Specification for

Vessels for use in heating systems —

Part 2: Tubular heat exchangers and storage vessels for building and industrial services

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Committees responsible for this British Standard

The preparation of this British Standard was entrusted to Technical Committee RHE/12, Calorifiers, upon which the following bodies were represented:

British Non-Ferrous Metals Federation
Chartered Institution of Building Services Engineers
Copper Development Association
Health and Safety Executive
Hevac Association
Institution of Mechanical Engineers
Marine Safety Agency
Waterheater Manufacturers' Association
Co-opted members

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Contents

Foreword

This Part of BS 853 has been prepared by Technical Committee RHE/12. It covers the requirements of vessels with higher duty requirements than BS 853-1, but for which the requirements of BS 5500 are unnecessarily stringent. BS 853 was first published in 1939 and revised in two Parts covering carbon steel and copper in 1960. A second revision was carried out in 1981, when the two Parts were again combined, and this was updated in 1990 to take account of current practice. With the introduction of this Part of BS 853, BS 853:1990 becomes BS 853-1:1996 without any technical changes. No provision has been included for thermal performance tests.

NOTE Information concerning SI units is given in BS 5555 and BS 5775.

WARNING. The use of asbestos is subject to the Control of Asbestos at Work Regulations, 1987 (SI 2115 as amended)[1], and the Health and Safety at Work etc. Act 1974[2]. Attention is drawn to the health hazards arising from asbestos dust. Further information is available in Health and Safety Executive Guidance Note EH/10, *Environmental Hygiene, Asbestos*[3].

This British Standard calls for the use of substances and/or procedures that may be injurious to health if adequate precautions are not taken. It refers only to technical suitability and does not absolve the user from legal obligations relating to health and safety at any stage.

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 44, 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.

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1 Scope

This Part of BS 853 specifies requirements for the design, methods of construction and testing of tubular heat exchangers and storage vessels for heating, cooling and heat recovery associated with building and industrial services with limiting conditions of 2 N/mm^2 design pressure, 300 °C maximum and 0 °C minimum temperature using such media as water, steam, thermal oils, glycol mixtures and oil.

In addition to the definitive requirements, this standard also requires the items detailed in clause **4** to be documented. For compliance with this standard, both the definitive requirements and the documented items have to be satisfied.

NOTE Attention is drawn to the dangers associated with heating flammable fluids above their flash points.

Three types of heat exchangers are covered by this standard:

a) fixed tubeplate at each end;

b) U-tubes (removable tube bundle);

c) straight tubes with floating head at rear end (removable tube bundle).

2 References

2.1 Normative references

This Part of BS 853 incorporates, by dated or undated reference, provisions from other publications. These normative references are made at the appropriate places in the text and the cited publications are listed on page 42. For dated references, only the edition cited applies: any subsequent amendments to or revisions of the cited publication apply to this British Standard only when incorporated in the reference by amendment or revision. For undated references, the latest edition of the cited publication applies, together with any amendments.

2.2 Informative references

This British Standard refers to other publications that provide information or guidance. Editions of these publications current at the time of issue of this standard are listed on the inside back cover, but reference should be made to the latest editions.

3 Definitions

For the purposes of this British Standard the following definitions apply.

3.1
design pressure

the value of pressure to be employed for calculation

3.2
design temperature

the value of temperature to be employed for calculation

3.3
design stress

the maximum allowable stress for the materials of construction at the design temperature

3.4
purchaser

the organization or individual who buys the heat exchanger for its own use or as an agent for the owner

3.5
inspecting authority

the body or association which checks that the design, materials and construction are in accordance with this standard

3.6
maximum working pressure

the pressure at which the safety relief devices are set to commence operation

3.7
set pressure

the pressure at which the safety relief devices commence to operate

4 Information and requirements to be agreed and to be documented

4.1 Information to be supplied by the purchaser

The following information to be supplied by the purchaser shall be fully documented. Both the definitive requirements specified throughout the standard and the following documented items shall be satisfied before a claim of compliance with the standard can be made and verified.

a) Sufficient details from the data sheet Annex A to enable the manufacturer to offer a heat exchanger suitable for the installation and duty.

b) Details of any independent inspection requirements.

NOTE The manufacturer is seldom in a position to determine the optimum fouling resistance when calculating the heating surface. The purchaser should therefore, where possible, specify the design fouling resistances for the particular service. When such information is not available the manufacturer should specify the fouling resistances that have been allowed for in the calculations. Typical fouling resistance data are given in Annex B.

4.2 Information to be supplied by the manufacturer

The manufacturer shall provide the purchaser with full information of the heat exchanger or storage vessel offered along with leading dimensions and details of tube bundle withdrawal space where applicable.

NOTE A typical data sheet for a heat exchanger is given in Annex A.

5 Materials

5.1 Selection

Table 1 gives a range of materials which conform to this standard and shall be used for the fabrication of heat exchangers and storage vessels and gives design stress values that shall be used for the scantling design (see clause **8**). Where other materials are used the manufacturer shall show that they are comparable with those given in Table 1 and that they have been submitted to equivalent tests and approvals.

Materials for use in the fabrication of heat exchangers and storage vessels specified in this standard shall be either:

a) selected from Table 1; or

b) other materials for which the manufacturer shall show that they are comparable with those given in Table 1 and that they have been submitted to equivalent tests and approvals.

Table 1 also gives design stress values that shall be used for the scantling design.

5.2 Bolting materials

Bolting materials shall be as given in Table 2.

5.3 Filler materials

Filler materials shall be as given in Table 3.

5.4 Forging and fittings

Forging materials and fittings shall be as given in Table 4.

6 Welding procedure and welder approval tests

6.1 General

The welding shall conform to approved welding procedures in accordance with **6.2** and welder approval tests shall be in accordance with **6.3**.

The preparation of welding procedures, the approval of welders, testing and the maintenance of records shall be the responsibility of the manufacturer.

6.2 Approval of procedure

Approval testing of welding procedures for steel shall be conducted, recorded and reported in accordance with BS EN 288-3:1992.

Approval testing of welding procedures for copper shall use the methods of testing welds given in BS 4206. The copper test piece shall be subject to visual examination, penetrant testing and destructive tests. The number of test specimens shall be as given in Table 5. The welding procedures shall be certified to BS 853-1 using relevant documentation and records conforming to BS EN 288.

Each welding procedure test and the accompanying test results shall be recorded as "Welding Procedure Approval Records" in accordance with Annex A of BS EN 288-3:1992.

Each welding procedure test shall be documented to include all items in accordance with clause **4** of BS EN 288-2:1992.

An approved welding procedure test shall only require reapproval when any changes are made which conform to BS EN 288-3:1992.

6.3 Welder approval

Approval testing of welders for steel shall be conducted, recorded and reported in accordance with Annex B of BS EN 287-1:1992.

Approval testing of welders for copper shall use the methods of testing welds in accordance with BS 4206. The copper test piece shall be subject to visual examination and destructive tests, augmented by penetrant testing if necessary. The number of test specimens shall be as given in Table 6.

Welder approval shall be certified to BS 853-1 using relevant documentation and records in accordance with BS EN 287-1:1992.

A welder's approval to weld to a particular procedure shall remain valid unless there are changes in the procedure in accordance with clause **8** of BS EN 288-3:1992.

Table 1 — Materials for the construction of heat exchangers (with design stress values *f*)

Material	British Standard designation		Relevant note(s)	Design stress values (*f*) for metal (see note 1)								
				Design temperature not exceeding:								
				50 °C	100 °C	150 °C	175 °C	200 °C	225 °C	250 °C	275 °C	300 °C
				N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2
Shells and branches:												
carbon steel plate	BS 1501	151 or 161 grade 430A or B		108	108	108	108	108	104	100	93	91
carbon steel tube	BS 3601	320.ERW 360.ERW.S		80	80	80	80	78	73	69	65	62
		(A106B)	11	90	90	90	90	89	83	77	73	69
		430.ERW.S		108	108	108	108	106	99	92	87	82
	BS 3602	360.ERW.S		90	90	90	90	90	90	89	83	78
		430.ERW.S		108	108	108	108	108	102	96	91	87
stainless steel plate, sheet and strip	BS 1501	Part 3. 304S11 (A240.304L) 304S31	11	114	102	89	85	81	78	76	74	72
		(A240.304) 316S11.S13	11	122	109	95	91	87	84	82	80	78
		(A240.316L) 316S31.S33	11	120	107	95	91	88	85	82	80	79
		(A240.316)	11	128	114	102	98	94	91	89	87	85
stainless steel tube	BS 3605	304S11										
		(A312.TP304L) 304S31	11	114	102	89	85	81	78	76	74	72
		(A312 TP304) 316S11.S13	11	122	109	95	91	87	84	82	80	78
		(A312 TP316L) 316 S31.S33	11	120	107	95	91	88	85	82	80	79
		(A312 TP316)	11	128	114	102	98	94	91	89	87	85

Table 1 — Materials for the construction of heat exchangers (with design stress values *f*)

Material	British Standard designation		Relevant note(s)	Design stress values (*f*) for metal (see note 1)								
				Design temperature not exceeding:								
				50 °C	100 °C	150 °C	175 °C	200 °C	225 °C	250 °C	275 °C	300 °C
				N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2
copper plate	BS 2875	C106-0		48	46	34	26	18	—	—	—	—
copper sheet	BS 2870	C106-0		48	46	34	26	18	—	—	—	—
copper tubes	BS 2871	Part 2. C106-0		41	40	34	26	18	—	—	—	—
90/10 CuNi plate	BS 2875	CN102-0		97	92	89	63	60	58	54	49	44
90/10 CuNi sheet	BS 2870	CN102-0		97	92	89	63	60	58	54	49	44
90/10 CuNi tube	BS 2871	Part 2. CN102-0		75	73	71	63	60	58	54	49	44
70/30 CuNi plate	BS 2875	CN107-0		117	109	105	76	75	73	72	71	70
70/30 CuNi sheet	BS 2870	CN107-0		117	109	105	76	75	73	72	71	70
70/30 CuNi tube	BS 2871	Part 2. CN107-0		93	87	84	76	75	73	72	71	70
Bars and sections (for welding):												
carbon steel	BS 1502	151 or 161		108	108	108	108	108	104	100	93	91
stainless steel	BS 1502	304S11		114	102	89	85	81	78	76	74	72
		304S31		122	109	95	91	87	84	82	80	78
		316S11.S13		120	107	95	91	88	85	82	80	79
		316S31.S33		128	114	102	98	94	91	89	87	85
copper	BS 2874	C106-0		41	40	34	26	18	—	—	—	—
Tubeplates:												
carbon steel	BS 1501	151 or 161 grade 430A or B		108	108	108	108	108	104	100	93	91
stainless steel	BS 1501	Part 3. 304S11 (A240.304L)	11	114	102	89	85	81	78	76	74	72
		304S31 (A240.304)	11	122	109	95	91	87	84	82	80	78
		316S11.S13 (A240.316L)	11	120	107	95	91	88	85	82	80	79
		316S31.S33 (A240.316)	11	128	114	102	98	94	91	89	87	85

Table 1 — Materials for the construction of heat exchangers (with design stress values *f*)

Material	British Standard designation		Relevant note(s)	Design stress values (*f*) for metal (see note 1)								
				Design temperature not exceeding:								
				50 °C	100 °C	150 °C	175 °C	200 °C	225 °C	250 °C	275 °C	300 °C
				N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2
70/30 brass	BS 2875	CZ105-0		72	71	69	54	25	15	—	—	—
aluminium bronze	BS 2875	CA105-0		180	175	174	103	94	87	77	71	63
90/10 CuNi	BS 2875	CN102-0		97	92	89	63	60	58	54	49	44
70/30 CuNi	BS 2875	CN107-0		117	109	105	76	75	73	72	71	—
aluminium brass	BS 2875	CZ110-0		72	71	69	54	25	15	—	—	—
naval brass	BS 2875	CZ112-0		117	108	105	55	17	—	—	—	—
60/40 brass	BS 2875	CZ123-0		93	93	92	55	17	—	—	—	—
Tubes:												
	BS 3059	Part 1. 320	10	80	80	80	80	78	73	69	65	62
carbon steel	BS 3059	Part 2. 360		90	90	90	90	90	90	89	83	78
	BS 3606	320		80	80	80	80	78	73	69	65	62
	BS 3606	400		100	100	100	100	98	93	88	82	77
stainless steel	BS 3059	Part 2. 304S51										
		(A213.TP304)		122	109	95	91	87	84	82	80	78
		316S51.S52										
		(A213.TP316)		128	114	102	98	94	91	89	87	85
	BS 3606	304S11										
		(A213 TP304L)	11	114	102	89	85	81	78	76	74	72
		304S31	11	122	109	95	91	87	84	82	80	78
		316S11.S13										
		(A213 TP316L)	11	120	107	95	91	88	85	82	80	79
		316 S31.S33		128	114	102	98	94	91	89	87	
copper	BS 2871	Part 3. C106-0		41	40	34	26	18	—	—	—	—
aluminium bronze	BS 2871	CA102-0		85	82	80	67	44	31	16	—	—
90/10 CuNi	BS 2871	CN102-0		75	73	71	63	60	58	54	49	44
70/30 CuNi	BS 2871	CN107-0		93	87	84	76	75	73	72	71	70
aluminium brass	BS 2871	CZ110-0	6	87	83	83	54	25	15	—	—	—
70/30 arsenical brass	BS 2871	CZ105-0	6	70	69	67	57	25	15	—	—	—

Table 1 — Materials for the construction of heat exchangers (with design stress values *f*)

Material	British Standard designation		Relevant note(s)	Design stress values (f) for metal (see note 1)								
				Design temperature not exceeding:								
				50 °C	100 °C	150 °C	175 °C	200 °C	225 °C	250 °C	275 °C	300 °C
				N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2
special 70/30 arsenical brass	BS 2871	CZ126-0	6	77	72	71	54	25	15	—	—	—
Castings:												
carbon steel	BS 1504	161-430A		107	97	88	86	83	82	81	77	73
grey cast iron	BS 1452	grade 180	7, 8 and 9	16	16	16	16	16	—	—	—	—
	BS 1452	grade 220	7, 8 and 9	19	19	19	19	19	—	—	—	—
spheroidal or nodular	BS 2789	SNG 24/17		60	60	60	60	60	60	—	—	—
graphite cast iron			—	—	—	—	—	—	—	—	—	
admiralty gunmetal	BS 1400	G1	2	55	55	55	52	49	46	25	—	—
leaded gunmetal	BS 1400	LG2	3	64	61	58	47	45	43	34	22	—
leaded gunmetal	BS 1400	LG4	3	59	59	59	56	51	—	—	—	—
naval brass	BS 1400	SCB4	4 and 5	—	—	—	—	—	—	—	—	—
brass for brazable castings	BS 1400	SCB6	4	30	30	20	—	—	—	—	—	—

NOTE 1 Linear interpolation may be used to determine the appropriate design stress value when the design metal temperature lies between two temperatures.
NOTE 2 May be welded only if the lead content is less than 0.5 %.
NOTE 3 Cannot be welded but may be brazed.
NOTE 4 Not suitable for operating temperatures greater than 150 °C.
NOTE 5 Suitable for non-pressure parts only.
NOTE 6 If these materials are to be screwed they should be ordered in the "M" instead of the "O" condition
NOTE 7 Grey cast iron is not suitable for design pressures in excess of 1.03 N/mm^2 or for temperatures exceeding 220 °C, at which temperature the values of f at 200 °C should be used.
NOTE 8 Refer to Figure 3 of BS 1452:1990 for estimated thicknesses of cast iron greater than 30 mm. The design stress used should be 10 % of the tensile stress for the grade of material selected and thickness required.
NOTE 9 Iron castings that are subjected to the working pressure should have the number and grade of the relevant British Standard cast upon them so that the maximum operating pressure and temperature for which they are suitable may be assessed at any time.
NOTE 10 Where necessary tubes should be ordered to special tolerances, so that the maximum diametral clearances specified in **10.1.4** are not exceeded.
NOTE 11 Acceptable ASTM materials shown in brackets.

