APROXIMACION AL ESTUDIO DEL RIESGO DEL BLEVE Y SUS EFECTOS EN LOS GENERADORES MARINOS DE VAPOR Y LOS TANQUES DE CARGA DE LOS BUQUES LNG-LPG. APLICACION COMPARATIVA DE LAS NORMAS QUE LO REGULAN Y PREVIENEN.

Autor: German de Melo Rodriguez
Director: Emilio Eguia López

Barcelona, mayo de 1994

ANEXO 6

```
AT961: PR [A9] 2, 7182818^(-(AT957/AT956)+AT958)
AU961: PR [A9] 2, 7182818^(-(AU957/AU956)+AU958)
AQ962: PR [A7] '
AR962: PR [A12] '
AS962: PR [A8] '
AT962: PR [A9] '
AU962: PR [A9] '
AP965: PR [A26] '
AP967: PR [A26] '      - (A / T) + B
AP968: PR [A26] 'P = e
AP969: PR [A26] '
AP973: PR [A26] 'Alt-q >>> VUELVE AL MENU
```

Chapter 10

Steam Raising Plant and Associated Pressure Vessels

CONTENTS

SECTION 1

General requirements

1.1 Application

1.1.1 The requirements of this Chapter are applicable to pressure vessels of seamless and fusion welded construction, and their mountings or fittings, for the following uses:

(a) Fired boilers.

(b) Exhaust gas heated boilers.

(c) Economizers, superheaters, reheaters and steam receivers for, and associated with, (a) and (b).

(d) Steam heated steam generators.

1.2 Definition of symbols

1.2.1 The symbols used in the various formulae in Sections 2 to 8, unless otherwise stated, are defined as follows and are applicable to the specific part of the pressure vessel under consideration:

d = diameter of hole or opening, in mm,

D_i = inside diameter, in mm,

D_o = outside diameter, in mm,

J = joint factor applicable to welded seams (*see* 1.9), or ligament efficiency between tube holes (expressed as a fraction, *see* 2.2),

p = design pressure (*see* 1.3), in bar (kgf/cm^2),

R_i = inside radius, in mm,

R_o = outside radius, in mm,

r_i = inside knuckle radius, in mm,

r_o = outside knuckle radius, in mm,

s = pitch, in mm,

T = design temperature, in °C,

t = minimum thickness, in mm,

σ = allowable stress (*see* 1.8), in N/mm^2 (kgf/cm^2).

1.3 Design pressure

1.3.1 The design pressure is the maximum permissible working pressure and is to be not less than the highest set pressure of any safety valve.

1.3.2 The calculations made to determine the scantlings of the pressure parts are to be based on the design pressure, adjusted where necessary to take account of pressure variations corresponding to the most severe operational conditions.

1.3.3 It is desirable that there should be a margin between the normal pressure at which the boiler or pressure vessel operates and the lowest pressure at which any safety valve is set to lift, to prevent unnecessary lifting of the safety valve.

1.4 Metal temperature

1.4.1 The metal temperature, T, used to evaluate the allowable stress, σ, is to be taken as the actual metal temperature expected under operating conditions for the pressure part concerned, and is to be stated by the manufacturer when plans of the pressure parts are submitted for consideration.

1.4.2 The following values are to be regarded as the minimum:

(a) For fired boilers, T is to be taken as not less than 250°C.

(b) For steam heated steam generators, secondary drums of double evaporation boilers, steam receivers and pressure parts of fired pressure vessels, not heated by hot gases and adequately protected by insulation, T is to be taken as the maximum temperature of the internal fluid.

(c) For pressure parts heated by hot gases, T is to be taken as not less than 25°C in excess of the maximum temperature of the internal fluid.

(d) For boiler, superheater, reheater and economizer tubes, T is to be taken as indicated in 7.1.3.

(e) For combustion chambers of the type used in horizontal wet-back boilers, T is to be taken as not less than 50°C in excess of the maximum temperature of the internal fluid.

(f) For furnaces, fireboxes, rear tube plates of dry-back boilers and pressure parts subject to similar rates of heat transfer, T is to be taken as not less than 90°C in excess of the maximum temperature of the internal fluid.

1.4.3 In general, any parts of boiler drums or headers not protected by tubes, and exposed to radiation from the fire or to the impact of hot gases, are to be protected by a shield of good refractory material or by other approved means.

1.4.4 Drums and headers of thickness greater than 30 mm are not to be exposed to combustion gases having an anticipated temperature in excess of 650°C unless they are efficiently cooled by closely arranged tubes.

1.5 Classification of fusion welded pressure vessels

1.5.1 For Rule purposes, pressure vessels are graded as Class 1 if they comply with the following conditions:

(a) for pressure parts of steam raising plant as indicat in 1.1.1(a), (b) and (c), where the design pressu exceeds 3,4 bar (3,5 kgf/cm²), or

(b) for pressure parts of steam heated steam generato where the design pressure exceeds 11,3 bar (11 kgf/cm²), or where the design pressure, in bar (k cm²), multiplied by the internal diameter of the sh in mm, exceeds 14 420 (14 700), or

(c) for other pressure vessels where the shell thickne exceeds 38 mm.

1.5.2 For Rule purposes, the pressure parts of boil and associated pressure vessels not included in Clas are graded as Class 2/1 and Class 2/2.

1.5.3 Pressure vessels which are constructed accordance with Class 2/1 or Class 2/2 standards indicated above) will, if manufactured in accordan with the requirements of a superior Class, be approve with the scantlings appropriate to that Class.

1.5.4 In special circumstances relating to serv conditions, materials, operating temperature, the c riage of dangerous gases and liquids, etc., it may required that certain pressure vessels be manufactu in accordance with the requirements of a superior Cla

1.5.5 Heat treatment, non-destructive examinatio and routine tests, where required, for the three classes fusion welded pressure vessels are indicated in Tab 10.1.1. Details are given in Chapter 17.

1.6 Plans

1.6.1 Plans of boilers, superheaters and economize are to be submitted in triplicate for consideration. Wh plans of water tube boilers are submitted for approva particulars of the safety valves and their disposition o boilers and superheaters, together with the estimate pressure drop through the superheaters, are to be stated The pressures proposed for the settings of boiler an superheater safety valves are to be indicated on th boiler plan.

1.6.2 Plans, in triplicate, showing full constructio features of fusion welded pressure vessels and dime sional details of the weld preparation for longitudinal a circumferential seams and attachments, together w particulars of the welding consumables and of t mechanical properties of the materials, are to b submitted before construction is commenced.

Table 10.1.1

Class	Radiographic examination	Heat treatment	Routine weld tests	Hydraulic test
1	Required *see* Chapter 17	*see* Chapter 17	Required	Required
2/1	Spot required *see* Chapter 17	*see* Chapter 17	Required	Required
2/2	–	*see* Chapter 17	Required	Required

1.7 Materials

1.7.1 Materials used in the construction are to be manufactured and tested in accordance with the requirements of the Society's *Rules for the Manufacture, Testing and Certification of Materials*.

1.7.2 The specified minimum tensile strength of carbon and carbon-manganese steel plates, pipes, forgings and castings is to be within the following general limits:

(a) For seamless and Class 1 and Class 2/1 fusion welded pressure vessels:
340 to 520 N/mm^2 (35 to 53 kgf/mm^2).

(b) For Class 2/2 fusion welded pressure vessels:
340 to 420 N/mm^2 (35 to 43 kgf/mm^2)

(c) For boiler furnaces, combustion chambers and flanged plates:
400 to 520 N/mm^2 (41 to 53 kgf/mm^2).

1.7.3 The specified minimum tensile strength of low alloy steel plates, pipes, forgings and castings is to be within the general limits of 400 to 500 N/mm^2 (41 to 51 kgf/mm^2), and pressure vessels made in these steels are to be either seamless or Class 1 fusion welded.

1.7.4 The specified minimum tensile strength of boiler and superheater tubes is to be within the following general limits:

(a) Carbon and carbon-manganese steels:
320 to 460 N/mm^2 (33 to 47 kgf/mm^2)

(b) Low alloy steels:
400 to 500 N/mm^2 (41 to 51 kgf/mm^2).

1.7.5 Where it is proposed to use materials other than those specified in the Society's *Rules for the Manufacture, Testing and Certification of Materials*, details of the chemical compositions, heat treatment and mechanical properties are to be submitted for approval. In such cases the values of the mechanical properties used for deriving the allowable stress are to be subject to agreement by the Society.

1.7.6 Where a fusion welded pressure vessel is to be made of alloy steel, and approval of the scantlings is required on the basis of the high temperature properties of the material, particulars of the welding consumables to be used, including typical mechanical properties and chemical composition of the deposited weld metal, are to be submitted for approval.

1.8 Allowable stress

1.8.1 The term 'allowable stress', σ, is the stress to be used in the formulae for the calculation of scantlings of pressure parts.

1.8.2 The allowable stress, σ, is to be the lowest of the following values:

$$\sigma = \frac{E_t}{1,6} \qquad \sigma = \frac{R_{20}}{2,7} \qquad \sigma = \frac{S_R}{1,5}$$

where E_t = specified minimum lower yield stress or 0,2 per cent proof stress at temperature T,
R_{20} = specified minimum tensile strength at room temperature,
S_R = average stress to produce rupture in 100 000 hours at temperature T,
T = metal temperature (*see* 1.4).

1.8.3 The allowable stress for steel castings is to be taken as 80 per cent of the value determined by the method indicated in 1.8.2, using the appropriate values for cast steel.

1.8.4 Where steel castings, which have been tested in accordance with the Society's *Rules for the Manufacture, Testing and Certification of Materials*, are also subjected to non-destructive tests, consideration will be given to increasing the allowable stress using a factor up to 90 per cent in lieu of the 80 per cent referred to in 1.8.3. Particulars of the non-destructive test proposals are to be submitted for consideration.

1.9 Joint factors

1.9.1 The following joint factors are to be used in the equations in Sections 2 to 8, where applicable. Fusion welded pressure parts are to be made in accordance with Chapter 17.

Class of pressure vessel	Joint factor
Class 1	1,0
Class 2/1	0,85
Class 2/2	0,75

1.9.2 The longitudinal joints for all classes of vessels are to be butt joints. Circumferential joints for Classes 1, 2/1 and 2/2 vessels are also to be butt joints.

1.10 Pressure parts of irregular shape

1.10.1 Where pressure parts are of such irregular shape that it is impracticable to desgn their scantlings by the application of the formulae in Sections 2 to 8, the suitability of their construction is to be determined by hydraulic proof test of a prototype or by an agreed alternative method.

1.11 Adverse working conditions

1.11.1 Where working conditions are adverse, special consideration may be required to be given to increasing the scantlings derived from the formulae, e.g. by increasing the corrosion or other allowance at present shown in the formulae, or by adopting a design pressure higher than defined in 1.3, to offset the possible reduction of life in service caused by the adverse conditions. In this connection, where necessary, account should also be taken of any excess of loading resulting from:

(a) impact loads, including rapidly fluctuating pressures,

(b) weight of the vessel and normal contents under operating and test conditions,

(c) superimposed loads such as other pressure vessels, operating equipment, insulation, corrosion-resistant or erosion-resistant linings and piping,

(d) reactions of supporting lugs, rings, saddles or other types of supports, or

(e) the effect of temperature gradients on maximum stress.

SECTION 2

Cylindrical shells and drums subject to internal pressure

2.1 Minimum thickness

2.1.1 The minimum thickness, t, of a cylindrical shell is to be determined by the following formula:

$$t = \frac{pR_i}{10\sigma J - 0{,}5p} + 0{,}75 \quad \text{mm}$$

$$\left(t = \frac{pR_i}{\sigma J - 0{,}5p} + 0{,}75 \quad \text{mm} \right)$$

where t, p, R_i and σ are as defined in 1.2,

J = efficiency of ligaments between tube holes or other openings in the shell or the joint factor of the longitudinal joints (expressed as a fraction). *See* 1.9 or 2.2, whichever applies. In the case of seamless shells clear of tube holes or other openings, $J = 1{,}0$.

2.1.2 The formula in 2.1.1 is applicable only where the resulting thickness does not exceed half the internal radius, i.e. where R_o is not greater than 1,5 R_i.

2.1.3 Irrespective of the thickness determined by the above formula, t is to be not less than:

(a) 9,5 mm for drum shell plates. In special cases where it is proposed to use a shell thickness less than 9,5 mm, the proposal will be subject to special consideration.

(b) For tube plates, such thickness as will give a minimum parallel seat of 9,5 mm, or such greater width as may be necessary to ensure tube tightness. *See* 14.9.

2.2 Efficiency of ligaments between tube holes

2.2.1 Where tube holes are drilled in a cylindrical shell in a line or lines parallel to its axis, the efficiency, J, of the ligaments is to be determined as in 2.2.2, 2.2.3 and 2.2.4.

2.2.2 **Regular drilling.** Where the distance between adjacent tube holes is constant (*see* Fig. 10.2.1),

$$J = \frac{s - d}{s}$$

Where s = pitch of tube holes, in mm,

d = the mean effective diameter of the tube holes, in mm, after allowing for any serrations, counterboring or recessing, or the compensating effect of the tube stub. *See* 2.3 and 2.4.

