

Australian Standard™

Design of rotating steel shafts



This Australian Standard was prepared by Committee ME-005, Cranes. It was approved on behalf of the Council of Standards Australia on 9 March 2004 and published on 21 May 2004.

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- Australian Elevator Association
- Australian Industry Group
- Australian Institute for Non-Destructive Testing
- Bureau of Steel Manufacturers of Australia
- Construction and Mining Equipment Association of Australia
- Crane Industry Council of Australia
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STANDARDS AUSTRALIA

RECONFIRMATION
OF
AS 1403—2004
Design of rotating steel shafts

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NOTES

Australian Standard™

Design of rotating steel shafts

Originated as AS B249—1969.
Previous edition AS 1403—1985.
Fourth edition 2004.

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Published by Standards Australia International Ltd
GPO Box 5420, Sydney, NSW 2001, Australia

ISBN 0 7337 5893 2

PREFACE

This Standard was prepared by the Joint Standards Australia/Standards New Zealand Committee ME-005, Cranes, to supersede AS 1403—1985.

The 1985 edition was reviewed by the Committee, and it was found to be technically valid. Hence, this edition does not include technical alterations to the 1985 edition; however, the referenced documents have been updated, and editorial changes have been made to align with current Standards Australia policy.

The term ‘informative’ has been used in this Standard to define the application of the appendix to which it applies. An informative appendix is only for information and guidance.

CONTENTS

	<i>Page</i>
FOREWORD.....	4
1 SCOPE.....	5
2 REFERENCED DOCUMENTS.....	5
3 DEFINITIONS.....	6
4 NOTATION.....	6
5 APPLIED LOADING	8
6 MINIMUM CALCULATED DIAMETER OF SHAFT	8
7 SHAFT DESIGN AND MANUFACTURE.....	9
8 SHAFT DESIGN FACTORS	10
APPENDICES	
A 'TRIAL' SHAFT DIAMETER	21
B ITERATIVE METHOD FOR CALCULATING MINIMUM DIAMETER OF SHAFT	22
C CHARACTERISTICS OF MOTOR CONTROLLERS AND TORQUE-LIMITING DEVICES	23
D TYPICAL WORKED EXAMPLE—CRANE DRIVING MECHANISM.....	26
E TYPICAL WORKED EXAMPLE—CRANE HOIST DRIVE.....	36
F TYPICAL WORKED EXAMPLE—CONVEYOR DRIVE.....	45

FOREWORD

The method of design in this Standard is the same as that contained in the 1984 issue and is based on calculations for infinite life of the shaft.

It is noted that, as the result of extensive laboratory and/or field tests, shafts of diameters smaller than those resulting from calculations in this Standard may be used.

Although the method is not theoretically precise, it gives results that are of sufficient accuracy for practical purposes. The method makes no allowance for corrosive or other abnormal conditions, such as the presence of buckling, ‘whipping’, cyclic vibrations and similar effects (see Clause 7.2).

In some cases, deflection may be the factor that determines the minimum value of the shaft diameter (see Clause 7.6).

Typical worked examples employing this method of shaft design are given in Appendices D, E and F.

A ‘trial’ diameter may be necessary in the application of the design method. The value may be assumed on the basis of the designer’s previous experience, or a simplified version of the method may be used to determine the order of magnitude of the shaft diameter, and the full method then applied to check the assumed trial diameter. A convenient quick method of estimating the ‘trial’ diameter is given in Appendix A, and an iterative method using programmable calculators is provided in Appendix B.

The process of carrying out an accurate and complete analysis of loadings applied to a shaft (see Clause 5) may involve much time and effort, and is warranted where it is desired to keep shaft diameters as small as possible. Where larger diameter shafts can be tolerated, it may be more appropriate to use approximations or to ignore inertia or other effects, but it is important to ensure that, where such inaccuracies are introduced, they result in increased rather than decreased margin of safety and stress-raising characteristics are avoided.

The effect of electric motors with high breakdown torque (also known as pull-out torque) or high locked-rotor torque (also known as starting torque) and the characteristics of motor controllers and torque-limiting devices are outlined in Appendix C.