Table 2 — Bolting materials (see note 1)

Material	British Standard designation	Diameter	Design temperature, ° not exceeding:				
			50 °C	100 °C	150 °C	200 °C	250 °C
		mm	N/mm^2	N/mm^2	N/mm^2	N/mm^2	N/mm^2
Carbon steel	BS 3692 Grade 8.8	≤ 68	192	174	156	139	129
1 % chromium molybdenum steel	BS 4882 Grade B7	≤ 63	193	181	174	167	158
Stainless steel	BS 4882 Grade B8 (304)	All	126	106	97	89	83
	BS 4882 Grade B8M (316)	All	129	109	101	94	87

NOTE 1 Linear interpolation may be used to determine the appropriate design stress when the design temperature lies between two temperatures

NOTE 2 The grade of the bolt material used should be specified on a separate and easily seen plate attached to the heat exchanger or storage vessel.

NOTE 3 Nuts used with high tensile steel bolts or studs shall have a specific minimum ultimate tensile strength which is not more than 77 N/mm^2 lower than that of the bolts or studs.

Table 3 — Filler and brazing materials

Material	British Standard designation	Relevant note(s)
Filler rods, wires and fluxes or welding:		
for manual metal-arc welding of carbon steel	BS EN 499	
for submerged arc welding of carbon steel	BS 4165	
for TIG and MIG welding of carbon steel	BS EN 440	
for TIG and MIG welding of copper	BS EN 440	
for gas welding of copper	BS 1453 C1	
for manual metal-arc welding of stainless steel	BS 2926	
for submerged arc welding of stainless steel	BS 5465	
for TIG and MIG welding of stainless steel	BS EN 440	
Brazing filler metals:		
copper-phosphorus	BS 1845 CP1, CP2 or CP4	1
silver	BS 1845 AG1 to AG18	1
Filler alloys for attaching steel non-pressure parts (e.g. support brackets) to copper shells:		
for bronze welding	BS 1453 C2, C4, C5 or C6	
for flame brazing	BS 1845 CZ3 to CZ8	
for soft soldering (tin content exceeding 33 %)	BS EN 29453	2

NOTE 1 For brazed seams exposed to aggressive water, which could give rise to dezincification or other forms of selective attack, brazing alloys in accordance with BS 1845 CP1 or CP2 should be used. For brazed seams which are not subject to such forms of attack, brazing alloys in accordance with BS 1845 AG1 to AG7 may be used.

NOTE 2 Soft solders may be used only for the external attachment of brackets and similar fittings and may only be applied to parts not in contact with either the heated or the heating medium in the heat exchanger. The operating temperature for soft solder should not exceed 150 °C.

Table 4 — Forgings and hot pressing stock

Material	British Standard designation
Carbon steel	BS 1503 BS 1640-1 and BS 1640-3 BS 1945 BS 1740-1 BS 3799
Stainless steel	BS 1503 BS 1640-2 and BS 1640-4 BS 3799
Copper	BS 2872 C106
Naval brass	BS 2872 CZ112
Aluminium bronze	BS 2872 CA 104
Steel pipe fittings (for screwed connections)	BS 1740

Table 5 — Number of test specimens required for procedure approval for copper

Test specimen	Butt joint in plate of thickness		Butt joint in pipe of thickness		Fillet weld in plate
	less than 10 mm	10 mm and over	less than 10 mm	10 mm and over	
Macro examination	1	1	2	2	2
Transverse tensile	1	1	1	1	1
Root bend	2	0	2	—	—
Face bend	1	0	1	—	—
Side bend	0	2	—	2	—
Fillet weld fracture (for test piece with only single side weld)	0	0	—	—	3

NOTE When more than one specimen of a particular type is required the specimens shall be taken as far apart as possible with one specimen for macro-examination taken from that part of the joint considered to have been welded in the most difficult welding position or from a stop/start position.

Table 6 — Number of test specimens required for welder approval for copper

Test specimen	Butt joint in plate of thickness		For fillet weld in plate
	less than 10 mm	10 mm and over	
Macro examination	2	2	2
Root bend	1	—	—
Face bend	1	—	—
Side bend	—	2	—
Fillet weld fracture (for test piece with only single side weld)	—	—	3

NOTE When more than one specimen of a particular type is required the specimens shall be taken as far apart as possible with one specimen for macro-examination taken from that part of the joint considered to have been welded in the most difficult welding position or from a stop/start position.

(Organization's symbol or logo)	Welder approval test certificate	Test record
Manufactor's name	Welder's name and identity no.	Issue no.

Declaration

I,the undersigned, declare that welder named above has been regularly and satisfactorily employed on work covered by this certificate during the six month preceding the date of my signature.

Date	Personal signature	Position or title

Figure 1 — Welder's history sheet

6.3.1 *Reapproval of welder*

For the purpose of this standard a welder's approval shall remain valid provided that it can be shown, as signified at intervals of 6 months by a senior responsible person in the firm that employs the welder, that the welder has, subsequent to the test, been employed with reasonable continuity on work within the extent of his approval and has continued to produce satisfactory welds as verified by traceable records by the type of production work. Reapproval shall be required if any of the following apply.

a) The welder is to be employed on work outside the extent of his current approval.

b) The welder changes his employer without the transfer of his test records.

c) Six months or more have elapsed since the welder was engaged in welding on work within the extent of his approval.

NOTE Subject to the agreement of the inspecting authority, a complete reapproval test may be waived provided the first production weld by the welder is supplemented with a non-destructive test for steel and a bend test for copper.

d) There is some specific reason to question the welder's ability.

Proof of the welder's continued use of the approved procedure shall be the maintenance of a welder's history sheet such as that shown in Figure 1.

7 Design pressure and design temperature

NOTE In establishing the design pressure and the design temperature it is necessary to take into account all conditions of service, including start-up, shut-down and cleaning.

7.1 Design pressure (data sheet item 26)

7.1.1 *Shell side design pressure*

The shell side design pressure shall be not less than the set pressure of the safety valve mounted on or adjacent to the shell (see clause **11**) and in no case less than two-thirds of the hydraulic test pressure.

7.1.2 *Tube side design pressure*

The tube side design pressure shall be not less than the highest pressure which can be reached in the tubes under all service conditions and in no case less than two-thirds of the hydraulic test pressure.

Protection against excess pressure shall be given to both the tube side and shell side of the heat exchanger in accordance with clause **11**.

NOTE It is recommended that when the shell or the tube side is subject to vacuum, that side should be designed for full negative pressure of 0.1 N/mm^2 (1 bar[1)]) unless a vacuum breaker valve or similar device is provided, in which case a lower design pressure may be used.

7.2 Design temperature (data sheet item 26)

For parts in contact with one fluid only, the design temperature shall be taken as the maximum fluid temperature in contact with each part. For parts subject to two different fluid temperatures, the design temperature shall be taken as either the actual temperature under operating conditions or the higher fluid temperature in contact with the part.

NOTE In establishing the actual metal temperatures due consideration needs to be given to such factors as the relative heat transfer coefficients of the two fluids contacting the part and the relative heat transfer area of the two sides of the part in contact with the fluids.

8 Scantling design

8.1 General

The thickness and construction of the components of shell, channel and tubeplate assemblies shall conform to **8.3** to **8.8** and, where appropriate, shall include the corrosion allowance in accordance with **8.2**.

8.2 Corrosion allowance

8.2.1 ***Pressure parts***

8.2.1.1 *Unprotected carbon steel pressure parts*

All unprotected carbon steel pressure parts shall have a corrosion allowance of 1 mm unless a different allowance is specified.

8.2.1.2 *Unprotected cast iron pressure parts*

All unprotected cast iron pressure parts shall have a corrosion allowance of 1 mm.

8.2.2 ***Tubes***

If no corrosion allowance is specified by the purchaser the manufacturer shall either:

a) select a material which is suitable for the service conditions without a corrosion allowance; or

b) include a corrosion allowance suitable for the material selected and the service conditions.

8.2.3 ***Internal covers***

Internal covers shall have corrosion allowance on both sides.

8.2.4 ***Tubeplates***

Tubeplates shall have corrosion allowance on both sides.

8.2.5 ***Grooved tubeplates and external covers***

Where the plates are grooved the depth of the pass-partition groove shall be considered available for corrosion allowance.

8.2.6 ***End flanges***

Corrosion allowance shall not be applied to the mating faces of flanges.

8.2.7 ***Non-pressure parts***

Non-pressure parts such as tie rods, spacers, baffles and support plates shall have no corrosion allowance.

8.2.8 ***Floating head backing devices***

Floating head backing devices and internal bolting shall have no corrosion allowance.

8.2.9 ***Copper and stainless steel components***

Copper, copper alloys and stainless steel components shall have no corrosion allowance unless it is definitely specified.

8.3 Shell and shell ends (data sheet item 30 and 31)

8.3.1 ***Cylindrical shells***

8.3.1.1 *Calculated shell thickness*

The calculated thickness of a cylindrical shell subject to pressure on its internal surface shall be calculated from the following equations.

$$t_c = \frac{PD_i}{2fJ - P} + c \qquad (1)$$

or

$$t_c = \frac{PD_o}{2fJ + P} + c \qquad (2)$$

where

t_c is the calculated thickness (in mm);

P is the design pressure (in N/mm^2);

D_i is the internal diameter of the shell without corrosion allowance (in mm);

D_o is the external diameter of the shell without corrosion allowance (in mm);

f is the design stress value for the shell material from Table 1 (in N/mm^2);

J is the joint factor, which has the following value:

a) for seamless shells $J = 1.0$

b) for shells with butt-welded longitudinal seams:

$J = 0.7$ for carbon steel and stainless steel

$J = 0.8$ for copper

c is the corrosion allowance (in mm).

1) 1 bar = 0.1 N/mm^2 = 0.1 MPa = 10.2 m head.

8.3.1.2 *Actual thickness for shells not subject to vacuum conditions*

The actual thickness of a cylindrical shell shall be not less than the greater of the values given in **8.3.1.2** a) and **8.3.1.2** b) as follows:

a) the value calculated in accordance with **8.3.1.1**,

b) the following value imposed by design limitations:

$0.006D_i$ or 3 mm whichever is the greater.

8.3.1.3 *Actual thickness for shells subject to vacuum conditions*

The actual thickness of a cylindrical shell shall be not less than the greater of the values given in **8.3.1.3** a) and **8.3.1.3** b) as follows:

a) the value calculated in accordance with **8.3.1.1**,

b) the following value imposed by design limitations:

1) carbon steel and stainless steel	$0.008D_o$ or 3 mm whichever is the greater
2) copper	$0.01D_o$ or 3 mm whichever is the greater

8.3.2 *Endplates*

8.3.2.1 *Domed ends*

The form of a domed end shall conform to the requirements of **7.2.1.1**, **7.2.1.2** and **7.2.1.3** of BS 853-1:1996.

8.3.2.2 *Calculated thickness of domed end subject to pressure on the concave side*

The thickness t_c(in mm) of the domed end shall be calculated as shown in **7.2.1.4** of BS 853-1:1996.

8.3.2.3 *Actual thickness of domed end material*

In no case shall the actual thickness of material used for the domed end prior to forming be less than t_c for the type of end concerned (see **8.3.2.2**) nor shall it be less than the thickness of material for the shell as defined in accordance with either **8.3.1.2** or **8.3.1.3**.

In no case shall the actual thickness at any point after forming be less than:

a) $0.9t_c$ for steel; or

b) $0.7t_c$ for copper.

8.3.3 *Overlapping plates*

Where there is an overlap of plates at the circumferential seam between the shell and the shell end, the minimum overlap shall conform to one of the following:

a) a double full fillet lap joint conforming to Figure 2; or

b) a single full fillet lap joint with plug welds to take at least 20 % of the total load. This is acceptable up to a maximum thickness of 14 mm as shown in Figure 3.

8.3.4 *Flat endplates without openings*

The calculated thickness for a flat endplate t_c (in mm) shall be determined by the use of the following equations.

a) For bolted-on flat endplates where the jointing surfaces and joint ring extend to the outer periphery of the endplate the following equation shall be used:

$$t_c = \sqrt{\frac{(pD_i^2)}{4F}} + c \qquad (3)$$

where

p	is the shell design pressure (in N/mm^2);
F	is the design stress value (see Table 1) (in N/mm^2);
D_i	is the diameter of the bolt pitch circle (in mm);
c	is the corrosion allowance.

b) For blind flanges with the gasket entirely within the bolt circle, reference shall be made to the equation in Figure 4.

c) For flat endplates that are flanged at the periphery for butt welding to the shell or header, the following equation shall be used:

$$t_c = \sqrt{\frac{(pD_i^2)}{4F}} + c \qquad (4)$$

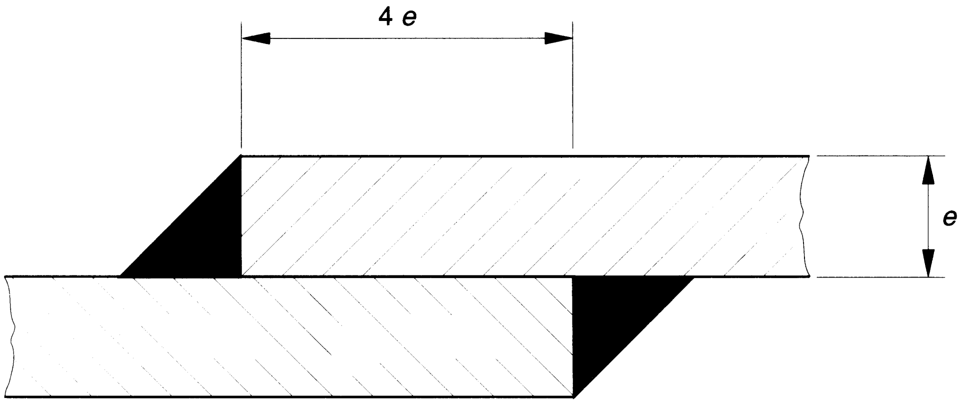
where p, F and c have the meanings given in **8.3.4** a) and D_i is the inside diameter of the shell or header (in mm).

d) For flat endplates that are inserted into, and adequately welded to, the shell or header in accordance with Figure 5, the following equation shall be used:

$$t_c = \sqrt{\frac{(pD_i^2)}{3F}} + c \qquad (5)$$

where p, F and c have the meanings given in **8.3.4** a) and D_i is the inside diameter of the shell or header (in mm).

If, in the case of welded ends/covers, the nominal design stress of the cylinder and end/cover are different, the lower value shall be used.



e is the thickness of thinner plate joined

Figure 2 — Double full fillet lap joint

Where an opening of diameter greater than *D*/2 is present, the flat head shall be designed in accordance with the basic principles given in **8.4.3**.

Flat heads that have an opening of diameter *D*/2 or less shall be provided with a total area of reinforcement in accordance with Figure 7 of BS 853-1:1996.

8.4 Main flanges and bolting materials

8.4.1 *Bolting materials*

8.4.1.1 *Selection*

Bolting materials shall be selected from Table 2.

8.4.1.2 *Corrosion resistant properties*

Internal bolts for floating heads shall have similar corrosion resistant properties to the materials used for the shell interior.

8.4.1.3 *Diameter*

Bolts shall be not less than 12 mm in diameter.