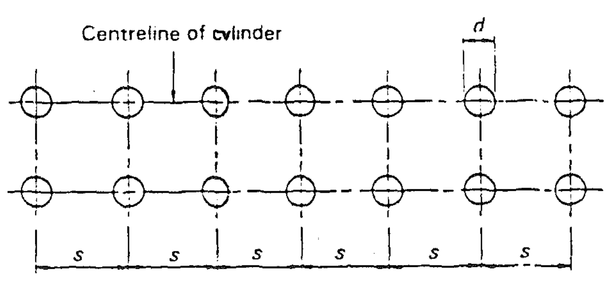


Fig. 10.2.1 Regular drilling

2.2.3 **Irregular drilling.** Where the distance betwe[en] centres of adjacent tube holes is not constant (see F[ig.] 10.2.2),

$$J = \frac{s_1 + s_2 - 2d}{s_1 + s_2}$$

where d is as defined in 2.2.2,

s_1 = the shorter of any two adjacent pitches, mm,

s_2 = the longer of any two adjacent pitches, mm.

2.2.4 When applying the formula in 2.2.3, the dou[ble] pitch ($s_1 + s_2$) chosen is to be that which makes J minimum, and in no case is s_2 to be taken as greater th[an] twice s_1.

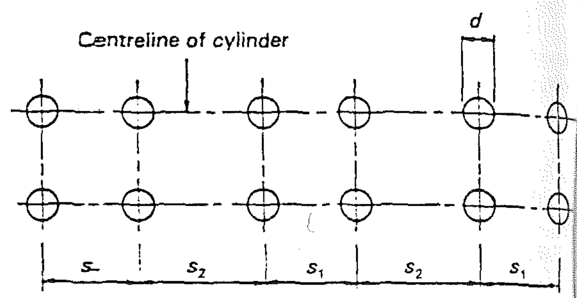


Fig. 10.2.2 Irregular drilling

2.2.5 Where the circumferential pitch between tu[be] holes measured on the mean of the external and intern[al] drum or header diameters is such that the circumferen[tial] ligament efficiency determined by the formulae in 2.2.[2] and 2.2.3 is less than one-half of the ligament efficien[cy] on the longitudinal axis, J in 2.1 is to be taken as tw[ice] the circumferential efficiency.

2.2.6 Where tube holes are drilled in a cylindrical sh[ell] along a diagonal line with respect to the longitudinal axi[s] the efficiency, J, of the ligaments is to be determined [as] in 2.2.7 to 2.2.10.

2.2.7 For spacing of tube holes on a diagonal line [as] shown in Fig. 10.2.3, or in a regular saw-tooth pattern [as] shown in Fig. 10.2.4, J is to be obtained from the se[t] of curves given in Fig. 10.2.6, where a and b, as sho[wn] in Figs. 10.2.3 and 10.2.4, are measured, in millimet[res] on the median line of the plate, and d is as define[d in] 2.2.2.

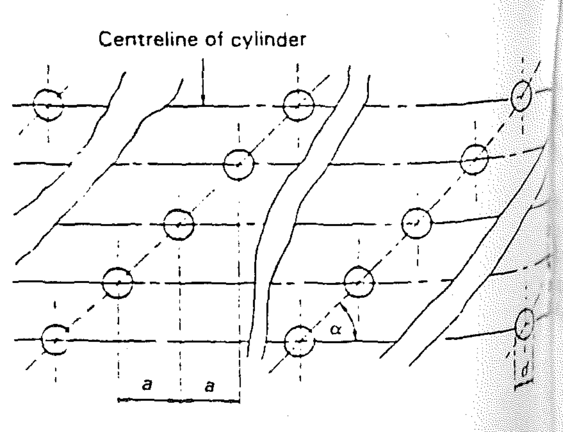


Fig. 10.2.3 Spacing of holes on a diagonal line

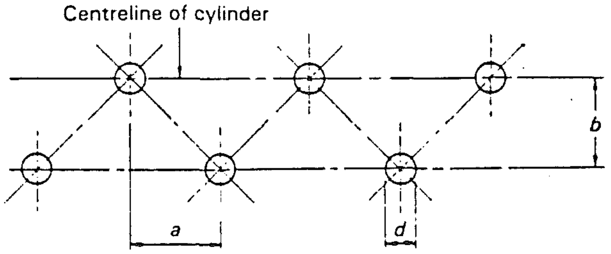


Fig. 10.2.4 Regular saw-tooth pattern of holes

2.2.8 The data for Fig. 10.2.6 is based on the following:

$$J = \frac{2}{A + B + \sqrt{(A-B)^2 + 4C^2}}$$

where $A = \dfrac{\cos^2\alpha + 1}{2\left(1 - \dfrac{d\cos\alpha}{a}\right)}$

$$B = 0{,}5\left(1 - \frac{d\cos\alpha}{a}\right)(\sin^2\alpha + 1)$$

$$C = \frac{\sin\alpha\cos\alpha}{2\left(1 - \dfrac{d\cos\alpha}{a}\right)}$$

$$\cos\alpha = \frac{1}{\sqrt{1 + \dfrac{b^2}{a^2}}}$$

$$\sin\alpha = \frac{1}{\sqrt{1 + \dfrac{a^2}{b^2}}}$$

α = angle between centreline of cylinder and centreline of diagonal holes.

2.2.9 For regularly staggered spacing of tube holes as shown in Fig. 10.2.5, the smallest value of the efficiency, J, of all ligaments (longitudinal, circumferential and diagonal) is obtained from Fig. 10.2.7 where a and b as shown in Fig. 10.2.5 are measured, in millimetres, on the median line of the plate; and d is as defined in 2.2.2.

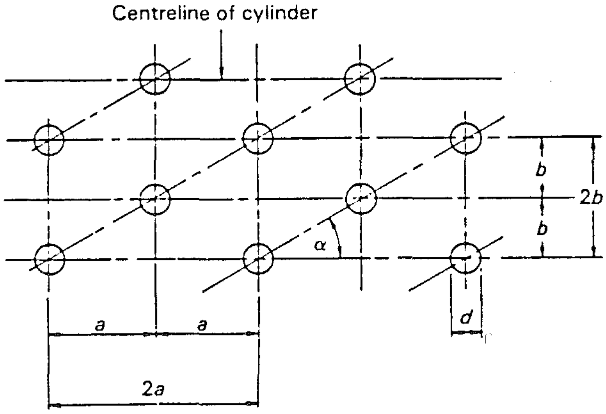


Fig. 10.2.5 Regular staggering of holes

2.2.10 For irregularly spaced tube holes whose centres do not lie on a straight line, the formula in 2.2.3 is to apply, except that an equivalent longitudinal width of the diagonal ligament is to be used. An equivalent longitudinal width is that width which gives, using the formula in 2.2.2, the same efficiency as would be obtained using Fig. 10.2.6 for the diagonal ligament in question.

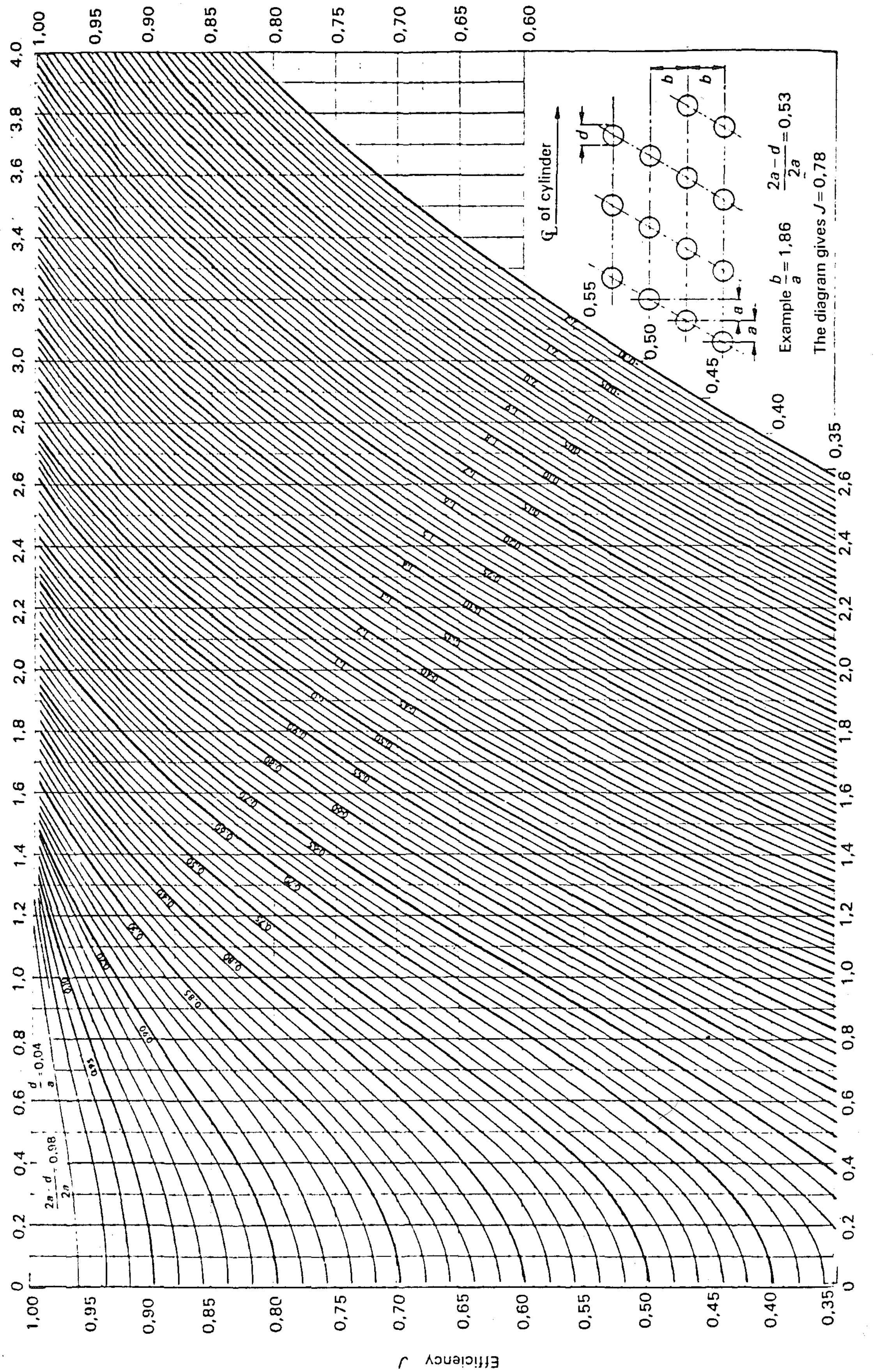


Fig. 10.2.6 Efficiency of ligament along a diagonal line

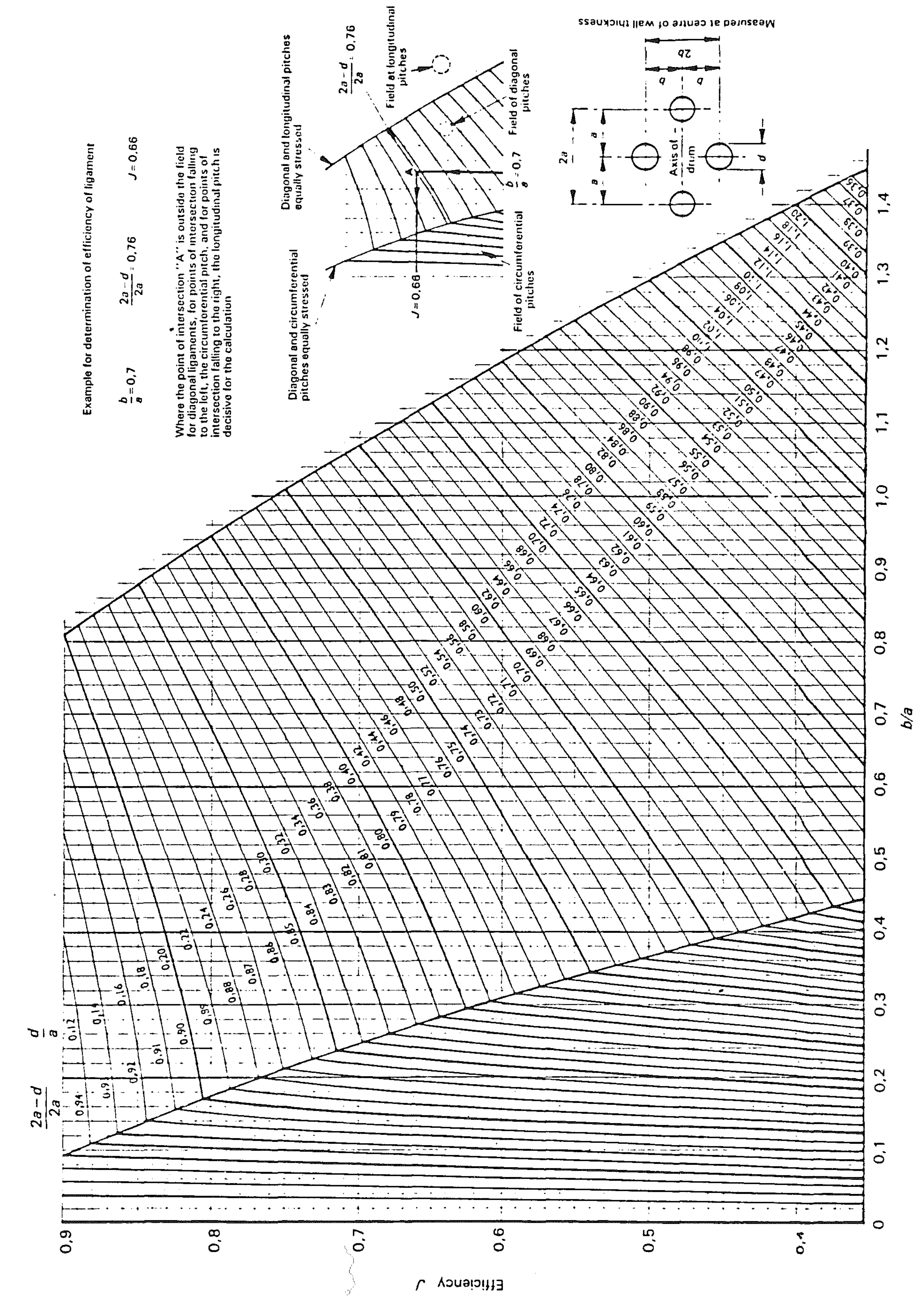


Fig. 10.2.7 Efficiency of ligaments between holes

2.3 Compensating effect of tube stubs

2.3.1 Where a drum or header is drilled for tube stubs fitted by strength welding, either in line or in staggered formation, the effective diameter of holes is to be taken as

$$d_e = d_a - \frac{A}{t}$$

where d_e = the equivalent diameter of the hole, in mm,
d_a = the actual diameter of the hole, in mm,
t = the thickness of the shell, in mm,
A = the compensating area provided by each tube stub and its welding fillets, in mm^2.

