BRITISH STANDARD

BS 2573-2: 1980

(Reprinted, incorporating Amendments No. 1 and No. 2 inclusive.)

Rules for the design of cranes —

Part 2: Specification for classification, stress calculations and design of mechanisms

UDC 621.873.03:621.873–23

Cooperating organizations

The Mechanical Engineering Standards Committee, under whose direction this British Standard was prepared, consists of representatives from the following:

Associated Offices Technical Committee*
Association of Consulting Engineers
Association of Hydraulic Equipment Manufacturers
Association of Mining Electrical and Mechanical Engineers
British Compressed Air Society
British Electrical and Allied Manufacturers' Association (BEAMA)
British Gas Corporation
British Gear Manufacturers' Association
British Internal Combustion Engine Manufacturers' Association
British Pump Manufacturers' Association
British Steel Corporation
British Steel Industry*
British Valve Manufacturers' Association Ltd.
Cbmpe
Chartered Institution of Building Services
Crown Agents for Oversea Governments and Administrations*
Department of Industry (Mechanical Engineering)*
Department of Industry (National Engineering Laboratory)*
Department of the Environment (PSA)
Department of Trade (Marine Division)
Department of Transport
Electricity Supply industry in England and Wales
Engineering Equipment Users' Association
Federation of Manufacturers of Construction Equipment and Cranes*
Health and Safety Executive*
Institution of Gas Engineers
Institution of Mechanical Engineers*
Institution of Plant Engineers
Institution of Production Engineers*
London Transport Executive
Machine Tool Industry Research Association
Ministry of Defence*
National Coal Board
Oil Companies Materials Association*
Process Plant Association
Society of Motor Manufacturers and Traders Limited
Telecommunication Engineering and Manufacturing Association (TEMA)

The organizations marked with an asterisk in the above list, together with the following, were directly represented on the Technical Committee entrusted with the preparation of this British Standard:

British Ports Association
British Railways Board
British Transport Docks Board
Bureau of Engineer Surveyors
Contractors' Plant Association
Control and Automation Manufacturers' Association (BEAMA)
Department of the Environment (Building Research Establishment)
Department of Transport
Federation of Civil Engineering Contractors
Federation of Wire Rope Manufacturers of Great Britain
High Commission of India
Independent Engineering Insurers' Committee
Institute of Materials Handling
Institute of Quality Assurance
Institution of Structural Engineers
National Federation of Building Trades Employers
Port of London Authority
Welding Institute
Individual manufacturers

This British Standard, having been prepared under the direction of the Mechanical Engineering Standards Committee, was published under the authority of the Executive Board and comes into effect on 31 December 1980

The following BSI references relate to the work on this standard:
Committee reference MEE/41
Draft for comment 78/77677 DC

ISBN 0 580 11763 4

Amendments issued since publication

Amd. No.	Date of issue	Comments
3952	September 1982	
5013	July 1986	Indicated by a sideline in the margin

Contents

		Page
Cooperating organizations		Inside front cover
Foreword		iii
1	Scope	1
2	References	1
3	Service life of a mechanism or component	1
4	Classification of mechanisms	1
4.1	General	1
4.2	Class of utilization	1
4.3	State of loading	1
4.4	Determination of group classification	2
5	Loadings to be considered in the design of mechanisms	2
5.1	General	2
5.2	Loading conditions	3
5.3	Load combinations	3
6	Verification of components	4
7	Working stresses	4
7.1	Checking for strength	4
7.2	Checking for fatigue	4
8	Selection of components	6
8.1	General	6
8.2	Shafts	6
8.3	Bearings	6
8.4	Gearing	6
8.5	Ropes and rope reeving components	7
8.6	Lifting hooks	9
8.7	Rail wheels	9
8.8	Determination of limiting bearing stress, P_L	10
8.9	Determination of the coefficient, c_1	10
8.10	Determination of the coefficient, c_2	10
8.11	Verification of bearing stress	10
8.12	Bolts	10
9	Welding	11
Appendix A Typical classification of crane mechanism		12
Appendix B Assessment of torque loading of crane mechanisms		16
Appendix C Determination of fatigue reference stress		17
Appendix D Gearing		21
Appendix E Derating of hooks		21
Figure 1 — Fatigue reference stress factor, $C_{f,\ r}$		19
Figure 2 — Size factor, C_2		20
Figure 3 — Surface finish factor, C_3		20
Table 1 — Classes of utilization of mechanisms		1
Table 2 — States of loading of mechanisms		2
Table 3 — Group classification of a mechanism		2
Table 4 — Permissible stress, P_c, for mechanism components		5
Table 5 — Mechanism duty factor, G		5
Table 6 — Minimum coefficient of utilization, Z_p		8
Table 7 — Values of H for rope reeving components		9
Table 8 — Limiting bearing stress, P_L		10

	Page
Table 9 — Coefficient, c_1, for rail wheels	11
Table 10 — Coefficient, c_2, for rail wheels	11
Table 11 — Surface finish factor, C3	11
Table 12 — Hours per day for gear design	21
Table 13 — Derating of hooks (Table 2 of BS 2903:1970)	21
Table 14 — Derating of hooks (Table 1 of BS 2903:1970)	21
Publications referred to	Inside back cover

Foreword

The preparation of a British Standard dealing with design stress levels in the mechanisms of cranes was begun in response to demand from the user industries and particularly the public sector. Whilst British Standards have existed, for many years, concerned with specific mechanical items that are used in cranes, such as steel wire ropes, hooks and antifriction bearings, the process of design of crane mechanisms has received only limited attention in the British Standards for particular types of crane. Following the successful introduction of BS 2573-1 to regulate and rationalize design stress levels in crane structures of all types, and its recent revision to align more closely with international policy in the International Organization for Standardization (ISO), it became increasingly urgent to produce a corresponding standard for crane mechanisms, and developments in ISO Technical Committee 96, Cranes, Lifting Appliances and Related Excavator Equipment, indicated the sort of approach that might be adopted.

Whereas in the case of crane structures there was considerable affinity with the fields of steel girder bridges and the use of structural steel in building which, to some extent, determined the analytical treatment, crane mechanisms involve specialized empirical design and problems can arise in the selection of proprietary components for which there is little precedent in other British Standards. As in the case of BS 2573-1, the "Rules for the design of hoisting appliances Section 1: heavy lifting equipment" of the Fédération Européenne de la Manutention (F.E.M.) were available for the technical committee to consider together with such work of ISO/TC 96 as had reached a definitive form.

A difficult decision had to be taken on the method of classification to adopt because the needs of specially designed cranes and series built standard cranes were to some extent in conflict. An approach, based on ISO work, was finally adopted which enables each mechanism of a crane to be classified in respect of the duty it performs, independently of the design classification of the whole crane. Whilst, to some users, the unfamiliarity of this concept may initially raise doubts, it is considered to give the best possible scope for precise definition of the purchaser's requirements on the one hand and for rational mechanism design and economic production on the other.

A concept of service life has been defined with the object of providing a basis for compatibility of mechanical components in a mechanism, whether selected proprietary items or purpose designed, in respect of the time during which each component is in motion. In practical terms this should ensure, for example, that the couplings, shafts, gears, bearings, etc., in a particular train all have a comparable basis of durability.

It was essential to recognize in the standard that experience of crane mechanisms under known service conditions is the most valuable and reliable basis for design, and that absolute stress computations are the exception rather than the rule. The content has also been generally confined to the permissible stress and design rule aspects, since any attempt to provide design guidance beyond this level would involve a very extensive practical design manual, which is not the intent of the standard. Mention is made of practical design details only to the extent that they are inseparably associated with the basis for the permissible stresses.

The committee found its scope limited in certain areas where the revision of some British Standards has been held back in order to take advantage of work which is being carried out in ISO. Hooks and the capacity of gears are particular instances in which it should be possible to enlarge the treatment when ISO progress permits.

The selection of steel wire ropes as specified in **8.5**, is based on ISO work which is known to have wide acceptance and, in some cases, permits lower factors of safety than have been used in the past. It should be noted that there will often be overriding regulations or other limitations on factors of safety for crane ropes, and also that the provisions in this standard presuppose regular inspection and discard criteria in accordance with established recommendations.

An outline procedure for fatigue verifications has been included, which defines recommended principles, but it is not possible, in the present state of the art, to include a comprehensive treatment, and reference to specialized text books may be necessary.

The rules included for determining rail wheel sizes have been based on F.E.M. and German work which is known to have given satisfactory results in service over a considerable period.

Because cranes lift and transport heavy loads, there is a major safety aspect in their design which has, to some extent, to be related to the level of design stresses permitted. Nevertheless, it has been the general experience of users, manufacturers and the safety authorities that, where design was found to contribute to an accident, it was nearly always caused by bad detail design or failure to appreciate the loadings involved rather than the selection of too high a permissible stress. Whilst, therefore, the present standard has a significant contribution to make to improved standards of industrial safety, it should be emphasized that the main responsibility for safe design will always remain with the manufacturer and designer making proper use of their specialist experience.

In line with current policy, the values in this Part of this British Standard have been expressed in SI units.

A British Standard does not purport to include all the necessary provisions of a contract. Users of British Standards are responsible for their correct application.

Compliance with a British Standard does not of itself confer immunity from legal obligations.

Summary of pages

This document comprises a front cover, an inside front cover, pages i to iv, pages 1 to 22, an inside back cover and a back cover.

This standard has been updated (see copyright date) and may have had amendments incorporated. This will be indicated in the amendment table on the inside front cover.