8.4.1.4 *Number and size*

The number and size of bolts for full face flanges shall be in accordance with **7.6** of BS 853-1:1996 and shall be calculated in accordance with that standard. For narrow faced flanges the number and size of bolts shall be derived in accordance with **8.4.3**.

8.4.2 *Full face flanges*

8.4.2.1 *Thickness of main flanges*

The thickness of main flanges for shells, shell covers and channels with the exception of cast iron, shall be calculated in accordance with the requirements of **7.6.2** and **7.6.5** of BS 853-1:1996.

8.4.2.2 *Thickness of flanges for cast iron bonnets*

Flanges for cast iron bonnets shall have full face joints and the thicknesses shall be calculated in accordance with **9.9.2.3** of BS 853-1:1996.

8.4.2.3 *Flange backing rings*

Full face flanges for copper shells, shell covers and channels that are flanged outwards and are supported by a steel backing ring shall conform to **7.6.4** of BS 853-1:1996.

8.4.3 *Narrow faced flanges*

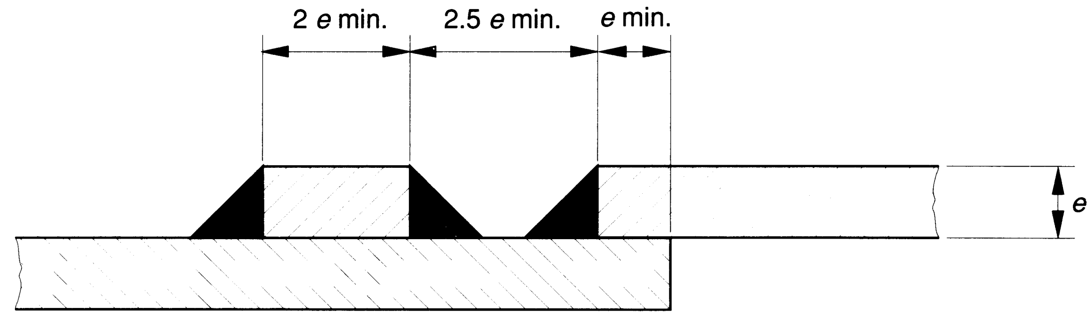
8.4.3.1 *General*

The design methods specified in **8.4.3** shall be applied to circular flanges under internal pressure.

8.4.3.2 *Notation*

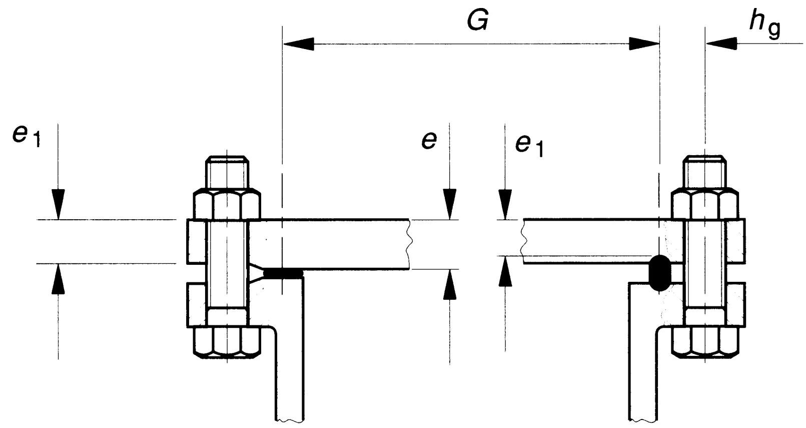
For the purposes of **8.4.3** and **8.4.4** the following symbols shall apply. Consistent units shall be used for equations.

NOTE All dimensions exclude corrosion allowances.



e is the thickness of thinner plate joined

Figure 3 — Single full fillet lap joint with plug weld



$$e = \sqrt{\frac{0.3G^2p}{S_{FO}} + \frac{1.909W_{m1}h_g}{GS_{FO}}}$$

and

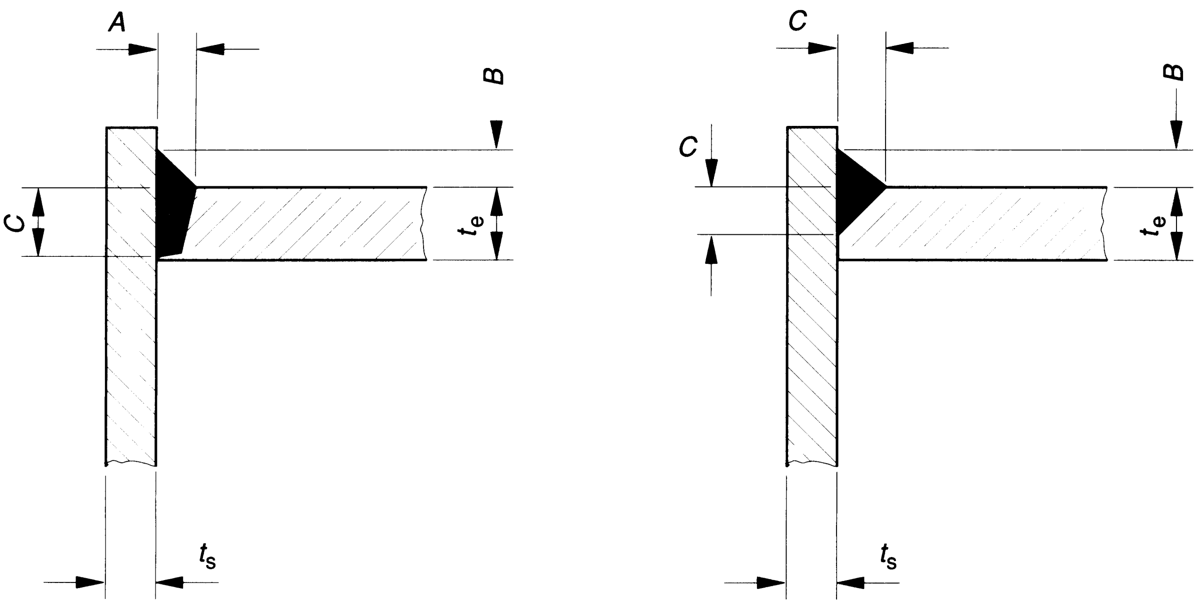
$$e_1 = \sqrt{\frac{1.909Wh_g}{GS_{FA}}}$$

or

$$e_1 = \sqrt{\frac{1.909W_{m1}h_g}{GS_{FO}}} \text{ whichever is the greater}$$

where G, W, W_{m1}, h_g, S_{FA} and S_{FO} are in accordance with **8.4.3.2**, and e_1 is the minimum thickness at, and beyond, the gasket.

Figure 4 — Typical blind flange with gasket within the bolt circle reference



where

$A = t_s$

$B = t_s$

$C = 2t_s$ or $(t_e - 1.5)$, whichever is less

Figure 5 — Welds for unflanged endplates

A is the outside diameter of the flange or, where slotted holes extend to outside of flange, the diameter to bottom of slots (in mm);

A_b is the actual total cross-sectional area of bolts at roof of thread or section of least diameter under stress (in mm^2);

A_m is the total required cross-sectional area of bolts, taken as the greater of A_{m1} and A_{m2} (in mm^2);

A_{m1} is the total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for operating conditions, $= W_{m1}/S_b$(in mm^2);

A_{m2} is the total cross-sectional area of bolts at root of thread or section of least diameter under stress, required for gasket seating, $= W_{m2}/S_a$(in mm^2);

B is the inside diameter of flange (in mm);

b_o is the basic gasket seating width (in mm) (see Table 7);

b is the effective gasket or joint-contact-surface seating width (in mm):

$b = b_o$ when $b_o \leq 6.3$ mm

$^a b = 2.52\sqrt{b_o}$ when $b_o \leq 6.3$ mm;

C is the bolt circle diameter (in mm);

C_F is the bolt pitch correction factor (in mm);

$$C_F = \sqrt{\frac{h}{2g + 6t/(m + 0.5)}}$$

NOTE Minimum value to be taken as 1.

g is the bolt diameter (in mm);

G is the diameter at location of gasket load reaction, for welded flanges (in mm) and is defined as follows (see Table 7 and Figure 6):

when $b_o \leq 6.3$ mm, G = mean diameter of gasket contact face,

when $b_o \geq 6.3$ mm, G = outside diameter of gasket contact face less $2b$;

G_1 is the diameter measured to line of mid-point of contact between flange and lap (in mm);

h is the bolt spacing (in mm);

H is the total hydrostatic end force, $= 0.785G^2P$ (in N);

H_D is the hydrostatic end force on area inside of flange (in N) (i.e. force applied via connection to flange) $= 0.785B^2p$;

H_T is the hydrostatic end force due to pressure on flange face, $= H - H_D$ (in N);

H_P is the compression load on gasket to ensure tight joint (in N), $= 2b \times 3.14\ Gmp$;

H_g is the gasket load (in N), $= H_p$ for operating condition, $= W$ for bolting-up condition;

h_D is the radial distance from bolt circle to circle on which H_D acts (in mm)

$= \dfrac{C - B}{2}$;

h_g is the radial distance from gasket load reaction to bolt circle (in mm)

$= \dfrac{C - G}{2}$;

h_T is the radial distance from bolt circle to circle on which H_T acts (in mm)

$= \dfrac{h_D + h_g}{2}$;

K is the ratio of outside diameter of flange to inside diameter of flange (in mm)

$= A/B$;

$M = \dfrac{M_o C_F}{B}$;

M_{atm} is the total moment acting upon flange for gasket seating conditions (in N·mm);

M_{op} is the total moment acting on flange for operating conditions (in N·mm);

M_o is the greater of M_{op} or $\dfrac{S_{FO}}{S_{FA}} M_{atm}$ (in N·mm);

m is the gasket factor, given in Table 8;

N is the dimension used to determine basic gasket seating width b_o (in mm), based upon possible contact width of gasket (see Table 7);

P is the design pressure (in N/mm^2);

P_e is the external design pressure (in N/mm^2);

S_a is the bolt nominal design stress at atmospheric temperature (in N/mm^2) (see Table 1);

S_b is the bolt nominal design stress at design temperature (in N/mm^2) (see Table 1);

S_{FA} is the nominal design stress (in N/mm^2) for flange material at atmospheric temperature from stress table (see Table 1);

S_{FO} is the nominal design stress for flange material at design temperature (in N/mm^2) (operating conditions) from stress table (see Table 1);

S_T is the calculated tangential stress in flange (in N/mm^2);

t is the flange thickness (in mm);

T is the gasket thickness (in mm);

W_{m1} is the minimum required bolt load for operating conditions (in N)

$= H_p + H$;

W_{m2} is the minimum required bolt load for gasket seating (in N);

W is the flange design bolt load (in N) derived from W_{m1} and W_{m2}

$W = 0.5(A_m + A_b)\ S_a$;

w is the dimension used to determine basic gasket seating width b_o (in mm), based upon contact with width between flange facing and gasket (see Table 7);

Y is a factor involving K (see Figure 7);

y is the gasket of joint-contact-surface unit seating load (in N).

[a] This relationship is valid only with dimensions expressed in millimetres.

8.4.3.3 *Bolt loads and areas*

Bolt loads and areas shall be calculated for both the operating and bolt-up conditions.

a) Operating conditions. The minimum bolt load, W_{m1}, shall be

$$W_{m1} = H + H_p \qquad (6)$$

where

$H = 0.785G^2p$ for gasket flanges and

$H_p = 6.28bGmp$

b) Bolting-up conditions. The minimum bolt load, W_{m2}, shall be

$$W_{m2} = 3.14bGy \qquad (7)$$

The minimum bolt area, A_m, shall be determined for W_{m1} or W_{m2} using the nominal bolt stress at the temperature appropriate to the two conditions, i.e. A_m is the greater of A_{m1} or A_{m2}

where $A_{m1} = W_{m1}/S_b$,

and $A_{m2} = W_{m2}/S_a$.

The actual bolt area provided, A_b, shall be not less than A_m.

NOTE See Table 8 and Figure 6 for suggested values of m, b and y.

8.4.3.4 *Flange moments*

Flange moments shall be calculated for both the operating condition and the bolting-up condition.

a) Operating conditions. The total flange moment shall be:

$$M_{op} = H_D h_D + H_T h_T + H_g h_g \tag{8}$$

b) Bolting-up condition. The total flange moment shall be:

$$M_{atm} = W h_g \tag{9}$$

8.4.3.5 *Flange stresses and stress limits*

8.4.3.5.1 *Flange stresses*

Flange stresses shall be calculated for the more severe of the operating or the bolting-up conditions so that:

$$M_o = M_{op} \tag{10}$$

or

$$M_o \text{ or } \frac{S_{FO}}{S_{FA}} M_{atm} \tag{11}$$

whichever is the larger.

For flanges having a rectangular cross section, the calculated tangential stress in flange shall be:

$$S_T = \frac{YM}{t^2} \tag{12}$$

8.4.3.5.2 *Stress limits*

The flange design stress as calculated in accordance with **8.4.3.5.1** shall not exceed the following value:

$S_T = S_{FO}$

or, when considering the bolting-up at atmospheric temperature case:

$S_T = S_{FA}$.

In the case of flanges with laps where the gasket is so located that the lap is subjected to shear, the shearing stress shall not exceed $0.8S_{FO}$ or $0.8S_{FA}$ for gasket seating and operating conditions respectively for the material of the lap, in accordance with **8.4.3.2**.

In the case of flanges attached by fillet welds, the shearing stress carried by the welds shall not exceed $0.8S_{FO}$ and $0.8S_{FA}$ for gasket seating and operating conditions respectively. The shearing stress shall be calculated on the basis of H_p or W_{m2} in accordance with **8.4.3.2**, whichever is greater.

Similar cases where flange parts are subjected to shearing stress shall be governed by the same requirements.

8.4.3.6 *Narrow flanges subject to external pressure*

The design of flanges for external pressure only, shall be calculated from the equations given in **8.4.3.2** except that for operating conditions:

$$M_{op} = H_D(h_D - h_g) + H_T(h_T - h_g) \tag{13}$$

(for gasket seating $M_{atm} = Wh_g$)

where

$$W = \frac{A_{m2} + A_b}{2} S_a$$

$H_D = 0.785B^2 P_e$

$H_T = H - H_D$

$H = 0.785G^2 P$

NOTE The combined force of external pressure and bolt loading may plastically deform certain gaskets and result in loss of gasket contact pressure when the connection is depressurized. To maintain a tight joint when the unit is repressurized, consideration should be given to gasket and facing details, so that excessive deformation of the gasket will not occur. Joints subject to pressure reversals, such as in heat exchanger floating heads, are in this type of service.