2.3.2 The compensating area, A, is to be measured in a plane through the axis of the tube stub parallel to the longitudinal axis of the drum or header and is to be calculated as follows (*see* Figs. 10.2.8 and 10.2.9):

The cross-sectional area of the stub, in excess of that required by 7.1 for the minimum tube thickness, from the interior surface of the shell up to a distance b from the outer surface of the shell;

plus the cross-sectional area of the stub projecting inside the shell within a distance b from the inner surface of the shell;

plus the cross-sectional area of the welding fillets inside and outside the shell;

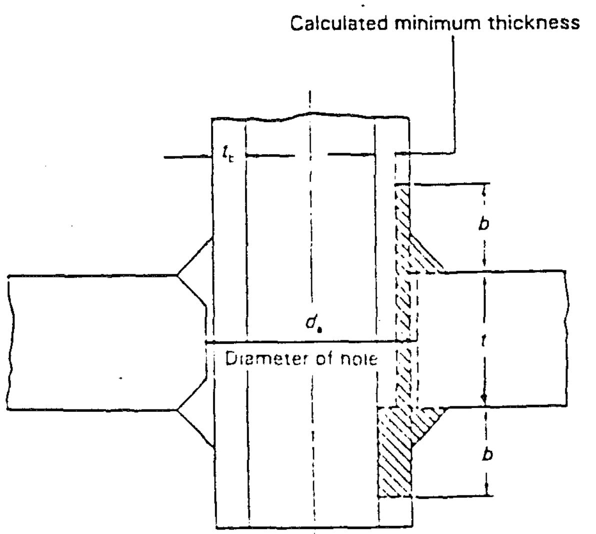
where $b = \sqrt{d_a t_b}$
t_b = actual thickness of the tube stub, in mm.

2.3.3 Where the material of the tube stub has an allowable stress lower than that of the shell, the compensating cross-sectional area of the stub is to be multiplied by the ratio:

$$\frac{\text{allowable stress of stub at design metal temperature}}{\text{allowable stress of shell at design metal temperature}}$$

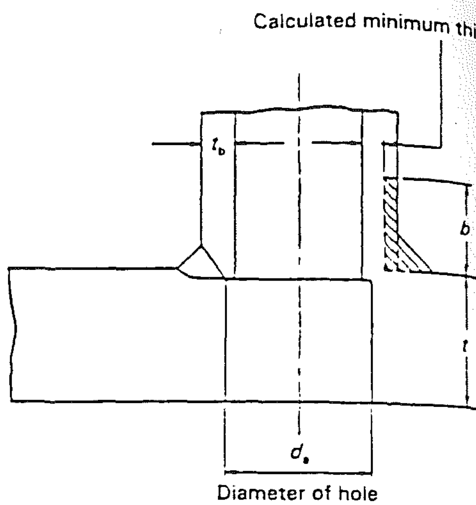
2.4 Unreinforced openings

2.4.1 Openings in a definite pattern, such as tube holes, may be designed in accordance with the Rules for ligaments in 2.2, provided that the diameter of the largest hole in the group does not exceed that permitted by 2.4.2.



The area shown hatched is half the area A

Fig. 10.2.8 Compensation of welded tube stubs
The calculated minimum thickness is to satisfy 7.1.



The area shown hatched is half the are

Fig. 10.2.9 Compensation of welded tube stubs
The calculated minimum thickness is to s

2.4.2 The maximum diameter, d, of any unre openings is to be obtained from the curves 10.2.10 or Fig. 10.2.11. The value of K to be calculated from the following formula:

$$K = \frac{pD_o}{18{,}2\sigma t} \qquad \left(K = \frac{pD_o}{1{,}82\sigma t}\right)$$

where p, D_o and σ are as defined in 1.2,
t = actual thickness of shell, in mm.

2.4.3 For elliptical or oval holes, d refers to t axis when this lies longitudinally or to the mea major and minor axes when the minor a longitudinally.

2.4.4 No unreinforced opening is to exceed in diameter.

2.5 Reinforced openings

2.5.1 The following notations are used 10.2.12:

A = calculated thickness of a shell without opening,
B = thickness calculated in accordance wit
$C = 0{,}8\sqrt{d_i t_b}$
$D = \sqrt{D_i t_a}$ and is not to exceed $0{,}5d_i$,
t_a = actual thickness of shell plate, in mm,
t_b = actual thickness of standpipe stem or b mm,
t_r = thickness of added reinforcement, in m be zero when there is no compensating the side of the shell under consideratio
d_i = internal diameter of standpipe or branch
D_i = internal diameter of cylindrical shell, in

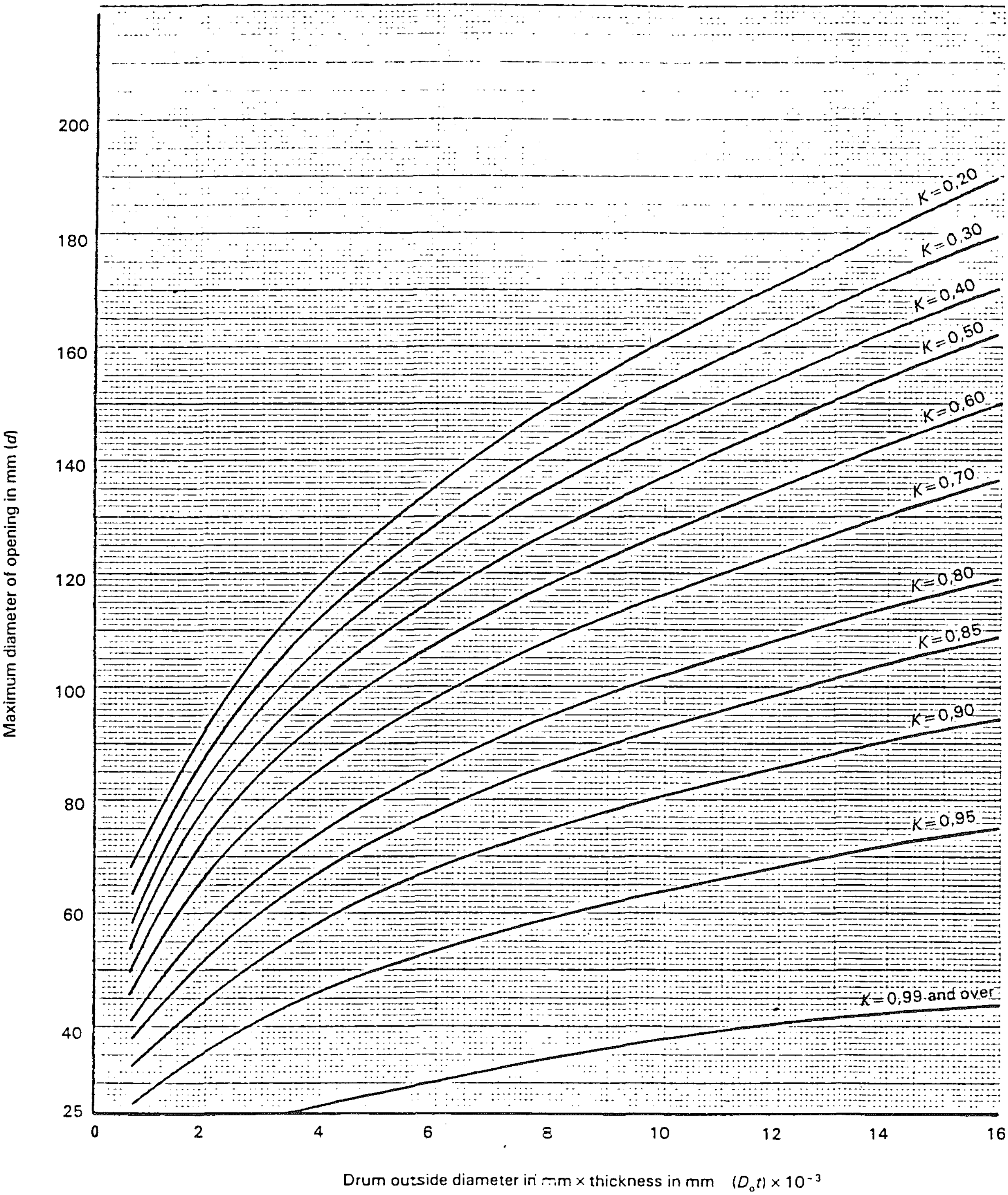


Fig. 10.2.10 Maximum diameter of unreinforced openings

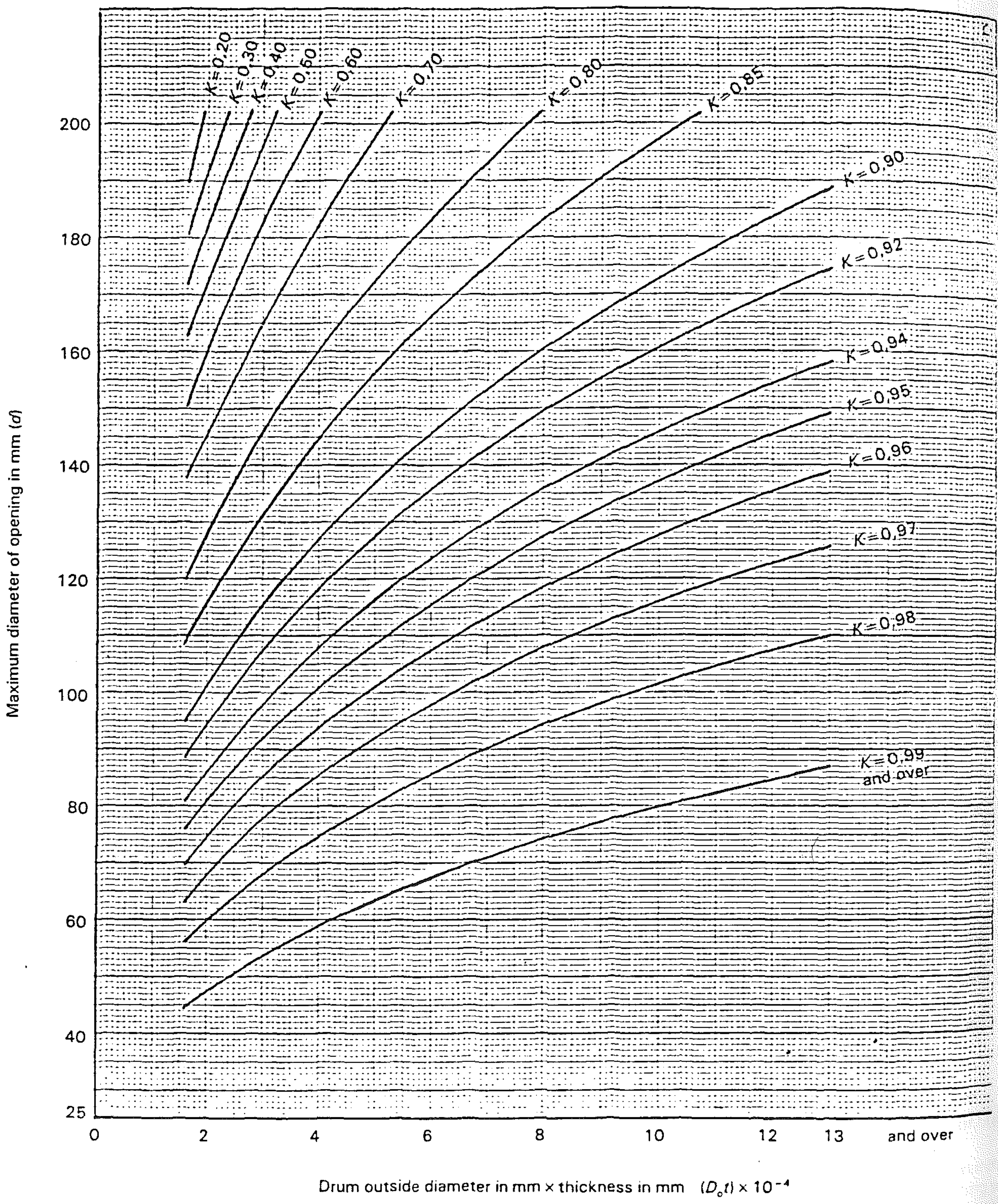
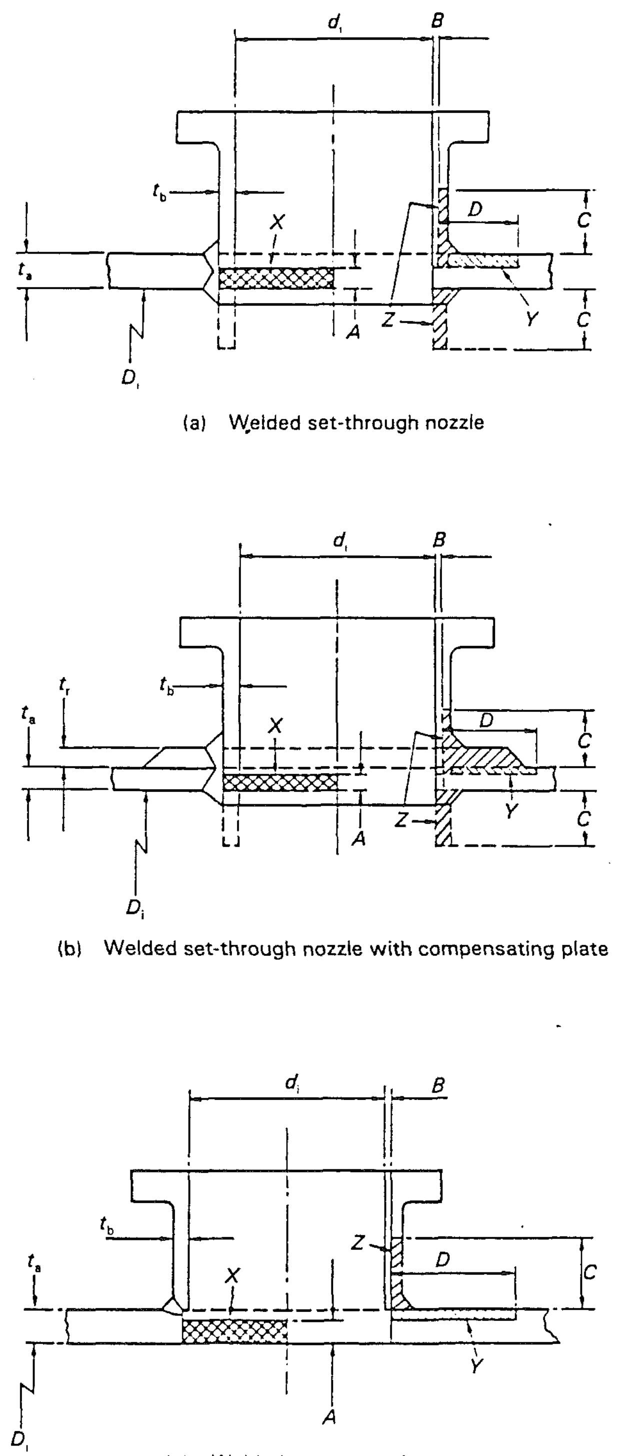