1 Scope

This Part of BS 2573 provides the basis for calculating stresses in components of crane mechanisms and specifies the way in which permissible stresses shall be determined. It also gives rules for the selection of proprietary items where these are used as mechanism components.

2 References

The titles of the standards publications referred to in this standard are listed on the inside back cover.

3 Service life of a mechanism or component

The service life of a mechanism or component is defined as "the total number of hours during which the mechanism or component is in motion". Mechanisms and components shall be designed or selected for a particular intended service life and one of the nominal values specified in Table 1 shall be used.

NOTE 1 In many cases it is desirable that separate components forming part of a mechanism (e.g. the couplings, shafts, gears, bearings, etc., in a particular train) should be designed or selected for the same service life. Where components of a mechanism are brought into use for only part of the time that the mechanism as a whole is in motion, the components can be designed for a shorter service life to take this into account.

NOTE 2 In some cases, economic considerations or technical limitations dictate that a component should be designed or selected for a shorter life than the remainder of the mechanism, so that renewal of the component will be necessary during the life of the mechanism: for example a rope with a service life shorter than that of the remainder of the hoist mechanism can be used.

The service life for which a mechanism or component is designed is a figure used only as a design parameter and shall not be taken as implying a guaranteed life.

4 Classification of mechanisms

4.1 General. For the purpose of applying this Part of this British Standard, mechanisms shall be classified into groups according to their desired service life and the conditions of loading to which they are subjected. The purposes of such classification is to provide a rational and uniform basis for the design of mechanisms and their components that takes account of these considerations. It also provides a framework of reference between the purchaser and the manufacturer. The classification system adopted follows that specified by ISO in ISO 4301.[1)]

The two factors considered in determining the group to which a mechanism belongs shall be its class of utilization and its state of loading.

4.2 Class of utilization. The class of utilization of a mechanism shall be determined from its assumed service life in hours. This is the total number of hours for which the mechanism will be in motion. Where the appliance of which the mechanism is part has known duties, such as would be the case where it performed part of a continuous, repetitive process, the service life shall be calculated from the actual daily utilization time for the mechanism, the number of working days per year and the number of years of expected service. In the absence of such information regarding the duty of the appliance, an assumed daily utilization time shall be used.

The class of utilization shall be selected according to the value of service life given in Table 1 that is nearest to, but not less than, the assumed service life determined by the procedure outlined in the previous paragraph.

Table 1 — Classes of utilization of mechanisms

Class of utilization	Service life hours	Remarks
T1	400	Irregular use
T2	800	
T3	1 600	
T4	3 200	Regular light use
T5	6 300	Regular intermittent use
T6	12 000	Regular intensive use
T7	25 000	Continuous intensive use
T8	50 000	
T9	> 50 000	

4.3 State of loading

4.3.1 The state of loading characterizes the extent to which a mechanism is subjected to its maximum loading and to smaller loadings.

[1)] In course of preparation.

4.3.2 Where details are available of the loading that the mechanism will experience, the nominal load spectrum factor, K_m, and corresponding state of loading shall be determined as follows:

let

- P_i be the individual load magnitudes (loading levels) characteristic of the duty of the mechanism (that is $P_1, P_2, P_3, ..., P_n$);
- P_{max} be the maximum loading applied to the mechanism;
- t_i be the individual durations of use of a mechanism at the individual loading levels P_1, P_2, P_3, etc., (that is $t_1, t_2, t_3, ..., t_n$);
- T be the total duration of use at all loading levels = $(t_1 + t_2 + t_3, ..., t_n)$;

then the spectrum factor, K'_m, is given by the formula

$$K'_m = \sqrt[3]{\frac{t_1}{T}\left(\frac{P_1}{P_{max}}\right)^3 + \frac{t_2}{T}\left(\frac{P_2}{P_{max}}\right)^3 ... + \frac{t_n}{T}\left(\frac{P_n}{P_{max}}\right)^3}$$

The nominal load spectrum factor and the corresponding state of loading are obtained from Table 2 by selecting the value of K_m that is nearest to, but not less than, the calculated value of, K'_m.

4.3.3 When details of the loading to which the mechanism will be subjected are not known, an appropriate nominal state of loading shall be selected and the descriptive remarks in Table 2 used to assist in such cases.

NOTE For purpose-built cranes, the selection of an appropriate nominal state of loading for each mechanism may be agreed between the purchaser and manufacturer.

4.4 Determination of group classification. When the class of utilization and the state of loading has been determined, the group classification of a mechanism shall be determined from Table 3.

The manufacturer shall in every case specify to the purchaser, and clearly and permanently mark on the crane, the group in which each mechanism of a crane has been classified for design purposes.

Appendix A gives examples of group classification ranges for the mechanisms of various types of crane in particular applications.

Table 2 — States of loading of mechanisms

States of loading	Nominal load spectrum factor, K_m	Remarks
L1 Light	0.50	Mechanisms subjected very rarely to their maximum load and, normally, to very light loads
L2 Moderate	0.63	Mechanisms occasionally subjected to their maximum load but, normally, to rather light loads
L3 Heavy	0.80	Mechanisms frequently subjected to their maximum load and, normally, to loads of medium magnitude
L4 Very heavy	1.00	Mechanisms regularly subjected to their maximum loads

Table 3 — Group classification of a mechanism

State of loading	Classes of utilization								
	T1	T2	T3	T4	T5	T6	T7	T8	T9
L1	M3	M3	M3	M3	M4	M5	M6	M7	M8
L2	M3	M3	M3	M4	M5	M6	M7	M8	M8
L3	M3	M3	M4	M5	M6	M7	M8	M8	M8
L4	M3	M4	M5	M6	M7	M8	M8	M8	M8

5 Loadings to be considered in the design of mechanisms

5.1 General. The following loadings shall be taken into account when designing a mechanism and its components.

- R_1 Loads due to the dead weight of the mechanism or component.
- R_2 Loads due to the dead weight of those parts of the crane acting on the mechanism or component under consideration.
- R_3 Loads due to the weight of the hook load.
- R_4 Loads due to the weight of the hook load augmented by the impact factor given in **5.3** of Part 1:1977 of this British Standard.

R_5 Dynamic loadings arising from the acceleration or braking of the motion and from skewing interaction between crane and track. These loads include those due to the inertia of the mechanism itself, its prime mover, brakes, the associated crane parts, the hook load, the concurrent operation of other motions, etc., as applicable.

R_6 Loads arising from frictional forces.

V_1 Loads due to the service wind acting horizontally in any direction, where applicable. (See **5.7** of Part 1:1977 of this British Standard.)

V_2 Loads due to the out of service wind acting horizontally in any direction where applicable. (See **5.7** of Part 1:1977 of this British Standard.)

B Loads due to collision with resilient buffers.

Loads shall be quantified by the designer having regard to the part played by the mechanism in the service and environment for which the crane is designed. Appendix B gives guidance on the assessment of torque loading of crane mechanisms.

5.2 Loading conditions. Each mechanism of the crane shall be designed to operate satisfactorily under the most unfavourable combination of loadings that can occur under the following conditions.

Case 1 Normal service without wind.

Case 2 Normal service with wind.

In addition to meeting the in-service requirements under cases 1 and 2, the crane mechanism shall, where appropriate, be designed to withstand the loadings to which they may be subjected under cases 3 and 4.

Case 3 Crane out of service.

Case 4 Exceptional loading conditions appropriate to the type of crane and its application that occur infrequently during its life (e.g.during erection of the crane).

5.3 Load combinations

5.3.1 *General.* In determining the loading used as the basis for design under each of the loading conditions given in **5.2**, the value taken shall be that resulting from the most unfavourable combination of the loads specified in this clause that could actually occur in practice.

NOTE It should be noted that this will not necessarily correspond to the combination of the maximum values of each of the individual loads.

5.3.2 *Case 1: normal service without wind*

a) Vertical motion: hoisting or lowering

1) $R_1 + R_2 + R_3 + R_5 + R_6$

2) $R_1 + R_2 + R_4$

b) Horizontal motion: traverse or travel

$R_1 + R_2 + R_3 + R_5 + R_6$

c) Horizontal motion: slewing

$R_1 + R_2 + R_3 + R_5 + R_6$

d) Horizontal/vertical motion (see note in **5.3.5**)

$R_1 + R_2 + R_3 + R_5 + R_6$

5.3.3 *Case 2: normal service with wind*

a) Vertical motion: hoisting or lowering

1) $R_1 + R_2 + R_3 + R_5 + R_6 + V_1$

2) $R_1 + R_2 + R_4 + V_1$

b) Horizontal motion: traverse or travel

$R_1 + R_2 + R_3 + R_5 + R_6 + V_1$

c) Horizontal motion: slewing

$R_1 + R_2 + R_3 + R_5 + R_6 + V_1$

d) Horizontal/vertical motion (see note in **5.3.5**)

$R_1 + R_2 + R_3 + R_5 + R_6 + V_1$

NOTE In computing the acceleration forces, R_5, it should be remembered that the accelerations achieved can be influenced by the presence of wind forces. In particular, a wind force that opposes a motion can significantly reduce the acceleration, with consequent reduction of the acceleration force.

5.3.4 *Case 3: crane out of service*

a) Vertical motion: hoisting or lowering

Not applicable

b) Horizontal motion: traverse or travel

$R_1 + R_2 + V_2$

c) Horizontal motion: slewing

$R_1 + R_2 + V_2$

d) Horizontal/vertical motion (see note in **5.3.5**)

$R_1 + R_2 + V_2$

5.3.5 *Case 4: exceptional loading conditions.* The combinations of loads to be considered for this case of loading will depend upon the type of crane, the application and the crane motion. Account shall be taken of any loading conditions that are known to apply but which are not covered under the other three cases of loading.