STANDARDS AUSTRALIA

Australian Standard Design of rotating steel shafts

1 SCOPE

This Standard provides formulae for the design of rotating steel shafts that are subjected to torsional, bending and axial-tensile loads either singly or in combination, on the basis of infinite life.

The Standard does not cover specially developed shafts, for example, those involving extensive laboratory and field testing, heat treatment and like developments, or to shafts for specific applications such as automotive and construction equipment transmissions.

NOTES:

- 1 Particular consideration has to be given to deflections of long shafts where the effects of out-of-balance forces tend to induce 'whipping', and to long shafts that are subject to buckling due to compressive loading (see Foreword).
- 2 The use of this method of shaft design enables justification of the use of shafts of minimal diameter. Where the designer conservatively sizes a shaft, then only those calculations necessary to ensure that the shaft complies with this Standard (see Clause 5) are required.
- 3 A 'trial' shaft diameter is given in Appendix A.
- 4 An interactive method for calculating minimum diameter of shaft is given in Appendix B.
- 5 Characteristics of motor controllers and torque-limiting devices are provided in Appendix C.
- 6 A typical worked example for crane driving mechanism is provided in Appendix D.
- 7 A typical worked example for crane hoist drive is provided in Appendix E.
- 8 A typical worked example for conveyor drive is provided in Appendix F.

2 REFERENCED DOCUMENTS

The following documents are referred to in this standard:

AS

- | | |
|--------|---|
| 1391 | Methods for tensile testing of metals |
| 1654 | ISO system for limits and fits |
| 1654.1 | Part 1: Bases of tolerances, deviations and fits |
| 1654.2 | Part 2: Tables of standard tolerance grades and limit deviations for holes and shafts |

ISO

- | | |
|------|--|
| 14 | Straight-sided splines for cylindrical shafts with internal centering; Dimensions, tolerances and verification |
| 4156 | Straight cylindrical involute splines; Metric module, side fit; Generalities, dimensions and inspection |

BS

- | | |
|------|---|
| 2059 | Specification for straight-sided splines and serrations |
|------|---|

3 DEFINITIONS

For the purpose of this Standard, the definitions below apply.

3.1 Applied loading

3.1.1 Axial loading

Force applied to a shaft in the direction of the shaft axis.

3.1.2 Bending loading

Bending moment applied to a shaft.

3.1.3 Torsional loading

Torque applied to a shaft.

3.2 Equivalent rotational mass moment of inertia of a shaft and its associated components

The linear mass moment of inertia and the rotational mass moment of inertia of the component, related to the rotational full-load speed of the braking or driving medium of the mechanism.

3.3 Stress-raising characteristic

A detail, of a shaft, that induces increased stress in the shaft, e.g. step in diameter, keyway, spline, interference fit with a component on the shaft, transverse hole, annular groove.

NOTE: Stress-raising effects are intensified by machining marks, notching and the like.

4 NOTATION

For the purpose of this Standard, the following notation applies:

D	=	minimum calculated diameter of shaft at cross-section under consideration, in millimetres (see also Clause 6.2)
D_a	=	estimated diameter in iterative method (see Appendix B)
D_{a-1}	=	previous estimated diameter in iterative method (see Appendix B)
D_I	=	internal diameter of hollow shaft, in millimetres
D_o	=	minimum calculated outside diameter of hollow shaft at cross-section under consideration, in millimetres
e	=	allowable deviation between successive iterations, expressed as a percentage (see Appendix B, Figure B1)
F_R	=	endurance limit of shaft material in reversed bending during rotation, based on tests of polished steel specimens of diameter between 8 mm and 10 mm, in megapascals
	=	$0.45 F_U$, where actual value is not known
F_S	=	safety factor
F_U	=	tensile strength of shaft material, in megapascals
F_Y	=	yield strength of shaft material, in megapascals
I_{Ei}	=	equivalent rotational mass moment of inertia of a shaft and its associated rotating and linear components, in kilogram metres squared
	=	$I_{Ri} + I_{Lj}$
I_{Ri}	=	equivalent rotational mass moment of inertia of a shaft and its associated rotating components, in kilogram metres squared

