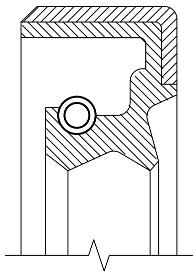
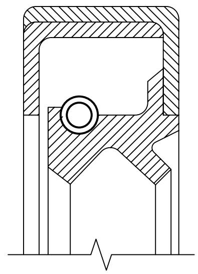


d	*D*	*B*	*h*	**Designation (SKF)**
10	26	4	1	Z 000
	30	4	1	Z 200
20	42	5	1.25	Z 004
	47	5	1.25	Z 204
	52	5	1.25	Z 304
30	55	5	1.25	Z 006
	62	6	1.5	Z 206
	72	6	1.5	Z 306
40	68	6	1.5	Z 008
	80	6	1.5	Z 208
	90	6	1.5	Z 308
50	80	6	1.5	Z 010
	90	6	1.5	Z 210
	110	6	1.5	Z 310
60	95	6	1.5	Z 012
	110	7	1.5	Z 212
70	110	7	1.5	Z 014
	125	7	1.5	Z 214
80	140	7	1.5	Z 216

Note: All dimensions are in mm.

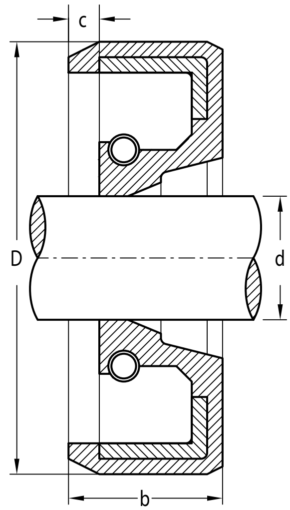
15.13 ROTARY SHAFT OILSEAL UNITS

Table 15.52 *Types of rotary shaft radial lip oilseals*

Type A	**Type B**	**Type C**	**Type D**
Spring-loaded rubber cased seal	Spring-loaded metal cased seal	Spring-loaded built up seal	Non spring-loaded rubber cased seal
			
Type E	**Type AS**	**Type BS**	**Type CS**
Non spring-loaded rubber cased seal	Spring-loaded rubber cased seal with secondary lip	Spring-loaded metal cased seal with secondary lip	Spring-loaded built up seal with secondary lip
			

Designation – The oilseal is designated by type, inner diameter, outer diameter and width. For example, an oilseal of type A of inner diameter 20 mm, outer diameter 40 mm and width 7 mm is designated by A $20 \times 40 \times 7$.

Table 15.53 *Dimensions of rotary shaft radial lip oilseal units*



d	*D*	*b* ± 2		*c* (Min.)
		Type A, B, D, E, AS and BS	Type C and CS	
10	19	7	—	0.3
	22			
	24			
	26			
20	30	7	—	0.3
	32			
	35			
	40			
	47			
30	40	7	—	0.4
	42			
	47			
	52		9	
	62			
40	52	7	—	0.4
	55			
	62		9	
	72			

(*Contd.*)

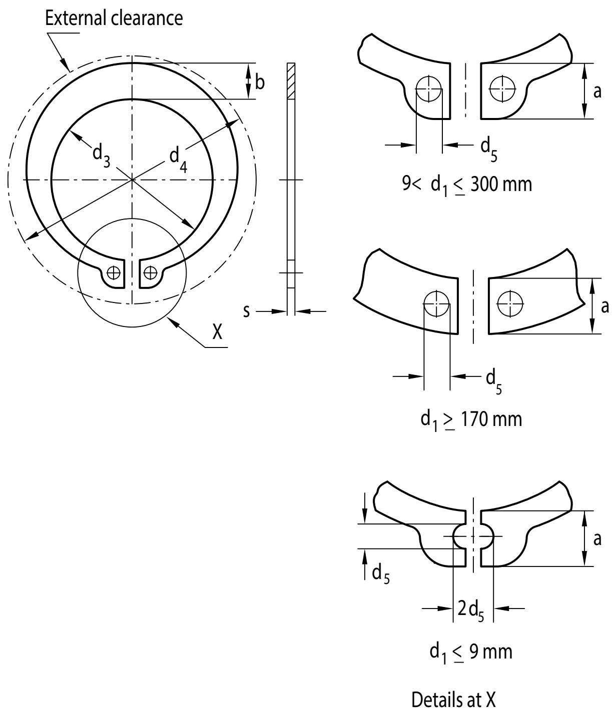
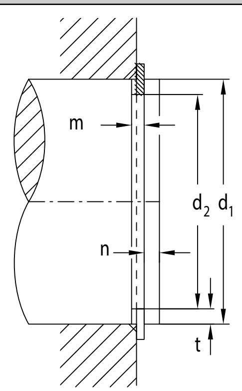
(*Contd.*)

50	65	8	—	0.4
	68			
	72		10	
	80			
60	75	8	—	0.4
	80			
	85		10	
	90			
70	90	10	12	0.5
	100			
80	100	10	12	0.5
	110			
90	110	12	15	0.8
	120			
100	120	12	15	0.8
	125			
	130			
110	130	12	15	0.8
	140			
120	150	12	15	0.8
	160			
130	160	12	15	0.8
	170			
140	170	15	15	1
150	180	15	15	1
160	190	15	15	1
170	200	15	15	1
180	210	15	15	1
190	220	15	15	1
200	230	15	15	1

Note: (i) All dimensions are in mm.
D = nominal bore diameter of housing or outside diameter of seal.
d = shaft diameter.

15.14 CIRCLIPS

Table 15.54 *Notations of circlips for shaft and groove dimensions*

Notations of circlips for shaft

Groove dimensions for circlip assembly


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Notations
d_1 = shaft diameter
d_2 = groove diameter
d_3 = internal diameter of circlip not under tension
d_4 = maximum symmetrical diameter of bore during fitting
d_5 = diameter of lug holes
a = radial width of lug
b = beam (radial width of circlip opposite the aperture)
m = groove width
n = edge margin
s = thickness of circlip
t = groove depth with nominal sizes of d_1 and d_2

Table 15.55 *Dimensions of circlips for shaft and groove dimensions (Normal type)*

Shaft dia d_1	Circlip					Groove				
	s	d_3	a Max.	$b \approx$	d_5 Min.	d_2 size	d_2 Tolerance	m H13	t	n Min.
10	1	9.3	3.3	1.8	1.5	9.6	h10	1.1	0.2	0.6
20	1.2	18.5	4	2.6	2	19	h11	1.3	0.5	1.5
30	1.5	27.9	5	3.5	2	28.6	h12	1.6	0.7	2.1
40	1.75	36.5	6	4.4	2.5	37.5	h12	1.85	1.25	3.8
50	2	45.8	6.9	5.1	2.5	47	h12	2.15	1.5	4.5
60	2	55.8	7.4	5.8	2.5	57	h12	2.15	1.5	4.5
70	2.5	65.5	8.1	6.6	3	67	h12	2.65	1.5	4.5
80	2.5	74.5	8.6	7.4	3	76.5	h12	2.65	1.75	5.3
90	3	84.5	8.8	8.2	3.5	86.5	h12	3.15	1.75	5.3
100	3	94.5	9.6	9	3.5	96.5	h12	3.15	1.75	5.3
110	4	103	10.1	9.6	3.5	106	h13	4.15	2	6
120	4	113	11	10.2	3.5	116	h13	4.15	2	6
130	4	123	11.6	10.7	4	126	h13	4.15	2	6
140	4	133	12	11.2	4	136	h13	4.15	2	6
150	4	142	13	11.8	4	145	h13	4.15	2.5	7.5
160	4	151	13.3	12.2	4	155	h13	4.15	2.5	7.5
170	4	160.5	13.5	12.9	4	165	h13	4.15	2.5	7.5
180	4	170.5	14.2	13.5	4	175	h13	4.15	2.5	7.5
190	4	180.5	14.2	14	4	185	h13	4.15	2.5	7.5
200	4	190.5	14.2	14	4	195	h13	4.15	2.5	7.5

(DIN: 471)

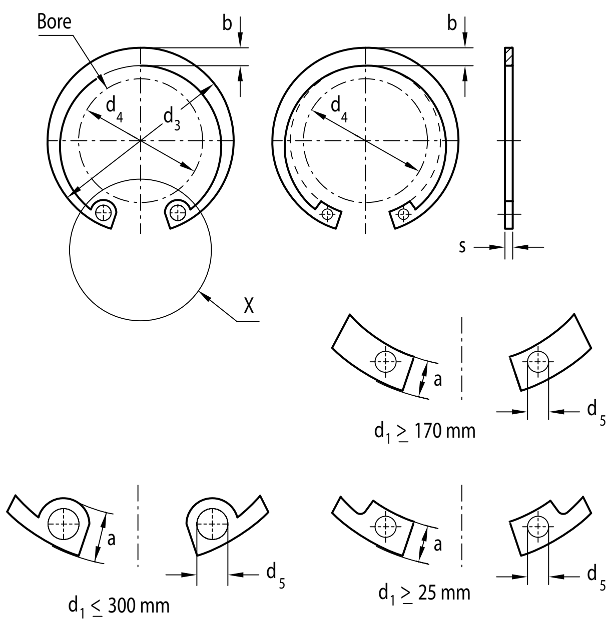
Note: (i) All dimensions are in mm.
(ii) The circlips are made of spring steels 70C6 or 75C6.
Designation – A circlip for shaft diameter (nominal size) d_1 = 80 mm and thickness 2.5 mm, normal type (N) is designated as, Circlip 80 × 2.5 N.

Table 15.56 *Hardness of circlips*

Nominal diameter of circlip (mm)		Hardness	
Over	**Up to**	**Vickers hardness**	**Corresponding Rockwell C hardness**
—	48	470–580	47–54
48	200	435–530	44–51
200	300	390–470	40–47

Table 15.57 *Notations of circlips for bores and groove dimensions*

Notations of circlips for bores



Details at X

(*Contd.*)

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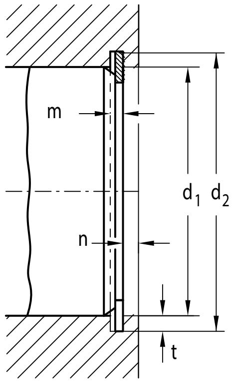
Groove dimensions for circlip assembly

Notations: d_1 = bore diameter d_2 = groove diameter d_3 = external diameter of circlip not under tension d_4 = maximum symmetrical diameter of bore during fitting d_5 = diameter of lug holes a = radial width of lug b = beam (radial width of circlip opposite the aperture) m = groove width n = edge margin s = thickness of circlip t = groove depth with nominal sizes of d_1 and d_2

Table 15.58 *Dimensions of circlips for bores and groove dimensions (Normal type)*

Bore dia d_1	Circlip					Groove				
	s	d_3	a Max.	$b \approx$	d_5 Min.	d_2		m H13	t	n Min.
						Size	Tolerance			
10	1	10.8	3.2	1.4	1.2	10.4	H11	1.1	0.2	0.6
20	1	21.5	4.2	2.3	2	21	H11	1.1	0.5	1.5
30	1.2	32.1	4.8	3	2	31.4	H12	1.3	0.7	2.1
40	1.75	43.5	5.8	3.9	2.5	42.5	H12	1.85	1.25	3.8
50	2	54.2	6.5	4.6	2.5	53	H12	2.15	1.5	4.5
60	2	64.2	7.3	5.4	2.5	63	H12	2.15	1.5	4.5
70	2.5	74.5	7.8	6.2	3	73	H12	2.65	1.5	4.5

(*Contd.*)

(*Contd.*)

80	2.5	85.5	8.5	7	3	83.5	H12	2.65	1.75	5.3
90	3	95.5	8.6	7.6	3.5	93.5	H12	3.15	1.75	5.3
100	3	105.5	9.2	8.4	3.5	103.5	H12	3.15	1.75	5.3
110	4	117	10.4	9	3.5	114	H13	4.15	2	6
120	4	127	11	9.7	3.5	124	H13	4.15	2	6
130	4	137	11	10.2	4	134	H13	4.15	2	6
140	4	147	11.2	10.7	4	144	H13	4.15	2	6
150	4	158	12	11.2	4	155	H13	4.15	2.5	7.5
160	4	169	13	11.6	4	165	H13	4.15	2.5	7.5
170	4	179.5	13.5	12.2	4	175	H13	4.15	2.5	7.5
180	4	189.5	14.2	13.2	4	185	H13	4.15	2.5	7.5
190	4	199.5	14.2	13.8	4	195	H13	4.15	2.5	7.5
200	4	209.5	14.2	14	4	205	H13	4.15	2.5	7.5
210	5	222	14.2	14	4	216	H13	5.15	3	9
220	5	232	14.2	14	4	226	H13	5.15	3	9
230	5	242	14.2	14	4	236	H13	5.15	3	9
240	5	252	14.2	14	4	246	H13	5.15	3	9
250	5	262	14.2	14	4	256	H13	5.15	3	9
260	5	275	16.2	16	5	268	H13	5.15	4	12
270	5	285	16.2	16	5	278	H13	5.15	4	12
280	5	295	16.2	16	5	288	H13	5.15	4	12
290	5	305	16.2	16	5	298	H13	5.15	4	12
300	5	315	16.2	16	5	308	H13	5.15	4	12

(DIN: 472)

Note: (i) All dimensions are in mm.
(iii) The circlips are made of spring steels 70C6 or 75C6.
Designation – A circlip for bore diameter (nominal size) d_1= 80 mm and thickness 2.5 mm, normal type (N) is designated as, Circlip 80 × 2.5 N.

15.15 SET COLLARS

Table 15.59 *Notations for light series set collars*

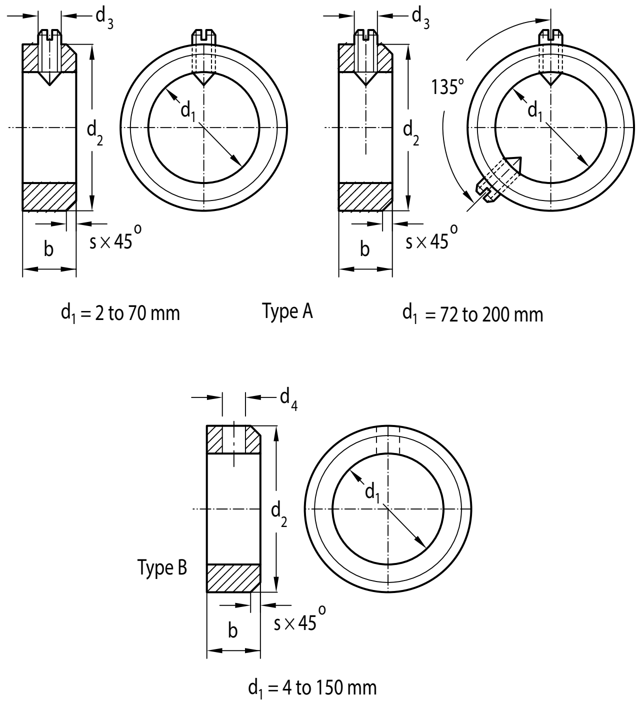


Table 15.60 *Dimensions for light series set collars*

Nominal bore d_1	b	d_2	d_3	d_4	s	Grub screw	Taper pins
H8	j14	h13		H11			
10	10	20	M5	3	1.0	M5 × 8	3 × 25
20	14	32	M6	5	1.6	M6 × 8	5 × 35
30	16	45	M8	6	1.6	M8 × 12	6 × 50
40	18	63	M10	8	1.6	M10 × 16	8 × 70
50	18	80	M10	10	1.6	M10 × 16	10 × 90
60	20	90	M10	10	1.6	M10 × 20	10 × 100
70	20	100	M10	10	1.6	M10 × 20	10 × 110
80	22	110	M12	10	2.5	M12 × 20	10 × 120
90	22	125	M12	12	2.5	M12 × 20	12 × 140

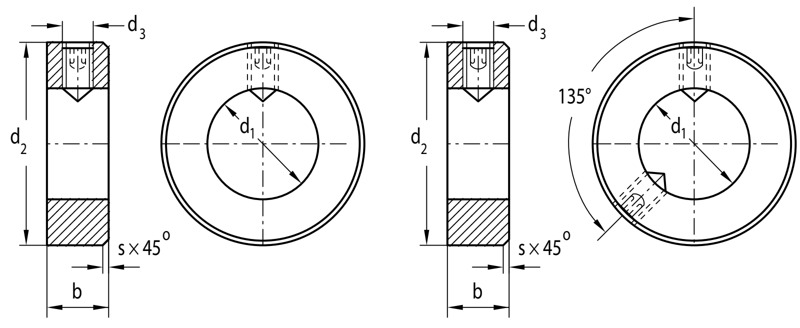
(*Contd.*)

(*Contd.*)

100	25	140	M12	12	2.5	M12 × 25	12 × 150
110	25	160	M12	12	2.5	M12 × 25	12 × 180
120	25	160	M12	12	2.5	M12 × 25	12 × 180
130	28	180	M16	16	4	M16 × 30	16 × 200
140	28	200	M16	16	4	M16 × 30	16 × 220
150	28	200	M16	16	4	M16 × 30	16 × 220
160	32	220	M20 × 2	—	4	M16 × 35	—
170	32	250	M20 × 2	—	4	M20 × 40	—
180	32	250	M20 × 2	—	4	M20 × 40	—
190	32	280	M20 × 2	—	4	M20 × 45	—
200	32	280	M20 × 2	—	4	M20 × 45	—

Note: (i) All dimensions are in mm.
(ii) The set collars are made of suitable steel such as St 37.
Designation – The set collar is designated by the series, the type, and the nominal bore. For example, Light set collar A 60.

Table 15.61 *Notations for heavy series set collars*



d_1 = 10 to 65 mm

d_1 = 68 to 150 mm

Table 15.62 *Dimensions for heavy series set collars*

Nominal bore (d_1)	b	d_2	d_3	s	**Grub screw**
H8	***j*14**	***h*13**			
10	12	25	M6	1.6	M6 × 8
20	20	50	M10	1.6	M10 × 16
30	22	63	M10	1.6	M10 × 16

(*Contd.*)

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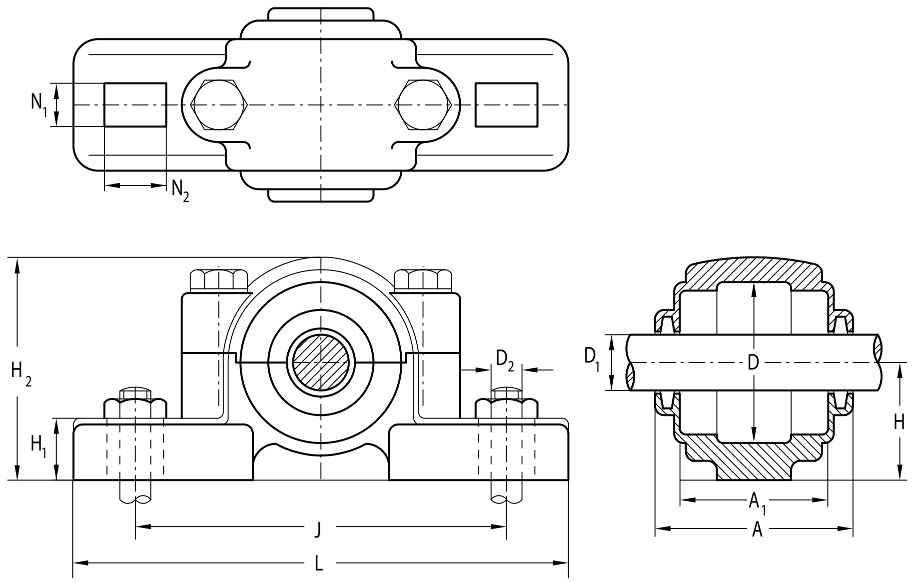
40	28	80	M12	1.6	M12 × 20
50	28	90	M12	1.6	M12 × 20
60	28	100	M12	1.6	M12 × 20
70	32	110	M16	2.5	M16 × 20
80	32	125	M16	2.5	M16 × 20
90	32	125	M16	2.5	M16 × 20
100	32	140	M16	2.5	M16 × 25
110	32	160	M16	2.5	M16 × 25
120	32	160	M16	2.5	M16 × 25
130	36	180	M16	4	M16 × 30
140	38	200	M20 × 2	4	M20 × 30
150	38	200	M20 × 2	4	M20 × 30

Note: All dimensions are in mm.

15.16 PLUMMER BLOCKS

Table 15.63 *Notations for plummer blocks*



Note: Grease nipple wherever desired, should be provided off centre. It is fitted on the side away from locknuts to ensure the grease supply reaching the bearing.

Table 15.64 *Dimensions for light series plummer blocks*

Housing seat dia. D (H7)	D_1	H	H_2 Approx.	J	D_2	N_1	N_2 Min.	A Max.	L Max.	A_1	H_1 Max.
47	17	35	65	115	M10	12	12	66	155	45	19
52	20	40	75	130	M12	15	15	67	170	46	22
62	25	50	90	150	M12	15	15	77	190	52	22
72	30	50	95	150	M12	15	15	82	190	52	22
80	35	60	110	170	M12	15	15	85	210	60	25
85	40	60	110	170	M12	15	15	85	210	60	25
90	45	60	115	170	M12	15	15	90	210	60	25
100	50	70	130	210	M16	18	18	95	270	70	28
110	55	70	135	210	M16	18	18	105	270	70	30
120	60	80	150	230	M16	18	18	110	290	80	30
125	60	80	155	230	M16	18	18	115	290	80	30
130	65	80	155	230	M16	18	18	115	290	80	30
140	70	95	175	260	M20	22	22	120	330	90	32
150	75	95	185	260	M20	22	22	125	330	90	32
160	80	100	195	290	M20	22	22	145	360	100	35
180	90	112	215	320	M24	26	26	160	400	110	40
200	100	125	240	350	M24	26	26	175	420	120	45
215	110	140	315	350	M24	26	26	185	420	120	45
230	115	150	335	380	M24	28	28	190	450	130	50
250	125	150	350	420	M30	33	33	205	510	150	50
270	133	160	370	450	M30	33	33	220	540	160	60
290	140	170	390	470	M30	33	33	235	560	160	60

Note: (i) All dimensions are in mm.
(ii) The plummer blocks are made of grey cast iron FG 150 or cast steel or spheroidal graphite cast iron SG 400/12.
(iii) Light series plummer blocks are suitable for bearings of dimension series 02.

Designation – A plummer block for radial roller bearing of dimension series 02 having housing seat diameter 100 mm is designated as, Plummer block 02 100

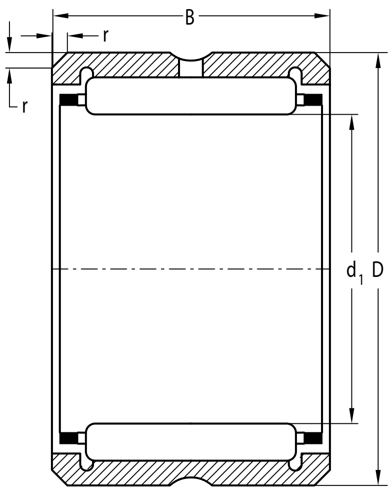
Table 15.65 *Dimensions for medium series plummer blocks*

Housing seat dia. *D* (H7)	D_1	*H*	H_2 Approx.	*J*	D_2	N_1	N_2 Min.	*A* Max.	*L* Max.	A_1	H_1 Max.
62	20	50	90	150	M12	15	15	82	190	52	22
72	25	50	95	150	M12	15	15	82	190	52	22
80	30	60	110	170	M12	15	15	90	210	60	25
90	35	60	115	170	M12	15	15	95	210	60	25
100	40	70	130	210	M16	18	18	105	270	70	28
110	45	70	135	210	M16	18	18	115	270	70	30
120	50	80	150	230	M16	18	18	120	290	80	30
130	55	80	155	230	M16	18	18	125	290	80	30
140	60	95	175	260	M20	22	22	130	330	90	32
150	60	95	185	260	M20	22	22	130	330	90	32
160	65	100	195	290	M20	22	22	140	360	100	35
170	70	112	210	290	M20	22	22	145	360	100	35
180	75	112	215	320	M24	26	26	155	400	110	40

Note: (i) All dimensions are in mm.
(ii) Medium series plummer blocks are suitable for bearings of dimension series 03.

15.17 NEEDLE BEARINGS

Table 15.66 *Dimensions and load capacities of (NRB) cage-guided needle bearings without inner ring*



Type and designation (NRB)	Dimensions (mm)				Basic capacities (N)	
	d_1 (F6)	*D*	*B*	*r*	*C*	C_o
NB 15 23 12	15	23	12	0.35	8300	8900
NB 15 23 16	15	23	16	0.35	12800	15700

(*Contd.*)

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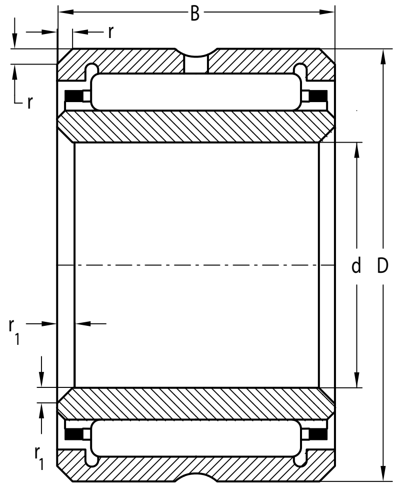
NB 15 23 20	15	23	20	0.35	12800	15700
NB 20 28 16	20	28	16	0.35	15100	20900
NB 20 28 20	20	28	20	0.35	18900	28000
NB 20 28 23	20	28	23	0.35	18900	28000
NB 25 33 16	25	33	16	0.35	17600	27400
NB 25 33 20	25	33	20	0.35	22000	36600
NB 30 40 20	30	40	20	0.35	26000	42300
NB 30 40 30	30	40	30	0.35	39000	71100
NB 30 47 30	30	47	30	0.35	47200	64500
NB 35 45 20	35	45	20	0.35	27800	48300
NB 35 45 30	35	45	30	0.35	41700	81400
NB 40 50 20	40	50	20	0.35	30100	56000
NB 40 50 30	40	50	30	0.35	45100	94000
NB 40 52 20	40	52	20	0.35	30100	56000
NB 45 55 20	45	55	20	0.35	32300	63700
NB 45 55 30	45	55	30	0.35	48500	107 400
NB 50 62 25	50	62	25	0.65	41400	90800
NB 50 62 35	50	62	35	0.65	57800	139 400
NB 50 65 25	50	65	25	0.65	41400	90800
NB 50 68 25	50	68	25	0.65	41400	90800
NB 55 68 25	55	68	25	0.65	41400	93600
NB 55 68 35	55	68	35	0.65	61000	154 500
NB 60 72 25	60	72	25	0.65	42500	108 100
NB 60 72 35	60	72	35	0.65	63100	166 000
NB 65 78 25	65	78	25	0.85	50600	114 800
NB 65 78 35	65	78	35	0.85	66100	181 100
NB 70 85 25	70	85	25	0.85	55100	131 600
NB 70 85 35	70	85	35	0.85	76900	202 000
NB 75 92 25	75	92	25	0.85	56000	129 000
NB 75 92 35	75	92	35	0.85	82200	204 400
NB 80 95 25	80	95	25	1.35	73000	148 000
NB 80 95 35	80	95	35	1.35	103 000	220 000
NB 80 100 35	80	100	35	1.35	97800	222 900
NB 90 110 25	90	110	25	1.35	75300	136 800
NB 90 110 35	90	110	35	1.35	102 800	247 400

Table 15.67 *Dimensions and load capacities of cage-guided needle bearings with inner ring*



Type and designation (NRB)	Dimensions (mm)					Basic capacities (N)	
	d	*D*	*B*	*r*	r_1	*C*	C_o
NBI 15 27 16	15	27	16	0.35	0.30	15200	20800
NBI 15 27 20	15	27	20	0.35	0.30	19400	27500
NBI 15 28 16	15	28	16	0.35	0.35	15100	20900
NBI 15 28 20	15	28	20	0.35	0.35	18900	28000
NBI 20 32 16	20	32	16	0.35	0.30	21400	34800
NBI 20 32 20	20	32	20	0.35	0.30	17600	27400
NBI 20 33 16	20	33	16	0.35	0.35	22000	36600
NBI 20 33 20	20	33	20	0.35	0.35	21700	26200
NBI 25 38 15	25	38	15	0.35	0.35	15500	13200
NBI 25 38 20	25	38	20	0.35	0.30	25300	40400
NBI 25 38 30	25	38	30	0.35	0.30	41500	73000
NBI 25 40 20	25	40	20	0.35	0.35	26000	42300
NBI 25 40 30	25	40	30	0.35	0.35	39000	71100
NBI 30 45 20	30	45	20	0.35	0.35	27800	48300
NBI 30 45 30	30	45	30	0.35	0.35	41700	81400
NBI 35 50 20	35	50	20	0.35	0.35	30100	56000
NBI 35 50 30	35	50	30	0.35	0.35	45100	94000
NBI 40 55 20	40	55	20	0.35	0.35	32300	63700

(*Contd.*)

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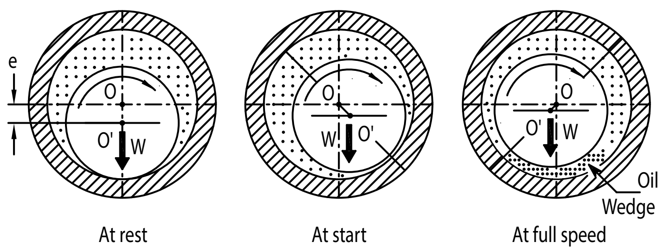
NBI 40 55 30	40	55	30	0.35	0.35	48500	107 400
NBI 45 62 25	45	62	25	0.65	0.35	41400	90800
NBI 45 62 35	45	62	35	0.65	0.65	57800	139 400
NBI 45 65 25	45	65	25	0.65	0.65	41400	90800
NBI 45 68 25	45	68	25	0.65	0.35	41400	90800
NBI 50 68 25	50	68	25	0.65	0.65	41400	93600
NBI 50 68 35	50	68	35	0.65	0.65	61000	154 500
NBI 55 72 25	55	72	25	0.65	0.65	42500	108 100
NBI 55 72 35	55	72	35	0.65	0.65	63100	166 000
NBI 65 90 35	65	90	35	0.85	0.85	98000	210 000
NBI 70 95 25	70	95	25	1.35	1.35	73000	148 000
NBI 70 95 35	70	95	35	1.35	1.35	103 000	230 000
NBI 70 100 35	70	100	35	1.35	1.35	88000	220 000
NBI 75 105 35	75	105	35	1.35	1.35	105 000	240 000
NBI 80 110 25	80	110	25	1.35	1.35	77000	165 000
NBI 80 110 35	80	110	35	1.35	1.35	108 000	255 000

Note: C = Dynamic load capacity, C_o = Static load capacity.

16 Sliding Contact Bearings

16.1 SLIDING CONTACT BEARINGS

Table 16.1 *Modes of lubrication*

Thick film lubrication
Thick film lubrication describes a condition of lubrication, where two surfaces of the journal and the bearing in relative motion are completely separated by a film of fluid. Since there is no contact between the surfaces, the properties of surface, like surface finish, have little or no influence on the performance of the bearing. The resistance to relative motion arises from the viscous resistance of the fluid. Therefore, the viscosity of the lubricant affects the performance of the bearing. Thick film lubrication is further divided into two groups: hydrodynamic and hydrostatic lubrication.
Hydrodynamic lubrication

Hydrodynamic lubrication is a system of lubrication in which the load supporting fluid film is created by the wedge shape and relative motion of the sliding surfaces. Since the pressure is created within the system due to rotation of the shaft, this type of bearing is known as 'self-acting' bearing. This mode of lubrication is seen in bearings mounted on engines and centrifugal pumps. A 'journal' bearing is a sliding contact bearing working on hydrodynamic lubrication and which supports the load in radial direction. There are two types of hydrodynamic journal bearings namely full journal bearing and partial bearing.

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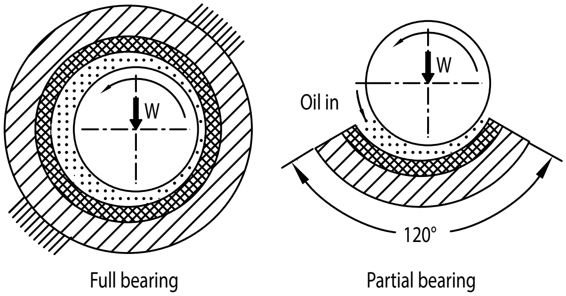
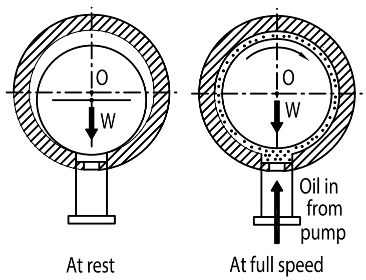
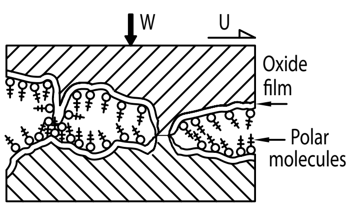
Full and Partial journal bearings

In full journal bearing, the angle of contact of the bushing with journal is 360°. Full journal bearing can take load in any radial direction. Most of the bearings used in industrial applications are full journal bearing. In partial bearings, the angle of contact between the bush and the journal is always less than 180°. Most of the partial bearings in practice have 120° angle of contact. Partial bearing can take load in only one radial direction. Partial bearings are used for railroad-cars.
Hydrostatic lubrication

Hydrostatic lubrication is a system of lubrication in which the load supporting fluid film, separating the two surfaces is created by an external source, like a pump, supplying sufficient fluid under pressure. Since the lubricant is supplied under pressure, this type of bearing is called 'externally pressurised' bearing. Hydrostatic bearings are used on vertical turbo generators, centrifuges and ball mills.
Thin film lubrication


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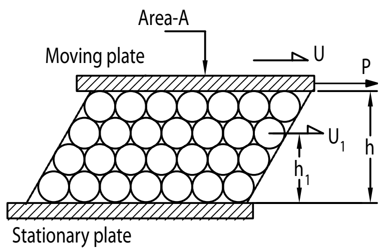
Thin film lubrication, which is also called boundary lubrication, is a condition of lubrication where the lubricant film is relatively thin and there is partial metal to metal contact. This mode of lubrication is seen in door hinges and machine tool slides. The conditions resulting in boundary lubrication are excessive load, insufficient surface area or oil supply, low speed and misalignment. There are certain fatty acids, which contain polar molecules. Molecules in which there is a permanent separation of positive and negative charges are called polar molecules. Their polarity has a tendency to orient and stick to the surface in a particular fashion. The clusters of polar molecules, cohering to one another and adhering to the surface, form a compact film which prevents metal to metal contact. This results in partial lubrication. There is also a zone where metal to metal contact take place, junctions are formed at high spots and shearing takes place due to relative motion.

16.2 VISCOSITY

Table 16.2 *Viscosity*

Newton's law of viscosity	
	
$P = \mu A\left(\dfrac{U}{h}\right)$ (16.1) $P = \mu A\left(\dfrac{dU}{dh}\right)$ (16.2)	P = tangential force (N) μ = absolute viscocity of oil (MPa-s) or (N-s/mm^2) A = area of moving plate (mm^2) U = velocity of moving plate (mm/s) h = film thickness (mm) **Note:** When the velocity distribution is non-linear with respect to h, the term (U/h) is replaced by (dU/dh)
Viscosity relationships	
$z_k = \left[0.22t - \dfrac{180}{t}\right]$ (16.3)	z_k = kinematic viscosity in centiStokes (cSt) t = viscosity in Saybolt Universal Seconds (SUS)
$z_k = \dfrac{z}{\rho}$ (16.4)	z = viscosity (cP) ρ = density of the lubricant (gm/cm^3)
$\mu = \dfrac{z}{10^9}$ (16.5)	μ = viscosity (MPa-s) or (N-s/mm^2)
Viscosity–Temperature relationship	
$\log \mu \approx A + \dfrac{B}{T}$ (16.6)	A, B = constants T = absolute temperature

(*Contd.*)

(*Contd.*)

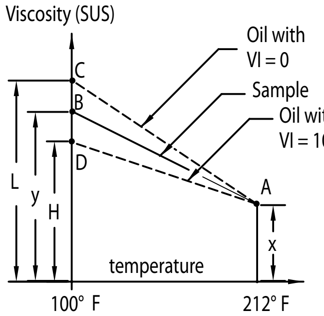
Viscosity index	
The rate of change of viscosity with respect to temperature is indicated by a number called viscosity index (V.I.). The viscosity index is defined as an arbitrary number used to characterize the variation of the kinematic viscosity of lubricating oil with temperature. In order to find out viscosity index of the oil, two groups of reference oils are considered. One group consists of oils having VI = 100 and these oils have very small change of viscosity with temperature. The other group consists of oils having VI = 0 and these oils have very large change of viscosity with temperature. The given oil is compared with these two reference oils, one with a viscosity index of 100 and the other of zero.	
	$$\mathrm{VI} = \left(\frac{L-y}{L-H}\right) \times 100\% \quad (16.7)$$

Table 16.3 *Viscosity–Temperature relationships—Graphs*

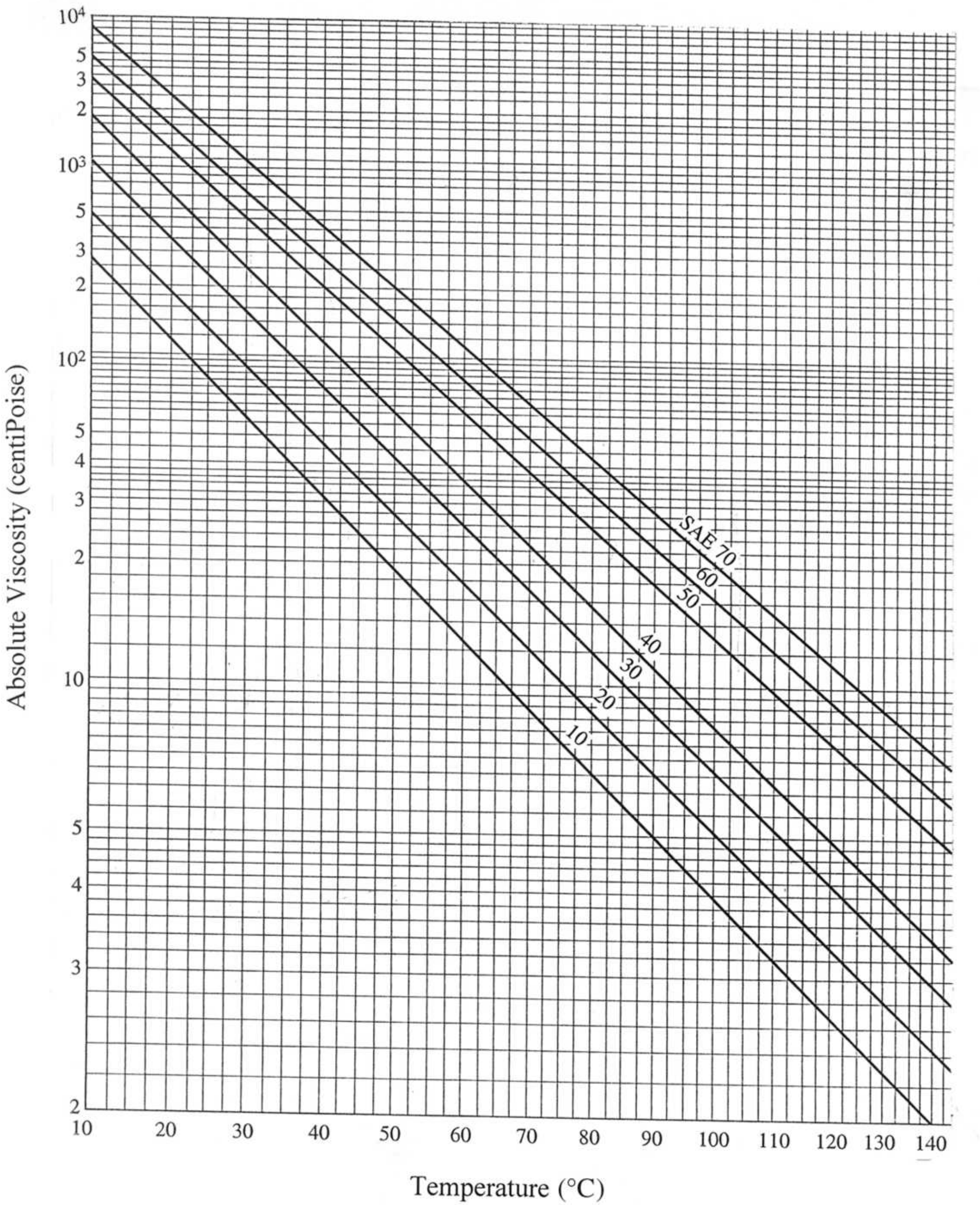


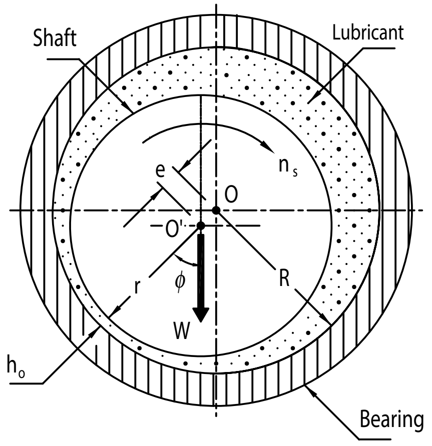
Table 16.4 *Viscosity–Temperature relationships—Analytical expressions*

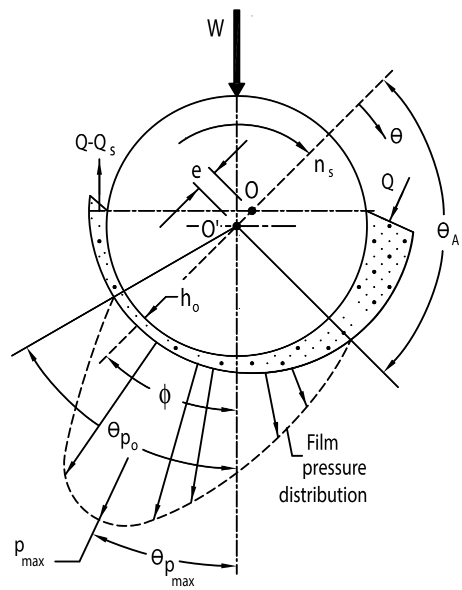
Approximate expression for Viscosity–Temperature functions		
$$\mu = \mu_o \exp\left[\frac{b}{(1.8T+127)}\right] \qquad (16.8)$$ μ = viscosity of oil at temperature T (mPa-s) T = temperature of oil (°C) μ_o, b = constants		
SAE Grade	**(μ_o)**	**(b)**
10	0.1089	1157.5
20	0.0937	1271.6
30	0.0971	1360.0
40	0.0827	1474.4
50	0.1171	1509.6
60	0.1288	1564.0

Note: 1 mPa-s = (10^{-9}) MPa-s = 1 cP

16.3 HYDRODYNAMIC BEARINGS

Table 16.5 *Dimensionless performance parameters*





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Eccentricity	
e = distance OO' = eccentricity (mm)	O = axis of bearing O' = axis of journal
Radial clearance	
$c = R - r$ (16.9)	c = radial clearance (mm) R = radius of bearing (mm) r = radius of journal (mm)
Eccentricity ratio	
$\varepsilon = \dfrac{e}{c}$ (16.10)	ε = eccentricity ratio (dimensionless)
Minimum film thickness	
$R = e + r + h_o$ (16.11)	h_o = minimum film thickness (mm)
Minimum film thickness variable	
$\varepsilon = 1 - \left(\dfrac{h_o}{c}\right)$ (16.12)	$\left(\dfrac{h_o}{c}\right)$ = minimum film thickness variable (dimensionless)
Sommerfeld number	
$S = \left(\dfrac{r}{c}\right)^2 \dfrac{\mu n_s}{p}$ (16.13)	S = Sommerfeld number (dimensionless) μ = viscosity of the lubricant (MPa-s) or (N-s/mm^2) n_s = journal speed (rev/s) p = unit bearing pressure i.e., load per unit of the projected area (MPa) or (N/mm^2)
Unit bearing pressure	
$p = \dfrac{W}{ld}$ (16.14)	W = radial load acting on bearing (N) l = axial length of bearing (mm) d = journal diameter (mm)
Attitude angle	
ϕ = attitude angle = angle of eccentricity (deg). **Note** (ϕ) = locates the position of minimum film thickness with respect to the direction of load.	
Coefficient of Friction Variable (CFV)	
$(\text{CFV}) = \left(\dfrac{r}{c}\right) f$ (16.15)	(CFV) = coefficient of friction variable (dimensionless) f = coefficient of friction
Flow Variable (FV)	
$(\text{FV}) = \dfrac{Q}{rcn_s l}$ (16.16)	(FV) = flow variable (dimensionless) Q = flow of the lubricant drawn into clearance space by journal (mm^3/s)

(*Contd.*)

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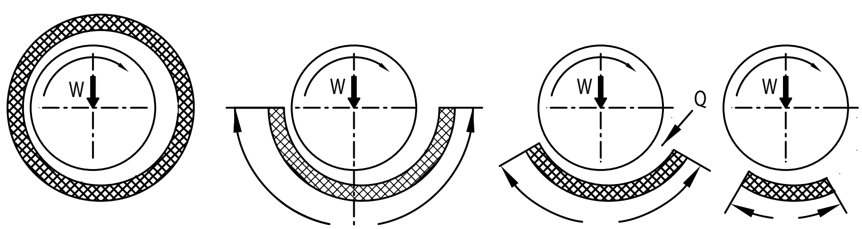
Flow Ratio Variable (FRV)	
$(\text{FRV}) = (Q/Q_s)$ (16.17)	(FRV) = flow ratio variable (dimensionless) Q_s = out-flow of lubricant from both sides of bearing or side leakage (mm^3/s)
Temperature Rise Variable (TRV)	
$(\text{TRV}) = \dfrac{\rho C_p \Delta t}{p}$ (16.18) $\Delta t = (t_2 - t_1)$ (16.19)	(TRV) = temperature rise variable (dimensionless) ρ = density of lubricating oil (gm/cm^3) ($\rho \approx 0.86$ to 0.88 gm/cm^3) C_p = specific heat of lubricating oil (kJ/kg °C) ($C_p \approx 1.68$ to 1.76 kJ/kg °C) Δt = temperature rise (°C) t_2 = outlet temperature of lubricant (°C) t_1 = inlet temperature of lubricant (°C)
Pressure Ratio (PR)	
$(\text{PR}) = \left(\dfrac{p}{p_{\max}}\right)$ (16.20)	(PR) = pressure ratio (dimensionless) $p_{\max}$ = maximum pressure developed in lubricant film (MPa) or (N/mm^2)
Position of Maximum film pressure ($\theta_{p_{\max}}$)	
$\theta_{p_{\max}}$ = Position of maximum film pressure with respect to the direction of load (deg)	
Position at which the lubricant film begins (θ_A)	
θ_A = Angle from line of centres to start of lubricant film, measured in the direction of rotation (deg)	
Position at which the lubricant film ends (θ_{p_o})	
θ_{p_o} = Position at which lubricant film terminates with respect to the direction of load (deg)	
Angular extent of bearing arc	
β = Angular extent of bearing arc (deg.) For example, β = 360° for full bearing	



β = 360° β = 180° β = 120° β = 60°

16.4 RAIMONDI AND BOYD CHARTS

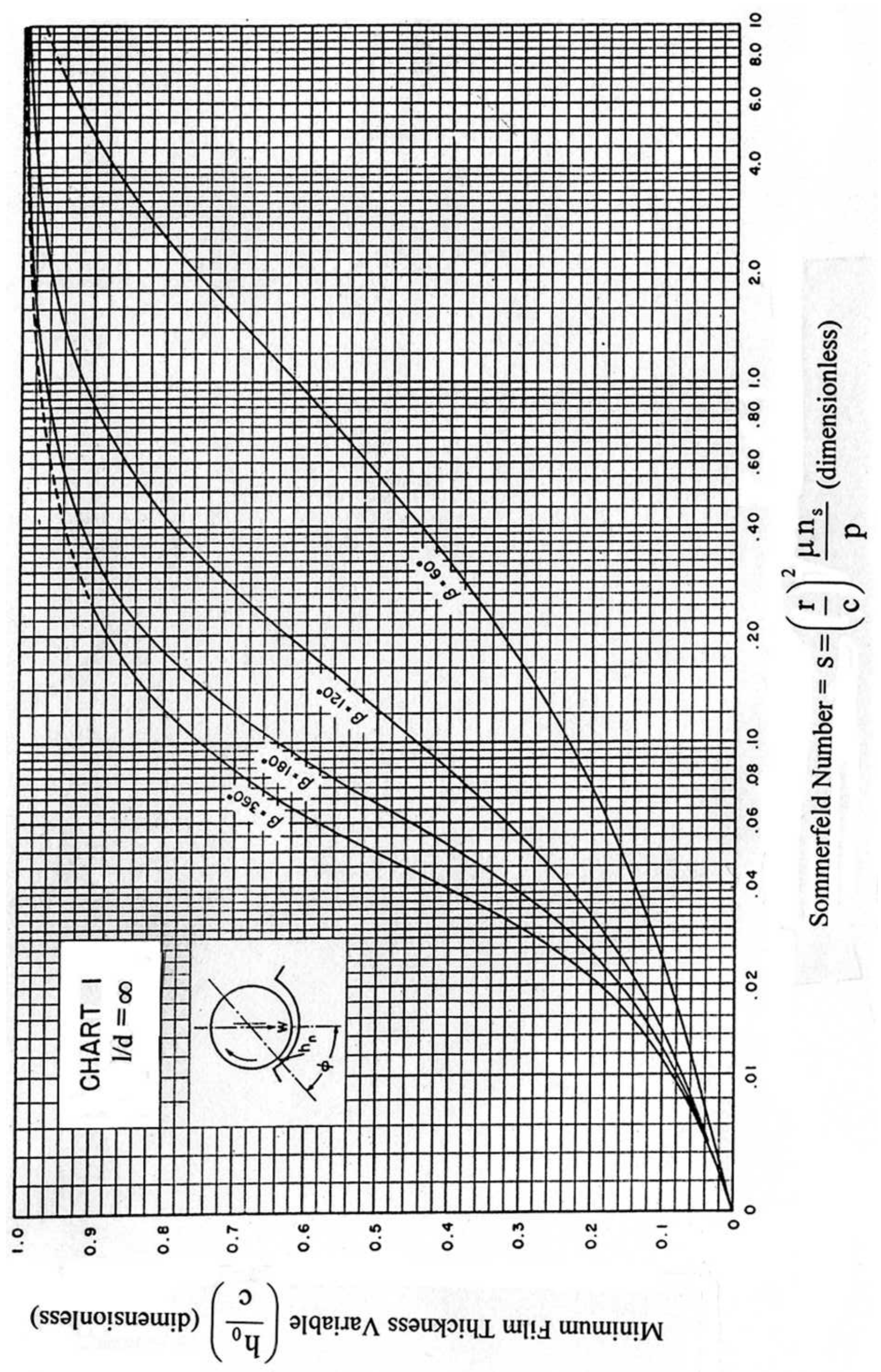


Chart 16.1 *Minimum film thickness variable v/s Sommerfeld number (1/d = ∞)*

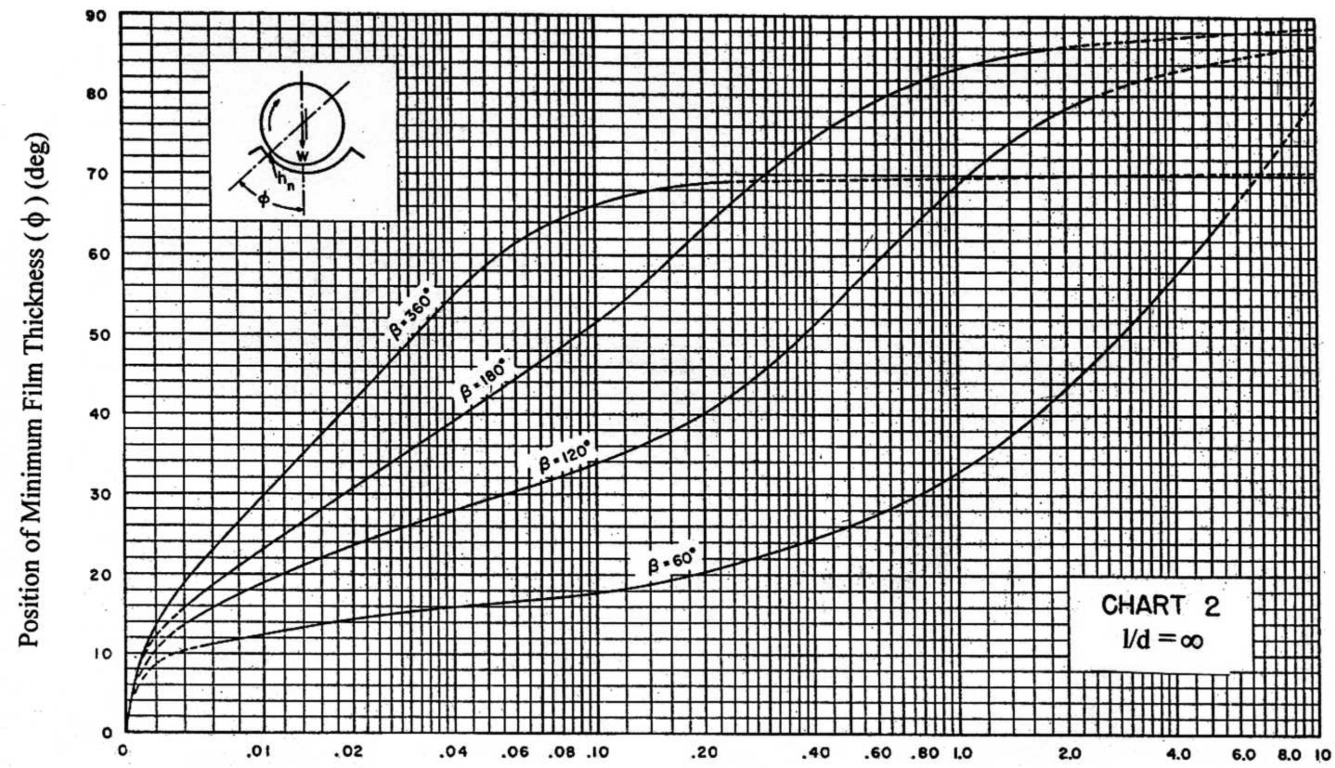


Chart 16.2 *Sommerfeld number* $s = \left(\frac{r}{c}\right)^2 \frac{\mu n_s}{p}$ *(dimensionless)*

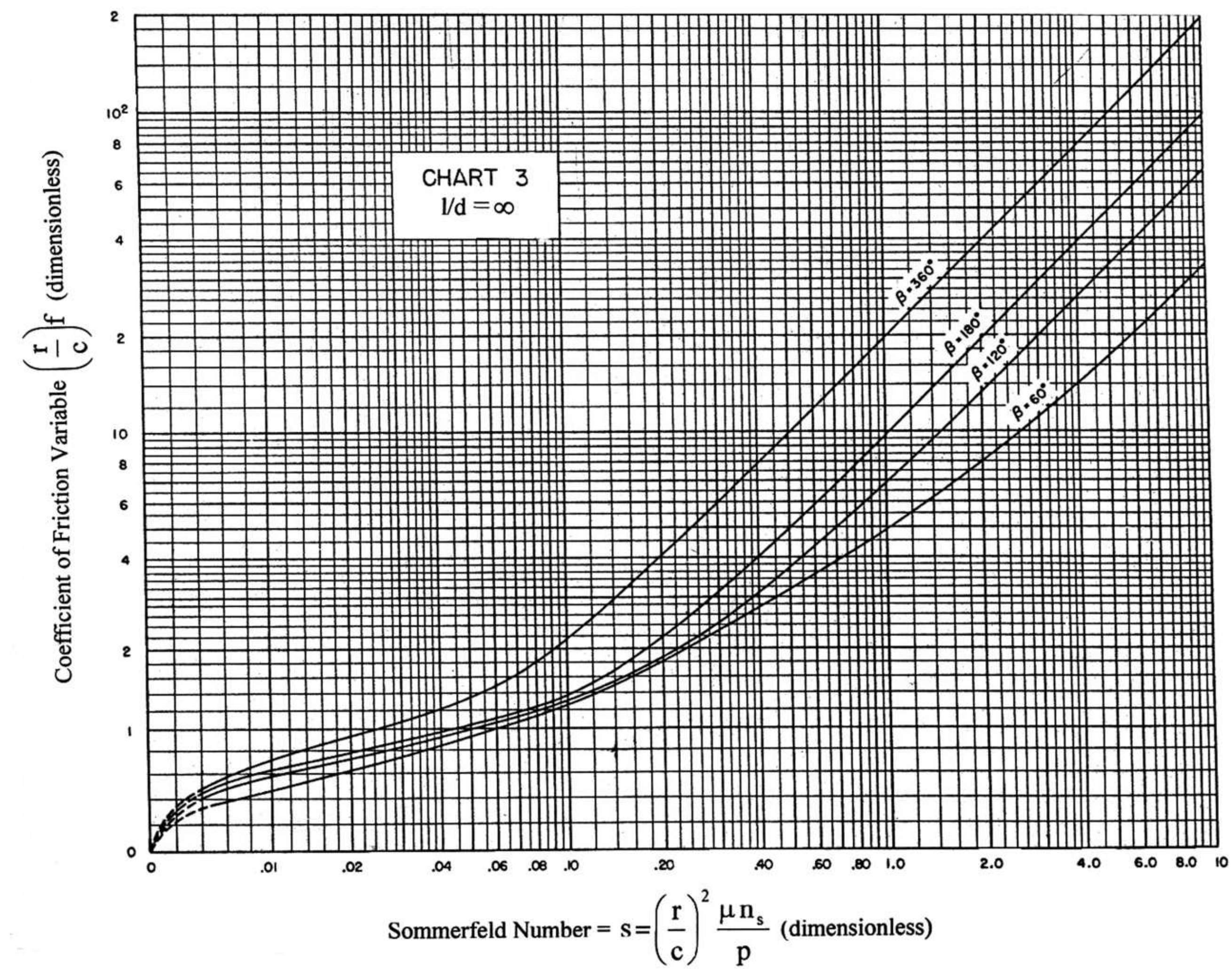


Chart 16.3 *Coefficient of friction variable v/s Sommerfeld number (1/d = ∞)*

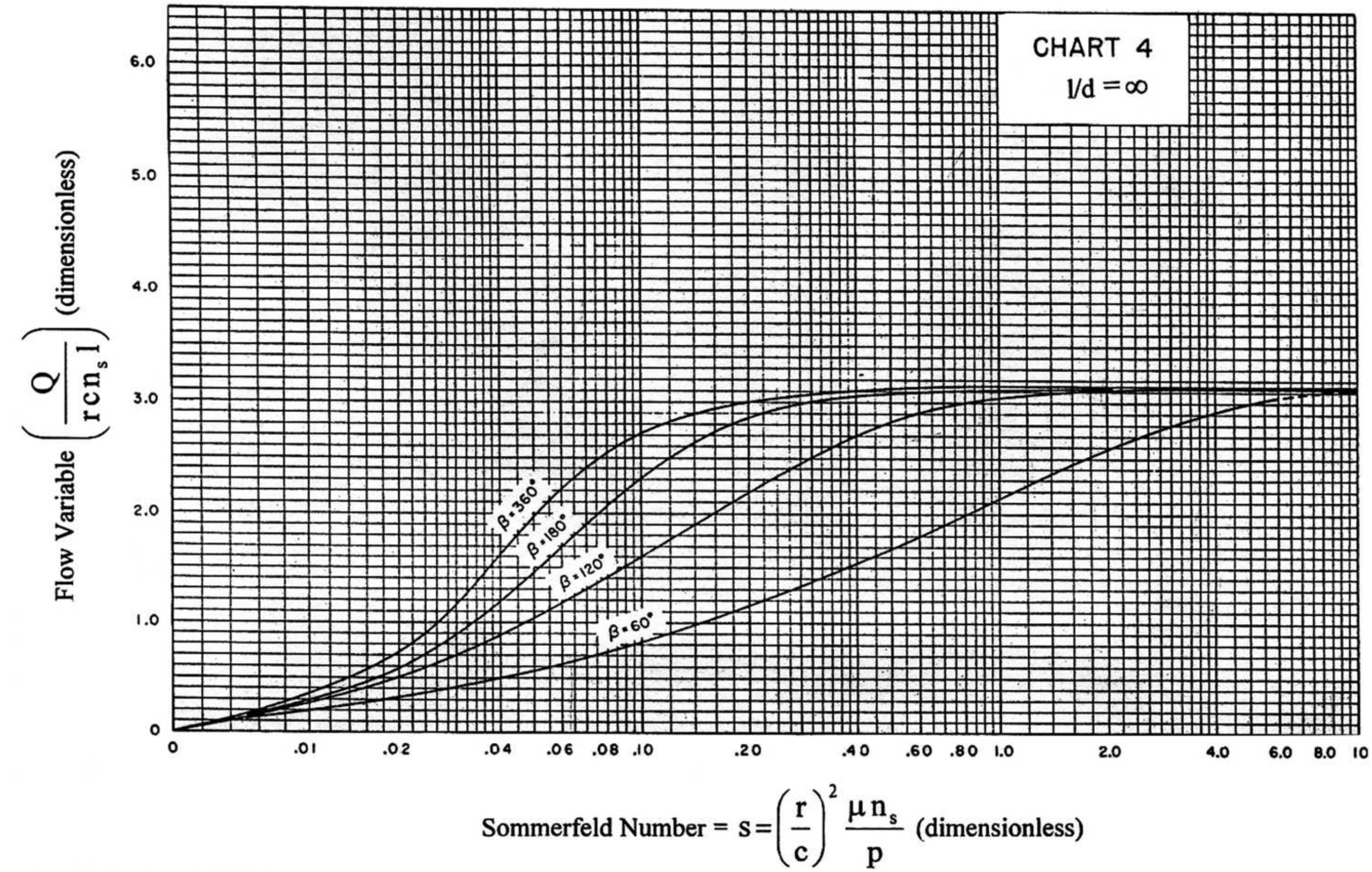


Chart 16.4 *Flow variable v/s Sommerfeld number (1/d = ∞)*

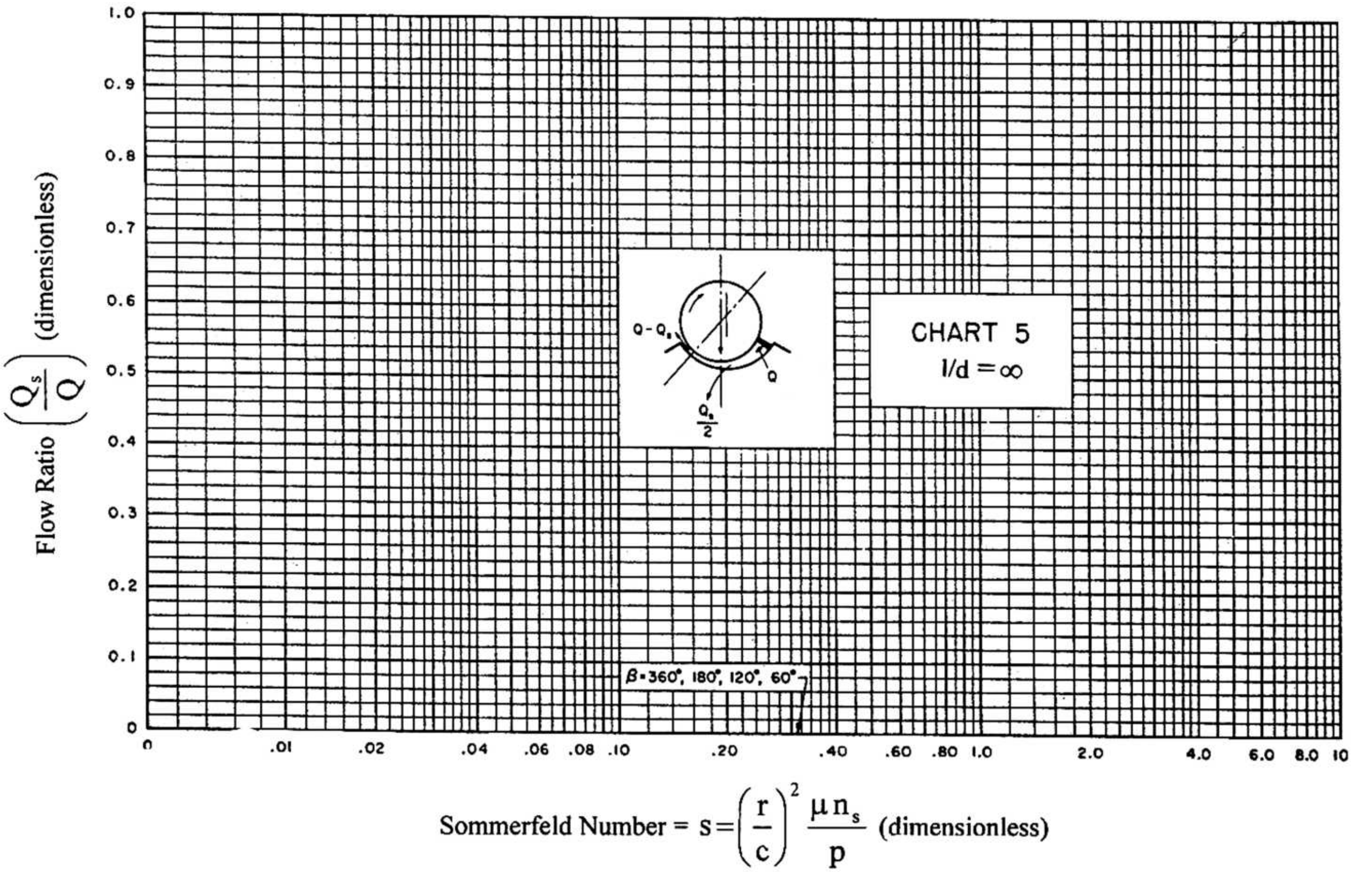


Chart 16.5 *Flow ratio v/s Sommerfeld number (1/d = ∞)*

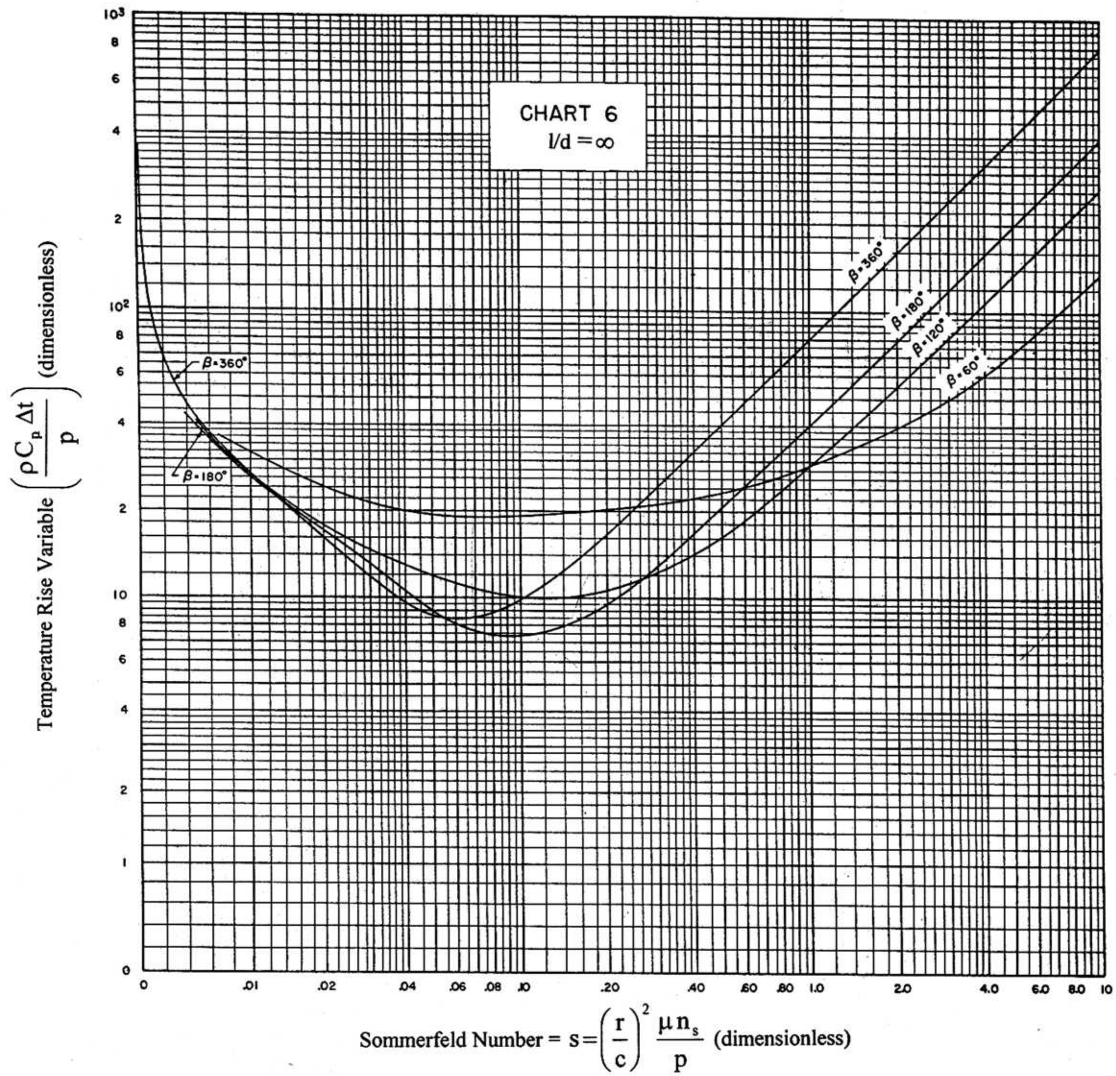


Chart 16.6 *Temperature rise variable v/s Sommerfeld number (1/d = ∞)*

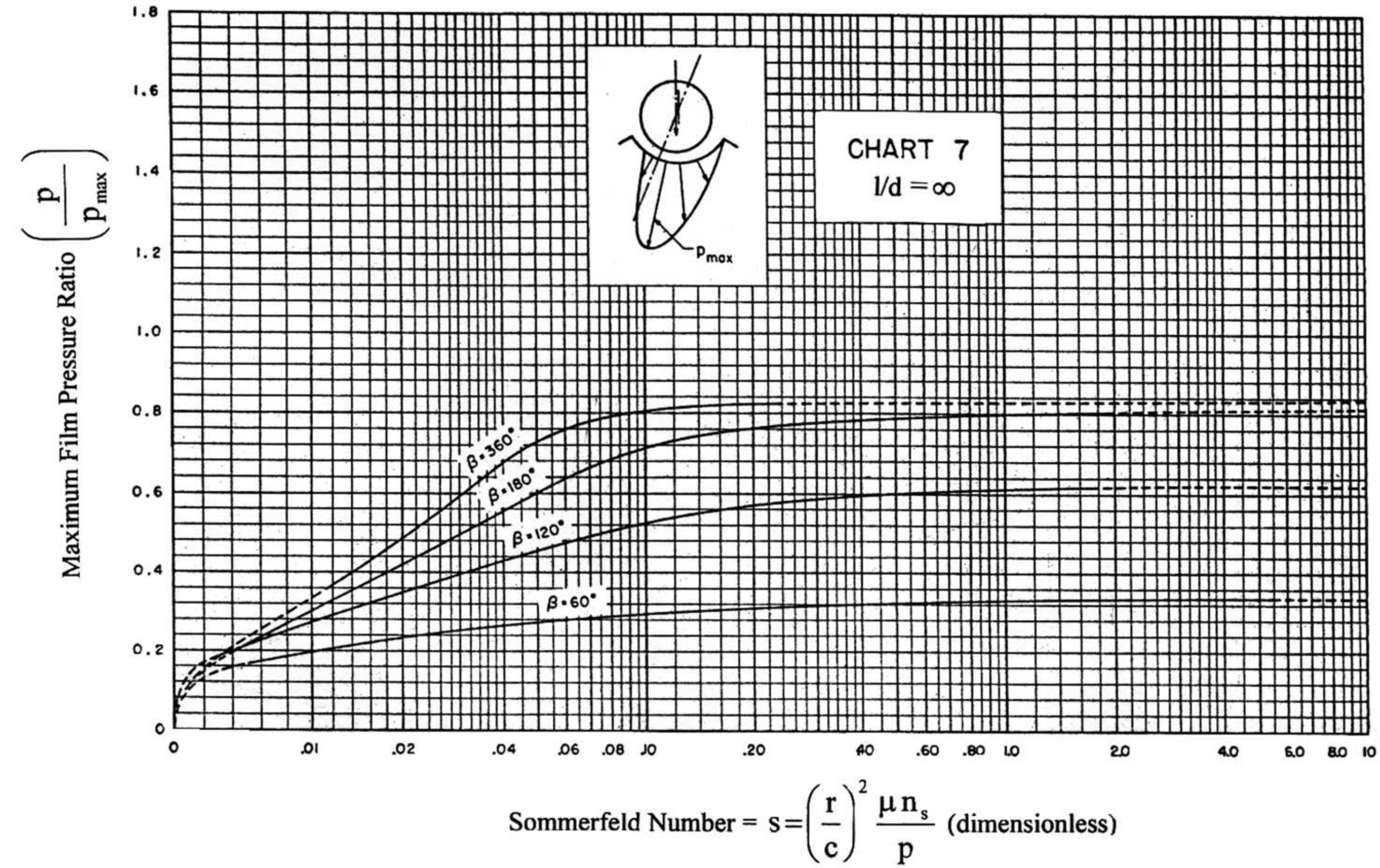


Chart 16.7 *Maximum film pressure ratio v/s Sommerfeld number (1/d = ∞)*

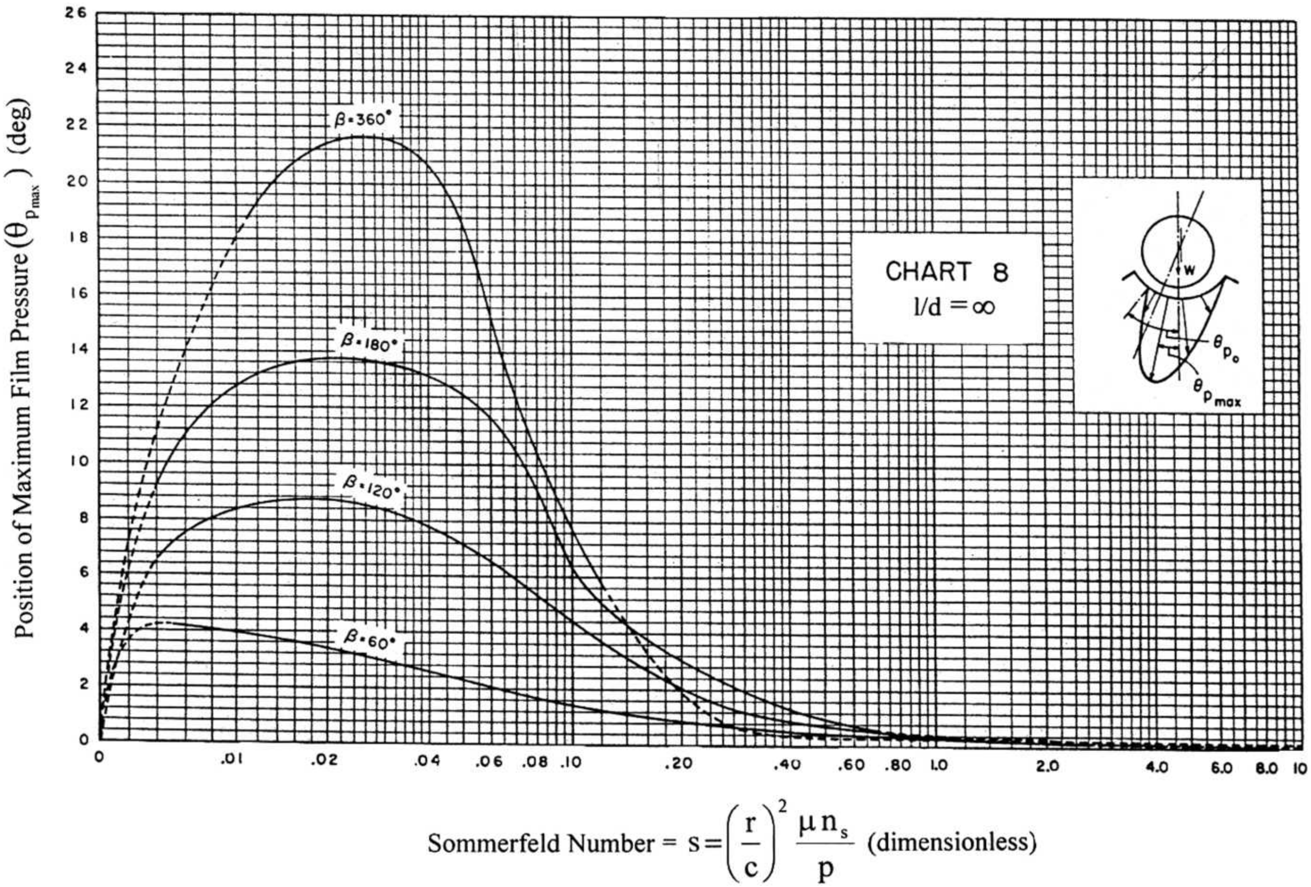


Chart 16.8 *Position of maximum film pressure v/s Sommerfeld number (1/d = ∞)*

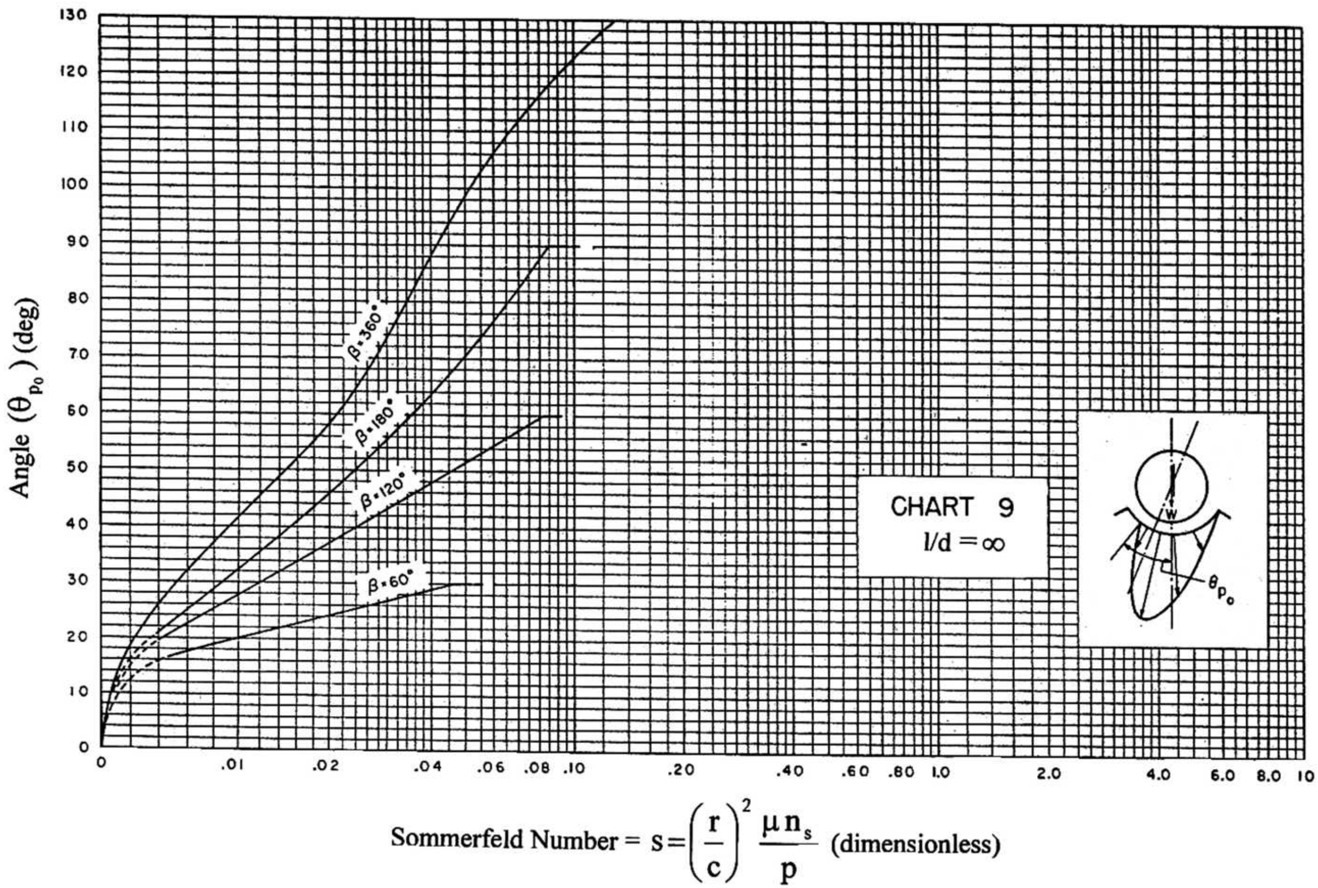


Chart 16.9 *Film terminating angle v/s Sommerfeld number (1/d = ∞)*

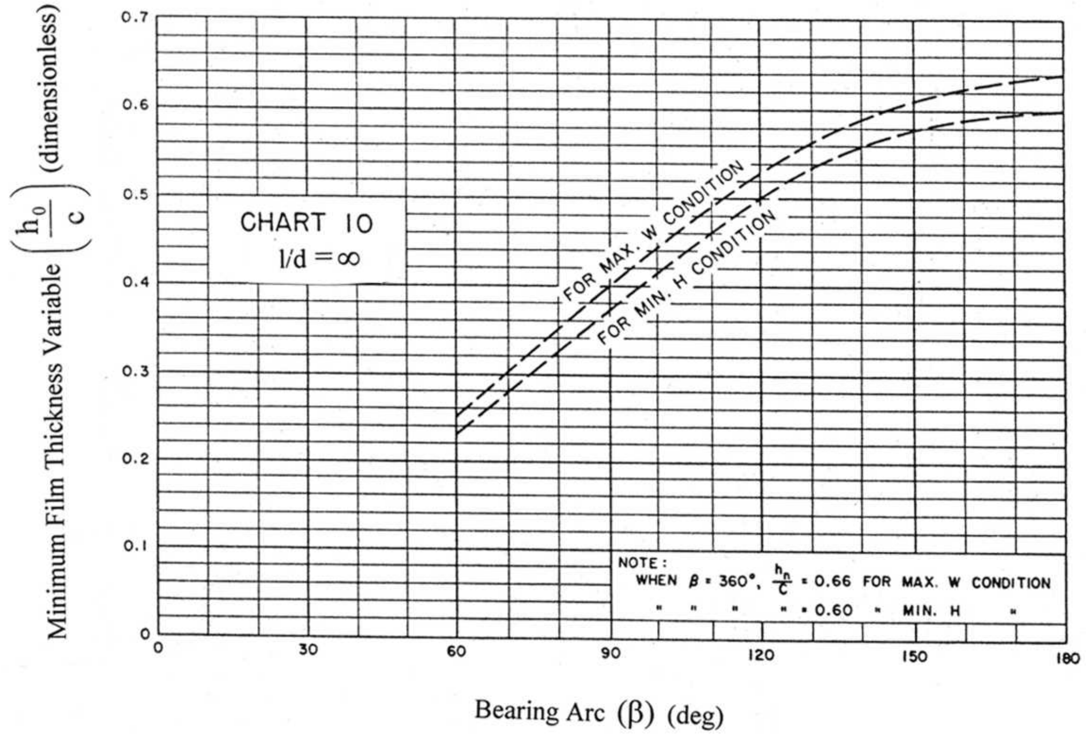


Chart 16.10 *Minimum film thickness variable v/s Bearing arc (Maximum load and Minimum power loss conditions) (1/d = ∞)*

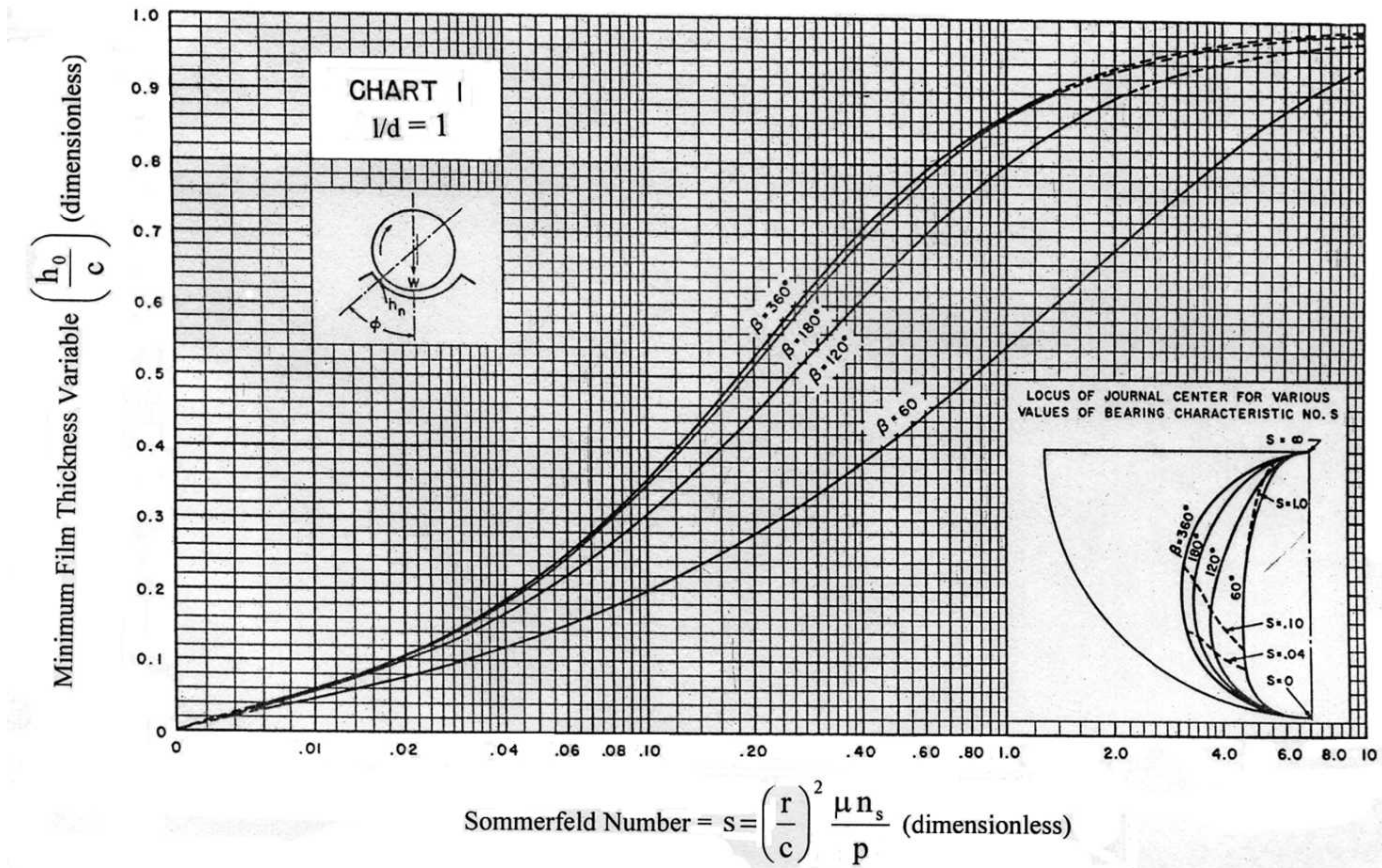
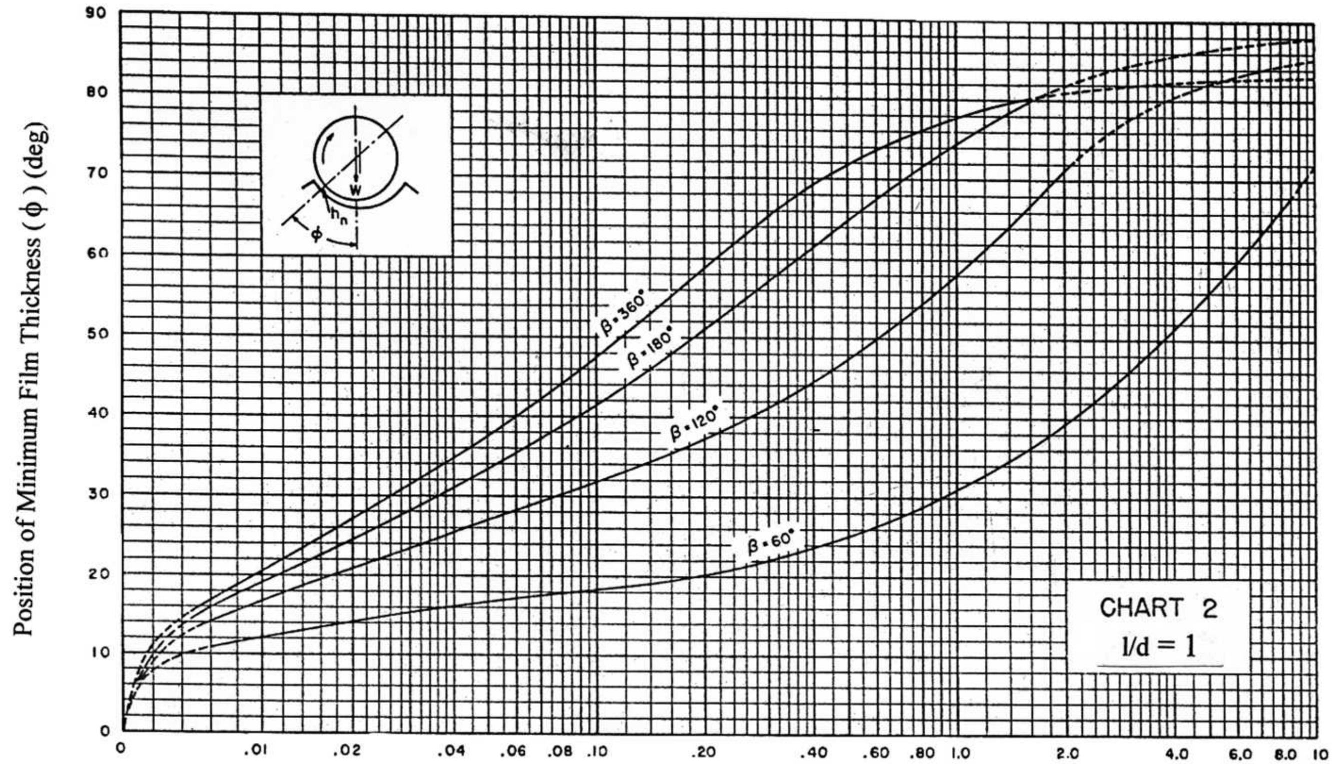


Chart 16.11 *Minimum film thickness variable v/s Sommerfeld number (1/d = 1)*



Sommerfeld Number = $S = \left(\frac{r}{c}\right)^2 \frac{\mu n_s}{p}$ (dimensionless)

Chart 16.12 *Position of minimum film thickness v/s Sommerfeld number (1/d = 1)*

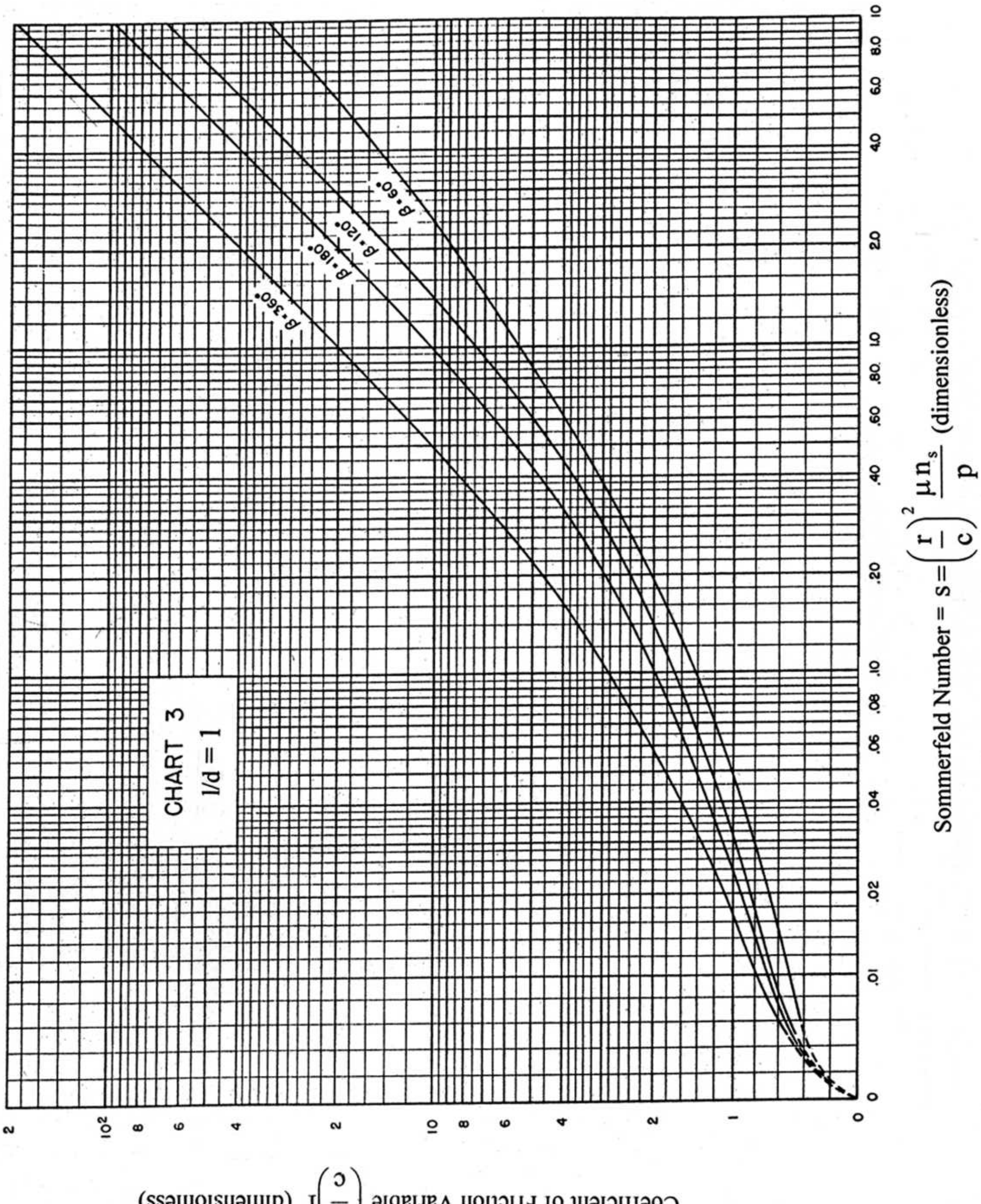


Chart 16.13 *Coefficient of friction variable v/s Sommerfeld number (1/d = 1)*

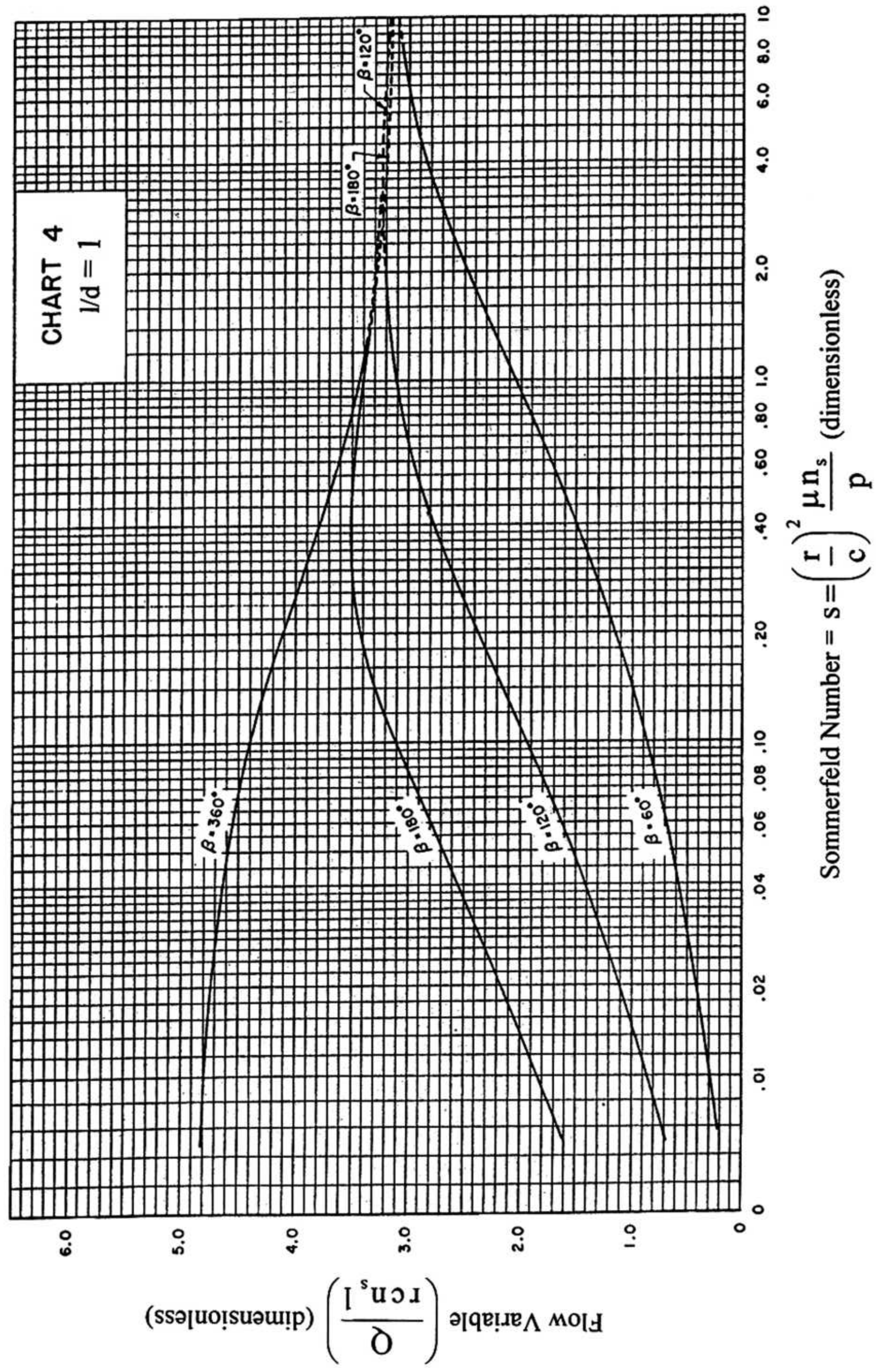


Chart 16.14 *Flow variable v/s Sommerfeld number (1/d = 1)*

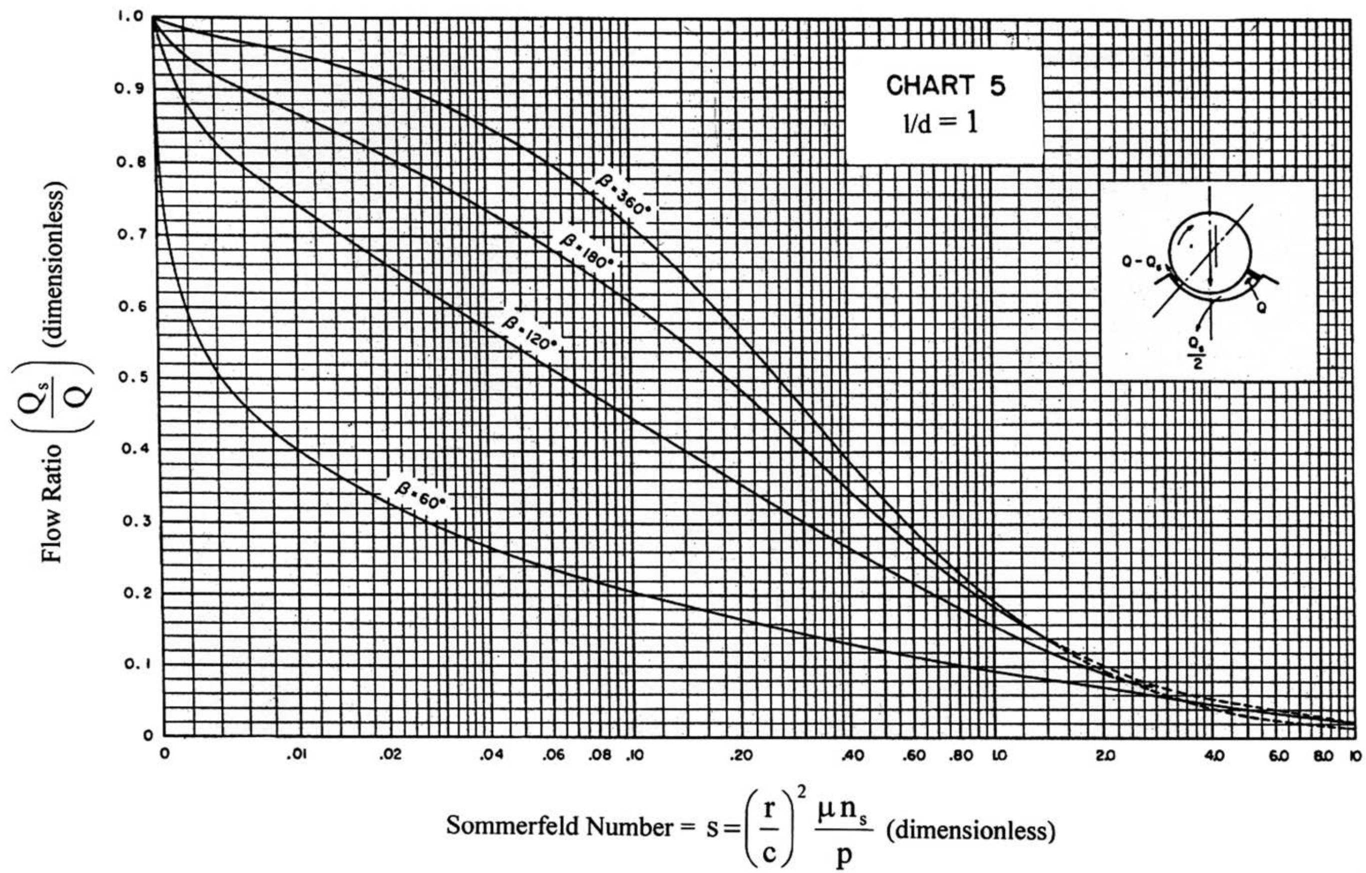


Chart 16.15 *Flow ratio v/s Sommerfeld number (1/d = 1)*

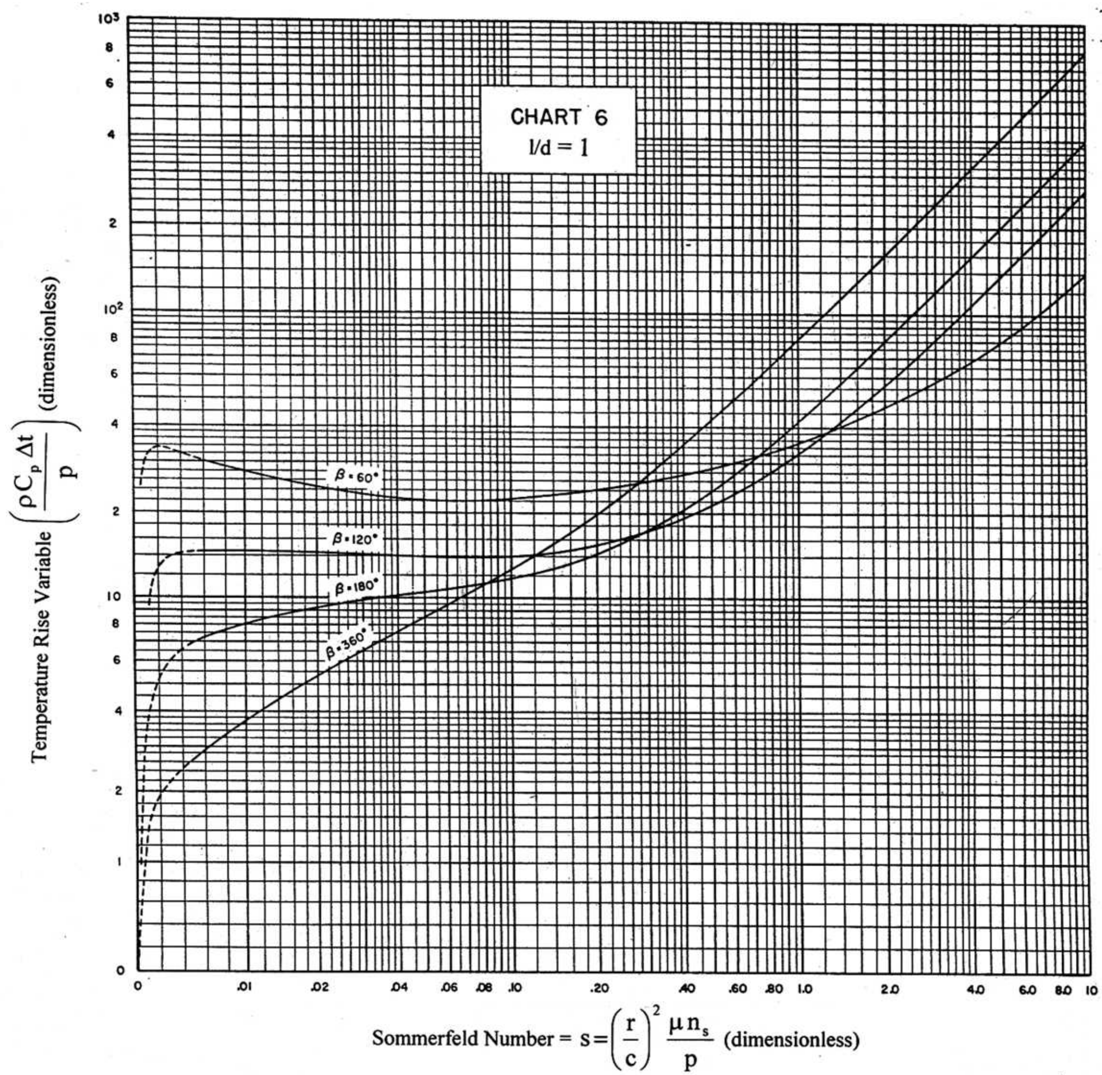


Chart 16.16 *Temperature rise variable v/s Sommerfeld number (1/d = 1)*

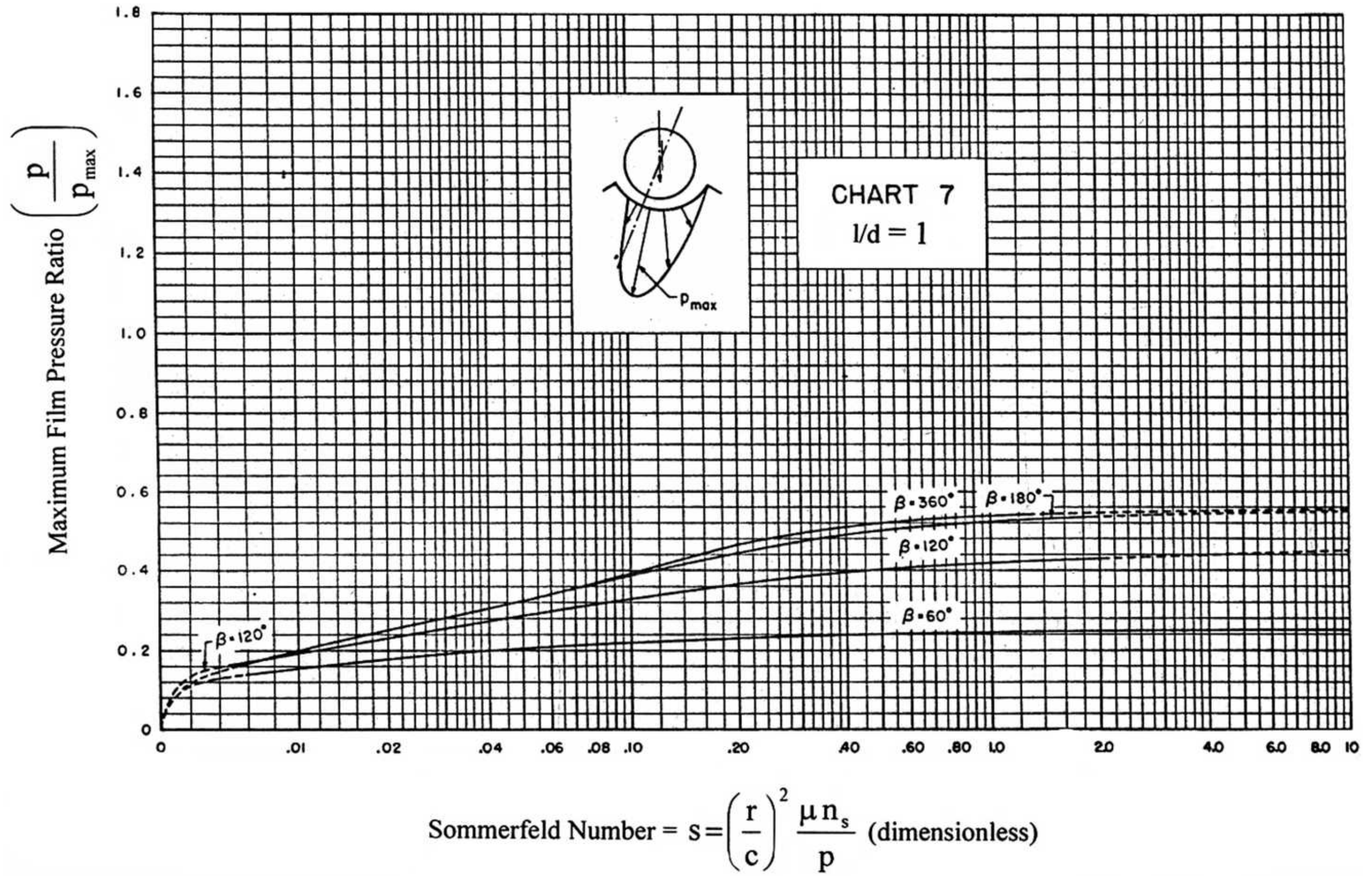


Chart 16.17 *Maximum film pressure ratio v/s Sommerfeld number (1/d = 1)*

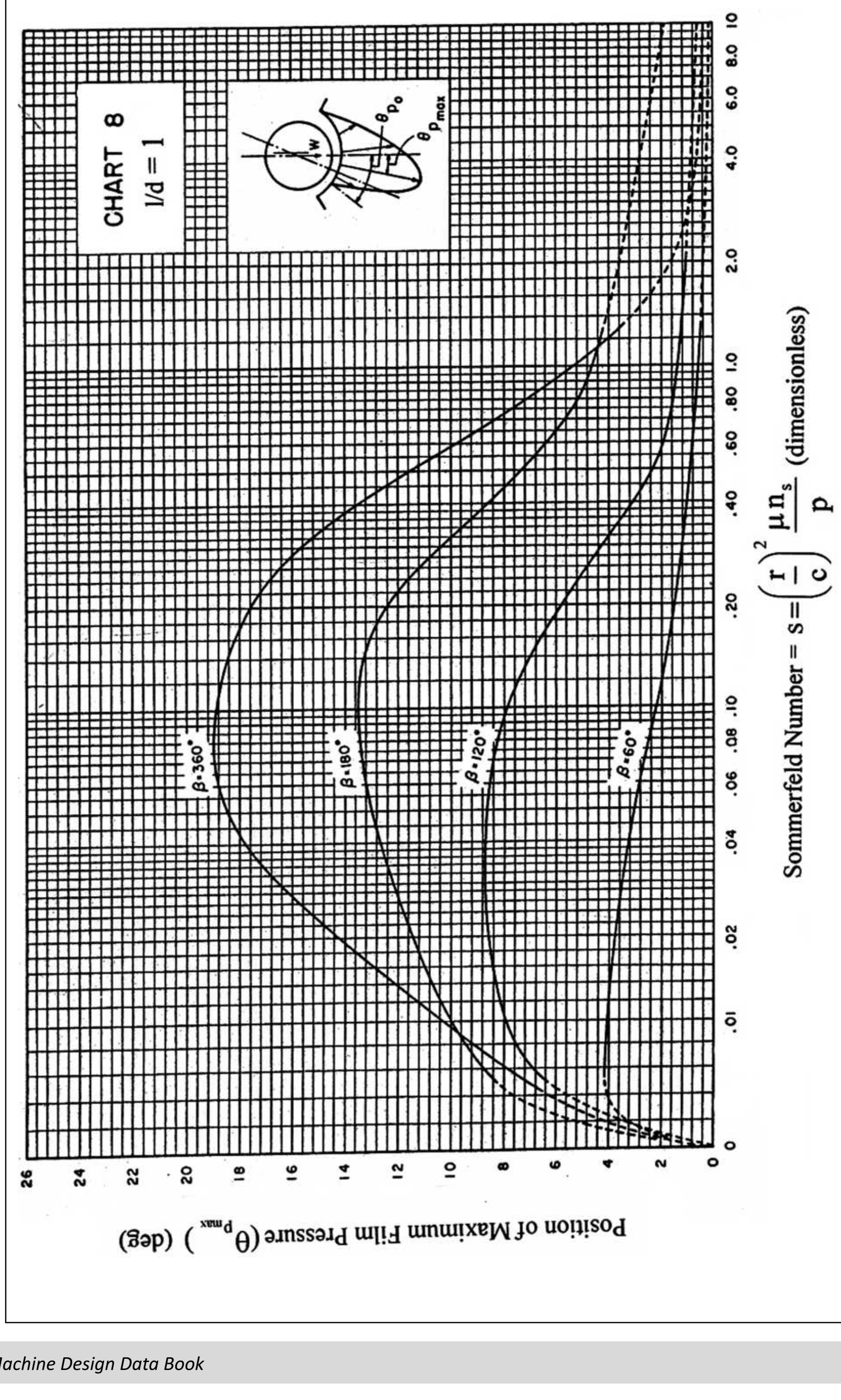
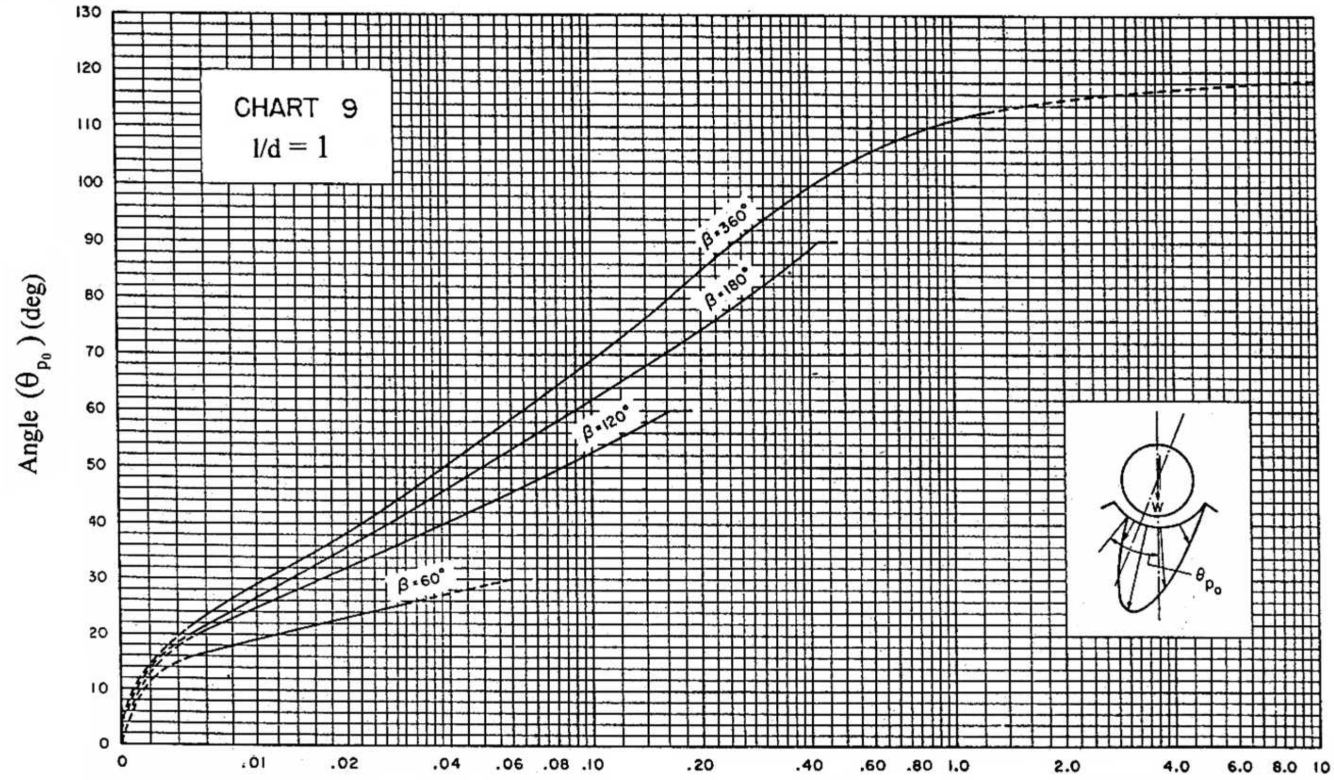


Chart 16.18 *Position of maximum film pressure v/s Sommerfeld number (l/d = 1)*



Sommerfeld Number = $S = \left(\frac{r}{c}\right)^2 \frac{\mu n_s}{p}$ (dimensionless)

Chart 16.19 *Film terminating angle v/s Sommerfeld number (1/d = 1)*

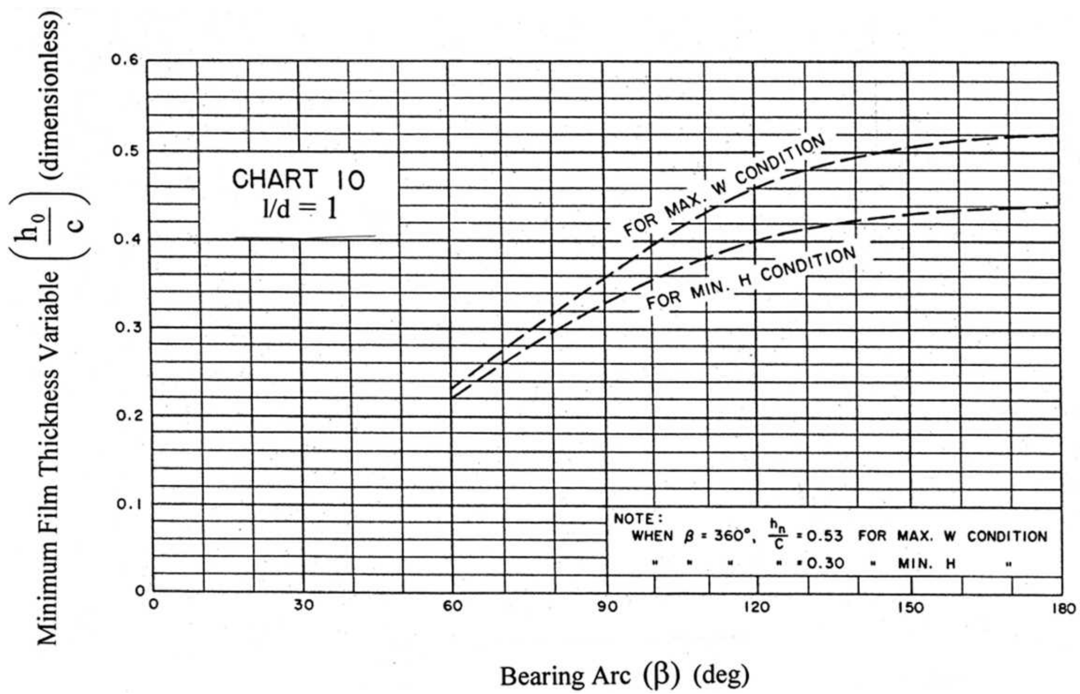


Chart 16.20 *Minimum film thickness variable v/s Bearing arc (Maximum load and Minimum power loss conditions) (1/d = 1)*