A specific case is buffer load B, which is covered by **5.6** of Part 1:1977 of this British Standard.

During erection or dismantling operations it should be noted that, unless the operation can be completed during a period when the wind does not exceed V_1 conditions, the parts being erected or dismantled should be so secured that they will withstand a wind of V_2 conditions.

NOTE Combined horizontal/vertical motion occurs during the following operating conditions:

a) luffing with a non-level luffing crane or telescoping;

b) travel or traverse on an inclined plane;

c) slew on an inclined plane.

6 Verification of components

Non-proprietary mechanism components shall be proportioned by checking that they have adequate safety against becoming unserviceable as a result of a single combination of loads causing fracture, bending, or other type of failure. They shall also, when appropriate, be checked against fatigue. Due regard shall also be taken of other factors, such as deflection or overheating, which could interfere with the correct functioning of the mechanism.

NOTE In cases where there is adequate service experience of similar or comparable components under known duty conditions, the checks may be confined to comparisons or extrapolations on a suitable basis.

In general, components shall be required not to fail under extreme loading conditions that are likely to occur occasionally during service. In this connection, consideration shall be given to the consequences of failure. For example, failure in a hoisting mechanism component is usually a more dangerous occurrence than failure of a horizontal motion mechanism. Components shall also be designed or selected to give an economic life in terms of durability that can be affected by fatigue, by wear, by temperature and by vibration. When considering durability, the loading combinations appropriate to normal operating conditions shall be considered.

7 Working stresses

7.1 Checking for strength

7.1.1 *General.* Mechanism components shall be checked for strength under case 1, 2, 3 and 4 loading conditions specified in **5.2** and the load combinations specified in **5.3**. The calculated stress, F_c, determined in accordance with **7.1.3**, shall not exceed the permissible stress, P_c, defined in **7.1.2**.

7.1.2 *Permissible stresses.* The permissible stress, P_c, is obtained from Table 4 according to the type of stress and case of loading under consideration.

In Table 4:

U_s is the ultimate tensile strength of the material.

G is a mechanism duty factor taken from Table 5 according to the group classification of the mechanism (see clause **3**). It takes account of the greater possibility in high speed mechanisms, and those in intensively used cranes handling loads near their maximum capacity, that combined stresses greater than those derived from calculation will occur.

7.1.3 *Calculated stresses.* Where a component is subjected to axial tensile, axial compressive, bending or shear stress, F_c shall be calculated in the normal manner.

Where a component is subjected to a combination of co-existent stresses, F_c is the combined stress which shall be calculated as follows:

Combined bending and tension

$$F_c = 1.25f_t + f_{bt}$$

Combined bending and compression

$$F_c = f_c + f_{bc}$$

Combined bending, tension and shear

$$F_c = \sqrt{(1.25f_t + f_{bt})^2 + 3f_q^{\,2}}$$

Combined bending, compression and shear

$$F_c = \sqrt{(f_c + f_{bc})^2 + 3f_q^{\,2}}$$

where

f_t is the calculated axial tensile stress

f_c is the calculated axial compressive stress

f_{bt} is the calculated maximum tensile stress due to bending about both principal axes

f_{bc} is the calculated maximum compressive stress due to bending about both principal axes

f_q is the calculated shear stress.

7.2 Checking for fatigue

7.2.1 *General.* Where it is necessary to check a component for fatigue, use shall be made of recognized techniques available for calculation and comparison of fatigue properties.

NOTE 1 The general method outlined in this clause does not purport to be the only method, and more accurate predictions of fatigue life can be attained by using other appropriate methods in particular circumstances.

NOTE 2 The selection procedures specified in this standard make allowance for fatigue in the case of ropes and gears. In general, similar allowances are included in the procedures given by manufacturers for the selection of proprietary anti-friction bearings.

Table 4 — Permissible stress, P_c, for mechanism components

Type of stress	Case of loading (see **5.1** and **5.2**)		
	1	2	3 and 4
Axial tension	$0.285 \times G \times U_s$	$0.322 \times G \times U_s$	$0.40 \times U_s$
Axial compression	$0.36 \times G \times U_s$	$0.40 \times G \times U_s$	$0.50 \times U_s$
Bending	$0.36 \times G \times U_s$	$0.40 \times G \times U_s$	$0.50 \times U_s$
Shear	$0.20 \times G \times U_s$	$0.226 \times G \times U_s$	$0.288 \times U_s$
Combined stresses	$0.36 \times G \times U_s$	$0.40 \times G \times U_s$	$0.50 \times U_s$

Table 5 — Mechanism duty factor, G

Group classification of mechanism						
	M3	M4	M5	M6	M7	M8
G	1.0	1.0	0.9	0.8	0.7	0.63

7.2.2 *Loading conditions.* Case 1 loading conditions specified in **5.2** shall be used as the basis for all fatigue checks.

7.2.3 *Determination of applied stresses and numbers of stress cycles.* The following characteristics shall be determined for each type of fluctuating stress (e.g. tension, bending or shear) occurring during an appropriate stress cycle in the component detail having regard to the loading that it will experience.

f_{max}, f_{min} The extreme values of stress occurring in the stress cycle, where f_{max} is the stress of greater magnitude and f_{min} is considered as negative if it is of opposite sense to f_{max}

$\frac{f_{min}}{f_{max}}$ The maximum degree of stress fluctuation.

n The total number of stress cycles during the service life of the component.

7.2.4 *Individual stresses.* Where a component detail is subject to a single type of fluctuating stress, f_{max} shall not exceed P_f, the permissible fatigue stress determined as as in **7.2.6**.

7.2.5 *Combined stresses.* The stress combination occurring most frequently in practice in a component detail under case 1 loading conditions is that of bending and torsion and a detail subjected to this combination shall be so designed that

$$\left(\frac{f_{b\,max}}{P_{f,b}}\right)^2 + \left(\frac{f_{q\,max}}{P_{f,q}}\right)^2 \leqslant 10$$

where

$f_{b\,max}$ is the maximum calculated bending stress

$f_{q\,max}$ is the maximum calculated shear stress

$P_{f,b}$ is the permissible fatigue stress in bending

$P_{f,q}$ is the permissible fatigue stress in shear

The permissible fatigue stresses, $P_{f,b}$ and $P_{f,q}$, shall be determined as in **7.2.6**.

7.2.6 *Permissible fatigue stress.* The permissible fatigue stress, P_f, for each type of stress (e.g. tension, bending or shear) is given by:

P_f = 0.80 $P_{f,r}$ for hoisting mechanisms

= 0.85 $P_{f,r}$ for all other mechanisms

where $P_{f,r}$, the fatigue reference stress, is that at which a component detail has a 90 % probability of survival. It shall be determined from test data, service experience, or by calculation.

For any component, the magnitude of the fatigue reference stress depends upon:

a) *the applied loadings*

1) the number of stress cycles, n;

2) the type of stress cycles (i.e. degree of stress fluctuation characterized by the ratio of f_{min}/f_{max});

b) *the characteristics of the component*

3) the quality of the material;

4) the size of the component;

5) the surface finish of the component;

6) the configuration of the individual detail under consideration.

Appendix C gives a method of calculating the fatigue reference stress which takes account of factors 1) to 6) above, and includes an allowance giving a 90 % probability of survival.

8 Selection of components

8.1 General. In cases where there is adequate service experience of similar or comparable components under known duty conditions, the checks shall be confined to comparisons or extrapolations on a suitable basis.

8.2 Shafts. The material for shafts shall be selected from steels having suitable physical properties for the duty required. Suitable steels are specified in BS 970.

Where other steels are used, the crane manufacturer shall show that the properties are comparable with the standard steels and that they have been subjected to equivalent tests.

BS 1134 deals with assessment of surface texture and gives guidance on the surface finishes resulting from various methods of production. Suitable surface roughness values for stressed portions of shafts shall be selected having regard to size, stress levels, severity of stress raising features and the needs of bearings, seals, etc. Stresses shall not, however, exceed those specified in clause **7**. In addition, adequate allowances shall be made for the effective losses in section due to keyways, splines, etc. In addition, the requirements for stiffness (lateral and torsional), fatigue, influence of bearings, fits of components, etc. shall be taken into account when determining shaft sizes.

Large changes in section shall be avoided wherever possible. Where such changes in section are unavoidable, a component manufactured from a purpose-made forging is preferable to one machined from bar.

8.3 Bearings. Bearing pedestals and mountings shall be capable of transmitting the load from the bearing to the supporting structure, suitable provision being made for cases where the resultant load is not acting as a compression load on a bedplate.

NOTE Because crane mechanisms are mounted on structures that are flexible to a greater or lesser extent, particular care is needed in selecting bearings of an appropriate type for each application.

8.3.1 ***Anti-friction bearings***

8.3.1.1 *General.* Anti-friction bearings shall be installed and fitted in accordance with the manufacturer's procedures, including adequate provision for their lubrication.

Arrangements for axial location shall be in accordance with correct practice so that unforeseen additional bearing loads are not introduced. Adequate provision shall be made for axial loading on the bearings of rail wheels, pulleys, drums, etc.

8.3.1.2 *Selection procedure*

8.3.1.2.1 *Maximum load capacity.* A check shall be made in accordance with the bearing manufacturer's recommendations to ensure that the bearing is capable of withstanding the greatest static and/or dynamic load it can be subjected to under whichever of loading cases 1, 2, 3 and 4 is the most unfavourable.