$$= I_i \left(\frac{N_i}{N} \right)^2$$

I_{Lj} = equivalent rotational mass moment of inertia of components with linear motion, associated with the shaft under consideration, in kilogram metres squared

$$= m_j \left(\frac{V_j}{2\pi N} \right)^2$$

$I_1, I_2, I_3 \dots I_n$ = rotational mass moment of inertia at first, second, third ... n th shafts and its components rotating at $N_1, N_2, N_3 \dots N_n$ revolutions per second, in kilogram metres squared

i = number defining the rotational motion component under consideration (1, 2, 3, ... n)

j = number defining the linear motion component under consideration (1, 2, 3 ... k)

k = total number of components with linear motion

K = stress-raising factor (see Clause 8.2):
for a stepped shaft, see Figure 4
for a shaft fitted with rolling-element bearing, see Figure 5
for a shaft with fitted component without key or spline, see Figure 6
for a shaft with keyed component, see Figure 7
for a splined shaft, see Figure 8
for a shaft with annular groove, see Figure 9
for a shaft with transverse hole, see Figure 10

K_S = size factor (see Clause 8.1 and Figure 1)

m_j = mass of linear-motion component, in kilograms

M_q = bending moment at shaft cross-section under consideration, in newton metres

n = total number of components with rotational motion

N = rotational full-load speed of driving or braking medium, in revolutions per second

N_i = rotational full-load speed of a shaft and its components, in revolutions per second

$N_1, N_2, N_3 \dots N_n$ = rotational full-load speeds of 1st, 2nd, 3rd .. n th rotating elements in gear train or other speed-changing device, in revolutions per second

P_q = maximum axial tensile force at shaft cross-section under consideration, in newtons

NOTE: For axial compressive force, see Clause 7.2.

S_O = original cross-sectional area within the gauge length (see Clause 7.1)

T_E = equivalent torque when calculating 'trial' diameter, in newton metres (see Appendix A)

T_q = maximum torque at shaft cross-section under consideration, in newton metres

T_M = torque applied to the mechanism by the braking or driving means or by the external load, in newton metres

V_j = velocity of linear-motion component, in metres per second

α	=	angular acceleration of driving medium or braking medium, in radians per second squared
$\alpha_1, \alpha_2, \alpha_3 \dots \alpha_n$	=	angular acceleration of the first, second, third ... <i>n</i> th shafts of a gear train or other speed changing device, in radians per second squared
Δ	=	correction factor (see Figure 3)
η	=	mechanical efficiency of system
$\eta_{i/i+1}$	=	mechanical efficiency of mating components
$\eta_{1/2}, \eta_{2/3}, \eta_{3/4}, \dots \eta_{n-1/n}$	=	mechanical efficiency of gear pair or other speed changing device between first/second, second/third, third/fourth ... (<i>n</i> – 1)th/ <i>n</i> th shafts

5 APPLIED LOADING

Determination of the axial, bending and torsional loading (see Clause 3.1) applied to a component shaft of a mechanism shall take into account the following effects:

- Torque applied by the driving or braking medium.
- Reactive torque of the element driven by the mechanism including, where applicable, dynamic effects.
- Energy gain and loss during acceleration and deceleration of the system.
- Mechanical efficiency.

NOTES:

- The process of completely and accurately analysing the loading applied to a component shaft of a mechanism may involve considerable time and effort (except where performed by computer with existing programs) and is necessary where it is desired to keep shaft diameter as small as possible.

Where the size of a shaft diameter is not critical or needs to be relatively large to provide adequate stiffness (see Clause 7.6) or to accommodate a component, experience may indicate that a shaft size selected need not be checked by calculation or may be checked by use of approximations or by ignoring mechanical efficiency, inertia and like effects. It is essential, however, that under such circumstances the margin of safety applicable to the shaft is increased rather than decreased.

- It may be more accurate and easier to calculate the loading of a particular component shaft of a mechanism from either the driving end or the driven end of the system depending on the characteristics of the system and the driven element.
- The presence and location of a flywheel in the system should be taken into account in the selection of the appropriate formula from Table 1, e.g. flywheel driving effect during braking.