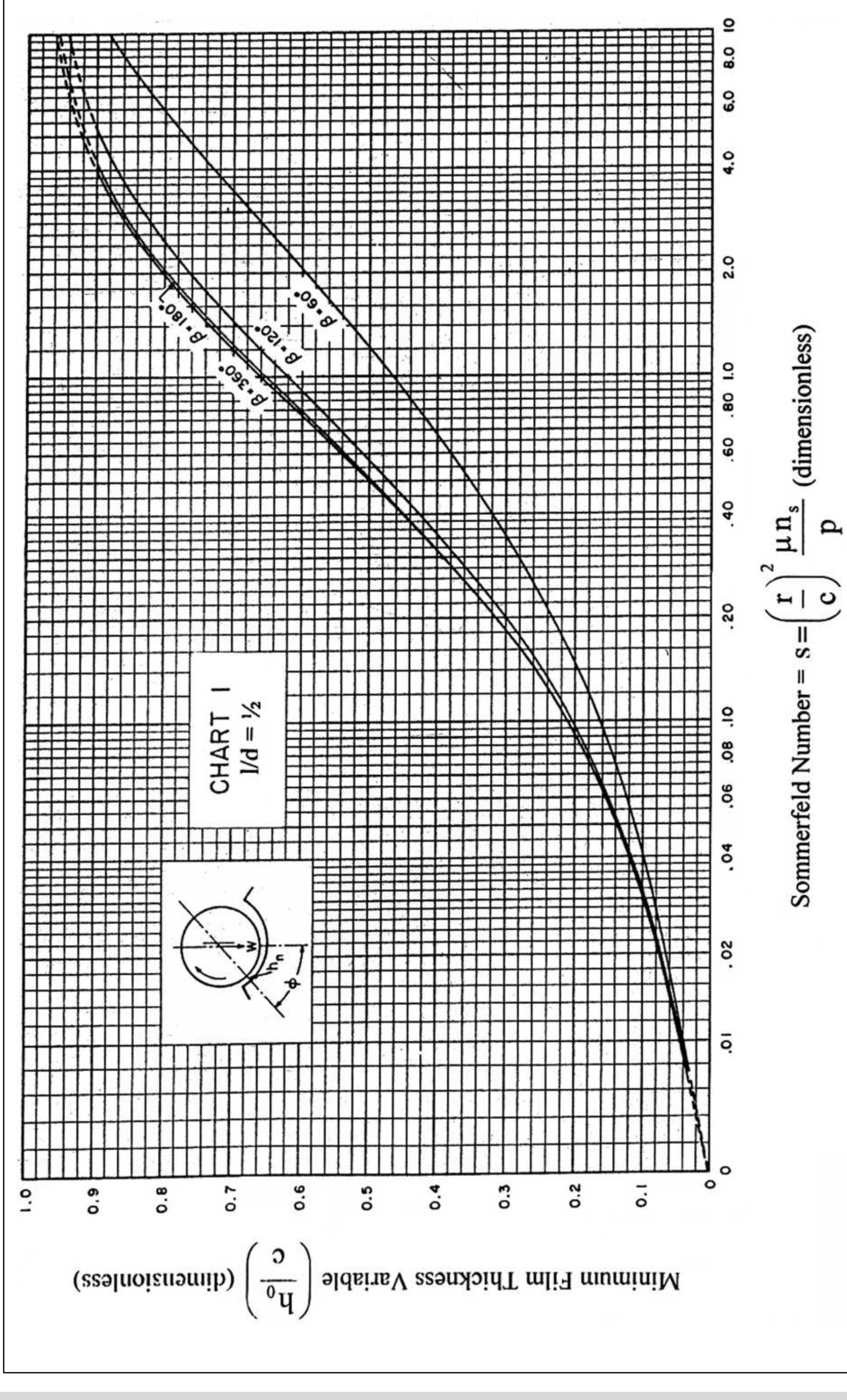
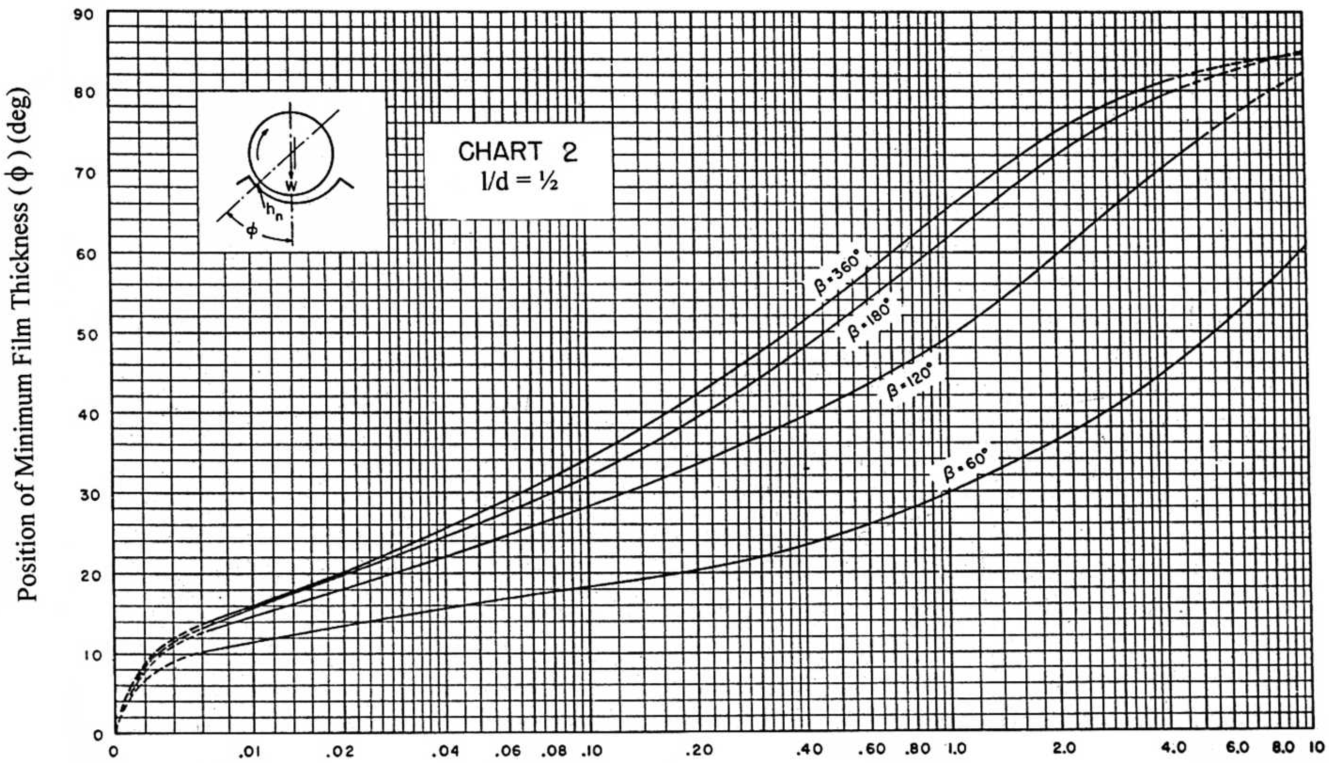


Chart 16.21 *Minimum film thickness variable v/s Sommerfeld number (1/d = 1/2)*



Sommerfeld Number = $S = \left(\frac{r}{c}\right)^2 \frac{\mu n_s}{p}$ (dimensionless)

Chart 16.22 *Position ot minimum film thickness v/s Sommerfeld number (1/d = 1/2)*

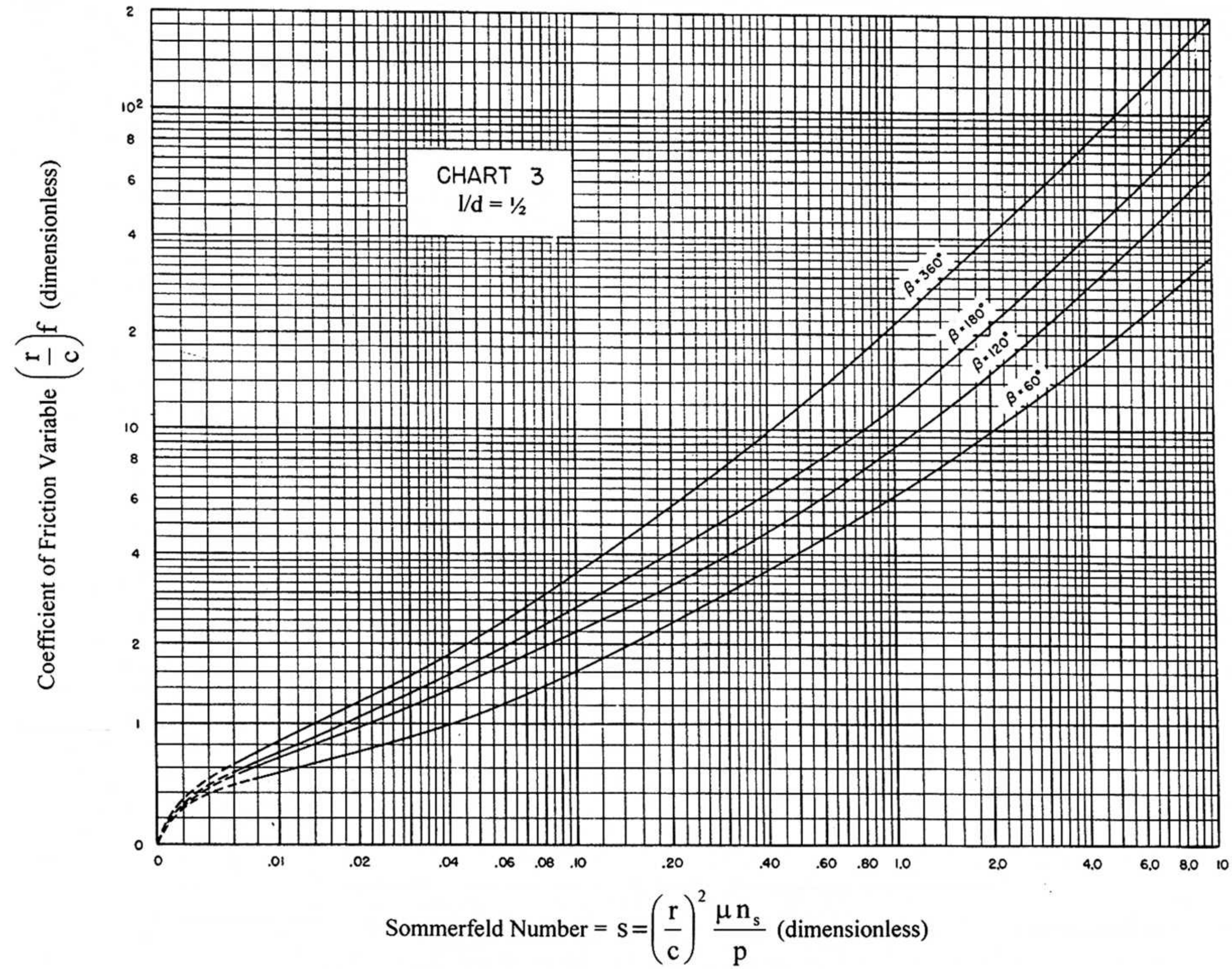


Chart 16.23 *Coefficient of friction variable v/s Sommerfeld number (1/d = 1/2)*

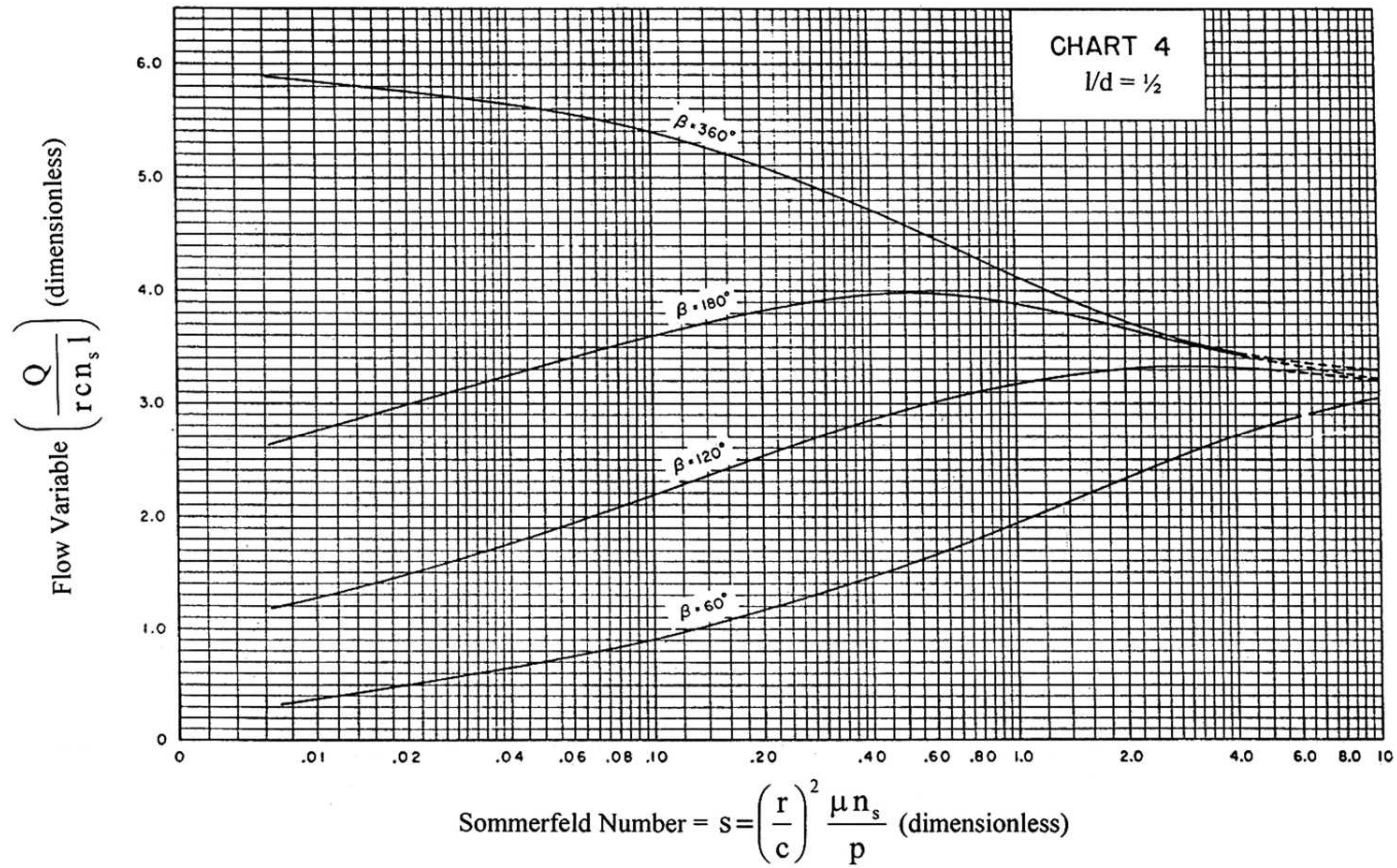


Chart 16.24 *Flow variable v/s Sommerfeld number (1/d = 1/2)*

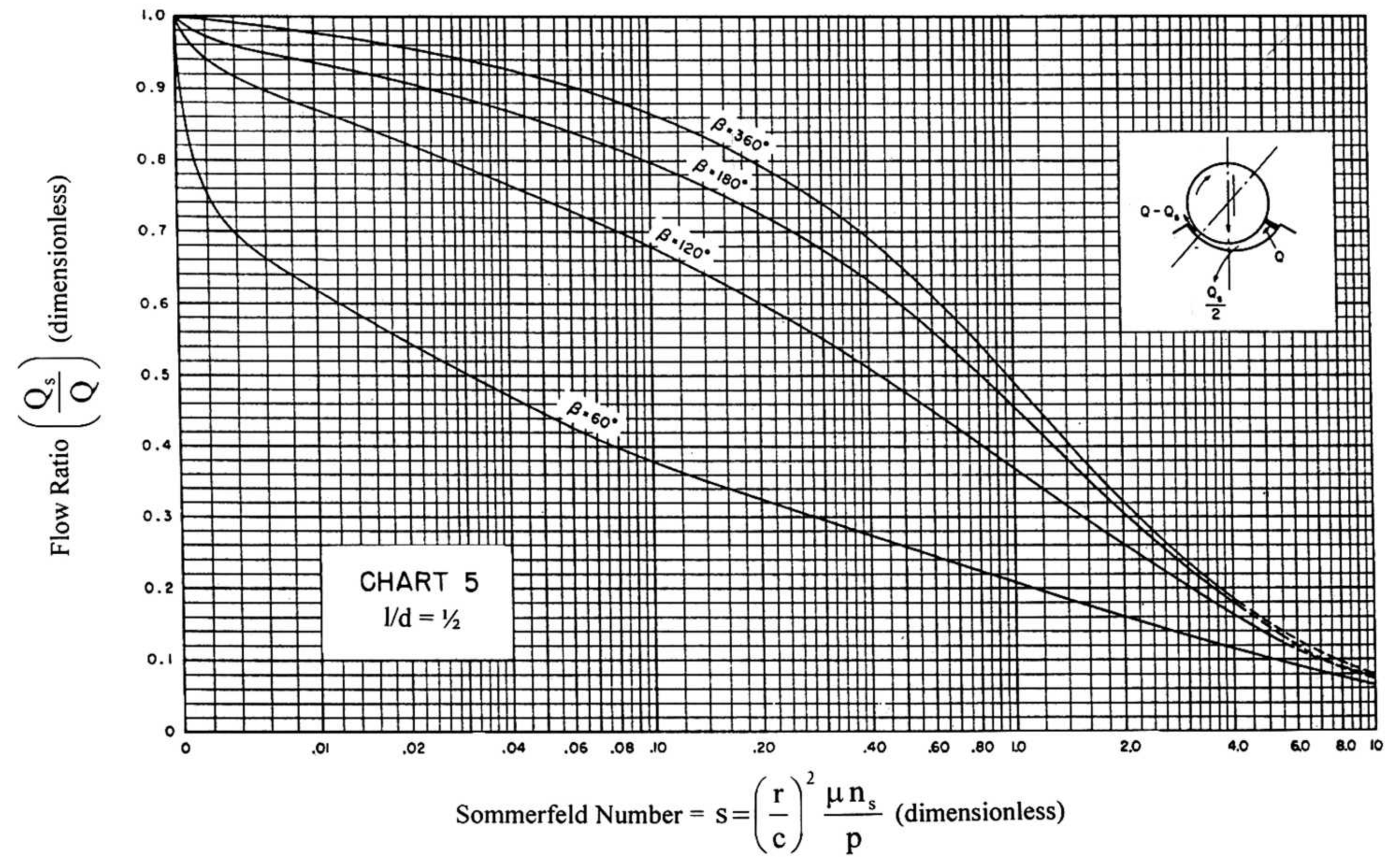


Chart 16.25 *Flow ratio v/s Sommerfeld number (1/d = 1/2)*

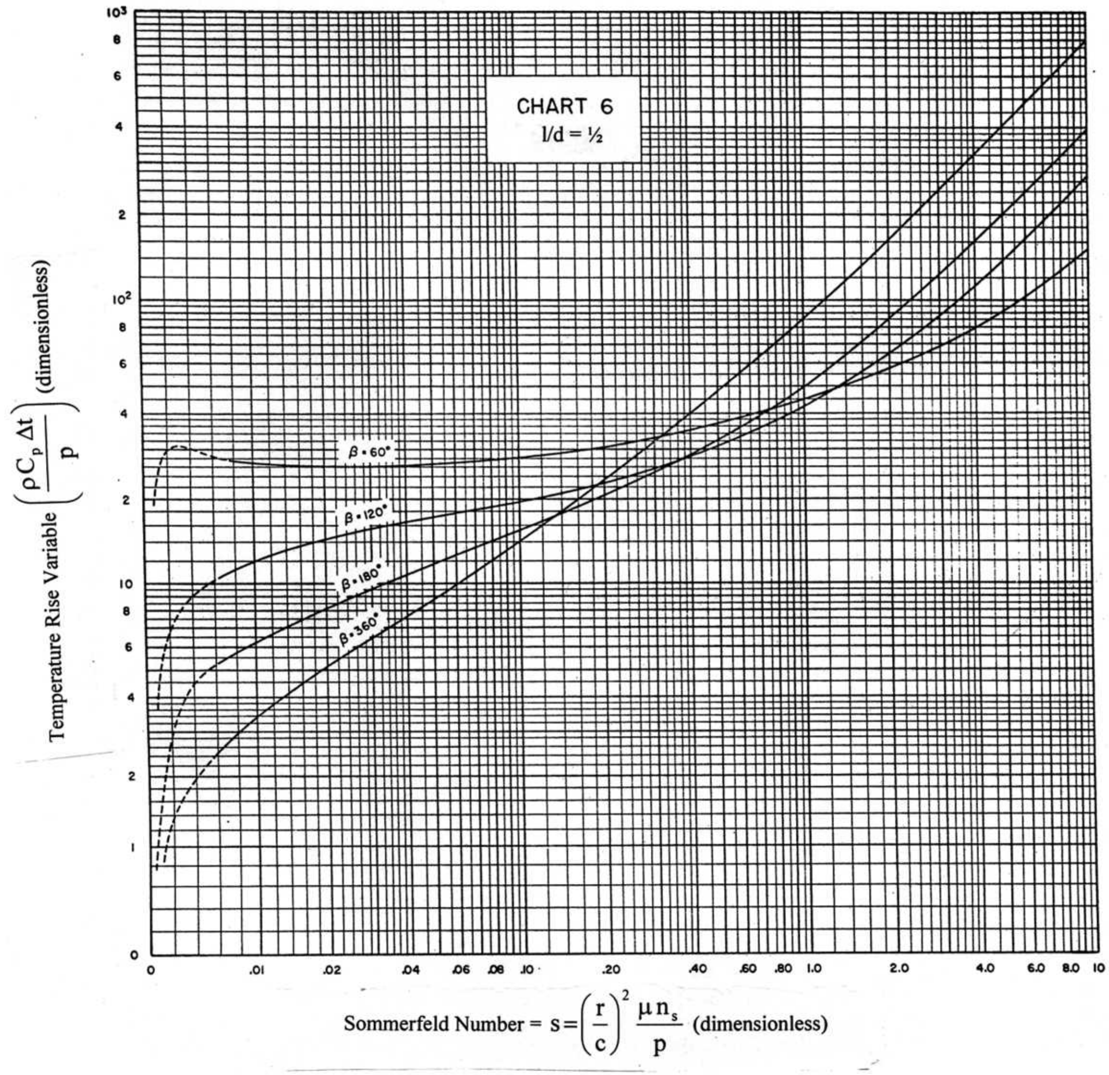
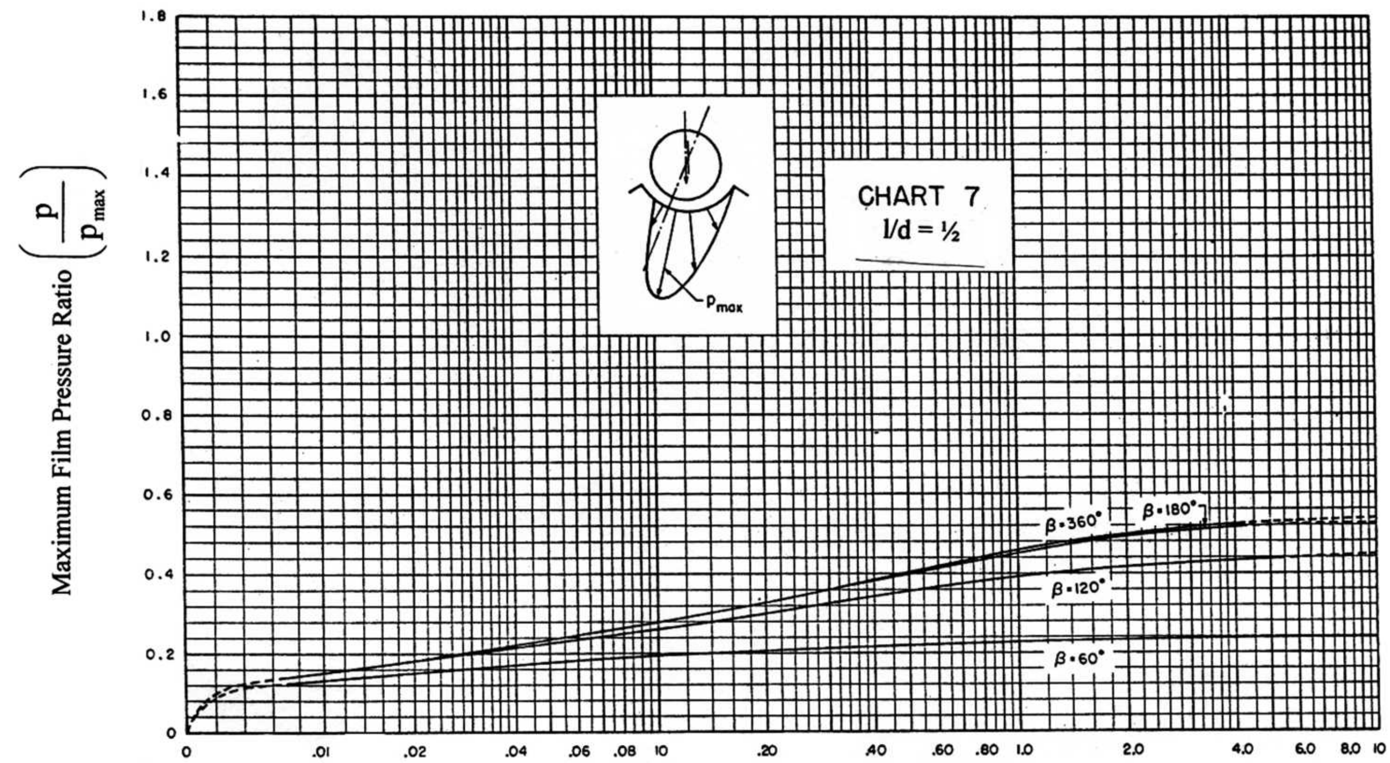


Chart 16.26 *Temperature rise variable v/s Sommerfeld number (1/d = 1/2)*



Sommerfeld Number = $S = \left(\frac{r}{c}\right)^2 \frac{\mu n_s}{p}$ (dimensionless)

Chart 16.27 *Maximum film pressure ratio v/s Sommerfeld number (1/d = 1/2)*

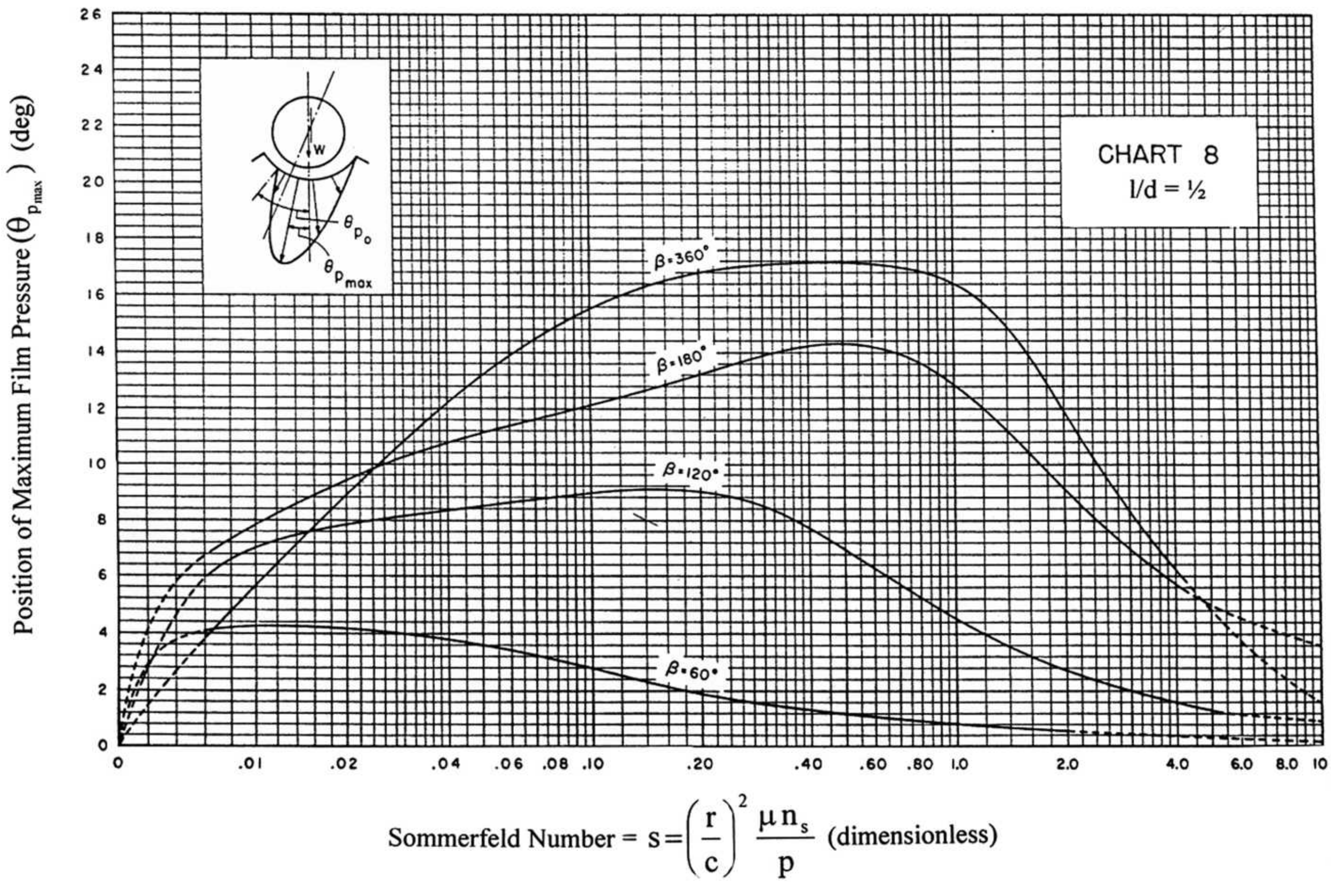


Chart 16.28 *Position of maximum film pressure v/s Sommerfeld number (1/d = 1/2)*

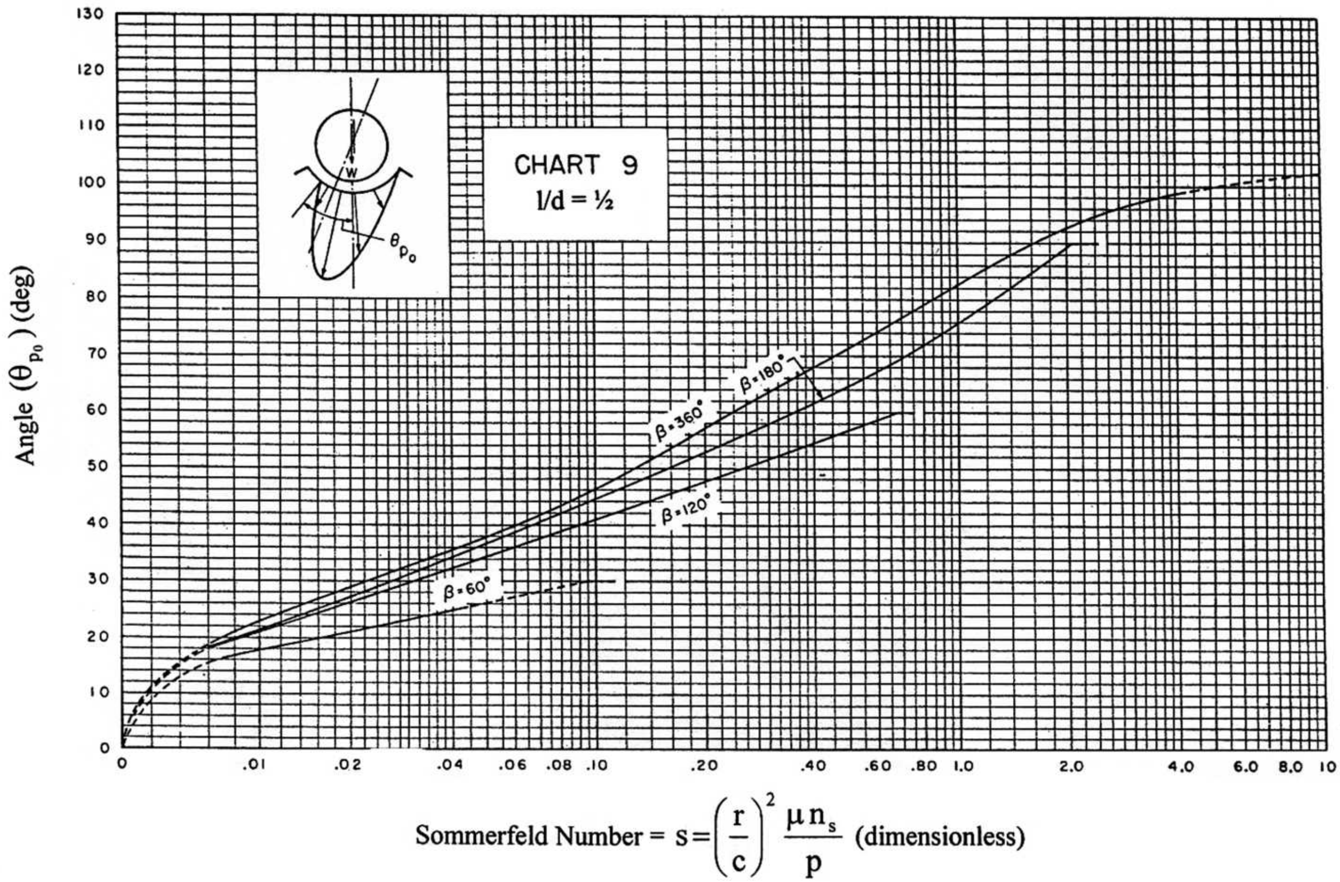


Chart 16.29 *Film terminating angle v/s Sommerfeld number (1/d = 1/2)*

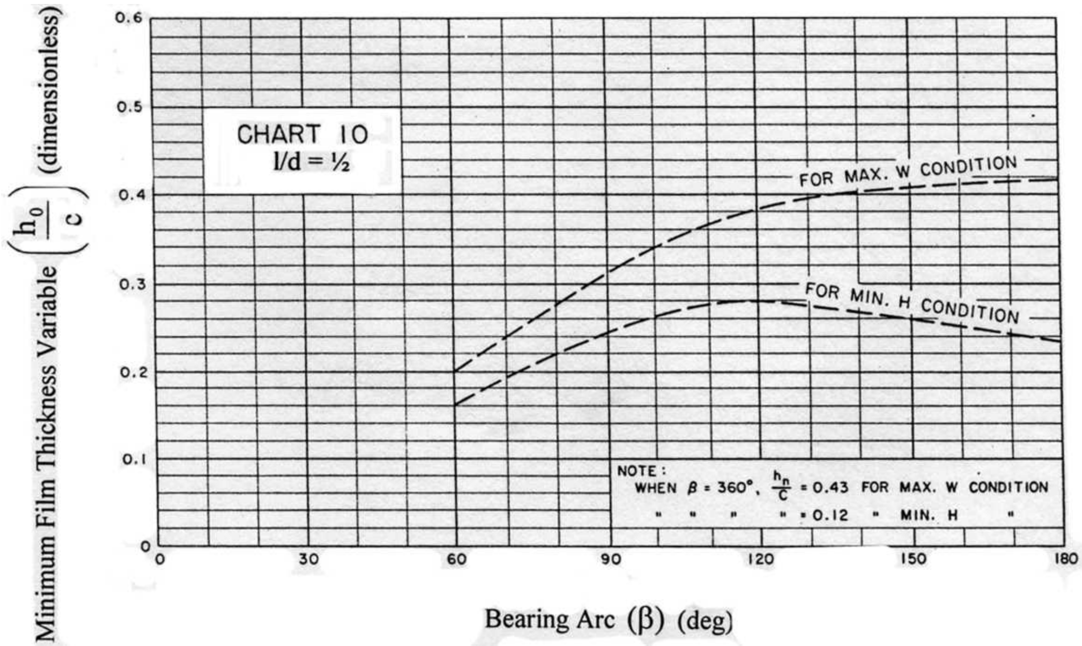


Chart 16.30 *Minimum film thickness variable v/s Bearing arc (Maximum load and Minimum power loss conditions (1/d = 1/2)*

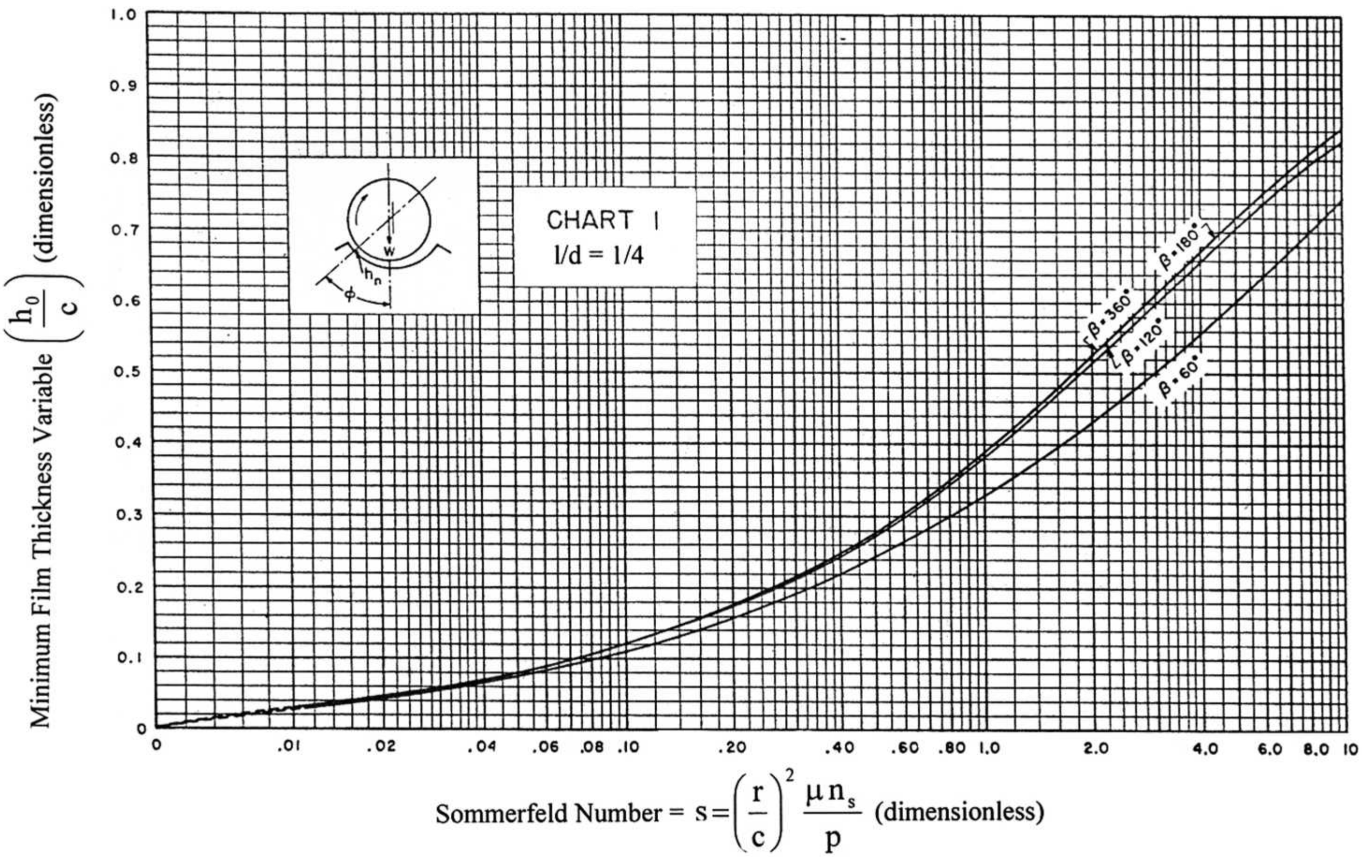
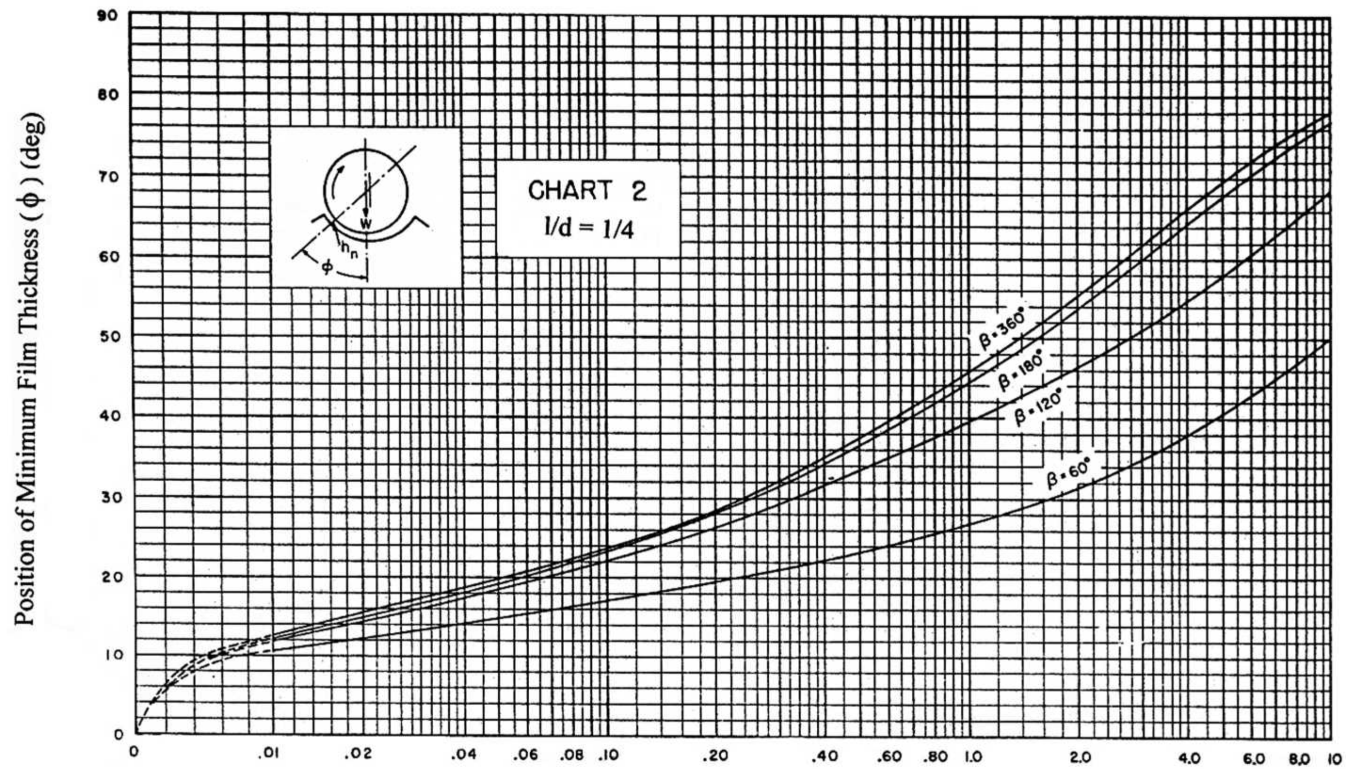


Chart 16.31 *Minimum film thickness variable v/s Sommerfeld number (1/d = 1/4)*



Sommerfeld Number = $s = \left(\frac{r}{c}\right)^2 \frac{\mu n_s}{p}$ (dimensionless)

Chart 16.32 *Position of minimum film thickness v/s Sommerfeld number (1/d = 1/4)*

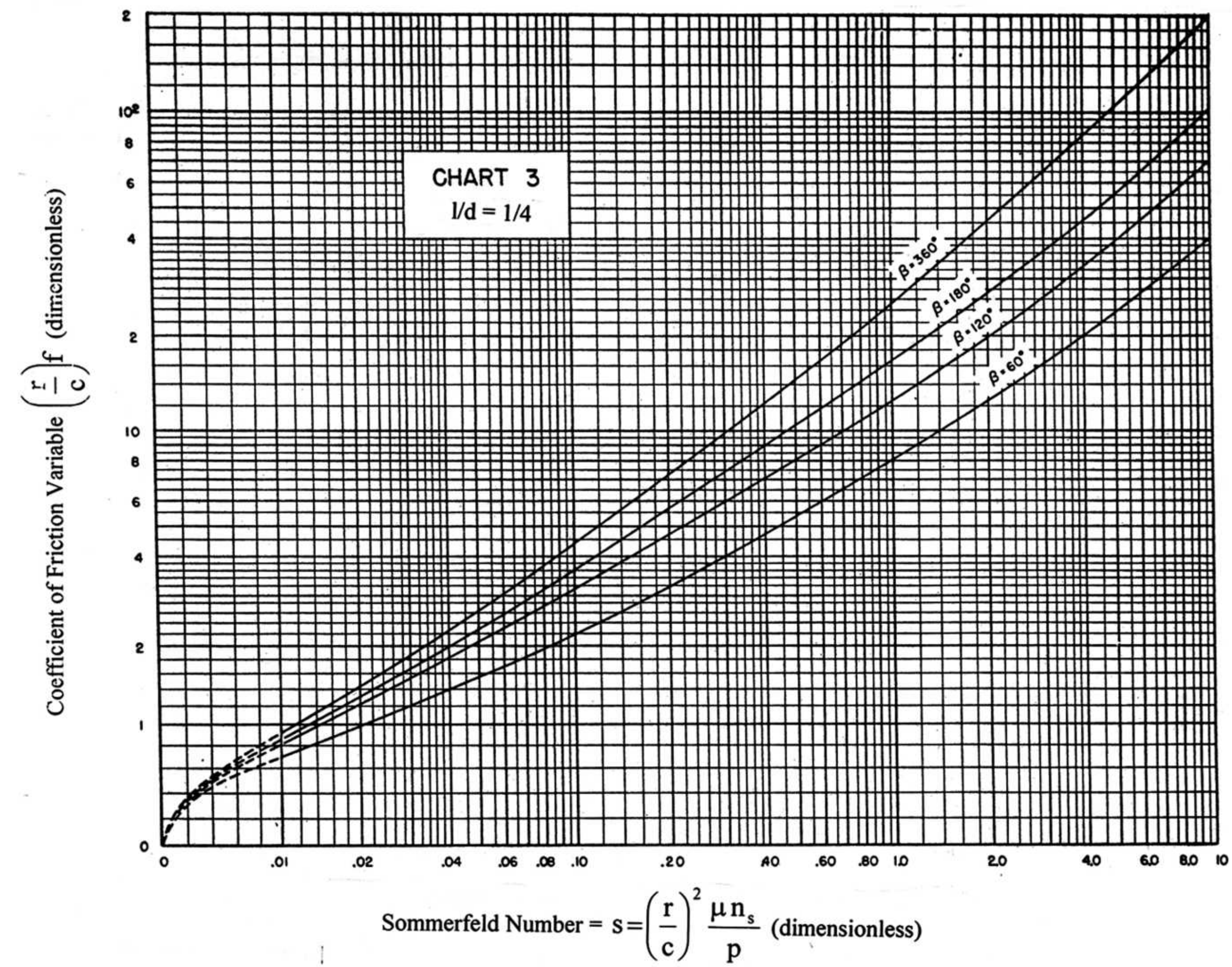


Chart 16.33 *Coefficient of friction variable v/s Sommerfeld number (1/d = 1/4)*

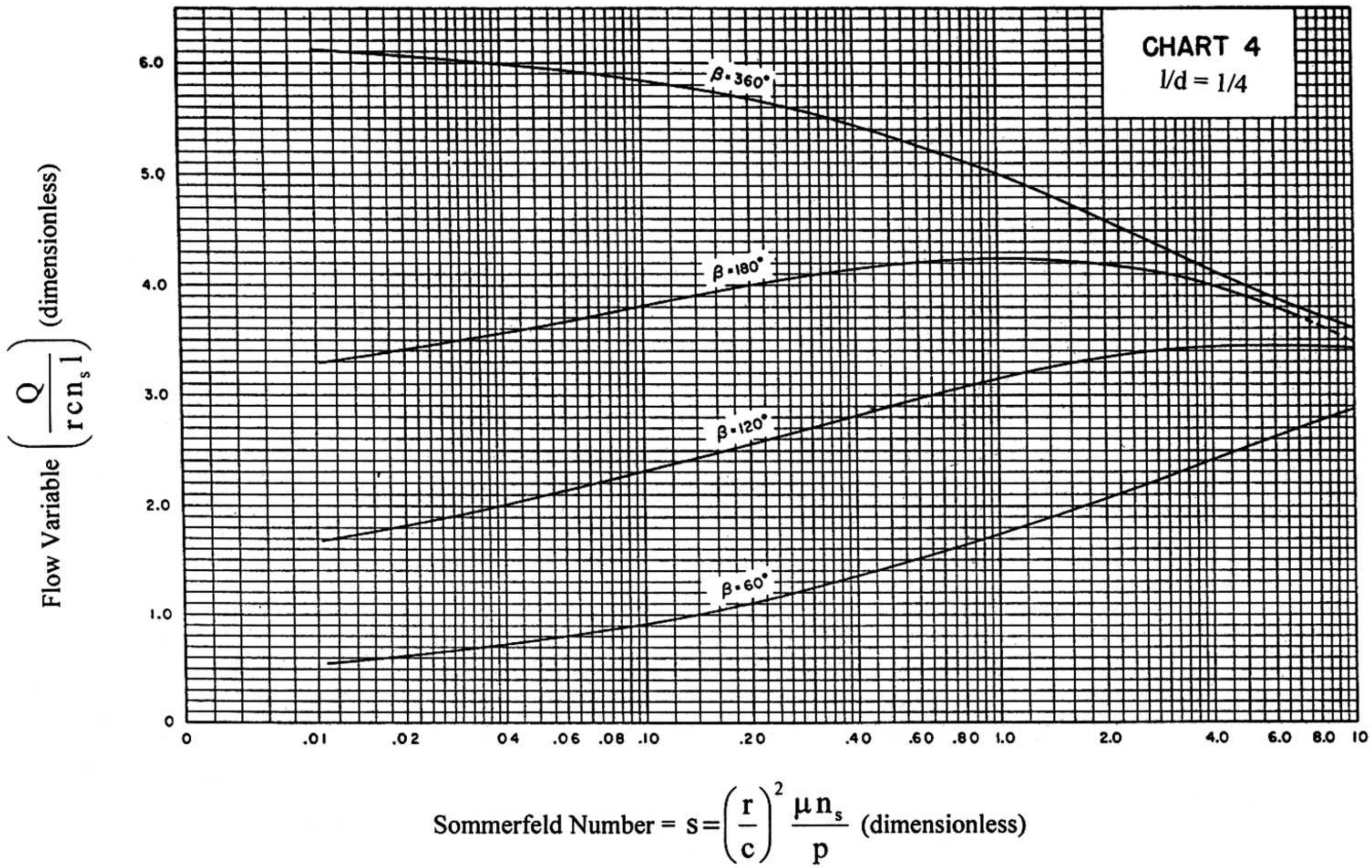


Chart 16.34 *Flow variable v/s Sommerfeld number (1/d = 1/4)*

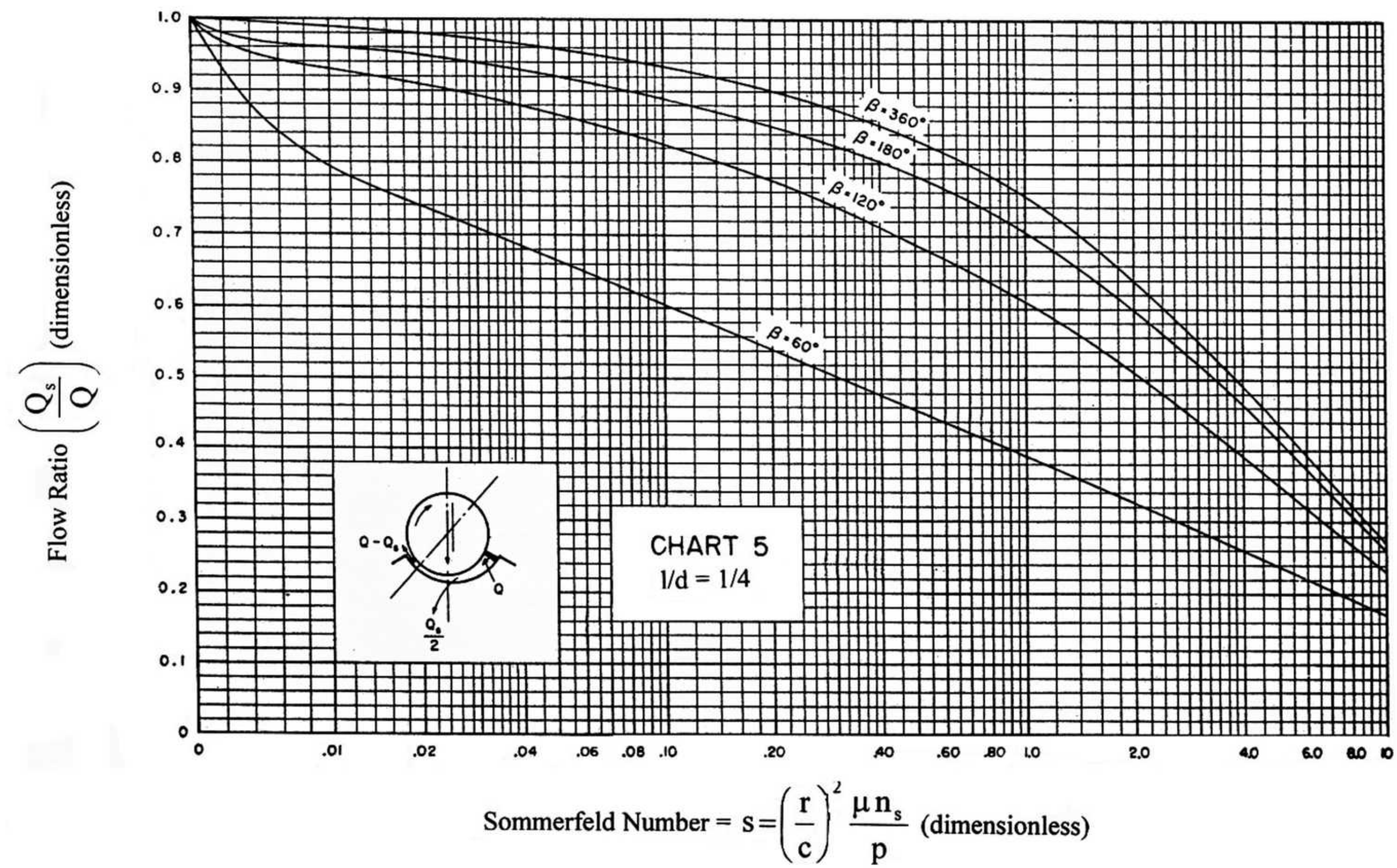


Chart 16.35 *Flow ratio v/s Sommerfeld number (1/d = 1/4)*

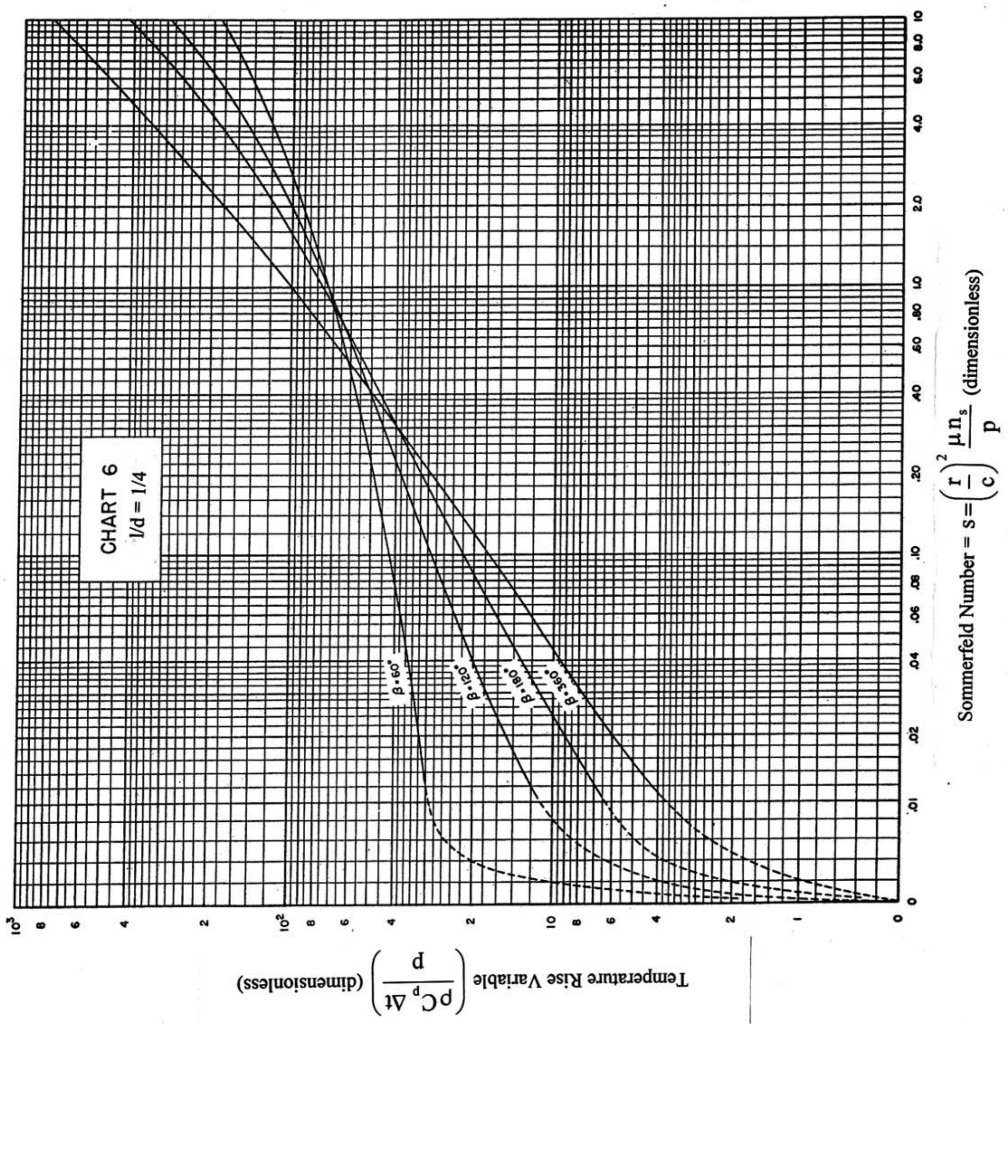


Chart 16.36 *Temperature rise variable v/s Sommerfeld number (1/d = 1/4)*

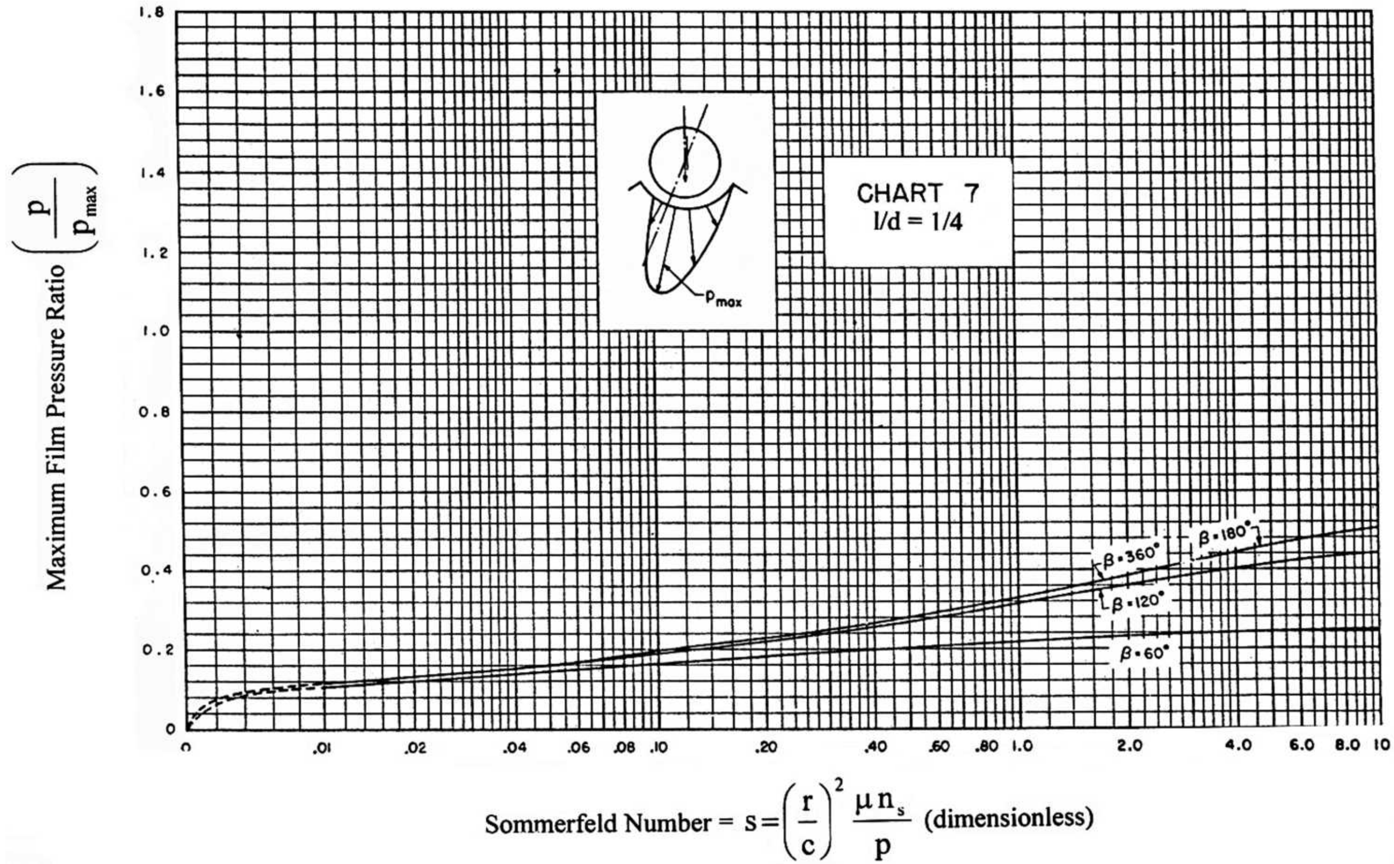


Chart 16.37 *Maximum film pressure ratio v/s Sommerfeld number (1/d = 1/4)*

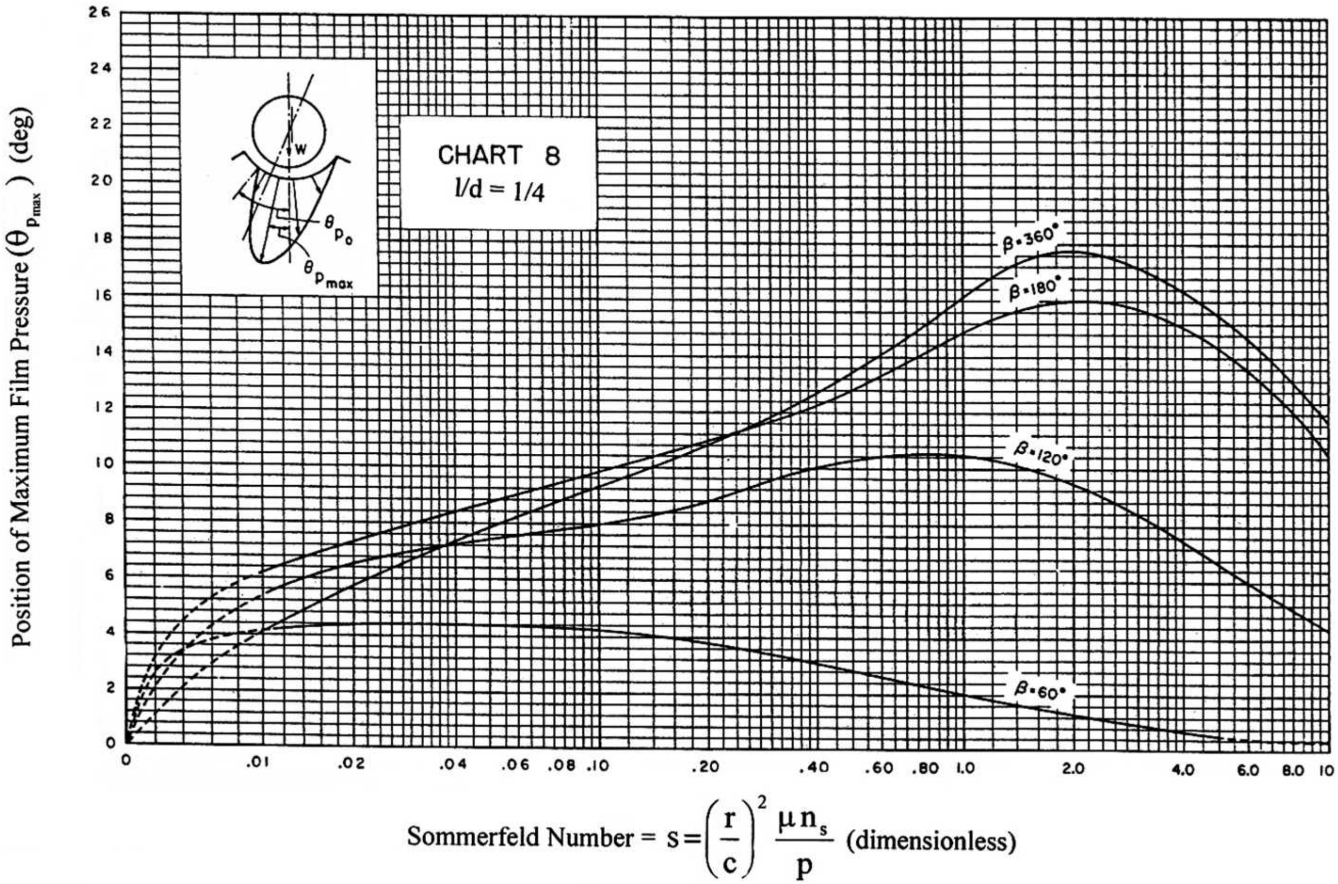


Chart 16.38 *Position of maximum film pressure v/s Sommerfeld number (1/d = 1/4)*

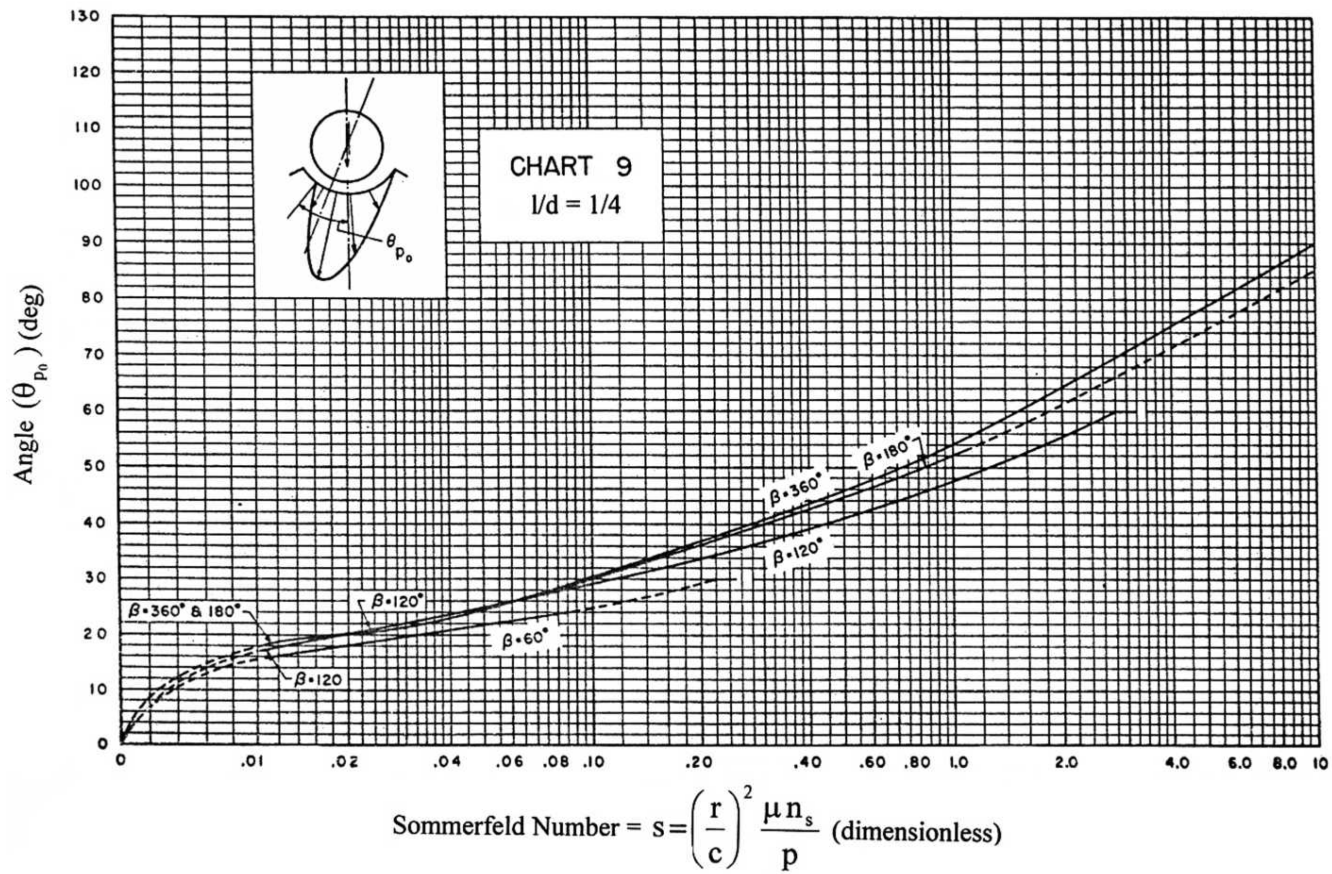


Chart 16.39 *Film terminating angle v/s Sommerfeld number (1/d = 1/4)*

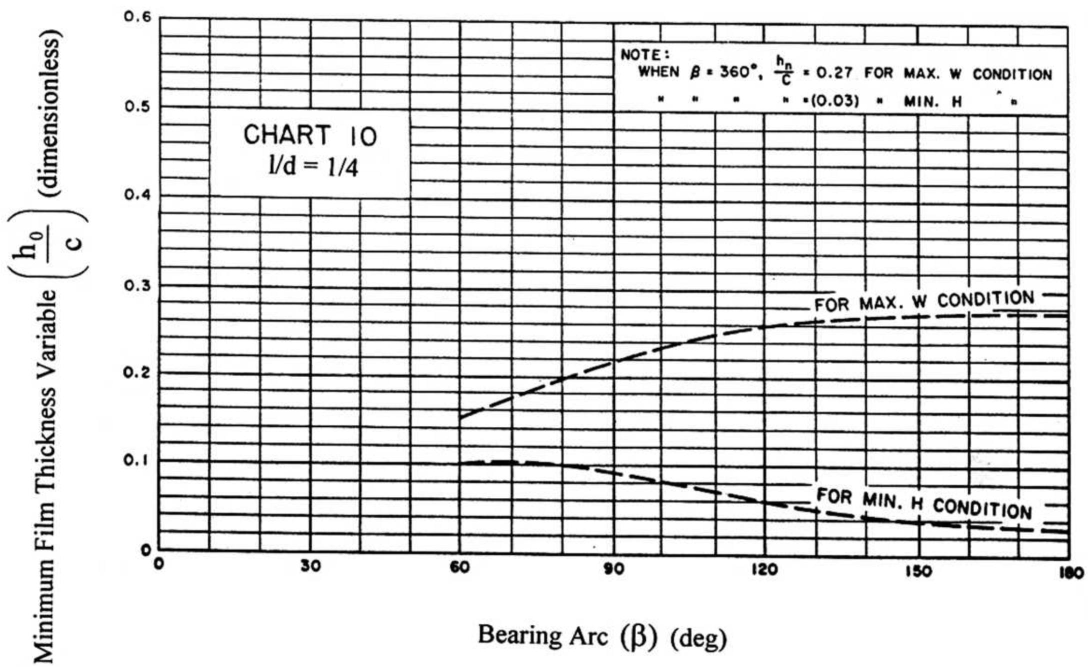


Chart 16.40 *Minimum film thickness variable v/s Bearing arc (Maximum load and Minimum power loss conditions) (1/d = 1/4)*

16.5 RAIMONDI AND BOYD TABLES

Table 16.6 *Performance characteristics for full journal bearings*

$\left(\frac{l}{d}\right)$	ε	$\left(\frac{h_o}{c}\right)$	θ_A	S	ϕ	$\left(\frac{r}{c}\right)f$	$\left(\frac{Q}{rcn_s l}\right)$	$\left(\frac{Q}{Q_s}\right)$	$\left(\frac{\rho C_p \Delta t}{p}\right)$	$\left(\frac{p}{p_{\max}}\right)$	$\theta_{p_{\max}}$	θ_{p_o}
∞	0	1.0	0	∞	(70.92)	∞	π	0	∞	—	0	(148.38)
	0.1	0.9	0	0.240	69.10	4.80	3.03	0	19.9	0.826	0.0	137
	0.2	0.8	0	0.123	67.26	2.57	2.83	0	11.4	0.814	5.6	128
	0.4	0.6	0	0.0626	61.94	1.52	2.26	0	8.47	0.764	14.4	107
	0.6	0.4	0	0.0389	54.31	1.20	1.56	0	9.73	0.667	20.8	86
	0.8	0.2	0	0.0210	42.22	0.961	0.760	0	15.9	0.495	21.5	58.8
	0.9	0.1	0	0.0115	31.62	0.756	0.411	0	23.1	0.358	19	44
	0.97	0.03	0	—	—	—	—	0	—	—	—	—
	1.00	0	0	0	0	0	0	0	∞	0	0	0
1	0	1.0	0	∞	(85)	∞	π	0	∞	—	0	(119)
	0.1	0.9	0	1.33	79.5	26.4	3.37	0.150	106	0.540	3.5	113
	0.2	0.8	0	0.631	74.02	12.8	3.59	0.280	52.1	0.529	9.2	106
	0.4	0.6	0	0.264	63.10	5.79	3.99	0.497	24.3	0.484	16.5	91.2
	0.6	0.4	0	0.121	50.58	3.22	4.33	0.680	14.2	0.415	18.7	72.9
	0.8	0.2	0	0.0446	36.24	1.70	4.62	0.842	8.00	0.313	18.2	52.3
	0.9	0.1	0	0.0188	26.45	1.05	4.74	0.919	5.16	0.247	13.8	37.3
	0.97	0.03	0	0.00474	15.47	0.514	4.82	0.973	2.61	0.152	7.1	20.5
	1.00	0	0	0	0	0	—	1.0	0	0	0	0
1/2	0	1.0	0	∞	(88.5)	∞	π	0	∞	—	0	(107)
	0.1	0.9	0	4.31	81.62	85.6	3.43	0.173	343	0.523	5.8	99.2
	0.2	0.8	0	2.03	74.94	40.9	3.72	0.318	164	0.506	11.9	92.5
	0.4	0.6	0	0.779	61.45	17.0	4.29	0.552	68.6	0.441	16.9	78.8
	0.6	0.4	0	0.319	48.14	8.10	4.85	0.730	33.0	0.365	17.1	64.3
	0.8	0.2	0	0.0923	33.31	3.26	5.41	0.874	13.4	0.267	15.3	44.2
	0.9	0.1	0	0.0313	23.66	1.60	5.69	0.939	6.66	0.206	11	33.8
	0.97	0.03	0	0.00609	13.75	0.610	5.88	0.980	2.56	0.126	3.8	19.1
	1.00	0	0	0	0	0	—	1.0	0	0	0	0

(*Contd.*)

(*Contd.*)

1/4	0	1.0	0	∞	(89.5)	∞	π	0	∞	—	0	(99)
	0.1	0.9	0	16.2	82.31	322.0	3.45	0.180	1287	0.515	7.4	98.9
	0.2	0.8	0	7.57	75.18	153.0	3.76	0.330	611	0.489	13.5	85
	0.4	0.6	0	2.83	60.86	61.1	4.37	0.567	245	0.415	17.4	70
	0.6	0.4	0	1.07	46.72	26.7	4.99	0.746	107	0.334	16.4	55.5
	0.8	0.2	0	0.261	31.04	8.80	5.60	0.884	35.4	0.240	11.5	39.7
	0.9	0.1	0	0.0736	21.85	3.50	5.91	0.945	14.1	0.180	8.6	27.8
	0.97	0.03	0	0.0101	12.22	0.922	6.12	0.984	3.73	0.108	4	17.7
	1.00	0	0	0	0	0	—	1.0	0	0	0	0

(A.S.L.E. Transactions)

Table 16.7 *Performance characteristics for 180° bearings—Centrally loaded*

$\left(\frac{l}{d}\right)$	ε	$\left(\frac{h_o}{c}\right)$	θ_A	S	ϕ	$\left(\frac{r}{c}\right)f$	$\left(\frac{Q}{rcn_s l}\right)$	$\left(\frac{Q}{Q_s}\right)$	$\left(\frac{\rho C_p \Delta t}{p}\right)$	$\left(\frac{p}{p_{\max}}\right)$	$\theta_{p_{\max}}$	θ_{p_o}
∞	0	1.0	0	∞	90	∞	π	0	∞	—	0	90
	0.1	0.9	17.000	0.347	72.90	3.55	3.04	0	14.7	0.778	1.6	90
	0.2	0.8	28.600	0.179	61.32	2.01	2.80	0	8.99	0.759	3.4	90
	0.4	0.6	40.000	0.0898	49.99	1.29	2.20	0	7.34	0.700	7.7	90
	0.6	0.4	46.900	0.0523	43.15	1.06	1.52	0	8.71	0.607	12.3	71.8
	0.8	0.2	56.700	0.0253	33.35	0.859	0.767	0	14.1	0.459	13.7	51.2
	0.9	0.1	64.200	0.0128	25.57	0.681	0.380	0	22.5	0.337	13.3	36.8
	0.97	0.03	74.650	0.00384	15.43	0.416	0.119	0	44.0	0.190	9.1	20.7
	1.00	0	90	0	0	0	0	0	∞	0	0	0
1	0	1.0	0	∞	90	∞	π	0	∞	—	0	90
	0.1	0.9	11.500	1.40	78.50	14.1	3.34	0.139	57.0	0.525	4.5	90
	0.2	0.8	21.000	0.670	68.93	7.15	3.46	0.252	29.7	0.513	5.9	90
	0.4	0.6	34.167	0.278	58.86	3.61	3.49	0.425	16.5	0.466	10.8	80.8
	0.6	0.4	45.000	0.128	44.67	2.28	3.25	0.572	12.4	0.403	13.4	66.8
	0.8	0.2	58.000	0.0463	32.33	1.39	2.63	0.721	10.4	0.313	12.9	48.5
	0.9	0.1	66.000	0.0193	24.14	0.921	2.14	0.818	9.13	0.244	11.3	35.2
	0.97	0.03	75.584	0.00483	14.57	0.483	1.60	0.915	6.96	0.157	8.2	20
	1.00	0	90	0	0	0	—	1.000	0	0	0	0

(*Contd.*)

(*Contd.*)

1/2	0	1.0	0	∞	90	∞	π	0	∞	—	0	90
	0.1	0.9	10.000	4.38	79.97	44.0	3.41	0.167	177	0.518	5.4	90
	0.2	0.8	17.800	2.06	72.14	21.6	3.64	0.302	87.8	0.499	9.1	90
	0.4	0.6	32.000	0.794	58.01	9.96	3.93	0.506	42.7	0.438	13.7	71.9
	0.6	0.4	45.000	0.321	45.01	5.41	3.93	0.665	25.9	0.365	14.1	59.1
	0.8	0.2	59.000	0.0921	31.29	2.54	3.56	0.806	15.0	0.273	12.1	43.9
	0.9	0.1	67.200	0.0314	22.80	1.38	3.17	0.886	9.80	0.208	10.4	31.5
	0.97	0.03	76.500	0.00625	13.63	0.581	2.62	0.951	5.30	0.132	6.9	18.1
	1.00	0	90	0	0	0	—	1.000	0	0	0	0
1/4	0	1.0	0	∞	90	∞	π	0	∞	—	0	90
	0.1	0.9	9.000	16.3	81.40	163	3.44	0.176	653	0.513	6.3	90
	0.2	0.8	16.300	7.60	73.70	79.4	3.71	0.320	320	0.489	12.4	80.9
	0.4	0.6	31.000	2.84	58.99	35.1	4.11	0.534	146	0.417	15.8	70.5
	0.6	0.4	45.000	1.08	44.96	17.6	4.25	0.698	79.8	0.336	14.8	53.6
	0.8	0.2	59.300	0.263	30.43	6.88	4.07	0.837	36.5	0.241	11.4	39
	0.9	0.1	68.900	0.0736	21.43	2.99	3.72	0.905	18.4	0.180	9.3	27.3
	0.97	0.03	77.680	0.0104	12.28	0.877	3.29	0.961	6.46	0.110	6.3	17.5
	1.00	0	90	0	0	0	—	1.000	0	0	0	0

(A.S.L.E. Transactions)

Table 16.8 *Performance characteristics for 120° bearings—Centrally loaded*

$\left(\frac{l}{d}\right)$	ε	$\left(\frac{h_o}{c}\right)$	θ_A	S	ϕ	$\left(\frac{r}{c}\right)f$	$\left(\frac{Q}{rcn_s l}\right)$	$\left(\frac{Q}{Q_s}\right)$	$\left(\frac{\rho C_p \Delta t}{p}\right)$	$\left(\frac{p}{p_{\max}}\right)$	$\theta_{p_{\max}}$	θ_{p_o}
∞	0	1.0	30	∞	90	∞	π	0	∞	—	0	60
	0.1	0.9007	53.300	0.877	66.69	6.02	3.02	0	25.1	0.610	0.4	60
	0.2	0.8	67.400	0.431	52.60	3.26	2.75	0	14.9	0.599	0.9	60
	0.4	0.6	81.000	0.181	39.02	1.78	2.13	0	10.5	0.566	2.4	60
	0.6	0.4	87.300	0.0845	32.67	1.21	1.47	0	10.3	0.509	5.1	60
	0.8	0.2	93.200	0.0328	26.80	0.853	0.759	0	14.1	0.405	8.2	44.2
	0.9	0.1	98.500	0.0147	21.51	0.653	0.388	0	21.2	0.311	8.7	32.9
	0.97	0.03	106.15	0.00406	13.86	0.399	0.118	0	42.4	0.199	6.6	19.6
	1.00	0	120	0	0	0	0	0	∞	0	0	0

(*Contd.*)

(*Contd.*)

1	0	1.0	30	∞	90	∞	π	0	∞	—	0	60
	0.1	0.9024	47.500	2.14	72.43	14.5	3.20	0.0876	59.5	0.427	1.1	60
	0.2	0.8	62.000	1.01	58.25	7.44	3.11	0.157	32.6	0. 420	1.3	60
	0.4	0.6	76.000	0.385	43.98	3.60	2.75	0.272	19.0	0.396	3.2	60
	0.6	0.4	84.500	0.162	35.65	2.16	2.24	0.384	15.0	0.356	6.5	60
	0.8	0.2	92.600	0.0531	27.42	1.27	1.57	0.535	13.9	0.290	8.6	43.6
	0.9	0.1	98.667	0.0208	21.29	0.855	1.11	0.657	14.4	0.233	8.5	32.5
	0.97	0.03	106.50	0.00498	13.49	0.461	0.694	0.812	14.0	0.162	6.3	19.3
	1.00	0	120	0	0	0	—	1.0	0	0	0	0
1/2	0	1.0	30	∞	90	∞	π	0	∞	—	0	60
	0.1	0.9034	45.000	5.42	74.99	36.6	3.29	0.124	149	0.431	1.2	60
	0.2	0.8003	56.650	2.51	63.38	18.1	3.32	0.225	77.2	0.424	2.4	60
	0.4	0.6	72.000	0.914	48.07	8.20	3.15	0.386	40.5	0.389	4.8	60
	0.6	0.4	81.500	0.354	38.50	4.43	2.80	0. 530	27.0	0.336	8.1	53.4
	0.8	0.2	92.000	0.0973	28.02	2.17	2.18	0.684	19.0	0.261	9.0	40.5
	0.9	0.1	99.000	0.0324	21.02	1.24	1.70	0.787	15.1	0.203	8.2	30.4
	0.97	0.03	107.00	0.00631	13.00	0.550	1.19	0.899	10.6	0.136	6.0	18.2
	1.00	0	120	0	0	0	—	1.0	0	0	0	0
1/4	0	1.0	30	∞	90	∞	π	0	∞	—	0	60
	0.1	0.9044	43.000	18.4	76.97	124	3.34	0.147	502	0.456	3.0	60
	0.2	0.8011	54.000	8.45	65.97	60.4	3.44	0.260	254	0.438	4.8	60
	0.4	0.6	68.833	3.04	51.23	26.6	3.42	0.442	125	0.389	8.4	60
	0.6	0.4	79.600	1.12	40.42	13.5	3.20	0.599	75.8	0.321	10.4	48.3
	0.8	0.2	91.560	0.268	28.38	5.65	2.67	0.753	42.7	0.237	9.4	35.8
	0.9	0.1	99.400	0.0743	20.55	2.63	2.21	0.846	25.9	0.178	7.8	26.9
	0.97	0.03	108.00	0.0105	12.11	0.832	1.69	0.931	11.6	0.112	5.5	17.4
	1.00	0	120	0	0	0	—	1.0	0	0	0	0

(A.S.L.E. Transactions)

Table 16.9 *Performance characteristics for 60° bearings—Centrally loaded*

$\left(\frac{l}{d}\right)$	ε	$\left(\frac{h_o}{c}\right)$	θ_A	S	ϕ	$\left(\frac{r}{c}\right)f$	$\left(\frac{Q}{rcn_s l}\right)$	$\left(\frac{Q}{Q_s}\right)$	$\left(\frac{\rho C_p \Delta t}{p}\right)$	$\left(\frac{p}{p_{\max}}\right)$	$\theta_{p_{\max}}$	θ_{p_o}
∞	0	1.0	60	∞	90	∞	π	0	∞	—	0	30
	0.1	0.9191	84.00	5.75	65.91	19.7	3.01	0	82.3	0.337	0.16	30
	0.2	0.8109	101.00	2.66	48.91	10.1	2.73	0	46.5	0.336	0.18	30
	0.4	0.6002	118.00	0.931	31.96	4.67	2.07	0	28.4	0.329	0.25	30
	0.6	0.4	126.80	0.322	23.21	2.40	1.40	0	21.5	0.317	0.54	30
	0.8	0.2	132.60	0.0755	17.39	1.10	0.722	0	19.2	0.287	1.7	30
	0.9	0.1	135.06	0.0241	14.94	0.667	0.372	0	22.5	0.243	3.2	25.5
	0.97	0.03	139.14	0.00495	10.88	0.372	0.115	0	40.7	0.163	4.2	16.9
	1.00	0	150	0	0	0	0	0	∞	0	0	0
1	0	1.0	60	∞	90	∞	π	0	∞	—	0	30
	0.1	0.9212	82.00	8.52	67.92	29.1	3.07	0.0267	121	0.252	0.3	30
	0.2	0.8133	99.00	3.92	50.96	14.8	2.82	0.0481	67.4	0.251	0.3	30
	0.4	0.6010	116.00	1.34	33.99	6.61	2.22	0.0849	39.1	0.247	0.54	30
	0.6	0.4	125.50	0.450	24.56	3.29	1.56	0.127	28.2	0.239	0.95	30
	0.8	0.2	131.60	0.101	18.33	1.42	0.883	0.200	22.5	0.220	2.2	30
	0.9	0.1	134.67	0.0309	15.33	0.822	0.519	0.287	23.2	0.192	3.5	25.9
	0.97	0.03	139.10	0.00584	10.88	0.422	0.226	0.465	30.5	0.139	4.2	16.9
	1.00	0	150	0	0	0	—	1.0	0	0	0	0
1/2	0	1.0	60	∞	90	∞	π	0	∞	—	0	30
	0.1	0.9223	81.00	14.2	69.00	48.6	3.11	0.0488	201	0.239	0	30
	0.2	0.8152	97.50	6.47	52.60	24.2	2.91	0.0883	109	0.239	0.03	30
	0.4	0.6039	113.00	2.14	37.00	10.3	2.38	0.160	59.4	0.233	0.45	30
	0.6	0.4	123.00	0.695	26.98	4.93	1.74	0.236	40.3	0.225	1.0	30
	0.8	0.2	130.40	0.149	19.57	2.02	1.05	0.350	29.4	0.201	2.2	30
	0.9	0.1	134.09	0.0422	15.91	1.08	0.664	0.464	26.5	0.172	3.8	25.4
	0.97	0.03	139.22	0.00704	10.85	0.490	0.329	0.650	27.8	0.122	4.2	16.6
	1.00	0	150	0	0	0	—	1.0	0	0	0	0

(*Contd.*)

(*Contd.*)

1/4	0	1.0	60	∞	90	∞	π	0	∞	—	0	30
	0.1	0.9251	78.50	35.8	71.55	121	3.16	0.0666	499	0.251	0	30
	0.2	0.8242	91.50	16.0	58.51	58.7	3.04	0.131	260	0.249	0.1	30
	0.4	0.6074	109.00	5.20	41.01	24.5	2.57	0.236	136	0.242	0.5	30
	0.6	0.4	119.80	1.65	30.14	11.2	1.98	0.346	86.1	0.228	1.5	30
	0.8	0.2	128.30	0.333	21.70	4.27	1.30	0.496	54.9	0.195	3.2	30
	0.9	0.1	133.10	0.0844	16.87	2.01	0.894	0.620	41.0	0.159	4.3	23.7
	0.97	0.03	139.20	0.0110	10.81	0.713	0.507	0.786	29.1	0.107	4.1	15.9
	1.00	0	150	0	0	0	—	1.0	0	0	0	0

(A.S.L.E. Transactions)

Note: According to Raimondi and Boyd method, interpolation of data from charts and tables for other (l/d) ratios can be done using following equation:

$$y = \frac{1}{(l/d)^3}\left[-\frac{1}{8}\left(1-\frac{l}{d}\right)\left(1-2\frac{l}{d}\right)\left(1-4\frac{l}{d}\right)y_\infty + \frac{1}{3}\left(1-2\frac{l}{d}\right)\left(1-4\frac{l}{d}\right)y_1 - \frac{1}{4}\left(1-\frac{l}{d}\right)\left(1-4\frac{l}{d}\right)y_{1/2} + \frac{1}{24}\left(1-\frac{l}{d}\right)\left(1-2\frac{l}{d}\right)y_{1/4}\right]$$

where

y = desired variable within the range of ∞ and (1/4)

$y_\infty, y_1, y_{1/2}, y_{1/4}$ = variables corresponding to (l/d) ratios of ∞, 1, 1/2, 1/4 respectively.

Table 16.10 *Values of* $\left(\frac{h_o}{c}\right)$ *for maximum load and minimum friction coefficient conditions*

β	$\left(\frac{l}{d}\right) = \infty$		$\left(\frac{l}{d}\right) = 1$		$\left(\frac{l}{d}\right) = \left(\frac{1}{2}\right)$		$\left(\frac{l}{d}\right) = \left(\frac{1}{4}\right)$	
	Max. W	**Min. f**	**Max. W**	**Min. f**	**Max. W**	**Min. f**	**Max. W**	**Min. f**
360°	0.655	0.60	0.533	0.30	0.427	0.116	0.272	(0.03)
180°	0.640	0.60	0.520	0.44	0.418	0.233	0.276	(0.03)
120°	0.530	0.50	0.460	0.40	0.382	0.279	0.259	(0.06)
60°	0.250	0.23	0.230	0.22	0.200	0.160	0.150	0.10

(A.S.L.E. Transactions)

Table 16.11 *Bearing parameters*

Length to diameter ratio	
The length-to-diameter ratio (l/d) affects the performance of the bearing. A long bearing has more load carrying capacity compared with a short bearing. A short bearing, on the other hand, has greater side flow, which improves heat dissipation. The long bearings are more susceptible to metal-to-metal contact at the two edges, when the shaft is deflected under load. The longer the bearing, more difficult it is to get sufficient oil flow through the passage between the journal and the bearing. Therefore, the design trend is to use (l/d) ratio as 1 or less than 1. When the shaft and the bearing are precisely aligned, the shaft deflection is within the limit and cooling of lubricant and bearing does not present a serious problem, the (l/d) ratio can be taken as more than 1. In practice, the (l/d) ratio varies from 0.5 to 2.0, but in the majority of applications, it is taken as 1 or less than 1.	
Unit bearing pressure	
The unit bearing pressure ($p = W/ld$) is the load per unit of projected area of the bearing in running condition. It depends upon number of factors, such as bearing material, operating temperature, the nature and frequency of load and service conditions.	
Permissible unit bearing pressures	
Application	**Unit bearing pressure (p) (MPa or N/mm^2)**
(i) Diesel engines: Main bearing Crank pin Gudgeon pin	 5–10 7–14 13–14
(ii) Automotive engines: Main bearing Crank pin	 3–4 10–14
(iii) Air compressors: Main bearing Crank pin	 1–1.5 1.5–3.0
(iv) Centrifugal pumps: Main bearing	 0.5–0.7
(v) Electric motors: Main bearing	 0.7–1.5
(vi) Transmission shafting: Light duty Heavy duty	 0.15 1.00
(vii) Machine tools: Main bearing	 2
Start-up load	
The unit bearing pressure for starting conditions should not exceed 2 MPa. The start-up load is static load when the shaft is stationary. It mainly consists of the dead weight of shaft and its attachments. The start-up load can be used to determine the minimum length of the bearing on the basis of starting conditions.	

(*Contd.*)

(*Contd.*)

Radial clearance
The radial clearance (c) should be small to provide the necessary velocity gradient. However, this requires costly finishing operations, rigid mountings of the bearing assembly and clean lubricating oil without any foreign particles. This increases the initial and maintenance costs. The practical value of radial clearance is 0.001 mm per mm of the journal radius or $$c = (0.001)\, r$$ The practical values of radial clearances for commonly used bearing materials are as follows:

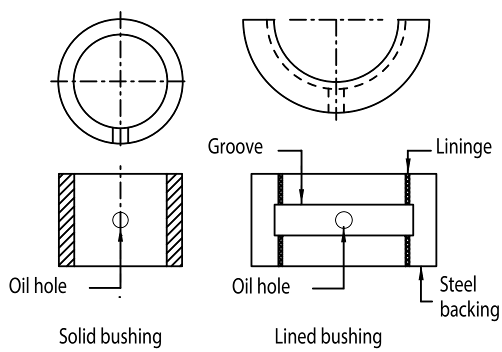
Material	Radial clearance (c)
Babbitts	(0.001) r to (0.00167) r
Copper-lead	(0.001) r to (0.01) r
Aluminium-alloy	(0.002) r to (0.0025) r

Minimum oil film thickness
The surface finish of the journal and the bearing is governed by the value of the minimum oil film thickness (h_o) selected by the designer and vice versa. There is a lower limit for the minimum oil film thickness, below which metal-to-metal contact occurs and hydrodynamic film breaks. This lower limit is given by, $$h_0 = (0.0002)r$$

Maximum oil film temperature
The lubricating oil tends to oxidise when the operating temperature exceeds 120°C. Also, the surface of babbitt bearing tends to soften at 125° C (for bearing pressure of 7 N/mm^2) and at 190° C (for bearing pressure of 1.4 N/mm^2). Therefore, the operating temperature should be kept within these limits. In general, the limiting temperature is 90° C for bearings made of babbitts.

Table 16.12 *Bearing constructions*

Solid and lined bushings



There are two types of bearing construction – solid bushing and lined bushing. A solid bushing is made either by casting or by machining from a bar. It is then finished by grinding and reaming operations. A typical example of this type of bushing is bronze bearing. A lined bushing consists of steel backing with a thin lining of bearing material like Babbitt. It is usually split into two halves.

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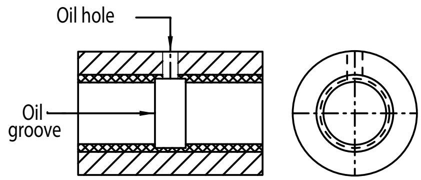
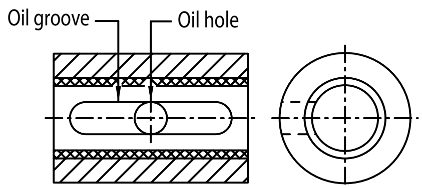
(*Contd.*)

Circumferential oil-groove bearing



Cylindrical oil-groove bearing

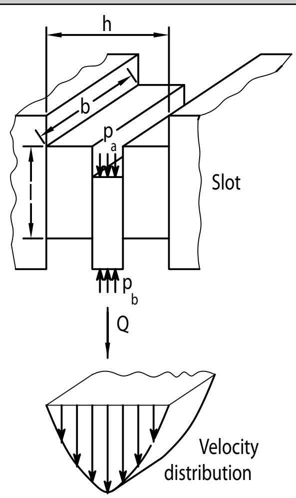


There are two basic types of patterns of oil grooves – circumferential and cylindrical. A circumferential oil-groove bearing in effect divides the bearing into two short bearings, each of length ($l/2$). The pressure developed in the bearing along the axis is reduced due to the groove. This results in a lower load carrying capacity. The centrifugal force acting on the oil in the circumferential groove may build pressure higher than the supply pressure, which may restrict the flow of the lubricant. Circumferential oil-groove bearings are used for connecting rod and crankshaft of automotive engines. The cylindrical oil-groove bearing has an axial groove almost along the full length of the bearing. It has a higher load carrying capacity compared with the circumferential oil-groove bearing; however, it is more susceptible to vibrations, known as oil-whip. Cylindrical oil-groove bearings are used for gearboxes and high-speed applications. In practice, different patterns of oil-groove are obtained by the combination of cylindrical and circumferential passages of the oil.

16.6 HYDROSTATIC STEP BEARINGS

Table 16.13 *Hydrostatic step bearing with circular pad*

Viscous flow through rectangular slot



$$Q = \frac{\Delta p b h^3}{12 \mu l} \qquad (16.21)$$

Q = flow of lubricating oil through the slot (mm^3/s)

l = length of the slot in the direction of flow (mm)

b and h = dimensions of the slot in a plane perpendicular to the direction of flow. (The dimensions b is very large compared with h) (h is fluid film thickness) (mm)

$\Delta p = (p_a - p_b)$ = pressure difference between two sides of slot (MPa or N/mm^2)

μ = viscosity of lubricating oil (MPa-s) or (N-s/mm^2)

(*Contd.*)

(*Contd.*)

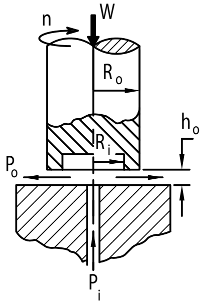
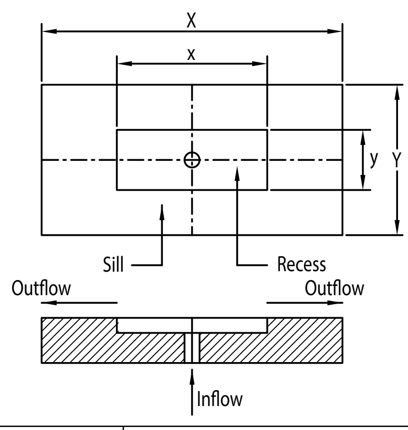
Hydrostatic step bearing	
	$$Q = \frac{\pi P_i h_o^3}{6\mu \log_e\left(\frac{R_o}{R_i}\right)} \quad (16.22)$$ $$W = \frac{\pi P_i}{2}\left[\frac{R_o^2 - R_i^2}{\log_e\left(\frac{R_o}{R_i}\right)}\right] \quad (16.23)$$ $$(\text{kW})_p = Q\,(P_i - P_o)\,(10^{-6}) \quad (16.24)$$ $$(\text{kW})_f = \left(\frac{1}{58.05 \times 10^6}\right)\frac{\mu n^2 (R_o^4 - R_i^4)}{h_o} \quad (16.25)$$ $$(\text{kW})_t = (\text{kW})_P + (\text{kW})_f \quad (16.26)$$ where, W = thrust load (N) R_0 = outer radius of the shaft (mm) R_i = radius of the recess or the pocket (mm) P_i = supply of inlet pressure (N/mm^2) or (MPa) P_o = outlet or atmospheric pressure (N/mm^2) or (MPa) h_o = fluid film thickness (mm) Q = flow of the lubricant (mm^3/s) μ = viscosity of the lubricant (N-s/mm^2) or (MPa-s) $(\text{kW})_p$ = pumping power (kW) $(\text{kW})_f$ = frictional power loss (kW) $(\text{kW})_t$ = total power required (kW)

Table 16.14 *Hydrostatic step bearing with rectangular pad*



x, y = linear dimensions of recess (mm)	X, Y = linear dimensions of pad (mm)
Area of pad	
$A_P = X\,Y$ (16.27)	A_P = total projected area of bearing pad (mm^2)

(*Contd.*)

(*Contd.*)

Load carrying capacity of bearing pad	
$W = a_f A_p p_r$ (16.28)	W = load acting on bearing (N) a_f = load coefficient (dimensionless) p_r = pressure of lubricating oil in recess (MPa or N/mm^2)
Amount of lubricant flow across pad	
$Q = q_f \left(\frac{W}{A_p}\right)\frac{h^3}{\mu}$ (16.29)	Q = volume of lubricating oil passing through the pad (mm^3/s) q_f = flow coefficient (dimensionless) h = film thickness (mm) μ = viscosity of the lubricant (N-s/mm^2) or (MPa-s)
Pumping power required by pad	
$H_B = p_r Q = H_f \left(\frac{W}{A_p}\right)^2 \frac{h^3}{\mu}$ (16.30) $H_f = \frac{q_f}{a_f}$ (16.31)	H_B = pumping power (N-mm/s) H_f = power coefficient (dimensionless)

Note: Refer to Charts 16.41 to 16.46 for determining the values of load coefficient, flow coefficient and power coefficient. The charts are prepared using electric analog field plotter.

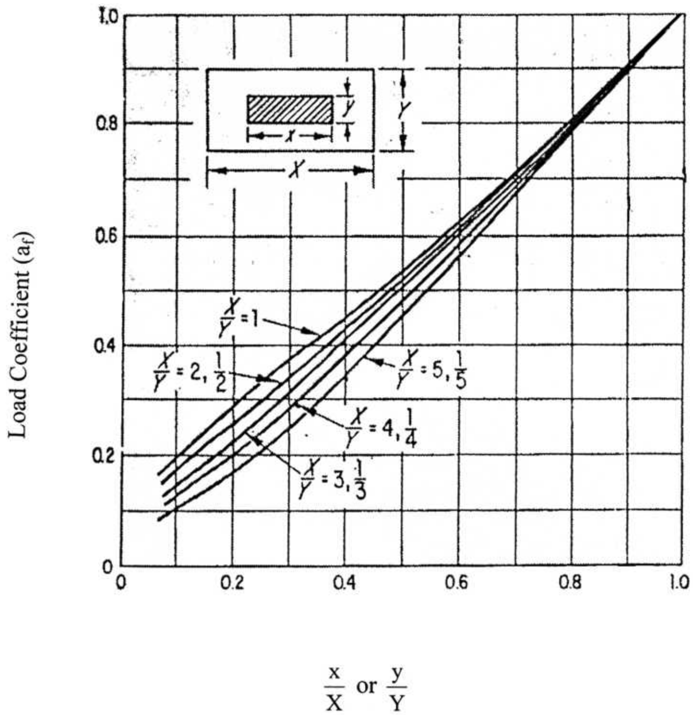


Chart 16.41 *Load Coefficients for Rectangular Pads having equally proportioned recess and pad perimeter (x/X = y/Y) (A_p = XY)*

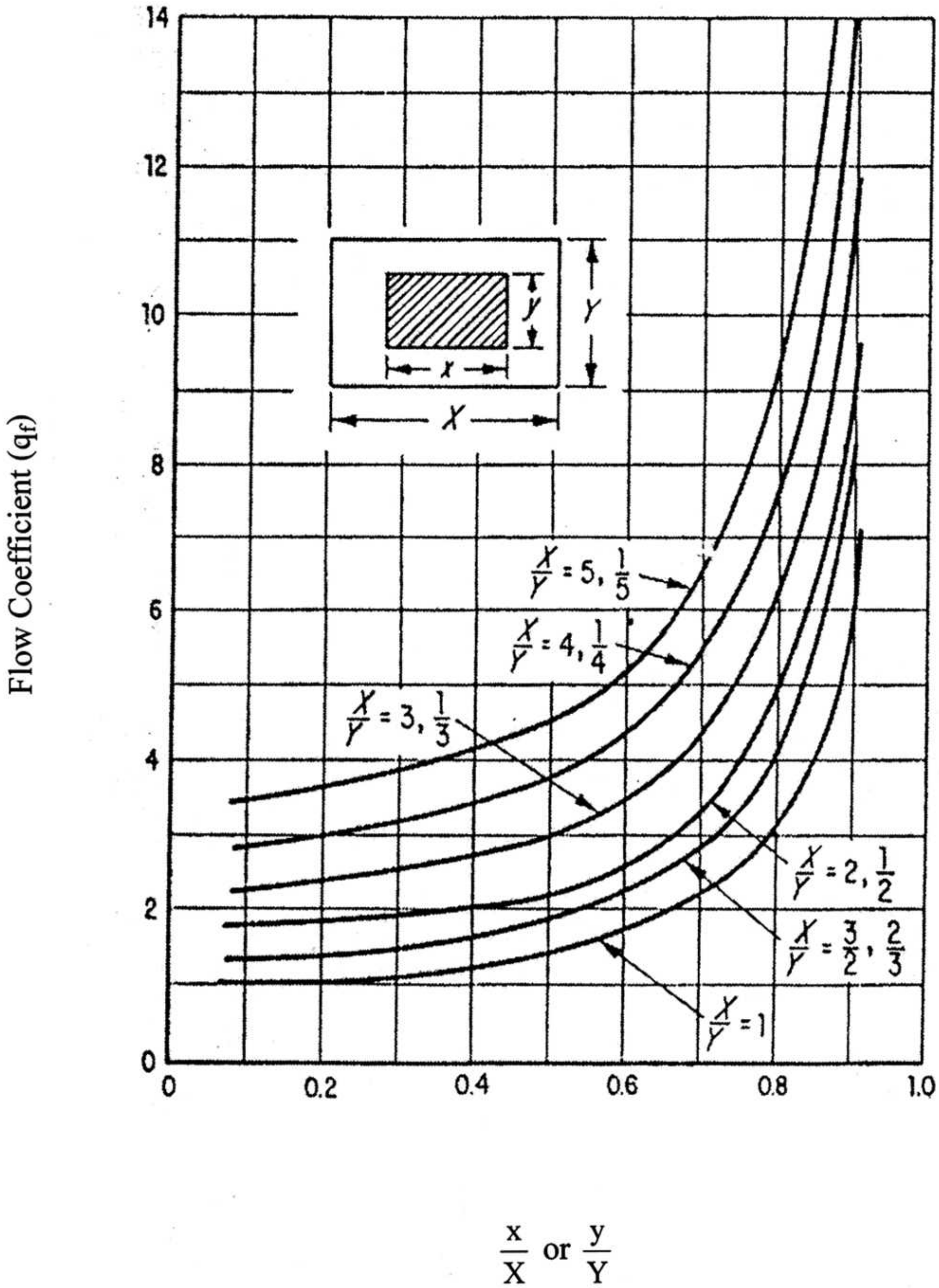


Chart 16.42 *Flow Coefficients for Rectangular Pads having equally proportioned recess and pad perimeter ($x/X = y/Y$) ($A_p = XY$)*

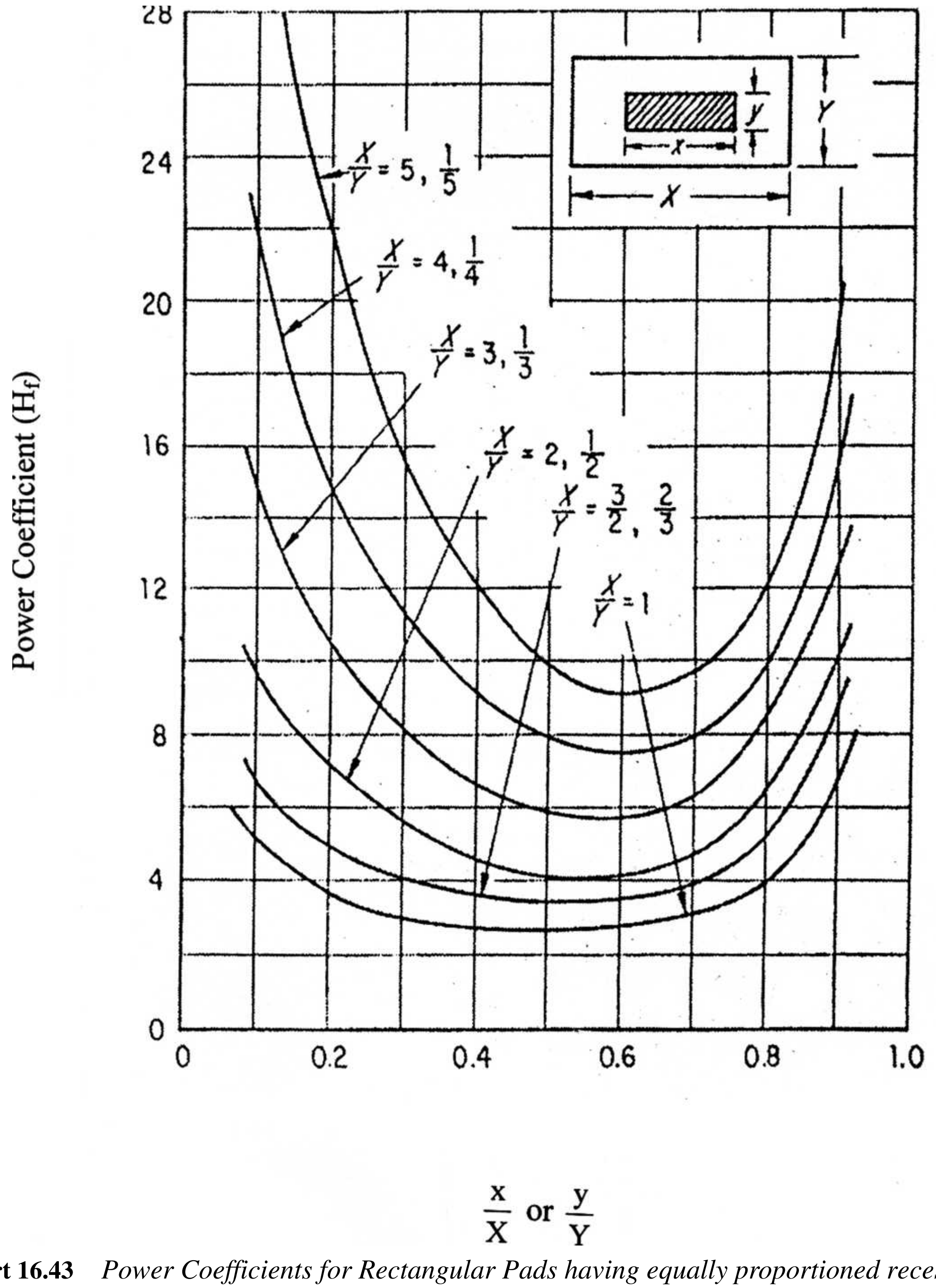


Chart 16.43 *Power Coefficients for Rectangular Pads having equally proportioned recess and pad perimeter ($x/X = y/Y$) ($A_p = XY$)*

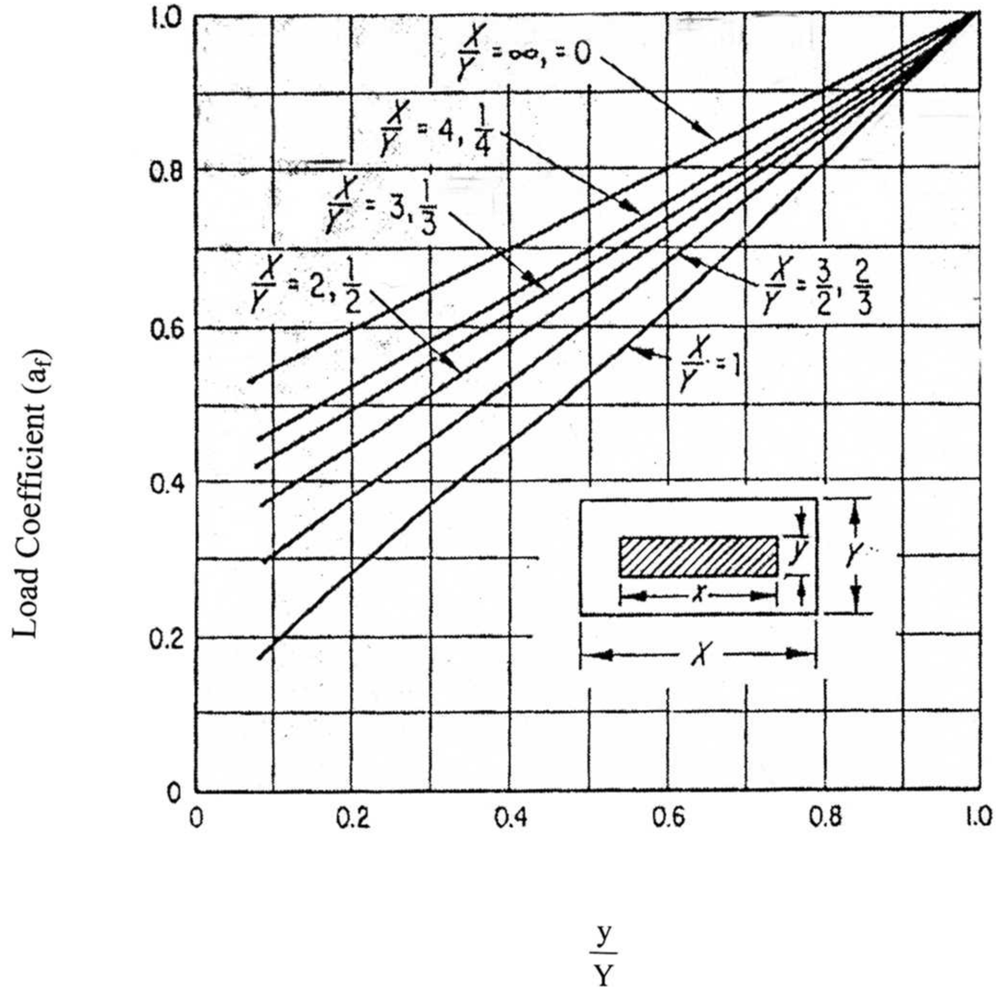
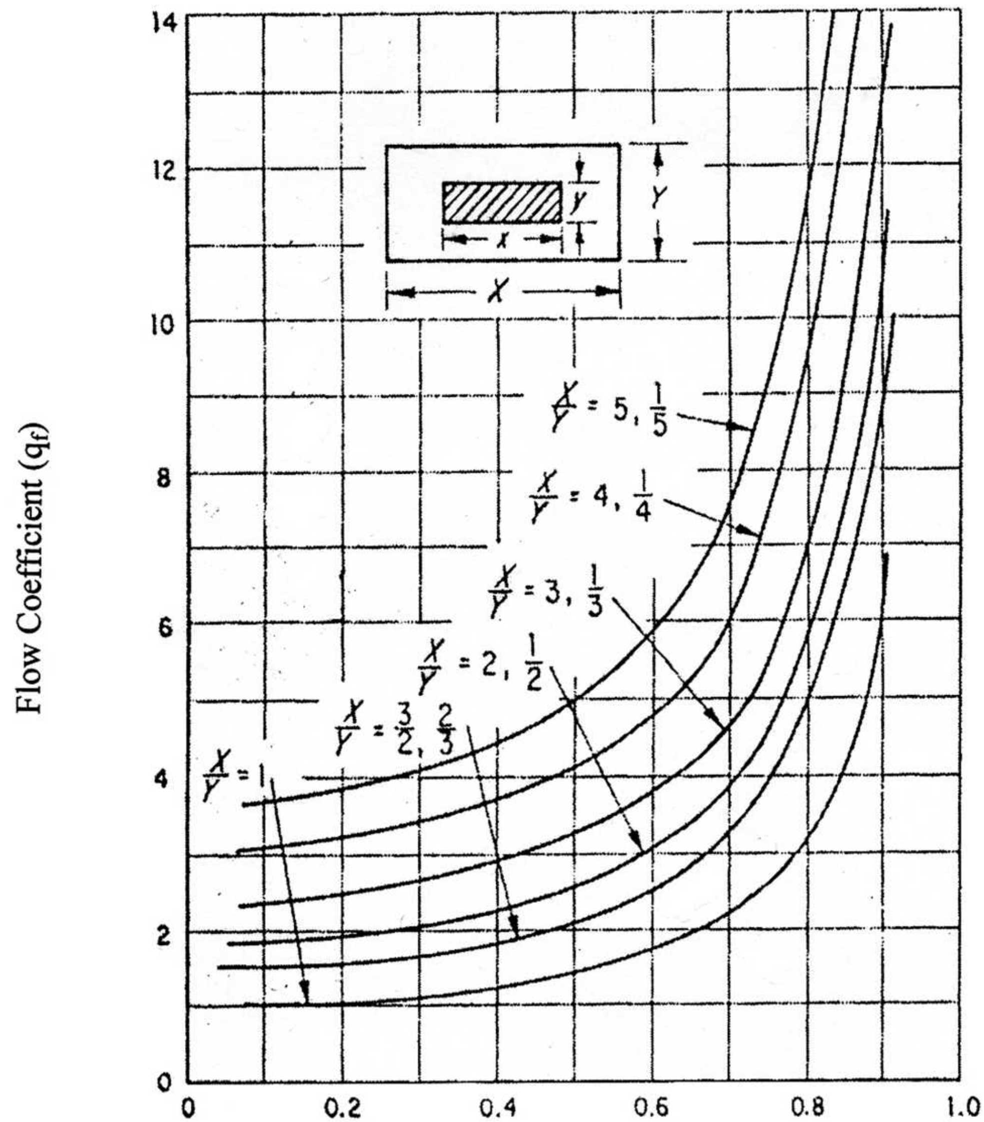


Chart 16.44 *Load Coefficients for Rectangular Pads having equal sill lengths ($X - x = Y - y$) ($A_p = XY$)*
Note: *Abscissa is y/Y for X/Y > 1 and x/X for X/Y < 1.*



$\frac{y}{Y}$

Chart 16.45 *Flow Coefficients for Rectangular Pads having equal sill lengths ($X - x = Y - y$) ($A_p = XY$)*
Note: *Abscissa is y/Y for X/Y > 1 and x/X for X/Y < 1.*

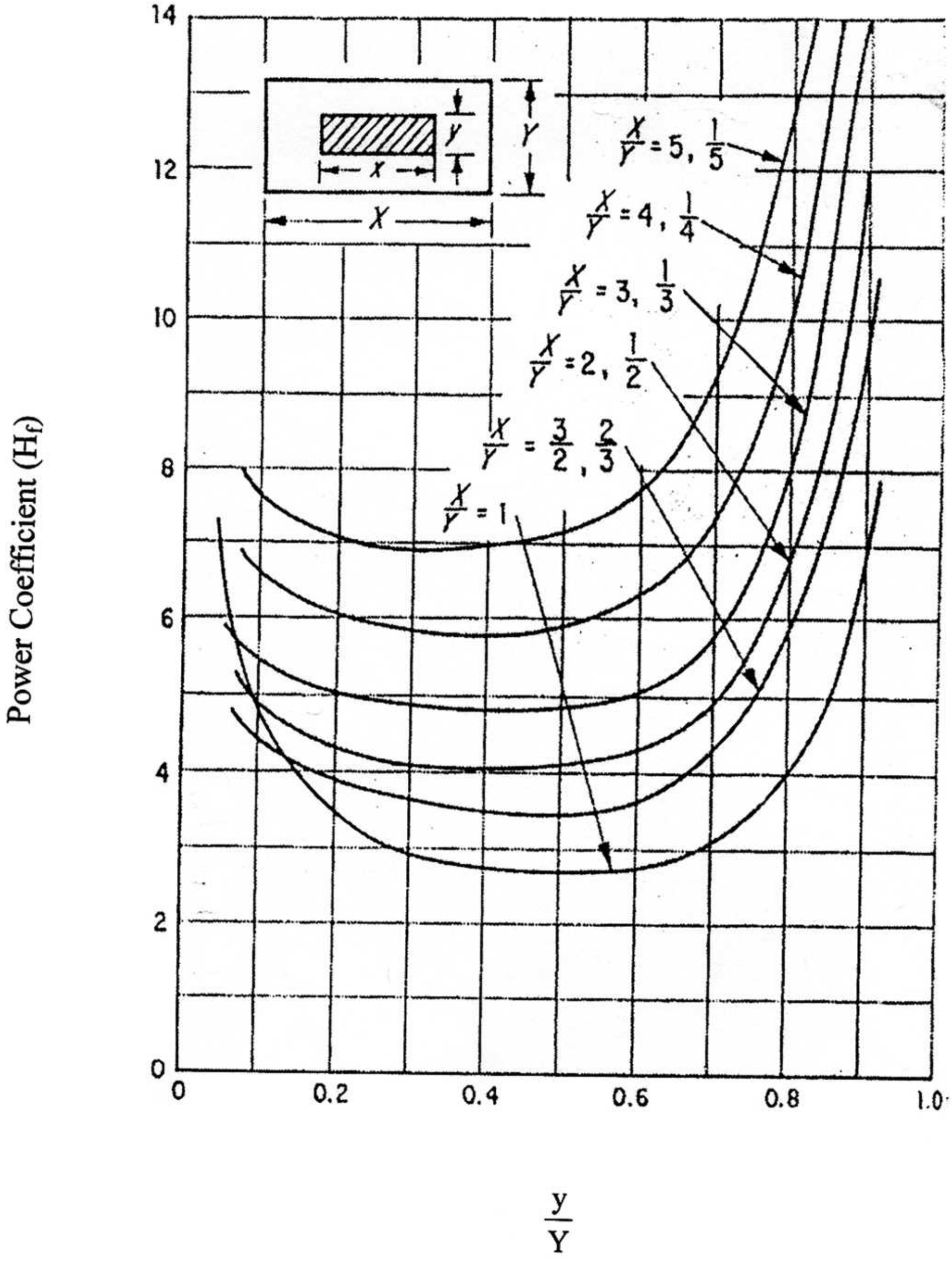


Chart 16.46 *Power Coefficients for Rectangular Pads having equal still lengths (X –x = Y – y) (A_p = XY)*

Note: *Abscissa is y/Y for X/Y > 1 and x/X for X/Y < 1.*

16.7 BEARING MATERIALS

Table 16.15 *Chemical composition of antifriction bearing alloys*

Grade	Alloying elements (percent)							
	Sn	**Sb**	**Pb**	**Cu**	**Ni**	**Cd**	**As**	**Zn**
90	90 Min.	6.5–7.5	0.3 Max.	2.5–3.5	—	—	—	—
84	84 Min.	9–11	0.3 Max.	5–6	—	—	—	—
75	74–76	10–12	Remainder	2.75–3.25	—	—	—	—
69	68–70	0.2 Max.	0.3 Max.	1.0–1.4	—	—	—	29–31
60	59–61	11–12	Remainder	2.5–3.5	—	—	—	—
20	19–21	14–16	Remainder	1.25–1.75	—	—	—	—
10	9–11	13–15	Remainder	0.50–1.0	—	—	—	—
6	5–7	14–16	Remainder	0.80–1.20	0.80–1.50	0.70–1.5	0.30–0.80	—
5	4.5–5.5	14–16	Remainder	0.30–0.70	—	—	—	—
1	0.75–1.25	15–16	Remainder	0.5 Max.	—	—	0.8–1.1	—

Table 16.16 *Pouring temperatures and applications of antifriction bearing alloys*

Grade	Pouring temperature (ºC)	Applications
90	340–390	Linings of petrol and diesel engine bearings and other bearings used at high speeds
84	430–460	
75	360–400	Mostly used for repair jobs in mills and marine installations
69	500 approx.	Bearing alloy for under water applications and gland packings
60	370–400	For lining of bearings working at medium speeds such as centrifugal pumps, circular saws, converters, dynamos and electric motors
20	370–410	For low speed bearings such as pulp crushers, concrete mixers and rope conveyors
10		
6	500–530	Heavy-duty bearings in rolling mills, sugar, rubber, paper and steel industries. Also bearings for diesel engines and turbines.
5	350–390	For bearings in mill shafting and railway carriage and wagon bearings
1	—	Used as thin line overlay on steel strips where the white metal lining is 0.076 mm thick

16.8 THIN-WALLED HALF BEARINGS

Table 16.17 *Composition of linings for thin-walled half-bearings*

Lining material	**Approximate percentage of**					
	Sn	**Sb**	**Cu**	**Pb**	**As**	
(1) Tin-based White metal						
(a) SnSb8Cu4	88–90	7–8	3–4	0.35 Max.	0.1 Max.	
(2) Lead-based White metal						
(a) PbSb10Sn6	5–7	9–11	0.7 Max.	80–86	0.25 Max.	
(b) PbSb15Sn10	9–11	14–16	0.7 Max.	71–77	0.6 Max.	
(c) PbSb15SnAs	0.9–1.7	13.5–15.5	0.7 Max.	80–84	0.8–1.2	
(3) Copper-based alloys						
(a) CuPb30	0.5 Max.	—	R	26–33	—	
(b) CuPb24Sn4	3–4.5	—	R	19–27	—	
(c) CuPb24Sn	0.6–2.0	—	R	14–27	—	
(d) CuPb8Sn4	3.5–4.5	—	R	7–9	—	
(4) Aluminium-based alloys						
	Al	Cu	Sn	Ni	Si	Mn
(a) AlSn20Cu	R	0.7–1.3	17.5–22.5	0.1 Max.	0.7 Max.	0.7 Max.
(b) AlSn6Cu	R	0.7–1.3	5.5–1.3	1.3	0.7 Max.	0.7 Max.
(5) Overlay						
	Pb	Sn	Cu			
(a) PbSn10	R	8–12	–			
(b) PbSn10Cu2	R	8–12	1.3			

Note: *R* = Remainder

Table 16.18 *Composition of backings for thin-walled half-bearings*

For white-metal alloys and aluminium-based alloys	
(Cold rolled bight annealed steel)	
Constituent	Percentage
Carbon	0.12 Max.
Manganese	0.5 Max.
Silicon	0.35 Max.
Sulphur	0.05 Max.
Phosphorus	0.05 Max.