Table 7 — Effective gasket width

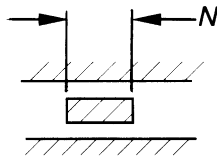
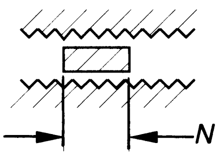
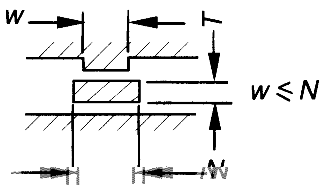
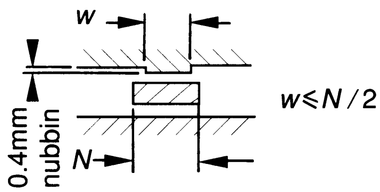
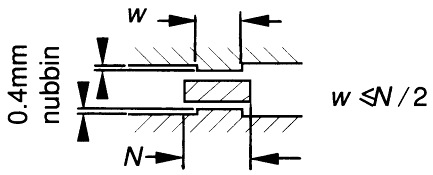
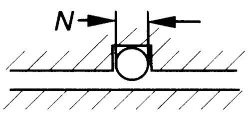
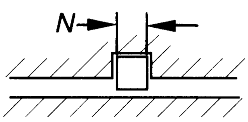
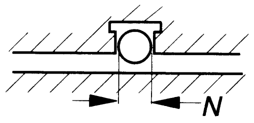
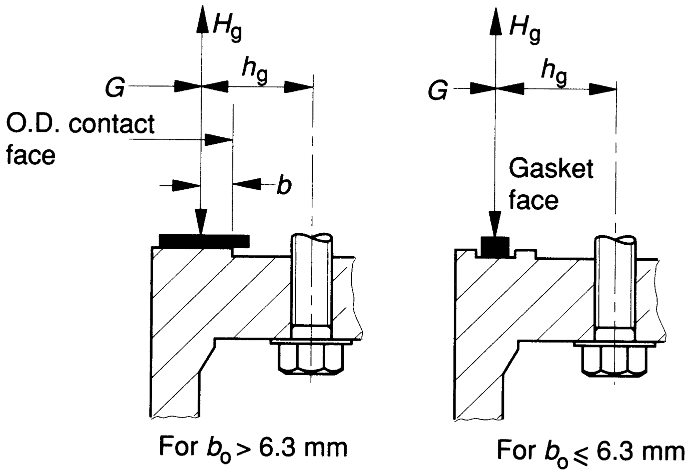
Facing sketch (exaggerated)		Basic gasket seating width b_o	
		Column I	Column II
		mm	mm
1 (a)		$\frac{N}{2}$	$\frac{N}{2}$
1 (b)			
1 (c)		$\frac{w+T}{2}; \left\{\frac{w+N}{4} \text{ max.}\right\}$	$\frac{w+T}{2}; \left\{\frac{w+N}{4} \text{ max.}\right\}$
2		$\frac{w+N}{4}$	$\frac{w+N}{8}$
3		$\frac{w}{2}; \left\{\frac{N}{4} \text{ min.}\right\}$	$\frac{w+N}{4}; \left\{\frac{3N}{8} \text{ min.}\right\}$
4		—	$\frac{N}{2}$
5	BS 4518, "O"-ring groove depth modified to give $\frac{1}{2}$ BS "O"-ring nip Side of square = cross section diameter of corresponding BS "O"-ring 	—	$\frac{N}{2}$

Table 7 — Effective gasket width

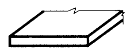
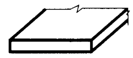
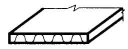
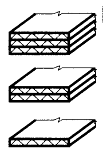
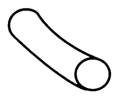
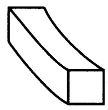
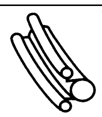
Facing sketch (exaggerated)		Basic gasket seating width b_o	
		Column I	Column II
		mm	mm
6	Special grove dimensions and protrusion required  	—	$\frac{N}{2}$



NOTE. The gasket seating width factors b_o and b shown apply only to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

Figure 6 — Location of gasket load reaction

Table 8 — Gasket materials and contact facings: gasket factor (*s*) for operating conditions and minimum design seating stress (*y*)

Dimension N (min.)	Gasket material		Gasket factor m	Min. design seating stress y	Sketches	Reference to Table 7 Use sketch	 Use column
mm				N/mm²			
	Rubber without fabric or a high percentage of asbestos fibre:						
	below 75° BS and IRH 75° NBS		0.50	0			
	and IRH of higher		1.00	1.4			
	Asbestos with a suitable binder for the operating conditions:						
	3.2 mm thick		2.0	11.0			
	1.6 mm thick		2.75	25.5			
	0.8 mm thick		3.50	44.8			
	Rubber with cotton fabric insertion		1.25	2.8		1(a),(b) (c), 2, 3	
10	Rubber with asbestos fabric insertion, with or without wire reinforcement						II
	3-ply		2.25	15.2			
	2-ply		2.50	20.0			
	1-ply		2.75	25.5			
	Corrugated metal	Soft aluminium	2.75	25.5		1(a),(b),(c)	
		Soft copper or brass	3.00	31.0			
		Iron or soft steel	3.25	37.9			
		Monel or 4 % to 6 % chromium steel	3.50	44.8			
		Stainless steels	3.75	52.4			II
	Rubber “O”-rings:					4 only	II
	below 70° BS		3°	0.7			
	between 75° and 85° BS and IRH		6°	1.4			
	Rubber square section rings:					5 only	II
	below 75° BS and IRH		4°	1.0			
	between 75° and 85° BS and IRH		9°	2.8			
	Rubber T-section rings:					6 only	II
	below 75° BS and IRH		4°	1.0			
	between 75° and 85° BS and IRH		9°	2.8			

NOTE This table gives a list of many commonly used gasket materials and contact facings with suggested design values of *a* and *y* that have generally proved satisfactory in actual service when using effective gasket seating width b given in Table 7. The design values and other details given in this table are suggested only and are not mandatory.

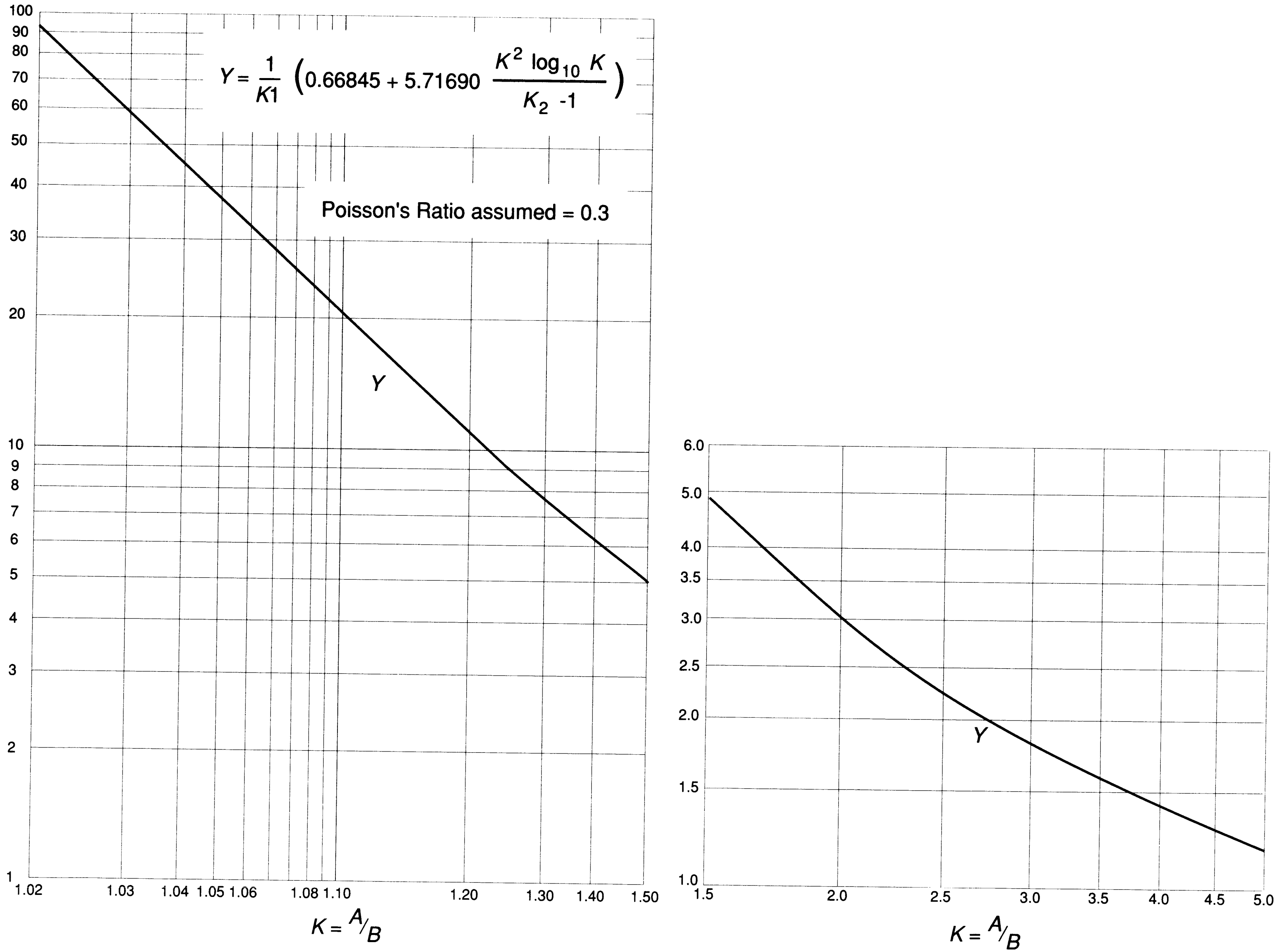


Figure 7 — Values of Y

8.5 Tubeplates (data sheet item 32)

8.5.1 ***Notation***

For the purposes of **8.5.2** to **8.5.4** the following symbols shall apply.

NOTE All dimensions exclude corrosion allowances except where otherwise indicated.

d outside diameter of tubes (in mm);

D outside diameter of shell (in mm);

D_j effective pressurized diameter of expansion joint bellows (in mm);

E elastic modulus of tubeplate material (in N/mm^2);

E_s elastic modulus of shell material (in N/mm^2);

E_t elastic modulus of tube material (in N/mm^2);

f design stress value for tubeplate material at design temperature (in N/mm^2);

f_A design stress value for tubeplate material at atmospheric temperature (in N/mm^2);

F 1.25 for U-tube exchangers, otherwise 1.0;

F_q tubeplate factor in accordance with **8.5.5.5.1**;

G diameter over which the pressure under consideration is acting (in mm):

a) when pressure is acting on the integral side of a tubeplate G is the inside diameter of the integral pressure part;

b) when pressure is acting on the gasketed side of a tubeplate G is the diameter at the location of the gasket load reaction in accordance with **8.4.3.2** for narrow faced flanges and for full faced flanges the diameter to the inside edge of the bolt holes less 5 mm.

K mean strain ratio, tube bundle/shell in accordance with **8.5.5.5**;

L tube length between inner faces of tubeplates (in mm);

N number of tube holes in tubeplate;

P effective design pressure for shell or tube side (in N/mm^2) whichever is greater, in accordance with **8.5.5.2** or **8.5.5.3**;

P_s shell side design pressure (in N/mm^2). For vacuum design P_s is negative;

P_t tube side design pressure (in N/mm^2); For vacuum design P_t is negative;

P'_s equivalent shell side pressure in accordance with **8.5.5.2**;

P'_t equivalent tube side pressure (in N/mm^2) given by equations in accordance with **8.5.5.3**;

P_d equivalent differential expansion pressure (in N/mm^2) in accordance with **8.5.5.5**;

P_{bop} equivalent bolting pressure (in N/mm^2) in accordance with **8.5.5.4**;

P_{batm} equivalent bolting pressure (in N/mm^2) in accordance with **8.5.5.4**;

t_s shell thickness (in mm);

t_t tube thickness (in mm);

T tubeplate thickness exclusive of corrosion allowance and partition grooves (in mm);

t_1 tube side fluid inlet temperature (°C);

t_2 tube side fluid outlet temperature (°C);

T_1 shell side fluid inlet temperature (°C);

T_2 shell side fluid outlet temperature (°C);

a_s thermal expansion coefficient of shell material (m/K);

a_t thermal expansion coefficient of tube material (m/K);

θ_s mean shell metal temperature less 10 °C;

θ_t mean tube wall metal temperature less 10 °C.

8.5.2 ***Minimum thickness***

The minimum thickness of flat heat exchanger tubeplates shall be calculated in accordance with **8.5.1** and **8.5.3** to **8.5.5**.

NOTE The analysis used to obtain the equations is based on the following assumptions:

a) the tubes are of uniform size;

b) where the exchanger has a pair of tubeplates, they are both of the same thickness;

c) the tubeplate is of constant thickness across the specified diameter;

d) the tubed area is uniformly perforated and nominally circular (untubed partition lanes in multipass units are accepted);

e) any untubed annular ring is sufficiently narrow to be treated as a ring whose cross section rotates without appreciable distortion (i.e. $G \leq$ outer tube limit circle diameter + $6T$);

f) the tubeplate thickness (less corrosion allowance) is not less than:

1) 0.75 × the tube outside diameter for tubes with a 25 mm outside diameter and less;

2) 22 mm for tubes with a 30 mm outside diameter and less;

3) 25 mm for tubes with a 40 mm outside diameter and less;

4) 30 mm for tubes with a 50 mm outside diameter and less.

When tubes are expanded into the tubeplate and not welded, the total thickness of the tubeplate minus the corrosion allowance in the area of expansion shall not be less than the tube's outside diameter. The minimum thickness, including corrosion allowance, shall in no case be less than 19 mm.

8.5.3 *Tubeplate thickness*

8.5.3.1 *Effective thickness*

The effective tubeplate thickness shall be the thickness measured at the bottom of the pass partition groove minus shell side corrosion allowance and corrosion allowance on the tube side in excess of the groove depth.

8.5.3.2 *Conditions for calculation*

The required effective tubeplate thickness for any type of heat exchanger shall be calculated in accordance with **8.5.3.3** and **8.5.3.4**, for both tube side and shell side conditions, using whichever is the greater.

8.5.3.3 *Calculation of thickness*

The thickness T of the tubeplate shall be calculated from the equation:

$$T = \frac{FG}{3}\sqrt{\frac{P}{\mu f}} \qquad (14)$$

where μ, the ligament efficiency, is given by either:

$$\mu = 1 - \left\{\frac{0.785}{(p/d)^2}\right\} \text{ for square tube pitch}$$

or

$$\mu = 1 - \left\{\frac{0.907}{(p/d)^2}\right\} \text{ for triangular tube pitch.}$$

where

p is the pitch of tubes.

8.5.3.4 *Minimum thickness*

When fixed tubeplates are extended for bolting to heads with ring type gaskets the minimum thickness, T_e, of the extended portion shall be the greater of the following:

$$T_e = \sqrt{\frac{1.909\ Wh_g}{Gf_A}} \qquad (15)$$

or

$$T_e = \sqrt{\frac{1.909\ W_{m1}h_g}{Gf}} \qquad (16)$$

NOTE The symbols h_g, W and W_{m1} are as defined in **8.4.3** for narrow faced flanges.

8.5.4 *Tubeplates of exchangers with floating heads and U-tubes*

8.5.4.1 *Floating heads*

Floating heads shall be completely immersed in the shell side fluid.

8.5.4.2 *Tubeplate thickness*

For heat exchangers with floating heads both tubeplates shall have the same thickness.

8.5.4.3 *Effective design pressure*

The effective design pressure, P, shall be determined as either:

a) $P = P_s$ (17)

where P_s is shell side design pressure; or

b) $P = P_t$ (18)

where P_t is tube side design pressure.

Values shall be corrected for vacuum when present on the opposite side of the shell or tube.

8.5.5 *Fixed tubeplate heat exchangers*

8.5.5.1 *General*

Fixed tubesheet heat exchangers shall be considered as being those having tubesheets fixed to both ends of shell, with or without a shell expansion joint.

8.5.5.2 *Shell side design pressure*

The effective shell side design pressure P shall be calculated using the following equations:

$$P = 0.5\ (P'_{\mathrm{s}} - P_{\mathrm{d}}) \tag{19}$$

or

$$P = P'_{\mathrm{s}} \tag{20}$$

or

$$P = P_{\mathrm{batm}} \tag{21}$$

or

$$P = 0.5\ (P'_{\mathrm{s}} - P_{\mathrm{d}} - P_{\mathrm{batm}}) \tag{22}$$

or

$$P = 0.5\ (P_{\mathrm{batm}} + P_{\mathrm{d}}) \tag{23}$$

or

$$P = (P'_{\mathrm{s}} - P_{\mathrm{batm}}) \tag{24}$$

whichever gives the greatest absolute value.