Fig. 10.2.11 Maximum diameter of unreinforced openings



(a) Welded set-through nozzle

(b) Welded set-through nozzle with compensating plate

(c) Welded set-on nozzle

Fig. 10.2.12 Compensation for welded standpipes or branches in cylindrical shells

2.5.2 Openings larger than those permitted by 2.4 are to be reinforced by the method shown in Fig. 10.2.12. Compensation will be considered adequate when:

$$Y + Z\frac{\sigma_s}{\sigma} \geqslant X$$

where X = the area to be compensated and is indicated by X in Fig. 10.2.12,

Y = the compensating area available in the she material and is indicated by Y in Fig. 10.2.12

Z = the compensating area available in the stand pipe material and is indicated by Z in Fig 10.2.12,

σ_s = the allowable stress of the standpipe materia at design temperature,

σ = the allowable stress of shell material a design temperature.

$\frac{\sigma_s}{\sigma}$ is to be taken as not greater than 1.

2.5.3 Area X is to be such that the reinforcement i: provided on all planes through the centre of the openinc and normal to the shell surface, and is to be calculated a: the product of the radius of the hole cut in the shell anc the thickness A that would be required for an equivalen seamless unpierced shell.

2.5.4 Area Y is to be measured in the same plane a: area X, and is to be calculated as the product of the difference between the actual shell thickness and the equivalent unpierced shell thickness, A, and the dimension from edge of opening in shell to limit D.

2.5.5 Area Z is to be measured in the same plane as area X, and is to be calculated as follows:

For that part of the standpipe which projects outside the shell, calculate the full cross-sectional area of the stem up to a distance C from the actual outer surface of the shell plate, and deduct from it the cross-sectional area which the stem would have if its thickness were as calculated in accordance with 7.1,

plus, in the case of set-through nozzles (*see* Fig. 10.2.12 (a) and (b)), the full cross-sectional area of that part of the stem which passes through the shell thickness defined by A and that part of the stem which projects inside the shell up to a distance C, from the inside surface of the shell,

plus the cross-sectional area of all appropriate fillet welds,

plus, if additional reinforcement is fitted as illustrated in Fig. 10.2.12 (b), the cross-sectional area of the reinforcement and the sectional area of its fillet welds.

2.5.6 The welds attaching standpipes and reinforcing plates to the shell are to be of sufficient size to transmit the full strength of the reinforcing areas and all other loadings to which they may be subjected.

SECTION 3

Spherical shells subject to internal pressure

3.1 Minimum thickness

3.1.1 The minimum thickness of a spherical shell is to be determined by the following formula:

$$t = \frac{pR_i}{20\sigma J - 0{,}5p} + 0{,}75 \text{ mm}$$

$$\left(t = \frac{pR_i}{2\sigma J - 0{,}5p} + 0{,}75 \text{ mm}\right)$$

where t, p, R_i, σ and J are as defined in 1.2.

3.1.2 The formula in 3.1.1 is applicable only where the resulting thickness does not exceed half the internal radius.

3.1.3 Openings in spherical shells are to comply with 4.3 to 4.6 where applicable.

SECTION 4

Dished ends subject to internal pressure

4.1 Minimum thickness

4.1.1 The thickness, t, of semi-ellipsoidal, torispherical and hemispherical unstayed ends, dished from plate, having pressure on the concave side and satisfying the conditions listed below, is to be determined by the following formula:

$$t = \frac{pD_oK}{20\sigma J} + 0{,}75 \text{ mm}$$

$$\left(t = \frac{pD_oK}{2\sigma J} + 0{,}75 \text{ mm}\right)$$

where t, p, D_o, σ and J are as defined in 1.2,
K = a shape factor, *see* 4.2 and Fig. 10.4.1.

4.1.2 For semi-ellipsoidal ends:
the external height, $H \geqslant 0{,}18\, D_o$
where D_o = the external diameter of the parallel portion of the end, in mm,

4.1.3 For torispherical ends:
the internal radius, $R_i \leqslant D_o$
the internal knuckle radius, $r_i \geqslant 0{,}1D_o$
the internal knuckle radius, $r_i \geqslant 3t$
the external height, $H \geqslant 0{,}18\, D_o$, and is determined as follows:

$$H = R_o - \sqrt{(R_o - 0{,}5D_o)\,(R_o + 0{,}5D_o - 2r_o)}$$

4.1.4 In addition to the formula in 4.1.1 the thickness, t, is to be not less than that determined by the following formula:

$$t = \frac{pR_i}{20\sigma J - 0{,}5p} + 0{,}75 \text{ mm}$$

$$\left(t = \frac{pR_i}{2\sigma J - 0{,}5p} + 0{,}75 \text{ mm}\right)$$

where t, p, R_i, σ and J are as defined in 1.2.

4.1.5 In all cases, H is to be measured from the commencement of curvature, *see* Fig. 10.4.2.

4.1.6 The minimum thickness of the head, t, is to be not less than 9,5 mm. In special cases where it is proposed to use less than 9,5 mm thickness, the proposal will be the subject of special consideration.

4.1.7 For ends which are butt welded to the dr shell (*see* 1.8), the thickness of the edge of the flange connection to the shell is to be not less than thickness of an unpierced seamless or welded sh whichever is applicable, of the same diameter a material and determined by 2.1.

4.2 Shape factors for dished ends

4.2.1 The shape factor, K, to be used in 4.1.1 is to obtained from the curves in Fig. 10.4.1, and depends the ratio of height to diameter $\frac{H}{D_o}$.

4.2.2 The lowest curve in the series provides t factor K for plain (i.e. unpierced) ends. For lower valu of H/D_o, K depends upon the ratio of thickness t diameter, t/D_o, as well as on the ratio H/D_o, and a tr calculation may be necessary to arrive at the corre value of K.

4.3 Dished ends with unreinforced openings

4.3.1 Openings in dished ends may be circular approximately elliptical.

4.3.2 The upper curves in Fig. 10.4.1 provide valu of K to be used in 4.1.1, for ends with unreinforc openings. The selection of the correct curve depends the value $\frac{d}{\sqrt{D_o\, t}}$ and a trial calculation is necessary t select the correct curve, where

d = the diameter of the largest opening in the e plate, in mm (in the case of an elliptic opening, the larger axis of the ellipse),
t = minimum thickness, after dishing, in mm,
D_o = outside diameter of dished end, in mm.

4.3.3 The following requirements must in any cas be satisfied:

$$\frac{t}{D_o} \leqslant 0{,}1$$

$$\frac{d}{D_o} \leqslant 0{,}7$$

4.3.4 From Fig. 10.4.1 for any selected ratio of $\frac{t}{D}$ the curve for unpierced ends gives a value for $\frac{d}{\sqrt{D_o\, t}}$ well as for K. Openings giving a value of $\frac{d}{\sqrt{D_o\, t}}$ no greater than the value so obtained may thus be pierce through an end designed as unpierced without an increase in thickness.

4.4 Flanged openings in dished ends

4.4.1 The requirements in 4.3 apply equally flanged openings and to unflanged openings cut in plate of an end. No reduction may be made in end p thickness on account of flanging.

4.4.2 Where openings are flanged, the radius, r, the flanging is to be not less than 25 mm, *see* 10.4.2 (*d*). The thickness of the flanged portion may less than the calculated thickness.