The theoretical value of the torque applied to a component shaft of a gear train or other speed-changing device shall be not less than the value calculated by means of the appropriate formulas given in Table 1.

6 MINIMUM CALCULATED DIAMETER OF SHAFT

6.1 Minimum calculated diameter

The minimum calculated diameter of each stressed section of shaft shall be calculated by means of the appropriate formula given in Table 2.

6.2 Solid shafts

The minimum calculated diameter for solid shafts shall comply with the following requirements:

- (a) Solid shafts without keyways or with maximum of two keyways. The outside diameter shall be not less than D .

NOTE: The influence of the keyway(s) is taken into account by Figure 7.

- (b) Solid shafts with splines. The root diameter of the splines shall be not less than D .

NOTE: The influence of the splines is taken into account by Figure 8.

- (c) Solid shafts with annular grooves. The diameter at the base of the groove shall be not less than D .

NOTE: The influence of the groove is taken into account by Figure 9.

6.3 Hollow shafts

Hollow shafts shall comply with the following requirements:

- (a) The shaft shall have a constant diameter longitudinal hole, coaxial with the outside diameter.
- (b) The wall thickness shall be greater than $0.15 D$.
- (c) The outside diameter shall be greater than the value of D_o calculated by the following formula:

$$D_o^3 = D^3 + 1.7 D_1^3 \quad \dots 5$$

7 SHAFT DESIGN AND MANUFACTURE

7.1 Shaft material

The steel from which the shaft is manufactured shall have an elongation of not less than 5% on a gauge length of $5.65\sqrt{S_o}$ when tested in accordance with AS 1391.

NOTE: Where the shaft is manufactured from a high strength steel, fatigue strength is generally affected more critically by stress-raising characteristics of the shaft than where the shaft is manufactured from a low strength steel.

7.2 Shaft diameters

The required diameter for each stressed section of the shaft shall be specified in accordance with Clause 6.1.

Where shafts are to operate under indeterminate or particularly arduous and abnormal conditions, and such aspects as corrosion fatigue, fretting fatigue or high temperature influences and the like may be involved, additional allowances shall be made for the deleterious effects of such conditions by making the actual diameter greater than the theoretical size.

Where a shaft is subjected to an axial compression force, the value of P_q is taken as 0 (zero), but if buckling, 'whipping', cyclic vibrations and like effects exist, it is essential that provision be made to ensure that—

- (a) the shaft diameter is adequate to withstand any additional stresses due to such effects; and
- (b) such effects are limited to the extent that the operation of the mechanism in which the shaft is incorporated is not adversely affected.

7.3 Stress-raising characteristics (see Clause 8.2)

The shaft shall be designed to minimize the effects of stress-raising characteristics. Special care shall be taken in the manufacture to achieve a smooth finish of highly stressed portions; this shall apply particularly where stress concentrations occur, e.g., at changes of shaft diameters. Radii at steps in diameter and at the bottoms of keyways shall be as large as practicable and large changes in diameter should be avoided.

7.4 Welding on shafts

Shafts for transmission of power shall not have welds in any position where the tensile stress exceeds 15 MPa and the number of stress cycles exceeds 1000 in the design life of the shaft, unless the designer makes adequate provision for—

- (a) the weld shape and quality in respect of the stress-raising effect of the weld;
- (b) the reduced fatigue performance of welded higher strength steels; and
- (c) the possible reduced mechanical properties of the shaft in the weld.

NOTE: The performance of shafts can be seriously impaired by welding. The design of shafts containing welds is not covered by this Standard.

7.5 Splines and keyways

Splines and keyways shall conform to one of the relevant standards, i.e., ISO 14, ISO 4156 and BS 2059.

7.6 Deflection

The deflection of the shaft under operating conditions shall not be greater than the values dictated by gear-separation tolerances, bearing manufacturers' recommendations, locking elements and like effects (see Foreword).

8 SHAFT DESIGN FACTORS

8.1 Size factor (K_s)

The value of the size factor (K_s) shall be as follows:

- (a) For shaft diameters up to 250 mm the value in Figure 1 that corresponds to the actual diameter of the stressed section of the shaft.
- (b) For shaft diameters greater than 250 mm 1.8.