8.3.1.2.2 *Theoretical life.* The rating of anti-friction bearings shall be in accordance with the bearing manufacturer's recommendations for the particular crane duty.

Anti-friction bearings shall be selected to give an L_{10} life in accordance with BS 5512-1 compatible with the service life of the mechanism.

Where the bearing manufacturer's selection procedure includes a method of determining an equivalent load for a spectrum of loading and this differs from the method described in clause **4**, the bearing manufacturer's method shall be followed.

8.3.2 *Slewing rings.* Attention shall be given to the method of mounting and the bolting requirements for which the manufacturer's recommendations shall be taken into account.

Design and selection shall have regard to the potentially serious consequences of failure.

NOTE In the case of proprietary slewing rings, it is particularly important that the manufacturer is consulted and given full details of the loads and duty involved.

8.3.3 *Plain bearings.* The selection, fitting and lubrication of plain bearings shall be in accordance with the bearing manufacturer's procedures. The pressure and rubbing velocity shall be in accordance with recommended practice and service experience for the material involved, having regard to the nature of the loads and operating conditions.

8.4 Gearing

8.4.1 *General.* The most reliable basis for porportioning gearing of a crane motion is service experience under conditions corresponding to the classification parameters involved.

Gear calculations shall, where appropriate, be in accordance with the following British Standards:

BS 436, BS 545 and BS 721.

However, crane gearing can be designed or selected on the basis of other recognized standards or codes which have been proved satisfactory in crane service.

In all cases the design parameters for gear life and loading shall be as stated in **8.4.2** and **8.4.3**.

8.4.2 *Service life.* The gearing shall be designed for a service life compatible with the service life of the mechanism of which the gearing forms a part.

8.4.3 *Design loading.* The design loading used for gearing calculations shall be determined from the case 1 load combinations in clause **5**, having regard to the prime mover, brake and transmission characteristics for the specific case.

The equivalent design load derived from case 1 load combinations is obtained either by multiplying the maximum loading arising in a cycle of operation by the spectrum factor given in clause **4** or by using the methods specified in the individual gear standard or code to assess the uniform load equivalent of the operating cycle.

Where the individual design standard or code requires the static strength of the gearing to be checked in addition to the strength during the operational cycle, case 3 loading conditions specified in **5.2** shall be used in assessing the highest load that can be developed in the motion.

When selecting a proprietary gear unit, the class of utilization and state of loading shall be used in establishing the service factor to be applied to the design rating of the gear unit.

The use of BS 436 requires a determination of the operating hours per day for a particular crane motion. Appendix D lists the hours per day which shall be used for design purposes for each class of utilization.

8.5 Ropes and rope reeving components.

This clause defines the minimum requirements governing the selection of ropes. It applies to ropes complying with the requirements of ISO 2408; it does not exclude, however, ropes that are not specified in ISO 2408. Attention is drawn to the need for thorough consultation between rope manufacturer and crane manufacturer.

The selection procedure adopted assumes that the ropes are lubricated correctly, that the winding diameters on the pulleys and the drums are selected in accordance with **8.5.3** and that, when in service, the ropes are properly maintained, inspected and periodically replaced in conformity with ISO 4309.[2)]

The selection of rope diameter (and winding diameter) shall be related to the group classification of the mechanism that winds the rope.

When a hoisting appliance is to be used for dangerous handling operations (e.g. molten materials, highly radioactive or corrosive products, etc.) the choice of ropes and winding diameters shall be based on the mechanism group next above that of the mechanism concerned, and only groups M5 to M8 inclusive shall be used. For a mechanism that is classified in group M8, the choice of ropes and winding diameters shall be based on group M8.

This clause, which is based on ISO 4308[2)], introduces a generalized method of rope selection covering all types of cranes and their mechanisms for a wide range of duties. In the present state of experience, values of Z_p below 4.5 shall not be used for running rope selection.

NOTE Attention is drawn to the existence in the UK and elsewhere of regulations concerning factors of safety for crane ropes which may override the requirements of this standard.

8.5.1.1 *Maximum tensile force S in the hoist rope* (grab ropes excepted). This shall be obtained by taking account of the following factors:

- the rated working load of the appliance;
- weight of the hook, pulley block and the hoist accessories, the dead weights of which are to be added to the load;
- the rope reeving system employed;
- the efficiency of the rope reeving system;
- rope inclination at the upper extreme position of the hook if the rope inclination with respect to the hoist axis exceeds 22½°

8.5.1.2 *Maximum tensile force, S, in ropes other than hoist ropes.* The determination of the maximum tensile force, *S*, in the various ropes which do not exclusively serve in the vertical hoisting of the load shall be based on the loads determined from the loading conditions specified in **5.2** together with the load combinations specified in **5.3** which can actually occur in normal operation of the appliance.

For ropes exerting forces in directions near the horizontal, account shall be taken of rolling resistance, friction of axles and pulleys and maximum inclination of the support along which a load is moved. Any pre-loading arrangements shall also be considered.

[2)] In course of preparation.

8.5.1.3 *Maximum tensile force, S, in grab ropes (holding and closing ropes).* In the case of appliances with grabs, where the weight of the load is not always equally distributed between the closing ropes and the holding ropes during the whole of the cycle, the value of S to be applied shall be determined as follows.

a) If the hoist system used automatically ensures an equal division of the hoisted load between the closing and holding ropes, and any difference between the loads carried by the ropes is limited to a short period at the end of the closing or the beginning of the opening, S shall be determined as follows:

1) closing ropes: S = 66 % of the weight of the loaded grab, divided by the number of closing ropes.

2) holding ropes: S = 66 % of the weight of the loaded grab, divided by the number of holding ropes.

b) If the system used does not automatically ensure an equal division of load between the closing ropes and the holding ropes during the hoisting motion, and in practice almost all the load is applied to the closing ropes, S shall be determined as follows:

1) closing ropes: S = total weight of the loaded grab divided by the number of closing ropes.

2) holding ropes: S = 66 % of the total weight of the loaded grab divided by the number of holding ropes.

8.5.2 ***Selection procedure***

8.5.2.1 *General.* The selection procedure given in **8.5.2.2** and **8.5.2.3** is in accordance with ISO 4308[3)] and with the method which has been used in the British Standards relating to particular types of crane.

8.5.2.2 *Minimum coefficient of utilization, Z_p.* Table 6 gives the minimum coefficient of utilization, Z_p, for each mechanism group.

8.5.2.3 *Determination of the breaking load of the rope.* The minimum breaking load of a rope intended for a particular duty shall be determined from:

$$F_o = S.Z_p$$

Where

F_o is the minimum required breaking load (in kN);

S is the maximum rope tension (in kN) as defined in **8.5.1**;

Z_p is the minimum practical coefficient of utilization as defined in Table 6 for the appropriate class of mechanism.

Table 6 — Minimum coefficient of utilization, Z_p

Group of mechanism	Value of Z_p	
	Running ropes	Static ropes[a]
M3	3.55[b]	3[b]
M4	4.0[b]	3.5[b]
M5	4.5	4
M6	5.6	4.5
M7	7.1	5
M8	9.0	5

[a] For a static rope the minimum coefficient of utilization, Z_p, is determined by the group of the mechanism with which it is associated. Where a static rope is not clearly associated with a particular mechanism, the minimum coefficient of utilization, Z_p, shall be taken as that corresponding to the hoist mechanism of the appliance.

[b] See **8.5**.

8.5.3 ***Pulleys and drums***

8.5.3.1 *Minimum winding diameter.* The minimum winding diameter of the rope shall be determined by checking compliance with the relationship:

$$D \geqslant H.d$$

Where

D is the minimum winding diameter (in mm) on pulleys, drums or compensating pulleys measured at the rope centres;

H is a coefficient depending upon the group of the mechanism;

d is the nominal diameter of the rope (in mm) which will achieve the minimum breaking load established in **8.5.2.3**.

The minimum values of the coefficient, H, according to the group in which the mechanism is classified are given in Table 7 for drums, pulleys and compensating pulleys.

[3)] In course of preparation.

Table 7 — Values of *H* for rope reeving components

Group of mechanism	Drums	Pulleys	Compensating pulleys
M3	14	16	12.5
M4	16	18	14
M5	18	20	14
M6	20	22.4	16
M7	22.4	25	16
M8	25	28	18

8.5.3.2 *Derivation of the theoretical rope diameter.* In the process of selecting a rope the situation can arise where the minimum breaking load achieved exceeds the requirement and leads to a higher coefficient of utilization than the minimum value quoted in Table 6. Under such conditions, the designer shall derive a theoretical rope diameter from the under-mentioned formula, and utilize this calculated value to select the drum and pulley winding diameters for the particular mechanism, provided the selected rope does not exceed the theoretical rope diameter by more than 25 %.

$$d_o = C\sqrt{S}$$

where

d_o is the theoretical rope diameter (in mm)

S is defined in **8.5.2.2**

C is a coefficient derived from

$$C = \sqrt{\frac{Z_p . d^2}{F_p}}$$

where

F_p is the minimum breaking load (in kN) of the rope selected for use

Under these circumstances the formula $D \geqslant H.d$ is modified to:

$$D \geqslant H.d_o$$

8.6 Lifting hooks. Point hooks and C hooks shall comply with the requirements of BS 2903 when applicable.

Ramshorn hooks shall comply with the requirements of BS 3017 when applicable.

Swivelling hooks used on power driven appliances shall be mounted on rolling bearings.

