BRITISH STANDARD

BS 4994:1987

*Reprinted, incorporating Amendment No. 1*

## Specification for

# Design and construction of vessels and tanks in reinforced plastics

UDC 621.642.3:678.6.067.5

# Committees responsible for this British Standard

The preparation of this British Standard was entrusted by the Pressure Vessel Standards Committee (PVE/-) to Technical Committee PVE/4, upon which the following bodies were represented:

Associated Offices Technical Committee
British Chemical Engineering Contractors' Association
British Plastics Federation
Chemical Industries Association
Department of Trade and Industry (National Physical Laboratory)
Engineering Equipment and Materials Users' Association
Health and Safety Executive
Institution of Mechanical Engineers
Process Plant Association

The following bodies were also represented in the drafting of the standard, through subcommittees and panels:

Electricity Supply Industry in England and Wales
Plastics and Rubber Institute

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# Contents

# Foreword

This British Standard has been prepared under the direction of the Pressure Vessel Standards Committee and is a revision of BS 4994:1973, which is now withdrawn together with its explanatory supplement PD 6480. Its purpose is to establish a general standard for the design, fabrication and use of vessels and tanks in reinforced plastics for industrial service.

This revision has been made to take account of experience with reinforced plastics vessels and tanks and opportunity has been taken to extend the scope to include filament winding and rectangular tanks and to categorize vessels in relation to duty.

In the 1973 edition of this standard the minimum value for the overall design factor was 6. In this edition that value has been increased to 8. The change is made because experience has shown that the overall strain limitation does not permit lower values than 8.

The manufacture of vessels and tanks in reinforced plastics is a wide field, involving a large number of materials, both plastics and reinforcing systems, and widely different methods of manufacture. It is not practicable to cover all aspects in a single standard and this standard covers part of the field, namely, the use of polyester, epoxy and furane resins in wet lay-up systems.

Information on the engineering properties of reinforced plastics is somewhat limited and this applies particularly to the changes in such properties over long periods. In the absence of comprehensive long-term properties, therefore, the material properties used for design are based on short-term tests. Material property data should be presented in the form recommended in BS 4618.

Metallic vessels, being made from materials which are normally isotropic, are conveniently designed by calculating permissible stresses, based on measured tensile and ductility properties. In contrast, laminar constructions are usually anisotropic and the design method in this standard, being based on unit loadings, is particularly suited to the design of composites of reinforced plastics.

The calculation of an appropriate laminate construction is based on the allowable unit loading and unit modulus for the type of composite proposed. In addition, the allowable strain in the laminate is limited to ensure that breakdown of the resin-reinforcement bond does not occur in any part of the structure.

NOTE 1 Debonding occurs at a strain of approximately 0.3 % for the resin-glass fibre composites at present in general use. (See references [1] to [8].)

Design factors are included to cover such variables as:

a) deterioration of the composite properties over a long period;

b) the effect of temperature on the properties of the composite;

c) repeated or alternating loading.

As a result of adopting what was effectively a minimum design factor of 8 and taking into account the other design factors, environmental stress cracking of the laminates in aqueous environments has not been a cause of failure in vessels and tanks produced in accordance with this standard. This mode of failure has been a common source of trouble in other laminates. Furthermore, test work has shown that the allowable strains are below the level at which environmental stress cracking is likely to occur.

It has been assumed in the drafting of this British Standard that the execution of its provisions is entrusted to appropriately qualified and experienced people.

NOTE 2 The numbers in square brackets used throughout the text of this standard relate to the bibliographic references in Appendix K.

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 viii, pages 1 to 188, 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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# Section 1. General

## 1 Scope

This British Standard specifies requirements for the design, materials, construction, inspection, testing and erection of vessels and tanks in reinforced plastics, consisting of a polyester, epoxy or furane resin system reinforced with glass fibres, manufactured by the wet lay-up process. Constructions both with and without a lining of thermoplastics are included.

It is implicit that vessels and tanks covered by this standard are made only by manufacturers who are competent and suitably equipped to comply with all the requirements of this standard. Compliance with these requirements may be proven by documentation of past experience, or prototype testing, to the satisfaction of the purchaser or Inspecting Authority, as appropriate (see clause **38**).

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.

This standard covers vessels and tanks subject to temperatures between – 30 °C and 110 °C.

NOTE 1 There is experience with vessels and tanks at temperatures above 110 °C but these require special consideration.

NOTE 2 The following vessels and tanks also require special consideration:

a) jacketed vessels and tanks;
b) vessels and tanks for the transport of liquids and gases;
c) buried tanks.

NOTE 3 The titles of the publications referred to in this standard are listed on the inside back cover.

## 2 Definitions

For the purposes of this British Standard the definitions of *plastics, resin, thermoset, thermoplastics* and *reinforced plastics* given in BS 1755-1 apply, together with the following definitions and those listed in Table 1.

NOTE Where the words purchaser, manufacturer and Inspecting Authority occur in the text, they are intended also to include representatives of the purchaser, manufacturer and Inspecting Authority.

**2.1**
**cure**

the chemical reaction resulting in the final polymerized product

NOTE It may be effected at ambient temperature or by the use of heat. In certain resin systems the cure has to be effected in two stages; the first stage may, and the second stage does, involve the application of heat. This second stage is known as the "post-cure".

**2.2**
**gel coat**

a thin layer of resin on the surface of a laminate, that may or may not be reinforced with a fabric, tissue or scrim

**2.3**
**laminate**

a resin sheet or moulding reinforced with a form of glass fibre or other suitable material

**2.4**
**laying-up**

a process of applying or producing laminates in position on a former prior to cure

**2.5**
**manufacturer**

the organization, that designs and fabricates the vessel or tank in accordance with the purchaser's requirements

NOTE The design and fabrication functions may be carried out by separate organizations.

**2.6**
**purchaser**

the organization or individual that buys the finished vessel or tank for its own use or as an agent for the owner

**2.7**
**inspecting authority**

the body or association that checks that the design, materials and construction comply with this standard

**2.8**
**vessel**

a closed container subject to applied pressure or vacuum, with or without hydrostatic head

NOTE The term vessel includes branches up to the first flanged connection.

**2.9**
**tank**

a container for the storage of fluids subject only to its own hydrostatic head and freely vented to atmosphere

NOTE The term tank includes branches up to the first flanged connection.

## 3 Nomenclature

Several terms relating to the strength and load-carrying capacity of individual layers of the composite laminate are used in this standard. Some have similar but quite distinct meanings and because of both their similarity and their application, particular care is required in their use.

The terms concerned are listed in Table 1, with their definitions, symbols and units.

## 4 Information and requirements to be agreed and to be documented

### 4.1 Information to be supplied by the purchaser

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

a) Process conditions.

1) Materials to be handled (names, concentrations and relative densities) including likely impurities or contaminants.

2) Design pressure (or vacuum) including test requirements and design temperature.

3) Operating pressure (or vacuum) and temperature.

4) Mode of operation, e.g. process cycling conditions.

5) Any abrasion or erosion problems which may be encountered.

b) Site conditions.

1) Nature of ambient atmosphere including any extremes of temperature.

2) Superimposed loads, e.g. wind, snow and associated pipework.

3) Loads imposed by personnel during erection and operation.

4) In the case of buried vessels and tanks, soil conditions and expected loading, e.g. traffic.

5) Seismic loading.

c) Special conditions.

1) Boiling but.

2) Vibration due to adjacent plant.

3) Agitation details.

4) Danger of mechanical impact and damage.

5) Loads imposed during transport.

6) Finish, e.g. if fire-resisting (see clause **8**).

e) Details of any special or additional tests or inspection required and where these are to be carried out (see clause **33** and **39.1**).

f) Exemption to apply a pigmented coating to the vessel before final inspection (see clause **33**).

g) Facilities for testing (see Table 4 and **39.1**).

h) Name of Inspecting Authority, if applicable.

i) Requirements for packaging, despatch and installation.

**Table 1 — Nomenclature, symbols and units**

| Term | Definition | Derivation | Symbol | Unit |
|---|---|---|---|---|
| Ultimate tensile unit strength (UTUS) | The strength of a reinforcement type, expressed as force per unit width, per unit mass of reinforcement | Obtained from the fracture load of a laminate of known construction, in a tensile test | $u$ | N/mm per kg/m$^2$ glass |
| Load-limited allowable unit loading | The load permitted to be applied to a reinforcement type per unit width, per unit mass of reinforcement, not taking strain into account | Determined by dividing the ultimate tensile unit strength by the design factor, $K$ | $u_L$ | N/mm per kg/m$^2$ glass |
| Strain-limited allowable unit loading | The load permitted to be applied to a reinforcement type per unit width, per unit mass of reinforcement, taking only strain into account | Determined by multiplying the unit modulus, $X$, by the allowable strain for the particular laminate layer | $u_S$ | N/mm per kg/m$^2$ glass |
| Design unit loading (layer design unit loading) | The load permitted to be applied to a reinforcement type per unit width per unit mass of reinforcement, for the vessel or tank under consideration | Whichever is the smaller of $u_S$ or $u_L$ at the design strain, $\epsilon_d$ | $u_z$ | N/mm per kg/m$^2$ glass |
| Unit modulus | The ratio of the load per unit width to the corresponding direct strain, in a loaded tensile test specimen per unit mass of a single glass reinforcement type | Obtained from the measured load and strain in a tensile test | $X$ | N/mm per kg/m$^2$ glass |
| Overall unit modulus | The calculated ratio of the load per unit width to the corresponding direct strain in full laminate of single or multiple glass reinforcement type | Obtained from a summation of the products of unit modulus and mass of each glass reinforcement type in a laminate [see equation (12)] | $X_{LAM}$ (may be identified by further subscript) | N/mm |
| Design strength (laminate design unit loading) | The load carrying capacity of a laminate, expressed as force per unit width | Obtained by summing the load carrying capacities of all the layers in the laminate ($u$, $m$, $n$ terms). The subscripts indicate a main ($U_{LAM}$) or an overlay ($U_{OVL}$) laminate | $U_{LAM}$ $U_{OVL}$ | N/mm width |
| Unit load | The force per unit width carried by a laminate resulting from pressure or other loads applied to the vessel or tank | Obtained from the appropriate design calculations for the portion of vessel or tank under consideration (see section 3) | $Q$ | N/mm width |

### 4.2 Requirements to be agreed and documented

The following items to be agreed between the purchaser, or the Inspecting Authority, where appropriate, and the manufacturer shall be fully documented. Both the definitive requirements specified throughout the standard and the documented items shall be satisfied before a claim of compliance with this standard can be made and verified.

a) Resin system to be used [see **6.1** a)].

b) Use of reinforcing materials other than those complying with BS 3396, BS 3496, BS 3691 or BS 3749, as appropriate (see **6.2**).

c) Mechanical properties of materials (see **6.4.1**).

d) Type of chemical barrier to be used (see **7.1** and **27.3**).

e) Where a thermoset lining is used on tanks and vessels which are constructed in accordance with categories I and II, whether it is permissible to reduce the backing layer (see **7.3**).

f) Design details (see clause **11**).

1) Essential dimensions, including tolerances (see also **30.2.7**), preferably on a drawing.

2) Design calculations with references.

3) Nominal thickness, including tolerance, of corrosion-resistant lining (thermoplastics or gel coat) which does not contribute to strength.

4) Form(s) of reinforcement including type, number and arrangement of individual layers.

5) Form(s) of local stiffening, where used.

6) Details of welds in thermoplastics linings.

7) Bolting and flange materials and details (see also Figure 24).

8) Gasket materials and details.

9) Details of external finish, including steelwork (see also **28.6**).

10) Requirements for access and inspection openings (see also **20.5**).

g) Where the design incorporates reinforcement with directional properties, the orientation of the fibres (see clause **11**).

h) Lining and laminate system to be employed (see **13.4**).

i) Supports (see **22.1**).

j) Any modification to the approved design (see clause **25**).

k) Where site fabrication is employed, the special procedures to be adopted (see clause **26**).

l) Repair of laminate defects and method of repair (see **27.2**).

m) Whether hot plate welding is to be used (see **27.3**).

n) For cylindrical shells, the tolerance on the circumference where this is not 5 mm for shells up to and including 600 mm outside diameter or 0.25 % of the calculated circumference for larger shells (see **30.2.2**).

o) Arrangements for access to manufacturer's premises (see **31.4**).

p) The provision of special test laminates and the extent of mechanical testing to be carried out either on cut-outs or prepared laminates (see **37.1**).

q) If the prototype tests are not to be witnessed by the purchaser, and the Inspecting Authority, where applicable, (see **38.1**).

r) The nature of prototype tests, the hydraulic test pressure if it is higher than the design pressure and the limits of cyclic variations to determine fatigue strength (see **38.3**).

## 5 Categories of vessels and tanks

### 5.1 General

The design, documentation and inspection and test requirements for vessels and tanks, shall be related to one of three categories that shall be determined from all the factors given in Table 2 when considered in combination.

### 5.2 Design documentation

The design documentation shall be as given in Table 3.

### 5.3 Quality control (tests and records)

The quality control shall be as given in Table 4.

**Table 2 — Minimum categories of vessel or tank**

| | Category I | Category II | Category III |
|---|---|---|---|
| **Contents**[a] | | | |
| Toxic | X | | |
| Highly corrosive | X | | |
| Corrosive | | | X |
| Flammable | X | | |
| Others | | | X |
| **Chemical compatibility of liner with process fluid** | | | |
| Known long-term compatibility based on service experience | | | X |
| Compatibility based on related performance data | | X | |
| Only specimen data (dip coupons) available | X | | |
| **Design temperature, *T*** | | | |
| $T < 60$ ° C and $T \leqslant$ (HDT[a] – 40 °C) | | | X |
| $T \geqslant 60$ ° C and T $\leqslant$ (HDT – 40 °C) | | X | |
| $T >$ (HDT – 40 °C) and $T \leqslant$ (HDT – 20 °C) | X | | |
| **Design pressure and/or vacuum** | | | |
| Static head only | | | X |
| < ± 5 mbar[b] (above static head) | | X | |
| $\geqslant$ ± 5 mbar (above static head) | X | | |
| **Size of vessel or tank (capacity)** | | | |
| < 10 $m^3$ | | | X |
| 50 $m^3$ $\leqslant$ capacity $\geqslant$ 10 $m^3$ | | X | |
| > 50 $m^3$ | X | | |
| **Geometry and supports** | | | |
| Flat bottom full support | | | X |
| Any other, e.g. legs, skirts, saddles, rings and frames | | X | |
| **Other criteria** | | | |
| If item is critical to safety | X | | |

[a] **Description of contents classification**

| | |
|---|---|
| Toxic. | The contents could present a significant risk to health of persons exposed. |
| Highly corrosive. | The contents could severely burn, blind, disfigure or maim an individual. |
| Corrosive. | The contents could cause damage to the skin or eye. |
| Flammable. | The contents have a flash point equal to or less than 55 °C. |
| Others. | The contents are not considered to burn, blind or injure individuals. |

[a] Heat distortion temperature of resin (see **6.4.2**).
[b] 1 mbar = 100 $N/m^2$ = 100 Pa.

**Table 3 — Design documentation and drawing requirements**

| | Category I | Category II | Category III |
|---|---|---|---|
| **Design calculations** | | | |
| a) Independent approval required | X | | |
| b) Calculations to cover | | | |
| 1) Hydrostatic loadings | X | X | X |
| 2) Applied pressure | X | X | |
| 3) Applied vacuum | X | X | |
| 4) Wind loads | X | X | |
| 5) Lifting arrangements | X | X | X |
| 6) Supporting | X | X | |
| 7) Seismic loading (if applicable) | X | X | |
| **Drawing requirements** | | | |
| Vessel/tank general arrangement | X | X | X |
| Full fabrication drawings showing method of manufacture | X | X | |
| Installation procedure | X | X | X |

**Table 4 — Quality control: tests and records**

| | Category I | Category II | Category III |
|---|---|---|---|
| **Material records** | | | |
| Record of resin type and quantity | X | | |
| Record of glass type and quantity | X | | |
| Record of personnel on the fabrication | X | X | |
| Record of layers and type of glass | X | X | X |
| Record of cure system | X | X | X |
| Record of post cure (when used) | X | X | |
| **Quality control tests** | | | |
| Spark test on thermoplastics liners | X | X | X |
| Adequate documented information on the mechanical properties of the particular resin/glass laminate to be provided. If this is not available for categories II or III, production test samples to be tested as required for category I. | X | X | X |
| A production test coupon to be laminated with the vessel or obtained from nozzle cut-outs. The coupons to be tested, as follows. | | | |
| Ultimate tensile unit strength | X | | |
| Unit modulus | X | | |
| Lap shear strength | X | | |
| Visual examination of nozzle cut-outs | X | X | X |
| Ash test on nozzle cut-outs | X | | |
| Thickness measurement[a] | X | X | |
| Acetone test (polyester resin) | X | X | X |
| Barcol hardness measurement | X | X | X |
| Residual styrene (polyester resin)[b] | X | X | |
| Also, in the case of thermoplastics lined glass reinforced plastics (GRP) tanks and vessels, the following to be demonstrated. | | | |
| Weld strength | X | X | |
| Bond strength | X | X | |
| **Quality control records** | | | |
| Hardness test (Barcol) | X | X | X |
| Thickness measurement | X | X | X |
| Nameplate details | X | X | X |
| Documentation requirements | | | |
| Pressure/vac./hydrostatic head | X | X | |
| Ultimate tensile unit strength | X | X | |
| Unit modulus | X | X | |
| Lap shear strength | X | X | |
| Weld strength | X | X | |
| Bond strength | X | X | |
| Ash test on cut-out | X | | |
| **Independent inspection** | X | | |

[a] To verify minimum design thickness. Particular attention is to be paid to points of discontinuity, e.g. nozzles and end attachments.
[b] See **37.2**.

# Section 2. Materials and design loadings

## 6 Materials

### 6.1 Thermosetting resin systems

The resin system(s) to be used shall be agreed between the purchaser and the manufacturer [see **4.2** a)].

The resin selected shall be of a suitable commercial grade which complies with the technical requirements of the application and for polyester resins the minimum requirements of BS 3532.

NOTE 1 The specified classes of resins used for the preparation of laminates are polyester resins, epoxy resins and furane resins. There are many resin systems in each class and the properties of these systems vary, especially with respect to chemical resistance and heat distortion temperature. See also Appendix A.

The recommendations of the resin suppliers for the use of hardeners, catalysts and accelerators shall be followed.

NOTE 2 The amount of these materials used is critical as it can affect both the rate of the reaction and extent of the cure.

Pigments shall not be used except in the external finish [see clause **33** j)].

NOTE 3 Limited use of thixotropic agents can be made provided that they do not interfere with visual inspection and chemical resistance of the laminate.

Data, procedures and figures in this standard refer to laminates containing no filler materials. Where special-purpose additives are used, special consideration shall be given to design and manufacture as there may be an effect on the laminate properties.

### 6.2 Reinforcing materials

The reinforcing material shall be a suitable grade of glass fibre having a glass finish compatible with the resin used and shall either comply with BS 3396, BS 3496, BS 3691 or BS 3749, as appropriate, or be the subject of agreement between the purchaser and the manufacturer [see **4.2** b)].

### 6.3 Thermoplastics lining material

A number of thermoplastics, for example, polyvinyl chloride (PVC), polypropylene (PP), polyvinylidene fluoride (PVDF), ethylene chlorotrifluoroethylene (ECTFE), chlorinated PVC (CPVC) and fluorinated ethylene propylene copolymer (FEP) are available as lining materials and the thermoplastics selected shall comply with **7.1**.

The thickness of PVC shall not be less than 2.5 mm. In the case of other thermoplastics, the minimum thickness shall be 2.00 mm except in the case of pipes and branches of 100 mm diameter or less, when the minimum thickness shall be 1.5 mm.

NOTE Experience has shown that the use of PVC liners in excess of 4.5 mm thickness can lead to problems in service when the operating temperature is above 40 °C.

### 6.4 Mechanical properties

**6.4.1** The mechanical properties of the laminate layers shall be agreed between the purchaser and the manufacturer [see **4.2** c)] and shall be not less than the values given in Table 5, when tested in accordance with Appendix B. The values given in Table 5 apply to laminates incorporating only E glass reinforcement complying with BS 3396, BS 3496, BS 3691 or BS 3749 and having a glass content by mass within the range 28 % to 45 % for chopped strand mat (CSM), 45 % to 55 % for woven roving (WR) cloth and 65 % to 75 % for filament windings and values for laminates including other types of reinforcement or other glass contents shall be on the basis of test results.

Where WR cloth with a directional bias is used, the figure for ultimate tensile unit strength shall be 500 $P$, where $P$ is the proportion of glass fibres running in the direction concerned. The unit modulus in the warp and weft direction will be different to that given in Table 5 and shall be established by test. Where the bias is greater than 5 to 1, the strength in the weft direction shall be ignored.

**6.4.2** The heat distortion temperature of the fully cured resin system used for the reinforced laminate shall be not less than 20 °C higher than the design temperature of the vessel when determined as described in BS 2782:Method 121A (see also **B.4**).

### 6.5 Chemical properties

As the chemical resistance of lining materials varies with the source and type of polymer, the suitability of the material, including adequate corrosion and erosion resistance, shall be established by tests; a suitable reference document is BS 4618-4.1. It is important in any such tests that the test liquors are fully representative of the process, particularly with respect to the presence of trace organics, and, where possible, the test specimens shall be subjected to a level of strain not less than the design strain.

**Table 5 — Minimum properties of reinforced laminate layers**

| Type of reinforcement | Ultimate tensile unit strength (UTUS) | Unit modulus | Lap shear strength |
|---|---|---|---|
| | N/mm width per kg/m$^2$ glass | N/mm width per kg/m$^2$ glass | N/mm$^2$ |
| CSM (resins other than furane) | 200 | 14 000 | 7.0 |
| CSM (furane) | 140 | 14 000 | 5.0 |
| WR cloth plain weave (warp and weft directions) (resins other than furane) | 250 | 16 000 | 6.0 |
| WR cloth plain weave (warp and weft directions) (furane) | 160 | 16 000 | 4.0 |
| Unidirectional filament (fibre direction) | 500 | 28 000 | 6.0 |

### 6.6 Water quality

When selecting materials, account shall be taken of the effects of materials on water quality (see Appendix C).

## 7 Construction of chemical barrier

**7.1** The type of chemical barrier to be used shall be agreed between the purchaser and the manufacturer [see **4.2** d)].

**7.2** Where a thermoplastics lining is used, the minimum bond strength of the reinforcement to the lining shall be 7 N/mm$^2$ in direct shear and 5 N/mm width in peel when tested in accordance with **B.10** and **B.11**.

NOTE This bond strength may be achieved by the inclusion of a minimum of 0.45 kg/m$^2$ chopped glass strand or CSM immediately behind the thermoplastics lining.

**7.3** Where a thermoset lining is used, in order to achieve the optimum properties the construction of the laminate in contact with the corrodent shall consist of the following.

a) *Surface layer*. A resin-rich surface layer reinforced with C glass surfacing mat, synthetic fibres or other suitable material, with a thickness between 0.25 mm and 0.50 mm.

b) *Backing layer*. A backing layer normally containing a minimum of 1.2 kg/m$^2$ chopped glass strand or CSM with a soluble binder with between 25 % and 33 % glass content, by mass.

For tanks and vessels which are constructed in accordance with categories II and III it is permissible to reduce the backing layer to 0.6 kg/m$^2$ chopped glass strand or CSM if agreed between the purchaser and the manufacturer [see **4.2** e)].

## 8 Flammability

Where a vessel or tank is intended to contain flammable fluids the external surface layers shall be modified so as to have a surface spread of flame characteristic which at least complies with the class 1 requirements of BS 476-7.

In other cases the finish shall be as specified by the purchaser [see **4.1** c) 6)].

## 9 Allowable and design unit loadings

### 9.1 Symbols

For the purposes of clause **9** the following symbols apply.

$F_\phi$ circumferential factor

$F_x$ longitudinal factor

$k_1$ factor relating to method of manufacture

$k_2$ factor relating to long-term behaviour

$k_3$ factor relating to temperature

$k_4$ factor relating to cyclic loading

$k_5$ factor relating to curing procedure

$K$ overall design factor determined from equation (1)

$u_z$ design unit loading (in N/mm per kg/m$^2$ glass) for layer of type $z$

$u_L$ load-limited allowable unit loading, i.e. ultimate tensile unit strength (in N/mm per kg/m$^2$ glass) divided by $K$

$u_S$ strain-limited allowable unit loading, i.e. unit modulus (in N/mm per kg/m$^2$ glass) multiplied by allowable strain

$X_\phi$ unit modulus in circumferential direction (in N/mm per kg/m$^2$ glass)

$X_x$ unit modulus in longitudinal direction (in N/mm per kg/m$^2$ glass)

$X_z$ unit modulus of layer of type $z$ (in N/mm per kg/m$^2$ glass)

$\epsilon_L$ strain under unit loading $u_L$

$\epsilon$ maximum allowable strain

$\epsilon_d$ least strain, determined from allowable loadings and resin properties

$\epsilon_R$ extension to failure (fracture strain) of unreinforced resin determined in accordance with Appendix B

$\theta$ angle to the longitudinal axis of filament winding

### 9.2 Design calculations

**9.2.1** When designing in reinforced plastics, it is desirable to work in terms of unit load, i.e. force per unit width, rather than stresses, i.e. force per unit area, and therefore the following calculations shall be adopted.

The maximum allowable unit load for each type of layer shall be determined from the material properties given in Table 5 as follows (see Appendix D for worked examples and abbreviated method).

Where the design calculations (see section 3) require the use of allowable compressive unit loadings, these shall be determined in accordance with **9.2.2** to **9.2.6**, substituting the ultimate compressive unit strength for the ultimate tensile unit strength in equation (2).

Ultimate compressive unit load shall be determined, when required, for each laminate layer concerned, in accordance with **B.8**.

**9.2.2** The design factor $K$ shall be determined from equation (1):

$$K = 3 \times k_1 \times k_2 \times k_3 \times k_4 \times k_5 \qquad (1)$$

where the factor 3 represents a constant which allows for the reduction of material strength caused by long-term loading (required even for loading in air), and $k_1$ to $k_5$ represent factors determined by the method of manufacture and operating conditions. No vessel or tank shall have a design factor $K$ of less than 8.

NOTE For a rarely occurring, short-term emergency condition (less than 10 times in the life of the vessel, each duration less than 30 min) a design factor less than the calculated design factor, but not less than 8, may be used.

The chemical liner, see clause **7**, shall be ignored in strength calculations.

Values for factors $k$ shall be determined as follows.

a) *Factor relating to method of manufacture,* $k_1$. This factor shall be the value taken from Table 6 appropriate to the method of manufacture to be adopted.

When rovings are chopped for spray application, the length of individual strands shall be not less than 32 mm.

**Table 6 — Factor relating to method of manufacture**

| Method of manufacture | Factor $k_1$ |
|---|---|
| Handwork | 1.5 |
| Machine-controlled filament winding | 1.5 |
| Machine-controlled spray application | 1.5 |
| Hand-held spray application | 3.0 |

b) *Factor for chemical environment (and associated strength loss),* $k_2$. This factor shall be 1.2 for vessels having a thermoplastics lining unless lack of experience or suitable test data suggests that a higher value is required. The factor for vessels without a thermoplastics lining shall be selected within the range 1.2 to 2.0.

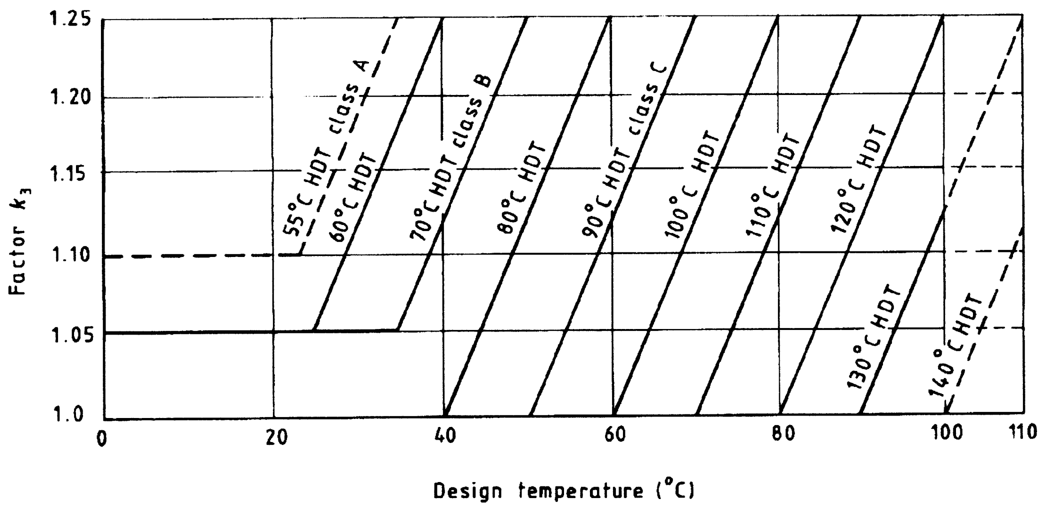
NOTE A guide to establishing the value of $k_2$ is given in Appendix E.

c) *Factor relating to temperature,* $k_3$. This factor shall be within the range 1 to 1.25, dependent upon the heat distortion temperature and be determined from Figure 1.

d) *Factor relating to cyclic loading,* $k_4$. This factor shall be determined from Figure 2, having regard to the expected operating conditions of the vessel.

e) *Factor relating to the curing procedure,* $k_5$. Where the vessel is subjected to a complete curing procedure, including a full post-cure at elevated temperature appropriate to the resin system, at the manufacturer's works the factor $k_5$ shall be taken as 1.1. Copies of the temperature chart shall be supplied to the Inspecting Authority (information to be obtained from the resin supplier). Vessels not subjected to full post-cure shall have the factor $k_5$ taken as 1.3 when they are designed for operating temperatures up to and including 45 °C and 1.5 when they are designed for operating temperatures over 45 °C.

NOTE The post-curing temperature should be chosen to ensure that the laminate will comply with the mechanical, temperature and chemical performance required by the design. High-performance resin systems generally require a post-cure temperature of at least 80 °C to develop their optimum properties and, wherever possible, the tank or vessel should be post-cured at the design temperature.




NOTE Experience in the range above 100 °C is limited and it is therefore recommended there is a full discussion between the manufacturer and the purchaser if a vessel is required to operate in this range.

**Figure 1 — Factor relating to temperature,** $k_3$

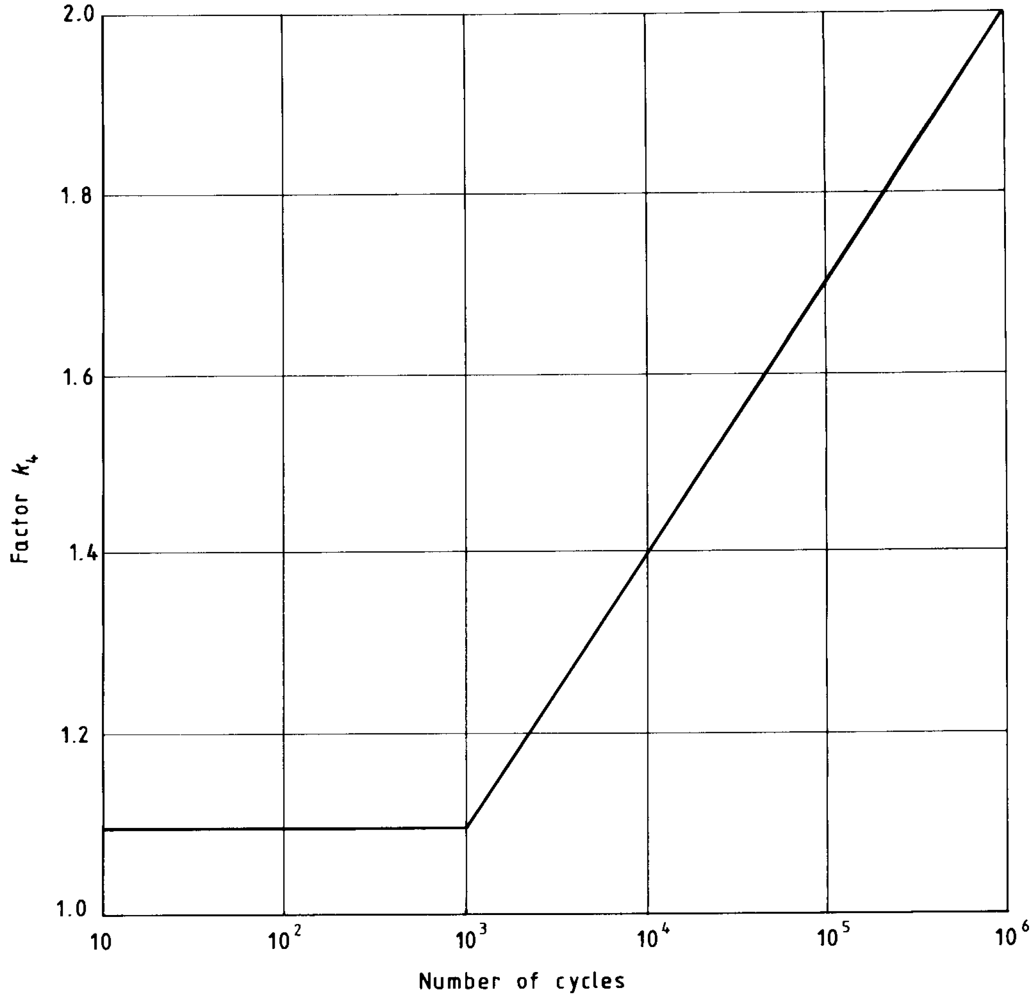



**Figure 2 — Factor relating to cyclic loading, $k_4$**

**9.2.3** The load-limited allowable unit loading, $u_L$, shall be determined from equation (2):

$$u_L = \frac{u}{K} \qquad (2)$$

where $u$ is UTUS from Table 5.

**9.2.4** The maximum allowable strain, $\epsilon$, shall be determined.

a) The maximum allowable strain shall not exceed 0.1 $\epsilon_R$ or 0.2 %, whichever is the smaller.

b) For chemically resistant resin surface layers, the extension to failure of the unreinforced resin shall be determined in accordance with **B.5**. The maximum allowable strain for the whole of the laminate shall be that value determined from the smallest extension to failure of the resins to be incorporated.

NOTE For thermoplastics linings a maximum allowable strain of 0.2 % will be satisfactory, so that these linings will not introduce a more severe strain limitation into the vessel design.

**9.2.5** The strain-limited allowable unit loading, $u_S$, shall be determined from equation (3):

$$u_S = X_z \epsilon \qquad (3)$$

**9.2.6** The design unit loading, $u_z$, for each type of layer shall be determined.

a) If $u_S$ is smaller than $u_L$ for all layers, the appropriate value of $u_S$ shall be taken as the design unit loading, $u_z$ for each layer.

NOTE A laminate design is strain-limited when $u_S$ is smaller than $u_L$ for all the layers.

b) If, for some or all of the layers, $u_L$ is smaller than $u_S$, the strain for each layer concerned shall be determined from equation (4):

$$\epsilon_L = \frac{u_L}{X_z} \quad (4)$$

Considering all the layers making up the laminate, the allowable strain for that laminate, $\epsilon_d$, shall be the smallest of the values of $\epsilon_L$ so determined. The design unit loading for each layer, $u_z$, shall then be determined from equation (5):

$$u_z = X_z \epsilon_d \quad (5)$$

The design unit loadings for each type of layer, $u_z$, determined from a) or b), as appropriate, shall then be substituted as required in the design equations in section 3.

**9.2.7** The procedure in **9.2.6** does not apply when continuous rovings are filament wound at an angle of $\theta$ (in degrees) to the vessel or tank axis and in this case values of circumferential and longitudinal unit modulus $X_\phi$ and $X_x$ for individual layers shall be obtained by reference to Figure 3. Values of circumferential and longitudinal design unit load shall be determined from equations 5(a) or (b) and by application of the factors given in Table 7, as appropriate.

In circumferential direction:

$$u_z = X_\phi \epsilon_d F_\phi \quad [5(a)]$$

In longitudinal direction:

$$u_z = X_x \epsilon_d F_x \quad [5(b)]$$

Where a laminate construction using filament winding is subjected to a simultaneous application of loading in the circumferential and longitudinal directions then, depending upon the combination of direction and type, i.e. tensile or compressive, of loading, the maximum allowable loads are not necessarily as great as the individual unit loads derived from the calculations of this section and, therefore, the acceptability of the combined loadings shall be determined by means of a biaxial design envelope, details of which are given in the worked examples of **D.4**.

**Table 7 — Factors to be applied to the design unit loading of continuous rovings for different winding angles**

| Filament winding angle to axis $\theta$ | Circumferential factor $F_\phi$ | Longitudinal factor $F_x$ |
|---|---|---|
| degrees | | |
| $0 < \theta \leqslant 15$ | 0 | 1 |
| $15 < \theta \leqslant 75$ | 0.5 | 0.5 |
| $75 < \theta \leqslant 90$ | 1 | 0 |
| NOTE Higher values may be used for these factors if a rigorous anisotropic elastic analysis is carried out. This analysis allows for the contribution from each layer in the laminate and for the interaction between normal and shear strains (see references [9], [10] and [11]). | | |

It shall be ensured that the strain transverse to the fibre direction is less than 0.1 %. In the absence of a rigorous anisotropic analysis, the axial strain shall be limited to 0.1 % for a winding angle greater than 75°.

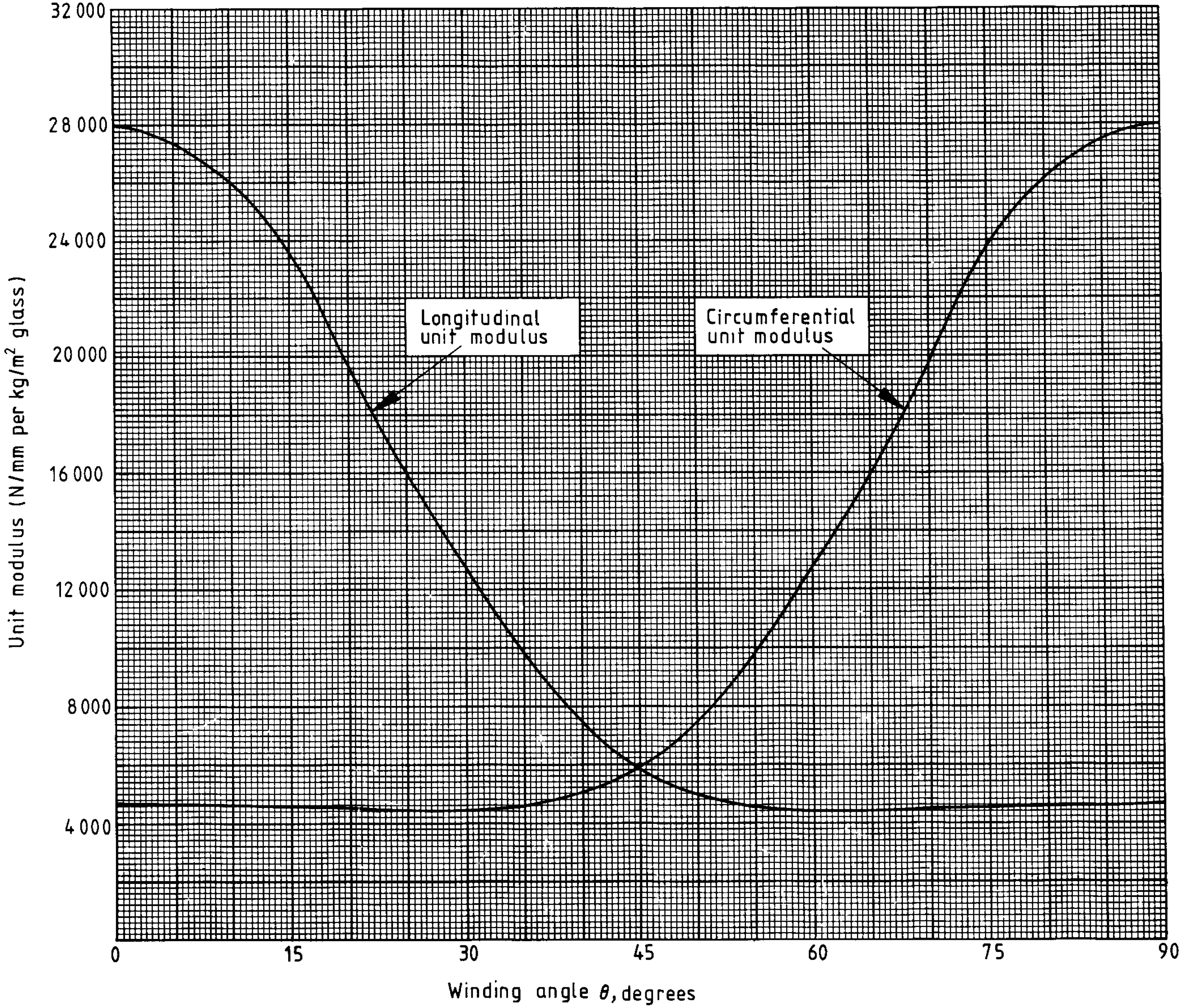



Figure 3 — Unit modulus/winding angle

# Section 3. Design

## 10 General

The vessel or tank shall be designed for the most severe combination of conditions which may include:

a) internal or external pressure;

b) static head of contents (working and test conditions);

c) weight of vessel or tank and contents;

d) design temperature;

e) superimposed and wind loads;

f) bending moments due to eccentric loads;

g) localized loads acting at the supports, lugs and other attachments;

h) shock loads;

i) loads due to heating or cooling and thermal gradients;

j) loads applied during transport or erection;

k) loads imposed by personnel during erection and operation;

l) fatigue.

## 11 Design basis

The vessel or tank shall be designed in accordance with this section. The actual design details shall be agreed between the purchaser and the manufacturer, and the Inspecting Authority, where applicable, [see **4.2** f)].

NOTE 1 The design calculations in this standard take advantage of the ease with which the laminate details can be varied to suit the loads imposed by operating and test conditions in the different regions.

Where the design incorporates reinforcement with directional properties (e.g. WR), the orientation of the fibres shall be agreed between the purchaser and the manufacturer [see **4.2** g)] and shall be specified on drawings and calculations in order to ensure that the structural properties required by the design are obtained.

NOTE 2 The objective of the calculations specified in this section is, therefore, to determine acceptable laminate constructions for each part and detail, e.g. shell, ends, manholes, branches and supports.

## 12 Design details

### 12.1 Design temperature

The design temperature shall be the maximum temperature it is possible for the vessel or tank to attain under operating conditions (including boil-out, where applicable).

NOTE Where necessary, provision should be made in the design to permit expansion or contraction, to avoid severe thermal loads. Thermal loads arising from temperature gradients may also require evaluation.

### 12.2 Design pressure

The design pressure, i.e. the pressure to be used in calculations to establish the required strength of the component parts of a vessel or tank, shall be not less than:

a) the pressure which will exist in such parts of the vessel or tank when the pressure-relieving device(s) start(s) to relieve or the set pressure of the pressure-relieving device(s), whichever is the higher;

b) the maximum pressure or head which can be achieved in service where this is not limited by relieving device(s).

Notwithstanding this, the strength of the laminates so calculated shall be increased where necessary to ensure that during hydraulic pressure or static head tests the laminate is at no point strained beyond the limiting values specified in clause **39**, i.e. 0.26 %, or 1.3 $\epsilon_d$, whichever is the smaller, where $\epsilon_d$ is the least strain, determined from allowable loadings and resin properties.

The value of the design pressure(s) to be used in calculation shall include the static head of content or test medium, as applicable, taking due account of relative density.

Vessels subject to external pressure shall be designed for the maximum differential pressure to which the vessel may be subjected in service.

NOTE 1 It is recommended that vessels subjected to vacuum be designed for a full negative pressure of 1 bar unless a vacuum break valve or similar device is provided, in which case a lower design pressure may be agreed between the purchaser and the manufacturer.

The total discharge capacity of the pressure-relieving device(s) shall be such that during discharge the pressure within the vessel or tank does not exceed the design pressure by more than 10 %.

NOTE 2 Where bursting discs are used, it is essential that their manufacturer be consulted before the design pressure is decided as the bursting pressure of the disc is dependent upon disc material and operating temperature. See also BS 2915.

# 13 Laminate design and thickness

## 13.1 Symbols

For the purposes of clause **13** the following symbols apply.

$m_z$ mass of reinforcement per unit area (in kg/m$^2$ glass) in one layer of type $z$

$n_z$ number of layers of type $z$ in construction under consideration (for filament winding a layer shall consist of two helixes wound at $\pm \theta°$)

$Q$ maximum applied unit load (in N/mm), i.e. maximum force per unit width, to be carried by the laminate

$u_z$ design unit loading (in N/mm per kg/m$^2$ glass) for layer of type $z$

## 13.2 Design calculation

For each region of the vessel or tank, a proposed laminate construction shall be determined by taking into account the design unit loading for each constituent laminate layer (as calculated in accordance with section 2). These loadings shall be related to the unit loads to be carried by the region concerned. Clauses **14** to **23**, inclusive, specify calculations for unit loads, determined from the vessel geometry and the greatest of these unit loads shall be used in equation (6).

The suitability of the proposed laminate construction shall then in every case be checked by using equation (6):

$$u_1 m_1 n_1 + u_2 m_2 n_2 + \ldots u_z m_z n_z \geqslant Q \tag{6}$$

NOTE 1 It may happen that the value of more than one $n_z$ is undetermined. The check should then be by a trial and error method and more than one acceptable solution may be found in these circumstances.

Alternatively, all but one (or two interdependent) values of $n_z$ may be fixed and the remaining value determined.

If the sum of the terms is less than $Q$, one or more of the values of $n$ shall be increased or a different laminate construction proposed and the calculations repeated until the required condition is satisfied. If the sum of the terms exceeds $Q$ by a large margin, the laminate is overdesigned for the region concerned.

NOTE 2 Worked examples of this design method are given in Appendix D and examples of construction are shown in Figure 4.

NOTE 3 Additional consideration is necessary if the vessel or tank is filament wound and subjected to biaxial loading. It is important to note that the response of a filament-wound vessel or tank to combinations of load applied simultaneously is different from the response of those loads applied independently.

To assess the behaviour of a filament-wound vessel or tank to combined loads either:

a) a complete anisotropic stress/strain analysis shall be carried out, the response of the material to those combined loads examined, and the shear or normal strain present within each layer shall be less than that calculated in **9.2.6**; or

b) a biaxial failure envelope shall be constructed as shown in the worked example given in Appendix D (see reference [16]).

NOTE 4 Where WR are to be used in the construction of vessels for use in vacuum service they should have a 1 to 1 warp to weft ratio.

## 13.3 Thickness

Where values of thickness are required in the equations in this section, the thickness of the laminate in the region under consideration shall be taken as the sum of the thicknesses of the individual layers making up that laminate.

The thickness of each layer, for design purposes, shall be determined from the glass content for that layer by using the graph (see Figure 5). Glass content shall be specified on the vessel drawings (see **B.2** for test method).

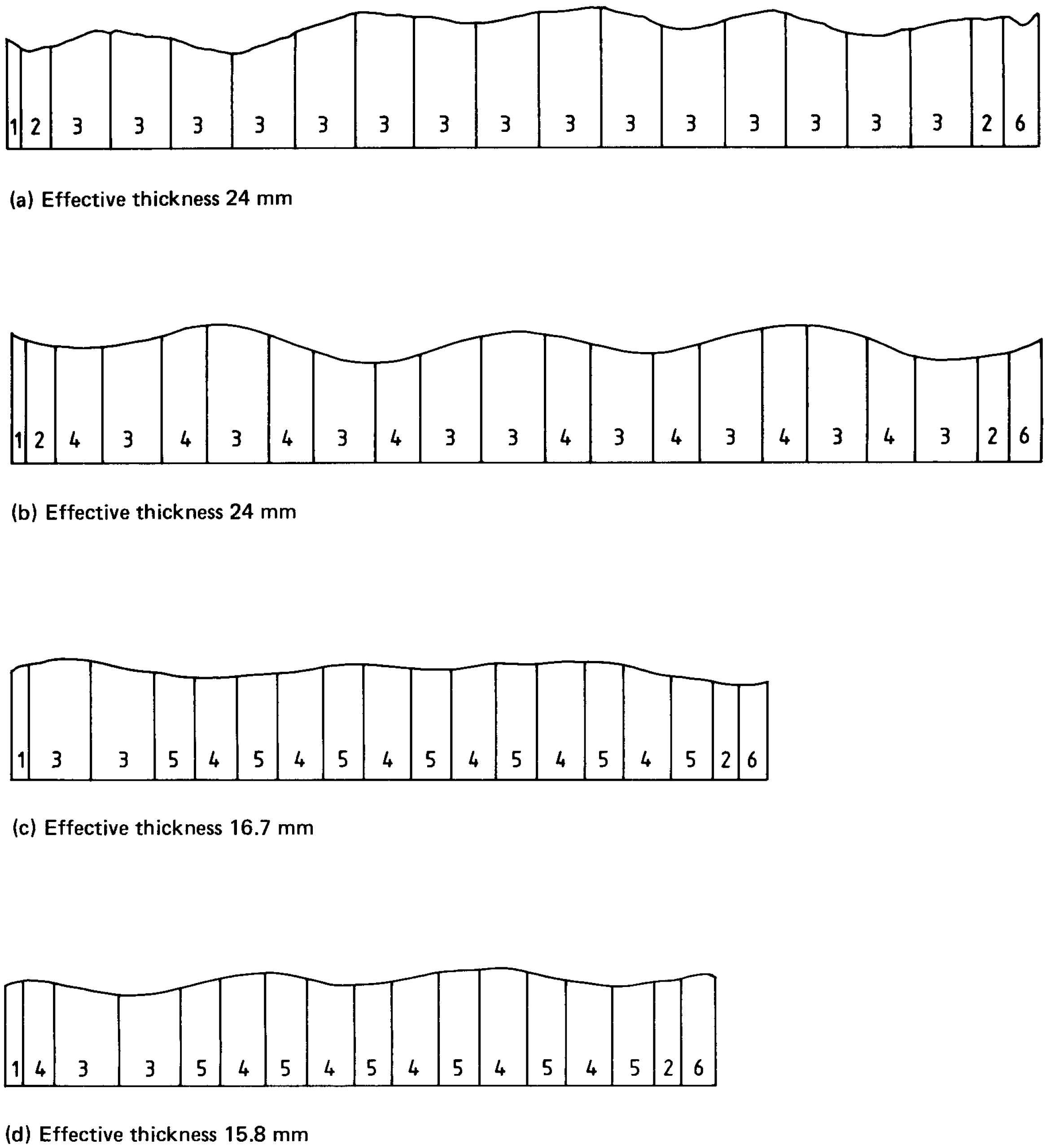
In no case shall the laminate thickness (excluding any lining) be less than 3 mm for tanks subject only to hydrostatic head of liquid contents, and 5 mm for vessels subject to internal pressure or vacuum.

Abrupt changes in laminate thickness shall be avoided. The blending taper between regions of differing thickness shall not be steeper than 1 in 6 (see Figure 6 and Figure 7).

### 13.4 Choice of construction

The lining and laminate system to be employed shall be specified in full on the drawing and agreed between the purchaser and the manufacturer before manufacture commences [see **4.2** h)].

NOTE The outermost layers of the vessel or tank should normally be CSM (300 g/m$^2$ minimum), and a tissue with a resin-rich surface.




**Key**
1 Chemical barrier (see clause **7** and **9.2.2**)
2 Chopped strand mat 0.3 kg/m$^2$
3 Chopped strand mat 0.6 kg/m$^2$
4 Chopped strand mat 0.45 kg/m$^2$
5 Woven roving cloth 0.8 kg/m$^2$
6 Resin-rich surface layer with binding tissue

NOTE Constructions (a), (b) and (d) all have a similar load carrying capacity (see Appendix D).

**Figure 4 — Four typical examples of laminate construction (without thermoplastics lining)**

## 14 Cylindrical and spherical shells

NOTE The equations in this clause are derived from thin shell theory and do not apply to vessels having a total laminate thickness greater than 10 % of the shell internal diameter.

### 14.1 Symbols

For the purposes of clause **14** the following symbols apply.

| | |
|---|---|
| $d$ | relative density of resin |
| $D_i$ | inside diameter of shell (in mm) |
| $D_o$ | outside diameter of shell (in mm) |
| $D_s$ | diameter of neutral axis of stiffening ring (in mm) |
| $E_{LAM}$ | Young's modulus (in $N/mm^2$)[a] of laminate under consideration |
| $F$ | factor of safety against collapse by buckling [see equations (13), (15), (16) and (19)] |
| $h_i$ | internal height of end (in mm) |
| $I_{s2}$ | second moment of area of stiffening ring (in $mm^4$) |
| $L$ | effective shell length (in mm) (see Figure 8) |
| $L_s$ | length of shell which may be regarded as contributing to the second moment of area of a stiffening ring (in mm) |
| $M$ | bending moment (in N mm) due to weight of vessel or tank and contents at point under consideration and wind or other applied loadings |
| $m_g$ | percentage glass content by mass |
| $m_z$ | mass of reinforcement per unit area (in $kg/m^2$ glass) in one layer of type $z$ |
| $n_z$ | number of layers of type $z$ in construction under consideration |
| $p$ | total effective pressure (in $N/mm^2$), i.e. design pressure (negative if vacuum) plus hydrostatic head at point under consideration |
| $Q_x$ | maximum longitudinal unit load (in N/mm) at point under consideration |
| $Q_\phi$ | maximum circumferential unit load (in N/mm) at point under consideration |
| $Q_p$ | maximum permissible compressive unit load (in N/mm) [see equation (13)] |
| $R_o$ | outside radius of spherical shell (in mm) |
| $t$ | laminate thickness of shell (in mm) calculated from laminate details |
| $t_m$ | laminate thickness of shell (in mm) derived from buckling criteria [see equations (15), (16) and (19)] |
| $T$ | taper length (in mm) |
| $W$ | weight of contents and/or vessel depending on support detail (in N) |
| $X_{LAM}$ | overall unit modulus (in N/mm) of laminate under consideration [see equation (12)] |
| $X_z$ | unit modulus (in N/mm per $kg/m^2$ glass) of layer of type $z$ |

[a] 1 $N/mm^2$ = 1 $MN/m^2$ = 1 MPa.

### 14.2 Cylindrical and spherical shells subject to internal pressure

The maximum circumferential unit load, $Q_\phi$, shall be determined from equations (7) or (8).

For cylindrical shells:

$$Q_\phi = \frac{pD_i}{2} \tag{7}$$

For spherical shells:

$$Q_\phi = \frac{pD_i}{4} \tag{8}$$

**14.3 Cylindrical shells subject to combined loads**

**14.3.1** *Vertical vessels and tanks.* The maximum longitudinal unit load, $Q_x$, resulting from the combined effects of:

a) total effective pressure;

b) bending moment due to wind loads (calculated in accordance with CP 3:Chapter V-2) or other causes;

c) total weight of vessel or tank, fittings, attachments and contents;

shall be determined from equations (9) and (10).

NOTE 1 Negative values of $Q_x$ denote compressive loads.

NOTE 2 Pressure, $p$, excludes static head. Negative values of $p$ denote external pressure.

For points above the plane of support:

$$Q_x = \frac{pD_i}{4} \pm \frac{4M}{\pi D_i^2} - \frac{W}{\pi D_i} \qquad (9)$$

where

$W$ is the weight of those parts of the vessel or tank and fittings, attachments and fluid content supported above the point considered.

For points below plane of support:

$$Q_x = \frac{pD_i}{4} \pm \frac{4M}{\pi D_i^2} + \frac{W}{\pi D_i} \qquad (10)$$

where

$W$ is the weight of those parts of the vessel or tank and fittings, attachments and fluid content supported below the point considered.

NOTE 3 In equation (10), $W$ normally includes the total fluid content by weight.