NOTE: When the shaft diameter is not known, a 'trial' diameter (see Appendices A and B) should be calculated and the factor K_s selected accordingly.

8.2 Stress-raising factor (K)

The value of the stress-raising factor (K) shall be as follows:

- (a) Where there is only one stress-raising characteristic the value read from Figures 4 to 10, as appropriate.
- (b) Where two stress-raising characteristics are separated by an axial distance greater than $0.25D$ the greater of the two values read from Figures 4 to 10, as appropriate (see Figure 2).
- (c) Where two stress-raising characteristics are separated by an axial distance between $0.16D$ and $0.25D$ the sum of the greater value and 0.1 times the lesser value (see Figure 2), both values being those read from Figures 4 to 10, as appropriate.
- (d) Where two stress-raising characteristics are coincident or separated by an axial distance not greater than $0.16D$ the sum of the greater value and 0.2 times the lesser value (see Figure 2) both values being those read from Figures 4 to 10, as appropriate.

NOTE: When the shaft diameter is not known, a 'trial' diameter (see Appendices A and B) should be calculated and the factor K selected accordingly.

TABLE 1
FORMULAS FOR SHAFT TORQUE T_q

System	Shaft	Torque	Inertia significant	Inertia not significant
Driving/Accelerating	1	Input	T_M	T_M
		Output	$T_M - I_{E1} \alpha$	
	2	Input	$(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \eta_{1/2}$	$T_M \frac{N_1}{N_2} \eta_{1/2}$
		Output	$(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \eta_{1/2} - I_{E2} \frac{N_1}{N_2} \alpha$	
	3	Input	$\left[(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \eta_{1/2} - I_{E2} \frac{N_1}{N_2} \alpha \right] \frac{N_2}{N_3} \eta_{2/3}$	$T_M \frac{N_1}{N_2} \eta_{1/2} \eta_{2/3}$
		Output	$\left[(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \eta_{1/2} - I_{E2} \frac{N_1}{N_2} \alpha \right] \frac{N_2}{N_3} \eta_{2/3} - I_{E3} \frac{N_1}{N_3} \alpha$	
	4	Input	$\left\{ \left[(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \eta_{1/2} - I_{E2} \frac{N_1}{N_2} \alpha \right] \frac{N_2}{N_3} \eta_{2/3} - I_{E3} \frac{N_1}{N_3} \alpha \right\} \frac{N_3}{N_4} \eta_{3/4}$	$T_M \frac{N_1}{N_2} \eta_{1/2} \eta_{2/3} \eta_{3/4}$
		Output	$\left\{ \left[(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \eta_{1/2} - I_{E2} \frac{N_1}{N_2} \alpha \right] \frac{N_2}{N_3} \eta_{2/3} - I_{E3} \frac{N_1}{N_3} \alpha \right\} \frac{N_3}{N_4} \eta_{3/4} - I_{E4} \frac{N_1}{N_4} \alpha$	
Braking/Decelerating	1	Input	T_M	T_M
		Output	$T_M - I_{E1} \alpha$	
	2	Input	$(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \frac{1}{\eta_{1/2}}$	$T_M \frac{N_1}{N_2} \frac{1}{\eta_{1/2}}$
		Output	$(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \frac{1}{\eta_{1/2}} - I_{E2} \frac{N_1}{N_2} \alpha$	
	3	Input	$\left[(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \frac{1}{\eta_{1/2}} - I_{E2} \frac{N_1}{N_2} \alpha \right] \frac{N_2}{N_3} \frac{1}{\eta_{2/3}}$	$T_M \frac{N_1}{N_3} \frac{1}{\eta_{1/2}} \frac{1}{\eta_{2/3}}$
		Output	$\left[(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \frac{1}{\eta_{1/2}} - I_{E2} \frac{N_1}{N_2} \alpha \right] \frac{N_2}{N_3} \frac{1}{\eta_{2/3}} - I_{E3} \frac{N_1}{N_3} \alpha$	
	4	Input	$\left\{ \left[(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \frac{1}{\eta_{1/2}} - I_{E2} \frac{N_1}{N_2} \alpha \right] \frac{N_2}{N_3} \frac{1}{\eta_{2/3}} - I_{E3} \frac{N_1}{N_3} \alpha \right\} \frac{N_3}{N_4} \frac{1}{\eta_{3/4}}$	$T_M \frac{N_1}{N_4} \frac{1}{\eta_{1/2}} \frac{1}{\eta_{2/3}} \frac{1}{\eta_{3/4}}$
		Output	$\left\{ \left[(T_M - I_{E1} \alpha) \frac{N_1}{N_2} \frac{1}{\eta_{1/2}} - I_{E2} \frac{N_1}{N_2} \alpha \right] \frac{N_2}{N_3} \frac{1}{\eta_{2/3}} - I_{E3} \frac{N_1}{N_3} \alpha \right\} \frac{N_3}{N_4} \frac{1}{\eta_{3/4}} - I_{E4} \frac{N_1}{N_4} \alpha$	