NOTE 1 For certain steelworks cranes, other heavy duty cranes and cranes which may be used in circumstances under which persons are subject to particularly high risk such as in mine shaft sinking and when raising and lowering persons, it is desirable to derate the capacities of the hooks to values below those stated in the appropriate standard in order to ensure compatibility with the lifting appliance and accessories, to reduce wear or to provide an increased margin of safety.

NOTE 2 Although de-rating is not mandatory for reasons of permissible stress resulting from the working load, it may be advisable from the point of view of obtaining increased endurance under arduous conditions of use. As a typical example for guidance, Appendix E shows a proposed basis for the derating of hooks in accordance with BS 2903 which has regard to their design factor of safety for class 1 crane duty when used on heavy industrial applications.

8.7 Rail wheels

8.7.1 *General.* This clause gives a method of determining the size of rail wheel required to meet a particular duty. The method given assumes that the rail track is maintained in good condition and that the rail wheel is correctly aligned.

The factors that need to be considered when determining the required diameter for a particular rail wheel are:

- the load on the wheel;
- the quality of the material in the wheel rim;
- the type of rail on which the wheel runs;
- the speed of rotation of the wheel;
- the group classification of the mechanism.

The calculation method given in this clause is applicable only to rail wheels that do not exceed 1.25 m in diameter. The use of wheels of greater diameter is not recommended.

Tyres for crane wheels shall, where appropriate, comply with the requirements of BS 3037.

In the case of rail wheels with tyres, consideration shall be given to the design of the tyre, which shall be sufficiently thick not to roll itself out.

8.7.2 *Loading conditions.* Rail wheels shall be checked under the loading conditions specified in **5.2**. In assessing the individual loadings arising under case 1and case 2 loading conditions, the hook load impact factor is not applied.

8.7.3 *Pressure between wheel and rail.* In checking the suitability of a rail wheel for a particular duty, it shall be checked that the pressure between the wheel and the rail is not excessive under the maximum loading conditions, and that the wheel is capable of allowing the crane to perform its normal duty without abnormal wear.

The wheel capacity shall be assessed by comparing the mean bearing stress based on the projected area with a permissible bearing stress determined from a knowledge of the rail/wheel material, the wheel rotational speed and the group classification of the mechanism.

8.7.4 *Determination of the maximum static wheel loading, P_s.* This shall be the maximum wheel load occurring with the wheel static upon the rail in any of the four cases of loading (see **5.2**).

8.7.5 *Determination of the mean wheel loading.* For loading conditions 1 and 2 specified in **5.2**, a mean wheel load shall be determined from the relationship

$$P_{\text{mean}} = \frac{2P_{\text{max}} + P_{\text{min}}}{3}$$

where

P_{max} is the maximum wheel load
P_{min} is the minimum wheel load

occurring during the normal duty of the appliance.

8.7.6 *Determination of the bearing stress, f_B.* The bearing stress, f_B, shall be determined from the maximum static wheel load or the mean wheel loading using the following relationship

$$f_B = \frac{p_s}{bD} \text{ or } \frac{P_{\text{mean}}}{bD}$$

where

D is the wheel diameter
b is the useful width of the rail determined as follows:
For a rail with a total width, w, having rounded corners of radius r at each side and having

a) flat bearing surface

$b = w - 2r$

b) convex bearing surface

$$b = w - \frac{4r}{3}$$

8.7.7 *Determination of the permissible bearing stress, P_B.*

For case 1 and case 2 mean loading conditions the permissible bearing stress, P_B, shall be determined from the relationship

$P_B = P_L \cdot c_1 \cdot c_2$

For the maximum static wheel loading occurring in any of the 4 cases of loading, the permissible bearing stress, P_B, shall be determined from the relationship

$P_B = 1.9\ P_L$

where

P_L is the limiting bearing stress defined in **8.8** as a function of the material of the rail wheel rim
c_1 is a coefficient dependent on the speed of rotation of the wheel as defined in **8.9**
c_2 is a factor dependent on the group classification of the mechanism as defined in **8.10**

8.8 Determination of limiting bearing stress, P_L

In the case of wheels made of high tensile steel and treated to ensure a very high surface hardness, the value of P_L shall be limited to that for the quality of the steel comprising the wheel prior to heat treatment, since a higher value would risk causing premature wear of the rail.

NOTE 1 For a given load, wheels of this type have a much longer useful life than wheels of lesser surface hardness, which makes their use worthwhile in the case of appliances performing intensive duty.

NOTE 2 The use of suitable cast iron with adequate surface hardness is not excluded for rail wheels, but it should be remembered that some such materials are brittle and should not be used where shock loading may occur. The material properties should be carefully matched to the duty, and the limiting bearing stress, P_L, should not exceed 5 N/mm^2

The material properties shall be carefully matched to the duty, and the limiting bearing stress, P_L, shall not exceed 5 N/mm^2.

Table 8 — Limiting bearing stress, P_L

Ultimate strength of metal used for rail wheel	P_L
N/mm^2	N/mm^2
500	5.0
600	5.6
700	6.5
800	7.2
NOTE The qualities of metal refer to cast, forged or rolled steels and spheroidal graphite cast iron.	

8.9 Determination of the coefficient, c_1. The value of the coefficient c_1 depends on the speed of rotation of the wheel and shall be as given in Table 9.

8.10 Determination of the coefficient, c_2. The coefficient, c_2, depends on the group classification of the mechanism and shall be as given in Table 10.

8.11 Verification of bearing stress. For each of the loading conditions specified in **5.2** it shall be verified that:

$f_B/P_B \leqslant 1.0$

where

f_B is determined in accordance with **8.7.6**
P_B is determined in accordance with **8.7.7**

8.12 Bolts. Where appropriate, the term "bolt" also applies to a setscrew or stud. Bolts required to transmit a load from one part of a mechanism to another (alternatively, to transmit a load between a mechanism and a part of the crane structure) shall comply with the requirements of clause **13** of Part 1:1977 of this British Standard.

Bolts other than friction grip bolts shall comply with the requirements of BS 3692, but this does not preclude the use of specialized fasteners where appropriate, in accordance with the manufacturer's recommendations.

Where bolts are tightened by controlled means to comply with the requirements of clause **13** of Part 1:1977 of this British Standard, a method of tightening which allows the preload requirement to be accurately and consistently achieved shall be used.

Table 9 — Coefficient, c_1 for rail wheels

Wheel rotational speed	c_1
r/min	
200	0.66
160	0.72
125	0.77
112	0.79
100	0.82
90	0.84
80	0.87
71	0.89
63	0.91
56	0.92
50	0.94
45	0.96
40	0.97
35.5	0.99
31.5	1.00
28	1.02
25	1.03
22.4	1.04
20	1.06
18	1.07
16	1.09
14	1.10
12.5	1.11
11.2	1.12
10	1.13
8	1.14
6.3	1.15
5.6	1.16
5	1.17

Table 10 — Coefficient, c_2 for rail wheels

Group classification of mechanism	c_2
M3	1.12
M4	1.12
M5	1.00
M6	0.90
M7	0.80
M8	0.80

9 Welding

Where a component is required to transmit a load from one part of the mechanism to another (alternatively to transmit a load from a mechanism to or from a part of the crane structure) and is fabricated by welding, the preparation and welding work shall be carried out in accordance with the requirements of the appropriate British Standards.

NOTE Attention is drawn to the need for special care in connection with weld processes and any heat treatment in mechanical components, and in determining stress levels to which welded joints may be subjected.

Appendix A Typical classification of crane mechanism

A.1 General. This appendix shows the range of crane mechanism classifications normally associated with particular types of crane; it should not be considered as a substitute for the process of determining classification specified in **4.1**, **4.2** and **4.3**.

A.2 Overhead travelling industrial type cranes (O.T.C.)

Type of crane	Motion	Mechanism classification		
		Class of utilization	State of loading	Group of mechanism
Power station O.T.C.	Hoist and aux. hoist	T3	L1 L2	M3
	Cross traverse	T3	L1 L2	M3
	Long travel	T3	L2	M3
O.T.C. for assembly and dismantling machinery	Hoist and aux. hoist	T3	L1 L2	M3
	Cross traverse	T3	L1 L2	M3
	Long travel	T3	L2	M3
O.T.C. stores	Hoist and aux. hoist	T4 T5	L1 L2	M3 M5
	Cross traverse	T4 T5	L1 L2	M3 M5
	Long travel	T4	L2 L3	M4 M5
O.T.C. workshop	Hoist and aux. hoist	T4 T5	L1 L2	M3 M5
	Cross traverse	T4 T5	L1 L2	M3 M5
	Long travel	T4	L2 L3	M4 M5
O.T.C. grabbing	Hoist	T5 T7	L3	M6 M8
	Closing motion	T5 T7	L3	M6 M8
	Cross traverse	T5 T7	L3	M6 M8
	Long travel	T5 T6	L3	M6 M7
O.T.C. scrapyard or magnet	Hoist and aux. hoist	T5 T6	L3	M6 M7
	Cross traverse	T5 T6	L3	M6 M7
	Long travel	T5	L3	M6
O.T.C. with magnet for transporting plates and like	Hoist	T5 T6	L3	M6 M7
	Cross traverse	T5 T6	L3	M6 M7
	Long travel	T5	L3	M6
O.T.C. for containers	Hoist	T5 T7	L2 L3	M5 M8
	Cross traverse	T5 T7	L3	M5 M8
	Long travel	T5 T7	L2 L3	M5 M8
O.T.C. foundry ladle. (For the selection of hoist rope for dangerous handling operation, see **8.5**.)	Hoist	T5 T6	L2 L3	M5 M7
	Cross traverse	T5 T6	L2 L3	M5 M7
	Long travel	T5	L3	M6
O.T.C. process cranes	Hoist	T6 T7	L2 L3	M6 M8
	Cross traverse	T5	L3	M6
	Long travel	T6	L3	M7