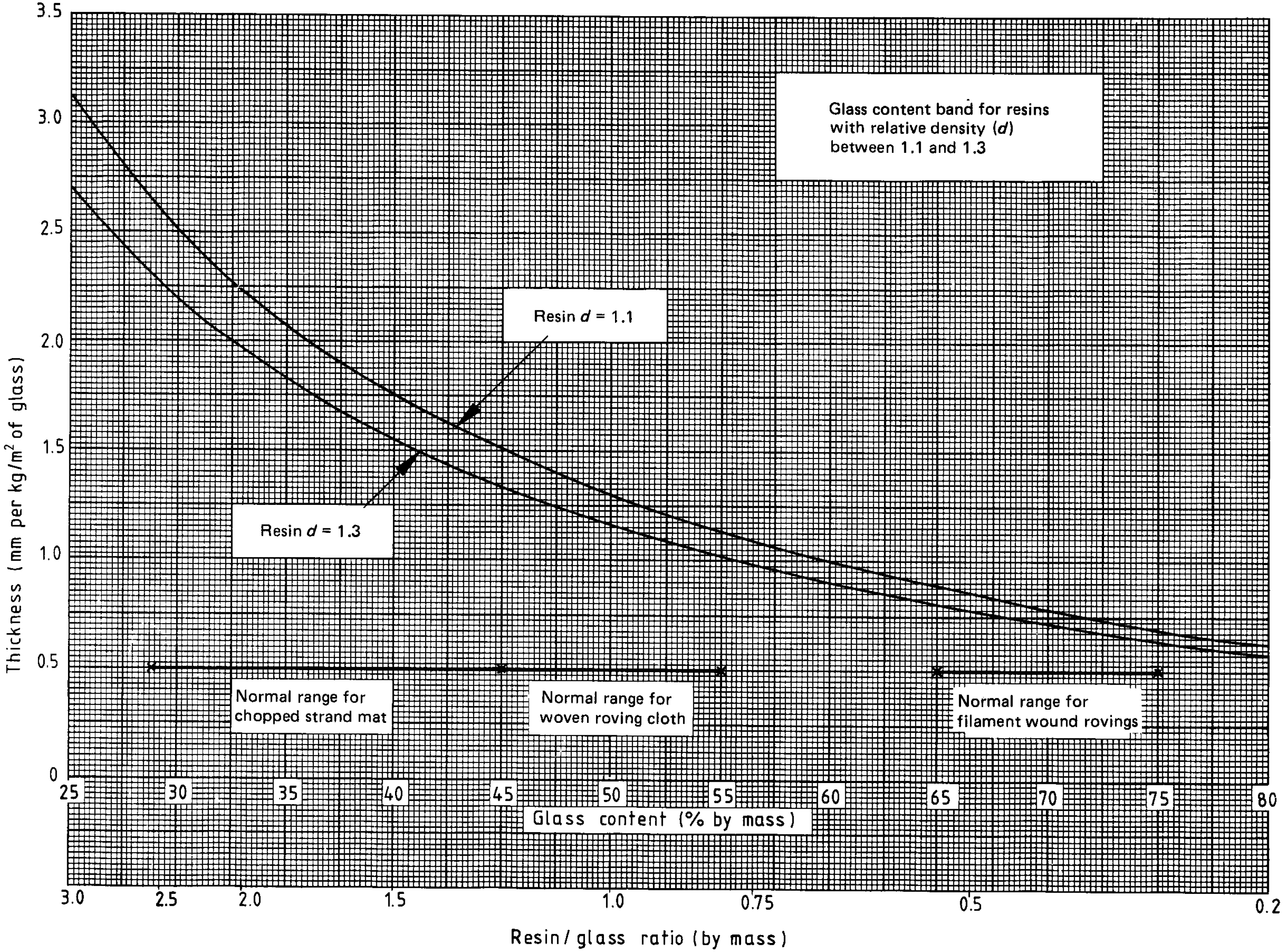



**Figure 5 — Relationship between thickness and glass content**

Values are calculated from the equation:

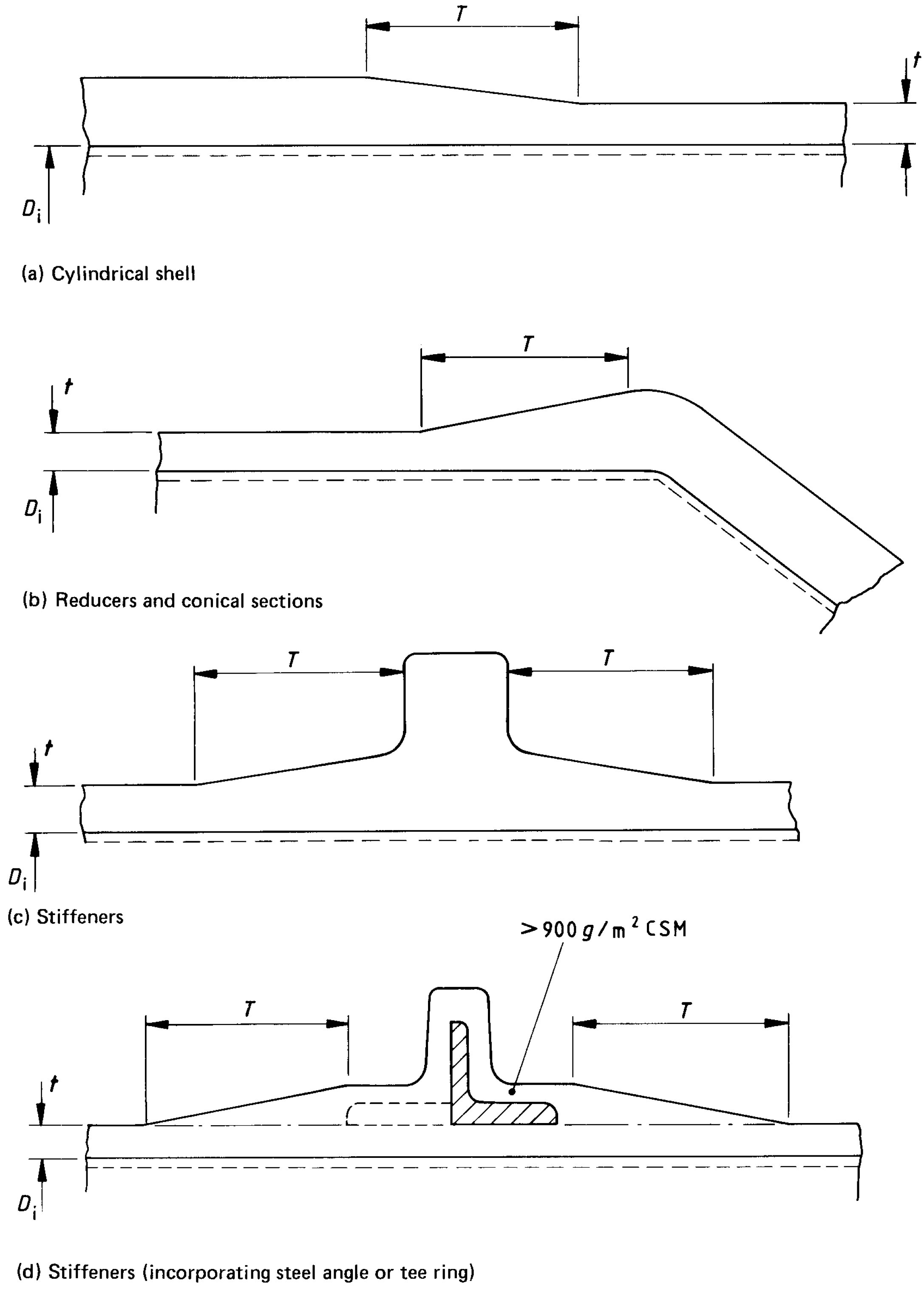
$$\frac{\text{Thickness (mm)}}{\text{Glass mass}} = \frac{1}{2.56} + \frac{100 - m_g}{m_g d}$$

where

$m_g$ is the percentage glass content by mass;
$d$ is the relative density of resin.

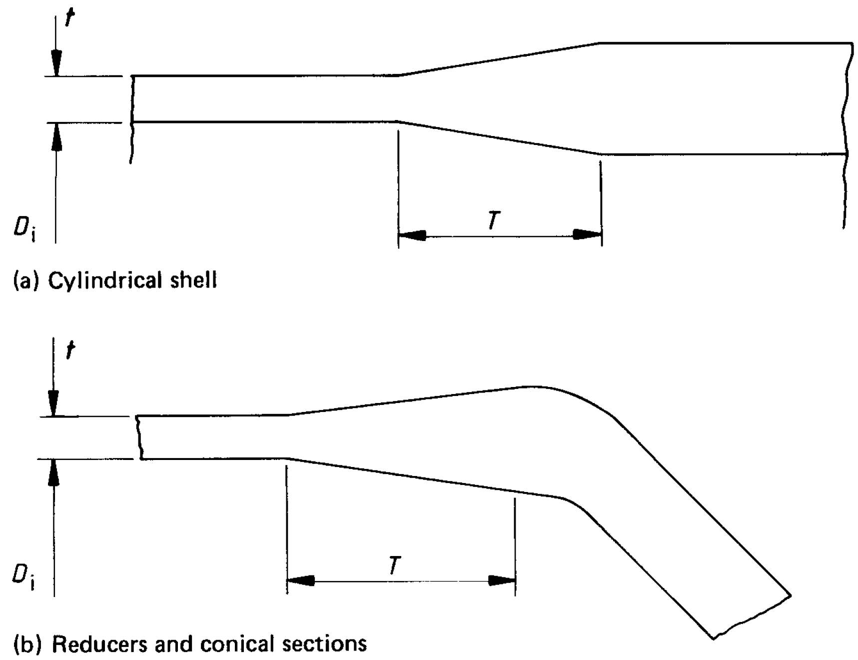
| **Glass content %** | 25 | 30 | 35 | 40 | 45 | 50 | 55 | 60 | 65 | 70 | 75 | 80 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Thickness (Resin $d$ = 1.1) mm per kg/m² of glass** | 3.12 | 2.51 | 2.08 | 1.75 | 1.50 | 1.30 | 1.13 | 1.00 | 0.88 | 0.78 | 0.69 | 0.62 |
| **Thickness (Resin $d$ = 1.3) mm per kg/m² of glass** | 2.70 | 2.19 | 1.82 | 1.54 | 1.33 | 1.16 | 1.02 | 0.90 | 0.80 | 0.72 | 0.65 | 0.58 |

**Figure 5 — Relationship between thickness and glass content** *(concluded)*




NOTE The tapers shown at $T$ are not to be steeper than 1 in 6 and are to extend over a distance not less than $\sqrt{D_i t}$.

**Figure 6 — Single taper blending at changes of thickness (suitable for vessels with or without thermoplastics linings)**




NOTE The tapers shown at $T$ are not to be steeper than 1 in 6 and are to extend over a distance not less than $\sqrt{D_i t}$.

**Figure 7 — Double taper blending at changes of thickness (suitable for vessels without thermoplastics linings)**

**14.3.2** *Horizontal vessels and tanks.* The maximum longitudinal unit load resulting from the combined effects of:

a) total effective pressure;

b) bending moment due to the weight of vessel or tank and contents, as determined by the support configuration;

shall be determined from equation (11).

$$Q_x = \frac{pD_i}{4} \pm \frac{4M}{\pi D_i^2} \qquad (11)$$

The maximum tensile and compressive values of longitudinal unit loads for the full range of possible loading conditions shall be determined.

**14.3.3** *Permissible compressive load.* Where there is a compressive axial load, check calculation shall be made to ensure that the region of the shell subject to the greatest compressive load is adequate to resist collapse by buckling. To make this check the overall unit modulus, $X_{LAM}$, for the laminate construction decided upon shall be determined from equation (12):

$$X_{LAM} = X_1\, m_1\, n_1 + X_2\, m_2\, n_2 + \ldots\, X_z m_z n_z \qquad (12)$$

The permissible maximum compressive unit load shall then be determined from equation (13):

$$Q_p = \frac{0.6 t X_{LAM}}{F D_o} \qquad (13)$$

$F$ shall be taken as 4.

The maximum compressive unit load shall in no case exceed the value of $Q_p$ so obtained.

NOTE The laminate construction may need to be modified, and the necessary calculations repeated until the condition is satisfied.

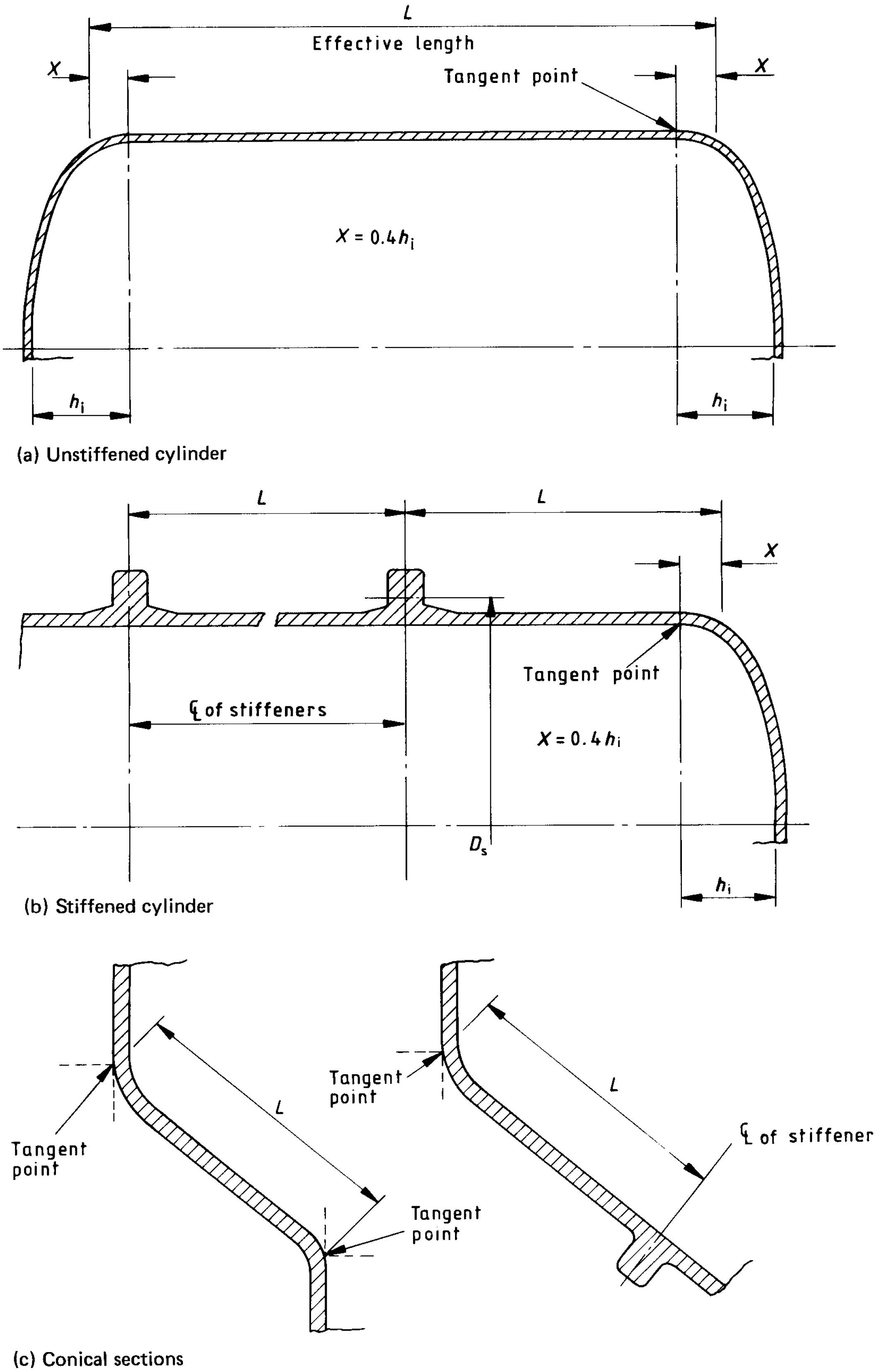



**Figure 8 — Effective length between stiffeners under external pressure**

Special consideration shall be given to the laminates where circumferential and longitudinal modulii are different, e.g. filament-wound tanks or vessels. In such circumstances a rigorous anisotropic analysis shall be carried out to assess the permissible maximum compressive unit load. In the absence of such an analysis the lower of the two modulii shall be used in equation (13).

See also **9.2.7** and **D.4** for cases of simultaneous application of circumferential and longitudinal loads to laminates incorporating filament winding, where a check shall be made using a biaxial design envelope.

**14.4 Cylindrical and spherical shells subject to external pressure**

**14.4.1** *Cylindrical shells.* The circumferential unit load, $Q_\phi$, shall be determined from equation (7). The maximum direct longitudinal unit load, $Q_x$, shall be determined from equations (9), (10) and (11), as appropriate. The greater of these values shall be substituted in equation (6). Using as a basis a laminate construction which complies with this requirement, the total thickness of the laminate, $t$, shall be determined in accordance with **13.3**. The composite modulus of the laminate shall also be determined from equation (14):

$$E_{LAM} = \frac{X_{LAM}}{t} \qquad (14)$$

The minimum permissible laminate thickness, $t_m$, to prevent buckling shall be determined from equations (15) or (16), as appropriate.

If $\frac{L}{D_o} \geqslant 1.35 \left(\frac{E_{LAM}}{pF}\right)^{0.17}$

Then

$$t_m = D_o \left(\frac{pF}{2E_{LAM}}\right)^{0.33} \qquad (15)$$

And if $\frac{L}{D_o} < 1.35 \left(\frac{E_{LAM}}{pF}\right)^{0.17}$

Then

$$t_m = D_o \left(\frac{0.4pF}{E_{LAM}} \times \frac{L}{D_o}\right)^{0.40} \qquad (16)$$

$F$ shall be taken as 4.

It shall be ensured that $t$ is not less than $t_m$.

NOTE $t_m/D_o$ increases as $L/D_o$ increases up to a limiting value. Beyond this value, $t_m/D_o$ is independent of $L/D_o$.

If the proposed design does not comply with this requirement, the design shall be changed either by redesigning the laminate or by providing additional stiffening rings. The calculations shall then be repeated until an acceptable construction is indicated.

**14.4.2** *Cylindrical shells with stiffening rings.* Where the calculations in **14.4.1** indicate an unacceptable laminate thickness it may be preferable to redesign to include stiffening rings and the first step shall be to obtain a new effective shell length, $L$, from:

a) direct subdivision of the effective length of the unstiffened shell;

b) requirements for the support of internal fittings, such as trays, etc.; or

c) considerations of the design of the supports, in the case of horizontal vessels or tanks.

For a proposed stiffening ring profile and composition it is then necessary to determine the diameter, $D_s$, of its neutral axis.

It shall then be ensured that the second moment of area $I_{s2}$ of the stiffening ring section is greater than or equal to:

$$0.18 D_o L D_s^2 \frac{p}{E_{LAM}} \qquad (17)$$

The appropriate $E_{LAM}$ shall be determined from equation (14).

Where a steel stiffening ring fully bonded to the shell [see Figure 6(d)] is used, the modulus for steel shall replace $E_{LAM}$. In this case the steel section only shall be considered as the stiffener.

The permissible length of shell, $L_s$, which may be regarded as effectively contributing to the second moment of area of the stiffening ring section shall be:

$$L_s = 0.75\sqrt{tD_o} \qquad (18)$$

but in no case shall $L_s$ be taken greater than $L$.

Stiffening rings shall extend completely round the circumference of the shell and any joints shall be so designed as to develop the full stiffness of the ring. There shall be no spaces between shell and ring.

The design shall be checked for stresses in the GRP caused by restraint to thermal expansion/contraction by stiffening rings where the design temperature is above 60 °C or below 0 °C.

**14.4.3** *Spherical shells.* The unit circumferential load, $Q_\phi$, shall be calculated from equation (8) and this value then substituted in equation (6). Using as a basis a laminate construction which complies with this requirement, the total thickness of the laminate, $t$, shall be determined in accordance with **13.3**.

The appropriate $E_{LAM}$ shall be determined from equation (14).

The minimum permissible laminate thickness, $t_m$, to prevent buckling shall be determined from equation (19):

$$t_m = 1.7R_o\sqrt{\frac{pF}{E_{LAM}}} \qquad (19)$$

$F$ shall be taken as 4.

It shall be ensured that $t$ is not less than $t_m$.

If the proposed design does not comply with this requirement, the laminate shall be redesigned and the calculations repeated until an acceptable laminate, construction is indicated.

## 15 Shells subjected to wind loading

### 15.1 Symbols

For the purposes of clause **15** the following symbols apply.

$D_o$ outside diameter of shell (in mm)

$F$ factor of safety against collapse by buckling

$h$ actual height of course or part of course under consideration (in mm)

$h_e$ equivalent height (in mm) at thickness $t_{min}$ of the shell course or part of course of thickness $t_h$ and height $h$ under consideration

$h_l$ actual height (in mm) from stiffener to top of course in which it is positioned

$h_{le}$ equivalent height (in mm) at thickness $t_{min}$ from stiffener to top of equivalent height course in which it is positioned

$H$ ($H_1$,$H_2$, etc.) equivalent height(s) (in mm) from proposed position(s) of stiffener(s) to top equivalent height tank

$H_A$ actual height (in mm) from proposed position(s) of stiffener(s) to top of tank

$H_e$ total equivalent height (in mm) at thickness $t_{min}$ of all shell courses of thickness(es) $t_{h1}$, $t_{h2}$ . . . $t_{hx}$ and height(s) $h_1$, $h_2$, . . ., $h_x$

$H_p$ maximum equivalent height (in mm) at thickness $t_{min}$ over which secondary stiffeners are not required

$L$ effective shell length (in mm) at thickness $t_{min}$ between stiffeners or other points of support

$p$ total external pressure (in N/mm$^2$), i.e. wind pressure plus any internal vacuum

$t_h$ thickness of course under consideration (in mm)

$t_{min}$ thickness of top course (in mm)

$X_{LAM\,h}$ overall unit modulus (in N/mm) of course under consideration

$X_{LAM\,min}$ overall unit modulus (in N/mm) of top course

### 15.2 Design calculations

As many vessels and tanks designed and constructed in accordance with this standard are situated in the open, consideration shall be given to wind loading which subjects the vessels and tanks not only to an overturning moment (requirements for which are specified in Appendix F) but to the local collapsing effect of the wind pressure to which, because of the relatively low elastic modulus of GRP, they are particularly susceptible.

The laminate thickness(es) of such vessels and tanks shall be checked for suitability for this pressure in accordance with the requirements for external pressure contained in this section and where there is also an internal vacuum condition, the pressure, $p$, used in the calculations shall be the sum of the wind pressure and the vacuum.

The wind pressure shall be determined in accordance with CP 3:Chapter V-2.

Tanks designed for static head of liquid content only are frequently designed with shell courses of varying laminate construction and thickness and in such cases an equivalent height of courses shall be established to determine the value of $L$, to be used in equations (15), (16) and (17) in conjunction with $X_{LAM}$ and thickness values for the top, weakest, course as follows.

$L = \Sigma h_e$ between adjacent stiffeners or supports (20)

where for each course or part thereof of height $h$ and thickness $t_h$:

$$h_e = h \left(\frac{X_{LAM\,min}}{X_{LAM\,h}}\right) \left(\frac{t_{min}}{t_h}\right)^{1.5} \quad (21)$$

The vertical positioning of secondary rings shall be calculated by first determining the height of a complete tank shell of equivalent stability, at the same diameter, and of the same thickness as the top course of the shell. An analysis of this equivalent tank shell in association with the required wind and vacuum design criteria shall be used to determine the number of secondary rings required.

In many cases these rings will be located on the top course, or on a course of similar thickness, but if the location is not on such a course the actual positioning shall be determined by converting back the equivalent shell course heights to their actual values.

A secondary ring shall not be located within 150 mm of a main circumferential tank seam or change of section.

In line with BS 2654, the equivalent height of the tank shall be the summation of all individual courses, i.e.:

$H_e = \Sigma h_e$ between tank base and equivalent tank top (22)

Values of $h_e$ are determined from equation (21).

Secondary wind stiffeners shall be necessary when the ratio of $H_e/H_p > 1$.

where

$$H_p = \frac{X_{LAM\,min}\, t_{min}^{1.5}}{0.4 F p D_o^{1.5}}$$

$F$ shall be taken as 4.

To obtain the number of secondary stiffeners, the integer of the ratio $H_e/H_p$ shall be used, i.e. if $H_e/H_p = 2.5$ then two wind stiffeners are required spaced, for example, at $H_1 = H_e/3$ and $H_2 = 2H_e/3$ from the top.

To find the actual position of the stiffeners the following method of calculation shall be used.

a) A table of relevant details including tank course height(s) $h$ and the corresponding equivalent height(s) $h_e$ shall be constructed (see **D.7**).

b) The course(s) in which the stiffener(s) will be positioned shall be located by inspection of the table.

c) The position(s) of stiffener(s) with the course(s) shall be determined as follows:

$h_{le} = H - \Sigma h_e$ between top of stiffened course under consideration and equivalent tank top

The true distance of the stiffener from the stiffened course top, $h_l$, shall be derived from equation (21):

$$h_l = h_{le} \left(\frac{X_{LAM\,h}}{X_{LAM\,min}}\right) \left(\frac{t_h}{t_{min}}\right)^{1.5}$$

The true distance, $H_A$, of the stiffener from the actual tank top shall be determined from equation (23):

$$H_A = h_1 + \Sigma h \quad \text{between top of stiffened course under consideration and actual tank top} \qquad (23)$$

NOTE A worked example is provided in Appendix D.

In the case of open-top or fixed-roof tanks, adequate primary stiffening at the upper edge of the top course either by external stiffener or inherent stiffness in the shell/roof junction shall be provided. The second moment of area of such stiffening, and of secondary stiffeners shall comply with **14.4**.

## 16 Rectangular tanks

### 16.1 Small unstiffened rectangular tanks

Small unstiffened rectangular tanks, without supporting ribs or frames shall be considered as a number of individual panels joined at the corners. Each such panel shall be designed in accordance with clause **17** using type 2 edge coding constants appropriate to its proportions, edge conditions and type of loading.

### 16.2 Stiffened rectangular tanks

It is permissible for the arrangement of stiffening ribs to be bidirectional but for economy the ribs are usually arranged in the vertical or horizontal axes; however arranged, the stiffening ribs may be fabricated into a framework. Such constructions shall be considered as a number of individual, rectangular panels, and designed in accordance with clause **17**. For calculation purposes the load to be carried by each rib shall be the load carried by the half of each adjacent panel, due account being taken of any snow or wind loads.

### 16.3 Design of stiffening ribs

**16.3.1** *General.* Stiffening ribs shall be of steel or reinforced plastics construction. In either case the proportions of these integral ribs shall be calculated using conventional bending theory.

The design shall be checked for stresses in the GRP caused by restraint to thermal expansion/contraction by stiffening ribs where the design temperature is above 60 °C or below 0 °C.

**16.3.2** *Steel stiffening ribs.* All assemblies of ribs shall be designed to ensure adequate strength at all welded junctions, and at butt welded connections the edges to be welded shall be so prepared to ensure full penetration welds developing the full strength of the sections.

The calculated stress in the steel section shall not exceed 0.25 × tensile strength. The calculated stress in fillet welds shall not exceed 0.1 × tensile strength of the steel section. The load-carrying area of a fillet weld shall be the product of its length and throat thickness, where the throat shall be taken as not more than 0.7 × weld leg length.

The calculated stress in butt welds shall not exceed 0.16 × tensile strength.

**16.3.3** *GRP stiffening ribs.* The ribs shall be fabricated using only CSM as the reinforcing material. The overlay laminate shall be extended either side of the rib for a sufficient distance to carry all shear loads and shall not be less than 75 mm.

It is permissible to regard the panel to which the rib is overlayed as contributing to the strength of the rib for a distance not exceeding 16 × panel thickness on either side of the rib provided there is such a clear distance available. No part of the panel shall be regarded as contributing to more than one rib. The material over which the rib is formed shall not be regarded as contributing to the strength of the rib.

## 17 Flat panels

### 17.1 General

Flat panels designed in accordance with this standard shall be manufactured using only chopped strand or CSM as the reinforcing material.

The equations in this clause are based on a consideration of both the maximum allowable unit skin loading, tensile or compressive, and an over-riding limiting deflection of $t$, where $t$ is the thickness of the panel and $t$ shall always be equal to or greater than $t_{min}$, where $t_{min}$ is the required minimum thickness of the panel.

NOTE 1 It is considered that the membrane component of loading arising when the maximum deflection equals the panel thickness does not significantly affect the validity of the bending analysis from which the equations were derived.

NOTE 2 Whilst it is recognized that edge conditions can be of great significance when calculating in this manner, it is considered that in many cases the effect of change in such conditions is only marginal and for the purposes of this clause the edge conditions have been directed into type 1 and type 2. These are illustrated in Figure 9 and Figure 10.

The edge conditions for a particular panel shall be either type 1 or type 2 or a combination of both depending on conditions.

### 17.2 Symbols

For the purposes of clause **17** the following symbols apply.

| | |
|---|---|
| $a$ | panel dimension, longest span (length) (in mm) (see Figure 9 and Figure 10) |
| $b$ | panel dimension, shortest span (breadth) (in mm) (see Figure 9 and Figure 10) |
| $c$ | depth of core (in mm) |
| $d$ | depth of sandwich (in mm) between neutral axes of faces |
| $D_p$ | panel dimension, circular panels (diameter) (in mm) (see Figure 9 and Figure 10) |
| $E_{LAM}$ | Young's modulus (in N/mm$^2$) of laminate under consideration |
| $F$ | factor of safety |
| $m_{CSM}$ | total mass of CSM reinforcement (in kg/m$^2$) |
| $M_d$ | design moment = total moment induced in the panel by the worst combination of distributed loads plus moment due to any local load (in N·mm) |
| $M_l$ | moment due to local load (in N·mm) |
| $M_p$ | moment due to distributed load (in N·mm) |
| $p$ | pressure (particular definitions are given throughout the text) (in N/mm$^2$) |
| $Q_s$ | maximum unit load (in N/mm) in skin of laminate |
| $r_l$ | radius of local load (in mm) |
| $r_p$ | panel radius (in mm) or in the case of an isosceles triangular plate, the length of one of the equal sides |
| $S$ | maximum shear force per unit width (in N/mm) acting on panel |
| $t$ | solid panel thickness (in mm) |
| $t_g$ | laminate thickness (in mm per kg/m$^2$ glass) determined from Figure 5 |
| $t_{min}$ | required minimum thickness of panel (in mm) |
| $t_s$ | skin thickness (in mm) |
| $t_t$ | total thickness (in mm) of sandwich to outside of the structural layers |
| $u_{CSM}$ | design unit loading for CSM layers (in N/mm per kg/m$^2$ glass) |
| $W_l$ | applied local load (in N) |
| $\alpha$ | constant appropriate to combination of edge conditions and type of loading |
| $\beta$ | constant appropriate to combination of edge conditions and type of loading |
| $\tau_c$ | maximum shear stress (in N/mm$^2$) in core and in skin to core interface |

### 17.3 Panel shapes and loads

The design of flat panels shall be undertaken in accordance with **17.4** to **17.10**, which cover panels of circular, rectangular, triangular and segmented shape subjected to uniformly distributed loads and uniformly varying loads, as in the case of a static head of liquid, and circular and rectangular panels for concentrated loads applied to the centre of the panels.

NOTE 1 This provides for panels subject to access loadings as given in BS 6399-1 and gives a worst case for fixed direct loads.

For loads remote from the panel centre, designs shall be based upon reference [11] for rectangular panels and references [11] and [12] for circular panels.

NOTE 2 The effects of external moments on flat panels cannot readily be assessed and such loadings should be avoided whenever possible.

### 17.4 Design method for all panel shapes

The design moment, $M_d$, shall be the total moment induced in the panel by the worst combination of distributed loads that could occur, e.g. pressure, vacuum, static head, wind and snow loads, plus the moment due to any local load, for example:

$$M_d = M_{vacuum} + M_{snow} + M_{wind} + M_{local\ load}$$

A check calculation shall be made to ensure that the thickness of the resulting laminate is such that the deflection of the panel does not exceed the panel thickness.

### 17.5 Rectangular panels

**17.5.1** *General.* One of the following four arrangements of edge conditions shall be adopted. The considered and appropriate constants shall be as given in Table 8.

*Case A.* Type 1 edge conditions on all four sides. For example, a bolted cover with a joint extending beyond the bolt holes or one of an assembly of panels with balanced fixing at stiffeners or corners.

*Case B.* Type 1 edge conditions on both short edges and one long edge with type 2 edge conditions on the other long edge. For example, one of an assembly of panels forming the side of an open topped tank with one long edge carried by the tank top flange or support frame.

*Case C.* As an alternative to case B, panels with type 1 edge conditions on two long edges and one short edge and type 2 on the other short edge.

*Case D.* Type 2 edge conditions on all four sides, e.g. a loose cover plate.

**17.5.2** *Rectangular panels under distributed load.* The moment, $M_p$, due to a uniformly distributed load on a rectangular panel shall be determined from equation (24):

$$M_p = \beta_1 p b^2 \qquad (24)$$

where

$\beta_1$ is obtained from Table 8;

$p$ is the uniform internal or external pressure, a uniform load such as snow or wind or such a combination of these as will give rise to the greatest differential pressure across the panel with due regard being taken to the sign or direction of application of each form of load.

The moment, $M_p$, due to a static head of liquid decreasing from maximum pressure at the base of the panel to zero at the top shall be determined from equation (25):

$$M_p = 0.67 \beta_1 p b^2 \qquad (25)$$

where

$\beta_1$ is obtained from Table 8;

$p$ is the pressure at the base of the panel, i.e. the maximum pressure on the panel.

The moment, $M_p$, due to a static head of liquid decreasing from a maximum pressure at the base of the panel but which is not zero at the top shall be determined from equation (24) using the maximum value of pressure for the panel concerned.

**17.5.3** *Rectangular panels under central local loads.* Cases A and D only shall be considered when panels are subjected to local loads.

NOTE Panels of case B may be considered as case D and panels of case C may be considered as case A for local loads only.

The moment, $M_l$, due to local load on a rectangular panel shall be determined from the greater value given by equations (26) and (27) for case A and equation (26) only for case D:

$$M_l = \frac{W_l}{4\pi}\left(1.3 \ln \frac{2b}{\pi r_l} + \beta_2\right) \qquad (26)$$

$$M_l = \beta_3 W_l \qquad (27)$$

where

$\beta_2$ and $\beta_3$ are obtained from Table 9.

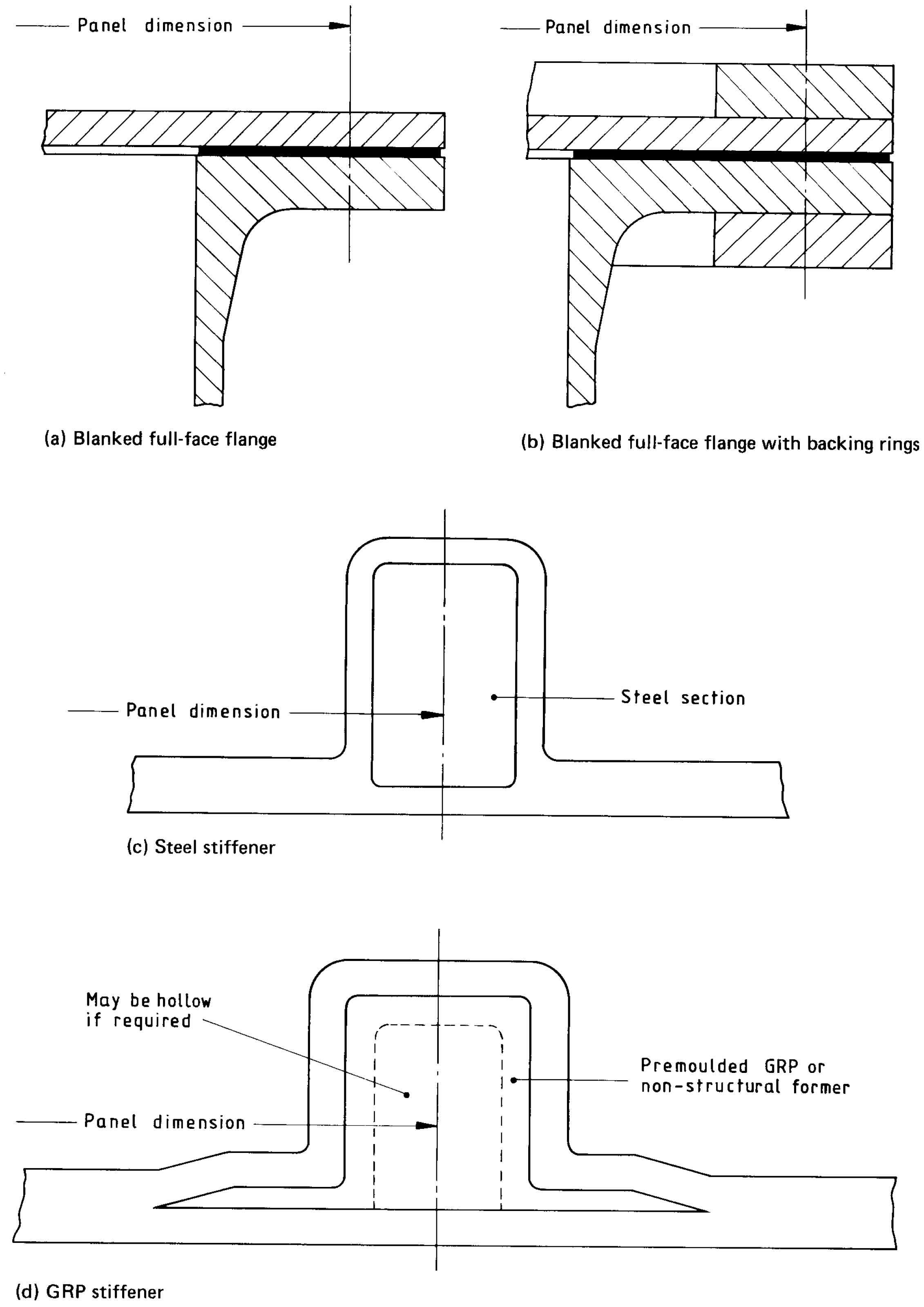



**Figure 9 — Type 1 edge fixings**

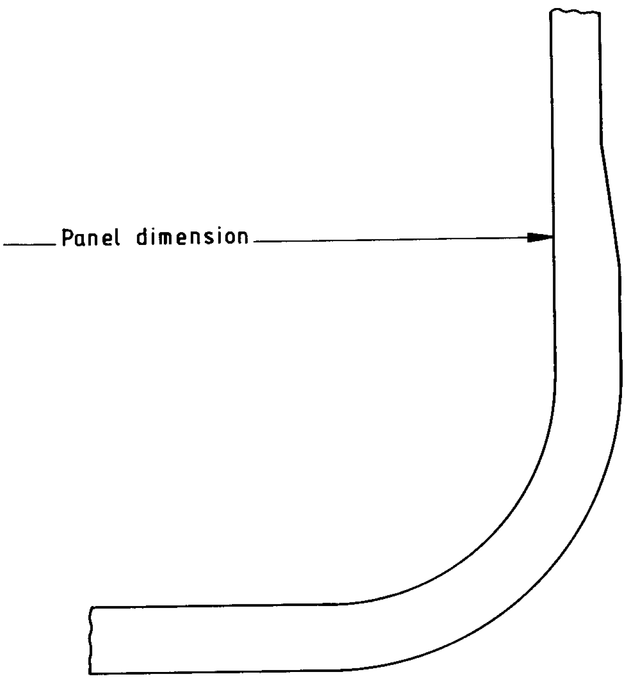



**Figure 9 — Type 1 edge fixings** *(concluded)*

**17.5.4** *Mass of reinforcement.* The total mass of CSM reinforcement, $m_{CSM}$ (in kg/m$^2$), shall be determined from equation (28):

$$m_{CSM} = \left( \frac{6M_d}{u_{CSM} \times t_g} \right)^{0.5} \quad (28)$$

where

$M_d$ is the total moment as defined in **17.4**.

A check of the laminate thickness shall be made to ensure that the deflection of the panel does not exceed the panel thickness. The required minimum thickness of the panel shall be determined from equation (29):

$$t_{min} = \left( \frac{\alpha_1 p b^4}{E_{LAM}} + \frac{\alpha_2 W_l b^2}{E_{LAM}} \right)^{0.25} \quad (29)$$

where

$\alpha_1$ is obtained from Table 8;

$\alpha_2$ is obtained from Table 9;

$p$ is the design unit pressure (worst combination of pressure, vacuum, snow, wind loading, etc.).

### 17.6 Circular panels

**17.6.1** *Circular panels under uniformly distributed load.* The moment, $M_p$, due to uniformly distributed load shall be determined from equation (30):

$$M_p = \beta_1 D^2 p \quad (30)$$

where

$\beta_1$ is a constant equal to:

0.03 125 for type 1 edge conditions;

0.0516 for type 2 edge conditions;

$p$ is the design unit pressure (worst combination of pressure, vacuum, snow, wind loading, etc.).

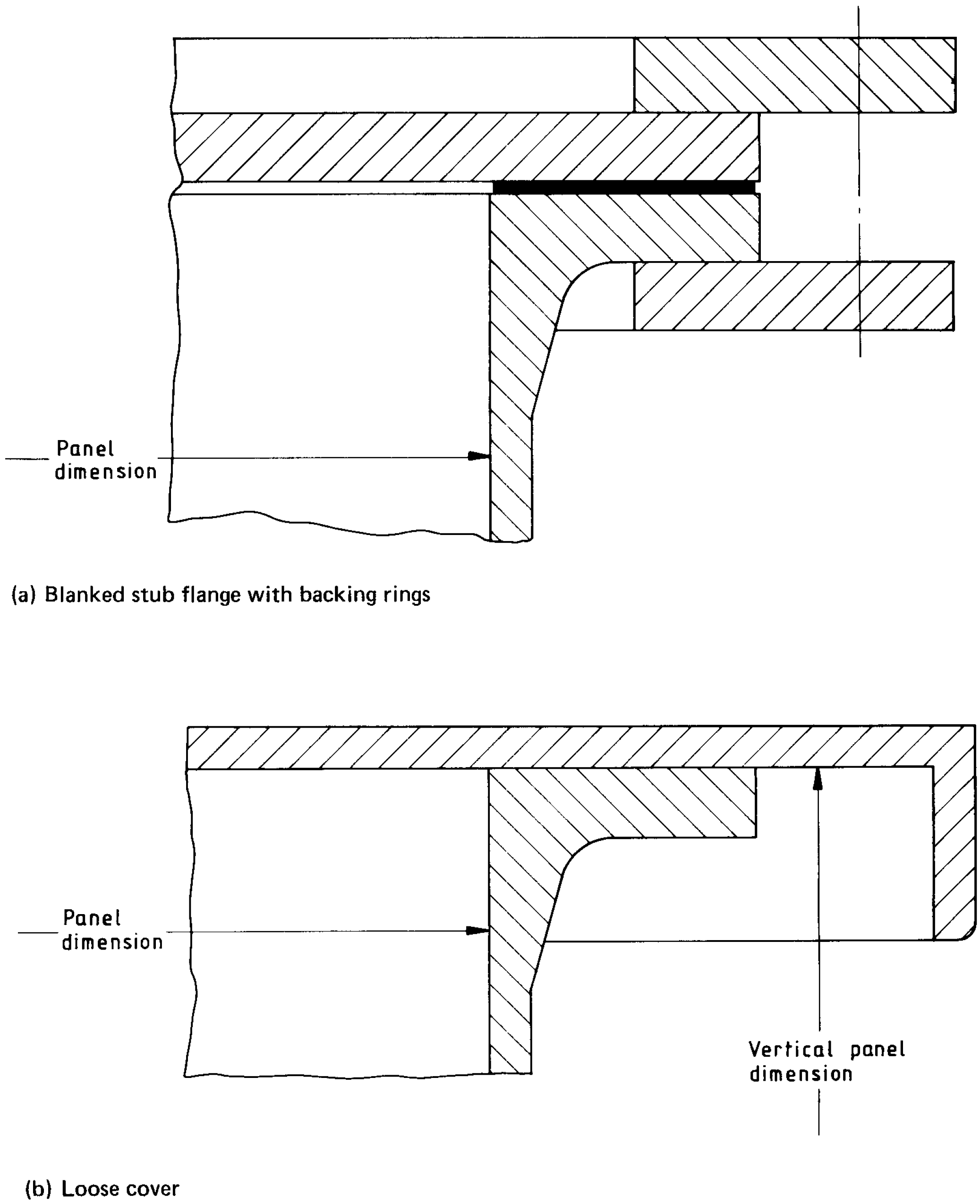



**Figure 10 — Type 2 edge fixings**

**Table 8 — Deflection and bending moment constants for flat plates**

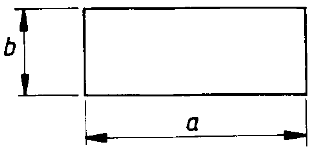
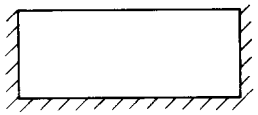
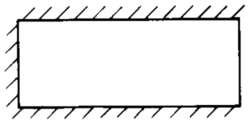
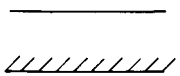
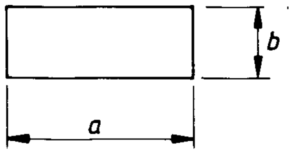
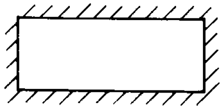
| Boundary condition | Constant | Length/breadth ratio *a/b* 1.0 | 1.25 | 1.5 | 1.75 | 2.0 | 2.5 | 3.0 | 4.0 | 5.0 | > 5.0 | Case |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
|   | $\beta_1$ | 0.048 | 0.067 | 0.081 | 0.093 | 0.102 | 0.113 | 0.119 | 0.125 | 0.125 | 0.125 | D |
| | $\alpha_1$ | 0.044 | 0.060 | 0.084 | 0.099 | 0.111 | 0.126 | 0.134 | 0.140 | 0.142 | 0.142 | D |
|  | $\beta_1$ | 0.051 | 0.067 | 0.075 | 0.080 | 0.083 | 0.083 | 0.083 | 0.083 | 0.083 | 0.083 | A |
| | $\alpha_1$ | 0.014 | 0.020 | 0.024 | 0.026 | 0.028 | 0.0281 | 0.0282 | 0.0284 | 0.0284 | 0.0284 | A |
|  | $\beta_1$ | 0.060 | 0.077 | 0.094 | 0.107 | 0.114 | 0.121 | 0.124 | 0.124 | 0.124 | 0.124 | B |
| | $\alpha_1$ | 0.017 | 0.028 | 0.037 | 0.044 | 0.049 | 0.055 | 0.057 | 0.058 | 0.058 | 0.058 | B |
|  | $\beta_1$ | 0.060 | 0.073 | 0.079 | 0.082 | 0.083 | 0.083 | 0.083 | 0.083 | 0.083 | 0.083 | C |
| | $\alpha_1$ | 0.017 | 0.022 | 0.025 | 0.027 | 0.028 | 0.0284 | 0.0284 | 0.0284 | 0.0284 | 0.0284 | C |
|  | Type 2 fixing, simply supported<br>Type 1 fixing, built-in edge | | | | | | | | | | | |

**Table 9 — Deflection and bending moment constants for central local loads on rectangular flat plates**

| Boundary condition | Constant | Length/breadth ratio *a/b* | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | **1.0** | **1.25** | **1.5** | **1.75** | **2.0** | **2.5** | **3.0** | **4.0** | **5.0** | **> 5.0** | **Case** |
|  | $\alpha_2$ | 0.1267 | 0.1518 | 0.1671 | 0.1763 | 0.1805 | 0.1821 | 0.1831 | 0.1842 | 0.1849 | 0.1851 | D |
| | $\beta_2$ | 0.435 | 0.691 | 0.838 | 0.917 | 0.958 | 0.976 | 0.986 | 0.997 | 1.000 | 1.000 | D |
|  | $\alpha_2$ | 0.0611 | 0.0720 | 0.0768 | 0.0785 | 0.0788 | 0.0790 | 0.0791 | 0.0791 | 0.0791 | 0.0791 | A |
| | $\beta_2$ | – 0.238 | – 0.052 | 0.036 | 0.067 | 0.067 | 0.067 | 0.067 | 0.067 | 0.067 | 0.067 | A |
| | $\beta_3$ | 0.1257 | 0.1530 | 0.1632 | 0.1665 | 0.1674 | 0.1677 | 0.1679 | 0.1680 | 0.1680 | 0.1680 | A |
| ——— | Type 2 fixing, simply supported | | | | | | | | | | | |
| /////// | Type 1 fixing, built-in edge | | | | | | | | | | | |

**17.6.2** *Circular panels under central local load.*

The moment $M_l$ due to local load shall be determined from the greater of equations (31) and (32) for type 1 edge condition and equation (33) only for type 2 edge condition.

Type 1:

$$M_l = 0.325 \frac{W_l}{\pi} \ln \frac{D_p}{2r_l} \quad (31)$$

or

$$M_l = \frac{W_l}{4\pi} \quad (32)$$

Type 2:

$$M_l = \frac{W_l}{4\pi}\left(1.3 \ln \frac{D_p}{2r_l} + 1\right) \quad (33)$$

**17.6.3** *Mass of reinforcement.* The required mass of CSM reinforcement, $m_{CSM}$ (in kg/m$^2$), shall be determined from equation (34):

$$m_{CSM} = \left(\frac{6M_d}{u_{CSM} \times t_g}\right)^{0.5} \quad (34)$$

A check of the laminate thickness shall be made to ensure that the deflection of the panel does not exceed its thickness.

The minimum laminate thickness shall be determined from equation (35):

$$t_{min} = \left(\frac{\alpha_1 p D_p^4}{E_{LAM}} + \frac{\alpha_2 W_l D_p^2}{E_{LAM}}\right)^{0.25} \quad (35)$$

where

$\alpha_1$ is a constant equal to:
- 0.01066 for type 1 edge condition;
- 0.04347 for type 2 edge condition;

$\alpha_2$ is a constant equal to:
- 0.05431 for type 1 edge condition;
- 0.13787 for type 2 edge condition;

$p$ is the design unit pressure (worst combination of pressure, vacuum, snow, wind loading, etc.).

**Table 10 — $\alpha$ and $\beta$ constants for sector and triangular plates**

| Angle | π | π/2 | π/3 | π/4 |
|---|---|---|---|---|
| **Type 1. Fixing on sector edges** | | | | |
| $\alpha$ | 0.0368 | 0.0144 | 0.0062 | 0.0031 |
| $\beta$ | 0.0756 | 0.0488 | 0.034 | 0.025 |
| **Type 2. Fixing on all sides** | | | | |
| $\alpha$ | 0.0886 | 0.0246 | 0.01 | 0.0054 |
| $\beta$ | 0.0868 | 0.0381 | 0.0255 | 0.0183 |

## 17.7 Plates in the form of a sector of a circle and triangular plates

Plates in the form of a sector of a circle and triangular plates shall be designed using the values of ($\alpha$ and $\beta$ obtained from Table 10.

The moment on the panel, $M_p$, shall be determined from equation (36):

$$M_p = \beta p r_p^2 \quad (36)$$

where

$\beta$ is obtained from Table 10;

$p$ is the design uniform pressure (worst combination of pressure, vacuum, snow, wind loading, etc.).

The required mass of CSM reinforcement, $m_{CSM}$, is then determined from equation (37):

$$m_{CSM} = \left(\frac{6M_p}{u_{CSM} \times t_g}\right)^{0.5} \qquad (37)$$

A check of the panel deflection shall be made using equation (38) to ensure the deflection does not exceed the panel thickness:

$$t_{min} = r_p \left(\frac{\alpha p}{E_{LAM}}\right)^{0.25} \qquad (38)$$

where

$\alpha$ is obtained from Table 10.

**17.8 Sandwich construction panels**

**17.8.1** *General.* A sandwich construction panel shall consist of two reinforced plastics skins separated by and bonded to a core of low load-carrying capability in order to improve the strength and stiffness of the flat panel with economy of reinforcement material.

In these calculations the core material is not considered to contribute any strength to the panel. However, its ultimate shear strength in any direction shall not be less than 0.2 $N/mm^2$ as determined in accordance with **B.13**.

**17.8.2** *Design conditions.* Where flat sandwich panels are used as an alternative to solid panels, the following conditions shall be satisfied.

a) The thickness of each skin shall in no case be less than the thickness specified in **13.3**.

b) The ultimate shear strength of the core shall not be less than 0.2 $N/mm^2$.

c) The design temperature shall not exceed 60 °C.

d) The flexural rigidity of the core shall be ignored and the skins shall be composed of CSM unless more detailed analysis of the panel is undertaken.

NOTE Consideration should be given to the ability of the core material to withstand local compressive loads at the intersections of panels and stiffening ribs.

**17.8.3** *Design calculations.* In order to obtain an initial construction, the thickness required for a solid panel shall be determined (necessary to limit the deflection to the panel thickness) in accordance with **17.4**. The overall thickness, $t_t$, of the sandwich panel shall then be obtained by assuming a core thickness and using equation (39):

$$t_t = (t^3 + c^3)^{0.33} \qquad (39)$$

The skin thickness, $t_s$, shall then be determined from equation (40):

$$t_s = \frac{(t_t - c)}{2} \qquad (40)$$

In no case shall $t_s$ be less than that specified in **17.8.2** a).

The skin unit loading shall be checked to ensure that the design unit loading is not exceeded using equation (41):

$$Q_s = \frac{M_d t_t}{d^2} \qquad (41)$$

This value of unit load shall be substituted in equation (6).

In addition the core shear stress shall be determined from equation (42):

$$\tau_c = \frac{S}{d} \qquad (42)$$

where the maximum shear force $S$ is determined for the loading present. $\tau_c$ shall not exceed:

$$\frac{\text{Ultimate shear strength of the core}}{F}$$

or

$$\frac{\text{Ultimate shear strength between the core and the skin}}{F}$$

$F$ shall be taken as not less than 4.

NOTE See worked example in Appendix D.

### 17.9 Panels at design temperature above ambient

For panels with design temperature above ambient, the following design cases shall be considered:

a) ambient temperature, design loading;

b) design temperature, design loading;

c) design temperature, no loading.

NOTE Further information on calculating thermally induced moments and deflections is given in reference [12].

## 18 Ends

### 18.1 Symbols

For the purposes of clause **18** the following symbols apply.

$A_b$ actual total cross-sectional area (in mm²) of bolts at root of thread or section of least diameter under stress

$A_m$ total required cross-sectional area of bolts (in mm²), taken as greater of $A_{m1}$ and $A_{m2}$

$A_{m1}$ total cross-sectional area of bolts (in mm²) at root of thread or section of least diameter under stress, required for operating conditions, $= W_{m1}/S_b$

$A_{m2}$ total cross-sectional area of bolts (in mm²) at root of thread or section of least diameter under stress, required for gasket seating, $= W_{m2}/S_a$

$b$ effective gasket width, $= 2.52\sqrt{N/2}$ (in mm)

$B$ inside diameter of flange (in mm) [see Figure 14(a)]

$D_m$ gasket mean diameter (in mm) [see Figure 13(a)]

$D$ outside diameter of gasket contact face (in mm)

$D_b$ pitch circle diameter of cover bolts (in mm) [see Figure 13(b) and Figure 14(b)]

$D_i$ inside diameter of shell (in mm)

$D_k$ inside diameter at large base of cone (in mm)

$D_o$ outside diameter of shell (in mm)

$f$ allowable design stress for material used for steel backing plate:

$$\frac{\text{Room temperature tensile strength}}{4}$$

or

$$\frac{\text{Room temperature yield or proof stress}}{1.6}$$

whichever is lower (in N/mm²)

$G$ diameter at location of gasket load reaction, $= D - 2b$ (in mm)

$h_g$ radial distance from gasket load reaction to the bolt circle, $= \dfrac{D_b - G}{2}$ (in mm)

$h_i$ inside head height (in mm), i.e. distance from tangent point between knuckle and parallel skirt (see Figure 11).