TABLE 2
FORMULAS FOR CALCULATING MINIMUM DIAMETER OF SHAFT D

Number of mechanism starts per year	Number of revolutions of shaft per year	Torque application conditions	Formula	Formulas	
≤600	≤900	Manually or power applied	$D^3 = \frac{10^4 F_S}{F_Y} \sqrt{\left(M_q + \frac{P_q D}{8000}\right)^2 + \frac{3}{4} T_q^2}$	1	
	>900	Power applied	$D^3 = \frac{10^4 F_S}{F_R} \sqrt{\left[K_S K \left(M_q + \frac{P_q D}{8000}\right)\right]^2 + \frac{3}{4} T_q^2}$	2	See Notes 2 and 3
>600	>900	Power applied torque reversals	$D^3 = \frac{10^4 F_S}{F_R} K_S K \sqrt{\left(M_q + \frac{P_q D}{8000}\right)^2 + \frac{3}{4} T_q^2}$	3	
		Power applied, no torque reversals (see Note 1)	$D^3 = \frac{10^4 F_S}{F_R} \sqrt{\left[K_S K \left(M_q + \frac{P_q D}{8000}\right)\right]^2 + \frac{3}{16} [(1 + K_S K) T_q]^2}$	4	

NOTES:

- Where the magnitude of the torque in one direction is not greater than 0.1 times the torque in the other direction, the torque application conditions may be considered as being non-reversing.
- These formulas are extracted from a paper titled ‘Shortcuts for Designing Shafts’ by H.A. Borchardt, published in *Machine Design*, Vol. 45, No.3, 8 February 1973, pp 139-141.
- The value of F_R is based on 10^6 stress cycles and is applicable for numbers of revolutions of shafts per year greater than 50 000.
- For numbers of revolutions of shafts per year from 50 000 down to 900, formulas 2, 3 and 4 result in progressively more conservative values for the theoretical diameter of shaft.
- The value of F_S is 2.0 for Formula 1 and 1.2 for Formulas 2, 3 and 4. Where severe injury, death or extensive equipment damage is likely to occur because of the failure of the shaft, higher factors of safety may be used.
- The values of K_S , K and the term $P_q D/8000$ may require the calculation of a ‘trial’ diameter (see Appendix A).