For shells with expansion joints equivalent shell side pressure (P'_{s}) shall be calculated as follows:

$$P'_{\mathrm{s}} = -\frac{P_{\mathrm{s}}}{2}\left(\frac{D_{\mathrm{j}}}{G} - 1\right) \tag{25}$$

For shells without expansion joints equivalent shell side pressure (P'_{s}) shall be calculated as follows:

$$P'_{\mathrm{s}} = 0.4P_{\mathrm{s}}\left\{\frac{1.5 + K(1.5 + F_{\mathrm{s}})}{1 + KF_{\mathrm{q}}}\right\} \tag{26}$$

where

$$F_{\mathrm{s}} = 1 - N\left(\frac{d}{G}\right)^2$$

8.5.5.3 *Tube side design pressure*

The effective tube side design pressure P shall be calculated from the following equations, whichever gives the greatest absolute value:

$$P = 0.5\ (P'_{\mathrm{t}} - + P_{\mathrm{bop}} + P_{\mathrm{d}}) \tag{27}$$

or

$$P = P'_{\mathrm{t}} + P_{\mathrm{bop}} \tag{28}$$

when P'_{s} is positive

and

$$P = 0.5 \times (P'_{\mathrm{t}} - P'_{\mathrm{t}} + P_{\mathrm{bop}} + P_{\mathrm{d}}) \tag{29}$$

or

$$P = P'_{\mathrm{t}} - P'_{\mathrm{s}} + P_{\mathrm{bop}} \tag{30}$$

when P'_{s} is negative

For shells with expansion joints equivalent tube side design pressure (P'_{t}) shall be calculated as follows:

$$P'_{\mathrm{t}} = P_{\mathrm{t}} \tag{31}$$

For shells without expansion joints equivalent tube side design pressure (P'_{t}) shall be calculated as follows:

$$P'_{\mathrm{t}} = P_{\mathrm{t}}\left\{\frac{1 + 0.4K\ (1.5 + F_{\mathrm{t}})}{1 + KF_{\mathrm{q}}}\right\} \tag{32}$$

where

$$F_{\mathrm{t}} = 1 - N\left\{\frac{(d - 2t_{\mathrm{t}})}{G}\right\}^2$$

8.5.5.4 *Equivalent bolting pressure*

When fixed tubeplates are extended for bolting to heads with ring type gaskets, the equivalent tube side and shell side pressures shall be calculated using the following equations:

$$P_{\mathrm{bop}}\ \frac{6.2M_{\mathrm{op}}}{F^2G^3} \tag{33}$$

and

$$P_{\mathrm{batm}} = \frac{6.2M_{\mathrm{atm}}}{F^2G^3} \tag{34}$$

Where full faced gaskets are fitted, the value of P_{bop} shall be such that

$P_{\mathrm{bop}} = P_{\mathrm{batm}} = 0.$

NOTE The values of M_{op} and M_{atm} are in accordance with those given in **8.4.3** for narrow faced flanges.

8.5.5.5 *Equivalent differential expansion pressure*

8.5.5.5.1 *Calculation*

On exchangers having tubeplates fixed at both ends of the shell the equivalent pressure due to differential thermal expansion shall be calculated as follows.

For shells with expansion joints equivalent differential expansion pressure (P_{d}) shall be calculated as follows:

$P_{\mathrm{d}} = 0$

except as limited in accordance with **8.5.6**.

For shells without expansion joints equivalent differential expansion pressure (P_{d}) shall be calculated as follows:

$$P_{\mathrm{d}} = \frac{4E_{\mathrm{s}}t_{\mathrm{s}}(a_{\mathrm{s}}\theta_{\mathrm{s}} - a_{\mathrm{t}}\theta_{\mathrm{t}})}{(D - 3t_{\mathrm{s}})(1 + KF_{\mathrm{q}})} \tag{35}$$

where

$$K = \frac{E_s t_s (D - t_s)}{E_t t_t N (d - t_t)}$$

and

$$F_q = 0.25 + (F - 0.6) \left\{ \frac{300 t_s E_s}{KLE} \left(\frac{G}{T} \right)^3 \right\}^{\frac{1}{4}}$$

NOTE These symbols are as defined in **8.5.5.5.2** and **8.5.1**.

When calculating the tubeplate factor F_q, the tubeplate thickness T shall be assumed. The value of T used in calculating F_q shall match the final calculated value of T within a tolerance of ± 1.5 %.

The calculated value of F_q or 1.0 shall be used, whichever is greater.

Unless sufficient data are available for the true metal temperature to be established using the relevant heat transfer coefficients, θ_s and θ_t shall be calculated (in °C) as follows:

$$\theta_s = 0.5 (T_1 + T_2) - 10 \quad (36)$$

and

$$\theta_t = \left\{ \frac{0.5 (T_1 + T_2) + 0.5 (t_1 + t_2)}{2} \right\} - 10 \quad (37)$$

8.5.5.5.2 *Special cases*

Special consideration shall be given to tubeplates with abnormal conditions of support or loading.

Example 1

Fixed tubeplates in exchangers with expansion joints which require considerable axial loads to produce required movements.

Example 2

Tubeplates with portions not adequately stayed by tubes, e.g. exchangers with large unpierced annular gap or area between the tube bundle and the shell.

8.5.6 *Tube loads*

8.5.6.1 *Maximum effective tube to tubeplate joint load*

On exchangers with unconfined heads having tubeplates fixed at both ends and without a shell expansion joint, the maximum effective tube to tubeplate joint load (W_j) shall be calculated as follows:

$$W_j = \frac{\pi F_q P_t^* G^2}{4N} \quad (38)$$

where

$$P_t^* = p_s$$

or

$$P_t^* = P_2 - P_3$$

or

$$P_t^* = -P_3$$

or

$$P_t^* = ZP_d$$

or

$$P_t^* = Z (P_2 + P_d)$$

or

$$P_t^* = Z (P_d - P_3)$$

or

$$P_t^* = Z (P_2 - P_3 + P_d)$$

whichever is the greatest absolute value and

$Z = 1.0$ if the algebraic sign is negative

$Z = 0.5$ if the algebraic sign is positive

and

$$P_2 = P'_t - \left(\frac{F_t P_t}{F_q} \right)$$

$$P_3 = P'_s - \left(\frac{F_s P_s}{F_q} \right)$$

8.5.6.2 *Tensile tube stress*

The tensile tube stress shall not exceed the allowable value given in Table 1.

8.5.6.3 *Tube compressive stress*

The allowable tube compressive stress (S_t) shall be calculated as follows:

$$S_t = \frac{\pi^2 r^2 E_t}{2 L_k^2} \qquad \text{When } C \leq \frac{L_k}{r} \quad (39)$$

or,

$$\frac{S_y}{F_y} \left(1 - \frac{L_k}{2rC} \right) \qquad \text{When } C > \frac{L_k}{r} \quad (40)$$

where

$$C = \sqrt{\frac{2\pi^2 E_t}{S_y}}$$

F_y is the factor of safety = $3.25 - 0.5 F_q$. F_y shall not be less than 1.25 and need not be greater than 2.0

L_k is the buckling length = lk where

l is the unsupported tube span (in mm)

k = 0.6 for unsupported spans between two tubeplates

0.8 for unsupported spans between a tubeplate and a tube support

1.0 for unsupported spans between two tube supports

r is the radius of gyration of tube (in mm)

$= 0.25\sqrt{d^2 + (d - 2_t)^2}$

S_y is the yield stress of tube material at design temperature (in N/mm^2).

NOTE The value of E_t is as given in **8.5.1**.

8.5.6.4 *Tube joint end load*

The allowable tube joint end load shall be limited to:

a) for joint types a and b (see Table 9)

$$W_{j\,max} = A_t f_t F_r \quad (41)$$

b) for joint types c, d and e (see Table 9)

$$W_{j\,max} = A_t f_t F_r F_e F_m \quad (42)$$

where

A_t is the cross-sectional area of tube (in mm^2)

f_t is the design stress value for tube material at design temperature (in mm^2)

F_r is the reliability factor (see Table 9)

F_e is the expansion factor (not greater than 1.0):

1) for grooved holes

$F_e = 1.0$

2) for plain holes

$$F_e = \frac{L_e}{\mathrm{d}}$$

where

L_e is the expanded length

F_m is the material factor (not greater than 1.0)

$$F_m = \frac{f}{f_t}$$

In the case of welded tube ends the procedure shall be in accordance with BS 4870-3.

Table 9 — Reliability factor (F_r) for joint types

Joint type	*Fr*
a) Welded with minimum weld throat ≥ tube thickness	0.8
b) Welded with minimum weld throat < tube thickness	0.55
c) Expanded and welded with minimum weld throat ≥ tube thickness	0.8
d) Expanded and welded with minimum weld throat < tube thickness	0.55
e) Expanded only	0.5
NOTE These values of F_r can be increased if the procedure is approved and checked with a pull out test.	

8.5.7 ***Longitudinal shell stress***

8.5.7.1 *Effective longitudinal shell stress*

On exchangers with unconfined heads having tubeplates fixed both ends without a shell expansion joint and if expansion bellows are not fitted, the effective longitudinal shell stress shall be calculated as follows:

$$S_s = \frac{P_s^*(D - t_s)}{4t_s} \tag{43}$$

where

$P_s^* = P_t - P'_t$

or

$P_s^* = P's$

or

$P_s^* = P_t - P'_t + P'_s$

or

$P_s^* = -YP_d$

or

$P_s^* = Y(P_t - P'_t - P_d)$

or

$P_s^* = Y(P'_s - P_d)$

or

$P_s^* = Y(P_t - P'_t + P'_s - P_d)$

whichever gives the greatest absolute value and

$Y = 1.0$ if the algebraic sign is negative

$Y = 0.5$ if the algebraic sign is positive

8.5.7.2 *Tensile shell stress*

The tensile shell stress shall not exceed the allowable value given in Table 1.

8.5.7.3 *Compressive stress*

The compressive shell stress shall be limited to 0.5 × allowable value given in Table 1.

8.5.8 ***Special cases***

Special consideration shall be given to tubeplates with abnormal conditions of support or loading.

Example 1

Fixed tubeplates in exchangers with expansion joints which require considerable axial loads to produce required movements.

Example 2

Tubeplates with portions not adequately stayed by tubes e.g. exchangers with large unpierced annular gap or area between the tube bundle and the shell.

8.6 Channels, channel covers and floating heads (data sheet items 31 and 32)

8.6.1 ***General***

In calculating the thickness of a channel, the channel shall be tested as a cylinder and any restraint imposed by the attachment of pass partitions shall be ignored.

8.6.2 ***Thickness of materials***

8.6.2.1 *Materials other than cast iron*

The minimum thickness of channels and channel covers in materials other than cast iron shall be not less than the calculated thickness for shell and shell ends obtained in accordance with **8.3.1.1**.

8.6.2.2 *Cast iron*

The minimum thickness of channels, bonnets and floating head covers in cast iron shall be not less than that obtained by calculation in accordance with **9.9** of BS 853-1:1996.

8.7 Compensation for branches

8.7.1 ***General***

NOTE Openings in the shell or headers of a heat exchanger should preferably be circular or elliptical.

Compensation of openings shall conform to **7.7** of BS 853-1:1996 using the design stress values given in Table 1.

The edges of openings shall be located at least 12 mm clear of welded seams. Where this is not practicable the compensation of openings of all sizes shall conform to **7.7** of BS 853-1:1996 except that the actual shell thickness used to determine area A4 for carbon steel shells and area A2 for copper shells shall be multiplied by the joint factor J in accordance with **8.3.1.1**.

8.7.2 ***Uncompensated openings***

Single openings at least 12 mm clear of welded seams shall not require compensation when either:

a) not larger than 80 mm diameter in shells or headers of thickness 10 mm or less, or

b) not larger than 50 mm diameter in shells or headers of thickness greater than 10 min.

8.7.3 ***Maximum openings***

The ratio of opening diameter to nominal shell diameter, or size of the opening, shall not exceed the following values:

a) for shells up to and including 300 mm diameter a ratio of 67 %;

b) for shell diameters between 300 mm and 400 mm the opening shall not exceed 200 mm.

c) for shells greater than 400 mm a ratio of 50 %.

8.8 Branches and connections

8.8.1 *Branch pipes*

The size for branch pipes shall be as given in Table 10.

Table 10 — Preferred tube sizes and thicknesses for branch pipes

Steel		Copper		
Nominal size mm	Thickness mm	Nominal size mm	Preferred Standard thickness mm	Other Standard thickness mm
15	2.3	10	0.8	1.0
20	2.3	12	0.8	1.0
25	2.7	16	1.0	1.5
32	3.1	20	1.0	2.0
40	3.1	25	1.5	2.0
50	3.7	30	1.5	2.0
65	3.7	38	1.5	2.0
80	4.7	44.5	1.5	2.0
100	4.7	57	1.5	2.0
125	5.5	76.1	2.0	2.5
150	5.5	88.9	2.5	3.0
200	6.2	108	2.5	3.0
250	7.9	133	2.5	3.0
300	7.9	159	2.5	3.0
350	7.9	193.7	3.0	3.5
400	8.4	219.1	3.0	4.0
450	8.4	267	3.0	4.0
500	9.5	323.9	4.0	4.5
600	9.5	368	4.0	4.5
		419	4.0	4.5
		457.2	4.0	4.5
		508	4.5	5.0

NOTE 1 For carbon steel the thickness should be increased by the amount of any required corrosion allowance.

NOTE 2 The design of branch pipe necks is governed by the following three main considerations.

a) Ability to withstand design pressure. For this purpose the minimum thickness of a branch neck should be calculated in accordance with the provisions for cylindrical shells (see **8.3.1**).

b) Ability to withstand superimposed loading by connected pipework or fittings. Not withstanding the minimum thickness as required for a) or to meet compensation requirements. The nominal thickness of a branch intended for connection to external piping should not be less than:

1) The value given in this table, increased by the amount of any required corrosion allowance, or

2) The nominal (as-built) thickness of the main portion of the vessel shell where this is less than i).

c) Suitability for the recommended forms of branch to shell attachment pads.

NOTE 3 Because of the ductile nature of copper every effort should be made to provide external support for branch connections to copper shells. No allowance should be made in reinforcement of copper shells for external nozzle loads.

8.8.2 ***Connections***

8.8.2.1 *General*

The length of thread in each of the connections below, from **8.8.2.2** to **8.8.2.4**, shall be not less than that given in Table 11.

Table 11 — Minimum length of thread

Thread designation R or G	Minimum length of thread mm
$\frac{1}{4}$ and $\frac{3}{8}$	8
$\frac{1}{2}$ and $\frac{3}{4}$	10
1 and $1\frac{1}{4}$	16
$1\frac{1}{2}$	20
2 to $2\frac{1}{2}$	23
3	26
4	32

8.8.2.2 *Screwed connections*

Screwed connections shall conform to BS 21 or BS 2779. Screwed mountings shall not exceed the requirements of type designation R2 of BS 21 or type designation G2 of BS 2779. Screwed pipe connections shall not exceed the requirements of type designation R4 of BS 21 or BS 2779.

8.8.2.3 *Screwed pipe connections*

Screwed pipe connections shall only be used for liquids that are below their flash point at the design temperature when subjected to atmospheric pressure, or for steam below 0.7 N/mm^2.

8.8.2.4 *Screwed connection bosses*

Screwed connection bosses shall be welded or brazed to the shell or channel.

8.8.3 ***Flanges, bolts and studs for branches***

Flanges, bolts and studs for branches and pads to which pipes or mountings are to be connected shall conform to BS 1560 or BS 4504.