A.3 Overhead travelling steel works cranes and transporters

Type of crane	Motion	Mechanism classification					
		Class of utilization		State of loading	Group of mechanism		
Ladle crane (For the selection of hoist rope for dangerous handling operation, see **8.5**.)	Hoist and aux. hoist	T6	T7	L4	M8	
	Cross traverse	T5	T6	L4	M7	M8
	Aux. traverse	T5	T6	L3	M6	M7
	Long travel	T6	T7	L4	M8	
Pig/scrap breaking crane	Hoist	T5	T6	L4	M7	M8
	Cross traverse	T5	T6	L4	M7	M8
	Long travel	T5		L3	M6	
Stripper, vertical ingot charger and soaking pit mould handling crane	Hoist	T7	T8	L3	M8	
	Aux.hoist	T5	T6	L2	M5	M6
	Cross traverse	T6	T8	L4	M8	
	Long travel	T6	T7	L4	M8	
	Slewing	T6	T7	L3	M7	M8
	Closing tong motion	T6	T7	L4	M8	
Furnace charging crane	Hoist and aux. hoist	T7	T8	L3	M8	
	Cross traverse	T7	T8	L3	M8	
	Aux. traverse	T5	T6	L2	M5	M6
	Long travel	T7	T8	L3	M8	
Forging crane	Hoist	T6	T8	L3	M7	M8
	Cross traverse	T5	T6	L3	M6	M7
	Long travel	T6	T8	L3	M7	M8
In line process crane	Hoist	T7		L3	M8	
	Cross traverse	T5		L3	M6	
	Long travel	T6		L3	M7	
Off line process crane	Hoist	T6	T7	L2	M6	M7
	Cross traverse	T5		L2	M5	
	Long travel	T6		L2	M6	
Heavy mill service	Hoist	T5	T6	L2	M5	M6
	Aux. hoist	T4		L2	M4	
	Cross traverse	T5		L2	M5	
	Long travel	T5	T6	L2	M5	M6
Service and maintenance crane	Hoist and aux. hoist	T4	T5	L1 – L2	M3	M5
	Cross traverse	T4	T5	L1 – L2	M3	M5
	Long travel	T4		L2 – L3	M4	M5
Transporter: medium duty general use	Hoist	T5		L2 – L3	M5	M6
	Cross traverse	T5		L2 – L3	M5	M6
	Long travel	T5		L2 – L3	M5	M6
	Boom hoist	T4		L1	M3	

A.3 Overhead travelling steel works cranes and transporters

Type of crane	Motion	Mechanism classification		
		Class of utilization	State of loading	Group of mechanism
Transporter: heavy duty intermittent grabbing and magnet work	Hoist (hold and close)	T6 T7	L3 L4	M7 M8
	Cross traverse	T6 T7	L3 L4	M7 M8
	Long travel	T6	L2 L3	M6 M7
	Boom hoist	T4	L1	M3
Transporter: extra heavy duty continuous grabbing and magnet work	Hoist (hold and close)	T8 T9	L4	M8
	Cross traverse	T7 T8	L4	M8
	Long travel	T7	L3	M8
	Boom hoist	T4	L1	M3

A.4 High pedestal or portal jib cranes and derrick cranes

Type of crane	Motion	Mechanism classification		
		Class of utilization	State of loading	Group of mechanism
(a) Cranes used in docks or elsewhere for handling general cargo and piece goods	Hoist	T5	L1 L2	M4 M5
	Slew	T5	L3	M6
	Luff	T4	L3	M5
	Travel	T3	L2	M3
(b) (1) Cranes used intermittently for grabbing or magnet work	Hoist Holding	T5	L3	M6
	Hoist Closing	T5	L3	M6
	Slew	T5	L3	M6
	Luff	T5	L3	M6
	Travel	T3	L2	M3
(b) (2) Cranes on full time freight container duty	Hoist	T5	L2	M5
	Slew	T6	L3	M7
	Luff	T5	L3	M6
	Travel	T3	L2	M3
(c) Cranes working on full time grabbing duty	Hoist Holding	T6	L3	M7
	Hoist Closing	T6	L3	M7
	Slew	T6	L3	M7
	Luff	T5	L3	M6
	Travel	T4	L2	M4
(d) Shipyard cranes	Hoist	T5	L1	M4
	Aux. hoist	T5	L2	M5
	Slew	T5	L3	M6
	Luff	T5	L2	M5
	Travel	T5	L2	M5

A.5 Tower cranes

Type of crane	Motion	Mechanism classification		
		Class of utilization	State of loading	Group of mechanism
(a) Cranes for normal duty for use on building sites	Hoist	T3	L1	M3
	Traverse	T3	L2	M3
	or Luff	T3	L3	M4
	Slew	T3	L3	M4
	Travel	T3	L2	M3
(b) Medium duty cranes for general use (included in this category are tower cranes that may be used on a permanent site)	Hoist	T5	L1	M4
	Traverse	T4	L2	M4
	or Luff	T4	L3	M5
	Slew	T5	L3	M6
	Travel	T3	L2	M3

A.6 Freight container cranes

Type of crane	Motion	Mechanism classification		
		Class of utilization	State of loading	Group of mechanism
Freight container transporter	Hoist	T5 T7	L2 L3	M5 M8
	Traverse	T5 T7	L2 L3	M5 M8
	Travel	T5 T7	L2 L3	M5 M8
	Boom hoist	T2	L3	M3
Goliath for container handling duty	Hoist	T5 T7	L2 L3	M5 M8
	Traverse	T5 T7	L2 L3	M5 M8
	Travel	T5 T7	L2 L3	M5 M8

A.7 Power driven mobile cranes

Type of crane	Mechanism classification		
	Class of utilization	State of loading	Group of mechanism
(a) Cranes for general use			
Hoist	T2	L3	M3
Slew			
Derrick/Boom telescope	T3	L2	M3
(b) Cranes for heavy duty including intermittent grabbing and magnet work			
Hoist	T5	L3	M6
Slew	T5	L2 L3	M6
Derrick	T3 T5	L2 L3	M6

Appendix B Assessment of torque loading of crane mechanisms

B.1 General. Torques applied to the components of crane mechanisms arise from the interaction of prime movers or brakes with gravitational loads, the inertias of various masses and rotating parts, and with combinations of both. The assessment of appropriate torque values for input to mechanism calculations requires a clear understanding both of the characteristics of the prime mover and of the way in which the mechanism will be required to perform in service. It is an important responsibility of the designer and manufacturer to ensure that torque values are correctly and appropriately quantified so that mechanism will perform as required in the service environment for the intended life. Failure of mechanism components represents a hazard of varying importance and can lead to consequential damage and cost which may be considerable. The question of reliability is critical in the case of cranes used in a process, or for intensive bulk or unit handling. For this purpose, the mechanism design has to be carried out with particular attention to integrity, taking into account the time required to effect repairs arising from accidental or other unforeseen circumstances outside normal service conditions. In **B.2** to **B.5** attention is drawn to some of the factors that have to be borne in mind when assessing torques for mechanism design in general crane applications. More specific requirements may be laid down in standards relating to the individual types of crane as in the case of BS 3579.

B.2 All mechanisms. In calculating the torque applied to a mechanism or component during an accelerating or decelerating motion, it may be necessary to take into account the torques required to accelerate or decelerate the moving parts of the mechanism and drive unit themselves. In some cases, for example, the torque required to accelerate or retard a motor armature or rotor complete with coupling, brake drum and pinion assembly may be proportionately high in relation to the torques required to perform the mechanism function.

Braking often produces higher torques than conditions of acceleration. In either case, account should be taken of the starting characteristics of the drive unit or the retarding properties of the brake system, as appropriate. The assumed rates of acceleration and deceleration should be consistent with the crane type, application and duty, and should also be compatible with the rating of the drive unit. With certain gear arrangements, the overhauling of the prime mover by the inertia of a crane or trolley may give rise to maximum loadings and, with some braking systems, the action of brakes under emergency stop or upon supply failure may give rise to loadings that require consideration in respect of component strength. Such cases are not, however, relevant for fatigue verifications.

Where cushioning devices are inserted in the drive, such as hydraulic couplings or electrical control features that limit the applied torque, the benefit of these devices may be used in assessing the torques transmitted to the mechanism.

When considering horizontal motions, i.e. travelling, traversing, slewing and level luffing, it must be remembered that there is generally a full reversal of torque from maximum in one direction to maximum in the opposite direction.

Further factors relating to specific mechanisms of a crane are discussed in **B.3** to **B.5**.

B.3 Hoisting mechanisms. When forces and torques in hoisting mechanisms are calculated, proper consideration should be given to the dynamic behaviour of the system having regard to the starting and braking characteristics of the drive unit, to the acceleration or deceleration of the hook load and the mechanism (which may, however, be negligible in some cases), to frictional effects, and to the effects of flexibility in the system as a whole. For some purposes, it is appropriate to allow for dynamic effects by considering a steady condition modified by the use of an impact factor. These approaches are implicit in the loading conditions given in clause **5**.

B.4 Horizontal motion mechanisms. When the driving unit of a travel or traverse mechanism has been selected to give high accelerations, the torque transmitted by the mechanism may be limited by skidding of the wheels on the rails. Similar effects may occur during braking or collision.