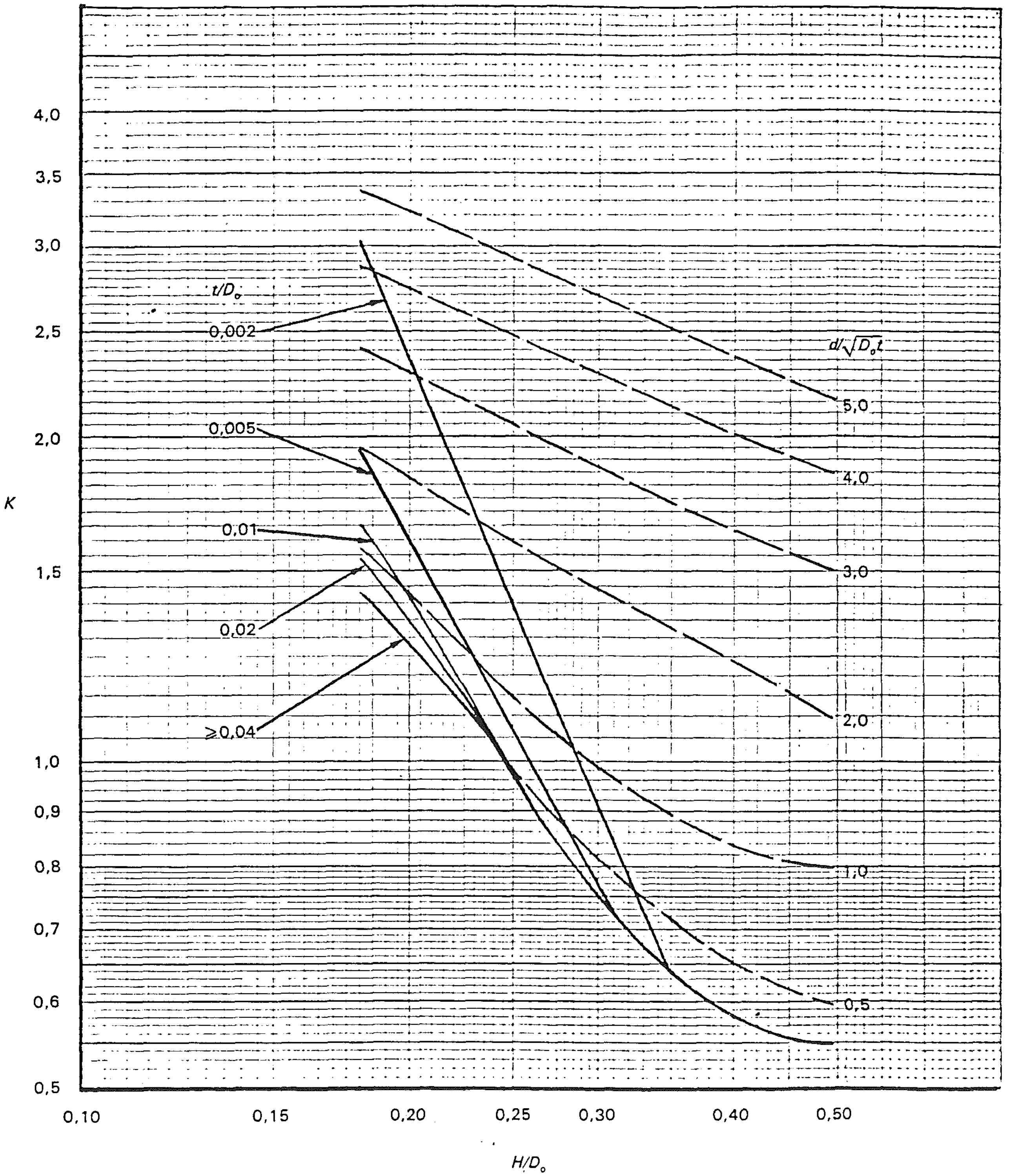
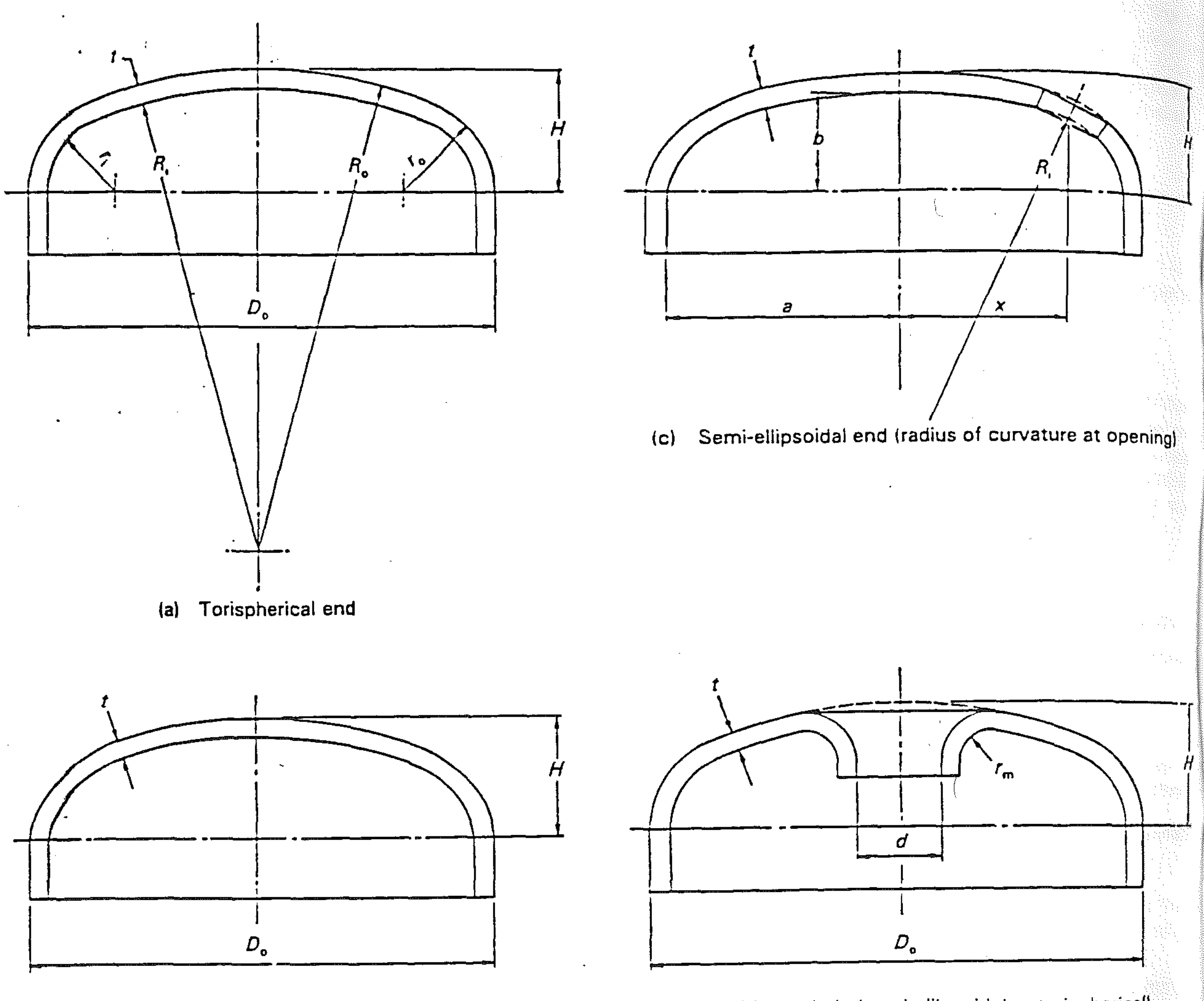


Fig. 10.4.1 Shape factor



(a) Torispherical end

(b) Semi-ellipsoidal end

(c) Semi-ellipsoidal end (radius of curvature at opening)

(d) End with manhole (semi-ellipsoidal or torispherical)

Fig. 10.4.2 Typical dished ends

4.5 Location of unreinforced and flanged openings in dished ends

4.5.1 Unreinforced and flanged openings in dished ends are to be so arranged that the distance from the edge of the hole to the outside edge of the plate and the distance between openings are not less than those shown in Fig. 10.4.3.

4.6 Dished ends with reinforced openings

4.6.1 Where it is desired to use a large opening on a dished end of less thickness than would be required by 4.3, the end is to be reinforced. This reinforcement may consist of a ring or standpipe welded into the hole, or of reinforcing plates welded to the outside and/or inside of the end in the vicinity of the hole, or a combination of both methods, *see* Fig. 10.4.4. Forged reinforcements may be used.

4.6.2 Reinforcing material within the following limits may be taken as effective reinforcement:

(a) The effective width l_1 of reinforcement is not to exceed $\sqrt{2R_i t}$ or $0,5d_c$ whichever is the lesser.

(b) The effective length l_2 of a reinforcing ring is not to exceed $\sqrt{d_c t_b}$

Where R_i = the internal radius of the spherical part of torispherical end, in mm, or

R_i = the internal radius of the meridian of the ellipse at the centre of the opening, of semi-ellipsoidal end, in mm, and is given by the following formula:

$$\frac{[a^4 - x^2(a^2 - b^2)]^{\frac{3}{2}}}{a^4 b}$$

where a, b and x are shown in Fig. 10.4.2(c),

t_b = actual thickness of ring or standpipe, in mm

d_c = external diameter of ring or standpipe, in mm

l_1 and l_2 are as shown in Fig. 10.4.4.

4.6.3 The shape factor, K, for a dished end having a reinforced opening can be read from Fig. 10.4.1 using the value obtained from:

$$\frac{d_c - \frac{A}{t}}{\sqrt{D_o t}} \text{ instead of from } \frac{d}{\sqrt{D_o t}}$$

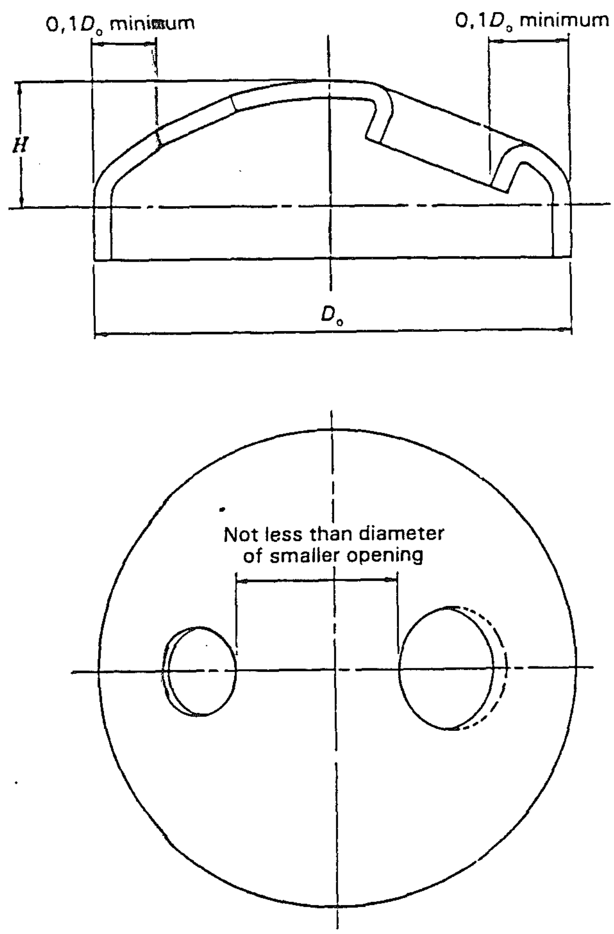
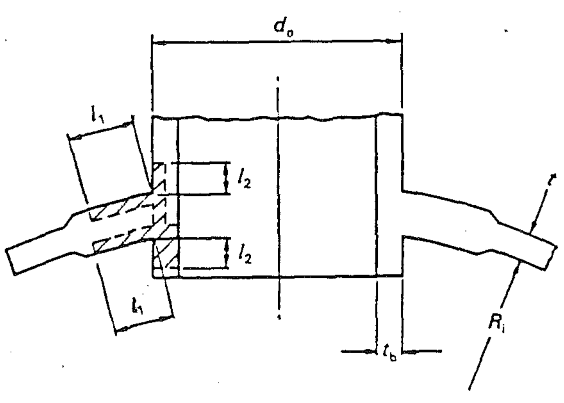


Fig. 10.4.3 **Opening in dished ends**



$l_1 = \sqrt{2R_i t}$ or 0,5d_o whichever is the lesser

l_2 is not to exceed $\sqrt{d_o t_b}$

Fig. 10.4.4 **Limits of reinforcement**

where A = the effective cross-sectional area of reinforcement and is to be twice the area shown shaded on Fig. 10.4.4.

As in 4.3, a trial calculation is necessary in order to select the correct curve.

4.6.4 The area shown in Fig. 10.4.4 is to be obtained as follows:

Calculate the cross-sectional area of reinforcement both inside and outside the end plate within the length l_1,

plus the full cross-sectional area of that part of the ring or standpipe which projects inside the end plate up to a distance l_2,

plus the full cross-sectional area of that part of the ring or standpipe which projects outside the internal surface of the end plate up to a distance l_2, and deduct the sectional area which the ring or standpipe would have if its thickness were as calculated in accordance with 7.1.

4.6.5 If the material of the ring or the reinforcing plates has an allowable stress value lower than that of the end plate, then the effective cross-sectional area, A, is to be multiplied by the ratio:

$$\frac{\text{allowable stress of reinforcing plate at design temperature}}{\text{allowable stress of end plate at design temperature}}$$

SECTION 5

Conical ends subject to internal pressure

5.1 General

5.1.1 Conical ends and conical reducing sections, as shown in Fig. 10.5.1, are to be designed in accordance with the equations given in 5.2.

5.1.2 Connections between cylindrical shell and conical sections and ends should preferably be by means of a knuckle transition radius. Typical permitted details are shown in Fig. 10.5.1. Alternatively, conical sections and ends may be butt welded to cylinders without a knuckle radius where the change in angle of slope, ψ, between the two sections under consideration does not exceed 30°.

5.1.3 Conical ends may be constructed of several ring sections of decreasing thickness, as determined by the corresponding decreasing diameter.

5.2 Minimum thickness

5.2.1 The minimum thickness, t, of the cylinder, knuckle and conical section at the junction and within the distance L from the junction is to be determined by the following formula:

$$t = \frac{pD_oK}{20\sigma J} + 0{,}75 \quad \text{mm}$$

$$\left(t = \frac{pD_oK}{2\sigma J} + 0{,}75 \quad \text{mm}\right)$$

where t, p, σ and J are as defined in 1.2,

D_o = outside diameter, in mm, of the conical section or end, *see* Fig. 10.5.1,

K = a factor, taking into account the stress in the knuckle, *see* Table 10.5.1.

5.2.2 If the distance of a circumferential seam from the knuckle or junction is not less than L, then J is to be taken as 1,0; otherwise J is to be taken as the weld joint factor appropriate to the circumferential seam,

where L = distance, in mm, from knuckle or junction within which meridional stresses determine the required thickness, *see* Fig. 10.5.1,

$$= 0{,}5\sqrt{\frac{D_o t}{\cos\psi}}$$

ψ = difference between angle of slope of two adjoining conical sections, *see* Fig. 10.5.1,
r_i = inside radius of transition knuckle, in mm, which is to be taken as $0{,}01D_c$ in the case of conical sections without knuckle transition.

5.2.3 The minimum thickness, t, of those parts of conical sections not less than a distance L from the junction with a cylinder or other conical section is to be determined by the following formula:

$$t = \frac{pD_c}{(20\sigma J - p)} \quad \frac{1}{\cos\alpha} + 0{,}75 \text{ mm}$$

$$\left(t = \frac{pD_c}{(2\sigma J - p)} \quad \frac{1}{\cos\alpha} + 0{,}75 \text{ mm}\right)$$

where D_c = inside diameter, in mm, of conical section or end at the position under consideration, *see* Fig. 10.5.1,
$\alpha, \alpha_1, \alpha_2$ = angle of slope of conical section (at the point under consideration) to the vessel axis, *see* Fig. 10.5.1.

5.2.4 The greater of the two thicknesses determined by the formulae in 5.2.1 and 5.2.2 is to apply.

5.2.5 The thicknesses of conical sections having an angle of inclination to the vessel axis of more than 75° is to be determined as for a flat plate.

SECTION 6

Standpipes and branches

6.1 Minimum thickness

6.1.1 The minimum wall thickness of standpipes and branches is to be not less than that determined by 7.1, making such additions as may be necessary on account of bending, static loads and vibration. The wall thickness, however, is to be not less than:

$$t = 0{,}04D_o + 2{,}5 \text{ mm}$$

where t and D_o are as defined in 1.2.