For a torispherical head:

$$h_i = R_i - \left\{ \left( R_i - \frac{D_i}{2} \right) \left( R_i + \frac{D_i}{2} - 2r_i \right) \right\}^{0.5}$$

$H$ total hydrostatic end force, = 0.785 $B^2 p$ (in N)

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

$K_{c1}$ shape factor for use with cone/cylinder intersections with knuckle radius, obtained from Table 12 to determine longitudinal unit loading [see equation (48)]

$K_{c2}$ shape factor for use with cone/cylinder intersections without a knuckle, where permitted in accordance with **18.4.1**, obtained from Table 13 to determine longitudinal unit loading [see equation (49)]

$K_e$ shape factor, for use with semi-ellipsoidal ends under external pressure, taken from Figure 15

$K_s$ shape factor for use with domed ends, obtained from Table 11

$L$ effective shell length (in mm)

$L_c$ distance (in mm) from cone/cylinder intersection for which both cone and cylinder are to be increased to thickness $t_c$, $\left( = \sqrt{\dfrac{D_i t_c}{\cos \phi}} \right)$

$m$ gasket factor = 1 for soft rubber without fabric or asbestos reinforcement

$m_{CSM}$ total mass of glass per unit area (in kg/m$^2$) in complete laminate required for ring supported flat covers

$N$ width of gasket in contact with flange face (in mm)

$p$ total effective pressure (in N/mm$^2$), i.e. design pressure including hydrostatic head at point under consideration

$Q$ maximum unit load (in N/mm), i.e. maximum force per unit width, to be carried by the laminate

$Q_{x1}$ longitudinal unit load (in N/mm) for cone/cylinder intersections with knuckle [see equation (48)]

$Q_{x2}$ longitudinal unit load (in N/mm) for cone/cylinder intersections without knuckle, where permitted in accordance with **18.4.1** [see equation (49)]

$r_i$ inside knuckle radius of dished end or of transition between conical end and cylinder (in mm)

$R$ radius of corner (in mm) (see Figure 16)

$R_i$ inside radius of spherical portion of dished end (in mm)

$R_o$ outside radius of spherical shell (in mm)

$S_a$ bolt nominal design stress at atmospheric temperature (see Table 15)

$S_b$ bolt nominal design stress at design temperature (see Table 15)

$t$ laminate thickness of end (in mm) calculated from laminate details

$t_c$ design thickness of cone/cylinder intersection (in mm). It extends along the cone and cylinder for a distance not less than $L_c$ (see **18.4.3**)

$t_g$ thickness per unit mass of glass determined from Figure 5 after glass content has been fixed (in mm per kg/m$^2$ glass)

$t_B$ thickness of steel backing plate (in mm)

$u_{CSM}$ design unit loading (in N/mm per kg/m$^2$ glass) for CSM layers (from **9.2**)

*W* flange design bolt load (in N), based on actual size bolts used to provide the greater of $W_{m1}$ and $W_{m2}$ and ends $\frac{(A_m + A_b) S_a}{2}$

$W_{m1}$ minimum required bolt load for operating conditions (in N), $= H + H_p$

$W_{m2}$ minimum required bolt load for gasket seating (in N), $= 3.14 \times bGy$

*y* is the gasket or joint contact surface unit seating stress = 2.32 N/mm² for soft rubber without fabric or asbestos reinforcement

$\phi$ angle of slope of conical section to vessel axis (see Figure 12)

**18.2 Limitations of shape**

Ends shall be one of the following shapes:

a) hemispherical [see Figure 11(a)];

b) semi-ellipsoidal: the ratio of inside head height, $h_i$, to inside diameter, $D_i$, shall be not less than 0.2 [see Figure 11(b)];

c) torispherical: the inside radius of dishing, $R_i$, shall be not greater than the inside diameter, $D_i$, and the inside knuckle radius, $r_i$, shall be not less than one-tenth of the inside diameter, $D_i$, [see Figure 11(c)];

d) conical [see Figure 12(a) and Figure 12(b)];

e) flat: fully supported or ribbed;

f) bolted flat covers (see Figure 13 and Figure 14).

NOTE The type of end in which the convex surface of the end forms part of the inner surface of the vessel or tank is not covered by this standard.

**18.3 Domed ends**

**18.3.1** *General.* Domed ends shall include shapes listed as **18.2** a), b), and c).

**18.3.2** *Domed ends subject to internal pressure.* The unit load to be substituted in equation (6) shall be determined from equation (43):

$$Q = 0.5p\, D_i\, K_s \tag{43}$$

Since the value of $K_s$ to be used is dependent on the thickness of the domed end, and this thickness cannot be accurately determined until the laminate has been designed, a value for the $t/D_i$ ratio shall be assumed in order to select an appropriate value $K_s$.

The value of $K_s$ so determined shall be substituted in equation (43) to determine the notional unit load, $Q$, and this value of $Q$ used in equation (6) for the determination of a suitable laminate.

The thickness of the laminate so calculated shall then be determined, in accordance with **13.3**, and the value of $t/D_i$ given by this thickness shall be compared with the ratio assumed. If the difference is significant, the calculations shall be repeated using a new value of $K_s$ appropriate to the likely thickness of the final laminate.

NOTE As an alternative to repeating the calculation, it may be found preferable to redesign the end with larger head height, $h_i$, and knuckle radius, $r_i$, thus reducing the value of $K_s$ to be used in the design calculations.

**18.3.3** *Hemispherical and torispherical ends subject to external pressure.* The unit load to be substituted in equation (6) shall be determined from equation (44):

$$Q = 0.66p\, D_i\, K_s \tag{44}$$

Additionally, the minimum permissible laminate thickness, $t_m$, to prevent buckling shall be determined from equation (19).

It shall be ensured that $t$ is not less than $t_m$.

If, in the proposed design, this condition is not fulfilled the laminate shall be redesigned and the calculations repeated until an acceptable laminate construction is indicated.

**18.3.4** *Semi-ellipsoidal ends subject to external pressure.* The unit load to be substituted in equation (6) shall be determined from equation (45):

$$Q = 0.66p\ D_i\ K_s \quad (45)$$

Additionally, the minimum permissible laminate thickness, $t_m$, to prevent buckling shall be determined from equation (19).

The value of $R_o$ to be used in equation (19) shall be determined from equation (46):

$$R_o = 0.5\ D_o\ K_e \quad (46)$$

It shall be ensured that $t$ is not less than $t_m$.

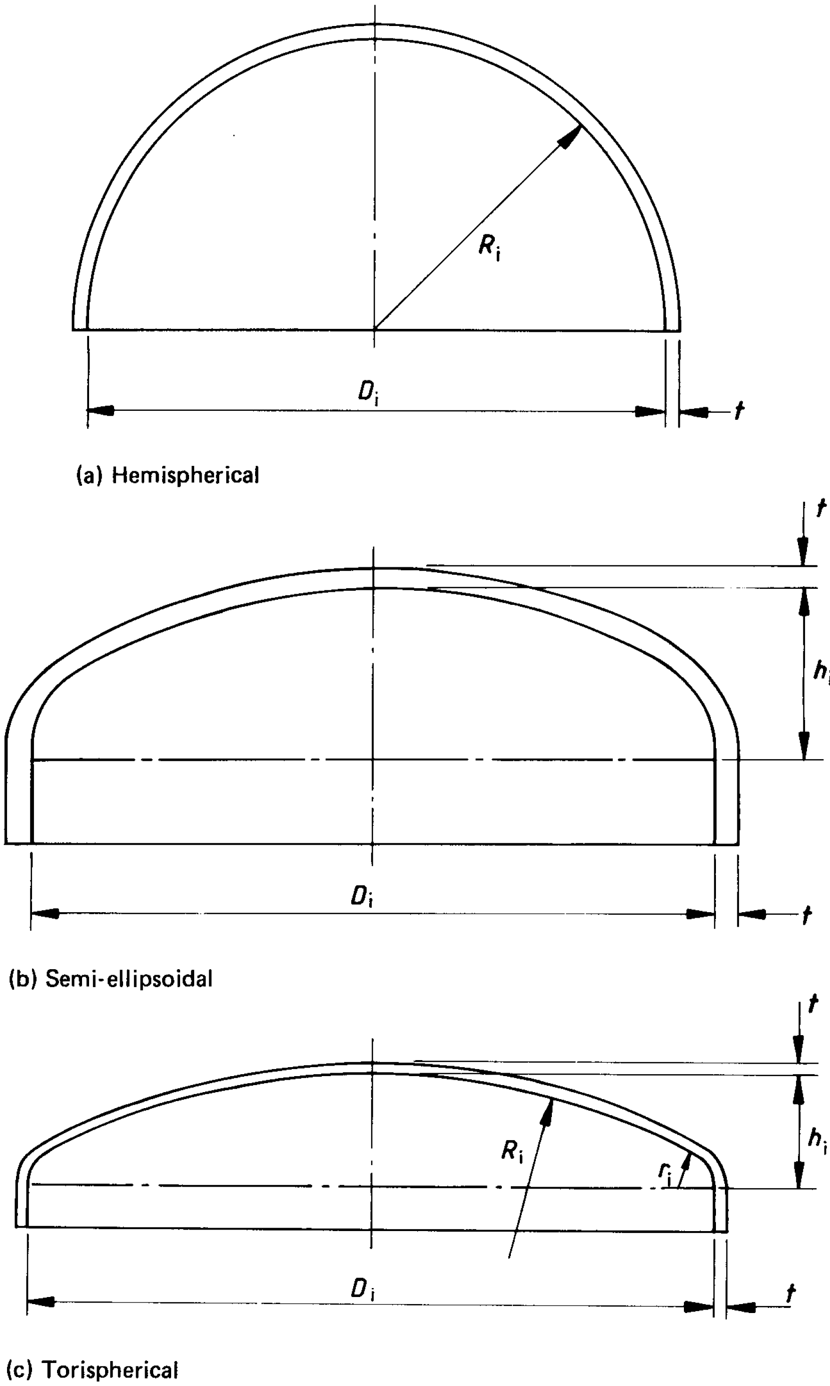



**Figure 11 — Domed ends**

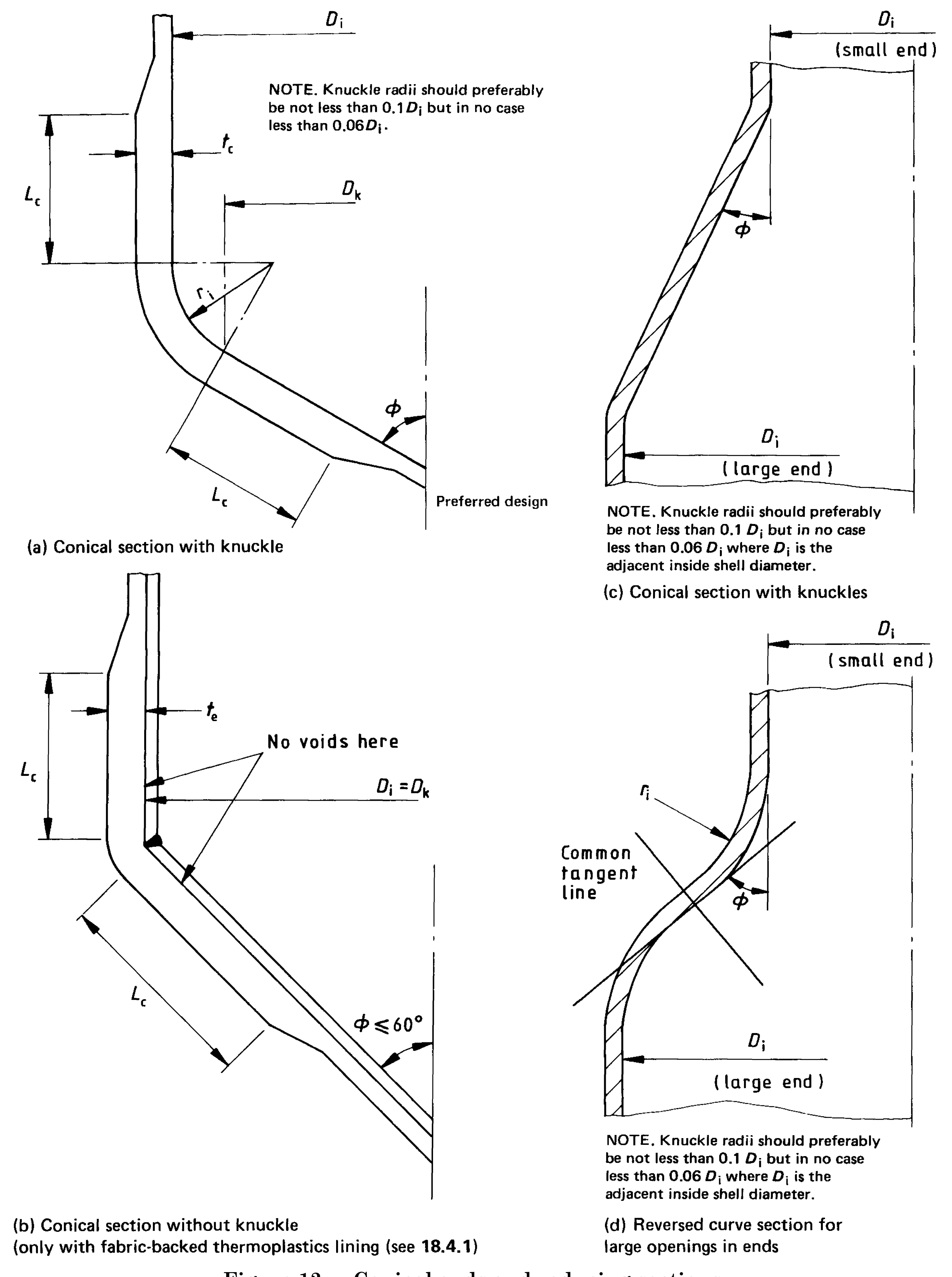
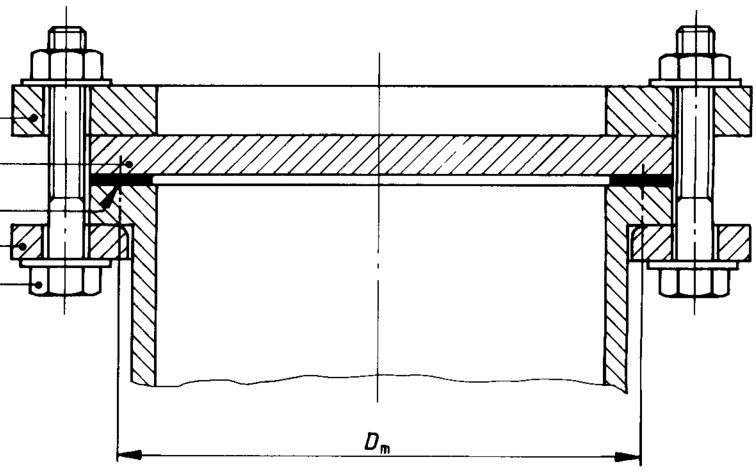
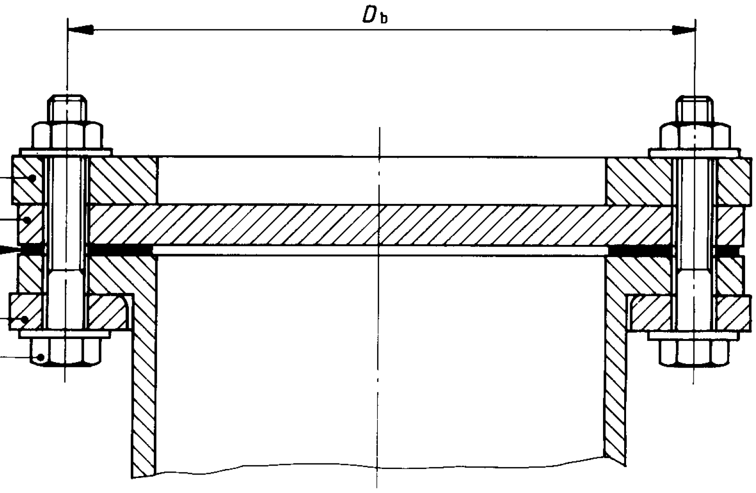



**Figure 12 — Conical ends and reducing sections**




(a) Gasket inside bolt circle




(b) Full-face gasket

**Figure 13 — Blind flanges: general**

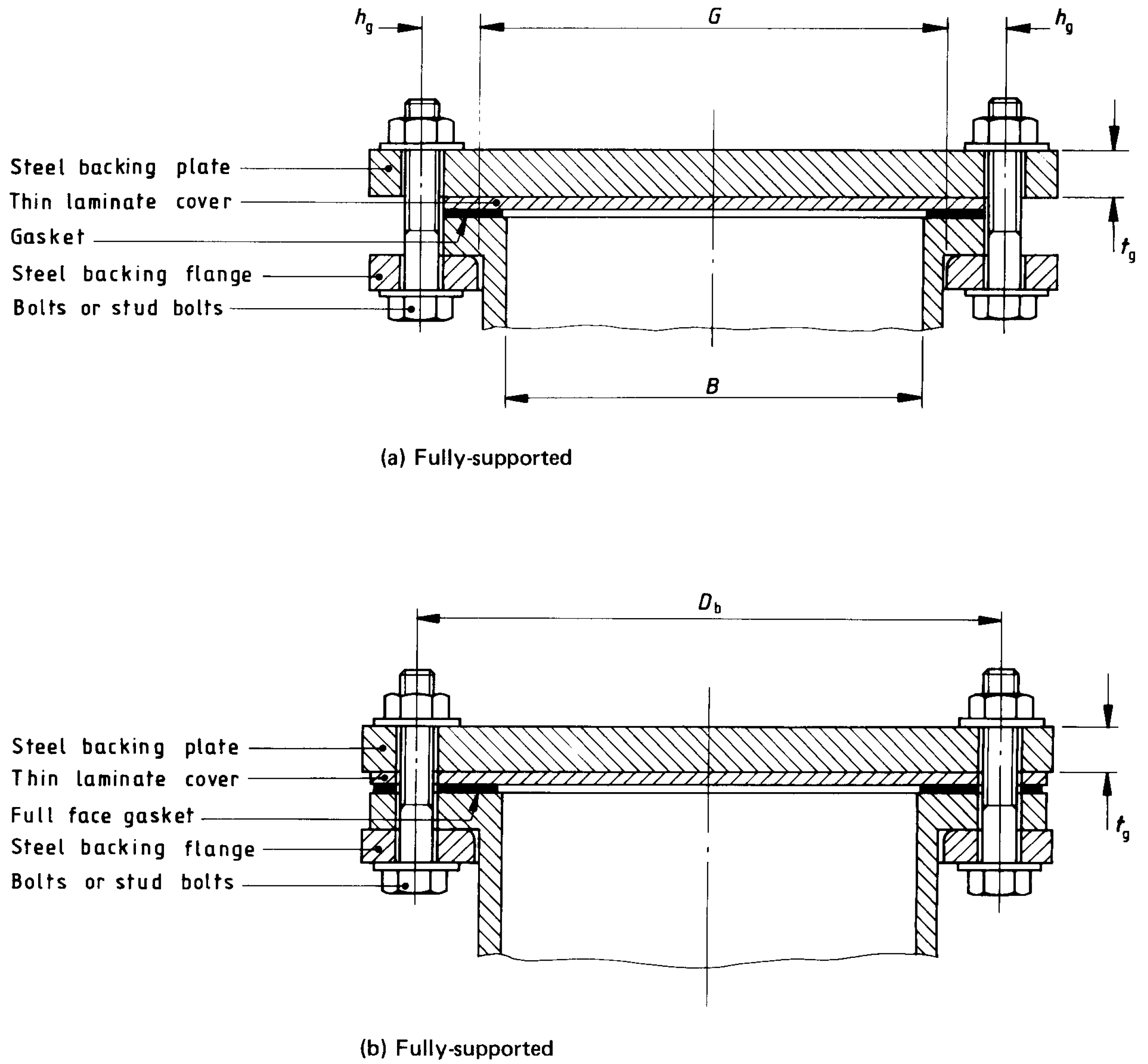



**Figure 14 — Blind flanges: not suitable for vacuum vessels**

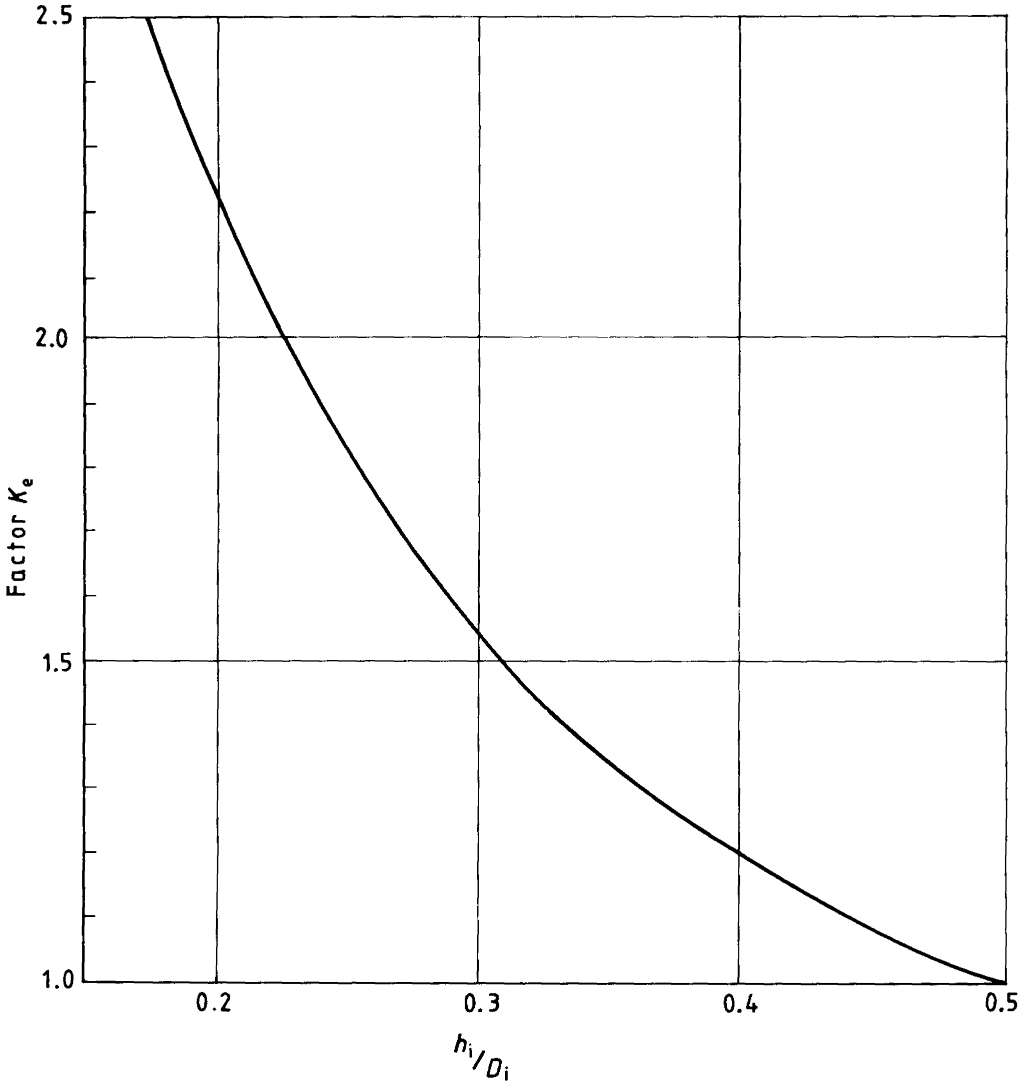



**Figure 15 — Shape factor for semi-ellipsoidal ends convex to pressure**

**Table 11 — Shape factor for domed ends**[a]

| $h_i/D_i$ | $t/D_i$ | Shape factor $K_s$ | | |
|---|---|---|---|---|
| | | Torispherical | | Semi-ellipsoidal |
| | | $0.1 \leqslant r_i/D_i \leqslant 0.15$ | $r_i/D_i > 0.15$ | |
| 0.20 | 0.005 | 2.95 | Not permitted since $R_i > D_i$ | 2.00 |
| | 0.01 | 2.85 | | 2.10 |
| | 0.02 | 2.65 | | 2.20 |
| | 0.04 | 2.35 | | 2.25 |
| | 0.05 | 2.25 | | 2.35 |
| 0.25 | 0.005 | 2.35 | 1.90 | 1.30 |
| | 0.01 | 2.25 | 1.80 | 1.35 |
| | 0.02 | 2.10 | 1.75 | 1.45 |
| | 0.04 | 1.85 | 1.70 | 1.45 |
| | 0.05 | 1.75 | 1.70 | 1.45 |
| 0.32 | | | $0.15 < r_i/D_i \leqslant 0.25$ | |
| | 0.005 | 1.95 | 1.45 | 0.85 |
| | 0.01 | 1.85 | 1.45 | 0.95 |
| | 0.02 | 1.60 | 1.40 | 1.0 |
| | 0.04 | 1.40 | 1.35 | 1.05 |
| | 0.05 | 1.30 | 1.30 | 1.10 |
| 0.50 (hemispherical) | All values | 0.6 | 0.6 | 0.6 |

[a] See references [13], [14], [15] and [16].

### 18.4 Conical ends and reducing sections

**18.4.1** *General.* Conical ends and conical reducing sections shall be of the forms shown in Figure 12.

For vessels and tanks other than those utilizing a fabric-backed thermoplastics liner, the form of cone/cylinder junction shall be as shown in Figure 12(a). In no case shall the knuckle inside radius $r_i$ be less than 6 % of the inside diameter of the adjacent cylindrical shell.

In the case only of vessels and tanks utilizing a fabric-backed thermoplastics liner, with the exception of uPVC and CPVC, it is permissible to use the form of cone/cylinder intersection shown in Figure 12(b) for angles of inclination to the vessel axis up to and including 60° but construction as shown in Figure 12(a) is the preferred design in all cases and shall be used for fabric-backed uPVC and CPVC.

Additional reinforcement added at the cone/cylinder intersection in accordance with equations (47) and (48) or (47) and (49), as appropriate, shall be distributed through the laminate in a balanced manner.

For constructions of the form shown in Figure 12(b) the intersection shall not be closer to another junction or discontinuity than twice $L_c$ determined from equation (50). In addition, the manufacturer shall fabricate a proof specimen as detailed in **36.2** d) to demonstrate to the purchaser, and/or Inspecting Authority, where appropriate, that the chosen method of manufacture produces a satisfactory laminate as defined in that clause.

Conical tops of static storage tanks which are not subject to internal or external pressure and are manufactured without a radiused knuckle region shall not be subject to the restrictions shown in Figure 12(b).

**18.4.2** *Shallow conical ends.* Conical ends having an angle of inclination to the axis of the vessel or tank of more than 75° shall be designed as flat ends in accordance with **18.5**.

**18.4.3** *Other conical ends subject to internal pressure.* For ends having an inclination to the axis of the vessel or tank of 75° or less, the circumferential unit load to be substituted in equation (6), for the conical portions shall be not less than:

$$Q = \frac{0.5 p D_k}{\cos \phi} \quad (47)$$

For construction of the form shown in Figure 12(a) the longitudinal unit load to be substituted in equation (6) for the knuckle portion shall be the greater of the values for $Q$ determined from equations (47) and (48):

$$Q_{x1} = 0.5p\, D_i\, K_{c1} \qquad (48)$$

where

$K_{c1}$ is obtained from Table 12.

For construction of the form shown in Figure 12(b), as permitted in accordance with **18.4.1**, the longitudinal unit load to be substituted in equation (6) for the cylinder/cone junction region shall be not less than:

$$Q_{x2} = 0.5p\, D_1\, K_{c2} \qquad (49)$$

where

$K_{c2}$ is obtained from Table 13.

In no case shall the longitudinal unit strength of the final laminate be less than the unit load determined in accordance with equation (47).

Since the value of $K_{c2}$ to be used in equation (49) is dependent upon the design thickness at the cylinder/cone junction, and this thickness cannot be accurately determined until the laminate has been designed, a value for the $t_c/D_i$ ratio shall be assumed in order to select a value for $K_{c2}$.

The value of $K_{c2}$ so determined shall be substituted in equation (49) to determine a notional unit load, $Q$, and this value of $Q$ used in equation (6) for the calculation of a suitable laminate.

The thickness of the laminate so calculated shall then be determined in accordance with **13.3** and the value of $t_c/D_i$ given by this thickness compared with the ratio assumed. If the difference is significant, the calculations shall be repeated using a new value for $K_{c2}$ appropriate to the likely thickness of the final laminate.

The calculated thickness for the knuckle or cylinder/cone intersection shall extend on to adjacent parts of the cylinder and cone by a distance not less than $L_c$ where:

$$L_c = \sqrt{D_i\, t_c / \cos \phi} \qquad (50)$$

In the case of conical ends to filament-wound tanks or vessels, a rigorous anisotropic elastic analysis shall be carried out. This shall allow for the contribution from each layer in the laminate and for the interaction between normal and shear strains. In the absence of such an analysis all of the additional design loading arising due to the change in geometry shall be taken by layers of CSM and WR cloth only. Care shall be taken to ensure that the longitudinal unit load used to calculate the laminate shall be not less than $Q$, $Q_{x1}$ or $Q_{x2}$, as appropriate.

**18.4.4** *Other conical ends subject to external pressure.* The laminate construction for conical ends and sections subject to external pressure shall be determined in accordance with **14.4.1**. Where a design incorporating stiffening rings is proposed, the calculations of **14.4.2** shall be used. In applying these calculations, the following modifications shall be incorporated.

a) $D_o/\cos \phi$ shall be used throughout in place of $D_o$, where $D_o$ is the major outside diameter of the conical section being considered;

b) the effective shell length $L$ shall be either the slant height of the conical section or the slant height between effective stiffening rings [see Figure 8(c)], whichever is the smaller.

**Table 12 — Shape factors for conical ends as shown in Figure 12(a)**

| $r_i/D_i$ | Shape factor $K_{c1}$ | | | | |
|---|---|---|---|---|---|
| | Angle $\phi$ | | | | |
| | 15° | 30° | 45° | 60° | 75° |
| 0.06 | | | 1.3 | 2.0 | 3.85 |
| 0.08 | | | 1.2 | 1.75 | 3.50 |
| 0.10 | | | 1.1 | 1.6 | 3.15 |
| 0.15 | 1.0 | | | 1.4 | 2.70 |
| 0.20 | | | | 1.25 | 2.40 |
| 0.30 | | | | | 1.55 |
| 0.40 | | | | | |
| 0.50 | | | | | |

**Table 13 — Shape factors for conical ends as shown in Figure 12(b)**

| $t_c/D_i$ | Shape factor $K_{c2}$ | | | |
|---|---|---|---|---|
| | Angle $\phi$ (not greater than 60°) | | | |
| | 15° | 30° | 45° | 60° |
| 0.002 | 2.94 | 5.62 | 8.90 | 13.60 |
| 0.005 | 2.05 | 3.70 | 5.80 | 8.70 |
| 0.01 | 1.60 | 2.75 | 4.12 | 6.30 |
| 0.02 | 1.24 | 2.00 | 3.00 | 4.40 |
| 0.04 | 1.00 | 1.55 | 2.20 | 3.20 |
| 0.05 | 1.00 | 1.45 | 2.00 | 2.75 |

### 18.5 Flat bottoms to vertical cylindrical tanks and other flat ends

**18.5.1** *General.* Flat bottoms to vertical cylindrical tanks shall be either fully supported or designed in accordance with clause **17**. Flat ends to other vessels and tanks shall be designed in accordance with clause **17**.

NOTE Flat roofs of tanks should be designed to comply with BS 6399-1.

**18.5.2** *Fully supported flat bottoms.* At the corners of fully supported bottom ends, the load-carrying capacity of the laminate in the longitudinal direction shall depend upon the form of construction. In the case of radiused corners, see Figure 16, the factor of not less than 1.5 shall be applied and in the case of square corners, see Figure 17, the factor of not less than 2 shall be applied to the required load-carrying capacity in the circumferential direction of the cylinder laminate for the internal pressure.

NOTE 1 The thickness of the bottom away from the junction between end and cylindrical shell is not critical.

When some fabric-backed thermoplastics are used as linings (uPVC and CPVC are specifically excluded) the methods illustrated in Figure 17 are permissible and they shall also comply with **36.2** f).

Small unsupported areas in way of outlets, etc. shall comply with clause **17** for the design of flat panels.

In the case of bottom corners to filament-wound tanks or vessels, either a rigorous anisotropic elastic analysis shall be carried out or all of the additional design loading arising due to the change in geometry, which shall be considered equivalent to the design loading in the shell, shall be taken by layers of CSM and WR cloth only. The anisotropic elastic analysis shall allow for the contribution from each layer in the laminate and for the interaction between normal and shear strains.

NOTE 2 Dependent upon the bedding arrangements for the base of the vessel or tank, it may be necessary to provide a flat base.

### 18.6 Bolted circular blind flanges

**18.6.1** *General.* Bolted blind flanges shall comply with one of the designs shown in Figure 13(a) or Figure 13(b).

**18.6.2** *Shell flange connection.* A flange on the end of the shell shall be designed in the same way as that on a large flanged branch connection (see clause **21**).

**18.6.3** *Ring-supported circular blind flanges.* Ring-supported circular blind flanges [see Figure 13(a) and Figure 13(b)] shall be manufactured using CSM only as the reinforcing material.

The total unit mass of glass reinforcement required shall be determined from equation (51) or (52), as appropriate.

Figure 13(a):

$$m_{CSM} = \sqrt{\frac{p D_m^2}{3.23 t_g u_{CSM}}} \qquad (51)$$

Figure 13(b)

$$m_{CSM} = \sqrt{\frac{p D_b^2}{5.33 t_g u_{CSM}}} \qquad (52)$$

NOTE 1 For Figure 13(a), the metallic backing flanges should either be selected from Table 14, or designed as "loose" flanges in accordance with BS 5500 (see clause **21**).

NOTE 2 For Figure 13(b), metallic spreading rings should not be less than 6 mm thick. Within this constraint they may be of nominal thickness only.

The laminate construction shall comprise any CSM layer arrangement which complies with this total mass requirement.

**18.6.4** *Circular blind flanges with edge moment.* Circular blind flanges with edge moment of such design that the gasket contact is wholly within the bolt circle and the action of the bolts subject the flange to an edge moment shall not be used.

**18.6.5** *Fully supported blind flanges.* Fully supported blind flanges [see Figure 14(a) and Figure 14(b)] shall be made only with a steel backing plate over the full area of the blind flange. They shall not be used for vacuum vessels.

The thickness of the steel backing plate, $t_B$, for the arrangement shown in Figure 14(a) shall be the greater of the following:

$$\text{a) } t_B = G\sqrt{\frac{p}{K_l f}} \qquad (53)$$

where

$$K_l = \frac{1}{0.3 + \left(\frac{1.5 W h_g}{H G}\right)}$$

or

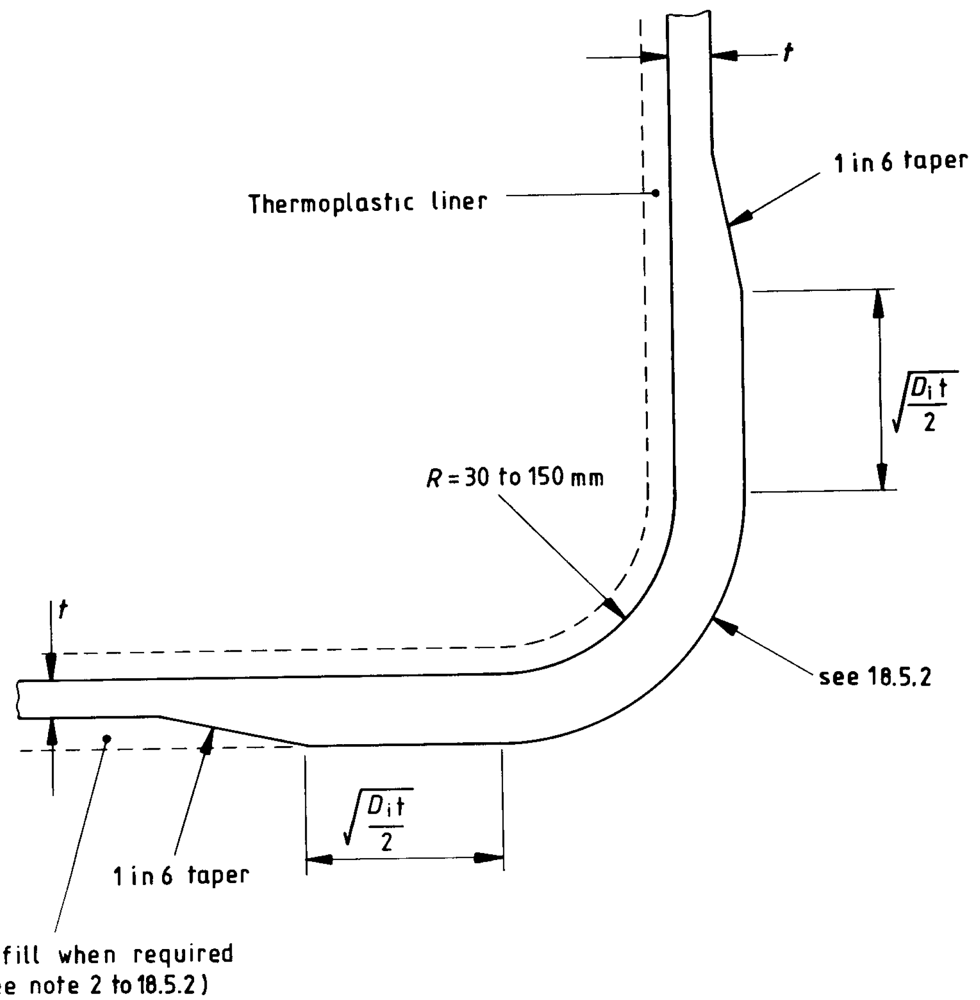
$$\text{b) } t_B = \sqrt{\frac{1.909 W h_g}{G f}} \qquad (54)$$

For the arrangement shown in Figure 14(b), where the gasket extends beyond the bolt circle, the thickness of the steel backing plate shall be not less than:

$$t_B = 0.42 D_b \sqrt{\frac{p}{f}} \qquad (55)$$

For the arrangements shown in Figure 14(a) and Figure 14(b) these requirements shall not preclude the use, as a steel backing plate, of blind flanges in accordance with BS 1560, BS 3293 or BS 4504 for the appropriate pressure and temperature design conditions.

NOTE For fully supported blind flanges, the thickness of the supported laminate need only be sufficient to act as an effective corrosion barrier and to have adequate rigidity during handling (subject to a minimum thickness of 3 mm). In the case of vessels or tanks lined with thermoplastics, a sheet of unreinforced thermoplastics, similar in thickness to the lining will be sufficient for all except very large openings.




NOTE With a radius, *R*, of 75 mm to 150 mm there is evidence from a stress analysis that the stress concentration in the knuckle is greater and additional support of the knuckle should be considered.

**Figure 16 — Lined and unlined tanks, knuckle radius 30 mm to 150 mm (preferred construction)**

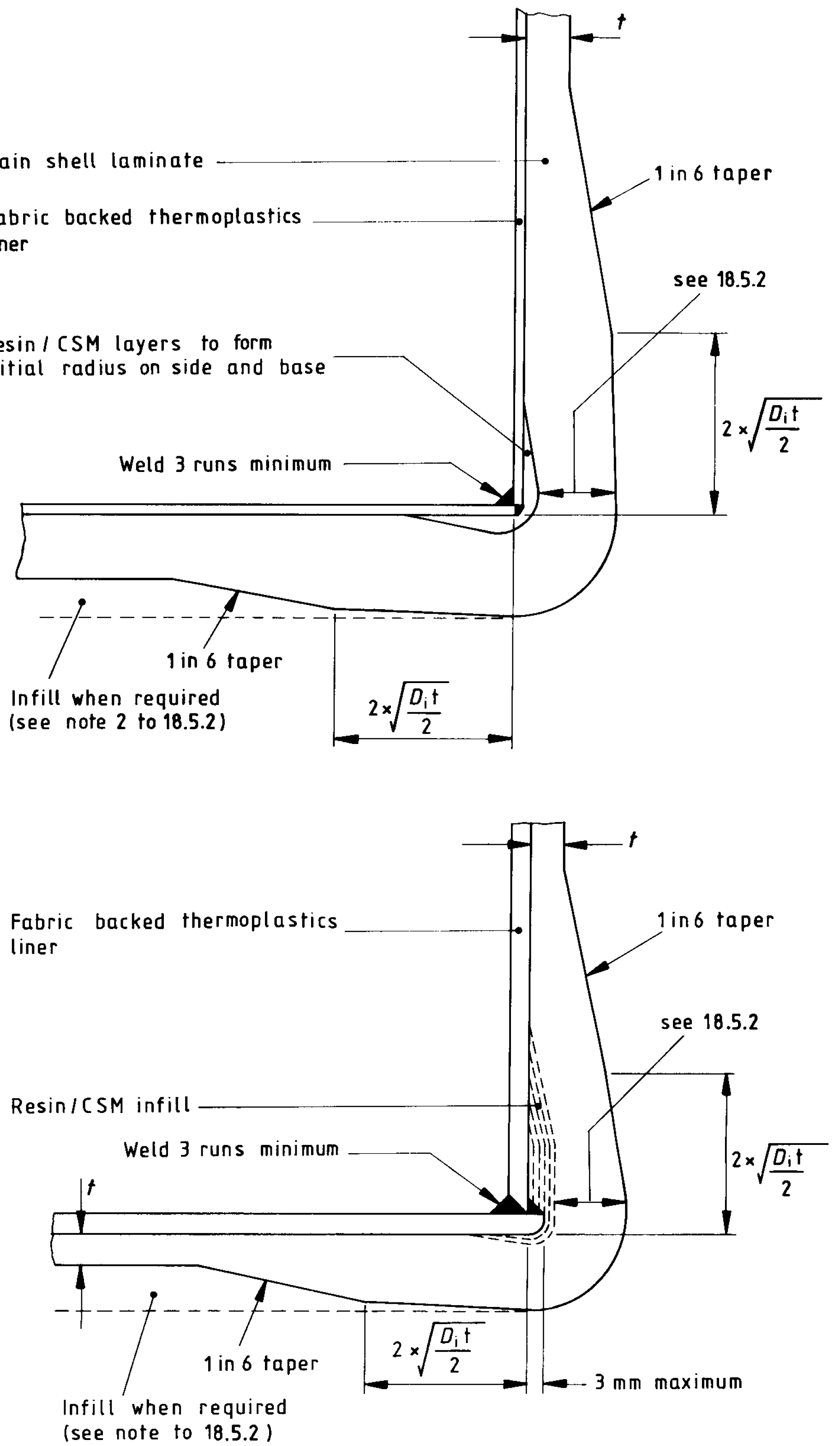



**Figure 17 — Lined tanks, fabric-backed liner without a formed knuckle**

## 19 Circumferential seams

### 19.1 Symbols

For the purposes of clause **19** the following symbols apply.

$K_{OVL}$ design factor determined from equation (1) for the type of laminate used and its method of manufacture in the construction of the overlay

$L_j$ minimum effective length of overlay (in mm) (see Figure 18 and Figure 19)

$m_z$ mass of reinforcement per unit area (in kg/m$^2$ glass) in one layer of type $z$

$n_z$ number of layers of type $z$ in laminate construction under consideration

$Q_a$ unit load to be used in equation (56) for calculation of $L_j$ determined from equations (8), (9), (10) or (11), as applicable

$u_z$ design unit loading (in N/mm per kg/m$^2$ glass) for layer of type $z$

$U_{LAM}$ design loading (in N/mm) for main laminate

$U_{OVL}$ design loading (in N/mm) for overlay laminate

### 19.2 General

The overlay laminate shall be designed so that its design loading $U_{OVL}$ in both circumferential and longitudinal directions is not less than that of the main shell laminate in the circumferential and longitudinal directions, respectively, except at the terminating tapers.

The overlay shall be terminated by a taper not steeper than 1 in 6, and shall be smoothly blended into the main laminate at its extremities (see Figure 18 and Figure 19).

Joints shall be positioned clear of regions of high local loadings, e.g. knuckles, branches or transitions.

The effective length for each type of joint shall be determined from equation (56):

$$L_j = \frac{Q_a K_{OVL}}{\text{Lap shear strength}} \qquad (56)$$

where the lap shear strength is the minimum value given in Table 5 for the type of reinforcement and resin system employed in the construction of the overlay and the main laminate in the region of the main seam.

### 19.3 Design loading of laminates

The design loading of the main shell laminate $U_{LAM}$, and of the overlay laminate, $U_{OVL}$, shall be determined from equation (57):

$$U_{LAM} \text{ or } U_{OVL} = u_1 m_1 n_1 + u_2 m_2 n_2 + \ldots u_z m_z n_z \qquad (57)$$

summed for the actual laminate construction selected.

## 20 Openings, branches and compensation

### 20.1 General

Openings in vessels or tanks shall be circular or oval, with a ratio of major to minor axis not exceeding 2.

NOTE It is preferable to locate all openings clear of changes in laminate construction, e.g. points of end attachment and welds in thermoplastics liners.

Compensated openings and branches in domed ends shall be located so that the attachment and any additional reinforcement is entirely within a circle of radius $0.4D_i$, concentric with the axis of the shell, where $D_i$ is the inside diameter of the shell.

Openings and branches in vessels subject to external pressure shall be designed in accordance with the requirements for vessels subject to internal pressure (see **20.2.3** to **20.3.2**).

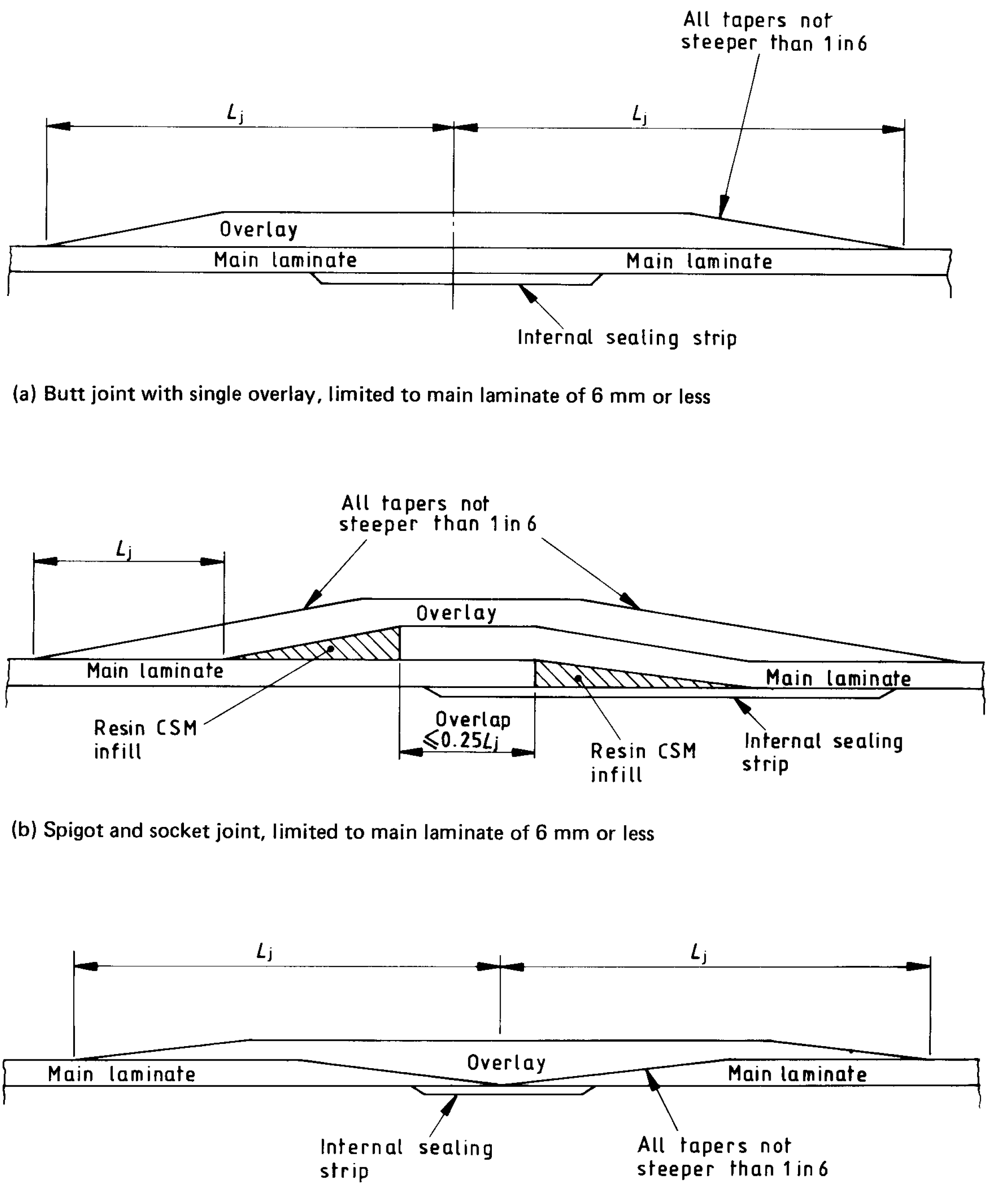
### 20.2 Compensation of openings

**20.2.1** *General.* All openings shall be compensated by the use of additional laminate designed in accordance with **20.2.4** for all planes through the axis of the openings normal to the vessel surface.

The maximum dimension of compensated openings in domed ends shall not exceed $0.5D_i$ and larger openings shall be designed as conical or reversed curve reducers (see Figure 12).

The maximum dimension of openings in cylindrical shells covered by this clause shall be $0.3D_i$.

NOTE Openings in cylindrical shells with a maximum dimension exceeding $0.3D_i$ and all openings of non-circular shape require special consideration.

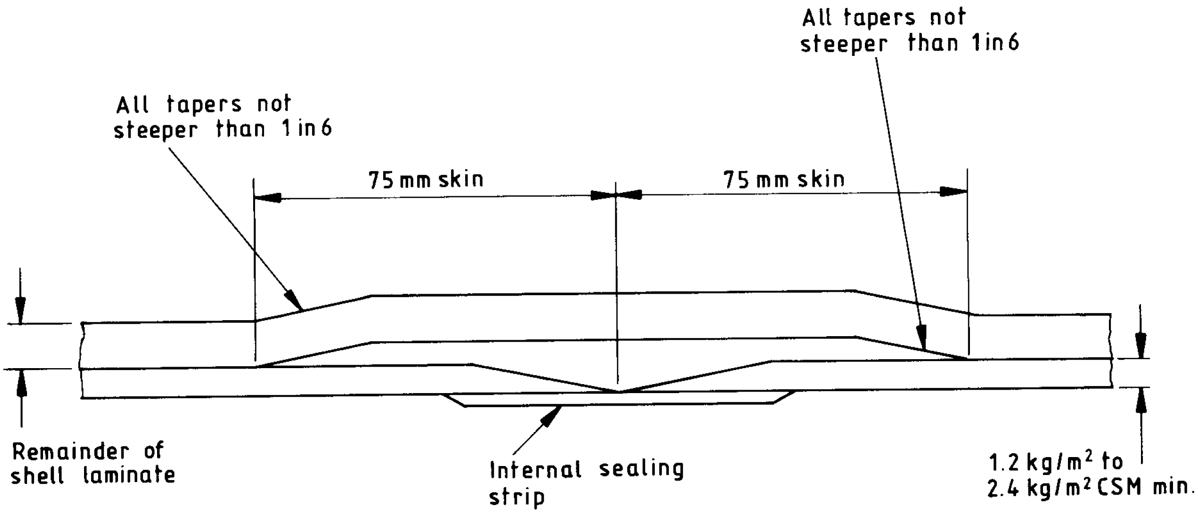



(a) Butt joint with single overlay, limited to main laminate of 6 mm or less

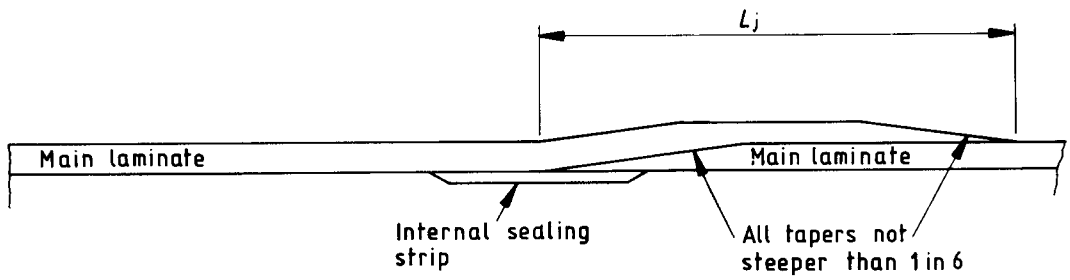
(b) Spigot and socket joint, limited to main laminate of 6 mm or less

(c) Feathered joint

**Figure 18 — Typical main seam joints for vessels without thermoplastics linings**




(d) Skin form




(e) Main laminate overlay joint

**Figure 18 — Typical main seam joints for vessels without thermoplastics linings** *(concluded)*

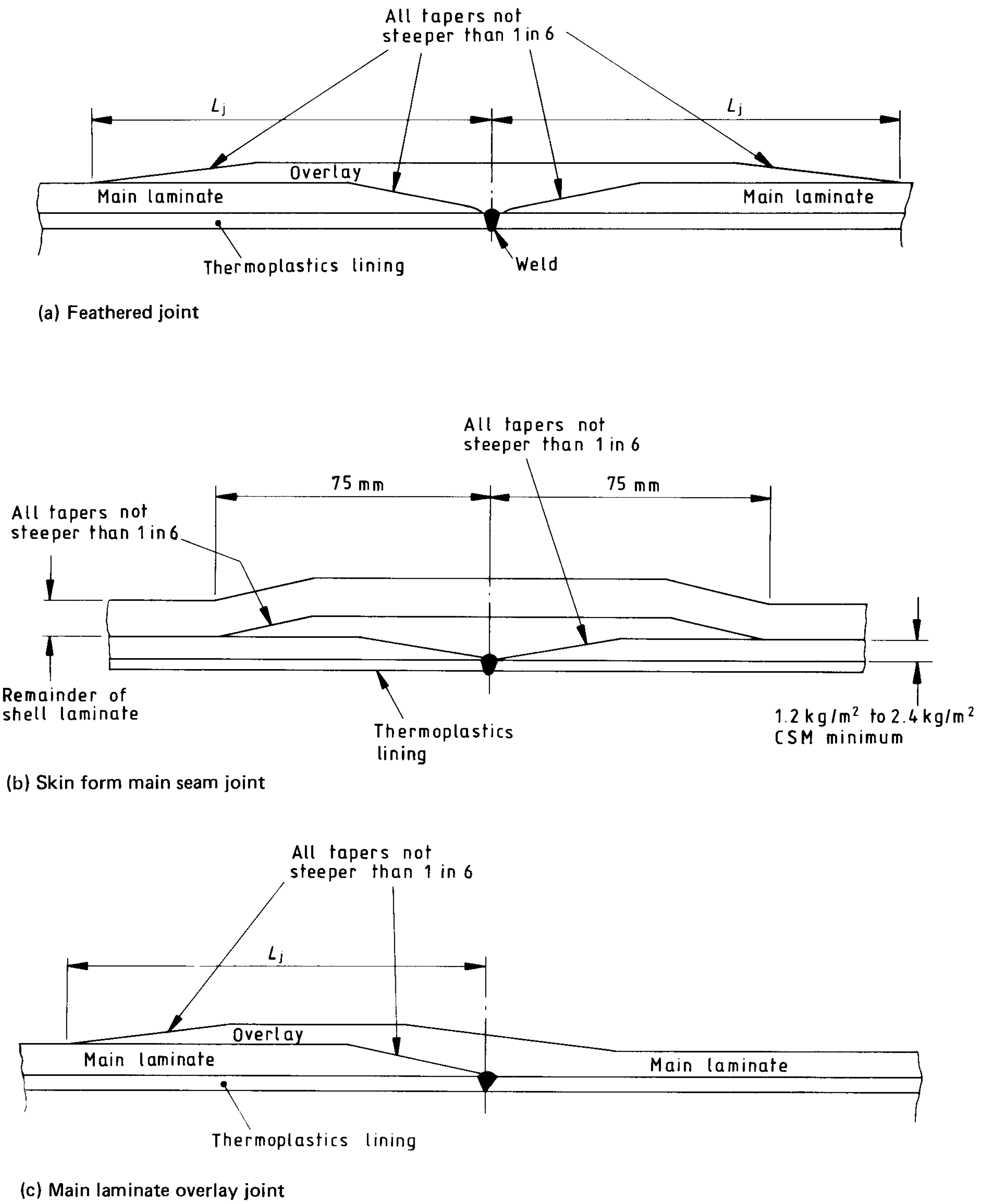



**Figure 19 — Typical main seam joint for vessels with thermoplastics linings**

**20.2.2** *Symbols*. For the purposes of **20.2** the following symbols apply.

$A_c$ effective load capacity of compensation (in N)

$A_{ci}$ load capacity (in N) of that portion of the total required compensation which is interleaved into the main shell laminate

$A_{ce}$ load capacity (in N) of the externally overlayed portion of the total required compensation, i.e. not interleaved into the main shell laminate

$A_E$ load capacity of excess strength in laminate as constructed (in N)

$A_L$ load capacity lost within diameter $d_c$ (in N)

$A_s$ effective shear capacity of compensation (in N)

$d_b$ inside diameter of branch (in mm) (see Figure 20 and Figure 22)

$d_c$ effective diameter of hole cut in vessel wall (in mm)

$d_r$ diameter of reinforcement compensation at start of taper (in mm)

$d_s$ diameter of studs in pad connections (in mm) (see Figure 23)

$D_i$ inside diameter of shell (in mm)

$F$ factor of safety

$K$ overall design factor determined from equation (1)

$K_{OVL}$ design factor determined from equation (1) for type of laminate used and its method of manufacture in the construction of the overlay

$p$ total effective pressure (in N/mm²) i.e. design pressure plus hydrostatic head at point under consideration

$Q_b$ unit load due to pull-out force (in N/mm)

$R$ radius of corner (in mm) (see Figure 16)

$t$ laminate thickness of shell (in mm) calculated from laminate details

$t_p$ design thickness of panel or end (in mm) calculated in accordance with **13.3**

$t_t$ total thickness of branch (in mm) [see Figure 20(d)]

$U_c$ required design loading of unpierced vessel wall (in N/mm)

$U_{LAM}$ design loading (in N/mm) for main laminate

$U_{OVL}$ design loading (in N/mm) for overlay laminate determined from equation (57)

**20.2.3** *Design of compensation*. The additional laminate to be provided at an opening shall be designed by taking into account each of the following factors.

a) The disposition of the additional laminate or compensation shall be as shown diagrammatically in Figure 20. The additional layers for compensation shall be $S_1 + S_2$. Where practicable, laminate layers $S_2$ shall be interleaved with the adjacent laminates.