(*Contd.*)

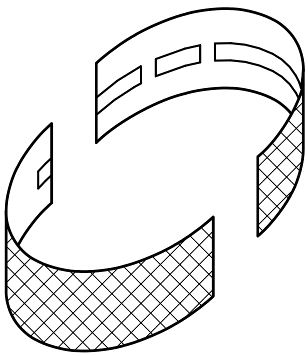
Hardness range = 100 to 120 HV10	
For copper-based alloys	
(Cold rolled steel)	
Constituent	Percentage
Carbon	0.15 Max.
Manganese	0.70 Max.
Silicon	0.35 Max.
Sulphur	0.05 Max.
Phosphorus	0.05 Max.
Hardness range = 170 to 210 HV10	

Table 16.19 *Lining thickness for thin-walled half-bearings*

Lining material	Minimum lining thickness
White metal (non-micro)	0.125 mm
White metal (micro)	0.005 mm
Copper-lead	0.125 mm
Tin-aluminium	0.125 mm

Note: For better fatigue strength, it is desirable to provide minimum lining thickness on steel backings.

Table 16.20 *Dimensions of thin-walled half-bearings (Plain bearings)*



Preferred housing diameter (D_1)	Wall thickness							
	1.5	1.75	2.0	2.5	3.0	3.5	4.0	5.0
	Inside diameter							
20	17	16.5						
21	18	17.5						
22	19	18.5						
24	21	20.5						

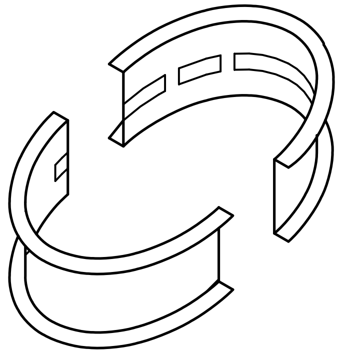
(*Contd.*)

(*Contd.*)

25	22	21.5						
26	23	22.5						
28	25	24.5						
30	27	26.5						
32	29	28.5	28					
34	31	30.5	30					
36	33	32.5	32					
38	35	34.5	34					
40		36.5	36	35				
42		38.5	38	37				
45		41.5	41	40				
48		44.5	44	43				
50		46.5	46	45				
53		49.5	49	48				
56		52.5	52	51				
60		56.5	56	55				
63		59.5	59	58				
67			63	62	61			
71			67	66	65			
75			71	70	69			
80			76	75	74			
85			81	80	79			
90				85	84	83		
95				90	89	88		
100				95	94	93		
105				100	99	98		
110				105	104	103		
120				115	114	113		
125					119	118	117	
130					124	123	122	
140					134	133	132	
150					144	143	142	
160					154	153	152	
170						163	162	160
180						173	172	170
190						183	182	180
200						193	192	190

Note: All dimensions are in mm.

Table 16.21 *Dimensions of thin-walled half-bearings (Flanged bearings)*



Preferred housing diameter (D_1)	Wall thickness						
	1.5	1.75	2.0	2.5	3.0	3.5	4.0
	Inside diameter						
38	35	34.5	34				
40		36.5	36	35			
42		38.5	38	37			
45		41.5	41	40			
48		44.5	44	43			
50		46.5	46	45			
53		49.5	49	48			
56		52.5	52	51			
60		56.5	56	55			
63		59.5	59	58			
67			63	62	61		
71			67	66	65		
75			71	70	69		
80			76	75	74		
85			81	80	79		
90				85	84	83	
95				90	89	88	
100				95	94	93	
105				100	99	98	
110				105	104	103	
120				115	114	113	

(*Contd.*)

(*Contd.*)

125					119	118	117
130					124	123	122
140					134	133	132
150					144	144	142

Note: All dimensions are in mm.
Designation – A thin walled half bearing having housing diameter 50 mm and wall thickness 2 mm, made of white metal (tin based) lining is designated as,
Thin-walled Half Bearing 50 – 2 WM

16.9 LUBRICATING OILS

Table 16.22 *Properties of automotive lubricating oils (Heavy duty)*

Characteristic	Grade						
	SAE 5 W	**SAE 10 W**	**SAE 20 W**	**SAE 20**	**SAE 30**	**SAE 40**	**SAE 50**
(1) Apparent viscosity							
(a) Min. cP at °C	3250 at –30	3500 at –25	3500 at –20	—	—	—	—
(b) Max. cP at °C	<3500 at –25	<3500 at –20	<4500 at –10	—	—	—	—
(2) Kinematic viscosity cSt at 100°C	3.8 Min.	4.1 Min.	5.6 Min.	5.6 to <9.3	9.3 to <12.5	12.5 to <16.3	16.3 to <21.9
(3) Borderline pumping temperature °C Max.	–30	–25	–15	—	—	—	—
(4) Viscosity index Min	100	100	95	95	95	90	90
(5) Pour point °C Max.	–33	–27	–21	–21	–6	–6	–6
(6) Flash point °C (Cleveland open cup), Min.							
(a) For Type 1 to 4	160	190	200	200	215	215	220
(b) For Type 5	160	205	210	210	220	225	230
(7) Stable Pour point °C Max.	–33	–27	—	—	—	—	—

Table 16.23 *Properties of axle oil*

Characteristic	Requirement for		
	Light grade	Medium grade	Heavy grade
Pour point (°C) Max.	–3	–3	–3
Flash point (°C) closed (Pensky-Martens), Min.	160	160	160
Kinematic viscosity at 60 °C (cSt)	64–71	76–86	90–103
Water content, percent by mass, Max.	0.5	0.5	0.5
Saponification value (mg of KOH/g of oil), Max.	1.5	1.5	1.5
Viscosity index, Min.	40	40	40

Note: Axle oil is used as lubricant for axle bearings.

Table 16.24 *Properties of 2T - oil*

1. Kinematic viscosity (cSt at 40°C)	27–33
2. Flash point (°C) (Min)	66
3. Pour point (°C) (Min)	–6

Note: 2T-oil consists of SAE30 as base oil and detergent additives. The recommended fuel/oil ratio varies from 12:1 to 50:1 depending upon the type of engine.

Table 16.25 *Properties of gear oils*

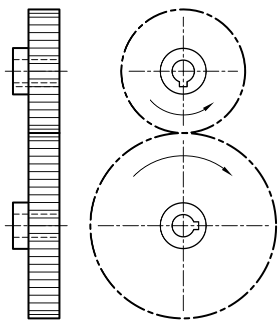
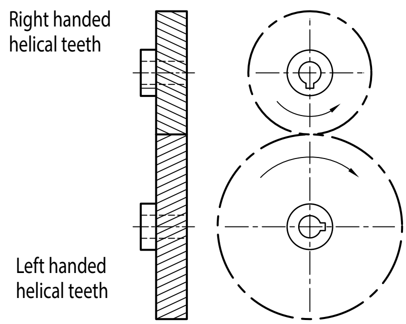
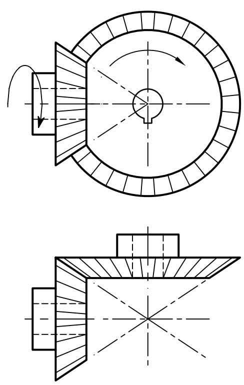
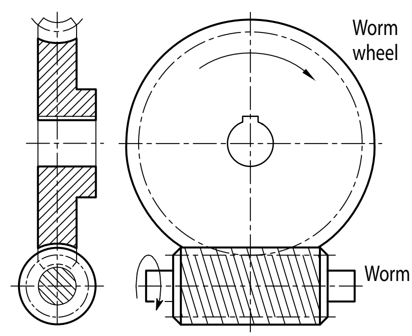
Properties	Servo Gear HP / Gear Super		
	80	90	140
1. SAE grade	80	90	140
2. Kinematic viscosity (cSt at 100°C)	9–11	16.5–18.5	31–33
3. Viscosity index (Min.)	85	85	80
4. Flash point (°C) (Min.)	165	180	190
5. Pour point (°C) (Max.)	–27	–9	0

Spur Gear Design Theoretical Approach

17

17.1 STANDARD TOOTH PROFILES

Table 17.1 *Basic types of gears*

Spur gears	Helical gears
	
Bevel gears	**Worm gears**
	

Note: In a pair of gears, the smaller gear is called as ‘pinion’, while the bigger gear is usually referred as ‘wheel’ or simply ‘gear’.

Table 17.2 *Module of gear*

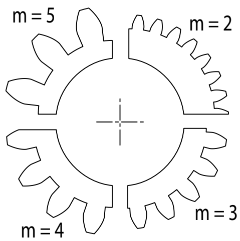
The module (m) is the inverse of diametral pitch. The diametral pitch (P) is the ratio of the number of teeth on gear to the pitch circle diameter of gear.

$$P = \frac{z}{d} \tag{17.1}$$

$$m = \frac{1}{P} = \frac{d}{z} \tag{17.2}$$

$$d = mz \tag{17.3}$$

P = diametral pitch
z = number of teeth on gear
d = pitch circle diameter of gear (mm)
m = module (mm)



The module specifies the size of the gear tooth. As the module increases, the size of gear tooth also increases. It can be said that module is the 'index' of the size of gear tooth. The values of module are standardized. (Table 17.3)

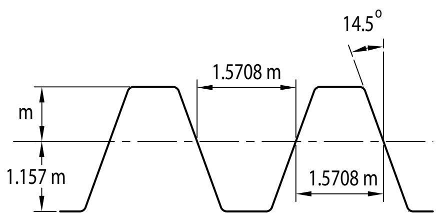
Table 17.3 *Recommended series of module (mm)*

Choice 1	1.0	1.25	1.5	2.0	2.5	3.0	4.0
	5.0	6.0	8.0	10	12	16	20
Choice 2	1.125	1.375	1.75	2.25	2.75	3.5	4.5
	5.5	7	9	11	14	18	
Choice 3	(3.25)	(3.75)	(6.5)				

Note: (i) Preference is given to use the modules under Choice 1.
(ii) Modules given under Choice 3 should be avoided as far as possible.

Table 17.4 *Standard systems of gear tooth*

14.5° Full depth involute system



The basic rack for this system is composed of straight sides except for the fillet arcs. In this system, interference occurs when the number of teeth on pinion is less than 23. This system is satisfactory when the number of teeth on the gears is large. When the number of teeth is small and the gears are made by generating process, undercutting is unavoidable.

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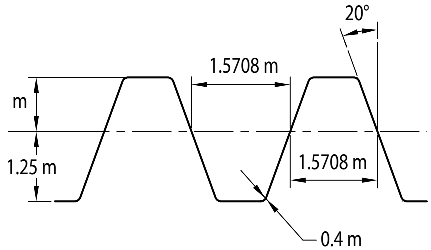
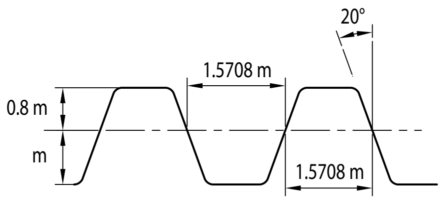
20° Full depth involute systems



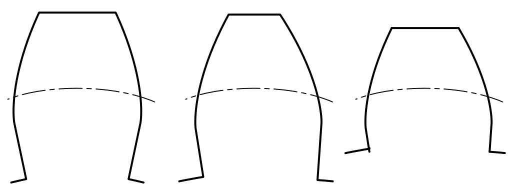
The basic rack for this system is also composed of straight sides except for the fillet arcs. In this system, interference occurs when the number of teeth on pinion is less than 17. The 20°-pressure angle system with full depth involute teeth is widely used in practice. It is also recommended by Bureau of Indian Standards.

20° Stub involute system



The gears in this system have shorter addendum and shorter dedendum. The interfering portion of the tooth, that is, a part of addendum is thus removed. Therefore, these teeth have still smaller interference. This also, reduces the undercutting. In this system, the minimum number of teeth on pinion to avoid interference is 14. Since the pinion is small, the drive becomes more compact. Stub teeth are stronger than full depth teeth because of smaller moment arm of bending force. Therefore, stub system transmits very high load. Stub teeth results in lower production cost, as less metal must be cut away. The main draw back of this system is that the contact ratio is reduced due to short addendum. Due to insufficient overlap, vibrations are likely to occur.

Comparative shapes of standard tooth profile



(i) 14.5° full depth (ii) 20° full depth (iii) 20° stub

Table 17.5 *Gear ratio and transmission ratio*

Gear ratio	
The gear ratio (i) is the ratio of the number of teeth on the wheel to that on the pinion	
$$i = \frac{n_p}{n_g} = \frac{z_g}{z_p} \quad (17.4)$$	n_p = speed of pinion (rpm) n_g = speed of wheel (rpm) z_p = number of teeth on the pinion z_g = number of teeth on the wheel

(*Contd.*)

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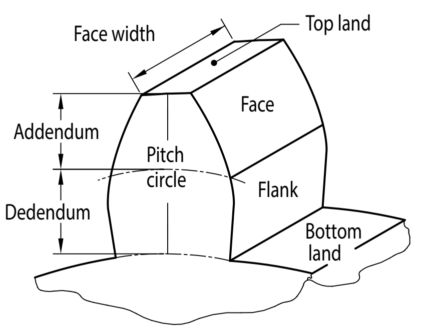
Transmission ratio	
The transmission ratio (i') is the ratio of the angular speed of the first driving gear to the angular speed of the last driven gear in a gear train.	
Requirement for compact gearbox	
For two-stage gearbox, $i = \sqrt{i'}$ (17.5)	For three-stage gearbox, $i = \sqrt[3]{i'}$ (17.6)

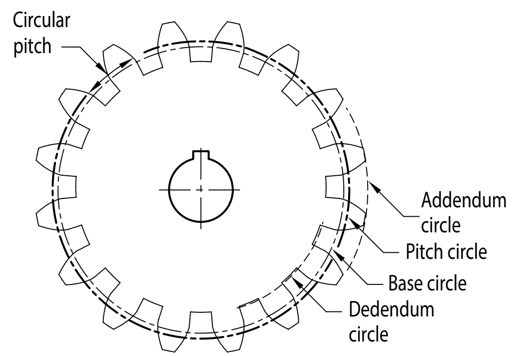
Table 17.6 *Values of Gear ratio*

Type of gear set	Minimum ratio	Maximum ratio
External spur gears	1:1	6:1
Internal spur gears	1.5:1	7:1
External helical gears	1:1	10:1
Internal helical gears	1.5:1	10:1
Cylindrical worm gears	3:1	100:1
Straight bevel gears	1:1	8:1
Spiral bevel gears	1:1	8:1
Epicyclic planetary gears	3:1	12:1
Epicyclic star gears	2:1	11:1

17.2 SPUR GEAR NOMENCLATURE

Table 17.7 *Spur gear terminology*





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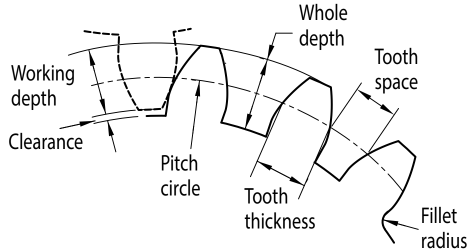

Pitch circle
The pitch circle is the curve of intersection of the pitch surface of revolution and the plane of rotation. It is an imaginary circle that rolls without slipping with the pitch circle of a mating gear. The pitch circles of a pair of mating gears are tangent to each other.
Pitch circle diameter (*d*)
The pitch circle diameter is the diameter of pitch circle. The size of the gear is usually specified by pitch circle diameter. It is also called 'pitch diameter'.
Base circle
The base circle is an imaginary circle from which the involute curve of the tooth profile is generated. The base circles of two mating gears are tangent to the pressure line.
Addendum circle
The addendum circle is an imaginary circle that borders the tops of gear teeth in the cross section.
Addendum (h_a)
The addendum (h_a) is the radial distance between pitch and the addendum circles. Addendum indicates the height of tooth above the pitch circle.
Dedendum circle
The dedendum circle is an imaginary circle that borders the bottom of spaces between teeth in the cross section. It is also called 'root' circle.
Dedendum (h_f)
The dedendum (h_f) is the radial distance between pitch and the dedendum circles. The dedendum indicates the depth of the tooth below the pitch circle.
Clearance (*c*)
The clearance is the amount by which the dedendum of a given gear exceeds the addendum of its mating tooth.
Face width (*b*)
Face width is width of the tooth measured parallel to the axis.
Circular tooth thickness
The length of the arc on pitch circle subtending a single gear tooth is called circular tooth thickness. Theoretically, circular tooth thickness is half of circular pitch.
Tooth space
The width of the space between two adjacent teeth measured along the pitch circle is called the tooth space. Theoretically, tooth space is equal to circular tooth thickness or half of circular pitch.

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Working depth (h_k)
The working depth is the depth of engagement of two gear teeth, that is, the sum of their addendums.
Whole depth (h)
The whole depth is the total depth of the tooth space, that is, the sum of addendum and dedendum. Whole depth is also equal to working depth plus clearance.
Centre distance
The centre distance is the distance between centres of pitch circles of mating gears. It is also the distance between centres of base circles of mating gears.
Pressure angle (α)
The pressure angle is the angle, which the line of action makes with the common tangent to the pitch circles. The pressure angle is also called the angle of obliquity.
Circular pitch (p)
The circular pitch is the distance measured along the pitch circle between two similar points on adjacent teeth.
Diametral pitch (P)
The diametral pitch is the ratio of the number of teeth to the pitch circle diameter.

Table 17.8 *Basic relationships*

$p = \dfrac{\pi d}{z}$ (17.7)	p = circular pitch (mm) d = pitch circle diameter (mm) z = number of teeth on gear
$P = \dfrac{z}{d}$ $P \times p = \pi$ (17.8)	P = diametral pitch
$a = \dfrac{m(z_p + z_g)}{2}$ (17.9)	a = centre-to-centre distance (mm) z_p = number of teeth on the pinion z_g = number of teeth on the wheel

Table 17.9 *Standard proportions of gear tooth for 20° full depth involute system*

Dimension	Notation	Proportion
Addendum	h_a	$h_a = m$
Dedendum	h_f	$h_f = 1.25$ m
Clearance	c	$c = 0.25$ m
Working depth	h_k	$h_k = 2$ m
Whole depth	h	$h = 2.25$ m
Tooth thickness	s	$s = 1.5708$ m
Tooth space	—	1.5708 m
Fillet radius	—	0.4 m

Note: In above table, '*m*' is the module of gear tooth.

17.3 GEAR MATERIALS

Table 17.10 *Mechanical properties of gear materials*

Material	Condition	Minimum tensile strength (MPa or N/mm^2)	Brinell Hardness (HB)
Grey cast iron			
FG 200	As cast	200	179 Min
FG 260		260	197 Min
FG 350		350	207 Min
FG 350	Heat-treated	350	300 Min
Malleable cast iron			
Whiteheart	—	280	217 Min
Blackheart	—	320	149 Min
Phosphor bronze			
Phosphor bronze castings (for Gear blanks)	Sand cast	160	60 Min
Phosphor bronze castings (for Gear blanks)	Chill cast	240	70 Min
Phosphor bronze castings (for Gear blanks)	Centrifugally cast	265	90
Cast steel			
Grade -1	—	550	145
Grade -1	Annealed	630	182 Min
Grade -1	Hardened and Tempered	800	229 Min
Grade -1	Surface hardened	630	165 (core) 460 (case)
High tensile steel castings			
Grade –1 (CS 640)	—	640	190 Min
Grade –2 (CS 700)	—	700	207 Min
Grade –3 (CS 840)	—	840	248 Min
Grade –4 (CS 1030)	—	1030	311 Min
Grade –5 (CS 1230)	—	1230	363 Min

(*Contd.*)

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Forged steel – Carbon steels			
30C8	Normalised	500	143
30C8	Hardened and Tempered	600	152
40C8	Normalised	580	152
40C8	Hardened and Tempered	600	179
55C8	Normalised	720	201
55C8	Hardened and Tempered	700	223
55C8	—	800	248
Forged steel – Carbon chromium steels			
55Cr3	Hardened and Tempered	900	225 Min
55Cr3		1000	285 Min
40Cr4		800	229 Min
Forged steel – Carbon manganese steels			
27C15	Normalised	550	—
27C15	Hardened and Tempered	600	170 Min
27C15		700	201 Min
37C15	—	700	201 Min
37C15	—	800	229 Min
Forged steel –Manganese molybdenum steels			
35Mn6Mo3	Hardened and Tempered	700	201 Min
35Mn6Mo3		800	229 Min
35Mn6Mo4		800	229 Min
35Mn6Mo4		900	255 Min
Forged steel –Chromium molybdenum steels			
40Cr4Mo2	Hardened and Tempered	700	201 Min
40Cr4Mo2		800	229 Min
40Cr4Mo2		900	225 Min
40Cr4Mo2		1000	285 Min

(*Contd.*)

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Forged steel –Nickel steels			
40Ni14	Hardened and Tempered	800	229 Min
40Ni14		900	255 Min
30Ni16Cr5		1540	444 Min
40Ni6Cr4Mo2		900	255 Min
40Ni6Cr4Mo2		1550	444 Min
31Ni10Cr3Mo6		900	255 Min
31Ni10Cr3Mo6		1550	444 Min
31Ni10Cr3Mo6		1100	311 Min
31Ni10Cr3Mo6		1000	285 Min
40Ni10Cr3Mo6		1550	444 Min
40Ni10Cr3Mo6		1200	341 Min
Surface hardened steel – Carbon steels			
0.4 % Carbon steel	—	550	145 (core) 460 (case)
0.55 % Carbon steel	—	700	200 (core) 520 (case)
Surface hardened steel – Carbon chromium steels			
0.55 % Carbon Chromium steel	—	860	250 (core) 500 (case)
1 % Chromium steel	—	700	500 (case)
Surface hardened steel – Nickel steel			
3.5 % Nickel steel	—	700	200 (core) 300 (case)
Surface hardened steel – Nickel chromium steels			
3.5 % Nickel Chromium steel	—	860	250 (core) 500 (case)
1.5 % Nickel, 1 % Chromium steel	—	860	250 (core)
Case hardened steel – Carbon steels			
0.12 to 0.22 % Carbon steel	—	500	650 (case)
0.15 % Carbon steel	—	500	140 (core)
0.20 % Carbon steel	—	500	140 (core) 640 (case)

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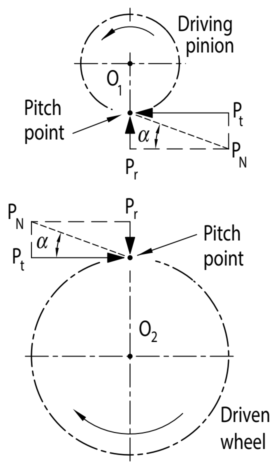
Case hardened steel – Nickel steels			
3 % Nickel steel	—	700	200 (case)
3.5 % Nickel steel	—	700	200 (core) 620 (case)
5 % Nickel steel	—	860	250 (core) 600 (case)

17.4 SPUR GEARS TOOTH FORCE

Table 17.11 *Spur gears —Components of tooth force*

Magnitude of two components



$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n_p} \tag{17.10}$$

$$P_t = \frac{2M_t}{d_p} \tag{17.11}$$

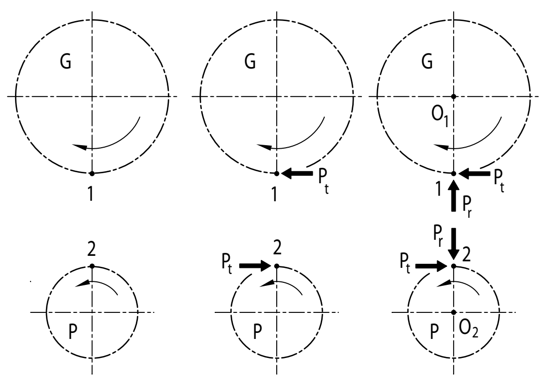
$$P_r = P_t \tan\alpha \tag{17.12}$$

$$d_p = m z_p$$

Notations:

M_t = torque transmitted by gears (N-mm)
(kW) = power transmitted by gears (kW)
n_p = speed of pinion (rpm)
P_t = tangential component of resultant tooth force (N)
P_r = radial component of resultant tooth force (N)
d_p = pitch circle diameter of pinion (mm)
α = pressure angle (°)

Direction of two components (Free body diagram of forces)



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Assumptions
(i) It is assumed that pinion is the driving element while gear is the driven element.
(ii) It is assumed that pinion rotates in anticlockwise direction. Therefore, the gear will rotate in clockwise direction.
(iii) In running condition, point 2 on pinion and point 1 on gear are in contact with each other.
Direction of tangential component
(i) The gear G is the driven element. It is made to rotate in clockwise direction. Therefore, at point 1 on gear G, the tangential component P_t will act towards left.
(ii) There will be equal and opposite reaction at point 2 on the pinion P. It is observed that the direction of tangential component P_t on driving element, that is, pinion is opposite to direction of rotation.
Direction of radial component
(i) The radial component acts towards the centre of respective gear. For pinion P, the radial component P_r acts at point 2 towards the centre O_2.
(ii) For gear G, the radial component P_r acts at point 1 towards the centre O_1.

17.5 DESIGN BY LEWIS AND BUCKINGHAM'S EQUATIONS

Table 17.12 *Minimum number of teeth*

Pressure angle (α)	14.5°	20°	25°
z_{min} (theoretical)	32	17	11
z_{min} (practical)	27	14	9

Note: The minimum number of teeth to avoid interference is given by,

$$z_{min} = \frac{2}{\sin^2 \alpha} \tag{17.13}$$

In practice, giving a slight radius to the tip of tooth can further reduce the value of z_{min}.

Table 17.13 *Face width of tooth*

If the face width (b) is too large, there is a possibility of concentration of load at one end of the gear tooth due to a number of factors, like misalignment, elastic deformation of shafts, and warping of gear tooth. On the other hand, gears with a small face width have a poor capacity to resist the shock and absorb vibrations. They also wear at a faster rate. A narrow face width results in a coarse pitch.
Optimum range of face width (8 m) < b < (12 m) or ($b \approx$ 10 m) (17.14)

Table 17.14 *Beam strength of gear tooth (Lewis equation)*

The beam strength (S_b) is the maximum value of the tangential force that the tooth can transmit without bending failure.	
$S_b = mb\sigma_b Y$ (17.15)	S_b = beam strength of gear tooth (N) σ_b = permissible bending stress (MPa or N/mm^2) Y = Lewis form factor (Table 17.15)
$\sigma_b = S_e = \left(\frac{1}{3}\right) S_{ut}$ (17.16)	S_e = endurance limit (MPa or N/mm^2) S_{ut} = ultimate tensile strength (MPa or N/mm^2)

Table 17.15 *Values of the Lewis form factor (Y) for 20° full depth involute system*

z	Y	z	Y	z	Y
15	0.289	27	0.348	55	0.415
16	0.295	28	0.352	60	0.421
17	0.302	29	0.355	65	0.425
18	0.308	30	0.358	70	0.429
19	0.314	32	0.364	75	0.433
20	0.320	33	0.367	80	0.436
21	0.326	35	0.373	90	0.442
22	0.330	37	0.380	100	0.446
23	0.333	39	0.386	150	0.458
24	0.337	40	0.389	200	0.463
25	0.340	45	0.399	300	0.471
26	0.344	50	0.408	Rack	0.484
(z = Number of teeth)					

Table 17.16 *Wear strength of gear tooth (Buckingham's equation)*

The wear strength (S_w) is the maximum value of the tangential force that the tooth can transmit without pitting failure.	
$S_w = bQd_p K$ (17.17)	S_w = wear strength of the gear tooth (N) Q = ratio factor d_p = pitch circle diameter of pinion (mm) K = load – stress factor (MPa or N/mm^2)
$Q = \dfrac{2z_g}{z_g + z_p}$ (for external gears) (17.18) $Q = \dfrac{2z_g}{z_g - z_p}$ (for internal gears) (17.19)	z_p = number of teeth on pinion z_g = number of teeth on wheel

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$$K = \frac{\sigma_c^2 \sin\alpha \cos\alpha (1/E_p + 1/E_g)}{1.4} \qquad (17.20)$$	σ_c = surface endurance strength of the material (MPa or N/mm^2) α = pressure angle E_p = modulus of elasticity of pinion materials (MPa or N/mm^2) (Table 17.17) E_g = modulus of elasticity of wheel materials (MPa or N/mm^2) (Table 17.17)
For steel gears with 20° pressure angle, $$K \approx 0.16\left(\frac{\text{BHN}}{100}\right)^2 \qquad (17.21)$$	BHN = surface hardness of gears (Brinell Hardness Number)
According to G. Niemann, $$\sigma_c = 0.27(\text{BHN})\ \text{kgf/mm}^2 = 0.27(9.81)(\text{BHN})\ \text{N/mm}^2 \qquad (17.22)$$	

Table 17.17 *Values of Modulus of elasticity and Poisson's ratio for gear materials*

Material	Modulus of elasticity E (MPa or N/mm^2)	Poisson's ratio
Steel	206 000	0.3
Cast steel	202 000	0.3
Spheroidal cast iron	173 000	0.3
Cast tin bronze	103 000	0.3
Tin bronze	113 000	0.3
Grey cast iron	118 000	0.3

Table 17.18 *Effective load on gear tooth*

Tangential force due to rated torque or rated power (P_t)	
$$M_t = \frac{60\times10^6(\text{kW})}{2\pi n_p} \qquad (17.23)$$ $$P_t = \frac{2M_t}{d_p} \qquad (17.24)$$	P_t = tangential force due to rated torque (N) M_t = rated torque (N-mm) kW = power transmitted by gears (kW) n_p = speed of pinion (rpm) d_p = pitch circle diameter of pinion (mm)
Effective load on Gear tooth (P_{eff}) – Preliminary gear design	
$$P_{eff} = \frac{C_S P_t}{C_v} \qquad (17.25)$$	P_{eff} = effective load on gear tooth (N) C_s = service factor (Tables 17.19) C_v = velocity factor (Table 17.22)
Effective load on Gear tooth (P_{eff}) – Final gear design	
$$P_{eff} = (C_s P_t + P_d) \qquad (17.26)$$	P_d = incremental dynamic load (N) (Buckingham's equation) (Table 17.23)

Table 17.19 *Service factor for speed reduction gearboxes (C_s)*

Working characteristics of Driving machine (Table 17.20)	Working characteristics of Driven machine (Table 17.21)		
	Uniform	*Moderate shock*	*Heavy shock*
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.5	1.75	2.25

Note: For electric motors, $C_s = \frac{\text{starting torque}}{\text{rated torque}}$ (17.27)

Table 17.20 *Examples of driving machines with different working characteristics*

Character of operation	Driving machines
Uniform	Electric motor, steam turbine, gas turbine
Light shock	Multi-cylinder internal combustion engine
Medium shock	Single cylinder internal combustion engine

Table 17.21 *Examples of driven machines with different working characteristics*

Character of operation	Driven machines
Uniform	Generator, belt conveyor, platform conveyor, light elevator, electric hoist, feed gears of machine tools, ventilators, turbo-blower, mixer for constant density material
Medium shock	Main drive to machine tool, heavy elevator, turning gears of crane, mine ventilator, mixer for variable density material, multi-cylinder piston pump, feed pump
Heavy shock	Press, shear, rubber dough mill, rolling mill drive, power shovel, heavy centrifuge, heavy feed pump, rotary drilling apparatus, briquette press, pug mill

Table 17.22 *Velocity factor (C_v)*

For ordinary and commercially cut gears made with form cutters and with (v < 10 m/s),

$$C_v = \frac{3}{3+v} \qquad (17.28)$$

For accurately hobbed and generated gears with (v < 20 m/s),

$$C_v = \frac{6}{6+v} \qquad (17.29)$$

For precision gears with shaving, grinding and lapping operations and with (v > 20m/s),

$$C_v = \frac{5.6}{5.6+\sqrt{v}} \qquad (17.30)$$

$v = \frac{\pi d_p n_p}{60 \times 10^3}$ (17.31)	v = pitch line velocity (m/s)

Table 17.23 *Incremental dynamic load (P_d) (Buckingham's equation)*

$$P_d = \frac{21v(Ceb + P_t)}{21v + \sqrt{(Ceb + P_t)}} \quad (17.32)$$	P_d = dynamic load or incremental dynamic load (N) v = pitch line velocity (m/s) C = deformation factor (MPa or N/mm^2) (Tables 17.24 and 17.25) e = sum of errors between two meshing teeth (mm) (Table17.26) b = face width of tooth (mm) P_t = tangential force due to rated torque (N)

Table 17.24 *Deformation factor (C)*

The deformation factor C depends upon modulii of elasticity of materials for pinion and gear and the form of tooth or pressure angle.	
$$C = \frac{k}{\left[\frac{1}{E_p} + \frac{1}{E_g}\right]} \quad (17.33)$$	k = constant depending upon the form of tooth E_p = modulus of elasticity of pinion material (MPa or N/mm^2) E_g = modulus of elasticity of wheel material (MPa or N/mm^2)
The values of k for various tooth forms are as follows, $k = 0.107$ (for 14.5° full depth teeth) $k = 0.111$ (for 20° full depth teeth) $k = 0.115$ (for 20° stub teeth)	

Table 17.25 *Values of deformation factor C*

Materials		14.5° full depth teeth	20° full depth teeth	20° stub teeth
Pinion material	**Gear material**			
Grey C.I.	Grey C.I.	5500	5700	5900
Steel	Grey C.I.	7600	7900	8100
Steel	Steel	11 000	11 400	11 900

Table 17.26 *Errors between two meshing teeth (e)*

$$e = e_p + e_g \quad (17.34)$$	e = sum of errors between two meshing teeth (mm) e_p = error for pinion(mm) (Table 17.27) e_g = error for wheel(mm) (Table 17.27)
The expected error on gear tooth is considered to be equal to tolerance.	
$$\phi = m + 0.25\sqrt{d} \quad (17.35)$$	ϕ = tolerance factor m = module (mm) d= pitch circle diameter (mm)

Table 17.27 *Tolerances on the adjacent pitch*

Grade	e (microns)
1	$0.80 + 0.06\phi$
2	$1.25 + 0.10\phi$
3	$2.00 + 0.16\phi$
4	$3.20 + 0.25\phi$
5	$5.00 + 0.40\phi$
6	$8.00 + 0.63\phi$
7	$11.00 + 0.90\phi$
8	$16.00 + 1.25\phi$
9	$22.00 + 1.80\phi$
10	$32.00 + 2.50\phi$
11	$45.00 + 3.55\phi$
12	$63.00 + 5.00\phi$

Note: In above table, the value of '*e*' is given in microns (μm) (1 μm = 10^{-3} mm)

Helical Gear Design Theoretical Approach

18

18.1 BASIC RELATIONSHIPS

Table 18.1 *Basic equations of helical gears*

$p_n = p \cos \psi$ (18.1)	p_n = normal circular pitch (mm) p = transverse circular pitch (mm) ψ = helix angle (deg) (15° to 25°)
$P_n = \dfrac{P}{\cos \psi}$ (18.2)	P_n = normal diametral pitch (mm) P = transverse diametral pitch (mm)
$m_n = m \cos \psi$ (18.3)	m_n = normal module (mm) m = transverse module (mm)
$p_a = \dfrac{p}{\tan \psi}$ (18.4)	p_a = axial pitch (mm)
$\cos \psi = \dfrac{\tan \alpha_n}{\tan \alpha}$ (18.5)	α_n = normal pressure angle (°) (usually 20°) α = transverse pressure angle (°)
$d = \dfrac{zm_n}{\cos \psi}$ (18.6)	d = pitch circle diameter (mm) z = number of teeth
$a = \dfrac{m_n(z_p + z_g)}{2 \cos \psi}$ (18.7)	a = centre-to-centre distance (mm) z_p = number of teeth on pinion z_g = number of teeth on wheel
$i = \dfrac{\omega_p}{\omega_g} = \dfrac{z_g}{z_p}$ (18.8)	i = speed ratio

Table 18.2 *Virtual number of teeth*

In the design of helical gears, an imaginary spur gear is considered in a plane which is perpendicular to tooth elements and which is having a pitch circle radius of r' and module m_n. It is called a 'formative' or 'virtual' spur gear. The pitch circle radius r' of the virtual gear is given by, $$r' = \frac{d}{2\cos^2\psi} \qquad (18.9)$$ The number of teeth z' on this imaginary spur gear is called the virtual number of teeth.	
$z' = \dfrac{z}{\cos^3\psi}$ (18.10)	z' = virtual number of teeth z = actual number of teeth

Table 18.3 *Recommended series of normal module (m_n)*

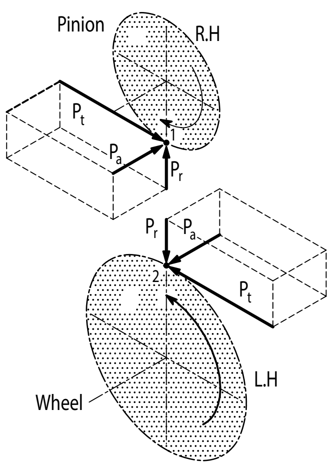
m_n (in mm) = 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8 and 10

Table 18.4 *Standard proportions of helical gears*

addendum (h_a) = m_n dedendum (h_f) = 1.25 m_n clearance (c) = 0.25 m_n	
$d_a = m_n\left[\dfrac{z}{\cos\psi} + 2\right]$ (18.11)	d_a = addendum circle diameter (mm)
$d_f = m_n\left[\dfrac{z}{\cos\psi} - 2.5\right]$ (18.12)	d_f = dedendum circle diameter (mm)

18.2 COMPONENTS OF TOOTH FORCE

Table 18.5 *Helical gears – Components of tooth force*

Magnitude of three components	
	$M_t = \dfrac{60\times10^6(\text{kW})}{2\pi n_p}$ (18.13) $P_t = \dfrac{2M_t}{d_p}$ (18.14) $P_r = P_t\left[\dfrac{\tan\alpha_n}{\cos\psi}\right]$ (18.15) $P_a = P_t\tan\psi$ (18.16) $d_p = \dfrac{z_p m_n}{\cos\psi}$ where suffix 'p' is used for pinion M_t = torque transmitted by gears (N-mm) (kW) = power transmitted by gears (kW) n_p = speed of pinion (rpm) P_t = tangential component of resultant tooth force (N) P_r = radial component of resultant tooth force (N) P_a = axial or thrust component of resultant tooth force (N) d_p = pitch circle diameter of pinion (mm) α_n = normal pressure angle (°)

(*Contd.*)

(*Contd.*)

Direction of Three Components
Following information is required in order to decide the direction of three components: (i) Which is the driving element? Which is the driven element? (ii) Whether the pinion is rotating in clockwise or anti-clockwise direction? (iii) What is the hand of helix? Whether right hand or left hand? (a) Tangential component (P_t): (i) The direction of tangential component for driving gear is opposite to the direction of rotation. (ii) The direction of tangential component for driven gear is same as the direction of rotation. (b) Radial component (P_r): (i) The radial component on pinion acts towards the centre of pinion. (ii) The radial component on wheel acts towards the centre of wheel. (c) Thrust component (P_a): The following guidelines can be used to determine the direction of the thrust component: (i) Select the driving gear from the pair. (ii) Use right hand for RH-helix and left hand for LH-helix. (iii) Keep the fingers in the direction of rotation of the gear and the thumb will indicate the direction of the thrust component for the driving gear. (iv) The direction of the thrust component for the driven gear will be opposite to that for the driving gear.

18.3 DESIGN BY LEWIS AND BUCKINGHAM'S EQUATIONS

Table 18.6 *Beam strength of gear tooth (Lewis' equation)*

Beam strength (S_b) indicates the maximum value of tangential force that the tooth can transmit without bending failure.	
$S_b = m_n b \sigma_b Y$ (18.17)	S_b = beam strength of gear tooth (N) σ_b = permissible bending stress (MPa or N/mm^2) Y = Lewis form factor based on virtual number of teeth (z') (Table 17.15)
$\sigma_b = S_e = \left(\frac{1}{3}\right) S_{ut}$ (18.18)	S_e = endurance limit (MPa or N/mm^2) S_{ut} = ultimate tensile strength (MPa or N/mm^2)

Table 18.7 *Wear strength of gear tooth (Buckingham's equation)*

Wear strength (S_w) indicates the maximum value of tangential force that the tooth can transmit without pitting failure.	
$S_w = \frac{bQd_pK}{\cos^2 \psi}$ (18.19)	S_w = wear strength of the gear tooth (N) Q = ratio factor d_p = pitch circle diameter of pinion (mm) K = load – stress factor (MPa or N/mm^2) ψ = helix angle (°)
$Q = \frac{2z_g}{z_g + z_p}$ (for external gears) (18.20) $Q = \frac{2z_g}{z_g - z_p}$ (for internal gears) (18.21)	z_p = number of teeth on pinion z_g = number of teeth on wheel

(*Contd.*)

(*Contd.*)

$K = \dfrac{\sigma_c^2 \sin\alpha_n \cos\alpha_n (1/E_p + 1/E_g)}{1.4}$ (18.22)	σ_c = surface endurance strength of the material (MPa or N/mm^2) α_n = normal pressure angle (°) E_p = modulus of elasticity of pinion materials (MPa or N/mm^2) (Table 17.17) E_g = modulus of elasticity of wheel materials (MPa or N/mm^2) (Table 17.17)
For steel gears with 20° pressure angle, $K \approx 0.16\left(\dfrac{\text{BHN}}{100}\right)^2$ (18.23)	BHN = surface hardness of gears (Brinell Hardness Number)
According to G. Niemann, $\sigma_c = 0.27(\text{BHN})\ \text{kgf/mm}^2 = 0.27(9.81)(\text{BHN})\ \text{N/mm}^2$ (18.24)	

Table 18.8 *Effective load on gear tooth*

Tangential force due to rated torque or rated power (P_t)	
$M_t = \dfrac{60 \times 10^6 (\text{kW})}{2\pi n_p}$ (18.25) $P_t = \dfrac{2M_t}{d_p}$ (18.26)	P_t = tangential force due to rated torque (N) M_t = rated torque (N-mm) kW = power transmitted by gears (kW) n_p = speed of pinion (rpm) d_p = pitch circle diameter of pinion (mm)
Effective load on Gear tooth (P_{eff}) – Preliminary gear design	
$P_{\text{eff}} = \dfrac{C_S P_t}{C_v}$ (18.27) $C_v = \dfrac{5.6}{5.6 + \sqrt{v}}$ (18.28)	P_{eff} = effective load on gear tooth (N) C_s = service factor (Tables 17.19) C_v = velocity factor v = pitch line velocity (m/s)
Effective load on gear tooth (P_{eff}) – Final gear design	
$P_{\text{eff}} = (C_s P_t + P_d)$ (18.29)	P_d = incremental dynamic load (N) (Buckingham's equation) (Table 18.9)

Table 18.9 *Incremental dynamic load (P_d) (Buckingham's equation)*

$P_d = \dfrac{21v(Ceb\cos^2\psi + P_t)\cos\psi}{21v + \sqrt{(Ceb\cos^2\psi + P_t)}}$ (18.30)	P_d = dynamic load or incremental dynamic load (N) v = pitch line velocity (m/s) C = deformation factor (N/mm^2) (Tables 17.24 and 17.25) e = sum of errors between two meshing teeth (mm) (Table17.26 and 17.27) b = face width of tooth (mm) P_t = tangential force due to rated torque (N) ψ = helix angle (°)

Bevel Gear Design Theoretical Approach

19

19.1 BASIC RELATIONSHIPS

Table 19.1 *Basic equations of bevel gears*

$z = \dfrac{D}{m}$ (19.1)	D = pitch circle diameter at large end of tooth (mm) m = module at large end of tooth (mm) z = number of teeth on gear
$\tan \gamma = \dfrac{z_p}{z_g}$ (19.2)	γ = pitch angle of pinion (°) z_p = number of teeth on pinion z_g = number of teeth on wheel
$\tan \Gamma = \dfrac{z_g}{z_p}$ (19.3) $\gamma + \Gamma = \dfrac{\pi}{2}$ (usually) (19.4)	Γ = pitch angle of wheel (°)
$A_o = \sqrt{\left(\dfrac{D_p}{2}\right)^2 + \left(\dfrac{D_g}{2}\right)^2}$ (19.5)	A_o = cone distance (mm) D_p = pitch circle diameter of pinion at large end of tooth (mm) D_g = pitch circle diameter of wheel at large end of tooth (mm)

Note: The dimensions of the bevel gear are always specified and measured at the large end of the tooth. The addendum h_a, the dedendum h_f and the pitch circle diameter D are specified at the large end of the tooth.

Table 19.2 *Virtual number of teeth*

In the design of bevel gears, an imaginary spur gear is considered in a plane which is perpendicular to tooth elements at the large end and which is having a pitch circle radius of r_b and module m. It is called a 'formative' or 'virtual' spur gear. The pitch circle radius r_b of the virtual gear is given by,

$$r_b = \frac{D}{2 \cos \gamma} \tag{19.6}$$

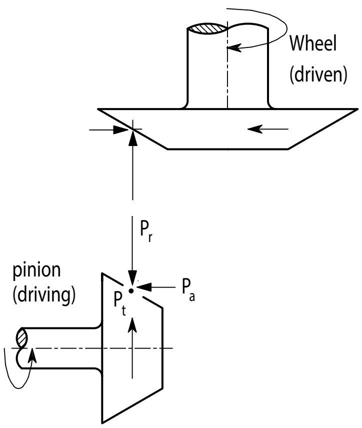
The number of teeth z' on this imaginary spur gear is called the virtual number of teeth.

$z' = \dfrac{z}{\cos \gamma}$ (19.7)	z' = virtual number of teeth z = actual number of teeth

19.2 COMPONENTS OF TOOTH FORCE

Table 19.3 *Bevel gears – Components of tooth force*

Magnitude of Three Components



$$M_t = \frac{60 \times 10^6 (\text{kW})}{2\pi n_p} \tag{19.8}$$

$$r_m = \left[\frac{D_p}{2} - \frac{b \sin \gamma}{2}\right] \tag{19.9}$$

$$P_t = \frac{M_t}{r_m} \tag{19.10}$$

$$P_r = P_t \tan \alpha \cos \gamma \tag{19.11}$$

$$P_a = P_t \tan \alpha \sin \gamma \tag{19.12}$$

where suffix '*p*' is used for pinion

Notations:

M_t = torque transmitted by gears (N-mm)
(kW) = power transmitted by gears (kW)
n_p = speed of pinion (rpm)
P_t = tangential component of resultant tooth force (N)
P_r = radial component of resultant tooth force (N)
P_a = axial or thrust component of resultant tooth force (N)
r_m = radius of the pinion at midpoint along the face width (mm)
D_p = pitch circle diameter of pinion at large end of tooth (mm)
b = face width of the tooth (mm)
α = pressure angle (°)
γ = pitch angle of pinion (°)

Note: The notations (P_t), (P_r) and (P_a) are used for components of tooth force on pinion only. The radial component on the wheel is equal to the axial component (P_a) on the pinion. Similarly, the axial component on the wheel is equal to the radia1 component (P_r) on the pinion.

Direction of Three Components

(a) Tangential component (P_t):
  (i) The direction of tangential component for driving gear is opposite to the direction of rotation.
  (ii) The direction of tangential component for driven gear is same as the direction of rotation.
(b) Radial component (P_r):
  (i) The radial component on pinion acts towards the centre of pinion.
  (ii) The radial component on wheel acts towards the centre of wheel.
(c) Thrust component (P_a):
  The following guidelines can be used to determine the direction of the thrust component:
  (i) The thrust component on pinion is equal and opposite of the radial component on the wheel.
  (ii) The thrust component on wheel is equal and opposite of the radial component on the pinion.
  The tendency of thrust components is to separate the meshing teeth. This fact is useful in deciding their directions.

Note: In two-dimensional representation of forces, very often (·) and (×) are used to indicate forces perpendicular to the plane of paper. (·) indicates a force that is perpendicular to the plane of paper and which is towards the observer. (×) indicates a force that is perpendicular to the plane of paper and which is away from the observer.

19.3 DESIGN BY LEWIS AND BUCKINGHAM'S EQUATIONS

Table 19.4 *Beam strength of gear tooth (Lewis equation)*

Beam strength (S_b) indicates the maximum value of tangential force at the large end of tooth that the tooth can transmit without bending failure.	
$S_b = mb\sigma_b Y\left[1 - \dfrac{b}{A_o}\right]$ (19.13)	S_b = beam strength of the tooth (N) m = module at the large end of the tooth (mm) b = face width (mm) σ_b = permissible bending stress(N/mm^2) Y = Lewis form factor based on formative number of teeth (Table 17.15) A_o = cone distance (mm)
$\sigma_b = S_e = \left(\dfrac{1}{3}\right)S_{ut}$ (19.14)	S_e = endurance limit (MPa or N/mm^2) S_{ut} = ultimate tensile strength (MPa or N/mm^2)
b = (10 m) or $b = (A_o/3)$ (whichever is smaller) (19.15)	

Table 19.5 *Wear strength of gear tooth (Buckingham's equation)*

Wear strength (S_w) indicates the maximum value of tangential force at the large end of tooth that the tooth can transmit without pitting failure.	
$S_w = \dfrac{0.75bQD_pK}{\cos\gamma}$ (19.16)	S_w = wear strength of the gear tooth (N) b = face width of gear (mm) Q = ratio factor D_p = pitch circle diameter of pinion at the large end of tooth (mm) K = load – stress factor (MPa or N/mm^2) γ = pitch angle of pinion (°)
$Q = \dfrac{2z_g}{z_g + z_p\tan\gamma}$ (19.17)	z_p = number of teeth on pinion z_g = number of teeth on wheel
$K = \dfrac{\sigma_c^2 \sin\alpha\cos\alpha(1/E_p + 1/E_g)}{1.4}$ (19.18)	σ_c = surface endurance strength of the material (MPa or N/mm^2) α = normal pressure angle (°) E_p = modulus of elasticity of pinion materials (MPa or N/mm^2) (Table 17.17) E_g = modulus of elasticity of wheel materials (MPa or N/mm^2) (Table 17.17)
For steel gears with 20° pressure angle, $K \approx 0.16\left(\dfrac{\text{BHN}}{100}\right)^2$ (19.19)	BHN = surface hardness of gears (Brinell Hardness Number)
According to G. Niemann, $\sigma_c = 0.27(\text{BHN})\ \text{kgf/mm}^2 = 0.27(9.81)(\text{BHN})\ \text{N/mm}^2$ (19.20)	

Table 19.6 *Effective load on gear tooth*

Tangential force due to rated torque or rated power (P_t)	
$M_t = \dfrac{60 \times 10^6 (\text{kW})}{2\pi n_p}$ (19.21) $P_t = \dfrac{2M_t}{D}$ (19.22)	P_t = tangential force due to rated torque (N) M_t = rated torque (N-mm) kW = power transmitted by gears (kW) n_p = speed of pinion (rpm) D = pitch circle diameter of pinion at the large end of tooth (mm)
Effective load on Gear tooth (P_{eff}) – Preliminary gear design	
$P_{\text{eff}} = \dfrac{C_S P_t}{C_v}$ (19.23) $C_v = \dfrac{6}{6+v}$ (for cut teeth) (19.24) $C_v = \dfrac{5.6}{5.6+\sqrt{v}}$ (for generated teeth) (19.25)	P_{eff} = effective load on gear tooth (N) C_s = service factor (Table 17.19) C_v = velocity factor v = pitch line velocity (m/s)
Effective load on Gear tooth (P_{eff}) – Final gear design	
$P_{\text{eff}} = (C_s P_t + P_d)$ (19.26)	P_d = incremental dynamic load (N) (Buckingham's equation) (Table 19.7)

Table 19.7 *Incremental dynamic load (P_d) (Buckingham's equation)*

$P_d = \dfrac{21v(Ceb + P_t)}{21v + \sqrt{(Ceb + P_t)}}$ (19.27)	P_d = dynamic load or incremental dynamic load (N) v = pitch line velocity (m/s) C = deformation factor (N/mm^2) (Tables 17.24 and 17.25) e = maximum expected error between two meshing teeth (mm) (Table19.8) b = face width of tooth (mm) P_t = tangential force due to rated torque (N)

Table 19.8 *Maximum expected error between two meshing teeth (mm)*

Module(m) (mm)	Class – 1	Class – 2	Class – 3
Up to 4	0.050	0.025	0.0125
5	0.056	0.025	0.0125
6	0.064	0.030	0.0150
7	0.072	0.035	0.0170
8	0.080	0.038	0.0190
9	0.085	0.041	0.0205
10	0.090	0.044	0.0220

Spur and Helical Gears Calculation of Load Capacities

20

20.1 BASIC DATA FOR LOAD CAPACITY (CONTACT STRESS)

Table 20.1 *Basic (given) data required for finding out the load capacity based on surface contact stress*

(1) Number of teeth on pinion (z_1)
(2) Number of teeth on wheel (z_2)
(3) Module (m) (mm) (for spur gears)
(4) Normal module (m_n) (mm) (for helical gears)
(5) Normal pressure angle (α_n) (deg)
(6) Helix angle (β) (deg.)
(7) Transmitted power (P) (W)
(8) Speed of pinion (n_1) (rpm)
(9) Pitch circle diameter of pinion (d_1) (mm)
(10) Pitch circle diameter of wheel (d_2) (mm)
(11) Outside diameter of pinion (d_{a1}) (mm)
(12) Outside diameter of wheel (d_{a2}) (mm)
(13) Working centre distance (a') (mm)
(14) Face width (b) (mm)
(15) Probability of failure (p_f)
(16) Life of gears (L_H) (hr)
(17) Accuracy grade for pinion
(18) Accuracy grade for wheel
(19) Hardness of pinion material
(20) Hardness of wheel material
(21) Viscosity grade of lubricant

20.2 PERMISSIBLE CONTACT STRESS

Table 20.2 *Permissible contact stress (σ_{HP})*

$$\sigma_{HP} = \left(\frac{\sigma_{H\lim} Z_L Z_l Z_R Z_S Z_V Z_W}{\sqrt{K_R}} \right) \tag{20.1}$$

σ_{HP} = Permissible contact stress (MPa or N/mm^2)
Z_L = Life factor for contact stress (Table 20.3)
Z_l = Lubrication factor for contact stress (Table 20.4)
Z_R = Roughness factor for contact stress (Table 20.6)
Z_S = Size factor for contact stress (Table 20.7)
Z_V = Velocity or speed factor for contact stress (Table 20.8)
Z_W = Work hardening factor for contact stress (Table 20.9)
K_R = Reliability factor (Table 20.10)
$\sigma_{H\lim}$ = Endurance limit for contact stress (MPa or N/mm^2) (Table 20.11 to 20.17)

(ISO: 6336 Part 2)

Note: (i) The endurance limit ($\sigma_{H\lim}$) is the repeated stress, which the material can sustain for at least 50×10^6 load cycles without fatigue failure. (ii) The permissible contact stress (σ_{HP}) is calculated separately for pinion and wheel from the above equation.

Table 20.3 *Life factor for contact stress (Z_L)*

Material	Condition	Z_L
(i) Through hardened steels (ii) Spheroidal graphite cast iron (iii) Malleable cast iron (iv) Surface hardened steels (with a permitted level of pitting)	$L_N \le 6 \times 10^5$	1.6
	$6 \times 10^5 \le L_N \le 10^7$	$\left(\frac{3 \times 10^8}{L_N}\right)^{0.0756}$
	$L_N \ge 10^9$	1
(i) Through hardened steels (ii) Spheroidal graphite cast iron (iii) Surface hardened steels (with no permitted level of pitting)	$L_N \le 10^5$	1.6
	$10^5 \le L_N \le 10^7$	$\left(\frac{5 \times 10^7}{L_N}\right)^{0.0756}$
	$L_N \le 5 \times 10^7$	1.0

(*Contd.*)

(*Contd.*)

(i) Gas-nitrided steels (ii) Grey cast iron	$L_N \le 10^5$	1.3
	$10^5 \le L_N \le 2\times10^6$	$\left(\dfrac{2\times10^6}{L_N}\right)^{0.0875}$
	$L_N \ge 2\times10^6$	1.0
(i) Liquid-nitrided steels	$L_N \le 10^5$	1.1
	$10^5 \le L_N \le 2\times10^6$	$\left(\dfrac{2\times10^6}{L_N}\right)^{0.0318}$
	$L_N \ge 2\times10^6$	1.0

Note: (i) The life factor (Z_L) takes into account the increase in permissible contact stress, if the number of stress cycles is less than the life at endurance limit.

(ii) Z_L = 1.0 for normal mechanical use with life more than 10^9 cycles.

(iii) $L_N = 60 \times n \times L_H$

L_N = life in number of cycles

n = speed of pinion or gear (rpm) (Given data – Table 20.1)

L_H = required life in hours (hr) (Given data – Table 20.1)

(iv) For single tooth mesh of gears [$L_N = L_N$]

(v) For intermediate gears [$L_N = 2L_N$]

(vi) For motor pinion with double mesh [$L_N = 2L_N$]

(vii) For planetary gear train with 'q' planet gears,

Sun gear [$L_N = qL_N$]

Ring gear [$L_N = qL_N$]

Planet gears [$L_N = 2L_N$]

Table 20.4 *Lubrication factor for contact stress (Z_l)*

$Z_l = C_{Zl} + \dfrac{4(1-C_{Zl})}{\left[1.2+\dfrac{80}{v_{50}}\right]^2}$ or $Z_l = C_{Zl} + \dfrac{4(1-C_{Zl})}{\left[1.2+\dfrac{134}{v_{40}}\right]^2}$	C_{Zl} = factor for calculating Z_l v_{50} = kinematic viscosity of lubricant at 50°C (mm²/s) (Table 20.5) v_{40} = kinematic viscosity of lubricant at 40°C (mm²/s) (Table 20.5)
$C_{Zl} = \left(\dfrac{\sigma_{H\lim} - 850}{350}\right) \times 0.08 + 0.83$ [if $850 \le \sigma_{H\lim} \le 1200$ MPa]	

(*Contd.*)

(*Contd.*)

$C_{Zl} = 0.83$ [if $\sigma_{H\,\lim} < 850$ MPa]
$C_{Zl} = 0.91$ [if $\sigma_{H\,\lim} > 1200$ MPa]
$\sigma_{H\,\lim}$ = endurance limit for contact stress (MPa or N/mm^2) (Table 20.11 to 20.17)

Note: The lubrication factor (Z_l) accounts for the influence of the type of lubricant and its viscosity on permissible contact stress.

Table 20.5 *Viscosity of lubricating oils (mm^2/s)*

ISO viscosity grade	Kinematic viscosity range at 40°C (ν_{40})	Kinematic viscosity range at 50°C (ν_{50})
ISO VG 2	1.98 – 2.42	1.69 – 2.03
ISO VG 3	2.88 – 3.52	2.37 – 2.83
ISO VG 5	4.14 – 5.06	3.27 – 3.91
ISO VG 7	6.12 – 7.48	4.63 – 5.52
ISO VG 10	9.00 – 11.0	6.53 – 7.83
ISO VG 15	13.5 – 16.5	9.43 – 11.3
ISO VG 22	19.8 – 24.2	13.3 – 16.0
ISO VG 32	28.8 – 35.2	18.6 – 22.2
ISO VG 46	41.4 – 50.6	25.5 – 30.3
ISO VG 68	61.2 – 74.8	35.9 – 42.8
ISO VG 100	90.0 – 110	50.4 – 60.3
ISO VG 150	135 – 165	72.5 – 86.9
ISO VG 220	198 – 242	102 – 123
ISO VG 320	288 – 352	144 – 172
ISO VG 460	414 – 506	199 – 239
ISO VG 680	612 – 748	283 – 339
ISO VG 1000	900 – 1100	400 – 479
ISO VG 1500	1350 – 1650	575 – 688
Guidelines (For students only) – (i) For high speed gears at low operating temperatures (10° to 16°C), low viscosity oil should be selected such as ISO VG 46. (ii) For medium speed gears with centre distances less than 200 mm, oils with viscosity grade ISO VG 68 or ISO VG 100 are reasonable. (iii) For medium speed gears, with centre distances more than 200 mm, oils with viscosity grade ISO VG 68 to ISO VG 220 are reasonable. (iv) For low speed gears at high operating temperatures up to 52°C, oils with viscosity grade ISO VG 150 to ISO VG 320 are satisfactory.		

Table 20.6 *Roughness factor for contact stress (Z_R)*

Gears	Z_R
Hardened and ground gears	1.0
Finish cut gears	0.85
Gears lapped after generation	0.9

Note: The roughness factor (Z_R) accounts for the influence of surface texture of tooth flanks on permissible contact stress.

Table 20.7 *Size factor for contact stress (Z_S)*

Heat Treatment	Z_S	Limiting Conditions
Case hardened gears (flame or induction hardening)	$1.05 - 0.005\, m_n$	$0.9 \le Z_S \le 1$
Nitrided gears	$1.08 - 0.011\, m_n$	$0.75 \le Z_S \le 1$
Structural or annealed gears	1	—

Note: (i) The size factor (Z_S) accounts for the influence of tooth dimensions on permissible contact stress. (ii) m_n = normal module (mm) (Given data-Table 20.1)

Table 20.8 *Velocity factor for contact stress (Z_V)*

$Z_V = C_{ZV} + \dfrac{2(1 - C_{ZV})}{\sqrt{0.8 + \dfrac{32}{v}}}$	C_{ZV} = factor for calculating Z_V v = pitch line velocity (m/s)
$C_{ZV} = 0.85 + \left(\dfrac{\sigma_{H\lim} - 850}{350}\right) \times 0.08$ [if $850 \le \sigma_{H\lim} \le 1200$ MPa]	
C_{ZV} = 0.85 [if $\sigma_{H\lim} < 850$ MPa]	
C_{ZV} = 0.93 [if $\sigma_{H\lim} > 1200$ MPa]	
$v = \dfrac{\pi d_1 n_1}{60 \times 10^3} = \dfrac{d_1 n_1}{19098}$	
$\sigma_{H\lim}$ = endurance limit for contact stress (MPa or N/mm^2) (Tables 20.11 to 20.17) d_1 = pitch circle diameter of pinion (mm) (Given data – Table 20.1) n_1 = speed of pinion (rpm) (Given data – Table 20.1)	

Note: The velocity or speed factor (Z_V) accounts for the influence of pitch line velocity on permissible contact stress.

Table 20.9 *Work hardening factor for contact stress (Z_W)*

$Z_W = 1.2 - \left(\dfrac{HB - 130}{1700}\right)$	HB = Brinell hardness of gear material (Given data – Table 20.1)
$Z_W = 1$ [if $130 < HB < 400$]	

Note: The work hardening factor (Z_W) accounts for the increase of surface durability due to meshing a through-hardened steel wheel with surface-hardened pinion. In all other cases, (Z_W) = 1.0.

Table 20.10 *Reliability factor for contact and bending stresses (K_R)*

$K_R = 1$ (corresponding to a probability of failure of 1 in 100 at rated load and required life) ($p_f = 0.01$) $K_R = [0.79 - 0.105\log(p_f)]$ (for applications requiring greater or smaller reliability factor)
p_f = probability of failure (Given data – Table 20.1)

Note: The reliability factor (K_R) is introduced in the gear design, to allow for design with a calculated risk or design with higher reliability.

20.3 DETERMINATION OF ENDURANCE LIMIT ($\sigma_{H\,LIM}$)

Table 20.11 *Determination of ($\sigma_{H\,lim}$) for normalised structural steel and cast steel*

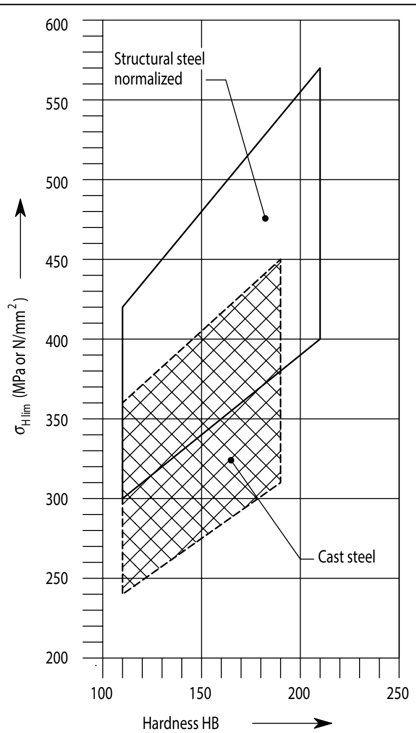


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Table 20.12 *Determination of ($\sigma_{H\,lim}$) for cast iron*

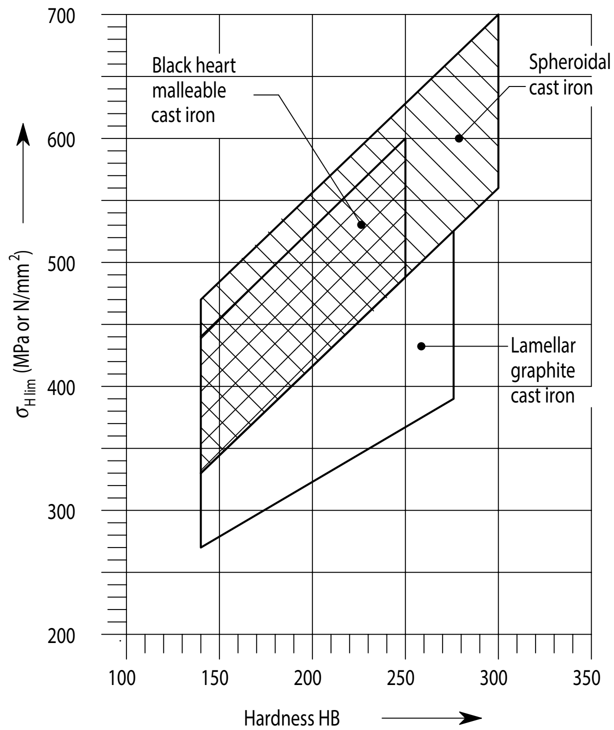


Table 20.13 *Determination of ($\sigma_{H\,lim}$) for through hardened alloy steels, carbon steels and cast steel*

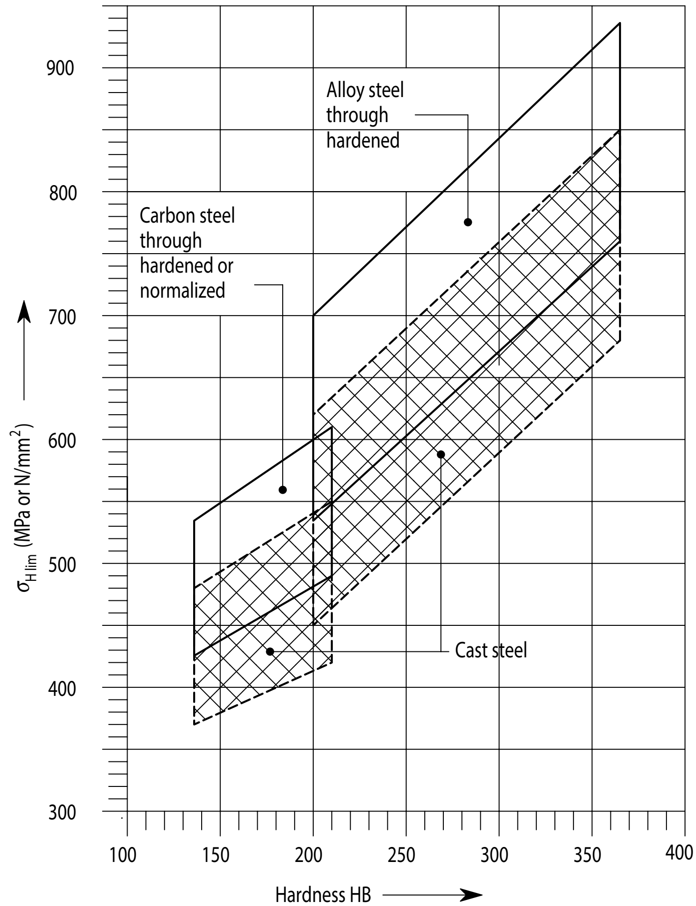


Table 20.14 *Determination of ($\sigma_{H\,lim}$) for hardenable steels and case-hardened alloy steels*

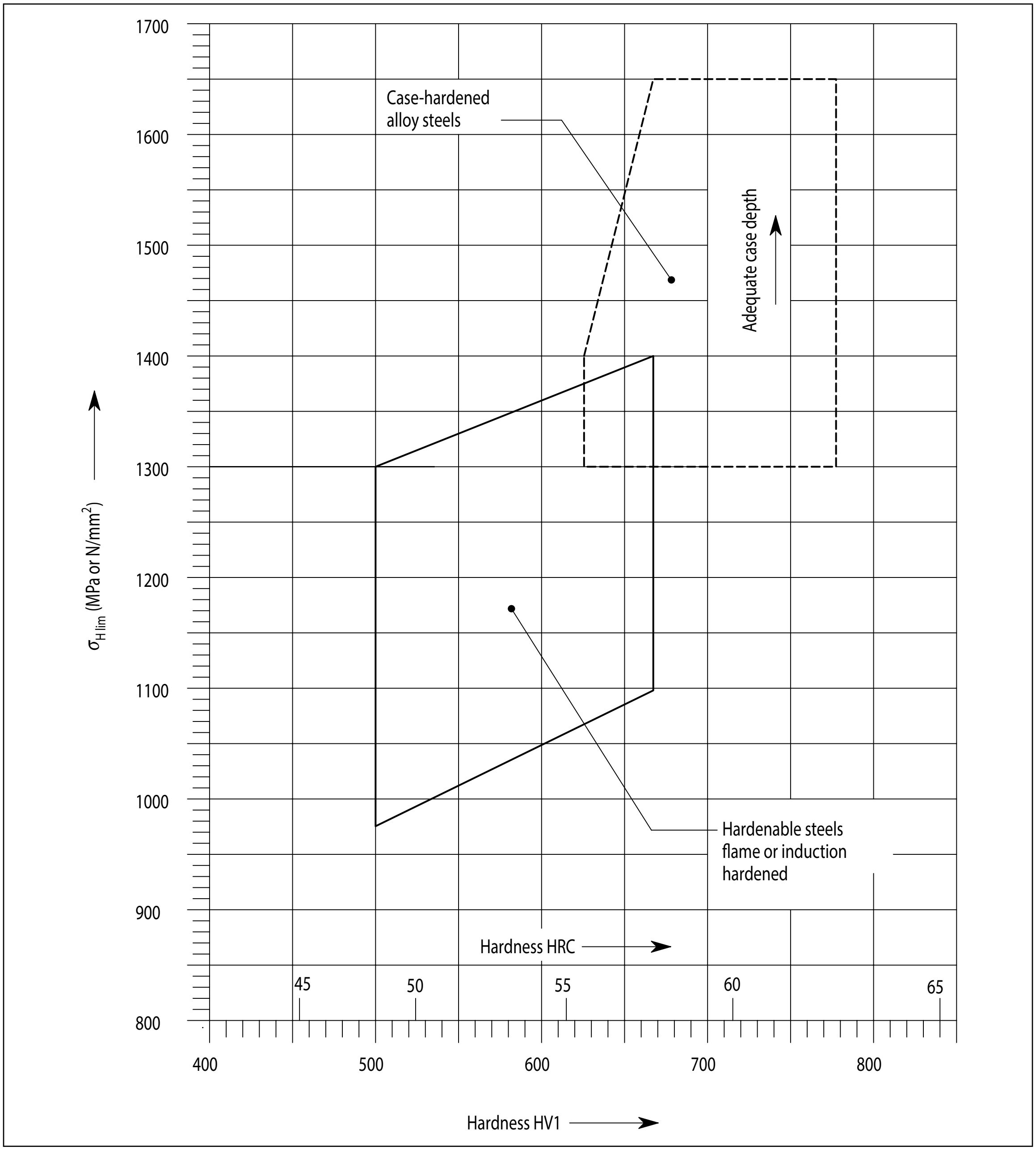
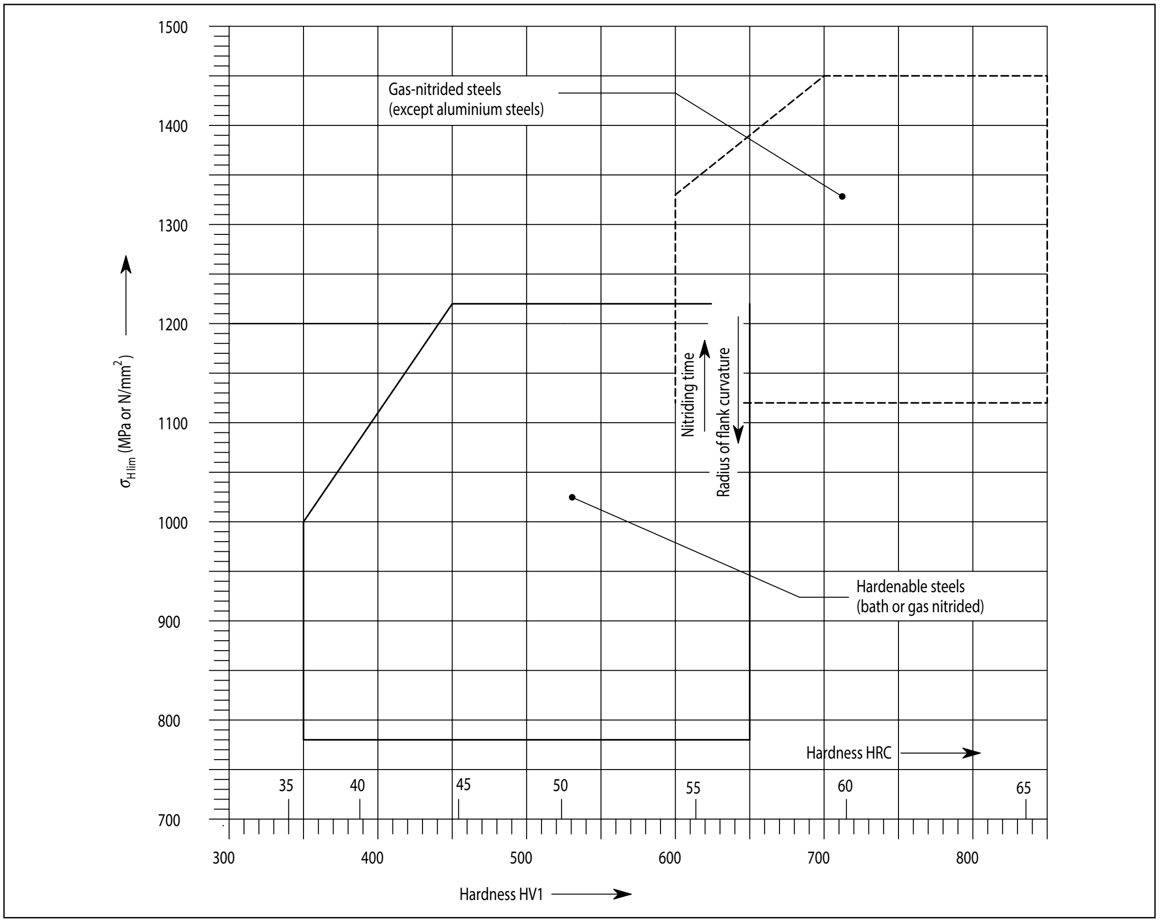


Table 20.15 *Determination of ($\sigma_{H\ lim}$) for nitriding steels*



20.4 DETERMINATION OF ENDURANCE LIMIT ($\sigma_{H\ LIM}$) – ALTERNATIVE METHOD

Table 20.16 *Endurance limit for contact stress ($\sigma_{H\ lim}$)*

The value of allowable endurance limit ($\sigma_{H\ \mathrm{lim}}$) for contact stress is given by, $$\sigma_{H\ \mathrm{lim}} = A \cdot x + B$$
x = Surface hardness (HBW or HV) (Given Data – Table 20.1) A, B = Constants for contact stress (Table 20.17) HBW, HV = units of hardness number (Table 20.17)

(ISO: 6336 – Part 5)

Note: The above equation for allowable endurance limit for contact stress ($\sigma_{H\ \mathrm{lim}}$) is based on experimental data of endurance tests of reference test gears under reference test conditions.

Table 20.17 *Constants A and B for contact stress in calculation of* ($\sigma_{H\,lim}$)

Material	Type	Quality	A	B	Units of Hardness	Min. hardness	Max. hardness
Low carbon steels	Wrought normalized low carbon steel	ML/MQ	1.00	190	HBW	110	210
		ME	1.52	250	HBW	110	210
Cast steels	—	ML/MQ	0.986	131	HBW	140	210
		ME	1.143	237	HBW	140	210
Cast iron	Black malleable cast iron	ML/MQ	1.371	143	HBW	135	250
		ME	1.333	267	HBW	175	250
	Nodular cast iron	ML/MQ	1.434	211	HBW	175	300
		ME	1.500	250	HBW	200	300
	Grey cast iron	ML/MQ	1.033	132	HBW	150	240
		ME	1.465	122	HBW	175	275
Through hard-ened wrought steels	Carbon steels	ML	0.963	283	HV	135	210
		MQ	0.925	360	HV	135	210
		ME	0.838	432	HV	135	210
	Alloy steels	ML	1.313	188	HV	200	360
		MQ	1.313	373	HV	200	360
		ME	2.213	260	HV	200	390
Case hardened wrought steels	—	ML	0.0	1300	HV	600	800
		MQ	0.0	1500	HV	660	800
		ME	0.0	1650	HV	660	800
Flame or induction hard-ened wrought steels	—	ML	0.740	602	HV	485	615
		MQ	0.541	882	HV	500	615
		ME	0.505	1013	HV	500	615
Nitrided steels	Nitrided steels	ML	0.0	1125	HV	650	900
		MQ	0.0	1250	HV	650	900
		ME	0.0	1450	HV	650	900
	Through hard-ened Nitrided steels	ML	0.0	788	HV	450	650
		MQ	0.0	998	HV	450	650
		ME	0.0	1217	HV	450	650

(*Contd.*)

(*Contd.*)

Nitro-carburised steels	Through hard-ened steels	ML	0.0	650	HV	300	650
		MQ/ME	1.167	425	HV	300	450

Note: ML, MQ and ME are material quality grades.

(i) ML stands for modest demands on material quality and on heat treatment process during gear manufacturing.

(ii) MQ stands for requirements that can be met by experienced manufacturers at moderate cost.

(iii) ME stands for requirements that must be realised when a high degree of operating reliability is required.

20.5 WORKING (ACTUAL) CONTACT STRESS

Table 20.18 *Working (actual) contact stress (σ_H)*

$$\sigma_H = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{bd_1}\left(\frac{u+1}{u}\right) K_A K_V K_{H\beta} K_{H\alpha}} \tag{20.2}$$

σ_H = working or actual contact stress (MPa or N/mm^2)
Z_H = zone factor for contact stress (Table 20.19)
Z_E = elasticity factor for contact stress (Table 20.20)
Z_ε = contact ratio factor for contact stress (Table 20.22)
Z_β = helix angle factor for contact stress (Table 20.24)
F_t = tangential force (nominal) at pitch circle appropriate to pinion or wheel calculations respectively (N)
b = face width (mm)
d_1= pitch circle diameter of pinion (mm)
u = gear ratio (z_2/z_1)
K_A = application factor (Table 20.25)
K_V = dynamic load factor (Table 20.28)
$K_{H\beta}$ = longitudinal load distribution factor for contact stress (Table 20.31)
$K_{H\alpha}$ = transverse load distribution factor for contact stress (Table 20.32)

$$F_t = \frac{P}{v} \text{ and } v = \frac{\pi d_1 n_1}{60 \times 10^3} = \frac{d_1 n_1}{19098} \tag{20.3}$$

P = transmitted power (W)
v = pitch line velocity (m/s)
n_1= speed of pinion (rpm)

(ISO: 6336 – Part 2)

Note: The working contact stress (σ_H) is calculated separately for pinion and wheel from the above equation.

Table 20.19 *Zone factor for contact stress (Z_H)*

1. Transverse module (m_t)
$$m_t = \frac{m_n}{\cos\beta}$$
m_n = normal module (mm) (Given data – Table20.1) β = helix angle (deg) (Given Data – Table 20.1)
2. Transverse pressure angle (α_t)
$$\tan\alpha_t = \frac{\tan\alpha_n}{\cos\beta}$$
α_n = normal pressure angle (deg) (Given data – Table20.1)
3. Transverse working pressure angle (α_{tw})
$$\cos\alpha_{tw} = \frac{m_t(z_1+z_2)\cos\alpha_t}{2a'}$$
a' = working centre distance (mm) (Given data – Table20.1) z_1 = number of teeth on pinion (Given data – Table 20.1) z_2 = number of teeth on wheel (Given data – Table 20.1)
4. Base helix angle (β_b)
$$\sin\beta_b = \sin\beta\cos\alpha_n$$
5. Zone factor (Z_H)
$$Z_H = \sqrt{\left(\frac{2\cos\beta_b}{\cos^2\alpha_t}\frac{\cos\alpha_{tw}}{\sin\alpha_{tw}}\right)}$$ For gears at standard centres, $$Z_H = 2\sqrt{\frac{\cos\beta_b}{\sin(2\alpha_t)}}$$

Note: The zone factor accounts for the influence of tooth flank curvature at pitch point on working (actual) contact stress and converts tangential force at reference cylinder to a normal force.

Table 20.20 *Elasticity factor for contact stress (Z_E)*

$$Z_E = \sqrt{\frac{1}{\pi\left[\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}\right]}}$$
E_1 = modulus of elasticity of pinion material (MPa or N/mm^2) (Table 20.21) E_2 = modulus of elasticity of wheel material (MPa or N/mm^2) (Table 20.21) μ_1 = Poisson's ratio of pinion material (Table 20.21) μ_2 = Poisson's ratio of wheel material (Table 20.21)

Note: The elasticity factor accounts for the influence of material properties viz. modulus of elasticity and Poisson's ratio on working (actual) contact stress.

Table 20.21 *Modulus of elasticity and Poisson's ratio for gear materials*

Material	Modulus of elasticity E (MPa or N/mm^2)	Poisson's ratio
Steel	206 000	0.3
Cast steel	202 000	0.3
Spheroidal cast iron	173 000	0.3
Cast tin bronze	103 000	0.3
Tin bronze	113 000	0.3
Grey cast iron	118 000	0.3

Table 20.22 *Contact ratio factor for contact stress (Z_ε)*

For spur gears,

$$Z_\varepsilon = \sqrt{\frac{4-\varepsilon_\alpha}{3}}$$

For helical gears,

If $\varepsilon_\beta < 1$ $$Z_\varepsilon = \sqrt{\frac{(4-\varepsilon_\alpha)(1-\varepsilon_\beta)}{3} + \frac{\varepsilon_\beta}{\varepsilon_\alpha}}$$

For helical gears,

If $\varepsilon_\beta \geq 1$ $$Z_\varepsilon = \sqrt{\frac{1}{\varepsilon_\alpha}}$$

ε_α = Transverse contact ratio (Table 20.23)
ε_β = Overlap ratio (Table 20.23)

Note: The contact ratio factor accounts for the load sharing influence of the transverse contact ratio and the overlap ratio on working (actual) contact stress.

Table 20.23 *Normal contact ratio ($\varepsilon_{\alpha n}$) and overlap ratio (ε_β)*

1. Base circle diameters of pinion and wheel (d_{b1} and d_{b2})

$$d_{b1} = z_1 m_n \frac{\cos\alpha_n}{\cos\beta_b}$$

$$d_{b2} = z_2 m_n \frac{\cos\alpha_n}{\cos\beta_b}$$

d_{b1} = base circle diameter of pinion (mm)
d_{b2} = base circle diameter of wheel (mm)
z_1 = number of teeth on pinion (Given data – Table 20.1)
z_2 = number of teeth on wheel (Given data – Table 20.1)

(*Contd.*)

m_n = normal module (mm) (Given data – Table 20.1) α_n = normal pressure angle (deg) (Given data – Table 20.1) β_b = base helix angle (deg) (Table 20.19)
2. Tip pressure angles for pinion and wheel (α_{a1} and α_{a2})
$$\cos\alpha_{a1} = \left(\frac{d_{b1}}{d_{a1}}\right)$$ $$\cos\alpha_{a2} = \left(\frac{d_{b2}}{d_{a2}}\right)$$
α_{a1} = tip pressure angle for pinion (deg) α_{a2} = tip pressure angle for wheel (deg) d_{a1} = outside diameter of pinion (mm) (Given data – Table 20.1) d_{a2} = outside diameter of wheel (mm) (Given data – Table 20.1)
3. Transverse contact ratio for pinion and wheel ($\varepsilon_{\alpha1}$ and $\varepsilon_{\alpha2}$)
$$\varepsilon_{\alpha1} = \frac{z_1(\tan\alpha_{a1} - \tan\alpha_{tw})}{2\pi}$$ $$\varepsilon_{\alpha2} = \frac{z_2(\tan\alpha_{a2} - \tan\alpha_{tw})}{2\pi}$$
$\varepsilon_{\alpha1}$ = transverse contact ratio for pinion $\varepsilon_{\alpha2}$ = transverse contact ratio for wheel α_{tw} = transverse working pressure angle (deg) (Table 20.19)
4. Transverse contact ratio (ε_α)
$$\varepsilon_\alpha = \varepsilon_{\alpha1} + \varepsilon_{\alpha2}$$
ε_α = transverse contact ratio
5. Normal contact ratio ($\varepsilon_{\alpha n}$)
$$\varepsilon_{\alpha n} = \frac{\varepsilon_\alpha}{\cos^2\beta_b}$$
6. Overlap ratio (ε_β)
$$\varepsilon_\beta = \frac{b\sin\beta}{\pi m_n}$$
ε_β = overlap ratio b = face width (Given data – Table 20.1)

Table 20.24 *Helix angle factor for contact stress (Z_β)*

$Z_\beta = \sqrt{\cos\beta}$
β = helix angle (deg) (Given data – Table 20.1)

Note: The helix angle factor accounts for the influence of helix angle on working (actual) contact stress.

Table 20.25 *Application factor for speed reducing gears (K_A)*

Working characteristics of Driving machine (Table 20.26)	Working characteristics of Driven machine (Table 20.27)		
	Uniform	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.5	1.75	2.25

Note: (i) The application factor accounts for dynamic overloads from the mean load due to sources external to gearing.
(ii) For step-up drives, multiply the values of (K_A) by 1.1.

Table 20.26 *Examples of driving machines with different working characteristics*

Character of operation	Driving machines
Uniform	Electric motor, steam turbine, gas turbine
Light shock	Multi-cylinder internal combustion engine
Medium shock	Single cylinder internal combustion engine

Table 20.27 *Examples of driven machines with different working characteristics*

Character of operation	Driven machines
Uniform	Generator, belt conveyor, platform conveyor, light elevator, electric hoist, feed gears of machine tools, ventilators, turbo-blower, mixer for constant density material
Moderate shock	Main drive to machine tool, heavy elevator, turning gears of crane, mine ventilator, mixer for variable density material, multi-cylinder piston pump, feed pump
Heavy shock	Press, shear, rubber dough mill, rolling mill drive, power shovel, heavy centrifuge, heavy feed pump, rotary drilling apparatus, briquette press, pug mill

Table 20.28 *Dynamic load factor (K_V)*

For spur gears $K_V = K_{V\alpha}$
For helical gears with ($\varepsilon_\beta \geq 1$) $K_V = K_{V\beta}$

(*Contd.*)

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For helical gears with ($\varepsilon_\beta < 1$) $K_V = K_{V\alpha} - \varepsilon_\beta(K_{V\alpha} - K_{V\beta})$
$K_{V\alpha}$ = factor for calculating (K_V) (Table 20.29) $K_{V\beta}$ = factor for calculating (K_V) (Table 20.30) ε_β = overlap ratio (Table 20.23)
$A = \frac{z_1 v}{100}\sqrt{\frac{u^2}{1+u^2}}$
A = factor for finding out values of ($K_{V\alpha}$) and ($K_{V\beta}$) z_1 = number of teeth on pinion (Given data – Table 20.1) v = pitch line velocity (m/s) (Table 20.18) u = gear ratio (z_2/z_1) (Given data – Table 20.1)

Note: The dynamic load factor (K_V) is the ratio of maximum force, which occurs at the mesh of an actual gear pair to corresponding force due to externally applied load. It accounts for the influence of tooth accuracy, natural frequency and inertia forces acting on pinion and wheel.

Table 20.29 *Factor ($K_{V\alpha}$)*

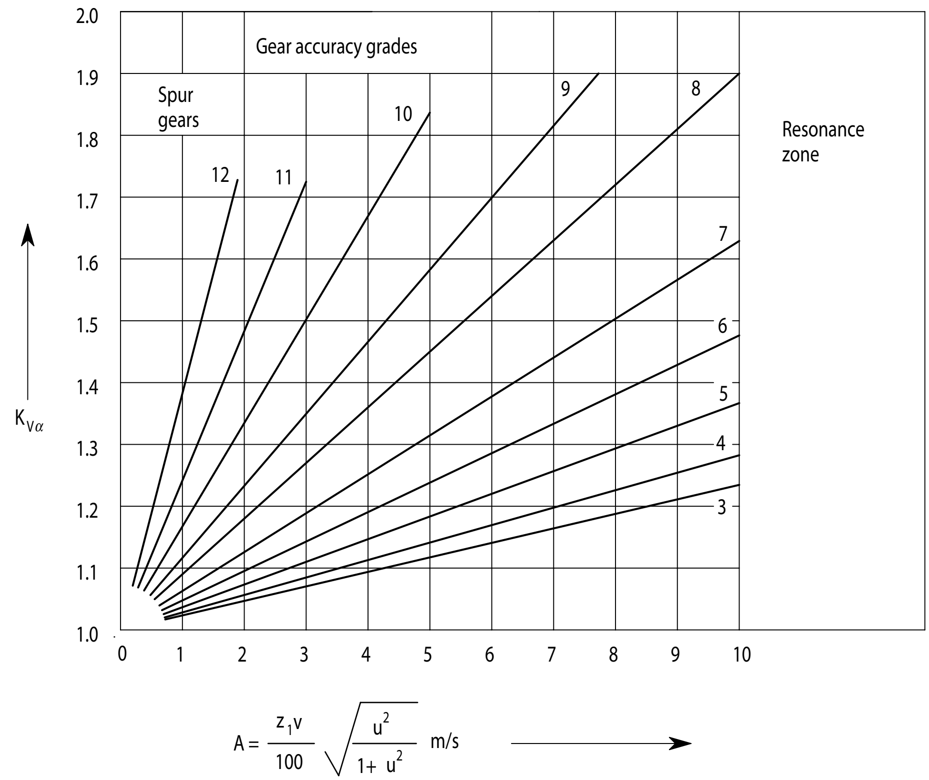

Table 20.30 *Factor ($K_{V\beta}$)*

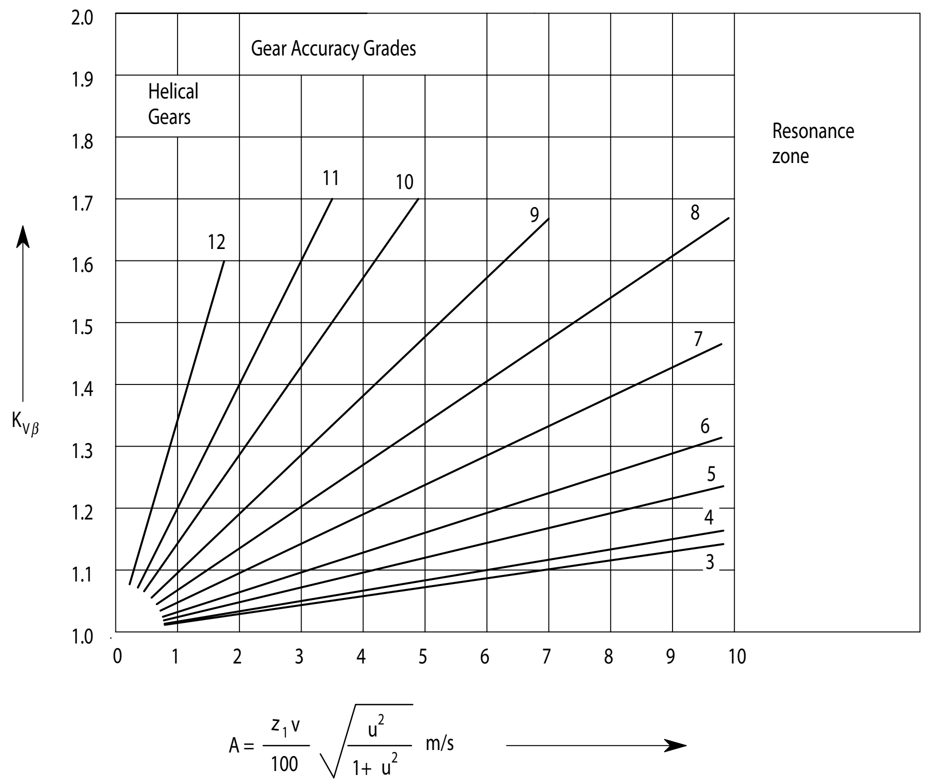

Table 20.31 *Longitudinal load distribution factors $K_{H\beta}$ and $K_{F\beta}$ [$K_{H\beta} \approx K_{F\beta}$]*

Gear quality	With no post assembly adjustment	With post assembly adjustment
Gr 6 and better	$1.135 + 0.18\left(\frac{b}{d_1}\right)^2 + 0.23 \times 10^{-3} \times b$	$1.1 + 0.18\left(\frac{b}{d_1}\right)^2 + 0.115 \times 10^{-3} \times b$
Gr 7	$1.15 + 0.18\left(\frac{b}{d_1}\right)^2 + 0.3 \times 10^{-3} \times b$	$1.11 + 0.18\left(\frac{b}{d_1}\right)^2 + 0.15 \times 10^{-3} \times b$
Gr 8	$1.17 + 0.18\left(\frac{b}{d_1}\right)^2 + 0.47 \times 10^{-3} \times b$	$1.12 + 0.18\left(\frac{b}{d_1}\right)^2 + 0.23 \times 10^{-3} \times b$
Gr 9	$1.23 + 0.18\left(\frac{b}{d_1}\right)^2 + 0.61 \times 10^{-3} \times b$	$1.15 + 0.18\left(\frac{b}{d_1}\right)^2 + 0.31 \times 10^{-3} \times b$

(*Contd.*)

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$K_{H\beta}$ = longitudinal load distribution factor for contact stress
$K_{F\beta}$ = longitudinal load distribution factor for bending stress
b = face width (mm) (Given data – Table 20.1)
d_1 = pitch circle diameter of pinion (mm) (Given data – Table 20.1)

Note: The longitudinal load distribution factor accounts for the non-uniform distribution of load across the face width.

Table 20.32 *Transverse load distribution factors $K_{H\alpha}$ and $K_{F\alpha}$ [$K_{H\alpha} \approx K_{F\alpha}$]*

Specific loading $\left(\frac{F_t K_A}{b}\right)$	**≥ 100 N/mm**							**< 100 N/mm**
Gear quality grade	Gr 6 and better	Gr 7	Gr 8	Gr 9	Gr 10	Gr 11	Gr 12	Gr 6 and worse
Hardened spur gears	1.0		1.1	1.2	$1/Z_\varepsilon^2 \geq 1.2$			
Hardened helical gears	1.0	1.1	1.2	1.4	$\varepsilon_{\alpha n} \geq 1.4$			
Unhardened spur gears	1.0			1.1	1.2	$1/Z_\varepsilon^2 \geq 1.2$		
Unhardened helical gears	1.0		1.1	1.2	1.4	$\varepsilon_{\alpha n} \geq 1.4$		

$K_{H\alpha}$ = transverse load distribution factor for contact stress
$K_{F\alpha}$ = transverse load distribution factor for bending stress
Z_ε = contact ratio factor for contact stress (Table 20.22)
ε_α = transverse contact ratio (Table 20.23)
$\varepsilon_{\alpha n}$ = normal contact ratio

$$\varepsilon_{\alpha n} = \frac{\varepsilon_\alpha}{\cos^2 \beta_b}$$

β_b = base helix angle (deg) (Table 20.19)

Note: (i) The transverse load distribution factor accounts for the distribution of actual load during the gear mesh.
(ii) For specific loading less than 100 N/mm and quality grade 5 and better, $K_{H\alpha}$ and $K_{F\alpha}$ may be taken as 1.0.

20.6 FACTOR OF SAFETY FOR CONTACT STRESS

Table 20.33 *Factor of safety for contact stress (S_H)*

$$S_{H1} = \frac{\sigma_{HP1}}{\sigma_H} \quad (20.4)$$

$$S_{H2} = \frac{\sigma_{HP2}}{\sigma_H} \quad (20.5)$$

S_{H1} = factor of safety for pinion based on contact stress
S_{H2} = factor of safety for wheel based on contact stress
σ_{HP1} = permissible contact stress for pinion (MPa or N/mm^2) (Table 20.2)
σ_{HP2} = permissible contact stress for wheel (MPa or N/mm^2) (Table 20.2)
σ_H = working (actual) contact stress (MPa or N/mm^2) (Table 20.18)

Table 20.34 *Minimum recommended factor of safety for contact stress (S_H)*

Application	Minimum value of (S_H)
For normal industrial applications	1.0 to 1.2
For high reliability and critical applications (involving high consequential damage or loss of life)	1.3 to 1.6

Note: The values given in above table are for the reference of students only.

20.7 BASIC DATA FOR LOAD CAPACITY (BENDING STRESS)

Table 20.35 *Basic (given) data required for finding out the load capacity based on bending stress*

(1) Number of teeth on pinion (z_1)
(2) Number of teeth on wheel (z_2)
(3) Module (m) (mm) (for spur gears)
(4) Normal module (m_n) (mm) (for helical gears)
(5) Normal pressure angle (α_n) (deg)
(6) Helix angle (β) (deg)
(7) Transmitted power (P) (W)
(8) Speed of pinion (n_1) (rpm)
(9) Pitch circle diameter of pinion (d_1) (mm)
(10) Pitch circle diameter of wheel (d_2) (mm)

(*Contd.*)

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(11)	Outside diameter of pinion (d_{a1}) (mm)
(12)	Outside diameter of wheel (d_{a2}) (mm)
(13)	Working centre distance (a') (mm)
(14)	Face width (b) (mm)
(15)	Profile correction factor for pinion (x_1)
(16)	Profile correction factor for wheel (x_2)
(17)	Probability of failure (p_f)
(18)	Mean roughness for pinion (R_{Z1})
(19)	Mean roughness for wheel (R_{Z2})
(20)	Accuracy grade for pinion
(21)	Accuracy grade for wheel
(22)	Hardness of pinion material
(23)	Hardness of wheel material
(24)	Viscosity grade of lubricant
(25)	Life of gears (L_H) (hr)

Table 20.36 *Supplementary data required for finding out the load capacity based on bending stress*

(1) Addendum of basic rack for pinion (h_{a01}) (mm)
(2) Addendum of basic rack for wheel (h_{a02}) (mm)
(3) Height at protuberance for pinion (h_{k1}) (mm)
(4) Height at protuberance for wheel (h_{k2}) (mm)
(5) Protuberance angle for pinion ($\alpha_{\text{pro }1}$) (deg)
(6) Protuberance angle for wheel ($\alpha_{\text{pro }2}$) (deg)
(7) Tip radius of basic rack of tool for pinion ($\rho_{ao\ 1}$) (mm)
(8) Tip radius of basic rack of tool for wheel ($\rho_{ao\ 2}$) (mm)
For standard gears, $h_{ao} = 1.25m_n$ or $1.167m_n$
For tools without protuberance, $\alpha_{\text{pro}} = \alpha_n$ $h_k = 0$

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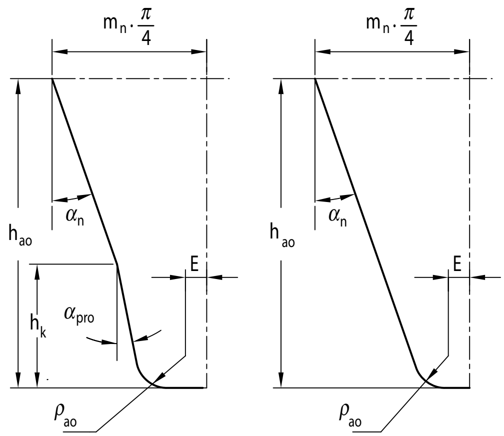


Dimensions of basic rack for tool

20.8 PERMISSIBLE LOAD CAPACITY BENDING STRESS

Table 20.37 *Permissible bending stress (σ_{FP})*

$$\sigma_{FP} = \left(\frac{\sigma_{FE}}{K_R}\right) Y_N Y_n Y_R Y_S \tag{20.6}$$

σ_{FP} = permissible bending stress at the root of tooth (MPa or N/mm^2)
K_R = reliability factor (Table 20.10)
Y_N = life factor for bending stress (Table 20.38)
Y_n = notch sensitivity factor (Table 20.39)
Y_R = factor of relative surface roughness (Table 20.42)
Y_S = size factor for bending stress (Table 20.43)
σ_{FE} = endurance limit of an un-notched specimen (MPa or N/mm^2) (Table 20.44 to Table 20.50)

(ISO: 6336 – Part 3)

Note: (i) The endurance limit (σ_{FE}) is the unidirectional pulsating stress with minimum stress of zero, which the material can sustain for at least 3×10^6 load cycles without fatigue failure.
(ii) The permissible contact stress (σ_{FP}) is calculated separately for pinion and wheel from the above equation.

Table 20.38 *Life factor for bending stress (Y_N)*

Material	Condition	Y_N
(i) Through hardened steels (ii) Spheroidal graphite cast iron (iii) Malleable cast iron	$L_N \le 10^4$	2.5
	$10^4 < L_N \le 3 \times 10^6$	$\left(\frac{3 \times 10^6}{L_N}\right)^{0.115}$
	$L_N > 3 \times 10^6$	1.0
(i) Carburising steels (ii) Surface hardened steels	$L_N \le 10^3$	2.5
	$10^3 < L_N \le 3 \times 10^6$	$\left(\frac{3 \times 10^6}{L_N}\right)^{0.115}$
	$L_N > 3 \times 10^6$	1.0
(i) Through hardened steels - Gas-nitrided steels (iii) Grey cast iron	$L_N \le 10^3$	1.6
	$10^3 < L_N \le 3 \times 10^6$	$\left(\frac{3 \times 10^6}{L_N}\right)^{0.059}$
	$L_N > 3 \times 10^6$	1.0
(i) Through hardened steels – Liquid-nitrided steels	$L_N \le 10^3$	1.2
	$10^3 < L_N \le 3 \times 10^6$	$\left(\frac{3 \times 10^6}{L_N}\right)^{0.012}$
	$L_N > 3 \times 10^6$	1.0

Note: (i) The life factor (Y_N) takes into account the case of limited life (less than 3×10^6 cycles) where a higher bending stress can be permitted. (ii) The life factor (Y_N) can be taken as unity if the required life is more than (3×10^6) cycles.

(iii) $L_N = 60 \times n \times L_H$

L_N = life in number of cycles

n = speed of pinion or gear (rpm) (Given data – Table 20.35)

L_H = required life in hours (hr) (Given data – Table 20.35)

(iv) For single tooth mesh of gears [$L_N = L_N$]

(v) For intermediate gears [$L_N = 2L_N$]

(vi) For motor pinion with double mesh [$L_N = 2L_N$]

(vii) For planetary gear train with 'q' planet gears,

Sun gear [$L_N = qL_N$]

Ring gear [$L_N = qL_N$]

Planet gears [$L_N = 2L_N$]

Table 20.39 *Notch sensitivity factor (Y_n)*

Material	Y_n
Cast iron	$0.35Y_K + 0.38$
Nitrided steels	$0.22Y_K + 0.61$
Soft steels	$0.18Y_K + 0.68$
Through hardened steels	$0.08Y_K + 0.86$
Case hardened steels	$0.04Y_K + 0.93$
Y_K = stress concentration factor (Table 20.40)	

Note: The actual reduction of resistance to fatigue due to notch sensitivity is often less than the value of stress concentration factor (Y_K). The notch sensitivity factor accounts for this effect.

Table 20.40 *Form factor (Y_{Fa}) and stress concentration factor (Y_K)*

1. Auxiliary parameter (G)

$$G = \frac{\rho_{ao}}{m_n} - \frac{h_{ao}}{m_n} + x$$

G = auxiliary parameter
ρ_{ao} = tip radius of basic rack (mm) (Supplementary data - Table 20.36)
m_n = normal module (mm) (Given data – Table 20.35)
h_{ao} = addendum of basic rack for tool (mm) (Supplementary data - Table 20.36)
x = profile correction factor (Given data – Table 20.35)
Note: The value of G may be negative for gears of normal design.

2. Auxiliary parameter (D)

$$D = \frac{\pi}{4} m_n - h_{ao} \tan \alpha_n + h_k (\tan \alpha_n - \tan \alpha_{\text{pro}}) - (1 - \sin \alpha_{\text{pro}}) \frac{\rho_{ao}}{\cos \alpha_{\text{pro}}}$$

h_k = height of protuberance (mm) (Supplementary data - Table 20.36)
α_{pro} = protuberance angle (deg) (Supplementary data - Table 20.36)

3. Auxiliary parameter (H)

$$H = \frac{2}{z_v}\left(\frac{\pi}{2} - \frac{D}{m_n}\right) - \frac{\pi}{3}$$

$$z_v = \frac{z}{\cos^3 \beta}$$

(*Contd.*)

(*Contd.*)

H = auxiliary parameter z_v = virtual number of teeth D = auxiliary parameter z = number of teeth (Given data – Table 20.35) β = helix angle (deg) (Given data – Table 20.35)
4. Auxiliary parameter (θ)
$$\theta = \frac{2G}{z_v}\tan\theta - H$$ The above transcendent equation is solved by substituting an initial value of $\left(\theta = \frac{\pi}{6}\right)$. The equation can be solved in two or three trials. In case of two solutions, the smaller value of θ is chosen. Note that the value of θ is in radians.
5. Auxiliary parameter (S_{Fn})
$$\frac{S_{Fn}}{m_n} = z_v \sin\left(\frac{\pi}{3} - \theta\right) + \sqrt{3}\left(\frac{G}{\cos\theta} - \frac{\rho_{ao}}{m_n}\right)$$
6. Involute function of tip pressure angle (α_a)
$$d_b = zm_n\left(\frac{\cos\alpha_n}{\cos\beta}\right)$$ $$\cos\alpha_a = \left(\frac{d_b}{d_a}\right)$$ $$\text{inv}\,\alpha_a = \tan\alpha_a - \alpha_a\left(\frac{\pi}{180}\right)$$
d_a = outside diameter of gear (mm) (Given data – Table 20.35)
7. Involute function of transverse pressure angle (α_t)
$$\tan\alpha_t = \frac{\tan\alpha_n}{\cos\beta}$$ $$\text{inv}\,\alpha_t = \tan\alpha_t - \alpha_t\left(\frac{\pi}{180}\right)$$
8. Auxiliary parameter (τ)
$$\tau = \left[\frac{1}{z}\left(\frac{\pi}{2} + 2x\tan\alpha_n\right) + \text{inv}\,\alpha_t - \text{inv}\,\alpha_a\right]\frac{180}{\pi}$$
9. Transverse pressure angle at the tooth tip (α_{ta})
$$\alpha_{ta} = \alpha_a - \tau$$

(*Contd.*)

(*Contd.*)

10. Helix angle at the tip circle (β_a)
$$\tan \beta_a = \frac{d_b \tan \beta}{d \cos \alpha_{ta}}$$
11. Normal pressure angle at the tooth tip (α_{an})
$$\tan \alpha_{an} = \tan \alpha_{ta} \cos \beta_a$$
12. Auxiliary parameter (h_{Fa})
$$\frac{h_{Fa}}{m_n} = \frac{1}{2}\left[\frac{z}{\cos \beta}\left(\frac{\cos \alpha_t}{\cos \alpha_{ta}} - 1\right) + z_v\left\{1 - \cos\left(\frac{\pi}{3} - \theta\right)\right\} - \frac{G}{\cos \theta} + \frac{\rho_{ao}}{m_n}\right]$$
13. Form factor (Y_{Fa})
$$Y_{Fa} = \frac{6\left(\frac{h_{Fa}}{m_n}\right)\cos \alpha_{an}}{\left(\frac{S_{Fn}}{m_n}\right)^2 \cos \alpha_n}$$
14. Auxiliary parameter (L_a)
$$L_a = \frac{S_{Fn}}{h_{Fa}}$$
15. Auxiliary parameter (ρ_F)
$$\frac{\rho_F}{m_n} = \frac{\rho_{ao}}{m_n} + \frac{2G^2}{\cos \theta (z_v \cos^2 \theta - 2G)}$$
16. Auxiliary parameter (q_n)
$$q_n = \frac{S_{Fn}}{2\rho_F}$$
17. Stress concentration factor (Y_K)
$$Y_K = (1.2 + 0.13 L_a) q_n^{\left[\frac{1}{1.21 + \frac{2.3}{L_a}}\right]}$$ $$Y_K = 1.0 \text{ [if } (q_n < 1) \text{ or } (q_n > 8)]$$

Table 20.41 *Manufacturing process and surface roughness*

Manufacturing process for gears	Mean surface roughness (R_Z) (μm)
Lapping	0.5–4.8
Grinding	2.4–4.8
Grinding –crisscross	4.8–9.6
Shaving	4.8–19.2
Shaping	9.6–75.0
Hobbing	19.2–75.0
Milling with form cutters	19.2–75.0

Note: CLA value = R_a $\quad R_a = \left(\frac{R_Z}{6}\right)$

Table 20.42 *Factor of relative surface roughness (Y_R)*

Material	Condition	Y_R
Through-hardened and case-hardened steels	$1\,\mu m \le R_Z \le 40\,\mu m$	$1.674 - 0.529(R_Z + 1)^{0.1}$
	$R_Z < 1\,\mu m$	1.12
Soft steels	$1\,\mu m \le R_Z \le 40\,\mu m$	$5.306 - 4.203(R_Z + 1)^{0.01}$
	$R_Z < 1\,\mu m$	1.07
Grey cast iron and nitrided steels	$1\,\mu m \le R_Z \le 40\,\mu m$	$4.299 - 3.259(R_Z + 1)^{0.005}$
	$R_Z < 1\,\mu m$	1.025

Note: The surface roughness factor takes into account the influence of surface roughness on the fatigue life of gears.

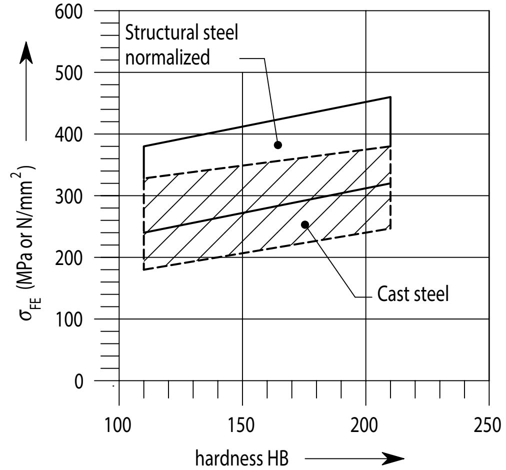
Table 20.43 *Size factor for bending stress (Y_S)*

Material	Condition	Y_Z
(i) Through hardened steels (ii) Spheroidal graphite cast iron (iii) Malleable cast iron	$5 < m_n < 30$	$1.03 - 0.006\ m_n$
	$30 \le m_n$	0.85
Surface hardened steels	$5 < m_n < 30$	$1.05 - 0.01\ m_n$
	$30 \le m_n$	0.75
Cast Materials	$5 < m_n < 25$	$1.075 - 0.015\ m_n$
	$m_n \ge 25$	0.7
For all materials under static load and ($m_n \le 5$)	—	1.0

Note: The size factor takes into account the decrease of fatigue strength with increasing size of gear tooth.

20.9 DETERMINATION OF ENDURANCE LIMIT (σ_{FE})

Table 20.44 *Determination of (σ_{FE}) for normalised structural steels and cast steel*



Note for all tables from Tables 20 .44 to 20.48: (i) In case of idler gears, the values obtained from these tables are to be multiplied by (0.7).
(ii) When the direction of load changes, the values obtained from these tables are to be multiplied by (1.7).

Table 20.45 *Determination of (σ_{FE}) for cast iron*

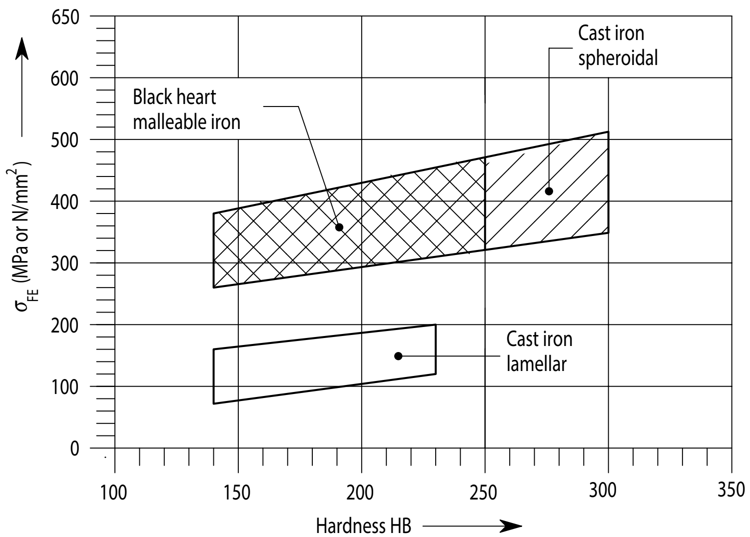


Table 20.46 *Determination of (σ_{FE}) for through hardened steels (> 0.32% C)*

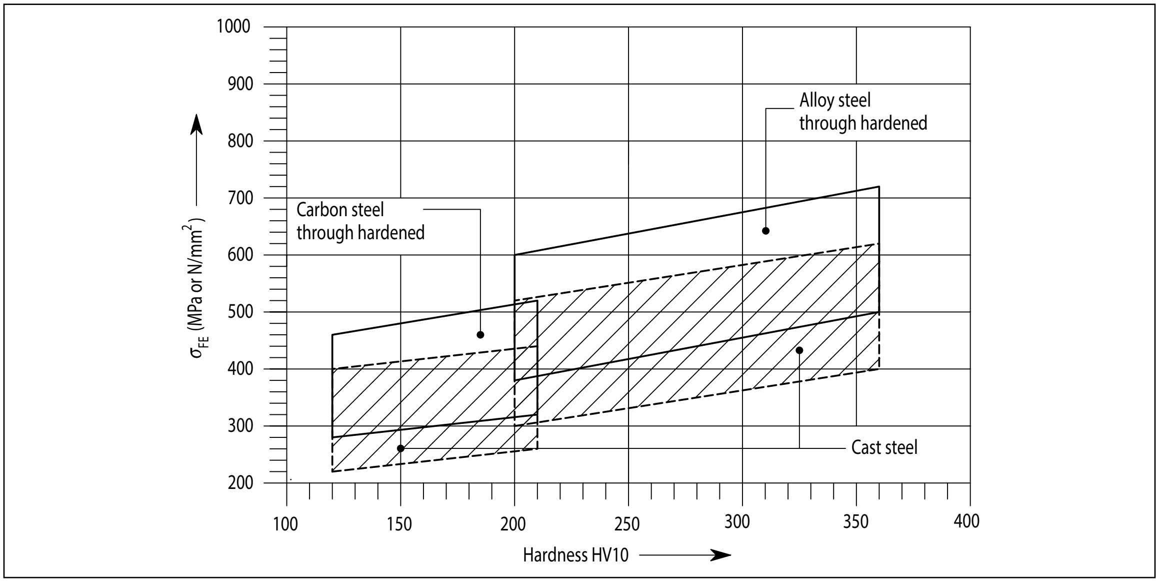
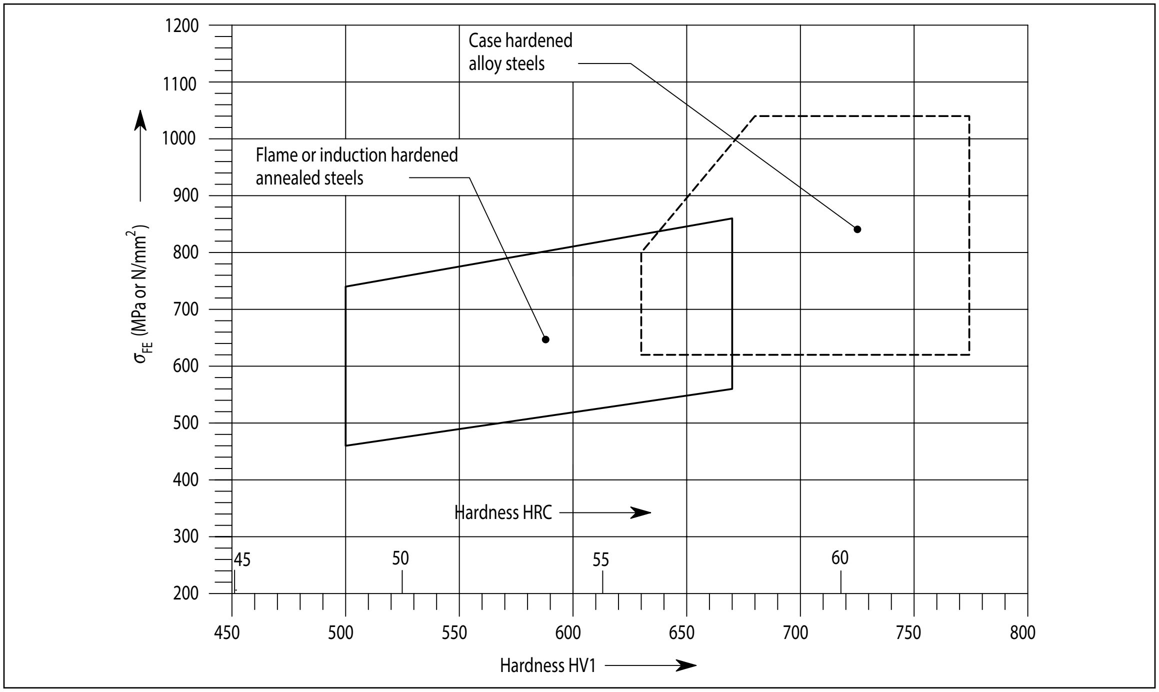
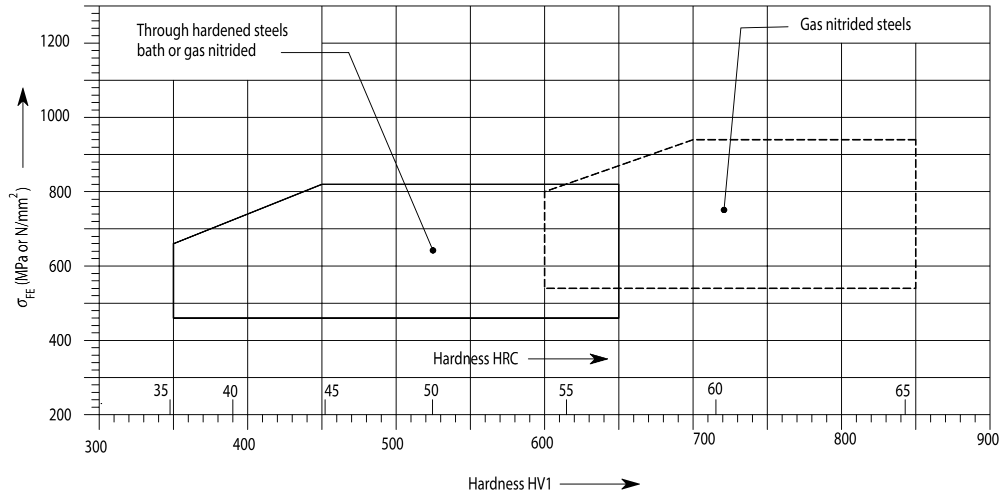


Table 20.47 *Determination of (σ_{FE}) for surface hardened steels*



Note: The endurance limit indicated for surface hardened test gears apply to case depth of (0.15 m_n) and above of the finished gears.

Table 20.48 *Determination of (σ_{FE}) for nitrided steels*



Note: The endurance limit indicated apply to a depth of nitration of 0.4 to 0.6 mm.

20.10 DETERMINATION OF ENDURANCE LIMIT (σ_{FE})—ALTERNATIVE METHOD

Table 20.49 *Nominal stress number in bending (σ_{FE})*

The value of nominal stress number in bending (σ_{FE}) is given by,

$$\sigma_{FE} = A \cdot x + B$$

x = Surface hardness (HBW or HV) (Given Data – Table 20.35)
A, B = Constants for bending stress (Table 20.50)
HBW, HV = units of hardness number (Table 20.50)

(ISO: 6336 – Part 5)

Note: The above equation for nominal stress number in bending (σ_{FE}) is based on experimental data of endurance tests of reference test gears under reference test conditions.