In making allowance for runner bearing friction, the ratio between tread diameter and bearing diameter has to be taken into account, especially when plain bearings are used.

B.5 Derricking and slewing mechanism. Maximum torques in derricking and slewing mechanisms normally occur during conditions of braking, unless a high rate of acceleration is called for.

Appendix C Determination of fatigue reference stress

C.1 General. Clause **7.2** sets out the procedure for checking the fatigue strength of a mechanical component and outlines the factors which affect its fatigue life. A fatigue reference stress is used in the process of evaluating the fatigue strength. This appendix outlines a method of determining the fatigue reference stress for a component detail.

C.2 Fatigue reference stress, $P_{f,r}$

$P_{f,r} = C_{f,r} U_s$

where

- U_s is the minimum ultimate tensile stress of the material
- $C_{f,r}$ is a fatigue reference stress factor that depends on:
  - a) the number and characteristics of the stress cycles to which a component is subjected; and
  - b) the value of the fatigue limit factor C_{lim}.

C.3 Determination of fatigue reference stress factor $C_{f,r}$. For steels having a minimum ultimate tensile stress not greater than 1 100 N/mm^2, the value of $C_{f,r}$ is given in Figure 1(a) for cases of bending stress and in Figure 1(b) for cases of torsional shear stress. The value of $C_{f,r}$ is determined in accordance with the appropriate number of stress cycles and with the appropriate value of the fatigue limit factor C_{lim}.

Alternatively the factor $C_{f,r}$ can be determined from the following formulae, in which N is the total number of cycles.

For N Š 10^6 $\qquad C_{f,r} = C_{lim}$

For $10^3 < N < 10^6$ $\qquad C_{f,r} = A\,C_{lim}^{(1/3 \log N - 1)}$

where for bending stresses $\qquad A = 1$

and for torsional shear stresses $\qquad A = 3^{(1/6 \log N - 1)}$

C.4 Fatigue limit factor C_{lim}. The fatigue limit factor C_{lim} is the value of $C_{f,r}$ corresponding to the fatigue endurance limit.

$$C_{lim} = \frac{C_1}{C_2\, C_3\, K_f}$$

where

- C_1 is a stress state factor
- C_2 is a size factor
- C_3 is a surface finish factor
- K_f is a fatigue notch factor equal to K_{bf} for bending stresses and K_{qf} for torsional shear stresses

C.5 Determination of C_1, C_2, C_3 and K factors

C.5.1 *Stress factor C_1*. The value of C_1 is given in accordance with the nature of the stress cycles as follows:

Unidirectional bending	$0 < f_{min.}/f_{max.} \leq 1$	C1 = 0.85
Reversing bending	$-1 \leq f_{min.}/f_{max.} < 0$	C1 = 0.50
Unidirectional torsion (shear)	$0 < f_{min.}/f_{max.} \leq 1$	C1 = 0.50
Reversing torsion (shear)	$-1 \leq f_{min.}/f_{max.} < 0$	C1 = 0.30

C.5.2 *Size factor C_2*. The size factor is derived from Figure 2.

C.5.3 *Surface finish factor C_3*. The surface finish factor is dependent on the ultimate tensile stress of the material and the machining process used in manufacture. The value of C_3 is given in Figure 3 in which R_t is the peak-to-valley height of surface roughness (in µm).

C.5.4 ***Fatigue notch factor K_{bf}, K_{qf}***

C.5.4.1 *Derivation of fatigue notch factor.* The presence of such details as shoulders, grooves, holes, keyways, threads, etc. results in a modification of the simple stress distribution that would occur in members which have a constant cross section with gradual changes of contour. The resulting localization of high stresses is known as the stress concentration or notch effect and is measured by the stress concentration factor.

The stress concentration factor is defined as

$$K_b = \frac{f_{b\,max}}{f_{b\,nom}} \quad \text{for bending stress}$$

$$K_q = \frac{f_{q\,max}}{f_{q\,nom}} \quad \text{for torsional shear stress}$$

where $f_{b\,max}$ and $f_{q\,max}$ are the maximum stresses in a component detail and $f_{b\,nom}$ and $f_{q\,nom}$ are the nominal stresses at the corresponding location.

The effect of a notch on the fatigue strength of a part varies considerably with material and notch geometry and is normally less than would be predicted by the use of the stress concentration factor. The general phenomenon is denoted "notch sensitivity".

For design purposes the following relations apply:

$K_{bf} = q\ (K_b - 1) + 1$

$K_{qf} = q\ (K_q - 1) + 1$

where

q is the notch sensitivity factor

K_{bf} is the fatigue notch factor (bending stress)

K_{qf} is the fatigue notch factor (torsional shear stress).

C.5.4.2 *Stress concentration factor*. The stress concentration factors K_b and K_q are to be taken from established reference works.

Two such works are:

a) *Stress Concentration Factors* by R.E. Peterson, published by John Wiley and Sons (1974).

b) *Data sheets* published by Engineering Sciences Data Unit, 251-259 Regent Street, London W1R 7AD.

C.5.4.3 *Notch sensitivity factor*. For ductile materials the value of q may be obtained from published reference works or from experimental data.

Where no data exist, then a value of $q = 1.0$ should be used in deriving K_{bf} and K_{qf}.

For brittle materials $q = 1.0$ shall be used in all cases.

C.5.4.4 *Stress concentrations which are superimposed or in close proximity*. Treatment of multiple stress concentrations in close proximity is outside of the scope of this appendix and where this situation arises reference should be made to established reference works.

Guidance on certain types of multiple stress concentration is given in ESDU Data Sheet No. 75022.