For vessels and tanks without thermoplastics lining, a sealing laminate with a minimum construction of 1.2 kg/m² CSM plus surface layer shall be provided as shown in Figure 20(a), Figure 20(b) and Figure 20(c).

The sealing laminate shall be ignored in strength calculations.

b) In addition to the specific requirements in c) to f), the total construction of additional laminate plus the shell thickness shall be adequate to carry all additional loads.

c) The additional laminate shall have a load capacity greater than:

$$A_L = d_c \times U_c \qquad (58)$$

where

$U_c$ is as follows.

1) Where the opening is in a cylindrical or conical shell, $U_c$ is the maximum required design loading within the diameter $d_c$.

2) Where the opening and its compensation are located entirely within the spherical portion of a dished end, $U_c$ is the design loading required for a sphere having a radius equal to the spherical portion of the end.

3) Where the opening and its compensation are in a semi-ellipsoidal end and are located entirely within a circle having a radius, measured from the centre of the end, of 0.40 of the shell diameter, $U_c$ is the design loading required for a sphere having a radius given by equation (46).

4) In the case of tanks with fully supported flat bottoms with shell openings located within the distances from the base of:

i) $R + \sqrt{\frac{D_i}{2} t}$ for constructions as shown in Figure 16;

ii) $2\sqrt{\frac{D_i}{2} t}$ for constructions as shown in Figure 17;

$U_c$ in both the circumferential and longitudinal directions is the circumferential design loading required for the basic cylindrical shell at its lowest point.

Such compensating laminate shall be in addition to the extra laminate required at the shell/base intersection in accordance with **18.5.2** and shown in Figure 16 and Figure 17.

NOTE Because of the complex loadings in this region it is recommended that, wherever possible, openings should be clear of the shell/base intersection and preferably situated at a height such that the opening and its compensation are above the thickened regions shown in Figure 16 and Figure 17.

The load carrying capacity of the effective compensating overlay shall be taken as:

$$A_c = (d_r - d_c) \times U_{OVL} \quad (59)$$

d) The bonds between the additional overlay and the main shell shall be adequate to cover the transferred shell load across the bond in shear.

Where all or part of the compensating layers of $S_2$ (see Figure 20) are interleaved with the main shell laminate there is a considerable increase in the shear area between compensating layers and the main laminate, which is of particular value in the case of small branches where $d_c$ is large in relation to $d_r$, and, provided there is proper interleaving of that portion of the compensating laminate layers and that they extend over a concentric diameter not less than $d_r$, it is permissible for the required shear area, between the externally overlayed layers of compensation of $S_2$ plus $S_1$ to be related only to the strength of those layers of compensation not interleaved into the main shell laminate as follows.

Total compensation:

$$A_c = A_{ci} + A_{ce} \geqslant A_L$$

The effective shear strength at the interface between external overlay and main laminate is given by:

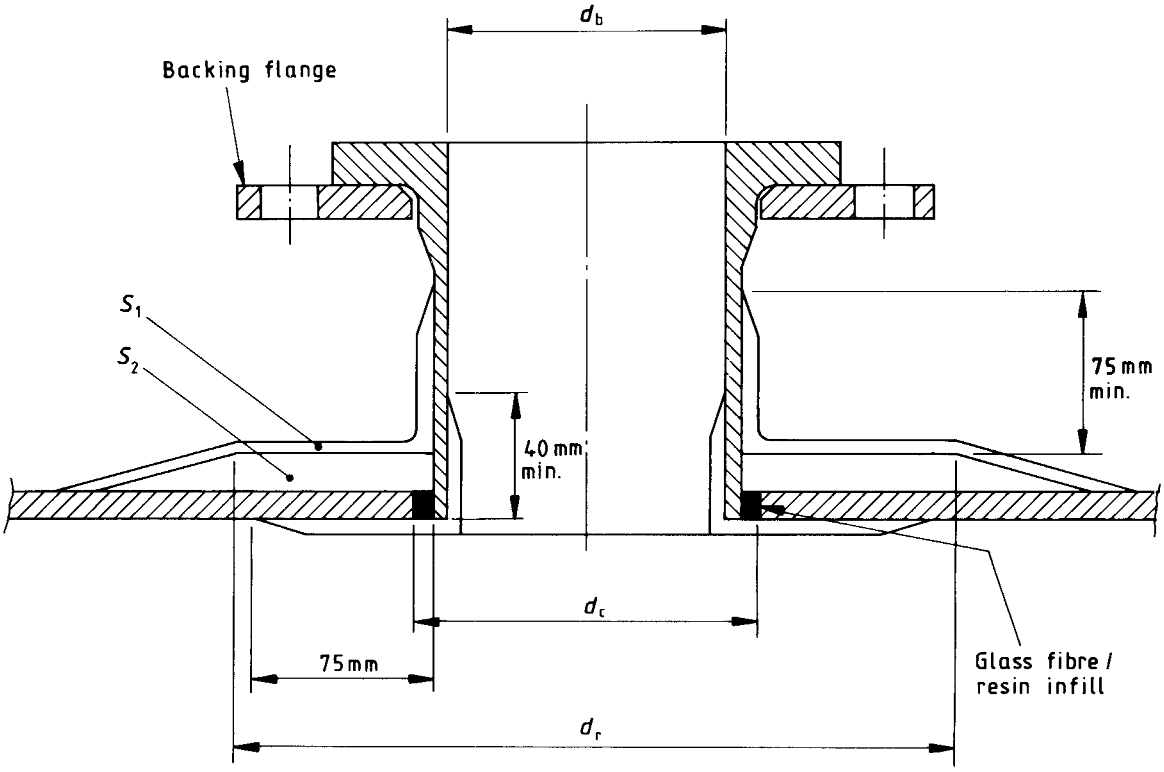
$$A_s = \frac{\text{Lap shear strength}}{F} \times 0.4 \times (d_r^2 - d_c^2) \quad (60)$$

Where all the compensation is applied externally, $A_s$ shall be not less than $A_L$.

Where a portion of the compensating layers are properly interleaved in the main laminate over a diameter not less than $d_r$, then $A_s$ shall be not less than $A_{ce}$. The factor $F$ shall be not less than $K$ or $K_{OVL}$ [see equation (1)].

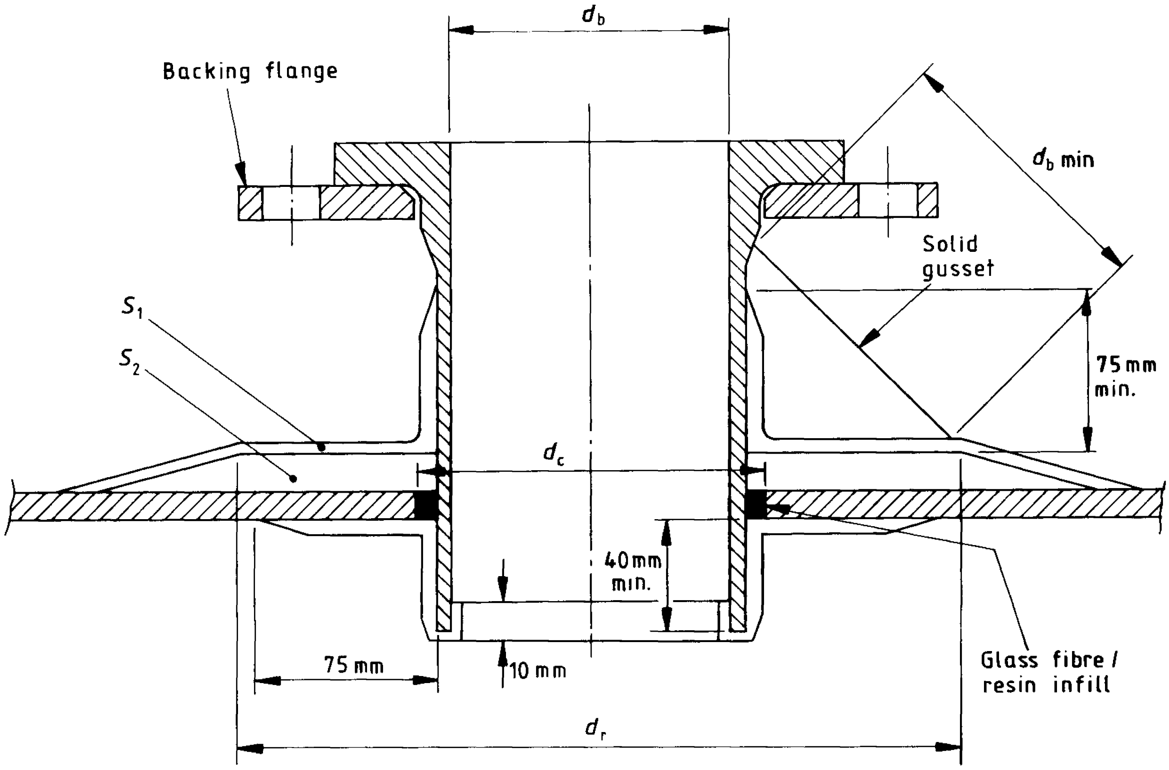
e) The overlay laminate up the branch shall be adequate to carry the pull-out load due to pressure as determined in accordance with **20.2.6** and equation (57).

f) The bond between the overlay laminate and the branch shall be adequate to carry the pull-out load across the bond in shear as determined in accordance with **20.2.6** and equation (56).



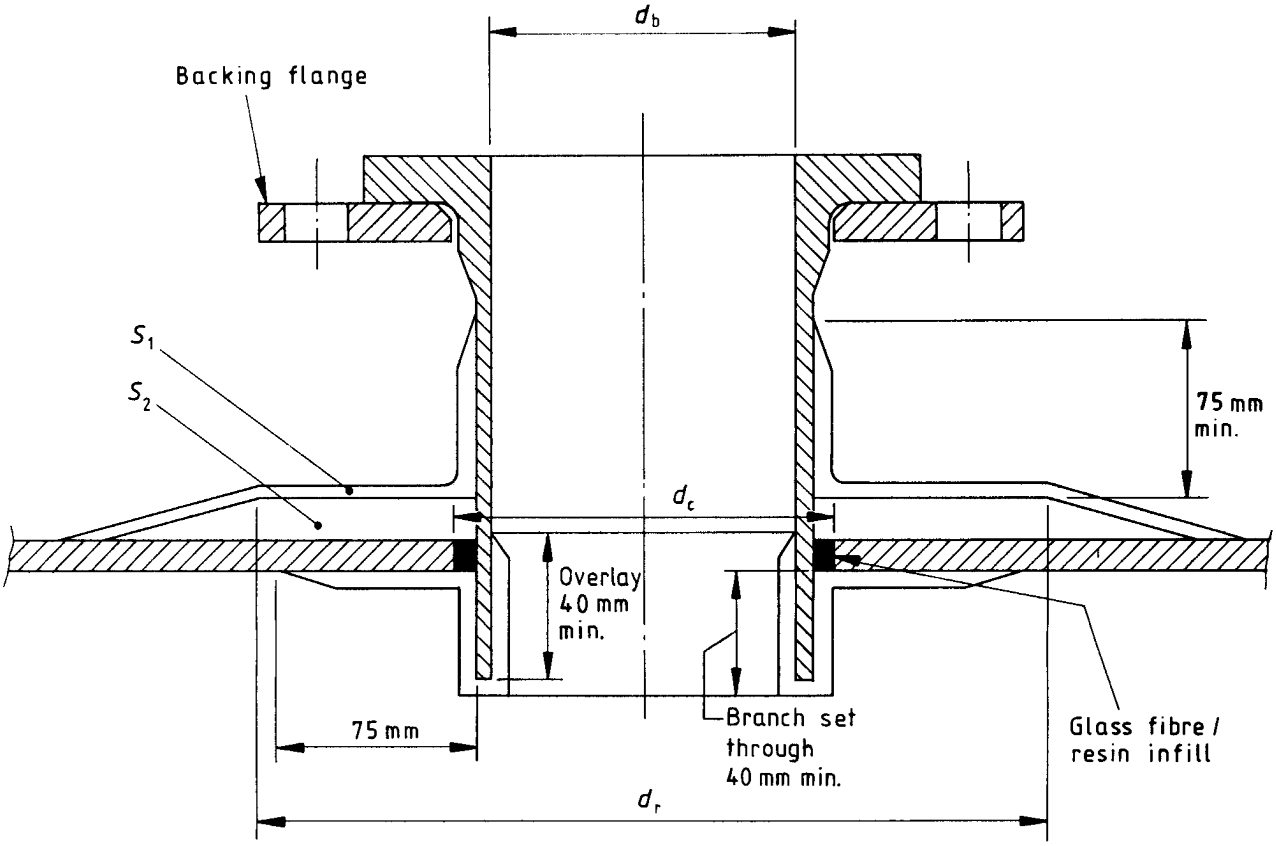

(a) Flush set-in branch without thermoplastics lining (greater than 100 mm bore)

**Figure 20 — Diagrams of typical branches**



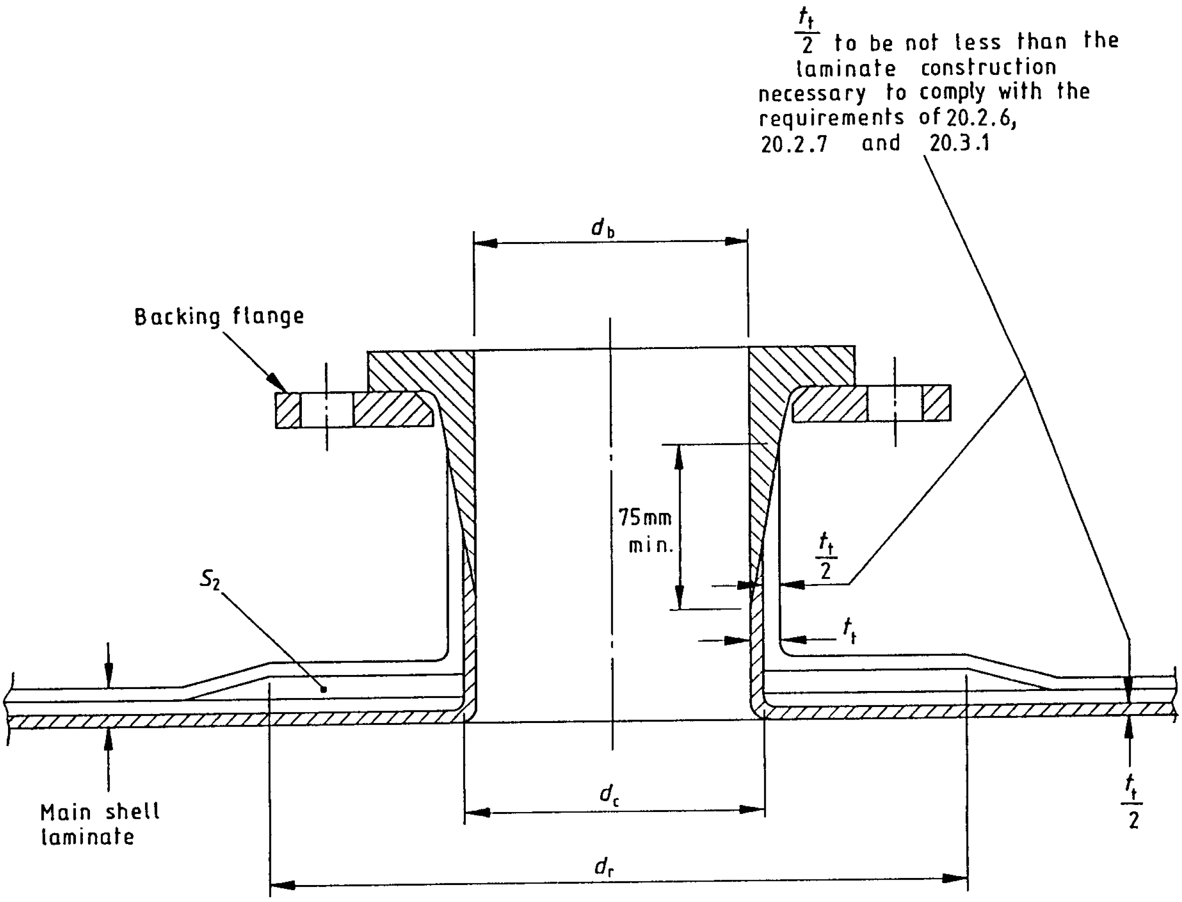

(b) Set through branch without thermoplastics lining (50 mm to 100 mm bore)

**Figure 20 — Diagrams of typical branches** *(continued)*




(c) Set through branch without thermoplastics lining (greater than 100 mm bore)

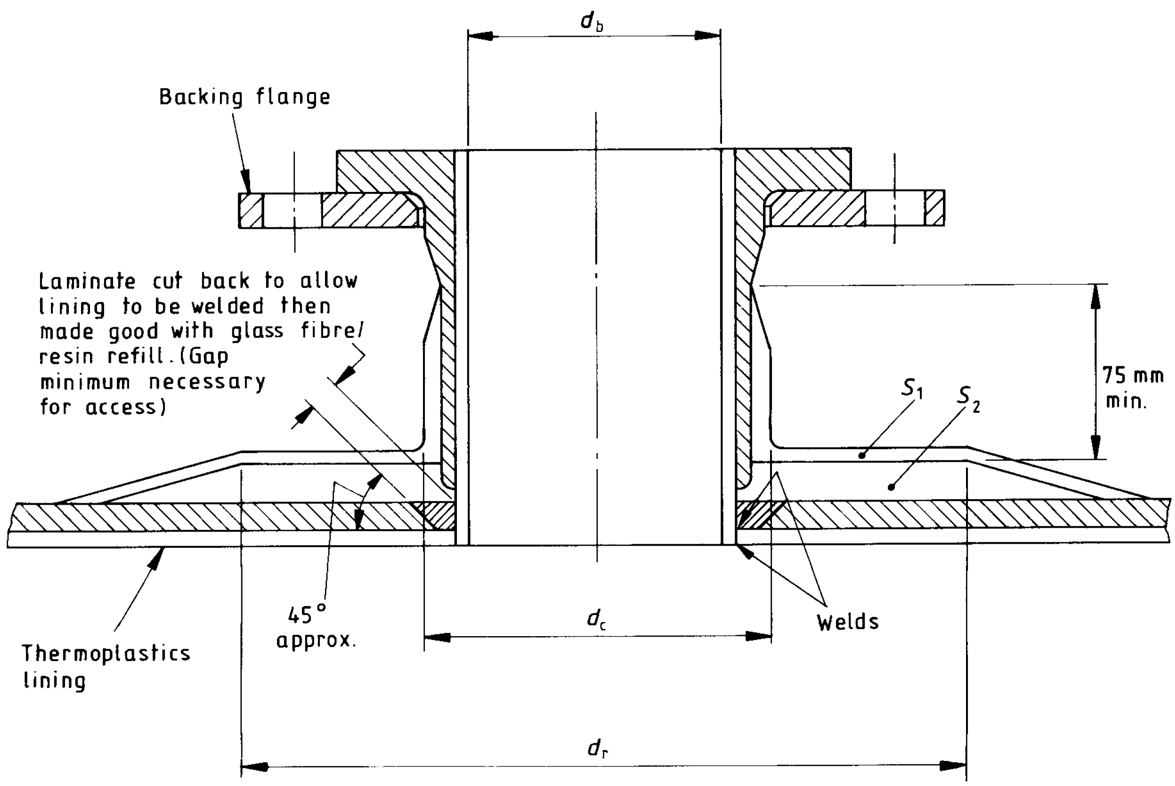
**Figure 20 — Diagrams of typical branches** *(continued)*




NOTE It should be noted that the inner and outer layers, $t_t/2$, which are continued to form the branch and its overlay, are a part of the main shell laminate and cannot be considered as branch compensation, i.e. with this form of construction there is no external overlay layer $S_1$ on the main shell.

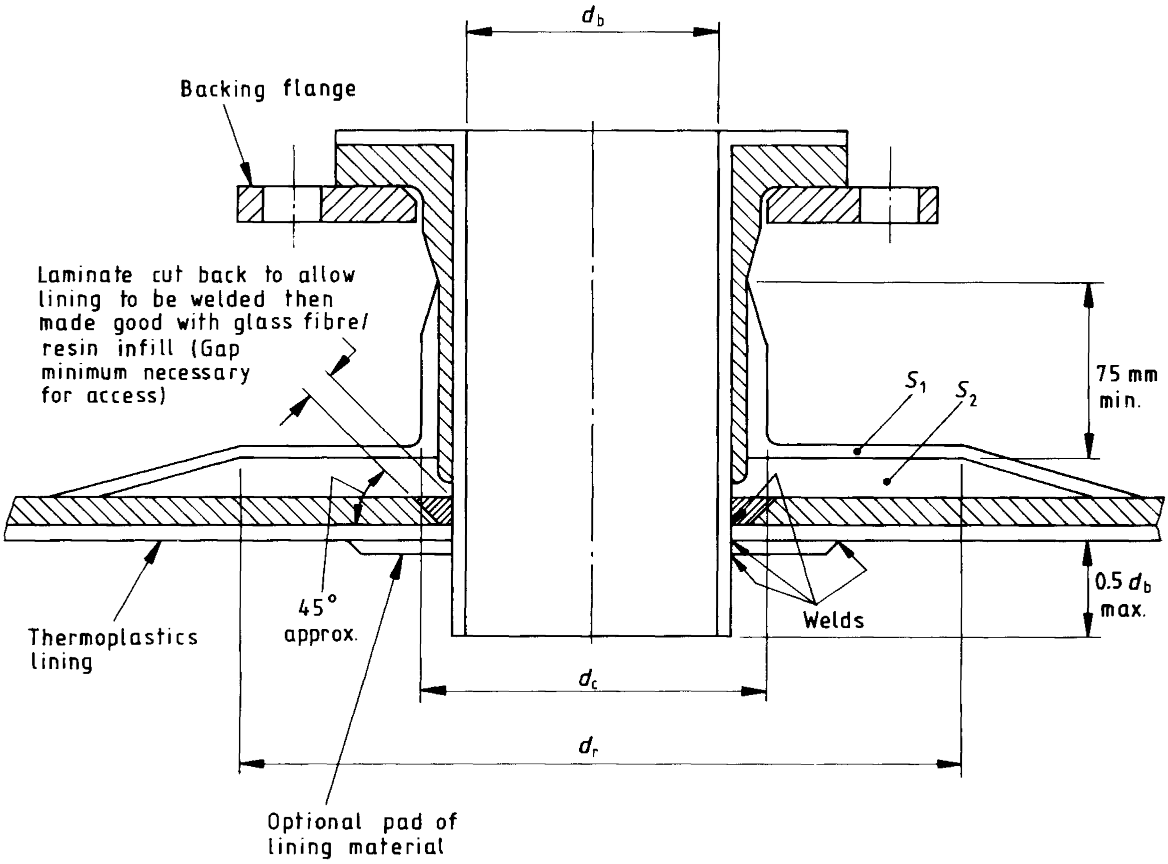
(d) Flush moulded-in branch without thermoplastics lining or with thermoplastics lining (not shown)

**Figure 20 — Diagrams of typical branches** *(continued)*




(e) Flush branch with thermoplastics lining

**Figure 20 — Diagrams of typical branches** *(continued)*




(f) Set through branch with thermoplastics lining

**Figure 20 — Diagrams of typical branches** *(concluded)*

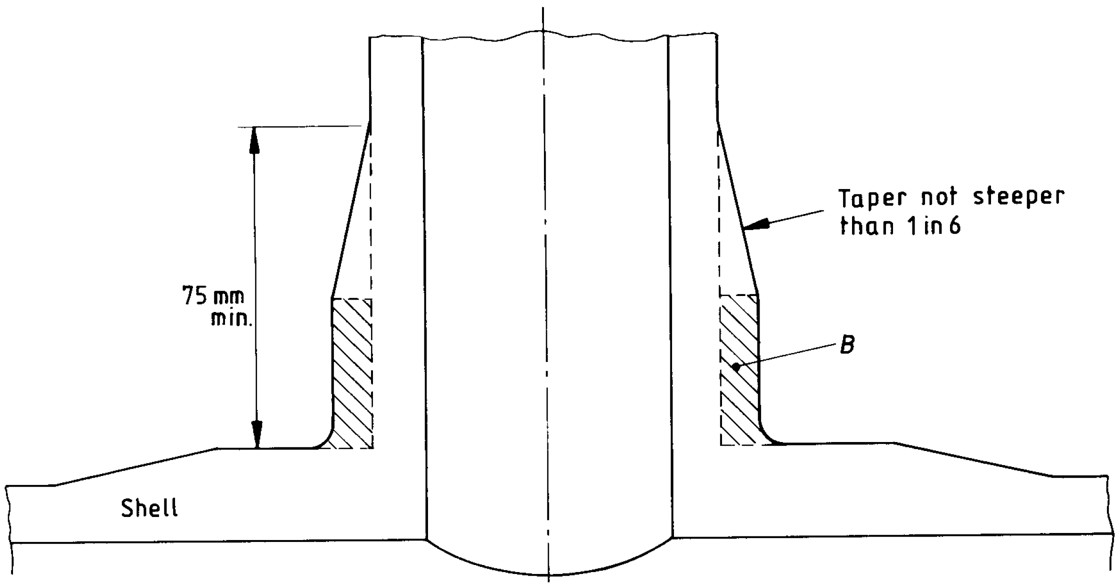



**Figure 21 — Arrangement of branch overlay for long branches**
(applicable to all types of branch construction in Figure 20)

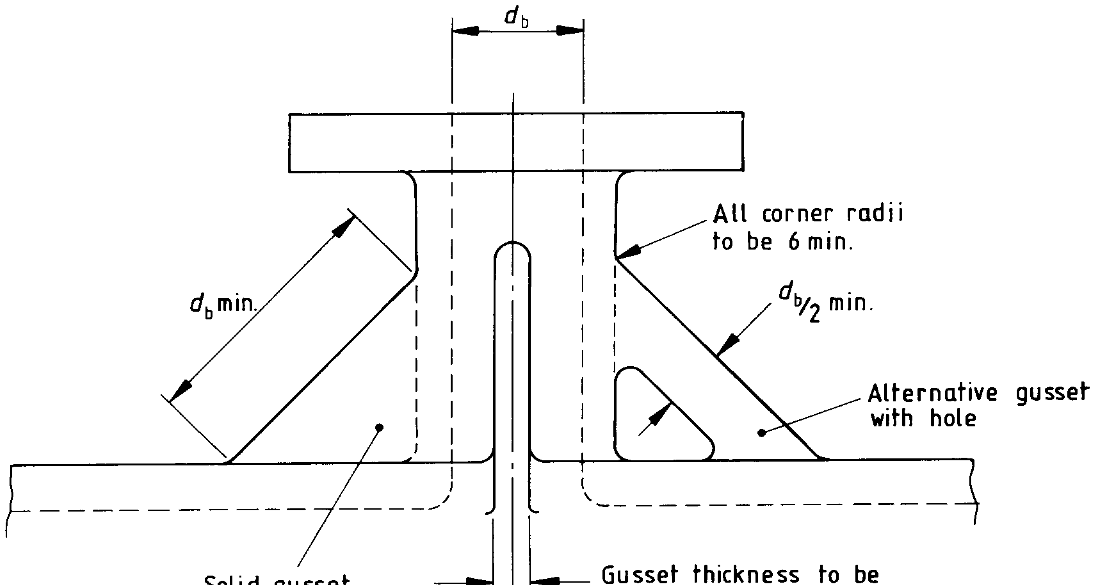
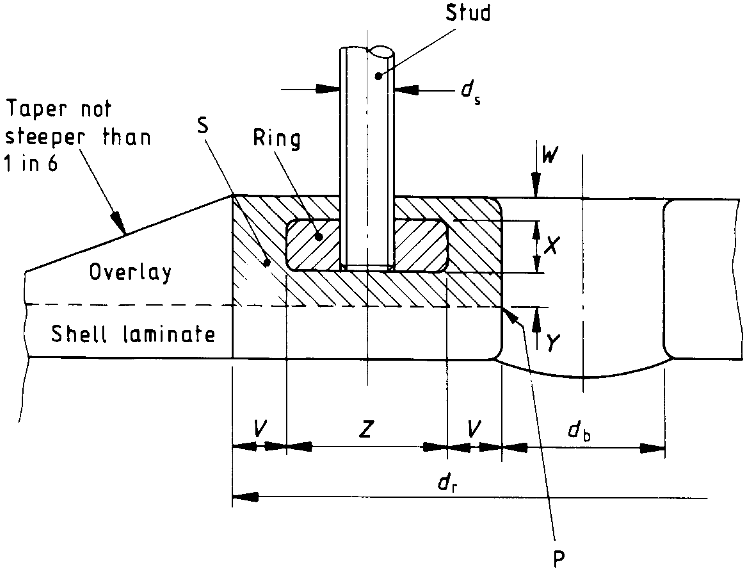
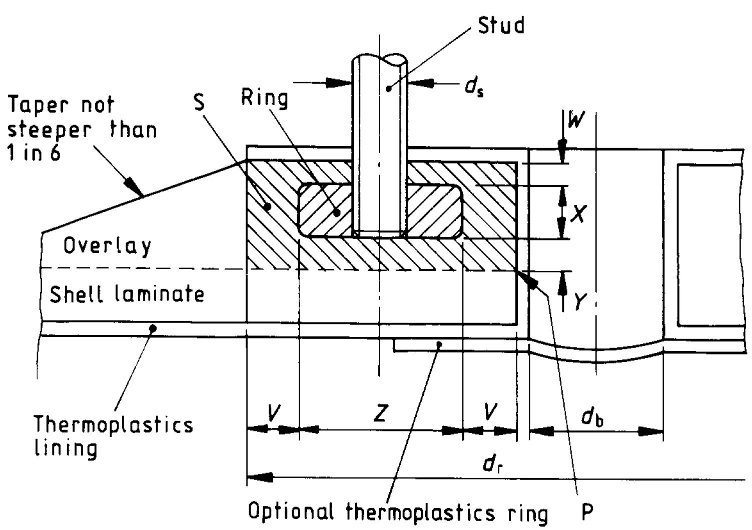



**Figure 22 — Gussets for branches**

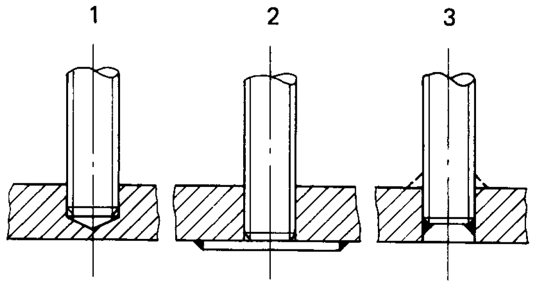



(a) Without thermoplastics lining




(b) With thermoplastics lining

| Dimension | To be not less than |
|---|---|
| *V* | $d_s$ |
| *W* | $0.75d_s$ |
| *X* | $d_s$ + 3 mm |
| *Y* | Thickness of compensation see **20.2** |
| *Z* | $3d_s$ |




1. Screwed into blind hole
2. Screwed with welded back plate
3. Screwed and welded

(c) Permissible fixings for studs into reinforcing ring

**Figure 23 — Pads for vessels**

**20.2.4** *Overlay laminate.* The unit load to be carried by the overlay laminate shall be the maximum unit load for the shell at the position of the opening, and this value shall be substituted in equation (6), for the determination of a suitable laminate construction.

The laminate thus calculated shall be applied to the area around the opening (dimension $d_b$ in Figure 20) to a diameter ($d_r$ in Figure 20) of between two and three times the diameter of the opening in the same plane ($2d_b \leqslant d_r \leqslant 3d_b$).

In those cases where the design strength of the laminate $U_{LAM}$, exceeds the required maximum design loading, $U_c$, it shall be permissible for such excess strength to be taken as contributing to the required load capacity of the compensation $A_L$ as follows:

$$A_E = \{(d_r - d_c)\ (U_{LAM} - U_c)\} \tag{61a}$$

and

$$A_E + A_c \geqslant A_L \tag{61b}$$

Where necessary, local loads shall be considered in addition to the requirements for compensation.

**20.2.5** *Openings in flat panels and flat ends.* In the case of openings in solid flat panels and flat ends, the compensation provided shall comply with all the following.

a) Compensation shall be of a similar type of construction as that of the laminate in which the opening is made.

b) $A_c$ shall be not less than $0.5A_L$ (see **20.2**).

c) The thickness of the compensation shall be not less than $0.5t_p$, within a diameter $d_r$ not less than $2d_c$.

NOTE The compensation of openings in panels of sandwich construction requires special consideration.

**20.2.6** *Pull-out load.* A check calculation shall be made to ensure that the overlay design loading, $U_{OVL}$, is equal to or greater than the pull-out unit load, $Q_b$, where:

$$Q_b = \frac{pd_b}{4} \tag{62}$$

If it is not, the overlay laminate shall be redesigned.

The length over which this overlay laminate is to be applied to the branch shall be not less than 75 mm for all sizes of branch.

**20.2.7** *Prevention of failure in peel.* If all the overlay laminate is applied outside the main shell, the total pull-out load exerted by the branch has to be resisted by the bond between the overlay laminate and the main shell laminate and the soundness of particular design details shall be either demonstrated by prototype testing or verified from experience.

Where the unit pull-out load, $Q_b$, as calculated in accordance with **20.2.6**, exceeds the peel strength of the laminate bond as determined by tests, the branch shall be redesigned.

NOTE In the case of vessels and tanks without thermoplastics linings, improved resistance to pull-out may be obtained from the internal sealing laminate specified in **20.2.3** [see Figure 20(a), Figure 20(b) and Figure 20(c)].

Type testing shall be used to demonstrate that with the resin being used in the manufacture of the vessel or tank and the preparation of the laminate prior to applying the overlay, the peel strength at the interface is at least 5 N/mm.

The peel strength shall be used in the design of the overlay in order to prevent failure in peel. A minimum safety factor of 4 shall be applied.

### 20.3 Branch pipes

#### 20.3.1 *Branch design*

NOTE Internal diameters of branch pipes will normally be determined by the process requirements but any diameter less than 50 mm is not advised.

The design of the branch shall be in accordance with one of the types shown in Figure 20 and Figure 21.

The laminate construction to be used for each branch pipe shall be determined in accordance with clauses **13** and **14**, treating the pipe as a cylindrical shell.

Branches shall be as short as possible but of sufficient length to permit removals of bolts from flanges, etc., and to permit the required overlay length for lap shear.

**20.3.2** *Gusseted branches.* All branches with an internal diameter of 100 mm or less, and branches of any size having a length exceeding the internal diameter shall be provided with supporting gussets which shall be fitted after completion of the main laminate and compensation (see Figure 20(b) and Figure 22). At least three, but preferably four, gussets shall be provided equally spaced around the branch. The length of the gusset leg shall not be less than the internal diameter of the branch, and the thickness shall not be less than the thickness of the branch (excluding compensation) or 6 mm, whichever is the greater. Gussets shall be either made by directly laminating on to the branch and shell, and blending into all curvatures or, if pre-laminated, shall be overlayed in position with a thickness equal to one-half of the gusset thickness on each side.

**20.4 Screwed connections**

Screwed connections, including nipples and tapped branches, shall not be used.

**20.5 Access and inspection openings**

All vessels shall be provided with inspection and/or access openings so located as to permit visual examination of the interior of the vessel or tank. [See also **4.1** f) 10).]

NOTE Attention is drawn to the Factories Act 1961:Section 30.

Manholes and inspection openings shall comply with BS 470.

**20.6 Pad connections**

**20.6.1** *General.* Studded pads shall comply with the minimum requirements of **20.6.2** to **20.6.4** and the minimum dimensions shown in Figure 23.

NOTE If high loads or moments from the piping are expected, or if it is intended to mount a valve directly on to the vessel or tank, the design of the pad will require special consideration.

**20.6.2** *Studs.* The studs shall be of material which will not be appreciably corroded if in contact with the vessel or tank contents.

The studs shall be fitted into a metal reinforcing ring by one of the following methods (see Figure 23):

a) screwing;

b) screwing and welding.

Screwing shall be used to permit stud replacement where the likelihood of damage to studs in service is high, or where machining of the joint face is required.

**20.6.3** *Metal ring.* The material of the metal ring shall be compatible with the material of the studs.

The dimensions of the ring shall be not less than the proportions shown in Figure 23, and the studs shall not protrude beyond the inner face. Studs which are retained by screwing only shall either be fitted into blind tappings in the ring, or alternatively use metal plates welded over the inner face of the ring to prevent the studs from being screwed too far.

The metal ring shall be cleaned to remove any scale or surface deposits and degreased before bonding into the pad.

**20.6.4** *Design of pads.* Pad connections should be designed to mate with flanges complying with either BS 1560 or BS 4504, reducing the actual bore where necessary.

Additional overlay laminate shall be designed to carry the following applied loads:

a) shell loads (see **20.2.3** a) dimension *Y* on Figure 23);

b) transferred loads in shear [see **20.2.3** d)];

c) peel loads [see **20.2.3** e) point P on Figure 23];

and to comply with the minimum overlay dimensions shown in Figure 23.

# 21 Bolted flanged circular connections

## 21.1 General

Where possible, stub and full-face flanged connections shall be designed to connect with class 150 of BS 1560 or Table 10 of BS 4504 as in BS 6464.

Flanged connections shall where possible be as shown in Figure 24 and given in Table 14.

Where this matching of flanges is not possible, e.g. bolted main flanges, flanged connections shall be designed using an appropriate flange design method, backed by prototype test and service experience, where available.

Where loose metallic backing flanges of split construction are used, due allowance shall be made for the effects of loss of continuity in the flange. The thickness of the flange shall be not less than 1.4 times the thickness calculated for an otherwise identical backing flange of solid construction.

NOTE 1 BS 5500, the specification for metallic pressure vessels, may be used as follows.

a) *Stub flange.* No method given explicitly in BS 5500. It is permissible that taper hub flanges may be adapted (see Figure **3.8.3** 1) b), sketch 2 of BS 5500:1985).

Contrary to **3.8.3.4.2** of BS 5500:1985, stresses are not to exceed the design stress, i.e.:

$$\frac{\text{design unit loading (CSM)}}{\text{thickness per unit weight of glass (CSM)}} = \frac{U_{CSM}}{t_g}$$

b) *Loose metal backing flange.* A method is given in **3.8.3.1** a) of BS 5500:1985.

Whatever design method is used, it is essential that there is a proper connection between flange and hub, see Figure 24.

NOTE 2 Loose metal main body backing flanges should be restrained locally away from the flange by local GRP stops.

## 21.2 Full-face flanges with or without backing and with soft ring-type gaskets not less than 1.5 mm thick and extending beyond the bolt holes

Full-faced flanges shall be designed in accordance with **21.2** to **21.7** which provide appropriate flange dimensions to withstand the hydrostatic end loads and bolt loads necessary to ensure a leak-tight joint in service. The mating flanges shall be full faced and flat. The gasket shall extend across the full width of the flange (see Figure 26). Two conditions shall be considered.

a) *Bolting-up condition:* which applies when the gasket or joint contact surface is seated during assembly of the joint at atmospheric temperature, and pressure; the minimum bolt load under this condition is a function of gasket material and the effective gasket contact area to be seated.

b) *Operating condition:* which applies when the hydrostatic end force due to the design pressure tends to part the joint and the bolt load has to maintain sufficient compression on the gasket or joint contact surface to ensure a tight joint at design temperature; the minimum bolt load under this condition is a function of design pressure, the gasket material and the effective gasket or contact area to be kept tight under pressure.

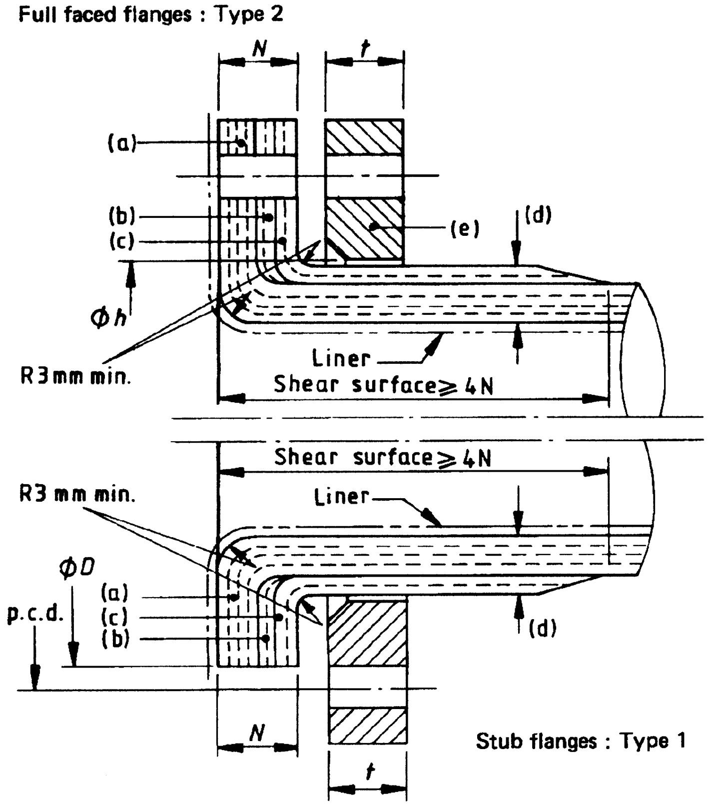
In all cases care shall be taken to ensure that the flange material is of adequate crushing strength to resist the applied bolt and joint loads.

As gasket seating is normally accomplished at atmospheric temperature whereas flange and bolt temperatures under operating conditions may be significantly above or below this temperature, the allowable stress for each temperature condition shall therefore be taken into account in the determination of the required flange design. In the design of large-diameter flanges, special consideration shall be given to the choice of gasket, size and pitch of bolts and sequence of bolt tightening when closing the joint, Special consideration shall be also given to applications where flanges are subject to significant additional loading.

**Table 14 — Flange and bolt details** (for guidance see Figure 24)

| Branch nominal size | Backing flange thickness[a], minimum | | Outside diameter and drilling in accordance with class 150 of BS 1560 | | | | | | Outside diameter and drilling in accordance with Table 10 of BS 4504 | | | | | Sub flange (type 1) thickness, *N* | | | | Full-faced flange (type 2) thickness, *N* | Stub flange outside diameter, $D^b$ | | Bolt tightening torque | | Branch nominal size |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Pressure 10 bar | | OD | PCD | Hole diameter | | Bolts | | OD | PCD | Hole diameter | Bolts | | Design strain | | | | | Class 150 of BS 1560 | Table 10 of BS 4504 | | | |
| | | | | | | | | | | | | | | 0.2 | 0.16 | 0.13 | 0.1 | 0.2 | | | | | |
| | Solid | Split | | | | | No. | Size | | | | No. | Size | Pressure 10 bar | | | | | | | | | |
| mm | mm | mm | mm | mm | in | mm | | in | mm | mm | mm | | | mm | mm | mm | mm | mm | mm | mm | N m | lbf ft | mm |
| 25 | — | — | 115 | 79.4 | 5/8 | 15.9 | 4 | 1/2 | 115 | 85 | 14 | 4 | M12 | — | — | — | — | 23 | — | — | — | — | 25 |
| 32 | — | — | 125 | 88.9 | 5/8 | 15.9 | 4 | 1/2 | 140 | 100 | 18 | 4 | M16 | — | — | — | — | 24 | — | — | — | — | 32 |
| 40 | — | — | 134 | 98.4 | 5/8 | 15.9 | 4 | 1/2 | 150 | 110 | 18 | 4 | M16 | — | — | — | — | 25 | — | — | — | — | 40 |
| 50 | 10 | 14 | 152 | 120.6 | 3/4 | 19.0 | 4 | 5/8 | 165 | 125 | 18 | 4 | M16 | 10 | 10 | 10 | 12 | 28 | 102 | 107 | 20 | 15 | 50 |
| 65 | 10 | 14 | 178 | 139.7 | 3/4 | 19.0 | 4 | 5/8 | 185 | 145 | 18 | 4 | M16 | 10 | 10 | 11 | 13 | 30 | 121 | 127 | 23 | 17 | 65 |
| 80 | 10 | 14 | 190 | 152.4 | 3/4 | 19.0 | 4 | 5/8 | 200 | 160 | 18 | 8 | M16 | 10 | 11 | 12 | 14 | 32 | 133 | 142 | 27 | 20 | 80 |
| 100 | 12 | 17 | 229 | 190.5 | 3/4 | 19.0 | 8 | 5/8 | 220 | 180 | 18 | 8 | M16 | 14 | 15 | 16 | 18 | 32 | 172 | 162 | 20 | 15 | 100 |
| 125 | 12 | 17 | 254 | 215.9 | 7/8 | 22.2 | 8 | 3/4 | 250 | 210 | 18 | 8 | M16 | 15 | 16 | 17 | 19 | 32 | 194 | 192 | 30 | 22 | 125 |
| 150 | 13 | 18 | 279 | 241.3 | 7/8 | 22.2 | 8 | 3/4 | 285 | 240 | 22 | 8 | M20 | 16 | 17 | 18 | 20 | 32 | 219 | 218 | 34 | 25 | 150 |
| 200 | 15 | 21 | 343 | 298.4 | 7/6 | 22.2 | 8 | 3/4 | 340 | 295 | 22 | 8 | M20 | 18 | 20 | 22 | 24 | 38 | 276 | 273 | 47 | 35 | 200 |
| 250 | 18 | 25 | 406 | 362.0 | 1 | 25.4 | 12 | 7/8 | 395 | 350 | 22 | 12 | M20 | 22 | 24 | 26 | 28 | 45 | 337 | 328 | 50 | 37 | 250 |
| 300 | 21 | 30 | 483 | 431.8 | 1 | 25.4 | 12 | 7/8 | 445 | 400 | 22 | 12 | M20 | 26 | 29 | 31 | 34 | 50 | 406 | 378 | 75 | 55 | 300 |
| 350 | 22 | 31 | 533 | 476.2 | 1 1/8 | 28.6 | 12 | 1 | 505 | 460 | 22 | 16 | M20 | 26 | 29 | 31 | 34 | 55 | 448 | 438 | 95 | 70 | 350 |
| 400 | 24 | 34 | 597 | 539.8 | 1 1/8 | 28.6 | 16 | 1 | 565 | 515 | 26 | 16 | M24 | 28 | 30 | 33 | 36 | 55 | 511 | 489 | 88 | 65 | 400 |
| 450 | 25 | 35 | 635 | 577.8 | 1 1/4 | 31.8 | 16 | 1 1/8 | 615 | 565 | 26 | 20 | M24 | 30 | 32 | 35 | 38 | 60 | 546 | 539 | 115 | 85 | 450 |
| 500 | 27 | 38 | 698 | 635.0 | 1 1/4 | 31.8 | 20 | 1 1/8 | 670 | 620 | 26 | 20 | M24 | 31 | 34 | 36 | 39 | 60 | 603 | 594 | 115 | 85 | 500 |
| 600 | 32 | 45 | 813 | 749.3 | 1 3/8 | 34.9 | 20 | 1 1/4 | 780 | 725 | 30 | 20 | M27 | 35 | 38 | 40 | 44 | 65 | 714 | 695 | 163 | 120 | 600 |
| | Pressure 6 bar | | Dimensions and drilling in accordance with class 150 of BS 3293 | | | | | | Table 10 of BS 4504 | | | | | Pressure 6 bar | | | | | Class 150 of BS 3293 | Table 10 of BS 4504 | | | |
| 700 | 29 | 41 | 927[c] | 864[c] | 1 3/8 | 34.9 | 28 | 1 1/4 | 895 | 840 | 30 | 24 | M27 | 36 | 39 | 43 | 47 | 55 | 829 | 810 | 108 | 80 | 700 |
| 800 | 32 | 45 | 1 061[c] | 978[c] | 1 5/8 | 41.3 | 28 | 1 1/2 | 1 015 | 950 | 33 | 24 | M30 | 39 | 42 | 46 | 50 | 60 | 937 | 917 | 156 | 115 | 800 |
| 900 | 35 | 49 | 1 168[c] | 1 086[c] | 1 5/8 | 41.3 | 32 | 1 1/2 | 1 115 | 1 050 | 33 | 28 | M30 | 43 | 47 | 50 | 55 | 65 | 1 045 | 1 017 | 190 | 140 | 900 |
| 1 000 | 39 | 55 | 1 289[c] | 1 200[c] | 1 5/8 | 41.3 | 36 | 1 1/2 | 1 230 | 1 160 | 36 | 28 | M33 | 46 | 50 | 54 | 59 | 70 | 1 159 | 1 124 | 197 | 145 | 1 000 |

[a] Based on rubber gaskets, 75 IRHD, with a seating stress of 2.32 N/mm$^2$.
[b] $D$ = pitch circle diameter (p.c.d) — bolt hole diameter.
[c] These values are metric conversions.




NOTE 1 Above 40 mm nominal bore to be used only by agreement between the purchaser and the manufacturer [see **4.2** f) 7)].

a) Pipe wall CSM and WR continuing into flange face.

b) Additional glass fibre reinforcement as required to give flange thickness. To be tapered on to the hub.

NOTE 2 The additional glass fibre reinforcement may be included in either a) or c).

c) Final overlay of 1.2 kg/m$^2$ CSM across flange into hub.

d) Hub thickness ⩾ $N/2$.

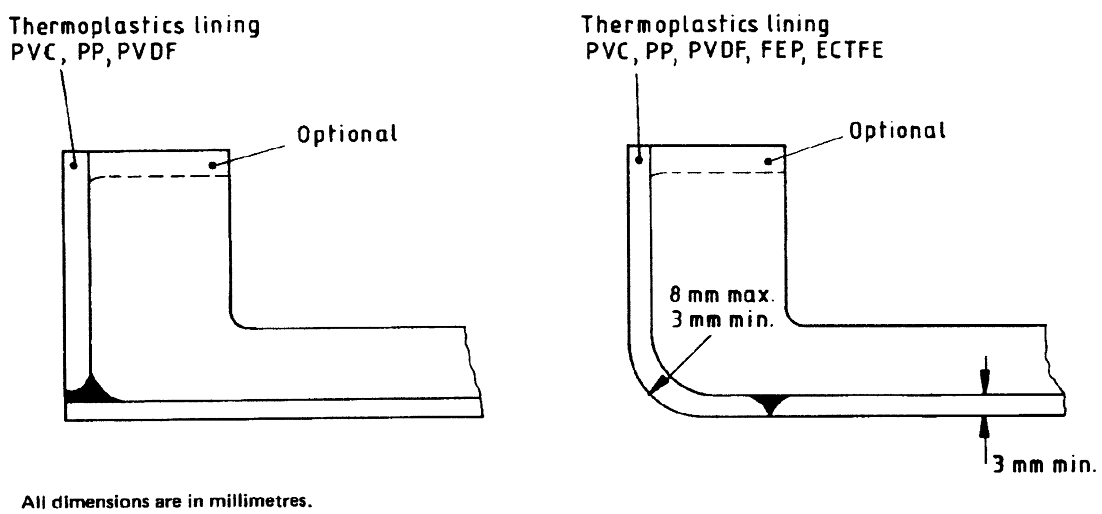
NOTE 3 This requirement may be satisfied by pipe wall reinforcement alone.

e) Steel backing flange not less than 6 mm thick and flat over the full bearing area to be used for pressure duties.

NOTE 4 The clearance between the bore of the backing flange and the outside diameter of the hub should be as small as practicable.

NOTE 5 For flange dimensions see Table 14.

**Figure 24 — Flange construction**




All dimensions are in millimetres.

**Figure 25 — Details of flanges for thermoplastics lined GRP vessels**

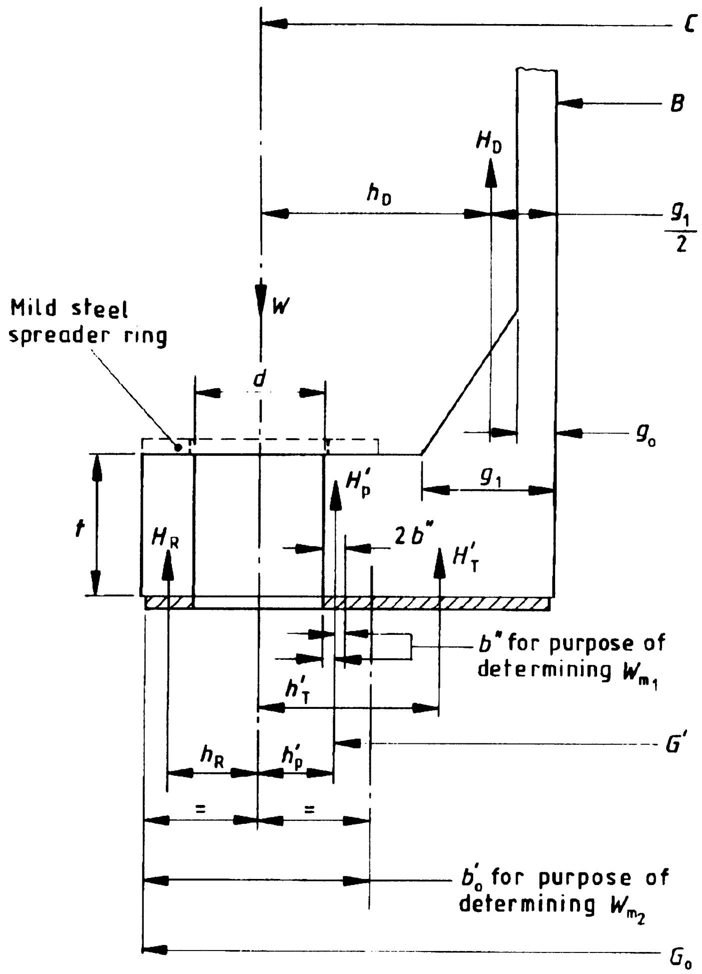



**Figure 26 — Full-faced flange with soft gasket**

### 21.3 General requirements for bolting

Bolts and studs shall have a nominal diameter of not less than 12 mm.