Table 20.50 *Constants A and B for bending stress in calculation of (σ_{FE})*

Material	Type	Quality	A	B	Units of Hardness	Min. hardness	Max. hardness
Low carbon steels	Wrought normalized low carbon steel	ML/MQ	0.455	69	HBW	110	210
		ME	0.386	147	HBW	110	210

(*Contd.*)

(*Contd.*)

Cast steels	-	ML/MQ	0.313	62	HBW	140	210
		ME	0.254	137	HBW	140	210
Cast iron	Black malleable cast iron	ML/MQ	0.345	77	HBW	135	250
		ME	0.403	128	HBW	175	250
	Nodular cast iron	ML/MQ	0.350	119	HBW	175	300
		ME	0.380	134	HBW	200	300
	Grey cast iron	ML/MQ	0.256	8	HBW	150	240
		ME	0.200	53	HBW	175	275
Through hardened wrought steels	Carbon steels	ML	0.250	108	HV	115	215
		MQ	0.240	163	HV	115	215
		ME	0.283	202	HV	115	215
	Alloy steels	ML	0.423	104	HV	200	360
		MQ	0.425	187	HV	200	360
		ME	0.358	231	HV	200	390
Case hardened wrought steels	-	ML	0.0	312	HV	600	800
		MQ	0.0	425	HV	660	800
		ME	0.0	525	HV	660	800
Flame or induction hardened wrought steels	-	ML	0.305	76	HV	485	615
		MQ	0.138	290	HV	500	570
		ME	0.271	237	HV	500	615
Nitrided steels	Nitrided steels	ML	0.0	270	HV	650	900
		MQ	0.0	420	HV	650	900
		ME	0.0	468	HV	650	900
	Through hardened Nitrided steels	ML	0.0	258	HV	450	650
		MQ	0.0	363	HV	450	650
		ME	0.0	432	HV	450	650
Nitro-carburized steels	Through hardened steels	ML	0.0	224	HV	300	650
		MQ/ME	0.653	94	HV	300	450

Note: ML, MQ and ME are material quality grades.

(i) ML stands for modest demands on material quality and on heat treatment process during gear manufacturing.
(ii) MQ stands for requirements that can be met by experienced manufacturers at moderate cost.
(iii) ME stands for requirements that must be realized when a high degree of operating reliability is required

20.11 WORKING (ACTUAL) BENDING STRESS

Table 20.51 *Working (actual) bending stress (σ_F)*

$$\sigma_F = \left(\frac{F_t}{bm_n}\right) Y_{Fa} Y_K Y_\varepsilon Y_\beta K_A K_V K_{F\beta} K_{F\alpha} \qquad (20.7)$$
σ_F = working (actual) bending stress (MPa or N/mm^2) F_t = tangential force (nominal) at pitch circle appropriate to pinion or wheel calculations respectively (N) b = face width (mm) (Given data – Table 20.35) m_n = normal module (mm) (Given data – Table 20.35) Y_{Fa} = form factor (Table 20.40) Y_K = stress concentration factor (Table 20.40) Y_ε = contact ratio factor for bending stress (Table 20.52) Y_β = helix angle factor for bending stress (Table 20.53) K_A = application factor (Table 20.25) K_V = dynamic load factor (Table 20.28) $K_{F\beta}$ = longitudinal load distribution factor for bending stress (Table 20.31) $K_{F\alpha}$ = transverse load distribution factor for bending stress (Table 20.32)
$$F_t = \frac{P}{v} \text{ and } v = \frac{\pi d_1 n_1}{60 \times 10^3} = \frac{d_1 n_1}{19098} \qquad (20.8)$$
P = transmitted power (W) (Given data – Table 20.35) v = pitch line velocity (m/s) n_1 = speed of pinion (rpm) (Given data – Table 20.35) d_1 = pitch circle diameter of pinion (rpm) (Given data – Table 20.35)

(ISO: 6336 – Part 3)

Note: The working bending stress (σ_F) is calculated separately for pinion and wheel from the above equation.

Table 20.52 *Contact ratio factor for bending stress (Y_ε)*

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_\alpha}$$
ε_α = transverse contact ratio (Table 20.23)

Note: The contact ratio factor (Y_ε) converts the applied load at the tip of tooth to the corresponding force at the decisive position of load application.

Table 20.53 *Helix angle factor for bending stress (Y_β)*

$$Y_\beta = 1 - \varepsilon_\beta\left(\frac{\beta}{120}\right)$$ $$Y_\beta \geq 1 - 0.25\varepsilon_\beta \text{ and } Y_\beta \geq 0.75$$
When $\varepsilon_\beta > 1$, use $\varepsilon_\beta = 1$ When $\beta > 30°$, use $\beta = 30°$
ε_β = overlap ratio (Table 20.23) β = helix angle (Given data – Table 20.35)

Note: The helix angle factor (Y_β) takes into account the difference between the actual helical gear and virtual spur gear in the normal plane.

20.12 FACTOR OF SAFETY FOR BENDING STRESS

Table 20.54 *Factor of safety for bending stress (S_B)*

$$S_{B1} = \frac{\sigma_{FP1}}{\sigma_{F1}} \quad (20.9)$$ $$S_{B2} = \frac{\sigma_{FP2}}{\sigma_{F2}} \quad (20.10)$$
S_{B1} = factor of safety for pinion based on bending stress S_{B2} = factor of safety for wheel based on bending stress σ_{FP1} = permissible bending stress for pinion (MPa or N/mm^2) (Table 20.37) σ_{FP2} = permissible bending stress for wheel (MPa or N/mm^2) (Table 20.37) σ_{F1} = working (actual) bending stress for pinion (MPa or N/mm^2) (Table 20.51) σ_{F2} = working (actual) bending stress for wheel (MPa or N/mm^2) (Table 20.51)

Table 20.55 *Minimum recommended factor of safety for bending stress (S_B)*

Application	Minimum value of (S_B)
For normal industrial applications	1.4 to 1.5
For high reliability and critical applications (involving high consequential damage or loss of life)	1.6 to 3.0

Note: The values given in above table are for the reference of students only.

Gear Design using Data Book

21

21.1 PARAMETERS OF GEAR DESIGN

Table 21.1 *Recommended series of module (m) for spur gears and normal module (m_n) for helical gears (mm)*

Choice-1	1.0	1.25	1.5	2.0	2.5	3.0	4.0
	5.0	6.0	8.0	10	12	16	20
Choice-2	1.125	1.375	1.75	2.25	2.75	3.5	4.5
	5.5	7	9	11	14	18	
Choice-3	(3.25)	(3.75)	(6.5)				

Note: (i) Preference is given to use the modules under Choice-1. (ii) Modules given under Choice-3 should be avoided as far as possible.

Table 21.2 *Gear drive accuracy recommendations*

Type of gear drive	**Peripheral velocity (m/s)**			
	Under 5	5 to 8	8 to 12.5	Over 12.5
	Grade of gears			
Spur gears	Gr 9	Gr 8	Gr 7	Gr 6
Helical gears	Gr 9	Gr 9	Gr 8	Gr 7
Straight bevel gears	Gr 8	Gr 7	—	—
Spiral bevel gears	Gr 9	Gr 9	Gr 8	—

(D. Chernilevsky)

Table 21.3 *Gear drive accuracy recommendations (for spur and helical gears)*

Minimum quality grade for gears	**Peripheral velocity of gears (m/s)**
Gr 5 to Gr 6	15 to 30
Gr 6 to Gr 7	7.5 to15
Gr 7 to Gr 8	5 to 7.5
Gr 8 to Gr 9	2.5 to 5
Gr 10 to Gr 12	Below 2.5

(Industrial Practice)

Table 21.4 *Allowable velocities and applications of gears of various accuracy grades*

Quality Grade of gears	Peripheral velocity (m/s)		Applications
	Spur gears Up to	Other gears Up to	
Gr 6 (Gears of above-normal accuracy)	20	31.5	High-speed drives, Indexing mechanisms
Gr 7 (Gears of normal accuracy)	12.5	20	High-speed and medium-load drives and vice versa
Gr 8 (Gears of below-normal accuracy)	8	12.5	Gear drives in general engineering
Gr 9 (Gears of coarse accuracy)	5	8	Low-speed drives of low-accuracy machinery

(D. N. Reshetov)

Table 21.5 *Standard gear ratios (i) for general-purpose speed reducer (Cylindrical gears)*

Series - 1	1.0	1.25	1.6	2.0	2.5	3.15	4.0	5.0	6.3	8.0	10
Series - 2	—	1.4	1.8	2.24	2.8	3.55	4.5	5.6	7.1	9.0	—

(D. Chernilevsky)

Note: Series-1 should be preferred to Series-2. $\left[i = \frac{Z_2}{Z_1}\right]$

Notations – i = gear ratio

Z_1 = number of teeth on pinion

Z_2 = number of teeth on wheel

Table 21.6 *Standard centre distances (a) for speed reducer (Cylindrical gears) (mm)*

Series–1	40	50	63	80	100	125	160	200	250	315	400	500
Series–2	—	—	71	90	112	140	180	224	280	355	450	560

(D. Chernilevsky)

Note: Series–1 should be preferred to Series–2. $\left[a = \frac{m(Z_1 + Z_2)}{2}\right]$

21.2 GEAR MATERIALS

Table 21.7 *Recommended combinations of materials for pinion and wheel*

Gear ratio (i)	Surface hardness (BHN)	Material of pinion	Material of wheel
		45C8	Fe490 35C8 Cast steel
		50C8	35C8 Cast steel

(*Contd.*)

(*Contd.*)

$i < 8$	BHN ≤ 350	55C8	Fe620 45C8 Cast steel
		40Cr4	50C8 55C8 Cast steel
		35Ni5Cr2	40Cr4 Cast steel
$i < 4$	BHN > 350	45C8 50C8	35C8
		55C8	Fe620 45C8
		40Cr4	50C8 55C8
		35Ni5Cr2	40Cr4
$i \geq 4$	BHN for pinion > 350 BHN for wheel ≤ 350	45C8	Cast steel
		50C8	35Ni5Cr2
		40Cr4	Cast steel 35Ni5Cr2
		40Ni6Cr4Mo3	45C8
		40Ni6Cr4Mo2	55C8

Note: (i) In general, the surface hardness of pinion should be more than the surface hardness of wheel by 40 BHN. (for BHN > 350 to 400). This minimises the risk of seizure and improves the reliability and load carrying capacity of cylindrical (both spur and helical) gears. (ii) The recommended grades of cast steel are CS 640 or CS 840

Table 21.8 *Properties of gear materials*

Material	Condition	Minimum tensile strength (σ_u) (MPa or N/mm^2)	Minimum yield stress (σ_y) (MPa or N/mm^2)	Surface hardness (HB)
Grey cast iron				
FG 200	As cast	200	—	160 Min
FG 260		260	—	180 Min
FG 350		350	—	207 Min
FG 350	Heat-treated	350	—	300 Min
High tensile steel castings				
CS 640	As cast	640	390*	190 Min
CS 700		700	580*	207 Min
CS 840		840	700*	248 Min
CS 1030		1030	850*	305 Min
CS 1230		1230	1000*	355 Min

(*Contd.*)

(*Contd.*)

Phosphor bronze				
Sand cast		160	—	60 Min
Chill cast		240	—	70 Min
Centrifugally cast		267.5	—	90 Min
Carbon steels				
Fe 490	—	490	290	—
Fe 620		620	380	—
30C8	Hardened and Tempered	600	400	179
35C8		600	400	—
40C8		600	380	217
45C8		600	380	229
50C8		700	460	241
55C8		700	460	265
Alloy steels				
37C15	Hardened and Tempered	590	390**	170 Min
40Cr4		690	490**	201 Min
40Cr4Mo2		700	490**	201 Min
35Ni5Cr2		690	490**	201 Min
40Ni14		790	550**	229 Min
30Ni16Cr5		1540	1240**	444 Min
40Ni6Cr4Mo2		790	550**	229 Min
40Ni6Cr4Mo3		790	550**	229 Min
40Ni10Cr3Mo6		990	750**	285 Min

Note: (i) * Yield stress is 0.5 percent proof stress. (ii) ** Yield stress is 0.2 percent proof stress. (iii) The values of hardness given in the table are for guidance only.

21.3 DESIGN TWISTING MOMENT

Table 21.9 *Design twisting moment [M_t]*

$[M_t] = M_t\, k_d\, k$	(21.1)	$[M_t]$ = design twisting moment or design torque (N-mm) M_t = nominal twisting moment or rated torque transmitted by pinion (N-mm) k = load concentration factor (Tables 21.10 and 21.11) k_d = dynamic load factor (Table 21.12)
$M_t = \dfrac{60 \times 10^6 (\text{kW})}{2\pi n_p}$	(21.2)	M_t = nominal twisting moment or rated torque transmitted by pinion (N-mm) kW = nominal or rated power transmitted by gears (kW) n_p = speed of rotation of pinion (rpm)
$M_t = 71620\left(\dfrac{hp}{n_p}\right)$	(21.3)	M_t = nominal twisting moment or rated torque transmitted by pinion (kgf-cm) (1 kgf = 9.807 N) hp = nominal or rated power transmitted by gears (hp) (1hp = 0.7457 kW)

(*Contd.*)

(*Contd.*)

Initially assume, $k_d\, k = 1.3$ (for symmetrical scheme) $k_d\, k = 1.5$ (for unsymmetrical and overhanging schemes)

(D. N. Reshetov)

Table 21.10 *Load concentration factor (k) for steel gears of Quality Gr–8 having hardness HB >350 (spur and helical gears)*

$\psi_p = \left(\frac{b}{d_1}\right)$	Bearings close to gears and symmetrical	Asymmetrical		Over hanging pinion
		* Very rigid shaft	** Less rigid shaft	
0.2	1	1	1.05	1.15
0.4	1	1.04	1.1	1.22
0.6	1.03	1.08	1.16	1.32
0.8	1.06	1.13	1.22	1.45
1.0	1.1	1.18	1.29	—
1.2	1.14	1.23	1.36	—
1.4	1.19	1.29	1.45	—
1.6	1.25	1.35	1.55	—

* $l/d_s < 3$ $\quad l$ = length of shaft

** $l/d_s > 3$ $\quad d_s$ = diameter of shaft $\quad \psi_p = \frac{\psi}{2}(i \pm 1)$

Table 21.11 *Load concentration factor (k) for bevel gears*

Surface hardness of gears (HB)	$\left(\frac{b}{d_{1av}}\right)$ ratio		
	≤ 1	1 to 1.6	1.6 to 1.8
> 350 for both gears	1.6	—	—
≤ 350 for both gears or at least for wheel	1.1	1.2	1.3

d_{1av} = average pitch diameter of bevel pinion (mm)

V = pitch line velocity (m/s)

$$d_{1av} = m_1 z_1 \left(\frac{R - 0.5b}{R}\right) \qquad V = \left(\frac{\pi d_{1av} n_1}{60 \times 1000}\right)$$

Table 21.12 *Dynamic load factor (k_d)*

Quality grade: Cylindrical gear	Quality grade: Conical gear	Pinion surface hardness HB	Spur and straight bevel gears: 1.0	3.0	8.0	12.0	Helical and spiral bevel gears: 3.0	8.0	12.0	18.0
			Pitch line velocity V (m/s) up to							
Gr 5	—	≤ 350	—	—	1.2	1.4	—	1	1.1	1.2
		> 350	—	—	1.2	1.3	—	1	1	1.1

(*Contd.*)

(*Contd.*)

Gr 6	Gr 5	≤ 350	—	1.25	1.45	—	1	1	1.2	1.3
		> 350	—	1.2	1.3	—	1	1	1.1	1.2
Gr 8	Gr 6	≤ 350	1	1.35	1.55	—	1.1	1.3	1.4	—
		> 350	—	1.3	1.4	—	1.1	1.2	1.3	—
Gr 10	Gr 8	≤ 350	1.1	1.45	—	—	1.2	1.4	—	—
		> 350	—	1.4	—	—	1.2	1.3	—	—
—	Gr 10	≤ 350	1.2	—	—	—				

Table 21.13 *Standard values of width factor (ψ) for speed reducer*

ψ	0.100	0.125	0.160	0.20	0.250	0.315	0.40	0.50	0.63	0.8	1.0	1.25

(D. N. Reshetov)

Note: Values from 0.63 to1.25 are for herringbone gears. $\left[\psi = \left(\frac{b}{a}\right)\right]$

Notations: ψ = width ratio
b = face width of gear (mm)
a = centre-to-centre distance between gears (mm)

Table 21.14 *Width factor (ψ) for speed reducer*

Gear arrangement with respect to bearings	Surface hardness	ψ
Symmetrical	Any	0.315, 0.4, 0.5
Non-symmetrical	Up to 350 BHN	0.315, 0.4
	Rockwell C 40 upwards	0.25, 0.315
One shaft cantilever	Up to 350 BHN	0.25
	Rockwell C 40 upwards	0.2
Herringbone gears	Any	0.4-0.63
Internal gears	Any	$\psi = 0.3(i \pm 1)$

(D. Chernilevsky)

Note: In general, [$\psi_m = 8 \text{ to } 12$] or [10] $\psi_m = \left(\frac{b}{m}\right)$
(b = 10 m)

21.4 DESIGN BENDING STRESS

Table 21.15 *Design bending stress [σ_b]*

$[\sigma_b] = \frac{1.4}{n}\frac{k_{bl}}{k_\sigma}\sigma_{-1}$ (21.4) (For gears rotating in one direction only) $[\sigma_b] = \frac{k_{bl}}{nk_\sigma}\sigma_{-1}$ (21.5) (For gears rotating in both directions)	$[\sigma_b]$ = design bending stress (MPa or N/mm^2) n = factor of safety (Table 21.16) k_{bl} = life factor for bending (Table 21.17) k_σ = fillet stress concentration factor (Table 21.20) σ_{-1} = endurance limit stress in bending for complete reversal of stresses (MPa or N/mm^2) (Table 21.21)

(V. Dobrovolsky)

Table 21.16 *Values of factor of safety (n)*

Material	Method of manufacturing	Heat treatment	Factor of safety (n)
Steel Cast iron	Casting	No heat treatment	2.5
		Tempering or normalising	2.0
Steel	Casting or forging	Case hardening	2.0
	Forging	Surface hardening	2.5
		Normalising	2.0

Table 21.17 *Life factor for bending (k_{bl})*

Material	Surface hardness (HB)	Life in number of stress cycles to be sustained (N)	k_{bl}
Steel	≤ 350	$\geq 10^7$	1
		$< 10^7$	$\sqrt[9]{\frac{10^7}{N}}$
	> 350	$\geq 25 \times 10^7$	0.7
		$< 25 \times 10^7$	$\sqrt[9]{\frac{10^7}{N}}$
Cast iron			$\sqrt[9]{\frac{10^7}{N}}$

Note: (i) When the case hardness HB>350 and the core hardness HB< 350, then the life factor (k_{bl}) is obtained for HB<350. (ii) The life in number of stress cycles to be sustained (N) can be calculated by formulae given in Table 21.18. (iii) The values of N are given in Table 21.19

Table 21.18 *Equivalent mean life (N)*

Condition of load	Formulae	Notations
Constant load	$N = 60\,nT$ (21.6)	N = life in number of stress cycles to be sustained n = speed of rotation (rpm) T = life in hours (hr)
Variable load	$N = \frac{60}{(M_{t1})^3} \Sigma (M_{ti})^3 T_i n_i$ (21.7)	

Note: When the gear is subjected to

(i) a constant torque of (M_{t1}) for T_1 hours at a mean speed of (n_1) rpm,

(ii) a constant torque of (M_{t2}) for T_2 hours at a mean speed of (n_2) rpm,

(iii) a constant torque of (M_{t3}) for T_3 hours at a mean speed of (n_3) rpm, and so on,

Then the equivalent number of cycles (N_{eq}) at a constant torque of (M_{t1}) is given by,

$$N_{eq} = 60\left[T_1 n_1 \left(\frac{M_{t1}}{M_{t1}}\right)^3 + T_2 n_2 \left(\frac{M_{t2}}{M_{t1}}\right)^3 + T_3 n_3 \left(\frac{M_{t3}}{M_{t1}}\right)^3 + \cdots\right]$$

$$= \frac{60}{(M_{t1})^3} \Sigma T_i n_i (M_{ti})^3 \tag{21.8}$$

Table 21.19 *Life in number of stress cycles to be sustained (N)*

Surface hardness of gears	BHN	Up to 200	250	300	350	—	—	—
	Rockwell C	—	—	—	—	40	50	60
N (cycles)		10×10^6	17×10^6	25×10^6	36×10^6	44×10^6	84×10^6	140×10^6

(D. Chernilevsky)

Table 21.20 *Values of fillet stress concentration factor (k_σ)*

Material and Heat Treatment	Addendum modification coefficient (X)		
	$X < 0$	$0 \le X \le 0.1$	$X > 0.2$
Steel Normalising, Surface hardening	1.4	1.5	1.6
Steel Case hardening	1.1	1.2	1.3
Cast iron	1.2	1.2	1.3

Note: Initially, assume $X = 0$ and $Z_1 \ge 17$

Table 21.21 *Endurance limit stress in bending for complete reversal of stresses (σ_{-1}) (MPa or N/mm^2)*

Material of mating gear	(σ_{-1}) (MPa or N/mm^2)
Forged steel	$\sigma_{-1} = 0.25(\sigma_u + \sigma_y) + 50$
Cast steels	$\sigma_{-1} = 0.22(\sigma_u + \sigma_y) + 50$
Alloy steels	$\sigma_{-1} = 0.35(\sigma_u) + 120$
Cast iron	$\sigma_{-1} = 0.45(\sigma_u)$

Note: (i) σ_u = minimum ultimate tensile strength (MPa or N/mm^2)
(ii) σ_y = minimum yield stress (MPa or N/mm^2)
(iii) Refer to Table 21.8 for values of (σ_u) and (σ_y).

21.5 DESIGN CONTACT STRESS

Table 21.22 *Design contact stress [σ_c]*

$$[\sigma_c] = C_B \text{ HB } k_{cl} \tag{21.9}$$

$$[\sigma_c] = C_R \text{ HRC } k_{cl} \tag{21.10}$$

$[\sigma_c]$ = design contact stress (MPa or N/mm^2)
C_B, C_R = coefficients depending upon method of measuring the surface hardness (Table 21.23)
HB = surface hardness in units of Brinell hardness number
HRC = surface hardness in units of Rockwell C hardness number
k_{cl} = life factor for contact stress (Table 21.24)

(V. Dobrovolsky)

Table 21.23 *Values of coefficients C_B and C_R*

Material	Heat treatment	Surface hardness	Coefficients C_B and C_R
Carbon steels and alloy steels of any type	Normalising or Hardening and tempering	HB ≤ 350	$C_B = 2.5$
High strength alloy steels – Nickel and chromium steels	Case-hardening	HRC = 55 to 63	$C_R = 31$
Alloy steels	Case-hardening	HRC = 55 to 63	$C_R = 28$
Carbon and Carbon - Manganese steels [Equivalent to C15, C20, C15Mn $\underline{85}$, C20Mn $\underline{85}$]	Case-hardening	HRC = 55 to 63	$C_R = 22$
Alloy steels and Carbon steels 40C8 and 45C8	Hardening and tempering	HRC = 40 to 55	$C_R = 26.5$
Alloy steels and Carbon steels 40C8 and 45C8	Surface-hardening	HRC = 40 to 55	$C_R = 23$
Grey cast iron FG 200, FG 260	—	HB = 170 to 200	$C_B = 2.0$
Grey cast iron FG 300, FG 350	—	HB = 200 to 260	$C_B = 2.3$

Table 21.24 *Life factor for contact stress (k_{cl})*

Material	Surface hardness (HB)	Life in number of cycles (N)	k_{cl}
Steel	≤ 350	$\geq 10^7$	1
		$< 10^7$	$\sqrt[6]{\frac{10^7}{N}}$
	> 350	$\geq 25 \times 10^7$	0.585
		$< 25 \times 10^7$	$\sqrt[6]{\frac{10^7}{N}}$
Cast iron			$\sqrt[6]{\frac{10^7}{N}}$

Note: (i) $k_{cl} = \sqrt[6]{\frac{10^7}{N}}$

(ii) The life in number of stress cycles to be sustained (N) can be calculated by formulae given in Table 21.18.

(iii) The values of N are given in Table 21.19.

21.6 DESIGN FORMULAE – BEAM STRENGTH

Table 21.25 *Formulae based on beam strength of pinion tooth (Bending strength)*

Formulae for designing gears		
Spur gears		
$m \geq 1.26 \sqrt[3]{\dfrac{[M_t]}{y[\sigma_b]\psi_m Z_1}}$	(21.11)	m = module of gear tooth (mm) $[M_t]$ = design twisting moment or design torque (N-mm) (Table 21.9) y = Lewis form factor (Table 21.26) $[\sigma_b]$ = design bending stress (MPa or N/mm^2) (Table 21.15) $\psi_m = \left(\dfrac{b}{m}\right)$ factor [ψ_m is usually 8 to 12] [$b \cong 10\ m$] b = face width of gear (mm) Z_1 = number of teeth on pinion
Helical and herringbone gears		
$m_n \geq 1.15 \cos\beta \sqrt[3]{\dfrac{[M_t]}{y_v[\sigma_b]\psi_m Z_1}}$	(21.12)	m_n = normal module of helical gear tooth (mm) β = helix angle β = 8° to 25° (for helical gears) β = 25° to 40° (for herringbone gears) y_v = Lewis form factor based on virtual number of teeth (Z_v) (Table 21.26) Z_v = formative or virtual number of teeth (Table 21.27)
Straight bevel gears		
$m_{av} \geq 1.28 \sqrt[3]{\dfrac{[M_t]}{y_v[\sigma_b]\psi_m Z_1}}$	(21.13)	m_{av} = mean or average module (mm) m_t = transverse module (mm)
$m_t = m_{av} + \left(\dfrac{b}{Z}\right)\sin\delta$	(21.14)	b = face width (mm) Z = number of teeth
$\tan\delta = i$ and $\delta_1 = 90 - \delta_2$	(21.15)	δ = reference angle/semi-cone angle/pitch angle
Spiral bevel gears		
$m_{av} \geq 1.15 \cos\beta_m \sqrt[3]{\dfrac{[M_t]}{y_v[\sigma_b]\psi_m Z_1}}$	(21.16)	β_m = average or mean spiral angle
Formulae for checking design of gears		
Spur gears		
$\sigma_b = \dfrac{(i \pm 1)}{amby}[M_t] \leq [\sigma_b]$	(21.17)	σ_b = calculated bending stress (MPa or N/mm^2) i = gear ratio (Z_2/Z_1) a = centre distance (mm)

(*Contd.*)

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Helical gears	
$\sigma_b = 0.7\dfrac{(i \pm 1)}{abm_n y_v}[M_t] \le [\sigma_b]$ (21.18)	
Herringbone gears	
$\sigma_b = 0.85\dfrac{(i \pm 1)}{abm_n y_v}[M_t] \le [\sigma_b]$ (21.19)	
Straight bevel gears	
$\sigma_b = \dfrac{R\sqrt{(i^2+1)}[M_t]}{(R-0.5b)^2\, bmy_v}\dfrac{1}{\cos\alpha} \le [\sigma_b]$ (21.20)	R = cone distance (mm) $R = 0.5m_t Z_1\sqrt{(i^2+1)}$ or $R = 0.5m_t\sqrt{Z_1^2 + Z_2^2}$ $m_t = m_{\text{av}} + \left(\dfrac{b}{Z}\right)\sin\delta$ $\tan\delta = i$ and $\delta_1 = 90 - \delta_2$ α = pressure angle
Spiral bevel gears	
$\sigma_b = \dfrac{0.7R\sqrt{(i^2+1)}[M_t]}{(R-0.5b)^2\, bm_n y_v} \le [\sigma_b]$ (21.21)	

(M. Movnin) and (V. Dobrovolsky)

Note: Use + sign for external gears and – sign for internal gears.

Table 21.26 *Lewis form factor (y) (For 20° full depth involute teeth) (f_0 = 1)*

Z	y	Z	y
12	0.308	35	0.452
14	0.330	40	0.465
16	0.355	45	0.471
18	0.377	50	0.477
20	0.389	60	0.490
22	0.402	80	0.499
24	0.414	100	0.505
26	0.427	150	0.515
28	0.434	300	0.521
30	0.440	Rack	0.550

Note: f_0 = height factor f_0 = 1 (for full depth teeth) f_0 = 0.8 (for stub teeth)

Table 21.27 *Virtual number of teeth*

Helical and herringbone gears	
$Z_v = \dfrac{Z}{\cos^3 \beta}$ (21.22)	Z = actual number of teeth Z_v = formative or virtual number of teeth β = helix angle
Straight bevel gears	
$Z_v = \dfrac{Z}{\cos \delta}$ (21.23)	δ = reference angle ($\tan \delta = i$)
Spiral bevel gears	
$Z_v = \dfrac{Z}{\cos \delta \cos^3 \beta_m}$ (21.24)	β_m = average or mean spiral angle

21.7 DESIGN FORMULAE–SURFACE CONTACT STRENGTH

Table 21.28 *Formulae based on surface contact strength*

Formulae for designing gears	
Spur gears	
$a \geq (i \pm 1) \sqrt[3]{\left(\dfrac{0.74}{[\sigma_c]}\right)^2 \dfrac{E[M_t]}{i\psi}}$ (21.25)	a = centre distance (mm) (Table 21.6) i = gear ratio (Z_2/Z_1) (Table 21.5) $[\sigma_c]$ = design contact stress (MPa or N/mm^2) (Table 21.22) E = equivalent Young's modulus (MPa or N/mm^2) (Table 21.29) $[M_t]$ = design twisting moment or design torque (N-mm) (Table 21.9) $\psi = \left(\dfrac{b}{a}\right)$ ratio or width factor (Tables 21.30 and 21.14)
Helical and herringbone gears	
$a \geq (i \pm 1) \sqrt[3]{\left(\dfrac{0.7}{[\sigma_c]}\right)^2 \dfrac{E[M_t]}{i\psi}}$ (21.26)	
Straight and spiral bevel gears	
$R \geq \psi_y \sqrt{(i^2+1)} \sqrt[3]{\left(\dfrac{0.72}{(\psi_y - 0.5)[\sigma_c]}\right)^2 \dfrac{E[M_t]}{i}}$ (21.27)	R = cone distance (mm) $R = 0.5 m_t Z_1 \sqrt{(i^2+1)}$ $m_t = m_{av} + \left(\dfrac{b}{Z}\right)\sin \delta$ $\tan \delta = i$ and $\delta_1 = 90 - \delta_2$ $\psi_y = \left(\dfrac{R}{b}\right)$ ratio (Table 21.31)

(*Contd.*)

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Formulae for checking design of gears	
Spur Gears	
$\sigma_c = 0.74\left(\frac{(i \pm 1)}{a}\right)\sqrt{\frac{(i \pm 1)}{ib} E[M_t]} \le [\sigma_c]$ (21.28)	σ_c = calculated contact stress (MPa or N/mm^2) b = face width of gear (mm)
Helical and herringbone gears	
$\sigma_c = 0.7\left(\frac{(i \pm 1)}{a}\right)\sqrt{\frac{(i \pm 1)}{ib} E[M_t]} \le [\sigma_c]$ (21.29)	
Straight and spiral bevel gears	
$\sigma_c = \frac{0.72}{(R - 0.5b)}\sqrt{\frac{\sqrt{(i^2 \pm 1)^3}}{ib} E[M_t]}$ (21.30)	

(M. Movnin) and (V. Dobrovolsky)

Note: Use + sign for external gears and – sign for internal gears

Table 21.29 *Equivalent Young's modulus (E)* $\left[E = \frac{2E_1E_2}{E_1 + E_2}\right]$

Pinion		Wheel			Equivalent Young's modulus (E) (MPa or N/mm^2)
Material	**Young's modulus (E_1) (MPa or N/mm^2)**	**Material**	**Tensile strength (σ_u) (MPa or N/mm^2)**	**Young's modulus (E_2) (MPa or N/mm^2)**	
Steel	215×10^3	Steel		215×10^3	215×10^3
		Cast iron	≤ 280	110×10^3	146×10^3
		Cast iron	> 280	140×10^3	170×10^3
		Bronze		120×10^3	155×10^3

Note: For other combinations of materials for pinion and wheel, use basic formulae $\left[E = \frac{2E_1E_2}{E_1 + E_2}\right]$

Table 21.30 *Recommended values of* (ψ) $\left[\psi = \left(\frac{b}{a}\right)\right]$

Construction of gear box	ψ
Open-type gear arrangement	0.1 to 0.315
Closed gear boxes (Speed reducer)	
(i) High speed (V = 8 to 25 m/s)	Up to 0.315
(ii) Medium speed (V = 3 to 8 m/s)	Up to 0.63
(iii) Low speed (V = 1 to 3 m/s)	Up to 1.0
Gear boxes with sliding gears	0.125 to 0.160

Note: (i) For light and medium duty applications, [$b \le d_1$]
(ii) For heavy duty applications, [$b \le 1.5d_1$]
b = face width (mm) d_1= pitch circle diameter of pinion (mm)
(iii) For standard values of (ψ), refer to Table 21.13.
(iv) Recommended values of (ψ) are also given in Table 21.14.

Table 21.31 *Recommended values of (ψ_y)* $\left[\psi_y = \left(\frac{R}{b}\right)\right]$

Construction of gear box			ψ_y
Speed reducers	Shafts mounted on roller bearings	i = 1 to 4	3
		i = 4 to 6	4
	Shafts mounted on journal and thrust bearings	i = 6	5

Note: The ratio (ψ_y) is usually from 3 to 4.

21.8 DESIGN CHECK FOR PLASTIC DEFORMATION OR BRITTLE FAILURE

Table 21.32 *Design check for plastic deformation or brittle crushing*

Check for plastic deformation or brittle crushing under surface contact stress	
$$\sigma_{c\max} = \sigma_c \sqrt{\frac{(M_t)_{\max}}{M_t}} \leq [\sigma_c]_{\max} \qquad (21.31)$$	$\sigma_{c\max}$ = maximum surface contact stress due to $(M_t)_{\max}$ (MPa or N/mm^2) σ_c = calculated surface contact stress (MPa or N/mm^2) (Table 21.28) $[\sigma_c]_{\max}$ = maximum allowable surface contact stress to avoid plastic deformation or brittle crushing (MPa or N/mm^2) (Table 21.33) $(M_t)_{\max}$ = instantaneous maximum twisting moment (during starting, braking or sudden stopping) (N-mm) M_t = nominal twisting moment or rated torque (N-mm) (Table 21.9)
Check for plastic deformation or brittle crushing under bending stress	
$$\sigma_{b\max} = \sigma_b \frac{(M_t)_{\max}}{M_t} \leq [\sigma_b]_{\max} \qquad (21.32)$$	$\sigma_{b\max}$ = maximum bending stress due to $(M_t)_{\max}$ (MPa or N/mm^2) σ_b = calculated bending stress (MPa or N/mm^2) (Table 21.25) $[\sigma_b]_{\max}$ = maximum allowable bending stress to avoid plastic deformation or brittle crushing (MPa or N/mm^2) (Table 21.34)

(V. Dobrovolsky)

Note: When instantaneous maximum twisting moment $(M_t)_{\max}$ is not known, assume, $[(M_t)_{\max} = 2\ M_t]$

(R/b) ratio

Table 21.33 *Maximum allowable surface contact stress $[\sigma_c]_{max}$*

Material	Surface hardness	$[\sigma_c]_{max}$ (MPa or N/mm^2)
Steel	HB ≤ 350	$3.1\sigma_y$
	HB > 350	42 HRC
Cast iron	HB ≤ 350	$1.8\sigma_u$

Note: (i) σ_u = minimum ultimate tensile strength (MPa or N/mm^2)
(ii) σ_y = minimum yield stress (MPa or N/mm^2)
(iii) HRC = Rockwell hardness C.
(iv) Refer to Table 21.8 for values of (σ_u) and (σ_y).

Table 21.34 *Maximum allowable bending stress $[\sigma_b]_{max}$*

Material	Core Hardness	$[\sigma_b]_{max}$ (MPa or N/mm^2)
Steel	No heat treatment HB ≤ 350	$0.8\sigma_y$
	Heat treated HB > 350	$0.36\left(\dfrac{\sigma_u}{k_\sigma}\right)$
Cast iron	—	$0.6\sigma_u$

Note: (i) σ_u = minimum ultimate tensile strength (MPa or N/mm^2)
(ii) σ_y = minimum yield stress (MPa or N/mm^2)
(iii) k_σ = fillet stress concentration factor (Table 21.20)

21.9 GEAR DIMENSIONS

Table 21.35 *Basic dimensions of spur gears*

Parameter	Formulae
Minimum number of teeth on pinion (Z_1)	$Z_1 \geq 17$ (for 20° full depth teeth) $Z_1 \geq 14$ (for 20° stub teeth) $Z_1 \geq 32$ (for 14.5° full depth teeth)
Number of teeth (Z_1 and Z_2)	$Z_1 = \dfrac{2a}{m(i+1)} \quad Z_2 = i Z_1$
Module (m)	$m = \dfrac{2a}{(Z_1 + Z_2)}$
Centre-to-centre distance (a)	$a = \dfrac{m(Z_1 + Z_2)}{2}$

(*Contd.*)

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Pitch circle diameter of pinion (d_1)	$d_1 = mZ_1$
Pitch circle diameter of wheel (d_2)	$d_2 = mZ_2$
Face width (b)	$b \cong 0.3$ a or ($\psi = 0.3$) b = (8 m to 12 m) or b = 10 m
Height factor (f_0)	f_0 = 1 (for full depth teeth) f_0 = 0.8 (for stub teeth)
Bottom clearance (c)	c = 0.25 m (for full depth teeth) c = 0.3 m (for stub teeth)
Tip or addendum circle diameter of pinion (d_{a1})	$d_{a1} = (Z_1 + 2f_0)m$
Tip or addendum circle diameter of wheel (d_{a2})	$d_{a2} = (Z_2 + 2f_0)m$
Root or dedendum circle diameter of pinion (d_{f1})	$d_{f1} = (Z_1 - 2f_0)m - 2c$
Root or dedendum circle diameter of wheel (d_{f2})	$d_{f2} = (Z_2 - 2f_0)m - 2c$
Tooth depth (h)	h = 2.25 m (for full depth teeth) h = 1.9 m (for stub teeth)

Table 21.36 *Basic dimensions of helical and herringbone gears*

Parameter	Formulae
Helix angle (β)	$\cos\beta = \dfrac{m_n(Z_1 + Z_2)}{2a}$ β = 8° to 25° (for helical gears) β = 25° to 40° (for herringbone gears)
Minimum number of teeth on pinion (Z_1)	$Z_1 \geq 17$ (for 20° full depth teeth) $Z_1 \geq 14$ (for 20° stub teeth) $Z_1 \geq 32$ (for 14.5° full depth teeth)
Number of teeth (Z_1 or Z_2)	$Z_1 = \dfrac{2a}{m_n}\dfrac{\cos\beta}{(i+1)}$ $\quad Z_2 = iZ_1$
Virtual number of teeth on pinion (Z_{v1})	$Z_{v1} = \dfrac{Z_1}{\cos^3\beta}$
Virtual number of teeth on wheel (Z_{v2})	$Z_{v2} = \dfrac{Z_2}{\cos^3\beta}$
Normal module (m_n)	$m_n = \dfrac{2a\cos\beta}{(Z_1 + Z_2)}$
Transverse module (m_t)	$m_t = \dfrac{m_n}{\cos\beta}$ or $m_t = \dfrac{2a}{(Z_1 + Z_2)}$
Centre-to-centre distance (a)	$a = \dfrac{m_n}{\cos\beta}\dfrac{(Z_1 + Z_2)}{2}$

(*Contd.*)

(*Contd.*)

Pitch circle diameter of pinion (d_1)	$d_1 = \frac{m_n}{\cos\beta} Z_1$
Pitch circle diameter of wheel (d_2)	$d_2 = \frac{m_n}{\cos\beta} Z_2$
Face width (b)	$b \cong 0.5$ a or ($\psi = 0.5$) $b = (8\ m_n$ to $12\ m_n)$ or $b = 10\ m_n$
Height factor (f_0)	$f_0 = 1$ (for full depth teeth) $f_0 = 0.8$ (for stub teeth)
Bottom clearance (c)	$c = 0.25\ m_n$ (for full depth teeth) $c = 0.3\ m_n$ (for stub teeth)
Tip or addendum circle diameter of pinion (d_{a1})	$d_{a1} = \left(\frac{Z_1}{\cos\beta} + 2f_0\right) m_n$
Tip or addendum circle diameter of wheel (d_{a2})	$d_{a2} = \left(\frac{Z_2}{\cos\beta} + 2f_0\right) m_n$
Root or dedendum circle diameter of pinion (d_{f1})	$d_{f1} = \left(\frac{Z_1}{\cos\beta} - 2f_0\right) m_n - 2c$
Root or dedendum circle diameter of wheel (d_{f2})	$d_{f2} = \left(\frac{Z_2}{\cos\beta} - 2f_0\right) m_n - 2c$
Tooth depth (h)	$h = 2.25\ m_n$ (for full depth teeth) $h = 1.9\ m_n$ (for stub teeth)

Table 21.37 *Minimum number of teeth on straight and spiral bevel pinion (Z_1)*

Speed ratio (*i*)	β_m		
	0° to 15°	**20° to 25°**	**30° to 40°**
1	17	17	17
1.6	15	15	14
2	13	12	11
≥ 3.15	12	10	8

Note: β_m = mean spiral angle.

Table 21.38 *Basic dimensions of straight and spiral bevel gears*

Parameter	**Formulae**
Mean spiral angle (β_m)	$\beta_m = 35°$ (most commonly used value) $\beta_m = 20°$ to $30°$ (for skew bevel gears) $\beta_m = 0°$ (for zerol bevel gears)

(*Contd.*)

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Reference angle/semi-cone angle/pitch angle (δ)	$\tan\delta = i$ and $\delta_1 = 90 - \delta_2$
Addendum angle (θ_a)	$\tan\theta_{a1} = \tan\theta_{a2} = \dfrac{m_t f_0}{R}$
Dedendum angle (θ_f)	$\tan\theta_{f1} = \tan\theta_{f2} = \dfrac{m_t(f_0 + c)}{R}$ $f_0 = 1$ and $c = 0.2$
Tip angle (δ_a)	$\delta_a = \delta + \theta_a$
Root angle (δ_f)	$\delta_f = \delta - \theta_f$
Normal module (m_n)	$m_n = m_t \cos\beta_m$
Transverse module (m_t)	$m_t = m_m + \left(\dfrac{b}{Z}\right)\sin\delta$ $m_t = m_m\left(\dfrac{\psi_y}{\psi_y - 0.5}\right)$ m_m = mean or average module
Cone distance (R)	$R = 0.5 m_t Z_1\sqrt{(i^2+1)}$ or $R = 0.5 m_t\sqrt{Z_1^2 + Z_2^2}$ $R = \dfrac{m_t Z_1}{2\sin\delta_1}$ $R = \dfrac{m_t Z_2}{2\sin\delta_2}$
Pitch circle diameter of pinion (d_1) (at the large end)	$d_1 = m_t Z_1$ (for straight bevel gears)
Pitch circle diameter of wheel (d_2) (at the large end)	$d_2 = m_t Z_2$ (for straight bevel gears)
Tip or addendum circle diameter of pinion (d_{a1})	$d_{a1} = m_t(Z_1 + 2\cos\delta_1)$
Tip or addendum circle diameter of wheel (d_{a2})	$d_{a2} = m_t(Z_2 + 2\cos\delta_2)$
Face width (b)	b = (0.3 R) or (10 m_t) whichever is smaller (For straight bevel gears)
Virtual number of teeth (Z_v)	$Z_v = \dfrac{Z}{\cos\delta}$ (For straight bevel gears) $Z_v = \dfrac{Z}{\cos\delta\cos^3\beta_m}$ (For spiral bevel gears)

Bevel Gears Calculation of Load Capacities

22

22.1 INITIAL DATA FOR LOAD CAPACITY

Table 22.1 *Initial data for bevel gears (Type–1)*

P	Nominal power transmitted by gears
n_1	Rotational speed of pinion
α_n	Normal pressure angle
z_1	Number of teeth on pinion
z_2	Number of teeth on wheel
Σ	Shaft angle
d_{e2}	Outer pitch diameter of wheel or (m_{mn}) Mean normal module
β_m	Mean spiral angle
b	Face width
Rz_1	Mean surface roughness of pinion
Rz_2	Mean surface roughness of wheel
C_1	Accuracy grade for pinion
C_2	Accuracy grade for wheel
HB_1	Surface hardness of pinion
HB_2	Surface hardness of wheel
L_H	Life of gears
ISO VG	Viscosity grade of lubricant
ρ_{a01}	Cutter edge radius for pinion
ρ_{a02}	Cutter edge radius for wheel
r_{c0}	Cutter radius

(*Contd.*)

(*Contd.*)

x_{sm1}	Thickness modification coefficient for pinion
x_{sm2}	Thickness modification coefficient for wheel
x_{hm1}	Profile shift coefficient for pinion
x_{hm2}	Profile shift coefficient for wheel
h^*_{f01}	Tool dedendum (related to m_{mn}) for pinion
h^*_{f02}	Tool dedendum (related to m_{mn}) for wheel
h^*_{a01}	Tool addendum (related to m_{mn}) for pinion
h^*_{a02}	Tool addendum (related to m_{mn}) for wheel
s_{pr1}	Protuberance for pinion
s_{pr2}	Protuberance for wheel

Note: (i) The initial data required for calculating load capacities is supplied in either of two commonly used forms: Type–1 (Table 22.1) or Type–2 (Table 22.2).
(ii) The equations for conversion of data between two forms: Type–1 and Type–2 are given in Table 22.3.
(iii) Refer to Table 22.4 for tool-dimensions with profile shift and thickness modification.

Table 22.2 *Initial data for bevel gears (Type–2)*

P	Nominal power transmitted by gears
n_1	Rotational speed of pinion
α_n	Normal pressure angle
z_1	Number of teeth on pinion
z_2	Number of teeth on wheel
Σ	Shaft angle
R_e	Outer cone distance or (m_{et}) Outer transverse module or (P_d) Outer diametral pitch
β_m	Mean spiral angle
b	Face width
δ_{a1}	Face angle of pinion
δ_{a2}	Face angle of wheel
θ_{f1}	Dedendum angle of pinion
θ_{f2}	Dedendum angle of wheel
Rz_1	Mean surface roughness of pinion
Rz_2	Mean surface roughness of wheel

(*Contd.*)

(*Contd.*)

C_1	Accuracy grade for pinion
C_2	Accuracy grade for wheel
HB_1	Surface hardness of pinion
HB_2	Surface hardness of wheel
L_H	Life of gears
ISO VG	Viscosity grade of lubricant
ρ_{a01}	Cutter edge radius for pinion
ρ_{a02}	Cutter edge radius for wheel
r_{c0}	Cutter radius
$s_{\mathrm{mn}1,2}$	Mean normal circular thickness or ($s_{\mathrm{mt}1,2}$) Mean transverse circular thickness or ($s_{\mathrm{amn}1,2}$) mean normal top land
h_{ae1}	Outer addendum for pinion
h_{ae2}	Outer addendum for wheel
h_{fe1}	Outer dedendum for pinion
h_{fe2}	Outer dedendum for wheel
s_{pr1}	Protuberance for pinion
s_{pr2}	Protuberance for wheel

22.2 FORMULAE FOR CALCULATION OF LOAD CAPACITIES

Table 22.3 *Basic formulae*

Gear ratio (u) z_2 = number of teeth on wheel z_1 = number of teeth on pinion δ_2 = pitch angle of wheel (º) δ_1 = pitch angle of pinion (º)	$$u = \frac{z_2}{z_1} = \frac{\sin \delta_2}{\sin \delta_1} \quad (22.1)$$
Pitch angle (δ) For Σ = 90º, Σ = shaft angle (º)	$$\tan \delta_1 = \frac{\sin \Sigma}{(\cos \Sigma + u)} \quad (22.2)$$ $$\delta_2 = (\Sigma - \delta_1) \quad (22.3)$$ $$\tan \delta_1 = \frac{1}{u} \quad \text{and} \quad \tan \delta_2 = u \quad (22.4)$$

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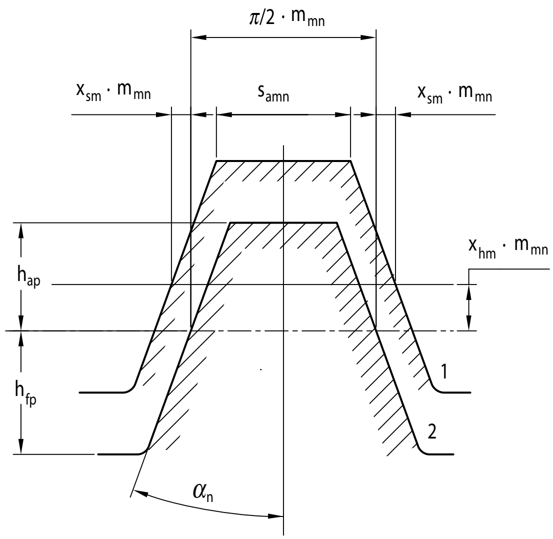
Outer cone distance (R_e) d_{e2} = outer pitch diameter of wheel (mm) d_{e1} = outer pitch diameter of pinion (mm)	$$R_e = \frac{0.5 d_{e2}}{\sin \delta_2} = \frac{0.5 d_{e1}}{\sin \delta_1}$$	(22.5)
Mean cone distance (R_m) b = face width (mm)	$$R_m = R_e - \left(\frac{b}{2}\right)$$	(22.6)
Outer transverse module (m_{et}) P_d = outer diametral pitch (mm^{-1})	$$m_{et} = \frac{d_{e2}}{z_2} = \frac{d_{e1}}{z_1} = \frac{1}{P_d}$$	(22.7)
Mean transverse module (m_{mt})	$$m_{mt} = \frac{R_m}{R_e} m_{et}$$	(22.8)
Mean normal module (m_{mn}) β_m = mean spiral angle (º)	$$m_{mn} = m_{mt} \cos \beta_m$$	(22.9)
Mean pitch diameter (d_m)	$$d_{m1} = d_{e1} - b \sin \delta_1 = \frac{m_{mn} z_1}{\cos \beta_m}$$ $$d_{m2} = d_{e2} - b \sin \delta_2 = \frac{m_{mn} z_2}{\cos \beta_m}$$	(22.10)
Addendum angle (θ_a) δ_{a1} = face angle of pinion (º) δ_{a2} = face angle of wheel (º)	$$\theta_{a1} = \delta_{a1} - \delta_1$$ $$\theta_{a2} = \delta_{a2} - \delta_2$$	(22.11)
Dedendum angle (θ_f) δ_{f1} = root angle of pinion (º) δ_{f2} = root angle of wheel (º)	$$\theta_{f1} = \delta_1 - \delta_{f1}$$ $$\theta_{f2} = \delta_2 - \delta_{f2}$$	(22.12)

Table 22.4 *Tool parameters*

Rack tooth profile

Note: In above figure, 1. Tooth profile with profile shift and thickness modification 2. Basic rack tooth profile
Tool equations for data of Type-1
In case of data of Type-1, values of (h_{f0}), (h_{a0}) and (ρ_{a0}) are generally taken from the manufacturer.
Dedendum of the tool related to mean normal module (h^*_{f0}) $$h^*_{f01} = \frac{h_{f01}}{m_{mn}} = \frac{h_{aP1}}{m_{mn}} = h^*_{aP1}$$ $$h^*_{f02} = \frac{h_{f02}}{m_{mn}} = \frac{h_{aP2}}{m_{mn}} = h^*_{aP2} \qquad (22.13)$$ h_{f01} = tool dedendum for pinion h_{f02} = tool dedendum for wheel h^*_{aP1} = addendum of basic rack tooth profile (related to m_{mn}) for pinion h^*_{aP2} = addendum of basic rack tooth profile (related to m_{mn}) for wheel
Addendum of tool related to mean normal module (h^*_{a0})
$$h^*_{a01} = \frac{h_{a01}}{m_{mn}} = \frac{h_{fP1}}{m_{mn}} = h^*_{fP1}$$ $$h^*_{a02} = \frac{h_{a02}}{m_{mn}} = \frac{h_{fP2}}{m_{mn}} = h^*_{fP2} \qquad (22.14)$$

(*Contd.*)

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h_{a01} = tool addendum for pinion
h_{a02} = tool addendum for wheel
h^*_{fP1} = dedendum of basic rack tooth profile (related to m_{mn}) for pinion
h^*_{fP2} = dedendum of basic rack tooth profile (related to m_{mn}) for wheel

Common values

$$\frac{\rho_{a0}}{m_{mn}} = 0.2 \text{ to } 0.4$$

$$h^*_{f0} = 1.0$$

$$h^*_{a0} = 1.25 \text{ to } 1.30 \qquad (22.15)$$

Tool depth at mid-face

Mean addendum (h_{am})

$$h_{am1} = m_{mn}(h^*_{f01} + x_{hm1})$$

$$h_{am2} = m_{mn}(h^*_{f02} + x_{hm2}) \qquad (22.16)$$

Mean dedendum (h_{fm})

$$h_{fm1} = m_{mn}(h^*_{a01} - x_{hm1})$$

$$h_{fm2} = m_{mn}(h^*_{a02} - x_{hm2}) \qquad (22.17)$$

Tool equations for data of Type-2

In case of data of Type-2, only the tool tip radius (ρ_{a0}) is given.

Mean addendum (h_{am})

$$h_{am1} = h_{ae1} - \frac{b}{2}\tan\theta_{a1}$$

$$h_{am2} = h_{ae2} - \frac{b}{2}\tan\theta_{a2} \qquad (22.18)$$

h_{ae1} = outer addendum of pinion
h_{ae2} = outer addendum of wheel
θ_{a1} = addendum angle of pinion
θ_{a2} = addendum angle of wheel

Mean dedendum (h_{fm})

$$h_{fm1} = h_{fe1} - \frac{b}{2}\tan\theta_{f1}$$

$$h_{fm2} = h_{fe2} - \frac{b}{2}\tan\theta_{f2} \qquad (22.19)$$

h_{fe1} = outer dedendum of pinion
h_{fe2} = outer dedendum of wheel
θ_{f1} = dedendum angle of pinion
θ_{f2} = dedendum angle of wheel

(*Contd.*)

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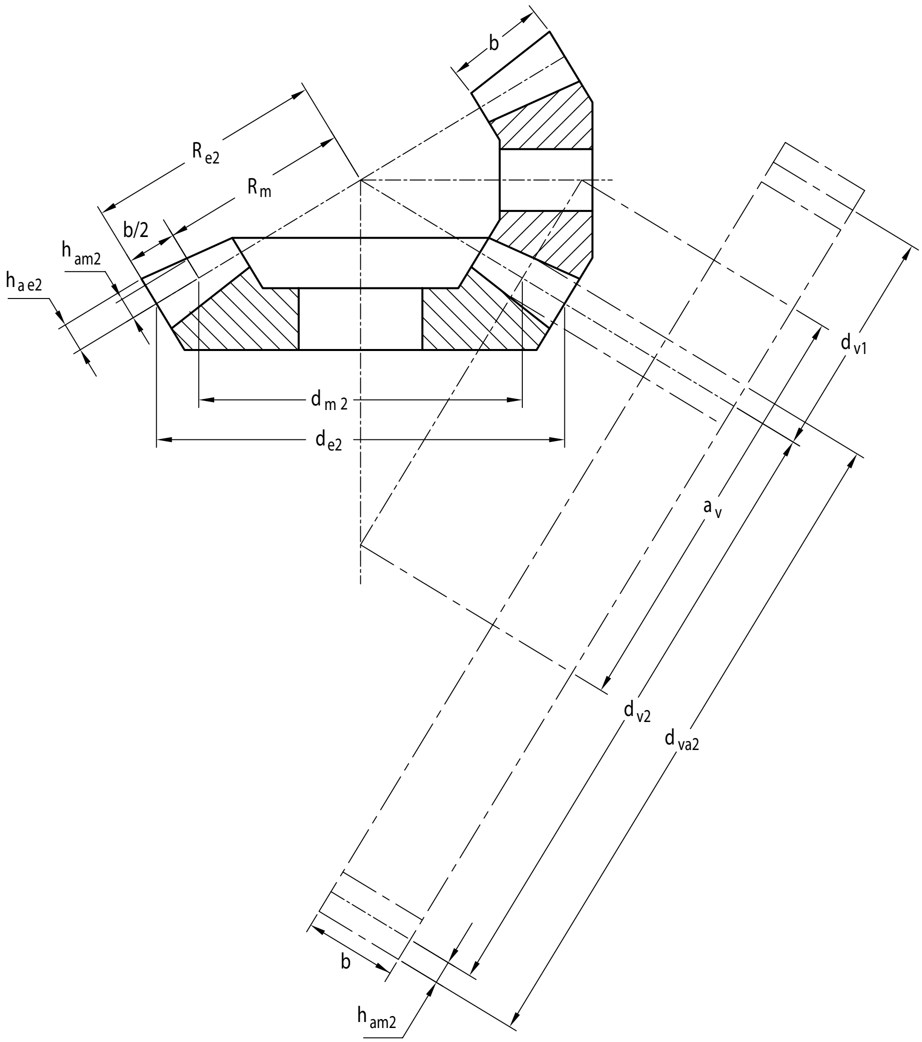
Profile shift coefficient (x_{hm}) $$x_{hm1} = \left(\frac{h_{am1} - h_{am2}}{2m_{mn}}\right)$$ $$x_{hm2} = \left(\frac{h_{am2} - h_{am1}}{2m_{mn}}\right) \qquad (22.20)$$

Table 22.5 *Virtual cylindrical gear in transverse section*



Note: For parameters of virtual cylindrical gears, shown in the above figure, the suffix '*m*', which indicates the conditions at mid face width, is not used.

Table 22.6 *Equations of virtual cylindrical gear in transverse section (at mid face width) (suffix v)*

Number of teeth (z_v) $$z_{v1} = \frac{z_1}{\cos \delta_1} \quad \text{and} \quad z_{v2} = \frac{z_2}{\cos \delta_2}$$ δ_1 = pitch angle of pinion δ_2 = pitch angle of wheel For $\Sigma = 90^\circ$, $$z_{v1} = z_1 \frac{\sqrt{u^2+1}}{u} \text{ and } z_{v2} = z_2\sqrt{u^2+1}$$ u = gear ratio of actual bevel gears	(22.21) (22.22)
Gear ratio (u_v) $$u_v = \frac{z_{v2}}{z_{v1}} = u \frac{\cos \delta_1}{\cos \delta_2}$$ For $\Sigma = 90^\circ$, $$u_v = \left(\frac{z_2}{z_1}\right)^2 = u^2$$	(22.23) (22.24)
Reference diameter (d_v) $$d_{v1} = \frac{d_{m1}}{\cos \delta_1} = \frac{d_{e1}}{\cos \delta_1} \times \frac{R_m}{R_e} \text{ and } d_{v2} = \frac{d_{m2}}{\cos \delta_2} = \frac{d_{e2}}{\cos \delta_2} \times \frac{R_m}{R_e}$$ For $\Sigma = 90^\circ$, $$d_{v1} = d_{m1}\frac{\sqrt{u^2+1}}{u} \text{ and } d_{v2} = u^2 d_{v1}$$ d_{m1} = mean pitch diameter of pinion d_{m2} = mean pitch diameter of wheel d_{e1} = outer pitch diameter of pinion d_{e2} = outer pitch diameter of wheel R_e = outer cone distance R_m = mean cone distance	(22.25) (22.26)
Centre distance (a_v) $$a_v = \frac{d_{v1} + d_{v2}}{2}$$	(22.27)
Tip diameter (d_{va}) $$d_{va1} = d_{v1} + 2h_{am1} \text{ and } d_{va2} = d_{v2} + 2h_{am2}$$ h_{am1} = mean addendum of pinion h_{am2} = mean addendum of wheel	(22.28)
Base diameter (d_{vb}) $$d_{vb1} = d_{v1} \cos \alpha_{vt} \text{ and } d_{vb2} = d_{v2} \cos \alpha_{vt}$$ $$\alpha_{vt} = \text{arc tan}\left(\frac{\tan \alpha_n}{\cos \beta_m}\right)$$ α_{vt} = transverse pressure angle of virtual cylindrical gear	(22.29) (22.30)

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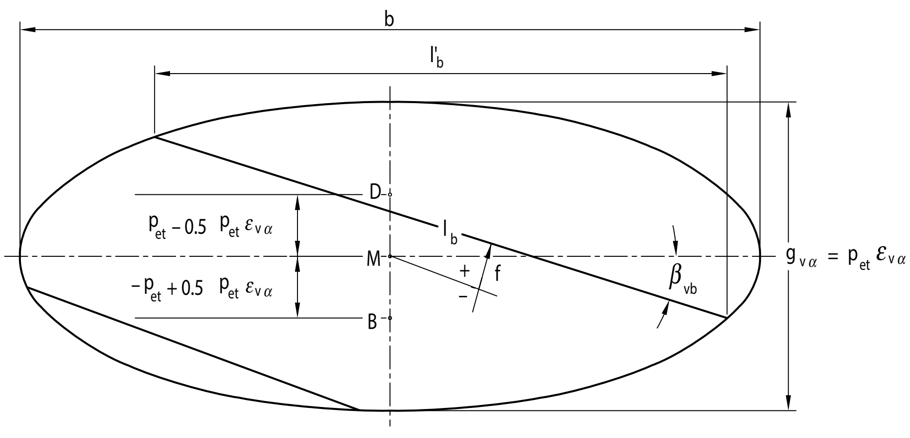
Helix angle at base circle (β_{vb})	$\beta_{vb} = \text{arc sin}\,(\sin \beta_m \cos \alpha_n)$	(22.31)
Transverse base pitch (p_{et})	$p_{et} = m_{mt}\, \pi \cos \alpha_{vt}$	(22.32)
Length of path of contact ($g_{v\alpha}$)	$g_{v\alpha} = \frac{1}{2}[\sqrt{(d_{va1}^2 - d_{vb1}^2)} + \sqrt{(d_{va2}^2 - d_{vb2}^2)}] - a_v \sin \alpha_{vt}$	(22.33)
Transverse contact ratio (ε_{va})	$\varepsilon_{v\alpha} = \frac{g_{v\alpha}}{p_{et}} = \frac{g_{v\alpha} \cos \beta_m}{m_{mn}\, \pi \cos \alpha_{vt}}$	(22.34)
Overlap ratio ($\varepsilon_{v\beta}$)	$\varepsilon_{v\beta} = \frac{b \sin \beta_m}{m_{mn}\, \pi}$	(22.35)
Modified contact ratio ($\varepsilon_{v\gamma}$)	$\varepsilon_{v\gamma} = \sqrt{\varepsilon_{v\alpha}^2 + \varepsilon_{v\beta}^2}$	(22.36)

Table 22.7 *Equations of virtual cylindrical gear in transverse section (at mid face width) (suffix v) Length of line of contact*



Since the bevel gears are usually crowned and barreled, it is assumed that the contact zone is bounded by an ellipse. The major axis of this ellipse is face width b as shown in the above figure.
B is the inner point of single contact
D is the outer point of single contact

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Length of line of contact (l_b)	
$$l_b = bg_{v\alpha}\frac{\sqrt{g_{v\alpha}^2\cos^2\beta_{vb}+b^2\sin^2\beta_{vb}-4f^2}}{g_{v\alpha}^2\cos^2\beta_{vb}+b^2\sin^2\beta_{vb}}$$ For $[g_{v\alpha}^2\cos^2\beta_{vb}+b^2\sin^2\beta_{vb}-4f^2]>0$	(22.37)
$$l_b = 0$$ For $[g_{v\alpha}^2\cos^2\beta_{vb}+b^2\sin^2\beta_{vb}-4f^2]\le 0$ f = distance to line of contact (Tables 22.8 and 22.9)	(22.38)
Equations (22.37) and (22.38) are calculated according to Tables 22.8 and 22.9 for (i) the tip line of contact with $f = f_t$; (ii) the middle line of contact with $f = f_m$; (iii) the root line of contact with $f = f_r$.	
Length of middle line of contact (l_{bm})	
$$l_{bm} = \frac{b\varepsilon_{v\alpha}}{\cos\beta_{vb}}\cdot\frac{\sqrt{\varepsilon_{v\gamma}^2-[(2-\varepsilon_{v\alpha})(1-\varepsilon_{v\beta})]^2}}{\varepsilon_{v\gamma}^2} \quad \text{for} \quad \varepsilon_{v\beta}<1$$	(22.39)
$$l_{bm} = \frac{b\varepsilon_{v\alpha}}{\cos\beta_{vb}\varepsilon_{v\gamma}} \text{ for } \varepsilon_{v\beta}\ge 1$$	(22.40)
Projected length of middle line of contact (l'_{bm})	
$$l'_{bm} = l_{bm}\cos\beta_{vb}$$	(22.41)

Table 22.8 *Distance f of the tip, middle and root line of contact in the zone of action (for contact stress)*

		For contact stress
$\varepsilon_{vb} = 0$	f_t	$-(p_{et}-0.5p_{et}\varepsilon_{v\alpha})\cos\beta_{vb}+p_{et}\cos\beta_{vb}$
	f_m	$-(p_{et}-0.5p_{et}\varepsilon_{v\alpha})\cos\beta_{vb}$
	f_r	$-(p_{et}-0.5p_{et}\varepsilon_{v\alpha})\cos\beta_{vb}-p_{et}\cos\beta_{vb}$
$0<\varepsilon_{v\beta}<1$	f_t	$-(p_{et}-0.5p_{et}\varepsilon_{v\alpha})\cos\beta_{vb}(1-\varepsilon_{v\beta})+p_{et}\cos\beta_{vb}$
	f_m	$-(p_{et}-0.5p_{et}\varepsilon_{v\alpha})\cos\beta_{vb}(1-\varepsilon_{v\beta})$
	f_r	$-(p_{et}-0.5p_{et}\varepsilon_{v\alpha})\cos\beta_{vb}(1-\varepsilon_{v\beta})-p_{et}\cos\beta_{vb}$
$\varepsilon_{v\beta}\ge 1$	f_t	$+p_{et}\cos\beta_{vb}$
	f_m	0
	f_r	$-p_{et}\cos\beta_{vb}$

Table 22.9 *Distance f of the tip, middle and root line of contact in the zone of action (for bending stress)*

		For bending stress
$\varepsilon_{v\beta}=0$	f_t	$(p_{et}-0.5p_{et}\,\varepsilon_{v\alpha})\cos\beta_{vb}+p_{et}\cos\beta_{vb}$
	f_m	$(p_{et}-0.5p_{et}\,\varepsilon_{v\alpha})\cos\beta_{vb}$
	f_r	$(p_{et}-0.5p_{et}\,\varepsilon_{v\alpha})\cos\beta_{vb}-p_{et}\cos\beta_{vb}$
$0<\varepsilon_{v\beta}<1$	f_t	$(p_{et}-0.5p_{et}\,\varepsilon_{v\alpha})\cos\beta_{vb}(1-\varepsilon_{v\beta})+p_{et}\cos\beta_{vb}$
	f_m	$(p_{et}-0.5p_{et}\,\varepsilon_{v\alpha})\cos\beta_{vb}(1-\varepsilon_{v\beta})$
	f_r	$(p_{et}-0.5p_{et}\,\varepsilon_{v\alpha})\cos\beta_{vb}(1-\varepsilon_{v\beta})-p_{et}\cos\beta_{vb}$
$\varepsilon_{v\beta}\geq 1$	f_t	$+p_{et}\cos\beta_{vb}$
	f_m	0
	f_r	$-p_{et}\cos\beta_{vb}$

Table 22.10 *Equations of virtual cylindrical gear in normal section (suffix vn)*

Number of teeth (z_{vn})

$$z_{vn1}=\frac{z_{v1}}{\cos^2\beta_{vb}\cos\beta_m} \tag{22.42}$$

$$z_{vn2}=u_v z_{vn1} \tag{22.43}$$

Reference diameter (d_{vn})

$$d_{vn1}=\frac{d_{v1}}{\cos^2\beta_{vb}}=z_{vn1}\,m_{mn} \tag{22.44}$$

$$d_{vn2}=u_v d_{vn1}=z_{vn2}\,m_{mn} \tag{22.45}$$

Centre distance (a_{vn})

$$a_{vn}=\frac{d_{vn1}+d_{vn2}}{2} \tag{22.46}$$

Tip diameter (d_{van})

$$d_{van1}=d_{vn1}+d_{va1}-d_{v1}=d_{vn1}+2h_{am1} \tag{22.47}$$

$$d_{van2}=d_{vn2}+d_{va2}-d_{v2}=d_{vn2}+2h_{am2} \tag{22.48}$$

Base diameter (d_{vbn})

$$d_{vbn1}=d_{vn1}\cos\alpha_n=z_{vn1}\,m_{mn}\cos\alpha_n \tag{22.49}$$

$$d_{vbn2}=d_{vn2}\cos\alpha_n=z_{vn2}\,m_{mn}\cos\alpha_n \tag{22.50}$$

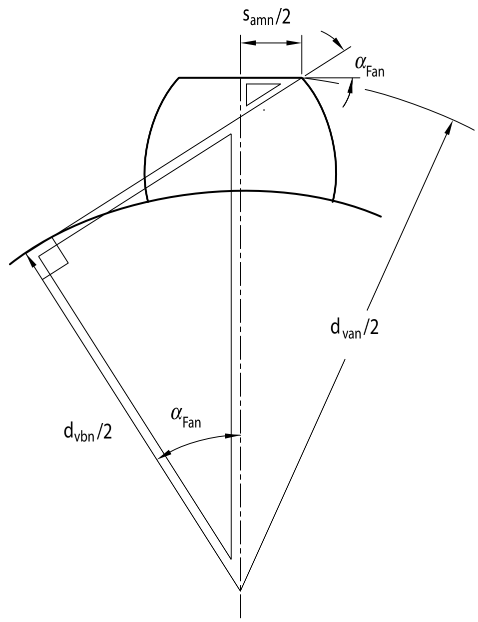
Length of path of contact ($g_{v\alpha n}$)

$$g_{v\alpha n}=\frac{1}{2}[\sqrt{(d_{van1}^2-d_{vbn1}^2)}+\sqrt{(d_{van2}^2-d_{vbn2}^2)}]-a_{vn}\sin\alpha_n \tag{22.51}$$

Contact ratio ($\varepsilon_{v\alpha n}$)

$$\varepsilon_{v\alpha n}=\frac{\varepsilon_{v\alpha}}{\cos^2\beta_{vb}} \tag{22.52}$$

Table 22.11 *Tooth thickness modification*



By similar triangles,

$$s_{amn} \sin \alpha_{Fan} = d_{van} \cos \alpha_{Fan} - d_{vbn} \tag{22.53}$$

s_{amn} = mean normal top land
α_{Fan} = load application angle at tip circle of virtual spur gear

Load application angle (α_{Fan})

$$\alpha_{Fan1} = \text{arc cos}\left(\frac{d_{van1} d_{vbn1} + s_{amn1}\sqrt{d_{van1}^2 + s_{amn1}^2 - d_{vbn1}^2}}{d_{van1}^2 + s_{amn1}^2}\right)$$

$$\alpha_{Fan2} = \text{arc cos}\left(\frac{d_{van2} d_{vbn2} + s_{amn2}\sqrt{d_{van2}^2 + s_{amn2}^2 - d_{vbn2}^2}}{d_{van2}^2 + s_{amn2}^2}\right) \tag{22.54}$$

α_{Fan1} = load application angle for pinion
α_{Fan2} = load application angle for wheel

Mean normal circular thickness (s_{mn})

$$s_{mn1} = \left[\sqrt{\left(\frac{d_{van1}}{d_{vbn1}}\right)^2 - 1} - \text{inv}\,\alpha_n - \alpha_{Fan1}\frac{\pi}{180}\right] d_{vn1}$$

$$s_{mn2} = \left[\sqrt{\left(\frac{d_{van2}}{d_{vbn2}}\right)^2 - 1} - \text{inv}\,\alpha_n - \alpha_{Fan2}\frac{\pi}{180}\right] d_{vn2} \tag{22.55}$$

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Thickness modification coefficient (x_{sm}) $$x_{sm1} = \frac{s_{mn1}}{2m_{mn}} - \frac{\pi}{4} - x_{hm1} \tan \alpha_n$$ $$x_{sm2} = \frac{s_{mn2}}{2m_{mn}} - \frac{\pi}{4} - x_{hm2} \tan \alpha_n \qquad (22.56)$$ x_{hm1} = profile shift coefficient for pinion x_{hm2} = profile shift coefficient for wheel
Mean transverse circular thickness (s_{mt}) $$s_{mt1} = s_{et1} \frac{m_{mt}}{m_{et}} = s_{et1} \frac{R_m}{R_e}$$ $$s_{mt2} = s_{et2} \frac{m_{mt}}{m_{et}} = s_{et2} \frac{R_m}{R_e} \qquad (22.57)$$ S_{et1} = transverse tooth thickness at the back cone of pinion S_{et2} = transverse tooth thickness at the back cone of wheel m_{et} = outer transverse module
Mean normal circular thickness (s_{mn}) $$s_{mn1} = s_{mt1} \cos \beta_m$$ $$s_{mn2} = s_{mt2} \cos \beta_m \qquad (22.58)$$

(ISO: 10300–1)

22.3 PERMISSIBLE CONTACT STRESS

Table 22.12 *Permissible contact stress (σ_{HP})*

$$\sigma_{HP} = \sigma_{H\lim} Z_{NT} Z_X Z_L Z_R Z_V Z_W \qquad (22.59)$$
σ_{HP} = permissible contact stress (MPa or N/mm^2) $\sigma_{H\lim}$ = endurance limit for contact stress (MPa or N/mm^2) (Tables 22.13 and 22.14) Z_{NT} = life factor (Table 22.15) Z_X = size factor (Table 22.16) Z_L = lubricant factor (Table 22.17 and 22.18) Z_R = roughness factor for contact stress (Table 22.19) Z_V = speed factor for contact stress (Table 22.20) Z_W = work hardening factor for contact stress (Table 22.22)

(ISO: 10300–2)

Note: (i) The endurance limit ($\sigma_{H\lim}$) is the repeated stress, which the material can sustain for at least 50×10^6 load cycles without fatigue failure.
(ii) The permissible contact stress (σ_{HP}) is calculated separately for pinion and wheel from the above equation.

Table 22.13 *Endurance limit for contact stress ($\sigma_{H\,lim}$)*

The value of allowable endurance limit ($\sigma_{H\,\mathrm{lim}}$) for contact stress is given by, $$\sigma_{H\,\mathrm{lim}} = A \cdot x + B$$
x = surface hardness (HBW or HV) (Given Data – Tables 22.1 and 22.2) A, B = constants for contact stress (Table 22.14) HBW, HV = units of hardness number (Table 22.14)

Note: The above equation for allowable endurance limit for contact stress ($\sigma_{H\,\mathrm{lim}}$) is based on experimental data of endurance tests of reference test gears under reference test conditions.

Table 22.14 *Constants A and B for contact stress in calculation of ($\sigma_{H\,lim}$)*

Material	Type	Quality	A	B	Units of Hardness	Min. Hardness	Max. hardness
Low carbon steels	Wrought normalized low carbon steel	ML/MQ	1.00	190	HBW	110	210
		ME	1.52	250	HBW	110	210
Cast steels	-	ML/MQ	0.986	131	HBW	140	210
		ME	1.143	237	HBW	140	210
Cast iron	Black malleable cast iron	ML/MQ	1.371	143	HBW	135	250
		ME	1.333	267	HBW	175	250
	Nodular cast iron	ML/MQ	1.434	211	HBW	175	300
		ME	1.500	250	HBW	200	300
	Grey cast iron	ML/MQ	1.033	132	HBW	150	240
		ME	1.465	122	HBW	175	275
Through hard-ened wrought steels	Carbon steels	ML	0.963	283	HV	135	210
		MQ	0.925	360	HV	135	210
		ME	0.838	432	HV	135	210
	Alloy steels	ML	1.313	188	HV	200	360
		MQ	1.313	373	HV	200	360
		ME	2.213	260	HV	200	390
Case hardened wrought steels	–	ML	0.0	1300	HV	600	800
		MQ	0.0	1500	HV	660	800
		ME	0.0	1650	HV	660	800
Flame or in-duction hard-ened wrought steels	–	ML	0.740	602	HV	485	615
		MQ	0.541	882	HV	500	615
		ME	0.505	1013	HV	500	615

(*Contd.*)

(*Contd.*)

Nitrided steels	Nitrided steels	ML	0.0	1125	HV	650	900
		MQ	0.0	1250	HV	650	900
		ME	0.0	1450	HV	650	900
	Through hardened Nitrided steels	ML	0.0	788	HV	450	650
		MQ	0.0	998	HV	450	650
		ME	0.0	1217	HV	450	650
Nitro-carburized steels	Through hardened steels	ML	0.0	650	HV	300	650
		MQ/ME	1.167	425	HV	300	450

Note: ML, MQ and ME are material quality grades.

(i) ML stands for modest demands on material quality and on heat treatment process during gear manufacturing.

(ii) MQ stands for requirements that can be met by experienced manufacturers at moderate cost.

(iii) ME stands for requirements that must be realised when a high degree of operating reliability is required.

Table 22.15 *Life factor for contact stress (Z_{NT})*

Material	**Number of load cycles (N_L)**	Z_{NT}
(i) Steels ($\sigma_B < 800$ N/mm^2) (ii) Through hardened steels ($\sigma_B \geq 800$ N/mm^2) (iii) Spheroidal cast iron (perlitic and bainitic) (iv) Black malleable cast iron (perlitic) (v) case hardened and flame or induction hardened steels	$N_L = 10^7$	1.3
	$N_L = 10^9$	1.0
	$N_L = 10^{10}$	0.85
(i) Steels ($\sigma_B \geq 800$ N/mm^2) (ii) Through hardened steels ($\sigma_B < 800$ N/mm^2) (iii) Spheroidal cast iron (perlitic and bainitic) (iv) Black malleable cast iron (perlitic) (v) Case hardened and flame or induction hardened steels (σ_B is tensile strength)	$N_L = 5 \times 10^7$	1.0
	$N_L = 10^{10}$	0.85
	$N_L = 10^{10}$ (optimum lubrication, material, manufacturing and experience)	1.0
(i) Grey cast iron (ii) Spheroidal cast iron (ferritic) (iii) Nitrided steels (iv) Through hardened and case hardened steels (nitrided)	$N_L = 2 \times 10^6$	1.0
	$N_L = 10^{10}$	0.85
	$N_L = 10^{10}$ (optimum lubrication, material, manufacturing and experience)	1.0
(i) Through hardened and case hardened steels (nitro-carburised)	$N_L = 2 \times 10^6$	1.0
	$N_L = 10^{10}$	0.85
	$N_L = 10^{10}$ (optimum lubrication, material, manufacturing and experience)	1.0

Note: (i) The life factor (Z_{NT}) accounts for increase in permissible contact stress, if the number of stress cycles is less than the life at endurance limit (that is 50×10^6 load cycles).

(ii) $N_L = 60 \times n \times L_H$

N_L = life in number of load cycles

n = speed of pinion or gear (rpm) (Given Data – Tables 22.1 and 22.2)

L_H = required life in hours (hr) (Given data – Tables 22.1 and 22.2)

Table 22.16 *Size factor for contact stress (Z_X)*

The size factor depends upon (i) material quality (furnace charge, cleanliness, forging) (ii) heat treatment, depth of hardening, distribution of hardening (iii) radius of flank curvature (iv) module in case of surface hardening, depth of hardened layer relative to the size of tooth (core supporting effect)
In absence of data, $Z_X = 1$

Note: The size factor (Z_X) accounts for the statistical evidence indicating that the stress level at which fatigue failure occurs decreases with an increase in size of gear tooth (due to larger number of discontinuities and irregularities in structure)

Table 22.17 *Lubricant factor for contact stress (Z_L)*

$Z_L = C_{ZL} + \dfrac{4(1 - C_{ZL})}{\left[1.2 + \dfrac{134}{\nu_{40}}\right]^2}$	C_{ZL} = factor for calculating Z_L ν_{40} = kinematic viscosity of lubricant at 40ºC (mm²/s) (Table 22.8)
$C_{ZL} = \left(\dfrac{\sigma_{H\,\lim} - 850}{350}\right) \times 0.08 + 0.83$ [if $850 \le \alpha_{H\,\lim} \le 1200$ MPa]	
$C_{ZL} = 0.83$ [if $\sigma_{H\,\lim} < 850$ MPa]	
$C_{ZL} = 0.91$ [if $\sigma_{H\,\lim} > 1200$ MPa]	
$\sigma_{H\,\lim}$ = Endurance limit for contact stress (MPa or N/mm²) (Tables 22.13 and 22.14)	

Note: The lubrication factor (Z_L) accounts for the influence of the type of lubricant and its viscosity on permissible contact stress.

Table 22.18 *Viscosity of lubricating oils (mm²/s)*

ISO viscosity grade	Kinematic viscosity range at 40ºC (ν_{40})	Kinematic viscosity range at 50ºC (ν_{50})
ISO VG 2	1.98 – 2.42	1.69 – 2.03
ISO VG 3	2.88 – 3.52	2.37 – 2.83
ISO VG 5	4.14 – 5.06	3.27 – 3.91

(*Contd.*)

(*Contd.*)

ISO VG 7	6.12 – 7.48	4.63 – 5.52
ISO VG 10	9.00 – 11.0	6.53 – 7.83
ISO VG 15	13.5 – 16.5	9.43 – 11.3
ISO VG 22	19.8 – 24.2	13.3 – 16.0
ISO VG 32	28.8 – 35.2	18.6 – 22.2
ISO VG 46	41.4 – 50.6	25.5 – 30.3
ISO VG 68	61.2 – 74.8	35.9 – 42.8
ISO VG 100	90.0 – 110	50.4 – 60.3
ISO VG 150	135 – 165	72.5 – 86.9
ISO VG 220	198 – 242	102 – 123
ISO VG 320	288 – 352	144 – 172
ISO VG 460	414 – 506	199 – 239
ISO VG 680	612 – 748	283 – 339
ISO VG 1000	900 – 1100	400 – 479
ISO VG 1500	1350 – 1650	575 – 688

Guidelines (For students only) –

(i) For high speed gears at low operating temperatures (10º to 16º C), low viscosity oil should be selected such as ISO VG 46.

(ii) For medium speed gears with centre distances less than 200 mm, oils with viscosity grade ISO VG 68 or ISO VG 100 are reasonable.

(iii) For medium speed gears, with centre distances more than 200 mm, oils with viscosity grade ISO VG 68 to ISO VG 220 are reasonable.

(iv) For low speed gears at high operating temperatures up to 52º C, oils with viscosity grade ISO VG 150 to ISO VG 320 are satisfactory.

Table 22.19 *Roughness factor for contact stress (Z_R)*

1. Radius of relative curvature (ρ_{red})

$$\rho_{\text{red}} = \frac{a_v \sin \alpha_{vt}}{\cos \beta_{vb}} \times \frac{u_v}{(1+u_v)^2}$$

ρ_{red} = radius of relative curvature (mm)
a_v = centre distance of virtual cylindrical gears (mm) (Table 22.6–Eq.22.27)
α_{vt} = transverse pressure angle of virtual cylindrical gears (º) (Table 22.6–Eq. 22.30)
β_{vb} = helix angle of base circle of virtual cylindrical gears (º) (Table 22.6–Eq. 22.31)
u_v = gear ratio of virtual cylindrical gears (Table 22.6– Eq. 22.23)

2. Mean roughness value Rz_{10}

$$Rz_{10} = \frac{Rz_1 + Rz_2}{2} \times \sqrt[3]{\frac{10}{\rho_{\text{red}}}}$$

Rz_{10} = mean roughness for gear pair with (ρ_{red} = 10 mm) (µm)
Rz_1 = mean roughness for pinion (µm)(Given Data – Tables 22.1 and 22.2)
Rz_2 = mean roughness for wheel (µm)(Given Data – Tables 22.1 and 22.2)

(*Contd.*)

(*Contd.*)

3. Factor C_{ZR} $$C_{ZR} = 0.12 + \left(\frac{1000 - \sigma_{H\,\lim}}{5000}\right) \quad [\text{if } 850 \le \sigma_{H\,\lim} \le 1200 \text{ N/mm}^2]$$ **Note:** (i) For $\sigma_{H\,\lim}$ below 850 N/mm^2, use $\sigma_{H\,\lim}$ = 850 N/mm^2 (ii) For $\sigma_{H\,\lim}$ above 1200 N/mm^2, use $\sigma_{H\,\lim}$ = 1200 N/mm^2	
4. Roughness factor for contact stress (Z_R) $$Z_R = \left(\frac{3}{Rz_{10}}\right)^{C_{ZR}}$$	

Note: (i) The roughness factor (Z_R) accounts for the influence of surface texture of tooth flanks on permissible contact stress.
(ii) When the roughness is given as Ra or CLA or AA value, the following approximation can be used,

$$Ra = \text{CLA} = \text{AA} = \left(\frac{Rz}{6}\right)$$

Table 22.20 *Speed factor for contact stress (Z_V)*

$Z_V = C_{ZV} + \dfrac{2(1.0 - C_{ZV})}{\sqrt{0.8 + \dfrac{32}{v_{mt}}}}$	C_{ZV} = factor for calculating Z_V v_{mt} = tangential velocity at reference cone at mid-face width (m/s)
$v_{mt} = \dfrac{d_{m1}n_1}{19098}$ or $v_{mt} = \dfrac{d_{m2}n_2}{19098}$	d_{m1} = mean pitch diameter of pinion (mm) (Table 22.3-Eq.22.10) d_{m2} = mean pitch diameter of wheel (mm) (Table 22.3-Eq.22.10) n_1 = rotational speed of pinion (rpm) (Given Data – Tables 22.1 and 22.2) n_2 = rotational speed of wheel (rpm)
$C_{ZV} = 0.85 + \left(\dfrac{\sigma_{H\,\lim} - 850}{350}\right) \times 0.08 \quad [\text{if } 850 \le \sigma_{H\,\lim} \le 1200 \text{ N/mm}^2]$	
$C_{ZV} = 0.85 \quad [\text{if } \sigma_{H\,\lim} < 850 \text{ N/mm}^2]$	
$C_{ZV} = 0.93 \quad [\text{if } \sigma_{H\,\lim} > 1200 \text{ N/mm}^2]$	

Note: The velocity or speed factor (Z_V) accounts for the influence of pitch line velocity on permissible contact stress.

Table 22.21 *Values of product of Z_L, Z_R and Z_V*

Condition	Product of Z_L, Z_R and Z_V
For through hardened, milled gear pairs	0.85
For gear pairs lapped after milling	0.92
For gear pairs ground after hardening, or for hard-cut gear pairs, with $Rz_{10} \le 4$ µm	1.0
For gear pairs ground after hardening, or for hard-cut gear pairs, with $Rz_{10} > 4$ µm	0.92

Table 22.22 *Work hardening factor for contact stress (Z_W)*

$Z_W = 1.2 - \left(\dfrac{\text{HB} - 130}{1700}\right)$	HB = Brinell hardness of tooth flanks of the softer gear of the pair (Given Data – Tables 22.1 and 22.2)
Z_W = 1.2 for HB < 130	
Z_W = 1.0 for HB > 470	
Z_W = 1.0 (if pinion and gear have the same hardness)	

Note: The work hardening factor (Z_W) accounts for the increase of surface durability due to meshing a through-hardened steel wheel with surface-hardened pinion having smooth tooth flanks ($Rz \le 6$ μm). In all other cases, (Z_W) = 1.0.

22.4 WORKING (ACTUAL) CONTACT STRESS

Table 22.23 *Working (actual) contact stress (σ_H)*

$$\sigma_H = \sigma_{HO}\sqrt{K_A K_V K_{H\beta} K_{H\alpha}} \le \sigma_{HP} \tag{22.60}$$

$$\sigma_{HO} = \sqrt{\frac{F_{mt}}{d_{v1} l_{bm}} \times \frac{u_v + 1}{u_v}} Z_{M-B} Z_H Z_E Z_{LS} Z_\beta Z_K \tag{22.61}$$

For shaft angle $\Sigma = \delta_1 + \delta_2 = 90°$,

$$\sigma_{HO} = \sqrt{\frac{F_{mt}}{d_{m1} l_{bm}} \times \frac{\sqrt{u^2 + 1}}{u}} Z_{M-B} Z_H Z_E Z_{LS} Z_\beta Z_K \tag{22.62}$$

where

$$F_{mt} = \frac{2000 T_1}{d_{m1}} \text{ and } T_1 = \frac{9549 P}{n_1} \tag{22.63}$$

σ_H = working or actual contact stress (MPa or N/mm^2)
σ_{HO} = nominal value of contact stress (MPa or N/mm^2)
K_A = application factor (Table 22.24, 22.25 and 22.26)
K_V = dynamic factor (Table 22.27)
$K_{H\beta}$ = face load factor for contact stress (Table 22.28)
$K_{H\alpha}$ = transverse load factor for contact stress (Table 22.30)
F_{mt} = nominal tangential force at reference cone at mid face width (N)
T_1 = nominal torque on pinion (Nm)
P = nominal power (kW) (Given Data – Tables 22.1 and 22.2)
n_1 = rotational speed of pinion (rpm) (Given Data – Tables 22.1 and 22.2)
d_{v1}= reference diameter of virtual cylindrical pinion (mm) (Table 22.6 – Eq. 22.25)
d_{m1} = mean pitch diameter of pinion (mm) (Table 22.3 – Eq. 22.10)
l_{bm} = length of middle line of contact (mm) (Table 22.7 – Eq. 22.39)
u_v = gear ratio of virtual cylindrical gear $\left(\dfrac{z_{v2}}{z_{v1}}\right)$ (Table 22.6 – Eq. 22.23)
u = gear ratio of bevel gears
Z_{M-B} = mid-zone factor for contact stress (Table 22.31)
Z_H = zone factor for contact stress (Table 22.33)
Z_E = elasticity factor for contact stress (Table 22.34)
Z_{LS} = load sharing factor (Table 22.36)
Z_β = spiral-angle factor for contact stress (Table 22.37)
Z_K = bevel gear factor (Table 22.38)

(ISO: 10300-2)

Note: The working contact stress (σ_H) is calculated separately for pinion and wheel from the above equation.

Table 22.24 *Application factor (K_A) for speed reducing gears*

Working characteristics of Driving machine (Table 22.25)	Working characteristics of Driven machine (Table 22.26)			
	Uniform	**Light shocks**	**Medium shocks**	**Heavy shocks**
	Application factor (K_A)			
Uniform	1.00	1.25	1.50	1.75 or higher
Light shocks	1.10	1.35	1.60	1.85 or higher
Medium shocks	1.25	1.50	1.75	2.00 or higher
Heavy shocks	1.50	1.75	2.00	2.25 or higher

Note: (i) The application factor accounts for dynamic overloads from the mean load due to sources external to gearing.
(ii) For speed-increasing devices, add ($0.01\ u^2$) to K_A, where $u = z_2/z_1 =$ gear ratio.

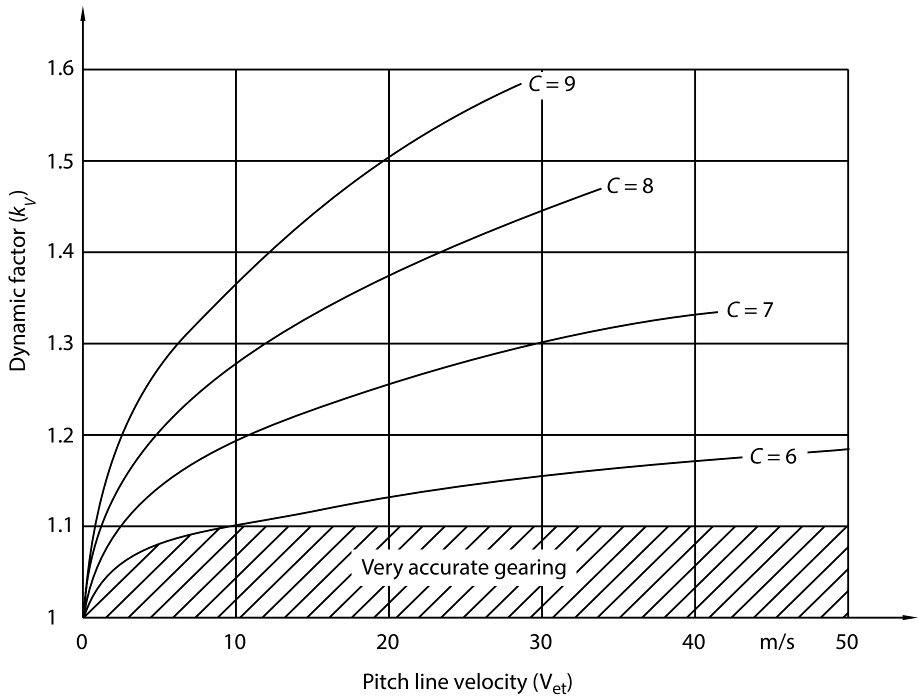
Table 22.25 *Examples of driving machines with different working characteristics*

Working characteristics	Driving machines
Uniform	Electric motor (e.g., dc motor), steam turbine or gas turbine with uniform operation
Light shocks	Steam turbine or gas turbine, hydraulic or electric motor(large, frequently occurring starting torques)
Medium shocks	Multi-cylinder internal combustion engine
Heavy shocks	Single cylinder internal combustion engine

Table 22.26 *Examples of driven machines with different working characteristics*

Working characteristics	Driven machines
Uniform	Steady load current generator, uniformly loaded belt conveyor or platform conveyor, worm conveyors, light lifts, feed drives for machine tools, ventilators, light weight centrifuges, centrifugal pumps, agitators and mixer for constant density material, presses, stamping machines, running gear.
Light shocks	Non-uniformly loaded belt conveyor or platform conveyor, machine tool main drive, heavy lifts, industrial and mine ventilator, heavy centrifuges, agitators and mixer for non-uniform density material, multi-cylinder piston pumps, rotating kilns.
Moderate shock	Rubber extruders, continuously operating mixers for rubber and plastics, light ball mills, wood working machinery, single cylinder piston pumps.
Heavy shock	Excavators, bucket chain drives, sieve drives, power shovels, heavy ball mill, stone crushers, foundry machines, heavy distribution pump, brick presses, briquette press.

Table 22.27 *Dynamic factor (K_V)*



Note: (i) The dynamic factor (K_V) accounts for the effect of gear tooth quality related to speed and load. It relates the total tooth load including internal dynamic load and transmitted tangential tooth load.

(ii) V_{et} = Tangential speed at outer end (heel) of reference cone (m/s).

(iii) $C = 6$, $C = 7$, are accuracy grades of gear tooth.

Table 22.28 *Face load factor for contact stress ($K_{H\beta}$)*

$K_{H\beta} = 1.5\ K_{H\beta-be}$ (for $b_e \geq 0.85\ b$)
$K_{H\beta} = 1.5 K_{H\beta-be} \cdot \dfrac{0.85}{b_e/b}$ (for $b_e < 0.85\ b$)
$K_{H\beta}$ = face load factor for contact stress $K_{H\beta-be}$ = bearing factor (Table 22.29) b = face width (mm) (Given Data – Tables 22.1 and 22.2) b_e = effective face width (mm)

Note: The face load factor accounts for the non-uniform distribution of load along the face width. $K_{H\beta}$ is defined as the ratio between the maximum load per unit face width and the mean load per unit face width.

Table 22.29 *Bearing or mounting factor ($K_{H\beta-be}$)*

Verification of contact pattern	Mounting conditions of pinion and wheel		
Contact pattern is checked:	*Neither member cantilever mounted*	*One member cantilever mounted*	*Both members cantilever mounted*
for each gear set in its housing under full load	1.00	1.00	1.00
for each gear set under light test load	1.05	1.10	1.25
for a sample of gear set and estimated for full load	1.20	1.32	1.50

Table 22.30 *Transverse load distribution factors $K_{H\alpha}$ and $K_{F\alpha}$ [$K_{H\alpha} \approx K_{F\alpha}$]*

Specific loading $\left(\frac{F_{mt}K_A}{b_e}\right)$	≥ 100 N/mm							< 100 N/mm
Gear quality grade as per ISO 1328 (using d_m and m_{mn})	Gr 6 and better	Gr 7	Gr 8	Gr 9	Gr 10	Gr 11	Gr 12	All accuracy grades
	$K_{H\alpha} \approx K_{F\alpha}$							
Surface hardened: Straight bevel gears	1.0		1.1	1.2	$K_{H\alpha} = 1/Z_{LS}^2$ or 1.2 whichever is greater $K_{F\alpha} = 1/Y_\varepsilon$ or 1.2 whichever is greater			
Surface hardened: Helical and spiral bevel gears	1.0	1.1	1.2	1.4	$\varepsilon_{v\alpha n}$ or 1.4 whichever is greater			
Unhardened: Straight bevel gears	1.0			1.1	1.2	$K_{H\alpha} = 1/Z_{LS}^2$ or 1.2 whichever is greater $K_{F\alpha} = 1/Y_\varepsilon$ or 1.2 whichever is greater		
Unhardened: Helical and spiral bevel gears	1.0		1.1	1.2	1.4	$\varepsilon_{v\alpha n}$ or 1.4 whichever is greater		

$K_{H\alpha}$ = transverse load distribution factor for contact stress
$K_{F\alpha}$ = transverse load distribution factor for bending stress
Z_{LS} = load sharing factor
$\varepsilon_{v\alpha n}$ = transverse contact ratio of virtual cylindrical gears in normal section (Table 22.10 – Eq. (22.52)

Note: (i) The transverse load distribution factor accounts for the distribution of actual load during the gear mesh.
(ii) For specific loading less than 100 N/mm and quality grade 5 and better, $K_{H\alpha}$ and $K_{F\alpha}$ may be taken as 1.0.

Table 22.31 *Mid-zone factor for contact stress (Z_{M-B})*

$$Z_{M-B} = \frac{\tan \alpha_{vt}}{\left[\sqrt{\left(\frac{d_{va1}}{d_{vb1}}\right)^2 - 1} - F_1 \cdot \frac{\pi}{z_{v1}}\right] \times \left[\sqrt{\left(\frac{d_{va2}}{d_{vb2}}\right)^2 - 1} - F_2 \cdot \frac{\pi}{z_{v2}}\right]}$$

Z_{M-B} = mid-zone factor
α_{vt} = transverse pressure angle of virtual cylindrical gear (°) (Table 22.6 – Eq. 22.30)
d_{va1} = tip diameter of virtual cylindrical pinion (mm) (Table 22.6 – Eq. 22.28)
d_{va2} = tip diameter of virtual cylindrical wheel (mm) (Table 22.6 – Eq. 22.28)
d_{vb1} = base diameter of virtual cylindrical pinion (mm) (Table 22.6 – Eq. 22.29)
d_{vb2} = base diameter of virtual cylindrical wheel (mm) (Table 22.6 – Eq. 22.29)
F_1, F_2 = auxiliary factors for calculation of mid-zone factor (Table 22.32)
z_{v1} = number of teeth on virtual cylindrical pinion (Table 22.6 – Eq. 22.21)
z_{v2} = number of teeth on virtual cylindrical wheel (Table 22.6 – Eq. 22.21)

Note: The mid-zone factor (Z_{M-B}) transforms zone factor Z_H and thereby contact pressure at the pitch point, to that at the determinant point of load application.

Table 22.32 *Auxiliary factors F_1 and F_2 for mid-zone factor*

	F_1	F_2
$\varepsilon_{v\beta} = 0$	2	$2(\varepsilon_{v\alpha} - 1)$
$0 < \varepsilon_{v\beta} < 1$	$2 + (\varepsilon_{v\alpha} - 2)\varepsilon_{v\beta}$	$2\varepsilon_{v\alpha} - 2 + (2 - \varepsilon_{v\alpha})\varepsilon_{v\beta}$
$\varepsilon_{v\beta} > 1$	$\varepsilon_{v\alpha}$	$\varepsilon_{v\alpha}$

ε_{va} = transverse contact ratio of virtual cylindrical gears (Table 22.6 – Eq. 22.34)
$\varepsilon_{v\beta}$ = overlap ratio of virtual cylindrical gears (Table 22.6 – Eq. 22.35)

Table 22.33 *Zone factor for contact stress (Z_H)*

For involute profile and x-zero bevel gears where $(x_1 + x_2 = 0)$ and $(\alpha_t = \alpha_{wt})$

$$Z_H = 2\sqrt{\frac{\cos \beta_{vb}}{\sin(2\alpha_{vt})}}$$

α_{vt} = transverse pressure angle of virtual cylindrical gear (°) (Table 22.6 – Eq. 22.30)
β_{vb} = helix angle at base circle of virtual cylindrical gear (°) (Table 22.6 – Eq. 22.31)

Note: The zone factor (Z_H) accounts for the influence of the flank curvature in the profile direction at the pitch point on contact stress.

Table 22.34 *Elasticity factor for contact stress (Z_E)*

$$Z_E = \sqrt{\frac{1}{\pi\left[\dfrac{1-\mu_1^2}{E_1} + \dfrac{1-\mu_2^2}{E_2}\right]}}$$

E_1 = modulus of elasticity of pinion material (MPa or N/mm^2) (Table 22.35)
E_2 = modulus of elasticity of wheel material (MPa or N/mm^2) (Table 22.35)
μ_1 = Poisson's ratio of pinion material (Table 22.35)
μ_2 = Poisson's ratio of wheel material (Table 22.35)

Note: (i) The elasticity factor accounts for the influence of material properties viz. modulus of elasticity and Poisson's ratio on working (actual) contact stress.
(ii) For steel on steel gear pair, Z_E = 189.8

Table 22.35 *Modulus of elasticity and Poisson's ratio for gear materials*

Material	Modulus of elasticity E (MPa or N/mm^2)	Poisson's ratio
Steel	206 000	0.3
Cast steel	202 000	0.3
Spheroidal cast iron	173 000	0.3
Cast tin bronze	103 000	0.3
Tin bronze	113 000	0.3
Grey cast iron	118 000	0.3

Table 22.36 *Load sharing factor (Z_{LS})*

$$Z_{LS} = 1 \quad (\text{For } \varepsilon_{v\gamma} \le 2)$$

$$Z_{LS} = \left(1 + 2\left[1 - \left(\frac{2}{\varepsilon_{v\gamma}}\right)^{1.5}\right]\sqrt{1 - \frac{4}{\varepsilon_{v\gamma}^2}}\right)^{-0.5} \quad (\text{For } \varepsilon_{v\gamma} > 2 \text{ and } \varepsilon_{v\beta} > 1)$$

$\varepsilon_{v\gamma}$ = modified contact ratio (Table 22.6 – Eq. 22.36)
$\varepsilon_{v\beta}$ = overlap ratio of virtual cylindrical gears (Table 22.6 – Eq. 22.35)

Note: The load sharing factor (Z_{LS}) accounts for load sharing between two or more pairs of teeth.

Table 22.37 *Spiral-angle factor for contact stress (Z_β)*

$$Z_\beta = \sqrt{\cos \beta_m}$$

β_m = mean spiral angle (º) (Given Data – Tables 22.1 and 22.2)

Note: The spiral-angle factor (Z_β) accounts for the influence of spiral angle on contact stress.

Table 22.38 *Bevel gear factor (Z_K)*

$Z_K = 0.8$

Note: The bevel gear factor (Z_K) accounts for the difference between bevel gear and cylindrical gear loading.

22.5 FACTOR OF SAFETY FOR CONTACT STRESS

Table 22.39 *Factor of safety for contact stress (S_H)*

$$S_H = \frac{\sigma_{HP}}{\sigma_H} \quad (22.64)$$

S_H = factor of safety based on contact stress
σ_{HP} = permissible contact stress (MPa or N/mm^2) (Table 22.12)
σ_H = working (actual) contact stress (MPa or N/mm^2) (Table 22.23)

Note: The factor of safety (S_H) is calculated separately for pinion and wheel from the above equation.

22.6 PERMISSIBLE BENDING STRESS

Table 22.40 *Permissible bending stress (σ_{FP})*

$$\sigma_{FP} = \sigma_{F\,\mathrm{lim}}\, Y_{ST}\, Y_{NT}\, Y_{\delta\mathrm{rel}\,T}\, Y_{R\,\mathrm{rel}\,T}\, Y_X \quad (22.65)$$

σ_{FP} = permissible bending stress at the root of tooth (MPa or N/mm^2)
$\sigma_{F\,\mathrm{lim}}$ = nominal stress number in bending (MPa or N/mm^2) (Tables 22.41 and 22.42)
Y_{ST} = stress correction factor for the dimensions of standard test gear (Y_{ST} = 2.0)
Y_{NT} = life factor (Table 22.43)
$Y_{\delta\mathrm{rel}\,T}$ = relative sensitivity factor (Table 22.44)
$Y_{R\,\mathrm{rel}\,T}$ = relative surface condition factor (Table 22.45)
Y_X = size factor (Table 22.46)

(ISO: 10300-3)

Note: The permissible bending stress (σ_{FP}) at the root of tooth is determined separately for pinion and wheel.

Table 22.41 *Nominal stress number in bending ($\sigma_{F\,lim}$)*

The value of nominal stress number in bending ($\sigma_{F\,\mathrm{lim}}$) is given by,

$$\sigma_{F\,\mathrm{lim}} = A \cdot x + B$$

x = surface hardness (HBW or HV) (Given Data – Table 22.1 and 22.2)
A, B = constants for bending stress (Table 22.42)
HBW, HV = units of hardness number (Table 22.42)

Note: The above equation for nominal stress number in bending ($\sigma_{F\,\mathrm{lim}}$) is based on experimental data of endurance tests of reference test gears under reference test conditions.

Table 22.42 *Constants A and B for bending stress in calculation of* ($\sigma_{F\ lim}$)

Material	Type	Quality	A	B	Units of hardness	Min. hardness	Max. hardness
Low carbon steels	Wrought normalized low carbon steel	ML/MQ	0.455	69	HBW	110	210
		ME	0.386	147	HBW	110	210
Cast steels	–	ML/MQ	0.313	62	HBW	140	210
		ME	0.254	137	HBW	140	210
Cast iron	Black malleable cast iron	ML/MQ	0.345	77	HBW	135	250
		ME	0.403	128	HBW	175	250
	Nodular cast iron	ML/MQ	0.350	119	HBW	175	300
		ME	0.380	134	HBW	200	300
	Grey cast iron	ML/MQ	0.256	8	HBW	150	240
		ME	0.200	53	HBW	175	275
Through hardened wrought steels	Carbon steels	ML	0.250	108	HV	115	215
		MQ	0.240	163	HV	115	215
		ME	0.283	202	HV	115	215
	Alloy steels	ML	0.423	104	HV	200	360
		MQ	0.425	187	HV	200	360
		ME	0.358	231	HV	200	390
Case hardened wrought steels	–	ML	0.0	312	HV	600	800
		MQ	0.0	425	HV	660	800
		ME	0.0	525	HV	660	800
Flame or induction hardened wrought steels	–	ML	0.305	76	HV	485	615
		MQ	0.138	290	HV	500	570
		ME	0.271	237	HV	500	615
Nitrided steels	Nitrided steels	ML	0.0	270	HV	650	900
		MQ	0.0	420	HV	650	900
		ME	0.0	468	HV	650	900
	Through hardened Nitrided steels	ML	0.0	258	HV	450	650
		MQ	0.0	363	HV	450	650
		ME	0.0	432	HV	450	650
Nitro-carburized steels	Through hardened steels	ML	0.0	224	HV	300	650
		MQ/ME	0.653	94	HV	300	450

Note: ML, MQ and ME are material quality grades.

(i) ML stands for modest demands on material quality and on heat treatment process during gear manufacturing.

(ii) MQ stands for requirements that can be met by experienced manufacturers at moderate cost.

(iii) ME stands for requirements that must be realised when a high degree of operating reliability is required.

Table 22.43 *Life factor (Y_{NT})*

Material	Number of load cycles (N_L)	Y_{NT}
(i) Through hardened steels ($\sigma_B \geq 800$ N/mm^2) (ii) Spheroidal cast iron (perlitic and bainitic) (iii) Black malleable cast iron (perlitic) (σ_B is tensile strength)	$N_L \leq 10^4$ static	2.5
	$N_L = 3 \times 10^6$ endurance	1.0
	$N_L = 10^{10}$ endurance	0.85
	$N_L = 10^{10}$ endurance (optimum conditions, material, manufacturing and experience)	1.0
(i) Case hardening steels (ii) Flame or induction hardened steels (including root fillet)	$N_L \leq 10^3$ static	2.5
	$N_L = 3 \times 10^6$ endurance	1.0
	$N_L = 10^{10}$ endurance	0.85
	$N_L = 10^{10}$ endurance (optimum conditions, material, manufacturing and experience)	1.0
(i) Steel ($\sigma_B < 800$ N/mm^2) (ii) Nitrided steels (iii) Through hardened and case hardened steels (nitrided) (iv) Grey cast iron (v) Spheroidal cast iron (ferritic)	$N_L \leq 10^3$ static	1.6
	$N_L = 3 \times 10^6$ endurance	1.0
	$N_L = 10^{10}$ endurance	0.85
	$N_L = 10^{10}$ endurance (optimum conditions, material, manufacturing and experience)	1.0
(i) Through hardened and case hardened steels (nitro-carburized)	$N_L \leq 10^3$ static	1.1
	$N_L = 3 \times 10^6$ endurance	1.0
	$N_L = 10^{10}$ endurance	0.85
	$N_L = 10^{10}$ endurance (optimum conditions, material, manufacturing and experience)	1.0

Note: (i) The life factor (Y_{NT}) accounts for increase in permissible bending stress, if the number of stress cycles is less than (3×10^6) load cycles.

(ii) $N_L = 60 \times n \times L_H$

N_L= life in number of load cycles

n = speed of pinion or gear (rpm) (Given Data – Tables 22.1 and 22.2)

L_H = required life in hours (hr) (Given Data – Tables 22.1 and 22.2)

Table 22.44 *Relative sensitivity factor ($Y_{\delta relT}$)*

$Y_{\delta \text{rel } T} = 1.00$ (for industrial gears with $q_s \geq 1.5$)
$Y_{\delta \text{rel } T} = 0.95$ (for industrial gears with $q_s < 1.5$)
q_s = notch parameter
$$q_s = \frac{s_{Fn}}{2\rho_F}$$
s_{Fn} = tooth root chord in calculation section (Table 22.48) ρ_F = fillet radius at contact point of 30° tangent (Table 22.48)

Note: The dynamic sensitivity factor ($Y_{\delta \text{rel } T}$) indicates the amount by which the theoretical stress peak exceeds the allowable stress number in case of fatigue failure.

Table 22.45 *Relative surface condition factor ($Y_{R\ rel\ T}$)*

Range ($Rz < 1$ µm) For through hardened and case hardened steels, $Y_{R \text{ rel } T} = 1.12$ For soft steels, $Y_{R \text{ rel } T} = 1.07$ For grey cast iron, nitrided, and nitro-carburized steels, $Y_{R \text{ rel } T} = 1.025$
Range (1 µm $\leq Rz \leq$ 40 µm) For through hardened and case hardened steels, $Y_{R\text{rel}T} = \frac{Y_R}{Y_{RT}} = 1.674 - 0.529(Rz + 1)^{1/10}$ For soft steels, $Y_{R\text{rel}T} = \frac{Y_R}{Y_{RT}} = 5.306 - 4.203(Rz + 1)^{1/100}$ For grey cast iron, nitrided, and nitro-carburized steels, $Y_{R\text{rel}T} = \frac{Y_R}{Y_{RT}} = 4.299 - 3.259(Rz + 1)^{1/200}$
Rz = Mean surface roughness at root (µm)

Note: The surface condition factor ($Y_{R \text{ rel } T}$) accounts for the dependence of the tooth root strength on surface condition at the root (roughness in root fillet)

Table 22.46 *Size factor (Y_X)*

For structural and through hardened steels, spheroidal cast iron, perlitic malleable cast iron, $Y_X = 1.03 - 0.006\ m_{mn}$ (with the restriction $0.85 \leq Y_X \leq 1.0$)

(*Contd.*)

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For case, flame, induction hardened steels, nitrided or nitro-carburized steels, $Y_X = 1.05 - 0.01\ m_{mn}$ (with the restriction $0.80 \le Y_X \le 1.0$)
For grey cast iron, $Y_X = 1.075 - 0.015\ m_{mn}$ (with the restriction $0.70 \le Y_X \le 1.0$)
m_{mn} = Mean normal module (mm) (Table 22.3 – Eq. 22.9)

Note: The size factor (Y_X) accounts for the decrease in strength with increasing size of tooth.

22.7 WORKING (ACTUAL) BENDING STRESS

Table 22.47 *Working (actual) bending stress (σ_F)*

$$\sigma_F = \left(\frac{F_{mt}}{bm_{mn}}\right) Y_{Fa} Y_{Sa} Y_\varepsilon Y_K Y_{LS} K_A K_V K_{F\beta} K_{F\alpha} \quad (22.66)$$

where,

$$F_{mt} = \frac{2000 T_1}{d_{m1}} \quad \text{and} \quad T_1 = \frac{9549 P}{n_1} \quad (22.67)$$

σ_F = working (actual) bending stress (tooth root stress) (MPa or N/mm^2)
F_{mt} = nominal tangential force at reference cone at mid face width (N)
T_1 = nominal torque on pinion (Nm)
P = nominal power (kW) (Given Data – Tables 22.1 and 22.2)
n_1 = rotational speed of pinion (rpm) (Given Data – Tables 22.1 and 22.2)
d_{m1} = mean pitch diameter of pinion (mm) (Table 22.3 – Eq. 22.10)
b = face width (mm)
m_{mn} = mean normal module (mm) (Table 22.3 – Eq. 22.9)
Y_{Fa} = tooth form factor (Tables 22.48 and 22.49)
Y_{Sa} = stress concentration factor (Table 22.50)
Y_ε = contact ratio factor (Table 22.51)
Y_K = bevel gear factor (Table 22.52)
Y_{LS} = load sharing factor (Table 22.53)
K_A = application factor (Tables 22.24, 22.25 and 22.26)
K_V = dynamic load factor (Table 22.27)
$K_{F\beta}$ = face load distribution factor for bending stress (Table 22.54)
$K_{F\alpha}$ = transverse load distribution factor for bending stress (Table 22.30)

(ISO: 10300-3)

Note: The working bending stress (σ_F) is calculated separately for pinion and wheel from the above equation.

Table 22.48 *Tooth form factor (Y_{Fa}) (for generated gear)*

1. Auxiliary parameter (E)
$$E = \left(\frac{\pi}{4} - x_{sm}\right) m_{mn} - h_{a0} \tan \alpha_n - \frac{\rho_{a0}(1 - \sin \alpha_n) - s_{pr}}{\cos \alpha_n}$$
x_{sm} = thickness modification coefficient (Table 22.11 – Eq. 22.56) m_{mn} = mean normal module (mm) (Table 22.3 – Eq. 22.9) h_{a0} = tool addendum (mm) (Given Data – Tables 22.1 and 22.2) α_n = normal pressure angle (°) (Given Data – Tables 22.1 and 22.2) ρ_{a0} = cutter edge radius (mm) (Given Data – Tables 22.1 and 22.2) s_{pr} = amount of protuberance (mm) (Given Data – Tables 22.1 and 22.2)
2. Auxiliary parameter (G)
$$G = \frac{\rho_{a0}}{m_{mn}} - \frac{h_{a0}}{m_{mn}} + x_{hm}$$
x_{hm} = profile shift coefficient (Table 22.4 – Eq. 22.20)
3. Auxiliary parameter (H)
$$H = \frac{2}{z_{vn}}\left(\frac{\pi}{2} - \frac{E}{m_{mn}}\right) - \frac{\pi}{3}$$
z_{vn} = Number of teeth on virtual cylindrical gear in normal section (Table 22.10 – Eq. 22.42)
4. Auxiliary parameter (υ)
$$\upsilon = \frac{2G}{z_{vn}} \tan \upsilon - H$$
For solution of above transcendent equation, substitute $\upsilon = \pi/6$ as initial value. In most of the cases, the equation converges after a few iteration steps.
5. Tooth root chord (s_{Fn})
$$\frac{s_{Fn}}{m_{mn}} = z_{vn} \sin\left(\frac{\pi}{3} - \upsilon\right) + \sqrt{3}\left(\frac{G}{\cos \upsilon} - \frac{\rho_{a0}}{m_{mn}}\right)$$
s_{Fn} = tooth root chord in calculation section (mm)
6. Fillet radius (ρ_F) at contact point of 30° tangent
$$\frac{\rho_F}{m_{mn}} = \frac{\rho_{a0}}{m_{mn}} + \frac{2G^2}{\cos \upsilon(z_{vn} \cos^2 \upsilon - 2G)}$$
ρ_F = fillet radius at contact point of 30° tangent (mm)

(*Contd.*)

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7. Bending moment arm (h_{Fa})
$$\alpha_{an} = \text{arc cos}\left(\frac{d_{vbn}}{d_{van}}\right)$$ $$\gamma_a = \frac{1}{z_{vn}}\left[\frac{\pi}{2} + 2(x_{hm} \tan \alpha_n + x_{sm})\right] + \text{inv } \alpha_n - \text{inv } \alpha_{an}$$ $$\alpha_{Fan} = \alpha_{an} - \gamma_a$$ $$\frac{h_{Fa}}{m_{mn}} = \frac{1}{2}\left[(\cos \gamma_a - \sin \gamma_a \tan \alpha_{Fan})\frac{d_{van}}{m_{mn}} - z_{vn} \cos\left(\frac{\pi}{3} - \upsilon\right) - \frac{G}{\cos \upsilon} + \frac{\rho_{a0}}{m_{mn}}\right]$$
d_{vbn} = base circle diameter of virtual cylindrical gear in normal section (mm) d_{van} = tip diameter of virtual cylindrical gear in normal section (mm) α_{Fan} = load application angle at the tip circle of virtual spur gear (º) γ_a = angle for tooth form and tooth correction factor (º)
8. Tooth form factor (Y_{Fa})
For generated gears $$Y_{Fa} = \frac{6\frac{h_{Fa}}{m_{mn}}\cos \alpha_{Fan}}{\left(\frac{s_{Fn}}{m_{mn}}\right)^2 \cos \alpha_n}$$

Note: The tooth form factor (Y_{Fa}) accounts for the influence of the tooth form on nominal bending stress in case of load application at the tip. It is determined separately for pinion and wheel.

Table 22.49 *Tooth form factor (Y_{Fa}) (for gears made by form cutting)*

1. Auxiliary parameter (E)
$$E = \left(\frac{\pi}{4} - x_{sm}\right)m_{mn} - h_{a0} \tan \alpha_n - \frac{\rho_{a0}(1 - \sin \alpha_n) - s_{pr}}{\cos \alpha_n}$$
x_{sm} = thickness modification coefficient (Table 22.11 – Eq. 22.56) m_{mn} = mean normal module (mm) (Table 22.3 – Eq. 22.9) h_{a0} = tool addendum (mm) (Given Data – Tables 22.1 and 22.2) α_n = normal pressure angle (º) (Given Data – Tables 22.1 and 22.2) ρ_{a0} = cutter edge radius (mm) (Given Data – Tables 22.1 and 22.2) s_{pr} = amount of protuberance (mm) (Given Data – Tables 22.1 and 22.2)
2. Tooth root thickness (s_{Fn2})
$$s_{Fn2} = \pi m_{mn} - 2E - 2\rho_{a02} \cos 30°$$
3. Fillet radius (ρ_{F2}) at contact point of 30º tangent
$$\rho_{F2} = \rho_{a02}$$

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4. Bending moment arm (h_{Fa2})
$$h_{Fa2} = h_{a02} - \frac{\rho_{a02}}{2} + m_{mn} - \left(\frac{\pi}{4} + x_{sm2} - \tan\alpha_n\right) m_{mn} \tan\alpha_n$$
5. Tooth form factor (Y_{Fa2})
For gears made by form cutting $$Y_{Fa2} = \frac{\dfrac{6h_{Fa2}}{m_{mn}}}{\left(\dfrac{s_{Fn2}}{m_{mn}}\right)^2}$$

Table 22.50 *Stress concentration factor (Y_{Sa})*

$$Y_{Sa} = (1.2 + 0.13L_a) q_s^{\left\{\frac{1}{1.21 + 2.3/L_a}\right\}} \quad (\text{for } 1 \le q_s < 8)$$ $$L_a = \frac{s_{Fn}}{h_{Fa}}$$ $$q_s = \frac{s_{Fn}}{2\rho_F}$$
s_{Fn} = tooth root thickness (Tables 22.48 and 22.49)
h_{Fa} = bending moment arm (Tables 22.48 and 22.49)
ρ_F = fillet radius at contact point of 30º tangent (Tables 22.48 and 22.49)

Note: The stress concentration factor (Y_{Sa}) accounts for the stress increasing effect of notch or root fillet. It converts the nominal bending stress into local tooth root stress.

Table 22.51 *Contact ratio factor (Y_ε)*

$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_{v\alpha}} \quad (Y_\varepsilon \ge 0.625)\ (\varepsilon_{v\beta} = 0)$$
$$Y_\varepsilon = 0.25 + \frac{0.75}{\varepsilon_{v\alpha}} - \varepsilon_{v\beta}\left(\frac{0.75}{\varepsilon_{v\alpha}} - 0.375\right) \quad (Y_\varepsilon \ge 0.625)\ (0 < \varepsilon_{v\beta} \le 1)$$
$$Y_\varepsilon = 0.625 \quad (\varepsilon_{v\beta} > 1)$$
$\varepsilon_{v\alpha}$ = transverse contact ratio (Table 22.6 – Eq. 22.34) $\varepsilon_{v\beta}$ = overlap ratio (Table 22.6 – Eq. 22.35)

Note: The contact ratio factor (Y_ε) converts the load application at the tooth tip to the decisive point of load application.

Table 22.52 *Bevel gear factor (Y_K)*

$$Y_K = \left(\frac{1}{2} + \frac{1}{2} \times \frac{l'_{bm}}{b}\right)^2 \frac{b}{l'_{bm}}$$
l'_{bm} = projected length of the middle line of contact (mm) (Table 22.7 – Eq. 22.41)

Note: The bevel gear factor (Y_K) accounts for the difference between bevel gear and cylindrical gear.

Table 22.53 *Load sharing factor (Y_{LS})*

$$Y_{LS} = Z_{LS}^2$$
Z_{LS} = load sharing factor (Table 22.36)

Note: The load sharing factor (Y_{LS}) accounts for load sharing between two or more pairs of teeth.