8.9 Drain and vent connections

8.9.1 ***General***

One drain and one vent connection shall be provided on both the shell and the tube side of each section, except where several sections are arranged in a vertical stack and complete drainage is possible from the lower section and complete venting from the upper section.

8.9.2 ***Protrusions***

Connections for vents and drains in channels shall not protrude beyond the inside contour of the shell.

8.10 Pressure gauge connection

One pressure gauge connection shall be provided on both the shell side and the tube side of any heat exchanger assembly.

9 Tubes (data sheet items 27 and 28)

9.1 Plain tubes

9.1.1 ***Nominal size and thickness***

The nominal size and wall thickness of the tubes shall be selected from:

a) BS 2871-2 for copper and copper alloys;

b) BS 3059 or BS 3606 for carbon, low alloy and austenitic stainless steels.

9.1.2 ***Minimum thickness***

The minimum wall thicknesses, t, of plain tubes shall be not less than the following:

$$t > t_c + b_a \qquad (44)$$

where

t_c is the wall thickness (in mm) calculated in accordance with the requirements of **9.1.3** for straight tubes,

b_a is the bending allowance (in mm) calculated in accordance with the requirements of **9.1.4** which is equal to zero for straight tubes.

NOTE Preferred nominal tube diameters and thicknesses are given in Table 12.

9.1.3 ***Calculated thickness***

9.1.3.1 *Straight tubes subject to internal pressure*

The calculated wall thickness, t_c, to be used for straight tubes subject to internal pressure shall, subject to the check for strength in accordance with **9.1.3.2**, be the greater of the following:

a) $$t_c = \frac{pd}{0.9\ (p + 2f)} \qquad (45)$$

b) $$t_c = d/26 \qquad (46)$$

where

d is the nominal outside diameter of the tube (in mm);

p is the tube side pressure (in N/mm^2);

f is the design stress value (in N/mm^2).

Table 12 — Preferred tube diameter and nominal thicknesses for heat exchange tubes

External tube diameter	Nominal thickness		
	Carbon and low alloy steels	Copper and copper alloys	Stainless steels
mm	mm	mm	mm
12.5	1.6	0.9	0.9
16	1.6	1.2	1.2
19	1.6	1.6	1.2
25	2.0	1.6	1.6
32	2.0	1.6	1.6
38	2.0	2.0	2.0

9.1.3.2 *Straight tubes subject to external pressure*

The calculated wall thickness, t_c, for straight tubes in accordance with **9.1.3.1** shall be checked for strength when the tube is subject to external pressure. Figure 8 gives the maximum external pressure, P_o, which shall be used to calculate the different ratios of t_c/d using the design stress, f, from Table 1.

If necessary the calculated thickness, t_c, shall be increased to satisfy the design stress and shell design pressure.

9.1.4 ***Bending allowance***

The bending allowance, b_a, shall be not less than that calculated from the following equation:

$$b_a = \frac{dt_c}{2.5r} \tag{47}$$

where

t_c is the calculated wall thickness for straight tubes (in mm);

d is the nominal outside diameter of the tube (in mm);

r is the radius of curvature measured to the centreline of the bend (in mm).

9.2 Extended surface tubes

Extended surface tubes shall be formed from tubes conforming to BS 2871.

The nominal wall thickness of a tube with external fins shall be in accordance with **9.1.2**, except that d shall be taken to be the diameter at the root of the fins in millimetres.

The nominal wall thickness of an internally finned tube with plain external surface shall be in accordance with **9.1.2**, where the thickness, t, shall be taken to be half the difference between the external diameter of the tube and the diameter at the contact point of the fin and tube.

9.3 Indented or other enhanced surface tubes

Indented tubes shall be formed from tubes conforming to BS 2871.

The nominal wall thickness of indented tubes or other enhanced surface tubes of similar form shall be in accordance with **9.1.2**.

10 Construction

10.1 Tubeplates

10.1.1 ***Tube arrangement***

Tubes shall be laid out either on an equilateral triangular pitch or on a square pitch.

10.1.2 ***Tube pitch***

With the exception of tubes that are welded to the tubeplate, tubes shall have a minimum pitch value 1.25 times their outside diameter nominal value and shall be subject to the following tolerances:

a) for nominal pitch value of 45 mm and less: ± 0.8 mm

b) for nominal pitch value of over 45 mm: ± 1.6 mm.

NOTE Special consideration should be given to the pitch of tubes and to tubeplate preparation when tubes are welded to the tubeplate. A typical weld construction is shown in Figure 9.

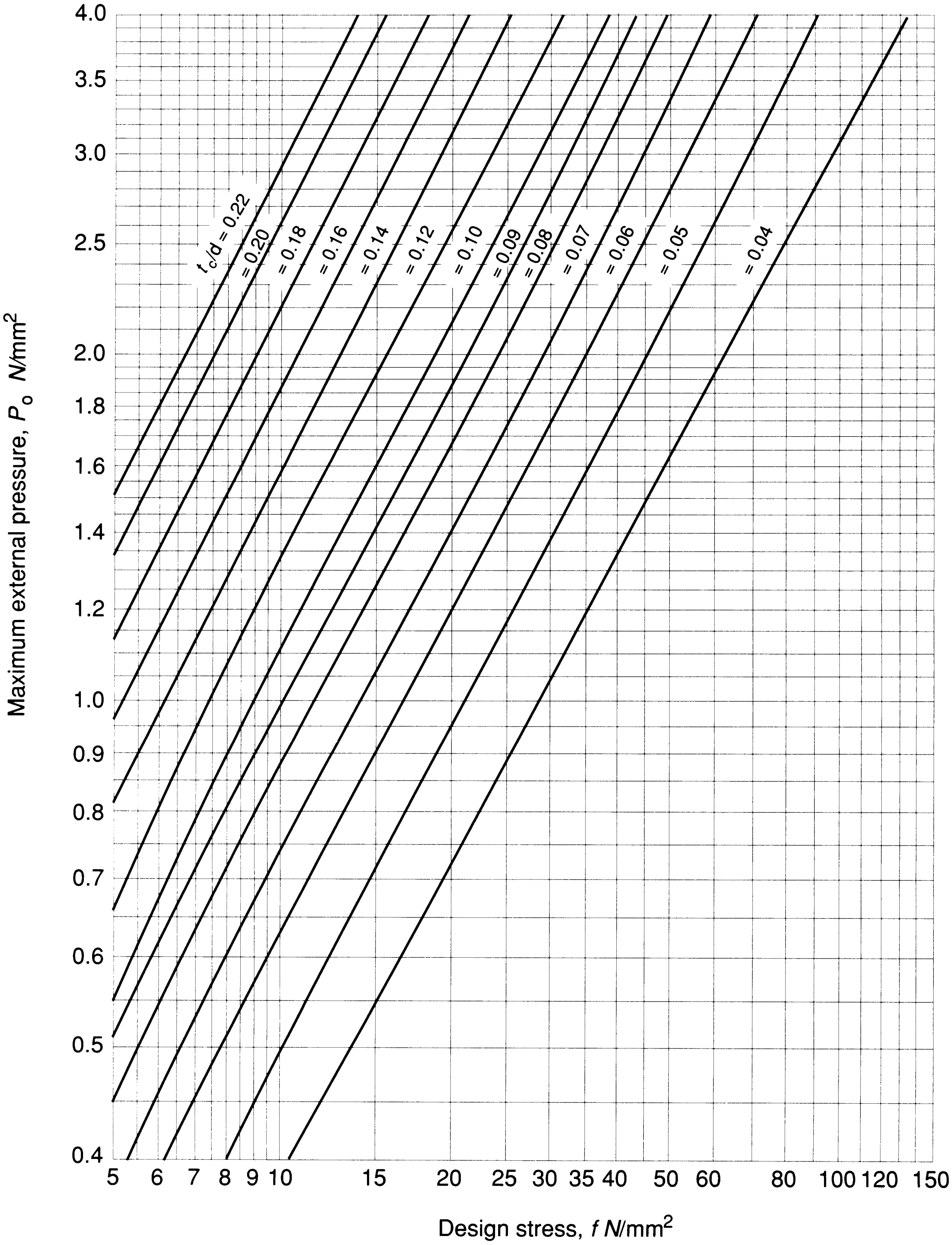
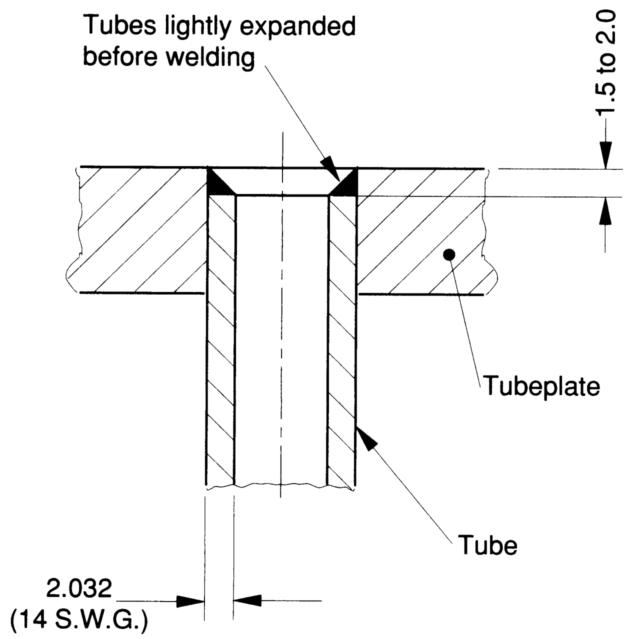


Figure 8 — Maximum external pressure of tubes for different ratios of t_c/d



Dimensions in mm

Figure 9 — Typical weld construction of tube to tubeplate

10.1.3 ***Tube hole ligaments***

The tolerances on the nominal values of tubeplate ligament widths shall be as follows:

a) for ligament nominal values of under 6 mm: ± 0.8 mm;

b) for ligament nominal values of 6 mm and over: ± 1.6 mm.

10.1.4 ***Tube holes***

10.1.4.1 *Size of tube holes*

For parallel expanded tubes that are not subject to work hardening the holes shall be drilled not more than:

a) 0.3 mm larger than the tube diameter for tubes up to 30 mm diameter, and

b) 0.5 mm larger than the tube diameter for tubes greater than 30 mm in diameter.

For tubes that are subject to work hardening the maximum diametral clearance shall be 0.1 min.

10.1.4.2 *Grooving of tube holes*

Except where excessive working of tube ends might lead to stress corrosion, for temperatures of 185 °C and above, tube holes shall have two circumferential anchor grooves 3 mm wide by 0.4 mm deep. The grooves shall be separated from one another by at least 3 mm and be a distance of at least 3 mm plus any corrosion allowance that is specified in **8.2** from either face of the tubeplate. (See Figure 10.)

The thickness of any tubeplate with anchor grooves shall be not less than 15 mm.

NOTE Where excessive working of the tube ends might lead to stress corrosion, alternative means such as welding may be required for securing the tubes.

10.1.4.3 *Finish of tubes and tube holes*

The edge of tube holes shall be free from burrs. The external surface of the tubes and the surface of the holes shall be smooth before the tubes are inserted.

10.1.5 ***Gasket seating surfaces***

Tubeplates used in conjunction with cast iron channels, bonnets or cast iron floating heads shall have the plate extended to the outside diameter of the cast iron flange and full face gaskets shall be used on both sides of the tubeplate.

10.1.6 ***Tie rod and spacer support holes***

Holes drilled in tubeplates for tie rods and spacer supports shall be blind.

10.2 Tube fixing

10.2.1 ***Expanding***

Where tubes are fixed by expanding only, the construction shall conform to **10.2.1** a) to **10.2.1** c) as follows.

a) The tube ends shall not project beyond the face of the tubeplate by more than 5 mm.

b) The expanding shall not extend beyond the inner face of the tubeplate.

c) Tube ends shall be softened if necessary to enable them to be expanded satisfactorily.

NOTE 1 With regard to **10.2.1** c) it is recommended that brass tubes are supplied in the annealed condition to conform to BS 2871-3.

NOTE 2 Tube ends may be bell mouthed.

NOTE 3 It is recommended that an automatic device be employed for the expanding operation to restrict and control the torque applied and consequently the degree and thinning of the tube wall.

10.2.2 ***Welding***

Where the tubes are welded into the tubeplates, the tubes shall be lightly expanded before welding except in the case of alloys whose heat resisting properties would be impaired by cold working.

NOTE Special consideration should be given to the method and details of expanding and welding of alloys whose heat resisting properties would be impaired by cold working.

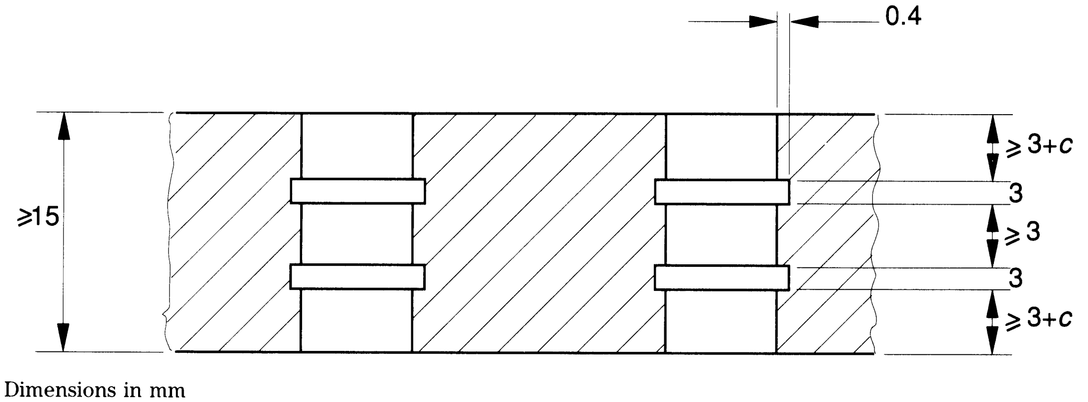


Figure 10 — Typical section through tubeplate showing arrangement of grooved tube holes

10.3 Means of breaking joints

Heat exchangers which have removable tube bundles and a nominal diameter exceeding 350 mm and/or an overall length exceeding 2 500 mm shall, where the design permits, be provided with means for breaking the joints between the stationary tubeplate and the shell flange.

NOTE Such means may be tapped holes in the tubeplate to receive jacking screws or pulling eyes.

10.4 Removal of channel or bonnet

For a heat exchanger whose channel or bonnet is to be removed without disturbing the tubeplate, collar bolts or stud bolts shall be provided in the tubeplate to prevent the inner joint being broken when the channel or bonnet is removed.

10.5 Internal floating heads

10.5.1 *Head design*

Internal floating heads shall be of a design that permits withdrawal of the tube bundle from the stationary tubeplate end of the shell. Where an end cover is fitted for easy access to the internal floating head without removing the tube bundle, the internal floating head shall be supported free from the end cover.

10.5.2 *Support plate thickness*

If a plate is used to support the floating head, the plate thickness shall be not less than given in Table 14 of an unsupported tube length of 3 000 mm.

10.6 Baffles and support plates (data sheet items 38 and 39)

NOTE 1 The details of **10.6** do not preclude the possibility of tube vibration due to pulsating conditions. It may be necessary to make special provision in the design of the baffles and support plates if this is likely to happen.

NOTE 2 Special consideration should be given to finned tubes where these pass through baffles and support plates and to longitudinal baffles subjected to large pressure differentials due to high shell side fluid pressure drop.

10.6.1 *Diametral clearance*

The diametral clearance between transverse baffles or support plates and the shell shall not exceed 6 mm for shell diameters up to and including 400 mm and shall not exceed 10 mm for shell diameters greater than 400 mm.

NOTE Reduced clearance may be necessary for improved heat transfer coefficients or to maintain the mean temperature difference but clearance should be sufficient for free movement of removable tube bundles.