NOTE Table 15 gives recommended bolt stresses for determining the appropriate number of bolts using the design calculations in clause **21**. These stress levels are nominal insofar as they may have to be exceeded in practice to provide against all conditions that tend to produce a leaking joint. However, there is sufficient margin to provide a satisfactory closure without having to overload or continually retighten the bolts. Torque spanners or other means for preventing the application of excessive load either on the bolt or flange may be necessary.

### 21.4 Symbols

For the purposes of clause **21** the following symbols apply and consistent units shall be used in the equations.

$A_b$ actual bolt area

$A_m$ minimum bolt area (the greater of $A_{m1}$ and $A_{m2}$)

$A_{m1}$ minimum bolt area for operating condition

$A_{m2}$ minimum bolt area for gasket seating

$b'_o$ basic gasket seating width effective under initial tightening up (symmetrically disposed relative to bolt circle), $= G_o - C$

$b'$ effective gasket seating width for calculation of minimum seating loads using $y$ factor $= 4\sqrt{b'_o}$ (valid only for dimensions in mm)

| | |
|---|---|
| $2b''$ | effective gasket contact width under pressure, taken to be 5 mm |
| $B$ | inside diameter of flange |
| $C$ | bolt circle diameter |
| $d$ | diameter of bolt holes |
| $d_b$ | bolt diameter |
| $E_{LAM}$ | Young's modulus of laminate |
| $g_o$ | thickness of hub at small end |
| $g_l$ | thickness of hub at back of flange |
| $G'$ | diameter at location of gasket load reaction $H'_p$, $= C - (d + 2b'')$ |
| $G_o$ | outside diameter of gasket or outside diameter of flange, whichever is the lesser |
| $h_D$ | radial distance from bolt circle to circle on which $H_D$ acts, $= R + 0.5g_l$ |
| $h'_T$ | radial distance from bolt circle to circle on which $H'_T$ acts, $= \dfrac{(C + d + 2b'') - B}{4}$ |
| $h'_p$ | radial distance from bolt circle to circle on which $H'_p$ acts, $= \dfrac{d + 2b''}{2}$ |
| $h_R$ | radial distance from bolt circle to circle on which $H_R$ acts, $= \dfrac{G_o - (C + d)}{4} + \dfrac{d}{2}$ |
| $H_D$ | hydrostatic end force on area inside of flange, i.e. force applied via connection to flange, $= 0.785B^2p$ |
| $H'$ | total hydrostatic end force, $= 0.785(C - d)^2p$ |
| $H'_T$ | hydrostatic end force due to pressure on flange face, $= H' - H_D$ |
| $H'_p$ | compression load on gasket to ensure tight joint, $= 2b'' \times 3.14\, G'\, mp$ |
| $H_R$ | balancing reaction force acting outside bolt circle in opposition to moments due to $H_D$, $H'_p$ and $H'_T$ |
| $m$ | gasket factor = 1 for soft rubber without fabric or asbestos reinforcement |
| $n$ | number of bolts |
| $p$ | design pressure |
| $R$ | $= \dfrac{C - B}{2} - g_l$ |
| $S_a$ | bolt nominal design stress at atmospheric temperature (see Table 15) |
| $S_b$ | bolt nominal design stress at design temperature (see Table 15) |

$$S_{CSM} = \frac{u_{CSM}}{t_g} = \frac{\text{design unit loading (CSM)}}{\text{thickness per unit mass of glass (CSM)}}$$

| | |
|---|---|
| $t$ | flange thickness |
| $W_{m1}$ | minimum required bolt load for operating conditions, $= H_p + H$ |
| $W_{m2}$ | minimum required bolt load for gasket seating |
| $W$ | flange design bolt load, based on actual size bolts used to provide greater of $W_{m1}$ and $W_{m2}$ |
| $Y$ | gasket or joint-contact-surface unit seating stress = 2.32 N/mm² for soft rubber without fabric or asbestos reinforcement |

**Table 15 — Recommended design stress values for flange bolting materials**

| Material | British Standard reference | Diameter | Recommended design stress for design temperatures (°C) not exceeding | | |
|---|---|---|---|---|---|
| | | | 50 | 100 | 200 |
| | | mm | N/mm² | N/mm² | N/mm² |
| Carbon steel | BS 1506-111 | ⩽ 152 | 93 | 89 | 85 |
| 1 % chromium molybdenum steel | BS 4882 grade B7, L7 | ⩽ 63 | 193 | 181 | 167 |

### 21.5 Bolt loads and areas

The operating and bolting up conditions shall be as follows.

a) *Operating conditions*. The minimum bolt load for the operating conditions, $W_{m1}$, shall be determined from equation (63):

$$W_{m1} = H' + H_p' + H_R \quad (63)$$

where

$$H_R = \frac{H_D h_D + H_T' h_T' + H_p' h_p'}{h_R}$$

b) *Bolting up conditions*. The minimum bolt load for gasket seating, $W_{m2}$, shall be determined from equation (64):

$$W_{m2} = 3.14\, C b' y = 12.56\, C y \sqrt{b_o'} \quad (64)$$

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 not be less than $A_m$.

### 21.6 Flange design

The flange thickness shall be not less than the value of $t$ determined from equation (65):

$$t = \sqrt{\frac{6M}{S_{CSM}\,(3.14C - nd)}} \quad (65)$$

where

$M = H_R\, h_R$

### 21.7 Bolting

The pitch of bolts shall not exceed the following, except where the minimum pitch is dictated by consideration of the space necessary to apply a spanner to the nuts and interference from gussets and other obstructions.

$$2d_b + \frac{6t}{m + 0.5}\left(\frac{E_{LAM}}{200\,000}\right)^{0.25} \quad (66)$$

where $E_{LAM}$ is expressed in N/mm².

## 22 Supports

### 22.1 General

Special attention shall be given to the design of supports for vessels and tanks in reinforced plastics, so as to avoid excessive loads at points of support. All supports shall be designed to accommodate temperature changes and to permit free expansion and contraction. The relatively large thermal expansion of non-metallic materials shall be taken into account.

Where there is steel bonded into the support, the design shall be checked for stresses in the GRP caused by restraint to thermal expansion/contraction where the design temperature is above 60 °C or below 0 °C.

NOTE Steel supporting structures which do not form part of the vessel or tank should comply with BS 449. When supports are to be constructed of reinforced concrete, CP 114 should be consulted.

The proposals for support shall be agreed between the purchaser and the manufacturer at the design stage [see **4.2** i)] and the agreed details shall be specified on the approved drawings. The principles set out in **22.2** and **22.3** shall apply.

Vessels and tanks fitted with an agitator shall have supports designed with a load multiplication factor of 1.25.

NOTE Vessels and tanks to be buried should be the subject of special consideration.

### 22.2 Vertical vessels and tanks

**22.2.1** *Flat-bottom configuration.* Continuous support shall be provided over the whole area of the flat bottom except where sumps or bottom run offs have to be accommodated.

The gaps in the support to allow for bottom branches shall be sized so that there is room for gussets and so that high local loads are not imposed on the additional reinforcement which is used for compensation of the branch opening.

NOTE Recommended supports are indicated in Appendix H.

Anchorages shall be provided if under the worst conditions there is a tendency for the shell to lift off its foundation; a design procedure to cover case of uplift in tanks is given in Appendix F and typical holding down arrangements are shown in Figure 27.

**22.2.2 *Dished-bottom configuration***

**22.2.2.1** *Drop-through.* Vessels or tanks supported through a building floor or within a structure shall either be provided with a ring girder or be supported by an adequate number of supporting brackets to support the load.

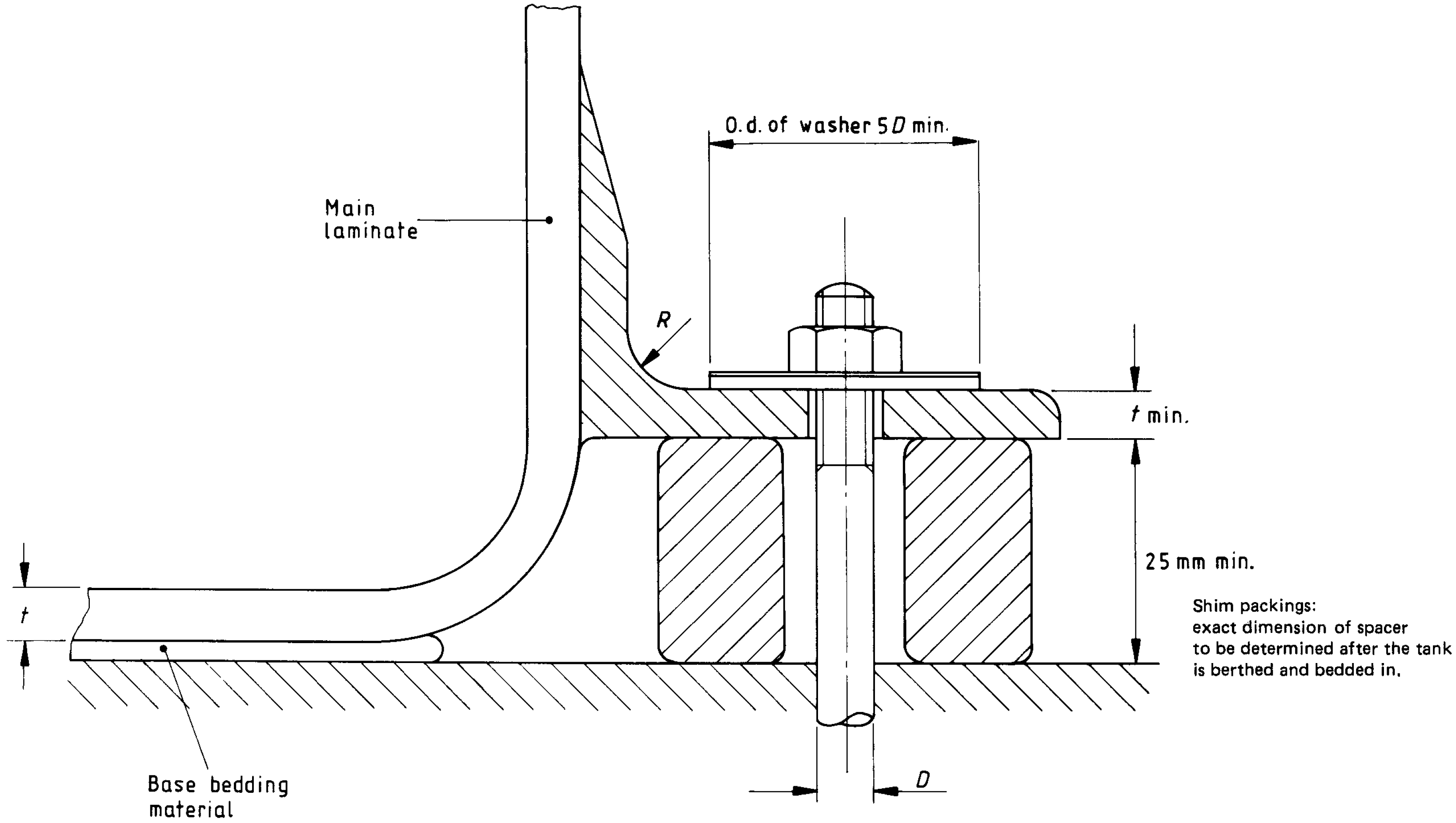
In either case, additional reinforcement shall be provided, see Figure 28. To avoid excessive bending loads, the supporting members shall be located as close as practicable to the shell.

**22.2.2.2** *Base-supported.* Supports shall be arranged in either of the following ways.

a) *On an angle ring bearing on a ring girder or legs [see Figure 29(a)].* The ring girder or legs shall have a horizontal top bearing surface of sufficient area to avoid compressive overload of the laminate between the angle ring and the support.

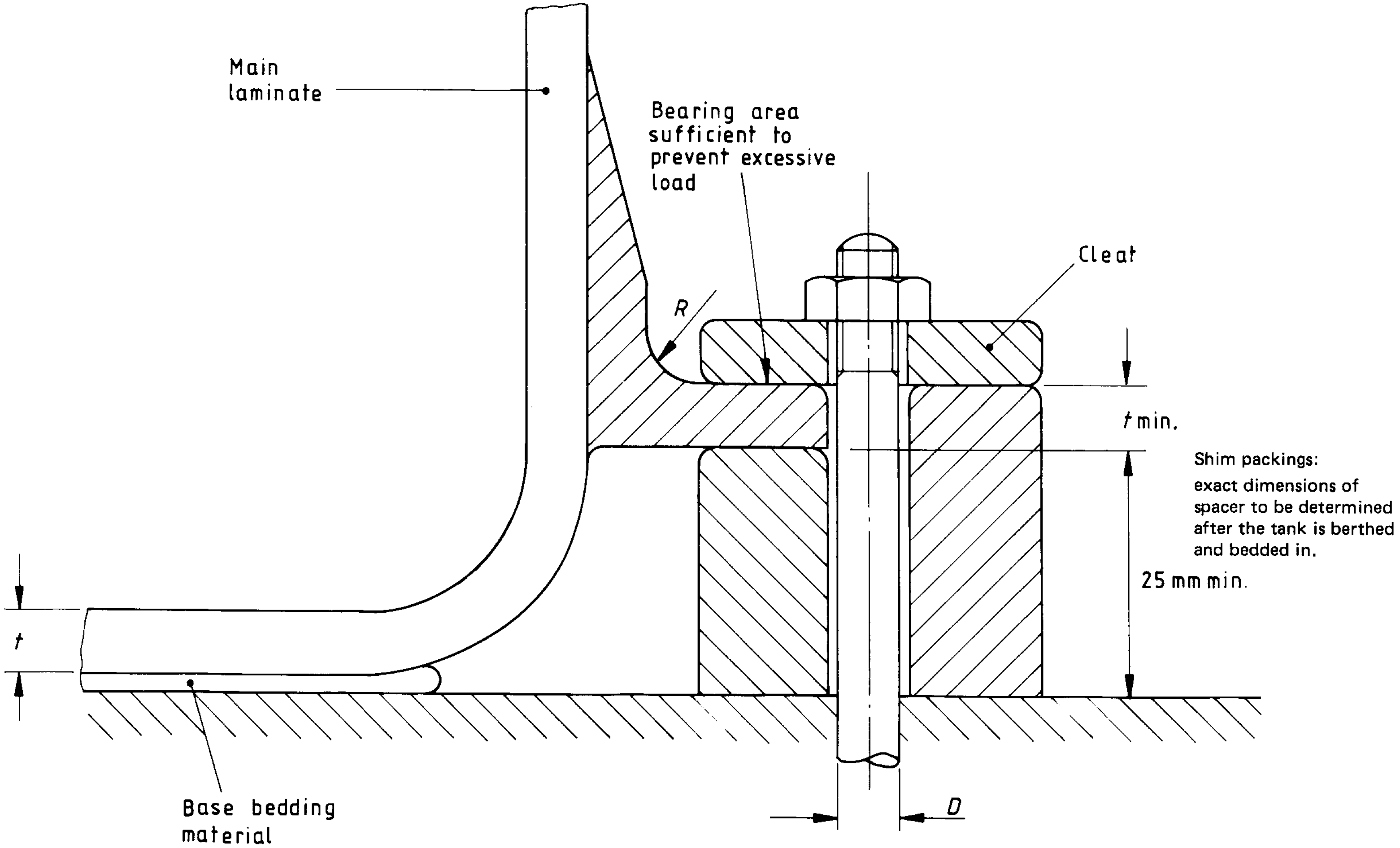
The overturning moment shall be calculated and arrangements made for holding-down bolts as appropriate.

b) *On a full circumference skirt [see Figure 29(b)].* The skirt shall be externally attached. The skirt laminate shall be designed and constructed in accordance with clause **14**. Where cutouts are required in the skirt, they shall be adequately reinforced and comply with the requirements for compensation of openings (see **20.2**) so as to ensure stability of the vessel or tank. The joint between the shell and the skirt shall be determined in accordance with equations (56) and (57) and the construction of the skirt shall be used as the basis for this calculation.



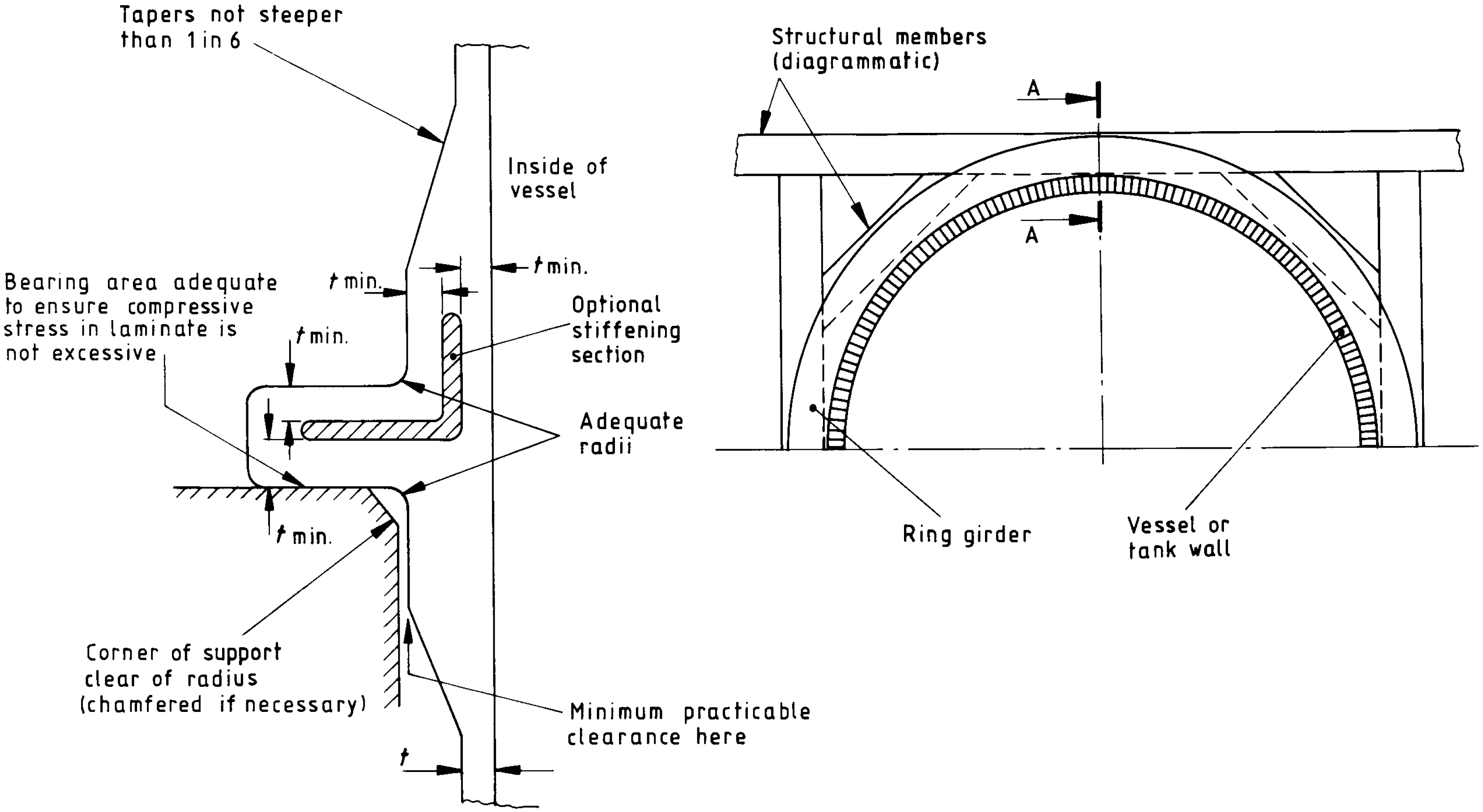

(a) Bolted anchorage

**Figure 27 — Typical anchorage arrangements for flat-bottomed tanks (brackets or ring)** *(continued)*



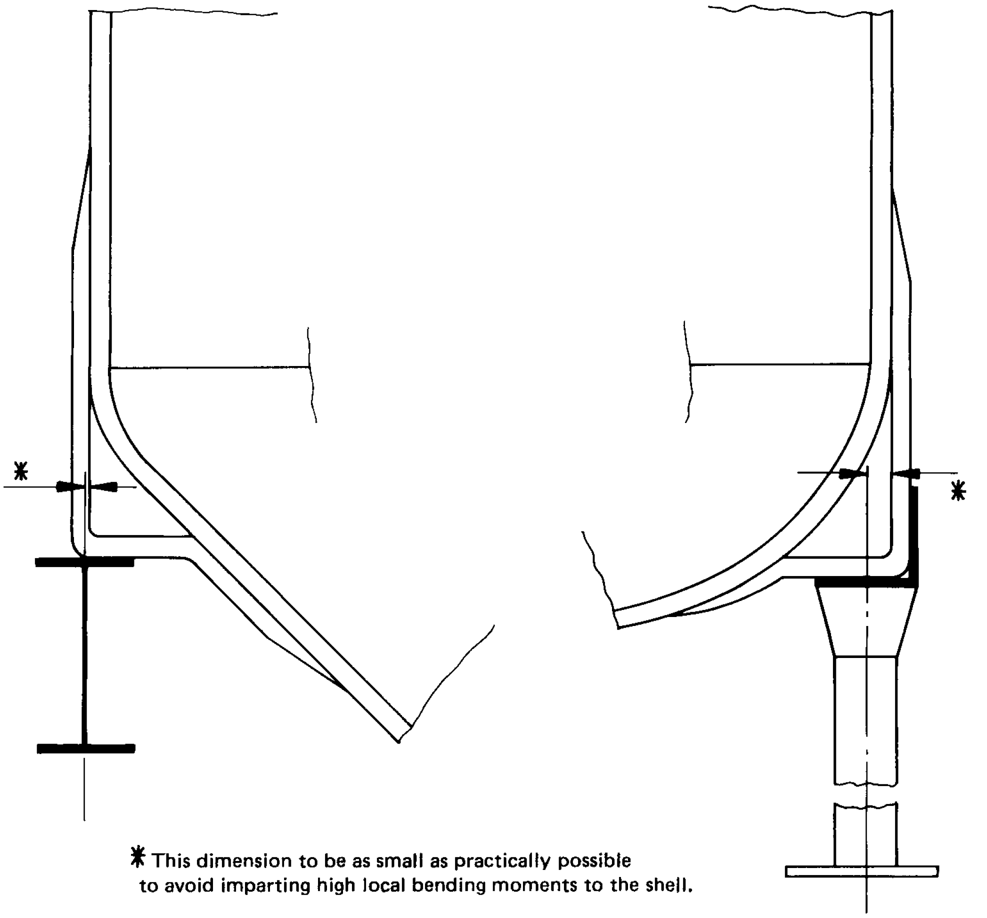

(b) Clamped anchorage

**Figure 27 — Typical anchorage arrangements for flat-bottomed tanks (brackets or ring)** *(concluded)*




Typical detail cross section of A-A ring girder or bracket

**Figure 28 — Supports for drop-through vessels and tanks**




(a) Angle ring

**Figure 29 — Support arrangements for base-supported vertical vessels and tanks** *(continued)*

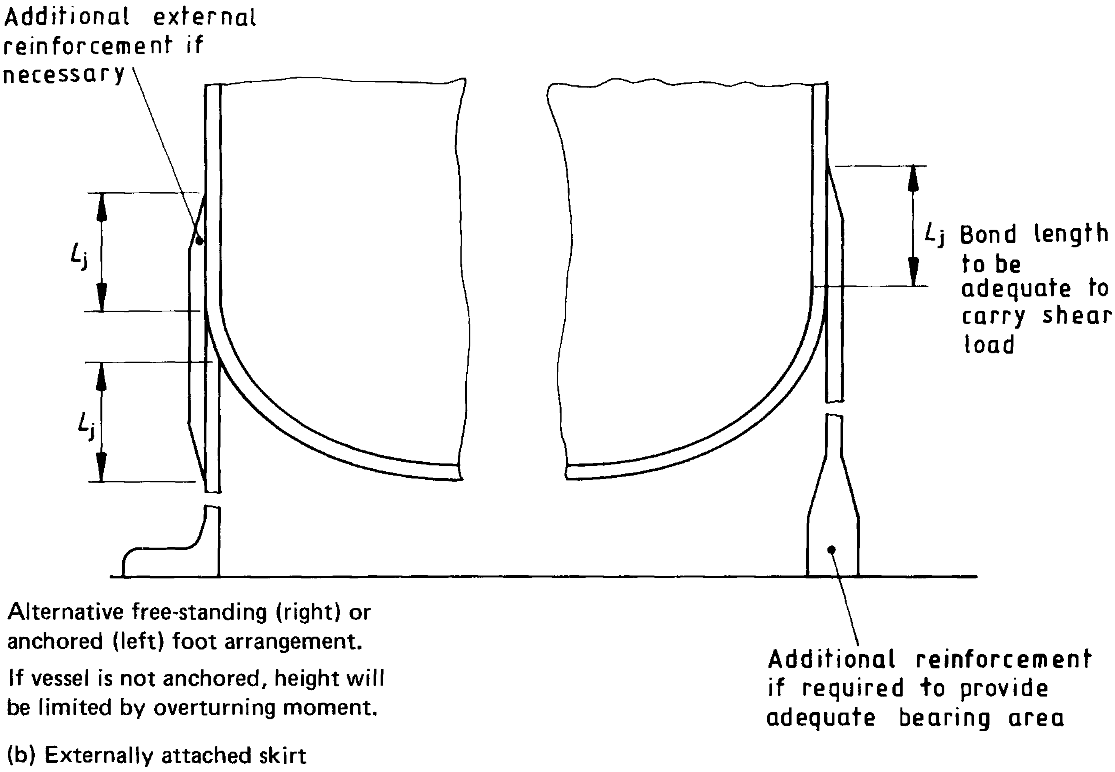



**Figure 29 — Support arrangements for base-supported vertical vessels and tanks** *(concluded)*

## 22.3 Horizontal vessels and tanks

**22.3.1** *General.* Horizontal vessels and tanks shall be supported on saddles or on full length channels, or suspended from sling straps. Provision shall be made to accommodate movements of the vessels or tanks relative to the support bearing areas.

NOTE It is important to ensure that the shell close to the supports remains round under load.

**22.3.2** *Longitudinal supports [see Figure 30(a)].* The longitudinal bearing areas shall extend over the full parallel length of the vessel or tank and their width shall be calculated so that the permissible compressive loading in the laminate is not exceeded.

The distance between the outermost edges of the supports shall be not less than three-quarters of the shell diameter.

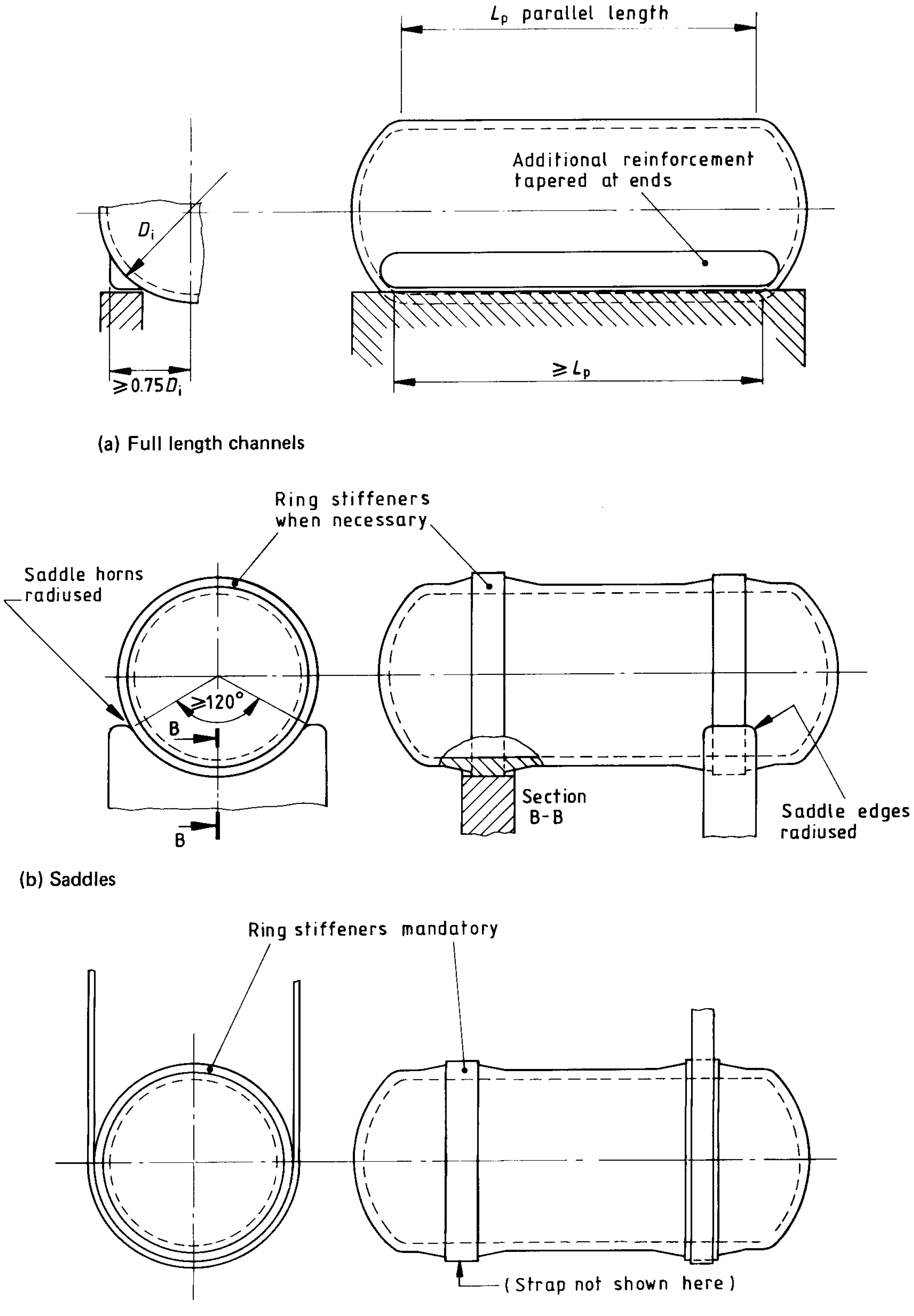
**22.3.3** *Saddles [see Figure 30(b)].* Wherever possible, saddle-supported vessels and tanks shall be designed for support at two cross sections only, although vessels with more supports have proved satisfactory in service. All saddles shall be carefully aligned. Reinforcement of the shell with stiffening rings at the saddle locations shall be provided where deflection of the shell would otherwise take place.

NOTE 1 For shells of internal diameter less than 1 500 mm, the curvature may provide adequate stiffening.

The included angle supported by a saddle shall be not less than 120° and the saddle width shall be calculated to ensure that the bearing pressure does not exceed the compressive loading for the laminate. The edges of saddle supports shall be radiused both in the circumferential and longitudinal directions over the saddle horns.

High local loads adjacent to the point of support, thermal loadings and the possibility of buckling shall all be taken into account in the design.

NOTE 2 A detailed analysis for calculations of the support of horizontal steel shells is given in Appendix G of BS 5500:1985. This analysis may be used for GRP vessels but caution has to be exercised in view of the strain limitation. (See Appendix G.)




(a) Full length channels

(b) Saddles

(c) Sling straps

**Figure 30 — Supports for horizontal vessels and tanks**

**22.3.4** *Sling straps [see Figure 30(c)].* All vessels, and tanks suspended from sling straps shall be reinforced with stiffening rings at the sling positions. The strap width shall be calculated to ensure that the bearing pressure does not exceed the compressive loading for the laminate. High local loads adjacent to the points of support and the possibility of buckling shall be taken into account in the design. The method of suspension shall be such as to ensure freedom of movement under thermal or fluctuating loading.

NOTE It is undesirable for a sling strap to encircle more than half the circumference of the shell.

## 23 Structures and fittings

### 23.1 General

All structures and fittings shall be arranged as far as practicable to avoid imposing local concentrated loads on the walls of the vessel or tank. The design shall be such that the local strain does not exceed the allowable strain (see **9.2.6**).

NOTE 1 For design of lifting lugs see Appendix J.

NOTE 2 It is recommended that the detailed design of such structures and fittings should be agreed between the purchaser and the manufacturer before fabrication commences.

### 23.2 Internal structures and fittings

The material used for internal fittings shall be suitable for the conditions of service.

NOTE 1 In general, for lined vessels this will be the same material as the thermoplastics lining.

Fittings carrying load, such as tray supports and all fittings in vessels and tanks fitted with agitators, shall be given special consideration to ensure adequate strength for the service conditions. Sufficient reinforcement shall be provided.

NOTE 2 In certain cases, steel reinforcement should be considered. For design temperatures greater than 60 °C, special consideration is required.

Local loads from internal structures or from contents shall be carried, where possible, directly to the supports and thus to the foundations, without overloading the walls or ends of the vessel or tank.

### 23.3 External structures and fittings

Where an agitator and drive are fitted, they shall be supported, either:

a) independently of the vessel or tank; or

b) on integral supports.

In the case where integral supports, which also act as strengthening ribs for the ends, are provided, they shall be designed to have adequate strength and rigidity for both purposes.

NOTE It is preferred to use support system a).

Where required, vessels and tanks shall be provided with lifting lugs or other suitable attachments for handling on site.

### 23.4 Protective shields

Consideration shall be given, where appropriate, to the incorporation of protective shields in areas of the vessel or tank which may be subject to accidental mechanical damage, e.g. beneath manholes or around branch openings. Where such shields are fitted they shall be of adequate size and of suitably resistant material for the conditions of use.

## 24 Local load analysis

### 24.1 General

This clause is concerned with the assessment of the effects of local loads and moments on the shell wall of a vessel, which may arise for example from support or lifting devices or from pipework connected to branches, and wherever possible, these loads shall be reduced to a minimum by careful design of the attachment or of the connecting pipework system.

### 24.2 Symbols

For the purposes of clause **24** the following symbols apply.

$E_C$ coupling modulus of laminate (in N/mm$^2$) [see equation (69)]. Subscript x indicates longitudinal and subscript $\phi$ indicates circumferential.

$E_F$ flexural rigidity of laminate (in N/mm$^2$) [see equation (69)]. Subscript x indicates longitudinal and subscript $\phi$ indicates circumferential.

$E_i$ Young's modulus of the $i$th layer (in N/mm$^2$)

$E_T$ tensile modulus of laminate (in N/mm$^2$) [see equation (69)]. Subscript x indicates longitudinal and subscript $\phi$ indicates circumferential.

$h_i$ thickness of $i$th layer (in mm)

$M_x$ moment resultant per unit circumference (in N·mm/mm) in the longitudinal or meridional direction (calculated in accordance with Appendix G)

$M_\phi$ moment resultant per unit length (in N·mm/mm) in the circumferential direction (calculated in accordance with Appendix G)

$n$ number of layers in laminate

$N_x$ force per unit circumference (in N/mm) in the longitudinal or meridional direction (calculated in accordance with Appendix G)

$N_\phi$ force per unit length (in N/mm) in the circumferential direction (calculated in accordance with Appendix G)

$t$ thickness of laminate (in mm)

$Z_i$ distance of the $i$th layer from the mid-plane of laminate (in mm)

$\epsilon_x$ strain in longitudinal or meridional direction

$\epsilon_\phi$ strain in circumferential direction

### 24.3 Calculation of force and moment resultants

Force and moment resultants in the shell wall shall be calculated. A suitable method is given in Appendix G and due care shall be taken as to the geometric limitations detailed in **G.2.2**, **G.2.3** and **G.2.4**.

If, for a cylindrical shell, the maximum applied force or moment does not lie in the longitudinal or circumferential directions these forces or moments shall be resolved into these directions and the effect of each individual component summed.

Due to the brittle nature of GRP, analytical techniques involving shakedown or other effects of plasticity shall not be used.

NOTE The published experimental work in support of this analysis has been carried out almost exclusively with steel and as a result care should be taken when applying it to GRP.

### 24.4 Calculation of laminate strains

As the type of layers and hence, the allowable stress, may vary through the thickness of the GRP laminate, the criterion of acceptance of a load system shall be that of maximum strain.

NOTE In the following two equations (67) and (68) the Poisson effect has been ignored.

For a shell made entirely from CSM or another isotropic material the resultant maximum strains in the shell are given by equation (67):

$$\epsilon_\phi = \frac{N_\phi}{E_{T\phi}\,t} \pm \frac{6M_\phi}{E_{T\phi}\,t^2}$$
$$\epsilon_x = \frac{N_x}{E_{Tx}\,t} \pm \frac{6M_x}{E_{Tx}\,t^2} \qquad (67)$$

For a shell laminate made up of layers of different properties, e.g. CSM and WR, account shall be taken of the disposition of the different layers through the laminate thickness.

The maximum strains for such a laminate (see Figure 31) are approximated by:

$$\epsilon_\phi = \frac{N_\phi}{E_{T\phi} t} + \frac{6M_\phi}{E_{F\phi} t^2}\left(\frac{2E_{C\phi}}{E_{T\phi}} \pm 1\right)$$

$$\epsilon_x = \frac{N_x}{E_{Tx} t} + \frac{6M_x}{E_{Fx} t^2}\left(\frac{2E_{Cx}}{E_{Tx}} \pm 1\right) \qquad (68)$$

where

$$E_T = \frac{1}{t}\sum_{i=1}^{n} E_i h_i$$

$$E_C = \frac{1}{t^2}\sum_{i=1}^{n} E_i h_i Z_i \qquad (69)$$

$$E_F = \frac{1}{t^3}\sum_{i=1}^{n} E_i \left(12 h_i Z_i^2 + h_i^3\right)$$

The values for the modulus of the individual layers $E_i$ shall be derived from the values given in Table 5.

To assess the acceptability of the shell strains derived from equations (67) and (68) these shall be added to those caused by other loads, e.g. pressure and self-weight (see **G.2.2.2.4**). These total strains shall then be compared to the allowable strain derived in **9.2**.

Where continuous rovings are filament wound at an angle ± $\theta$ to the vessel or tank axis, values of circumferential and longitudinal modulus for individual layers shall be obtained by reference to Figure 3.

**24.5 Additional reinforcement**

If the calculations in **21.3** and **21.4** indicate applied strains in excess of the allowable, additional reinforcement shall be applied in the region of the overstrain. This additional reinforcement shall be interleaved within the main shell laminate.

Further calculations shall be carried out to ensure that the applied strain is within acceptable limits at the point of application of the local load and at the point where any extra reinforcement ends.

If extra reinforcement is required due to external loadings on a branch, the area of such reinforcement shall be at least that specified in **20.2.4**.

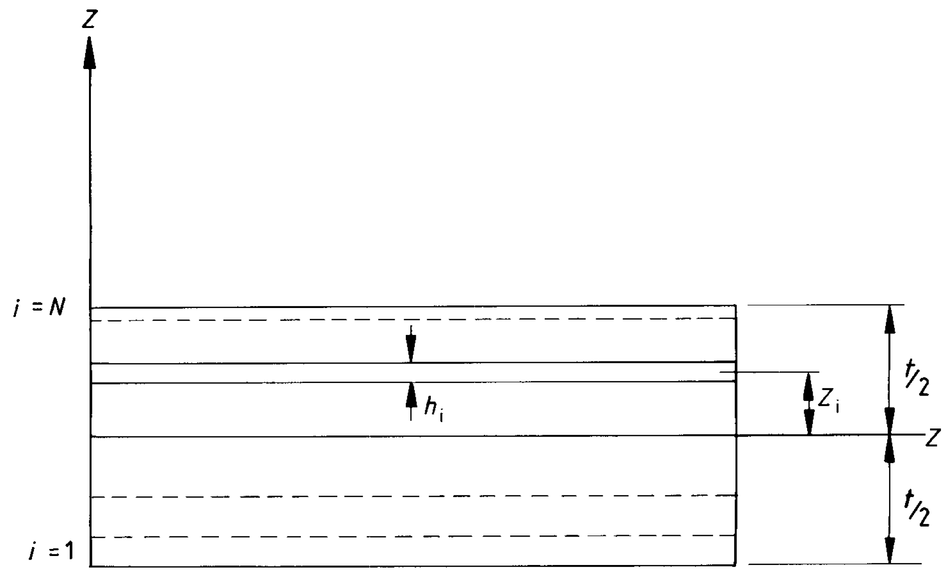



**Figure 31 — *n* ply laminated beam**

# Section 4. Construction and workmanship

## 25 Approval of design and construction details

Except for stock items previously approved and inspected in accordance with this standard, the manufacturer shall, before commencing manufacture, submit for approval by the purchaser all drawings showing details of design and construction as given in Table 3.

No modification shall be made to the approved design except with prior agreement between the purchaser and the manufacturer [see **4.2** j)].

## 26 Conditions in works

NOTE In the construction of GRP equipment, it is important that the environmental conditions are closely controlled during the manufacturing processes so as to ensure that the mechanical and chemical properties required are actually achieved in the product. It is therefore preferable that vessels should be shop fabricated, where variations in, for example, temperature and air movement can be minimized.

Manufacture shall be under controlled environmental conditions compatible with producing a satisfactory vessel or tank, i.e. the control of temperature, humidity, air movement and light.

The working area shall be suitably divided into clearly defined sections for preparation of reinforcement, mixing of resins, application, trimming and finishing.

Materials shall be stored and used in compliance with the suppliers' instructions; reinforcement materials shall be stored dry.

When laminating is carried out at air temperatures below 15 °C, special procedures may be required but in no case shall the maximum proportion of curing agents, e.g. peroxides, naphthenates or amines, exceed the recommendations of the resin supplier.

All laminating shall be discontinued whenever the air temperature falls to 10 °C or the dew-point is reached (condensation occurs) unless special arrangements are made to ensure that the resin system will gel and cure satisfactorily.

As it is essential to guard against styrene loss (in polyester resins), resin drainages and low exotherm, especially in laminates less than 6 mm thick, an elevated temperature post cure shall be applied unless satisfactory cure at ambient temperature can be ensured.

Site fabrication shall only be carried out where construction in a factory is impracticable or uneconomic. Where site fabrication is employed, the special procedures to be adopted, particularly for resin formulation and method of cure, shall be stated and agreed between the purchaser and the manufacturer [see **4.2** k)] prior to commencement of fabrication.

## 27 Manufacturing procedure

### 27.1 General

Consistency in both materials and fabrication shall be ensured and appropriate supervision shall be provided at all stages of manufacture.

### 27.2 Lay-up

The requisite amounts of resin, catalyst or hardener and any other ingredient, such as accelerator or permitted filler, shall be accurately measured and thoroughly mixed. The amounts of mixed resin and reinforcement used in each layer of the laminate and the number and type of layers applied shall be recorded: the records shall be made available to the purchaser, or inspecting Authority, where applicable.

Whilst good rolling is essential, excessive rolling pressure shall be avoided so as not to disturb the distribution of the reinforcement or to break the fibre strands.

It shall be ensured that good adhesion is obtained between successive layers of the laminate and between the shell and added fittings either by appropriate scheduling of the manufacturing operation or by removing the surface of the cured resin, to expose the fibres.

In the case of laminates used as a chemical barrier, adjacent pieces of reinforcement shall be overlapped by not less than 25 mm. In all other laminates, adjacent pieces of laminate, including feather-edged reinforcement, shall be overlapped by not less than 50 mm. So far as is practicable, all joints shall be staggered through the thickness of the laminate and in no case shall joints coincide in adjacent layers. When rovings are chopped for spray applications, the length of individual strands shall not be less than 32 mm.

When directionally biased reinforcement is used, care shall be taken to ensure that the high strength fibres are adequately aligned in the correct direction to give the required strength.

Any spillage, drips or runs (which may flake loose) shall be suitably removed.

The limits for laminate defects shall be as given in Table 16.

Subject to agreement of the purchaser and the Inspecting Authority, defects outside the limits given in Table 16 shall be repaired, the method of repair shall be agreed between the purchaser, or the Inspecting Authority, where appropriate, and the manufacturer [see **4.2** l)].

All surfaces shall have a smooth contour and irregularities which may cause high local loads shall be repaired.

**Table 16 — Permissible limits for laminate defects**

| Defect | Inner (process) surface (thermoset linings only) | Outer (nonprocess) surface |
|---|---|---|
| Blisters | None | Maximum 6 mm diameter, 1.5 mm high |
| Chips | None | Maximum 6 mm, provided it does not penetrate the reinforcing laminate |
| Cracks | None | None |
| Crazing | None | Slight |
| Dry spots | None | Maximum 10 per $m^2$ with total not greater than 100 $mm^2$ in area |
| Entrapped air | None at surface. If in laminate, not greater than 1.5 mm diameter and not more than 2/100 $mm^2$ | 3 mm diameter maximum; no more than 3 % of area |
| Exposed glass | None | None |
| Exposed cut edges | None | None |
| Foreign matter | None | None if it affects the properties of the laminate |
| Pits | Maximum 3 mm diameter, 0.5 mm deep, number shall not exceed $1/10^4$ $mm^2$ | Maximum 3 mm diameter, 1.5 mm deep |
| Scores | Maximum 0.2 mm deep | Maximum 0.5 mm deep |
| Surface porosity | None | None |
| Wrinkles | Maximum deviation 20 % of wall thickness but not exceeding 3 mm | Maximum deviation 20 % of wall thickness but not exceeding 4.5 mm |
| Sharp discontinuity | Maximum 0.5 mm | Maximum 1 mm |

### 27.3 Thermoplastics liners

The details of the fabrication process shall be chosen to suit a selected thermoplastics liner.

There shall be adhesion between the thermoplastics liner and the laminate (see **7.1**). The bond shall be achieved by:

a) chemical etching and priming;

b) use of selected resins; or

c) use of glass and fabric-backed material.

In the case of thermoplastic liners, e.g. polypropylene, it is permissible for small-diameter nozzles (less than 75 mm) to use a liner tube which is not fabric backed and which cannot be effectively treated to promote adhesion.

The manufacturer shall state the method to be employed when the materials of construction are agreed [see **4.2** d)].

When PVC is a chosen lining material, all sheet shall be stress relieved in an oven at a temperature between 120 °C and 140 °C for a time appropriate to the material and its dimensions.

All sheet-forming operations requiring high deformation such as right-angled folds or small-radius bends shall be performed hot.

NOTE 1 The layout of the lining sheet should be arranged to avoid welds in corners. Longitudinal welds should be staggered as far as is practicable. All welds should be located to avoid areas of high local strain, e.g. nozzles and supports. Lap welds are not permitted: an exception may be made when FEP is the lining material, for example in the attachment of nozzles.

All welders engaged in the fabrication of thermoplastics liners shall have successfully carried out specified approval tests before fabrication commences, see clause **34**.

Before welding of the sheet material commences, the edges to be welded, together with the filler rod, shall be suitably cleaned. In the case of glass-backed materials the glass-backing shall be stripped back for a distance of 3 mm to 6 mm on either side of the welds to ensure that no glass filaments are included in the welded joints.

Normally sheet welding shall be done by the hot gas filler rod technique. Subject to agreement between the purchaser and the manufacturer, hot plate welding is permitted [see **4.2** m)].

For hot gas filler rod welding an inert gas or air can be used providing it is free from moisture, dirt and oil and in all cases, the grade of material of the filler rod shall be fully compatible with the grade of sheet being welded.

All welds shall be fully penetrating and free from notches and pin-holes and shall have at least 85 % of the strength of the parent material in the case of welds made from two sides and 70 % of the strength of the parent material when made from one side, when tested in accordance with Appendix B.

The weld stand out shall not exceed 1 mm on the side to be laminated.

NOTE 2 The first laminate layer applied to the weld area may be electrically conducting in order to assist inspection, see clause **35**.

## 28 Constructional details

### 28.1 Main seams

Main seam joints shall be formed by the addition of overlay layers and comply with one of the types specified in clause **19** and illustrated in Figure 18 and Figure 19.

The overlay laminate shall be applied only to fully prepared surfaces, e.g. abraded, of the main shell and shall be worked so as to ensure a bond between shell and overlay. The overlay shall be smoothly terminated at its edges so that an abrupt change of section is avoided. Tapers between laminates of differing thickness shall be not steeper than 1 in 6.

Misalignment between the sections to be joined shall be kept as small as possible. Where, owing to the size of the vessel or tank or the conditions of fabrication, misalignment exceeds 0.2 % of the diameter or one-half of the shell laminate thickness, whichever is the smaller, additional overlay layers shall be added so as to increase the overlay thickness by the amount of the misalignment.

Where an internal sealing strip is to be fitted, this shall consist of a minimum of 1 200 g/m$^2$ CSM, plus a surfacing veil or tissue, and shall be finished with a resin-rich gel coat.

NOTE A minimum width of 100 mm is recommended for the sealing strip.

Where a thermoplastics lining is to be used, the parts to be joined shall be firmly clamped in position and the weld in the thermoplastics lining shall be completed and tested before the overlay lay-up is added.

### 28.2 Flanged joints

**28.2.1** After the completion of cure, flange and pad faces shall be flat within tolerances, see clause **30**. If machining is carried out, the face of the flange shall be restored with its original chemically resistant finish.

**28.2.2** Backing flanges shall be a snug fit against the back face of the flange and adequately chamfered to provide clearance for the radius behind the flange.

### 28.3 Branches in vessels and tanks with thermoplastics linings

Where branches are to be fitted in vessels and tanks with thermoplastics linings, the external laminate on both shell and branch shall be cut back, without damaging the lining, for a sufficient distance to ensure adequate access for welding of the lining. The lining shall be welded with at least three runs applied externally and one internally, wherever practicable. Additional laminate, including the necessary compensation, shall then be added after the lining welds have been tested and approved so that good bonding to the original laminate is achieved over the whole area. All the additional reinforcement shall be on the outside of the vessel or tank.

### 28.4 Branches in vessels and tanks without thermoplastics linings

The required additional laminate shall be provided in accordance with **20.2.3**. Wherever additional laminate is provided, it shall be applied in such a way as to ensure that a good bond to the original laminate is achieved over the whole area.

### 28.5 Pads

In building up a pad connection, the resin shall be forced into all parts of the pad and the glass fibre reinforcement shall be uniformly distributed.

NOTE 1 Of particular importance is the presence of reinforcement in the annular spaces inside and outside the metal ring.

Special attention shall be given to:

a) ensuring a sound bond between the additional laminate of the pad and the main laminate; and

b) the provision of adequate laminate as an overlay to cover both the pad and the surrounding area of the shell.

All transition areas between the pad and the shell shall be blended with a generous radius or gradual taper.

NOTE 2 The configuration of the transition in a plane normal to the vessel axis requires careful attention particularly in the case of small-diameter vessels and tanks.

### 28.6 Steelwork

All steelwork whether or not totally encased in GRP shall be blast cleaned and protected, see **4.2** f) 9).

## 29 Curing

All items shall be cured in accordance with the resin supplier's instructions. Wherever possible this shall be done at the manufacturer's works. Where this is not possible due to size limitations or other reasons, the curing procedure shall be determined according to the design requirements and resin system used.

Vessels or tanks shall not show any evidence of incomplete cure, e.g. appearance, tackiness, resin pick-up by acetone or low hardness, see **37.2**.

NOTE The vessel or tank may be subsequently considered as complying with this standard if a satisfactory rectification procedure can be shown to be fully effective.

## 30 Tolerances

### 30.1 General

The general arrangement and overall dimensions shall be in accordance with the purchase order and approved drawings.

The thickness of any item shall be nowhere less than that specified for that item and local variations shall be not greater than 10 % of the average thickness or 3 mm, whichever is least.

### 30.2 Cylindrical shells

**30.2.1** *General.* Completed shell sections shall be circular within the limits defined in **30.2.3**. Measurements shall be made to the surface of the parent material and not to a fitting or other raised part.

Shell sections may be measured for out-of-roundness either when laid flat on their sides or when set up on end. When the shell sections are checked whilst lying on their side, each measurement for diameter shall be repeated after turning the shell through 90° about its longitudinal axis. The two measurements for each diameter shall be averaged and the amount of out-of-round calculated from the average values so determined. Any local departure from circularity shall be gradual.

**30.2.2** *Circumference.* Unless otherwise agreed between the purchaser and the manufacturer [see **4.2** n)], the external circumference of the completed shell shall not depart from the calculated circumference (based upon nominal inside diameter and the actual material thickness) by more than 5 mm for shells up to and including 600 mm outside diameter and 0.25 % of the calculated circumference for larger shells.

**30.2.3** *Out-of-roundness.* The difference between the maximum and minimum internal diameters measured at any one cross section shall not exceed the amount given in Table 17.

The profile measured on the inside or outside of the shell by means of a gauge of the designed form of the shell and having a length equal to one-quarter of the internal diameter of the shell shall not depart from the designed form by more than the amount given in Table 17. (This amount corresponds to $x$ in Figure 32.)

**30.2.4** *Straightness.* Unless otherwise shown on the drawing, the maximum deviation of the shell from a straight line shall not exceed 0.3 % of the cylindrical length.

**30.2.5** *Domed ends.* The tolerances and dimensions of the skirt of a dished end shall be in accordance with **30.1** to **30.2.3** inclusive as for cylindrical shells.

The depth of dishing, measured from the plane passing through the point where the straight flange joins the knuckle radius shall in no case be less than the theoretical depth, nor shall this depth be exceeded by more than 1.25 % of the diameter or 38 mm, whichever is least.

Variations of the profile shall not be abrupt but shall merge gradually into the specified shape. Any departure from form shall not be more than the amount given in Table 17. In torispherical ends, the knuckle radius shall not be less than specified in **18.2** c) and shall have common tangents with both the straight flange and the dished profile at each join.

**30.2.6** *Flanges.* All flanges shall comply with the following.

a) *Flatness.* Flange faces shall not be concave in the direction of the axis. Flange faces and flange back faces shall have a flatness tolerance, see BS 308-3, as follows:

1) up to and including 450 mm nominal diameter: 1 mm;

2) over 450 mm and up to and including 1 000 mm nominal diameter: 1.5 mm;

3) above 1 000 mm nominal diameter: 3 mm.

b) *Squareness.* Flanges shall be square to the axis of the branch to within 1° up to 100 mm diameter and to within 0.5° above 100 mm diameter.

**30.2.7** *Flatness of base.* The base of a vessel or tank shall be flat to the tolerance agreed between the purchaser and the manufacturer [see also **4.2** f) 1)].