Table 22.54 *Face load distribution factor for bending stress ($K_{F\beta}$)*

$$K_{F\beta} = \frac{K_{H\beta}}{K_{FO}}$$
$K_{H\beta}$ = face load factor for contact stress (Table 22.28) K_{FO} = lengthwise curvature factor for bending
For straight bevel or zerol gears, $$K_{FO} = 1.0$$
For spiral bevel gears, $$K_{FO} = 0.211\left(\frac{r_{cO}}{R_m}\right)^q + 0.789$$
r_{cO} = cutter radius (mm) R_m = mean cone distance (mm) (Table 22.3 – Eq. 22.6)
$$q = \frac{0.279}{\log_{10}(\sin \beta_m)}$$
β_m = mean spiral angle (Given Data – Tables 22.1 and 22.2)
If calculated value of K_{FO} is greater than 1.15, then make K_{FO}=1.15; whereas if calculated value of K_{FO} is less than 1.0, then make K_{FO}=1.0.

Note: The face load distribution factor for bending stress ($K_{F\beta}$) accounts for the effect of load distribution across the face width on bending stress.

22.8 FACTOR OF SAFETY FOR BENDING STRESS

Table 22.55 *Factor of safety for bending stress (S_F)*

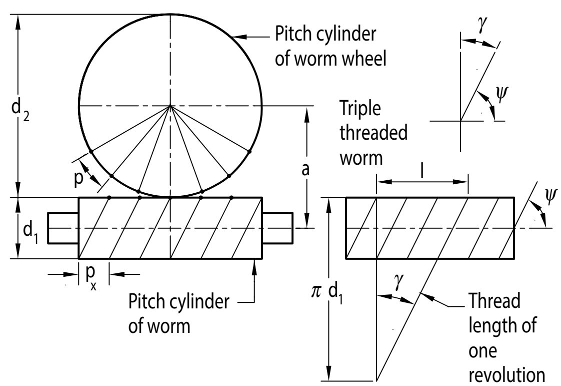
$$S_F = \frac{\sigma_{FP}}{\sigma_F}$$	(22.68)
S_F = factor of safety based on bending stress σ_{FP} = permissible bending stress (MPa or N/mm^2) (Table 22.40) σ_F = working (actual) bending stress (MPa or N/mm^2) (Table 22.47)	

Note: The factor of safety (S_F) is calculated separately for pinion and wheel from the above equation.

23 Worm Gear Design

23.1 BASIC PARAMETERS

Table 23.1 *Basic equations of worm gears*

A pair of worm gears is specified and designated by four quantities in the following manner, $z_1/z_2/q/m$	z_1 = number of starts on worm z_2 = number of teeth on worm wheel q = diametral quotient m = axial module (mm)
$q = \dfrac{d_1}{m}$ (23.1)	d_1 = pitch circle diameter of worm (mm)
	
$l = p_x z_1$ (23.2) $p_x = \pi m$ (23.3) $l = \pi m z_1$ (23.4)	l = lead of worm (mm) p_x = axial pitch of worm (mm)
$d_2 = m z_2$ (23.5)	d_2 = pitch circle diameter of worm wheel (mm)
$\tan\gamma = \dfrac{l}{\pi d_1}$ (23.6) $\tan\gamma = \dfrac{z_1}{q}$ (23.7)	γ= lead angle of worm (°)

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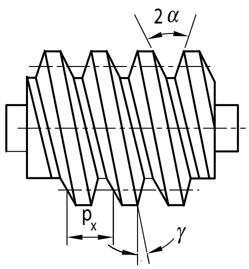
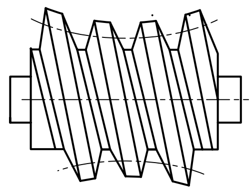
$$\gamma + \psi = \frac{\pi}{2} \quad (23.8)$$	ψ = helix angle of worm (°)
$$a = \frac{1}{2}(d_1 + d_2) \quad (23.9)$$ $$a = \frac{1}{2}m(q + z_2) \quad (23.10)$$	a = centre-to-centre distance between worm and worm wheel (mm)
$$i = \frac{z_2}{z_1} \quad (23.11)$$	i = speed ratio
Single enveloping worm	Double enveloping worm
	

Table 23.2 *Standard centre distances for worm gear transmission*

Centre distance (a) (mm)	80	100	125	160	200	250	315

Table 23.3 *Standard speed ratios for worm gear transmission*

Speed ratio (i)	10/1	12.5/1	16/1	20/1	25/1	31.5/1	40/1	50/1	63/1

Table 23.4 *Recommended number of starts (z_1) on worm*

Speed ratio (i)	10 to 12.5	16 to 25	over 30
Number of starts (z_1)	4	2	1

Table 23.5 *Possible values of (z_2) for worm gear transmission*

Base number	32	36	40	45	50	56	63
Allowable values	31	35	39 40	44 46	48 49 51 52	54 55 57 58	61 62 64 65

Table 23.6 *Standard values of diametral quotient (q) for worm gear transmission*

q	8	10	12.5	16	20	25

Table 23.7 *Standard values of module for worm gear transmission*

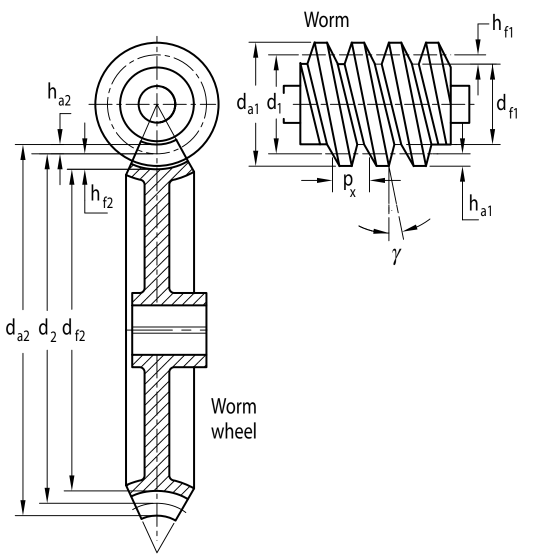
Module (m) (mm)	2.0	2.5	3.15	4.0	5.0	6.3	8.0	10.0	12.5

Table 23.8 *Recommended values of ($z_1/z_2/q/m$) for general-purpose worm gearing*

Transmission ratio (approx.)	Centre distance (mm)						
	80	100	125	160	200	250	315
10/1	3/31/9/4	3/29/7.5/5.5	3/31/7.5/6.5	4/41/8.5/6.5	4/39/8/8.5	4/40/7.5/10.5	5/49/8/11
12.5/1	2/25/7/5	2/24/7/6.5	3/37/8.5/5.5	3/38/8/7	3/37/7.5/9	4/51/8/8.5	4/49/8/11
15/1	2/31/9/4	2/29/7.5/5.5	2/31/7.5/6.5	2/31/7/8.5	3/45/8.5/7.5	3/45/7.5/9.5	3/45/7.5/12
20/1	1/20/7/6	2/41/9/4	2/41/9/5	2/41/8.5/6.5	2/39/8/8.5	2/40/7.5/10.5	2/41/7.5/13
25/1	1/25/7/5	1/24/7/6.5	1/25/6.5/8	1/25/7/10	2/49/8.5/7	2/51/8/8.5	2/49/8/11
30/1	1/31/9/4	1/29/7.5/5.5	1/31/7.5/6.5	1/31/7/8.5	1/30/6.5/11	1/29/6.5/14	2/61/9/9
40/1	1/40/9.5/3.25	1/41/9/4	1/41/9/5	1/41/8.5/6.5	1/39/8/8.5	1/40/7.5/10.5	1/41/7.5/13
50/1	1/51/11/2.6	1/51/11/3.25	1/50/9/4.25	1/49/9/5.5	1/51/8.5/6.75	1/51/8/8.5	1/49/8/11
60/1	1/59/12/2.25	1/61/12/2.75	1/61/11/3.5	1/61/10/4.5	1/60/9.5/5.75	1/60/9/7.25	1/61/9/9

23.2 DIMENSIONS OF WORM AND WORM WHEEL

Table 23.9 *Dimensions of worm and worm wheel*



Dimensions of worm		
$h_{a1} = m$	(23.12)	h_{a1} = addendum of worm (mm)

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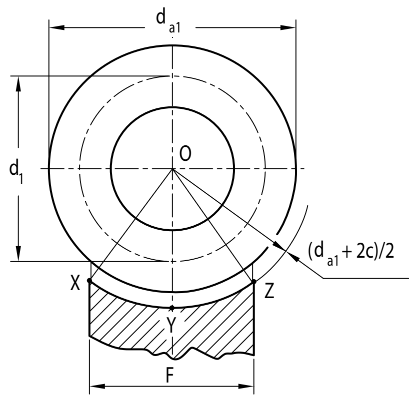
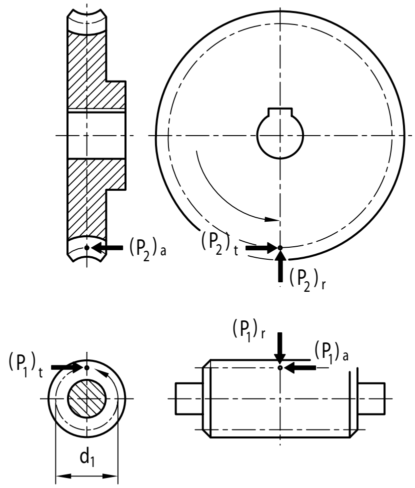
$h_{f1} = (2.2 \cos \gamma - 1)m$	(23.13)	h_{f1} = dedendum of worm (mm)
$c = 0.2\, m \cos \gamma$	(23.14)	c = clearance (mm)
$d_{a1} = m\,(q + 2)$	(23.15)	d_{a1} = outside diameter of the worm (mm)
$d_{f1} = m\,(q + 2 - 4.4 \cos \gamma)$	(23.16)	d_{f1} = root diameter of the worm (mm)
Dimensions of worm wheel		
$h_{a2} = m\,(2 \cos \gamma - 1)$	(23.17)	h_{a2} = addendum at the throat of worm wheel (mm)
$h_{f2} = m\,(1 + 0.2 \cos \gamma)$	(23.18)	h_{f2} = dedendum in the median plane (mm)
$d_{a2} = m\,(z_2 + 4 \cos \gamma - 2)$	(23.19)	d_{a2} = throat diameter of the worm wheel (mm)
$d_{f2} = m\,(z_2 - 2 - 0.4 \cos \gamma)$	(23.20)	d_{f2} = root diameter of the worm wheel (mm)



$F = 2m\sqrt{(q + 1)}$	(23.21)	F = effective face width of worm wheel (mm)
$l_r = (d_{a1} + 2c)\sin^{-1}\left[\dfrac{F}{(d_{a1} + 2c)}\right]$	(23.22)	l_r = length of arc *XYZ* l_r = length of root of worm wheel teeth (mm)

23.3 COMPONENTS OF TOOTH FORCE

Table 23.10 *Worm gears – Components of tooth force*



Assumptions: (i) The worm is the driving element, while the worm wheel is the driven element. (ii) The worm has right-handed threads. (iii) The worm rotates in anti-clockwise direction.

$(P_1)_t = \dfrac{2M_t}{d_1}$	(23.23)	M_t = torque transmitted by gears (N-mm) d_1 = pitch circle diameter of worm (mm) γ = lead angle of worm (°) α = normal pressure angle (°) μ = coefficient of friction $(P_1)_t$ = tangential component on the worm (N) $(P_1)_a$ = axial component on the worm (N) $(P_1)_r$ = radial component on the worm (N)
$(P_1)_a = (P_1)_t \times \dfrac{(\cos\alpha\cos\gamma - \mu\sin\gamma)}{(\cos\alpha\sin\gamma + \mu\cos\gamma)}$	(23.24)	
$(P_1)_r = (P_1)_t \times \dfrac{\sin\alpha}{(\cos\alpha\sin\gamma + \mu\cos\gamma)}$	(23.25)	
$(P_2)_t = (P_1)_a$ $(P_2)_a = (P_1)_t$ $(P_2)_r = (P_1)_r$	(23.26) (23.27) (23.28)	$(P_2)_t$ = tangential component on the worm wheel (N) $(P_2)_a$ = axial component on the worm wheel (N) $(P_2)_r$ = radial component on the worm wheel (N)

23.4 FRICTION IN WORM GEARS

Table 23.11 *Rubbing speed and efficiency*

$v_s = \dfrac{\pi d_1 n_1}{60000\cos\gamma}$	(23.29)	v_s = rubbing velocity (m/s) **Note:** The rubbing speed is the relative velocity between the worm and the wheel.
$\eta = \dfrac{(\cos\alpha - \mu\tan\gamma)}{(\cos\alpha + \mu\cot\gamma)}$	(23.30)	η = efficiency of worm gear drive μ = coefficient of friction (Tables 23.12 and 23.13)

Table 23.12 *Coefficient of friction in worm gears*

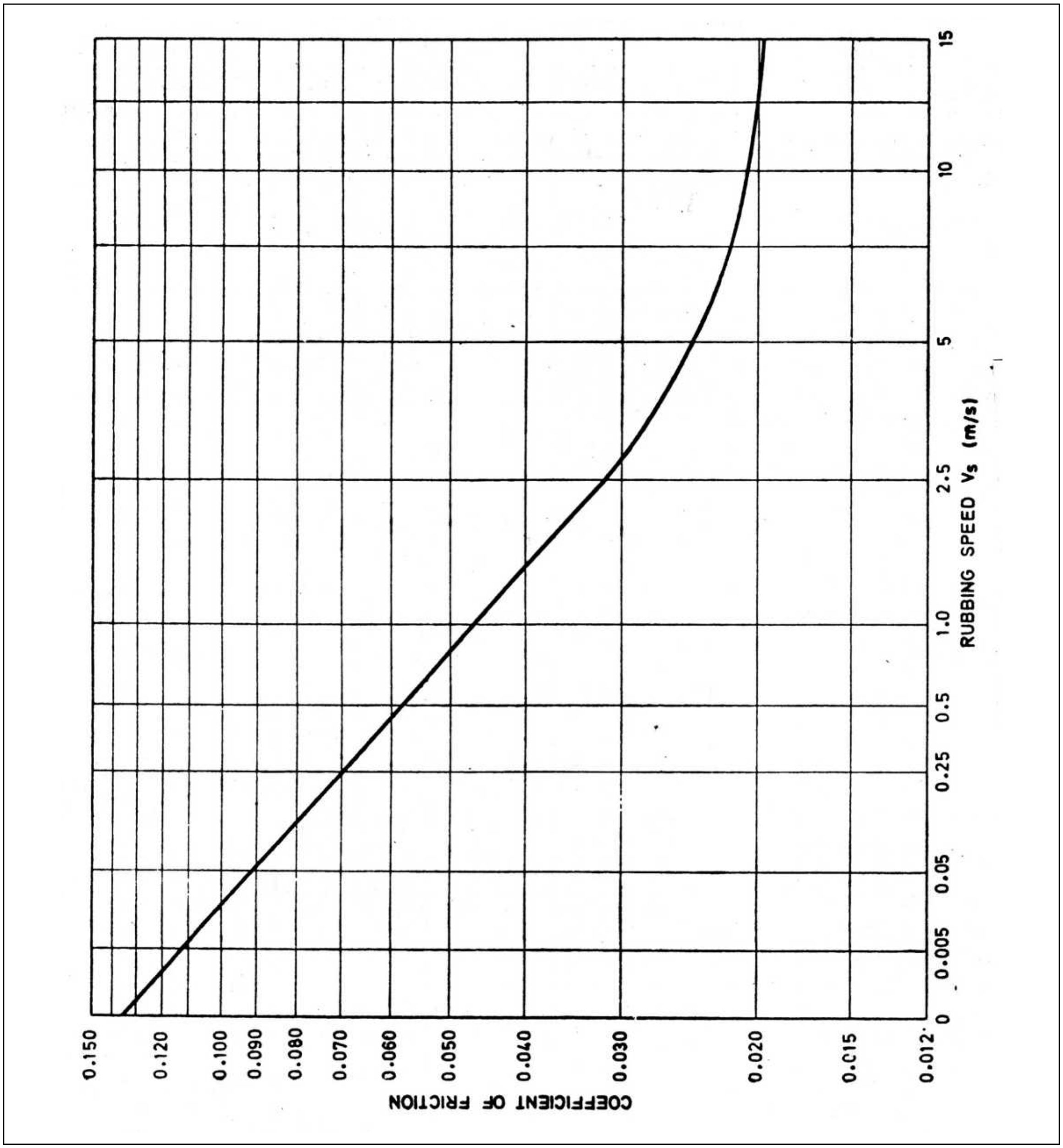


Table 23.13 *Coefficient of friction in worm gears (Alternative approach)*

v_s (m/s)	μ
0.01	0.10 to 0.12
0.1	0.08 to 0.09
0.25	0.065 to 0.075
0.5	0.055 to 0.065
1	0.045 to 0.055
1.5	0.04 to 0.05
2	0.035 to 0.045
2.5	0.03 to 0.04
3	0.028 to 0.035
4	0.023 to 0.03
7	0.018 to 0.026
10	0.016 to 0.024
15	0.014 to 0.020

Note: The values of coefficient of friction in Tables 23.12 and 23.13 are based on the two assumptions: (i) The worm wheel is made of phosphor-bronze, while the worm is made of case-hardened steel. (ii) The gears are lubricated with a mineral oil having a viscosity of 16 to 130 centiStokes at 60°C.

23.5 STRENGTH RATING OF WORM GEARS

Table 23.14 *Strength rating of worm gears*

The worm gears are usually designed according to national and international codes. There are two basic equations: beam strength and wear strength equations. The maximum permissible torque that the worm wheel can withstand without bending failure is given by the lower of the following two values:

$$(M_t)_1 = 17.65 X_{b1} S_{b1} m l_r d_2 \cos\gamma \quad (23.31)$$

$$(M_t)_2 = 17.65 X_{b2} S_{b2} m l_r d_2 \cos\gamma \quad (23.32)$$

$(M_t)_1$, $(M_t)_2$ = permissible torque on the worm wheel (N-mm)
X_{b1}, X_{b2} = speed factors for strength of worm and worm wheel (Table 23.16)
S_{b1}, S_{b2} = bending stress factors of worm and worm wheel (Table 23.15)
m = module (mm)
l_r = length of the root of worm wheel teeth (mm) (Table 23.9)
d_2 = pitch circle diameter of worm wheel (mm)
γ = lead angle of the worm (°)

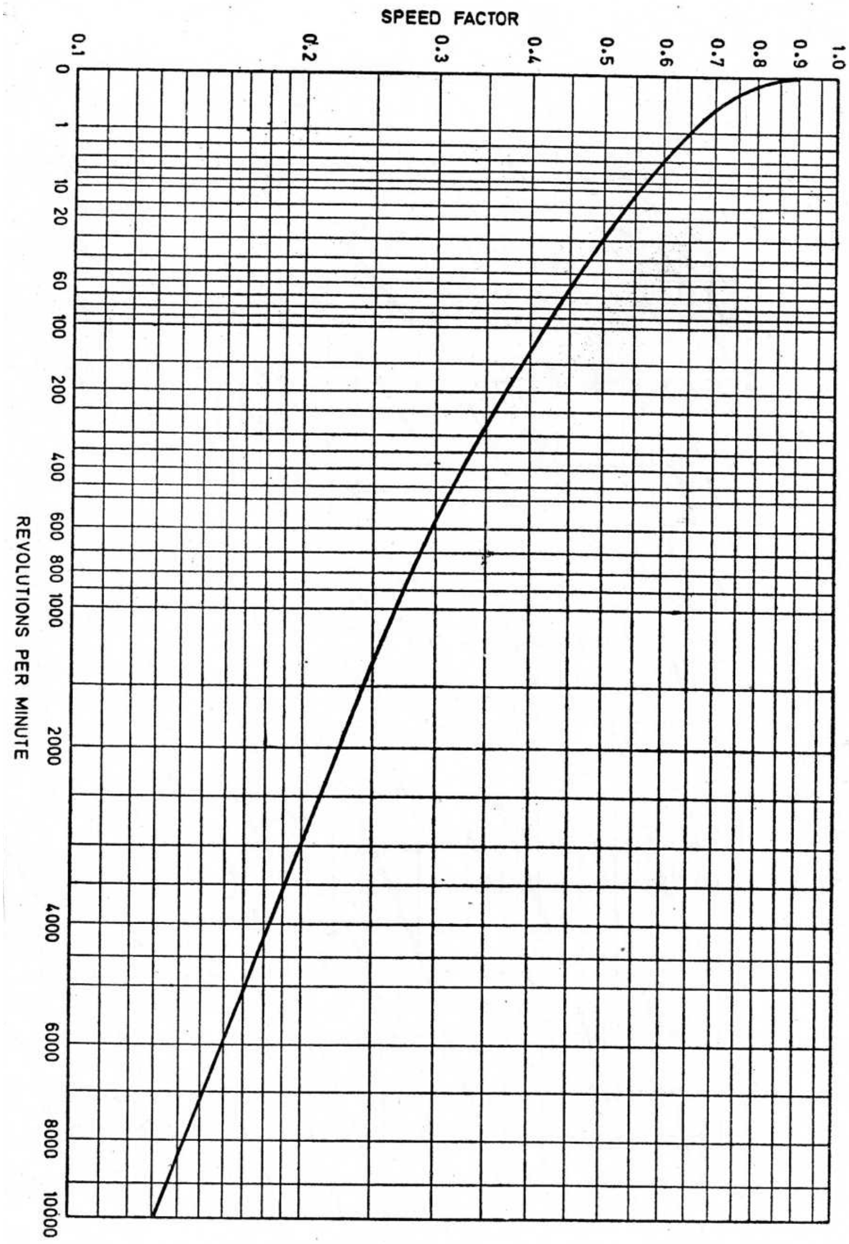
$$\text{kW} = \frac{2\pi n_2 M_t}{60 \times 10^6} \quad (23.33)$$

kW = power transmitting capacity based on beam strength (kW)
(M_t) = lower value between $(M_t)_1$ and $(M_t)_2$ (N-mm)
n_2 = speed of worm wheel (rpm)

Table 23.15 *Values of bending stress factor (S_b)*

Material	S_b
Phosphor-bronze (centrifugally cast)	7.00
Phosphor-bronze (sand-cast and chilled)	6.40
Phosphor-bronze (sand-cast)	5.00
0.4% Carbon steel-normalised (40C8)	14.10
0.55% Carbon steel-normalised (55C8)	17.60
Case-hardened carbon steels (10C4, 14C6)	28.20
Case-hardened alloy steels (16Ni80Cr60 and 20Ni2Mo25)	33.11
Nickel-chromium steels (13Ni3Cr80 and 15Ni4Crl)	35.22

Table 23.16 *Speed factors of worm gears for strength (X_b)*



23.6 WEAR RATING OF WORM GEARS

Table 23.17 *Wear rating of worm gears*

The maximum permissible torque that the worm wheel can withstand without pitting failure, is given by the lower of the following two values:

$$(M_t)_3 = 18.64 X_{c1} S_{c1} Y_z (d_2)^{1.8} m \tag{23.34}$$

$$(M_t)_4 = 18.64 X_{c2} S_{c2} Y_z (d_2)^{1.8} m \tag{23.35}$$

$(M_t)_3, (M_t)_4 =$ permissible torque on the worm wheel (N-mm)

$X_{c1}, X_{c2} =$ speed factors for the wear of worm and worm wheel (Table 23.20)

$S_{c1}, S_{c2} =$ surface stress factors of the worm and worm wheel (Table 23.18)

$Y_z =$ zone factor (Table 23.19)

$$\text{kW} = \frac{2\pi n_2 M_t}{60 \times 10^6} \quad (23.36)$$	kW = power transmitting capacity based on wear strength (kW) (M_t) = lower value between $(M_t)_3$ and $(M_t)_4$ (N-mm) n_2 = speed of worm wheel (rpm)

Table 23.18 *Values of the surface stress factor (S_c)*

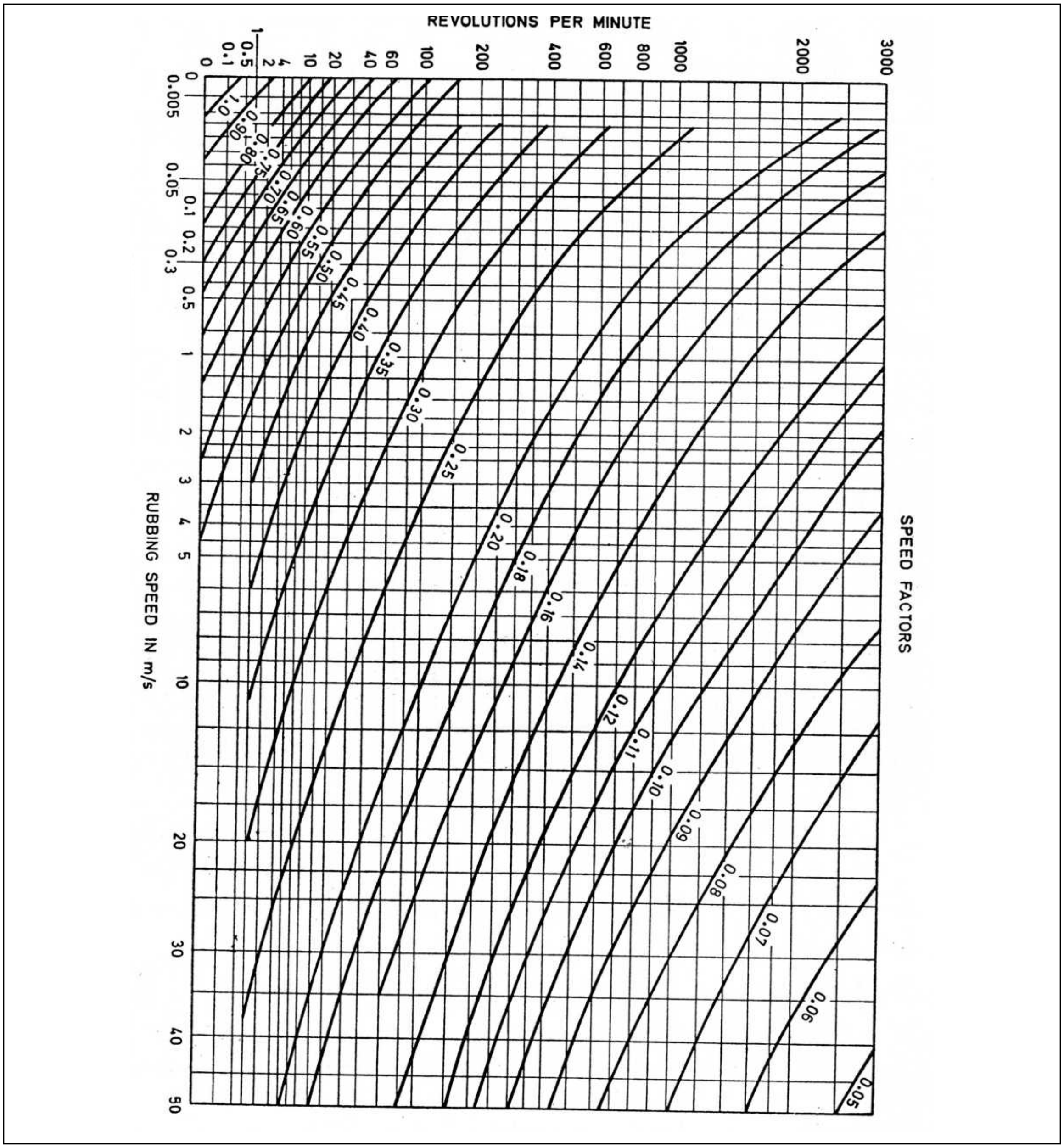
Materials		Values of S_c when running with			
		A	B	C	D
A	Phosphor-bronze (centrifugally cast)	—	0.85	0.92	1.55
	Phosphor-bronze (sand cast and chilled)	—	0.63	0.70	1.27
	Phosphor-bronze (sand-cast)	—	0.47	0.54	1.06
B	0.4% Carbon steel-normalised (40C8)	1.1	—	—	—
C	0.55%Carbon steel-normalised (55C8)	1.55	—	—	—
D	Case- hardened carbon steel (10C4 , 14C6)	4.93	—	—	—
	Case-hardened alloy steel (16Ni80Cr60 , 20Ni2Mo25)	5.41	—	—	—
	Nickel-Chromium steel (13Ni3Cr80, 15Ni4Cr1)	6.19	—	—	—

Table 23.19 *Values of the zone factor (Y_z)*

q	$z_1 = 1$	$z_1 = 2$	$z_1 = 3$	$z_1 = 4$
6	1.045	0.991	0.822	0.826
6.5	1.048	1.028	0.890	0.883
7	1.052	1.055	0.969	0.981
7.5	1.065	1.099	1.109	1.098
8	1.084	1.144	1.209	1.204
8.5	1.107	1.183	1.260	1.301
9	1.128	1.214	1.305	1.380
9.5	1.137	1.223	1.333	1.428
10	1.143	1.231	1.350	1.460
11	1.160	1.250	1.365	1.490
12	1.202	1.280	1.393	1.515
13	1.260	1.320	1.422	1.545
14	1.318	1.360	1.442	1.570
16	1.374	1.418	1.502	1.634
17	1.402	1.447	1.532	1.666
18	1.437	1.490	1.580	1.710
20	1.508	1.575	1.674	1.798

Note: The above values are based on assumption that $[b = 2m\sqrt{(q+1)}]$ and symmetrical about the central plane of the wheel.

Table 23.20 *Speed factors of worm gears for wear (X_c)*



23.7 THERMAL RATING OF WORM GEARS

Table 23.21 *Thermal equations*

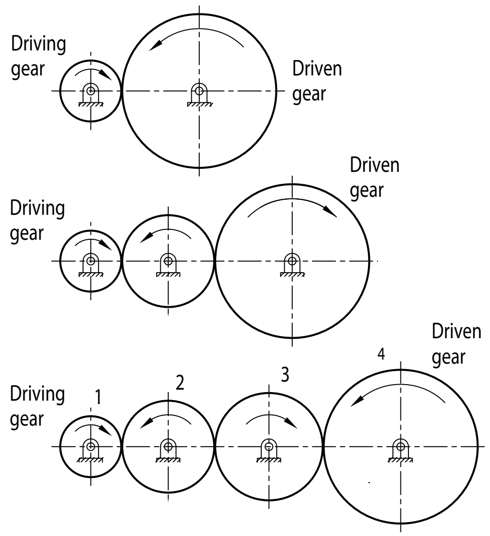
$$\text{kW} = \frac{k(t - t_o)A}{1000(1 - \eta)} \quad (23.37)$$ $$t = t_o + \frac{1000(1 - \eta)\text{kW}}{kA} \quad (23.38)$$	kW = power transmitted by gears on the basis of thermal consideration (kW) k = overall heat transfer coefficient of housing walls (W/m^2 °C) t = temperature of the lubricating oil (°C) t_o = temperature of the surrounding air (°C) A = effective surface area of housing (m^2) η = efficiency of worm gears (fraction)
The overall heat transfer coefficient under normal working conditions with natural air circulation is 12 to 18 W/m^2 °C. This value can be further increased by providing a fan on the worm shaft and arranging the fins horizontally along the stream of air. In such cases, the value of the overall heat transfer coefficient can be taken as 20 to 28 W/m^2 °C. It has been observed, that the maximum permissible temperature for commonly used lubricating oils is 95°C, above which it loses its properties and there is a danger of gear tooth failure due to seizure.	

Construction of Gear Boxes

24

24.1 GEAR TRAINS

Table 24.1 *Types of gear train*

<table>
<tr><th colspan="2">Gear train</th></tr>
<tr><td colspan="2">A gear train consists of two or more gears transmitting power from driving shaft to driven shaft.</td></tr>
<tr><th colspan="2">Simple gear train</th></tr>
<tr><td>
</td><td>A simple gear train is one in which each shaft carries only one gear. In this type of train, the velocity ratio is equal to the number of teeth on last driven gear to the number of teeth on first driving gear.

$$\frac{n_1}{n_4} = \frac{z_4}{z_1} \qquad (24.1)$$

The gears other than driving and driven gears are called idler gears.
The rules regarding direction of rotation are as follows:
(i) If odd number of idler gears is used, the first and last shafts rotate in the same direction.
(ii) If even (or zero) number of idler gears is used, the first and last shaft rotate in the opposite direction.
The main drawback of simple gear train is large overall dimensions and weight.</td></tr>
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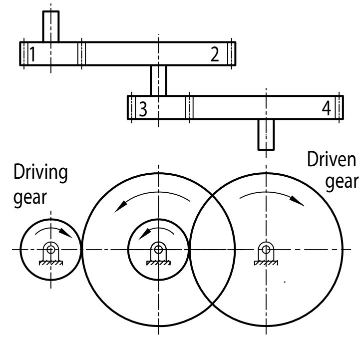
<table>
<tr><th colspan="2">Compound gear train</th></tr>
<tr><td>



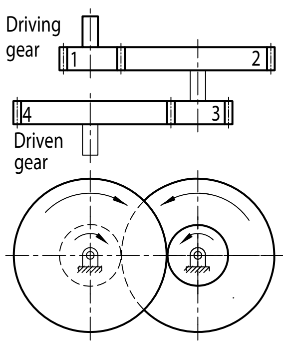
</td><td>

A compound gear train is one in which at least one shaft carries two gears.

$$\frac{n_1}{n_4} = \left(\frac{z_2}{z_1}\right)\left(\frac{z_4}{z_3}\right) \qquad (24.2)$$

Compound gear train is compact in construction compared with simple gear train.

</td></tr>
<tr><th colspan="2">Reverted gear train</th></tr>
<tr><td>



</td><td>

When the number of teeth on various gears in compound gear train are selected in such a way, that the centre distance between gears 1 and 2 is equal to the center distance between gears 3 and 4, then the driving and driven shafts can be located on the same centre line. This type of arrangement is called 'reverted' gear train,

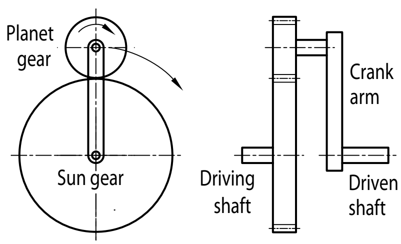
$$m_1(z_1 + z_2) = m_2(z_3 + z_4) \qquad (24.3)$$

where,

m_1 = module for gears 1 and 2
m_2 = module for gears 3 and 4

Reverted gear train is very useful in clocks and instruments where it is desirable to have two pointers on concentric shafts moving with specific velocity ratio.

</td></tr>
<tr><th colspan="2">Epicyclic gear train</th></tr>
<tr><td>



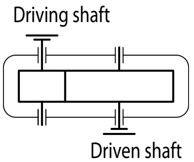
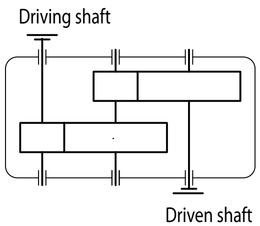
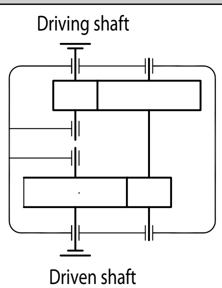
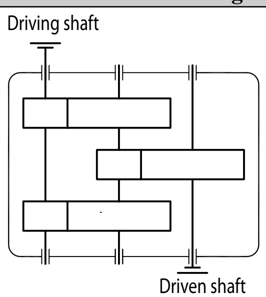
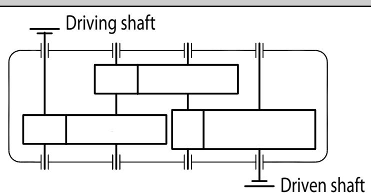
</td><td>

It is a gear train in which one gear is fixed and the meshing gear has a motion composed of two parts namely, a rotation about its own axis and rotation about the axis of the fixed gear. This type of train is also called 'planetary' gear train. The fixed gear is called sun gear while the revolving gear is called planet gear.

Epicyclic gear train has compact construction.

</td></tr>
</table>

Table 24.2 *Gear trains for reducing gears*

<table>
<tr><th colspan="2">Reducing gears</th></tr>
<tr><td colspan="2">Reducing gear box is a device, which is used to reduce the speed and increase the torque.</td></tr>
<tr><th colspan="2">Single-stage reducing gears</th></tr>
<tr><td>
</td><td>Single-stage reducing gears are used for speeds ratios up to 10. In general, the speed ratio is up to 6, in order to keep the size and overall dimensions of gear box as small as possible.</td></tr>
<tr><th colspan="2">Two-stage reducing gears (three-shaft type)</th></tr>
<tr><td>
</td><td>Two-stage reducing gears (three-shaft type) are widely used in about 65% of the industrial applications. The speed ratios range from 8 to 40. The nonsymmetrical arrangement of gears on shafts results in concentration of load along the face width of the teeth. This is the major drawback of this type of construction.</td></tr>
<tr><th colspan="2">Two-stage reducing gears (coaxial shaft type)</th></tr>
<tr><td>
</td><td>This type of arrangement results in compact construction with small overall dimensions.</td></tr>
<tr><th colspan="2">Two-stage reducing gears (double high-speed stage)</th></tr>
<tr><td>
</td><td>In this type of arrangement, there are two pairs of gears on the driving shaft, which transfer equal torque. The uniform loading of two pairs is obtained by using helical gears, one pair with right-hand helix and other with left-hand helix. One of the two shafts of this stage is mounted in bearings which allow for axial self adjustment.</td></tr>
<tr><th colspan="2">Three-stage reducing gears</th></tr>
<tr><td>
</td><td>This type of arrangement is the simplest type of arrangements for three stage reduction unit.</td></tr>
</table>

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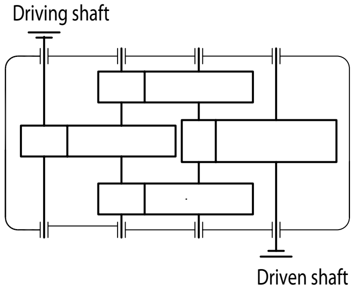
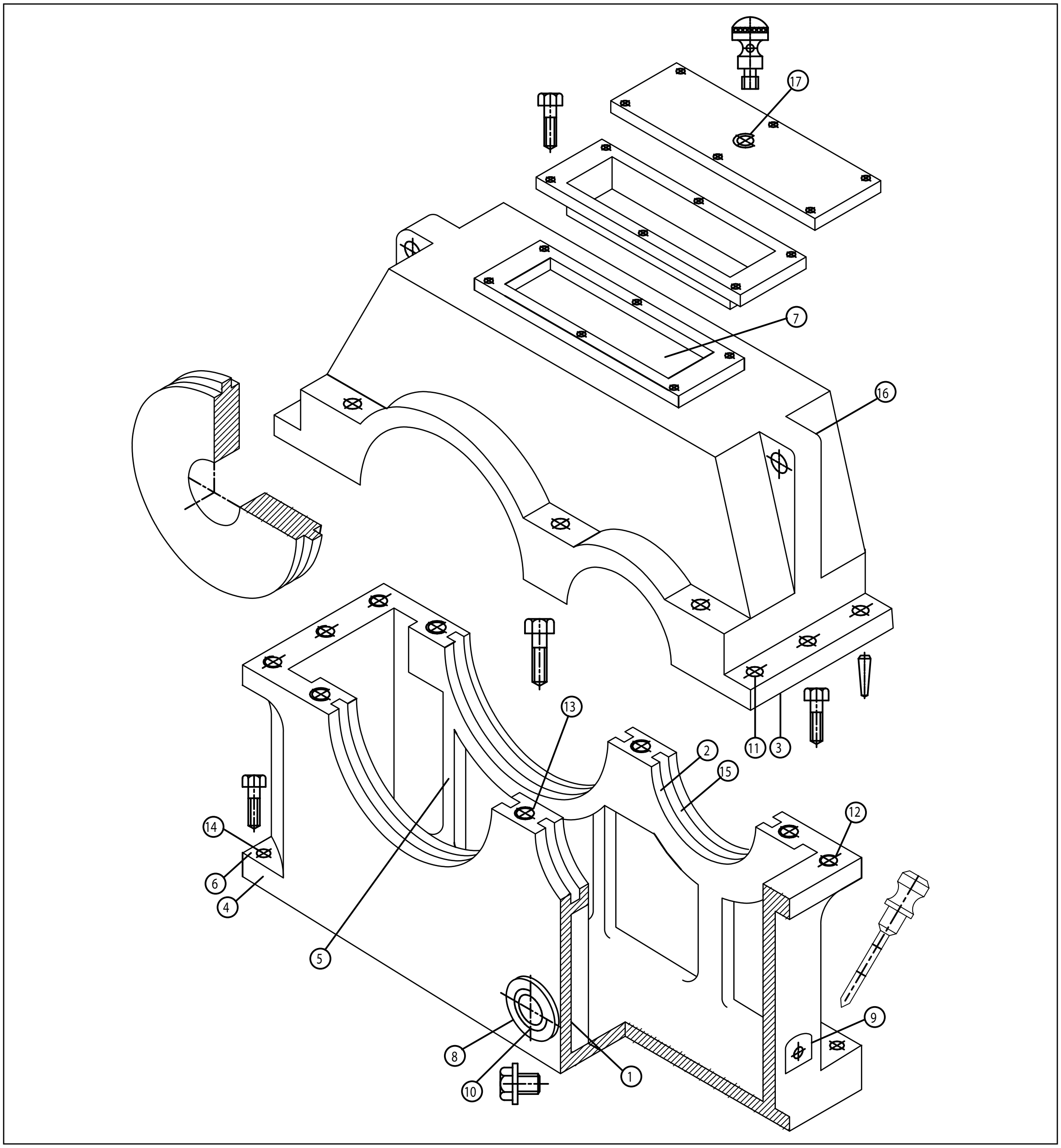
Three-stage reducing gears (double intermediate stage)	
	In this type of arrangement, there are two pairs of gears on the intermediate shafts, which transfer equal torque. This is an improved version of simple three-stage reducing gears. Double intermediate stage provides favorable operating conditions for both high and low speed stages.

24.2 HOUSING FOR GEAR BOX

Table 24.3 *Common features of housing*

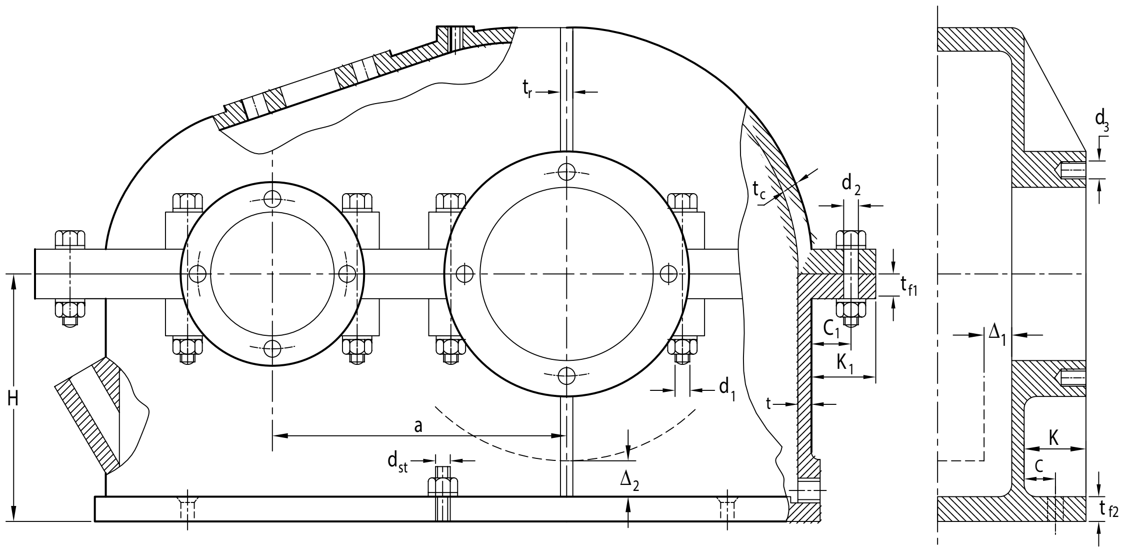
Housing for Gearboxes	
The function of housing is to support the transmission elements like gears, shafts and bearings in correct position and to take up all the forces developed in the speed reducer during its operation.	
Construction of housing	
The housing for the gearboxes is usually split into two parts; lower part or base and upper part or cover that are joined in a plane passing through the axes of shafts. This plane is usually horizontal.	
Common elements of housing	
1.	Wall of housing
2.	Bosses for bearings
3.	Flanges for lower part of housing and cover
4.	Flanges for connecting lower part of housing to base plate or frame
5.	Stiffening ribs
6.	Seat for nuts or screw heads for joining lower part of housing to base plate or frame
7.	Inspection opening
8.	Boss for drain plug
9.	Boss with threaded hole for oil gauge
10.	Threaded hole for drain plug
11.	Holes for puller bolts
12.	Holes for dowel pins
13.	Threaded holes for screws or studs joining lower part of housing to cover
14.	Threaded holes for screws or studs joining lower part of housing to base plate or frame
15.	Grooves for cap collars of bearings
16.	Lifting eyes
17.	Threaded hole for air vent

Table 24.4 *Construction of housing*



Note: Certain lines in the figure are purposefully omitted in order to clarify the construction and avoid confusion.

Table 24.5 *Proportions of housing*



t	Thickness of housing wall	$t = 0.025a + 5$
t_c	Thickness of cover wall	$t_c = 0.02a + 5$
t_{f1}	Thickness of flanges between housing and cover	$t_{f1} = 1.5t$
t_{f2}	Thickness of foundation flanges	$t_{f2} = (2 \text{ to } 2.4)t$
t_r	Thickness of ribs	$t_r = 0.85t$
d_{st}	Diameter of foundation bolts	$d_{st} = 0.036a + 12$
d_1	Diameter of bearing bolts	$d_1 = 0.75d_{st}$
d_2	Diameter of bolts for securing cover and housing	$d_2 = (0.5 \text{ to } 0.6)d_{st}$
d_3	Diameter of bolts for bearing cap	$d_3 = (0.4 \text{ to } 0.5)d_{st}$
C	Distance of foundation bolt axis from housing wall	$C \approx 1.2d_{st} + 5$
C_1	Distance of bolt (d_2) axis from housing wall	$C_1 \approx 1.2d_2 + 5$
K	Width of foundation flange	$K = C + d_{st} + (2 \text{ to } 5)$
K_1	Width of flanges between housing and cover	$K_1 = C_1 + d_2 + (2 \text{ to } 5)$

Note: (i) a = centre distance (mm)
(ii) All dimensions in mm

Table 24.6 *Proportions of housing (Alternative relationships)*

t	Thickness of housing wall	$t = 2\sqrt[4]{0.1T_2} \geq 6$
t_c	Thickness of cover wall	$t_c = 0.9t \geq 6$
t_{f1}	Thickness of flanges between housing and cover	$t_{f1} = t$
t_{f2}	Thickness of foundation flanges	$t_{f2} = 1.5d_{st}$
t_r	Thickness of ribs	$t_r = t$
d_{st}	Diameter of foundation bolts	$d_{st} = \sqrt[3]{2T_2} \geq 12$
d_2	Diameter of bolts for securing cover and housing	$d_2 = \sqrt[3]{T_2} \geq 10$
K	Width of foundation flange	$K = (2.1 \text{ to } 2.5)d_{st}$
K_1	Width of flanges between housing and cover	$K_1 = (2.1 \text{ to } 2.5)d_2$
H	Height of shaft axes from lower surface of housing	$H = (1 \text{ to } 1.12)a$
Δ_1	Axial clearance between gear side and protruding inner elements of housing	$\Delta_1 = 0.8t$
Δ_2	Radial clearance between gear face from bottom of housing	$\Delta_2 = 1.2t$

Note: (i) T_2 = torque on low speed shaft (N-mm)
a = centre distance (mm)
(ii) All dimensions are in mm

24.3 HOUSING ATTACHMENTS

Table 24.7 *Dowel pins*

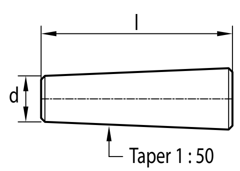
Dowel pins are used for precise alignment of lower part of housing and cover and to prevent their relative movement in assembly. Dowel pins are tapered pins that are machined with tight tolerances, while the corresponding holes in two flanges of housing and cover are reamed. Usually, two pins are placed along the flange diagonal. In mechanical assembly, dowel holes are often used as reference points for geometrical tolerances.				
				
Dimensions of dowel pins (mm)				
d	4	6	8	10
l	16–25	20–40	25–50	30–60
Preferred values of l (mm) – 16, 20, 25, 30, 36, 40, 50, 55, 60				

Table 24.8 *Air vent*

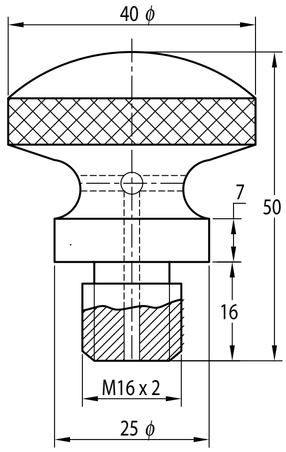
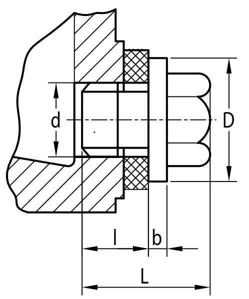
<table>
<tr>
<td>
</td>
<td>During the operation, the pressure inside the housing increases due to heating of oil and air as a result of frictional heat. This causes the lubricating oil to be ejected outside the housing through seals and flanges in the form of oil mist. Air vents are provided in top portion of the cover for communication between the inner space of housing and the surrounding. This keeps the pressure inside the housing under control.</td>
</tr>
</table>

Table 24.9 *Drain plug*

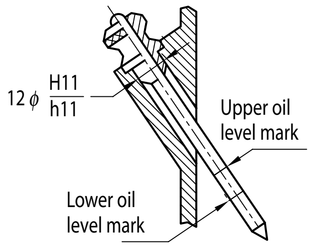
In service, the lubricating oil is gradually contaminated with grit and foreign particles and loses its original lubricating properties. It is to be replaced at regular intervals. Drain plug is provided to drain off waste oil from the housing without dismantling it. The drain plug is provided at the bottom of the housing. The bottom surface of housing is usually given a slight slope towards the drain hole so that the oil can be drained off completely.



Dimensions of drain plug (mm)

d	l	L	b	D
M12 × 1.25	12	22	3	20
M16 × 1.5	13	24	3	25
M20 × 1.5	13	25	3	30
M24 × 1.5	13	28	4	34

Table 24.10 *Dip stick*

	The level of lubricating oil in the housing of speed reducer is checked with dip stick. The dip stick is inserted in housing wall with sliding fit. It has two markings indicating the prescribed upper and lower levels for the lubricating oils.

24.4 SECTIONAL VIEWS OF GEARBOXES

Table 24.11 *Single-stage spur gearbox*

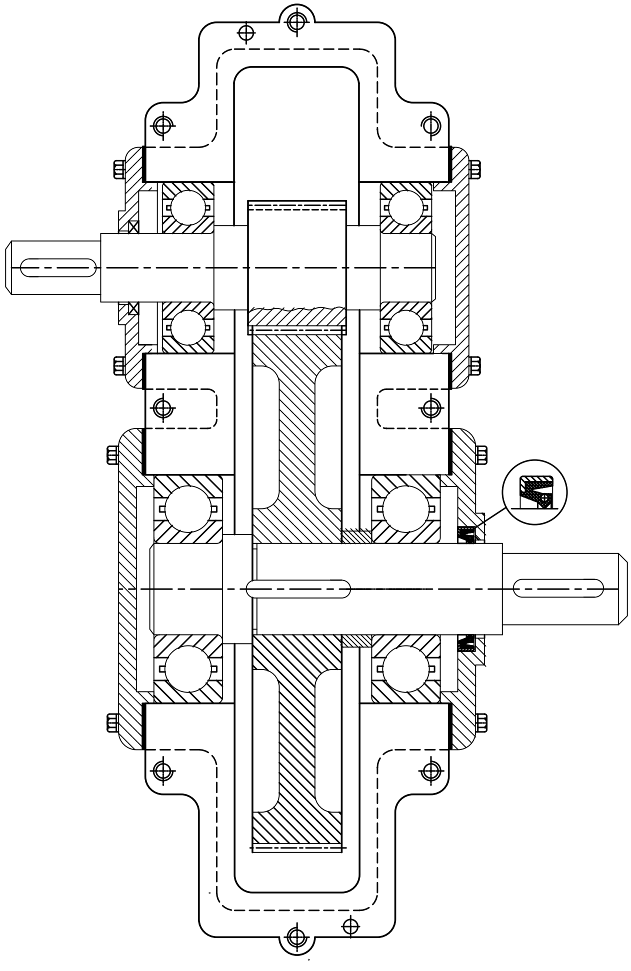


Table 24.12 *Single-stage helical gearbox*

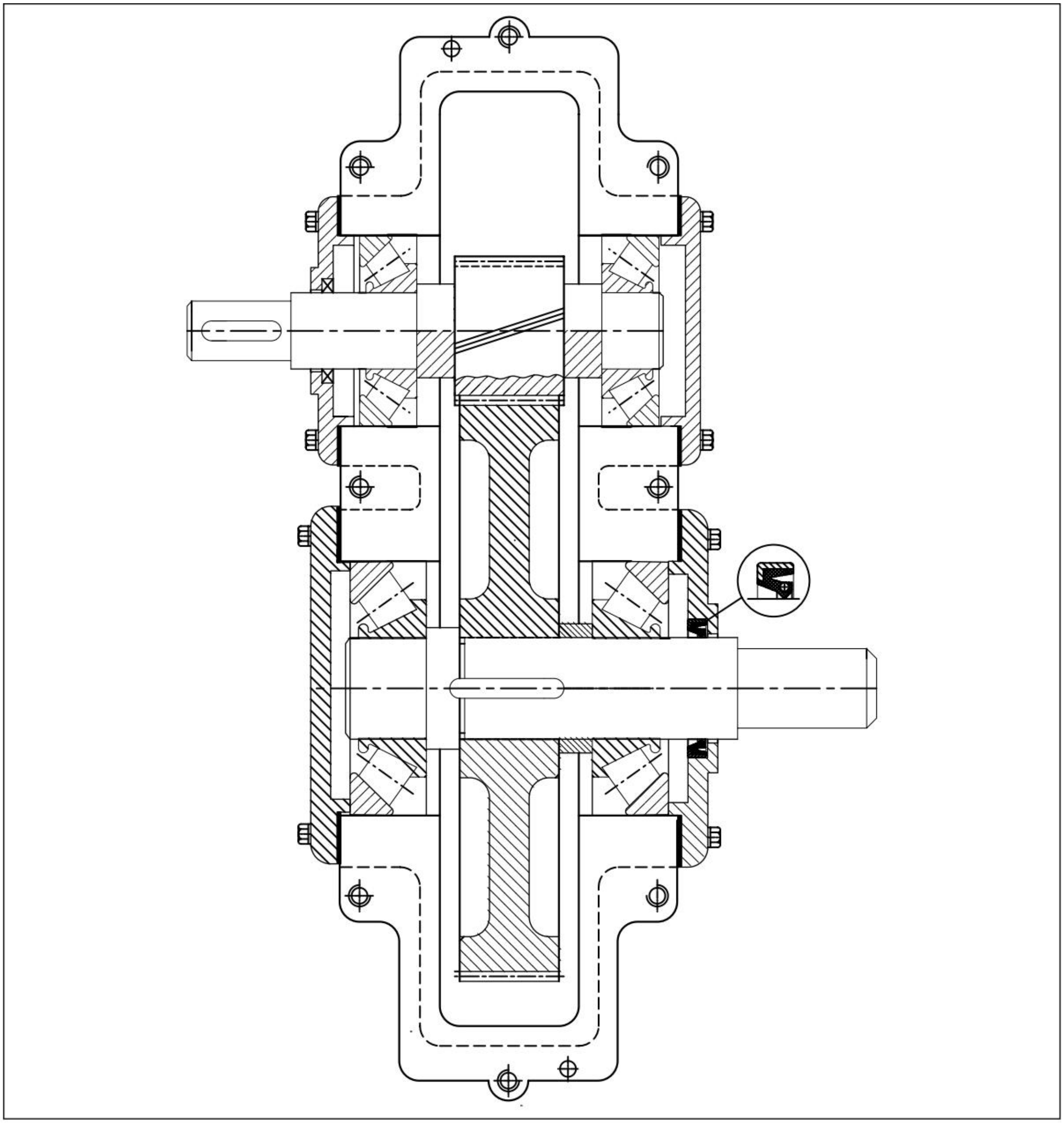


Table 24.13 *Two-stage spur gearbox*

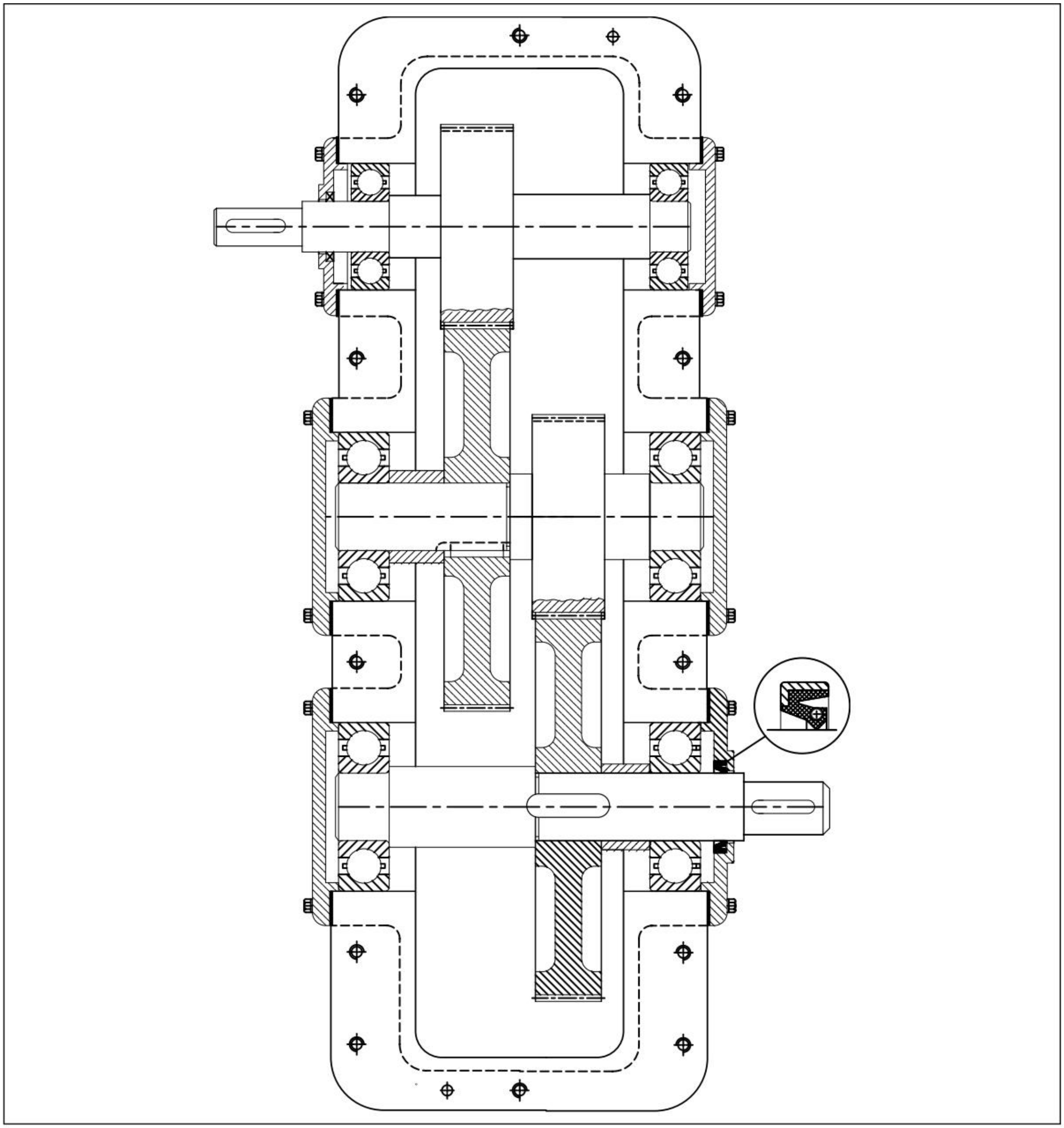


Table 24.14 *Two-stage helical gearbox*

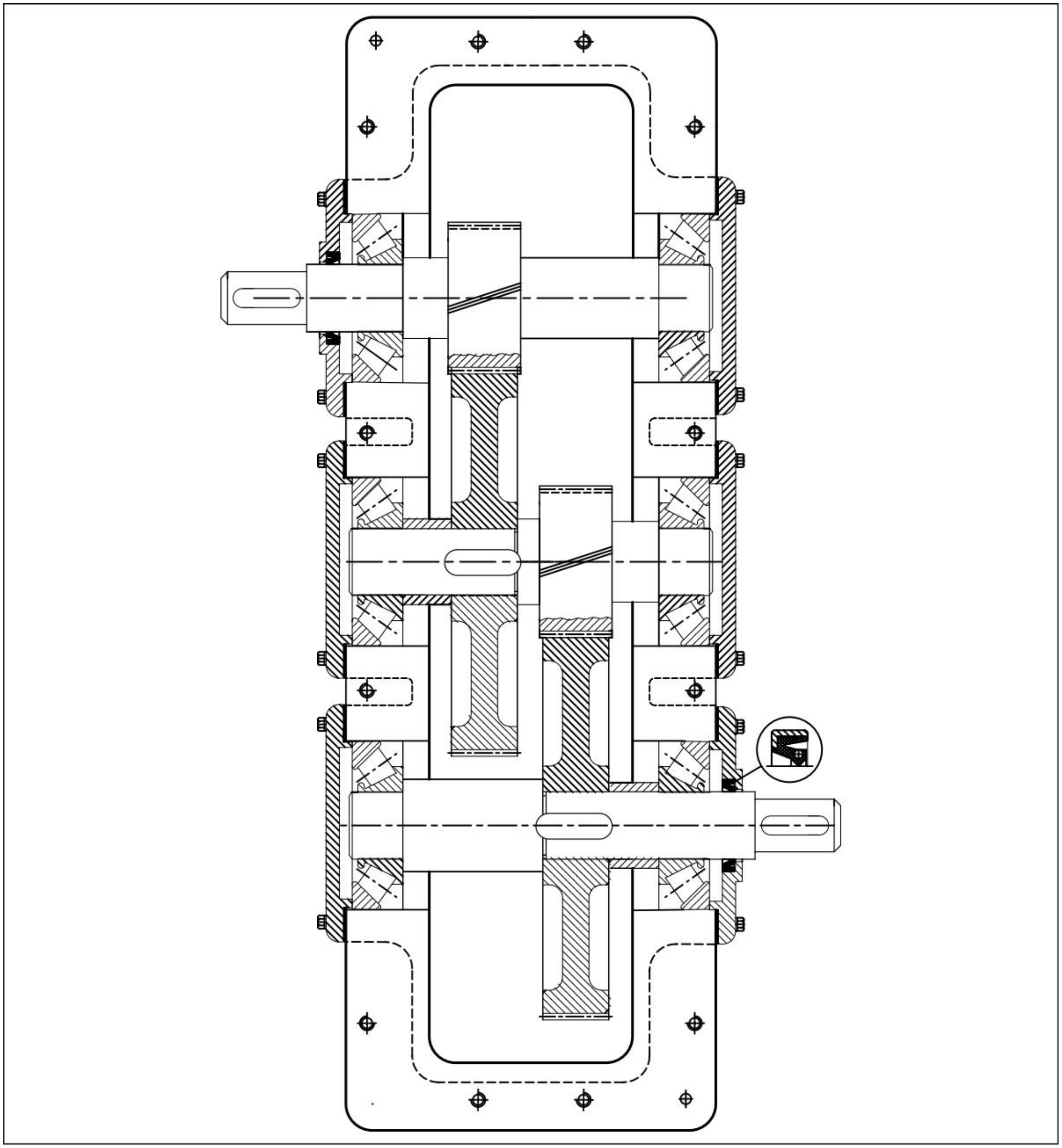


Table 24.15 *Two-stage helical gearbox (Reverted)*

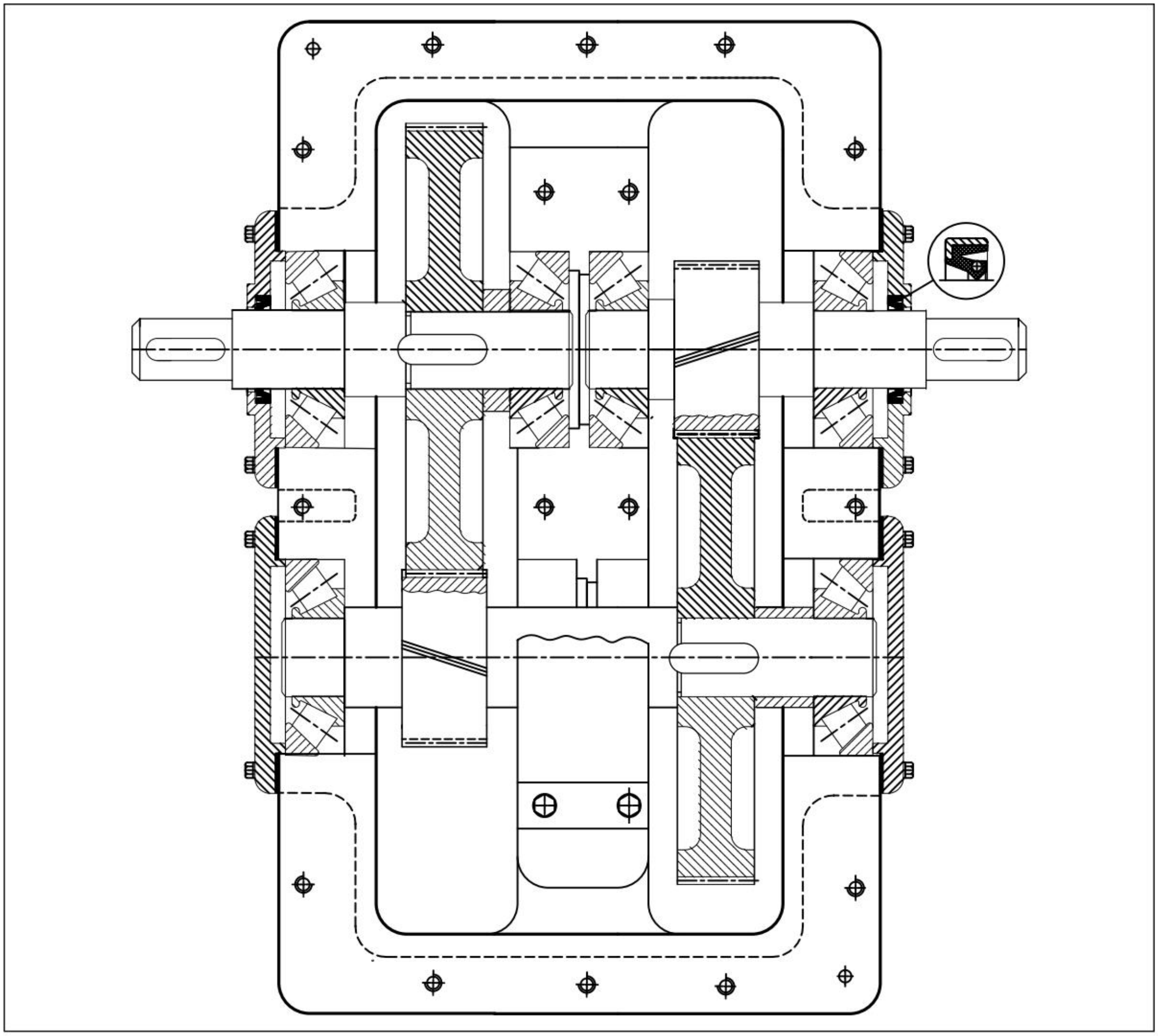


Table 24.16 *Bevel gearbox*

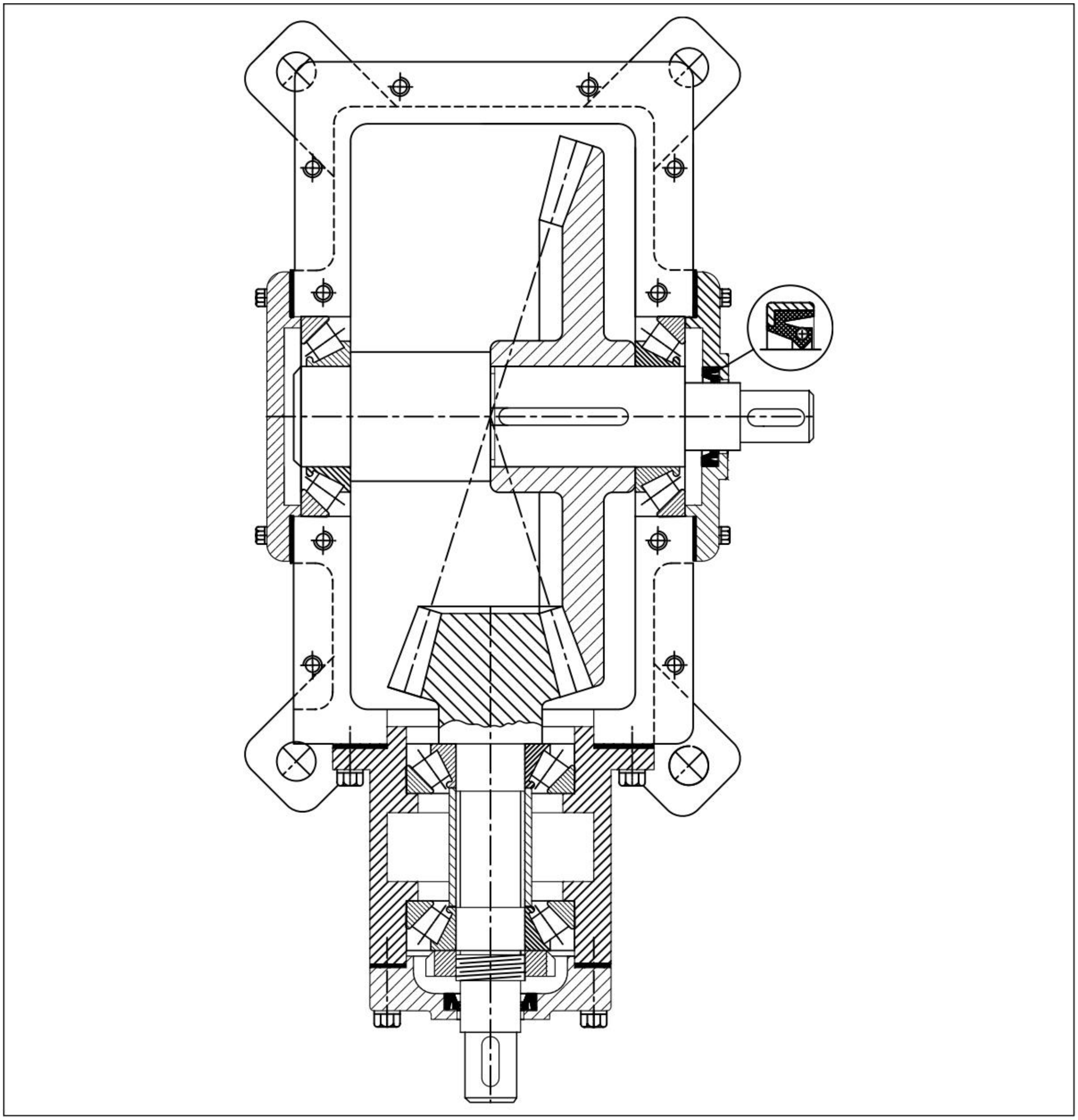


Table 24.17 *Two-stage spur and bevel gearbox*

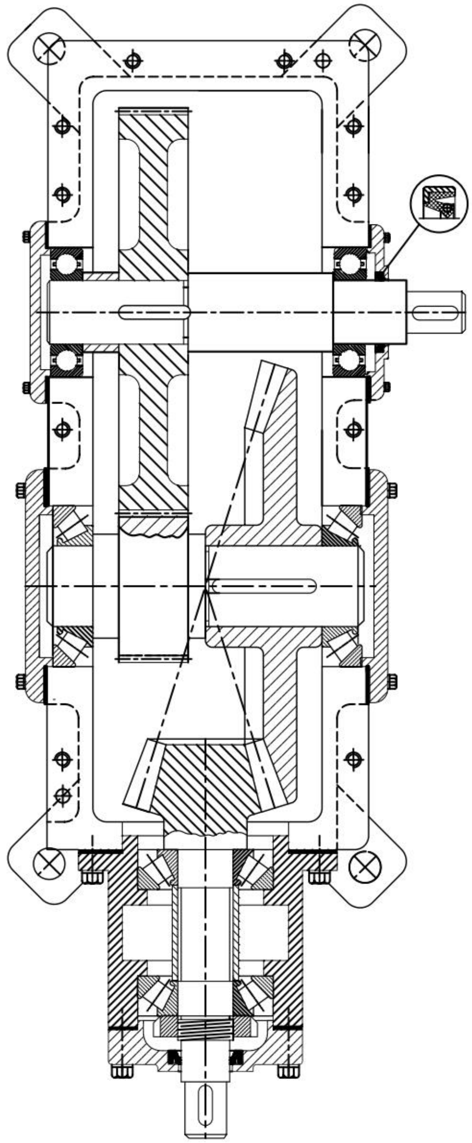


Table 24.18 *Worm gearbox*

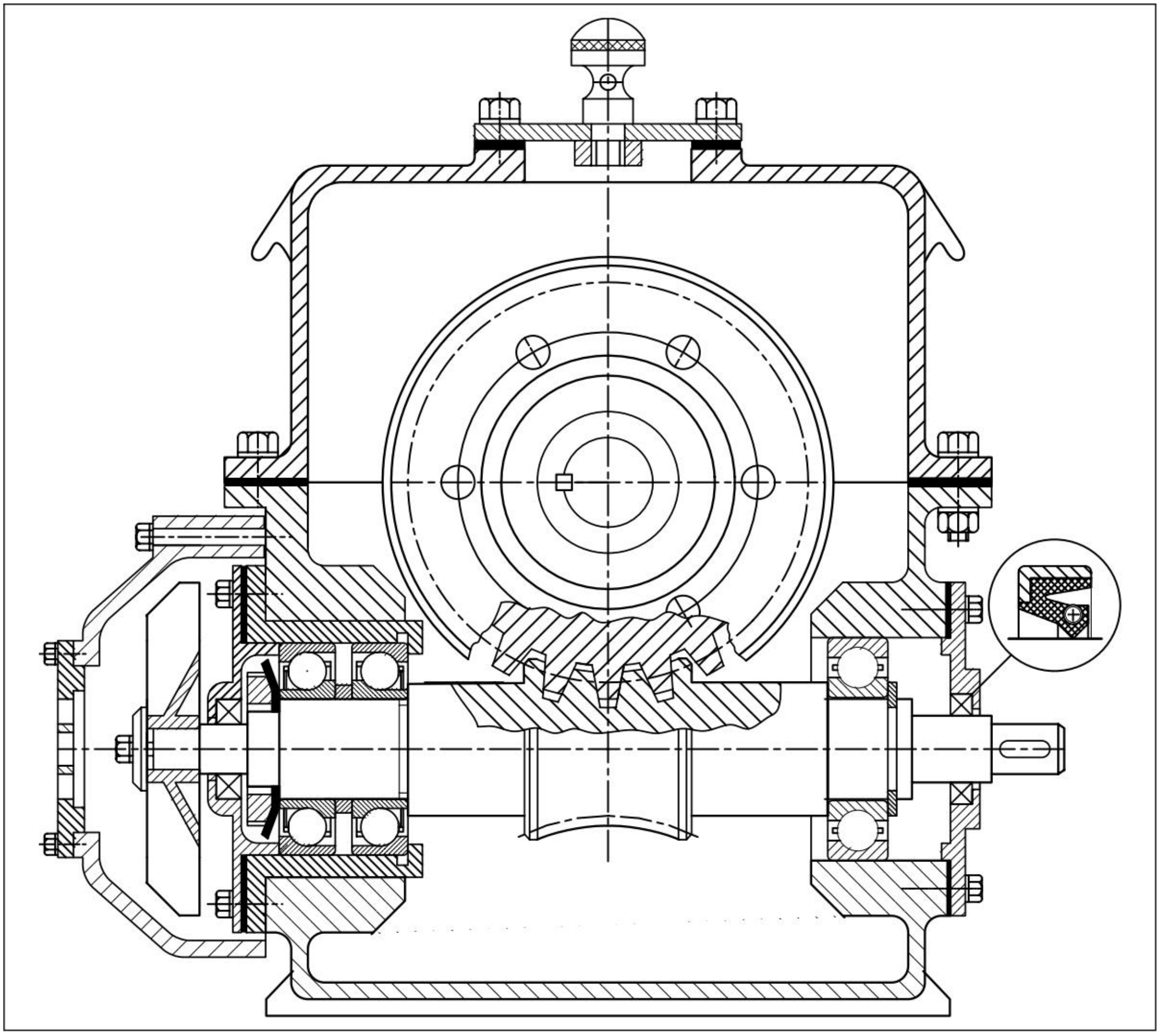


Table 24.19 *Two views of spur gearbox*

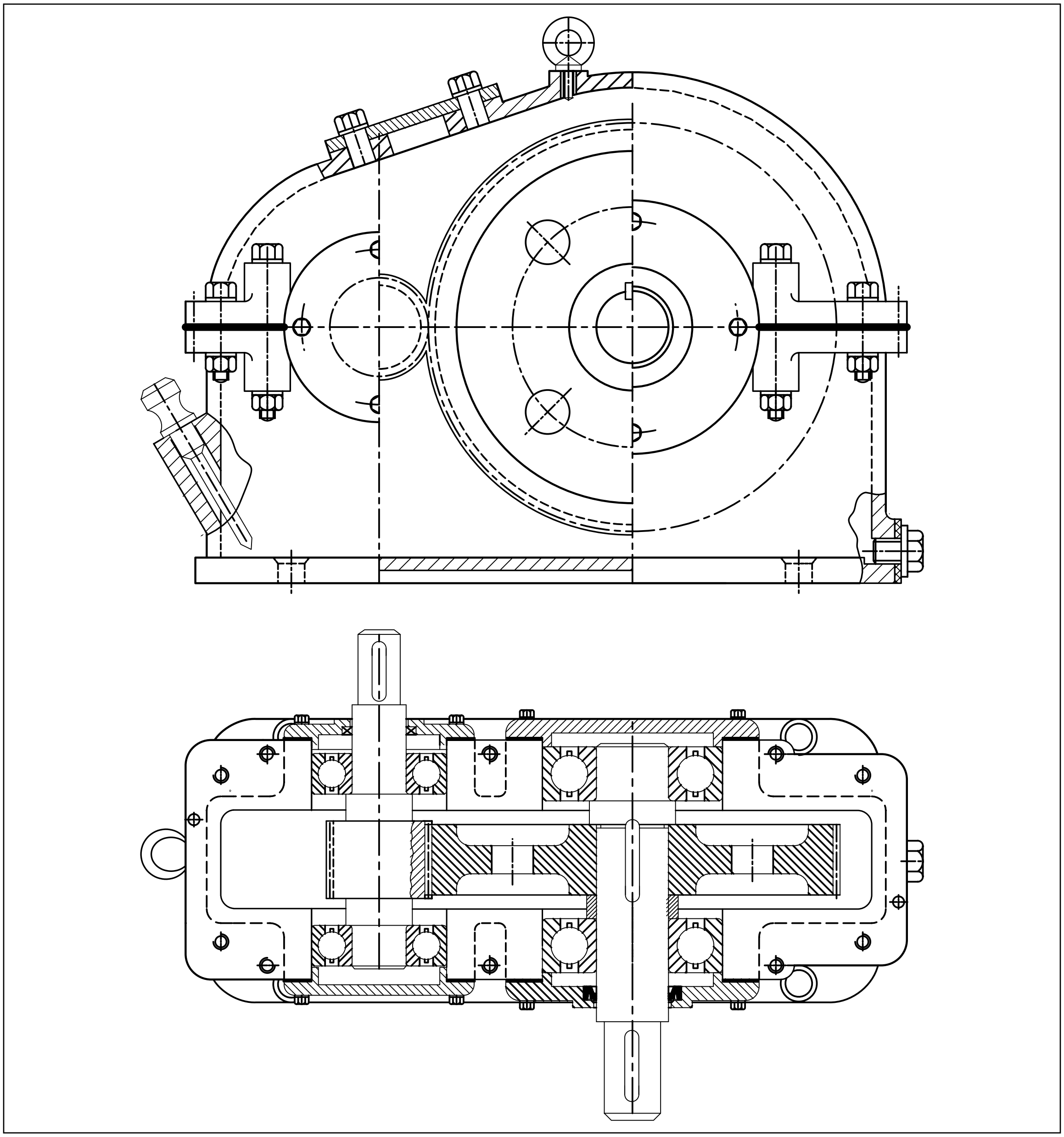
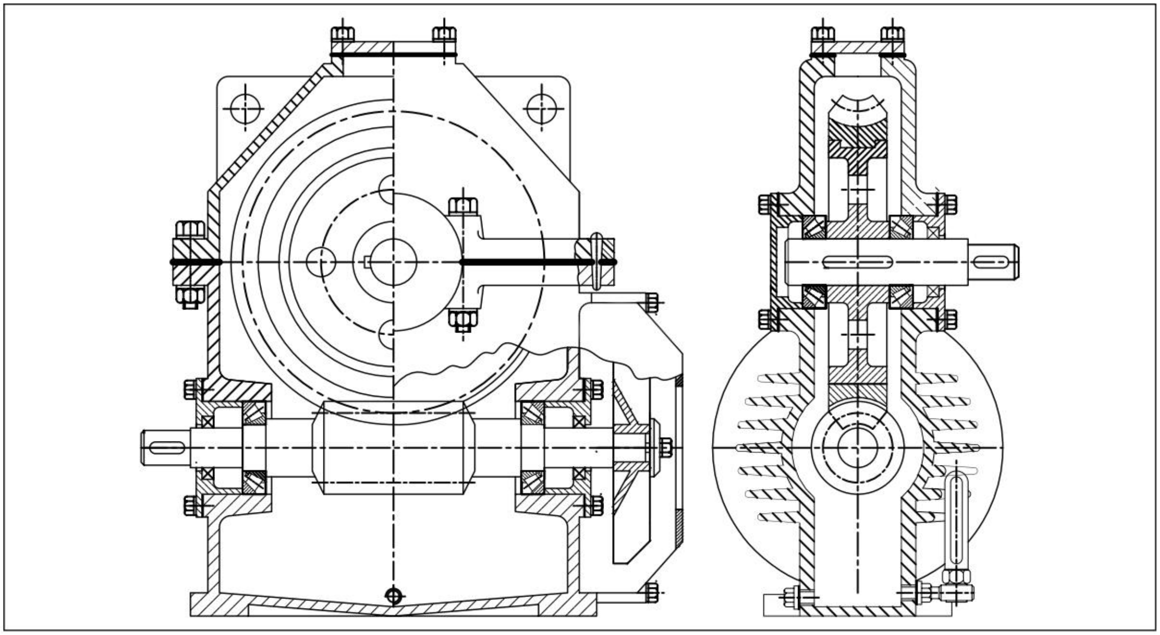
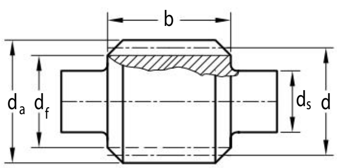
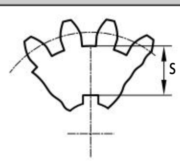


Table 24.20 *Two views of worm gearbox*



24.5 CONSTRUCTIONS OF GEAR BLANK

Table 24.21 *Small-size gears – Integral construction*

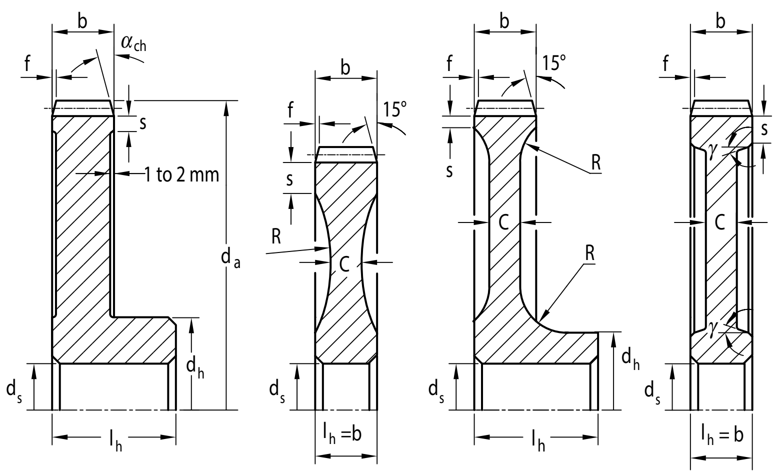
<table>
<tr><th colspan="2">Integral construction</th></tr>
<tr><td colspan="2">A pinion with root diameter near to the required diameter of shaft is made integral with the shaft</td></tr>
<tr><th colspan="2">Rule of thumb - 1</th></tr>
<tr><td>
</td><td>‘If the diameter of dedendum circle (d_f) exceeds the diameter of shaft (d_s), at the point where the pinion is fitted, by less than (d_s/2), the pinion is made integral with the shaft’.
$$d_f - d_s \le (d_s/2)$$</td></tr>
<tr><th colspan="2">Rule of thumb – 2</th></tr>
<tr><td>
</td><td>‘If the distance ‘s’ between the bottom land of tooth and the top of key-slot or spline-periphery is smaller than the safe value (2.5 m), the pinion is made integral with the shaft’.
$$s \le 2.5 \text{ m}$$
where m is the module</td></tr>
</table>

(*Contd.*)

(*Contd.*)

Dimensions of pinion
(i) Pitch circle diameter = $d = mz$ (ii) Addendum circle diameter = $d_a = d + 2h_a = mz + 2(m) = m\,(z + 2)$ (iii) Dedendum circle diameter = $d_f = d - 2h_f = mz - 2(1.25\text{ m}) = m(z - 2.5)$ (iv) Shaft diameter = d_s (v) Width of gear = face width = b
Advantages of integral construction
(i) It reduces the amount of machining since there is no need to cut keyways on the shaft and the pinion. (ii) It reduces the number of parts since there is no key. This reduces the cost. (iii) It increases the rigidity of the shaft and also increases the accuracy of contact.
Disadvantages of integral construction
(i) The shaft has to be fabricated from the same material as that of pinion, which is often of higher quality and costly. (ii) When the pinion is to be replaced because of wear or tooth break down, the shaft has to be discarded as well.

Table 24.22 *Medium-size gears – Disk-type construction*



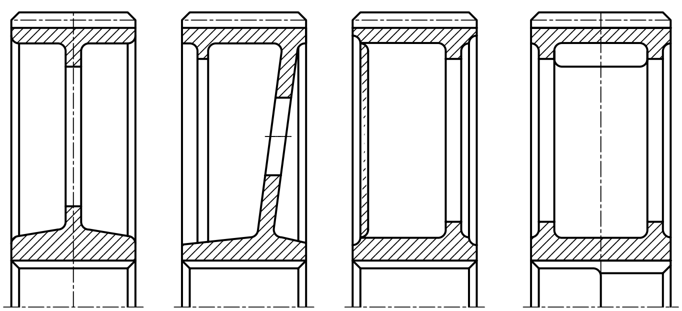
There are two methods to manufacture the medium-size gears. They include machining from bar stock and forging. Gears of addendum circle diameter up to 150 mm are machined on rolled steel bars. The gear blanks in this case are turned on lathe. Gears of addendum circle diameter from 150 mm to 400 mm are mostly forged in open or closed dies.
Advantages of forged gears
(i) The factor of material utilisation is equal to (1/3) when the gear is machined from bar stock. In case of forgings, material utilisation factor is (2/3), which is twice. This reduces the cost of material. (ii) Forged gears have lightweight construction, which reduces inertia and centrifugal forces. (iii) The fibre lines of the forged gears are arranged in a predetermined way to suit the direction of external force. In case of gears, prepared by machining methods, the original fibre lines of rolled stock are broken. Therefore, the forged gear is inherently strong compared with machined gear.

(*Contd.*)

(*Contd.*)

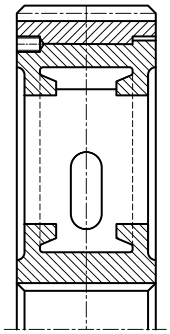
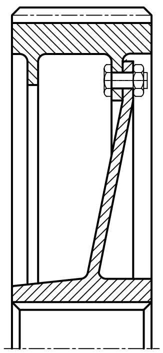
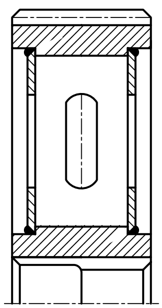
Disadvantages of forged gears				
The limiting factor governing the choice of forged gears is their high cost. The equipment and tooling required to make forged gears is costly. Forged gears become economical only when they are manufactured on large scale.				
Disk with hub				
Gear wheels with addendum circle diameter up to 250 mm are made as disks with or without hubs. Where hubs are used, they can be arranged symmetrically or non- symmetrically. Placing the hub on only one side of the wheel, allows the gear to be secured to the shaft by means of a setscrew. The gear sides are preferably recessed by 1 to 2 mm. In this case, the large surface of gear which serves as datum plane in teeth cutting, will not have to be machined to high degree of accuracy.				
Empirical proportions for medium-size gears				
$d_h = 1.5d_s + 10$ mm	d_h = diameter of hub (mm) d_s = diameter of shaft (mm)			
s = 2.5 m + 2 mm	m = module (mm)			
α_{ch} = 45º (for spur and helical gears with surface hardness <350BHN) α_{ch} = 15º (for helical gears with surface hardness >350BHN)	α_{ch} = chamfer angle for edges of gear disk (º)			
Chamfer for hub and rim f				
d_s	20–30	30–40	40–50	50–80
f	1.0	1.2	1.6	2.0
d_s = shaft diameter (mm)	f = chamfer for hub and rim (mm)			
Web thickness (C)				
Solid gear wheels with web thickness ($C = 0.5b$) are widely used. This type of construction is less susceptible to deformation caused by heat treatment. It also results in less noisy operation at high speeds. Where mass of the wheel is critical, for example, in aircraft, the web thickness is reduced to ($C = 0.25b$), the arc radii are kept to minimum and 4 to 6 holes are drilled in the web to reduce the weight.				

Table 24.23 *Large-size gears – Solid-cast constructions*



Solid cast gears are extensively used due to low manufacturing cost. However, they have low power transmitting capacity compared with forged gears. Gears, up to a diameter of 900 mm, are usually made of one single web. For larger diameters and widths, there are two webs.

Table 24.24 *Large-size gears – Rimmed, bolted and welded constructions*

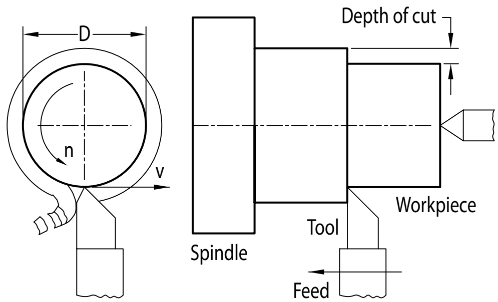
Rimmed gears	
	A rimmed gear consists of a steel rim fitted on the central cast iron hub. The rim is press fitted with interference. The composite construction results in saving of costly high quality steel. However, it is more expensive to manufacture. Wide-face gears, with width more than 500 mm, are made with two rims side by side. Setscrews are used to prevent displacement of the rim with respect to cast iron hub. The thickness of rim from inside diameter to the root circle diameter of teeth should be (7 m to 8 m) for spur gears and (7 m_n to 8 m_n) for helical gears.
Bolted gears	
	A bolted gear consists of a steel rim bolted on the central cast iron hub. In this case, the bolts are fitted in reamed holes and ground holes. Bolted gears have the same power transmitting capacity as rimmed type of gears. However, they are lighter in weight.
Welded gears	
	A welded gear consists of a steel rim welded on the central steel hub. The rim is either made as one piece or it is roll-formed of steel strip and butt welded in the tooth space.

24.6 MULTI-SPEED GEAR BOXES

Table 24.25 *Geometric progression*

Geometric progression	
When a sequence is formed by calculating each new term by multiplying the previous term by a fixed quantity, the sequence is called geometric progression. The fixed amount by which each term is multiplied is called 'step ratio' or 'progression ratio'.	
$a, a\,r, a\,r^2, a\,r^3, a\,r^4, a\,r^5, \ldots$	(24.4)
n^{th} term of the series = $a\,r^{n-1}$	(24.5)
Properties of geometric progression	
(i) If from a geometric progression having progression ratio r, elements are so removed that only every x^{th} member remains, then the resulting series is a geometric progression having progression ratio r^x. Consider a series, $a, a\,r, a\,r^2, a\,r^3, a\,r^4, a\,r^5, a\,r^6 \ldots$ Suppose we eliminate all elements except every second element (x = 2) and form a new sequence, then the resulting series is given by, $a, a\,r^2, a\,r^4, a\,r^6 \ldots$ progression ratio = $\frac{a\,r^2}{a} = \frac{a\,r^4}{a\,r^2} = r^2 = r^x$	
(ii) If a geometric progression having progression ratio r is multiplied by a constant C, the resulting series is a geometric progression having the same progression ratio but whose elements are C times more. Consider a series, $a, a\,r, a\,r^2, a\,r^3, a\,r^4, a\,r^5, a\,r^6 \ldots$ Suppose each element is multiplied by C. The resulting series is given by, $(aC), (aC)\,r, (aC)\,r^2, (aC)\,r^3, (aC)\,r^4, (aC)\,r^5, (aC)\,r^6 \ldots$ Substituting, $aC = a_1$ the above series is written as, $a_1, a_1\,r, a_1\,r^2, a_1\,r^3, a_1\,r^4, a_1\,r^5, a_1\,r^6 \ldots$ It is a geometric progression.	
(iii) If a geometric progression having a progression ratio r is multiplied by a factor r^x, the resulting series is a geometric progression, which is shifted by x elements. Consider a series, $a, a\,r, a\,r^2, a\,r^3, a\,r^4, a\,r^5, a\,r^6 \ldots$ Multiply all elements of above series by r^3 or (x = 3). The resultant series is given by, $a\,r^3, a\,r^4, a\,r^5, a\,r^6, a\,r^7, \ldots$ It is observed that the terms of above series are shifted by three elements compared with the original series.	
Application of geometric progression	
Geometric progression is used to specify various spindle speeds of the machine tools.	

Table 24.26 *Step ratio*



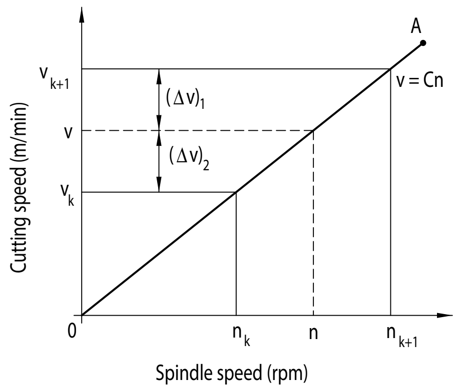
$$v = \frac{\pi D n}{1000} \tag{24.6}$$

v = cutting speed or cutting velocity (m/min)
D = diameter of surface being machined (mm)
n = spindle speed or angular velocity of work piece (rpm)

Range ratio

$$R = \frac{n_{max}}{n_{min}} \tag{24.7}$$

R = range ratio of spindle speed
n_{max} = maximum spindle speed
n_{min} = minimum spindle speed



If optimum spindle speed n is not available and suppose it lies between two spindle speeds n_k and n_{k+1}. The corresponding cutting speeds are denoted as v_k and v_{k+1} respectively.

Step ratio or progression ratio

$$\phi = \frac{n_{k+1}}{n_k} \tag{24.8}$$

ϕ = step ratio or progression ratio of the series

(*Contd.*)

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$$\frac{\Delta v}{v} = \frac{\phi - 1}{\phi + 1}$$	(24.9)
$(\Delta v/v)$ = proportional loss of speed.	
Conclusion: In order to have constant proportional loss of speed in the whole range of speed, the step ratio ϕ must be constant. The whole range means spindle speeds from n_{max} to n_{min}. Since ϕ is constant, the series of spindle speeds must be in geometric progression.	
In machine tool applications, the value of step ratio or geometric progression ratio ϕ varies from 1 to 2. $$1 \le \phi \le 2$$	(24.10)

Table 24.27 *Values of range ratio* $\left(R = \frac{n_{max}}{n_{min}}\right)$ *of spindle speed*

Machine tool	$\left(R = \frac{n_{max}}{n_{min}}\right)$
Centre lathe	40–60
Boring machine	40–60
Milling machine	30–50
Shaping machine	10
Drilling machine	15–30
Planing machine	6–12
Automatic turrets	10–30
Semi-automatic lathe	16–24
Grinding machine	1–13

Table 24.28 *Standard values of* (ϕ) *in machine tools*

ϕ	$\frac{\Delta v}{v} = \left(\frac{\phi - 1}{\phi + 1}\right) \times 100$
1.06	3
1.12	5
1.26	12
1.41	17
1.58	22
1.78	28
2.00	33

Table 24.29 *Recommended values of (ϕ) in machine tools*

Machine tool	ϕ
Heavy duty machine tools and automats	1.12
Large-size and medium-size general purpose machine tools and automats	1.26
Medium-size general purpose machine tools	1.41
Medium-size and small-size general purpose machine tools	1.58

Table 24.30 *Standard spindle speeds for various values of (ϕ)*

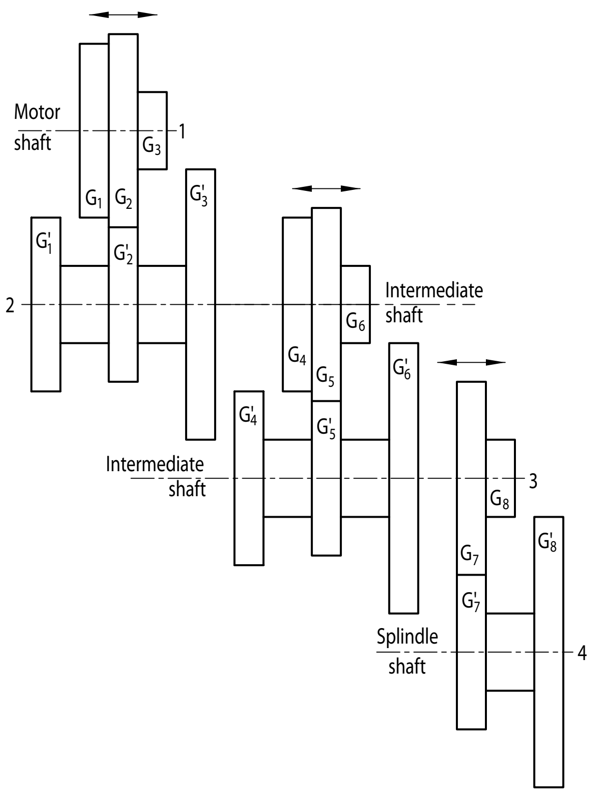
ϕ **= 1.06**	ϕ **= 1.12**	ϕ **= 1.26**	ϕ **= 1.41**	ϕ **= 1.58**	ϕ **= 1.78**	ϕ **= 2**
1.00	1.00	1.00	1.00	1.00	1.00	1.00
1.06						
1.12	1.12					
1.18						
1.25	1.25	1.25				
1.32						
1.40	1.40		1.40			
1.50						
1.60	1.60	1.60		1.60		
1.70						
1.80	1.80				1.80	
1.90						
2.00	2.00	2.00	2.00			2.00
2.12						
2.24	2.24					
2.36						
2.50	2.50	2.50		2.50		
2.65						
2.80	2.80		2.80			
3.00						
3.15	3.15	3.15			3.15	
3.35						
3.55	3.55					
3.75						
4.00	4.00	4.00	4.00	4.00		4.00
4.25						
4.50	4.50					

(*Contd.*)

(*Contd.*)

4.75						
5.00	5.00	5.00				
5.30						
5.60	5.60		5.60		5.60	
6.00						
6.30	6.30	6.30		6.30		
6.70						
7.10	7.10					
7.50						
8.00	8.00	8.00	8.00			8.00
8.50						
9.00	9.00					
9.50						
10.00	10.00	10.00	10.00	10.00	10.00	10.00

Table 24.31 *Eighteen-speed gear box – Kinematic layout*



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$$\text{Total number of spindle speeds} = p_1 \times p_2 \times p_3 \tag{24.11}$$

where,

p_1 = number of simple gear trains on shaft-1

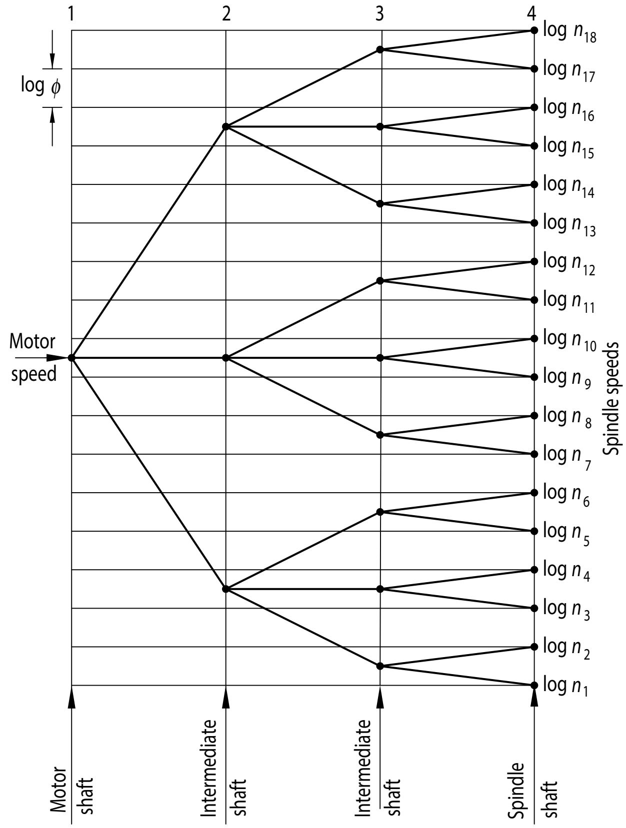
p_2 = number of simple gear trains on shaft-2

p_3 = number of simple gear trains on shaft-3

Table 24.32 *Eighteen-speed gear box – Combination of gear pairs*

Speed number	Gear pair on shafts 1 and 2	Gear pair on shafts 2 and 3	Gear pair on shafts 3 and 4
n_{18}	$G_1 - G'_1$	$G_4 - G'_4$	$G_7 - G'_7$
n_{17}	$G_1 - G'_1$	$G_4 - G'_4$	$G_8 - G'_8$
n_{16}	$G_1 - G'_1$	$G_5 - G'_5$	$G_7 - G'_7$
n_{15}	$G_1 - G'_1$	$G_5 - G'_5$	$G_8 - G'_8$
n_{14}	$G_1 - G'_1$	$G_6 - G'_6$	$G_7 - G'_7$
n_{13}	$G_1 - G'_1$	$G_6 - G'_6$	$G_8 - G'_8$
n_{12}	$G_2 - G'_2$	$G_4 - G'_4$	$G_7 - G'_7$
n_{11}	$G_2 - G'_2$	$G_4 - G'_4$	$G_8 - G'_8$
n_{10}	$G_2 - G'_2$	$G_5 - G'_5$	$G_7 - G'_7$
n_9	$G_2 - G'_2$	$G_5 - G'_5$	$G_8 - G'_8$
n_8	$G_2 - G'_2$	$G_6 - G'_6$	$G_7 - G'_7$
n_7	$G_2 - G'_2$	$G_6 - G'_6$	$G_8 - G'_8$
n_6	$G_3 - G'_3$	$G_4 - G'_4$	$G_7 - G'_7$
n_5	$G_3 - G'_3$	$G_4 - G'_4$	$G_8 - G'_8$
n_4	$G_3 - G'_3$	$G_5 - G'_5$	$G_7 - G'_7$
n_3	$G_3 - G'_3$	$G_5 - G'_5$	$G_8 - G'_8$
n_2	$G_3 - G'_3$	$G_6 - G'_6$	$G_7 - G'_7$
n_1	$G_3 - G'_3$	$G_6 - G'_6$	$G_8 - G'_8$

Table 24.33 *Eighteen-speed gear box –Ray diagram*



Note: Ray diagram or structure diagram is a graphical representation of speeds.

Table 24.34 *Number of speeds and preferred steps*

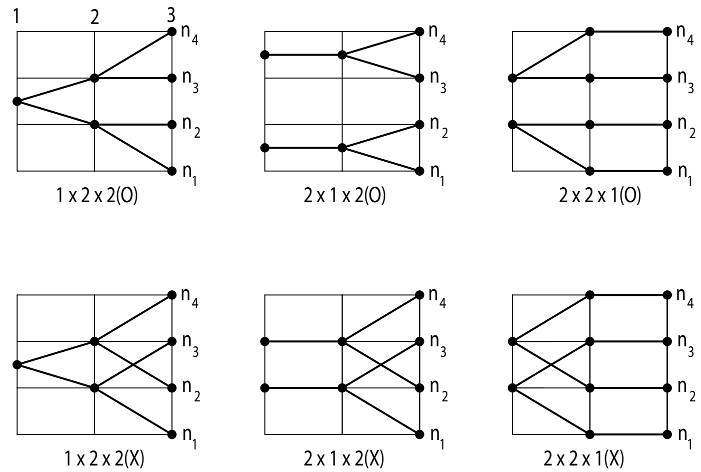
Number of total speeds	Steps of speeds	Minimum number of shafts
2	1 × 2	2
3	1 × 3	2
4	1 × 2 × 2	3
6	1 × 2 × 3 or 1 × 3 × 2	3
8	1 × 2 × 2 × 2	4
9	1 × 3 × 3	3
12	1 × 2 × 2 × 3 or 1 × 2 × 3 × 2	4

(*Contd.*)

(*Contd.*)

16	$1 \times 2 \times 2 \times 2 \times 2$	5
18	$1 \times 2 \times 3 \times 3$	4
24	$1 \times 2 \times 2 \times 2 \times 3$	5
27	$1 \times 3 \times 3 \times 3$	4
32	$1 \times 2 \times 2 \times 2 \times 2 \times 2$	6
36	$1 \times 2 \times 2 \times 3 \times 3$	5

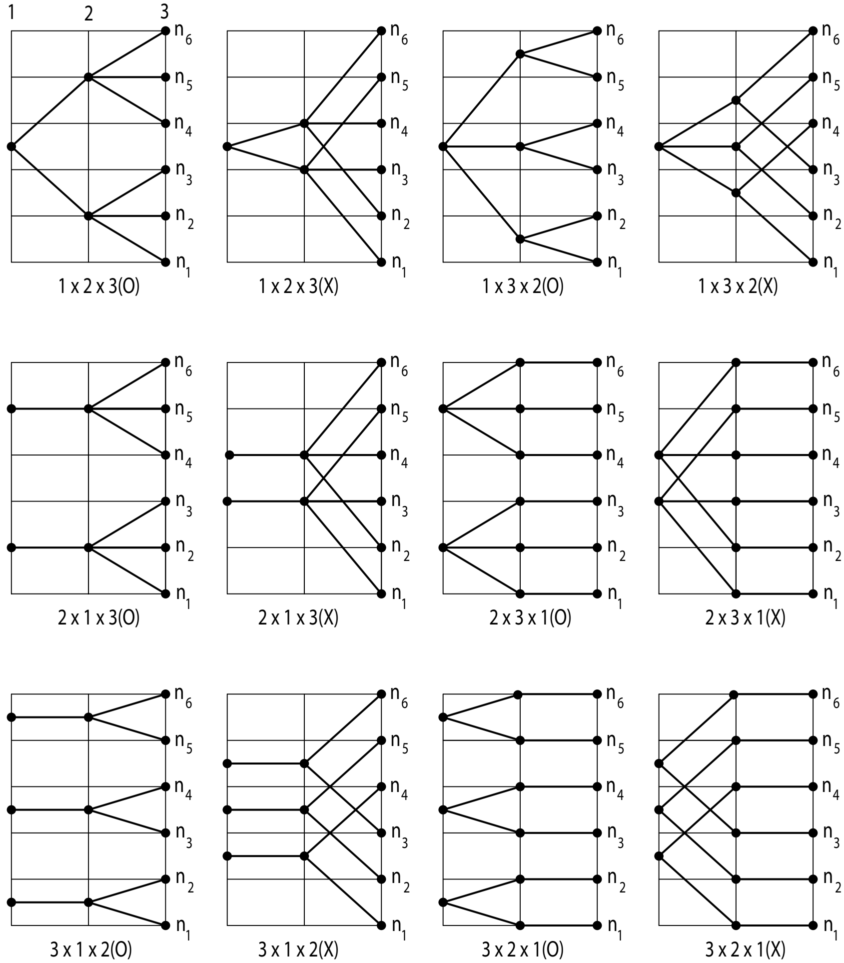
Table 24.35 *Ray diagrams for four-speed two-stage gear box*



Note: There are two types of structure or ray diagram – open and crossed. The types are based on distributive connections between input and output points. When the paths do not cross each other, it is called 'open' type of distribution pattern. If the paths cross each other, it is called 'crossed' type of distribution pattern.

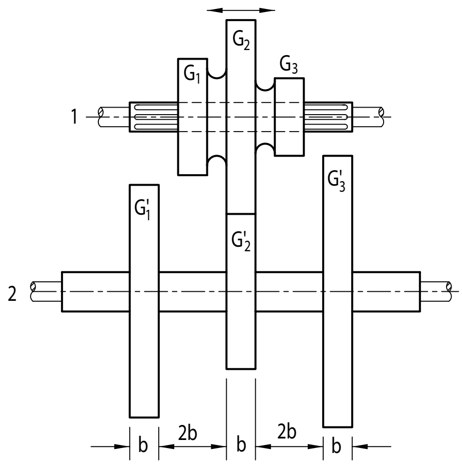
Notations : n_1 = minimum speed
n_4 = maximum speed
1 = input shaft
2 = intermediate shaft
3 = output shaft
O = open-type structure diagram
X = crossed-type structure diagram

Table 24.36 *Ray diagrams for six-speed two-stage gear box*



Notations: n_1 = minimum speed
n_6 = maximum speed
1 = input shaft
2 = intermediate shaft
3 = output shaft
O = open-type structure diagram
X = crossed-type structure diagram

Table 24.37 *Kinematic rules for sliding-mesh gear box*



Rule 1: The minimum number of teeth on the smallest gear in the gear box should be 17 in order to avoid interference (20° pressure angle).

$$z_{\min} \geq 17 \tag{24.11}$$

Rule 2: The sum of the number of teeth of meshing gears in a given stage must be the same, for same module.

$$(z_1 + z_1') = (z_2 + z_2') = (z_3 + z_3') \tag{24.12}$$

where,

z_1 = number of teeth on gear G_1
z_2 = number of teeth on gear G_2
z_1' = number of teeth on gear G_1'
z_2' = number of teeth on gear G_2'

Rule 3: The minimum difference between the numbers of teeth on adjacent gears should be 4.

$$z_2 - z_3 \geq 4 \tag{24.13}$$

Rule 4: The axial gap between adjacent gears on a shaft should be such that one set of gears is completely disengaged before the other set begins to mesh. The axial gap between two adjacent gears must be equal to at least twice the face width (b) of gears.

24.7 PREFERRED NUMBER OF TEETH ON PINION AND WHEEL

Table 24.38 *Preferred number of teeth on pinion and wheel for a given gear ratio* (z_p+z_g) *= 40 to 49*

i	$(z_p + z_g)$									
	40	**41**	**42**	**43**	**44**	**45**	**46**	**47**	**48**	**49**
1	20:20		21:21		22:22		23:23		24:24	
1.06		20:21		21:22		22:23		23:24		23:26
1.12	19:21							22:25		
1.26		18:23		19.24		20:25				
1.41		17:24					19:27			
1.58					17:27					19:30

z_p = number of teeth on pinion
z_g = number of teeth on wheel
i = gear ratio $\left(\dfrac{z_g}{z_p}\right)$

Table 24.39 *Preferred number of teeth on pinion and wheel for a given gear ratio* $(z_p + z_g)$ *= 50 to 59*

i	$(z_p + z_g)$									
	50	**51**	**52**	**53**	**54**	**55**	**56**	**57**	**58**	**59**
1	25:25		26:26		27:27		28:28		29:29	
1.06							27:29		28:30	
1.12		24:27		25:28		26:29		27:30		28:31
1.26	22:28		23:29		24:30		25:31			26:33
1.41		21:30		22:31		23:32			24:34	
1.58		19:32	20:32		21:33			22:35		23:36
1.78	18:32			19:34			20:36		21:37	
2		17:34			18:36			19:38		
2.24						17:38			18:40	

Table 24.40 *Preferred number of teeth on pinion and wheel for a given gear ratio* $(z_p + z_g)$ *= 60 to 69*

i	$(z_p + z_g)$									
	60	**61**	**62**	**63**	**64**	**65**	**66**	**67**	**68**	**69**
1	30:30		31:31		32:32		33:33		34:34	
1.06	29:31		30:32		31:33		32:34		33:35	
1.12			29:33		30:34		31:35		32:36	

(*Contd.*)

(*Contd.*)

1.18		28:33		29:34	29:35		30:36		31:37	
1.26		27:34		28:35		29:36	29:37		30:38	
1.41	25:35			26:37		27:38		28:39	28:40	
1.58	23:37			24:38		25:40		26:41		
1.78		22:39			23:41			24:43		25:44
2	20:40				21:42		22:44			23:46
2.24		19:42	19:43			20:45			21:47	
2.51	17:43			18:45			19:47	19:48		
2.82						17:48			18:50	18:51

Table 24.41 *Preferred number of teeth on pinion and wheel for a given gear ratio* $(z_p + z_g) = 70$ *to* 79

i	$(z_p + z_g)$									
	70	**71**	**72**	**73**	**74**	**75**	**76**	**77**	**78**	**79**
1	35:35		36:36		37:37		38:38		39:39	
1.06	34:36		35:37		36:38		37:39		38:40	
1.12	33:37		34:38		35:39		36:40	36:41	37:41	37:42
1.26	31:39		32:40		33:41	33:42		34:43		35:44
1.41	29:41		30:42	30:43		31:44		32:45		33:46
1.58	27:43		28:44	28:45		29:46		30:42	30:48	
1.78	25:45		26:46			27:48			28:50	
2			24:48			25:50			26:52	
2.24		22:49	22:50		23:51	23:52		24:53	24:54	
2.51	20:50	20:51		21:52	21:53			22:55	22:56	
2.82			19:53	19:54			20:56	20:57		
3.16	17:53	17:54				18:57				19:60
3.55							17:60	17:61		

Table 24.42 *Preferred number of teeth on pinion and wheel for a given gear ratio* $(z_p + z_g) = 80$ *to* 89

i	$(z_p + z_g)$									
	80	**81**	**82**	**83**	**84**	**85**	**86**	**87**	**88**	**89**
1	40:40		41:41		42:42		43:43		44:44	
1.06	39:41		40:42	40:43	41:43	41:44	42:44	42:45	43:45	43:46
1.12	38:42	38:43		39:44		40:45		41:46		42:47

(*Contd.*)

(*Contd.*)

1.26		36:45	36:46	37:46	37:47		38:48		39:49	
1.41	33:47		34:48		35:49	35:50		36:51		37:52
1.58	31:49		32:50	32:51		33:52	33:53		34:54	
1.78	29:51	29:52		30:53	30:54		31:55			32:57
2		27:54			28:56		29:57	29:58		30:59
2.24		25:56			26:58	26:59		27:60	27:61	
2.51	23:57	23:58			24:60	24:61		25:62	25:63	
2.82	21:59	21:60			22:62			23:64	23:65	
3.16	19:61			20:63	20:64			21:66	21:67	
3.55		18:63	18:64	18:65			19:67	19:68		
3.98					17:67	17:68	17:69			18:71

Table 24.43 *Preferred number of teeth on pinion and wheel for a given gear ratio ($z_p + z_g$) = 90 to 99*

i	**($z_p + z_g$)**									
	90	**91**	**92**	**93**	**94**	**95**	**96**	**97**	**98**	**99**
1	45:45		46:46		47:47		48:48	49:48	49:49	50:49
1.06	44:46	44:47	45:47	45:48	46:48	46:49	47:49	47:50		48:51
1.12		43:48	43:49	44:49	44:50	45:50	45:51	46:51	46:52	47:52
1.26	40:50	40:51	41:51	41:52		42:53		43:54		44:55
1.41	37:53	38:53	38:54		39:55		40:56	40:57		41:58
1.58	35:55	35:56		36:57		37:58	37:59		38:50	38:61
1.78		33:58	33:59		34:60	34:61		35:62	35:63	
2	30:60		31:61	31:62		32:63	32:64		33:65	33:66
2.24	28:62	28:63		29:64	29:65			30:67	30:68	
2.51		26:65	26:66		27:67	27:68			28:70	28:71
3.16		22:69	22:70			23:72	23:73			24:75
3.55	20:70	20:71	20:72			21:74	21:75			22:77
3.98	18:72	18:73			19:75	19:76	19:77			20:79

Flywheel and Cams

25

25.1 FLYWHEEL

Table 25.1 *Coefficient of fluctuation of speed (C_s)*

The ratio of maximum fluctuation of speed to the mean speed of flywheel is called the coefficient of fluctuation of speed. $$C_s = \frac{\omega_{max} - \omega_{min}}{\omega} \quad (25.1)$$ $$\omega = \frac{\omega_{max} + \omega_{min}}{2} \quad (25.2)$$
$$C_s = \frac{2(\omega_{max} - \omega_{min})}{(\omega_{max} + \omega_{min})} \quad \text{or} \quad C_s = \frac{2(n_{max} - n_{min})}{(n_{max} + n_{min})} \quad (25.3)$$
C_s = coefficient of fluctuation of speed ω_{max} = maximum angular velocity of flywheel (rad/s) ω_{min} = minimum angular velocity of flywheel (rad/s) ω = mean angular velocity of flywheel (rad/s) n_{max} = maximum speed of flywheel (rpm) n_{min} = minimum speed of flywheel (rpm)
Coefficient of steadiness (m)
The coefficient of steadiness is the reciprocal of the coefficient of fluctuation of speed. $$m = \frac{1}{C_s} = \frac{\omega}{\omega_{max} - \omega_{min}} = \frac{n}{n_{max} - n_{min}} \quad (25.4)$$
m = coefficient of steadiness

Table 25.2 *Coefficient of fluctuation of speed (C_s) for machinery*

Type of equipment	$\mathbf{C_s}$
Crushing machinery	0.200
A-C generators – direct drive	0.0035
D-C generators – direct drive	0.002

(*Contd.*)

(*Contd.*)

Engines with belt transmission	0.030
Flour milling machinery	0.020
Gear drives	0.020
Hammering machinery	0.200
Machine tools	0.030
Paper making machinery	0.025
Reciprocating pumps and Compressors	0.030–0.050
Shearing machinery	0.030–0.050
Spinning machinery	0.010–0.020
Textile machinery	0.025

Table 25.3 *Coefficient of fluctuation of speed (C_s) for engines*

Type of application	**C_s**
Engines driving agricultural equipment	0.050
Engines driving workshop machinery	0.030
Engines driving weaving and spinning machines	0.010 – 0.020
Engines driving D.C. generators	0.006

Table 25.4 *Coefficient of fluctuation of energy (C_e)*

The coefficient of fluctuation of energy is the ratio of the maximum fluctuation of energy to the work done per cycle. $$C_e = \frac{\text{maximum fluctuations of energy}}{\text{work done per cycle}} = \frac{U_0}{\text{work done per cycle}} \quad (25.5)$$
U_o = difference between the maximum kinetic energy and minimum kinetic energy in the cycle (Nm or J)
$$\text{Work done/cycle} = (2\pi)T_m \qquad \text{(for two stroke engine)} \quad (25.6)$$
$$\text{Work done/cycle} = (4\pi)T_m \qquad \text{(for four stroke engine)} \quad (25.7)$$
T_m = mean torque during the cycle (Nm or J)

Table 25.5 *Coefficient of fluctuation of energy (C_e) for engines*

Type of engine	**Number of cylin-ders**	**Angle between cranks**	**C_e**	
			Four-stroke cycle	**Two-stroke cycle**
Single acting	1	360	2.35–2.40	0.95–1.00
Twin	2	360	0.92–1.04	0.75–0.85

(*Contd.*)

(*Contd.*)

Opposed	2	180	0.92–1.04	0.75–0.85
Tandem	2	0	0.92–1.04	0.75–0.85
Twin	2	180	1.50–1.60	0.20–0.25
Opposed	2	360	1.50–1.60	0.20–0.25
Vertical	3	120	0.60–0.75	0.15–0.18
Vertical	4	180 and 90	0.15–0.20	0.075–0.10
Vertical	6	180 and 60	0.10–0.12	0.016–0.02
Double acting	1	—	1.50–1.60	0.20–0.24
Twin or tandem	2	—	0.18–0.20	0.18–0.20
Twin-tandem	4	90	0.08–0.09	0.07–0.08

Table 25.6 *Mass density of flywheel materials*

Material	Mass density (kg/m^3) (ρ)
Grey cast iron	
FG 150	7050
FG 200	7100
FG 220	7150
FG 260	7200
FG 300	7250
Steels	
Carbon steels	7800

Table 25.7 *Moment of inertia of flywheel*

$$I = \frac{U_o}{\omega^2 C_s} \tag{25.8}$$

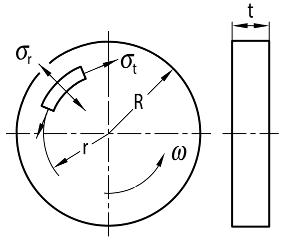
I = mass moment of inertia of flywheel (kg-m^2)
U_o = difference between the maximum kinetic energy and minimum kinetic energy in the cycle (Nm or J)
ω = mean angular velocity of flywheel (rad/s)
C_s = coefficient of fluctuation of speed

$$\omega = \frac{\omega_{\max} + \omega_{\min}}{2}$$

$\omega_{\max}$ = maximum angular velocity of flywheel (rad/s)
$\omega_{\min}$ = minimum angular velocity of flywheel (rad/s)

25.2 SOLID DISK FLYWHEEL

Table 25.8 *Solid disk flywheel—Moment of inertia and stresses*



$$I = \frac{m\,R^2}{2} \tag{25.9}$$

I = mass moment of inertia of disk (kg-m^2)
m = mass of disk (kg)
R = outer radius of disk (m)

$$m = \pi\,R^2\,t\,\rho \tag{25.10}$$

t = thickness of disk (m)
ρ = mass density of flywheel material (kg/m^3)

$$I = \frac{\pi}{2}\rho\,t\,R^4 \tag{25.11}$$

$$\sigma_t = \frac{\rho\,v^2}{10^6}\left(\frac{\mu+3}{8}\right)\left[1-\left(\frac{3\mu+1}{\mu+3}\right)\left(\frac{r}{R}\right)^2\right] \tag{25.12}$$

$$\sigma_r = \frac{\rho\,v^2}{10^6}\left(\frac{\mu+3}{8}\right)\left[1-\left(\frac{r}{R}\right)^2\right] \tag{25.13}$$

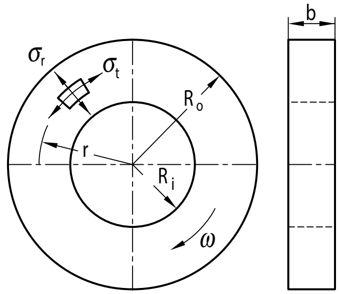
σ_t = tangential stress at radius r (N/mm^2)
σ_r = radial stress at radius r (N/mm^2)
μ = Poisson's ratio (0.3 for steel and 0.27 for cast iron)
v = peripheral velocity (m/s) ($R\omega$)

$$(\sigma_t)_{\max} = (\sigma_r)_{\max} = \frac{\rho\,v^2}{10^6}\left(\frac{\mu+3}{8}\right) \tag{25.14}$$

The maximum tangential stress and maximum radial stress are equal and both occur at ($r = 0$).