10.6.2 *Drainage*

Provision shall be made in the profile of the baffles for drainage from the shell.

10.6.3 *Tube hole clearance*

The diametral clearance between tubes and the tube holes drilled in baffles or support plates shall be not greater than 1 mm. All burrs shall be removed from the holes.

10.6.4 *Location of tube supports*

The maximum unsupported length of straight tube shall not exceed the value given in Table 13 for the material used.

For U-tube exchangers either the support plates or baffles adjacent to the bends shall be so located that for any individual bend the sum of the bend diameter plus the straight lengths of both legs from supports to the bend tangents does not exceed the value given in Table 13, otherwise special provision shall be made to support the U-bends.

10.6.5 *Thickness*

10.6.5.1 *Transverse baffles and support plates*

The thickness of materials for transverse baffles and tube support plates for tubes without fins shall be not less than the values given in Table 14.

The thickness of materials for transverse baffles and tube support plates for finned tubes shall be not less than the values given in Table 14 nor span fewer than five fins.

10.6.5.2 *Longitudinal baffles*

The minimum thickness of longitudinal carbon steel baffles shall not be less than 6 mm. Alloy steel and non-ferrous baffles shall be not less than 3 mm thick.

10.6.6 ***Impingement baffles***

10.6.6.1 *Shell side fluid*

An impingement baffle or other protective device shall be fitted when the entrance conditions of the shell side fluid exceed the following values:

a) for non-corrosive, non-abrasive fluids (in kg/m s^2):

$\rho V^2 > 2\ 250$

b) for all other fluids including a liquid at its boiling point (in kg/m s^2):

$\rho V^2 > 750$

where

ρ is the density (in kg/m^3);

V is the velocity of the fluid (in m/s).

The shell entrance area as defined in **10.6.6.2** shall not allow the value of ρV^2 to exceed 6 000 kg/m s^2

Table 13 — Maximum unsupported length of straight tubes

Min. tube outside diameter (including fins)	Tube materials	
	Carbon steel, steel alloys, stainless steel	Copper and copper alloys
mm	mm	mm
6	650	550
10	875	750
12	1 100	950
16	1 300	1 125
19	1 500	1 300
22	1 675	1 450
25	1 850	1 600
32	2 200	1 900
38	2 500	2 200
50	3 000	2 750

Table 14 — Minimum transverse baffle or support plate thickness

Maximum plate diameter	Minimum transverse baffle or support plate thickness for unsupported tube length[a] (mm) not exceeding:				
	600	900	1 200	1 500	3 000
mm	mm	mm	mm	mm	mm
350	3	5	6	10	10
700	5	6	10	10	13
1 000	6	8	10	13	16

[a] The maximum unsupported tube length refers to the straight section of the tube bundle and does not include the U-bend section.

10.6.6.2 *Shell entrance area*

When an impingement plate is fitted, the shell entrance area shall be defined as the unrestricted area between the impingement plate and adjacent parts of the heat exchanger shell (see Figure 11).

The shell entrance area A_s shall be not less than the nozzle bore area A_n, that is $A_s \geq A_n$ (see Figure 11).

When no impingement plate is fitted the shell entrance area in the presence of tubes shall be defined as the area within the projected nozzle circle formed between the tubes and the shell inside diameter plus the flow area between the first row of tubes.

NOTE When an axial inlet nozzle is used on the tube side of the heat exchanger or when liquid velocity in the tubes exceeds 3 m/s consideration should be given to protecting the tube ends against erosion. In the case of U-tubes it is recommended that liquid velocities in the tubes should not exceed 2 m/s.

10.7 Tie rods

Tie rods and spacers or other equivalent devices shall be provided to retain all transverse baffles and tube support plates securely in position and shall be of material compatible with the baffles or plates.

The number and diameter of tie rods shall be in accordance with Table 15.

10.8 Gaskets (data sheet item 37)

10.8.1 ***General***

Gaskets shall conform to **10.8.2** to **10.8.5**, as appropriate to the gasket type.

NOTE Table 8 lists gasket materials and contact facings together with gasket factors (m) for operating conditions and minimum design seating stress (y).

10.8.2 ***Peripheral ring gaskets***

The minimum width of peripheral ring gaskets for external joints shall be 10 mm.

10.8.3 ***Gaskets with cast iron flanges***

Gaskets used where either or both flanges are of cast iron shall extend to the full face of the flanges (see **10.1.5**).

10.8.4 ***Compressed asbestos or other composition gaskets***

Compressed asbestos or other composition gaskets shall be of one piece construction throughout.

10.8.5 ***Metal jacketed or solid metal gaskets***

Metal jacketed or solid metal gaskets shall have no break in continuity of the metal in contact with the flange surface to be sealed.

NOTE It is recommended that any joint in a metal gasket should be on the gasket's external periphery.

10.9 "O"-rings

"O"-rings shall be used only where they are confined in rectangular grooves and shall conform to the recommendations of the "O"-ring manufacturer for size, material and service conditions.

11 Pressure relief devices

11.1 General

The heat exchanger shall be protected against excess pressure by pressure relief devices.

Where mountings are fitted by the manufacturer they shall be in accordance with the requirements of **11.2** and **11.3**. Where they are not fitted by the manufacturer provision shall be made for their attachment.

NOTE 1 Heat exchangers connected together by piping of adequate capacity which does not contain any valve that can isolate individual parts may be considered as a single system for pressure relief.

NOTE 2 When the source of pressure (or temperature) is external to the vessel and is such that the pressure cannot exceed the design pressure, a protective device need not be mounted directly on the heat exchanger.

11.2 Safety valves

11.2.1 ***General***

The construction of safety valves shall be in accordance with the requirements of BS 6759. Valves shall be arranged vertically and shall be attached directly to or as close as possible to the pressure relief point. The connection shall have a bore at least equal to the area of the safety valve with no intervening valve between the pressure relief point and the safety valve.

11.2.2 ***Relieving capacity***

11.2.2.1 *Total capacity of the safety valves*

The total capacity of the safety valves mounted on or adjacent to the heat exchanger shall be sufficient to discharge the maximum quantity of fluid, liquid or gas that can be generated without occurrence of a rise in pressure of more than 10 % above the design pressure.

NOTE Advantage may be taken of the pressure drop across any intermediate valve or restriction to limit the flow or pressure reaching the heat exchanger when the safety valves are operating.

11.2.2.2 *Low temperature side*

The low temperature side of a heat exchanger shall be protected against over pressure due to isolation from the system when the high temperature side is functioning.

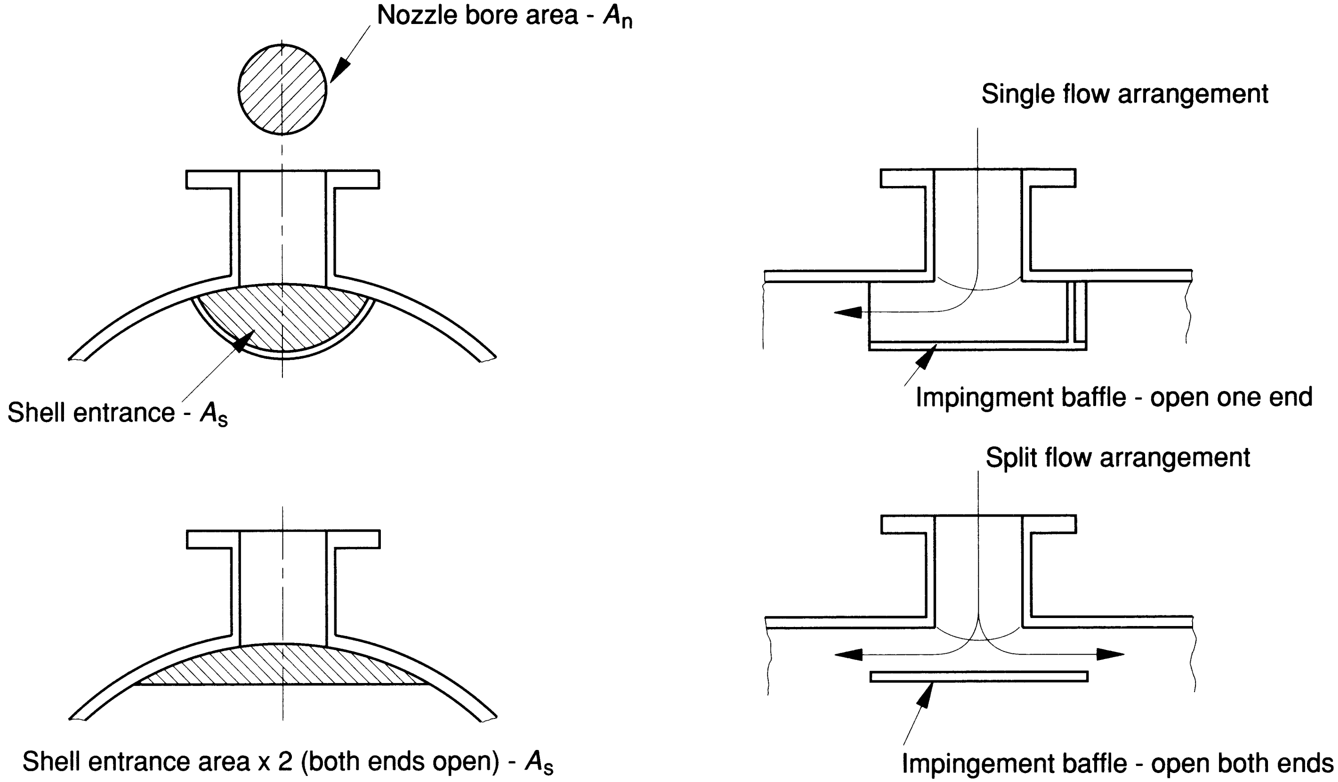


Figure 11 — Impingement baffles

Table 15 — Minimum number and diameter of tie rods

Maximum inside shell diameter mm	Minimum tie rod diameter mm	Minimum number of rods
350	6	4
450	8	4
700	8	6
850	12	6
1 000	12	8

11.2.2.3 *Sizing for pressurized (unvented) systems*

The safety valve shall be selected from those made in accordance with **21.5.5** of BS 6759-1:1984 as a result of the application of the following equation:

$$A_f = \frac{R}{0.329 P_r K_{dr}} \qquad (48)$$

where

- R is the required rating of the safety valve (in kW);
- P_r is the maximum absolute relieving pressure {(shell design pressure × 1.10)} + 1.0 (in bar absolute);
- A_f is the flow area (in mm^2);
- K_{dr} is the derated coefficient of discharge (i.e. the coefficient of discharge as stated for a particular valve by the manufacturer).

The nominal size of the safety valve shall be not less than 20 mm (¾ in).

NOTE The safety valve selected may have a higher rating than that required of the heat exchanger.

11.2.2.4 *Sizing for open vented systems*

The size of the safety valve shall be selected as a result of the application of the following equation:

$$A_f = \frac{0.5R}{0.329 P_r K_{dr}} \qquad (49)$$

The nominal size of valve shall be not less than 20 mm (¾ in).

NOTE 1 Symbols are as given in **11.2.2.2**.

NOTE 2 The value of 0.5*R* takes account of the extra discharge capacity of the vent.

11.2.2.5 *Set pressure*

The minimum set pressure shall be determined in accordance with the following equation:

$$\text{set pressure} = 1.10 \times \text{working pressure}^{2)} \qquad (50)$$

[2] If working pressure is in metres head, multiply by 0.0098 to obtain the working pressure (in N/mm^2).

In no case shall the margin between the working pressure and that at which the safety valve is set be less than 0.035 N/mm^2 (0.35 bar). In hot water heating systems with accelerated circulation, the set pressure shall be not less than 0.07 N/mm^2 (0.7 bar) in excess of the working pressure.

11.3 Bursting discs

Where the failure of a tube suddenly increases the fluid pressure on the other side of a heat exchanger beyond its design pressure, the low pressure side shall be protected with a bursting disc which shall be in accordance with BS 2915:1984.

NOTE When assessing the possibility of a sudden pressure rise the nature of the two fluids should be considered. In general both fluids will be both liquids or both vapours for the pressure rise to be sufficiently rapid to warrant a bursting disc.

11.4 Discharge

A pressure relief device shall require a discharge pipe equal to or greater than the size of the outlet port of the safety device.

NOTE It is important for safety reasons that the discharge pipe is laid with a continuous downward gradient clear of the heat exchanger to a place where the discharge is visible and cannot injure any person.

11.5 Pressure setting of pressure relieving devices

11.5.1 ***Single safety valve***

When a safety valve is fitted it shall be set to operate at a nominal pressure not exceeding the design pressure of the heat exchanger at the operating temperature.

11.5.2 ***Multiple safety valves***

If the capacity is provided by more than one safety valve, one of the valves shall be set to operate as required by **11.5.1**. The additional valve or valves shall be set to operate at a pressure not more than 5 % in excess of the design pressure at the operating temperature, provided there is conformity to the overall requirement of **11.2.2**.

12 Inspection and testing

12.1 General

The purchaser shall advise the manufacturer of details of any independent inspection requirements.

NOTE It is recommended that the purchaser and manufacturer should discuss and agree the extent of these requirements and a mutually-acceptable programme before the order stage and if possible at the enquiry stage.

The purchaser shall have the right to inspect, at any reasonable time, the fabrication at any stage and to reject any parts which do not conform to this standard.

12.2 Hydraulic testing

The hydraulic test shall be carried out in accordance with **11.2** of BS 853-1:1996.

The conditions of test and procedure adopted shall take the following aspects into account:

a) the tube side test pressure is equal to or in excess of shell side test pressure;

b) the shell side test pressure is in excess of tube side test pressure;

c) there is no limitation on which side of the exchanger has the higher test pressure.

The hydraulic test shall be carried out using either service gaskets to permit the heat exchanger to be despatched in the as tested condition or test gaskets (see note 3). If test gaskets are used they shall be of the same type as the service gaskets.

NOTE 1 To permit inspection of all joints (bolted, welded, etc.) it is sometimes necessary, depending on design, to test components of the heat exchanger before final assembly. To facilitate this, suitable gland rings and clamp rings may be necessary.

NOTE 2 The manufacturer may incorporate additional test rings to those referred for inspection and testing where this is agreed with the purchaser. This may include the use of special test shells where numbers of identical units are involved.

NOTE 3 The use of service gaskets or test gaskets in the hydraulic test is a matter for agreement between the manufacturer and the purchaser.

NOTE 4 If the hydraulic test is to be supplemented or replaced by a pneumatic test the manufacturer should be aware of his responsibility for safety and the recommendations of the latest Health and Safety Executive Guidance Note GS4 — *Safety in pressure testing*. Reference should also be made to BS 5500.

12.3 Proof hydraulic testing of cast chests and other components

The maximum design pressure, P_R, (in N/mm^2), of cast chests of other than spherical shape and of other cast components shall be based on testing identical components to destruction and shall be calculated from the following equation:

$$P_R = 0.15\,P_B \qquad (51)$$

where

P_R is the maximum design pressure within the limits of service conditions for temperature and pressure given in the notes to Table 1 (N/mm^2);

P_B is the destruction test pressure (in N/mm^2).

NOTE If a hydraulic proof test pressure of 7.0 N/mm^2 (70 bar) is reached without destruction of the component a maximum design pressure of 1.03 N/mm^2 (10.3 bar) can be used for identical components.

All components of the same material, design and construction shall be hydraulically tested in accordance with **12.2**.

CAUTION. Attention is drawn to the hazards involved in the proof testing of components and precautions should be taken to protect personnel against injury.