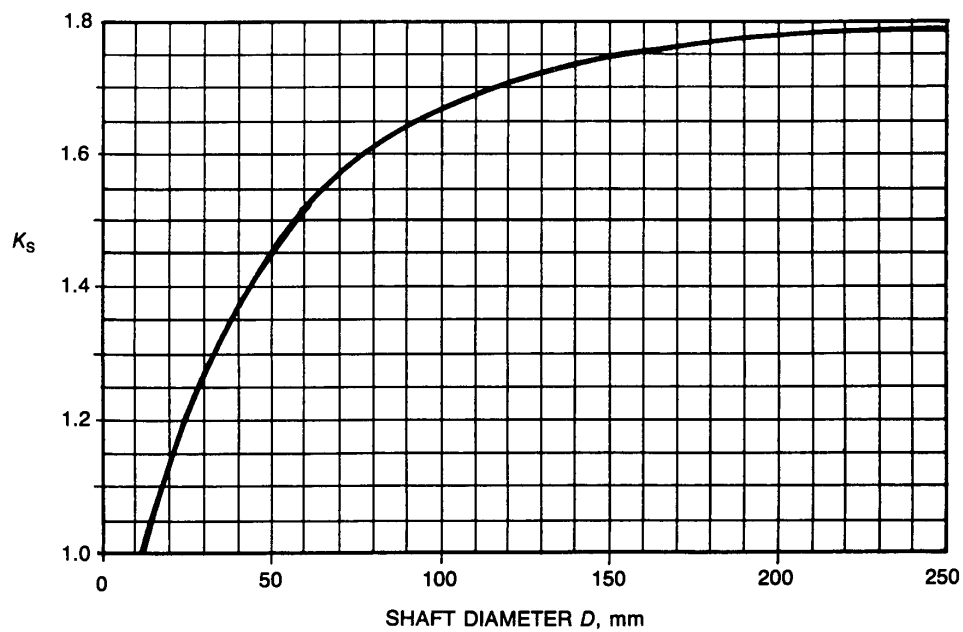
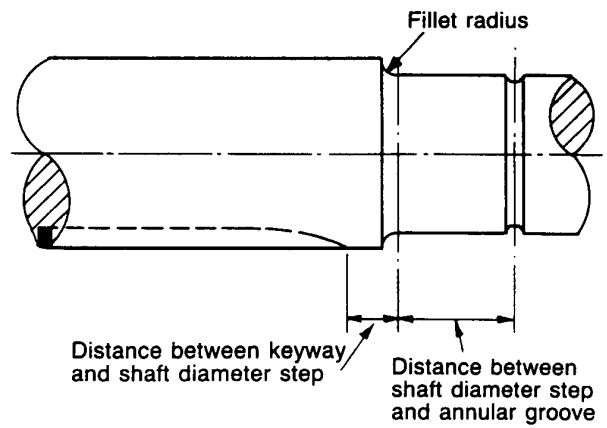
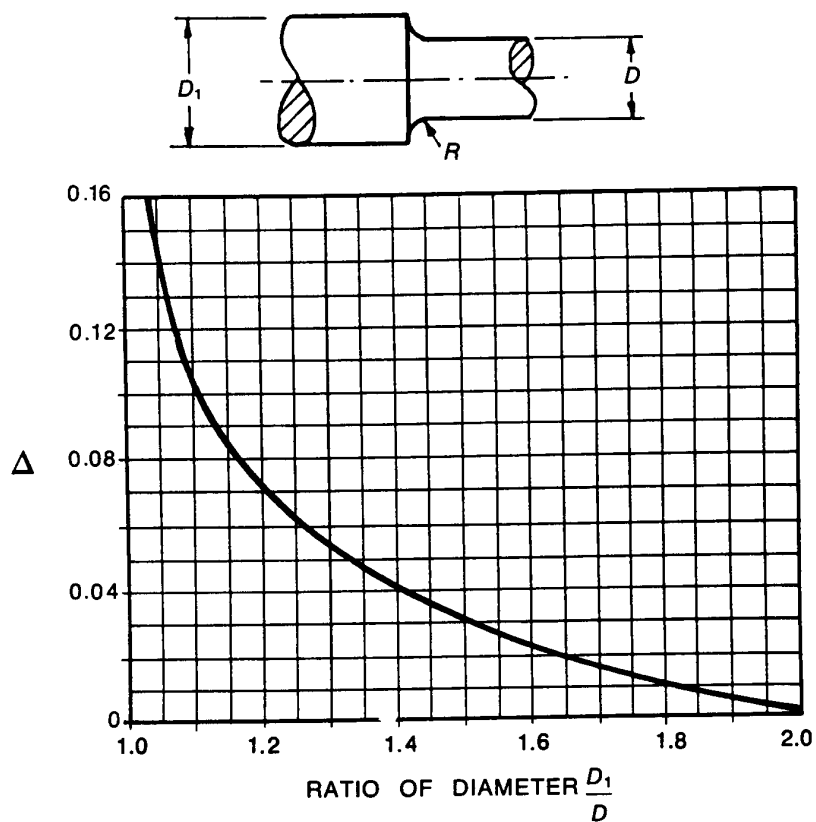
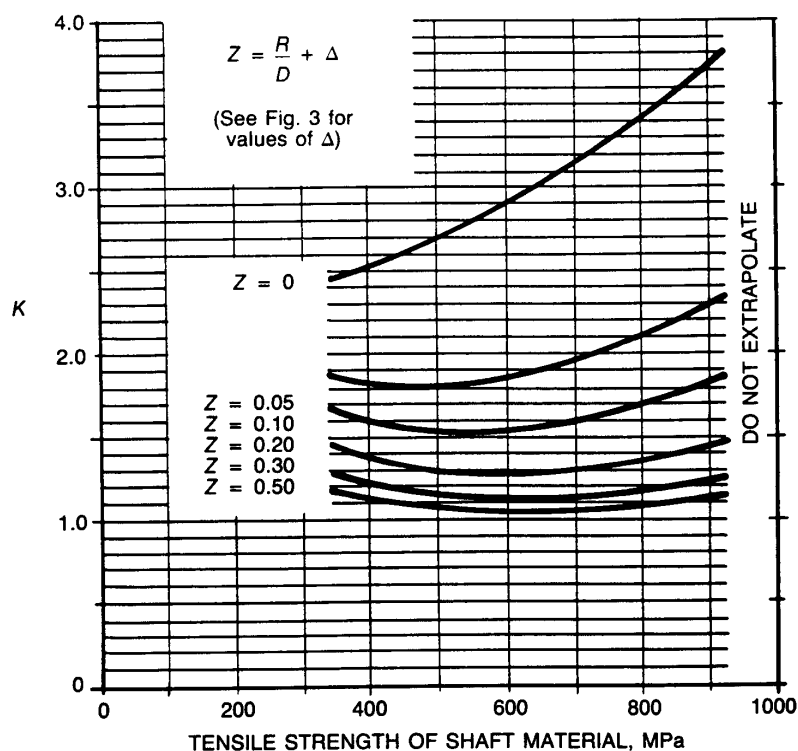
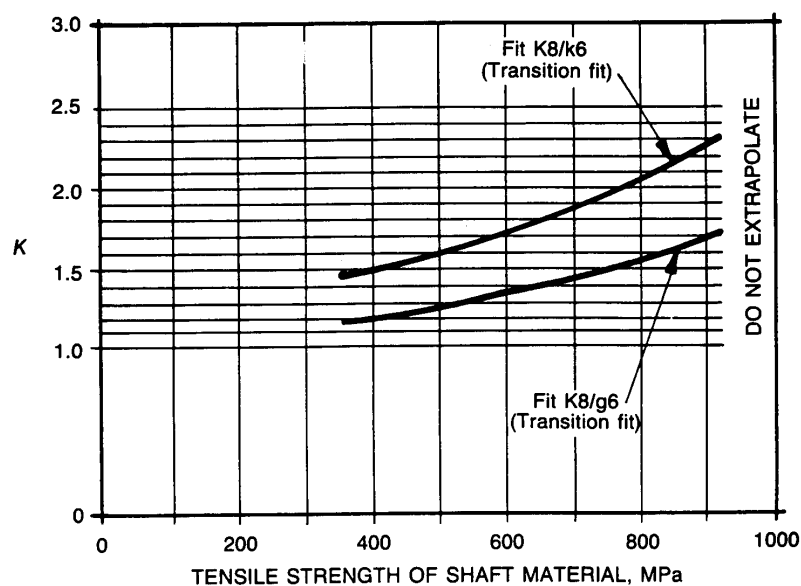
FIGURE 1 SIZE FACTOR K_s 

FIGURE 2 TYPICAL EXAMPLES OF STRESS-RAISING CHARACTERISTICS

FIGURE 3 CORRECTION FACTOR Δ

FIGURE 4 STRESS-RAISING FACTOR K FOR STEPPED SHAFT

NOTES:

- 1 Values may be interpolated for fits between K8/k6 and K8/g6 which are recommended by the bearing manufacturers.
- 2 For tolerances, see AS 1654.

FIGURE 5 STRESS-RAISING FACTOR K FOR SHAFT FITTED WITH ROLLING ELEMENT BEARING