6.1.2 For boilers having a working pressure exceeding 50 bar (51 kgf/cm²) and safety valves of full lift or full bore type, the thickness of the branch pipe carrying the superheater or drum safety valves is to be not less than:

$$t = \frac{1}{\sigma}\left[1{,}7d + \frac{DWK}{1{,}3d^2}\right] \text{ mm}$$

$$\left(t = \frac{10{,}2}{\sigma}\left[1{,}7d + \frac{DWK}{1{,}3d^2}\right] \text{ mm}\right)$$

where t and σ are as defined in 1.2,
d = inside diameter of branch, in mm,
D = inside diameter of safety valve discharge, in mm,
W = total valve throughput, in kg/h,
K = 2 for superheater safety valves,
= 1 for drum safety valves.

6.1.3 The offset from the centre line of the waste steam pipe to the centre line of the safety valve is not to exceed four times the outside diameter of the safety valve discharge pipe. The waste steam pipe system is to be supported and arrangements made for expansion such that no direct loading is imposed on the safety valve chests and the effects of vibration are to be minimized.

6.1.4 The pipe or header which carries the superheater safety valve is to be suitably thickened but is to be not less than the thickness required for the branch for a distance of $\sqrt{D_2 t}$ on either side of the opening,
where D_2 = inside diameter of the pipe or header,
t = thickness required for the branch.

6.1.5 Except as required by 6.1.4, in no case need the wall thickness exceed that of the shell.

6.1.6 Where a standpipe or branch is connected by screwing, the thickness is to be measured at the root of the thread.

6.1.7 For boiler, superheater or economizer tubes, the minimum thickness of the drum or the header connection or tube stub is to be calculated as part of the tube in accordance with 7.1.

SECTION 7

Boiler tubes subject to internal pressure

7.1 Minimum thickness

7.1.1 The minimum wall thickness of straight tubes subject to internal pressure is to be determined by the following formula:

$$t = \frac{pD_o}{20\sigma + p} \text{ mm} \qquad \left(t = \frac{pD_o}{2\sigma + p} \text{ mm}\right)$$

where t, p, D_o and σ are as defined in 1.2.

NOTE 1. Provision must be made for minus tolerances where necessary and also in cases where abnormal corrosion or errosion is expected in service. For bending allowances, *see* 7.2.

NOTE 2. The thickness is in no case to be less than the minimum shown in Table 10.7.1.

Table 10.7.1 Minimum thicknesses of tubes

Nominal outside diameter of tube	Minimum thickness	*Minimum thickness
mm	mm	mm
⩽38	1,75	2,95
>38 ⩽50	2,16	
>50 ⩽70	2,40	
>70 ⩽75	2,67	3,28
>75 ⩽95	3,05	
>95 ⩽100	3,28	
>100⩽125	3,50	3,50

*Applicable to tubes subject to internal pressure and fitted in cylindrical boilers, and also for the tubes of low pressure water tube boilers having a design pressure of 17,2 bar (17,5 kgf/cm²) and under with open feed systems.

7.1.2 It is recommended that the thickness of tubes determined by the formula in 7.1.1 be increased by 0,25 mm for tubes subject to internal pressure and fitted in cylindrical boilers, and also tubes of low pressure water tube boilers having a design pressure of 17,2 bar (17,5 kgf/cm²) and under, with open feed systems.

7.1.3 The minimum thickness of boiler, superheater, reheater and economizer tubes is to be determined by using the design stress appropriate to the mean wall temperature, which will be considered to be the metal temperature. Unless it is otherwise agreed between the manufacturer and the Society, the metal temperature used to decide the value of σ for these tubes is to be determined as follows:

(a) The calculation temperature for boiler tubes is to be taken as not less than the saturated steam temperature, plus 25°C for tubes mainly subject to convection heat, or plus 50°C for tubes mainly subject to radiant heat.

(b) The calculation temperature for superheater and reheater tubes is to be generally taken as not less than the steam temperature expected in the part being considered, plus 35°C for tubes mainly subject to convection heat. For tubes mainly subject to radiant heat the calculation temperature is generally to be taken as not less than the steam temperature expected in the part being considered, plus 50°C, but the actual metal temperature expected is to be stated when submitting plans.

(c) The calculation temperature for economizer tubes is to be taken as not less than 35°C in excess of the maximum temperature of the internal fluid.

7.1.4 The minimum thickness of downcomer tubes and pipes which form an integral part of the boiler and which are not exposed to combustion gases is to comply with the requirements for steam pipes.

7.2 Tube bending

7.2.1 Where boiler, superheater, reheater and economizer tubes are bent, the resulting thickness of the tubes at the thinnest part is to be not less than that required for straight tubes, unless it can be demonstrated that the method of forming the bend results in no decrease in strength at the bend. The manufacturer is to demonstrate in connection with any new method of tube bending that this condition is satisfied.

7.2.2 Tube bending, and subsequent heat treatment, where necessary, is to be carried out as to ensure that residual stresses do not adversely affect the strength of the tube for the design purpose intended.

CROSS-REFERENCES

For details of manholes, sight holes and doors, *see* 14.1.
For details of tube holes and fitting of tubes, *see* 14.9.

SECTION 8
Headers

8.1 Circular section headers

8.1.1 The minimum thickness of circular section headers is to be calculated in accordance with the formula for cylindrical shells in 2.1.

8.2 Rectangular section headers

8.2.1 The thickness of flat surfaces of rectangular solid forged headers is to be not less than $(t + 0{,}75)$ mm where t = the greatest basic thickness, in mm, derived by the use of Fig. 10.8.1.

8.2.2 Fig. 10.8.1 shows values of $\dfrac{t}{B}$ corresponding to values of a term K, for parameters of $\dfrac{A}{B}$

where A = the distance, in mm, between the centreline of the openings and the limit of the effective width, B, of the header. Where there is more than one row of holes, A is the distance to the row showing the lowest efficiency,

B = the effective width, in mm, of the pierced surface under consideration measured between the supporting sides of the headers, minus one corner radius. This effective width is to be taken as not less than 0,9 of the full distance between the sides,

$$K = 10\frac{\sigma J}{p} \qquad \left(\frac{\sigma J}{p}\right)$$

where p and σ are as defined in 1.2,

J = the efficiency of the ligaments as calculated in 2.2. Where there are several rows of tube holes the lowest calculated efficiency is to be used.

8.2.3 Investigations of two stresses are necessary:

the stress at the corner of the header, where $\dfrac{A}{B} = 0$ and $J = 1$, *see* Fig. 10.8.1, and
the stress in the ligaments between tube holes or other openings piercing the flat face of the header.

8.2.4 The corner radius is to be not less than 6,5 mm.

8.2.5 Where the header surfaces are machined locally at hand holes, the total thickness may be reduced by a maximum of 4 mm.

8.2.6 Except for small areas not exceeding 3,25 cm^2, where a reduction of designed thickness up to 50 per cent may be permitted, the thickness derived from use of Fig. 10.8.1 is to be the minimum. Such minimum is in no case to be less than 7,5 mm or, where tube holes are drilled, to be less than:

$$t = 0{,}5\sqrt{d} + 6{,}35 \text{ mm}$$

where d = the diameter of the tube hole, in mm.

8.3 Header ends

8.3.1 The shape and thickness of ends forged integrally with the bodies of headers are to be the subject of special consideration.

8.3.2 Where sufficient experience of previous satisfactory service of headers with integrally forged ends cannot be shown, the suitability of a proposed form of end is to be proved in accordance with the provisions of 1.10.

8.3.3 Ends attached by welding are to be designed as follows:

Dished ends: these are to be in accordance with 4.1.

Flat ends: the minimum thickness of flat end plates is to be determined by the following formula:

$$t = d\sqrt{\frac{pC}{10\sigma}} \qquad \left(t = d\sqrt{\frac{pC}{\sigma}}\right)$$

where p and σ are as defined in 1.2.

10.5.2 The inside radii of dishing and flanging are to be as required by 10.3.

10.6 Dished and flanged ends for unsupported vertical boiler furnaces

10.6.1 The minimum thickness, t, of dished and flanged ends for vertical boiler furnaces that are subject to pressure on the convex side and are without support from stays of any kind, is to be determined by the following formula, but is in no case to be less than the thickness of the firebox:

$$t=\frac{CpR_o}{660}+0{,}75 \quad \text{mm}$$

$$\left(t=\frac{CpR_o}{675}+0{,}75 \quad \text{mm}\right)$$

where t and p are as defined in 1.2,

R_o = outside radius of the crown plate, in mm (in no case is $\frac{R_o}{t}$ to exceed 88),

$C=\frac{2x}{x+\sigma}$ or 0,85 whichever is the greater,

x = specified minimum 0,2 per cent proof stress in N/mm^2 (kgf/mm^2), at a temperature 90°C above the saturated steam temperature corresponding to the design pressure for carbon and carbon-manganese steel with a specified minimum tensile strength of 400 N/mm^2 (41 kgf/mm^2),

σ = specified minimum 0,2 per cent proof stress in N/mm^2 (kgf/mm^2), at a temperature 90°C above the saturated steam temperature corresponding to the design pressure for the steel actually used.

10.6.2 The inside radius of curvature, R_i, of the end plate is to be not greater than the external diameter of the cylinder to which it is atttached.

10.6.3 The inside knuckle radius, r_i, *see* Fig. 10.4.2 (a), of the arc joining the cylindrical flange to the spherical surface of the end is to be not less than four times the thickness of the end plate and in no case less than 65 mm.

SECTION 11

Cylindrical furnaces subject to external pressure

11.1 Maximum thickness

11.1.1 Furnaces, plain or corrugated, are not to exceed 22,5 mm in thickness.

11.2 Corrugated furnaces

11.2.1 The minimum thickness, t, of corrugated furnaces is to be determined by the following formula:

$$t=\frac{pD_o}{C}+0{,}75 \quad \text{mm}$$

where p is as defined in 1.2,

D_o = external diameter of the furnace measured at the bottom of the corrugations, in mm,

t = thickness of the furnace plate measured at the bottom of the corrugations, in mm,

C = 1060 (1080) for Fox, Morison and Deighton corrugations,

= 1130 (1150) for Suspension Bulb corrugations.

11.3 Plain furnaces, flue sections and combustion chamber bottoms

11.3.1 The minimum thickness, t, of plain furnaces, furnaces strengthened by stiffening rings, of flue sections and of the cylindrical bottoms of combustion chambers is to be determined by the following formulae, the greater of the two thicknesses obtained being taken,

$$t=\sqrt{\frac{pD_o(L+610)}{102\,400}}+0{,}75 \quad \text{mm}$$

$$\left(t=\sqrt{\frac{pD_o(L+610)}{104\,400}}+0{,}75 \quad \text{mm}\right)$$

$$t=\frac{CpD_o}{1100}+\frac{L}{320}+0{,}75 \quad \text{mm}$$

$$\left(t=\frac{CpD_o}{1120}+\frac{L}{320}+0{,}75 \quad \text{mm}\right)$$

where t and p are as defined in 1.2,

D_o = external diameter of the furnace, flue or combustion chamber, in mm,

L = length of section between the centres of points of substantial support, in mm,

$C=\frac{2x}{x+\sigma}$

x and σ are as defined in 10.6.

11.4 Plain furnaces of vertical boilers

11.4.1 The thickness of plain furnaces not exceeding 1700 mm in external diameter is to be determined by the formulae given in 11.3.1, the greater of the two thicknesses being taken,

where D = external diameter of the furnace, in mm. Where the furnace is tapered, the diameter to be taken for calculation purposes is to be the mean of that at the top and that at the bottom where it meets the substantial support from flange, ring or row of stays.

L = effective length, in mm, of the furnace between the points of substantial support as indicated in Fig. 10.11.1.

11.4.2 For furnaces under 760 mm in external diameter, the thickness is to be not less than 8 mm, and for furnaces 760 mm in external diameter and over, the thickness is to be not less than 9,5 mm. Furnaces exceeding 1700 mm in external diameter will be the subject of special consideration.

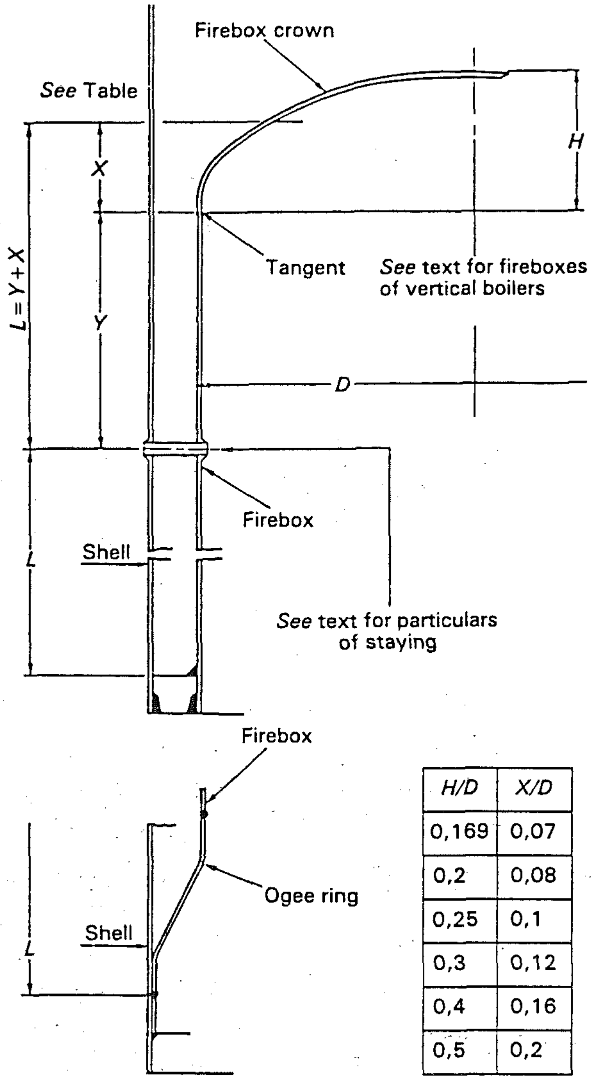
11.4.3 A circumferential row of stays connecting the furnace to the shell will be considered to provide substantial support to the furnace, provided that:

the diameter of the stay is not less than 22,5 mm or twice the thickness of the furnace, whichever is the greater. In the case of screwed stays the diameter is to be measured over the threads.

the pitch of the stays at the furnace does not exceed 14 times the thickness of the furnace.