25.3 CIRCULAR PLATE WITH CENTRAL HOLE

Table 25.9 *Circular plate with central hole—Moment of inertia and stresses*



$$I = \frac{1}{2} m_o R_o^2 - \frac{1}{2} m_i R_i^2 \qquad (25.15)$$

I = mass moment of inertia of disk (kg-m^2)
m_o = mass of disk with radius R_o (kg)
m_i = mass of disk with radius R_i (kg)
R_o = outer radius of disk (m)
R_i = inner radius of disk (m)

$$m_o = \pi R_O^2\, t\, \rho \qquad (25.16)$$

$$m_i = \pi R_i^2\, t\, \rho \qquad (25.17)$$

$$m = m_o - m_i \qquad (25.18)$$

m = mass of disk (kg)
t = thickness of disk (m)
ρ = mass density of flywheel material (kg/m^3)

$$I = \frac{\pi}{2} \rho\, t\, (R_o^4 - R_i^4) \qquad (25.19)$$

$$\sigma_t = \frac{\rho v^2}{10^6}\left(\frac{\mu+3}{8}\right)\left[1 + \left(\frac{R_i}{R_o}\right)^2 - \left(\frac{3\mu+1}{\mu+3}\right)\left(\frac{r}{R_o}\right)^2 + \left(\frac{R_i}{r}\right)^2\right] \qquad (25.20)$$

$$\sigma_r = \frac{\rho v^2}{10^6}\left(\frac{\mu+3}{8}\right)\left[1 + \left(\frac{R_i}{R_o}\right)^2 - \left(\frac{r}{R_o}\right)^2 - \left(\frac{R_i}{r}\right)^2\right] \qquad (25.21)$$

σ_t = tangential stress at radius r (N/mm^2)
σ_r = radial stress at radius r (N/mm^2)
μ = Poisson's ratio (0.3 for steel and 0.27 for cast iron)
v = peripheral velocity (m/s) ($v = R_o\, \omega$)

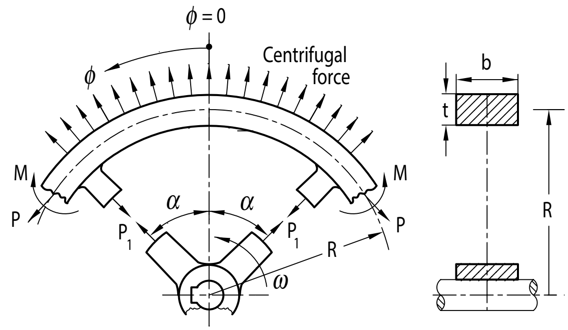
(*Contd.*)

(*Contd.*)

The maximum tangential stress occurs at ($r = R_i$) and given by,
$$(\sigma_t)_{max} = \frac{\rho v^2}{10^6}\left(\frac{\mu+3}{4}\right)\left[1+\frac{1-\mu}{\mu+3}\left(\frac{R_i}{R_o}\right)^2\right] \quad (25.22)$$
The maximum radial stress occur at ($r = \sqrt{R_i R_o}$) and given by,
$$(\sigma_r)_{max} = \frac{\rho v^2}{10^6}\left(\frac{\mu+3}{8}\right)\left(1-\frac{R_i}{R_o}\right)^2 \quad (25.23)$$

25.4 RIMMED FLYWHEEL

Table 25.10 *Rimmed flywheel—Moment of inertia and stresses*



Mass of flywheel rim
$I_r = K\,I \quad (25.24)$
I_r = mass moment of inertia of rim (kg-m^2) I = required moment of inertia of complete flywheel (kg-m^2) K = factor to account for the effect of spokes, hub and shaft
The factor K is equal to 1 when it is assumed that the entire moment of inertia is due to the rim alone. The factor K is taken as 0.9, when it is assumed that the rim contributes 90% of the required moment of inertia.
$$I_r = \frac{U_o K}{\omega^2 C_s} \quad (25.25)$$
U_o = difference between the maximum kinetic energy and minimum kinetic energy in the cycle (Nm or J) ω = mean angular velocity of flywheel (rad/s) C_s = coefficient of fluctuation of speed
$$\omega = \frac{\omega_{max} + \omega_{min}}{2}$$

(*Contd.*)

(*Contd.*)

ω_{max} = maximum angular velocity of flywheel (rad/s) ω_{min} = minimum angular velocity of flywheel (rad/s)	
$$I_r = m_r R^2$$	(25.26)
m_r = mass of the rim (kg) R = mean radius of the rim (m)	
Stresses in spokes	
$$\sigma_t = \frac{P_1}{A_1}$$	(25.27)
P_1 = tensile force in each spoke (N) A_1 = cross-sectional area of spoke (mm^2) σ_t = tensile stress in spokes (N/mm^2)	
$$P_1 = \frac{2}{3}\left[\frac{(1000)mv^2}{C}\right]$$	(25.28)
m = mass of the rim per mm of circumference (kg/mm) v = velocity at the mean radius (m/s) C = constant	
$$m = bt\rho$$	(25.29)
ρ = mass density of flywheel material (kg/mm^3) b = width of the rim (mm) t = thickness of the rim (mm)	
Stresses in rim	
$$\sigma_t = \frac{(1000)mv^2}{bt}\left[1 - \frac{\cos\phi}{3C\sin\alpha} \pm \frac{2(1000)R}{Ct}\left(\frac{1}{\alpha} - \frac{\cos\varphi}{\sin\alpha}\right)\right]$$	(25.30)
σ_t = tensile stress in the rim (N/mm^2) R = mean radius of rim (m) 2α = angle between two consecutive spokes (rad) ϕ = angle from centre line between spokes to the section where the stress is being calculated (rad)	
Expressions for constant C	
For 4 spokes: $$2\alpha = \frac{\pi}{2}$$ $$C = \left[\frac{72\,960\,R^2}{t^2} + 0.643 + \frac{A}{A_1}\right]$$	(25.31)
For 6 spokes: $$2\alpha = \frac{\pi}{3}$$ $$C = \left[\frac{20\,280\,R^2}{t^2} + 0.957 + \frac{A}{A_1}\right]$$	(25.32)

(*Contd.*)

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For 8 spokes:	$2\alpha = \dfrac{\pi}{4}$	
	$C = \left[\dfrac{9120\ R^2}{t^2} + 1.274 + \dfrac{A}{A_1}\right]$	(25.33)
A = cross-sectional area of rim (mm^2) (bt) A_1 = cross-sectional area of spokes (mm^2)		

25.5 CAMS AND FOLLOWERS

Table 25.11 *Types of cams*

Wedge cam

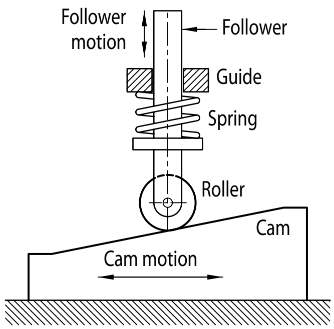
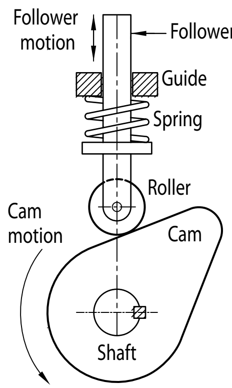



Plate cam


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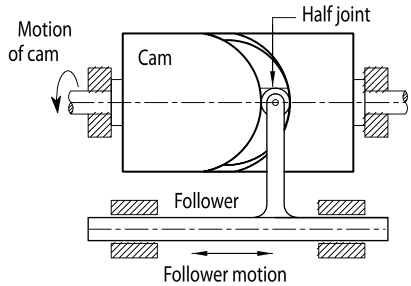
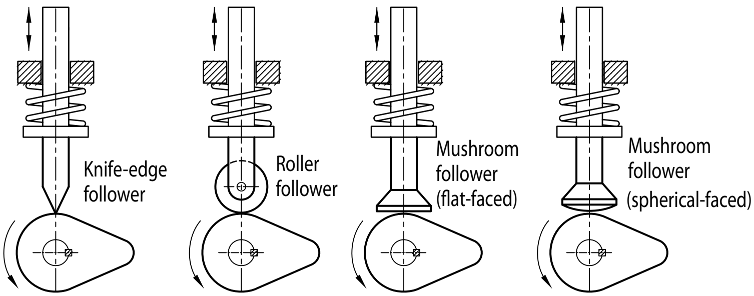
Cylindrical cam


Note: (i) The wedge cam requires reciprocating motion, rather than continuous rotary motion. There are very few practical applications of wedge cam due this limitation.
(ii) Plate cams are extensively used in internal combustion engines for opening of inlet and exhaust valves.
(iii) Cylindrical cams are used in automatic lathes and machine tools.
(iv) The term 'half joint' means 'roll-slide' joint, because it allows both rolling as well as sliding motion.

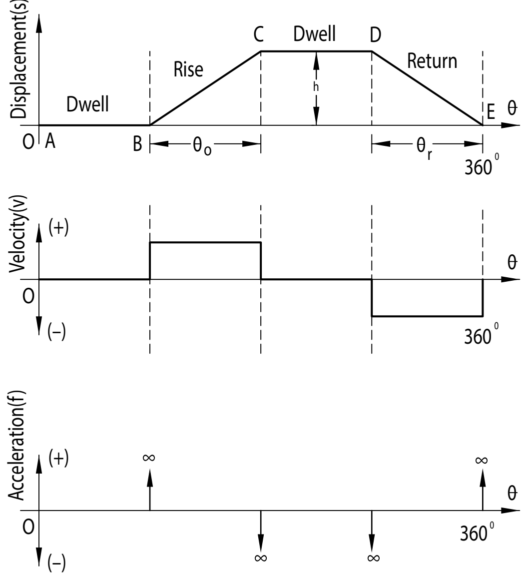
Table 25.12 *Types of followers*



Note: (i) Knife-edge follower is rarely used in practical applications due to rapid rate of wear at the knife-edge.
(ii) In stationary petrol and diesel engines, more space is available and roller follower is given preference over mushroom follower. Roller followers are frequently used in production machinery because of their easy replacement and availability from the manufactures' stock in any quantity.
(iii) Mushroom followers are used for operating the inlet and exhaust valves in cylinders of automobile engines.

25.6 FOLLOWER MOTIONS

Table 25.13 *Uniform velocity motion*



Displacement function

$$s = \left(\frac{h}{\theta_o}\right)\theta \qquad (25.34)$$

Velocity function

$$v = \left(\frac{h}{\theta_o}\right)\omega \ = \text{constant} \qquad (25.35)$$

Acceleration function

$$f = 0 \qquad (25.36)$$

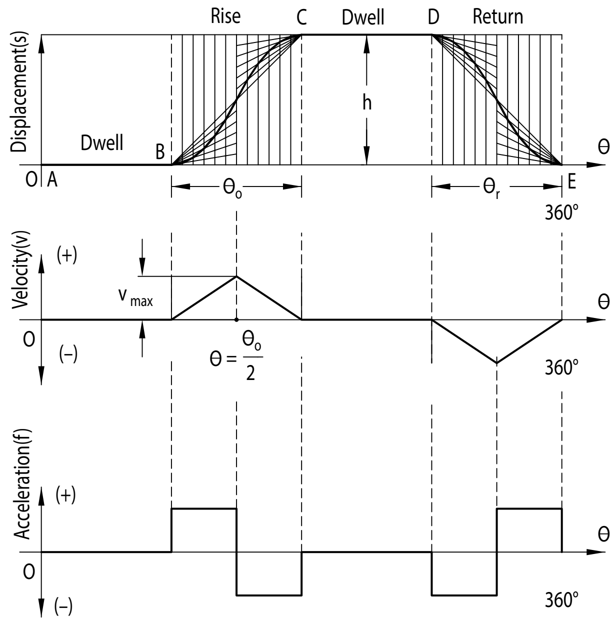
At point *B*, *C*, *D* or *E*,

$$f_B = \frac{v-0}{t \to 0} = \frac{v}{0} = \infty \qquad (25.37)$$

Notations: (For Tables 25.13 to 25.15)

s = instantaneous displacement of the follower
θ = instantaneous angle of rotation of the cam
h = lift or maximum displacement of the follower
θ_o = cam rotation during rise
θ_r = cam rotation during return
v = velocity of the follower
f = acceleration of the follower

Table 25.14 *Uniform acceleration and retardation motion*



Displacement function

For $\left(\dfrac{\theta}{\theta_o}\right) \le 0.5 \qquad s = 2h\dfrac{\theta^2}{\theta_o^2}$ (25.38)

For $\left(\dfrac{\theta}{\theta_o}\right) \ge 0.5 \qquad s = h\left[1 - 2\left(1 - \dfrac{\theta}{\theta_o}\right)^2\right]$ (25.39)

Velocity function

For $\left(\dfrac{\theta}{\theta_o}\right) \le 0.5 \qquad v = \dfrac{4h\omega\theta}{\theta_o^2}$ (25.40)

For $\left(\dfrac{\theta}{\theta_o}\right) \ge 0.5 \qquad v = \dfrac{4h\omega}{\theta_0}\left(1 - \dfrac{\theta}{\theta_o}\right)$ (25.41)

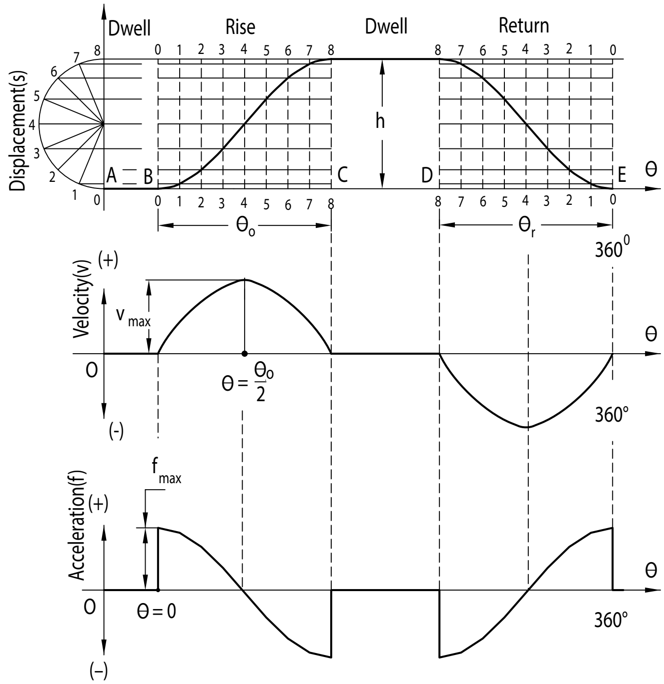
$v_{\max} = \left(\dfrac{2h\omega}{\theta_0}\right)$ (25.42)

Acceleration function

For $\left(\dfrac{\theta}{\theta_o}\right) \le 0.5 \qquad f = \left(\dfrac{4h\omega^2}{\theta_o^2}\right)$ (25.43)

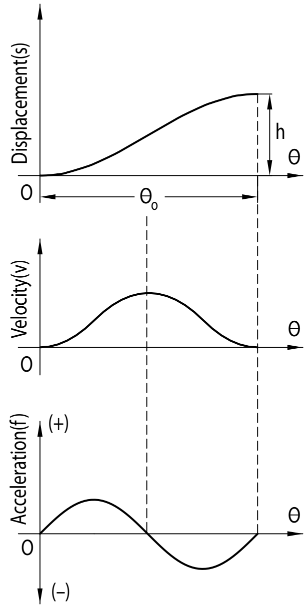
For $\left(\dfrac{\theta}{\theta_o}\right) \ge 0.5 \qquad f = -\left(\dfrac{4h\omega^2}{\theta_o^2}\right)$ (25.44)

Table 25.15 *Simple harmonic motion*



Displacement function	
$s = \frac{h}{2}\left[1 - \cos\left(\frac{\pi\theta}{\theta_o}\right)\right]$ or $s = \frac{h}{2}\left[1 - \cos\left(\frac{\pi\omega t}{\theta_o}\right)\right]$	(25.45)
Velocity function	
$v = \frac{h}{2}\left(\frac{\pi\omega}{\theta_o}\right)\sin\left(\frac{\pi\theta}{\theta_o}\right)$	(25.46)
$v_{max} = \frac{h}{2}\left(\frac{\pi\omega}{\theta_o}\right)$ (For outward stroke)	(25.47)
$v_{max} = -\frac{h}{2}\left(\frac{\pi\omega}{\theta_r}\right)$ (For return stroke)	(25.48)
Acceleration function	
$f = \frac{h}{2}\left(\frac{\pi\omega}{\theta_o}\right)^2\cos\left(\frac{\pi\theta}{\theta_o}\right)$	(25.49)
$f_{max} = \frac{h}{2}\left(\frac{\pi\omega}{\theta_o}\right)^2$ (For outward stroke)	(25.50)
$f_{max} = \frac{h}{2}\left(\frac{\pi\omega}{\theta_r}\right)^2$ (For return stroke)	(25.51)

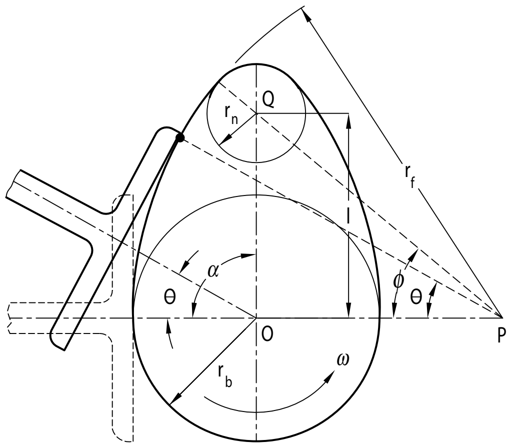
Table 25.16 *Sine acceleration or cycloidal motion*



Displacement function	
$$s = h\left[\frac{\theta}{\theta_0} - \frac{1}{2\pi}\sin\frac{2\pi\theta}{\theta_o}\right]$$	(25.52)
Velocity function	
$$v = \frac{h\omega}{\theta_o}\left[1 - \cos\frac{2\pi\theta}{\theta_o}\right]$$	(25.53)
$$v_{max} = \frac{2h\omega}{\theta_o} \text{ at } \left(\theta = \frac{\theta_o}{2}\right)$$	(25.54)
Acceleration function	
$$f = \frac{2\pi h\omega^2}{\theta_o^2}\sin\left(\frac{2\pi\theta}{\theta_o}\right)$$	(25.55)
$$f_{max} = \frac{2\pi h\omega^2}{\theta_o^2} \text{ at } \left(\theta = \frac{\theta_o}{4}\right)$$	(25.56)

25.7 CAMS WITH SPECIFIED CONTOURS

Table 25.17 *Circular arc cam with flat faced reciprocating follower*

Notations
s = instantaneous displacement of the follower v = instantaneous velocity of the follower f = instantaneous acceleration of the follower θ = instantaneous angle of rotation of the cam r_b = base circle radius of cam r_n = nose radius of cam r_f = radius of circular flank of cam l = distance between cam centre and nose centre α = angle of lift
Case I - Follower in contact with circular arc flank

$$s = (r_f - r_b)(1 - \cos\theta) \qquad (25.57)$$ $$v = \omega(r_f - r_b)\sin\theta \qquad (25.58)$$ $$f = \omega^2(r_f - r_b)\cos\theta \qquad (25.59)$$
Note: (i) The above three equations are applicable only while the follower is in contact with circular arc which has the centre of curvature at P, i.e., for values of θ form 0 to ϕ. When θ is greater than ϕ, the follower is in contact with nose with centre of curvature at Q. (ii) The velocity of follower is maximum when $\theta = \phi$. $$v_{max} = \omega(r_f - r_b)\sin\phi \qquad (25.60)$$ (iii) The acceleration of follower is maximum when $\theta = 0$. $$f_{max} = \omega^2(r_f - r_b) \qquad (25.61)$$

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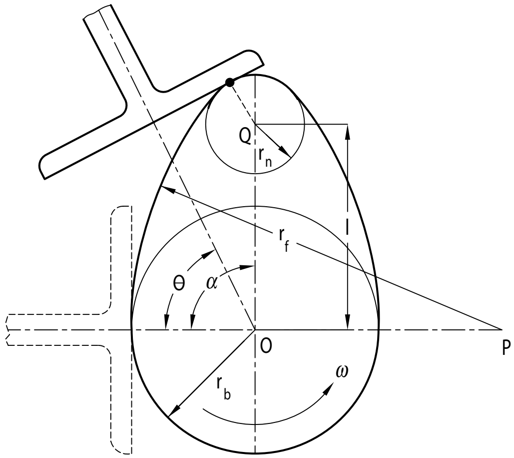
Case II - Follower in contact with circular nose	
	
$$s = r_n + l\cos(\alpha - \theta) - r_b$$	(25.62)
$$v = l\omega\sin(\alpha - \theta)$$	(25.63)
$$f = -l\omega^2\cos(\alpha - \theta)$$	(25.64)
Note: (i) The negative sign for acceleration (f) indicates that the follower is retarded while in contact with the nose of the cam. The above three equations are applicable only while the follower is in contact with the nose portion of the cam with centre of curvature at Q. (ii) The retardation of the follower is maximum when $(\alpha - \theta) = 0$, i.e., at the end of the lift or the follower is at the apex of the nose. $$f_{max} = -l\omega^2$$	(25.65)

Table 25.18 *Tangent cam with roller reciprocating follower*

Notations
s = instantaneous displacement of the follower v = instantaneous velocity of the follower f = instantaneous acceleration of the follower θ = instantaneous angle of rotation of the cam ϕ = rotation of cam from highest position of follower r_b = base circle radius of cam r_n = nose radius of cam r_r = radius of roller l = distance between cam centre and nose centre α = angle of lift

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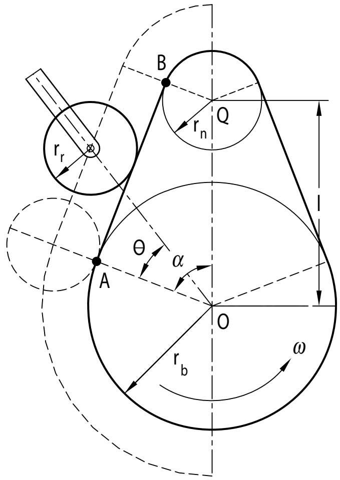
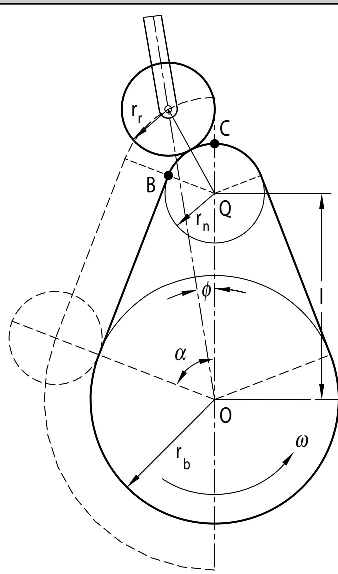
Case I - Roller in contact with straight flank

$s = (r_b + r_r)\sec\theta - (r_b + r_r)$ (25.66) $v = \omega(r_b + r_r)\sec\theta\tan\theta$ (25.67) $f = \omega^2(r_b + r_r)(2\sec^3\theta - \sec\theta)$ (25.68)
Note: (i) The above three equations are applicable only while the roller is in contact with flank from *A* to *B*. (ii) The velocity increases from zero at *A* to a maximum value at *B*. (iii) The acceleration increases from a minimum value at *A* to a maximum value at *B*. (iv) If the angle of rotation of cam is β while the roller moves from *A* to *B*, then $\tan\beta = \left(\dfrac{l\sin\alpha}{r_b + r_r}\right)$ (25.69) (v) The maximum velocity and acceleration on the flank are obtained by substituting ($\theta = \beta$) in above equations.
Case II - Roller in contact with circular nose


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$$s = l[\cos\phi + \sqrt{(n^2 - \sin^2\phi)}] - (r_b + r_r) \quad (25.70)$$ $$v = -\omega l\left[\sin\phi + \frac{\sin 2\phi}{2\sqrt{(n^2 - \sin^2\phi)}}\right] \quad (25.71)$$ $$f = -\omega^2 l\left[\cos\phi + \frac{\sin^4\phi + n^2\cos 2\phi}{(n^2 - \sin^2\phi)^{3/2}}\right] \quad (25.72)$$ where, $n = \left(\frac{r_n + r_r}{l}\right)$
Note: (i) The above three equations are applicable only while the roller is in contact with the nose portion of the cam. (ii) The negative sign for acceleration (f) indicates that the follower is retarded while in contact with the nose of the cam. (iii) The velocity decreases from a maximum value at B to zero at C. (iv) The acceleration decreases from a maximum value at B to a minimum value at C. (v) The negative acceleration at the highest point C is given by, $$f = -\omega^2 l\left(1 + \frac{1}{n}\right) \quad (25.73)$$ (vi) The deceleration of follower while the roller is in contact with nose at B is obtained by substituting $\phi = (\alpha - \beta)$

Table 25.19 *Circular arc cam with roller reciprocating follower*

Notations
s = instantaneous displacement of the follower v = instantaneous velocity of the follower f = instantaneous acceleration of the follower θ = instantaneous angle of rotation of the cam ϕ = rotation of cam from highest position of follower r_b = base circle radius of cam r_n = nose radius of cam r_r = radius of roller r_f = radius of circular flank of cam l = distance between cam centre and nose centre α = angle of lift

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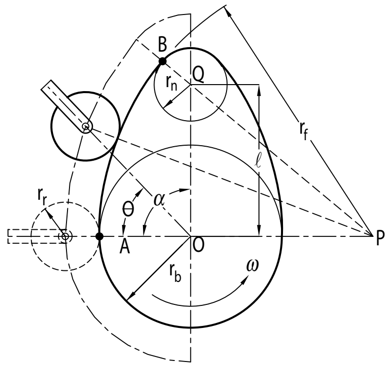
Case I - Roller in contact with circular flank



$$A = (r_f - r_b) \tag{25.74}$$

$$B = (r_f + r_r) \tag{25.75}$$

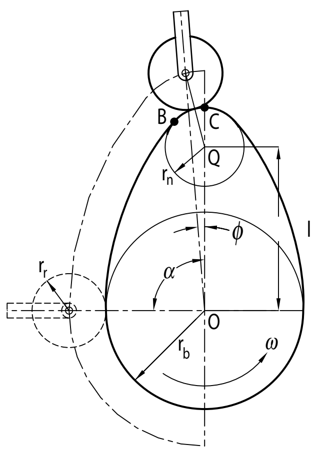
$$s = \sqrt{(B^2 - A^2 \sin^2 \theta)} - A \cos \theta - (r_b + r_r) \tag{25.76}$$

$$v = \omega A \left[\sin \theta - \frac{A \sin 2\theta}{2\sqrt{B^2 - A^2 \sin^2 \theta}} \right] \tag{25.77}$$

$$f = \omega^2 A \left[\cos \theta - \frac{A \cos 2\theta}{(B^2 - A^2 \sin^2 \theta)^{1/2}} - \frac{A^3 \sin^2 2\theta}{4(B^2 - A^2 \sin^2 \theta)^{3/2}} \right] \tag{25.78}$$

Note: The above three equations are applicable only while the roller is in contact with circular flank from *A* to *B*.

Case II - Roller in contact with circular nose



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$$s = l[\cos\phi + \sqrt{(n^2 - \sin^2\phi)}] - (r_b + r_r) \quad (25.79)$$

$$v = -\omega l\left[\sin\phi + \frac{\sin 2\phi}{2\sqrt{(n^2 - \sin^2\phi)}}\right] \quad (25.80)$$

$$f = -\omega^2 l\left[\cos\phi + \frac{\sin^4\phi + n^2\cos 2\phi}{(n^2 - \sin^2\phi)^{3/2}}\right] \quad (25.81)$$

where, $n = \left(\frac{r_n + r_r}{l}\right)$

Note: (i) The above three equations are applicable only while the roller is in contact with the nose portion of the cam.

(ii) The negative sign for acceleration (f) indicates that the follower is retarded while in contact with the nose of the cam.

(iii) The velocity decreases from a maximum value at B to zero at C.

(iv) The acceleration decreases from a maximum value at B to a minimum value at C.

(v) The negative acceleration at the highest point C is given by,

$$f = -\omega^2 l\left(1 + \frac{1}{n}\right) \quad (25.82)$$

(vi) The deceleration of follower while the roller is in contact with nose at B is obtained by substituting $\phi = (\alpha - \beta)$.

where,

$$\tan\beta = \left(\frac{l\sin\alpha}{r_b + r_r}\right) \quad (25.83)$$

Table 25.20 *Eccentric cam with flat faced follower*

Notations
s = instantaneous displacement of the follower v = instantaneous velocity of the follower f = instantaneous acceleration of the follower θ = instantaneous angle of rotation of the cam e = eccentricity of cam R = radius of circular disc h = lift of follower O = axis of cam rotation P = centre of cam

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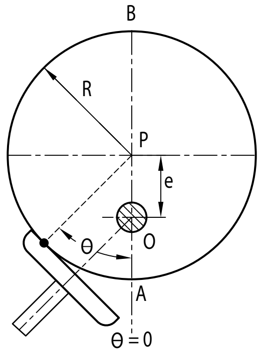

$$h = 2e \tag{25.84}$$

$$s = e(1 - \cos\theta) \tag{25.85}$$

$$v = \omega e \sin\theta \tag{25.86}$$

$$f = \omega^2 e \cos\theta \tag{25.87}$$

Note: (i) The follower moves with simple harmonic motion.
(ii) There are no periods of dwell in this type of cam.
(iii) The velocity increases from zero at A to a maximum value at ($\theta = 90°$) and then falls again to zero at B.
(iv) The acceleration has its maximum positive value at A and maximum negative value at B. It is zero at ($\theta = 90°$).

25.8 POLYNOMIAL CAMS

Table 25.21 *Polynomial equation*

$$s = C_0 + C_1\left(\frac{\theta}{\beta}\right)^1 + C_2\left(\frac{\theta}{\beta}\right)^2 + C_3\left(\frac{\theta}{\beta}\right)^3 + C_4\left(\frac{\theta}{\beta}\right)^4 + C_5\left(\frac{\theta}{\beta}\right)^5 + \cdots \tag{25.88}$$

s = instantaneous displacement of the follower
v = instantaneous velocity of the follower
f = instantaneous acceleration of the follower
θ = instantaneous angle of rotation of the cam
β = total angle of any segment such as 'rise'
$C_0, C_1, C_2, \ldots$ = constant coefficients

Note: (i) The value of β represents the total travel of θ for the respective section of cam such as 'rise'. Therefore, the magnitude of the ratio $\left(\frac{\theta}{\beta}\right)$ varies from 0 to 1.
(ii) The constants $C_0, C_1, C_2, \ldots$ are evaluated by assuming suitable boundary conditions.

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(iii) The degree of a polynomial is defined as the highest power present in any term of the equation. The polynomial of degree n will have $(n + 1)$ terms due to first term C_0 or $C_0\left(\frac{\theta}{\beta}\right)^0$.
(iv) The number of boundary conditions determines the degree of resulting polynomial. If 'k' represents the number of boundary conditions, there will be k equations in k unknowns C_0, C_1, C_2, ... and the degree of polynomial will be $(k - 1)$.
Velocity equation
$$v = \frac{\omega}{\beta}\left[C_1 + 2C_2\left(\frac{\theta}{\beta}\right) + 3C_3\left(\frac{\theta}{\beta}\right)^2 + 4C_4\left(\frac{\theta}{\beta}\right)^3 + 5C_5\left(\frac{\theta}{\beta}\right)^4 + \cdots\right] \qquad (25.89)$$
Acceleration equation
$$f = \frac{\omega^2}{\beta^2}\left[2C_2 + 6C_3\left(\frac{\theta}{\beta}\right) + 12C_4\left(\frac{\theta}{\beta}\right)^2 + 20C_5\left(\frac{\theta}{\beta}\right)^3 + \cdots\right] \qquad (25.90)$$

Table 25.22 *The 2-3 polynomial cam*

Polynomial equation
$$s = C_0 + C_1\left(\frac{\theta}{\beta}\right)^1 + C_2\left(\frac{\theta}{\beta}\right)^2 + C_3\left(\frac{\theta}{\beta}\right)^3 \qquad (25.91)$$
Boundary conditions
When $\left(\frac{\theta}{\beta}\right) = 0 \qquad s = h \qquad v = 0$ (25.92)
When $\left(\frac{\theta}{\beta}\right) = 1 \qquad s = 0 \qquad v = 0$ (25.93)
Solution of polynomial equation
$C_0 = h \quad C_1 = 0 \quad C_2 = -3h \quad C_3 = 2h$ (25.94)
Displacement
$$s = h\left[1 - 3\left(\frac{\theta}{\beta}\right)^2 + 2\left(\frac{\theta}{\beta}\right)^3\right] \qquad (25.95)$$
Velocity
$$v = \frac{\omega h}{\beta}\left[-6\left(\frac{\theta}{\beta}\right) + 6\left(\frac{\theta}{\beta}\right)^2\right] \qquad (25.96)$$
Acceleration
$$f = \frac{\omega^2 h}{\beta^2}\left[-6 + 12\left(\frac{\theta}{\beta}\right)\right] \qquad (25.97)$$

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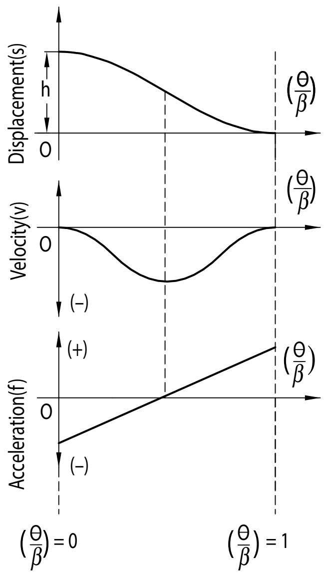

Table 25.23 *The 3-4-5 polynomial cam*

Polynomial equation	
$$s = C_0 + C_1\left(\frac{\theta}{\beta}\right)^1 + C_2\left(\frac{\theta}{\beta}\right)^2 + C_3\left(\frac{\theta}{\beta}\right)^3 + C_4\left(\frac{\theta}{\beta}\right)^4 + C_5\left(\frac{\theta}{\beta}\right)^5$$	(25.98)
Boundary conditions	
When $\left(\frac{\theta}{\beta}\right) = 0$ $\quad s = h \quad v = 0 \quad f = 0$	(25.99)
When $\left(\frac{\theta}{\beta}\right) = 1$ $\quad s = 0 \quad v = 0 \quad f = 0$	(25.100)
Solution of polynomial equation	
$C_0 = h \quad C_1 = 0 \quad C_2 = 0 \quad C_3 = -10h \quad C_4 = 15\,h \quad C_5 = -6\,h$	(25.101)
Displacement	
$$s = h\left[1 - 10\left(\frac{\theta}{\beta}\right)^3 + 15\left(\frac{\theta}{\beta}\right)^4 - 6\left(\frac{\theta}{\beta}\right)^5\right]$$	(25.102)
Velocity	
$$v = \frac{\omega h}{\beta}\left[-30\left(\frac{\theta}{\beta}\right)^2 + 60\left(\frac{\theta}{\beta}\right)^3 - 30\left(\frac{\theta}{\beta}\right)^4\right]$$	(25.103)

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Acceleration	
$$f = \frac{\omega^2 h}{\beta^2}\left[-60\left(\frac{\theta}{\beta}\right) + 180\left(\frac{\theta}{\beta}\right)^2 - 120\left(\frac{\theta}{\beta}\right)^3\right]$$	(25.104)

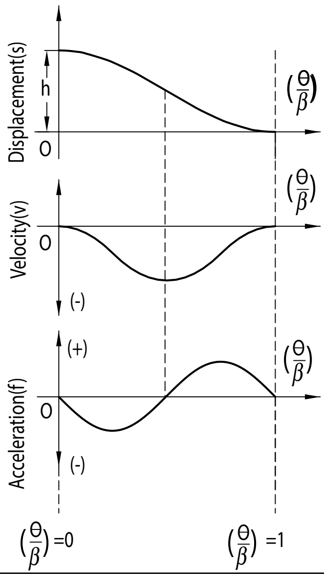

Table 25.24 *The 4-5-6-7 polynomial cam*

Polynomial equation	
$$s = C_0 + C_1\left(\frac{\theta}{\beta}\right)^1 + C_2\left(\frac{\theta}{\beta}\right)^2 + C_3\left(\frac{\theta}{\beta}\right)^3 + C_4\left(\frac{\theta}{\beta}\right)^4 + C_5\left(\frac{\theta}{\beta}\right)^5 + C_6\left(\frac{\theta}{\beta}\right)^6 + C_7\left(\frac{\theta}{\beta}\right)^7$$	(25.105)
Boundary conditions	
When $\left(\frac{\theta}{\beta}\right) = 0 \quad s = h \quad v = 0 \quad f = 0 \quad j = 0$	(25.106)
When $\left(\frac{\theta}{\beta}\right) = 1 \quad s = 0 \quad v = 0 \quad f = 0 \quad j = 0$	(25.107)
Solution of polynomial equation	
$C_0 = h \quad C_1 = 0 \quad C_2 = 0 \quad C_3 = 0$ $C_4 = -35\,h \quad C_5 = 84\,h \quad C_6 = -70\,h \quad C_7 = 20\,h$	(25.108)

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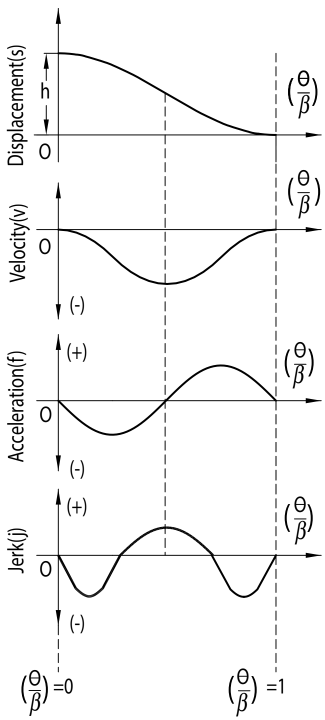
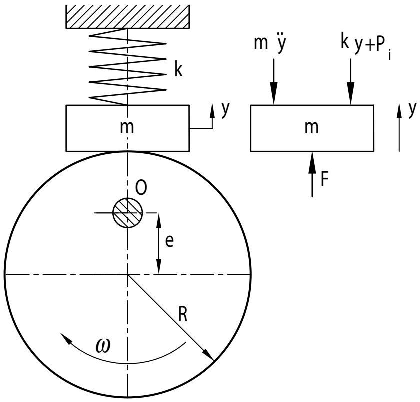
Displacement	
$$s = h\left[1 - 35\left(\frac{\theta}{\beta}\right)^4 + 84\left(\frac{\theta}{\beta}\right)^5 - 70\left(\frac{\theta}{\beta}\right)^6 + 20\left(\frac{\theta}{\beta}\right)^7\right]$$	(25.109)
Velocity	
$$v = \frac{\omega h}{\beta}\left[-140\left(\frac{\theta}{\beta}\right)^3 + 420\left(\frac{\theta}{\beta}\right)^4 - 420\left(\frac{\theta}{\beta}\right)^5 + 140\left(\frac{\theta}{\beta}\right)^6\right]$$	(25.110)
Acceleration	
$$f = \frac{\omega^2 h}{\beta^2}\left[-420\left(\frac{\theta}{\beta}\right)^2 + 1680\left(\frac{\theta}{\beta}\right)^3 - 2100\left(\frac{\theta}{\beta}\right)^4 + 840\left(\frac{\theta}{\beta}\right)^5\right]$$	(25.111)
Jerk	
$$j = \frac{\omega^3 h}{\beta^3}\left[-840\left(\frac{\theta}{\beta}\right) + 5040\left(\frac{\theta}{\beta}\right)^2 - 8400\left(\frac{\theta}{\beta}\right)^3 + 4200\left(\frac{\theta}{\beta}\right)^4\right]$$	(25.112)



25.9 DESIGN OF CAM DRIVES

Table 25.25 *Jump phenomenon*



Notations

m = mass of follower
k = stiffness of spring
P_i = initial preload in spring (force exerted by the spring when $y = 0$)
e = eccentricity of cam
ω = angular velocity of cam
y = displacement of follower
$\ddot{y}$ = acceleration of follower
$m\ \ddot{y}$ = inertia force
$(ky + P_i)$ = total spring force
F = contact force between cam and follower

Force analysis

$$F = m\ddot{y} + (ky + P_i) \tag{25.113}$$

$$y = e(1 - \cos\theta) = e(1 - \cos\omega t) \tag{25.114}$$

$$\ddot{y} = \omega^2 e \cos\theta = \omega^2 e \cos\omega t \tag{25.115}$$

$$F = m\omega^2 e \cos\omega t + ke(1 - \cos\omega t) + P_i$$

or

$$F = (m\omega^2 - k)e \cos\omega t + (P_i + ke) \tag{25.116}$$

Jump

The contact force F consists of a constant term $(P_i + ke)$ with a cosine component of force superimposed on it. The force is maximum when $\theta = 0$ and minimum when $\theta = 180°$. The cosine term has amplitude that depends upon the square of cam shaft speed (ω^2). As the velocity of cam (ω) increases, the cosine component of force increases at a greater rate (ω^2). At a certain speed (ω_j), the contact force becomes zero at $\theta = 180°$. When this happens, there is some impact between cam and follower, resulting in noisy operation. This result is called jump phenomenon. The noise occurs when the contact is reestablished.

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Condition for jump	
$$\omega t = \pi \quad F = 0 \quad \omega = \omega_j$$	(25.117)
$$\omega_j = \sqrt{\frac{P_i + 2ke}{me}}$$	(25.118)
Note: The jump will not occur if, $$P_i > e(m\omega^2 - 2k)$$	(25.119)

Table 25.26 *Comparison of follower motions*

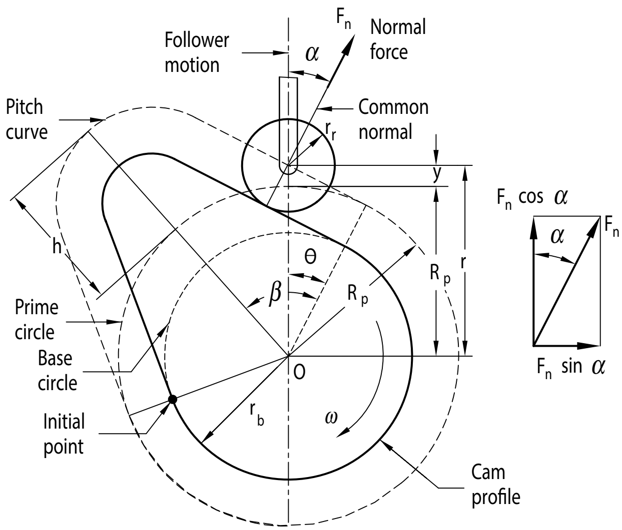
Follower motion	Maximum velocity	Maximum acceleration	Maximum jerk	Design features
Uniform acceleration	$2.000\left(\frac{h}{\beta}\right)$	$4.000\left(\frac{h}{\beta^2}\right)$	Infinite	Infinite jerk results in poor design
Simple harmonic motion	$1.571\left(\frac{h}{\beta}\right)$	$4.945\left(\frac{h}{\beta^2}\right)$	Infinite	Infinite jerk results in poor design
Trapezoidal acceleration	$2.000\left(\frac{h}{\beta}\right)$	$5.300\left(\frac{h}{\beta^2}\right)$	$44\left(\frac{h}{\beta^3}\right)$	–
Modified trapezoidal acceleration	$2.000\left(\frac{h}{\beta}\right)$	$4.888\left(\frac{h}{\beta^2}\right)$	$61\left(\frac{h}{\beta^3}\right)$	It has low acceleration but rough jerk
Modified sine acceleration	$1.760\left(\frac{h}{\beta}\right)$	$5.528\left(\frac{h}{\beta^2}\right)$	$69\left(\frac{h}{\beta^3}\right)$	It has low velocity and good acceleration
3-4-5 polynomial displacement	$1.875\left(\frac{h}{\beta}\right)$	$5.777\left(\frac{h}{\beta^2}\right)$	$60\left(\frac{h}{\beta^3}\right)$	It results in good compromise design
Cycloidal motion	$2.000\left(\frac{h}{\beta}\right)$	$6.283\left(\frac{h}{\beta^2}\right)$	$40\left(\frac{h}{\beta^3}\right)$	It has smooth acceleration and jerk
4-5-6-7 polynomial displacement	$2.188\left(\frac{h}{\beta}\right)$	$7.526\left(\frac{h}{\beta^2}\right)$	$52\left(\frac{h}{\beta^3}\right)$	It has high acceleration but smooth jerk

Note: h = total lift of follower (rise or fall) of any one section of cam profile
β = total cam angle of any one section of cam profile (rise or fall)

Table 25.27 *Pressure angle*

Notations

α = pressure angle
θ = instantaneous angle of rotation of the cam
r_b = base circle radius of cam
r_r = radius of roller
R_p = prime circle radius
r = distance between cam centre and roller centre
y = instantaneous displacement of follower
ω = angular velocity of cam
h = lift or maximum displacement of the follower
F_n = normal force between the cam and the follower
$F_n \cos \alpha$ = component of (F_n) that lifts the follower against the spring force
$F_n \sin \alpha$ = component of (F_n) that acts as side thrust on the guide of follower



Pressure angle

The pressure angle (α) is the angle between the axis of follower stem and the line of force exerted by the cam on the follower. This line is the normal to the pitch curve through the trace point. Pressure angle varies in magnitude during the rotation of cam. The component of force along the line of motion ($F_n \cos \alpha$) is useful component in overcoming the output load. The perpendicular component ($F_n \sin \alpha$) should be kept as small as possible to reduce friction between the follower and its guide way. Following guidelines should be considered for pressure angle:

(i) When ($\alpha = 0°$), complete transmitted force goes into motion of the follower and there is no side thrust on the guides of follower.
(ii) When ($\alpha = 90°$), there is no force acting on the follower with the result that there will be no motion of the follower.
(iii) According to rule of thumb, the pressure angle should be up to 30° for translating follower. If the follower is oscillating on the pivoted arm, pressure angle up to 35° is acceptable.

Equation for pressure angle

$$\tan \alpha = \frac{1}{r}\frac{dy}{d\theta} \tag{25.120}$$

$$r = (R_p + y) = (r_b + r_r + y) \tag{25.121}$$

Equation for maximum pressure angle

$$\tan(\alpha_{\max}) = \frac{\left(\dfrac{dy}{dt}\right)_{\max}}{\omega(R_p + y)} \quad \text{(Approx.)} \tag{25.122}$$

The above equation gives satisfactory results for parabolic, harmonic and cycloidal motions of follower.

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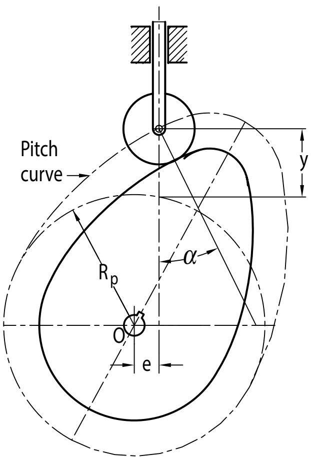
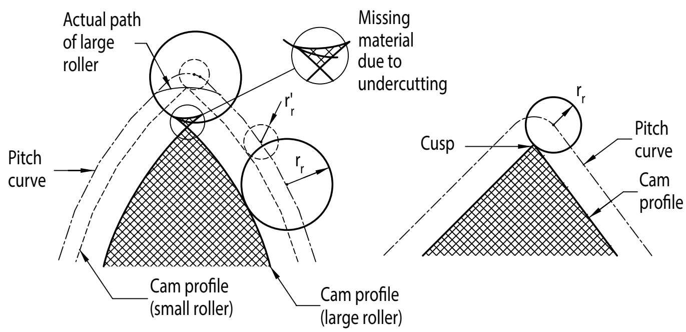
Equation for pressure angle with offset follower	
	$$\tan\alpha = \frac{y' - e}{\sqrt{(R_p^2 - e^2)} + y} \quad (25.123)$$ where $y' = \frac{dy}{d\theta}$ e = eccentricity **Note:** (i) It is observed from the above equation that once the displacement equations for y have been determined, the two parameters R_p and e can be adjusted to obtain a suitable pressure angle. (ii) Small value of eccentricity e can be used to reduce the pressure angle α during the rise motion when y' is positive. However, it is only at the cost of increased pressure angle during the return motion when y' is negative. Since the magnitude of force is more during the rise, it is a common practice to offset the follower to take advantage of reduced pressure angle. (iii) Significant reduction in pressure angle can be achieved by increasing the prime circle radius R_p. Alternatively, permissible pressure angle can be assumed and the value of R_p can be evaluated.

Table 25.28 *Undercutting*



The cam profile must be continuous curve without any loop. If the curvature of the pitch curve is too sharp, then the part of the cam shape would be lost during machining and thereafter the intended cam motion would not be achieved. Such a cam is said to be undercut.

The cam profile generated by large roller is undercut and doubles over itself. If the cam is machined as per this profile, it results in cusp (pointed cam) which does not produce the desired motion of the follower. On the other hand a small roller moving over the same pitch curve generates a satisfactory cam profile. Similarly, if the size of prime circle and the cam are increased, the larger roller will also operate satisfactorily.

Radius of curvature of pitch curve

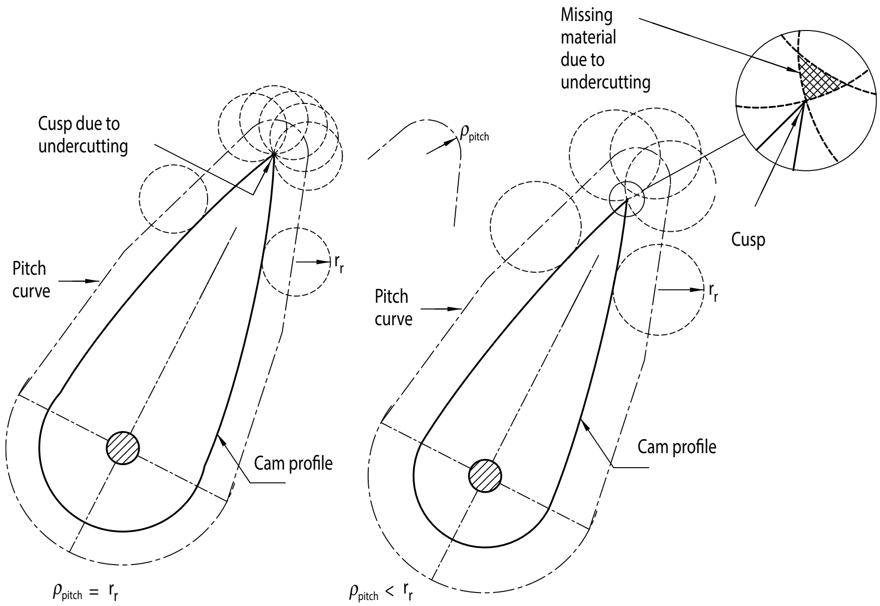
$$\rho_{\text{pitch}} = \frac{\left[(R_p + y)^2 + \left(\frac{1}{\omega}\frac{dy}{dt}\right)^2\right]^{3/2}}{(R_p + y)^2 + 2\left(\frac{1}{\omega}\frac{dy}{dt}\right)^2 - (R_p + y)\left(\frac{1}{\omega^2}\frac{d^2y}{dt^2}\right)} \quad (25.124)$$

The above equation is applicable to convex pitch curve only.

(*Contd.*)

(*Contd.*)

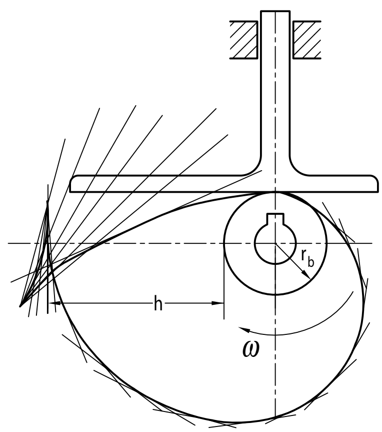
ρ_{pitch} = radius of curvature of pitch curve
R_p = prime circle radius
y = instantaneous displacement of follower
ω = angular velocity of cam



Note: (i) When the radius of curvature of pitch curve is more than roller radius ($\rho_{pitch} > r_r$), there is no undercutting and the profile is satisfactory.
(ii) When the radius of curvature of pitch curve is equal to roller radius ($\rho_{pitch} = r_r$), it results in cusp or pointed cam.
(iii) When the radius of curvature of pitch curve is less than roller radius ($\rho_{pitch} < r_r$), undercutting is unavoidable.

Plate cam with reciprocating flat-face follower

(*Contd.*)



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Radius of curvature	
$$\rho = r_b + y + \left(\frac{1}{\omega^2}\frac{d^2y}{dt^2}\right)$$	(25.125)
ρ = radius of curvature r_b = base circle radius y = instantaneous displacement of follower ω = angular velocity of cam	
Undercutting	
In plate cam with flat-face follower, undercutting results when the lift is too large and the corresponding cam rotation too small with small-size cam. It can be avoided by: (i) decreasing the required lift (h) (ii) increasing the cam rotation (β) corresponding to lift (iii) increasing the base circle radius (r_b) of the cam which will result in large-size cam. When undercutting occurs, the radius of curvature changes its sign from positive to negative. On the verge of undercutting, the radius of curvature (ρ) will be zero for some value of θ. To avoid high contact stresses and eliminate undercutting, the radius of curvature (ρ) should be more than some specified value (ρ_{min}). $\rho > \rho_{min}$	
Minimum face width of flat-face follower	
Rule: The distance of travel of the point of contact to either side of the cam rotation centre corresponds precisely with the plot of y'. Therefore, the minimum face width of the flat-face follower must extend at least ($+y'_{max}$) to the right and ($-y'_{max}$) to the left of cam centre in order to maintain the contact. Face width > $[y'_{max} - y'_{min}]$ $$\text{Face width} > \left[\left(\frac{1}{\omega}\frac{dy}{dt}\right)_{max} - \left(\frac{1}{\omega}\frac{dy}{dt}\right)_{min}\right]$$	(25.126)

Table 25.29 *Contact stress between cam and roller*

$$\sigma_c = 0.591\sqrt{\frac{F_n}{b}\frac{E_1E_2}{(E_1+E_2)}\left(\frac{1}{r_r} \pm \frac{1}{\rho_{c\,min}}\right)}$$	(25.127)
σ_c = contact stress between the cam and the roller (MPa or N/mm^2) F_n = normal force between the cam and the follower at the point of contact (N) b = face width of cam (mm) E_1, E_2 = moduli of elasticity of cam and follower materials (MPa or N/mm^2) r_r = radius of roller (mm) $\rho_{c\,min}$ = minimum radius of curvature of cam profile (mm)	

(*Contd.*)

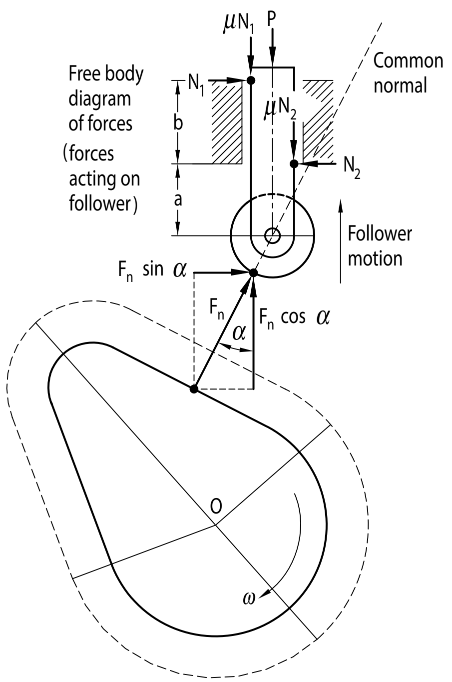
Table 25.30 *Permissible contact stress for cam materials*

(Roller material = Case hardened steel)

Cam material	Hardness	Permissible contact stress (σ_c)(MPa or N/mm^2)	
		With no sliding between cam and roller	With maximum 8.3% sliding between cam and roller
Grey cast iron	140-160 HB	400	360
Grey cast iron	200-220 HB	500	450
Grey cast iron	225-255 HB	600	540
Grey cast iron	255-300 HB	670	530
Carbon steel 20C8	130-150 HB	630	500
Carbon steel 20C8 (carburised)	50-58 HRC	1770	1130
Alloy steel 40Cr4Mo2	270-300 HB	1470	1100
Carbon manganese free cutting steel (induction hardened) 40C15S12	45-55 HRC	1530	1225

Note: The above values are given for the reference of students only.

Table 25.31 *Force analysis of cam and roller follower*



$$F_n = \frac{P}{\cos\alpha - \mu\left(\frac{2a+b}{b}\right)\sin\alpha} \tag{25.128}$$

(*Contd.*)

(*Contd.*)

$$N_1 = \left(\frac{a}{b}\right) F_n \sin\alpha$$	(25.129)
$$N_2 = \left(\frac{a+b}{b}\right) F_n \sin\alpha$$	(25.130)
P = [initial preload in spring + spring force due to rise y + inertia force + weight attached to follower] $$P = P_i + ky + m\frac{d^2y}{dt^2} + mg$$	(25.131)
F_n = normal force between the cam and the follower at the point of contact P = total force acting on follower along the line of stroke μ = coefficient of friction between follower and its guide α = pressure angle N_1, N_2 = reactions of guide on the follower	
Torque required to drive the cam	
$$M_t = \frac{Pv}{\omega}$$	(25.132)
M_t = torque required to drive the cam P = total force acting on follower along the line of stroke v = velocity of follower ω = angular velocity of cam	

Table 25.32 *Cam factor and pitch circle radius*

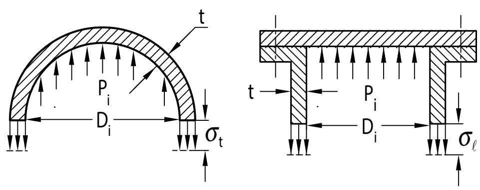
Cam factor		
Type of motion	Cam factor (f)	
Straight line motion	$\cot\alpha_m$	
Simple harmonic motion	$\left(\frac{\pi}{2}\right)\cot\alpha_m$	
Cycloidal motion	$2\cot\alpha_m$	
Parabolic motion	$2\cot\alpha_m$	
Pitch circle radius		
$$R_a = \frac{fh}{\beta}$$		(25.133)
R_a = pitch circle radius f = cam factor h = lift of follower β = total cam angle corresponding to lift (h) α_m = maximum allowable pressure angle α_m = 30° (for reciprocating roller follower) α_m = 35° (for oscillating roller follower)		

Pressure Vessels

26

26.1 THIN CYLINDERS

Table 26.1 *Thin cylinders*



Stresses in cylinder

$$\sigma_t = \frac{P_i D_i}{2t} \tag{26.1}$$

σ_t = circumferential or tangential stress (N/mm^2)
P_i = internal pressure (N/mm^2)
D_i = inner diameter of cylinder (mm)
t = cylinder wall thickness (mm)

$$\sigma_l = \frac{P_i D_i}{4t} \tag{26.2}$$

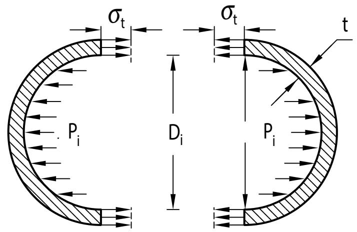
σ_1 = longitudinal stress (N/mm^2)

Thickness of cylinder

$$t = \frac{P_i D_i}{2\sigma_t} \tag{26.3}$$

σ_t = permissible tensile stress for cylinder material (N/mm^2)

Table 26.2 *Thin spherical vessels*



Thickness of spherical shell

$$t = \frac{P_i D_i}{4\sigma_t} \tag{26.4}$$

t = thickness of spherical shell (mm)
P_i = internal pressure (N/mm^2)
D_i = inner diameter of spherical shell (mm)
σ_t = permissible tensile stress (N/mm^2)

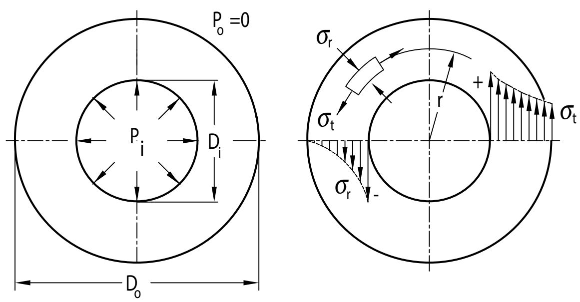
Diameter of spherical shell

$$D_i = \left(\frac{6V}{\pi}\right)^{1/3} \tag{26.5}$$

V = volume of spherical shell (mm^3)

26.2 THICK CYLINDERS

Table 26.3 *Thick cylinders subjected to internal pressure – Principal stresses*



Stresses at radius *r*

$$\sigma_t = +\frac{P_i D_i^2}{(D_o^2 - D_i^2)}\left[\frac{D_o^2}{4r^2} + 1\right] \tag{26.6}$$

(*Contd.*)

(*Contd.*)

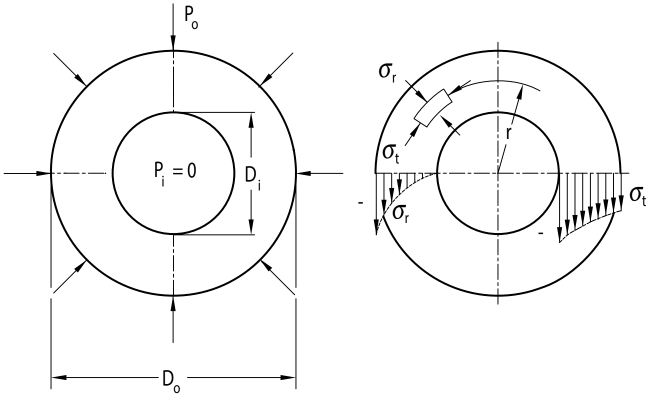
$$\sigma_r = -\frac{P_i D_i^2}{(D_o^2 - D_i^2)}\left[\frac{D_o^2}{4r^2} - 1\right]$$	(26.7)
$$\sigma_l = \frac{P_i D_i^2}{(D_o^2 - D_i^2)}$$	(26.8)
σ_t = circumferential or tangential stress at radius r (N/mm^2) σ_r = radial stress at radius r (N/mm^2) σ_l = longitudinal stress (uniform over cylinder wall) (N/mm^2) P_i = internal pressure (N/mm^2) D_i = inner diameter of cylinder (mm) D_o = outer diameter of cylinder (mm)	
$$t = \left(\frac{D_o - D_i}{2}\right)$$	(26.9)
t = cylinder wall thickness (mm)	
Stresses at inner surface at $\left(r = \frac{D_i}{2}\right)$	
$$\sigma_t = +\frac{P_i(D_o^2 + D_i^2)}{(D_o^2 - D_i^2)}$$	(26.10)
$$\sigma_r = -P_i$$	(26.11)
Stresses at outer surface at $\left(r = \frac{D_o}{2}\right)$	
$$\sigma_t = +\frac{2P_i D_i^2}{(D_o^2 - D_i^2)}$$	(26.12)
$$\sigma_r = 0$$	(26.13)

Note: (i) When the ratio of the inner diameter of the cylinder to the wall thickness is less than 15, the cylinder is called a 'thick-walled' cylinder or simply 'thick' cylinder.

(ii) In thin cylinders, it is assumed that the tangential stress (σ_t) is uniformly distributed over the cylinder wall thickness. In thick cylinders, the tangential stress (σ_t) has highest magnitude at the inner surface of the cylinder and gradually decreases towards the outer surface.

(iii) The radial stress (σ_r) is neglected in thin cylinders, while it is of significant magnitude in case of thick cylinders.

Table 26.4 *Thick cylinders subjected to external pressure – Principal stresses*



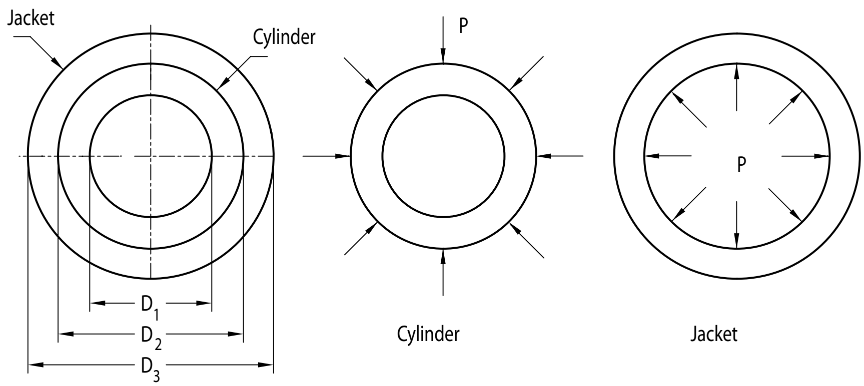
Stresses at radius *r*	
$$\sigma_t = -\frac{P_o D_o^2}{(D_o^2 - D_i^2)}\left[1 + \frac{D_i^2}{4r^2}\right]$$	(26.14)
$$\sigma_r = -\frac{P_o D_o^2}{(D_o^2 - D_i^2)}\left[1 - \frac{D_i^2}{4r^2}\right]$$	(26.15)
σ_t = circumferential or tangential stress at radius r (N/mm^2) σ_r = radial stress at radius r (N/mm^2) P_o = external pressure (N/mm^2) D_i = inner diameter of cylinder (mm) D_o = outer diameter of cylinder (mm)	
Stresses at inner surface at $\left(r = \frac{D_i}{2}\right)$	
$$\sigma_t = -\frac{2P_o D_o^2}{(D_o^2 - D_i^2)}$$	(26.16)
$$\sigma_r = 0$$	(26.17)
Stresses at outer surface at $\left(r = \frac{D_o}{2}\right)$	
$$\sigma_t = -\frac{P_o(D_o^2 + D_i^2)}{(D_o^2 - D_i^2)}$$	(26.18)
$$\sigma_r = -P_o$$	(26.19)

Table 26.5 *Equations of Lame, Clavarino, Birnie and Barlow*

Lame's equation	
$t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma_t + P_i}{\sigma_t - P_i}} - 1\right]$	(26.20)
t = cylinder wall thickness (mm) D_i = inner diameter of cylinder (mm) P_i = internal pressure (N/mm^2) $\sigma_t = \frac{S_{ut}}{(fs)}$ σ_t = permissible tensile stress (N/mm^2) S_{ut} = ultimate tensile strength of cylinder material (N/mm^2) (fs) = factor of safety	
Clavarino's equation	
$t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma + (1-2\mu)P_i}{\sigma - (1+\mu)P_i}} - 1\right]$	(26.21)
μ = Poisson's ratio $\sigma = \frac{S_{yt}}{(fs)}$ σ = permissible tensile stress (N/mm^2) S_{yt} = yield strength of cylinder material (N/mm^2) (fs) = factor of safety	
Birnie's equation	
$t = \frac{D_i}{2}\left[\sqrt{\frac{\sigma + (1-\mu)P_i}{\sigma - (1+\mu)P_i}} - 1\right]$	(26.22)
Barlow's equation	
$t = \frac{P_i D_o}{2\sigma_t}$	(26.23)
D_o = outer diameter of cylinder (mm)	

Note: (i) Lame's equation is applicable to cylinders made of brittle materials.
(ii) Clavarino's equation is applicable to cylinders with closed ends and made of ductile materials.
(iii) Birnie's equation is applicable to open cylinders made of ductile material.
(iv) Barlow's equation is applicable to high-pressure oil and gas pipes.

Table 26.6 *Compound cylinder*



$$\delta_j = \frac{D_2 P}{E}\left[\frac{(D_3^2 + D_2^2)}{(D_3^2 - D_2^2)} + \mu\right] \tag{26.24}$$

$$\delta_c = \frac{D_2 P}{E}\left[\frac{(D_2^2 + D_1^2)}{(D_2^2 - D_1^2)} - \mu\right] \tag{26.25}$$

δ_j = increase in inner diameter of jacket (mm)
δ_c = decrease in outer diameter of cylinder (mm)
D_1 = inner diameter of cylinder (mm)
D_2 = outer diameter of cylinder or inner diameter of jacket (mm)
D_3 = outer diameter of jacket (mm)
P = shrinkage pressure (N/mm^2)
E = modulus of elasticity (N/mm^2)
μ = Poisson's ratio

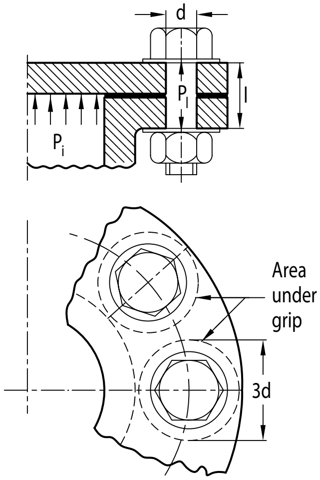
$$\delta = \delta_j + \delta_c \tag{26.26}$$

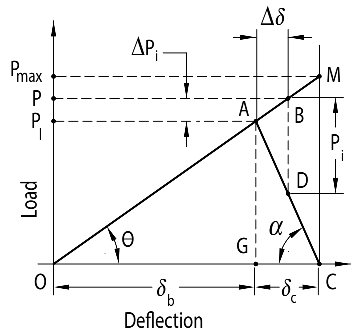
$$\delta = \frac{PD_2}{E}\left[\frac{2D_2^2(D_3^2 - D_1^2)}{(D_3^2 - D_2^2)(D_2^2 - D_1^2)}\right] \tag{26.27}$$

δ = total deformation or the amount of interference (mm)

26.3 GASKETED JOINT – THEORETICAL APPROACH

Table 26.7 *Gasketed joint for cylinder cover*





Stiffness of bolt (k_b)

$$k_b = \left(\frac{\pi}{4} d^2\right) \frac{E}{l} \tag{26.28}$$

k_b = stiffness of bolt (N/mm)
d = nominal diameter of bolt (mm)
l = total thickness of parts held together by bolt (mm)
E = modulus of elasticity of bolt material (N/mm^2)

Combined stiffness of cylinder cover, cylinder flange and gasket (k_c)

$$\frac{1}{k_c} = \frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_g} \tag{26.29}$$

k_1 = stiffness of cylinder cover (N/mm)
k_2 = stiffness of cylinder flange (N/mm)
k_g = stiffness of gasket (N/mm)

Approximate stiffness of cylinder cover, cylinder flange or gasket

Assumption: A hollow circular area of (3d) and (d) as outer and inner diameters respectively is under the grip of the bolt. Then,

$$k = \frac{2\pi d^2 E}{t} \tag{26.30}$$

t = thickness of the member under compression (mm)

Bolt load	
$$P = P_l + \Delta P_i$$	(26.31)
$$\Delta P_i = P_i \left[\frac{k_b}{k_b + k_c} \right]$$	(26.32)
P = resultant load on bolt (N) P_l = initial preload due to bolt tightening (N) P_i = external load due to pressure inside cylinder (N)	
Capacity of cylinder to bear the load	
$$P_{\max} = P_l \left[\frac{k_b + k_c}{k_b} \right]$$	(26.33)
$P_{\max}$ = maximum load (force) inside the cylinder when the joint is on the verge of opening or capacity of cylinder to bear the load (N)	

Table 26.8 *Gasket materials – Modulus of elasticity*

Material	Modulus of elasticity (E) (MPa or N/mm^2)
Cork	86
Compressed asbestos	480
Copper-asbestos	93 000
Copper (pure)	121 000
Plain rubber	69
Spiral wound	280
Teflon	240
Vegetable fiber	120

Table 26.9 *Combined stiffness of cylinder cover or cylinder flange (k_m)*

(Approximate empirical equation by Wileman) (Alternative method)

$$k_m = dEAe^{b(d/l)}$$	(26.34)
k_m = combined stiffness of cylinder cover or cylinder flange or material stiffness (N/mm) d = nominal diameter of bolt (mm) E = modulus of elasticity of bolt material (N/mm^2) l = clamped length (mm) A, b = parameters (Table 26.10)	

Table 26.10 *Stiffness parameters A and b*

Material	Poisson's ratio (μ)	Modulus of elasticity (MPa or N/mm^2)	A	b
Steel	0.291	206.8×10^3	0.78715	0.62873
Aluminium	0.334	71×10^3	0.79670	0.63816
Copper	0.326	118.6×10^3	0.79568	0.63553
Grey cast iron	0.211	100×10^3	0.77871	0.61616
General expression (averaging all four tested materials)			0.78952	0.62914

Table 26.11 *Shapes of cylinder gaskets*

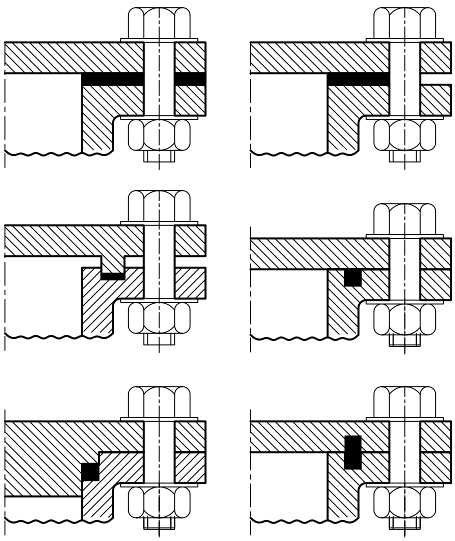


Table 26.12 *Selection of gasket material based on temperature*

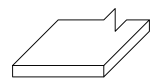
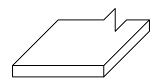
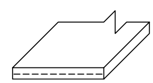
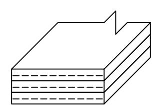
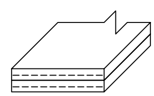
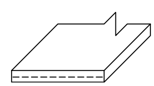
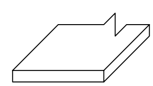
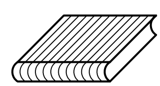
Temperature range	Gasket material
Up to 150°C	Cellulose, asbestos compositions, cork or conventional elastomers
150°C to 260°C	Asbestos compositions or special elastomers
Above 260°C	Metal- reinforced asbestos compositions or metallic constructions

Table 26.13 *Selection of gasket material based on internal pressure*

Pressure range	Gasket material
0 – 0.7 N/mm^2	Low density cellulose or asbestos compositions
0.7 – 3.5 N/mm^2	Medium to high density cellulose or asbestos compositions
3.5 – 7.0 N/mm^2	High density compressed asbestos
Above 7.0 N/mm^2	Metal- reinforced asbestos composites or metallic constructions

26.4 GASKET DESIGN AS PER PRESSURE VESSEL CODE

Table 26.14 *Gasket materials, gasket factor (m) and minimum design seating stress (y)*

Gasket materials	Gasket factor (*m*)	Minimum design seating stress (N/mm^2)	Sketches
Rubber without fabric or a high percentage of asbestos fibre: (i) Below 75 Shore Durometer (ii) 75A or higher Shore Durometer	 0.50 1.00	 0 1.4	
Asbestos with a suitable binder for the operating conditions: (i) 3.2 mm thick (ii) 1.6 mm thick (iii) 0.8 mm thick	 2.00 2.75 3.50	 11.0 26.0 45.0	
Rubber with cotton fabric insertion	1.25	2.8	
Rubber with asbestos fabric insertion, with or without wire reinforcement: (i) 3-ply (ii) 2-ply (iii) 1-ply	 2.25 2.50 2.75	 15.0 20.0 26.0	  
Vegetable fibre	1.75	7.6	
Spiral-wound metal, asbestos filled: (i) Carbon steel (ii) Stainless steel or monel metal	 2.50 3.00	 20.0 31.0	

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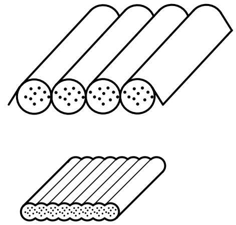
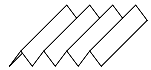
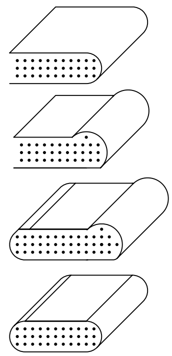
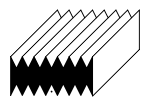
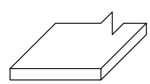
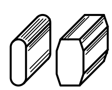
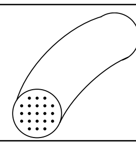
Corrugated metal, asbestos inserted or			
Corrugated metal, jacketed asbestos filled:			
(i) Soft aluminium	2.50	20.0	
(ii) Soft copper or brass	2.75	26.0	
(iii) Iron or soft steel	3.00	31.0	
(iv) Monel metal or 4-6 percent chrome steel	3.25	38.0	
(v) Stainless steels	3.50	45.0	
Corrugated metal:			
(i) Soft aluminium	2.75	26.0	
(ii) Soft copper or brass	3.00	31.0	
(iii) Iron or soft steel	3.25	38.0	
(iv) Monel metal or 4–6 percent chrome steel	3.50	45.0	
(v) Stainless steels	3.75	52.0	
Flat metal jacketed asbestos filled:			
(i) Soft aluminium	3.25	38.0	
(ii) Soft copper or brass	3.50	45.0	
(iii) Iron or soft steel	3.75	52.0	
(iv) Monel metal or 4–6 percent chrome steel	3.50	55.0	
(v) Stainless steels	3.75	62.0	
Grooved metal:			
(i) Soft aluminium	3.25	38.0	
(ii) Soft copper or brass	3.50	45.0	
(iii) Iron or soft steel	3.75	52.0	
(iv) Monel metal or 4–6 percent chrome steel	3.75	62.0	
(v) Stainless steels	4.25	70.0	
Solid flat metal:			
(i) Soft aluminium	4.0	61.0	
(ii) Soft copper or brass	4.75	90.0	
(iii) Iron or soft steel	5.50	124.0	
(iv) Monel metal or 4–6 percent chrome steel	6.00	150.0	
(v) Stainless steels	6.50	180.0	
Ring joint:			
(i) Iron or soft steel	5.50	124.0	
(ii) Monel metal or 4–6 percent chrome steel	6.00	150.0	
(iii) Stainless steels	6.50	180.0	
Rubber O-rings:			
(i) Below 75 IRHD	3	0.70	
(ii) 75 to 85 IRHD	6	1.50	

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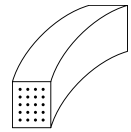
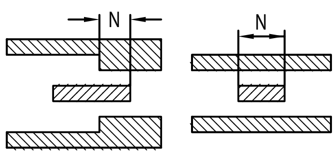
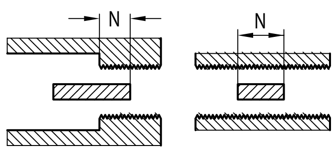
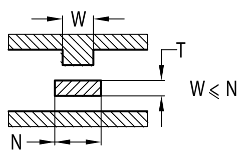
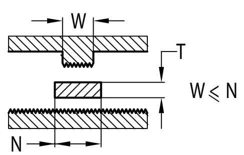
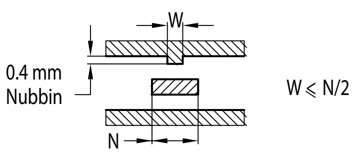
Rubber square section rings: (i) Below 75 IRHD (ii) 75 to 85 IRHD	 4 9	 1.0 2.8	

Note: (i) The minimum design seating stress (y) is the initial compressive stress in N/mm^2 on the contact area of the gasket that is required to provide a seal and prevent leaks in the joint as the system is pressurised.
(ii) The gasket factor or gasket maintenance factor (m) provides the additional preload needed in the flange fasteners to maintain the compressive load on the gasket after internal pressure is applied to the joint.

Table 26.15 *Basic seating width of gasket (b_o)*

Type	Flange and gasket diagram	Basic seating width of gasket (b_o)	
		Column I (Solid flat metal and ring joint gaskets)	**Column II (Spiral wound, metal jacketed, corrugated metal, grooved metal gaskets)**
1a		$\left(\frac{N}{2}\right)$	$\left(\frac{N}{2}\right)$
1b*			
1c		$\left(\frac{W+T}{2}\right)$, $\left(\frac{W+N}{4}\text{max}\right)$	$\left(\frac{W+T}{2}\right)$, $\left(\frac{W+N}{4}\text{max}\right)$
1d*			
2		$\left(\frac{W+N}{4}\right)$	$\left(\frac{W+3N}{8}\right)$

(*Contd.*)

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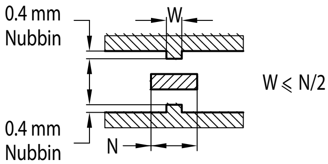
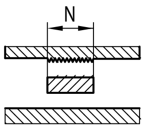
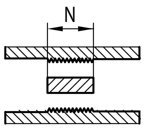
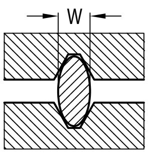
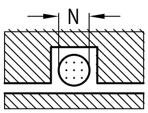
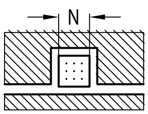
3		$\left(\frac{N}{4}\right)$	$\left(\frac{3N}{8}\right)$
4*		$\left(\frac{3N}{8}\right)$	$\left(\frac{7N}{16}\right)$
5*		$\left(\frac{N}{4}\right)$	$\left(\frac{3N}{8}\right)$
6		$\left(\frac{W}{8}\right)$	—
7		—	$\left(\frac{N}{2}\right)$
8		—	$\left(\frac{N}{2}\right)$

Note: (*) Where serrations do not exceed 0.4 mm depth and 0.8 mm width spacing, choose 1b or 1d.
Notations:

N = width of gasket (mm)
W = width of contact area (raised face or serrations) (mm)
T = thickness of gasket (mm)
b_o = basic seating width of gasket (mm)

Table 26.16 *Effective seating width of gasket (b) and location of gasket load reaction (G)*

Effective seating width of gasket (*b*)	
$b = 2.5\sqrt{b_o}$ when $b_o > 6.3$ mm (26.35)	$b = b_o$ when $b_o \le 6.3$ mm (26.36)
b_o = basic seating width of gasket (mm) (Table 26.15) b = effective seating width of gasket (mm)	

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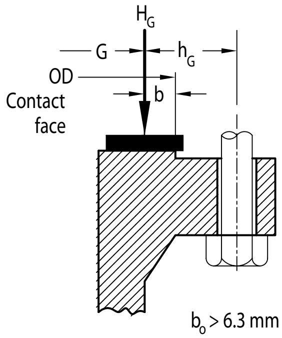	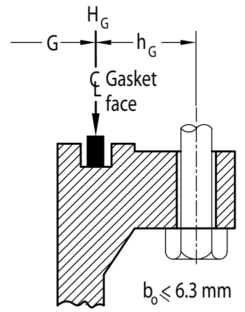
H_G = gasket load (N) G = diameter at location of gasket load reaction (mm) h_G = radial distance from gasket load reaction to pitch circle of bolts (mm)	
Diameter at location of gasket load reaction (G)	
G = (outside diameter of gasket contact face – 2 b) (when $b_o > 6.3$ mm) (26.37)	G = mean diameter of gasket contact face (when $b_o \le 6.3$ mm) (26.38)
$$h_G = \left(\frac{C - G}{2}\right) \quad (26.39)$$ C = pitch circle diameter of bolts (mm)	

Table 26.17 *Required bolt load*

Total hydrostatic end force (H)
$$H = \left(\frac{\pi}{4} G^2\right) P_i \quad (26.40)$$
H = total hydrostatic end force (N) G = diameter at location of gasket load reaction (mm) (Table 26.16) P_i = design pressure (N/mm^2)
Total joint-contact-surface compressive load (H_p)
$$H_p = 2\pi b G m P_i \quad (26.41)$$
H_p = total joint-contact-surface compressive load (N) b = effective seating width of gasket (mm) (Table 26.16) m = gasket factor (Table 26.14)
Minimum required bolt load for operating conditions (W_{m1})
$$W_{m1} = H + H_p = \left(\frac{\pi}{4} G^2\right) P_i + 2\pi b G m P_i \quad (26.42)$$
W_{m1} = minimum required bolt load for operating conditions (N)

(*Contd.*)

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Minimum required initial bolt load for gasket seating (W_{m2})
$W_{m2} = \pi b G y$ (26.43)
W_{m2} = minimum required initial bolt load for gasket seating (N) y = minimum design seating stress (N/mm^2) (Table 26.14)

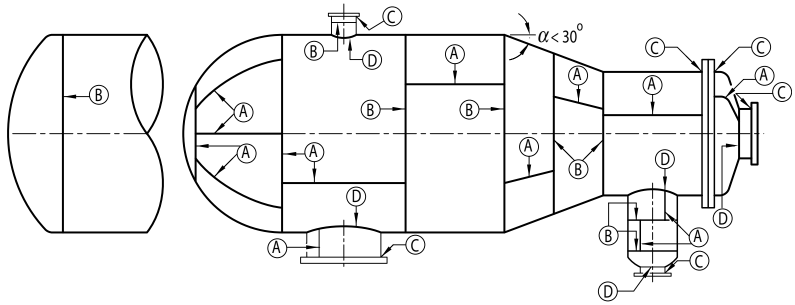
Note: The greater of the two bolt loads W_{m1} and W_{m2} is used to find out the required area of the bolts.

26.5 UNIFIRED PRESSURE VESSELS

Table 26.18 *Pressure vessels*

Pressure vessel
A pressure vessel is a vessel or a pipeline for carrying, storing or receiving steam, gases or liquids at pressures above the atmospheric pressure.
Fired pressure vessel
Fired pressure vessel (boiler) means closed vessel used to heat water or other liquids to generate steam or other vapors under pressure or vacuum by the application of heat resulting from combustion of fuels, electricity, atomic energy or waste gases.
Unfired pressure vessel
Unfired pressure vessel (pressure vessel) means an enclosed vessel in which pressure is obtained from an external source, or by applying heat from indirect source or from a direct source other than boilers.
Maximum working pressure
The maximum working pressure is the maximum gauge pressure, at the co-incident metal temperature, that is permitted for the vessel in operation. It is determined by the technical requirements of the process.
Design pressure
The design pressure (P) is the pressure that is used in design calculations for such quantities as the shell thickness and also in the design of other attachments like nozzles and openings. The design pressure is obtained by adding a minimum 5% to the maximum working pressure.
Hydrostatic test pressure
The hydrostatic test pressure is the maximum pressure to which the vessel is subjected to, during the final hydrostatic test. The hydrostatic test pressure is taken as 1.3 times the design pressure.
Joint factor or Weld joint efficiency factor (J)
Joint factor or weld joint efficiency factor (J) is the ratio of an arbitrary strength of the welded joint to the strength of the plates expressed as a decimal. The magnitude of weld joint efficiency depends upon two factors—the type of weld and the method of weld inspection.
Corrosion allowance (CA)
Corrosion allowance (CA) is additional metal thickness over and above that required to withstand the internal pressure, to compensate for the corrosion expected during the lifetime of the vessel. For carbon and low alloy steels, where severe corrosion is not expected, a minimum corrosion allowance of 2 mm should be used. Where more severe conditions are anticipated, the corrosion allowance is increased to 4 mm. Most design codes specify a minimum corrosion allowance of 1 mm.

Table 26.19 *Categories of welded joints in pressure vessels*



Category A	(i) Longitudinal welded joints in the main shell (ii) Longitudinal welded joints in the communicating chambers, transitions in diameter and nozzles (iii) Any welded joint in a formed or flat head
Category B	(i) Circumferential welded joints in the main shell (ii) Circumferential welded joints in the communicating chambers and nozzles (iii) Circumferential welded joints between the transitions in diameter and a cylinder at either the large or small end (iv) Circumferential welded joints connecting formed heads to main shell
Category C	(i) Welded joints connecting flanges and flat heads to the main shell (ii) Welded joints connecting tube sheets to the main shell (iii) Any welded joints connecting one side plate to another side plate of flat sided vessels
Category D	(i) Welded joints connecting communicating chambers and nozzles to the main shell (ii) Welded joints connecting communicating chambers and nozzles to flat sided vessels

Note: (i) The term category defines only the location of welded joint in the pressure vessel and never implies the type of welded joint.
(ii) Communicating chamber is an appurtenance to the vessel which intersects the shell or head of the vessel and forms an integral part of the pressure containing enclosure.

Table 26.20 *Classification of pressure vessels*

Class 1 vessels
Class 1 pressure vessels are the vessels that are used to contain lethal and toxic substances. Also, Class 1 pressure vessels are the vessels that are designed for operation below –20°C. Lethal substances means poisonous gases and liquids of such a nature that a very small amount of gas or vapour mixed or unmixed with air is dangerous to human life when inhaled, e.g., hydrocyanic acid, carbonyl chloride, cyanogens, mustard gas and xylyl bromide. Liquefied petroleum gas (such as propane, butane or butadiene), natural gas and vapours of any other petroleum product are not classified as a lethal substance. Any material that is allowed in the code is used for construction of theses vessels. There are two types of welded joints used in these vessels-double welded butt joint with full penetration and single welded butt joint with backing strip. Welded joints of Class 1 pressure vessels are fully radiographed.

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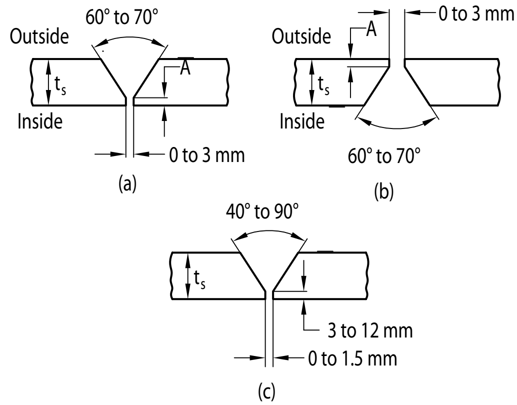
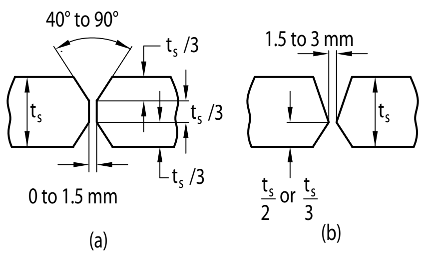
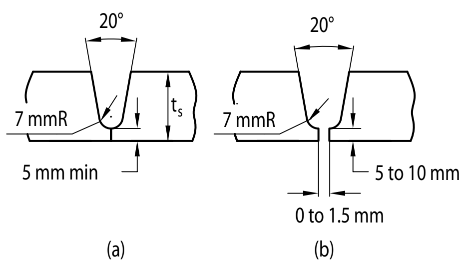
Class 2 vessels
Class 2 pressure vessels are the vessels that do not fall under Class 1 or Class 3 types. The maximum thickness of the main shell in this case is limited to 38 mm after adding corrosion allowance. Any material that is allowed in the code is used for construction of theses vessels. There are two types of welded joints used in these vessels—double welded butt joint with full penetration and single welded butt joint with backing strip. However, in Class 2 pressure vessels, the welded joints are spot radiographed.
Class 3 vessels
Class 3 pressure vessels are the vessels for relatively light duties, having plate thickness not in excess of 16 mm and built for working pressure not exceeding 3.5 kgf/cm^2 (1 kgf = 9.81 N) vapour pressure or 17.5 kgf/cm^2 hydrostatic design pressure. They are not recommended for service when the operating temperature is less than 0°C or more than 250°C. They are made from carbon and low alloy steels. Welded joints in Class 3 pressure vessels are not radiographed.

Table 26.21 *Joint factor or weld joint efficiency factor (J)*

Requirement	Class 1	Class 2	Class 3		
Joint factor or weld joint efficiency factor (J)	1.00	0.85	0.70	0.60	0.50
Material	Any material allowed	Any material allowed	Carbon and low alloy steels	Carbon and low alloy steels	Carbon and low alloy steels
Radiography	Fully radiographed	Spot radiographed	No radiography	No radiography	No radiography
Type of joints	(i) Double welded butt joints with full penetration excluding butt joints with metal backing strips which remain in place (ii) Single welded butt joints with backing strip (J = 0.9).	(i) Double welded butt joints with full penetration excluding butt joints with metal backing strips which remain in place (ii) Single welded butt joints with backing strip (J = 0.8).	(i) Double welded butt joints with full penetration excluding butt joints with metal backing strips which remain in place (ii) Single welded butt joints with backing strip (J = 0.65).	(i) Single welded butt joints with backing strip not over 16 mm thickness or over 600 mm outside diameter (ii) Single welded butt joints without backing strip (J = 0.55).	(i) Single full fillet lap joints for circumferential seams only

Table 26.22 *Unfired pressure vessels—Acceptable types of Butt welded joints*

(i) Double-welded Butt joint with single '*V*'

Application – Longitudinal and circumferential butt welds in plates from 5 to 20 mm thickness. The '*V*' should be on the inside of small diameter vessels as shown in (b).
(ii) Double-welded Butt joint with double '*V*'

Application – Longitudinal and circumferential butt welds where the plate thickness is more than 20 mm. Each side is welded in several layers.
(iii) Double-welded Butt joint with single '*U*'

Application – Longitudinal and circumferential butt welds where the plate thickness is more than 20 mm. Each side is welded in several layers.

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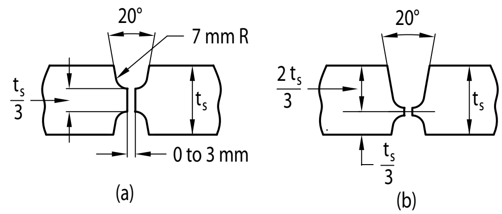
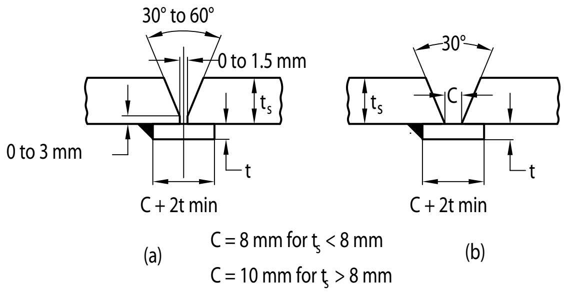
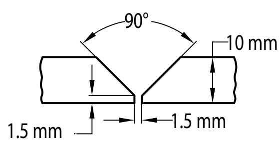
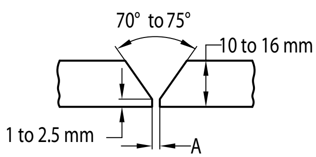
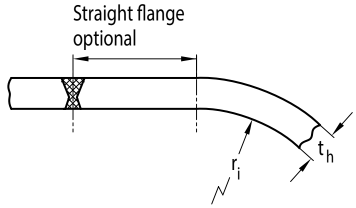
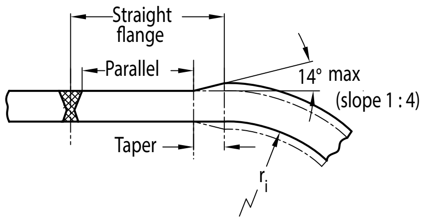
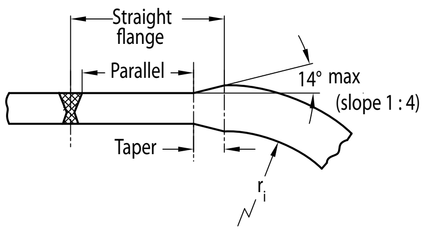
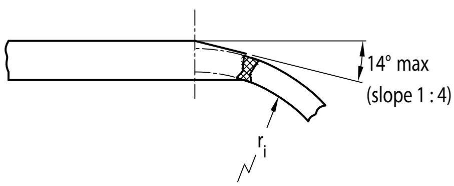
(iv) Double-welded Butt joint with double '*U*'

Application – Longitudinal and circumferential butt welds where the plate thickness is more than 20 mm. Each side is welded in several layers.
(v) Single-welded Butt joint with backing strip

Application – The joint is welded with backing bar in several layers. The device is essential to prevent slag or powder running through welding.
(vi) Single-welded Butt joint with '*V*' groove without backing strip

Application – The joint is used for plates with thickness not exceeding 10 mm.
(vii) Single-welded Butt joint with '*V*' groove without backing strip

A = 1.5 mm for thickness < 10 mm A = 3.0 mm for thickness >10 mm
Application – The joint is used for plates with thickness not exceeding 16 mm.

Table 26.23 *Unfired pressure vessels—Welded joints for end connections*









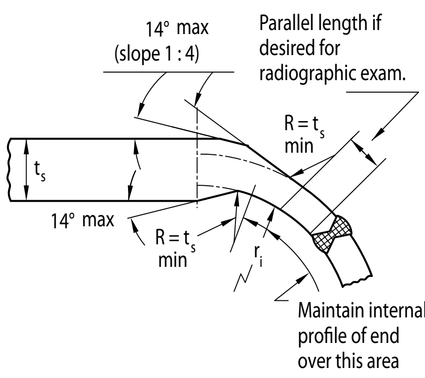


Table 26.24 *Allowable stresses for carbon and low alloy steels for pressure vessels (at room temperature) (f)*

Material designation	**Mechanical Properties**			**Allowable stress up to 50°C (*f*) (N/mm^2)**
	Tensile strength (Min.) (N/mm^2) R_{20}	**Yield stress (Min.) (N/mm^2)**	**Percentage elongation (%)**	
Plates				
IS 2002 – Grade I	362	0.55 R_{20}	26	120
IS 2002 – Grade 2A	412	0.50 R_{20}	25	137
IS 2002 – Grade 2B	510	0.50 R_{20}	20	169
IS 2041 – 20Mo55	470	274	20	156
IS 2041 – 20Mn2	510	294	20	169
IS1570 – 15Cr90Mo55	490	294	20	162
IS1570 – C15Mn75	412	225	25	137
Forgings				
IS 2004 – Class 1	362	0.50 R_{20}	–	120
IS 2004 – Class 2	431	0.50 R_{20}	15	143
IS 2004 – Class 3	490	0.50 R_{20}	21	162
IS 2004 – Class 4	618	0.50 R_{20}	15	206
IS1570 – 20Mo55	470	274	20	156
IS2611 – 15Cr90Mo55	490	294	20	162
IS1570 – 10Cr2Mo1	490	313	20	162
Tubes and pipes				
IS 3609 – 1%Cr-0.5%Mo Tube Normalized and tempered	431	235	–	143
IS 3609 – 2.5%Cr-1%Mo Tube Normalized and tempered	480	245	–	159
IS1570 – 20Mo55	451	245	25	150
IS1914 – Steel with Min. Tensile strength of $32 kgf/mm^2$	313	0.50 R_{20}	–	103
IS1914 – Steel with Min. Tensile strength of $43 kgf/mm^2$	421	0.50 R_{20}	–	140
IS2416 – Steel with Min. Tensile strength of 32 kgf/mm^2	313	0.50 R_{20}	–	103
IS1978 – St 18	309	172	–	103

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IS1978 – St 20	330	193	–	109
IS1978 – St 21	330	206	–	109
IS1978 – St 25	413	241	–	137
IS1979 – St 30	413	289	–	137
IS1979 – St 32	434	316	–	144
IS1979 – St 37	455	359	–	151
Sections, plates and bars				
IS226 – St 42-S	412	235	23	137
IS961 – St 55-HTW	490	284	20	162
IS2062 – St 42-W	412	225	23	137
IS3503 – Grade 1	362	0.50 R_{20}	26	120
IS3503 – Grade 2	412	0.50 R_{20}	25	137
IS3503 – Grade 3	431	0.50 R_{20}	23	143
IS3503 – Grade 4	461	0.50 R_{20}	22	153
IS3503 – Grade 5	490	0.50 R_{20}	21	162
IS3945 – Grade A-N	431	235	23	143
IS3945 – Grade B-N	490	279	20	162

Note: Some of the designations of materials in above table are changed. The information of these designations is given in Table 26.26.

Table 26.25 *Allowable stresses for carbon and low alloy steels for pressure vessels (at elevated temperatures) (f)*

Material designation	**Allowable stresses at Design temperature °C (*f*) (N/mm^2)**								
	Up to 50°C	**Up to 100°C**	**Up to 150°C**	**Up to 200°C**	**Up to 250°C**	**Up to 300°C**	**Up to 350°C**	**Up to 375°C**	**Up to 400°C**
Plates									
IS 2002 – Grade I	120	120	112	102	93	85	76	73	70
IS 2002 – Grade 2A	137	126	116	105	96	88	79	75	72
IS 2002 – Grade 2B	169	155	144	130	118	108	98	93	81
IS 2041 – 20Mo55	156	156	156	150	140	129	120	116	112
IS 2041 – 20Mn2	169	169	166	151	137	125	113	107	81
IS1570 – 15Cr90Mo55	162	162	162	162	156	149	141	135	131
IS1570 – C15Mn75	137	137	127	115	104	96	87	82	79

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Forgings									
IS 2004 – Class 1	120	110	102	93	84	77	69	66	63
IS 2004 – Class 2	143	131	121	110	100	91	83	78	75
IS 2004 – Class 3	162	150	138	125	114	104	94	89	81
IS 2004 – Class 4	206	189	174	157	144	131	119	112	81
IS1570 –20Mo55	156	156	156	150	140	129	120	116	112
IS2611 – 15Cr90Mo55	162	162	162	162	156	149	141	135	131
IS1570 – 10Cr2Mo1	162	162	162	162	162	162	160	157	154
Tubes and Pipes									
IS 3609 – 1%Cr-0.5%Mo Tube Normalized and tempered	143	143	139	133	125	118	112	108	104
IS 3609 – 2.5%Cr-1%Mo Tube Normalized and tempered	159	153	147	142	137	132	125	123	120
IS1570 – 20Mo55	150	150	143	133	125	115	107	103	101
IS1914 – Steel with Min. Tensile strength of 32kgf/mm^2	103	96	88	80	72	66	60	56	54
IS1914 – Steel with Min. Tensile strength of 43kgf/mm^2	140	128	119	107	98	90	81	77	74
IS2416 – Steel with Min. Tensile strength of 32 kgf/mm^2	103	96	88	80	72	66	60	56	54
IS1978 – St 18	103	103	97	88	80	73	65	62	60
IS1978 – St 20	109	109	109	99	90	82	74	70	67
IS1978 – St 21	109	109	109	105	96	88	79	75	72
IS1978 – St 25	137	137	136	123	112	103	93	88	81
IS1979 – St 30	137	137	137	137	135	123	112	105	81
IS1979 – St 32	144	144	144	144	144	135	122	115	81
IS1979 – St 37	151	151	151	151	151	151	138	131	81
Sections, plates and bars									
IS226 – St 42-S	137	137	133	120	96	88	79	-	-
IS961 – St 55-HTW	162	162	160	145	114	104	94	-	-
IS2062 – St 42-W	137	137	127	115	96	88	79	-	-
IS3503 – Grade 1	120	110	102	93	84	77	69	66	63
IS3503 – Grade 2	137	126	116	105	96	88	79	75	72
IS3503 – Grade 3	143	131	121	110	100	91	83	78	75

(*Contd.*)

(*Contd.*)

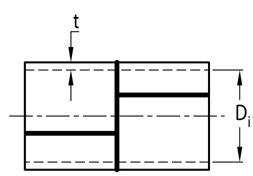
IS3503 – Grade 4	153	141	130	118	107	98	89	81	81
IS3503 – Grade 5	162	150	138	125	114	104	94	89	81
IS3945 – Grade A-N	143	143	133	120	96	88	79	–	–
IS3945 – Grade B-N	162	162	157	143	114	104	94	–	–

Note: Some of the designations of materials in above table have been changed. The information of these designations is given in Table 26.26.

Table 26.26 *Meaning of designations of materials in Tables 26.24 and 26.25*

20Mo55	Steel with average 0.2% Carbon and 0.55% Molybdenum
20Mn2	Steel with average 0.2% Carbon and 1.5% Manganese
15Cr90Mo55	Steel with average 0.15% Carbon, 0.90% Chromium and 0.55% Molybdenum
C15Mn75	Steel with average 0.15% Carbon and 0.75% Manganese (New Designation – 15C8)
10Cr2Mo1	Steel with maximum 0.15% Carbon, 2% Chromium and 1 % Molybdenum
St18	Steel with minimum yield stress of 17.6 kgf/mm^2
St20	Steel with minimum yield stress of 19.7 kgf/mm^2
St21	Steel with minimum yield stress of 21.1 kgf/mm^2
St25	Steel with minimum yield stress of 24.6 kgf/mm^2
St30	Steel with minimum yield stress of 29.5 kgf/mm^2 (New designation – Fe290)
St32	Steel with minimum yield stress of 32.3 kgf/mm^2 (New designation – Fe310)
St37	Steel with minimum yield stress of 36.6 kgf/mm^2 (New designation – Fe360)
St42	Steel with minimum tensile strength of 42 kgf/mm^2 (New designation – Fe410)

Table 26.27 *Thickness of pressure vessels shell*



Cylindrical shell subjected to Internal pressure

$$t = \frac{P_i D_i}{2fJ - P_i} + CA \tag{26.44}$$

$$t = \frac{P_i D_0}{2fJ + P_i} + CA \tag{26.45}$$

(*Contd.*)

(*Contd.*)

Spherical shell subjected to internal pressure

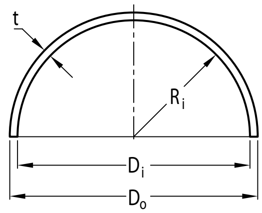
$$t = \frac{P_i D_i}{4 f J - P_i} + CA \qquad (26.46)$$

$$t = \frac{P_i D_0}{4 f J + P_i} + CA \qquad (26.47)$$

t = minimum thickness of shell plates (mm)
P_i = design (internal) pressure (N/mm^2)
D_i = inside diameter of shell (mm)
D_o = outside diameter of shell (mm)
f = allowable stress for plate material (N/mm^2) (Tables 26.24 and 26.25)
J = joint factor or weld joint efficiency factor (Table 26.21)
CA = corrosion allowance (mm)

Table 26.28 *Design of domed heads*

Hemispherical head

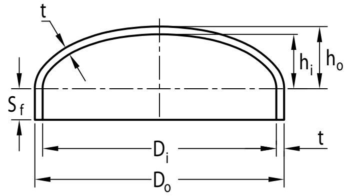


$$t = \frac{P_i R_i}{2 f J - 0.2 P_i} + CA \text{ or}$$

thickness of hemispherical head = 0.6 (thickness of cylindrical shell) (26.48)

t = minimum thickness of hemispherical head (mm)
P_i = design (internal) pressure (N/mm^2)
R_i = inside radius of head (mm)
f = allowable stress for plate material (N/mm^2) (Tables 26.24 and 26.25)
J = joint factor or weld joint efficiency factor (Table 26.21)
CA = corrosion allowance (mm)

Ellipsoidal head



(*Contd.*)

(*Contd.*)

$$t = \frac{P_i D_i}{2fJ - 0.2P_i} + CA \quad (26.49)$$

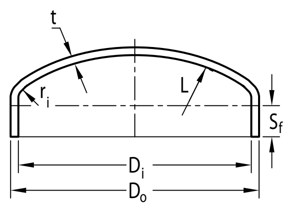
In ellipsoidal head, the ratio of the major axis to the minor axis is taken as 2:1. or

$$h_i = \frac{D_i}{4} \quad (26.50)$$

$$S_f = 3\ t \text{ or } 20 \text{ mm (whichever is more)} \quad (26.51)$$

D_i = inside diameter of head (mm)
h_i = inside height of head (mm)
S_f = length of straight flange (mm)

Torispherical head



$$t = \frac{0.885 P_i L}{fJ - 0.1P_i} + CA \quad (26.52)$$

$$S_f = 3\ t \text{ or } 20 \text{ mm (whichever is more)} \quad (26.53)$$

The knuckle radius r_i is taken as 6% of the crown radius.

$$r_i = 0.06\ L \quad (26.54)$$

The crown radius L should not be greater than the outside diameter of the cylindrical shell. Therefore, $L < D_o$

L = inside crown radius of head (mm)
r_i = inside knuckle (corner) radius of head (mm)

Torispherical head (Non-standard)

$$C_s = \frac{1}{4}\left[3 + \sqrt{\frac{L}{r_i}}\right] \quad (26.55)$$

$$t = \frac{P_i L C_s}{2fJ - 0.2P_i} + CA \quad (26.56)$$

C_s = stress concentration factor for torispherical head

Note: There are two common types of constructions for torispherical heads – Kloepper head and Korbbogen head

Torispherical head (Kloepper head)

$$L = D_o \text{ and } r_i = 0.1\ D_o \quad (26.57)$$

Also, $$S_f \geq 3.5t \quad (26.58)$$

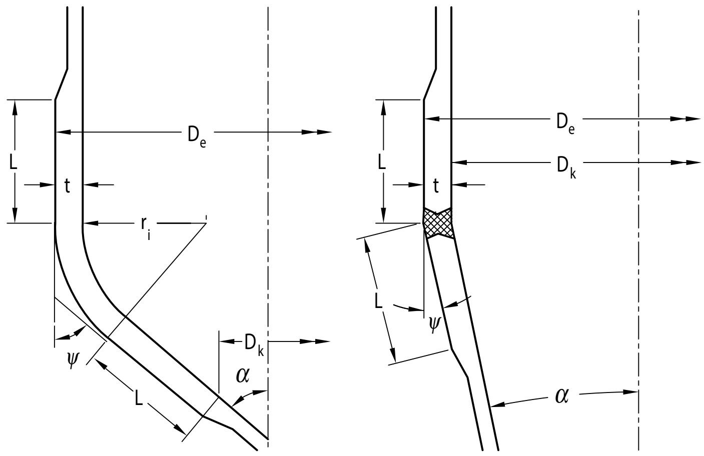
(*Contd.*)

(*Contd.*)

Torispherical head (Korbbogen head)		
	$L = 0.8\ D_o$ and $r_i = 0.154\ D_o$	(26.59)
Also,	$S_f \geq 3t$	(26.60)

Note: The end cap on a cylindrically shaped pressure vessel is called a head. Hemispherical, ellipsoidal and torispherical heads are collectively referred to as domed heads. Torispherical heads are often called dished heads or shallow dished heads.

Table 26.29 *Design of conical heads or conical reducing sections*



(a) Cone with knuckle (b) Cone without knuckle

Notations
t = minimum thickness of conical head or section (mm)
D_e = outside diameter of conical head or section (mm)
D_k = inside diameter of conical head or section at the position under consideration (mm)
r_i = inside radius of transition knuckle (mm)
L = length of thicker section (mm)
α = half of cone apex angle (°)
ψ = difference between angle of slope of two adjoining conical sections (°)
C = factor taking into account the stress in knuckle (Table 26.30)
P_i = design (internal) pressure (N/mm^2)
f = allowable stress for plate material (N/mm^2) (Tables 26.24 and 26.25)
J = joint factor or weld joint efficiency factor (Table 26.21)
CA = corrosion allowance (mm)

(*Contd.*)

(*Contd.*)

Inside radius of transition knuckle (r_i)	
$r_i = 0.01D_k$ (for conical section without knuckle transition)	(26.61)
Length of thicker section (L)	
$L = 0.5\sqrt{\dfrac{D_e \cdot t}{\cos\psi}}$	(26.62)
Thickness of conical head or section (t)	
The thickness of conical head or section within distance L from the junction is given by, $t = \dfrac{P_i D_e C}{2fJ - P_i} + CA$	(26.63)
The thickness of conical head or section not less than a distance L away from the junction is given by, $t = \dfrac{P_i D_k}{2fJ - P_i} \times \dfrac{1}{\cos\alpha} + CA$	(26.64)

Note: Conical ends can be constructed of several ring sections each with decreasing diameter and corresponding decreasing thickness.

Table 26.30 *Values of C*

ψ	Ratio of (r_i/D_e)											
	0.01	0.02	0.03	0.04	0.06	0.08	0.10	0.15	0.20	0.30	0.40	0.50
	C											
10°	0.70	0.65	0.60	0.60	0.55	0.55	0.55	0.55	0.55	0.55	0.55	0.55
20°	1.00	0.90	0.85	0.80	0.70	0.65	0.60	0.55	0.55	0.55	0.55	0.55
30°	1.35	1.2	1.1	1.0	0.90	0.85	0.80	0.70	0.55	0.55	0.55	0.55
45°	2.05	1.85	1.65	1.5	1.3	1.2	1.1	0.95	0.90	0.70	0.55	0.55
60°	3.2	2.85	2.55	2.35	2.0	1.75	1.6	1.4	1.25	1.00	0.70	0.55
75°	6.8	5.85	5.35	4.75	3.85	3.5	3.15	2.7	2.4	1.55	1.0	0.55

Table 26.31 *Design of unstayed flat heads*

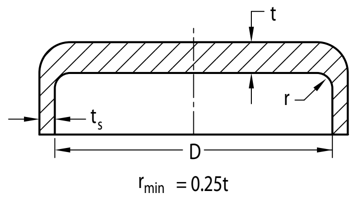
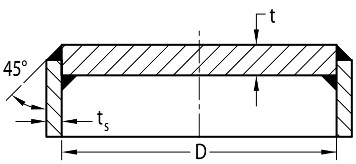
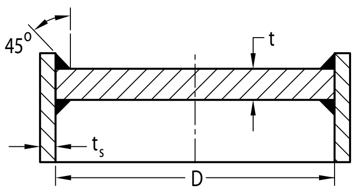
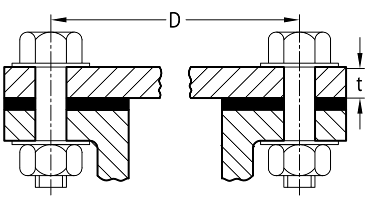
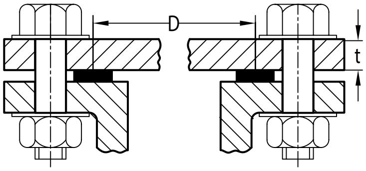
Notations
D = diameter or short span as shown in figures below (mm) t = minimum thickness of flat head or cover (mm) t_s = thickness of shell (mm) P_i = design (internal) pressure (N/mm^2) f = allowable stress for plate material (N/mm^2) (Tables 26.24 and 26.25) C = factor depending upon method of attachment of cover to shell or edge constraint

(*Contd.*)

(*Contd.*)

The thickness of flat unstayed circular heads and covers is given by,

$$t = CD\sqrt{\frac{P_i}{f}} + \text{CA} \tag{26.65}$$

Forged plate	
	$C = 0.5$
Welded plate	
	$C = 0.55$
Welded plate	
	$C = 0.55$
Bolted plate with full gasket	
	$C = 0.42$
Bolted plate with narrow faced gasket	
	$C = 0.55$

Materials Handling Equipment

27

27.1 CLASSIFICATION OF CRANES

Table 27.1 *Classification of cranes and hoists*

Class number	Working period	Effective load	Dynamic effect
Class 1	Short	Low	Low
Class 2	Long	Low	Low
	Short	High	Low
	Short	High	High
Class 3	Long	High	Low
	Long	Low	High
	Short	High	High
Class 4	Long	High	High

Note: (i) All applications used for raising or lowering persons irrespective of working period, effective load and dynamic effect should come under Class 4.

(ii) The working period of any crane or hoist is considered as short if it operates for less than 500 hours per year and long if it operates for more than 500 hours per year.

(iii) The effective load of any crane or hoist is considered as low unless it lifts loads greater than two-thirds of its safe working load on more than 1000 occasions per year. The effective load shall otherwise be considered as high.

(iv) In case of overhead traveling crane, the dynamic effect is considered low, if the speed of traveling of both the crab and the crane or hoist are each less than 100 m/min, or 130 m/min if the active surfaces of the respective track rails are uninterrupted by gaps or joints. Dynamic effect is considered high if the crane or the hoist is used for any purpose likely to produce greater shock effects than those caused by traveling on steel track rails at the aforesaid speeds.

(v) Dynamic effect is considered low for mobile cranes or mobile hoists having well-sprung road wheels and traveling at moderate speeds on surfaces not less regular than closely laid decking of sawn timber. Dynamic effect is considered high for other mobile cranes or mobile hoists.

Table 27.2 *Classification of electric overhead traveling (EOT) cranes*

Applications	Number of hours in service per year	Class	Number of cycles for fatigue calculations	Impact factor (applies in vertical plane)
Cranes for occasional use only, such as engine and power house cranes, hand and light power operated cranes	Up to 1000	Class 1 or Class 2	10^5	1.1
Medium-duty industrial cranes for intermittent use in stores and light machine shops, such as maintenance cranes, giant cranes, fixed and traveling gantries cranes and ice works cranes	Up to 2000	Class 2	6×10^5	1.3
Cranes for general use in factories, workshops and warehouses, such as heavy-duty industrial cranes for non-ferrous foundries, heavy engineering shops, stockyard, railway goods yards, light iron foundries, under-slug jib cranes, machine shop secondary cranes, and ship building cranes	2000 to 3000	Class 2	6×10^5	1.3
Cranes for steelworks service and light process cranes, heavy-duty foundry work, light magnet and grabbing duty, such as overhead traveling cranes not elsewhere included, and traveling gantry derrick cranes	Over 3000	Class 3	2×10^6	1.4
Continuous process cranes for steel-works such as continuous magnet work, continuous grabbing duty, and skull breaker cranes	Over 4000	Class 4	4×10^6	1.5

Note: The impact factor applied to the motion of hook in a vertical direction covers inertia forces including shock. The rated lifted load is multiplied by impact factor to find out the live loads in members of the structure. The impact factor does not apply to the dead weight of the crane.

Table 27.3 *Classification of electrically driven jib cranes mounted on a high pedestal or portal carriage*

Applications	Number of hours in service per year	Class	Number of cycles for fatigue calculations	Impact factor (applies in vertical plane)
Cranes for lifting occasional heavy loads but whose use at full load is infrequent, such as fitting-out cranes, tower and portal cranes, and hammer-headed cranes, cupola hoists for light and medium cranes, and over-braced or under-braced jib cranes	Up to 2000	Class 2	6×10^5	1.3
Cranes designed for general working of cargo	2000 to 3000	Class 2	6×10^5	1.3
Cranes designed for grabbing and magnet duties, handling full rated load for long periods, such as lifting magnets	Over 3000	Class 3	2×10^6	1.4

Table 27.4 *Classification of mobile power-driven cranes*

Applications	Number of hours in service per year	Class	Number of cycles for fatigue calculations	Impact factor (applies in vertical plane)
Cranes for ordinary duty, such as floating cranes, and stacking cranes	Up to 2000	Class 2	6×10^5	1.3
Cranes for severe duty as at decks, such as ship building cranes, and sprung mobile cranes (other than vehicular used in building and construction works)	2000 to 3000	Class 2	6×10^5	1.3
Cranes for severe duty dock cranes, such as unsprung mobile cranes, bach and front end loaders, and fork lift trucks	Over 3000	Class 3	2×10^6	1.4

Table 27.5 *Classification of traveling jib cranes (contractor's type)*

Applications	Number of hours in service per year	Class	Number of cycles for fatigue calculations	Impact factor (applies in vertical plane)
Ordinary duty cranes	2000 to 3000	Class 2	6×10^5	1.3
Severe duty (power) vehicular cranes used in building and constructional works	Over 3000	Class 3	2×10^6	1.4

Table 27.6 *Classification of Derrick cranes*

Applications	Number of hours in service per year	Class	Number of cycles for fatigue calculations	Impact factor (applies in vertical plane)
Hand operated cranes	Up to 1000	Class 1	10^5	1.1
Power-driven cranes for ordinary duty	Up to 3000	Class 2	6×10^5	1.3
Power-driven cranes for severe duty as for grabbing duties or at docks	Over 3000	Class 3	2×10^6	1.4

Table 27.7 *Factors for dynamic effects*

Class	Duty factor	Impact factor (applies in vertical plane)	Factor for horizontal Forces
Class 1	1.0	1.1	0.04
Class 2	0.95	1.3	0.05

(*Contd.*)

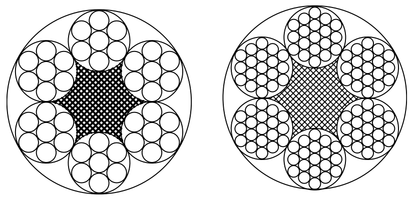
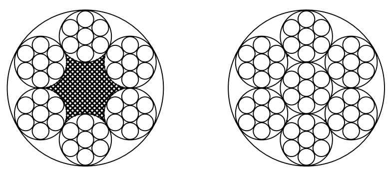
(*Contd.*)

Class 3	0.90	1.4	0.06
Class 4	0.85	1.5	0.08

Note: The forces acting on the crane or its parts are multiplied by the factors in above table in order to find out dynamic load.