13 Marking

13.1 General

Each heat exchanger shall have a plate securely attached in an accessible position. The plate shall bear the information required in **13.2** and shall be made from a corrosion resistant material. Where the heat exchanger is to be lagged the plate shall be mounted so that is will not be obscured by the lagging.

13.2 Plate marking

The plate specified in accordance with **13.1** shall indicate:

a) then name and/or mark of the manufacturer and inspecting authority;

b) the manufacturer's serial number;

c) the design pressure — shell ... N/mm^2 (or bar); tubes ... N/mm^2 (or bar);

d) the design temperature, — shell ... °C; tubes ... °C;

e) the hydraulic test pressure, — shell ... N/mm^2; tubes ... N/mm^2;

f) the letters SR if stress relieved;

g) the date of the hydraulic test;

h) the number of this British Standard, i.e. BS 853-2:1996[a].

[a] Marking BS 853-2:1996 on or in relation to a product represents a manufacturer's declaration of conformity, i.e. a claim by or on behalf of the manufacturer that the product meets the requirements of the standard. The accuracy of the claim is solely the claimant's responsibility. Such a declaration is not to be confused with third party certification of conformity, which may also be desirable.

13.3 Shell cover, floating head cover, channel and collar bolt marking

The shell cover, floating head cover and channel shall be clearly marked on the edges of the flanges with the manufacturer's serial number of the heat exchanger. The position of collar bolts shall be indicated by the letter "C" stamped on the edges of the flanges adjacent to each other bolt.

13.4 Orientation marking

For heat exchangers with removeable tube bundles, the stationary tubeplate shall be marked on its edge so as to indicate its correct orientation in the shell.

14 Test certificates

14.1 Materials for pressure bearing components

The manufacturer shall supply on request the test certificates or letters of conformity, where appropriate, giving chemical analysis and physical properties.

NOTE The manufacturer's quality assurance programme should show that only certified materials are used for pressure bearing components. Full material traceability during construction is not a mandatory requirement of this standard.

Annex A (informative) Typical heat exchanger data sheet

1.	Customer			Job No			
				Ref. No			
2.	Address			Enquiry No			
3.	Plant location			Date			
4.	Description			Item No			
5.	Size	Shells per unit		Arrangement			
6.	Type	Surface per shell		Surface per unit			
7.	Vented or unvented system						
8.	Operating data for one unit						
9.		Units		Shell side		Tube side	
10.	Description of fluid						
11.	Liquid	(kg/h)	In	Out	In	Out	
12.	Steam	(kg/h)	In	Out	In	Out	
13.	Absolute viscosity	(cp)	In	Out	In	Out	
14.	Molecular mass						
15.	Specific heat (cp)	kJ/(kg·K)		....			
16.	Latent heat	kJ/(kg)		....			
17.	Thermal conductivity	W/(m·K)		....			
18.	Temperature	°C	In	Out	In	Out	
19.	Working pressure	N/mm^2 (gauge)	In	Out	In	Out	
20.	Number of passes			....		....	
21.	Velocity	(m/s)					
22.	Fouling resistance	$w/m^2 \cdot k$					
23.	Heat exchanged	kJ/h M.T.D.* (corrected)				°C	
24.	Overall heat transfer coefficient (service)				w(m.°C)		
25.	Construction and materials						
26.	Design pressure and temperature		bar (gauge)	°C	bar (gauge)	°C	
27.	Test pressure and temperature		bar (gauge)	°C	bar (gauge)	°C	
28.	Tubes	Material					
29.	Tubes	No./shell	o.d .... (mm)	wall thickness .... (mm)	Length ...(mm)	Pitch .....(mm)	
30.	Shell			Nom. i.d .... (mm)	Thickness .... (mm)		
31.	Shell cover			Floating head cover			
32.	Channel			Channel cover			
33.	Tubeplates		Stationary	Floating			
34	Baffles (cross)			Type	Thickness .... (mm)		
35	Baffles (longitudinal)			Type	Thickness .... (mm)		
36	Tube support			Type	Thickness .... (mm)		
37.	Gaskets						
38.	Branches Shell		In .... (mm)	Out .... (mm)	Flanges		
	Channel		In .... (mm)	Out .... (mm)	Flanges		
39.	Corrosion allowance		Shell side	Tube side			
40.	Design code						
41.	Special regulations effecting the heat exchanger or storage vessel, e.g. local water bye-laws or building regulations						
42.	Are test certificates to be provided						
43.	Are hydraulic test certificates to be provided						
44.	Inspection requirements						
45.	Mass	Each exchanger complete-dry		Tube bundle...	Unit full of water .... (kg)	....	
46.	Information supplied by			For duty	For surface area		
47.	Remarks						

* M.T.D. (mean temperature difference)

Annex B (informative) Typical fouling resistance values

Substance	**Fouling resistance** (w/m^2 k)
Water:	
sea	0.0001762
brackish	0.0003524
cooling tower	0.0003524
river	0.0003524
distilled/closed cycle	0.0000881
treated boiler feed	0.0001762
boiler blowdown	0.0003524
Other liquids:	
fuel oil	0.000881
lubricating oil	0.0001762
thermal oil	0.0001762
hydraulic fluid	0.0001762
kerosene	0.0001762
caustic solutions	0.0003524
vegetable oil	0.0005286
Gases and vapours:	
steam (non-oil bearing)	0.0000881
exhaust steam	0.0001762

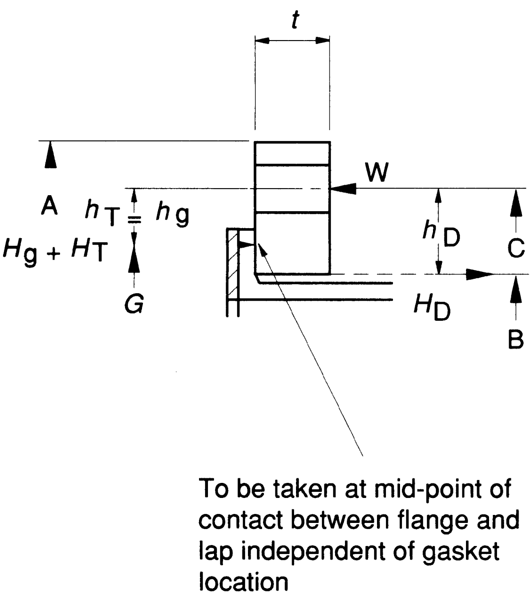
Annex C (informative)
Typical flange design procedure: work sheet

Design pressure	P	N/mm^2	Gasket details	$b = b_o$ if $b_o \le 6.3$ mm	N	
Design temperature		°C			b_o	
Flange material			Facing details Type: Gasket o.d.:	$b = 2.52\sqrt{b_o}$ if $b_o > 6.3$ mm	b	
Bolting material					y	
Corrosion allowance		mm	W_{m2}	$= 3.14bGy =$	m	
Allowable bolt stress — Atm. temp.	S_b	N/mm^2	H_p	$= 6.28bGmp =$		
Allowable bolt stress — Design temp.	S_a	N/mm^2	H	$= 0.785G^2p =$	$H_p + H =$	
Allowable flange stress — Design temp.	S_{FO}	N/mm^2	A_m	$=$ greater of $\frac{W_{m2}}{S_a}$ or $\frac{H_p + H}{S_b} =$		
Allowable flange stress — Atm. temp.	S_{FA}	N/mm^2	A_b	$=$		
All dimensions are shown in the corroded condition			W	$= 0.5(A_m + A_b)\,S_a =$		
			W_{m1}	$= H_p + H =$		

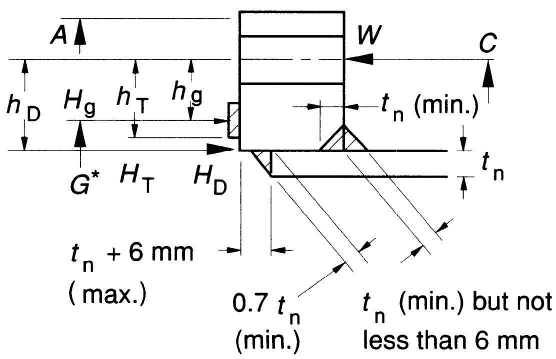
Operating condition		
$H_D = 0.785B^2p =$	$h_D = 0.5(C - B) =$	$H_D h_D =$
$H_g = H_P =$	$h_D = 0.5(C - G) =$	$H_g h_g =$
$H_T = H - H_D =$	(1) $h_T = 0.5\,(C - G) =$	$H_T h_T =$
	(2) $h_T = 0.5\,(h_D + h_g) =$	$M_{op} =$

Bolting-up condition		
$H_g = W =$	$h_g = 0.5(C - G) =$	$M_{atm} = H_g h_g =$
$M_o =$ greater of M_{op} or $M_{atm} \times \frac{S_{FO}}{S_{FA}} =$	Apply factor C_F when $C_F = \sqrt{\frac{h}{2(g) + 6t/(m + 0.5)}} \ge 1$	$M = \frac{M_o C_F}{B} =$

1) Lap joint flange



2) Welded flange



Shape constants:

$$K = \frac{A}{B}$$

Stress calculation:

$$t = \sqrt{\frac{YM}{S_{FO}}}$$

where Y is the factor for figure 7 based on K

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List of references

Normative references

BSI publications

BRITISH STANDARDS INSTITUTION, London

BS 21:1985, *Specification for pipe threads for tubes and fittings where pressure-tight joints are made on the threads (metric dimensions).*

BS 853-1:1996, *Specification for calorifiers and storage vessels for central heating and hot water supply.*

BS 1400:1985, *Specification for copper alloy ingots and copper alloy and high conductivity copper castings.*

BS 1452:1990, *Specification for flake graphite cast iron.*

BS 1453:1972, *Specification for filler materials for gas welding.*

BS 1501, *Steels for pressure purposes.*

BS 1501-3:1990, *Specification for corrosion- and heat-resisting steels: plates, sheet and strip.*

BS 1502:1982, *Specification for steels for fired and unfired pressure vessels: sections and bars.*

BS 1503:1989, *Specification for steel forgings for pressure purposes.*

BS 1504:1976, *Specification for steel castings for pressure purposes.*

BS 1560, *Circular flanges for pipes valves and fittings (Class designated).*

BS 1560-3, *Steel, cast iron and copper alloy flanges.*

BS 1640, *Specification for stell butt-welding pipe for the petroleum industry.*

BS 1640-1:1962, *Wrought carbon and ferritic alloy steel fittings.*

BS 1640-2:1962, *Wrought and cast austenitic chromium-nickel steel fittings.*

BS 1640-3:1968, *Wrought carbon and ferritic alloy steel fittings. Metric units.*

BS 1640-4:1968, *Wrought and cast austenitic chromium nickel steel fittings. Metric units.*

BS 1740, *Specification for wrought steel pipe fittings (screwed BS 21 R-series thread).*

BS 1740-1:1971, *Metric units.*

BS 1845:1984, *Specification for filler metals for brazing.*

BS 1945:1971, *Specification for fireguards for heating appliances (gas, electric and oil-burning).*

BS 2779:1986, *Specification for pipe threads for tubes and fittings where pressure-tight joints are not made on the threads (metric dimensions).*

BS 2789:1985, *Specification for spheroidal graphite or nodular graphite cast iron.*

BS 2870:1980, *Specification for rolled copper and copper alloys: sheet, strip and foil.*

BS 2871, *Specification for copper and copper alloys. Tubes.*

BS 2871-2:1972, *Tubes for general purposes.*

BS 2871-3:1972, *Tubes for heat exchangers.*

BS 2872:1989, *Specification for copper and copper alloy forging stock and forgings.*

BS 2874:1986, *Specification for copper and copper alloy rods and sections (other than forging stock).*

BS 2875:1969, *Specification for copper and copper alloys. Plate.*

BS 2915:1990, *Specification for bursting discs and bursting disc devices.*

BS 2926:1984, *Specification for chromium and chromium-nickel steel electrodes for manual metal-arc welding.*

BS 3059, *Steel boiler and superheater tubes.*

BS 3059-1:1987, *Specification for low tensile carbon steel tubes without specified elevated temperature properties.*

BS 3059-2:1990, *Specification for carbon, alloy and austenitic stainless steel tubes with specified elevated temperature properties.*

BS 3601:1987, *Specification for carbon steel pipes and tubes with specified room temperature properties for pressure purposes.*

BS 3602, *Specification for steel pipes and tubes for pressure purposes: carbon and carbon manganese steel with specified elevated temperature properties.*

BS 3602-1:1987, *Specification for seamless and electric resistance welded including induction welded tubes.*

BS 3602-2:1991, *Specification for longitudinally arc welded tubes.*

BS 3605, *Austenitic stainless steel pipes and tubes for pressure purposes.*

BS 3605-1:1991, *Specification for seamless tubes.*

BS 3605-2:1992, *Specification for longitudinally welded tubes.*

BS 3606:1992, *Specification for steel tubes for heat exchangers.*

BS 3692:1967, *Specification for ISO metric precision hexagon bolts, screws and nuts. Metric units.*

BS 3799:1974, *Specification for steel pipe fittings, screwed and socket-welding for the petroleum industry.*

BS 4165:1984, *Specification for electrode wires and fluxes for the submerged arc welding of carbon steel and medium-tensile steel.*

BS 4206:1967, *Methods of testing fusion welds in copper and copper alloys.*

BS 4504, *Circular flanges for pipes, valves and fittings. (PN designated).*

BS 4504-3, *Steel, cast iron and copper alloy flanges.*

BS 4504-3.1:1989, *Specification for steel flanges.*

BS 4504-3.2:1989, *Specification for cast iron flanges.*

BS 4504-3.3:1989, *Specification for copper alloy and composite flanges.*

BS 4518:1982, *Specification for metric dimensions of toroidal sealing rings (“O” rings) and their housings.*

BS 4870, *Specification for approval testing of welding procedures.*

BS 4870-3:1985, *Arc welding of tube to tube-plate joints in metallic materials.*

BS 4882:1990, *Specification for bolting for flanges and pressure containing purposes.*

BS 5465:1987, *Specification for electrodes and fluxes for the submerged arc welding of austenitic stainless steels.*

BS 5500:1994, *Specification for unfired fusion welded pressure vessels.*

BS 6759, *Safety valves.*

BS 6759-1:1984, *Specification for safety valves for steam and hot water.*

BS EN 287, *Approval testing of welders for fusion welding.*

BS EN 287-1:1992, *Steels.*

BS EN 287-2:1992, *Aluminium and aluminium alloys.*

BS EN 288, *Specification and approval of welding procedures for metallic materials.*

BS EN 288-2:1992, *Welding procedures specification for arc weldings.*

BS EN 288-3:1992, *Welding procedure tests for the arc welding of steels.*

BS EN 440:1995, *Welding consumables. Wire electrodes and deposits for gas shielded metal arc welding of non alloy and fine grain steels. Classification.*

BS EN 499:1995, *Welding consumables. Covered electrodes for manual metal arc welding of non alloy and fine grain steels. Classification.*

BS EN 29453:1994, *Soft solder alloys. Chemical compositions and forms.*

Informative references

BSI Publications

BRITISH STANDARDS INSTITUTION, London

BS 5555:1993, *Specification for SI units and recommendations for the use of their multiples and of certain other units.*

BS 5775, *Specification for quantities, units and symbols.*

Other references

[1] GREAT BRITAIN. Control of Asbestos at Work Regulations, 1987 (SI 2115 as amended). London: HMSO.

[2] GREAT BRITAIN. Health and Safety at Work etc. Act 1974. London: HMSO.

[3] HEALTH AND SAFETY EXECUTIVE. Guidance note EH/10, *Environmental Hygiene, Asbestos*. London: HMSO.

[4] HEALTH AND SAFETY EXECUTIVE. Guidance note GS4, *Safety in pressure testing*. London: HMSO.

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