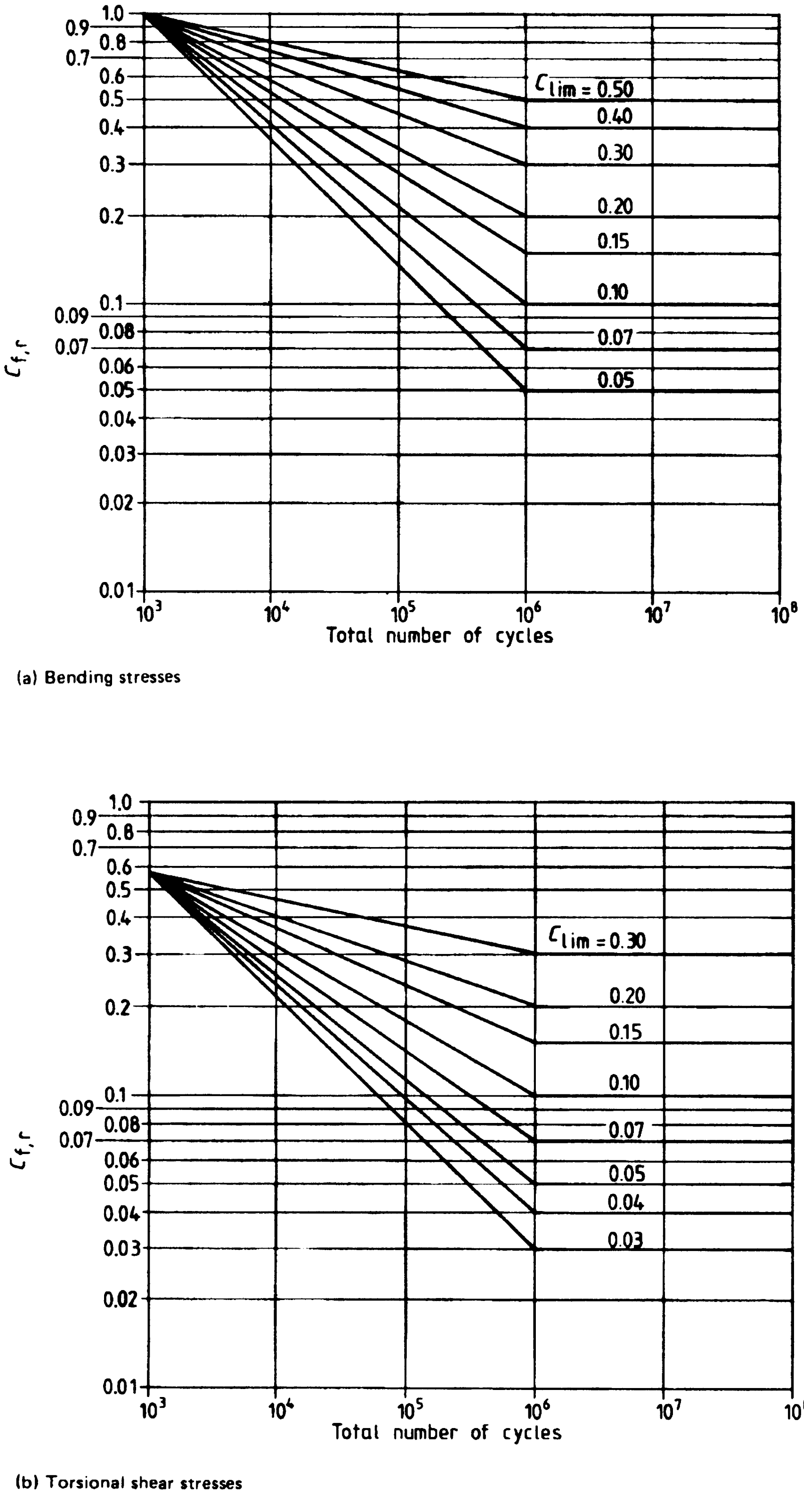

Figure 1 — Fatigue reference stress factor, $C_{f,r}$

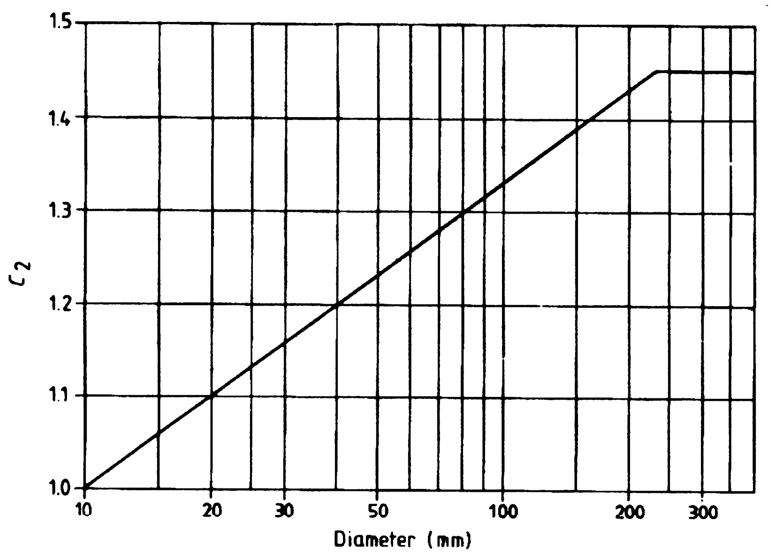

Figure 2 — Size factor, C_2

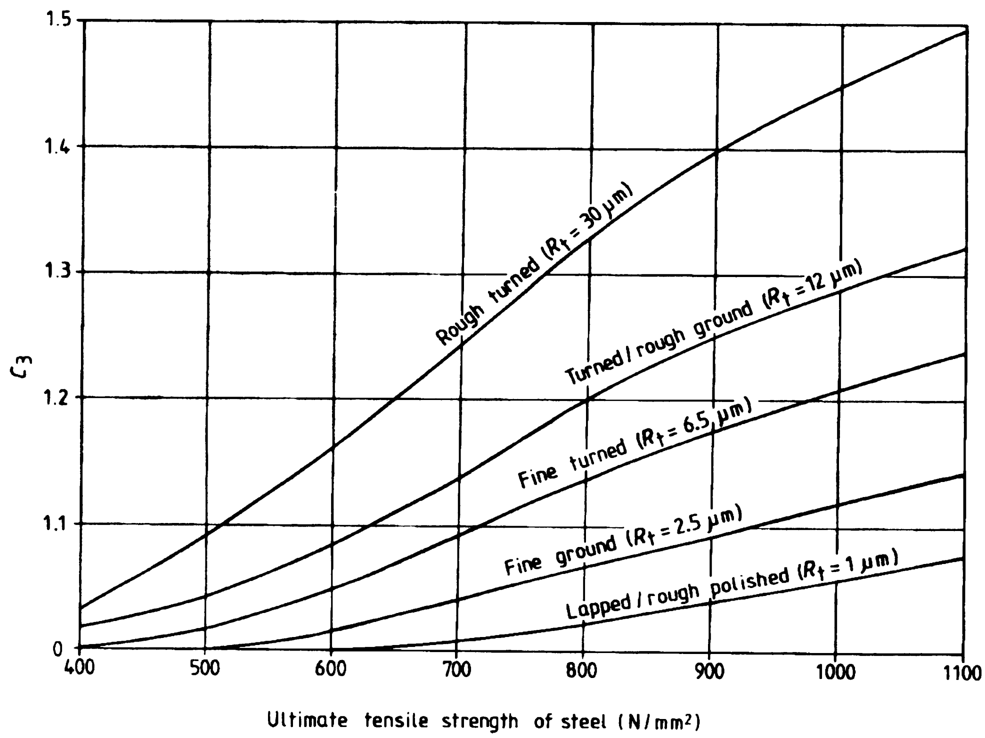

Figure 3 — Surface finish factor, C_3

Appendix D Gearing

When designing spur gearing in accordance with the requirements of BS 436, it is necessary to determine the hours per day of operation for the gearing under consideration. The hours per day to be used for gear design purposes is derived from the class of utilization for the particular mechanism. Table 12 gives the hours per day to be used for design purposes for each class of utilization when designing gearing in accordance with BS 436.

Table 12 — Hours per day for gear design

Class of utilization	Hours per day for gear design
T1	0.2
T2	0.4
T3	0.8
T4	1.6
T5	3.2
T6	6.3
T7	12.3
T8	24.0
T9	24.0

Appendix E Derating of hooks

Table 13 and Table 14 give a rational basis for derating the capacities of higher tensile steel hooks, as stated in BS 2903, when they are used with heavier lifting gear on steelworks cranes or in similar heavy duty applications and when wear can be a problem. The basis for derating suggested for these cases is the overall crane classification (see clause **3** and Table 1 of BS 2573-1:1980).

Table 13 — Derating of hooks

Table showing sizes of point hooks as listed in Table 2 of BS 2903:1970 (hooks with increased internal diameter) when derated.

Seat dia., *C*	Proof load	Capacity for M3-M4 classification crane	Capacity for M5 classification crane	Capacity for M6 classification crane
mm	t	t	t	t
25	1.0	0.5	0.40	
28	1.25	0.63	0.50	
32	1.6	0.8	0.63	0.5
36	2.0	1.0	0.80	0.63
40	2.5	1.25	1.00	0.80
45	3.2	1.6	1.25	1.00
51	4.0	2.0	1.60	1.25
57	5.0	2.5	2.00	1.60
64	6.3	3.2	2.50	2.0
72	8.0	4.0	3.20	2.5
80	10.0	5.0	4.0	3.2
90	12.5	6.3	5.0	4.0

NOTE Capacities are rounded off in line with R10 series.

Table 14 — Derating of hooks

Table showing selected sizes of point hooks, as listed in Table 1 of BS 2903:1970, and maximum lifting capacities when derated.

Seat dia., *C*	Proof load	Capacity for M3-M4 classification crane	Capacity for M5 classification crane	Capacity for M6 classification crane	Capacity for M7-M8 classification crane
mm	t	t	t	t	t
26	1.6	0.8	0.63		
37	3.2	1.6	1.25		
46	5.0	2.5	2.0		
58	8.0	4.0	3.2	2.5	
65	10.0	5.0	4.0	3.2	2.5
73	12.6	6.3	5.0	4.0	3.2
83	16.0	8.0	6.3	5.0	4.0
92	20.0	10.0	8.0	6.3	5.0
103	25.0	12.5	10.0	8.0	6.3
117	32.0	16.0	12.5	10.0	8.0
131	40.0	20.0	16.0	12.5	10.0
146	50.0	25.0	20.0	16.0	12.5
159	59.0	32.0	25.0	20.0	16.0
173	69.0	40.0	32.0	25.0	20.0
191	81.0	50.0	40.0	32.0	25.0
205	97.0	63.0	50.0	40.0	32.0

NOTE Capacities are rounded off in line with R10 series.

blank

Publications referred to

BS 436, *Spur and helical gears*.
BS 545, *Bevel gears (machine cut)*.
BS 721, *Worm gearing*.
BS 970, *Wrought steels in the form of blooms, billets, bars and forgings*.
BS 1134, *Method for the assessment of surface texture*.
BS 2573, *British Standard Rules for the design of cranes*.
BS 2573-1, *Specification for classification, stress calculations and design criteria for structures*.
BS 2903, *Higher tensile steel hooks*.
BS 3017, *Mild steel forged ramshorn hooks*.
BS 3037, *Tyres for crane rail wheels*.
BS 3579, *Heavy duty electric overhead travelling and special cranes for use in steel works*.
BS 3692, *ISO metric precision hexagon bolts, screws and nuts*.
BS 5512, *Specification for rolling bearings — Dynamic load rating and rating life*.
BS 5512-1, *Calculation methods*.
ISO 2408, *Steel wire ropes for general purposes — Characteristics*.
ISO 4301, *Lifting appliances — Classification*[4].
ISO 4308, *Cranes — Selection of wire ropes*[4].
ISO 4309, *Wire rope for lifting appliances — Code of practice for examination and discard*[4].

[4] In course of preparation.

BSI — British Standards Institution

BSI is the independent national body responsible for preparing British Standards. It presents the UK view on standards in Europe and at the international level. It is incorporated by Royal Charter.

Revisions

British Standards are updated by amendment or revision. Users of British Standards should make sure that they possess the latest amendments or editions.

It is the constant aim of BSI to improve the quality of our products and services. We would be grateful if anyone finding an inaccuracy or ambiguity while using this British Standard would inform the Secretary of the technical committee responsible, the identity of which can be found on the inside front cover. Tel: 020 8996 9000. Fax: 020 8996 7400.

BSI offers members an individual updating service called PLUS which ensures that subscribers automatically receive the latest editions of standards.

Buying standards

Orders for all BSI, international and foreign standards publications should be addressed to Customer Services. Tel: 020 8996 9001. Fax: 020 8996 7001.

In response to orders for international standards, it is BSI policy to supply the BSI implementation of those that have been published as British Standards, unless otherwise requested.

Information on standards

BSI provides a wide range of information on national, European and international standards through its Library and its Technical Help to Exporters Service. Various BSI electronic information services are also available which give details on all its products and services. Contact the Information Centre. Tel: 020 8996 7111. Fax: 020 8996 7048.

Subscribing members of BSI are kept up to date with standards developments and receive substantial discounts on the purchase price of standards. For details of these and other benefits contact Membership Administration. Tel: 020 8996 7002. Fax: 020 8996 7001.

Copyright

BSI
389 Chiswick High Road
London
W4 4AL