**Table 17 — Tolerance for shells**

| Nominal internal diameter of shell | Difference between maximum and minimum diameters | Maximum departure from designed form |
|---|---|---|
| mm | | mm |
| Up to and including 250 | 1.0 % of shell diameter | 1 |
| Over 250 up to and including 500 | | 2 |
| Over 500 up to and including 900 | | 3 |
| Over 900 up to and including 1 400 | | 4 |
| Over 1 400 up to and including 1 800 | | 5 |
| Over 1 800 up to and including 2 200 | 18 mm | 6 |
| Over 2 200 up to and including 2 500 | | 7 |
| Over 2 500 | 18 mm or 0.4 % of shell diameter, whichever is greater | 8 mm or 0.2 % of shell diameter, whichever is greater |

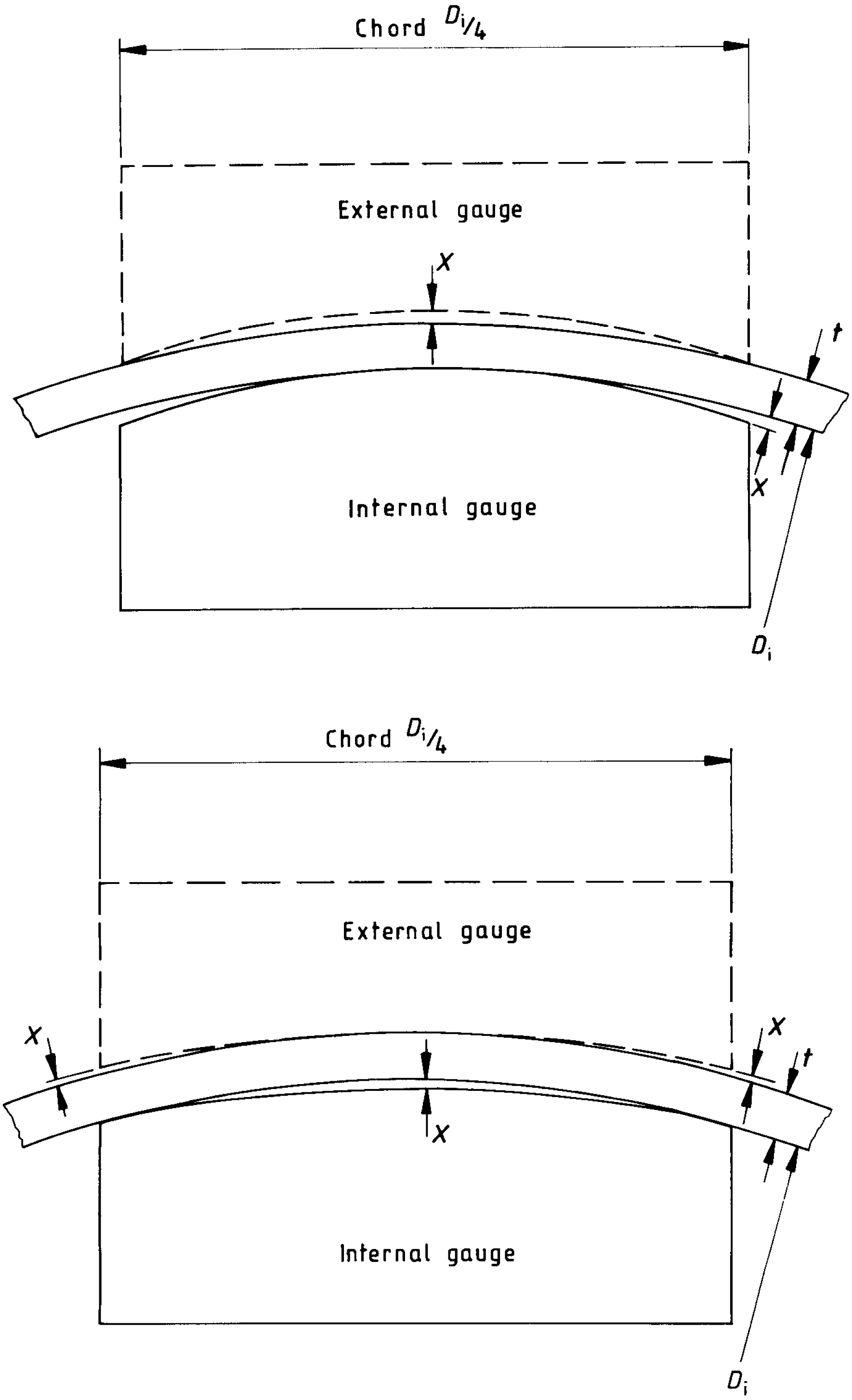



**Figure 32 — Tolerances for cylindrical shells**

# Section 5. Inspection and tests

## 31 Inspection and test requirements

### 31.1 General

All vessels and tanks shall be subject to the inspections and tests specified in this section and section 1.

The authority responsible for the inspection shall be determined in accordance with **31.2** and **31.3**.

### 31.2 Category I inspection

All vessels and tanks which, by reason of volume, pressure, temperature, hazardous nature of content, or any other factor, are designated as category I against any relevant heading in Table 2 shall be manufactured under the survey of a competent engineering Inspecting Authority or Organization in addition to the manufacturer's own inspection.

This requirement shall be satisfied where inspection is carried out by a recognized independent authority or competent personnel of a separate engineering inspection department maintained by the purchaser of the vessel. An inspection department maintained by the manufacturer does not satisfy this requirement except when:

a) specific responsibilities are delegated at the discretion of the Inspecting Authority or Organization; or

b) vessels are for the manufacturer's own use and not for resale.

Inspection by the Inspecting Authority shall not absolve the manufacturer from his responsibility to exercise such quality assurance procedures as will ensure that the requirements and intent of this standard are satisfied.

### 31.3 Categories II and III inspection

For those vessels and tanks which, by reason of their small volume, low pressure temperature or hazard associated with content and any other factors or any combination thereof, are designated as category II or category III against all relevant headings in Table 2, independent inspection shall not be required.

### 31.4 Facilities

The manufacturer shall furnish and prepare the necessary test pieces and make the tests specified either at his own works or elsewhere.

When required by the order or drawing, test pieces shall be made available for test in the purchaser's laboratories [see **4.1** d)]. Unless otherwise specified, all tests shall be witnessed by the Inspecting Authority and due notice shall be given by the manufacturer to permit this [see **4.2** o)].

## 32 Certificate of inspection and testing

In cases of joint responsibility for the inspection and testing of a vessel or tank, signed documentary evidence of the results of all the previously completed inspections and tests shall be forwarded to the Inspecting Authority responsible for witnessing the final tests, prior to the conducting of the final tests.

Upon satisfactory completion of the vessel or tank, the organizations responsible for design, construction and inspection shall furnish duplicate copies of a certificate to the purchaser, stating that the design, construction and testing complies with this standard. Where applicable, the actual test results obtained shall be stated on or with the certificate.

## 33 Principal stages of inspection

Unless otherwise required by the order [see **4.1** e)], inspection shall include the following stages, as appropriate.

a) Inspection of workshop conditions where manufacture will be carried out.

b) Inspection of works records relating to the control and issue of materials, resin mixing, etc., for each vessel or tank constructed.

c) Identification of the materials of construction and their storage condition.

d) Approval of laminating procedures and tests for operators, and check of validating documents, see clause **36**.

e) Approval of weld procedures and tests for welders of thermoplastics linings, and check of validating documentation, see clause **34**.

f) Witnessing of spark tests on welds in thermoplastics linings.

g) Examination during hand lay-up, spray application, winding and jointing of resin-glass laminates.

h) Examination of any repairs carried out during construction.

i) Verification of cure of laminate.

j) Examinations on completion of construction, during pressure testing, and after boiling out tests before any pigmented coating is applied. A pigmented coating shall not be applied to the vessel until after final inspection unless prior exemption is given by the purchaser [see **4.1** f)].

## 34 Welding procedure and welder approval tests for thermoplastics linings

The test piece shall incorporate a 300 mm long butt weld made by joining two pieces of the material to be used for a lining, each 300 mm long and 125 mm wide. The weld shall be made in the same way as the production welds and shall include at least one stop and start in each run. Weld procedure for the test welds shall comply with **27.3**.

After completion, the test weld shall be examined visually and by the use of a high-frequency spark tester at a voltage from 20 kV to 25 kV. Any weld showing evidence of notches, lack of fusion or pinholes shall be deemed not to comply with this standard.

Test pieces shall then be machined from the welded sample and subjected to the tensile test described in **B.6**. The tensile strength across the weld shall be not less than 85 % or 70 % of the tensile strength of unwelded sheet dependent upon whether the weld is made on two sides or on one side, respectively.

## 35 Production weld tests for thermoplastics linings

All production welds shall be examined visually and by high-frequency spark test equipment (20 kV to 25 kV) at the following stages:

a) after the first weld run has been completed;

b) after completion of the external weld;

c) after pressure or static head tests; and

d) after any boil out test.

Tests a) and b) shall be completed before any primer or reinforcement is applied to the weld area. A temporary earthing strip shall be provided behind the welds. This strip shall be removed before any reinforcement is applied to the weld areas. At this stage the visual inspection shall be made. There shall be no obvious undercutting, degradation of the thermoplastics or breaks in the weld run. When such defects exist, an appropriate length of weld shall be removed, the surfaces prepared again and a new weld completed.

NOTE The first laminate layer applied to the weld area may be conducting in order to assist inspection at stage c) and in service.

The repaired welds shall be subject to the full inspection procedure.

## 36 Laminating procedures and approval of operators

### 36.1 General

At the completion of the fabrication of a tank or vessel it is not possible by non destructive methods to determine the details of the laminates and therefore procedures shall be established to cover the features described in **36.2**. These procedures, including the resin and reinforcement, shall be recorded, together with the test work and case histories which demonstrate that the procedures are satisfactory.

NOTE The satisfactory documentation will normally be sufficient evidence to validate those procedures but a purchaser may wish to have demonstration pieces prepared and tested. The provision of such test pieces should be specified under **4.1** d).

In addition to establishing laminating procedures it is necessary for fabricators to demonstrate that their operators have the basic skills of good laminating and that they are familiar with various procedures specified in **36.2**. Records shall be kept of the training received by operators and the type of work on which they have been employed. In the absence of satisfactory records of experience, the manufacturer shall arrange operator approval tests in accordance with **36.3**.

### 36.2 Laminating procedures

The laminating procedures requiring approval shall include the following.

a) *General.* Laminates of a basically similar construction to that proposed for the barrel and ends of the vessel to be fabricated shall be tested in accordance with **37.2**.

b) *Flange/nozzle construction.* Samples of the manufacturer's standard forms of such details shall be laminated and their construction demonstrated as satisfactory by visual inspection of the finished sample, followed by inspection of cross sections taken through the flange/nozzle on two planes at right angles to each other.

For each type of construction, there shall be at least one test piece each to represent details up to 50 mm nominal bore and of greater than 50 mm nominal bore.

c) *Nozzle/shell attachment.* Samples of the manufacturer's standard forms of nozzle to shell attachments shall be laminated and their construction demonstrated as satisfactory by visual inspection of the finished sample, followed by inspection of cross sections taken through the nozzle/shell on two planes at right angles to each other.

For each type of construction, there shall be at least one test piece each to represent details up to 50 mm nominal bore and of greater than 50 mm nominal bore.

d) *Special forms of construction.* Any special form of construction shall be demonstrated as giving satisfactory results by visual inspection of the finished sample and by examination of sections cut through the critical areas of the construction.

e) *Butt jointing of vessel main seams.* These shall be demonstrated as satisfactory by visual inspection of a finished mock up and of three sections cut through the sample on planes normal to that of the joints.

f) *Tank bottom corner construction.* Where fabric-backed thermoplastics lined tanks are constructed without a knuckle radius, a mock up of the corner laminate construction shall be sectioned in at least three places to demonstrate that lamination is in accordance with **18.5.2**.

### 36.3 Approval testing of operators

During training every operator shall laminate a test sample for each type of construction specified in **36.2** on which he is to be employed, and each sample shall be visually examined on completion and at cross sections cut in accordance with **36.2**.

The manufacturer shall maintain fully documented records of each operator approval test and the results thereof.

If an operator has not carried out laminating of a particular detail for a period of 3 months or if there is any reason to doubt his ability to laminate satisfactorily such a detail, he shall be required to demonstrate his skill by preparing appropriate test pieces as specified in **36.2**.

### 36.4 Acceptance criteria

Every test sample shall show satisfactory surface finish, full impregnation and consolidation and proper distribution of the specified mass and type of reinforcement.

All compensation and overlays shall be in full contact with and satisfactorily bonded to the preceding layers with specified overlap.

All reinforcement shall be continuous round corners and into the mating section, i.e. flange to nozzle or shell to nozzle (see Figure 24).

## 37 Production samples for mechanical tests on laminates

### 37.1 General

Samples from waste areas such as cut-outs for branch openings shall be made available for inspection and appropriate testing, provided that they are typical of the laminate they represent.

NOTE In the majority of cases such samples will not be flat and therefore there may be difficulty in carrying out all the mechanical tests which could normally be done on a flat sample. The properties of laminates are determined to a considerable extent by the type of glass reinforcement, the type of resin and the glass resin ratio and what is required from production samples is a clear indication that the properties of the laminate produced match the properties used in design. This information may be obtained by measuring the thickness of the laminate, burning off the resin to obtain the glass content, and examining the glass which remains to determine the type of glass and the number of layers. If the lap shear strength of the laminate is also determined, there will be enough information to provide an adequate guide to the mechanical properties of the laminate.

As an alternative to using cut-outs for test samples, specimen laminates shall be prepared by the operators working on the production laminate. These specimens shall be laid up at the same time with the same materials and in the same manner as the item they represent and cured under the same conditions as the main laminate.

The provision of special test laminates and the extent of the mechanical testing to be carried out either on cut outs or prepared laminates, shall be agreed between the purchaser and the manufacturer [see **4.2** p)].

**37.2 Testing**

The following tests shall be carried out on all samples to verify laminate construction and determine typical properties as given in Table 4:

a) visual examination of a prepared cross section of the cut-out or prepared samples;

b) thickness of the laminate;

c) lap shear strength of laminate (see **B.9**);

d) shear and peel strength where a thermoplastic lining is used (see **B.10** and **B.11**);

e) glass content (BS 2782:Method 1002);

f) number of layers of glass and the type;

g) acetone test (see **B.18**);

h) Barcol hardness of the inner and outer surfaces (see **B.14**); the minimum requirement shall be at least 80 % of the hardness value published by the resin manufacturer for the particular system.

When polyester resins are used for the fabrication of category I and category II vessels or tanks, and, if following the Barcol test there is any doubt about the state of cure of the laminate, a residual styrene test shall be carried out (see **B.17**). The residual styrene content of the laminate shall not exceed 2 % by mass of the resin content of the sample.

The following tests shall be carried out when it is required to verify the mechanical properties of the laminate:

1) ultimate tensile unit strength (see **B.6**);

2) unit modulus (see **B.7**);

3) ultimate compressive unit strength (see **B.8**).

NOTE Tests on laminates of mixed types of reinforcement may not achieve the full unit load capacity of $u \times K$ due to the different strain to failure values of the individual layers of reinforcement (see **B.6**) where $K$ is the overall design factor and $u$ is the maximum allowable unit load (in N/mm).

## 38 Prototype testing

**38.1 General**

Where the manufacturer shall demonstrate his ability to design and/or produce a satisfactory vessel or tank, and acceptable documentary evidence of past experience is not available, a prototype shall be made and tested to determine its integrity.

Unless otherwise agreed, the prototype tests shall be witnessed by the purchaser or the Inspecting Authority, where appropriate, [see **4.2** q)].

**38.2 Manufacture of prototype vessel**

The prototype shall consist of one of the following:

a) a complete vessel identical in design and manufacture to the vessel or vessels proposed;

b) a specially made test vessel designed to withstand repeated test procedures, into which any novel details can be incorporated for test purposes; or

c) the first of a series of identical production vessels, tested only by the procedure detailed in **38.3** a) using strain gauges.

When a manufacturer produces vessels of varying design and size the sample vessel or vessels shall incorporate details of construction representative of the range, including in particular:

1) materials and laminates;

2) methods of construction;

3) dimensions;

4) liners.

### 38.3 Tests to be applied to prototype vessel

The nature of the tests to be performed shall be by agreement between the purchaser, or the Inspecting Authority, where applicable, and the manufacturer [see **4.2** r)]. The tests shall demonstrate resistance to specified modes of failure, and may include one or more of the following:

a) determination of general and local strains, by measurement, strain gauges or other suitable means, when the vessel is hydraulically pressurized to the design pressure or agreed higher pressure;

b) determination of the fatigue strength of the vessel or detail, by cyclic variations of pressure or temperature or both between agreed limits;

c) determination of the factor of safety to failure and the mode of failure, by hydraulically pressurizing the vessel until failure occurs (not applicable to vessels subject only to hydrostatic head of contents).

The tests shall be carried out at the design temperature and the behaviour of the vessel in respect of all modes of failure shall be observed and recorded.

### 38.4 Performance during prototype testing

The sample vessel shall withstand not less than the following.

a) In the strain determination test [see **38.3** a)]: 1.3 times the design pressure without failure in any form.

b) In the fatigue test [see **38.3** b)]: 10 times the estimated number of pressure cycles in the life of the vessel.

c) In the test to failure [see **38.3** c)]: 6 times the design pressure without bursting.

### 38.5 Records of tests

Records of all prototype tests shall be retained by the manufacturer and be made available to the purchaser, or Inspecting Authority, where appropriate, as required.

## 39 Testing after completion of fabrication

### 39.1 General

Vessels and tanks shall be subjected to the appropriate hydraulic pressure or static head test at ambient temperature in accordance with **39.2** to **39.8**.

The test shall be applied either:

a) at the manufacturer's works and repeated after installation; or

b) after installation

as specified by the purchaser [see **4.1** e) and g)].

NOTE 1 To avoid the risk of freezing, it is recommended that the temperature of the water during the test should not be less than 7 °C.

NOTE 2 In the case of all such testing it is important that the specified test pressure is not exceeded since over-pressurization may lead to laminate damage which would be cause for rejection of the vessel.

All care should also be taken to ensure that the vessel or tank, its supports and foundations are adequate to withstand the total loads that will be imposed upon them during this test.

### 39.2 Static head tests

Tanks subject only to hydrostatic head of content shall be tested in their normal working attitude by filling with water to the top of the tank.

NOTE When the operating contents of the tank have a relative density greater than 1.0, consideration should be given to the adoption of an increased static head to simulate the actual operating head at the tank base. This will however increase the loadings on the base and the upper parts of the tank which should be designed to ensure that at test no part of the tank is strained beyond 0.26 % or 1.3 $\epsilon_d$, whichever is the smaller, where $\epsilon_d$ is the least strain, determined from allowable loadings and resin properties. For flat-bottomed vessels, the holding down arrangement for the test should be checked in accordance with Appendix F and the value of $p$ should be not less than the static pressure at the top of the shell cyclinder divided by 1.3.

### 39.3 Hydraulic pressure test in the working attitude

Vessels subjected to internal pressure shall be hydraulically tested to not less than 1.3 times the design pressure, due allowances being made for static head of content and of test medium.

At test no part of the vessel shall be strained beyond 0.26 % or 1.3 $\epsilon_d$, whichever is smaller.

### 39.4 Hydraulic pressure test in other than the working attitude

Vertical vessels may sometimes have to be pressure tested in a horizontal position and in such cases due consideration shall be given to the method of supporting the vessel during test and to the resulting static and pressure loadings imposed on the various parts of the vessel or tank.

At test no part of the vessel shall be strained beyond 0.26 % or 1.3 $\epsilon_d$, whichever is smaller.

### 39.5 Vacuum test

Vessels to operate under vacuum conditions shall be tested under vacuum or applied external pressure to simulate vacuum conditions. In either case the resulting external pressure on the vessel shall be 1.3 times the design external pressure where this is less than full vacuum.

When the vessel is designed for full vacuum the internal test condition shall be between 0 kPa and 4 kPa (0 mmHg and 30 mmHg) absolute.

Where a test under vacuum or applied external pressure is not reasonably practicable, vessels shall be given an internal pressure test of 1.5 times the design external pressure subject to the strain under such test not exceeding 0.26 % or 1.3 $\epsilon_d$, whichever is the smaller.

NOTE There are often variations at different levels of a vessel or tank between the design pressure appropriate to the process conditions and the equivalent design pressure appropriate to test conditions because differences in density between content and test medium has resulted in a compensatory increase of "top" pressure at test in order to achieve the required minimum test pressure at vessel/tank base.

In such cases where $\epsilon_d$ has been dictated by a load limitation (see **9.2.6**) and the value of the design factor $K$ unfavourably influenced by environmental temperature and/or cyclic conditions (factors $k_2$, $k_3$ and $k_4$) a temporary reassessment of the design factor $K$ may be permitted to arrive at the permissible limiting strain while under static/hydraulic test.

For initial test at ambient temperature prior to entry into service, lower appropriate values may be adopted for factors $k_2$, $k_3$ and/or $k_4$ but if a similar test pressure is to be applied periodically while in service, then the original values of $k_2$ and $k_4$ have to be maintained and the scope for adjustment is limited to $k_3$, i.e. between design. and test temperatures.

So far as the basic laminate design is concerned, the value of $\epsilon_d$ to be used is that arrived at strictly in accordance with **9.2** based on the process design conditions.

### 39.6 Pressure test and vacuum test procedure

The test pressure shall be gradually increased to a value of 50 % of the specified final value, thereafter the pressure shall be increased in stages of approximately 10 % of the specified test pressure until this is reached. At no stage shall the vessel be approached for close inspection until the pressure has been positively reduced to 90 % of the pressure previously attained and which has been held for not less than 15 min.

The required maximum test pressure shall be maintained for not less than 1 h.

Any indications of leakage or excessive strain shall be cause for failing the test. Vessels or tanks shall not be hammered.

NOTE Care should be taken to ensure that the specified test pressure is not exceeded during hydraulic testing, by the fitting of an appropriate pressure-relief device.

### 39.7 Pneumatic testing

In exceptional circumstances when the possibility of hydraulic testing is excluded and pneumatic testing is necessary, reference shall be made to the following precautions:

a) the adequacy of blast protection;

b) the extent of area cleared for test safety purposes;

c) the degree of confidence in stress analysis of vessel details;

d) the extent of remote monitoring provided during test.

### 39.8 Examination after testing

On completion of pressure or vacuum tests the vessel or tank shall be inspected internally and externally. Any indication of cracking, resin crazing or excessive strain shall be cause for failing the test. Vessels and tanks with thermoplastics linings shall again be spark tested after completion of testing; any evidence of weld defects shall be cause for failing the test.

On completion of any necessary repairs the tests shall be repeated.

## 40 Marking

Each vessel and tank constructed in accordance with this standard shall be marked with the following information.

a) The number and date of this standard, i.e. BS 4994:1987[1)].

b) The manufacturer's name.

c) The manufacturer's serial number.

d) The design pressure (in bar gauge) and/or maximum head and relative density of contents.

e) The maximum test pressure (in bar gauge).

f) The maximum working temperature coincident with the safe working pressure (in °C).

g) The year in which vessel or tank was completed.

h) The resin reference(s).

i) When vessels or tanks have a thermoplastic lining, the type of thermoplastic lining shall be marked, e.g. uPVC lined.

j) The category of inspection undergone by the vessel or tank (see **31.1**).

k) The certifying mark of the Inspecting Authority, when applicable.

These markings shall be legibly stencilled with letters and figures at least 12 mm high on a conspicuous part of the exterior of the vessel or tank, or stamped or engraved on a plate which is subsequently bonded to the vessel or tank.

Engraving or stamping on the vessel or tank itself or on a plate already bonded to it shall not be used.



[1)] Marking BS 4994:1987 on or in relation to a product is a claim by the manufacturer that the product has been manufactured to the requirements of the standard. The accuracy of such a claim is therefore solely the manufacturer's responsibility. Enquiries as to the availability of third party certification should be addressed to the appropriate certification body.

# Section 6. Erection

## 41 Preparation for shipment

The vessel shall be cleaned inside to be free of all traces of release agents, fibres, dust or other foreign matter.

All branches and other openings shall be blanked either with service blanks or with temporary covers suitable for protecting the flange faces and preventing the ingress of foreign matter for the conditions likely to be met during transport and storage of the vessel. Care shall be taken, however, to ensure that the vessel is vented at all times.

To prevent overstressing of vessels during handling and transportation, temporary stiffening/supporting devices such as spiders or saddles shall be provided when necessary. If the vessel or vessel section has a transport mass in excess of 200 kg, a drawing shall be provided showing lifting positions, handling techniques and equipment required together with any other information necessary for safe handling of the vessel.

If lifting positions have special restrictions on angle of lift, minimum number of positions used, etc., details shall be recorded on the drawing and a permanent notice affixed to the vessel referring to these restrictions.

## 42 Handling

When lifting lugs are not provided, the vessel shall only be lifted using webbing or rope slings passed round the girth of the vessel. Chair or wire ropes are permitted if adequate provision is made to protect the vessel from local loads and abrasion. When lifting, care shall be taken not to apply shock loads to the lugs or vessel. Nozzles shall not be used as lifting positions unless a design check has been made.

NOTE A design procedure for lifting lugs or trunnions is given in Appendix J.

## 43 Transportation

The loads imposed by transportation shall be considered and temporary supports used where necessary (see clause **41**).

The vessel shall be adequately secured with slings or other fixings to prevent movement. Slings shall comply with clause **42**.

Loose parts shall not be placed inside the vessel unless they are secured to prevent movement and suitably protected to prevent damage.

## 44 Temporary storage

The vessel shall be stored on a level surface free from sharp protrusions and adequately supported to prevent local damage.

The storage location shall be selected to minimize the risk of accidental impact damage from forklift trucks, cranes, etc.

Possible movement due to wind shall be considered and temporary anchors provided, when necessary.

## 45 Installation

All vessels shall be placed upon their supports in such a way as to ensure that loads are distributed as intended in design, e.g. flat-bottomed vessels shall be provided with continuous support as specified in **22.2.1**.

Care shall be taken to ensure that all bolts are an easy fit in the mating holes of the base and stand so that no loads due to misaligned holes are imparted to the laminate. Drifts or bars shall not be used.

## 46 Support of associated pipework

All pipework shall be supported so that total loads local to the branches on the vessel or tank do not exceed the design values.

## Appendix A General information on the materials used for reinforced plastics construction

### A.1 General

Plastics, reinforced with glass or other materials, are finding increasing use in plant items. Some of these have a resin surface, others are lined with suitable thermoplastics, usually unplasticized polyvinyl chloride (uPVC) or polypropylene. Reinforced plastics have properties quite different from those of metallic materials of construction and it is important that designers, fabricators and users thoroughly understand the nature of the materials.

Reinforced plastics are not isotropic and may show very different strength properties in different directions. This characteristic is due to the method of constructing the laminate by blending layers of resin and reinforcement. In view of the wide range of combinations of matrix material (resin) and reinforcement it is not convenient to use conventional mechanical properties for design purposes, and layer properties (such as unit strength and unit modulus) have been developed.

The reinforcing materials used all have much higher elastic moduli than the matrix materials and are, therefore, the main load-carrying elements. Loads applied to a laminate are transferred from fibre to fibre through the matrix and the strongest possible bond between the constituents is thus required. Research into the nature of the bond and into its improvement is continuing. Because of the difference between the modulus of the matrix and that of the reinforcement, damage by debonding, that is local failure of the bond between the two, can occur at small values of overall strain and is often the first sign of irreversible damage to the laminate. Higher applied loads lead to breakage of fibres, as well as further debonding, and ultimately to complete laminate failure.

Because of the range of materials and manufacturing techniques available, the success of the finished item is dependent to a considerable degree on the quality of workmanship. It is therefore essential to ensure adequate supervision during fabrication, good control over raw materials and clean, consistent working conditions.

### A.2 Thermoplastics

Thermoplastics are polymeric materials which soften on heating and harden on cooling, reversibly, without undergoing a chemical change. They can therefore be moulded by the application of heat and then cooled to retain their moulded shape.

For linings of PVC the unplasticized grade (called uPVC) is used in sheet form, being shaped and welded as required. When using PVC as a vessel lining, the reinforcement is "chemically" bonded to the lining using specially formulated surface primers. For some primers, degreasing of the PVC is all that is required to effect an adequate bond; primers in general use, however, require the PVC surface to be roughened by abrasion and then degreased to ensure that a good bond is obtained. The resin supplier normally issues instructions for use with a particular resin system.

Polypropylene sheet to be used as a lining is normally supplied with a layer of open-weave glass fibre cloth containing a small proportion of polypropylene fibres fused to one surface. Care should be taken to ensure that a grade of polypropylene suitable for the working conditions is specified.

### A.3 Thermosetting plastics

Generally known as thermosets, thermosetting plastics, also usually polymeric, undergo an irreversible chemical change during manufacture. Moderate heating does not cause softening, as with thermoplastics, but excessive temperature will cause degradation.

The commonest thermoset used for reinforced plastics equipment is an unsaturated polyester resin, because it can be cold cured without the evolution of volatile by-products. The choice of resin, however, depends on the duty and a large selection is available. It is recommended that the resin supplier should be consulted when determining the best system for a given application.

Polyester resins complying with BS 3532 usually have a three-part formulation, comprising resin, catalyst and accelerator. The method of mixing and the relative amounts of the components are important.

Although of rather low strength and comparatively brittle, suitably reinforced resins of this type produce strong and relatively light laminates, which are useful for vessels and tanks to be used on a great variety of duties.

Other thermosetting resins, such as epoxy and vinylester, have also been successfully used in the fabrication of items for chemical and other plants. Different curing systems are used and the recommendations of the resin supplier should be closely followed.

**A.4 Glass fibre reinforcement**

Glass fibres are the most commonly used reinforcing materials for increasing the strength of thermosetting resins.

E type glass, a low-alkali borosilicate glass is usually specified for tank and vessel construction but the glass manufacturer's advice should be sought.

Although the most widely used glass fibre reinforcements are E type (borosilicate) glass, A type glass fibres (soda-lime glass) may be used. C type glass (another borosilicate glass of lower alkali content) has very good chemical and water resistance but is at present available only in the form of a light mat for the surface reinforcement of gelcoats.

For use as reinforcement, the glass fibres are coated with a surface-active agent, designed to ensure adequate adhesion to appropriate resin systems. When ordering glass fibres, the resin type to be used should be specified so that compatible surface agents may be applied. Glass fibres are available in several forms.

a) *Chopped strand mat (CSM).* This is the most frequently used material and consists of chopped glass fibres (usually between 25 mm and 50 mm long) which are bonded together in random orientation by a resinous binder to form sheet. The thickness of the sheet is expressed as mass per unit area and is normally supplied in the range 300 g/m$^2$ to 900 g/m$^2$ in increments of 150 g/m$^2$ (see BS 3496). The properties of laminates made using CSM depend on the proportion of glass present and this proportion should be carefully controlled during fabrication.

Generally, all CSM laminates have good interlaminar adhesion, reasonable impact resistance and relatively little anisotropy.

b) *Sprayed cut roving (spray).* This is where a continuous roving is chopped and sprayed together with a resin directly on to the mould by means of a special gun. The resultant random orientation of fibre, laminate properties, etc. are similar to CSM but unless the application is controlled by machine, the distribution of the reinforcement may not be so uniform as with a mat. This is reflected in the values for factor $k_1$ given in Table 6.

c) *Continuous roving.* This consists of a number of long parallel strands, bundled but not twisted together (see BS 3691). Laminates incorporating rovings exhibit marked anisotropy.

d) *Woven fabrics.* These are available in a variety of weaves and are made from rovings or glass yarns. (Further details of these materials can be found in BS 3396 and BS 3749.) Laminates incorporating these materials usually have higher glass contents than those with CSM and provide higher strengths but may have poorer interlaminar adhesion. Directional properties are usually present to some degree.

**A.5 Other reinforcements**

Cloths and tissues of organic synthetic fibres are commonly used for the reinforcement of chemically resistant gelcoats.

Asbestos[2)] and carbon fibres are available but are used for special applications, the latter where weight and stiffness are at a premium.

More massive reinforcements such as strips or sections of steel or other metals are occasionally used to provide additional local load-carrying capacity at heavily loaded locations such as in flanges, around branches or cut outs and at points of support or attachment of fittings.

[2)] Attention is drawn to the legislation covering the use of asbestos.

# Appendix B Methods of test

## B.1 General

### B.1.1 *Tests*

This appendix describes methods for the testing of resins, laminates and thermoplastics for vessels and tanks in reinforced plastics. Tests are described for the determination of the following properties.

- **B.2** Glass content
- **B.3** Tensile strength of thermoplastics sheet and welds
- **B.4** Heat distortion temperature of unreinforced resin
- **B.5** Extension to failure of unreinforced resin
- **B.6** Ultimate tensile unit strength of laminate
- **B.7** Unit modulus of laminate
- **B.8** Ultimate compressive unit strength of laminate
- **B.9** Lap shear strength of laminate
- **B.10** Shear strength of bond between thermoplastics lining and laminate
- **B.11** Peel strength of bond between thermoplastics lining and laminate
- **B.12** Peel strength of bond between laminate layers
- **B.13** Shear strength of sandwich panel
- **B.14** Barcol hardness
- **B.15** Water absorption
- **B.16** Electrical properties
- **B.17** Residual styrene (polyester resins)
- **B.18** Acetone test (polyester resins)

### B.1.2 *Accuracy of testing equipment*

Testing machines shall be calibrated in accordance with BS 1610 and shall be maintained to grade A.

Extensometers, including ancillary or autographic equipment, shall be calibrated in accordance with BS 3846 and shall satisfy at least grade E requirements.

## B.2 Glass content

The form and number of test specimens and the test procedure shall be as described in BS 2782:Method 1002.

## B.3 Tensile strength of thermoplastics sheet and welds

The form and number of test specimens and the test procedure shall be as described in BS 2782:Method 320C. The welded specimens shall be dressed as per production welds. Welds shall be in the middle of the test specimen.

The tensile strength of the specimen weld shall be calculated from the maximum load and the original area of cross section and shall be expressed in terms of percentage of the tensile strength of the unwelded sheet material.

## B.4 Heat distortion temperature of unreinforced resin

The form and number of test specimens shall be in accordance with BS 3532, and the apparatus, procedure and report shall be as described for casting and laminating systems in Method 121A of BS 2782:Methods 121A to 121C:1976.

The temperature of deflection under load at 1.81 N/mm$^2$ shall be reported as the heat distortion temperature for the resin system.

## B.5 Extension to failure of unreinforced resin

The specimen shall be of the form and dimensions described in BS 2782:Method 320C. The test procedure shall be as described in BS 2782-3, with the rate of grip separation 5 mm/min.

## B.6 Ultimate tensile unit strength of laminate

**B.6.1** The form and number of test specimens and the test procedure shall be as described in BS 2782:Method 1003, type I, and the speed of separation shall be 2 mm/min.

**B.6.2** If comparison with the values given in Table 5 is required, then each type of reinforcement shall be tested individually and the resulting loads per unit width divided by the number of kg/m² of glass in the specimen:

$$u = \frac{F}{bm}$$

where

$u$ is the ultimate tensile unit strength (in N/mm per kg/m² glass);
$F$ is the maximum tensile force (in N);
$b$ is the mean initial width of test specimen gauge length (in mm);
$m$ is the mass of the single type of reinforcement (in kg/m²) of glass in the specimen.

**B.6.3** Where only combined laminates are available (e.g. when testing cut-outs from a vessel) the specimen may not achieve full tensile unit strength.

The minimum properties of Table 5 can then be assessed by determination of the minimum tensile load that the test specimen shall carry without failure.

NOTE The different strain capabilities of each type of layer will cause failure in one type before full load is achieved in other types of reinforcement. The resulting load transfer will cause premature failure of the specimen.

For each type of layer the strain $\epsilon_z$ of a layer of type $z$ at minimum ultimate unit strength shall be determined from equation (70):

$$\epsilon_z = \frac{u}{X_z} \qquad (70)$$

where

$X_z$ is the unit modulus of layer of type $z$ (in N/mm per kg/m² glass).

The lowest strain $\epsilon_{min}$ shall be used to determine the tensile unit strengths of the other types of reinforcement at that strain from equation (71):

$$\text{tensile unit strength} = X_z \, \epsilon_{min} \qquad (71)$$

The resulting tensile unit strength of each layer shall then be multiplied by the total mass of each type of reinforcement and the width of the specimen to obtain the minimum required load.

## B.7 Unit modulus of laminate

The form and number of test specimens and the test procedure shall be as described in BS 2782:Method 1003, type I, and the speed of separation shall be 2 mm/min.

Apply a small initial tensioning force (not exceeding 10 % of the expected force at 0.2 % strain) to straighten the test specimen and at this force set the indicating device to zero.

Increase the force steadily until an extension, $Z_x$, of 0.1 mm is reached on the specimen gauge length of 50 mm (0.2 % strain) and note the value of the force, $F_x$, at this point.

Where it is expected that 0.2 % strain may give rise to danger of fracture of the test piece, it is permissible to carry out the test at 0.1 % strain (corresponding to an apparent extension of 0.05 mm over the 50 mm gauge length).

Reduce the initial force to straighten the specimen correspondingly.

The unit modulus of the laminate, $X_{LAM}$ (in N/mm), shall be determined from equation (72):

$$X_{LAM} = \frac{(F_2 - F_1)}{(Z_2 - Z_1)} \times \frac{L}{b} \text{ N/mm} \qquad (72)$$

where

$(Z_2 - Z_1)$ is the change in apparent extension (in mm) corresponding to a change in applied force $(F_2 - F_1)$ (in N) (see Figure 33);
$L$ is the gauge length of the specimen (50 mm);
$b$ is the mean width of the test specimen over the gauge length (in mm).

For a laminate consisting of a single type of reinforcement, the value of $X_z$ (in N/mm per kg/m$^2$ glass) is given by:

$$X_z = \frac{X_{LAM}}{m} \qquad (73)$$

where

$m$ is the mass of the single type of reinforcement of glass in the specimen (in kg/m$^2$).

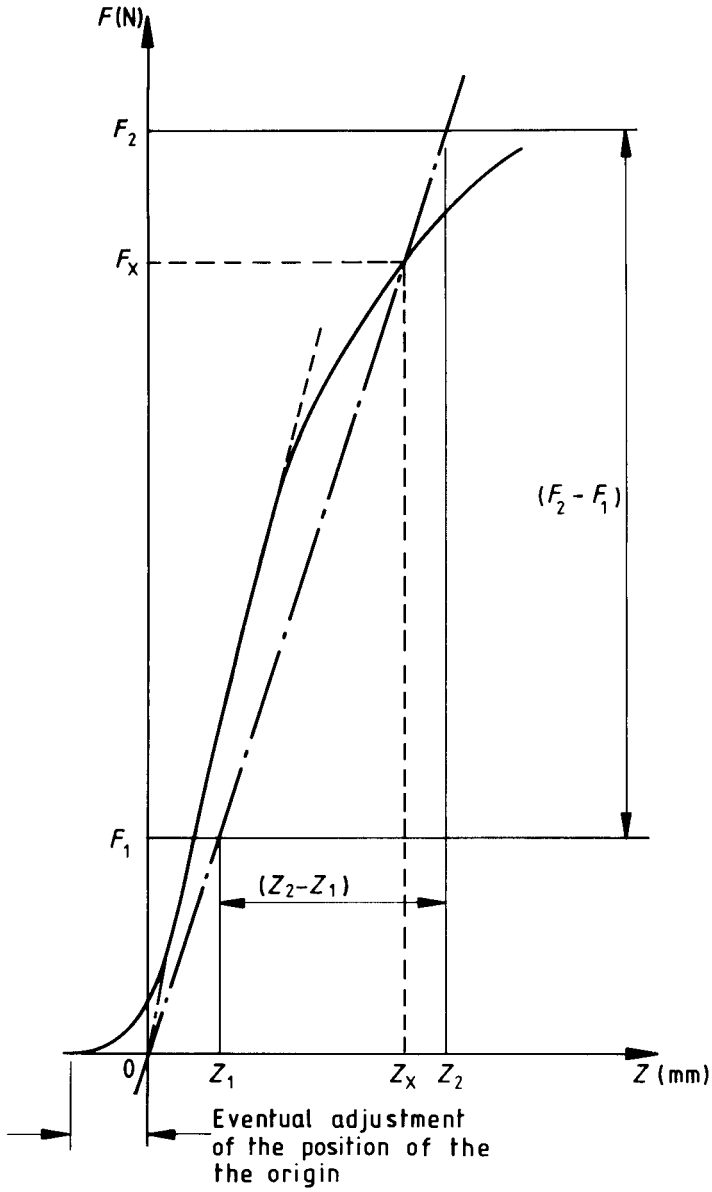



**Figure 33 — Chart of apparent extension, *Z*, versus force, *F***

## B.8 Ultimate compressive unit strength of laminate

The form and number of the test specimens and the test procedure shall be as described in BS 2782:Method 345A.

The ultimate compressive unit strength, $U_c$ (in N/mm), shall be determined from equation (74):

$$U_c = \frac{F}{bm} \tag{74}$$

where

$F$ is the compressive force (in N);

$b$ is the minimum width of the test specimen (in mm);

$m$ is mass of a single type of reinforcement of glass in the specimen (in kg/m$^2$).

## B.9 Lap shear strength of laminate

### B.9.1 *Form of test specimen*

Test specimens shall be of the form and dimensions shown in Figure 34. They shall have a minimum thickness of 3 mm and the length of the specimens may be varied to accommodate the requirements of the available testing equipment. The edges of the specimen shall be smooth but not round or bevelled. Two parallel cuts, one in each opposite face of the specimen and 12.5 mm apart, shall be sawn across the entire width of the specimen. They shall be as narrow as practicable and shall be parallel within 0.8 mm. The incision shall be half the laminate thickness plus the thickness of one layer or half the laminate thickness + 0.1, – 0 mm if the number of layers or thickness per layer is unknown.

### B.9.2 *Number of test specimens*

Five specimens shall be tested when they consist of laminates made entirely from CSM or by spraying.

Whenever the laminate contains WR or other directional reinforcement, 10 specimens, five parallel with each principal axis of anisotropy, shall be tested.

Specimens that fail prematurely or at an obvious flaw shall be discarded and retests made.

### B.9.3 *Procedure*

Condition the specimens by storing them at 20 ± 5 °C for a minimum of 3 h immediately before testing.

Insert the specimen in the grips of the tensile testing machine and stress it until rupture occurs in the form of a shear-type failure with some peeling at the interlaminar bond.

Ensure that the speed of testing, i.e. the relative rate of motion of the grips or test fixtures during test, is within the range of 5.0 mm/min to 6.5 mm/min.

### B.9.4 *Results*

The lap shear strength, $S$ (in N/mm$^2$), shall be determined from equation (75):

$$S = \frac{W}{ab} \tag{75}$$

where

$W$ is the maximum load (in N);

$a$ is the distance between sawcuts (in mm);

$b$ is the width of the specimen (in mm).

The lap shear strength of the laminate under test shall be reported as the arithmetic mean of the lap shear strengths of the specimens.

### B.9.5 *Report*

The report shall state:

a) the lap shear strength of the laminate;

b) the individual test results;

c) the conditioning of the test specimens.

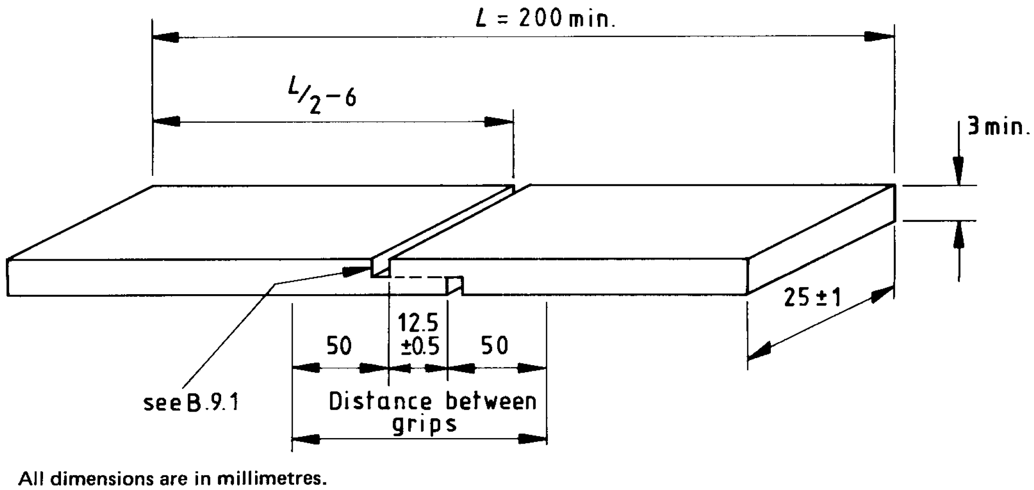



**Figure 34 — Test specimen for the determination of lap shear strength of laminate**

### B.10 Shear strength of bond between thermoplastics lining and laminate

**B.10.1** ***Form of test specimen***

The specimen shall be cut from the full thickness of the laminate and lining and shall be of the form and dimensions shown in Figure 35.

**B.10.2** ***Number of test specimens***

Three specimens shall be used.

**B.10.3** ***Procedure***

Make two thin saw cuts in the specimen at right angles to the major axis, 20 mm apart and symmetrically about the transverse centre line. Make one cut through the full thickness of the thermoplastics material but not into the laminate and the second through the full thickness of the laminate but not into the thermoplastics material.

Keep the specimens at 20 ± 5 °C for not less than 3 h immediately before testing, and carry out the tests at 20 ± 5 °C.

Clamp the specimen in the serrated jaws of a suitable tensile testing machine and in axial alignment with the direction of pull. Strain the specimen at a constant rate of 25 ± 6 mm/min.

The shear strength of the bond shall be calculated from the maximum load and the area under shear and expressed in newtons per square millimetre. The bond shear strength of the material under test shall be reported as the arithmetic mean of the bond shear strengths of the test specimens. If the specimen breaks other than at the interface and the calculated shear strength is less than that specified in **7.1** the test shall be repeated.

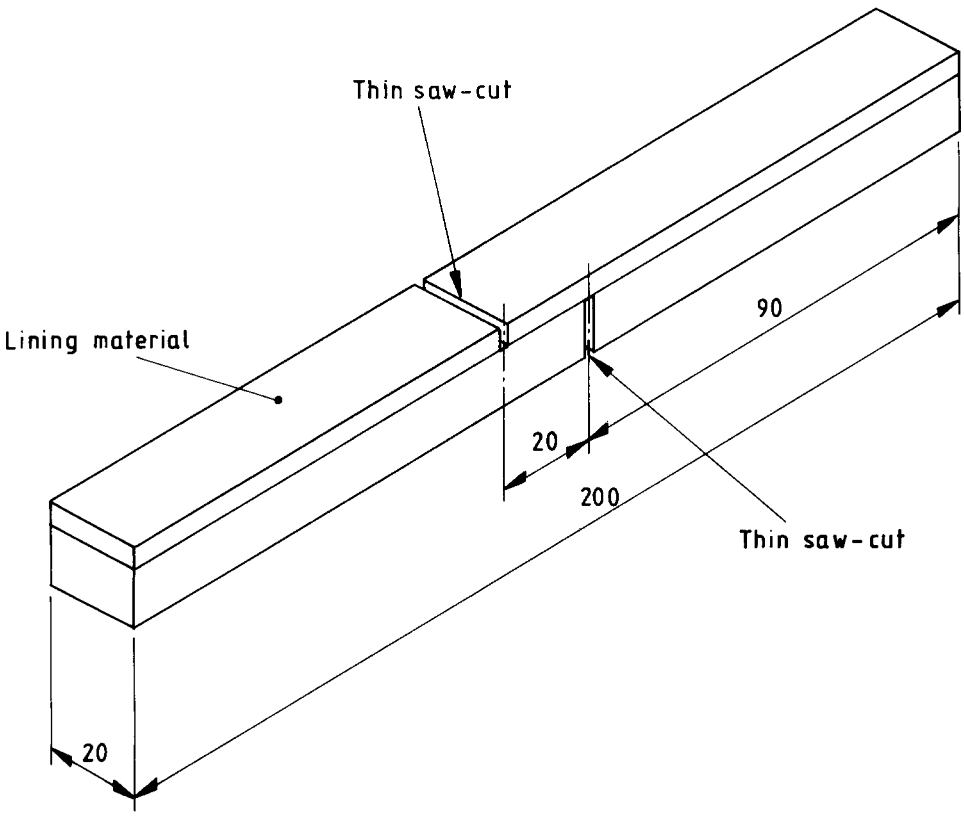
**B.10.4** ***Report***

The report shall state:

a) the bond shear strength of the thermoplastics lining-laminate combination;

b) the individual test results;

c) the conditioning of the test specimens.




All dimensions are in millimetres.

**Figure 35 — Test specimen for the determination of shear strength of bond between thermoplastics lining and laminate**

**B.11 Peel strength of bond between thermoplastics lining and laminate**

**B.11.1** ***Form of test specimen***

The specimen shall be cut from the full thickness of the laminate and lining and shall be of the form and dimensions shown in Figure 36(a).

**B.11.2** ***Number of specimens***

Five specimens shall be used.

**B.11.3** ***Procedure***

Make a saw cut at one end of the specimen at the interface of the laminate and thermoplastics material, across the width of the specimen and for 20 mm along its length.

NOTE The saw cut should include, as far as possible, equal amounts of laminate and thermoplastics material.

Keep the specimens at 20 ± 5 °C for not less than 3 h immediately before testing and carry out the tests at 20 ± 5 °C.

Grip the laminate horizontally in the jaws of a vice or clamp and apply the load to the thermoplastics lining by means of weights until the load is just sufficient to peel the lining from the laminate. During this operation, ensure that the plane of the load remains normal to the laminate thermoplastics interface [see Figure 36(c)].

The peel strength of the load shall be calculated from the total load at peel and the measured width of the specimen, and expressed in newtons per millimetre width. The bond peel strength of the material under test shall be reported as the arithmetic mean of the bond peel strengths of the test specimens.

**B.11.4** ***Report***

The report shall state:

a) the bond peel strength of the thermoplastics lining-laminate combination;

b) the individual test results;

c) the conditioning of the test specimens.

**B.12 Peel strength of bond between laminate layers**

**B.12.1** ***Form of test specimen***

The specimen shall be cut from the full thickness of laminate and shall be of the form and dimensions shown in Figure 36(b). The saw cut shall be as thin as practicable, and shall be made as accurately as possible along the interface between the two laminate layers whose peel strength is required (for example, between a main and an overlay laminate).

**B.12.2** ***Number of specimens***

Five specimens shall be used.

**B.12.3** ***Procedure***

Keep the specimens at 20 ± 5 °C for not less than 3 h immediately before testing, and carry out the tests at 20 ± 5 °C.

Grip the main laminate horizontally in the jaws of a vice or clamp and apply the load to the secondary laminate by means of weights or in a suitable testing machine until the load is just sufficient to peel the two laminate layers apart. During this operation ensure that the plane of the load remains normal to the laminate interface [see Figure 36(c)].

The peel strength of the load shall be calculated from the total load at peel and the measured width of the specimen and expressed in newtons per millimetre width. The bond peel strength of the laminate under test shall be reported as the arithmetic mean of the bond peel strengths to the test specimens.

**B.12.4** ***Report***

The report shall state:

a) the bond peel strength of the laminate;

b) the lay-up of the laminates incorporated in the test specimens;

c) the individual test results;

d) the conditioning of the test specimens.

**B.13 Shear strength of sandwich panel**

**B.13.1** ***Form of test specimen***

Test specimens shall be of the form and dimensions shown in Figure 37.

The core shall be of such a thickness that the centre line of the core lies on the line of axis of the loading. The skins shall consist of a minimum construction of 1.35 kg/m$^2$ CSM each.

**B.13.2** ***Number of specimens***

Five specimens shall be used.

If the core has directional properties, 10 specimens, five parallel with each axis of anisotropy, shall be tested.

**B.13.3** ***Apparatus***

The apparatus shall be as shown in Figure 38. It shall be ensured that each grip bears only on the skin and is at least 0.5 mm from the core/skin interface. The faces of the clamp plates shall be coated with lubricant and tightened just sufficiently to prevent slipping.

**B.13.4** ***Procedure***

Keep the specimens at 20 ± 5 °C for not less than 3 h immediately before testing and carry out the test at 20 ± 5 °C.

Record the dimensions of the specimen, place the specimen in the grips and stress it until failure occurs.

Separate the grips at a constant rate between 1.0 mm/min and 1.5 mm/min.

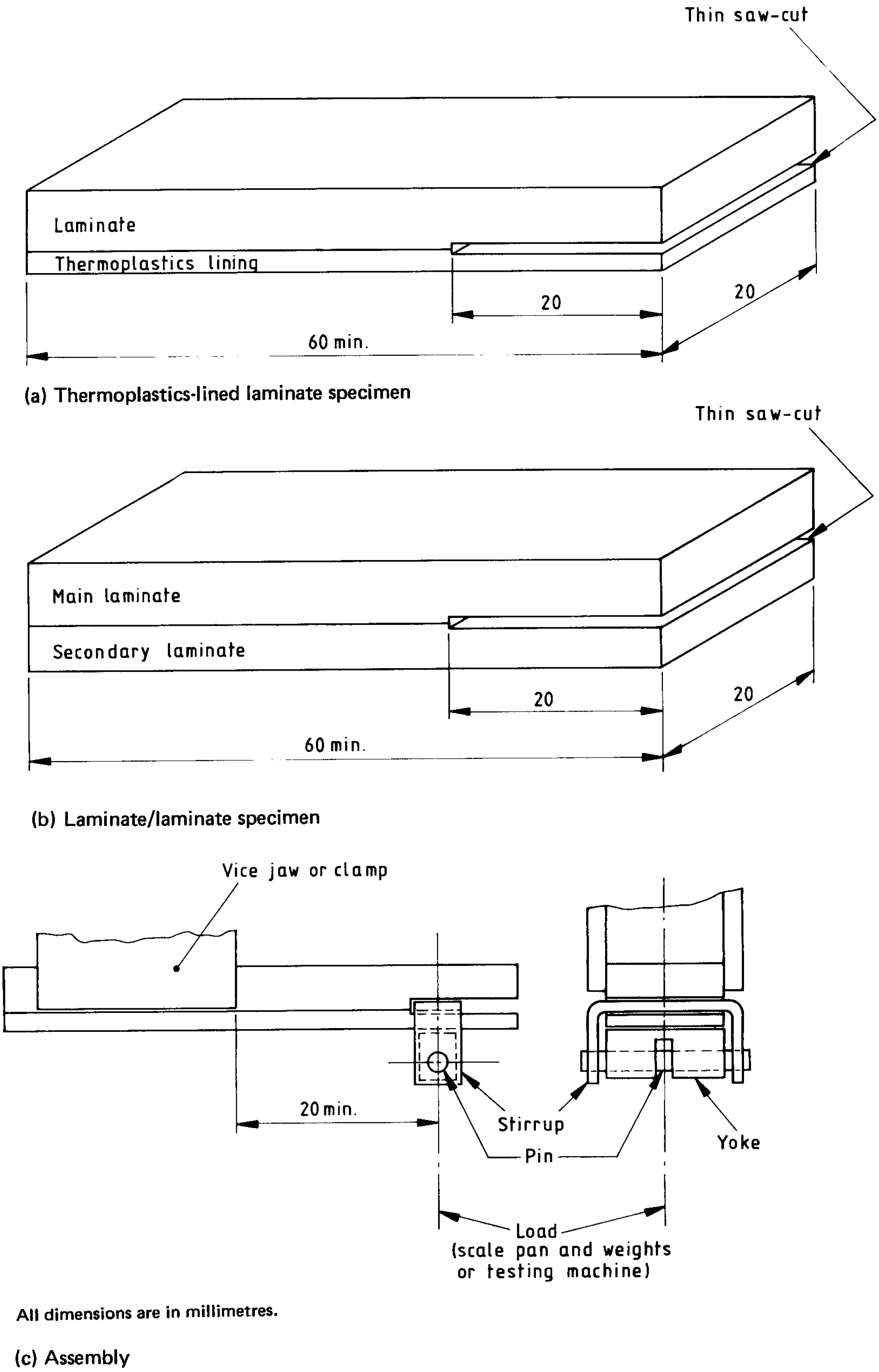



**Figure 36 — Test for determination of peel strength of bond**

**B.13.5** ***Results***

The shear strength, $s$ (in N/mm$^2$), shall be determined from equation (76):

$$s = \frac{W}{ab} \qquad (76)$$

where

- $W$ is the maximum load (in N);
- $a$ is the width of specimen (in mm);
- $b$ is the length of specimen (in mm).

The shear strength of the panel under test shall be reported as the arithmetic mean of the shear strengths of the specimens.

**B.13.6** ***Report***

The report shall state:

a) the shear strength of the core;

b) the individual test results;

c) the conditioning of the test specimens;

d) a brief description of the failure mode.