11.5 Hemispherical furnaces

11.5.1 The minimum thickness, t, of unsupported hemispherical furnaces subject to pressure on the convex surface is to be determined by the following formula:



H/D	X/D
0,169	0,07
0,2	0,08
0,25	0,1
0,3	0,12
0,4	0,16
0,5	0,2

Fig. 10.11.1 Effective length, L, for use in 11.4

$$t = \frac{CpR_o}{608} + 0{,}75 \quad \text{mm}$$

$$\left(t = \frac{CpR_o}{620} + 0{,}75 \quad \text{mm}\right)$$

where t and p are as defined in 1.2,

R_o = outer radius of curvature of the furnace, in mm,

$C = \dfrac{2x}{x+\sigma}$ or 0,85 whichever is the greater,

x and σ are as defined in 10.6.

11.5.2 In no case is the minimum thickness, t, to exceed 22,5 mm, or the ratio $\dfrac{R_o}{t - 0{,}75}$ to exceed 100.

11.6 Ogee rings

11.6.1 The minimum thickness, t, of the ogee ring which connects the bottom of the furnace to the shell of a vertical boiler and sustains the whole vertical load on the furnace is to be determined by the following formula:

$$t = \sqrt{\frac{pD_i(D_i - D_o)}{9\,900}} + 0{,}75 \quad \text{mm}$$

$$\left(t = \sqrt{\frac{pD_i(D_i - D_o)}{10\,110}} + 0{,}75 \quad \text{mm}\right)$$

where t and p are as defined in 1.2,

D_i = inside diameter of boiler shell, in mm,

D_o = outside diameter of the lower part of the furnace where it joins the ogee ring, in mm.

11.7 Uptakes of vertical boilers

11.7.1 The minimum thickness, t, of internal uptakes of vertical boilers is to be determined by the following formulae, the greater of the two thicknesses obtained being taken:

$$t = \sqrt{\frac{pD_o(L + 610)}{102\,400}} + 4 \quad \text{mm}$$

$$\left(t = \sqrt{\frac{pD_o(L + 610)}{104\,400}} + 4 \quad \text{mm}\right)$$

$$t = \frac{pD_o}{1100} + \frac{L}{320} + 4 \quad \text{mm}$$

$$\left(t = \frac{pD_o}{1120} + \frac{L}{320} + 4 \quad \text{mm}\right)$$

where t and p are as defined in 1.2,

D_o = external diameter of uptake, in mm,

L = length of uptake between the centres of points of substantial support, in mm.

SECTION 12

Boiler tubes subject to external pressure

12.1 Plain tubes

12.1.1 The thickness of plain tubes is to be in accordance with Table 10.12.1 for the appropriate outside diameter and design pressure.

12.1.2 Plain tubes may be seal welded at both ends, seal welded at the inlet end and expanded at the outlet end, or expanded at both ends. Where plain tubes are seal welded, the weld detail is to be shown in Fig. 10.9.4 and the tubes are to be expanded into the tube plates in addition to welding.

12.1.3 Where plain tubes are expanded only, the process is to be carried out with roller expanders, and the expanded portion of the tube is to be parallel through the full thickness of the tube plate. In addition to expanding, tubes may be bell-mouthed or beaded at the inlet end.

12.1.4 Where the total number of tubes is arranged in one nest and no stay tubes are fitted (*see* 9.2.1), the ends of all tubes are to be welded or expanded and beaded at the inlet end, and welded or expanded at the outlet end.

12.2 Cross tubes

12.2.1 Cross tubes are not to exceed 300 mm internal diameter. The minimum thickness, t, is to be determined by the following formula, but is in no case to be less than 9,5 mm:

Table 10.12.1 Thickness of plain tubes under external pressure

Design pressure, in bar (kgf/cm²)											
Outside diameter, in mm											Thickness,
38	44,5	51	57	63,5	70	76	82,5	89	95	102	in mm
–	–	–	–	–	–	–	–	–	26,9 (27,4)	25,2 (25,7)	5,89
–	–	–	–	–	–	–	26,2 (26,7)	24,1 (24,6)	22,8 (23,2)	21,4 (21,8)	5,38
–	–	–	–	–	–	24,1 (24,6)	22,1 (22,5)	20,7 (21,1)	19,3 (19,7)	17,9 (18,3)	4,88
–	–	–	27,6 (28,1)	24,8 (25,3)	22,8 (23,2)	20,7 (21,1)	19,3 (19,7)	17,9 (18,3)	16,6 (16,9)	15,9 (16,2)	4,47
–	29,3 (29,9)	25,5 (26,0)	22,8 (23,2)	20,7 (21,1)	18,9 (19,3)	17,3 (17,6)	15,9 (16,2)	14,8 (15,1)	13,7 (14,0)	12,7 (13,0)	4,06
26,6 (27,1)	22,8 (23,2)	20,7 (21,1)	17,9 (18,3)	15,9 (16,2)	14,8 (15,1)	13,1 (13,4)	12,4 (12,6)	11,4 (11,6)	10,3 (10,5)	9,6 (9,8)	3,66
20,3 (20,7)	16,9 (17,2)	14,8 (15,1)	13,1 (13,4)	12,1 (12,3)	11,0 (11,2)	9,6 (9,8)	8,9 (9,1)	8,2 (8,4)	7,6 (7,7)	6,9 (7,0)	3,25
14,8 (15,1)	12,4 (12,6)	10,7 (10,9)	9,6 (9,8)	8,6 (8,8)	7,6 (7,7)	–	–	–	–	–	2,95

$$t = \frac{pD_i}{440} + 6{,}5 \quad \text{mm}$$

$$\left(t = \frac{pD_i}{450} + 6{,}5 \quad \text{mm}\right)$$

where t and p are as defined in 1.2,

D_i = internal diameter of cross tube, in mm.

12.3 Pitch of tubes

12.3.1 The spacing of tube holes is to be such that the minimum width, in mm, of any ligament between the tube holes is not less than:

$$0{,}125d + 12{,}5 \quad \text{mm}$$

where d = diameter of the tube hole, in mm.

SECTION 13

Stay tubes and bar stays for cylindrical boilers

13.1 Stay tubes

13.1.1 Each stay tube is to be designed to carry its due proportion of the load on the plates which it supports. No stay tube is to be less than 5 mm thick at its thinnest part.

13.1.2 The thickness of stay tubes welded to tube plates is to be such that the maximum stress on the thinnest part of the tube does not exceed 69 N/mm² (705 kgf/cm²).

13.1.3 Welded-in stay tubes are to be expanded into the tube plate in addition to welding. Typical examples of welded stay tube attachments are shown in Fig. 10.9.5.

13.1.4 Stay tubes may be welded into the boiler after stress relief, provided they are not adjacent in the same tube nest.

13.2 Combustion chamber and longitudinal b stays

13.2.1 The permissible stress in combustion chamb and other similar stays, calculated on the minimu sectional area, is not to exceed 62 N/mm² (633 kgf/cm

13.2.2 The diameter of any stay is to be not less tha 19 mm.

13.2.3 The permissible stress in longitudinal stay calculated on the minimum cross-sectional area, is n to exceed:

$$\frac{\text{minimum specified tensile strength, in N/mm}^2\ (\text{kgf/cm}^2)}{5{,}3}$$

13.2.4 In no case is the diameter of the longitudinal st at any section to be less than 25 mm.

13.3 Loads on stay tubes and bar stays

13.3.1 Stay tubes and bar stays are to be designed t carry the whole load due to the pressure on the area to b supported.

13.3.2 For a stay tube or bar stay, the net area to b supported is to be the area, in mm², enclosed by the line bisecting at right angles the lines joining the stay and th adjacent points of support, less the area of any tubes stays embraced. In the case of a stay tube or bar stay the boundary rows, the support afforded by the flat pla margin, where applicable, should be taken into accoun

13.3.3 Where there are no stay tubes in the tube nes the area to be supported by a bar stay is to extend to th tangential boundary of the tube nest.

15.2 Safety valves

15.2.1 Boilers and steam generators are to be fitted with not less than two safety valves, each having a minimum internal diameter of 38 mm if of ordinary type or 25 mm if of high lift or full lift type, but those having a total heating surface of less than 50 m^2 may have one valve not less than 50 mm diameter.

15.2.2 The valves, spindles, springs and compression screws are to be so encased and locked or sealed that the safety valves and pilot valves, after setting to the working pressure, cannot be tampered with or overloaded in service; the spring casing of superheater safety valves should be ventilated, or other arrangement provided to protect the springs from excessive temperature.

15.2.3 Valves are to be so designed that in the event of fracture of springs they cannot lift out of their seats.

15.2.4 Easing gear is to be provided for lifting the safety valves and is to be operable by mechanical means at a safe position from the boiler or engine room platforms.

15.2.5 Safety valves are to be made with working parts having adequate clearances to ensure complete freedom of movement.

15.2.6 Valve seats are to be effectively secured in position. Any adjusting devices which control discharge capacity are to be positively secured so that the adjustment will not be affected when the safety valves are dismantled at surveys.

15.2.7 All the safety valves of each boiler and steam generator may be fitted in one chest, which is to be separate from any other valve chest and is to be connected directly to the shell by a strong and stiff neck, the passage through which is to be of cross-sectional area not less than the aggregate area of the safety valves in the chest in the case of full lift valves, and one-half of that area in the case of other valves. For the meaning of aggregate area, *see* 15.2.11.

15.2.8 Each safety valve chest is to be drained by a pipe fitted to the lowest part and led with a continuous fall to the bilge or to a tank, clear of the boilers. No valves or cocks are to be fitted to these drain pipes. It is recommended that the bore of the drain pipes be not less than 19 mm.

15.2.9 Safety valves for shell type exhaust gas steaming economizers are to incorporate fail safe features which will ensure operation of the valve even with solid matter deposits on the valve and guide. Alternatively, a bursting disc discharging to a suitable waste steam pipe is to be fitted. These emergency devices are to function at a pressure not exceeding 1,5 times the economizer approved design pressure. Full particulars of the proposed arrangements are to be submitted for consideration.

15.2.10 Where the receiver is fitted with safety valves to relieve the steam output of the economizer and the economizer cannot be isolated from the receiver the requirements of 15.2.9. may be waived.

15.2.11 The designed discharge capacities of the safety valves on each boiler and steam generator are to be found from the following formulae:

Saturated steam safety valves:

$$E = \frac{AC(p+1{,}03)}{98{,}1}$$

$$\left(E = \frac{AC(p+1{,}05)}{100}\right)$$

Superheated steam safety valves:

$$E = \frac{AC(p+1{,}03)}{98{,}1}\sqrt{\frac{V_S}{V_H}}$$

$$\left(E = \frac{AC(p+1{,}05)}{100}\sqrt{\frac{V_S}{V_H}}\right)$$

where E = the maker's specified peak load evaporation, in kg/hour (including all evaporation from water walls, integral, or steaming economizers and other heating surfaces in direct communication with the boiler). In no case is the designed evaporation to be based on less than 29 kg/m^2 hour of heating surface for fired boilers and 14,5 kg/m^2 hour for exhaust gas heated boilers,

A = for ordinary, high lift or improved high lift safety valves, the aggregate area, in mm^2, of the orifices through the seatings of the valves, neglecting the area of guides and other obstructions,

= for full lift safety valves, the net aggregate area, in mm^2, through the seats after deducting the area of the guides or other obstructions when the valves are fully lifted,

C = 4,8 for valves of ordinary type having a minimum lift of $\frac{D}{24}$,

= 7,2 for valves of high lift type, having a minimum lift of $\frac{D}{16}$,

= 9,6 for valves of improved high lift type having a minimum lift of $\frac{D}{12}$,

= 19,2 for valves of full lift type having a minimum lift of $\frac{D}{4}$,

D = bore of valve seat, in mm,

p = design pressure, in bar (kgf/cm^2) gauge,

V_S = specific volume of saturated steam (m^3/kg),

V_H = specific volume of superheated steam (m^3/kg).

15.2.12 When the discharge capacity of a safety valve of approved design has been established by type tests, carried out in the presence of the Surveyors or by an independent authority recognized by the Society, on valves representative of the range of sizes and pressures intended for marine application, consideration will be given to the use of a constant higher than $C = 19{,}2$, based on 90 per cent of the measured capacity up to a maximum of $C = 45$ for full lift safety valves.

15.3 Waste steam pipes

15.3.1 For ordinary, high lift and improved high lift type valves, the cross-sectional area of the waste steam pipe and passages leading to it is to be at least 10 per cent greater than the aggregate area of the safety valves as used in the formulae in 15.2.11. For full lift and other approved valves of high discharge capacity, the cross-sectional area of the waste steam pipe and passages is to be not less than 0,1*C* times the aggregate valve area.