27.2 WIRE ROPES

Table 27.8 *Steel wire ropes*

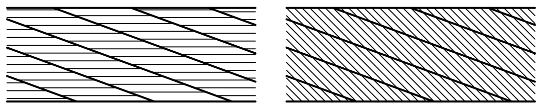
Construction of wire rope
 (a) 6 × 7(6/1) (b) 6 × 19(12/6/1)
The wire rope consists of number of strands, each strand comprising several steel wires. The number of wires in each strand is generally 7, 19 or 37, while the number of strands is usually six. The individual wires are first twisted into the strand and then the strands are twisted around a fibre or steel core.
Core of wire rope
 (a) Wire rope with fibre core (b) Wire rope with steel core
The central portion of the wire rope is called core. There are three types of cores-fibre, wire and synthetic material. The fibre core consists of natural fibres like sisal, hemp, jute or cotton. The fibre core is flexible and suitable for all conditions except when the rope is subjected to severe crushing e.g., when working under high load. The steel core consists of another strand of fairly soft wires with lower tensile strength. Wire core is used where the wire rope is subjected to severe heat or crushing conditions. Plastic cores are used in special purpose wire ropes. It can be plastic impregnated fibre core, plastic covered fibre core or solid plastic core
Specification of wire rope
The specification of wire ropes includes two numbers, such as 6 × 7 or 6 × 19. The first number indicates the number of strands in the wire rope, while the second gives the number of steel wires in each strand. The popular constructions of steel wire ropes are as follows, 6 × 7 (6/1) 6 × 19 (12/6/1) 6 × 37 (18/12/6/1)

(*Contd.*)

(*Contd.*)

Tensile designations of wire ropes	
The tensile designation of wires, such as 1570 or 1770, indicates the minimum tensile strength (in N/mm^2) of the individual wires used for making the wire rope.	
Tensile designation	**Range of tensile strength (N/mm^2)**
1230	1230–1620
1420	1420–1810
1570	1570–1960
1770	1770–2150
1960	1960–2340

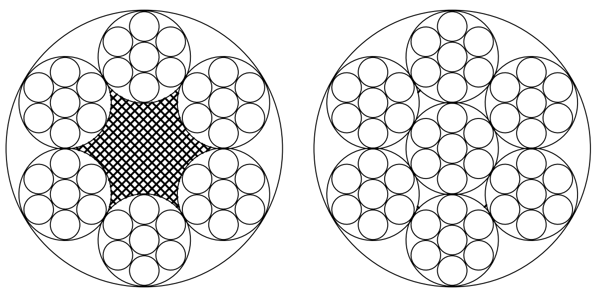
Lay of wire rope



(a) Regular lay (b) Lang's lay

The lay of wire rope refers to the manner in which the wires are helically laid into strands and the strands into the rope. If the wires in the strand are twisted in the same direction as the strands, then the rope is called a Lang's lay rope. When the wires in the strand are twisted in a direction opposite to that of the strands, the rope is said to be regular-lay or ordinary-lay.

Table 27.9 *Breaking load and mass for 6 × 7 (6/1) construction wire ropes*



(a) Wire rope with fibre core (b) Wire rope with steel core

Nominal diameter (mm) (d)	Approximate mass (kg/100 m)		Minimum breaking load corresponding to tensile designation of wires of (kN)					
			1570		1770		1960	
	Fibre core	Steel core	Fibre core	Steel core	Fibre core	Steel core	Fibre core	Steel core
8	22.9	25.2	33	36	38	41	42	45
9	28.9	31.8	42	46	48	51	53	57

(*Contd.*)

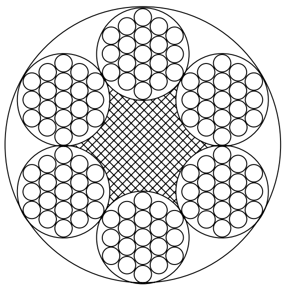
(*Contd.*)

10	35.7	39.1	52	56	59	64	65	70
11	43.2	47.6	63	68	71	77	79	85
12	51.5	56.6	75	81	85	91	94	101

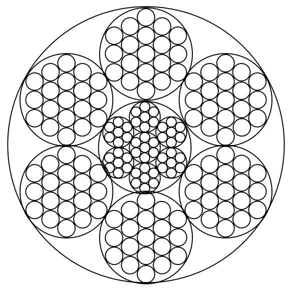
(ISO: 2408)

Note: The flexibility of the wire rope is an important consideration where sheaves are small or where the rope makes many bends. Flexibility in wire ropes is achieved by using a large number of small-diameter wires. The wire rope of 6 × 7 construction consists of a few wires of relatively large size. It is too stiff for hoisting purposes. The 6 × 19 or 6 × 37 constructions are flexible wire ropes, and are commonly used in hoists. The 6 × 7 construction is suitable as haulage and guy rope.

Table 27.10 *Breaking load and mass for 6 × 19 (12/6/1) construction wire ropes*



(a) Wire rope with fibre core



(b) Wire rope with steel core

Nominal diameter (*d*) (mm)	Approximate mass (kg/100 m)		Minimum breaking load corresponding to tensile designation of (kN)					
			1570		1770		1960	
	Fibre core	Steel core	Fibre core	Steel core	Fibre core	Steel core	Fibre core	Steel Core
8	22.1	24.3	31	33	35	37.6	39	41.6
9	28.0	30.8	39	42	44	47.5	49	52.6
10	34.6	38.0	48	52	54	58.7	60	65
11	41.9	46	58	63	66	71.0	73	70.7
12	49.8	54	69	75	78	84.6	87	93.6
13	58.5	64.3	82	88	92	99	102	110
14	67.8	74.5	95	102	107	115	118	127
16	88.6	97.4	124	133	139	150	154	166
18	112	123.0	156	160	176	190	195	210

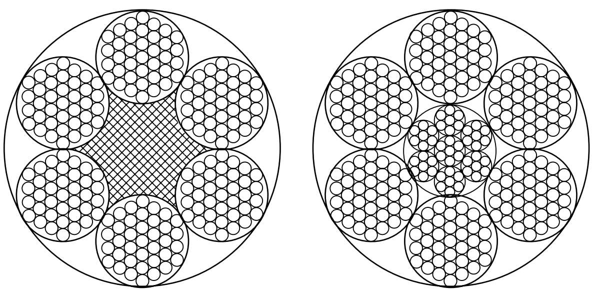
(*Contd.*)

(*Contd.*)

19	125	137	174	188	196	212	217	234
20	138	152.0	193	208	218	235	241	260
22	167	184.0	234	252	263	284	292	314
24	199	219.0	278	300	318	338	347	375
26	234	257	326	352	368	397	407	439
28	271	–	378	–	426	–	472	–
32	354	–	494	–	557	–	617	–
36	448	–	625	–	705	–	781	–
38	499	–	697	–	785	–	870	–
40	554	–	772	–	870	–	964	–
44	670	–	934	–	1053	–	1166	–
48	797	–	1112	–	1253	–	1388	–
52	936	–	1305	–	1471	–	1629	–

(ISO: 2408)

Table 27.11 *Breaking load and mass for 6 × 37 (18/12/6/1) construction wire ropes*



(a) Wire rope with fibre core (b) Wire rope with steel core

Nominal diameter (*d*) (mm)	Approximate mass (kg/100 m)		Minimum breaking load corresponding to tensile designation of (kN)					
			1570		1770		1960	
	Fibre core	Steel core	Fibre core	Steel core	Fibre core	Steel core	Fibre core	Steel Core
8	22.1	24.4	30	32	33	36	37	40
9	28.0	30.8	37	40	42	46	47	51
10	34.6	38.1	46	50	52	56	58	62
11	41.9	46.1	56	60	63	68	78	76

(*Contd.*)

(*Contd.*)

12	49.8	54.8	67	72	75	81	83	90
13	58.5	64.3	78	84	88	95	98	105
14	67.8	74.6	91	98	102	110	113	122
16	88.6	97.4	118	128	134	144	148	160
18	112	123.0	150	162	169	183	187	202
19	125	137	167	180	188	203	209	225
20	138	152.0	185	200	209	225	231	250
22	167	184.0	224	242	253	273	280	302
24	199	219.0	267	288	301	325	333	359
26	234	257	313	338	353	381	391	422
28	271	297	363	392	409	442	453	489
32	354	389	474	512	534	577	592	639
36	448	492	600	648	676	730	749	809
38	499	549	668	722	753	813	834	901
40	554	608	741	880	835	902	924	998
44	678	–	896	–	1010	–	1119	–
48	797	–	1066	–	1202	–	1331	–
52	936	–	1252	–	1411	–	1562	–
56	1085	–	1451	–	1636	–	1812	–

(ISO: 2408)

Table 27.12 *Factors of safety for wire ropes used in cranes and hoists*

Application	**Minimum factor of safety**		
	Class 1	**Class 2 and 3**	**Class 4**
Fixed guys, unreeved rope bridles of jib cranes or ancillary applications like lifting beams, ropes which are straight between terminal fittings	3.5	4.0	4.5
Hoisting, luffing and reeved bridle systems of inherently flexible cranes such as mobile, crawler tower, guy derrick (where jibs are supported by ropes or where equivalent shock absorbing devices are incorporated in the jib supports)	4.0	4.5	5.5
Cranes and hoists in general hoist blocks	4.5	5.0	6.0

Table 27.13 *Factor of safety for wire ropes used in mining applications*

Application	Minimum factor of safety
(a) Mining ropes for shafts of varying depths:	
Up to 300 m	10
300 m–500 m	9
500 m–700 m	8
700 m–1000 m	7
1000 m– 1500 m	6
Over 1500 m	5
(b) Haulages ropes	7

27.3 ROPE SHEAVES AND DRUMS

Table 27.14 *Ratio of drum and sheave diameter to rope diameter (D/d)*

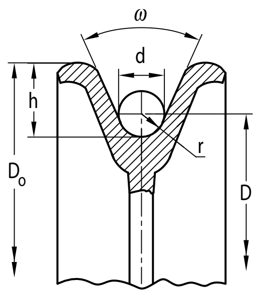
<table>
<tr><th colspan="2">Application</th><th colspan="3">Minimum ratio</th></tr>
<tr><td colspan="2">Mining installation
Winder
Haulage up to 50 kW
Haulage 100 kW and above</td><td colspan="3">
100
50
60</td></tr>
<tr><td rowspan="6">Cranes and hoists</td><td>Construction</td><td>Class-1</td><td>Class-2 and 3</td><td>Class-4</td></tr>
<tr><td>6 × 37
8 × 19</td><td>15</td><td>17</td><td>22</td></tr>
<tr><td>8 × 19
8 × 19 seale
34 × 7 (non rotating)</td><td>17</td><td>18</td><td>24</td></tr>
<tr><td>6 × 19 filler wire</td><td>18</td><td>20</td><td>23</td></tr>
<tr><td>6 × 19
17 × 7 and 18 × 7 (non rotating)</td><td>19</td><td>23</td><td>27</td></tr>
<tr><td>6 × 19 seale</td><td>24</td><td>28</td><td>35</td></tr>
</table>

Note: (i) The ratios of diameters specified in above table are valid for rope speeds up to 50 m/min. For higher speeds, the drum or sheave diameter should be increased by 8% for each additional 50 m/min of rope speed. Large ratio should be used particularly for non-rotating ropes

(ii) D = pitch circle diameter of rope sheave (mm)

d = nominal diameter of wire rope (mm)

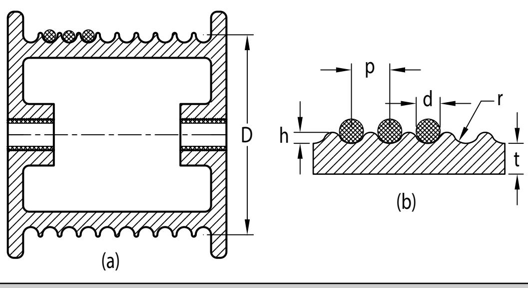
Table 27.15 *Proportions of rope sheave*



Notations
D_o = outside diameter of sheave (mm) h = depth of sheave groove (mm) ω = angle between sides of sheave (°) d = nominal diameter of wire rope (mm) r = radius of groove in the sheave (mm) D = pitch circle diameter of rope sheave (mm)
Proportions of sheave (Indian standard)
ω = 45º to 55º h = (d) to (1.5 d) r (minimum) = 1.05 (nominal rope radius)
Proportions of sheave (International standard)
ω = 30º to 60º h (minimum) = (1.5d) r = (0.525d) to (0.550d) r (optimum) = (0.5375d)

Note: Sheaves are usually made of grey cast iron FG 200.

Table 27.16 *Proportions of rope drum*



Notations
h = depth of drum groove (mm)
d = nominal diameter of wire rope (mm)

(*Contd.*)

r = radius of groove in the drum (mm) D = pitch circle diameter of rope drum (mm) p = pitch of ropes (mm) t = thickness of drum shell (mm)
Proportions of drum
h (minimum) = 0.375 d (for helically grooved drum) r (minimum) = 1.05 (nominal rope radius) r = (0.525d) to (0.550 d) r (optimum) = (0.5375 d) p (minimum) = 2.065 × groove radius (for single layer drum) p (maximum) = 2.18 × groove radius (for single layer drum)
t = 0.02D + (6 to 10 mm) (empirical relationship)

Note: Drums are usually made of grey cast iron FG 200.

Table 27.17 *Clearance between adjacent turns of rope – Annular or concentric grooved drums*

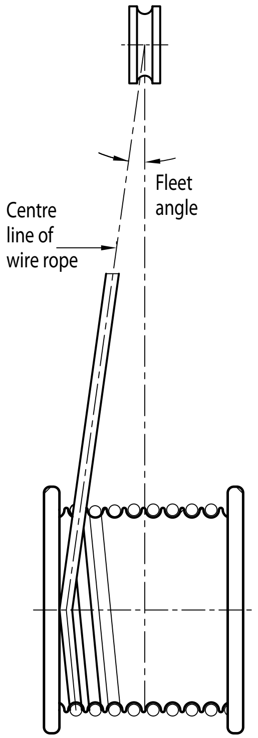
Rope diameter (d) (mm)	Clearance (mm)
Up to 12 mm	1.5
12 to 28	2.5
28 to 38	3.0

Table 27.18 *Clearance between adjacent turns of rope – Helically grooved drums*

Rope diameter (d) (mm)	Clearance (mm)
Up to 32 mm	1.5
32 to 44	2.5
44 to 51	3.0

Note: The grooves on the drum should be pitched so that there is clearance (as given in Tables 27.17 and 27.18) between neighboring turns of rope on the drum.

Table 27.19 *Fleet angle*

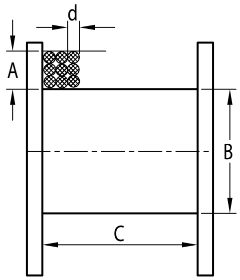


The angle between the vertical plane of rotation of the sheave and the rope leading to the drum is called the fleet angle. The fleet angle should be kept as small as possible in order to avoid excessive wear of sheave and to prevent chafing of the oncoming rope against previous wraps on the drum. The fleet angle can be reduced by

(i) decreasing the drum width or increasing drum diameter; and
(ii) Increasing distance between the sheave and the drum.

The fleet angle should kept between following limits:
Mining Installations = 1° 30′
Cranes and Hoists = 5°

Table 27.20 *Holding capacity of reel (drum)*



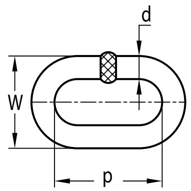
$$L = \frac{\pi(A+B) \times A \times C}{1000 d^2} \quad (27.1)$$

L = rope length or rope capacity of reel (m)
A = depth of rope layer (mm)
B = diameter of reel barrel (mm)
C = width between reel flanges (mm)
d = rope diameter (mm)

Note: The formula is based on assumption of uniform rope winding.

27.4 WELDED LIFTING CHAINS

Table 27.21 *Chain terminology*



Nominal size of chain (*d*)
The nominal size of chain is the nominal diameter of the round section steel wire or bar from which the chain is made.
Pitch of chain (*p*)
The pitch is the internal length of the link.
Manufacturing proof force of chain (MPF)
Proof force is the force to which the whole chain is subjected to during the manufacture. A test is conducted on the standard testing machine, where a straight length of the chain is subjected to tensile force at uniform rate. Proof force is the force which the chain has withstood. The proof force is usually twice the working load limit.
Breaking force (BF)
The breaking force is the maximum force that the chain can withstand during the course of a static tensile test to destruction. The minimum breaking force is the theoretical or calculated force at which chain will break under the test. For example, for grade M (4) chain, a mean stress of 400 N/mm^2 in the chain will result in minimum breaking force.

(*Contd.*)

(*Contd.*)

Working load limit (WLL)
The working load limit is the maximum mass which the chain is authorized to sustain, in vertical condition, in general lifting service.
Total ultimate elongation (A)
The total ultimate elongation is the total extension at the point of fracture of the chain, expressed as the percentage of the internal length of the test sample.
Factor of safety
The factor of safety is the ratio of the minimum breaking force to the working load limit of chain. It is usually 4 : 1for some grades of link chain.

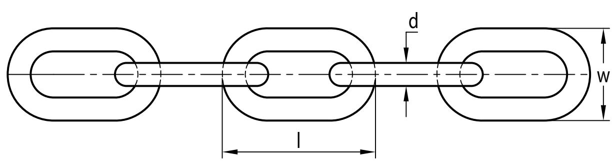
Table 27.22 *Grading system for link chains, hooks, shackles and other chain accessories (Symbols for grade)*

Grade		Mean stress at the specified minimum breaking force (N/mm^2)
Fine tolerance	**Medium tolerance**	
L	3	315
M	4	400
P	5	500
S	6	630
T	8	800
V	10	1000

Note: (i) Chains are classified into grades, which relate to the mechanical properties of the finished product. Each grade is designated by a letter for fine tolerance chain or number for medium tolerance chain such as *L*(3), *M*(4), *P*(5), *S*(6) *T*(8) or *V*(10). The letter or number indicates the mean stress at minimum breaking force. This grading system is also applicable to hooks, shackles and other accessories attached to chain, indicating their strength compatibility with link chain.

(ii) The stresses in the cross-section of chain link are not uniform. The maximum stress occurs at the extrados, which is considerably greater than the mean stress obtained by dividing the force acting on the chain by cross-sectional area of both legs of the link.

Table 27.23 *Short link chain of Grade L(3)*



The short link chains are electric resistance butt welded from round steel bar. The non-calibrated type of load chain of Grade L (3) is used for lifting purposes. Their sizes range from 6 to 45 mm.

(*Contd.*)

(*Contd.*)

Proportions of chain
l (max.) = 5 d l (min.) = 4.75 d w (max.) = 3.5 d
d = nominal size of chain (mm) l = outside length of link (mm) w = outside width of link (mm)
Material of chain
The steel used for chain is produced by open hearth or electric process or by oxygen-blown process. The maximum sulphur and phosphorus contents are limited to 0.045% and 0.040% respectively. The austenitic grain size is 5 or finer.

Note: (i) The nominal size (d) of link chain is the diameter of the rod from which the link is made.
(ii) Link chains are classified as calibrated and non-calibrated. Calibrated chain is manufactured with close tolerances and tested and stretched in such a way that individual links are more uniform and conform to a specified pitch.
(iii) Welded link chains are used in low capacity hoists, winches and hand operated cranes. They are used as slings for suspending the load from the hook or other device.
(iv) As a rule, chains are manufactured in lengths of the required size.

Table 27.24 *Limiting working load for short link chains of Grade L(3)*

Nominal size (d) (mm)	Proof force to which the whole chain is subjected (kN)	Minimum breaking force (kN)	Limiting working load (tonnes)
6	8.9	17.8	0.44
7.1	12.5	25.0	0.63
8	15.9	31.8	0.8
9	20.1	40.2	1.0
10	24.8	49.6	1.25
11	30.0	60.0	1.5
12	35.7	71.4	1.8
14	48.6	97.2	2.4
16	63.5	127.0	3.2
18	80.0	160.0	4.0
20	99.0	198.0	5.0
22	120.0	240.0	6.0
25	155.0	310.0	7.8
28	194.5	389.0	9.7
32	254.0	508.0	12.7
36	321.5	643.0	16.0
40	397.0	794.0	19.8
45	502.0	1004.0	25.0

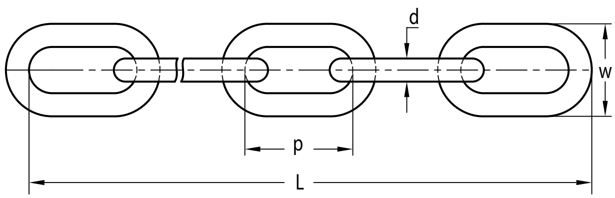
Note: 1 tonne is a metric unit of 1000 kg mass. It is also called metric ton.

Table 27.25 *Mechanical properties of short link chains of Grade L(3)*

Mechanical properties	Requirement
1. Mean stress at specified minimum breaking force, or $\dfrac{F_m(\text{min})}{2\left(\dfrac{\pi d^2}{4}\right)}$	315 MPa
2. Mean stress at proof force, or $\dfrac{F_e}{2\left(\dfrac{\pi d^2}{4}\right)}$	157.5 MPa
3. Ratio of proof force to specified minimum breaking force	50%
4. Specified minimum total ultimate elongation	20%
5. Mean stress at working load limit	78.75 MPa

Note: The stresses mentioned in above table are obtained by dividing the force acting on chain by the total cross-sectional area of both legs of the link. They are mean stresses. In practice, the stress is not uniform and the maximum stress at the extrados is considerably higher than the mean stress.

Table 27.26 *Short link chains of Grade S (6) (Calibrated load chain for pulley blocks)*



The short link chains are electric resistance butt welded from round steel bar. The calibrated type of load chain of Grade S (6) is used for pulley blocks and similar other lifting appliances. Their sizes range from 5 to 36 mm.

Proportions of chain

$p = 3d$
$w = 3.25d$

d = nominal size of chain (mm)
p = pitch or inside length of link (mm)
w = outside width of link (mm)

Material of chain

The steel used for chain is produced by open hearth or electric process or by oxygen-blown process. The maximum sulphur and phosphorus contents are limited to 0.035% each.

Note: (i) The nominal size (d) of link chain is the diameter of the rod from which the link is made.
(ii) Link chains are classified as calibrated and non-calibrated. Calibrated chain is manufactured with close tolerances and tested and stretched in such a way that individual links are more uniform and conform to a specified pitch.
(iii) As a rule, chains are manufactured in lengths of the required size.

Table 27.27 *Limiting working load for short link chains of Grade S (6) (Calibrated load chain for pulley blocks)*

Nominal size (*d*) (mm)	Proof force to which the whole chain is subjected (kN)	Minimum breaking force (kN)	Limiting working load (tonnes)
5	12.4	24.8	0.63
6	17.9	35.8	0.9
7.1	25	50	1.25
8	31.7	63.4	1.6
9	40.1	80.2	2.0
10	49.5	99	2.5
11	60	120	3.0
12	72	144	3.6
14	99	198	5.0
16	127	254	6.3
18	161	322	8.0
20	198	396	10
22	244.5	489	12.5
25	314	628	16
28	393	786	20
32	507	1014	25
36	642	1284	32

Note: 1 tonne is a metric unit of 1000 kg mass. It is also called metric ton.

Table 27.28 *Lifting capacity and chain size for Grade S (6) calibrated load chains*

Lifting capacity (tonne)	Chain size
1.0	6
1.6	8
2.5	10
4.0	12
6.3	16
10	20

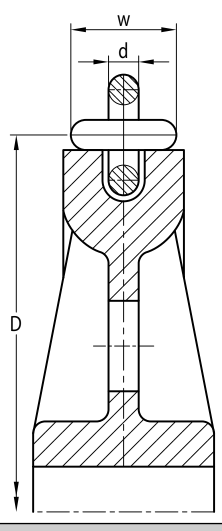
Table 27.29 *Mechanical properties of short link chains of Grade S(6)*

Mechanical properties	Requirement
1. Mean stress at specified minimum breaking force, or $$\frac{F_m(\text{min})}{2\left(\frac{\pi d^2}{4}\right)}$$	630 MPa
2. Mean stress at proof force, or $$\frac{F_e}{2\left(\frac{\pi d^2}{4}\right)}$$	315 MPa
3. Ratio of proof force to specified minimum breaking force	50%
4. Specified minimum total ultimate elongation	13%
5. Mean stress at working load limit	157.5 MPa

Note: The stresses mentioned in above table are obtained by dividing the force acting on chain by the total cross-sectional area of both legs of the link. They are mean stresses. In practice, the stress is not uniform and the maximum stress at the extrados is considerably higher than the mean stress.

27.5 CHAIN WHEELS

Table 27.30 *Sheaves for welded link chains*



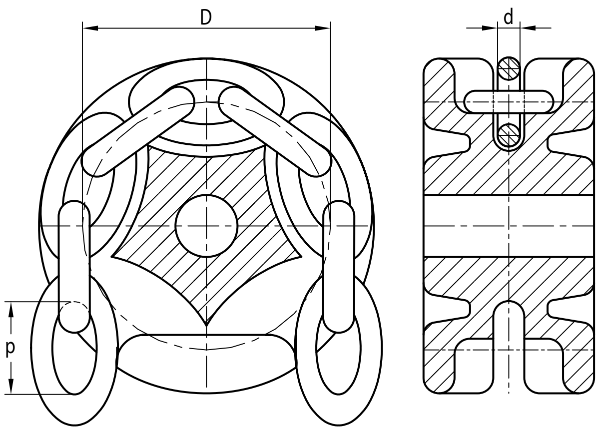
Diameter of sheave (D)
For hand-driven mechanisms, $D \geq 20d$ For power drives, $D \geq 30d$

(*Contd.*)

(*Contd.*)

Design Features:	(i) Sheaves are made of cast iron. (ii) The rim is not machined and the chain fits in the groove with a clearance. (iii) The sheaves are usually mounted freely on their axles. Since the speed is low, sheave hubs are designed without bronze bushing. (iv) The efficiency of chain sheave is 95%.
$$W = Q\frac{d}{R}\mu \qquad (27.2)$$	
W = resistance of welded chain running over the sheave to bending (N) Q = tension in chain (N) μ = coefficient of friction in link joints (μ = 0.1 to 0.2) d = nominal size of chain (mm) R = radius of sheave (D/2) (mm)	

Table 27.31 *Sprockets for welded link chains*



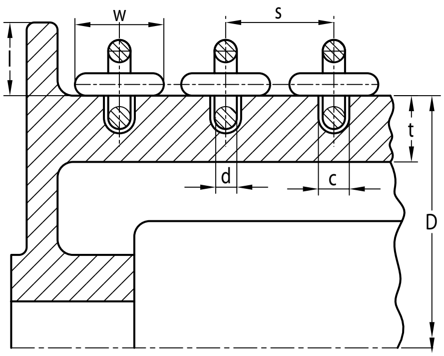
Diameter of rolling circle (*D*)
$$D = \sqrt{\left(\frac{p}{\sin\left(\frac{90}{z}\right)}\right)^2 + \left(\frac{d}{\cos\left(\frac{90}{z}\right)}\right)^2} \qquad (27.3)$$
D = diameter of rolling circle i.e., centre to centre distance between chain runs (mm) p = inside length of chain or pitch of chain (mm) d = nominal size of chain (mm) z = number of teeth in sprocket $$z_{min} = 4$$
When ($z \geq 6$) and ($d \leq 16$ mm), the first term under the square root is far greater than the second one and the above equation is written as, $$D = \frac{p}{\sin\left(\frac{90}{z}\right)} \qquad (27.4)$$

(*Contd.*)

(*Contd.*)

Design Features: (i) Sprockets are made of grey cast iron or cast steel. (ii) Pockets are provided around the periphery of the sprocket which fully conform to the shape and size of the oval links of chain. The sprocket catches the running-on chain and the links seat in the pockets. This prevents the chain from slipping off the rim. (iii) As a rule, sprockets are made with a small number of teeth and small in size resulting in compact construction. (iv) There is considerable friction when the chain runs over the sprocket. The efficiency of chain sprocket is 93%. (v) Usually, the sprocket is freely mounted on the axle. (vi) The arc of contact between the sprocket and the chain should be at least 180º. If the arc is less than 180º, a chain guide is provided.

Table 27.32 *Chain drums for welded link chains*



Chain drums are used only in exceptional cases for hand operated cranes with lifting capacity up to 5 tons. These drums are provided with helical grooves to guide the chain.
Diameter of drum (D)
$D \geq 20d$
D = diameter of drum (mm) d = nominal size of chain (mm)
Dimensions of groove
$s = w + (2 \text{ to } 3)$ mm $c = 1.2\, d$ $l > w$
s = pitch of groove (mm) w = outside width of chain link (mm) c = width of groove (mm) l = height of side flange of drum (mm)
Wall thickness of drum
$t = (0.75 \text{ to } 1.3)\, d$ cm
t = wall thickness of cast iron drum (cm) **Note:** The wall thickness at the weakest place should not be less than 15 to 20 mm.
$$W = Q\frac{d}{2R}\mu \qquad (27.5)$$

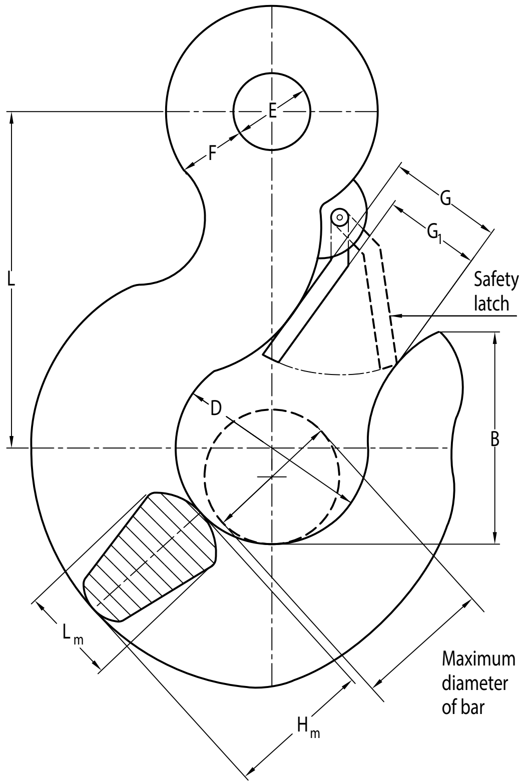
(*Contd.*)

(*Contd.*)

W = resistance of welded chain running over the drum to bending (N) Q = tension in chain (N) μ = coefficient of friction in link joints (μ = 0.1 to 0.2) d = nominal size of chain (mm) R = radius of drum (D/2) (mm)
Design Features: (i) Sprockets are usually made of grey cast iron. (ii) There is considerable friction when the chain runs over the drum. The efficiency of chain drum is 94% to 96%.

27.6 CHAIN HOOKS

Table 27.33 *Proportions of eye hooks for link chains*



G = 2.9 d (Min.) G_1 = 2.7 d (Min.) D = 3.8 d (Min.) H_m = 4.3 d (Max.) L_m = 2.9 d (Max.) E = 1.75 d (Min.) F = 1.8 d (Max.) L = 15.5 d (Max.)	G = throat opening (mm) D = seat diameter (mm) H_m = depth of cross-section (mm) L_m = width of cross-section (mm) d = nominal diameter of chain (mm)

(*Contd.*)

(*Contd.*)

Note: (i) The point height *B* should be greater than the throat opening *G*. (ii) The throat opening *G* should not exceed 95% of seat diameter *D*. (iii) When safety latch is fitted, it should be capable of closing over the maximum diameter of the bar which can be admitted through the opening G_1.	
Grades: There are four grades of eye hooks – Grade L(3), Grade M(4), Grade S(6) and Grade T(8). They are used with corresponding grades of link chains. **Material** – Eye hooks are made of mild steel, high tensile steel and alloy steel. **Manufacture** – Eye hooks are usually forged or drop forged from 0.2% carbon low alloy steel or 0.2% carbon low-manganese steel.	

Table 27.34 *Lifting capacities of eye hooks for link chains*

Nominal dia. of chain (*d*) (mm)	Lifting capacity for Grades (tonne)				Proof load for Grades (kN)			
	L	M	S	T	L	M	S	T
6.0	0.50	0.63	1.00	1.25	10.00	12.50	20.00	25.00
7.1	0.63	0.80	1.25	1.60	12.50	16.00	25.00	31.50
8.0	0.80	1.00	1.60	2.00	16.00	20.00	31.50	40.00
9.0	1.00	1.25	2.00	2.50	20.00	25.00	40.00	50.00
10.0	1.25	1.60	2.50	3.20	25.00	31.50	50.00	63.00
11.0	1.60	2.00	3.20	4.00	31.50	40.00	63.00	80.00
12.0	2.00	2.50	4.00	5.00	40.00	50.00	80.00	100.00
14.0	2.50	3.20	5.00	6.30	50.00	63.00	100.00	125.00
16.0	3.20	4.00	6.30	8.00	63.00	80.00	125.00	160.00
18.0	4.00	5.00	8.00	10.0	80.00	100.00	160.00	200.00
20.0	5.00	6.30	10.0	12.50	100.00	125.00	200.00	250.00
22.0	6.30	8.00	12.50	16.0	125.00	160.00	250.00	315.00
25.0	8.00	10.0	16.0	20.0	160.00	200.00	315.00	400.00
28.0	10.0	12.50	20.0	25.0	200.00	250.00	400.00	500.00
32.0	12.50	16.0	25.0	32.0	250.00	315.00	500.00	630.00
36.0	16.0	20.0	32.0	40.0	315.00	400.00	630.00	800.00
40.0	20.0	25.0	40.0	50.0	400.00	500.00	800.00	1000.0
45.0	25.0	32.0	50.0	63.0	500.00	630.00	1000.0	1260.0

Table 27.35 *Dimensions of eye hooks for link chains*

Nominal dia. of chain (*d*) (mm)	*G* (Min) (2.9*d*) (mm)	G_1 (Min) (2.7*d*) (mm)	*D* (Min) (3.8*d*) (mm)	H_m (Max) (4.3*d*) (mm)	L_m (Max) (2.9*d*) (mm)	*E* (Min) (1.75*d*) (mm)	*F* (Max) (1.8*d*) (mm)	*L* (Max) (15.5*d*) (mm)
6.0	17.40	16.20	22.80	25.80	15.40	10.50	10.80	93.00
7.1	20.60	19.20	27.00	30.50	20.60	12.40	12.80	110.0

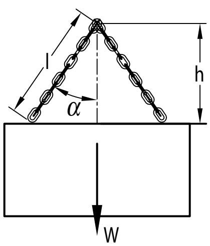
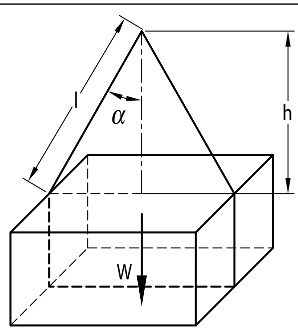
(*Contd.*)

(*Contd.*)

8.0	23.20	21.60	30.40	34.40	23.20	14.00	14.40	124.0
9.0	26.10	24.30	34.20	38.70	26.10	15.80	16.20	139.5
10.0	29.00	27.00	38.00	43.00	29.00	17.50	18.00	155.0
11.0	31.90	29.70	41.80	43.70	31.90	19.30	19.80	170.5
12.0	34.80	32.40	45.60	51.60	34.80	21.00	21.60	186.0
14.0	40.60	37.80	53.20	60.20	40.60	24.50	25.20	217.0
16.0	46.40	43.20	60.80	68.80	46.40	28.00	28.80	248.0
18.0	52.20	48.60	68.40	77.40	52.20	31.50	32.40	279.0
20.0	58.00	54.00	76.00	86.00	58.00	35.00	36.00	310.0
22.0	65.00	59.40	83.60	94.60	63.80	38.50	39.60	341.0
25.0	72.20	67.50	95.00	107.5	72.50	43.80	45.00	387.5
28.0	81.20	75.60	106.4	120.4	81.20	49.00	50.40	434.0
32.0	92.80	86.40	121.6	137.6	92.80	56.00	57.60	496.0
36.0	104.4	97.20	136.8	154.8	104.4	63.00	64.80	558.0
40.0	116.0	108.0	152.0	172.0	116.0	70.00	72.00	620.0
45.0	130.5	121.5	171.0	193.0	130.5	78.80	81.00	697.5

27.7 SLINGS

Table 27.36 *Chain slings*

<table>
<tr><td colspan="2">
</td></tr>
<tr><td colspan="2">Chain slings are made of un-calibrated welded chains with eyes or hooks at the end for suspension of load.</td></tr>
<tr><td>
</td><td>Two-leg sling
$$T = \frac{W}{2\cos\alpha} = \frac{W}{2}\left(\frac{l}{h}\right) \qquad (27.6)$$</td></tr>
</table>

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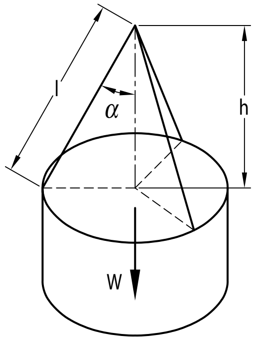
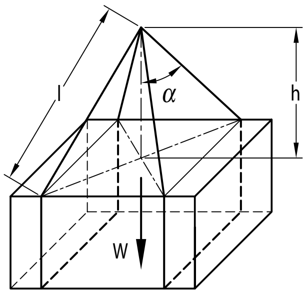
	Three-leg sling $$T = \frac{W}{3\cos\alpha} = \frac{W}{3}\left(\frac{l}{h}\right) \quad (27.7)$$
	Four-leg sling $$T = \frac{W}{4\cos\alpha} = \frac{W}{4}\left(\frac{l}{h}\right) \quad (27.8)$$

W = suspended mass (kg)
T = tension in each leg (kg)
l = length of sling chain (m)
h = vertical height of sling chain (m)
α = angle between the legs and the vertical (°)

General equation

$$T = \frac{W}{n\cos\alpha} = \frac{W}{n}\left(\frac{l}{h}\right) \quad (27.9)$$

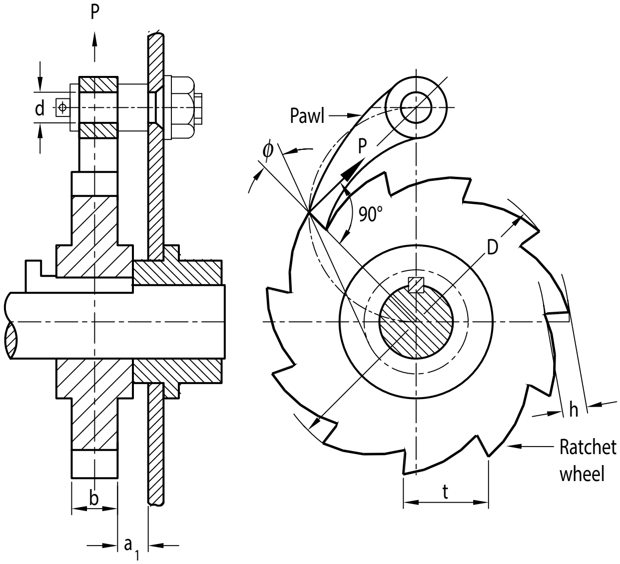
n = number of legs in the sling

Note: (i) The above equations are derived on the assumption that the suspended mass will be equally shared between the legs. In practice, three-leg or four-leg slings need special attention, because rarely do all legs take an equal share of the load. The reason for this situation is that it is virtually impossible to have equal length for each leg. When the length of one leg is slightly more, it becomes slack and do not take any load. It is best to assume that only two legs are sharing the load, with third leg or fourth leg only acting as balancer.

(ii) As the included angle (α) increases, the useful load taken by the sling legs decreases. Therefore, it is recommended that the included angle (α) should be preferably 30°. In any case, it should not exceed 60°.

27.8 ARRESTING GEAR

Table 27.37 *Ratchet and pawl*



Arresting gear or ratchet gearing is used to hold the load lifted by the winches and prevent it from falling down due to gravity. Ratchet gearing consists of a ratchet wheel fitted to the shaft of the drive and a pawl whose pivot receives support from a fixed structural part. The teeth of ratchet are shaped in such a way that the ratchet runs freely when the load is being raised. The pawl engaging the ratchet, arrests its motion and consequently that of the drive when the load is being lowered but does not interference ratchet operation in hoisting. The pawl is to be disengaged prior to hoisting. As shown in figure, the best engagement is obtained at a point of contact between the tangent passing through the pawl pivot and outside diameter of the ratchet wheel. In this case the force *P* acting on pawl is directed along the tangent to outside circle of ratchet wheel.

$$D = mz \tag{27.10}$$

$$\psi = \frac{b}{m} \tag{27.11}$$

$$b = \frac{P}{p} \tag{27.12}$$

D = outside diameter of ratchet (mm)
z = number of teeth on ratchet
m = module of ratchet (mm)
b = width of pawl end face (mm)
ψ = width to module ratio
P = tangential force (N)
p = allowable pressure per unit length (N/mm)

(*Contd.*)

(*Contd.*)

Note: 1. The width to module ratio (ψ) varies from 1.5 to 3.0.

$$\psi = 1.5 \text{ to } 3.0$$

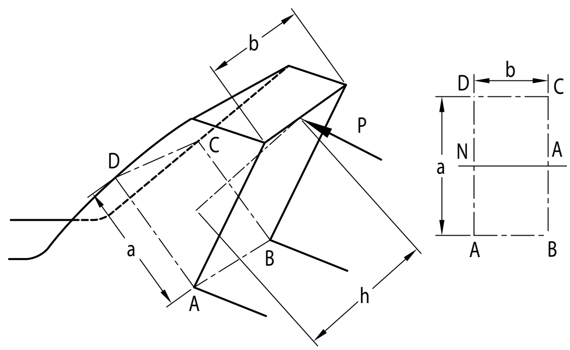
2. The values of allowable pressure per unit length (p) are as follows:
   (a) For steel pawl and cast iron ratchet wheel, $p = 50$ to 100 N/mm
   (b) For steel pawl and steel ratchet wheel, $p = 150$ to 300 N/mm

(N. Rudenko)

Table 27.38 *Number of teeth on ratchet wheel*

Application	Number of teeth (z)
1. Rack and pinion jacks, ratchets and brakes applied by the lifted load (worm hoists)	$z = 6$ to 8
2. Independent ratchet arresters	$z = 12$ to 20
3. Ratchet type brakes	$z = 16$ to 25 and more

Table 27.39 *Bending strength of ratchet*



The bending stresses at section ABCD are given by,

$$\sigma_b = \frac{M_b y}{I} \tag{27.13}$$

$$M_b = Ph \quad I = \frac{ba^3}{12} \quad y = \frac{a}{2} \quad P = \frac{2M_t}{D} \quad D = zm$$

$$Ph = \frac{a^2 b}{6}[\sigma_b] \tag{27.14}$$

Usually, $a = m$ $\quad h = 0.75$ m $\quad b = \psi m$. Therefore,

$$m = 2.08\sqrt[3]{\frac{M_t}{z\psi[\sigma_b]}} \tag{27.15}$$

(*Contd.*)

(*Contd.*)

$$m \approx 2\sqrt[3]{\frac{M_t}{z\psi[\sigma_b]}}$$	(27.16)
M_t = transmitted torque (N-mm) z = number of teeth on ratchet ψ = width to module ratio $[\sigma_b]$ = permissible bending stress (N/mm^2)	
$$[\sigma_b] = \frac{S_{ut}}{(fs)} \text{ (for cast iron)}$$	(27.17)
$$[\sigma_b] = \frac{S_{yt}}{(fs)} \text{ (for steel)}$$	(27.18)
S_{ut} = ultimate tensile strength (N/mm^2) S_{yt} = yield strength (N/mm^2) (fs) = factor of safety For cast iron, (fs) = 5 and For steel, (fs) = 3	

Table 27.40 *Design data for ratchet and pawl*

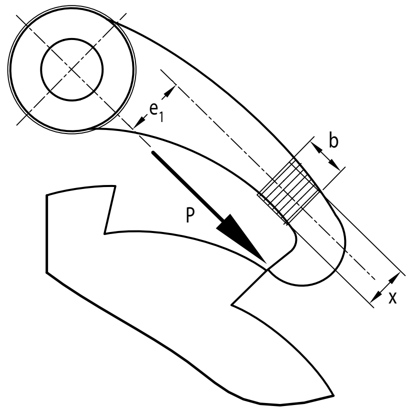
Material of ratchet	$\boldsymbol{\psi = \frac{b}{m}}$	***p*** **(N/mm)**	**Factor of safety (*fs*)**
Grey cast iron	2 – 6	150	5
0.35% C or 0.55%C cast steel	1.5 – 4	300	4
0.3% C steel	1 – 2	350	3
0.45% C steel with additives	1 – 2	400	3

Note: The values of '*p*' are for hand operated or light duty power operated installations. These values should be increased by 25 to 30 per cent for installations with more severe duties.

(M.P. Alexandrov)

Table 27.41 *Stresses in pawl and pawl pin*

Cross section of pawl



Stresses due to eccentric load:

$$M_b = Pe_1 \tag{27.19}$$

$$\sigma = \frac{6M_b}{bx^2} + \frac{P}{bx} \tag{27.20}$$

$$\sigma \le [\sigma_t] \quad \text{and} \quad [\sigma_t] = \frac{S_{yt}}{(fs)} \tag{27.21}$$

Pawl-Pin

Pawl pin is treated as cantilever beam subjected to bending.

$$Pl = 0.1 d^3 [\sigma_b] \tag{27.22}$$

$$l = \frac{b}{2} + a_1 \quad \text{and} \quad P = \frac{2M_t}{zm} \tag{27.23}$$

$$d = 2.71 \sqrt[3]{\frac{P}{2[\sigma_b]}\left(\frac{b}{2} + a_1\right)} \tag{27.24}$$

d = diameter of pawl pin (mm)

a_1= thickness of collar on pawl pin (mm)

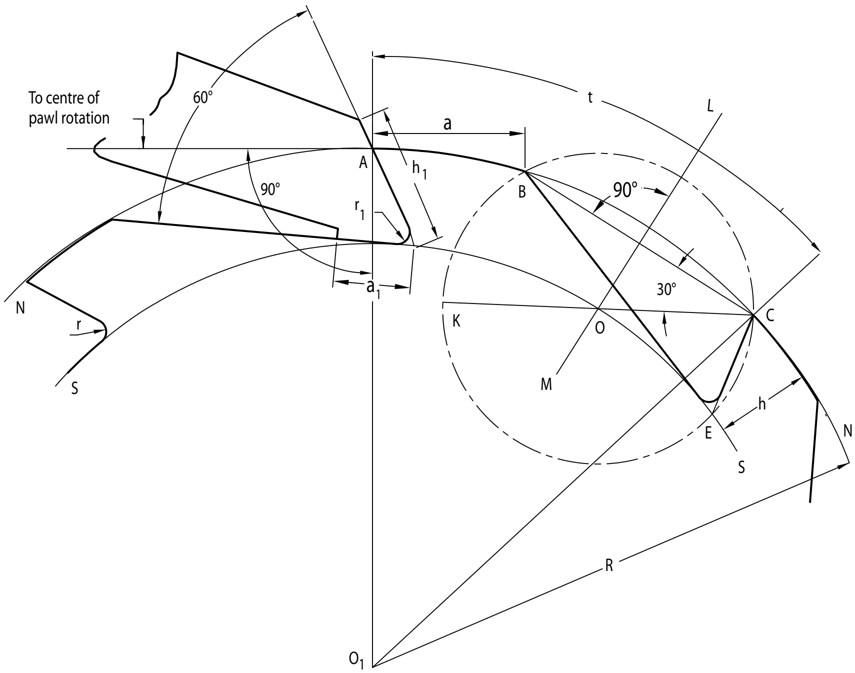
$[\sigma_b]$ = permissible bending stress for pawl pin (30-50 N/mm^2 for 0.45%C steel)

b = face width of ratchet (mm)

P = peripheral force (N)

Table 27.42 *Tooth profile of ratchet wheel*

Ratchet tooth profile



Pitch circle radius of ratchet wheel

$$R = \frac{mz}{2}$$

R = pitch circle radius of ratchet wheel (mm)
m = module (mm)
z = number of teeth on ratchet wheel
Note: R is also the addendum circle of ratchet wheel.

Dedendum or root circle radius of ratchet wheel

Dedendum or root circle radius = $(R - h)$

$h = 0.75$ m

h = tooth height (mm)

Length of chord *AB*

$$a = m$$

a = length of chord AB (mm)

Circumferential pitch

$$t = \pi m$$

t = circumferential pitch (mm)

(N. Rudenko)

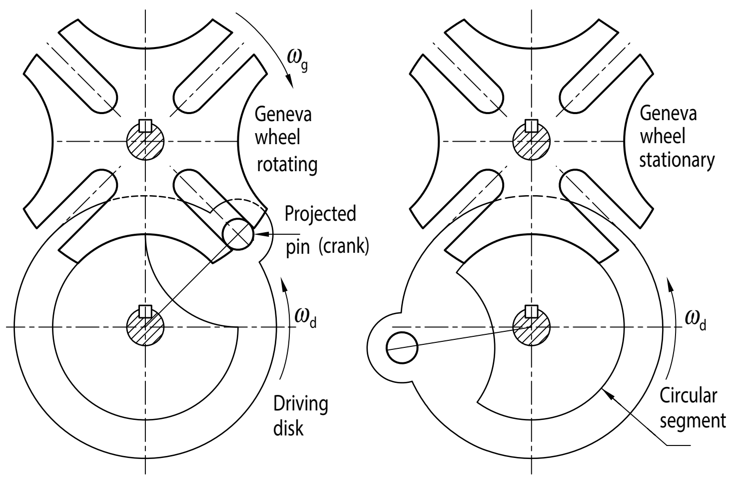
Note: The dimensions of ratchet tooth and pawl are given in Table 27.43.

Table 27.43 *Dimensions of ratchet tooth and pawl (in mm)*

m		6	8	10	12	14	16	18	20	22	24	26	30
z		From 6 to 30											
Ratchet wheel	t	18.85	25.13	31.42	37.70	43.98	50.27	56.55	62.83	69.12	75.40	81.68	94.25
	h	4.5	6	7.5	9	10.5	12	13.5	15	16.5	18	19.5	22.5
	a	6	8	10	12	14	16	18	20	22	24	26	30
	r	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5	1.5
Pawl	h_1	6	8	10	12	14	14	16	18	20	20	22	25
	a_1	4	4	6	6	8	8	12	12	14	14	14	16
	r_1	2	2	2	2	2	2	2	2	2	2	2	2

27.9 GENEVA MECHANISM

Table 27.44 *Geneva mechanism*

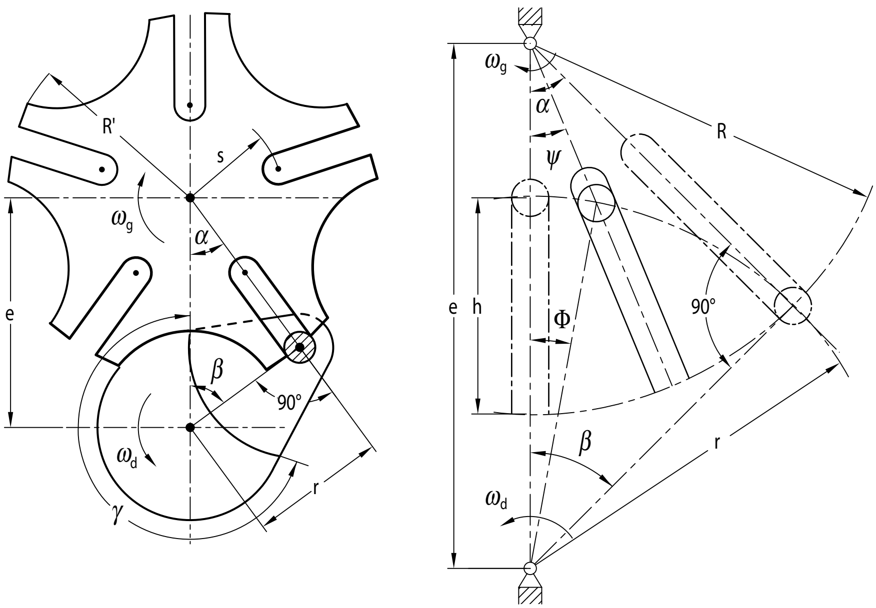


Geneva mechanism is a cam-like mechanism that provides intermittent rotary motion. Geneva wheel is also called Maltese cross. The Geneva mechanism consists of two parts viz. Geneva wheel and driving disk with a projected pin. The Geneva wheel is a star shaped plate containing number of slots that open on the outer periphery. The number of slots varies from three to eight. The driving disk, which is also called crank, has a circular segment and an oval shaped projection containing a pin. The projected pin acts like a crank and carries a roller to engage with the slots of Geneva wheel. The driving disk rotates at constant angular velocity ω_d. During one revolution of the driving disk, the Geneva wheel rotates a fractional part of the revolution. The construction of Geneva wheel and driving disk is such that, at the instant of entry of the crank pin in the slot, the radius of driving crank is tangent to the Geneva wheel. The motion of Geneva wheel consists of two parts- rotary motion during engagement with crank and no motion during engagement with circular segment. The driving crank rotates the Geneva wheel during the period of its engagement. During the remaining period, the circular segment of driving disk effectively locks the Geneva wheel against rotation. This results in intermittent rotary motion of Geneva wheel.

Applications
Geneva mechanism was originally developed as a stop to prevent over winding of watches. Now, it is extensively used as indexing devices in machine tools. The automatic machines require special attachments to hold number of tools and at the time of required operation the concerned tool is brought to the work piece. For carrying out such operations, different tools such as drilling tool, facing tool etc are brought to rotating work piece one after another at the required instant. Indexing means rotating the tool carrier through a definite angle. The applications of Geneva wheel include the indexing of automatic and semi-automatic lathes, turrets and multiple-station rotary tables. It is also used in motion picture projector to provide intermittent advance of film.
Geneva wheel is usually made of alloy steel 40Cr4. It is hardened and tempered to hardness of 45 to 50 HRC. The rollers in slot are made of either alloy steel 105Cr5 or 15Cr3. They are hardened to 60 HRC.

(N. Acherkan)

Table 27.45 *Geneva mechanism – Basic relationships*



$$2\alpha = \frac{360°}{Z} \quad \text{or} \quad \alpha = \frac{180°}{Z} = \frac{\pi}{Z} \tag{27.25}$$

α = semi-indexing angle of Geneva wheel (°)
Z = number of slots on Geneva wheel

Since, at the instant of entry of the crank pin in the slot, the radius of driving crank is tangent to the Geneva wheel,

$$\alpha + \beta = 90° = \frac{\pi}{2} \tag{27.26}$$

β = semi-indexing angle of driving disk (°)

(*Contd.*)

(*Contd.*)

Two dimensionless parameters λ and λ_1 are defined as, $$\lambda = \frac{r}{e} \quad \text{and} \quad \lambda_1 = \frac{R}{e} \qquad (27.27)$$	
λ, λ_1 = dimensionless parameters r = radius of driving crank (mm) R = radius of Geneva wheel (mm) e = centre distance between Geneva wheel and driving disk (mm)	
$$\sin\alpha = \frac{r}{e} = \lambda$$	(27.28)
$$\cos\alpha = \frac{R}{e} = \lambda_1$$	(27.29)
$$\sin^2\alpha + \cos^2\alpha = 1$$ $$\lambda^2 + \lambda_1^2 = 1 \text{ or } \lambda_1 = \sqrt{1-\lambda^2}$$	(27.30)
$$\beta = \frac{\pi}{2} - \alpha = \frac{\pi}{2} - \frac{\pi}{Z} = \frac{\pi}{2}\left(1 - \frac{2}{Z}\right)$$	(27.31)
$$\beta = \frac{\pi(Z-2)}{2Z}$$	(27.32)
The length of slot 'h' in Geneva wheel is given by, $$h = r + R - e$$	(27.33)
$$h = e[\sin\alpha + \cos\alpha - 1]$$	(27.34)
$$h = e\left(\sin\frac{\pi}{Z} + \cos\frac{\pi}{Z} - 1\right)$$	(27.35)
$$h = e[\lambda + \sqrt{1-\lambda^2} - 1]$$	(27.36)
Suppose, T = time required for one full revolution of driving disk (s) t_i = time required for indexing the Geneva wheel (s) t_r = time during which the Geneva wheel rests (s)	
$$\frac{t_i}{T} = \frac{2\beta}{2\pi} = \frac{\beta}{\pi} \text{ and } \beta = \frac{\pi(Z-2)}{2Z}$$ $$\frac{t_i}{T} = \frac{Z-2}{2Z}$$	(27.37)
$$\frac{t_r}{T} = \frac{2\pi - 2\beta}{2\pi} = 1 - \frac{\beta}{\pi} = 1 - \frac{Z-2}{2Z} = \frac{Z+2}{2Z}$$ $$\frac{t_r}{T} = \frac{Z+2}{2Z}$$	(27.38)

(*Contd.*)

(*Contd.*)

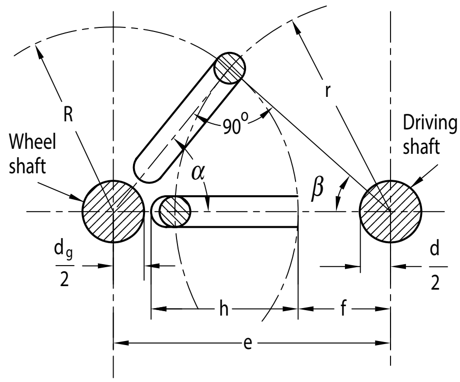
Since, $T = \frac{60}{n}$ sec n = speed of driving shaft (rpm) $$t_i = \left(\frac{Z-2}{2Z}\right)T = \left(\frac{Z-2}{2Z}\right)\frac{60}{n} = \left(\frac{Z-2}{Z}\right)\left(\frac{30}{n}\right)$$ $$t_i = \left(\frac{Z-2}{Z}\right)\left(\frac{30}{n}\right) \text{ sec} \quad (27.39)$$ $$t_r = \left(\frac{Z+2}{2Z}\right)T = \left(\frac{Z+2}{2Z}\right)\frac{60}{n} = \left(\frac{Z+2}{Z}\right)\left(\frac{30}{n}\right)$$ $$t_r = \left(\frac{Z+2}{Z}\right)\left(\frac{30}{n}\right) \text{ sec} \quad (27.40)$$
$$k = \frac{t_i}{t_r} = \frac{Z-2}{Z+2} \quad (27.41)$$ k = working time coefficient of Geneva wheel (dimensionless)
If the time t_r that the wheel is to be at rest is given, the required speed of driving shaft is given by, $$n = \left(\frac{Z+2}{Z}\right)\left(\frac{30}{t_r}\right) \text{ rpm} \quad (27.42)$$

(N. Acherkan)

Table 27.46 *Geneva mechanism –Parameters t_i , t_r and k*

Z	**3**	**4**	**5**	**6**	**8**
$t_i = \left(\frac{Z-2}{Z}\right)\left(\frac{30}{n}\right)$	$\left(\frac{10}{n}\right)$	$\left(\frac{15}{n}\right)$	$\left(\frac{18}{n}\right)$	$\left(\frac{20}{n}\right)$	$\left(\frac{22.5}{n}\right)$
$t_r = \left(\frac{Z+2}{Z}\right)\left(\frac{30}{n}\right)$	$\left(\frac{50}{n}\right)$	$\left(\frac{45}{n}\right)$	$\left(\frac{42}{n}\right)$	$\left(\frac{40}{n}\right)$	$\left(\frac{37.5}{n}\right)$
$k = \frac{t_i}{t_r} = \frac{Z-2}{Z+2}$	0.2	0.333	0.429	0.50	0.60

Table 27.47 *Geneva mechanism –Shaft diameter*



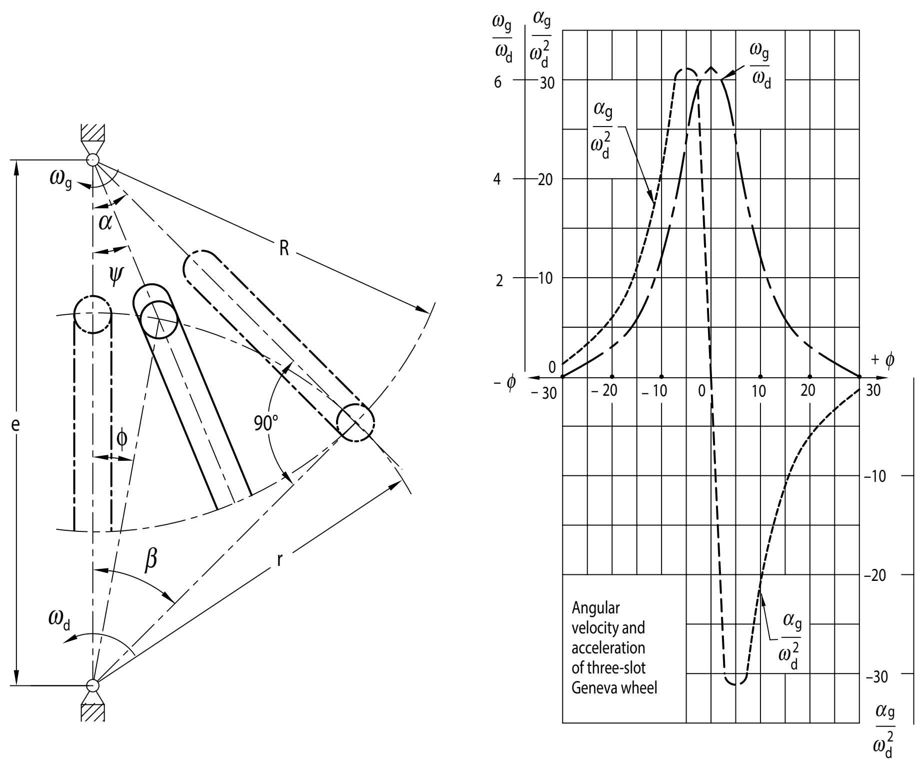
Diameter of shaft for driving disk (d)	
In order that the shaft of driving disk does not interfere with Geneva disk, following condition should be satisfied, $$\left(\frac{d}{2}\right) < f \quad \text{or } d < 2f$$ $$f = e - R$$ $$d < 2(e - R)$$ $$\left(\frac{d}{e}\right) < 2\left(1 - \frac{R}{e}\right) \quad \text{or} \quad \left(\frac{d}{e}\right) < 2(1 - \cos\alpha) \quad \text{or} \quad \left(\frac{d}{e}\right) < 2\left(1 - \cos\frac{\pi}{Z}\right)$$	
$$\left(\frac{d}{e}\right) < 2\left(1 - \cos\frac{\pi}{Z}\right)$$	(27.43)
Diameter of shaft for Geneva wheel (d_g)	
In order that the shaft of Geneva wheel does not interfere with driving disk, following condition should be satisfied, $$\left(\frac{d_g}{2}\right) < (e - r)$$ $$d_g < 2(e - r)$$ $$\left(\frac{d_g}{e}\right) < 2\left(1 - \frac{r}{e}\right) \quad \text{or} \quad \left(\frac{d_g}{e}\right) < 2(1 - \sin\alpha) \quad \text{or} \quad \left(\frac{d_g}{e}\right) < 2\left(1 - \sin\frac{\pi}{Z}\right)$$	
$$\left(\frac{d_g}{e}\right) < 2\left(1 - \sin\frac{\pi}{Z}\right)$$	(27.44)
As a first approximation, d_g is taken as, $$d_g = \left(\frac{R - h}{1.5}\right)$$ The above value is finally checked for torque capacity.	(27.45)

Note: The above relationships are numerically calculated and given in Table 27.48.

Table 27.48 *Geneva mechanism – Parameters for shaft diameter*

Z	3	4	5	6	8
λ	0.866	0.707	0.588	0.50	0.383
λ_1	0.5	0.707	0.809	0.866	0.924
h/e	0.366	0.414	0.397	0.366	0.307
$(d/e) <$	1.0	0.58	0.38	0.26	0.15
$(d_g/e) <$	0.26	0.58	0.82	1.0	1.23

Table 27.49 *Geneva mechanism –velocity and acceleration analysis*



For any angular position ψ of Geneva wheel,

$$\tan \psi = \frac{\lambda \sin \phi}{1 - \lambda \cos \phi} \tag{27.46}$$

Angular velocity of Geneva wheel (ω_g)

$$\omega_g = \frac{d\psi}{dt} = \left[\frac{\lambda(\cos \phi - \lambda)}{1 - 2\lambda \cos \phi + \lambda^2}\right]\omega_d \tag{27.47}$$

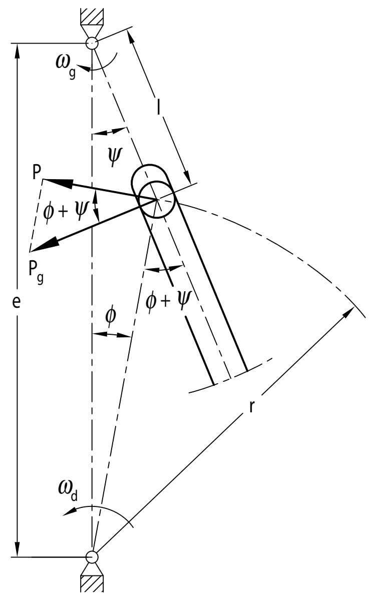
(*Contd.*)

(*Contd.*)

Angular acceleration of Geneva wheel (α_g)	
$$\alpha_g = \frac{d^2\psi}{dt^2} = \pm\left[\frac{\lambda(1-\lambda^2)\sin\phi}{(1-2\lambda\cos\phi+\lambda^2)^2}\right]\omega_d^2$$	(27.48)
Maximum angular velocity of Geneva wheel (ω_g)$_{max}$	
The velocity is maximum, when $\phi = 0$, $$(\omega_g)_{\max} = \left(\frac{\lambda}{1-\lambda}\right)\omega_d$$	(27.49)
Angular acceleration at engagement or disengagement of Geneva wheel	
At engagement or disengagement of Geneva wheel, $$(\alpha_g) = \pm\omega_d^2 \tan\left(\frac{\pi}{Z}\right)$$	(27.50)
The Geneva wheel is always accompanied by impact load at the engagement. The fewer the number of slots, the heavier the impact will be. Therefore, for increased service life of Geneva wheel, it is advantageous to increase the number of slots. In practice, six to eight slots are used for Geneva wheel in indexing mechanisms.	

Note: In above analysis, suffix '*g*' is used for Geneva wheel such as ω_g or α_g and suffix '*d*' is use for driving disk such as ω_d.

Table 27.50 *Geneva mechanism – Torque analysis*



$$M_t \cdot \omega_d = M_{tg} \cdot \omega_g$$

$$M_t = M_{tg}\left(\frac{\omega_g}{\omega_d}\right) \qquad (27.51)$$

(*Contd.*)

(*Contd.*)

Considering efficiency η,

$$M_t = M_{tg}\left(\frac{\omega_g}{\omega_d}\right)\left(\frac{1}{\eta}\right) \quad (27.52)$$

M_t = torque on driving shaft (N-mm)
M_{tg} = torque required at Geneva wheel shaft (N-mm)
ω_d = angular velocity of driving shaft (radian/s)
ω_g = angular velocity of Geneva wheel shaft (radian/s)
η = efficiency of Geneva mechanism

$\eta \cong 0.8$ to 0.9 (for Geneva wheel mounted on shaft running in sleeve bearings)
$\eta \cong 0.95$ (for Geneva wheel mounted on shaft running in antifriction bearings)
$\eta \cong 0.75$ (for Geneva wheel integral with spindle carrier, drum, etc. and if the diameter of bearing surface is larger than outside diameter of Geneva wheel)

The torque required at Geneva wheel shaft (M_{tg}) consists of two factor – torque required to overcome frictional resistance at bearings and at slotted guides (M_{tf}) and inertia torque (M_{ti}).

$$M_{tg} = M_{tf} + M_{ti} \quad (27.53)$$

$$M_{ti} = I_g \alpha_g \quad (27.54)$$

I_g = mass moment of inertia of all masses attached to Geneva wheel (kg-m^2)
α_g = angular acceleration of Geneva wheel (rad/s)

Instantaneous power requirement

$$(\text{kW})_{\text{inst}} = \frac{M_t \omega_d}{10^6} \quad (27.55)$$

Average power requirement

$$M_{tg} = M_{tf} + M_{ti} \quad (27.56)$$

The inertia torque M_{ti} is zero for one complete cycle.

For First half,

$$M_{t\,\text{ave}} = \frac{2}{Z-2}\left(M_{tf} + \frac{ZI_g}{2\pi}\left(\frac{\lambda}{1-\lambda}\right)^2 \omega_d^2\right)\frac{1}{\eta} \quad (27.57)$$

$$(\text{kW})_{\text{ave}} = \frac{M_{t\,\text{ave}}\,\omega_d}{10^6} \quad (27.58)$$

Maximum power requirement

$$M_{t\max} \cong M_{tf}\frac{\omega_{g\max}}{\omega_d}\frac{1}{\eta} + \frac{1}{\omega_d}(\alpha_g \omega_g)_{\max}\frac{1}{\eta}$$

(*Contd.*)

(*Contd.*)

$$M_{t\max} \cong M_{tf}\frac{\lambda}{1-\lambda}\frac{1}{\eta} + \frac{\lambda^2(1-\lambda^2)(\cos\phi_m - \lambda)\sin\phi_m}{(1-2\lambda\cos\phi_m + \lambda^2)^3} I_g \omega_d^2 \frac{1}{\eta}$$	(27.59)
$$(\text{kW})_{\max} = \frac{M_{t\max}\,\omega_d}{10^6}$$	(27.60)
ϕ_m = angular position of crank at which the product ($\alpha_g\ \omega_g$) is maximum (°) (Table 27.52)	
Contact force	
$$P_{g\max} = P_{f\max} + P_{i\max}$$	(27.61)
$$P_{f\max} = \frac{M_{tf}}{l_{\min}} = \frac{M_{tf}}{r}\frac{\lambda}{1-\lambda}$$	(27.62)
$P_{g\max}$ = maximum contact force between roller pin and slot in Geneva wheel (N) $P_{f\max}$ = maximum value of frictional component of contact force (N) $P_{i\max}$ = maximum value of inertia component of contact force (N) l = instantaneous length of action of Geneva wheel (arm of moment) (mm) $l_{\min}$ = minimum value of arm of moment (mm)	
The force P_i due to inertia torque, reaches its maximum value at values of angle ϕ, which depend upon number of slots Z. It is given in Table 27.51.	

Table 27.51 *Geneva mechanism – Maximum force $P_{i\,max}$*

Z	3	4	5	6	8
$P_{i\max}$: $\frac{I_g n^2}{r}$	1.966	0.126	0.0318	0.0131	0.00424

Table 27.52 *Geneva mechanism – Parameters of maximum power*

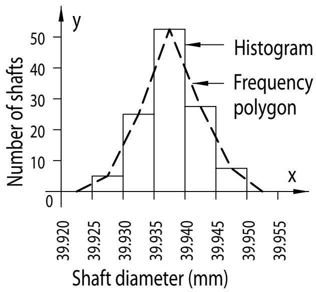
Z	ϕ_m	$\left(\frac{d\psi}{d\phi}\right)_{\max} = \frac{\omega_{g\max}}{\omega_d}$	$\left(\frac{d^2\psi}{d\phi^2}\right)_{\max} = \frac{\alpha_{g\max}}{\omega_d^2}$	$\left(\frac{d^2\psi}{d\phi^2}\right)_{\text{initial}} = \frac{\alpha_{g\,\text{initial}}}{\omega_d^2}$	$\left(\frac{d^3\psi}{d\phi^3}\right)_{\phi=0}$
3	4° 46′	6.46	31.44	1.732	–672
4	11° 24′	2.41	5.409	1.000	–48.04
5	17° 34′	1.43	2.299	0.7265	–13.32
6	22° 54′	1.00	1.350	0.5774	–6.00
7	27° 33′	0.766	0.9284	0.4816	–3.429
8	31° 38′	0.620	0.6998	0.4142	–2.249
9	35° 16′	0.520	0.5591	0.3640	–1.611
10	38° 30′	0.447	0.4648	0.3249	–1.236

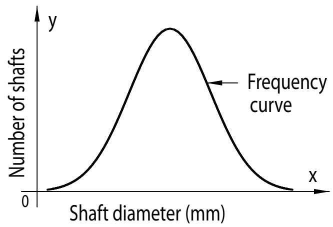
Statistical Considerations in Design

28

28.1 BASIC RELATIONSHIPS

Table 28.1 *Statistical terminology*





Population and sample
Population is defined as a collection of all elements we are studying and about which we are trying to draw conclusions. On the other hand, sample is defined as a collection of some, but not all, of the elements of the population. Sample is a part of population but the converse is not true.
Data
A data is defined as the collection of numbers belonging to observations of one or more variables. For example, the diameter of shaft is a variable. Raw data is a data before it is arranged or analyzed by any statistical method. Raw data does not show any pattern or trend of population and does not lead to any conclusions.
Frequency distribution
Frequency distribution is defined as an organised display of data that shows the number of observations that fall into different classes. There are two methods of representing the frequency distribution namely, histogram and frequency polygon.
Histogram
The histogram consists of a set of rectangles. The base of the rectangle on the *x*-axis is equal to the class width, and the area of the rectangle is proportional to the class frequency. When the class widths are of equal size, the heights of rectangles are proportional to the class frequencies.

(*Contd.*)

(*Contd.*)

Frequency polygon
The frequency polygon is a line graph of class frequency plotted against class marks or midpoints of class intervals.
Frequency curve
When a large number of observations are taken and very small class widths are selected, the frequency polygon becomes an approximate curve called frequency curve. One such frequency curve, which is widely used in the statistical analysis, is the normal curve.
Central tendency and dispersion
It is observed from the frequency polygon and normal curve, that most of the observations in engineering applications present a well behaved picture, which rises and falls smoothly. There is always a central tendency, where most of the observations cluster. Central tendency is the middle point of distribution. It is sometimes, referred as the 'measure of location'. There are certain observations that tend to spread about an average value called 'variation' or 'dispersion' of a population. Dispersion is defined as the spread of the data in a distribution, that is, the extent to which the observations are scattered. The central tendency and dispersion are the two important characteristics of frequency distribution.
Mean of population
The most popular unit to measure the central tendency is the arithmetic mean denoted by letter μ.
Suppose the population consists of n observations $x_1, x_2, ..., x_n$. The mean is given by $$\mu_x = \frac{x_1 + x_2 + \cdots + x_n}{n} \quad \text{or} \quad \mu_x = \frac{\Sigma x_i}{n} \qquad (28.1)$$
Standard deviation of complete population
The most popular unit for dispersion is the standard deviation denoted by letter $\hat{\sigma}$. The standard deviation is defined as the root mean square deviation from the mean.
$$\hat{\sigma}_x = \sqrt{\frac{(x_1 - \mu)^2 + (x_2 - \mu)^2 + \cdots + (x_n - \mu)^2}{n}} \quad \text{or} \quad \hat{\sigma}_x = \sqrt{\frac{\Sigma(x_i - \mu)^2}{n}} \qquad (28.2)$$ Alternatively, $$(\hat{\sigma}_x)^2 = \frac{\Sigma x_i^2 - \frac{(\Sigma x_i)^2}{n}}{n} \qquad (28.3)$$
Standard deviation of sample
When observations belong to a sample of a population, it has been observed that by replacing 'n' by '(n – 1)' in the denominator of above equation, a better estimate of standard deviation is obtained. Therefore, the standard deviation of sample (s_x) is given by, $$(s_x)^2 = \frac{\Sigma x_i^2 - \frac{(\Sigma x_i)^2}{n}}{n - 1} \qquad (28.4)$$

(*Contd.*)

(*Contd.*)

Standard variable (*z*)
The standard variable measures the deviation from the mean in the units of the standard deviation. A standard variable z is defined as, $$z = \frac{x - \mu}{\hat{\sigma}} \qquad (28.5)$$

Population combinations
There are many problems in machine design where it is required to combine two or more populations in a specific manner to obtain the resultant population. As an example, there are two populations in journal bearing – a population consisting of inner diameter of bearings and a population consisting of outer diameter of shafts. Statistically, both populations are random variables. The system is interchangeable and a shaft should match with any bearing selected at random. Further, they are fitted in such a way that there is a proper clearance between the bearing and the shaft. In this case, subtracting the shaft-population from the bearing-population, can form a third population consisting of clearances. The clearance population is a random variable. Sometimes, it is required to divide or multiply one population by another. In some cases, it is required to square a population or to take an inverse of the population.

Note: Table 28.2 shows various types of combinations of population and how to obtain the mean and the standard deviation of resultant combination.

Table 28.2 *Mean and standard deviation for simple algebraic operations on independent random variables*

Function	Mean μ_z	Standard deviation $\hat{\sigma}_z$	
$z = a$	a	0	(28.6)
$z = x$	μ_x	$\hat{\sigma}_x$	(28.7)
$z = ax$	$a\,\mu_x$	$a\ \hat{\sigma}_X$	(28.8)
$z = x + a$	$\mu_x + a$	$\hat{\sigma}_X$	(28.9)
$z = x - a$	$\mu_x - a$	$\hat{\sigma}_X$	(28.10)
$z = x + y$	$\mu_x + \mu_y$	$\sqrt{(\hat{\sigma}_X)^2 + (\hat{\sigma}_Y)^2}$	(28.11)
$z = x - y$	$\mu_x - \mu_y$	$\sqrt{(\hat{\sigma}_X)^2 + (\hat{\sigma}_Y)^2}$	(28.12)
$z = xy$	$\mu_x\,\mu_y$	$\sqrt{\mu_X^2(\hat{\sigma}_Y)^2 + \mu_Y^2(\hat{\sigma}_X)^2 + (\hat{\sigma}_X)^2(\hat{\sigma}_Y)^2}$	(28.13)
$z = x/y$	μ_x/μ_y	$\frac{1}{\mu_Y}\left[\frac{\mu_X^2(\hat{\sigma}_Y)^2 + \mu_Y^2(\hat{\sigma}_X)^2}{\mu_Y^2 + (\hat{\sigma}_Y)^2}\right]^{1/2}$	(28.14)
$z = x^2$	$\mu_x^2 + \hat{\sigma}_x^2$	$\frac{1}{2}\left(\frac{\hat{\sigma}_X}{\mu_X}\right)[4\mu_X^2 + (\hat{\sigma}_X)^2]$	(28.15)

(*Contd.*)

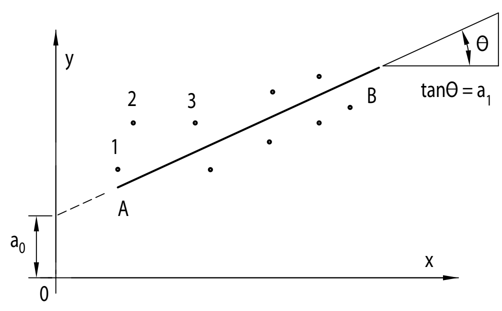
(*Contd.*)

$z = x^3$	$\mu_X^3 + 3\mu_X(\hat{\sigma}_X)^2$	$3\mu_X^2(\hat{\sigma}_X) + 3(\hat{\sigma}_X)^3$ (28.16)
$z = x^4$	$\mu_x^4\left[1+6\left(\frac{\hat{\sigma}_x}{\mu_x}\right)^2\right]$	$4\mu_x^4\left(\frac{\hat{\sigma}_x}{\mu_x}\right)\left[1+\frac{9}{4}\left(\frac{\hat{\sigma}_x}{\mu_x}\right)^2\right]$ (28.17)
$z = x^n$	$\mu_x^n\left[1+\frac{n(n-1)}{2}\left(\frac{\hat{\sigma}_x}{\mu_x}\right)^2\right]$	$(n)\mu_x^n\left(\frac{\hat{\sigma}_x}{\mu_x}\right)\left[1+\frac{(n-1)^2}{4}\left(\frac{\hat{\sigma}_x}{\mu_x}\right)^2\right]$ (28.18)
$z = 1/x$	$\frac{1}{\mu_X}\left[1+\left(\frac{\hat{\sigma}_X}{\mu_X}\right)^2\right]$	$\frac{\hat{\sigma}_X}{\mu_X^2}\left[1+\left(\frac{\hat{\sigma}_X}{\mu_X}\right)^2\right]$ (28.19)
$z = 1/x^2$	$\frac{1}{\mu_x^2}\left[1+3\left(\frac{\hat{\sigma}_X}{\mu_X}\right)^2\right]$	$\frac{2}{\mu_X^2}\left(\frac{\hat{\sigma}_X}{\mu_X}\right)\left[1+\frac{9}{4}\left(\frac{\hat{\sigma}_X}{\mu_X}\right)^2\right]$ (28.20)
$z = 1/x^3$	$\frac{1}{\mu_x^3}\left[1+6\left(\frac{\hat{\sigma}_X}{\mu_X}\right)^2\right]$	$\frac{3}{\mu_X^3}\left(\frac{\hat{\sigma}_X}{\mu_X}\right)\left[1+4\left(\frac{\hat{\sigma}_X}{\mu_X}\right)^2\right]$ (28.21)
$z = 1/x^4$	$\frac{1}{\mu_x^4}\left[1+10\left(\frac{\hat{\sigma}_X}{\mu_X}\right)^2\right]$	$\frac{4}{\mu_X^4}\left(\frac{\hat{\sigma}_X}{\mu_X}\right)\left[1+\frac{25}{4}\left(\frac{\hat{\sigma}_X}{\mu_X}\right)^2\right]$ (28.22)
$z = \sqrt{x}$	$\sqrt{\mu_x}\left[1-\frac{1}{8}\left(\frac{\hat{\sigma}_X}{\mu_X}\right)^2\right]$	$\frac{\sqrt{\mu_x}}{2}\left(\frac{\hat{\sigma}_X}{\mu_X}\right)\left[1+\frac{1}{16}\left(\frac{\hat{\sigma}_X}{\mu_X}\right)^2\right]$ (28.23)

Note: (i) z is a function of two independent random variables x and y and a is constant. (ii) Some expressions in above table are approximate.

28.2 CURVE FITTING

Table 28.3 *Linear regression*



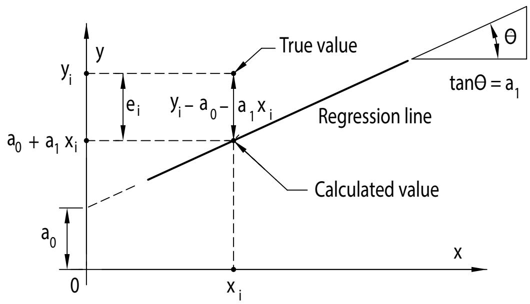
Linear regression is a process of statistical analysis to obtain a straight line which 'best' fits a set of experimental data points. The experimentally derived data points 1, 2, 3,..., n have coordinates (x_1, y_1), (x_2, y_2), (x_3, y_3), ...(x_n, y_n) respectively. The data points are approximated by a straight line AB. The equation of the line AB is given by,

(*Contd.*)

(*Contd.*)

$$y = a_0 + a_1 x \quad (28.24)$$ a_0 = intercept of line *AB* on *y* axis a_1 = slope of the line *AB*	

Criterion of best fit



The mathematical expression for straight line is written as,

$$y = a_0 + a_1 x + e \tag{28.25}$$

e = Error or residual

The error or residual is the discrepancy between true value of y and approximate value ($a_0 + a_1 x$) calculated by linear equation.

The sum of squares of residuals (S_r) between true value of y and calculated value ($a_0 + a_1 x$) is given by,

$$S_r = \sum_{i=1}^{n} (e_i)^2 = \sum_{i=1}^{n} (y_i - a_0 - a_1 x_i)^2 \tag{28.26}$$

The best line is based on the criterion of minimising (S_r). This analysis is often called 'least square' method. The coefficients a_0 and a_1 are obtained by differentiating (S_r) with respect to a_0 and a_1 and setting the results to zero.

$$\frac{\partial S_r}{\partial a_0} = -2\Sigma(y_i - a_0 - a_1 x_i) = 0$$

$$\frac{\partial S_r}{\partial a_1} = -2\Sigma[(y_i - a_0 - a_1 x_i)x_i] = 0$$

Simplifying,

$$\Sigma y_i - \Sigma a_0 - \Sigma a_1 x_i = 0$$

$$\Sigma y_i x_i - \Sigma a_0 x_i - \Sigma a_1 x_i^2 = 0$$

Also, $\Sigma a_0 = n a_0$

Substituting, final equations for a_1 and a_0 are obtained.

Equations for a_1 and a_0

$$a_1 = \frac{n\Sigma x_i y_i - \Sigma x_i \Sigma y_i}{n\Sigma x_i^2 - (\Sigma x_i)^2} \tag{28.27}$$

$$a_0 = \mu_y - a_1 \mu_x \tag{28.28}$$

(*Contd.*)

(*Contd.*)

$\Sigma x_i = x_1 + x_2 + x_3 + \ldots + x_n$

$\Sigma y_i = y_1 + y_2 + y_3 + \ldots + y_n$

$\Sigma x_i y_i = x_1 y_1 + x_2 y_2 + x_3 y_3 + \ldots + x_n y_n$

$\Sigma x_i^2 = x_1^2 + x_2^2 + x_3^2 + \cdots + x_n^2$

$(\Sigma x_i)^2 = (x_1 + x_2 + x_3 + \ldots + x_n)^2$

$$\mu_x = \frac{\Sigma x_i}{n} \text{ and } \mu_y = \frac{\Sigma y_i}{n} \tag{28.29}$$

n = total number of points

Coefficient of determination (r^2)

$$r^2 = \frac{S_t - S_r}{S_t} \tag{28.30}$$

$$S_t = \Sigma(y_i - \mu_y)^2 \tag{28.31}$$

Correlation coefficient (r)

The correlation coefficient indicates how well x and y correlate with each other.

(i) When the data points are scattered all over xy plane, there is no correlation.
(ii) When all data points coincide with line AB, there is perfect correlation.
(iii) The correlation coefficient has value between these two extreme conditions viz. no correlation and perfect correlation.

$$r = \frac{n\Sigma x_i y_i - (\Sigma x_i)(\Sigma y_i)}{\sqrt{n\Sigma x_i^2 - (\Sigma x_i)^2}\ \sqrt{n\Sigma y_i^2 - (\Sigma y_i)^2}} \tag{28.32}$$

r = correlation coefficient

(i) The correlation coefficient 'r' has a range of $-1 \leq r \leq +1$.
(ii) When $r = \pm 1$, there is perfect correlation
(iii) When $r = 0$, there is no correlation
(iv) When r is negative, it indicates that line AB has negative slope.

Standard error of estimate

$$s_{y/x} = \sqrt{\frac{S_r}{n-2}} \tag{28.33}$$

($s_{y/x}$) is called standard error of estimate. The subscript 'y/x' indicates that the error is for a predicted value of y corresponding to a particular value of x. It should be noted that the denominator is $(n - 2)$, because of the two degrees of freedom viz. two data derived estimates – a_0 and a_1 – are used to compute S_r.

Dictionary meaning: Regression is a measure of the relationship between the mean value of one variable (e.g., output) and corresponding value of other variable (e.g., time or cost).

Table 28.4 *Linearisation of non-linear relationships – Exponential equation*

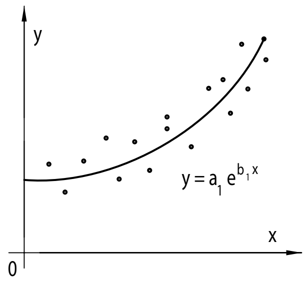
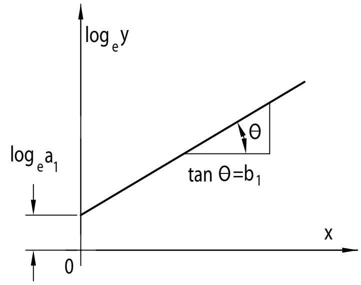
Exponential equation
$$y = a_1 e^{b_1 x} \qquad (28.34)$$ a_1 and b_1 = constant coefficients

Transformation to linear equation
Taking natural logarithm of both side of above equation, $$\log_e y = \log_e a_1 + b_1 x \log_e e$$ Since, $\log_e e = 1$, $$\log_e y = \log_e a_1 + b_1 x$$ The plot of $\log_e y$ versus x is a straight line with a slope of (b_1) and an intercept of $(\log_e a_1)$ on y-axis.
Transformed linear equation
$$\log_e y = \log_e a_1 + b_1 x \qquad (28.35)$$ $(\log_e a_1)$ = intercept on y-axis = constant b_1 = slope of straight line = constant

In this transformed form, linear regression technique can be used to evaluate the constant coefficients a_1 and b_1.

Table 28.5 *Linearisation of non-linear relationships – Power equation*

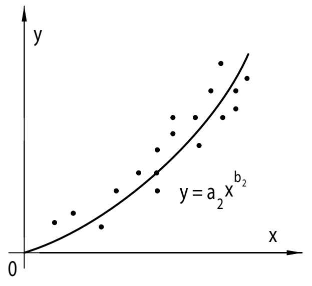
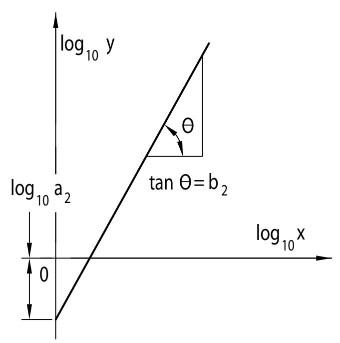
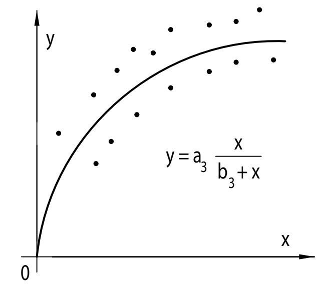
Power equation
$$y = a_2 x^{b_2} \qquad (28.36)$$ a_2 and b_2 = constant coefficients

Transformation to linear equation
Taking logarithm to the base 10, of both side of above equation: $$\log_{10} y = \log_{10} a_2 + b_2 \log_{10} x$$ The plot of $\log_{10} y$ versus $\log_{10} x$ is a straight line with a slope of (b_2) and an intercept of ($\log_{10} a_2$) on y-axis.
Transformed linear equation
$$\log_{10} y = \log_{10} a_2 + b_2 \log_{10} x \qquad (28.37)$$ ($\log_{10} a_2$) = intercept on y-axis = constant b_2 = slope of straight line = constant

In this transformed form, linear regression technique can be used to evaluate the constant coefficients a_2 and b_2.

Table 28.6 *Linearisation of non-linear relationships – 'Saturation-Growth-Rate' equation*

Saturation-Growth-Rate equation

$$y = a_3 \frac{x}{b_3 + x} \qquad (28.38)$$

a_3 and b_3 = constant coefficients



Transformation to linear equation

Inverting both side of above equation,

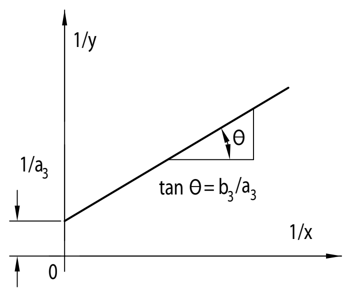
$$\frac{1}{y} = \frac{1}{a_3} + \left(\frac{b_3}{a_3}\right)\frac{1}{x}$$

The plot of $(1/y)$ versus $(1/x)$ is a straight line with a slope of (b_3/a_3) and an intercept of $(1/a_3)$ on y-axis.

Transformed linear equation

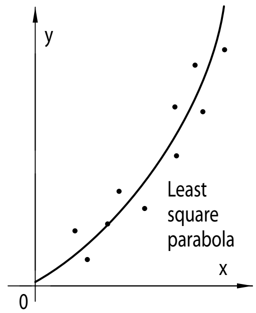
$$\frac{1}{y} = \frac{1}{a_3} + \left(\frac{b_3}{a_3}\right)\frac{1}{x} \qquad (28.39)$$

$(1/a_3)$ = intercept on y-axis = constant
(b_3/a_3) = slope of straight line = constant



In this transformed form, linear regression technique can be used to evaluate the constant coefficients a_3 and b_3.

Table 28.7 *Polynomial regression – Second-order polynomial*



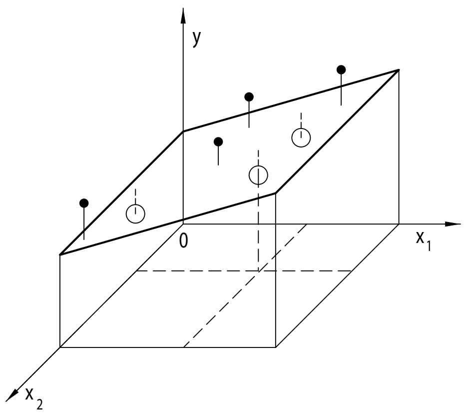
Second-order polynomial (quadratic) equation	
$y = a_0 + a_1 x + a_2 x^2$	(28.40)
Fitting of equation	
$y = a_0 + a_1 x + a_2 x^2 + e$	(28.41)
Sum of squares of residuals	
$S_r = \sum_{i=1}^{n} (y_i - a_0 - a_1 x_i - a_2 x_i^2)^2$	(28.42)
Criterion of best fit	
The best line is based on the criterion of minimising (S_r) that is 'least square' method. The coefficients a_0, a_1 and a_2 are obtained by differentiating (S_r) with respect to a_0, a_1 and a_2 respectively and setting the results to zero. $\frac{\partial S_r}{\partial a_0} = -2\Sigma(y_i - a_0 - a_1 x_i - a_2 x_i^2) = 0$ $\frac{\partial S_r}{\partial a_1} = -2\Sigma[x_i(y_i - a_0 - a_1 x_i - a_2 x_i^2)] = 0$ $\frac{\partial S_r}{\partial a_2} = -2\Sigma[x_i^2(y_i - a_0 - a_1 x_i - a_2 x_i^2)] = 0$	
Equations for a_1, a_0 and a_2	
$(n)a_0 + (\Sigma x_i)a_1 + (\Sigma x_i^2)a_2 = \Sigma y_i$	(28.43)
$(\Sigma x_i)a_0 + (\Sigma x_i^2)a_1 + (\Sigma x_i^3)a_2 = \Sigma x_i y_i$	(28.44)
$(\Sigma x_i^2)a_0 + (\Sigma x_i^3)a_1 + (\Sigma x_i^4)a_2 = \Sigma x_i^2 y_i$	(28.45)
The above three equations are linear and have three unknowns viz. a_0, a_1 and a_2. Therefore, the problem of determining the least-square second-order polynomial is equivalent to solving a set of three simultaneous equations.	

(*Contd.*)

(*Contd.*)

$$\begin{bmatrix} n & \Sigma x_i & \Sigma x_i^2 \\ \Sigma x_i & \Sigma x_i^2 & \Sigma x_i^3 \\ \Sigma x_i^2 & \Sigma x_i^3 & \Sigma x_i^4 \end{bmatrix} \begin{Bmatrix} a_0 \\ a_1 \\ a_2 \end{Bmatrix} = \begin{Bmatrix} \Sigma y_i \\ \Sigma x_i y_i \\ \Sigma x_i^2 y_i \end{Bmatrix} \quad (28.46)$$
The above equations are solved by using techniques like Gauss elimination method.
Standard error of estimate
$$s_{y/x} = \sqrt{\frac{S_r}{n-3}} \quad (28.47)$$
It should be noted that the denominator is $(n - 3)$; because of the three degrees of freedom viz. three data derived estimates – a_0, a_1 and a_2 – are used to compute S_r.
Coefficient of determination (r^2)
$$r^2 = \frac{S_t - S_r}{S_t}$$ $$S_t = \Sigma(y_i - \mu_y)^2$$

Table 28.8 *Multiple linear regression*



Equation for y as a linear function of two independent variable x_1 and x_2
$$y = a_0 + a_1x_1 + a_2x_2 \quad (28.48)$$ The regression 'line' becomes a 'plane' in this two-dimensional case.
Fitting of equation
$$y = a_0 + a_1x_1 + a_2x_2 + e \quad (28.49)$$

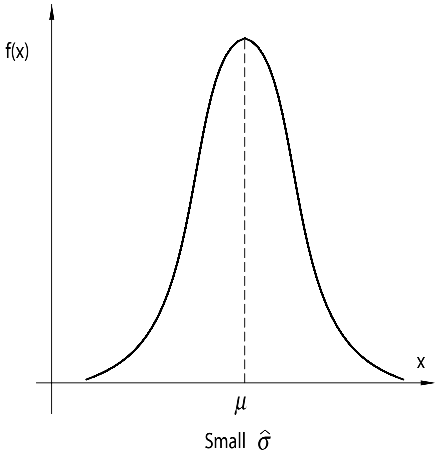
(*Contd.*)

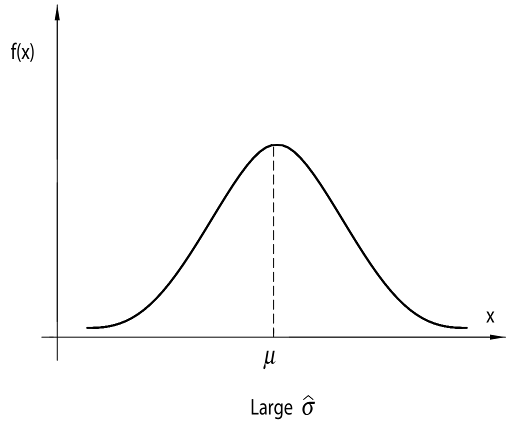
(*Contd.*)

Sum of squares of residuals	
$$S_r = \sum_{i=1}^{n} (y_i - a_0 - a_1 x_{1i} - a_2 x_{2i})^2$$	(28.50)
Criterion of best fit	
The best line is based on the criterion of minimising (S_r) that is 'least square' method. The coefficients a_0, a_1 and a_2 are obtained by differentiating (S_r) with respect to a_0, a_1 and a_2 respectively and setting the results to zero. $$\frac{\partial S_r}{\partial a_0} = -2\Sigma(y_i - a_0 - a_1 x_{1i} - a_2 x_{2i}) = 0$$ $$\frac{\partial S_r}{\partial a_1} = -2\Sigma[x_{1i}(y_i - a_0 - a_1 x_{1i} - a_2 x_{2i})] = 0$$ $$\frac{\partial S_r}{\partial a_2} = -2\Sigma[x_{2i}(y_i - a_0 - a_1 x_{1i} - a_2 x_{2i})] = 0$$	
Equations for a_1, a_0 and a_2 in matrix form	
$$\begin{bmatrix} n & \Sigma x_{1i} & \Sigma x_{2i} \\ \Sigma x_{1i} & \Sigma x_{1i}^2 & \Sigma x_{1i} x_{2i} \\ \Sigma x_{2i} & \Sigma x_{1i} x_{2i} & \Sigma x_{2i}^2 \end{bmatrix} \begin{Bmatrix} a_0 \\ a_1 \\ a_2 \end{Bmatrix} = \begin{Bmatrix} \Sigma y_i \\ \Sigma x_{1i} y_i \\ \Sigma x_{2i} y_i \end{Bmatrix}$$	(28.51)
The above three equations are linear and have three unknowns viz. a_0, a_1 and a_2. Therefore, the problem of determining the least-square polynomial is equivalent to solving a set of three simultaneous equations. The above equations are solved by using techniques like Gauss elimination method.	
Standard error of estimate	
$$s_{y/x} = \sqrt{\frac{S_r}{n-3}}$$	(28.52)
It should be noted that the denominator is (n – 3); because of the three degrees of freedom viz. three data derived estimates – a_0, a_1 and a_2 – are used to compute S_r.	
Coefficient of determination (r^2)	
$$r^2 = \frac{S_t - S_r}{S_t}$$ $$S_t = \Sigma(y_i - \mu_y)^2$$	

28.3 NORMAL DISTRIBUTION

Table 28.9 *Normal distribution*





Equation of normal (Gaussian) curve

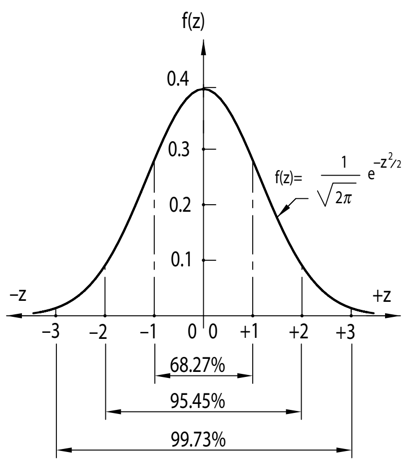
$$f(x) = \frac{1}{\hat{\sigma}\sqrt{2\pi}} \exp\left[-\frac{1}{2}\left(\frac{x-\mu}{\hat{\sigma}}\right)^2\right] \qquad (28.53)$$

Equation in terms of standard variable (z)

$$f(z) = \frac{1}{\sqrt{2\pi}} \exp\left[-\frac{z^2}{2}\right] \qquad (28.54)$$

z = standard variable

$$z = \frac{x-\mu}{\hat{\sigma}}$$



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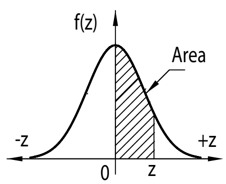
Characteristics of normal curve	
The total area below normal curve from $z = -\infty$ to $z = +\infty$ is one or unity. The curve is symmetrical about the ordinate. Therefore, the area below normal curve from $z = -\infty$ to $z = 0$ is 0.5. Also, the area below normal curve from $z = 0$ to $z = +\infty$ is 0.5. The areas included between different values of z are as follows:	
z	**Percentage of total area**
$z = -1$ to $z = +1$	68.27 % (28.55)
$z = -2$ to $z = +2$	95.45 % (28.56)
$z = -3$ to $z = +3$	99.73 % (28.57)
Applications	
Normal distribution is widely used in mechanical engineering. The dimensions of the components manufactured on shop floor are normally distributed. The test results of properties of materials such as tensile strength, yield strength or endurance limit have been found to follow normal distribution. Therefore, normal distribution is used in statistical analysis of tolerances and probabilistic approach to margin of safety.	

Table 28.10 *Areas under normal curve from 0 to z*



z	0	1	2	3	4	5	6	7	8	9
0.0	.0000	.0040	.0080	.0120	.0160	.0199	.0239	.0279	.0319	.0359
0.1	.0398	.0438	.0478	.0517	.0557	.0596	.0636	.0675	.0714	.0754
0.2	.0793	.0832	.0871	.0910	.0948	.0987	.1026	.1064	.1103	.1141
0.3	.1179	.1217	.1255	.1293	.1331	.1368	.1406	.1443	.1480	.1517
0.4	.1554	.1591	.1628	.1664	.1700	.1736	.1772	.1808	.1844	.1879
0.5	.1915	.1950	.1985	.2019	.2054	.2088	.2123	.2157	.2190	.2224
0.6	.2258	.2291	.2324	.2357	.2389	.2422	.2454	.2486	.2518	.2549
0.7	.2580	.2612	.2642	.2673	.2704	.2734	.2764	.2794	.2823	.2852
0.8	.2881	.2910	.2939	.2967	.2996	.3023	.3051	.3078	.3106	.3133
0.9	.3159	.3186	.3212	.3238	.3264	.3289	.3315	.3340	.3365	.3389
1.0	.3413	.3438	.3461	.3485	.3508	.3531	.3554	.3577	.3599	.3621

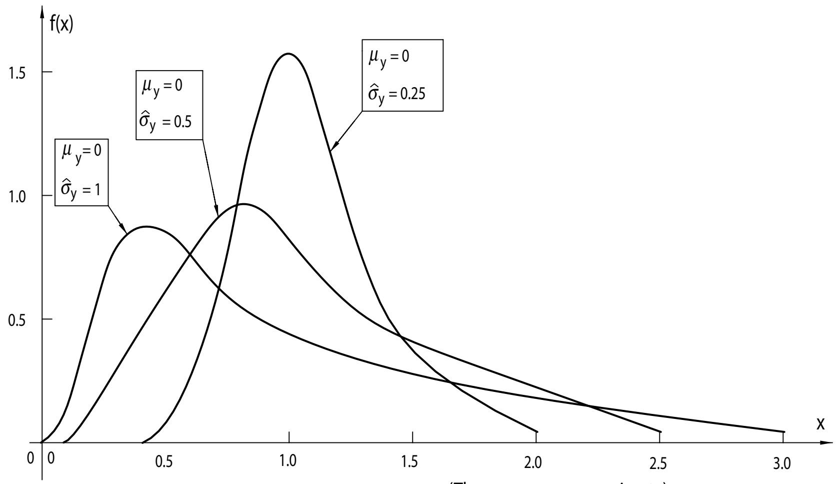
(*Contd.*)

(*Contd.*)

1.1	.3643	.3665	.3686	.3708	.3729	.3749	.3770	.3790	.3810	.3830
1.2	.3849	.3869	.3888	.3907	.3925	.3944	.3962	.3980	.3997	.4015
1.3	.4032	.4049	.4066	.4082	.4099	.4115	.4131	.4147	.4162	.4177
1.4	.4192	.4207	.4222	.4236	.4251	.4265	.4279	.4292	.4306	.4319
1.5	.4332	.4345	.4357	.4370	.4382	.4394	.4406	.4418	.4429	.4441
1.6	.4452	.4463	.4474	.4484	.4495	.4505	.4515	.4525	.4535	.4545
1.7	.4554	.4564	.4573	.4582	.4591	.4599	.4608	.4616	.4625	.4633
1.8	.4641	.4649	.4656	.4664	.4671	.4678	.4686	.4693	.4699	.4706
1.9	.4713	.4719	.4726	.4732	.4738	.4744	.4750	.4756	.4761	.4767
2.0	.4772	.4778	.4783	.4788	.4793	.4798	.4803	.4808	.4812	.4817
2.1	.4821	.4826	.4830	.4834	.4838	.4842	.4846	.4850	.4854	.4857
2.2	.4861	.4864	.4868	.4871	.4875	.4878	.4881	.4884	.4887	.4890
2.3	.4893	.4896	.4898	.4901	.4904	.4906	.4909	.4911	.4913	.4916
2.4	.4918	.4920	.4922	.4925	.4927	.4929	.4931	.4932	.4934	.4936
2.5	.4938	.4940	.4941	.4943	.4945	.4946	.4948	.4949	.4951	.4952
2.6	.4953	.4955	.4956	.4957	.4959	.4960	.4961	.4962	.4963	.4964
2.7	.4965	.4966	.4967	.4968	.4969	.4970	.4971	.4972	.4973	.4974
2.8	.4974	.4975	.4976	.4977	.4977	.4978	.4979	.4979	.4980	.4981
2.9	.4981	.4982	.4982	.4983	.4984	.4984	.4985	.4985	.4986	.4986
3.0	.4987	.4987	.4987	.4988	.4988	.4989	.4989	.4989	.4990	.4990
3.1	.4990	.4991	.4991	.4991	.4992	.4992	.4992	.4992	.4993	.4993
3.2	.4993	.4993	.4994	.4994	.4994	.4994	.4994	.4995	.4995	.4995
3.3	.4995	.4995	.4995	.4996	.4996	.4996	.4996	.4996	.4996	.4997
3.4	.4997	.4997	.4997	.4997	.4997	.4997	.4997	.4997	.4997	.4998
3.5	.4998	.4998	.4998	.4998	.4998	.4998	.4998	.4998	.4998	.4998
3.6	.4998	.4998	.4999	.4999	.4999	.4999	.4999	.4999	.4999	.4999
3.7	.4999	.4999	.4999	.4999	.4999	.4999	.4999	.4999	.4999	.4999
3.8	.4999	.4999	.4999	.4999	.4999	.4999	.4999	.4999	.4999	.4999
3.9	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000

28.4 OTHER TYPES OF DISTRIBUTIONS

Table 28.11 *Log - Normal distribution*



(The curves are approximate)

Log - Normal distribution (Galton distribution)

In mechanical engineering analysis, some times the random variables have the following two characteristics:

(i) The distribution is not symmetrical about the mean.

(ii) The variables have only positive values.

The normal distribution cannot be used in such circumstances. Log-normal distribution is applicable in above cases. The log-normal distribution is defined as the distribution in which the logarithms of the variate (see note below) have a normal distribution. Suppose,

$$y = \log_e x$$

If 'y' is normally distributed random variable with a mean of (μ_y) and standard deviation of ($\hat{\sigma}_y$), then 'x' is said to be log-normally distributed.

Equation of Log - Normal distribution

$$f(x) = \frac{1}{x\hat{\sigma}_y\sqrt{2\pi}} \exp\left[-\frac{1}{2}\left(\frac{y-\mu_y}{\hat{\sigma}_y}\right)^2\right] \quad (28.58)$$

where ($y = \log_e x$)

Relationships between (μ_y and μ_x) and ($\hat{\sigma}_y$ and $\hat{\sigma}_x$)

$$\hat{\sigma}_x^2 = \mu_x^2[\exp(\hat{\sigma}_y^2) - 1] \quad (28.59)$$

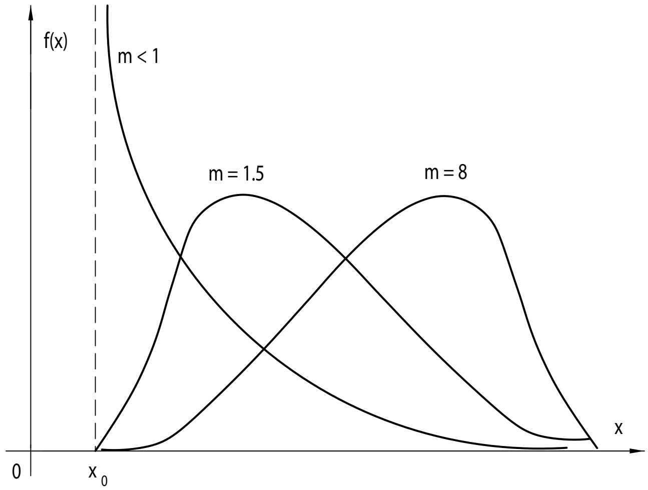
$$\mu_x = \exp\left[\mu_y + \frac{\hat{\sigma}_y^2}{2}\right] \quad (28.60)$$

Applications

The log-normal distribution is applicable in analysis of fatigue life under stress and the wear life of rolling contact bearings. In reliability analysis, this distribution is used to model times to repair a maintainable system.

Note: The term 'variate' is often used as synonym to 'variable'.

Table 28.12 *Weibull distribution*



There are two forms of Weibull distribution viz. three-parameter and two-parameter distributions. The two-parameter distribution is a special case of three-parameter distribution.

Three-parameter distribution

$$f(x) = \frac{m}{\theta - x_0}\left(\frac{x - x_0}{\theta - x_0}\right)^{(m-1)}\left\{\exp\left[-\left(\frac{x - x_0}{\theta - x_0}\right)^{m}\right]\right\} \quad (x \geq x_0) \tag{28.61}$$

x_0 = guaranteed value of x ($x_0 \geq 0$)
θ = characteristic or scale value ($\theta \geq x_0$)
m = shape parameter ($m > 0$)

Note: (i) The Weibull distribution is bonded by x_0 on the lower tail and therefore x_0 is called the 'guaranteed' value of x.

(ii) The characteristic variate θ is similar to mean and represents a value of x below which lie 63.2% of the observations.

(iii) The shape parameter m controls the skewness of the distribution. For large value of 'm', the distribution is skew to the left and for small value of 'm', the distribution is skew to the right. If ($3.3 < m < 3.5$) the distribution is more or less symmetrical and it can be approximated as normal distribution. When $m = 1$, the distribution is exponential.

Two-parameter distribution

If x_0 is taken as zero, the three-parameter density function becomes two-parameter function.

$$f(x) = \frac{m}{\theta}\left(\frac{x}{\theta}\right)^{(m-1)}\left\{\exp\left[-\left(\frac{x}{\theta}\right)^{m}\right]\right\} \quad (x \geq 0) \tag{28.62}$$

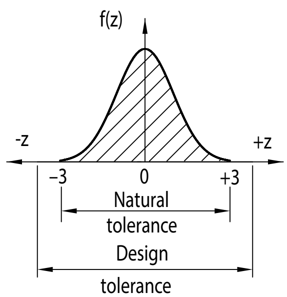
(*Contd.*)

(*Contd.*)

Mean and standard deviation
$$\mu_x = x_0 + (\theta - x_0)\Gamma\left(1 + \frac{1}{m}\right) \quad (28.63)$$ $$\hat{\sigma}_x = (\theta - x_0)\left[\Gamma\left(1 + \frac{2}{m}\right) - \Gamma^2\left(1 + \frac{1}{m}\right)\right]^{1/2} \quad (28.64)$$ where Γ is gamma function. **Note:** The calculation of mean and standard deviation for Weibull distribution is difficult and involves integration by parts and then resorting to table of definite integrals.
Applications
Weibull distribution is frequently used in reliability analysis. It is widely used for predicting fatigue life of rolling contact bearings. The readings required for reliability analysis are obtained from laboratory experiments and field trails. Such readings can be ideally fitted in Weibull distribution due its flexibility. By choosing proper parameters, the curves can take a variety of shapes suitable for experimental data. It contains good approximation of normal distribution as well as exact representation of exponential distribution. It is due to this flexibility, that Weibull distribution is popular in the field of mechanical reliability.

28.5 TOLERANCES

Table 28.13 *Design and natural tolerances*

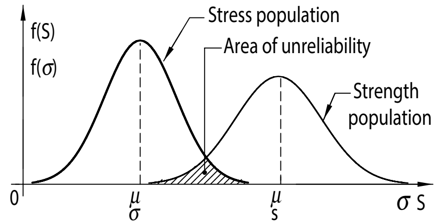


Natural tolerance
The natural tolerance is defined as the actual capabilities of the process, and can be considered as limits within which all but a given allowable fraction of items will fall. In general, the natural tolerance of a process is the spread of the normal curve that includes 99.73% of the total population. Referring to the normal curve, the values of the standard variables z_1 and z_2 corresponding to this population are: –3 and +3. Since, $z = \frac{x - \mu}{\hat{\sigma}}$ or ($x = \mu + z\hat{\sigma}$) $$x_1 = \mu + 3(\hat{\sigma}) \text{ and } x_2 = \mu - 3(\hat{\sigma}) \quad (28.65)$$ Therefore, the natural tolerances are ±3 ($\hat{\sigma}$). $$\text{Natural tolerances} = \pm 3\ (\hat{\sigma}). \quad (28.66)$$
Design tolerance
The design tolerances are specification limits, set somewhat arbitrarily by the designer from considerations of the proper matching of the two components and functioning of the assembly.

28.6 MARGIN OF SAFETY

Table 28.14 *Probabilistic approach to margin of safety*

Strength and stress populations

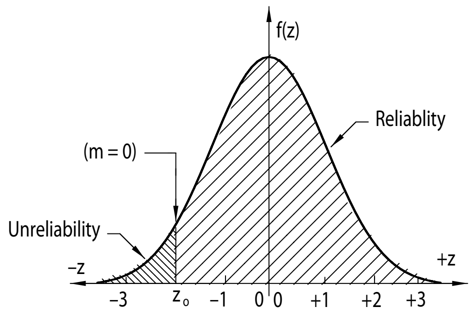


(i) The ultimate tensile strength or yield strength is not constant but subjected to statistical variation. The population of strength; denoted by S is under statistical control. It is normally distributed with a mean of μ_S and standard deviation $\hat{\sigma}_S$.

(ii) The stress induced in the component is not constant but subjected to statistical variation. The population of stress, denoted by σ, is normally distributed with a mean of μ_σ and standard deviation of $\hat{\sigma}_\sigma$.

It should be noted that the mean of strength population must be more than the mean of the stress population. However, there is an overlapping area between these two curves. The forward tail of stress distribution is overlapping the rear tail of the strength population. This overlapping area represents the region of unreliability. Some failures may occur in this region.

Population of margin of safety



A third population of margin of safety is formed by subtracting the population of stress from the population of the strength. The population of margin of safety is denoted by m.

$$\mu_m = \mu_S - \mu_\sigma \tag{28.67}$$

$$\hat{\sigma}_m = \sqrt{(\hat{\sigma}_S)^2 + (\hat{\sigma}_\sigma)^2} \tag{28.68}$$

The normal curve for population of margin of safety is in terms of standard variable z where,

$$z = \frac{m - \mu_m}{\hat{\sigma}_m} \tag{28.69}$$

When the stress is equal to strength, margin of safety is zero and failure may occur. Therefore, condition of failure is given by,

$$m = 0 \tag{28.70}$$

and corresponding standard variable z_0 is given by,

$$z_0 = \frac{0 - \mu_m}{\hat{\sigma}_m} = -\frac{\mu_m}{\hat{\sigma}_m} \tag{28.71}$$

Table 28.15 $(\hat{\sigma}/\mu)$ *values for properties of engineering metals*

Property of engineering metals	$(\hat{\sigma}/\mu)$ value
Ultimate tensile strength (S_{ut})	0.05
Tensile yield strength (S_{yt})	0.07
Endurance limit for steel (S'_e)	0.08
Brinell hardness of steel (BHN)	0.05
Fracture toughness	0.07

Note: The above values are approximate and can be used for design purposes in absence of experimental data.

Table 28.16 $(\hat{\sigma}/\mu)$ *values for modulus of elasticity for engineering metals*

Engineering Metal	$(\hat{\sigma}/\mu)$ value
Steels	≈ 0.03
Aluminium	≈ 0.03
Titanium	≈ 0.09
Nodular cast iron	≈ 0.04

Note: (i) The above values are approximate and can be used for design purposes in absence of experimental data. (ii) For tolerances, [3 $\hat{\sigma} = \pm$ tolerance]

IC Engine Components 29

29.1 ENGINE DESIGN – SELECTION OF PARAMETERS

Table 29.1 *Selection of engine weight*

Specific weight of engine	
$$\text{specific weight} = \frac{\text{engine weight}}{\text{rated power}} \quad (29.1)$$	
Values of (weight/power) ratio for spark ignition engine (kg/kW)	
Small motorcycle engines-two or four stroke	5.5 – 2.5
Passenger car engines-four stroke	4 – 2
Truck engines-four stroke	6.5 – 2.5
Large gas engines-two or four stroke	23 – 35
Values of (weight/power) ratio for compression ignition engines (kg/kW)	
Passenger car engines-four stroke	5 – 2.5
Truck engines (naturally aspirated)-four stroke	7 – 4
Truck engines (turbo-charged)-four stroke	7 – 3.5
Locomotive, industrial and marine engines- two or four stroke	6 – 18
Large engines, marine and stationary engines- two stroke	12 – 50

Table 29.2 *Selection of engine volume*

Specific volume of engine	
$$\text{specific volume} = \frac{\text{engine volume}}{\text{rated power}} \quad (29.2)$$	
Values of (power per unit volume) for spark ignition engine (kW/dm^3)	
Small motorcycle engines-two or four stroke	20 – 60

(*Contd.*)

(*Contd.*)

Passenger car engines-four stroke	20 – 50
Truck engines-four stroke	25 – 30
Large gas engines-two or four stroke	3 – 7
Values of (power per unit volume) for compression ignition engines (kW/dm^3)	
Passenger car engines-four stroke	18 – 22
Truck engines (naturally aspirated)-four stroke	15 – 22
Truck engines (turbo-charged)-four stroke	18 – 26
Locomotive, industrial and marine engines- two or four stroke	5 – 20
Large engines, marine and stationary engines- two stroke	2 – 8

Note: [$1\ dm^3 = 0.001\ m^3 = 1$ litre]

Table 29.3 *Selection of power and type of engine*

Application	Approximate range of engine power (kW)	Spark-ignition or Compression-ignition	Two-stroke or Four-stroke	Air-cooled or Water-cooled
Road vehicles				
Motorcycles and scooters	0.75 – 70	Spark-ignition	Two-stroke/ Four-stroke	Air-cooled
Small passenger cars	15 – 75	Spark-ignition	Four-stroke	Air-cooled/ Water-cooled
Large passenger cars	75 – 200	Spark-ignition	Four-stroke	Water-cooled
Light commercial	35 –150	Spark-ignition / Compression-ignition	Four-stroke	Water-cooled
Heavy commercial (long-distance)	120 – 400	Compression-ignition	Four-stroke	Water-cooled
Off-road vehicles				
Light vehicles (factory or airport)	1.5 – 15	Spark-ignition	Two-stroke/ Four-stroke	Air-cooled/ Water-cooled
Agricultural	3 – 150	Spark-ignition / Compression -ignition	Two-stroke/ Four-stroke	Air-cooled/ Water-cooled
Earth moving	40 – 750	Compression -ignition	Two-stroke/ Four-stroke	Water-cooled
Railroad				
Locomotive	400 –3000	Compression -ignition	Two-stroke/ Four-stroke	Water-cooled

(*Contd.*)

(*Contd.*)

Marine				
Ships	3500 – 22000	Compression-ignition	Two-stroke/ Four-stroke	Water-cooled
Aircrafts				
Airplanes	45 – 2700	Spark-ignition	Four-stroke	Air-cooled
Home-use				
Lawn mowers	0.7 – 3	Spark-ignition	Two-stroke/ Four-stroke	Air-cooled
Light Tractors	2 – 8	Spark-ignition	Four-stroke	Air-cooled
Stationary				
Building service	7 – 400	Compression-ignition	Two-stroke/ Four-stroke	Water-cooled
Electric power	35 –22000	Compression-ignition	Two-stroke/ Four-stroke	Water-cooled

Table 29.4 *Selection of rated speed for engine (N)*

Values of rated speed for spark ignition engines (N) (rpm)	
Small motorcycle engines-two or four stroke	4500 – 7500
Passenger car engines-four stroke	4500 – 6500
Truck engines-four stroke	3600 – 5000
Large gas engines-two or four stroke	300 – 900
Values of rated speed for compression ignition engines (N) (rpm)	
Passenger car engines-four stroke	4000 – 5000
Truck engines (naturally aspirated)-four stroke	2100 – 4000
Truck engines (turbo-charged)-four stroke	2100 – 4000
Locomotive, industrial and marine engines- two or four stroke	425 – 1800
Large engines, marine and stationary engines- two stroke	110 – 400
Mean piston speed	
Mean piston speed is more appropriate parameter than crank shaft speed (in rpm). Resistance to gas flow into the cylinder and inertia forces due to reciprocating parts limit the maximum mean speed of piston. It is usually in the range of 8 to 15 m/s. Automotive engines operate at the higher side of this range and large marine diesel engines operate at the lower side of this range.	