## B.14 Barcol hardness

The form and number of test specimens and the procedure shall be as described in BS 2782:Method 1001.

NOTE In the Barcol hardness test, the impressor penetrates the surface of the material and should, therefore, be used with caution on surfaces which depend for their chemical resistance on a resin-rich layer.

## B.15 Water absorption

The form and number of test specimens and the procedure shall be as described in BS 2782:Method 551A.

## B.16 Electrical properties

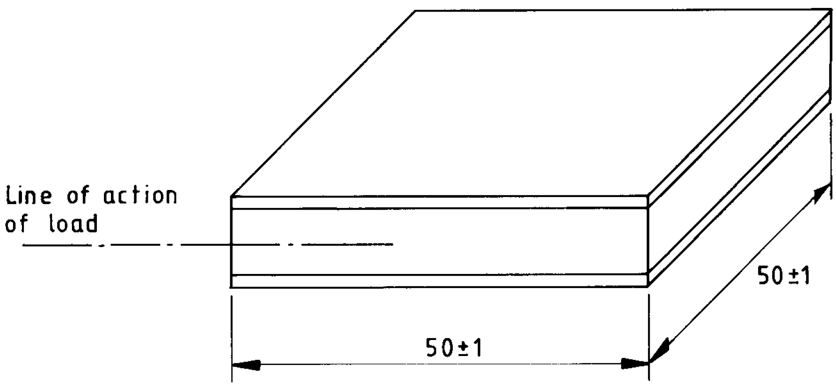
The form and number of test specimens and the procedure shall be as described in BS 2782:Methods 221 and 232.

## B.17 Residual styrene (polyester resins)

The form and number of test specimens and the procedure shall be as described in BS 2782:Method 453A.

## B.18 Acetone test (polyester resins)

The test shall be carried out by laying a cloth, soaked in acetone over selected areas of the lining, for 3 min. After this time the lining shall show no sign of tackiness.




All dimensions are in millimetres.

**Figure 37 — Test specimen for shear strength of sandwich panel**

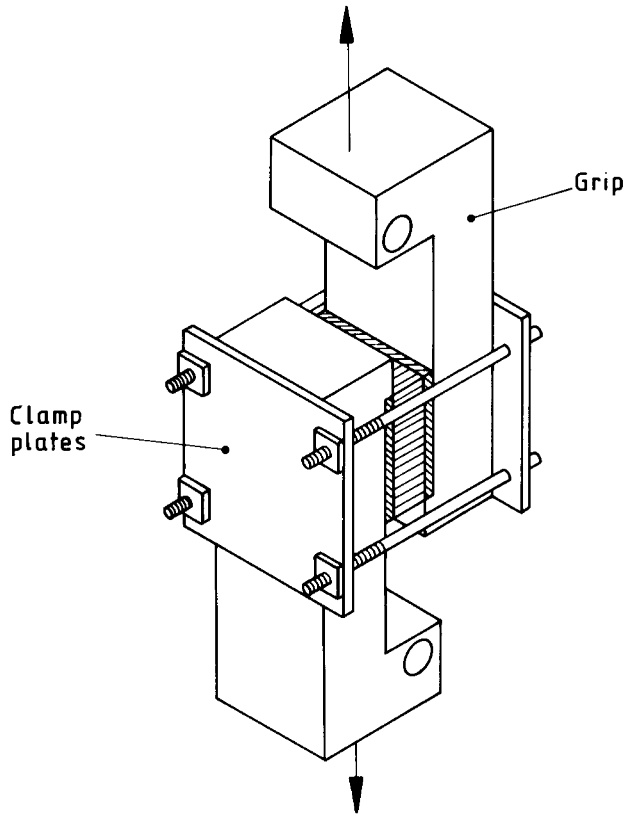



Figure 38 — Assembly of test specimen and grips

## Appendix C Effects of materials on water quality

When used under the conditions for which they are designed, non-metallic materials in contact with or likely to come into contact with potable water shall not constitute a toxic hazard, shall not support microbial growth and shall not give rise to unpleasant taste or odour, cloudiness or discoloration of the water.

Concentrations of substances, chemicals and biological agents leached from materials in contact with potable water, and measurements of organoleptic/physical parameters shall not exceed the maximum values recommended by the World Health Organization in its publication "Guidelines for drinking water quality" Vol 1 "Recommendations" (WHO, Geneva 1984) or as required by the EEC Council Directive of 15 July 1980 relating to the quality of water intended for human consumption (Official Journal of the European Communities L229 pp 11–29), whichever in each case is the more stringent.

NOTE 1 Requirements for the testing of non-metallic materials in these respects are set out in the UK Water Fittings Byelaws Scheme Information and Guidance Note No. 5-01-02, ISSN 0267-0313 obtainable from the Water Research Centre, Water Byelaws Advisory Service, 660 Ajax Avenue, Slough, Berkshire SL1 4BG.

NOTE 2 Pending the determination of suitable means of characterizing the toxicity of leachates from materials in contact with potable water, materials approved by the Department of the Environment Committee on Chemicals and Materials of Construction for use in Public Water Supply and Swimming Pools are considered free from toxic hazard for the purposes of compliance with this appendix. A list of approved chemicals and materials is available from the Technical Secretary of that Committee at the Department of the Environment, Water Division, Romney House, Marsham Street, London SW1P 3PY.

NOTE 3 Products manufactured for installation and use in the United Kingdom which are verified and listed under the UK Water Fittings Byelaws scheme administered by the Water Research Centre, (address as in note 1) are deemed to satisfy the requirements detailed in this appendix.

# Appendix D Worked examples of the design calculations specified in sections 2 and 3

## D.1 General

The design method in this standard, being based on unit loadings, is particularly suited to the design of laminar constructions. Correctly applied, the method ensures that each layer of the composite carries the proportion of the total load appropriate to its strength, and that excessive strains which might lead to local debonding and subsequent service failure cannot occur anywhere in the laminate.

The results of many tests (see reference [8]) done on resin/glass fibre composites have shown that both the ultimate tensile unit strength and the unit modulus values obtained are proportional to the mass of glass reinforcement contained in the laminate layer. For each of these properties a single value may, therefore, be utilized over the full range of glass contents normally used for each type of reinforcement (see Table 5), and the design calculations are thus considerably simplified.

The thickness of a laminate layer is also very largely controlled by the mass of glass specified and the glass content, being subject to only small variation resulting from manufacturing technique. Layer thicknesses for given contents and masses may be obtained, from Figure 5.

In none of the following worked examples is the construction/thickness of the chemical barrier included (see clause **7** and **9.2.2**).

## D.2 Shell and end design (internal pressure)

### D.2.1 *Design criteria*

Consider the design of a cylindrical vessel to be of 1 750 mm internal diameter and for a total effective pressure of 2 bar* (gauge) (0.2 N/mm$^2$).

From equation (7) in **14.2**, the circumferential unit load will be:

$$Q_\phi = \frac{0.2 \times 1750}{2} = 175 \text{ N/mm}$$

For the purposes of this example, it is assumed that the combined longitudinal unit load, $Q_x$, does not exceed 175 N/mm so that a laminate designed to withstand this load will be satisfactory.

Using equation (6) in **13.2** the laminate is designed so that the design strength of the laminate is not less than this calculated value of $Q_\phi$.

### D.2.2 *All CSM construction*

If the vessel was constructed entirely of resin and CSM layers the design unit loading per layer would be determined in accordance with **9.2**, thus:

a) Determine the design factor, $K$, (see **9.2.2**).

| | |
|---|---|
| Factor $k_1$ method of manufacture (hand lay-up) | = 1.5 |
| Factor $k_2$ long-term behaviour | = 2.0 |
| Factor $k_3$ operating temperature (assumed operation at 40 °C using a resin having a heat distortion temperature of 80 °C or higher) | = 1.0 |
| Factor $k_4$ cyclic stressing (assuming occasional filling and emptying) | = 1.1 |
| Factor $k_5$ curing procedure (assumed hot post-cure) | = 1.1 |

Hence

$K = 3.0 \times 1.5 \times 2.0 \times 1.0 \times 1.1 \times 1.1 = 10.89$

b) Determine the load-limited allowable unit loading, $u_L$ (see **9.2.3**):

$$u_L = \frac{\text{Ultimate tensile unit strength for CSM}}{\text{Design factor, } K}$$

$$= \frac{200}{10.89} \text{ (assuming a resin other than furane)}$$

= 18.36 N/mm per kg/m$^2$ glass

c) Determine the allowable strain, $\epsilon$, on laminate layers (see **9.2.4**).

Assuming a resin extension to failure of 3 % then:

$\epsilon = 0.1 \times 3 = 0.3$ %

This value is greater than the maximum strain permitted, therefore take $\epsilon = 0.2$ %.

d) Determine the strain-limited allowable unit loading, $u_S$ (see **9.2.5**):

$u_S$ = unit modulus for CSM (see Table 5) × allowable strain

$= 14\,000 \times 0.2 \times 10^{-2}$

$= 28.0$ N/mm per kg/m² glass

e) Determine the design unit loading, $u_z$, (see **9.2.6**).

Since $u_L$ is less than $u_S$, the $u_L$ value is taken for design purposes, that is, the design is load-limited.

NOTE A strain-limited design will normally occur when either:
1) a comparatively brittle resin is selected; or
2) the method of manufacture and working conditions lead to a low design factor, $K$.

Using equation (4), the design strain, $\epsilon_L$, on each CSM layer will be:

$$\epsilon_L = \frac{18.36 \times 100}{14000} = 0.13\ \%$$

and the design unit loading, $u_z$, at that strain level will be:

$u_z = u_L = 18.36$ N/mm per kg/m² glass

f) The total quantity of reinforcement, which in a laminate consisting wholly of CSM is the product $m_z\, n_z$, can be found simply from equation (6) in **13.2**, thus:

$Q = u_z\,(m_z\, n_z)$

So that total mass $(m_z\, n_z) = \dfrac{175}{18.36} = 9.53\ \text{kg/m}^2$.

A total mass of glass of 9.53 kg/m² is thus necessary and the distribution of this would be selected according to manufacturers' practice.

Thus, one suitable laminate would be [see Figure 4(a)]:

| | |
|---|---|
| 2 layers 0.3 kg/m² (one at each surface) | = 0.6 kg/m² |
| 15 layers 0.6 kg/m² | = 9.0 kg/m² |
| Total | = 9.6 kg/m² |

Equally acceptable would be a construction of [see Figure 4(b)]:

| | |
|---|---|
| 2 layers 0.3 kg/m² (one at each surface) | = 0.6 kg/m² |
| 9 layers 0.6 kg/m² | = 5.4 kg/m² |
| 8 layers 0.45 kg/m² | = 3.6 kg/m² |
| Total | = 9.6 kg/m² |

**D.2.3** ***CSM and WR construction***

**D.2.3.1** *Full procedure.* In practice, an all-CSM construction with such a large number of layers is unlikely to be selected and the construction of the vessel will be simplified by incorporating layers of WR. A typical construction of this type is shown in Figure 4(c) where alternate layers of RP25 WR, with the fibres suitably aligned relative to the direction of the maximum unit load, and 0.45 kg/m² CSM are used, with a layer of 0.3 kg/m² CSM at the outside surface and 1.2 kg/m² CSM at the inside surface. It is necessary to calculate the number of layers required.

a) Determine the design factor, $K$, (see **9.2.2**).

Assume the same factors as for all-CSM construction, so that $K = 10.89$ as in **D.2.2** a).

b) Determine the load-limited allowable unit loading, $u_L$ (see **9.2.3**).

For CSM layers, $u_L$ = 18.36 N/m per kg/m$^2$ glass as in **D.2.2** b).

For WR layers, $u_L = \frac{250}{10.89} = 22.95$ N/mm per kg/m$^2$ glass (assuming a resin other than furane).

c) Determine the allowable strain, $\epsilon$, on laminate layers (see **9.2.4**).

Assuming a resin extension to failure of 3 %, $\epsilon$ is limited to 0.2 % as in **D.2.2** c).

d) Determine the strain-limited allowable unit loading, $u_S$ (see **9.2.5**).

For CSM layers, $u_S$ = 28.0 N/mm per kg/m$^2$ glass as in **D.2.2** d).

For WR layers, $u_S = 16\,000 \times 0.2 \times 10^{-2}$

$= 32.0$ N/mm per kg/m$^2$ glass.

e) Determine the design unit loading, $u_z$, (see **9.2.6**).

In this example, $u_L$ is lower than $u_S$ for both types of layer so that the design is load-limited, just as it was for all-CSM construction.

Therefore, the strain in each layer when loaded with $u_L$ is next determined.

For CSM, $\epsilon_L = \frac{18.36 \times 100}{14000} = 0.13\ \%$ [as in **D.2.2** e)].

For WR, $\epsilon_L = \frac{22.95 \times 100}{16000} = 0.14\ \%.$

To avoid overloading the CSM layers in the laminate the design strain, $\epsilon_d$, has to be limited to 0.13 %, so that the design unit loadings equivalent to that strain level will be:

For CSM, $u_z = u_L = 18.36$ N/mm per kg/m$^2$ glass.

For WR, $u_z = X_z\, \epsilon_d = 16\,000 \times 0.13 \times 10^{-2}$

$= 20.8$ N/mm per kg/m$^2$ glass.

f) The laminate construction may now be determined from equation (6) in **13.2**.

If the number of layers of rovings required is $n$, the number of layers of 0.45 kg/m$^2$ CSM will be $(n - 1)$. In addition, there will be 1.5 kg/m$^2$ of CSM, distributed 0.3 kg/m$^2$ at the outside surface and 1.2 kg/m$^2$ at the inside surface.

The equation reads:

| | | | |
|---|---|---|---|
| $u_1\, m_1\, n_1$ | $+ u_2\, m_2\, n_2$ | $+ u_3\, m_3\, n_3$ | $\geqslant Q$ |
| $20.8 \times 0.8n$ | $+ 18.36 \times 0.45\,(n - 1)$ | $+ 18.36 \times 15$ | $\geqslant 175$ |
| $n$ layers | $(n - 1)$ layers | surface layers | |
| WR | 0.45 kg/m$^2$ CSM | 1.5 kg/m$^2$ CSM | |

Solving, the minimum value of $n$ to satisfy this condition is **6.26**.

Since the number of layers has to be an integer, $n$ has to be taken for this construction as 7, so the complete laminate would comprise [see Figure 4(c)]:

| | |
|---|---|
| chemical barrier (see clause **7** and **9.2.2**) | (internal surface) |
| 1.2 kg/m$^2$ CSM | |
| 0.8 kg/m$^2$ WR<br>0.45 kg/m$^2$ CSM | repeated six times |
| 0.8 kg/m$^2$ WR | |
| 0.3 kg/m$^2$ CSM | |
| resin-rich layer with binding tissue | (external surface) |

g) However, as the value of $n$ is very close to 6 it is possible by adding a further layer of 0.45 kg/m² behind the gel coat to reduce the actual calculated $n$ below 6. This will result in a saving of one layer of 0.8 kg/m² WR. The equation would then read:

| | | | |
|---|---|---|---|
| $u_1\, m_1\, n_1$ | $+ u_2\, m_2\, n_2$ | $+ u_3\, m_3\, n_3$ | $\geqslant Q$ |
| $20.8 \times 0.8n$ | $+ 18.36 \times 0.45\,(n-1)$ | $+ 18.36 \times 1.95$ | $\geqslant 175$ |
| $n$ layers | $(n-1)$ layers | surface layers | |
| WR | 0.45 kg/m² CSM | 1.95 kg/m² CSM | |

Solving, the minimum value of $n$ to satisfy this condition is 5.92 and the corresponding laminate construction will be:

| | |
|---|---|
| chemical barrier (see clause **7** and **9.2.2**) | (internal surface) |
| 1.65 kg/m² CSM | |
| 0.8 kg/m² WR<br>0.45 kg/m² CSM | repeated five times |
| 0.8 kg/m² WR | |
| 0.3 kg/m² CSM | |
| resin-rich layer with binding tissue | (external surface) |

**D.2.3.2** *Abbreviated procedure.* It is possible to abbreviate the procedure given in **D.2.3.1** if the minimum strain is firstly selected.

Allowable strain for resin = $0.1 \times \epsilon_r$.

$$\text{Allowable strain for CSM laminate} = \frac{\text{UTUS}^{a}(\text{CSM})}{X(\text{CSM}) \times K}$$

$$\text{Allowable strain for WR laminate} = \frac{\text{UTUS (WR)}}{X(\text{WR}) \times K}$$

Maximum allowable strain = 0.2 %.

Design strain, $\epsilon_d$ = least of above strains.

The design unit loadings would, therefore, be:

$U_{CSM} = \epsilon_d \times X_{CSM}$

$U_{WR} = \epsilon_d \times X_{WR}$

*Example*

$K$ factor = 10.89

Determine the allowable strains.

a) Allowable strain of resin

= 0.1 × 3 % = 0.3 %.

b) Allowable strain for CSM

= $200/(14000 \times 10.89 \times 10^{-2})$ = 0.13 %.

c) Allowable strain for WR

= $250/(16000 \times 10.89 \times 10^{-2})$ = 0.14 %.

Least strain above is 0.13 % maximum allowable strain is 0.2 %.

CSM layers limit and $\epsilon_d$ = 0.13 %.

Determine allowable unit loadings.

1) For CSM layers = 0.13 × 14000/100 = 18.2 N/mm.

2) For WR layers = 0.13 × 16000/100 = 20.8 N/mm.

[a] Ultimate tensile unit strength (see Table 5).

**D.2.4** ***Thickness calculations***

Assume a glass content of 30 % for CSM and 55 % for WR with a resin of relative density 1.1.

From Figure 5, the thickness expected for these constructions will be:

CSM, 2.5 mm per kg/m² glass.

WR, 1.1 mm per kg/m² glass.

The all-CSM construction of example **D.2.2** requires a total of 9.6 kg/m² glass and the overall thickness of this laminate irrespective of detailed construction will thus be 2.5 × 9.6 = 24 mm [see Figure 4(a) and Figure 4(b)].

Similarly, the mixed construction of example **D.2.3** may be summarized as follows.

*Case 1* [see **D.2.3** f)]:

1.5 kg/m² of CSM

six layers of 0.45 kg/m² CSM

seven layers of 0.8 kg/m² WR

This gives a total of:

| | |
|---|---|
| 1.5 + 6 × 0.45 = 4.2 kg/m² CSM having a thickness of 2.5 × 4.2 | 10.5 mm |
| and seven layers of WR having a thickness of 1.1 × 0.8 × 7 | 6.2 mm |
| The total design thickness of this laminate construction [see Figure 4(c)] is thus | 16.7 mm |

*Case 2* [see **D.2.3** g)]. The mixed construction using six layers of WR:

1.95 kg/m² of CSM

five layers of 0.45 kg/m² CSM

six layers of 0.8 kg/m² WR

This gives a total of:

| | |
|---|---|
| 1.95 + 5 × 0.45 = 4.2 kg/m² CSM having a thickness of 2.5 × 4.2 | 10.5 mm |
| and six layers of WR having a thickness of 1.1 × 0.8 × 6 | 5.3 mm |
| The total design thickness of this construction [see Figure 4(d)] is thus | 15.8 mm |

**D.2.5** ***Design of dished ends***

If a shallow end of torispherical shape is to be adopted typical values might be 0.20 for $h_i/D_i$ and 0.10 for $r_i/D_i$.

For an end of this shape, the value of shape factor, $K_s$, to be used in design, as determined from Table 11, will be between 2.25 (for a thick end) and 2.95 (for a thin end).

Assume an initial value for $K_s$ of, for example, 2.5. Then from equation (43) in **18.3.2** determine the unit load to be substituted in equation (6) in **13.2**:

$$Q = 0.5 \times 0.2 \times 1750 \times 2.5$$
$$= 437.5 \text{ N/mm}$$

If the laminate construction for the end is to be of CSM and WR, similar to that determined for the vessel shell, use of equation (6) will give:

$$u_1\, m_1\, n_1 + u_2\, m_2\, n_2 + u_3\, m_3\, n_3 \geqslant Q$$

$$20.8 \times 0.8n + 18.36 \times 0.45\,(n - 1) + 18.36 \times 1.5 \geqslant 437.5$$

WR — 0.45 kg/m² CSM — 1.5 kg/m² CSM

The minimum value of $n$ to satisfy this condition is **16.8**, so 17 layers of rovings would be required.

A laminate of this construction will have a total thickness of:

1.5 × 2.5 + 17 × 1.1 × 0.8 + 16 × 0.45 × 2.5 = 36.71 mm

(surface mats CSM) — (WR) — (main mat CSM)

For an actual laminate thickness of 36.71 mm, the ratio $t/D_i$ = 36.71/1750 = 0.021.

From Table 11, the $K_s$ value corresponding to this thickness/diameter ratio should be 2.63.

Repeating the procedure using this value of $K_s$ gives $Q$ = 460.3 N/mm, and produces a laminate construction [obtained by solving equation (6) with this value of $Q$] containing 18 layers of rovings.

This laminate will have a total thickness of:

$1.5 \times 2.5 + 18 \times 1.1 \times 0.8 + 17 \times 0.45 \times 2.5 = 38.72$ mm

This new thickness will give rise to a new value of $t/D_i$ = 38.72/1750 = 0.022 and thus to a slightly different value of $K_s$ = 2.62 and a revised value for $Q$ = 458.5 N/mm so that the laminate construction is now correctly determined, and the actual laminate will thus consist of:

reinforced gel coat
1.2 kg/m$^2$ CSM
0.8 kg/m$^2$ WR
0.45 kg/m$^2$ CSM } repeated 17 times
0.8 kg/m$^2$ WR }
0.3 kg/m$^2$ CSM
resin-rich outer lay with tissue

with a total thickness, of 38.72 mm.

Where a domed end is designed to incorporate WR (or other directionally oriented reinforcement) it is essential that, in order to achieve the calculated strength, the fibre directions are disposed correctly as the end is laid up during manufacture.

This is most easily done by laying in the rovings in sectors around the circumference so that the warp or weft directions are radial/circumferential over the knuckle and blending into successive layers, each angled so as to achieve approximately equal strength in all directions over the crown of the end.

Alternatively, to avoid the problems of directional properties, the end may be designed using an all-CSM construction. Because of the lower permissible unit loading for CSM, the end will be thicker than its CSM/WR counter-part and should be calculated as follows.

Assume an initial value of 2.4 for $K_s$.

The unit load $Q$, from equation (43), will then be:

$Q = 0.5 \times 0.2 \times 1750 \times 2.4$
$= 420$ N/mm

Use of the simplified form of equation (6) for an all-CSM laminate gives:

$$\text{total glass mass} = \frac{Q}{u} = \frac{420}{18.36} = 22.88 \text{ kg/m}^2$$

Assuming a glass content for the CSM layers of 30 % this mass of glass should produce a laminate having a thickness of:

$22.88 \times 2.5 = 57.2$ mm

For a laminate of this thickness $t/D_i$ = 57.2/1750 = 0.0327 and $K_s$ has a value of about 2.45, so that a total of 23.4 kg/m$^2$ glass is required to make the end, with a laminate thickness of 58.5 mm (more than 1.5 times the thickness required for a laminate incorporating rovings).

However, a more economical laminate may be designed by accepting more deeply dished ends.

If, for example, $h_i/D_i$ were taken as 0.25 and $r_i/D_i$ as 0.15, the corresponding value for shape factor $K_s$ will be about 1.78. At this value of $K_s$ the unit load to be carried is:

$Q = 0.5 \times 0.2 \times 1750 \times 1.78$
$= 311.5$ N/mm

Using the same type of laminate construction and substituting the new value of $Q$ into equation (6), the corresponding value of $n$ now becomes 11.7 so that 12 layers of rovings would be adequate.

An end with this number of roving layers will have a total thickness of:

$1.5 \times 2.5 + 12 \times 1.1 \times 0.80 + 11 \times 0.45 \times 2.5 = 26.68$ mm

This thickness gives a $t/D_i$ ratio very close to 0.015, so that the factor $K_s$ has been correctly chosen and further calculation is, in this instance, not necessary.

## D.3 Shell design (external pressure)

### D.3.1 *General*

Consider a cylindrical vessel having an internal diameter of 1 500 mm and an effective length (as shown in Figure 8) of 1 200 mm, which has to withstand a 500 mbar vacuum (i.e. an external pressure of $5 \times 10^{-2}$ N/mm² gauge).

It is to be lined with a chemically resistant polyester resin for process reasons and is intended for operation under nominally steady pressure conditions at ambient temperatures. The vessel is to be made by hand lay-up, with both resins having a heat distortion temperature of more than 80 °C, and is to be fully post-cured at elevated temperature.

The design procedure is initially similar to that in **D.2**.

### D.3.2 *All-CSM construction*

It is necessary to calculate the number of layers required.

a) Determine the design factor $K$ (see **9.2.2**).

| | |
|---|---|
| Factor $k_1$ method of manufacture (hand lay-up) | = 1.5 |
| Factor $k_2$ long-term behaviour (resin with proved performance) | = 1.2 |
| Factor $k_3$ operating temperature (ambient) | = 1.0 |
| Factor $k_4$ cyclic stressing (1 000 cycles) | = 1.1 |
| Factor $k_5$ curing procedure (hot post-cure) | = 1.1 |

Hence

$K = 3 \times 1.5 \times 1.2 \times 1.0 \times 1.1 \times 1.1 = 6.54$

However a minimum value of 8 has to be used (see **9.2.2**).

b) Determine the load-limited allowable unit loading, $u_L$ (see **9.2.3**):

$$u_L = \frac{\text{Ultimate tensile unit strength for CSM}}{\text{Design factor } (K)}$$

$$= \frac{200}{8} = 25.0 \text{ N/mm per kg/m}^2 \text{ glass}$$

c) Determine the allowable strain, ϵ, in laminate layers (see **9.2.4**).

Assuming the use of a flexible resins with extensions to failure of at least 2 %, ϵ will be 0.2 % for both lining and reinforcement layers.

d) Determine the strain-limited allowable unit loading, $u_S$ (see **9.2.5**):

$u_S$ = unit modulus for CSM (see Table 5) × allowable strain

$= 14000 \times 0.2 \times 10^{-2}$

= 28.0 N/mm per kg/m² glass

e) Determine the design unit loading, $u_z$ (see **9.2.6**):

In this example $u_L$ is less than $u_S$, the value of $u_L$ is to be taken as the design unit loading, $u_z$, and the design is thus load-limited.

f) Cylindrical shell calculation (see **14.4**).

Circumferential unit load calculated from equation (7) in **14.2**:

$$Q_\phi = \frac{5 \times 10^{-2} \times 1500}{2} = 37.5 \text{ N/mm}$$

Longitudinal unit load:

$Q_x = 40$ N/mm

NOTE This value is assumed for the purposes of this example. In practice it has to be calculated taking into account such things as the method of vessel support, wind and other external loads and the weight of liquid contents at various stages of filling, as appropriate to the vessel under consideration.

Since $Q_x$ is higher than $Q_\phi$, the $Q_x$ value has to be used in equation (6) in **13.2** to determine the laminate construction. For all-CSM construction, the simplified version of equation (6) may be used as in **D.2.2**:

$Q = u_z\ (m_z\ n_z)$

Thus total mass, $m_z\ n_z = \dfrac{40}{25} = 1.6\ \text{kg/m}^2$

A total glass mass of 1.6 kg/m² is thus adequate to carry the circumferential and longitudinal unit loads. However, the resulting thickness of 1.6 × 2.5 = 4.0 mm is less than the permitted minimum of 5.0 mm (see **13.3**) for which a glass content of 2 kg/m² will be necessary.

| | |
|---|---|
| For example two layers 0.3 kg/m² (one at each surface) | = 0.6 kg/m² |
| plus additional layers of CSM to make up 1.4 kg/m² | = 1.4 kg/m² |
| Total = | 2.0 kg/m² |

The design is not, however, complete without a check calculation for the possibility of buckling under external pressure and axial load.

g) Determine the thickness and overall unit modulus of the proposed laminate.

Making the same assumption as in **D.2.2** regarding glass content (30 % for CSM), the thickness per kg/m² of glass will be 2.5 mm, as before, so the total laminate thickness is 2.0 × 2.5 = 5.0 mm.

Overall unit modulus (see **14.3.3**):

$$X_{LAM} = X_1\ (m_1\ n_1)$$
$$= 14000 \times 2$$
$$= 28000 \text{ N/mm}$$

Composite modulus (see **14.4.1**):

$$E_{LAM} = \frac{X_{LAM}}{t}$$
$$= \frac{28000}{5}$$
$$= 5600 \text{ N/mm}^2$$

h) Determine the minimum thickness to prevent buckling, $t_m$, from equations (15) and (16) in **14.4.1**.

As these equations use $D_o$, the outside diameter, it is necessary to assume a value of $t$.

As a start point assume the value of $t$ as the largest yet calculated, e.g. for internal pressure or axial load or required minimum thickness.

In this case assume $t = 5.0$ mm (for minimum required thickness)

Then

$D_o = D_i + 2t = 1500 + 2 \times 5 = 1510$ mm.

The thickness to prevent buckling is then determined from equation (15) or (16).

To determine which equation to use calculate $\frac{L}{D_o}$ and

$$1.35\left(\frac{E_{LAM}}{pF}\right)^{0.17}$$

$$\frac{L}{D_o} = \frac{1200}{1510} = 0.795$$

$$1.35\left(\frac{E_{LAM}}{pF}\right)^{0.17} = 1.35\left(\frac{5600}{5 \times 10^{-2} \times 4}\right)^{0.17} = 7.7$$

$$\text{As } \frac{L}{D_o} < 1.35\left(\frac{E_{LAM}}{pF}\right)^{0.17} \text{ use equation (16)}$$

$$t_{min} = D_o\left(\frac{0.4\,pF}{E_{LAM}} \times \frac{L}{D_o}\right)^{0.40}$$

$$= 1510\left(\frac{0.4 \times 5 \times 10^{-2} \times 4}{5600} \times \frac{1200}{1510}\right)^{0.40}$$

$$= 15.89 \text{ mm}$$

which is greater than the 5.0 mm previously determined.

A thicker laminate is therefore necessary for stability under external pressure and the procedure has to be repeated until the calculated thickness is equal to or less than the assumed thickness.

By trial a thickness of 16.1 mm has been established as likely to be required, giving $D_o$ = 1532.2 mm.

Checking again by equation (16):

$$t_{min} = D_o\left(\frac{0.4\,pF}{E_{LAM}} \times \frac{L}{D_o}\right)^{0.4}$$

$$= 1532.2\left(\frac{0.4 \times 5 \times 10^{-2} \times 4}{5600} \times \frac{1200}{1532.2}\right)^{0.4}$$

$$= 16.02 \text{ mm}$$

As this is less than the assumed thickness, 16.1 mm should be adequate to resist circumferential buckling.

It is now necessary to check that a laminate of this thickness will be stable under the assumed longitudinal load of 40 N/mm, using equation (13) in **14.3.3**.

For a thickness of 16.1 mm the required glass mass will be:

$$\frac{16.1}{2.5} = 6.44 \text{ kg/m}^2 \text{ for 30 \% CSM by mass}$$

For this laminate $X_{LAM}$ = 6.44 × 14000 = 90 160 N/mm.

From equation (13) the maximum permissible longitudinal compressive unit load is:

$$Q_p = \frac{0.6 \times 16.1 \times 90\,160}{4 \times 1532.2} = 142 \text{ N/mm}$$

This is greater than the assumed longitudinal unit load of 40 N/mm therefore the thickness of 16.1 mm in 30 % CSM is adequate for stability under both the external pressure and the axial load.

To achieve the thickness of 16.1 mm with a glass mass of 6.44 kg/m² a suitable laminate would be:

| | | |
|---|---|---|
| 2 layers 0.3 kg/m² CSM (one at each surface) | = | 0.6 kg/m² |
| 9 layers 0.6 kg/m² CSM | = | 5.4 kg/m² |
| 1 layer 0.45 kg/m² CSM (in centre) | = | 0.45 kg/m² |
| Total | = | 6.45 kg/m² |

NOTE For the laminate thickness of 5 mm determined by the initial calculation, the maximum permissible longitudinal unit load would have been:

$$Q_p = \frac{0.6 \times 5 \times 28\,000}{4 \times 1510} = 13.9 \text{ N/mm}$$

which is considerably less than the assumed required longitudinal unit load of 40 N/mm.

All other CSM constructions having a minimum of 6.44 kg/m² total mass of glass and a thickness of 16.1 mm would be equally acceptable.

Alternatively the vessel may be redesigned using a similar glass/resin ratio.

If the stiffener spacing is reduced to 500 mm, equation (16) will again be required. This time assume a thickness of 12 mm when $D_o = 1500 + 2 \times 12 = 1524$ mm

$$t_{min} = D_o \left( \frac{0.4\,pF}{E_{LAM}} \times \frac{L}{D_o} \right)^{0.4}$$

$$= 1524 \left( \frac{0.4 \times 5 \times 10^{-2} \times 4}{5600} \times \frac{500}{1524} \right)^{0.4}$$

$$= 11.25 \text{ mm}$$

As this is less than the assumed value of 12 mm it would appear the latter is satisfactory against collapse under external pressure. A check is still necessary for suitability for the assumed longitudinal unit load of 40 N/mm, using equation (13).

The required glass mass for a 12 mm thick laminate using 30 % CSM by mass:

$$= \frac{12}{2.5} = 4.8 \text{ kg/m}^2$$

$$X_{LAM} = 4.8 \times 14\,000 = 67\,200 \text{ N/mm}$$

$$Q_p = \frac{0.6 \times 12 \times 67\,200}{4 \times 1524} = 79.4 \text{ N/mm}$$

which is greater than the required value of 40 N/mm.

With the unsupported length reduced to 500 mm the laminate thickness of 12 mm would be satisfactory to resist both circumferential and longitudinal buckling. The laminate glass mass is reduced by approximately 25 %.

An appropriate laminate construction would be:

| | | |
|---|---|---|
| 2 layers 0.3 kg/m² CSM (one at each surface) | = | 0.6 kg/m² |
| 7 layers 0.6 kg/m² CSM | = | 4.2 kg/m² |
| Total | = | 4.8 kg/m² |

**D.3.3** ***CSM and WR construction***

As in **D.2**, a CSM and WR construction could be used for this vessel. Assuming, then, a general construction of the type shown in Figure 4(c) (where alternate layers of RP25 WR, with the fibres suitably aligned relative to the direction of the maximum unit load, and 0.45 kg/m² CSM are used, with a layer of 0.3 kg/m² CSM at the outside surface and 1.2 kg/m² CSM at the inside surface. It is necessary to calculate the number of layers required.

a) Determine the design factor, $K$ (see **9.2.2**).

Assume the same factors as for all-CSM construction, so that $K = 8.0$, as in **D.3.2** a).

b) Determine the load-limited allowable unit loading, $u_L$ (see **9.2.3**).

For CSM layers, $u_L = 25.0$ N/mm per kg/m² glass as in **D.3.2** b).

For WR layers, $u_L = \frac{250}{8.0} = 31.25$ N/mm per kg/m² glass.

c) Determine the allowable strain, $\epsilon$, on laminate layers (see **9.2.4**).

Assuming a resin extension to failure of at least 2 %, $\epsilon$ is limited to 0.2 % as in **D.3.2** c).

d) Determine the strain-limited allowable unit loading, $u_S$ (see **9.2.5**).

For CSM layers, $u_S$ = 28.0 N/mm per kg/m² glass as in **D.3.2** d).

For WR layers, $u_S = 16000 \times 0.2 \times 10^{-2}$

$= 32.0$ N/mm per kg/m² glass.

e) Determine the design unit loading, $u_z$, (see **9.2.6**).

As both types of layer are load limited determine the strain in each layer.

$$\text{For CSM, } \epsilon_d = \frac{25 \times 100}{14000} = 0.178\ \%$$

$$\text{For WR, } \epsilon_d = \frac{21.25}{16000} \times 100 = 0.195\ \%$$

To avoid overloading the CSM layers limit the WR layer to 0.178 % strain = $\epsilon_d$

For WR, $u_z = X \epsilon_d$

$= 16000 \times 0.178 \times 10^{-2}$

$= 28.48$ N/mm (per kg/m² glass)

f) The circumferential and longitudinal unit loads to be carried out will be the same as in the all-CSM example, so the same value of $Q$ (= 40 N/mm) is used in equation (6) in **13.2** to determine the laminate construction.

If the number of layers of WR required is $n$, the number of layers of 0.45 kg/m² CSM will be ($n$ – 1). In addition there will be 1.5 kg/m² CSM, distributed 0.3 kg/m² at the outside surface and 1.2 kg/m² at the inside surface.

The equation reads:

$u_1\, m_1\, n_1 \quad + u_2\, m_2\, n_2 \quad + u_3\, m_3\, n_3 \quad \geqslant Q$

$28.48 \times 0.8n + 25.0 \times 0.45(n - 1) + 25.0 \times 1.5 \geqslant 40$

| | | |
|---|---|---|
| $n$ layers WR | ($n$ – 1) layers 0.45 kg/m² CSM | surface layers 1.5 kg/m² CSM |

Solving, the minimum value of $n$ to satisfy this condition is 1.06, so that $n$ has to be taken as 2. However, the design is not complete without check calculations for the possibility of buckling.

g) Determine the thickness and overall unit modulus of the proposed laminate.

Making the same assumptions as in **D.2** regarding glass content (30 % for CSM and 55 % for WR), the laminate thickness will be:

| | |
|---|---|
| CSM (1.5 + 1 × 0.45) × 2.5 | = 4.88 mm |
| WR 2 × 0.8 × 1.1 | = 1.76 mm |
| Total | = 6.64 mm |

Unit modulus (see **14.3.3**):

$X_{LAM}$

$= X_1\, m_1\, n_1 \quad + X_2\, m_2\, n_2 \quad + X_3\, m_3\, n_3$

$= 16000 \times 0.8 \times 2 + 14000 \times 0.45 \times 1 + 14000 \times 1.5$

| | | |
|---|---|---|
| WR | 450 g/m² CSM | surface layers CSM |

$= 52900$ N/mm

Composite modulus (see **14.4.1**):

$$E_{LAM} = \frac{X_{LAM}}{t} = \frac{52900}{6.64} = 7966\ \text{N/mm}^2$$

h) Determine the minimum thickness to prevent buckling, $t_m$, from equations (15) or (16) in **14.4.1**.

As in **D.3.2** h) assume a value of thickness equal to the largest calculated. In this case $t$ = 6.64 mm

$$D_o = D_i + 2t = 1500 + 2 \times 6.64$$
$$= 1513.3 \text{ mm}$$
$$t_{min} = D_o \left( \frac{0.4pF}{E_{LAM}} \times \frac{L}{D_o} \right)^{0.4}$$
$$= 1513.3 \left( \frac{0.4 \times 5 \times 10^{-2} \times 4}{7966} \times \frac{1200}{1513.3} \right)^{0.4}$$
$$= 13.8 \text{ mm}$$

As this is larger than the assumed thickness a new value has to be assumed. As the thickness change will affect the value of $E_{LAM}$ a construction has to be chosen based on the previous calculated value.

Assuming a construction of the type shown in Figure 4(c), the proportion of WR and hence the value of $E_{LAM}$ will increase, decreasing the required thickness. The under-estimate of thickness will have the opposite effect.

Assume a value of 13.5 mm.

The number of layers of WR can then be determined as follows again assuming the same glass contents.

$$1.1 \times 0.8 \times n + 2.5 \times 0.45\ (n - 1) + 2.5 \times 1.5 = 13.5$$

This gives a value of $n$ = 5.42.

As this value is close to 5 the thickness can be conveniently made up with an extra layer of 0.45 kg/m² CSM to give a construction of:

reinforced gel coat

1.2 kg/m² CSM

0.8 kg/m² WR

0.45 kg/m² CSM } Repeated four times
0.8 kg/m² WR }

0.45 kg/m² CSM

0.3 kg/m² CSM

$$t = 1.1 \times 0.8 \times 5.0 + 2.5 \times 0.45 \times 4 + 2.5 \times 1.95$$
$$= 13.77 \text{ mm}$$

To find unit modulus (see **14.3.3**):

$$X_{LAM} = X_1\, m_1\, n_1 + X_2\, m_2\, n_2 + X_3\, m_3\, n_3$$
$$= 16000 \times 0.8 \times 5 + 14000 \times 0.45 \times 4$$
$$+ 14000 \times 1.95$$
$$= 116500 \text{ N/mm}$$

Composite modulus (see **14.4.1**):

$$E_{LAM} = \frac{X_{LAM}}{t} = \frac{116\,500}{13.77} = 8460 \text{ N/mm}^2$$

Recalculate $t_{min}$ ($D_o$ = 1527.6 mm)

$$t_{min} = D_o \left( \frac{0.4pF}{E_{LAM}} \times \frac{L}{D_o} \right)^{0.4}$$
$$= 1527.6 \left( \frac{0.4 \times 5 \times 10^{-2} \times 4}{8460} \times \frac{1200}{1527.6} \right)^{0.4}$$
$$= 13.6 \text{ mm}$$

This is less than but close to the assumed thickness which is thus the minimum required to resist external pressure.

A check is now required to ensure suitability for the longitudinal compressive loading of 40 N/mm, using equation (13) in **14.3.3**:

$$Q_p = \frac{0.6 \times 13.6 \times 116\,500}{4 \times 1527.6} = 155.6 \text{ N/mm}$$

which is greater than 40 N/mm hence the proposed laminate with a thickness of 13.77 mm is satisfactory.

Alternatively the vessel may be redesigned.

If the stiffener spacing is reduced to 500 mm, from equation (16) and again using $t$ = 6.64 mm as an initial value, a thickness of 9.74 mm is obtained.

This thickness is used to assume a new construction of:

1.5 kg/m$^2$ CSM

0.8 kg/m$^2$ WR } repeated three times
0.45 kg/m$^2$ CSM }

Which yields the following properties:

$t$ = 9.76 mm

$X_{LAM}$ = 78 300 N/mm

$E_{LAM}$ = 8023 N/mm$^2$

A further calculation from equation (16) using the new values gives a thickness of 9.72 mm which is less than but close to the assumed thickness.

The assumed construction is therefore sufficient to carry the external pressure but a check is necessary in respect of the longitudinal compressive loading using equation (13):

$$Q_p = \frac{0.6 \times 9.76 \times 78300}{4 \times 1519.5} = 75.44 \text{ N/mm}$$

This is greater than the required value of 40 N/mm therefore the laminate should also be suitable for the longitudinal compressive loading.

This design required that two circumferential stiffeners symmetrically disposed along the vessel and 500 mm apart be included. The second moment of area of the stiffeners would require to be determined in accordance with equation (17) in **14.4.2**.

In practice, if the vessel is to be mounted horizontally the stiffeners would probably be required at the planes of the supports and this might determine the effective length between stiffeners to be used in calculations.

**D.4 Mat and filament wound construction**

**D.4.1** Consider the cases of two alternative forms of barrel construction using continuous filament winding to reinforce an inner CSM backing to the chemical barrier. (The chemical barrier is not considered to contribute to the strength of the laminate.)

For the purpose of this example and to simplify calculation the following design data is assumed:

Circumferential loading $Q_\phi$ = 80 N/mm

Longitudinal loading $Q_x$ = 40 N/mm plus 10 N/mm compressive load

Design factor $K$ = 10

The following calculations are made.

a) *Strains*. Maximum allowable strains:

$$\text{CSM strain} = \frac{200}{10 \times 14\,000} \times 100 = 0.143\ \%$$

Continuous filaments strain

$$= \frac{500}{10 \times 28\,000} \times 100 = 0.179\ \%$$

$\therefore$ Design strain $\epsilon_d$ = 0.143 % (0.00143)

b) *Type of construction*. To be layers of 0.6 kg/m$^2$ CSM applied to the chemical barrier and helically over-wound by unidirectional filaments.

In case i) the proposed winding angle is ± 55°.

In case ii) the proposed winding angle is ± 80°.

c) *Allowable loads*

1) CSM per 0.6 kg/m$^2$ layer:

$U = 14\,000 \times 0.00143 \times 0.6 = 12.01$ N/mm

2) Unidirectional filaments per kg/m$^2$ (from **9.2.7**).

In case i) (winding angle ± 55°):

Circumferential $U_\phi = 9500 \times 0.00143 \times 0.5$ [from equation 5(a)]
$= 6.79$ N/mm

Longitudinal $U_x = 4500 \times 0.00143 \times 0.5$ [from equation 5(b)]
$= 3.22$ N/mm

In case ii) (winding angle ± 80°):

Circumferential $U_\phi = 26\,000 \times 0.00147 \times 1.0$ [from equation 5(a)]
$= 37.18$ N/mm

Longitudinal $U_x = 4400 \times 0.00143 \times 0$ [from equation 5(b)]
$= 0$

d) *Proposed construction* (above chemical barrier)

1) Two layers of 0.6 kg/m$^2$ CSM; 11 kg/m$^2$ unidirectional filament at ± 55°.

Allowable unit loads:

Circumferential $U_{LAM\,\phi} = 2 \times 12.01 + 11 \times 6.79$
$= 98.71$ N/mm (> 80)

Longitudinal $U_{LAM\,x} = 2 \times 12.01 + 11 \times 3.22$
$= 59.44$ N/mm
(> 40 (40 – 10) – 10)

hence proposed construction satisfactory.

2) Two layers of 0.6 kg/m$^2$ CSM; 3 kg/m$^2$ unidirectional filament at ± 80°.

Allowable unit loads:

Circumferential $U_{LAM\,\phi} = 2 \times 12.01 + 3 \times 37.18$
$= 135.56$ N/mm (> 80)

Longitudinal $U_{LAM\,x} = 2 \times 12.01 + 3 \times 0$
$= 24.02$ N/mm
(< 40, (40 – 10) – 10)

therefore greater axial strength required.

Revise construction to:

Four layers of 0.6 kg/m$^2$ CSM;

3 kg/m$^2$ unidirectional filament at ± 80°.

Allowable unit loads:

Circumferential $U_{LAM\,\phi} = 4 \times 12.01 + 3 \times 37.18$
$= 159.58$ N/mm (> 80)

Longitudinal $U_{LAM\,x} = 4 \times 12.01 + 3 \times 0$
$= 48.04$ N/mm
(> 40, (40 – 10) – 10)

This appears satisfactory but as the winding angle is greater than 75° it is necessary to check that the longitudinal strain does not exceed 0.1 % (see **9.2.7**).

From equation (12) in **14.3.3** and using Table 5 and Figure 3, as appropriate.

In longitudinal direction:

$X_{LAM\,x} = 4 \times 0.6 \times 14\,000 + 3.0 \times 1.0 \times 4400$
$= 33\,600 + 13\,200$
$= 46\,800$ N/mm

$$\text{Longitudinal strain} = \frac{40}{46800} = 0.000855$$
$$= 0.0855\,\% < 0.1\,\%$$

which is satisfactory

If, however, the circumferential and longitudinal loads are applied simultaneously then, from **9.2.7**, the maximum allowable loads may be less than the foregoing values calculated for individual unit loads.

This has to be checked by means of a biaxial design envelope the four quadrants of which can be constructed from values of $U_{LAM\phi}$) and $U_{LAM\,x}$ (positive for tension and negative for compression, as appropriate) as follows.

If $Q_\phi > 0$ and $Q_x > 0$ (i.e. both tensile).

Then $U_\phi = U_{LAM\,\phi}$ and $U_x = U_{LAM\,x}$.

If $Q_\phi < 0$ and $Q_x < 0$ (i.e. both compressive).

Then $U_\phi = U_{LAM\,\phi}$ and $U_x = U_{LAM\,x}$.

If $Q_\phi > 0$ and $Q_x < 0$

or

$Q_\phi < 0$ and $Q_x > 0$

(i.e. $Q_\phi$ and $Q_x$ of opposite sign)

Then, for a given required value of $U_x \leqslant U_{LAM\,x}$ the maximum allowable value for $U_\phi$ is equal to:

$$U_{LAM\,\phi}\left(1 - \frac{U_x}{U_{LAM\,x}}\right)$$

and, for a given required value of $U_\phi \leqslant U_{LAM\phi}$ the maximum allowable value for $U_x$ is equal to:

$$U_{LAM\,x}\left(1 - \frac{U_\phi}{U_{LAM\,\phi}}\right)$$

where $U_\phi$ and $U_x$ are the maximum allowable simultaneous circumferential and longitudinal unit loads under biaxial loadings.

If these are less than the required design values the laminate has to be redesigned.

The biaxial design envelopes for the above example are shown in Figure 39, indicating the required coincident loads:

$Q_\phi = 80$ N/mm and $Q_x = 40$ N/mm, point A

$Q_\phi = 80$ N/mm and $Q_x = (40 - 10) = 30$ N/mm, point B

$Q_\phi = 80$ N/mm and $Q_x = -10$ N/mm, point C

each of which falls within the boundaries of envelopes.

**D.4.2** Consider constructions as in **D.4.1** applied to a static head tank where:

$Q_\phi = 80$ N/mm (circumferential loading due to static head of content)

$Q_{x1} = \pm 18$ N/mm (longitudinal loading due to bending under wind pressure)

$Q_{x2} = -2$ N/mm (longitudinal loading due to dead weight of tank barrel and crown)

The effective compressive longitudinal loading $Q_x$ is thus the algebraic sum of $Q_{x1}$ and $Q_{x2}$:

$Q_x = (-18 - 2) = -20$ N/mm

The design envelopes of **D.4.1** apply and reference to Figure 39 shows that for the value of $Q_\phi = 80$ N/mm, case i) permits an approximate maximum allowable compressive value for $Q_x$ of little more than – 11 N/mm, which is inadequate, whereas case ii) allows almost – 24 N/mm which is satisfactory for the required coincident values of $Q_\phi = 80$ N/mm and $Q_x = (-18 - 2) = -20$ N/mm (point D).

These maximum permitted values for compressive loading would be calculated from:

$$U_x = U_{LAM\,\phi}\left(1 - \frac{U_\phi}{U_{LAM\,x}}\right)$$

using the values appropriate to constructions i) and ii) respectively (from **D.4.1**):

a) Construction i) (windings at ± 55°).

$$U_x = -59.44\left(1 - \frac{80}{98.71}\right)$$

$$= -11.16 \text{ N/mm } (< -20 \text{ N/mm})$$

Inadequate.

b) Construction ii) (windings at ± 80°).

$$U_x = -48.04\left(1 - \frac{80}{159.58}\right)$$

$$= -23.96 \text{ N/mm } (> -20 \text{ N/mm})$$

Satisfactory.

## D.5 Branch design

### D.5.1 *General*

The design conditions are taken from **D.2**. It is proposed to fit a 150 mm branch in the cylindrical portion of the vessel. It is assumed that the branch will be subject only to the design pressure of 0.2 N/mm$^2$.

Branch calculations are mainly of two separate parts. The first part sizes the branch laminate and the second part sizes the attachment or overlay laminate.

### D.5.2 *Branch laminate calculation*

From equation (7) in **14.2** the circumferential unit loading will be:

$$Q_\phi = \frac{0.2 \times 150}{2} = 15.0 \text{ N/mm}$$

As pipework and fittings are supported such that they impose no excessive loads on the vessel branches, it is assumed for this example that $Q_x$ will not exceed $Q_\phi$.

As in **D.2.2** the value of $K = 10.89$ and $u_z = 18.36$ N/mm per kg/m$^2$ glass.

Assuming an all-CSM construction and using the simpler form of equation (6) (as in **D.2.2**):

$Q = u_z\,(m_z\,n_z)$

Total mass $(m_z\,n_z) = 15/18.36 = 0.82$ kg/m$^2$.

A total mass of glass to satisfy the pressure requirements would thus be 0.9 kg/m$^2$. In practice, however, it might be considered preferable to make the branch with a minimum of 1.2 kg/m$^2$ and the final thickness including overlay laminates, etc. should be not less than the minimum thickness required by **13.3**.

### D.5.3 *Thickness calculation*

Assume a glass content of 30 % for CSM. From Figure 5 the thickness would be:

$2.5 \times 1.2 = 3.0$ mm,

which may require to be increased as detailed in **D.5.2**.

**D.5.4** ***Design of compensation*** (from **20.2.3**)

The compensating laminate shall have a load capacity $A_L$.

$A_L = d_c \times U_c$

As in this example there is a 150 mm branch in the cylinder of the vessel of **D.2**:

$d_c$ = 160 mm approx. (for set-in branch)

$U_c = Q_\phi$ (of **D.2**) = 175 N/mm

$A_L = 160 \times 175$

= 28 000 N

This has to be resisted by $A_c$, the effective load capacity of the compensation:

$A_c = (d_r - d_c) \times U_{OVL}$

$d_r$ will be chosen from the detail branch construction as shown in Figure 20.

For this example assume a value of $d_c$ of 160 mm and choose a value of $d_r$ as required by **20.2.4**.

Assume $d_r = 2.5d_b$ = 375 mm:

$$U_{OVL} = \frac{A_c}{(d_r - d_c)}$$

$$= \frac{28\ 000}{375 - 160}$$

$$= 130.2 \text{ N/mm}$$

Assuming an all-CSM construction as used for the shell in **D.2**:

$U_{OVL} = U_z\ (m_z\ n_z)$

So that total mass $(m_z\ n_z)$ is equal to:

$$\frac{130.2}{18.36} = 7.09 \text{ kg/m}^2$$

One convenient construction would consist of 12 layers of 0.6 kg/m² CSM and resin-rich surface layer with binding tissue.

Check the lap shear area is adequate to transfer shell loads in shear:

$$A_s = \frac{\text{lap shear strength}}{F} \times 0.4 \times (d_r^2 - d_c^2)$$

From Table 5 the lap shear strength for CSM layers with resins other than furane = 7 N/mm²:

$F$ = the design factor $K$ = 10.89

$$A_s = \frac{7 \times 0.4}{10.89} \times (375^2 - 160^2)$$

$$= 29575 \text{ N}$$

This is greater than the required $A_L$ of 28 000 N.

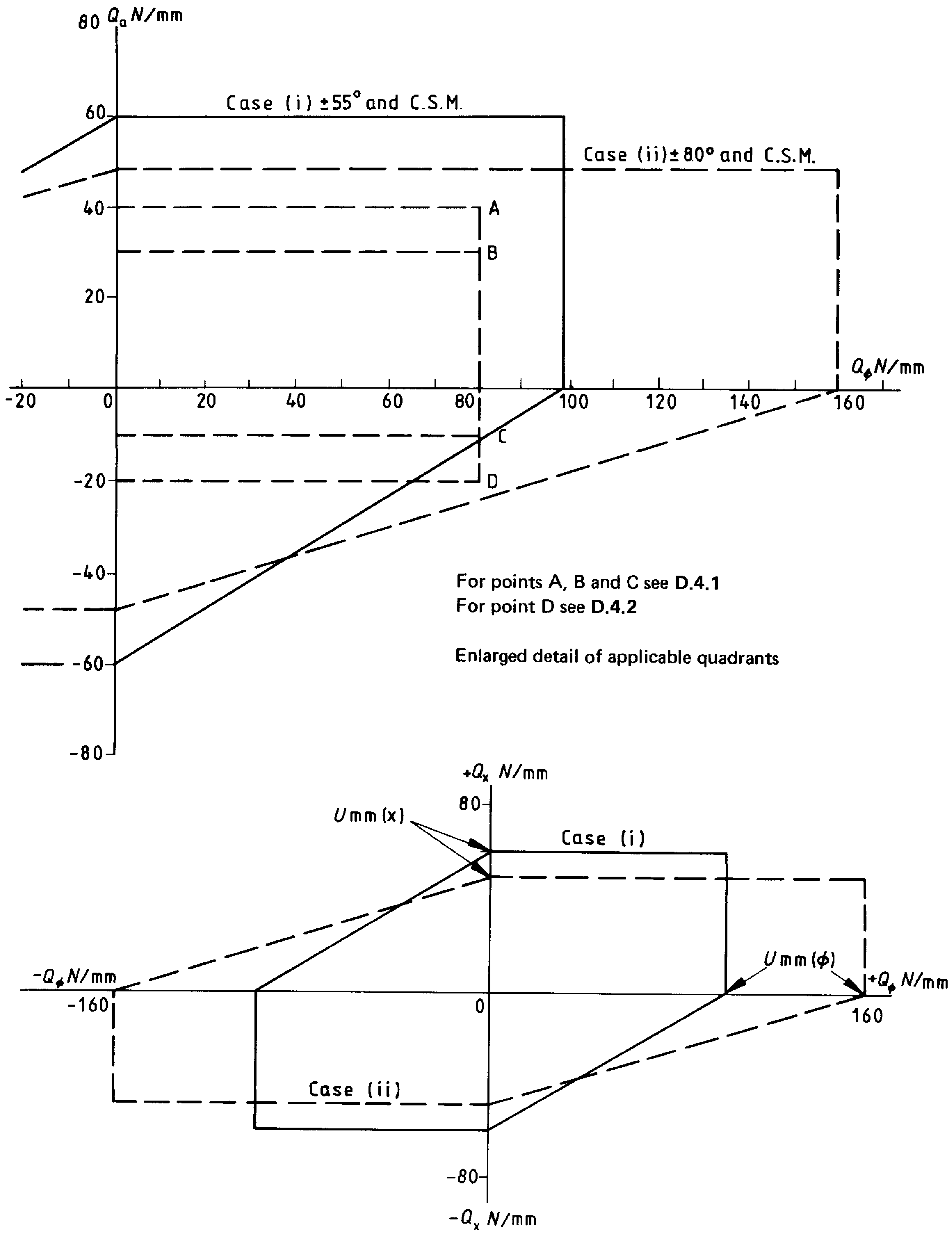



**Figure 39 — Biaxial design envelopes**

Alternatively, find the effect of interleaving five layers of the 0.6 kg/m$^2$ CSM compensation into the main shell laminate layers.