15.3.2 The cross-sectional area of the main waste steam pipe is to be not less than the combined cross-sectional areas of the branch waste steam pipes leading thereto from the boiler safety valves.

15.3.3 Waste steam pipes are to be led to the atmosphere and are to be adequately supported and provided with suitable expansion joints, bends or other means to relieve the safety valve chests of undue loading.

15.3.4 The scantlings of waste steam pipes and silencers are to be suitable for the maximum pressure to which the pipes may be subjected in service, and in any case not less than 10 bar (10,2 kgf/cm^2).

15.3.5 Silencers fitted to waste steam pipes are to be so designed that the clear area through the baffle plates is not less than that required for the pipes.

15.3.6 The safety valves of each exhaust gas heated economizer and exhaust gas heated boiler which may be used as an economizer are to be provided with entirely separate waste steam pipes.

15.3.7 External drains and exhaust steam vents to atmosphere are not to be led to waste steam pipes.

15.3.8 It is recommended that a scale trap and means for cleaning be provided at the base of each waste steam pipe.

15.4 Adjustment and accumulation tests

15.4.1 All safety valves are to be set under steam to a pressure not greater than the approved design pressure of the boiler. As a working tolerance the setting is acceptable provided the valves lift at not more than 103 per cent of the approved design pressure. During a test of 15 minutes with the stop valves closed and under full firing conditions the accumulation of pressure is not to exceed 10 per cent of the design pressure. During this test no more feed water is to be supplied than is necessary to maintain a safe working water level.

15.5 Stop valves

15.5.1 One main stop valve is to be fitted to each boiler and secured directly to the shell. There are to be as few auxiliary stop valves as possible so as to avoid piercing the boiler shell more than is absolutely necessary.

15.5.2 Where two or more boilers are connected together:

Stop valves of self-closing or non-return type are to be fitted.

Essential services are to be capable of being supplied from at least two boilers.

15.6 Water level indicators

15.6.1 Every boiler is to be fitted with at least two independent means of indicating the water level in it, one of which is to be a glass gauge. The other means is to be either an additional glass gauge or an approved equivalent device.

15.6.2 A set of not less than two test cocks will be accepted as the approved equivalent device mentioned in 15.6.1 for boilers having a design pressure less than 8,2 bar (8,4 kgf/cm^2) or an internal diameter less than 1,83 m. The test cocks are to be fitted, where practicable, direct to the boiler plating.

15.6.3 The water gauges are to be readily accessible and placed so that the water level is clearly visible. The lowest visible part of the glass of the water gauge and the lower test cock, where test cocks are fitted, are to be situated at the lowest safe working water level.

15.6.4 The level of the highest part of the effective heating surfaces, e.g. combustion chamber top of a horizontal boiler and the furnace crown of a vertical boiler, is to be clearly marked in a position adjacent to the glass water gauge.

15.6.5 The cocks of all water gauges are to be operable from positions free from danger in the event of the glass breaking.

15.6.6 If the water gauges are not fitted directly to the shell of the boiler, but to stand pillars or columns, it is desirable that these pillars or columns should be bolted directly to the shell of the boiler. If they are connected to the boiler by means of pipes, the pipes are to be fitted with terminal cocks, not valves, secured direct to the boiler shell. For boilers exceeding 3 m in diameter, the pillars are to be not less than 63 mm, and the connecting pipes not less than 38 mm internal diameter. For boilers exceeding 2,3 m but not exceeding 3 m in diameter, the pillars are to be not less than 50 mm and the pipes not less than 32 mm internal diameter. For boilers 2,3 m in diameter and under, the pillars are to be not less than 45 mm and the pipes not less than 25 mm internal diameter. The upper ends of the connecting pipes are to be so arranged that there is no pocket or bend where an accumulation of water from the condensation of the steam can lodge. They should not pass through the uptake if they can be otherwise arranged. If, however, this condition cannot be complied with, they may pass through it by means of a passage at least 50 mm clear of the pipe all round, open for ventilation.

15.7 Low water level fuel shut-off and alarm

15.7.1 Each fired boiler is to be fitted with a system of water level detection which is to be independent of any other mounting and which will operate audible and visible alarms and shut off automatically the fuel supply to the burners when the water level falls to a predetermined low level.

15.8 Feed check valves

15.8.1 Two feed check valves, connected to separate feed lines, are to be provided for all main and auxiliary boilers which are required for essential service, with the exception of boilers in which steam is generated exclusively by exhaust gas or steam, where one feed check valve will be accepted. *See* Ch 14,6.

15.8.2 The feed check valves are to be attached wherever practicable, direct to the boiler, but where the arrangements necessitate the use of standpipes between the boiler and the check valves, these pipes are to be of steel or other approved material. For boiler feed water systems, *see* Ch 14,6.

15.9 Pressure gauges

15.9.1 Each boiler is to be provided with a separate steam pressure gauge.

15.9.2 The gauges are to be placed where they are easily read.

15.10 Blow-down and scum valves

15.10.1 Each boiler is to be fitted with at least one blow-down valve secured direct to the lower part of the boiler.

15.10.2 Where it is not practicable to attach the blow-down valve direct to water tube boilers, the valve may be placed immediately outside the boiler casing with a steel pipe of substantial thickness fitted between the boiler and valve. The pipe and valve are to be suitably supported and any pipe which may be exposed to direct heat from the furnace is to be adequately protected.

15.10.3 The blow-down valve and its connections to the sea need not be more than 38 mm, and is not to be less than 19 mm internal diameter. For cylindrical boilers the size of the valve may be generally 0,0085 times the diameter of the boiler.

15.10.4 Blow-down valves and scum valves (where the latter are fitted) of two or more boilers may be connected to one common discharge, but where thus arranged there are to be screw-down non-return valves fitted for each boiler to prevent the possibility of the contents of one boiler passing to another.

15.10.5 For blow-down valves or cocks on the ship's side and attachments, *see* Ch 13,2.

15.11 Salinometer valve or cock

15.11.1 Each boiler is to be provided with a salinometer valve or cock secured direct to the boiler in a convenient position. The valve or cock is not to be on the water gauge standpipe.

SECTION 16

Mountings and fittings for water tube boilers

16.1 General

16.1.1 Mountings and fittings not mentioned in this Section are to be in accordance with the requirements in Section 15.

16.2 Safety valves

16.2.1 Water tube boilers are to be fitted with not less than two safety valves of area and design in general accordance with the requirements of 15.2, except that the minimum diameter of high discharge type valves may be 25 mm or equivalent free area.

16.2.2 Each saturated steam drum and each superheater are to be provided with at least one safety valve.

16.2.3 Where the superheater forms an integral part of the boiler, the relieving capacity of the superheater safety valve(s), based on the reduced pressure at the superheater outlet, may be included as part of the total relieving capacity required for the boiler. As some National Authorities limit the proportion of the superheater safety valve relieving capacity which may be credited towards the total capacity for the boiler, builders should give attention to any relevant statutory requirements of the National Authority of the country in which the ship is to be registered.

16.2.4 The boiler and superheater valves are to be so disposed and proportioned between saturated steam drum and superheater outlet that the superheater will be protected from overheating under all service conditions, including an emergency stop of the ship at full power.

16.2.5 Where it is proposed to fit full bore safety valves operated by independent pilot valves, the arrangements are to be submitted for consideration. The pipes connecting pilot valves and main valves are to be of ample bore and wall thickness to minimize the possibility of obstruction and damage.

16.2.6 Where it is impracticable to attach safety valves directly to the superheater, the valves are to be located as near as possible thereto and fitted to a branch piece connected to the superheater outlet pipe.

16.2.7 In high temperature installations the drains from safety valves are to be led to a tank or other place where high temperature steam can be safely discharged.

16.3 Safety valve settings

16.3.1 All boiler and superheater safety valves are to be set under steam to their respective working pressures, which are not to be greater than the approved design pressure of the boiler. As a working tolerance the setting is acceptable provided the valves lift at not more than 103 per cent of the approved pressure.

16.3.2 In the setting of superheater safety valves, allowance is to be made for the pressure drop through the superheater so that under discharge conditions the pressure in the boiler will not exceed the approved boiler pressure.

16.3.3 In no case is the superheater safety valve setting to exceed by more than 3 per cent the pressure for which the steam piping is approved.

16.4 Waste steam pipes

16.4.1 The waste steam pipe and passages leading to it from the safety valves are to be in general accordance with the requirements of 15.3.

16.4.2 In installations operating with a high degree of superheat, consideration is to be given to the high temperatures which waste steam pipes, silencers and surrounding spaces will attain when the superheater safety valves are blowing during accumulation tests and in service, adequate protection against heat effects is to be provided to the Surveyor's satisfaction.

16.4.3 Waste steam pipes are to be led well clear of electric cables and any parts or structures sensitive to heat or likely to distort; the pipes are to be insulated where necessary. In these installations each boiler should have a separate waste steam pipe system to atmosphere, with supporting and expansion arrangements such that no direct loading is imposed on the safety valve chests.

16.5 Accumulation tests

16.5.1 Tests for accumulation of pressure are to be carried out with the stop valve closed and under full firing conditions for a period not exceeding 7 minutes. The accumulation is not to exceed 10 per cent of the design pressure.

16.5.2 Where accumulation tests might endanger the superheaters, consideration will be given in cases of fired boilers to the omission of these tests, provided that application is made when the boiler plan and sizes of safety valves are submitted for approval, and that the safety valves are of an approved type for which the capacity has been established by test in the presence of the Surveyors or an approved independent authority, or for which the Society is satisfied, by long experience of accumulation tests, that the capacity is adequate. When it is agreed to waive accumulation tests, it will be required that the valve makers provide a certificate for each safety valve, stating its rated capacity at the approved working conditions of the boilers and that the boiler makers provide a certificate for each boiler stating its maximum evaporation.

16.5.3 The safety valves are to be found satisfactory in operation under working conditions during the trials of the machinery on board ship.

16.6 Water level indicators

16.6.1 Every boiler is to be fitted with at least two independent means of indicating the water level in it, one of which is to be a glass water gauge. The other means is to be either an additional glass water gauge, or an approved equivalent device other than test cocks.

16.6.2 Where a steam and water drum exceeding 4 m in length is fitted athwartships, two glass water gauges are to be fitted in suitable positions, one near each end of the drum.

16.6.3 The position of the glass water gauges of boilers in which the tubes are entirely drowned when cold is to be such that water is just showing in the glass when the water level in the steam drum is just above the top of the uppermost tubes when the boiler is cold.

16.6.4 In boilers, the tubes of which are not entirely drowned when cold, the glass water gauges are to be placed, to the Surveyor's satisfaction, in the positions which have been found by experience to indicate satisfactorily that the water content is sufficient for safety when the boiler is worked under all service conditions.

16.7 Low water level fuel shut-off and alarm

16.7.1 Each fired boiler is to be fitted with two systems of water level detection which are to be independent of each other and of any other mounting on the boiler. Both systems are to operate audible and visible alarms and shut off automatically the fuel supply to the burners when the water level falls to a predetermined low level.

16.7.2 Any proposals to depart from these requirements in the case of small auxiliary boilers will be the subject of special consideration.

16.7.3 For 'UMS' notation, *see* Pt 6, Ch 1.

16.8 Feed check valves and water level regulators

16.8.1 Two feed check valves, connected to separate feed lines, are to be provided for each boiler and are to be attached, wherever practicable, direct to the boiler or to an economizer which forms an integral part of the boiler.

16.8.2 Where the arrangements necessitate the use of a common inlet pipe on the economizer for both main and auxiliary feed systems, this pipe is to be as short as practicable, and the arrangements of check valves is to be such that either feed line can be effectively isolated without interruption of the feed water supply to the boiler.

16.8.3 At least one of the feed water systems is to be fitted with an approved feed water regulator whereby the water level in the boilers is controlled automatically. *See* Ch 14,6 for arrangements and details of boiler feed systems.

16.8.4 The feed check valves are to be fitted with efficient gearing, whereby they can be satisfactorily worked from the stokehold floor, or other convenient position.

16.8.5 Standpipes on boilers, for feed inlets, are to be designed with an internal pipe to prevent direct contact between the feed pipe and the boiler shell or end plate with the object of minimizing thermal stresses in these plates. Similar arrangements are to be provided for desuperheater and other connections where significant temperature differences occur in service.

SECTION 17

Hydraulic tests

17.1 General

17.1.1 Boilers and pressure vessels, together with their components, are to withstand the following hydraulic tests without any sign of weakness or defect.

17.1.2 Having regard to the variation in the types and design of boilers, the hydraulic test may be carried out by either of the methods indicated below:

(a) boilers are to be tested on completion to a pressure 1,5 times the approved design pressure, or

(b) where construction permits, all components of the boiler are to be tested on completion of the work including heat treatment to 1,5 times the design pressure. In the case of components such as drums or headers, which are to be drilled for tube holes, the test may be made before drilling the tube holes, but is to be after the attachment of standpipes, stubs and similar fittings and also after heat treatment has been carried out. Where all the components have been tested as above, each completed boiler after assembly is to be tested to 1,25 times the design pressure.

17.2 Mountings

17.2.1 All boiler mountings are to be subjected to a hydraulic test of twice the approved design pressure with the exception of feed check valves and other mountings connected to the main feed system which are to be tested to 2,5 times the approved boiler design pressure, or twice the maximum pressure which can be developed in the feed line in normal service, whichever is greater.