Table 29.5 *Specifications of existing representative engines*

Type of engine	D (mm)	l (mm)	CR	No. of Cylrs.	Maximum torque condition		Maximum power condition		$(sfc)_b$ (kg/kW.h)
					$(p_m)_b$ (MPa)	Speed (rpm)	$(p_m)_b$ (MPa)	Speed (rpm)	
SI/4S/NA	96.8	86	8.6	6	0.910	2500	0.750	4300	0.274
SI/4S/NA	84.5	88	8.5	4	0.966	2800	0.767	5200	0.274
SI/4S/NA	86	86	8.5	4	0.910	3500	0.758	5000	0.274
SI/4S/NA	96	80	9.5	4	0.998	2800	0.796	5400	0.274
SI/4S/TC	92	80	7.5	4	1.241	3800	1.024	5400	0.274
SI/4S/TCAC	96	80	8.7	4	1.356	2900	1.144	5300	0.274
SI/2S/C	64	54	–	2	0.686	7000	0.590	8200	0.340
CI/4S/NA	76.5	80	18.5	4	0.735	2800	0.600	5000	0.246
CI/4S/NA	102	100	17	4	0.886	2200	0.782	3500	0.221
CI/4S/NA	115	135	16	6	0.851	1400	0.777	2700	0.204
CI/4S/TC	115	135	–	6	1.098	1500	0.941	2500	0.203
CI/2S/TC	98.4	114.3	18	3,4,6	1.065	1500	0.952	2500	0.226

Notations:

SI = Spark-ignition, CI = Compression-ignition,
4S = Four-stroke, 2S = Two-stroke,
NA = Naturally aspirated, TC = Tturbo-charged,
TCAC = Turbo-charged and after-cooled,
C = Crankcase compression of scavenging mixture
D = Cylinder bore, l = Length of stroke, CR = Compression ratio,
$(p_m)_b$ = Brake mean effective pressure
$(sfc)_b$ = Brake specific fuel consumption

Table 29.6 *Energy balance for automotive engines at maximum power condition (in percentage of fuel heating value)*

	Spark ignition engine	Compression ignition engines
Energy converted into brake power	25 – 28%	34 – 38%
Energy transferred to cooling medium	17 – 26%	16 –35%
Energy transferred by convection and radiation from external surface of engine	3 –10%	2 – 6%
Energy loss due to incomplete combustion of fuel	2 – 5%	1 – 2%
Energy carried away by exhaust gases	34 – 45%	22 – 35%

Note: [Fuel heating value = Fuel flow rate × Lower calorific value]

Table 29.7 *Selection of brake specific fuel consumption $(sfc)_b$*

Brake specific fuel consumption $(sfc)_b$	
$$(sfc)_b = \frac{\text{mass flow of fuel per unit time}}{\text{brake power}} \quad (29.3)$$	$(sfc)_b$ = specific fuel consumption (kg per brake kW per h)
Typical 'best' values for specific fuel consumption $(sfc)_b$	
(i) For spark-ignition engines, $(sfc)_b$ = 0.27 kg/kW.h (ii) For compression-ignition engines, $(sfc)_b$ = 0.2 kg/kW.h	
Values of $(sfc)_b$ for spark ignition engines (kg/kW.h)	
Small motorcycle engines-two or four stroke	0.35
Passenger car engines-four stroke	0.27
Truck engines-four stroke	0.3
Large gas engines-two or four stroke	0.2
Values of $(sfc)_b$ for compression ignition engines(kg/kW.h)	
Passenger car engines-four stroke	0.25
Truck engines (naturally aspirated)-four stroke	0.21
Truck engines (turbo-charged)-four stroke	0.2
Locomotive, industrial and marine engines-two or four stroke	0.19
Large engines, marine and stationary engines-two stroke	0.18

Table 29.8 *Selection of compression ratio*

Compression ratio	
$$\text{CR} = \frac{V_d + V_c}{V_c} \quad (29.4)$$	CR = compression ratio (dimensionless) V_d = displaced or swept volume (mm^3) V_c = clearance volume (mm^3)
$$V_d = \left(\frac{\pi}{4}\right) D^2 l \quad (29.5)$$	D = cylinder bore (mm) l = length of stroke (mm)
Range of values for compression ratio	
(i) For Spark Ignition engines, CR = 8 to 12 (ii) For Compression Ignition engines, CR = 12 to 24	
Values of compression ratio for spark ignition engines	
Small motorcycle engines-two or four stroke	6 – 11
Passenger car engines-four stroke	8 – 10
Truck engines-four stroke	7 – 9
Large gas engines-two or four stroke	8 – 12

(*Contd.*)

(*Contd.*)

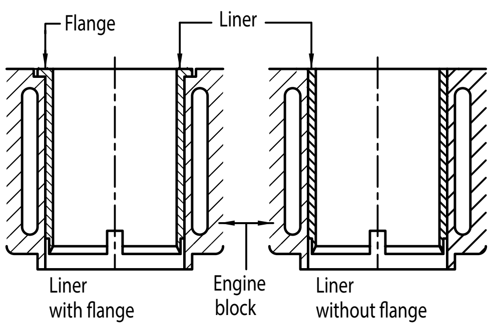
Values of compression ratio for compression ignition engines	
Passenger car engines-four stroke	17 – 23
Truck engines (naturally aspirated)-four stroke	16 – 22
Truck engines (turbo-charged)-four stroke	14 – 20
Locomotive, industrial and marine engines- two or four stroke	12 – 18
Large engines, marine and stationary engines- two stroke	10 – 12

Table 29.9 *Selection of air/fuel ratio*

Air/Fuel ratio
$$\text{Air/Fuel ratio} = \frac{\text{air mass flow rate}}{\text{fuel mass flow rate}} \quad (29.6)$$
Range of values for air/fuel ratio
(i) For spark-ignition engines, air/fuel ratio = 12 to 18 (ii) For compression-ignition engines, air/fuel ratio = 18 to 70

29.2 CYLINDER AND LINER

Table 29.10 *Construction of cylinder liners*

Cylinder liner
A cylinder liner is a separate cylindrical sleeve inserted in the engine block which serves as cylinder. There are two types of cylinder liners- dry and wet. Dry liner does not come in direct contact with the coolant. Instead, it fits closely against the wall of cooling jacket in the engine block. In wet liner, the coolant comes in direct contact with the liner. Wet liners usually have a cooling water space between the engine block and the liner. Dry liners have relatively thin wall compared with the wet liners.
Dry liners


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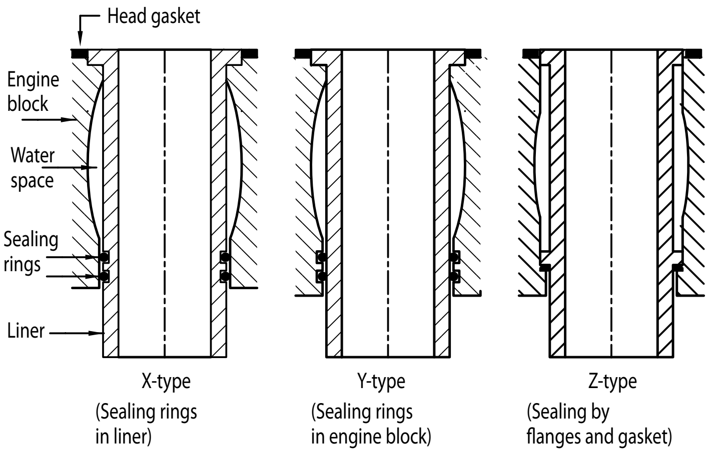
There are two types of dry liners – dry liner with a flange and dry liner without a flange. A dry liner with a flange has a flange at the top that fits into a recess in the surface of engine block. It is not tight fit. On the other hand, dry liner without flange is press or shrink fitted into the bore of engine block.
Wet liners

There are three types of wet liners – *X*, *Y* and *Z* depending upon the sealing arrangement for coolant at the lower end. In *X* and *Y* types, the liner is suspended from a flange at the top that fits into a recess in the surface of engine block. In *X*-type wet liner, the coolant space is sealed at the bottom by sealing rings fitted in machined grooves in the liner. On the other hand, in *Y*-type wet liner, the coolant space is sealed at the bottom by sealing rings fitted in machined grooves in the engine block. In *Z*-type wet liner, the coolant space is sealed at the bottom by a gasket between the flanges of liner and engine block.
Liner materials
The cylinder liners are made of grey cast iron. In some cases, grey cast iron alloyed with chromium, vanadium and molybdenum is used. The liners are manufactured either by centrifugal casting process or by gravity die casting process in metal or sand moulds.

Table 29.11 *Selection of mechanical efficiency*

Mechanical efficiency	
$\eta = \dfrac{BP}{IP}$ (29.7)	η = mechanical efficiency (in fraction) BP = brake power or power developed at the crankshaft (useful power) (W) IP = indicated power or power produced inside the cylinder (W)
Values for mechanical efficiency	
For Automotive engines at wide open or full throttle condition, (i) At speeds below 1800 to 2400 rev./min, η = 90% (ii) At maximum rated speed, η = 75%	

Note: When data is insufficient, the mechanical efficiency may be assumed as 80% or 0.8.

Table 29.12 *Selection of brake mean effective pressure*

Brake mean effective pressure	
$$\text{BP} = \frac{(p_m)_b lAn}{60} \qquad (29.8)$$ $$A = \frac{\pi D^2}{4} \qquad (29.9)$$	BP= brake power or power developed at the crankshaft (useful power) (W) $(p_m)_b$= brake mean effective pressure (MPa or N/mm^2) l = length of stroke (m) (note 'l' is in metres) A = cross-sectional area of cylinder (mm^2) D = bore or inner diameter of cylinder (mm) n = number of working strokes per minute N = engine speed (rpm)
For two-stroke engines $$n = N \qquad (29.10)$$	For four-stroke engines $$n = \frac{N}{2} \qquad (29.11)$$

Broad range of brake mean effective pressure

Spark ignition engines

(i) For naturally aspirated spark-ignition engines, (at engine speed for maximum torque condition) $(p_m)_b$ = 0.850 to 1.05 MPa
(ii) For naturally aspirated spark-ignition engines, (at engine speed for maximum power condition) $(p_m)_b$ = 10 to 15 per cent lower than above values.
(iii) For turbo charged automotive spark-ignition engines (at engine speed for maximum torque condition), $(p_m)_b$ = 1.25 to 1.7 MPa
(iv) For turbo charged automotive spark-ignition engines (at engine speed for maximum power condition), $(p_m)_b$ = 0.9 to 1.4 MPa

Compression ignition engines

(i) For naturally aspirated four-stroke compression–ignition engines, (at engine speed for maximum torque condition) $(p_m)_b$ = 0.7 to 0.9 MPa
(ii) For naturally aspirated four-stroke compression–ignition engines, (at engine speed for maximum power condition), $(p_m)_b$ = 0.7 MPa
(iii) For turbo charged four-stroke compression–ignition engines, $(p_m)_b$ = 1 to 1.2 MPa
(iv) For turbo charged after cooled compression–ignition engines, $(p_m)_b$ = 1.4 MPa
(v) For turbo charged four-stroke compression–ignition engines (at engine speed for maximum power condition), $(p_m)_b$ = 0.85 to 0.95 MPa
(vi) For large low-speed two-stroke compression–ignition engines, $(p_m)_b$ = 1.6 MPa

Brake mean effective pressure for specific applications

Values of brake mean effective pressure for spark ignition engines (MPa)	
Small motorcycle engines-two or four stroke	0.4 – 1
Passenger car engines-four stroke	0.7 – 1
Truck engines-four stroke	0.65 – 0.7
Large gas engines-two or four stroke	0.68 – 1.2
Values of brake mean effective pressure for compression ignition engines (MPa)	
Passenger car engines-four stroke	0.5 – 0.75

(*Contd.*)

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Truck engines (naturally aspirated)-four stroke	0.6 – 0.9
Truck engines (turbo-charged)-four stroke	1.2 – 1.8
Locomotive, industrial and marine engines- two or four stroke	0.7 – 2.3
Large engines, marine and stationary engines- two stroke	0.9 – 1.7

Note: (i) In engine terminology, 'bore' means the inner diameter of cylinder liner. (ii) For selection of engine speed (N) in various applications, refer to Table 29.4. (iii) For selection of (l/*D*) ratio in various applications, refer to Table 29.13.

(iv) For selection of bore diameter, refer to Tables 29.15 and 29.16.

(v) The length of cylinder is more than the length of stroke. There is clearance on both sides of the stroke. The total clearance on two sides can be taken as 15% of stroke length or

$$L = 1.15\, l \tag{29.12}$$

where L = length of cylinder (mm)

Table 29.13 *Selection of (l/d) ratio*

(*l*/*D*) ratio	
$(l/D) \text{ ratio} = \left(\dfrac{l}{D}\right) \quad (29.13)$	l = length of stroke (mm) D = cylinder bore (mm)
Broad range of (*l*/*D*) ratio	
(i) For small and medium size engines, (*l*/*D*) ratio = 1.25 to 0.83 (ii) For large and slow-speed compression-ignition engines, (*l*/*D*) ratio = 2	
(*l*/*D*) ratio for specific applications	
Values of (l/D) ratio for spark ignition engines	
Small motorcycle engines–two or four stroke	1.2 – 0.9
Passenger car engines–four stroke	1.1 – 0.9
Truck engines–four stroke	1.2 – 0.7
Large gas engines–two or four stroke	1.1 – 1.4
Values of (l/D) ratio for compression ignition engines	
Passenger car engines–four stroke	1.2 – 0.9
Truck engines (naturally aspirated)–four stroke	1.3 – 0.8
Truck engines (turbo-charged)–four stroke	1.3 – 0.8
Locomotive, industrial and marine engines–two or four stroke	1.1 – 1.3
Large engines, marine and stationary engines–two stroke	1.2 – 3.0

Note: For insufficient data, the (*l*/*D*) ratio may be assumed from 1.25 to 2.

Table 29.14 *Indicated power*

$$\text{I.P.} = \frac{p_m l A n}{60}$$	(29.14)
p_m = indicated mean effective pressure (N/mm^2 or MPa) l = length of stroke (m) (note 'l' is in metres) A = cross-sectional area of cylinder (mm^2) $\left(\frac{\pi D^2}{4}\right)$ n = number of working strokes/min D = bore or diameter of cylinder (mm)	
$$\eta = \frac{\text{BP}}{\text{IP}} = \frac{(p_m)_b}{p_m}$$	(29.15)
$$p_m = \frac{(p_m)_b}{\eta}$$	(29.16)

Note: For number of working strokes/min, refer to Table 29.12.

Table 29.15 *Selection of cylinder bore for engine*

Values of cylinder bore for spark ignition engines (mm)	
Small motorcycle engines–two or four stroke	50 – 85
Passenger car engines–four stroke	70 – 100
Truck engines–four stroke	90 – 130
Large gas engines–two or four stroke	220 – 450
Values of cylinder bore for compression ignition engines (mm)	
Passenger car engines–four stroke	75 – 100
Truck engines (naturally aspirated)–four stroke	100 – 150
Truck engines (turbo-charged)–four stroke	100 – 150
Locomotive, industrial and marine engines–two or four stroke	150 – 400
Large engines, marine and stationary engines–two stroke	400 – 1000

Table 29.16 *First preference values of bore diameters (mm)*

30, 32, 34, 36, 38, 40, 44, 46, 48, 50, 52, 54, 56, 58, 60, 65, 70, 75, 80, 85, 90, 95, 100, 105, 110, 115, 120, 125, 130, 135, 140, 145, 150, 155, 160, 165, 170, 175, 180, 185, 190, 195, 200, 210, 220, 230, 240, 250, 260, 270, 280, 290, 300, 310, 320, 330, 340 and 350

Table 29.17 *Thickness of cylinder wall*

$$t = \frac{p_{\max} D}{2\sigma_c} + C \tag{29.17}$$

t = thickness of cylinder wall (mm)
$p_{\max}$ = maximum gas pressure inside the cylinder (N/mm^2 or MPa)
D = inner diameter of cylinder or cylinder bore (mm)
σ_c = permissible circumferential (hoop) stress for cylinder material (N/mm^2)
C = reboring allowance (mm) (Table 29.18)

Apparent stresses

$$\sigma_c = \frac{p_{\max} D}{2t} \tag{29.18}$$

$$\sigma_l = \frac{p_{\max} D^2}{(D_o^2 - D^2)} \tag{29.19}$$

σ_c = apparent or principal circumferential stress (N/mm^2)
σ_l = apparent or principal longitudinal stress (N/mm^2)
D_o = outer diameter of cylinder (mm)

Net stresses

Two principal stresses – the circumferential hoop stress σ_c and longitudinal stress σ_l are tensile stresses and they act at right angles to each other. Therefore, net stresses in their respective directions are reduced. The net stresses are given by,

$$(\sigma_c)_{\text{net}} = \sigma_c - \mu\sigma_l \tag{29.20}$$

$$(\sigma_l)_{\text{net}} = \sigma_l - \mu\sigma_c \tag{29.21}$$

$(\sigma_c)_{\text{net}}$ = net circumferential stress (N/mm^2)
$(\sigma_l)_{\text{net}}$ = net longitudinal stress (N/mm^2)
μ = Poisson's ratio (For cast iron, μ = 0.25)

Note: (i) When the data about maximum gas pressure inside the cylinder is not available, it may be assumed as 10 times of indicated mean effective pressure. Or,

$$p_{\max} = 10\ (p_m) = 10\left(\frac{(p_m)_b}{\eta}\right) \tag{29.22}$$

(ii) The circumferential hoop stress σ_c is allowable tensile stress (σ_t). Since the cylinder material is brittle,

$$\sigma_c = \sigma_t = \frac{S_{ut}}{(fs)} \tag{29.23}$$

(iii) When the data about ultimate tensile strength of cylinder material and factor of safety is not available, the allowable circumferential stress (σ_c) may be taken as 35 to 100 N/mm^2.

(iv) Reboring is required to compensate uneven wear on inner wall of the cylinder. Reboring allowance is additional metal thickness over and above that required to withstand maximum gas pressure inside the cylinder. It is provided to compensate for reboring at intervals during the lifetime of cylinder.

Table 29.18 *Reboring allowance (C) for IC engine cylinders*

D	75	100	150	200	250	300	350	400	450	500
C	1.5	2.4	4.0	6.3	8.0	9.5	11.0	12.5	12.5	12.5

Note: D and C are in mm.

Table 29.19 *Empirical relationships for cylinder dimensions*

(i) Thickness of cylinder wall (t)	$0.045\,D + 1.6$ (mm)
(ii) Thickness of dry liner	$0.03\,D$ to $0.035\,D$ (mm)
(iii) Thickness of water jacket wall	$(1/3)\,t$ to $(3/4)\,t$ or $0.032\,D + 1.6$ (mm)
(iv) Water space between outer cylinder wall and inner jacket wall	9 mm for 75 mm cylinder bore to 75 mm for 750 mm cylinder bore, or $0.08\,D + 6.5$ mm
(v) Thickness of cylinder flange (d = Nominal diameter of bolt or stud)	$1.2\,t$ to $1.4\,t$ or $1.25\,d$ to $1.5\,d$
(vi) Radial distance between outer diameter of flange and pitch circle diameter of studs	$(d + 6)$ to $(1.5d)$ mm

Table 29.20 *Thickness of cylinder head*

$$t_h = D\sqrt{\frac{K p_{\max}}{\sigma_c}} \tag{29.24}$$

t_h = thickness of cylinder head (mm)
D = inner diameter of cylinder or cylinder bore (mm)
K = constant ($K = 0.162$)
$p_{\max}$ = maximum gas pressure inside the cylinder (N/mm^2 or MPa)
σ_c = allowable circumferential stress (N/mm^2)

Note: (i) Since the material of cylinder head is brittle,

$$\sigma_c = \sigma_t = \frac{S_{ut}}{(fs)} \tag{29.25}$$

(ii) When the data about ultimate tensile strength of cylinder head material and factor of safety is not available, the allowable circumferential stress (σ_c) may be taken as 30 to 50 N/mm^2.

Table 29.21 *Studs for cylinder head*

Number of studs	
Minimum number of studs = 0.01 D + 4	(29.26)
Maximum number of studs = 0.02 D + 4	(29.27)
Diameter of studs	
$\left(\frac{\pi D^2}{4}\right) p_{\max} = z\left(\frac{\pi d_c^2}{4}\right)\sigma_t$	(29.28)
$d \approx \frac{d_c}{0.8}$	(29.29)
D = inner diameter of cylinder or cylinder bore (mm) $p_{\max}$ = maximum gas pressure inside the cylinder (N/mm^2 or MPa) d_c = core or minor diameter of studs (mm) z = number of studs σ_t = allowable tensile stress for stud material (N/mm^2) d = nominal diameter of studs (mm)	
Pitch of studs	
$D_p = D + 3d$	(29.30)
Pitch of studs = $\frac{\pi D_P}{z}$	(29.31)
Minimum pitch = $18\sqrt{d}$	(29.32)
Maximum pitch = $28.5\sqrt{d}$	(29.33)
D_p = pitch circle diameter of studs (mm)	

Note: (i) The studs are made of steel and since the material is ductile,

$$\sigma_t = \frac{S_{yt}}{(fs)} \tag{29.34}$$

(iii) When the data about the yield strength of stud material and factor of safety is not available, the allowable tensile stress (σ_t) may be taken as 35 to 70 N/mm^2.

29.3 PISTON

Table 29.22 *Construction of piston*

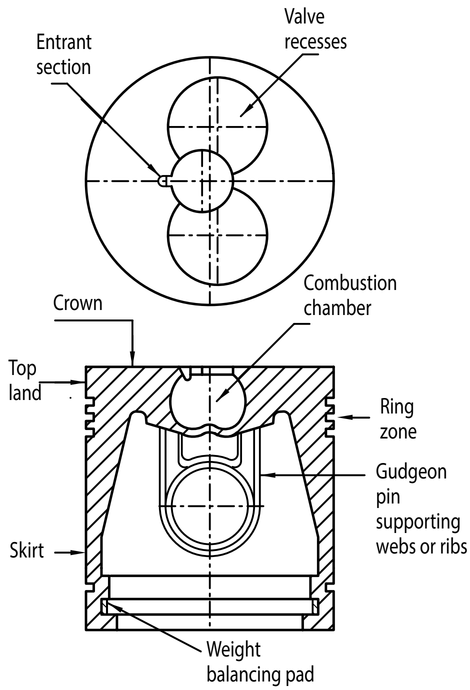
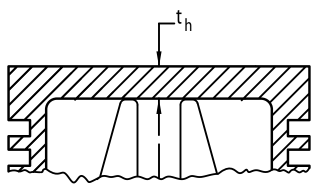
Solid skirt piston

Parts of piston
(i) Piston head or crown: It is the top portion of piston, which withstands the gas pressure inside the cylinder. It has flat, concave or convex shape depending upon the construction of combustion chamber. (ii) Piston rings: They act as seal and prevent the leakage of gas past the piston. Piston rings are also called 'compression' rings. (iii) Oil scraper ring: It prevents the leakage of lubricating oil past the piston into the combustion chamber. (iv) Piston skirt: It is the lower part of the piston below the piston rings which acts as bearing surface for the side thrust exerted by the connecting rod. (v) Piston pin: It connects the piston to the connecting rod. It is also called 'gudgeon' pin or 'wrist' pin.
Piston materials
Commonly used materials for IC engine pistons are cast iron or aluminium alloys. Cast iron pistons are used for moderately rated engines with piston speed below 6 m/s. Aluminium alloy pistons are used for highly rated engines with piston speeds above 6 m/s. The main advantages of aluminium alloy pistons are light weight and high thermal conductivity. Aluminium alloy pistons are either cast or forged.

Table 29.23 *Mechanical and physical properties of aluminium alloys for pistons*

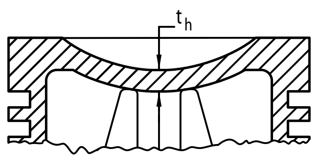
Alloy designation		Hardness (HB)	Tensile strength (MPa or N/mm^2)		Coefficient of thermal expansion (20–200°C) (cm/cm°C $\times 10^{-6}$)
Casting	*Forging*		*Chill casting*	*Forging*	
2285	24850	90–130	225–275	345–410	23–24
4625	46258	90–140	195–245	295–365	20.5–21.5
4928-A	49285	90–125	175–215	225–295	18.5–19.5
4928-B	–	90–125	165–205	–	17–18

Note: The piston castings or forgings are either solution treated or precipitation treated (WP) or stabilized (S).

Table 29.24 *Thickness of piston head*



Flat plate type head



Cup type head

Thickness of piston head – Strength criterion

Grashoff's formula

$$t_h = D\sqrt{\frac{3}{16}\frac{p_{\max}}{\sigma_b}} \tag{29.35}$$

t_h= thickness of piston head (mm)

D = cylinder bore (mm)

$p_{\max}$ = maximum gas pressure or explosion pressure (MPa or N/mm^2)

σ_b = permissible bending stress (N/mm^2)

$$\sigma_b = \sigma_t = \frac{S_{ut}}{(fs)} \tag{29.36}$$

(*Contd.*)

(*Contd.*)

(i) When data is not available, the allowable bending stress (σ_b) for grey cast iron may be taken from 35 to 40 N/mm^2. For aluminium alloy, it may be assumed from 50 to 90 N/mm^2. (ii) Maximum gas pressure (p_{max}) may rise up to 8 MPa. The average value of maximum gas pressure may be taken as 4 to 5 MPa or N/mm^2.
Thickness of piston head – Empirical relationship
Held and Favary formula
$t_h = 0.032\, D + 1.5$ mm (29.37)
Thickness of piston head – Heat dissipation criterion
$t_h = \left[\dfrac{H}{12.56k\,(T_c - T_e)}\right] \times 10^3$ (29.38)
t_h = thickness of piston head (mm) H = amount of heat conducted through piston head (W) k = thermal conductivity factor (W/m/ºC) (Table 29.26) T_c = temperature at the center of piston head (ºC) T_e = temperature at the edge of piston head (ºC)
(i) The approximate values of thermal conductivity factor (k) are as follows, For grey cast iron, k = 46.6 W/m/ºC For aluminium alloy, k = 175 W/m/ºC (ii) The approximate values of permissible temperature difference ($T_c - T_e$) are as follows, For grey cast iron, ($T_c - T_e$) = 220°C For aluminium alloy, ($T_c - T_e$) = 75°C
The amount of heat conducted through piston head (H) is given by, $H = [C \times \text{HCV} \times m \times BP] \times 10^3$ (29.39)
HCV = higher calorific value of fuel (kJ/kg) (Table 29.25) m = mass of fuel used per brake power per second(kg/kW/s) (Table 29.7) BP = brake power of the engine per cylinder (kW) C = ratio of heat absorbed by the piston to the total heat developed in the cylinder (C = 5% or C = 0.05)
(i) The higher calorific values of fuels are as follows: For diesel, HCV = 44 × 10^3 kJ/kg For petrol, HCV = 47 × 10^3 kJ/kg (ii) The average consumption of fuel in diesel engine is 0.24 to 0.30 kg/kW/hr $m = \left[\dfrac{0.24 \text{ to } 0.3}{60 \times 60}\right]$ kg/kW/s (29.40)

Table 29.25 *Properties of fuels*

	Gasoline	Light diesel	Heavy diesel	Natural gas
Fuel formula	$C_nH_{1.87n}$	$C_nH_{1.8n}$	$C_nH_{1.7n}$	$C_nH_{3.8n}N_{0.1n}$
Specific gravity	0.72–0.78	0.84–0.88	0.82–0.95	≈0.79
Specific heat (liquid stage) (kJ/kg.K)	2.4	2.2	1.9	–

(*Contd.*)

(*Contd.*)

Specific heat (vapor stage) (kJ/kg.K)	≈1.7	≈1.7	≈1.7	≈2
Higher calorific value (kJ/kg)	47.3×10^3	44.8×10^3	43.8×10^3	50×10^3
Lower calorific value (kJ/kg)	44.0×10^3	42.5×10^3	41.4×10^3	45×10^3
$(A/F)_s$	14.6	14.5	14.4	14.5

Note: The suffix's' in $(A/F)_s$ means 'stoichiometric'(chemically correct or theoretical) proportions of air and fuel. In this case, there is just enough oxygen for conversion of all fuel into completely oxidised products.

Table 29.26 *Thermal properties of wall materials*

Material	Thermal conductivity (k) (W/m.K)	Density (ρ) (kg/m^3)	Specific heat (c) (J/kg.K)	Thermal diffusivity (α) (m^2/s)
Cast iron	54	7.2×10^3	480	1.57×10^{-5}
Aluminium	155	2.75×10^3	915	6.2×10^{-5}
Reaction-bonded silicon nitride	5–10	2.5×10^3	710	2.8×10^{-6}
Sprayed zirconia	1.2	5.2×10^3	732	3.2×10^{-7}

Table 29.27 *Thumb rules for piston ribs and cup*

Piston ribs

Ribs strengthen the piston head against the gas pressure. They increase the rigidity and prevent distortion of piston head. Ribs transmit a large portion of combustion heat from the piston head to the piston rings.

(i) When the thickness of piston head is 6 mm or less, no ribs are required. When the thickness of piston head is more than 6 mm, suitable number of ribs are required.

$$t_h \le 6 \text{ mm} \quad \text{(no ribs)}$$
$$t_h > 6 \text{ mm} \quad \text{(provide ribs)} \tag{29.41}$$

(ii) The number of ribs is given by,

$$\text{Number of ribs} = 4 \text{ to } 6 \tag{29.42}$$

(iii) The thickness of ribs is given by,

$$t_R = \left(\frac{t_h}{3}\right) \text{ to } \left(\frac{t_h}{2}\right) \tag{29.43}$$

where,
t_R = thickness of ribs (mm)
t_h = thickness of piston head (mm)

Piston cup

Cup provides additional space for combustion of fuel.

(i) When the ratio of stroke length to bore (l/D) is up to 1.5, a cup is required on the top of piston

$$(l/D) \le 1.5 \quad \text{(cup required)}$$
$$(l/D) > 1.5 \quad \text{(no cup required)} \tag{29.44}$$

(ii) The radius of cup is given by,

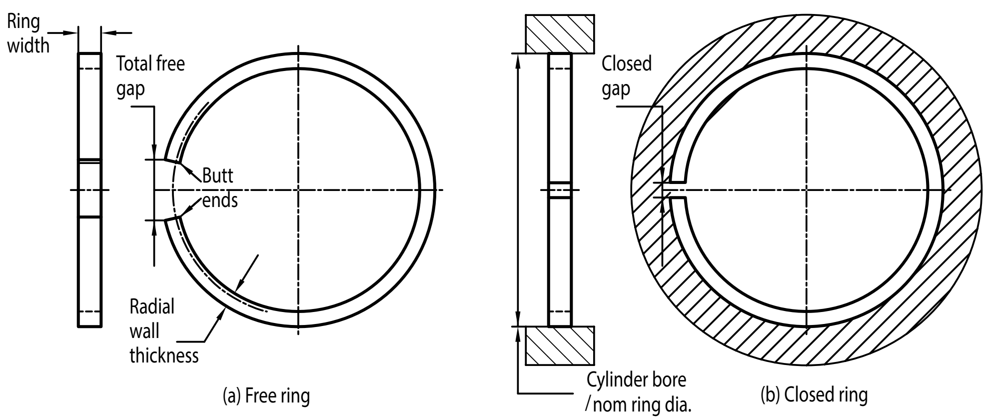
$$\text{radius of cup} = 0.7\,D \tag{29.45}$$

Table 29.28 *Properties of piston ring materials*

Class	Mechanical properties (MPa or N/mm²)		Material	Minimum hardness		Heat treatment and Micro-structure
	Modulus of elasticity	Minimum bending strength		HRB	HRC	
10	90 000	300	Grey cast iron	93	–	Non-heat treated
	90 000	350		95	–	
	1 00 000	390		95	–	
20	1 15 000	450	Grey cast iron	–	23	Heat treated
		450		–	28	
		450		–	40	
		500		–	32	
	1 30 000	650		–	37	
30	1 45 000	550	Carbidic cast iron	–	25	Heat treated pearlitic
		500		–	30	Heat treated martensitic
40	1 60 000	600	Malleable cast iron	95	-	Heat treated pearlitic
		600		–	22	Heat treated martensitic
		600		–	30	Heat treated martensitic
		1000		–	27	Heat treated Carbidic
50	1 60 000	1100	Spheroidal graphite cast iron	–	23	Heat treated martensitic
		1300		–	23	Heat treated martensitic
		1300		–	28	Heat treated martensitic
		1300		95	–	Pearlitic
		1300		97	–	Ferritic
		1300		–	35	Heat treated martensitic

Table 29.29 *Piston ring-terminology*

Nomenclature of piston ring



(a) Free ring
(b) Closed ring

(*Contd.*)

(*Contd.*)

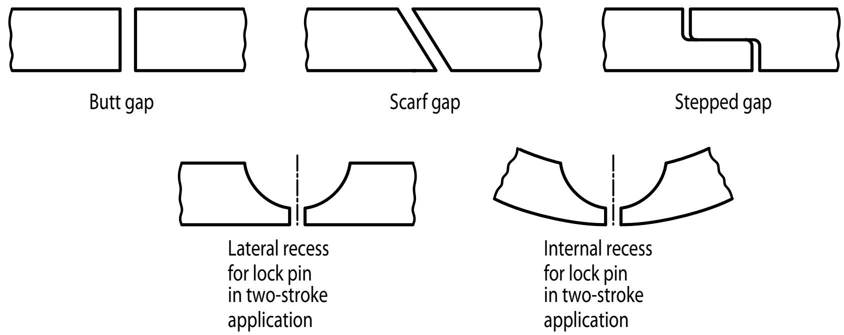
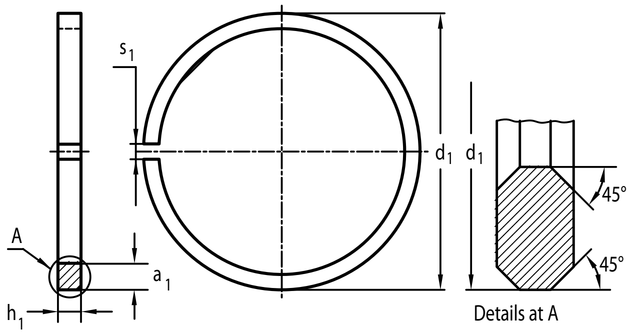
Types of ring gaps


Table 29.30 *R-rings – terminology*

R-rings
A compression ring with a rectangular cross section is called rectangular ring or R-ring. It has simple geometrical shape and it provides adequate seal under normal operating conditions.

d_1 = nominal diameter of ring/ cylinder bore (mm) h_1 = axial width of ring (mm) a_1 = radial wall thickness (mm) s_1 = closed gap for ring (mm)

Note: The internal and sliding edges of R-ring are chamfered.

Table 29.31 *Dimensions of R-rings*

d_1	a_1	h_1			s_1
		1	**2**	**3**	
50	2.1	1.5	2.0	2.5	$0.20^{+0.20}_{-0.00}$
55	2.3	1.5	2.0	2.5	$0.20^{+0.20}_{-0.00}$
60	2.55	1.5	2.0	2.5	$0.25^{+0.20}_{-0.00}$
65	2.75	1.5	2.0	2.5	$0.25^{+0.20}_{-0.00}$
70	2.95	1.5	2.0	2.5	$0.25^{+0.20}_{-0.00}$
75	3.15	1.5	2.0	2.5	$0.30^{+0.20}_{-0.00}$
80	3.35	1.5	2.0	2.5	$0.30^{+0.20}_{-0.00}$
85	3.6	1.5	2.0	2.5	$0.30^{+0.20}_{-0.00}$
90	3.8	2.0	2.5	3.0	$0.40^{+0.25}_{-0.00}$
95	4	2.0	2.5	3.0	$0.40^{+0.25}_{-0.00}$
100	4.2	2.0	2.5	3.0	$0.40^{+0.25}_{-0.00}$
110	4.6	2.5	3.0	3.5	$0.40^{+0.25}_{-0.00}$
120	5	2.5	3.0	3.5	$0.40^{+0.25}_{-0.00}$
130	5.4	2.5	3.0	3.5	$0.50^{+0.25}_{-0.00}$
140	5.7	3.0	3.5	4.0	$0.50^{+0.25}_{-0.00}$
150	6	3.0	3.5	4.0	$0.60^{+0.25}_{-0.00}$
160	6.4	3.0	3.5	4.0	$0.60^{+0.25}_{-0.00}$
170	6.7	3.0	3.5	4.0	$0.60^{+0.25}_{-0.00}$
180	7.1	3.5	4.0	-	$0.70^{+0.25}_{-0.00}$
190	7.4	3.5	4.0	-	$0.70^{+0.25}_{-0.00}$
200	7.7	3.5	4.0	-	$0.70^{+0.25}_{-0.00}$

Note: (i) All dimensions are in mm.

(ii) The tolerances for axial width (h_1) are $\left[h_1{}^{-0.010}_{-0.022}\right]$

(iii) The tolerances for radial wall thickness (a_1) are as follows,

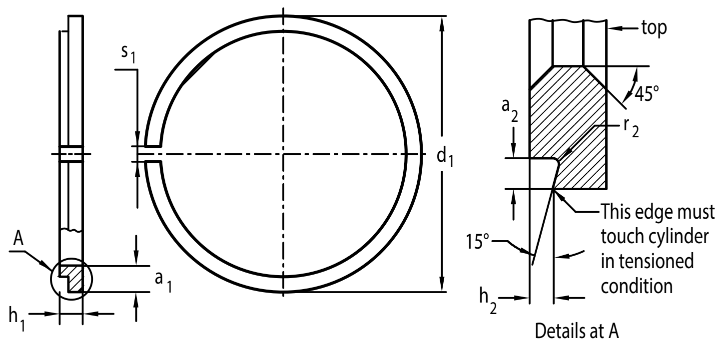
(a) When (a_1) is between 2.1 to 3.15 mm, $\left[a_1{}^{+0.10}_{-0.20}\right]$ with a maximum variation of 0.15 in a ring.

(b) When (a_1) is between 3.35 to 6.7 mm, $\left[a_1{}^{+0.10}_{-0.25}\right]$ with a maximum variation of 0.18 in a ring.

(c) When (a_1) is between 7.1 to 7.7 mm, $\left[a_1{}^{+0.15}_{-0.30}\right]$ with a maximum variation of 0.23 in a ring.

Table 29.32 *N-rings – Terminology*

N-rings
A scrapper compression ring with an undercut step is called Napier ring or N-ring. It is used to scrape the oil from cylinder wall.



d_1 = nominal diameter of ring/ cylinder bore (mm)
h_1 = axial width of ring (mm)
a_1 = radial wall thickness of ring (mm)
h_2 = axial width of groove (mm)
a_2 = radial depth of groove (mm)
r_2 = fillet radius of groove (mm)
s_1 = closed gap for ring (mm)

Note: The internal edges of N-ring are chamfered.

Table 29.33 *Dimensions of N-rings*

d_1	a_1	h_1			s_1	h_2			a_2
		1	**2**	**3**		**1**	**2**	**3**	
50	2.1	2	2.5	–	$0.20^{+0.20}_{-0.00}$	0.5	0.6	–	0.6
55	2.3								0.7
60	2.55				$0.25^{+0.20}_{-0.00}$				
65	2.75	2	2.5	3		0.5	0.6	0.75	0.8
70	2.95								
75	3.15	2.5	3	–	$0.30^{+0.20}_{-0.00}$	0.6	0.75	–	0.9
80	3.35								1
85	3.6								

(*Contd.*)

(*Contd.*)

<table>
<tr><td>90</td><td>3.8</td><td rowspan="3">2.5</td><td rowspan="3">3</td><td rowspan="3">3.5</td><td rowspan="5">$0.40^{+0.25}_{-0.00}$</td><td rowspan="3">0.6</td><td rowspan="3">0.75</td><td rowspan="3">0.9</td><td rowspan="2">1.1</td></tr>
<tr><td>95</td><td>4</td></tr>
<tr><td>100</td><td>4.2</td><td>1.2</td></tr>
<tr><td>110</td><td>4.6</td><td rowspan="2">3</td><td rowspan="2">3.5</td><td rowspan="2">–</td><td rowspan="2">0.75</td><td rowspan="2">0.9</td><td rowspan="2">–</td><td>1.3</td></tr>
<tr><td>120</td><td>5</td><td>1.4</td></tr>
<tr><td>130</td><td>5.4</td><td>3</td><td>3.5</td><td>4</td><td rowspan="2">$0.50^{+0.25}_{-0.00}$</td><td>0.75</td><td>0.9</td><td>1</td><td>1.5</td></tr>
<tr><td>140</td><td>5.7</td><td rowspan="7">3.5</td><td rowspan="7">4</td><td rowspan="7">–</td><td rowspan="7">0.9</td><td rowspan="7">1</td><td rowspan="7">–</td><td>1.6</td></tr>
<tr><td>150</td><td>6</td><td rowspan="3">$0.60^{+0.25}_{-0.00}$</td><td>1.7</td></tr>
<tr><td>160</td><td>6.4</td><td>1.8</td></tr>
<tr><td>170</td><td>6.7</td><td rowspan="2">1.9</td></tr>
<tr><td>180</td><td>7.1</td><td rowspan="3">$0.70^{+0.25}_{-0.00}$</td></tr>
<tr><td>190</td><td>7.4</td><td rowspan="2">2</td></tr>
<tr><td>200</td><td>7.7</td></tr>
</table>

Note: (i) All dimensions are in mm.

(ii) The tolerances for axial width (h_1) are $\left[h_{1\,-0.022}^{\,-0.010}\right]$

(iii) The tolerances for radial wall thickness (a_1) are as follows:

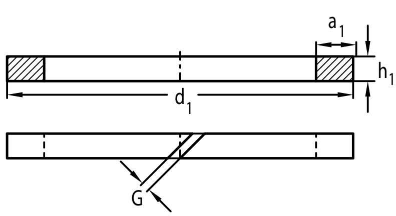
(a) When (a_1) is between 2.1 to 3.15 mm, $\left[a_{1\,-0.20}^{\,+0.10}\right]$ with a maximum variation of 0.15 in a ring.

(b) When (a_1) is between 3.35 to 6.7 mm, $\left[a_{1\,-0.25}^{\,+0.10}\right]$ with a maximum variation of 0.18 in a ring.

(c) When (a_1) is between 7.1 to 7.7 mm, $\left[a_{1\,-0.30}^{\,+0.15}\right]$ with a maximum variation of 0.23 in a ring.

(iv) The tolerances for axial width (h_2) of groove are ($h_2 \pm 0.1$)

Table 29.34 *Thumb rules for piston design*

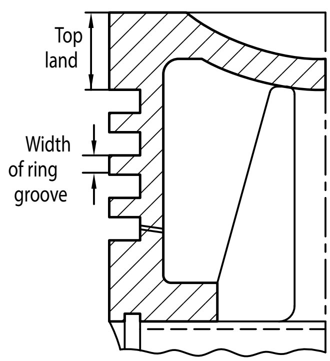


Number of piston rings

There are no strict rules for deciding the number of compression rings. The number of compression rings in automobile and aircraft engines is usually 3 to 4. In stationary diesel engines, 5 to 7 compression rings are used. The number of oil scraper rings is usually 1 to 3.

(*Contd.*)

(*Contd.*)

Radial wall thickness	
$$a_1 = d_1\sqrt{\frac{3p_w}{\sigma_t}}$$	(29.46)
a_1 = radial wall thickness of ring (mm) d_1 = nominal diameter of ring/cylinder bore (mm) p_w = allowable radial pressure on cylinder wall (N/mm^2) σ_t = permissible tensile stress for ring material (N/mm^2) **Note:** (i) The radial wall pressure is usually taken from 0.025 to 0.042 MPa. (ii) The permissible tensile stress for cast iron rings is taken from 85 to 110 N/mm^2.	
Axial width	
$$h_1 = (0.7\,a_1) \text{ to } a_1$$	(29.47)
$$(h_1)_{\min} = \left(\frac{d_1}{10z}\right)$$	(29.48)
h_1 = axial width of ring (mm) z = number of piston rings	
Gap between free ends	
$G = 3.5\,a_1$ to $4\,a_1$ (before assembly)	(29.49)
$G = 0.002\,d_1$ to $0.004\,d_1$ (after assembly in cylinder)	(29.50)
	
Top land	
The distance from the top of piston to the first ring groove is called top land. It is given by, Top land = (t_h) to (1.2 t_h)	(29.51)
t_h = thickness of piston head (mm)	
Width of ring groove	
The distance between two consecutive ring grooves is called width of ring groove and given by, Width of ring groove = 0.75 h_1 to h_1	(29.52)

(*Contd.*)

(*Contd.*)

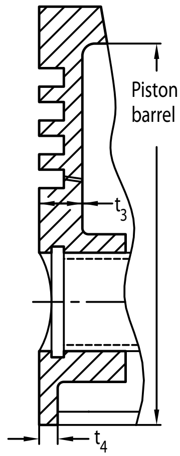
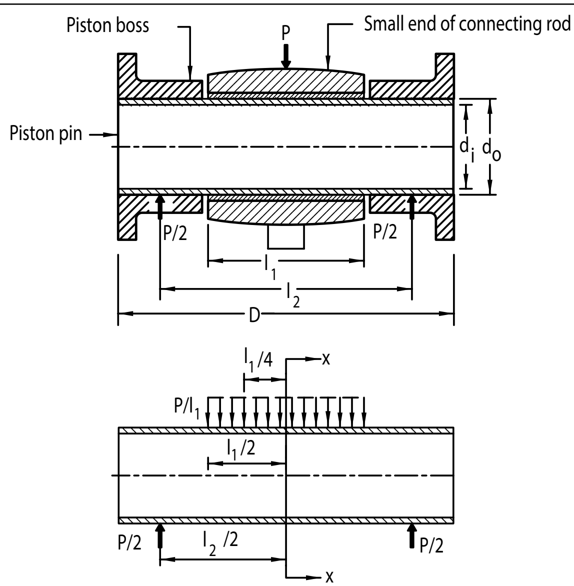
Piston barrel	
	
$t_3 = (0.03\ d_1 + a_1 + 4.9)$	(29.53)
t_3 = thickness of piston barrel at the top end (mm)	
$t_4 = (0.25\ t_3)$ to $(0.35\ t_3)$	(29.54)
t_4 = thickness piston barrel at open end (mm)	

Table 29.35 *Piston pin*



Proportions	
$l_1 = 0.45\ D$	(29.55)
$d_i = 0.6\ d_o$	(29.56)

(*Contd.*)

(*Contd.*)

l_1 = length of piston pin in the bush of small end of connecting rod (mm) D = cylinder bore (mm) d_i = inner diameter of piston pin (mm) d_o = outer diameter of piston pin (mm)
Bearing consideration
$$\left(\frac{\pi D^2}{4}\right) p_{max} = (p_b)_1 \times d_o \times l_1 \quad (29.57)$$
p_{max} = maximum gas pressure inside the cylinder (MPa or N/mm^2) $(p_b)_1$ = bearing pressure at the bushing of small end of connecting rod (MPa or N/mm^2) **Note:** In absence of data, the bearing pressure at the bushing of small end of connecting rod $(p_b)_1$ may be taken as 25 MPa.
Bending consideration
$$\sigma_b = \frac{M_b y}{I} \quad (29.58)$$ $$M_b = \left(\frac{PD}{8}\right) \quad \text{and} \quad P = \left(\frac{\pi D^2}{4}\right) p_{max} \quad (29.59)$$ $$y = \left(\frac{d_o}{2}\right) \quad \text{and} \quad I = \frac{\pi(d_o^4 - d_i^4)}{64} \quad (29.60)$$
Note: The allowable bending stress for the piston pin should not exceed the following values, σ_b = 84 N/mm^2 (for case hardened carbon steel) σ_b = 140 N/mm^2 (for heat treated alloy steels)

Table 29.36 *Piston pin materials – Case hardening steels*

Material	Ultimate tensile strength (Min) (MPa)	Elongation (%)	Symbol
15Cr3	590	13	A
16Mn5Cr4	790	10	B
15C8	500	17	C
15Ni5Cr4Mo1	990	9	D
13Ni13Cr3	840	12	E

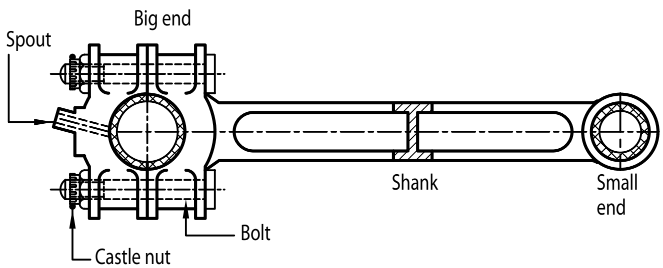
Table 29.37 *Piston pin materials – Nitriding steel*

Material	Ultimate tensile strength (MPa)	0.2% Proof stress (Min) (MPa)	Elongation (%)	Brinell hardness (HB)	Symbol
40Cr7Al10Mo2	690–840	490	18	201–248	F
	790–940	550	16	229–277	
	890–1040	650	15	255–311	

29.4 CONNECTING ROD

Table 29.38 *Connecting rod*

Connecting rod



Connecting rod consists of an eye at the small end to accommodate the piston pin, a long shank and a big end opening split into two parts to accommodate the crank pin. The basic function of connecting rod is to transmit the push and pull forces from the piston pin to the crank pin. Connecting rod transmits the reciprocating motion of piston to the rotary motion of crankshaft. It also transfers lubricating oil from the crank pin to the piston pin and provides splash or jet of oil to piston assembly. The connecting rod of IC engine is made by drop forging process and the outer surfaces are left unfinished. Most internal combustion engines have conventional two-piece connecting rod. The whole rod is forged in one piece; the bearing cap is cut off, faced and bolted in place for final machining of the big end. The small end of the rod is generally made as solid eye and then machined.

Materials for connecting rod

The materials used for connecting rod are either medium carbon steels or alloy steels. The medium carbon steels contain 0.35 to 0.45 percent carbon. The alloy steels include nickel-chromium or chromium- molybdenum steels. Medium carbon steels are used for the connecting rods of industrial engines. Alloy steels are used for connecting rods of automobile and aero engines.

Material	Ultimate tensile strength (MPa)	0.2% Proof stress (Min) (MPa)	Elongation (%)	Limiting ruling section (mm)
40C8	580–680	–	18	–
35Mn6Mo3	690–840 790–940 890–1040 990–1140	490 550 650 750	14 12 12 10	150 100 63 30

(*Contd.*)

(*Contd.*)

35Mn6Mo4	790–940 890–1040 990–1140	550 650 750	16 15 13	150 100 63
40Cr4	690–840 790–940 890–1040	490 550 650	14 12 11	100 63 30
40Cr4Mo2	700–850 800–950 900–1050 1000–1150	490 550 650 750	13 12 11 10	150 100 63 30
40Ni6Cr4Mo2	790–940 890–1040 990–1140 1090–1240	550 650 750 830	16 15 13 11	150 100 63 30

Methods of lubrications

There are two methods of lubrication of bearings at the two ends — splash lubrication and pressure feed lubrication. In splash lubrication, a spout is attached to the big end of connecting rod and set at an angle to the axis of the rod. The spout dips into the sump of lubricating oil during downward motion of connecting rod and splashes the oil as the connecting rod moves up. The splashed up oil finds its way into small end bearing. In pressure feed system, oil is fed under pressure to the crank pin bearing through the holes drilled in crankshaft. From the crank pin bearing, the oil is fed to small end bearing through the hole drilled in the shank of connecting rod.

Length of connecting rod

(L/r) ratio

$$(L/r) \text{ ratio} = \left(\frac{L}{r}\right) \qquad (29.61)$$

L = length of connecting rod (mm)
r = crank radius (mm)

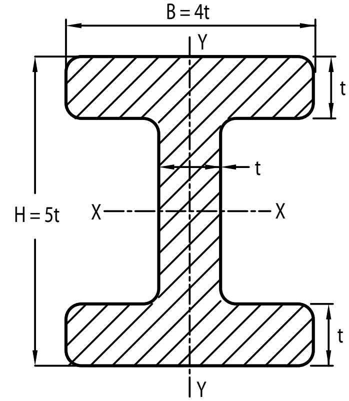
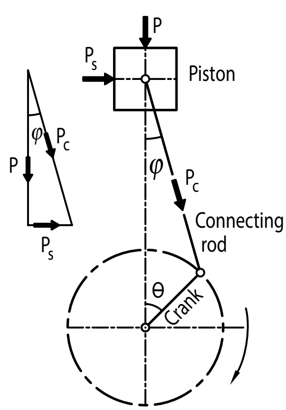
Range of values for (L/r) ratio

(i) For small and medium size engines, (L/r) ratio = 3 to 4
(ii) For large and slow-speed compression-ignition engines, (L/r) ratio = 5 to 9

Cross section of connecting rod

Most of the connecting rods in high-speed engines have I-section. It reduces the weight and inertia forces. It is also easy for forging. Most rods have rifle-drilled hole throughout the length from small end to big end to carry the lubricating oil to piston pin bearing. In low speed engines, circular cross-section is used.

Table 29.39 *Buckling of connecting rod*

<table>
<tr><th colspan="2">Cross section for connecting rod</th></tr>
<tr><td colspan="2">
</td></tr>
<tr><td>$A = 11\ t^2$</td><td>(29.62)</td></tr>
<tr><td>$I_{xx} = \left(\frac{419}{12}\right)t^4$ and $I_{yy} = \left(\frac{131}{12}\right)t^4$</td><td>(29.63)</td></tr>
<tr><td>$\frac{I_{xx}}{I_{yy}} = 3.2$</td><td>(29.64)</td></tr>
<tr><td>$k_{xx} = 1.78t$</td><td>(29.65)</td></tr>
<tr><th colspan="2">Force analysis</th></tr>
<tr><td colspan="2">
</td></tr>
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The maximum gas load occurs shortly after the dead centre position and at this instant ($\varphi = 3.3°$).

$$\varphi = 3.3° \quad \text{and} \quad \cos\varphi = 0.9983 \cong 1 \tag{29.66}$$

$$P_c = \left(\frac{\pi D^2}{4}\right) p_{max} \tag{29.67}$$

P = force acting on piston due to maximum gas pressure (N)
P_s = side thrust on cylinder wall (N)
P_c = force acting on connecting rod (N)
φ = angle of inclination of connecting rod with line of stroke
θ = angle of inclination of crank from top dead centre position
p_{max} = maximum gas pressure inside the cylinder (MPa or N/mm^2)
D = cylinder bore (mm)

Rankine's formula

$$P_{cr} = \frac{\sigma_c A}{1 + a\left(\frac{L}{k_{xx}}\right)^2} \tag{29.68}$$

$$P_{cr} = P_c\,(fs) \tag{29.69}$$

P_{cr} = critical buckling load (N)
σ_c = compressive yield stress (N/mm^2)
A = cross-sectional area of connecting rod (mm^2)
a = constant depending upon material and end fixity coefficient
L = length of connecting rod (mm)
k_{xx} = radius of gyration (mm)
(fs) = factor of safety (usually from 5 to 6)

Note: (i) In the plane of rotation, both ends are hinged and the equivalent length is equal to actual length of connecting rod. The constant 'a' for steel material is given by,

$$a = \frac{1}{7500} \tag{29.70}$$

(ii) For connecting rod made of mild steel and plain carbon steel,

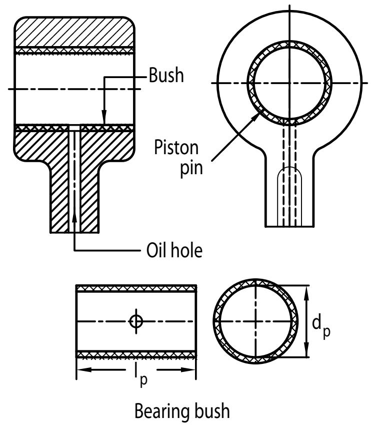
$$\sigma_c = 330 \text{ N/mm}^2 \tag{29.71}$$

(iii) The width B is kept constant throughout the length of connecting rod.
(iv) The height H varies from big end to small end in following way:

at the middle section, $H = 5\,t$
at the small end, $H_1 = 0.75\,H$ to $0.9\,H$
at the big end, $H_2 = 1.1\,H$ to $1.25\,H$ (29.72)

Table 29.40 *Connecting rod – Big and small end bearings*

Small end bearing



The piston pin bearing is usually phosphor bronze bush of 3 mm thickness. It is a solid one-piece bushing made by casting and then finished by grinding and reaming operations

$$P_c = \left(\frac{\pi D^2}{4}\right) p_{\max} \tag{29.73}$$

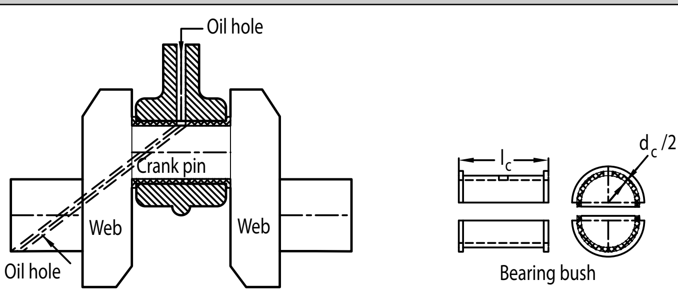
$$P_c = d_p l_p (p_b)_p \tag{29.74}$$

P_c = force acting on connecting rod (N)
$p_{\max}$ = maximum gas pressure inside the cylinder (MPa or N/mm^2)
D = cylinder bore (mm)
d_p = diameter of piston pin or inner diameter of bush on piston pin (mm)
l_p = length of piston pin or length of bush on piston pin (mm)
$(p_b)_p$ = allowable bearing pressure for piston pin bush (N/mm^2)

Note: (i) In absence of catalogue from bearing manufacturers, the allowable bearing pressure for piston pin bush may be taken from 10 to 12.5 MPa.
(ii) The (l/d) ratio for piston pin bush is taken from 1.5 to 2.

$$\left(\frac{l_p}{d_p}\right) = 1.5 \text{ to } 2 \tag{29.75}$$

Big end bearing



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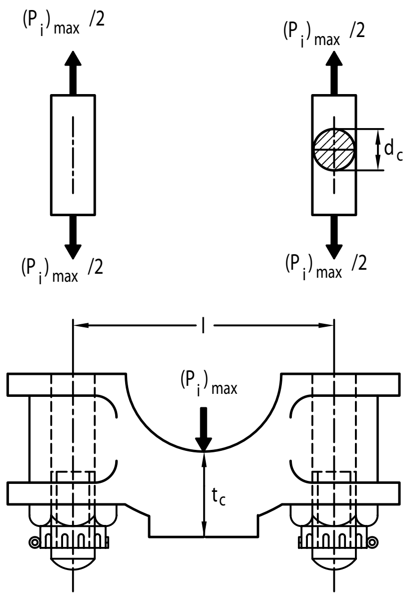
The crank pin bearing is a lined bearing split into two halves. The lined bushing consists of steel backing with a thin lining of bearing material like Babbitt.
$$P_c = d_c l_c (p_b)_c \qquad (29.76)$$
d_c = diameter of crank pin or inner diameter of bush on crank pin (mm) l_c = length of crank pin or length of bush on crank pin (mm) $(p_b)_c$ = allowable bearing pressure for crank pin bush (N/mm^2)

Note: (i) In absence of catalogue from bearing manufacturers, the allowable bearing pressure for crank pin bush may be taken from 5 to 10 N/mm^2.
(ii) The (l/d) ratio for crank pin bush is taken from 1.25 to 1.5.

$$\left(\frac{l_c}{d_c}\right) = 1.25 \text{ to } 1.5 \qquad (29.77)$$

Table 29.41 *Connecting rod – Big end cap and bolts*



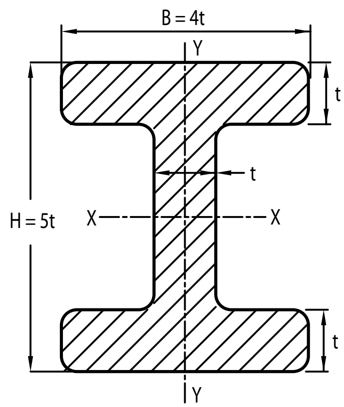
The maximum force acting on the cap and two bolts consists only of inertia force at the top dead centre on the exhaust stroke.
$$P_i = m_r \omega^2 r \left[\cos\theta + \frac{\cos 2\theta}{n_1}\right] \qquad (29.78)$$
P_i = inertia force on cap or bolts (N) m_r = mass of reciprocating parts (kg) ω = angular velocity of crank or angular speed of the engine (rad/s) r = crank radius (m) n_1 = ratio of length of connecting rod to crank radius = $\left(\frac{L}{r}\right)$

(*Contd.*)

(*Contd.*)

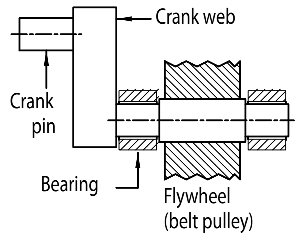
L = length of connecting rod (m) θ = angle of inclination of crank from top dead centre position	
m_r = [mass of piston assembly + $\left(\frac{1}{3}\right)$rd mass of connecting rod]	(29.79)
$$\omega = \left(\frac{2\pi N}{60}\right)$$ N = crank speed (rpm)	(29.80)
$$r = \left(\frac{l}{2}\right)$$ l = length of stroke (m)	(29.81)
The inertia force will be maximum at the top dead centre position where ($\theta = 0$). When ($\theta = 0$), $$\cos\theta = 0 \quad \text{and} \quad \cos 2\theta = 0$$	(29.82)
$$(P_i)_{\max} = m_r \omega^2 r \left[1 + \frac{1}{n_1}\right]$$	(29.83)
Diameter of bolts	
The bolts are subjected to tensile force. Since there are two bolts, each share the inertia force equally	
$$(P_i)_{\max} = 2\left(\frac{\pi d_c^2}{4}\right)\sigma_t$$	(29.84)
d_c = core diameter of bolts (mm) σ_t = permissible tensile stress for bolt material (N/mm^2)	
Thickness of cap	
The cap is subjected to inertia force $(P_i)_{\max}$. It is treated as a beam freely supported at the bolt centers and loaded in a manner intermediate between uniformly distributed and centrally concentrated load in which case the bending moment is ($Wl/6$). $$M_b = \frac{(P_i)_{\max}\, l}{6}$$ M_b = bending moment acting on cap (N-mm) l = span length or distance between the bolt centers (mm) **Note:** The distance between the centers of bolts is given by, l = diameter of crankpin + 2 [thickness of bush (3 mm)] + Nominal diameter of bolt (d) + Clearance (3 mm)	(29.85)
$$\sigma_b = \frac{M_b\, y}{I}$$	(29.86)
$$I = \left[\frac{(b_c)(t_c)^3}{12}\right] \quad \text{and} \quad y = \left(\frac{t_c}{2}\right)$$	(29.87)
b_c = width of cap (mm) It is equal to length of crank pin or big end bearing (l_c). t_c = thickness of big end cap (mm)	

Table 29.42 *Connecting rod –Whipping stresses*

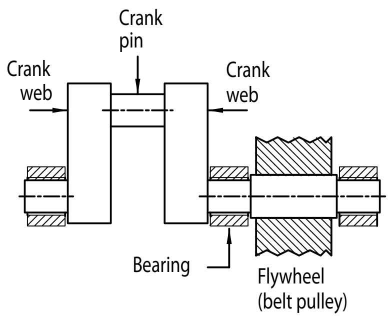
Whipping stresses	
The small end of the connecting rod is subjected to pure translation motion while the big end is subjected to pure rotary motion. The intermediate points on connecting rod move in an elliptical orbit. The lateral oscillations of connecting rod induce inertia forces that act all along the length of connecting rod causing bending. This type of action is called 'whipping'. The bending stress due to inertia force is called 'whipping stress'.	
	
$$\sigma_b = \frac{M_b y}{I}$$	(29.88)
$$I_{xx} = \left(\frac{419}{12}\right)t^4 \quad \text{and} \quad y = \left(\frac{5t}{2}\right)$$	(29.89)
$$(M_b)_{\max} = m_l \omega^2 r \frac{L^2}{9\sqrt{3}}$$	(29.90)
$(M_b)_{max}$ = maximum bending moment [that occurs at a distance of $(L/\sqrt{3})$ from piston pin] (N-m) m_1 = mass of connecting rod per metre length (kg/m) r = crank radius (m) L = length of connecting rod (m) ω = angular velocity of crank or angular speed of the engine (rad/s)	
$m_1 = (11\, t^2)\, \rho$ kg/m	(29.91)
t = thickness of web or flanges (m) ρ = mass density of connecting rod (kg/m^3) For steels, ρ = 7800 kg/m^3	
$$\omega = \left(\frac{2\pi N}{60}\right)$$ N = crank speed (rpm)	

29.5 CRANKSHAFT

Table 29.43 *Crankshaft*



(a) Side crankshaft



(b) Centre crankshaft

Crankshaft converts the reciprocating motion of piston into rotary motion through the connecting rod. The crankshaft consists of three portions- crank pin, crank web and shaft. The big end of connecting rod is attached to the crank pin. The crank web connects the crank pin to shaft portion. The shaft portion rotates in the main bearings and transmits power to the outside source through the belt drive, gear drive or chain drive.

There are two types of crankshafts-side crankshaft and centre crankshaft. The side crankshaft is also called 'overhung' crankshaft. It has only one crank web and requires only two bearings for support. It is used in medium size engines and large size horizontal engines. The centre crankshaft has two webs and three bearings for support. It is used in radial aircraft engines, stationary engines and marine engines. It is more popular in automotive engines. Crankshafts are also classified as single-throw and multi-throw crankshafts depending upon the number of crank pins used in the assembly. Crankshafts used in multi-cylinder engines have more than one crank pin. They are called multi-throw crankshafts. Crankshafts are made by drop forging process.

Materials for crankshaft

Material	Ultimate tensile strength (MPa)	0.2% Proof stress (Min) (MPa)	Elongation (%)	Limiting ruling section (mm)
40C8	600–750 700–850	380 * 480 *	18 17	100 30
55C4	700–850 800–950	460 * 540 *	15 13	63 30

(*Contd.*)

(*Contd.*)

35Mn6Mo3	690–840 790–940 890–1040 990–1140	490 550 650 750	14 12 12 10	150 100 63 30
35Mn6Mo4	790–940 890–1040 990–1140	550 650 750	16 15 13	150 100 63
40Cr4Mo2	700–850 800–950 900–1050 1000–1150	490 550 650 750	13 12 11 10	150 100 63 30
15Cr13Mo6	690–840 790–940 890–1040 990–1140 1090–1240 1540 Min	490 550 650 750 830 1240	14 12 11 10 9 8	150 150 150 150 100 63
35Ni5Cr2	690–840 790–940 890–1040	490 550 650	14 12 10	150 100 63
40Ni10Cr3Mo6	990–1140 1090–1240 1190–1340 1540 Min	750 830 930 1240	12 11 10 8	150 150 150 100

* Minimum yield stress.

Average bearing pressure

Application	***Average radial load per unit area (MPa or N/mm^2)***
Automotive engines	
Main bearings	4–5
Connecting rod bearings	10–15
Diesel engines	
Main bearings	6–12
Connecting rod bearings	8–15

Table 29.44 *Crankshaft – Thumb rule proportions*

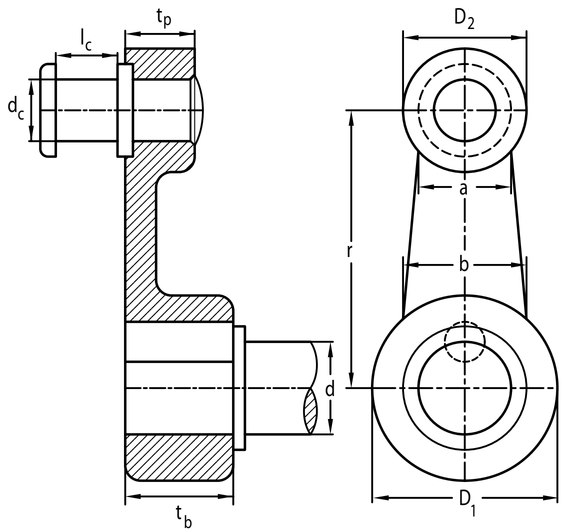
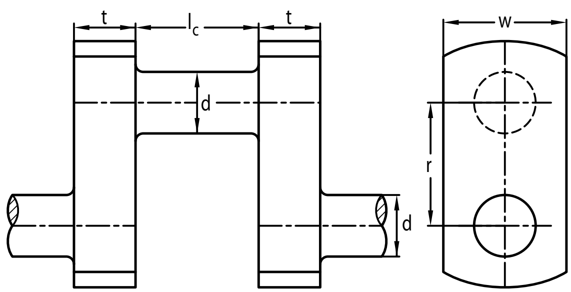
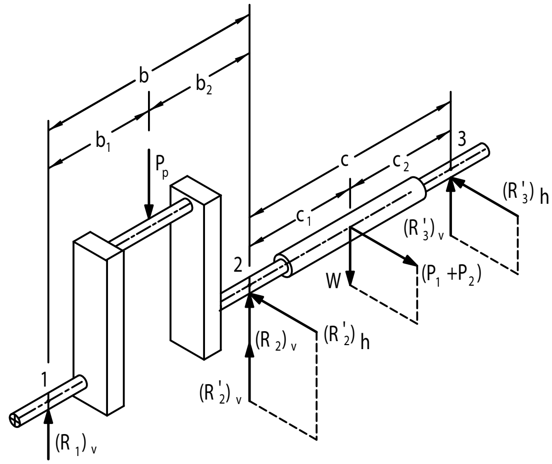
Side crankshaft	
	$a = 1.5d_c$ $b = 1.35d$ $l_c = 1.25d_c$ $D_1 = 1.75d$ $D_2 = 2d_c$ $t_b = 1$ to $1.25d$ $t_p = 1.4d_c$
Centre crankshaft	
	$t = 0.7d$ $l_c = 1.1d$ $w = 1.14d$
r = crank radius	

Table 29.45 *Centre crankshaft at top dead centre position*



Assumptions: (i) The engine is vertical and the crank is at the top dead centre position.
(ii) The belt drive is horizontal.
(iii) The crankshaft is simply supported on bearings.

A crankshaft is subjected to bending and torsional moments due to following three forces:

(i) Force exerted by connecting rod on crank pin.
(ii) Weight of flywheel (W) acting downward in vertical direction.
(iii) Resultant belt tensions acting in horizontal direction ($P_1 + P_2$)

Bearing reactions	
$$P_p = \left(\frac{\pi D^2}{4}\right) p_{\max} \qquad (29.92)$$	P_p = force acting on crank pin (N) D = diameter of piston (mm) $p_{\max}$ = maximum gas pressure inside the cylinder (MPa or N/mm^2)
$$(R_1)_v = \frac{P_p \times b_2}{b} \qquad (29.93)$$ $$(R_2)_v = \frac{P_p \times b_1}{b} \qquad (29.94)$$	$(R_1)_v$ = vertical reaction at bearing-1 (N) $(R_2)_v$ = vertical reaction at bearing-2 (N) b = distance between main bearings 1 and 2 (mm) $b_1 = b_2 = \left(\frac{b}{2}\right)$
$$(R_3')_v = \frac{W \times c_1}{c} \qquad (29.95)$$ $$(R_2')_v = \frac{W \times c_2}{c} \qquad (29.96)$$	W = weight of flywheel (N) $(R_3')_v$ = vertical reaction at bearing-3 (N) $(R_2')_v$ = vertical reaction at bearing-2 (N) c = distance between bearing 2 and 3 (mm) $c_1 = c_2 = \left(\frac{c}{2}\right)$

(*Contd.*)

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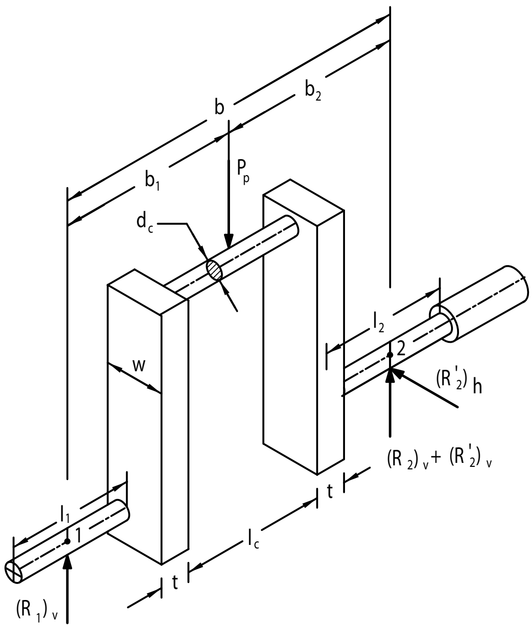
$$(R_3')_h = \frac{(P_1+P_2)\times c_1}{c} \quad (29.97)$$ $$(R_2')_h = \frac{(P_1+P_2)\times c_2}{c} \quad (29.98)$$	P_1 = tension in tight side of belt (N) P_2 = tension in slack side of belt (N) $(R_3')_h$ = horizontal reaction at bearing-3 (N) $(R_2')_h$ = horizontal reaction at bearing-2 (N)
Resultant reactions	
$$R_1 = (R_1)_v \quad (29.99)$$ $$R_2 = \sqrt{[(R_2)_v + (R_2')_v]^2 + [(R_2')_h]^2} \quad (29.100)$$ $$R_3 = \sqrt{[(R_3')_v]^2 + [(R_3')_h]^2} \quad (29.101)$$ **Note:** When distance '*b*' between bearings 1 and 2 is not specified, it is assumed by following empirical relationship, $b = 2 \times$ Piston diameter or $b = 2D$	
Dimensions of crank pin	
	
$$(M_b)_c = (R_1)_v b_1 \quad (29.102)$$ $$(M_b)_c = \left(\frac{\pi d_c^3}{32}\right)\sigma_b \quad (29.103)$$	$(M_b)_c$ = bending moment at the central plane (N-mm) d_c = diameter of crank pin (mm) σ_b = allowable bending stress for crank pin (N/mm^2) **Note:** In absence of data, the allowable bending stress for crank pin can be assumed as 75 N/mm^2.
$$l_c = \frac{P_p}{d_c p_b} \quad (29.104)$$	l_c = length of crank pin (mm) p_b = allowable bearing pressure at the crank pin bush (MPa or N/mm^2) (Table 29.43)

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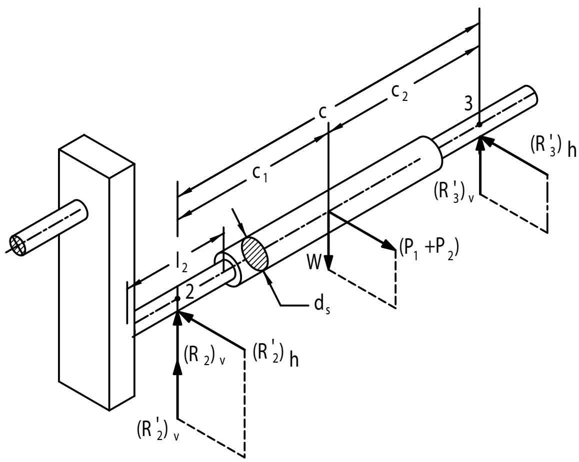
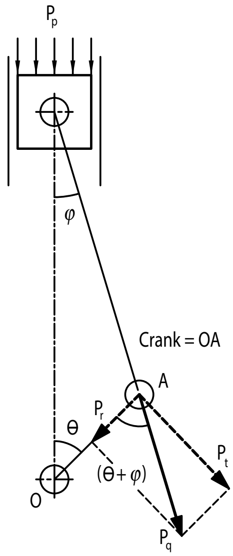
Left-hand crank web	
$t = 0.7\,d_c$ (29.105) $w = 1.14\,d_c$ (29.106)	w = width of crank web (mm) t = thickness of crank web (mm) d_c = diameter of crank pin (mm)
Compressive stresses in left-hand crank web	
$\sigma_c = \dfrac{(R_1)_v}{wt}$ (29.107) $\sigma_b = \dfrac{6(R_1)_v\left[b_1 - \dfrac{l_c}{2} - \dfrac{t}{2}\right]}{wt^2}$ (29.108) $(\sigma_c)_t = \sigma_c + \sigma_b$ (29.109)	σ_c = direct compressive stress (MPa or N/mm^2) σ_b = compressive stress due to bending moment (MPa or N/mm^2) $(\sigma_c)_t$ = total compressive stress (MPa or N/mm^2) **Note:** The total compressive stress should be less than the allowable bending stress
Right-hand crank web	
The right-hand and left-hand webs should be identical from balancing considerations. Therefore, the thickness and width of right-hand crank web are made equal to those of left hand crank web.	
Shaft under flywheel	
	
$(M_b)_v = (R_3')_v c_2$ (29.110) $(M_b)_h = (R_3')_h c_2$ (29.111) $M_b = \sqrt{(M_b)_v^2 + (M_b)_h^2}$ (29.112) $M_b = \left(\dfrac{\pi d_s^3}{32}\right)\sigma_b$ (29.113)	d_s = diameter of shaft under flywheel (mm) σ_b = allowable bending stress (MPa or N/mm^2)

Table 29.46 *Centre crankshaft at angle of maximum torque*



The torque is maximum when the tangential component of force on crank pin is maximum. For this condition, the crank angle from top dead centre position (θ) is usually 25° to 35° for petrol engines and 30° to 40° for diesel engines.

$$P_p = \left(\frac{\pi D^2}{4}\right) p' \quad (29.114)$$

P_p = force acting on piston top due to gas pressure (N)
p' = gas pressure on the piston top for maximum torque condition (MPa or N/mm^2)
D = diameter of piston (mm)

$$\sin \varphi = \frac{\sin \theta}{(L/r)} \quad (29.115)$$

φ = angle of inclination of connecting rod with the line of dead centres (deg)
θ = angle of inclination of crank with line of dead centres (deg)
(L/r) = ratio of length of connecting rod to radius of crank

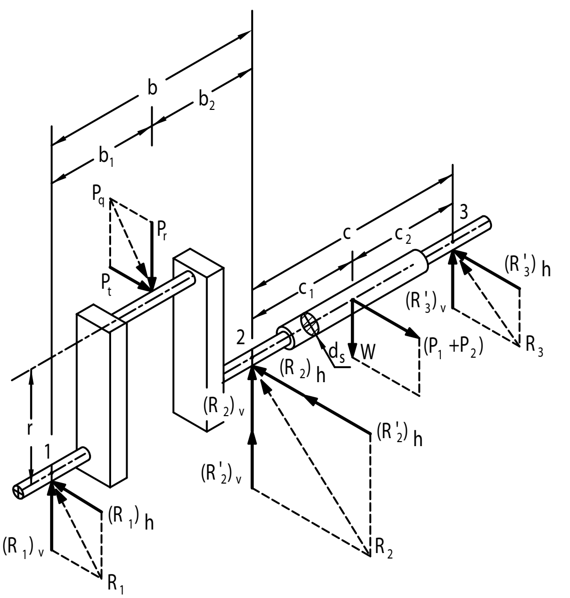
$$P_q = \frac{P_p}{\cos \varphi} \quad (29.116)$$

$$P_t = P_q \sin(\theta + \varphi) \quad (29.117)$$

$$P_r = P_q \cos(\theta + \varphi) \quad (29.118)$$

P_q = thrust on connecting rod (N)
P_t = tangential component of P_q at the crank pin (N)
P_r = radial component of P_q at the crank pin (N)

Bearing reactions



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$(R_1)_h = \dfrac{P_t \times b_2}{b}$ (29.119) $(R_2)_h = \dfrac{P_t \times b_1}{b}$ (29.120)	$(R_1)_h$ = horizontal reaction at bearing-1 due to (P_t) (N) $(R_2)_h$ = horizontal reaction at bearing-2 due to (P_t) (N)
$(R_1)_v = \dfrac{P_r \times b_2}{b}$ (29.121) $(R_2)_v = \dfrac{P_r \times b_1}{b}$ (29.122)	$(R_1)_v$ = vertical reaction at bearing-1 due to (P_r) (N) $(R_2)_v$ = vertical reaction at bearing-2 due to (P_r) (N)
$(R_2')_v = \dfrac{W \times c_2}{c}$ (29.123) $(R_3')_v = \dfrac{W \times c_1}{c}$ (29.124)	$(R_2')_V$ = vertical reaction at bearing-2 due to (W) (N) $(R_3')_V$ = vertical reaction at bearing-3 due to (W) (N)
$(R_2')_h = \dfrac{(P_1 + P_2) \times c_2}{c}$ (29.125) $(R_3')_h = \dfrac{(P_1 + P_2) \times c_1}{c}$ (29.126)	$(R_2')_h$ = horizontal reaction at bearing-2 due to $(P_1 + P_2)$ (N) $(R_3')_h$ = horizontal reaction at bearing-3 due to $(P_1 + P_2)$ (N)
Resultant reactions	
$R_1 = \sqrt{[(R_1)_v]^2 + [(R_1)_h]^2}$ (29.127) $R_2 = \sqrt{[(R_2)_v + (R_2')_v]^2 + [(R_2)_h + (R_2')_h]^2}$ (29.128) $R_3 = \sqrt{[(R_3')_v]^2 + [(R_3')_h]^2}$ (29.129)	
Shaft under flywheel	
$M_b = (R_3) \times c_2$ (29.130) $M_t = P_t \times r$ (29.131) $d_s^3 = \dfrac{16}{\pi\tau}\sqrt{(M_b)^2 + (M_t)^2}$ (29.132) $d_s^3 = \dfrac{16}{\pi\tau}\sqrt{[R_3 \times c_2]^2 + [P_t \times r]^2}$ (29.133)	M_b = bending moment at central plane (N-mm) M_t = torsional moment at central plane (N-mm) d_s = diameter of shaft under flywheel (mm) τ = allowable shear stress (MPa or N/mm^2) **Note:** In absence of data, the allowable shear stress can be taken as 40 N/mm^2

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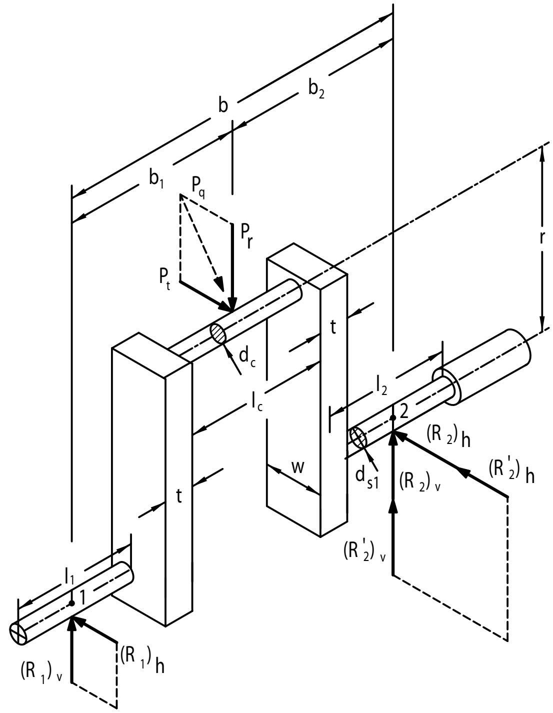
Dimensions of crank pin	
	
$M_b = (R_1)_v \times b_1$ (29.134) $M_t = (R_1)_h \times r$ (29.135) $d_c^3 = \dfrac{16}{\pi\tau}\sqrt{(M_b)^2 + (M_t)^2}$ (29.136) $d_c^3 = \dfrac{16}{\pi\tau}\sqrt{[(R_1)_v \times b_1]^2 + [(R_1)_h \times r]^2}$ (29.137)	M_b = bending moment at central plane (N-mm) M_t = torsional moment at central plane (N-mm) d_c = diameter of crank pin (mm) τ = allowable shear stress (MPa or N/mm^2) **Note:** In absence of data, the allowable shear stress can be taken as 40 N/mm^2
$l_c = \dfrac{P_q}{d_c p_b}$ (29.138)	l_c = length of the crank pin (mm) p_b = allowable bearing pressure at the crank pin bush (MPa or N/mm^2) (Table 29.43)
Shaft under flywheel at the juncture of right-hand crank web	
$(M_b)_v = (R_1)_v\left[b_1 + \dfrac{l_c}{2} + \dfrac{t}{2}\right] - P_r\left[\dfrac{l_c}{2} + \dfrac{t}{2}\right]$ (29.139) $(M_b)_h = (R_1)_h\left[b_1 + \dfrac{l_c}{2} + \dfrac{t}{2}\right] - P_t\left[\dfrac{l_c}{2} + \dfrac{t}{2}\right]$ (29.140) $M_b = \sqrt{[(M_b)_v]^2 + [(M_b)_h]^2}$ (29.141)	

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$$M_t = P_t \times r \tag{29.142}$$

$$d_{s1}^3 = \frac{16}{\pi\tau}\sqrt{(M_b)^2 + (M_t)^2} \tag{29.143}$$

$(M_b)_v$ = bending moment in vertical plane (N-mm)
$(M_b)_h$ = bending moment in horizontal plane (N-mm)
d_{s1} = diameter of shaft at the juncture of right hand crank web (mm)
τ = allowable shear stress (MPa or N/mm^2)
</td></tr>
<tr><td colspan="2">Right-hand crank web</td></tr>
<tr><td colspan="2">
Right-hand crank web is subjected to following stresses:
(i) Bending stresses in vertical and horizontal planes due to radial component P_r and tangential component P_t respectively.
(ii) Direct compressive stress due to radial component P_r.
(iii) Torsional shear stresses
</td></tr>
<tr><td>

$$(M_b)_r = (R_2)_v\left[b_2 - \frac{l_c}{2} - \frac{t}{2}\right] \tag{29.144}$$

$$(M_b)_r = (\sigma_b)_r\left[\frac{1}{6}wt^2\right] \tag{29.145}$$

</td><td>
$(M_b)_r$ = bending moment due to radial component (N-mm)
$(\sigma_b)_r$ = bending stress due to radial component (MPa or N/mm^2)
</td></tr>
<tr><td>

$$(M_b)_t = P_t\left[r - \frac{d_{s1}}{2}\right] \tag{29.146}$$

$$(M_b)_t = (\sigma_b)_t\left[\frac{1}{6}tw^2\right] \tag{29.147}$$

</td><td>
$(M_b)_t$ = bending moment due to tangential component at the juncture of crank web and shaft (N-mm)
$(\sigma_b)_t$ = bending stress due to tangential component (MPa or N/mm^2)
d_{s1} = diameter of shaft at the juncture of right hand crank web (mm)
</td></tr>
<tr><td>

$$(\sigma_c)_d = \frac{P_r}{2wt} \tag{29.148}$$

</td><td>
$(\sigma_c)_d$ = direct compressive stress due to radial component (MPa or N/mm^2)
</td></tr>
<tr><td>

$$\sigma_c = (\sigma_b)_r + (\sigma_b)_t + (\sigma_c)_d \tag{29.149}$$

</td><td>
σ_c = maximum compressive stress (MPa or N/mm^2)
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$$M_t = (R_1)_h\left[b_1 + \frac{l_c}{2}\right] - P_t\left[\frac{l_c}{2}\right] = (R_2)_h\left[b_2 - \frac{l_c}{2}\right] \tag{29.150}$$

M_t = torsional moment on the arm (N-mm)
</td></tr>
<tr><td>

$$\tau = \frac{M_t}{Z_p} = \frac{4.5M_t}{wt^2} \tag{29.151}$$

</td><td>
τ = torsional shear stress (MPa or N/mm^2)
</td></tr>
<tr><td>

$$(\sigma_c)_{\max} = \frac{\sigma_c}{2} + \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2} \tag{29.152}$$

</td><td>
$(\sigma_c)_{\max}$ = maximum compressive stress
</td></tr>
<tr><td colspan="2">Left-hand crank web</td></tr>
<tr><td colspan="2">
The left-hand crank web is not severely stressed to the extent of right-hand crank web. Therefore, it is not necessary to check the stresses in the left-hand crank web. The thickness and width of left-hand crank web are made equal to those of right-hand crank web from balancing consideration.
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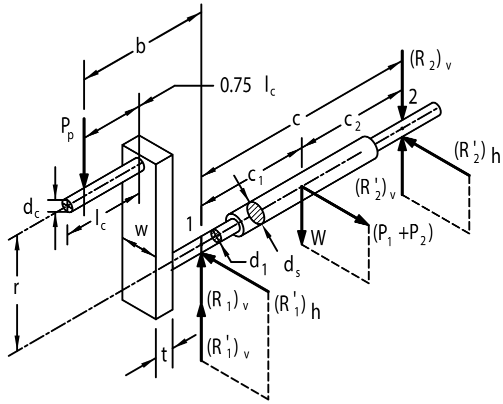
Crankshaft bearing	
$R_2 = \sqrt{[(R_2)_v + (R_2')_v]^2 + [(R_2)_h + (R_2')_h]^2}$ $l_2 = \dfrac{R_2}{d_{s1}\, p_b}$ (29.153)	ds_1 = diameter of journal at bearing 2 (mm) l_2 = length of bearing (mm) p_b = allowable bearing pressure at crank pin bush (MPa or N/mm^2) (Table 29.43)

Table 29.47 *Side crankshaft at top dead centre position*

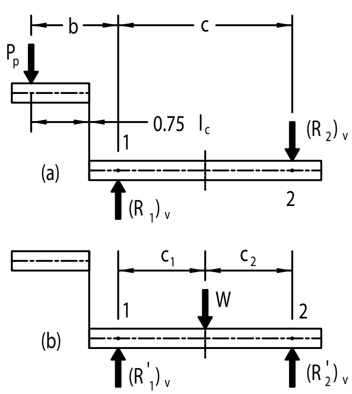


Assumptions: (i) The engine is vertical and the crank is at the top dead centre position.
(ii) The belt drive is horizontal.
(iii) The crankshaft is simply supported on bearings 1 and 2.

A crankshaft is subjected to bending and torsional moments due to following three forces:

(i) Force exerted by connecting rod on crank pin (P_p)
(ii) Weight of flywheel (W) acting downward in vertical direction
(iii) Resultant belt tensions acting in horizontal direction ($P_1 + P_2$)

Bearing reactions



$$P_p = \left(\frac{\pi D^2}{4}\right) p_{\max} \qquad (29.154)$$

P_p = force acting on crank pin (N)

D = diameter of piston (mm)

$p_{\max}$ = maximum gas pressure inside the cylinder (MPa or N/mm^2)

$$(R_1)_v = \frac{P_p \times (b+c)}{c} \qquad (29.155)$$

$$(R_2)_v = \frac{P_p \times b}{c} \qquad (29.156)$$

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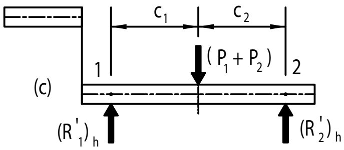	$(R_v)_1$ = vertical reaction at bearing-1 due to (P_p) (N) $(R_v)_2$ = vertical reaction at bearing-2 due to (P_p) (N) b = overhang distance of force P_p from bearings 1 (mm) c = centre distance between bearing 1 and 2 (mm)
$(R_1')_v = \dfrac{W \times c_2}{c}$ (29.157) $(R_2')_v = \dfrac{W \times c_1}{c}$ (29.158) $c_1 = c_2 = \dfrac{c}{2}$ (29.159)	$(R_1')_V$ = vertical reaction at bearing-1 due to (W) (N) $(R_2')_V$ = vertical reaction at bearing-2 due to (W) (N) W = weight of flywheel cum belt pulley (N)
$(R_1')_h = \dfrac{(P_1 + P_2) \times c_2}{c}$ (29.160) $(R_2')_h = \dfrac{(P_1 + P_2) \times c_1}{c}$ (29.161)	$(R_1')_h$ = horizontal reaction at bearing-1 due to $(P_1 + P_2)$ (N) $(R_2')_h$ = horizontal reaction at bearing-2 due to $(P_1 + P_2)$ (N) P_1 = tension in tight side of belt (N) P_2 = tension in slack side of belt (N)
Dimensions of crank pin	
$p_b = \dfrac{P_p}{d_c l_c}$ (29.162) $\left(\dfrac{l_c}{d_c}\right) = 0.60 \text{ to } 1.4$ (29.163)	d_c = diameter of crank pin (mm) l_c = length of crank pin (mm) p_b = allowable bearing pressure at the crank pin bush (10 to 12 MPa or N/mm^2)
Bending stresses in crank pin	
$M_b = \left(\dfrac{3}{4}\right) P_p l_c$ (29.164) $M_b = \sigma_b \left(\dfrac{\pi d_c^3}{32}\right)$ (29.165)	σ_b = allowable bending stress for crank pin (N/mm^2)
Bearings 1 and 2	
$t = 0.45\, d_c$ to $0.75\, d_c$ (29.166) $M_b = P_p[0.75 l_c + t + 0.5 l_1]$ (29.167) $l_1 = 1.5 d_c \text{ to } 2 d_c$ (29.168) $M_b = \sigma_b \left(\dfrac{\pi d_1^3}{32}\right)$ (29.169)	t = thickness of web (mm) d_1 = diameter of journal or shaft at bearing 1 (mm) l_1 = length of bearing 1 (mm) σ_b = allowable bending stress for bearing shaft (N/mm^2)
$R_1 = \sqrt{[(R_1)_v + (R_1')_v]^2 + [(R_1')_h]^2}$ (29.170) $p_b = \dfrac{R_1}{d_1 l_1}$ (29.171)	p_b = allowable bearing pressure at the bearing bush (10 to 12 MPa or N/mm^2) **Note:** Bearings 1 and 2 are made identical.

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Crank web		
$\sigma_c = \dfrac{P_P}{wt}$	(29.172)	w = width of crank web (mm) σ_c = direct compressive stress (MPa or N/mm^2) M_b = bending moment at the central plane (N-mm) $(\sigma_c)_t$ = total compressive stress (MPa or N/mm^2) **Note:** The total compressive stress should be less than the allowable bending stress)
$M_b = P_P\left[0.75l_c + \dfrac{t}{2}\right]$	(29.173)	
$\sigma_b = \dfrac{M_b}{Z}$		
$Z = \left(\dfrac{wt^2}{6}\right)$	(29.174)	
$(\sigma_c)_t = \sigma_c + \sigma_b$	(29.175)	
Shaft under flywheel		
$(M_b)_v = P_p(b + c_1) - [(R_1)_v + (R_1')_v](c_1)$	(29.176)	For the cross section of shaft at midway between two bearings 1 and 2, $(M_b)_v$ = bending moment in vertical plane (N-mm) $(M_b)_h$ = bending moment in horizontal plane (N-mm) d_s = diameter of shaft under flywheel (mm) σ_b = allowable bending stress for shaft (N/mm^2)
$(M_b)_h = (R_1')_h c_1$	(29.177)	
$M_b = \sqrt{(M_b)_v^2 + (M_b)_h^2}$	(29.178)	
$M_b = \left(\dfrac{\pi d_s^3}{32}\right)\sigma_b$	(29.179)	

Table 29.48 *Side crankshaft at angle of maximum torque*

The torque is maximum when the tangential component of force on crank pin is maximum. For this condition, the crank angle from top dead centre position (θ) is usually 25° to 35° for petrol engines and 30° to 40° for diesel engines.		
$P_p = \left(\dfrac{\pi D^2}{4}\right)p'$	(29.180)	P_p = force acting on piston top due to gas pressure (N) D = diameter of piston (mm) p' = gas pressure on the piston top for maximum torque condition (MPa or N/mm^2)
$\sin\varphi = \dfrac{\sin\theta}{(L/r)}$	(29.181)	φ = angle of inclination of connecting rod with the line of dead centres (deg) θ = angle of inclination of crank with line of dead centres (deg) (L/r) = ratio of length of connecting rod to radius of crank
$P_q = \dfrac{P_p}{\cos\varphi}$	(29.182)	P_q = thrust on connecting rod (N) P_t = tangential component of P_q at the crank pin (N) P_r = radial component of P_q at the crank pin (N)
$P_t = P_q\sin(\theta + \varphi)$	(29.183)	
$P_r = P_q\cos(\theta + \varphi)$	(29.184)	

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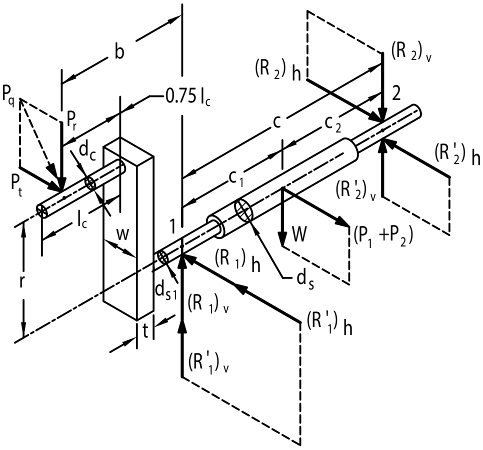
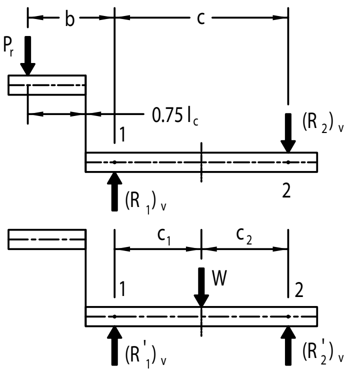
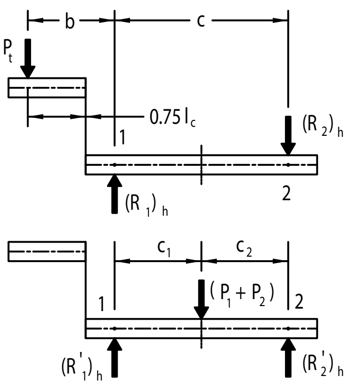
Bearing reactions	
	
 (a) Vertical plane	$$(R_1)_v = \frac{P_r \times (b+c)}{c} \quad (29.185)$$ $$(R_2)_v = \frac{P_r \times b}{c} \quad (29.186)$$ $(R_1)_v$ = vertical reaction at bearing-1 due to (P_r) (N) $(R_2)_v$ = vertical reaction at bearing-2 due to (P_r) (N)
	$$(R_1')_v = \frac{W \times c_2}{c} \quad (29.187)$$ $$(R_2')_v = \frac{W \times c_1}{c} \quad (29.188)$$ $(R_1')_V$ = vertical reaction at bearing-1 due to (W) (N) $(R_2')_V$ = vertical reaction at bearing-2 due to (W) (N)
 (b) Horizontal plane	$$(R_1)_h = \frac{P_t \times (b+c)}{c} \quad (29.189)$$ $$(R_2)_h = \frac{P_t \times b}{c} \quad (29.190)$$ $(R_1)_h$ = horizontal reaction at bearing-1 due to (P_t) (N) $(R_2)_h$ = horizontal reaction at bearing-2 due to (P_t) (N)
	$$(R_1')_h = \frac{(P_1 + P_2) \times c_2}{c} \quad (29.191)$$ $$(R_2')_h = \frac{(P_1 + P_2) \times c_1}{c} \quad (29.192)$$ $(R_1')_h$ = horizontal reaction at bearing-1 due to $(P_1 + P_2)$ (N) $(R_2')_h$ = horizontal reaction at bearing-2 due to $(P_1 + P_2)$ (N)

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Bending stresses in crank pin	
$(M_b)_v = P_r \times (0.75 l_c)$ (29.193) $(M_b)_h = P_t \times (0.75 l_c)$ (29.194) $M_b = \sqrt{[(M_b)_v]^2 + [(M_b)_h]^2}$ (29.195) $M_b = \sigma_b \left(\frac{\pi d_c^3}{32} \right)$ (29.196)	For the cross section of crank pin at the crank web, $(M_b)_v$ = bending moment in vertical plane (N-mm) $(M_b)_h$ = bending moment in horizontal plane (N-mm) M_b = resultant bending moment (N-mm) d_c = diameter of crank pin (mm) σ_b = allowable bending stress for crank pin (N/mm^2) **Note:** In absence of data, the allowable bending stress for crank pin can be assumed as 75 N/mm^2.
Compressive stresses in crank web	
The crank web is subjected to the following stresses: (i) Bending stresses in vertical and horizontal planes due to radial component P_r and tangential component P_t respectively. (ii) Direct compressive stress due to radial component P_r. (iii) Torsional shear stresses due to tangential component P_t.	
$(M_b)_r = P_r\ [0.75 l_c + 0.5t]$ (29.197) $(M_b)_r = (\sigma_b)_r \left[\frac{1}{6} w t^2 \right]$ (29.198)	t = thickness of web (mm) w = width of crank web (mm) $(M_b)_r$ = bending moment due to radial component at the central plane (N-mm) $(\sigma_b)_r$ = bending stress due to radial component (MPa or N/mm^2)
$(M_b)_t = P_t \left[r - \frac{d_{s1}}{2} \right]$ (29.199) $(M_b)_t = (\sigma_b)_t \left[\frac{1}{6} t w^2 \right]$ (29.200)	r = radius of crank (mm) $(M_b)_t$ = bending moment due to tangential component at the juncture of crank web and shaft (N-mm) $(\sigma_b)_t$ = bending stress due to tangential component (MPa or N/mm^2) d_{s1} = diameter of shaft at the juncture of crank web (mm)
$(\sigma_c)_d = \frac{P_r}{wt}$ (29.201)	$(\sigma_c)_d$ = direct compressive stress due to radial component (MPa or N/mm^2)
$\sigma_c = (\sigma_b)_r + (\sigma_b)_t + (\sigma_c)_d$ (29.202)	σ_c = resultant compressive stress (MPa or N/mm^2)
$M_t = P_t [0.75 l_c + 0.5t]$ (29.203) $Z_p = \frac{wt^2}{4.5}$ (29.204) $\tau = \frac{M_t}{Z_p} = \frac{4.5 M_t}{wt^2}$ (29.205)	M_t = torsional moment on the arm (N-mm) τ = torsional shear stress (MPa or N/mm^2)
$(\sigma_c)_{max} = \frac{\sigma_c}{2} + \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2}$ (29.206)	$(\sigma_c)_{max}$ = maximum compressive stress (MPa or N/mm^2)

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Torsional shear stresses in shaft at juncture of crank web	
The cross section of shaft at the juncture of crank web is subjected to the following moments: (i) Bending moment in vertical plane $(M_b)_v$ due to P_r. (ii) Bending moment in horizontal plane $(M_b)_h$ due to P_t. (iii) Torsional moment M_t due to tangential component P_t.	
$(M_b)_v = P_r[0.75 l_c + t]$ (29.207) $(M_b)_h = P_t[0.75 l_c + t]$ (29.208) $M_b = \sqrt{[(M_b)_v]^2 + [(M_b)_h]^2}$ (29.209) $M_t = P_t \times r$ (29.210) $\tau = \dfrac{16}{\pi d_{s1}^3}\sqrt{(M_b)^2 + (M_t)^2}$ (29.211)	d_{s1} = diameter of shaft at the juncture of crank web (mm) τ = allowable shear stress (MPa or N/mm^2)
Torsional shear stresses in shaft under flywheel	
$(M_b)_v = P_r(b + c_1) - [(R_1)_v + (R_1')_v]c_1$ (29.212) $(M_b)_h = P_t(b + c_1) - [(R_1)_h + (R_1')_h]c_1$ (29.213) $M_b = \sqrt{[(M_b)_v]^2 + [(M_b)_h]^2}$ (29.214) $M_t = P_t \times r$ (29.215) $\tau = \dfrac{16}{\pi d_s^3}\sqrt{(M_b)^2 + (M_t)^2}$ (29.216)	For the cross section of shaft at midway between two bearings 1 and 2, $(M_b)_v$ = bending moment in vertical plane (N-mm) $(M_b)_h$ = bending moment in horizontal plane (N-mm) d_s = diameter of shaft under flywheel (mm) τ = allowable shear stress (MPa or N/mm^2)

29.6 VALVE GEAR MECHANISM

Table 29.49 *Valve gear mechanism*

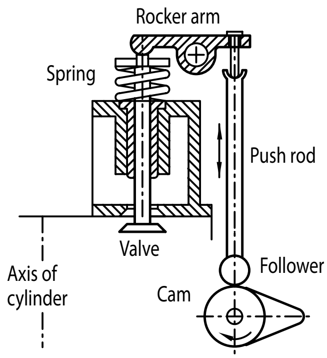
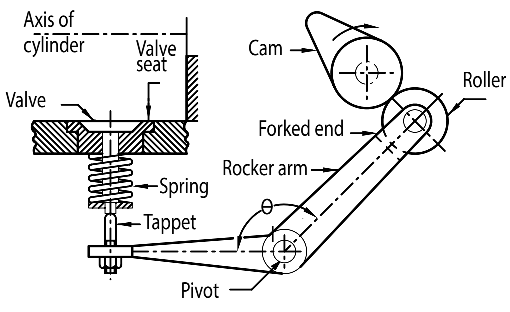
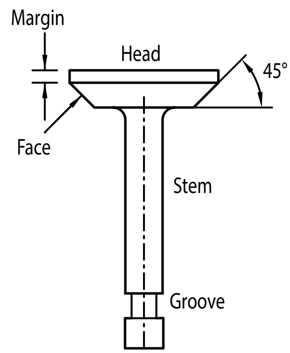
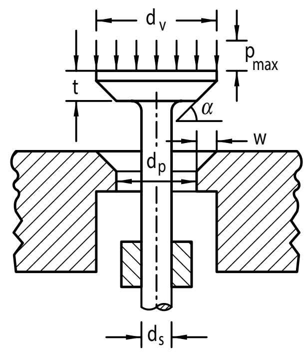
Valve gear mechanism	
Valve gear mechanism for vertical engine	*Valve gear mechanism for horizontal engine*
	
Valve gear mechanism is a subassembly of IC engine and its function is to open and close the inlet and exhaust valves at a proper time with respect to the position of piston and crankshaft. The fuel is admitted into the cylinder when the inlet valve is open. Also, the burnt gases are escaped when the exhaust valve is open.	

Table 29.50 *Poppet or mushroom valves*





Valve materials	
The inlet valve is subjected to comparatively less temperature than exhaust valve. Therefore, inlet valves are made of nickel-chromium steel. The exhaust valves are made of heat resistant silicon-chromium steel. For heavy duty engines, valves are made of chromium-vanadium steel. The valves are heat treated and surface hardness for inlet and exhaust valves is in the range of 250 to 300 HB. In slow-speed engines, valves have composite construction with cast iron head and steel stem. In high-speed engines, one piece construction is used and valves are forged.	
Diameter of valve port	
$a_p = \dfrac{av}{v_p}$ (29.217)	a = area of piston (mm^2) v = mean velocity of piston (m/s) a_p = area of port (mm^2) v_p = mean velocity of gas flowing through the port (m/s)
$a_p = \left(\dfrac{\pi d_p^2}{4}\right)$ (29.218)	d_p = diameter of port (mm)
$v = 2l\left(\dfrac{N}{60}\right)$ (29.219)	v = mean velocity of piston (m/s) l = length of stroke (m) N = engine speed (rpm)

Allowable mean velocities of gas (v_p)

Type of engine	*Mean velocity of gas (m/s)*	
	Inlet valve	*Exhaust valve*
Low-speed engine	33–40	40–50
Medium-speed engine	35–45	50–60
High-speed engine	80–90	90–100

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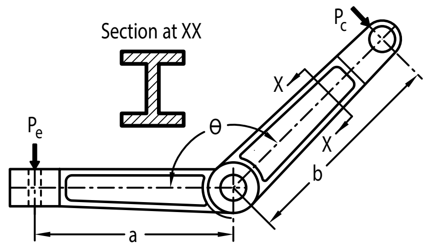
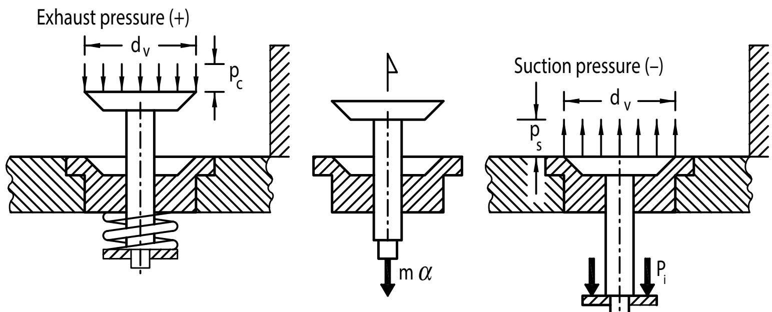
The inlet ports are made 20% to 40% larger than the exhaust ports for better cylinder charging and scavenging.	
Diameter of valve head	
$w = (0.05 \text{ to } 0.07)\, d_p$ or $w = 0.06\, d_p$ (29.220)	w = projected width of valve seat (mm) d_v = diameter of valve head (mm)
For seat angle of 45°, $d_v = (d_p + 2w)$ (29.221)	
Thickness of valve disk	
$t = kd_p\sqrt{\dfrac{p_{max}}{\sigma_b}}$ (29.222)	t = thickness of valve disk (mm) k = constant (k = 0.42 for steel and k = 0.54 for cast iron) d_p = diameter of port (mm) p_{max} = maximum gas pressure (MPa or N/mm^2) σ_b = permissible bending stress (N/mm^2) For carbon steel σ_b = 50 to 60 N/mm^2 For alloy steel σ_b = 100 to 120 N/mm^2
Margin or thickness of valve disk at edges = 0.75 to 0.85*t*	
Diameter of valve stem	
$d_s = \left[\dfrac{d_p}{8} + 6.35\right] \text{ to } \left[\dfrac{d_p}{8} + 11\right]$ (29.223) d_s = diameter of valve stem (mm)	
Tensile stress in valve	
$\sigma_t = \dfrac{1.4P_s}{t^2}\left[1 - \dfrac{2d_s}{3d_p}\right]$ (29.224)	P_s = spring force (N)
Maximum valve lift	
$h_{max} = \dfrac{d_p}{4\cos\alpha}$ (29.225) For flat headed valves, ($\alpha = 0$) ($\cos\alpha = 1$) $h_{max} = \dfrac{d_p}{4}$ (29.226)	h_{max} = maximum lift of valve (mm)

Table 29.51 *Rocker arm*

		(i) Rocker arms are made of grey cast iron, malleable cast iron or cast steel. (ii) In four-stroke cycle engines, the rocker arm of exhaust valve is more heavily loaded. On the other hand, the force required to operate inlet valve is comparatively less. However, it is usual practice to make rocker arms for inlet and exhaust valves identical. This results in ease of manufacturing. (iii) Two arm of rocker arm are often made equal in moderate and low speed engines. In high-speed engines, the (a/b) ratio is taken as (1/1.3). (iv) In order to reduce the weight and inertia force, the cross section of rocker arm is I-section. The arms of rocker arm are made of uniform strength by tapering the width and depth dimensions. However, the thickness of web and flanges is kept uniform.
Forces acting on the rocker arm of exhaust valve		
(a) Load due to gas pressure (valve open) (b) Inertia force (valve moving up) (c) Initial compression of spring (valve closed)		
$P_g = \left(\dfrac{\pi d_v^2}{4}\right) p_c$	(29.227)	P_g = force due to gas pressure (N) d_v = diameter of the valve head (mm) p_c = cylinder pressure or back pressure when the exhaust valve opens (MPa)
$P_a = m_v\,\alpha$	(29.228)	P_a = inertia force on valve (N) m_v = mass of valve (kg) α = acceleration of the valve (m/s^2)
$P_i = \left(\dfrac{\pi d_v^2}{4}\right) p_s$	(29.229)	P_i = initial spring force (N) p_s = maximum suction pressure (MPa)
$P_e = P_g + P_a + P_i$ (for exhaust valve) $P_e = P_a + P_i$ (for inlet valve)	(29.230) (29.231)	P_e = total force exerted on rocker arm (N)

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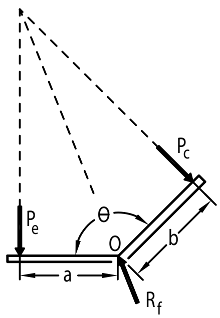
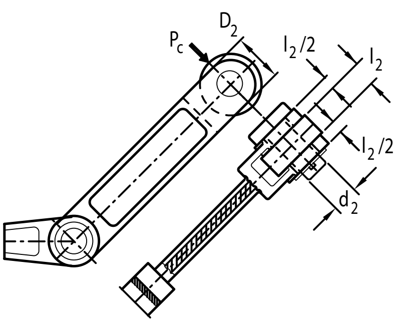
Fulcrum pin	
	$$R_f = \sqrt{(P_e)^2 + (P_c)^2 - 2P_eP_c\cos\theta} \quad (29.232)$$ R_f = reaction at the fulcrum pin (N) When arms are equal, $$a = b \quad \text{and} \quad P_e = P_c \quad (29.233)$$ $$p_b = \frac{R_f}{d_1 l_1} \quad (29.234)$$ $$\left(\frac{l_1}{d_1}\right) = 1.25 \quad (29.235)$$ d_1 = diameter of fulcrum pin (mm) l_1 = length of fulcrum pin (mm) p_b = permissible bearing pressure (5 MPa or N/mm^2) $$\tau = \frac{R_f}{2\left(\frac{\pi d_1^2}{4}\right)} \quad (29.236)$$ τ = permissible shear stress for fulcrum pin (42 N/mm^2)
Forked end	
	$$p_b = \frac{P_c}{d_2 l_2} \quad (29.237)$$ $$\left(\frac{l_2}{d_2}\right) = 1.25 \quad (29.238)$$ $$\tau = \frac{P_c}{2\left(\frac{\pi d_2^2}{4}\right)} \quad (29.239)$$ P_c = force acting on roller pin (N) d_2 = diameter of roller pin (mm) l_2 = length of roller pin (mm) p_b = allowable bearing pressure (5 MPa) τ = permissible shear stress for fulcrum pin (42 N/mm^2) **Dimensions of eye:** Thickness of each eye = $l_2/2$ (29.240) Outer diameter of eye = $D_2 = 2d_2$ (29.241)

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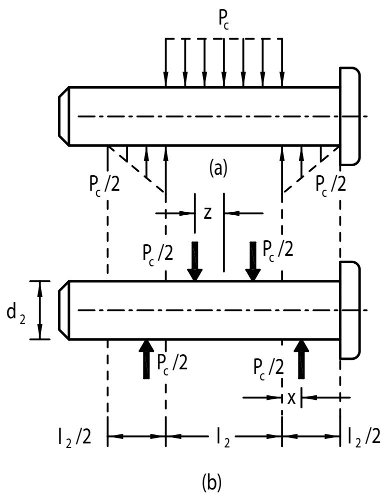
Roller pin



Note: (i) The roller pin is fixed in the eye. The forked end carries a roller that revolves on the pin.
(ii) The diameter of roller should be more than the outer diameter of eye by at least 5 mm.

Bending stresses in roller pin:

$$x = \left(\frac{l_2}{6}\right) \quad (29.242)$$

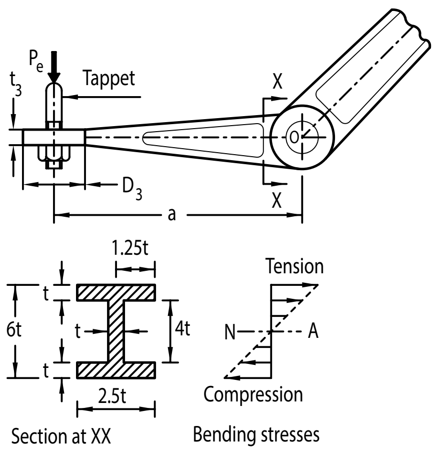
$$z = \left(\frac{l_2}{4}\right) \quad (29.243)$$

$$M_b = \left(\frac{5}{24}\right) P_c l_2 \quad (29.244)$$

$$Z = \frac{\pi d_2^3}{32} \text{ and } \sigma_b = \frac{M_b}{Z} \quad (29.245)$$

M_b = bending moment for cross-section of roller pin in central plane (N-mm)
σ_b = permissible bending stress (MPa or N/mm^2)

Cross section of rocker arm



Consider the cross section of arm near the boss of fulcrum pin. The outside diameter of boss of rocker arm at the fulcrum pin is twice the diameter of fulcrum pin ($2d_1$). Therefore,

$$M_b = P_e[a - d_1] \quad (29.246)$$

$$I = (37t^4) \text{ and } y = (3t) \quad (29.247)$$

$$\sigma_b = \frac{M_b y}{I}$$

The permissible values of bending stresses are as follows:

For cast steel, σ_b = 50 to 60 N/mm^2
For forged steel, σ_b = 70 to 80 N/mm^2

Tappet

$$\sigma_c = \frac{P_e}{\left(\frac{\pi d_c^2}{4}\right)} \quad (29.248)$$

$$d = \left(\frac{d_c}{0.8}\right) \quad (29.249)$$

d_c = core diameter of stud (mm)
d = nominal diameter of stud (mm)
σ_c = permissible compressive stress for stud (50 N/mm^2)

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$D_3 = 2\,d$ (29.250) $t_3 = 2\,d$ (29.251)	D_3 = diameter of circular end of rocker arm (mm) t_3 = depth of circular end of rocker arm (mm)

Table 29.52 *Valve spring*

Assumptions: (i) The spring is made of oil-hardened and tempered valve spring wire of Grade-VW. (ii) The stiffness of spring is 10 N/mm. (iii) The allowable torsional shear stress for spring material is 250 to 350 N/mm^2. (iv) The spring index (D/d) is 8. (v) The spring has square and ground ends.	
Force acting on spring	
The total force acting on the spring of exhaust valve consists of two factors, viz. the initial spring force and force required to lift the valve.	
$P_{max} = P_i + k\,\delta$ (29.252) $P_i = \left(\dfrac{\pi d_v^2}{4}\right) p_s$ (29.253)	P_i = initial spring force (N) p_s = maximum suction pressure (MPa) d_v = diameter of the valve head (mm) k = stiffness of spring (N/mm) δ = maximum lift of valve (mm)
Wire diameter	
$\tau = K\left(\dfrac{8P_{max}C}{\pi d^2}\right)$ (29.254) $K = \dfrac{4C-1}{4C-4} + \dfrac{0.615}{C}$ (29.255)	τ = allowable torsional shear stress (250 to 350 N/mm^2) C = spring index (8) d = wire diameter (mm) K = Wahl factor
Mean coil diameter	
$D = C\,d$ (29.256)	D = mean coil diameter (mm)
Number of active turns	
$N = \dfrac{Gd^4}{8D^3k}$ (29.257)	N = number of active turns G = modulus of rigidity (84×10^3 N/mm^2) k = stiffness of spring (10 N/mm)
Natural frequency of spring	
$\omega = \dfrac{1}{2}\sqrt{\dfrac{k}{m}}$ (29.258)	ω = natural frequency of spring (cycles/s) k = stiffness of spring (N/m) m = mass of spring (kg)

Table 29.53 *Push rod*

Push rods are used in overhead valve and side valve engines. It is a long column introduced between the cam and rocker arm so that the camshaft can be located at a lower level. The push rods are made of bright drawn steel tubes with 4% carbon or duralumin tubes. When the push rod is guided, the ends are flat plugs. When the push rod is not guided, ball and socket joints are used at the ends. The flat plugs or sockets are force fitted at the ends of push rod.

(*Contd.*)

(*Contd.*)

$\left(\frac{d_i}{d_o}\right) = 0.6 \text{ to } 0.8$ (29.259)	d_o = outer diameter of push rod (mm) d_i = inner diameter of push rod (mm)
$I = \frac{\pi(d_o^4 - d_i^4)}{64}$ (29.260)	$A = \frac{\pi(d_o^2 - d_i^2)}{4}$ (29.261)
$I = A\,k^2$ (29.262)	$k^2 = \frac{(d_o^2 + d_i^2)}{16}$ (29.263)
Rankine's formula	
$P = \frac{\sigma_c A}{1 + a\left(\frac{l}{k}\right)^2}$ (29.264) $a = \frac{1}{7500}$ (29.265)	P = force acting on the push rod (N) σ_c = permissible compressive stress (N/mm^2) A = cross-sectional area of push rod (mm^2) a = constant depending upon material and end fixity coefficient l = actual length of push rod (mm) k = radius of gyration (mm)
For push rod made of mild steel and plain carbon steel, σ_c = 70 N/mm^2 (Initial value)	

Permissible compressive stress (σ_c)(N/mm^2)

(l/d_o)	10	20	30	40	50	60	70	80	90	100
σ_c	100	70	50	35	28	20	15	12	9	7.5

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31. Justice S Malik, *Indian Boiler Regulations 1950*, Law Publishers (India).
32. Chuse, R, *Unfired Pressure Vessel*, McGraw Hill**.**

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1. Seireg, A S and S Dandage, *Empirical Design Procedure for the Thermodynamic Behavior of Journal Bearings*, Journal of Lubrication Technology, April 1982.
2. Raimondi, A A and John Boyd, *A Solution for the Finite Journal Bearing and its Application to Analysis and Design*, A.S.L.E. Transactions, Vol.1, No. 1, April 1958 (in three parts).
3. Rippel, Harry C, *Cast Bronze Hydrostatic Bearing Design Manual*, Cast Bronze Bearing Institute, USA.

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1. Dunlop, *Dunlop Transmission Belting - Technical Data*
2. Dunlop, *Polyester Cord V-Belt Catalogue*

3. Good Year, *V-Belt Catalogue*
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5. Diamond, *Industrial Roller Chain Catalogue*
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10 David Brown Radicon, *Worm Gear Catalogue*
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2. ISO 286-1: ISO System of Limits and Fits – Part1: Bases of Tolerances, Deviations and Fits
3. ISO 286-2: ISO System of Limits and Fits – Part2: Tables for Standard Tolerance Grades and Limit Deviations for Holes and Shafts
4. ISO 1101: Geometrical Product Specifications – Geometrical Tolerancing – Tolerances of form, orientation, location and run-out
5. ISO 1302: Geometrical Product Specifications – Indication of Surface Texture in technical product documentation
6. ISO 2901: ISO metric trapezoidal screw threads – Basic profile and maximum material profile
7. ISO 2902: ISO metric trapezoidal screw threads – General plan
8. ISO 2903: ISO metric trapezoidal screw threads – Tolerances
9. ISO 2904: ISO metric trapezoidal screw threads – Basic dimensions
10. ISO 724: ISO general purpose metric screw threads- Basic dimensions
11. ISO 965-1: ISO general purpose metric screw threads- Tolerances – Part 1: Principles and Basic data
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13. ISO 4017: Hexagon head screws– Product grades A and B
14. ISO 4032: Hexagon nut, style 1– Product grades A and B
15. ISO 4035: Hexagon thin nut (chamfered) - Product grades A and B
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28. ISO 898-1: Mechanical properties of Fasteners made of steel and alloy steel- Part1: Bolts, screws and studs with specified property class – Coarse thread and Fine pitch thread
29. ISO 228: Pipe threads where pressure-tight joints are not made on threads
30. ISO 2553: Welded, brazed and soldered joints – Symbolic representation on drawings
31. ISO 3117: Tangential keys and keyways
32. ISO 3912: Woodruff keys and keyways
33. ISO 14: Straight-sided splines for cylindrical shafts with internal centering
34. ISO 8458 -2: Steel wire for mechanical springs – Part 2: Patented cold-drawn non-alloy steel wire
35. ISO 8458 -3: Steel wire for mechanical springs – Part 3: Oil-hardened and tempered wire
36. ISO 6931-1: Stainless steels for springs – Part 1: Wire
37. ISO 606: Short pitch transmission precision Roller chains and Chain wheels
38. ISO 6336: Calculation of the load capacity of Spur and Helical gears
    - Part 1: Basic principles, introduction and general influence factors
    - Part 2: Calculation of surface durability
    - Part 3: Calculation of tooth bending strength
    - Part 5: Strength and quality of materials
39. ISO – 10300: Calculation of Load Capacity of Bevel Gears
    - Part 1: Introduction and general influence factors
    - Part 2: Calculation of surface durability
    - Part 3: Calculation of tooth root strength
40. ISO 2408: Steel wire ropes for general purposes - Characteristic

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1. DIN 513-1: Metric Buttress threads – Thread profile
2. DIN 513-2: Metric Buttress threads – General plan
3. DIN 513-3: Metric Buttress threads – Deviations and tolerances
4. DIN 917: Hexagon Cap nuts
5. DIN 1587: Hexagon Domed Cap nut
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7. DIN 508: Nuts for T-slots
8. DIN 127: Spring lock washers
9. DIN 431: Pipe nuts Whitworth pipe threads R1/8 to R6
10. DIN 5481: Internal and external serrations
11. DIN 471: Circlips for shafts – normal type and heavy type
12. DIN 472: Circlips for bores - normal type and heavy type

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