From these calculations:

$A_c$ = (375 – 160) × 12 × 0.6 × 18.36 = 28 421 N

$A_{ci}$ = (375 – 160) × 5 × 0.6 × 18.36 = 11 842 N

$A_{ce}$ = (375 – 160) × 7 × 0.6 × 18.36 = 16 579 N

$A_s$ = 29 575 > $A_{ce}$

From **20.2.3** d), where layers of compensation are interleaved with the main shell laminate layers it is required that $A_s$ be not less than $A_{ce}$. This condition is complied with.

The required minimum of 1.2 kg/m$^2$ CSM plus a reinforced gel coat (see **20.2.3**) is applied to the inside of the branch and vessel and the outer 1.5 kg/m$^2$ CSM of the compensation ($S_l$) is overlaid on the outside of the branch wall:

$Q$ for outer overlay $= u_z\,(m_z\,n_z)$

$= 18.36 \times 1.5$

$= 27.5$ N/mm

Pull out load from **20.2.6** [equation (62)]:

$$Q_p = \frac{p d_b}{4}$$

$$Q_p = \frac{0.2 \times 150}{4} = 7.5 \text{ N/mm}$$

As the overlay laminate is designed to withstand 27.5 N/mm it will be adequate.

Because the pull out load may be resisted by the internal overlay the external overlay is not necessarily subject to the full peel load.

Maximum unit loading which can be applied to the external overlay will be:

$m \times u_z$ = load carrying capacity of external overlay

1.5 × 18.36 = 27.54 N/mm

where

$m$ is the total mass in the external overlay;

$u_z$ is the design unit loading of CSM.

The minimum length over which the overlay has to be applied to carry the interlaminar shear loads will be $L_o$ or 75 mm, whichever is the greater, where:

$$L_o = \frac{F \times Q_p}{\text{Lap shear strength}}$$

and where $F$ is taken as $K$ or $K_{OVL}$

(the lap shear strength is taken as 7 N/mm$^2$, from Table 5, for this example)

$$L_o = \frac{10.89 \times 7.5}{7.0} = 11.7 \text{ mm}$$

Thus, the required overlay length will be 75 mm.

### D.6 Loaded rectangular panel

A rectangular panel 1 000 mm wide and 4 000 mm long is required to withstand a distributed load of 0.000 75 N/mm or a live load of 900 N on a 150 m radius circular area. The edges of the panel can be assumed to have type 1 edge conditions.

Calculate the thickness of a solid panel using only deflection criteria.

For distributed load:

$$t = \left(\frac{\alpha_1 p b^4}{E_{LAM}}\right)^{0.25} \quad \text{(second term zero)}$$

$$= \left(\frac{0.0284 \times 0.000\ 75 \times 1000^4}{5772}\right)^{0.25}$$

$$= 7.79 \text{ mm}$$

where $a/b = 4$. From Table 8, $\alpha_1 = 0.0284$.

For local load:

$$t = \left(\frac{\alpha_2 W_l b^2}{E_{LAM}}\right)^{0.25} \quad \text{(first term zero)}$$

$$= \left(\frac{0.0791 \times 900 \times 1000^2}{5772}\right)^{0.25}$$

$$= 10.54 \text{ mm}$$

where $a/b = 4$. From Table 9, $\alpha_2 = 0.0791$.

The larger thickness will be used for determining the sandwich construction. Assume a 12 mm thick core.

From equation (39) in **17.8.3**:

$$t_t = (t^3 + c^3)^{0.33}$$

$$= (10.54^3 + 12^3)^{0.33}$$

$$= 13.88 \text{ mm}$$

From equation (40):

$$t_s = \frac{(t_t - c)}{2}$$

$$= \frac{13.88 - 12}{2}$$

$$= 0.94 \text{ mm}$$

In order to comply with **17.8.2** a) a minimum thickness of 3 mm is required.

The number of layers in the skin under consideration, $n_{CSM}$ is given by:

$$n_{CSM} = \frac{t_s}{t_g m_{CSM}} = \frac{3}{2.3 \times 0.45} = 2.9$$

$$= 3 \text{ layers of } 0.45 \text{ kg/m}^2 \text{ CSM}$$

In order to check that the design unit loading is not exceeded the bending moment has to be determined for each case.

For distributed load:

$$M = \beta_1 p b^2$$

$$= 0.083 \times 0.000\ 75 \times 1000^2$$

$$= 62.25 \text{ N·mm/mm}$$

For local load:

$$M_l = \frac{W_l}{4\pi}\left(1.3 \ln\left(\frac{2b}{\pi r}\right) + \beta_2\right)$$

$$M_l = \frac{900}{4\pi}\left(1.3 \ln\left(\frac{2 \times 1000}{\pi \times 150}\right) + 0.067\right)$$

$$= 139.4 \text{ N·mm/mm}$$

$$M_l = \beta_3 W_l$$

$$= 0.168 \times 900$$

$$= 151.2 \text{ N·mm/mm}$$

Use the largest moment for calculating $Q_s$

From equation (41) in **17.8.3**:

$$Q_s = \frac{M_d t_t}{d^2}$$
$$= \frac{151.2 \times 18}{15^2}$$
$$= 12.1 \text{ N/mm}$$

For three layers of 0.45 kg/m² CSM (taking nominal values):

$$Q = u_{CSM} m_{CSM} n_{CSM}$$
$$= 18.03 \times 0.45 \times 3$$
$$= 24.3 \text{ N/mm}$$

Check shear stress in core for distributed load:

$$S = \frac{pab}{2a + 2b} = \frac{\text{pressure load}}{\text{periphery}}$$
$$= \frac{0.000\,75 \times 1000 \times 4000}{(2 \times 1000) + (2 \times 4000)}$$
$$= 0.3 \text{ N/mm}$$

For local load:

$$S = \frac{W_l}{2\pi r} = \frac{\text{local load}}{\text{circumference of circle of application}}$$
$$= \frac{900}{2\pi 150}$$
$$= 0.955 \text{ N/mm}$$

From equation (42):

$$\tau = \frac{S}{d}$$
$$= \frac{0.955}{15}$$
$$= 0.0636 \text{ N/mm}^2$$

Use a core with a minimum ultimate shear stress of 0.255 N/mm² to give the minimum factor of **4** (see **17.8.3**).

**D.7 Stiffener requirements for vertical multi-straked tank**

A vertical cylindrical tank with a fully supported flat bottom and a shallow conical roof is required to also withstand a vacuum of 6 mbar and an external wind pressure. The tank is 4 m inside diameter and 10 m high over parallel shell. Find the required number(s) and position(s) of stiffeners to resist collapse using the calculations of clause **15**.

Take dynamic wind pressure $q$ = 1 000 N/m².

[This would be calculated from design wind speed and coefficients of CP 3:Chapter V-2 (see also clause **15**).]

Design partial vacuum = 6 mbar = 600 N/m².

Design external pressure

= Dynamic wind pressure + design partial vacuum

= 1 000 + 6 000

= 1 600 N/m²

Tank has eight strakes of varying heights, thickness and laminate construction. Using these characteristics first calculate the equivalent height, $h_e$, as related to the thickness of the top strake, for the other seven strakes using equation (21).

Construct a table as follows (using appropriate values for the proposed construction).

| Strake no. | Strake height, $h$ | Overall unit modulus $X_{LAM}$ | Strake thickness, $t_h$ | Equivalent height, $h_e$ |
|---|---|---|---|---|
| | mm | N/mm | mm | mm |
| 1 (top) | 850 | 51 546 | 7.81 ($t_{min}$) | 850.0 |
| 2 | 1 700 | 72 612 | 10.11 | 819.4 |
| 3 | 1 700 | 93 678 | 12.40 | 467.6 |
| 4 | 1 700 | 114 744 | 14.69 | 296.0 |
| 5 | 1 700 | 135 810 | 16.99 | 201.1 |
| 6 | 850 | 156 876 | 19.28 | 72.0 |
| 7 | 850 | 164 496 | 20.66 | 61.9 |
| 8 (bottom) | 650 | 177 942 | 21.57 | 41.0 |

i.e. $$h_{e2} = 1700 \left(\frac{51\,546}{72\,612}\right)\left(\frac{7.81}{10.11}\right)^{1.5} = 819.4\text{ mm}$$

$$h_{e3} = 1700 \left(\frac{51\,546}{93\,678}\right)\left(\frac{7.81}{12.4}\right)^{1.5} = 467.6\text{ mm}$$

and so on for remaining strakes.

$H_e$ is the sum of all the values for $h_e$ and equals 2 809 mm.

Maximum permitted height between stiffeners, $H_p$, at thickness, $t_{min}$, is given by:

$$H_p = \frac{X_{LAM\,min}\,t_{min}^{1.5}}{0.4FpD_o^{1.5}}$$

$$= \frac{51\,546 \times 7.81^{1.5}}{0.4 \times 4.0 \times 1600 \times 10^{-6} \times 4015.6^{1.5}}$$

$$= 1727\text{ mm}$$

$H_e/H_p = 2809/1727 = 1.627$

i.e. $H_e > H_p$ and one stiffener is required, for example, at $H = H_e/2 = 1\,405$ mm from tank top.

By inspection of column 5 of the table it is obvious that this point is in the second strake from the top.

To find the equivalent distance from top of second course, $h_{ie}$, $\Sigma h_e$ for top course is 850 mm and $h_{ie} = 1405 - 850 = 555$ mm.

The true distance from the top of stiffened strake:

$$h_i = h_{ie} \times \frac{X_{LAM\,n}}{X_{LAM\,min}} \times \left(\frac{t_h}{t_{min}}\right)^{1.5}$$

$$= 555 \left(\frac{72\,612}{51\,546}\right)\left(\frac{10.11}{7.81}\right)^{1.5}$$

$$= 1151.5\text{ mm}$$

The true distance from stiffener to top of tank from equation (23):

$H_A = h_i + \Sigma h$

$= 1151.5 + 850$

$= 2001.5$ mm

The proportions of the intermediate stiffener should now be determined to comply with **14.4.2**, i.e. the minimum required second moment of area for the stiffener should be calculated from equation (17) using:

in this case $L = H = \left(\frac{H_e}{2}\right) = 1405$ mm

and

$$E_{LAM} = \frac{X_{LAM\,min}}{t_{min}} = \frac{51546}{7.81} = 6600 \text{ N/mm}^2$$

A check should be made of the primary stiffening at the upper edge of the top course, either external stiffener or inherent stiffness, again using equation (17), but in this case $L$ will equal $H/2$ = 702.5 mm and the value of $E_{LAM}$ will be unchanged. As the tank is subjected to axial compressive loading due to bending under wind pressure plus self weight and possibly snow loading, further checks using equation (13) in **14.3.3** are necessary to ensure that the laminate of each strake is capable of taking the axial load imposed at that level.

## D.8 Design of conical end with fabric-backed thermoplastics liner

Consider the vessel detailed in **D.2** of all-CSM construction, but with fabric-backed thermoplastics liner, for which a conical end is to be provided.

| | |
|---|---|
| Inside diameter of shell | 1 750 mm. |
| Design pressure | 2 bar. |

It is desired to construct the cone without a transition knuckle at the shell/cone intersection as the configuration is within the permitted parameters of **18.4.1**.

Assume a glass content of 30 % by mass and a resin of relative density 1.1. From Figure 5 the anticipated thickness of laminate is 2.5 mm per kg/m$^2$ glass.

From **D.2** the design unit loading $u_z$ is 18.36 N/mm per kg/m$^2$ glass.

The shell circumferential load $Q$ is 175 N/mm, proposed laminate 9.6 kg/m$^2$ CSM with a thickness, $t$, of 24 mm.

The design of the cone and the shell/cone intersection are required to comply with equations (47) and (49) in **18.4.3**.

From equation (47) the unit loading in the hoop direction has to be not less than:

$$Q = \frac{0.5 \times 0.2 \times 1750}{\cos 45} = 247.49 \text{ N/mm}$$

Check to see if this satisfies equation (49) for the minimum required unit load in the axial direction.

For this unit load of 247.49 N/mm the required mass of CSM is:

$$m_{CSM} = \frac{247.49}{18.36} = 13.48 \text{ kg/m}^2$$

The thickness appropriate to this glass mass is:

$t_c = 13.48 \times 2.5 = 33.7$ mm

The ratio of thickness to inside diameter is thus:

$$\frac{t_c}{D_i} = \frac{33.7}{1750} = 0.0193$$

By interpolation from Table 13:

$K_{c2} = 3.05$

Substituting this value in equation (49) gives:

$Q_{x2} = 0.5 \times 0.2 \times 1750 \times 3.05 = 533.75$ N/mm

As this is greater than $Q$ calculated from equation (47) some increase in thickness and unit load is necessary at the shell/cone intersection and further iterative calculation is required.

*First trial*

Assume $K_{c2}$ = 2.75 and recalculate equation (49):

$Q_{x2} = 0.5 \times 0.2 \times 1750 \times 2.75 = 481.25$ N/mm

For this unit load the required glass mass is:

$$m_{CSM} = \frac{481.25}{18.36} = 26.21 \text{ kg/m}^2$$

The thickness appropriate to this glass mass is:

$t_c = 26.21 \times 2.5 = 65.53$ mm

The ratio of thickness to inside diameter becomes:

$$\frac{t_c}{D_i} = \frac{65.53}{1750} = 0.0374$$

By interpolation from Table 13:

$K_{c2}$ = 2.5 (assumed 2.75)

This suggests that the assumed value of 2.75 was too high and a second trial is necessary.

*Second trial*

Assume $K_{c2}$ = 2.5 and recalculate equation (49):

$Q_{x2} = 0.5 \times 0.2 \times 1750 \times 2.5 = 437.5$ N/mm

For this unit load the required glass mass is:

$$m_{CSM} = \frac{437.5}{18.36} = 23.83 \text{ kg/m}^2$$

The thickness appropriate to this glass mass is:

$t_c = 23.83 \times 2.5 = 59.58$ mm

The ratio of thickness to inside diameter becomes:

$$\frac{t_c}{D_i} = \frac{59.58}{1750} = 0.034$$

By interpolation from Table 13:

$K_{c2}$ = 2.4 (assumed 2.5)

This suggests that a further small reduction could be made to obtain optimum value for $K_{c2}$ therefore make a further trial using a value of 2.45.

*Third trial*

Assume $K_{c2}$ = 2.45 and recalculate equation (49):

$Q_{x2} = 0.5 \times 0.2 \times 1750 \times 2.45 = 428.75$ N/mm

The required glass mass becomes:

$$m_{CSM} = \frac{428.75}{18.36} = 23.35 \text{ kg/m}^2$$

The thickness appropriate to this glass mass is:

$$t_c = 23.35 \times 2.5 = 58.38 \text{ mm}$$

$$\frac{t_c}{D_i} = \frac{58.38}{1750} = 0.0334$$

By interpolation from Table 13:

$K_{c2}$ is 2.45.

There is thus close agreement between the assumed and the calculated values for $K_{c2}$ and the required value can be taken as 2.45 from the preceding calculation.

$Q_{x2}$ = 428.75 N/mm

The required mass of CSM is 23.35 kg/m$^2$ with a thickness of 58.38 mm. This has to extend on to the shell and on to the cone by at least the distance, $L_c$, determined by equation (50):

$$L_c = \left(\frac{1750 \times 58.38}{\cos 45}\right)^{0.5} = 380 \text{ mm}$$

Beyond this distance and its associated tapers the laminate is not to be less than that calculated for the cylindrical shell and cone, respectively.

**D.9 Design of conical end of mixed laminate construction with fabric-backed thermoplastics liner**

For a mixed laminate construction the same process would be followed as in **D.8** but it should be noted that the reinforcement is required to be distributed through the laminate in a balanced manner (see **18.4.1**).

A first estimate of the required value for $K_{c2}$ can be derived from Figure 40 but its accuracy should be confirmed by calculation as in **D.8**.

## Appendix E Selection procedure for factor $k_2$

**E.1 General**

The factor $k_2$ is related to the loss in strength of an unstressed GRP laminate when exposed to the process conditions for the design lifetime of the vessel, in cases where this data is directly available, then method A (see **E.2**) may be used. In the absence of such data, then reference should be made to the resin manufacturer's chemical resistance tables for the selected resin in conjunction with method B (see **E.3**).

**E.2 Method A**

If loss in strength is $\leqslant$ 20 % of original ultimate tensile strength then use $k_2$ = 1.2.

If loss in strength is > 20 % and $\leqslant$ 50 % then interpolate between

$k_2$ = 1.2 for 20 % strength loss; and

$k_2$ = 2.0 for 50 % strength loss.

If loss in strength is > 50 % then material is unsuitable.

**E.3 Method B**

A full description of a method specific to GRP is given in ASTM C 581. A more general approach covering all plastics is given in BS 4618. The latter does not cover details of GRP sample preparation contained in ASTM C 581.

**E.4 Examples**

*Example A.* Select $k_2$ for 100 % carbon tetrachloride at 20 °C.

Since this is the most severe duty possible, $k_2$ = 2.0.

*Example B.* Select $k_2$ for aqueous calcium bisulphite at 60 °C.

For solution at 80 °C, $k_2$ = 2.0

For solution at 20 °C, $k_2$ = 1.2

For solution at 60 °C, $k_2 = 1.2 + 0.8\ \dfrac{(60-20)}{(80-20)}$

= 1.73

*Example C.* Select $k_2$ for 10 % sulphuric acid at 80 °C.

For 10 % acid at 100 °C, $k_2$ = 2.0

For 10 % acid at 20 °C, $k_2$ = 1.2

For 10 % acid at 80 °C, $k_2 = 1.2 + 0.8 \times \dfrac{60}{80} = 1.8$

*Example D.* Select $k_2$ for 70 % sulphuric acid at 60 °C.

For 70 % acid at 70 °C, $k_2$ = 2.0

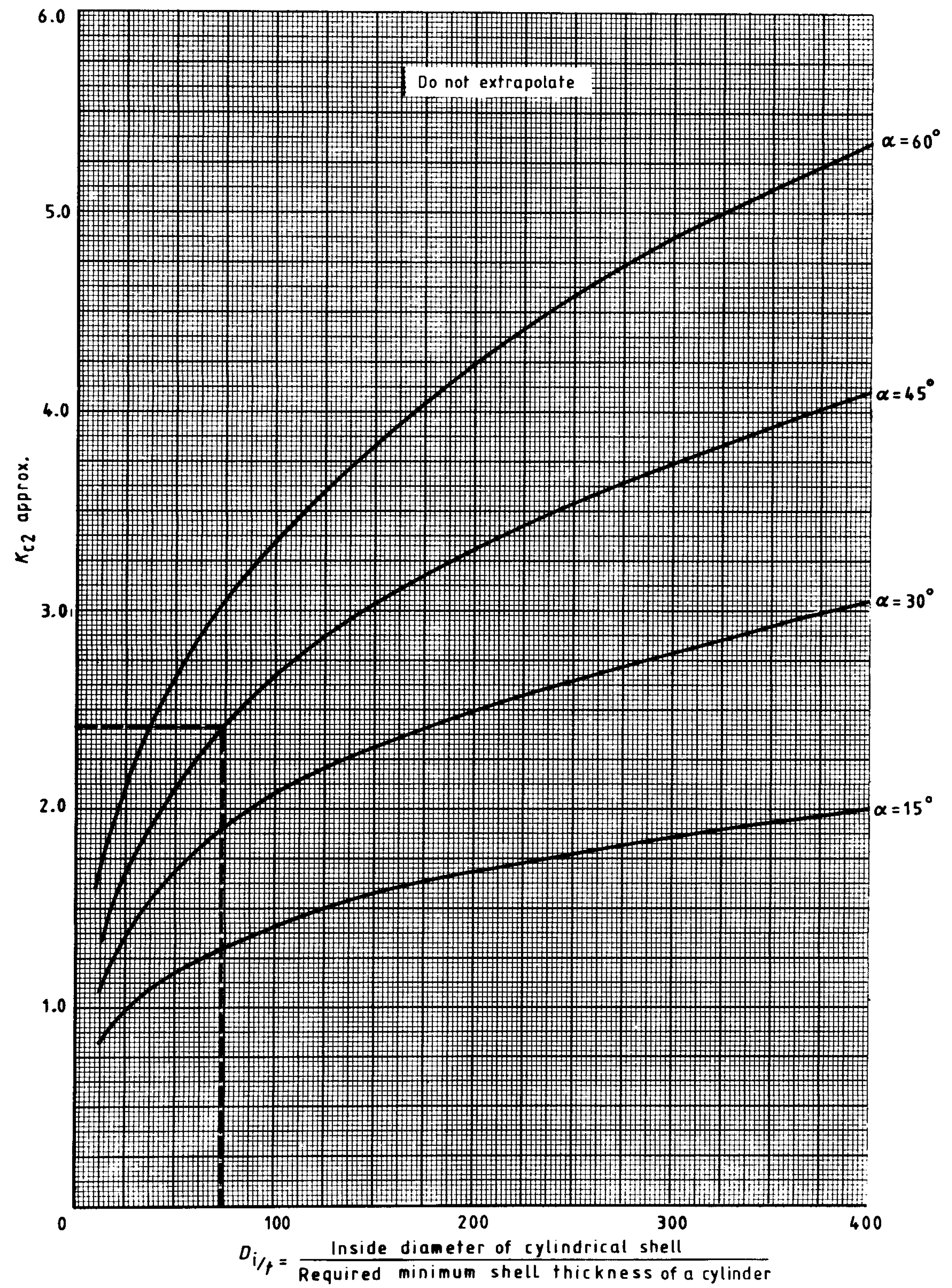



**Figure 40 — Approximate values of $K_{c2}$ [for first trial in equation (49)]**

A judgement now has to be taken on whether 70 % acid at 20 °C can be associated with $k_2$ = 1.2 or whether a higher value should be chosen. In the absence of manufacturer's data for this application assume that a higher value should be assigned, again using an interpolation technique.

For 70 % acid at 20 °C:

$$k_2 = 1.2 + 0.8\ \frac{(100 - 70)}{(100 - 20)}$$

$k_2$ = 1.5

Hence for 70 % acid at 60 °C interpolate between 1.5 and 2.0:

$$k_2 = 1.5 + 0.5\ \frac{(60 - 20)}{(70 - 20)}$$

$k_2$ = 1.9

This procedure is illustrated in Figure 41 and it may often be preferable to estimate $k_2$ values graphically in this way rather than by direct calculation.

*Example E.* Select $k_2$ for 50 % sulphuric acid at 50 °C.

A judgement should be made on whether it is allowable to interpolate for 50 % acid between the published data for 10 % and 70 % acid. In this example it is permissible to do so but note that this may not always be the case, e.g. if the mechanism of chemical attack changes from acidic hydrolysis to oxydizing acidic.

The interpolation is shown in Figure 42 and the value of $k_2$ can be read off:

$k_2$ = 1.7

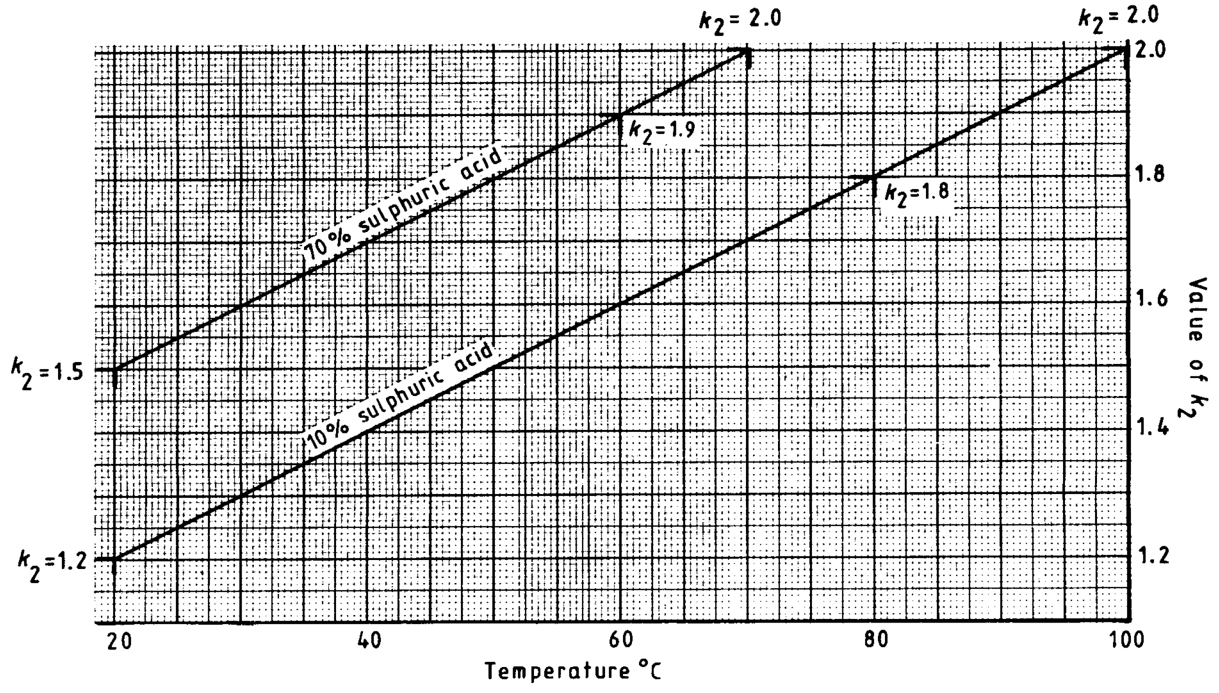



**Figure 41 — Graphical determination of $k_2$: example D**

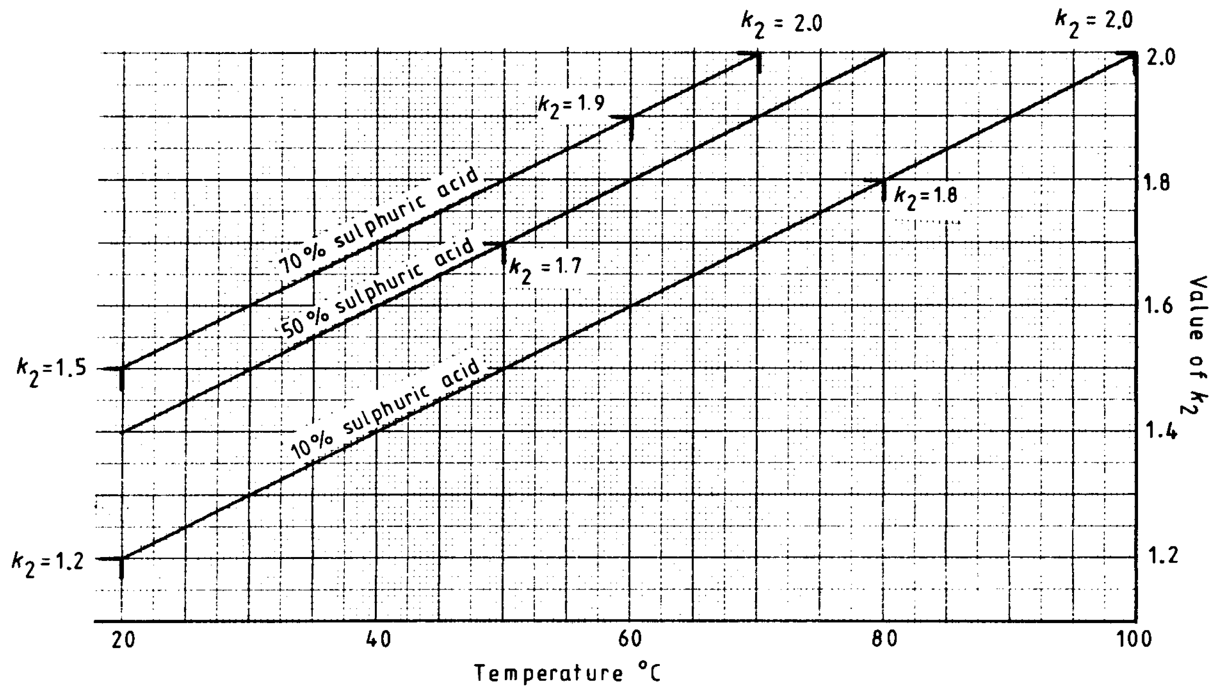



**Figure 42 — Graphical determination of $k_2$: example E**

## Appendix F Design calculation for tank and vessel anchorage

NOTE This appendix applies only to vertical cylindrical tanks or vessels where there may be a tendency for the base to lift off its support.

**F.1** The design uplift on the anchorage shall take into account the following conditions:

a) hydrostatic head and/or internal pressure;

b) wind overturning pressure;

c) seismic force, not in combination with b), where applicable;

d) weight of cylindrical shell, roof and associated structure;

e) weight of shell and roof insulation where fitted;

f) test pressure (see also note to **39.2**).

**F.2** The overturning force due to wind loading shall be calculated in accordance with CP 3:Chapter V-2.

**F.3** The anchorage shall be designed to withstand the maximum overturning moment due to wind loads and other causes. The number and size of anchorage bolts shall be calculated by taking the total resisting moment as equal to the maximum overturning moment.

NOTE It is recommended that the number of anchorage points be a multiple of four, that they be evenly spaced around the diameter and that the spacing should not exceed 1 500 mm.

If the contribution to the resisting moments of the tank weight is ignored and if only those bolts on the windward side of the cylinder are assumed to act at the base, then the load per anchor bolt, $F$ (in N), is given by:

$$F = \left( \frac{4M}{D_{pc}} - W + \frac{p \pi D_i^2}{4} \right) \frac{1}{N_b} \qquad (77)$$

where

$M$ is the total overturning moment (in N/mm);
$D_i$ is the internal diameter of vessel (in mm);
$D_{pc}$ is the pitch circle diameter of anchorage bolts (in mm);
$W$ is the minimum weight of vessel shell (in N);
$N_b$ is the number of anchorage bolts;
$p$ is either:

for vessels and tanks with fully supported flat bottoms: $p$ = the internal pressure at the top of the shell cylinder (in N/mm$^2$) (see also **39.2**);

for other vessels: $p = 0$.

NOTE If $N_b < 8$ use above equation with $W = 0$.

**F.4** The design temperature of the anchorage and anchorage attachments shall be the design temperature of the tank. Heat transfer from hot parts shall be such that unacceptable characteristics do not develop which may lead to failure of the anchorage or its bond to the shell of the tank.

## Appendix G Stresses from local loads, etc: recommended methods of calculation

### G.1 General

This appendix is based on Appendix G of BS 5500:1985 and has been modified to take account of the difference in properties between GRP and metals. The method should be used up to the points of establishing the unit bending moments $M_\phi$ and $M_x$, and the unit membrane forces $N_\phi$ and $N_x$ to evaluate the laminate strain. The criterion of acceptance for a GRP load system is that of maximum strain. The evaluation is carried out in accordance with section 3. In no case should the principle of "shake down" be applied to a GRP vessel or tank.

This appendix is in accordance with reference [17], and it deals with methods of calculating stresses due to local attachments on pressure vessels in some common cases.

The information in reference [18] may be useful in simplifying the approach to local load calculations.

The following figures are reproduced by courtesy of the American Welding Research Council.

Figure 72 was originally published as Figure 2 on page 12 of WRC Bulletin 90 September 1963.

Figure 73 was originally published as Figure 3 on page 14 of WRC Bulletin 90 September 1963.

Figure 74 was originally published as Figure 7 on page 23 of WRC Bulletin 90 September 1963.

Figure 75 was originally published as Figure 8 on page 24 of WRC Bulletin 90 September 1963.

Figure 76 was originally published as Figure 9 on page 31 of WRC Bulletin 90 September 1963.

Figure 77 was originally published as Figure 10 on page 33 of WRC Bulletin 90 September 1963.

Figure 78 was originally published as Figure 11 on page 41 of WRC Bulletin 90 September 1963.

Figure 79 was originally published as Figure 12 on page 42 of WRC Bulletin 90 September 1963.

### G.2 Local loads on pressure vessel shells

#### G.2.1 *General*

This appendix is concerned with the effect on the shell of a pressure vessel of local forces and moments which may come from supports, equipment supported from the vessel, or from thrusts from pipework connected to branches. Limits on vessel/attachment geometry, without which the methods given may be unreliable, are also stated.

Stresses due to local loads and moments applied to cylindrical shells through attachments, including nozzles, are dealt with in **G.2.2** and **G.2.3**.

The methods in **G.2.2** cover the determination of stresses at the edge of the loaded areas (**G.2.2.2**), stresses away from the edge of the loaded area (**G.2.2.3**) and deflections in a cylindrical shell due to the application of radial load (**G.2.2.4**).

Details are given in **G.2.3** of how to treat circumferential moments (**G.2.3.2**) and longitudinal moments (**G.2.3.3**) in order to determine the maximum stresses at the outer edge of the actual loaded area (**G.2.3.4**) and the rotation of the attachment due to the application of these moments (**G.2.3.5**) to a cylindrical shell.

Stresses due to local loads and moments applied to spherical shells through attachments including nozzles are dealt with in **G.2.4** and **G.2.5**.

A method is given in **G.2.4** for calculating stresses and deflections due to radial loads (**G.2.4.3**) and stresses and deflections and slopes due to an external moment (**G.2.4.4**) when applied to a spherical shell. **G.2.5** deals with the method of calculating stresses arising at a nozzle/shell junction due to application of pressure, external load and external moment to a spherical shell. The method is based on the analysis given in reference [19].

The data are presented in the form of charts in terms of non-dimensional functions of the variables so that any convenient system of consistent units may be used. The following symbols represent the notation used in this appendix and recommended for general use.

*Notation*

| | |
|---|---|
| $C$ | half length of side of square loading area (in mm) |
| $C_1$ | half side of equivalent square loading area (in mm) |
| $C_x$ | half length of rectangular loading area in longitudinal direction (in mm) |
| $C_\phi$ | half length of rectangular loading area in circumferential direction (in mm) |
| $d$ | distance from centre of applied load to mid-length of vessel (in mm) |
| $E$ | modulus of elasticity (in N/mm$^2$) |
| $f_x$ | resultant longitudinal stress (in N/mm$^2$) |
| $f_\phi$ | resultant circumferential stress (in N/mm$^2$) |
| $i$ | rotation of a fitting by an external moment (in radians) |
| $C_z$ | longitudinal length of loading area for an external longitudinal moment (see Figure 64) (in mm) |
| $C_\theta$ | circumferential length of loading area for an external circumferential moment (see Figure 63) (in mm) |
| $L$ | length of cylindrical part of shell (in mm) |
| $L_e$ | equivalent length of shell (in mm) |
| $M$ | external moment applied to a branch or fitting (in N/mm) |
| $M_x$ | longitudinal or meridional bending moment per unit circumference (in N·mm/mm) |
| $M_\phi$ | circumferential bending moment per unit length (in N·mm/mm) |
| $N_x$ | longitudinal membrane force per unit circumference (in N/mm) |
| $N_\phi$ | circumferential membrane force per unit length (in N/mm) |
| $r$ | mean radius of cylinder or sphere (in mm) |
| $r_o$ | mean radius of branch (in mm) |
| $t$ | wall thickness of shell (in mm) |
| $W$ | external load distributed over loading area (in N) |
| $x$ | longitudinal distance of a point in vessel wall from centre of a loading area (in mm) |
| $\delta$ | deflection of a cylinder at load or at any point of a sphere (in mm) |
| $\delta_1$ | deflection of a cylinder or sphere at positions detailed in **G.2.3.5** and **G.2.4.4** (in mm) |
| $\theta$ | polar co-ordinate of a point on a spherical vessel (in radians) |
| $\phi$ | cylindrical co-ordinate of a point in vessel wall (in radians) |

**G.2.2** ***Radial loads on cylindrical shells***

**G.2.2.1** *General.* The methods in this clause are not considered applicable in cases where the length of the cylinder $L$ is less than its radius $r$ (reference [20]). This applies either to an open-ended cylinder or a closed-ended cylinder where the stiffness is appreciably modified from the case considered. For off-centre attachments the distance from the end of the cylinder to the edge of the attachment should be not less than $r/2$.

In addition the $C_\phi/r$ ratio should not exceed that given in Figure 43, depending on the value of *r*/*t* for the vessel (see reference [20]). This is because in thin shells, the longitudinal axis is relatively flexible and free to deform in relation to the transverse axis, causing the latter to carry a disproportionate share of the load. The applicability of the methods to thick shells is also limited in specific cases by the range of *r*/*t* values against which data is given.

For values of $C_x/r > 0.25$, the data should be used with caution (see reference [17]).

These restrictions apply only in relation to the method of analysis in this appendix. They are not intended for practical cases where experimental or other evidence may support the validity of the design falling outside these restrictions.

In cases where the applicability of the method given in this clause may be in doubt further data may be found in reference [20].

**G.2.2.2** *Stresses at the edge of the loaded area*

**G.2.2.2.1** *General.* The maximum stresses are at the edge of the loaded area. Figure 44 shows a cylindrical vessel subjected to a radial load distributed over a central rectangular area $2C_x \times 2C_\phi$.

The cylindrical shell wall of the vessel is assumed to be simply supported at the ends, which means that the radial deflections, the bending moments and the membrane forces in the shell wall are assumed to be zero there. Since the stresses and deflection due to the load are local and die out rapidly away from the loaded area, this is equivalent to assuming that the loaded area is remote from the ends.

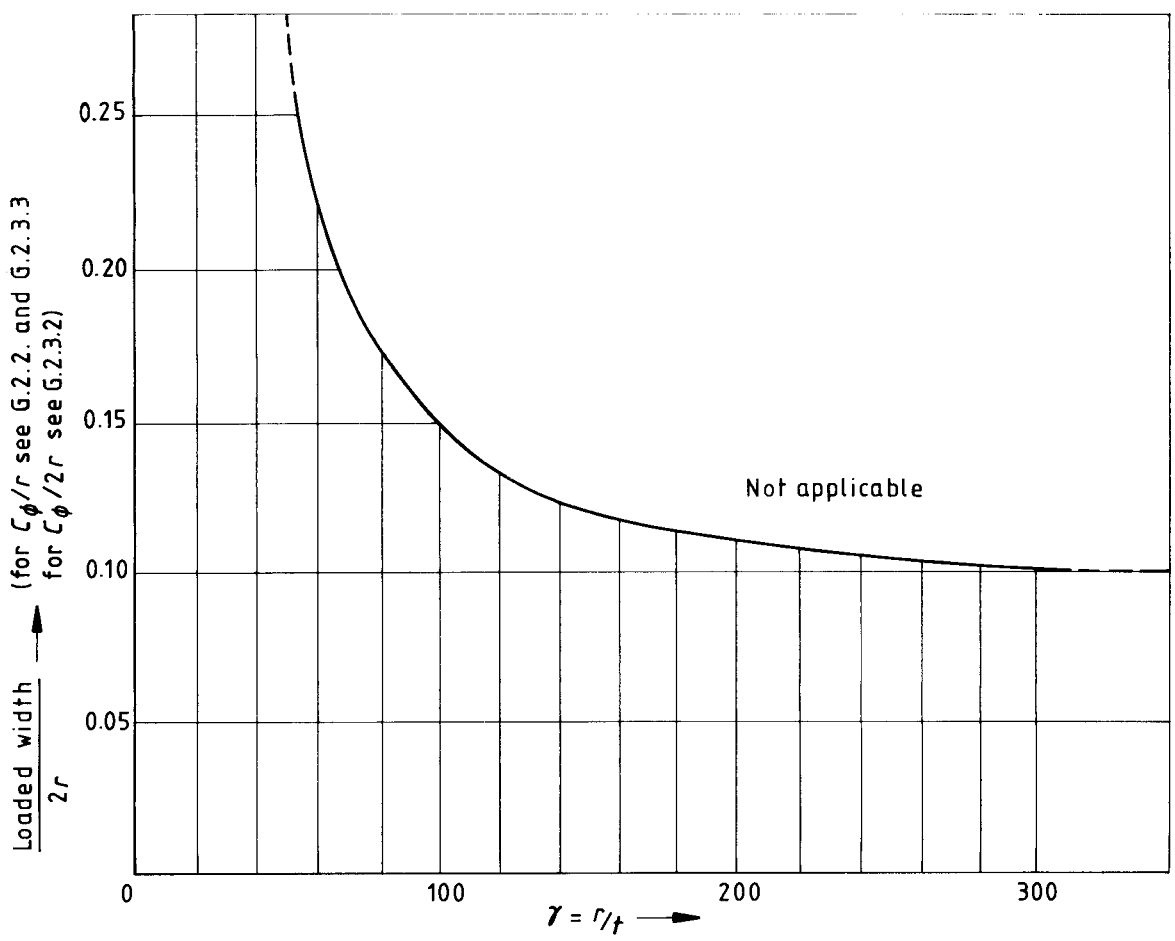



**Figure 43 — Restriction on vessel/attachment geometry** (see **G.2.2** and **G.2.3**)

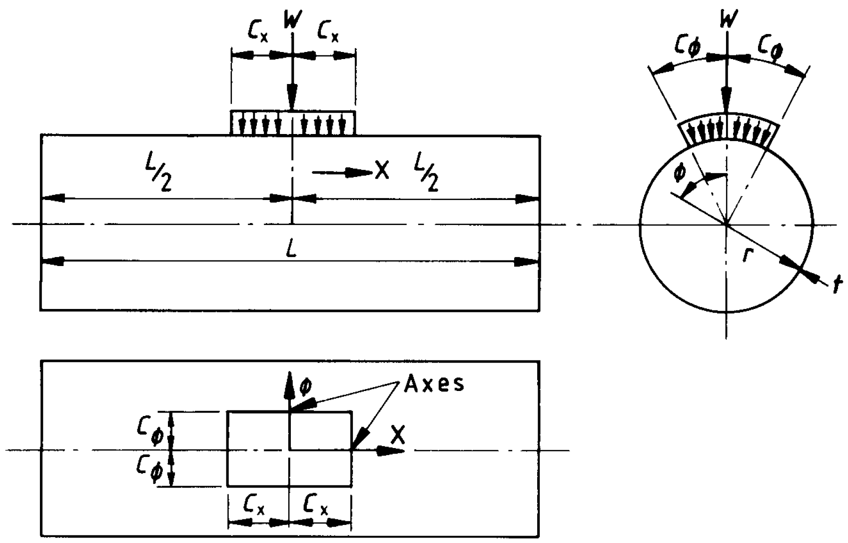



**Figure 44 — Vessel with central radial load**

**G.2.2.2.2** *Off centre loading.* If the loaded area is distant $d$ from the centre of the length of a vessel of length $L$, the deflections, bending moments and membrane forces may be assumed to be equal to those in a vessel of length $L_e$ loaded at its mid-length. $L_e$ is called the equivalent length and can be found from:

$$L_e = L - \frac{4d^2}{L} \quad (78)$$

Figure 45 shows a cylindrical shell loaded in this way and Figure 46 gives a graph of $L_e/L$ against $d/L$ which can be used to find $L_e$.

**G.2.2.2.3** *Determination of stresses.* The resultant longitudinal stresses in the shell is given by:

$$f_x = \frac{N_x}{t} \pm \frac{6M_x}{t^2} \quad (79)$$

The resultant circumferential stress is given by:

$$f_\phi = \frac{N_\phi}{t} \pm \frac{6M_\phi}{t^2} \quad (80)$$

$N_x$ and $N_\phi$ are positive for tensile membrane stresses.

$M_x$ and $M_\phi$ are positive when they cause compression at the outer surface of the shell.

These quantities depend on the ratios:

$$\frac{\text{longitudinal length of loaded area}}{\text{actual or equivalent length}} = \frac{2C_x}{L}$$

and

$$\frac{\text{circumferential length of loaded area}}{\text{longitudinal length of loaded area}} = \frac{2C_\phi}{2C_x}$$

For a radial nozzle or a circular area of radius $r_o$, $C_\phi$ and $C_x$ should be taken as $0.85r_o$.

For an oblique nozzle or elliptical area, $C_\phi$ and $C_x$ should be taken as 0.42 × the major and minor axis of the intersection of the shell or area as appropriate. Non-dimensional functions of each can be expressed in terms of the non-dimensional group $64 \frac{r}{t}\left(\frac{C_x}{r}\right)^2$. The numerical factor 64 is a scale factor without theoretical significance and the value of the expression can be found by calculation or from Figure 47 when *r, t* and $C_x$ are known. The moments and membrane forces are found by interpolation from the graphs of Figure 48, Figure 49, Figure 50 and Figure 51.

Each of the four graphs in each set is for a given value of the ratio $2C_x/L$ and has curves for four values of the ratio $C_\phi/C_x$.

The circumferential moment $M_\phi$ is found from Figure 48.

The longitudinal moment $M_x$ is found from Figure 49.

The circumferential membrane force $N_\phi$ is found from Figure 50. The longitudinal membrane force $N_x$ is found from Figure 51.

A moment is considered as positive if it causes compression at the outside of the vessel.

A membrane force is considered as positive if it causes tension in the vessel wall.

**G.2.2.2.4** *Effect of internal and external pressure.* A conservative result is obtained for total stresses if the stresses due to the pressure are simply added to those due to local radial loads already calculated.

The method cannot be used for vessels under external pressure because the deflection due to the radial load always increases the out-of-roundness of the shell. For the same reason it should not be applied to a cylindrical shell subject to a longitudinal load as well as a radial load. In these cases the deflection due to the radial load should be found as in **G.2.2.4** and the effect thereof assessed in relation to shape requirements given in Table 17 for such vessels.

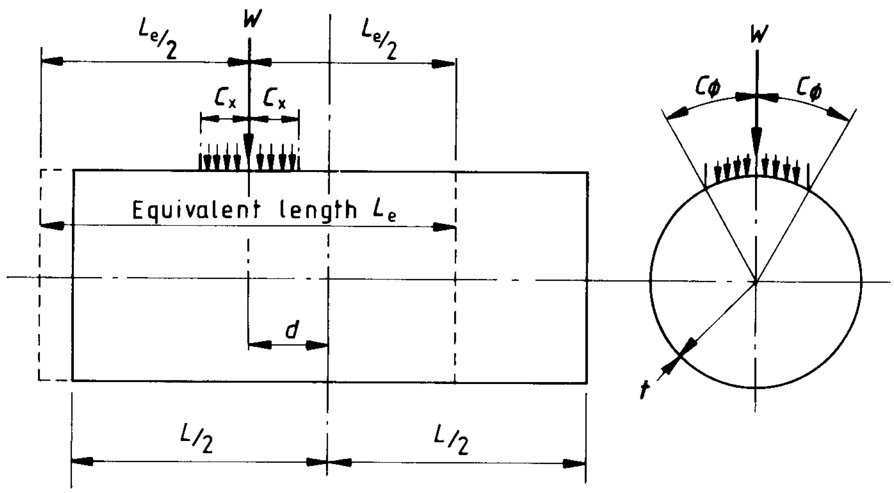



**Figure 45 — Vessel with radial load out of centre**

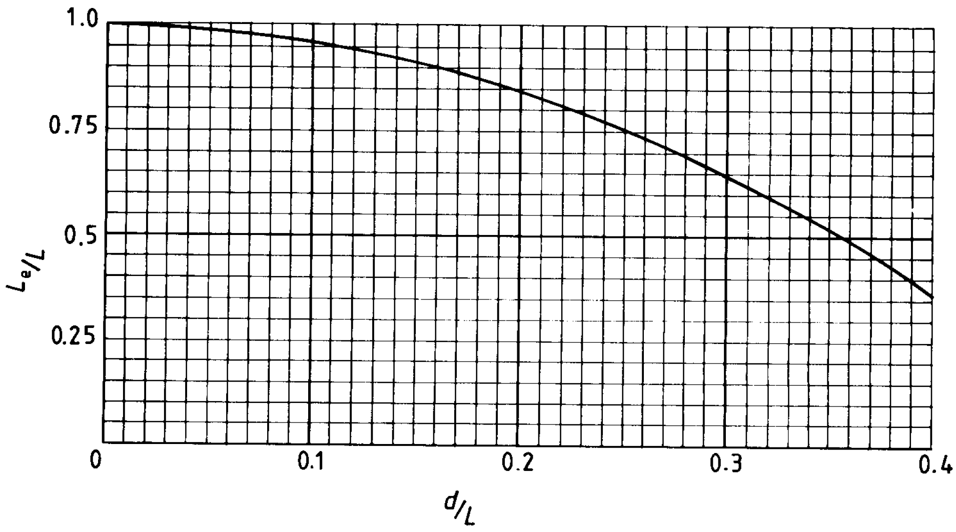



**Figure 46 — Graph for finding equivalent length $L_e$**

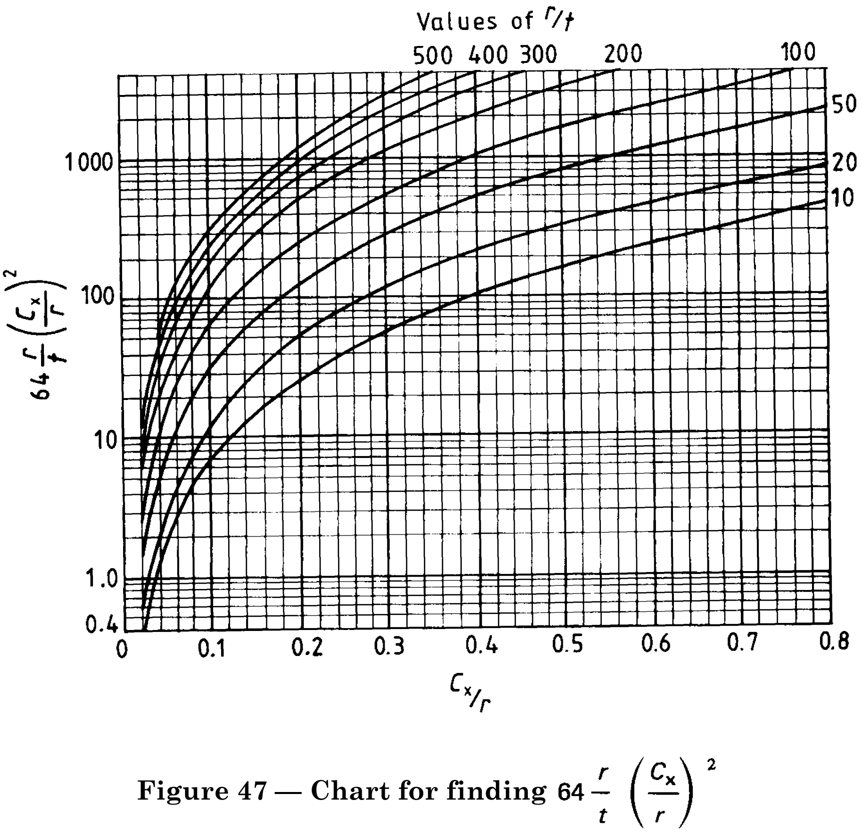
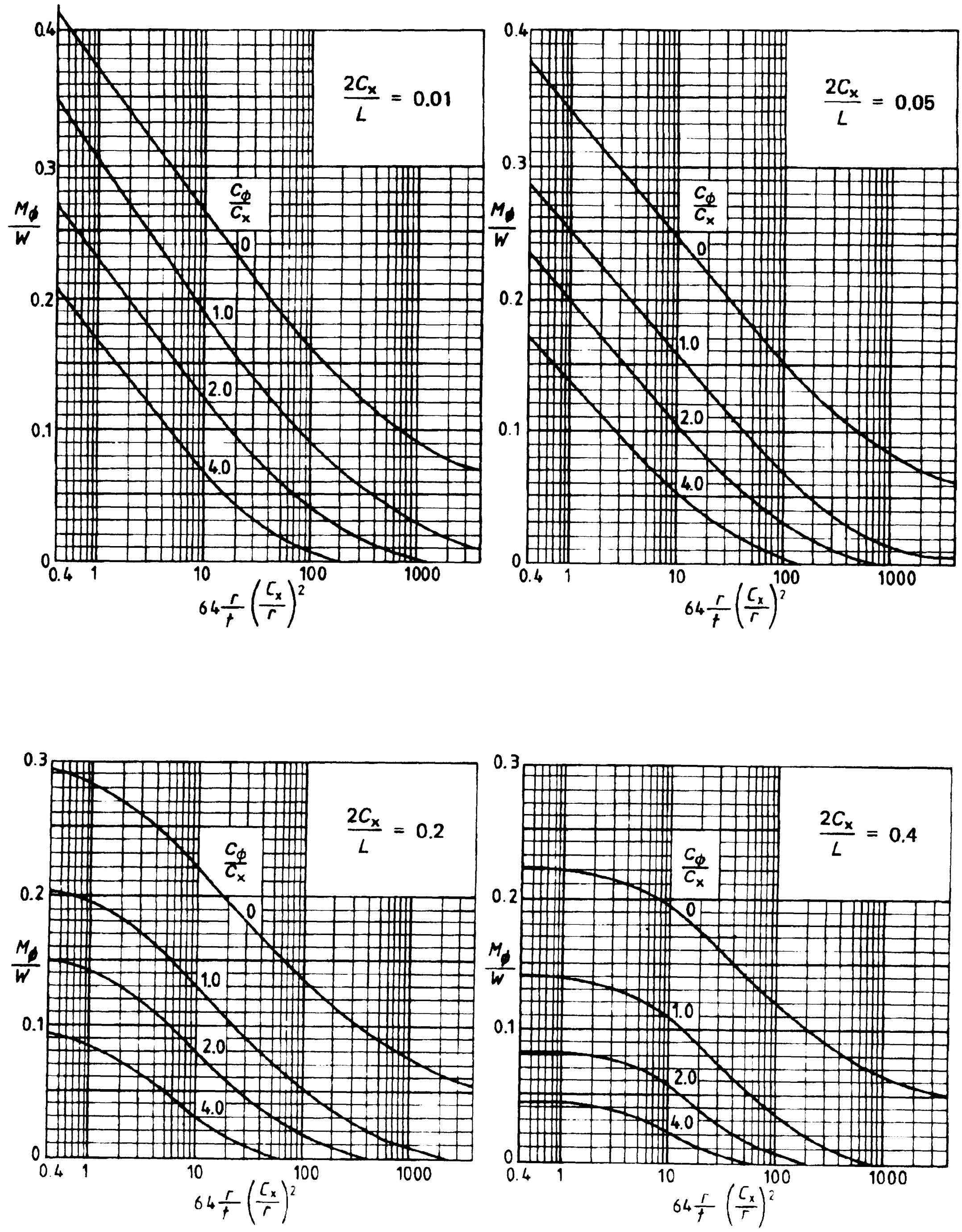
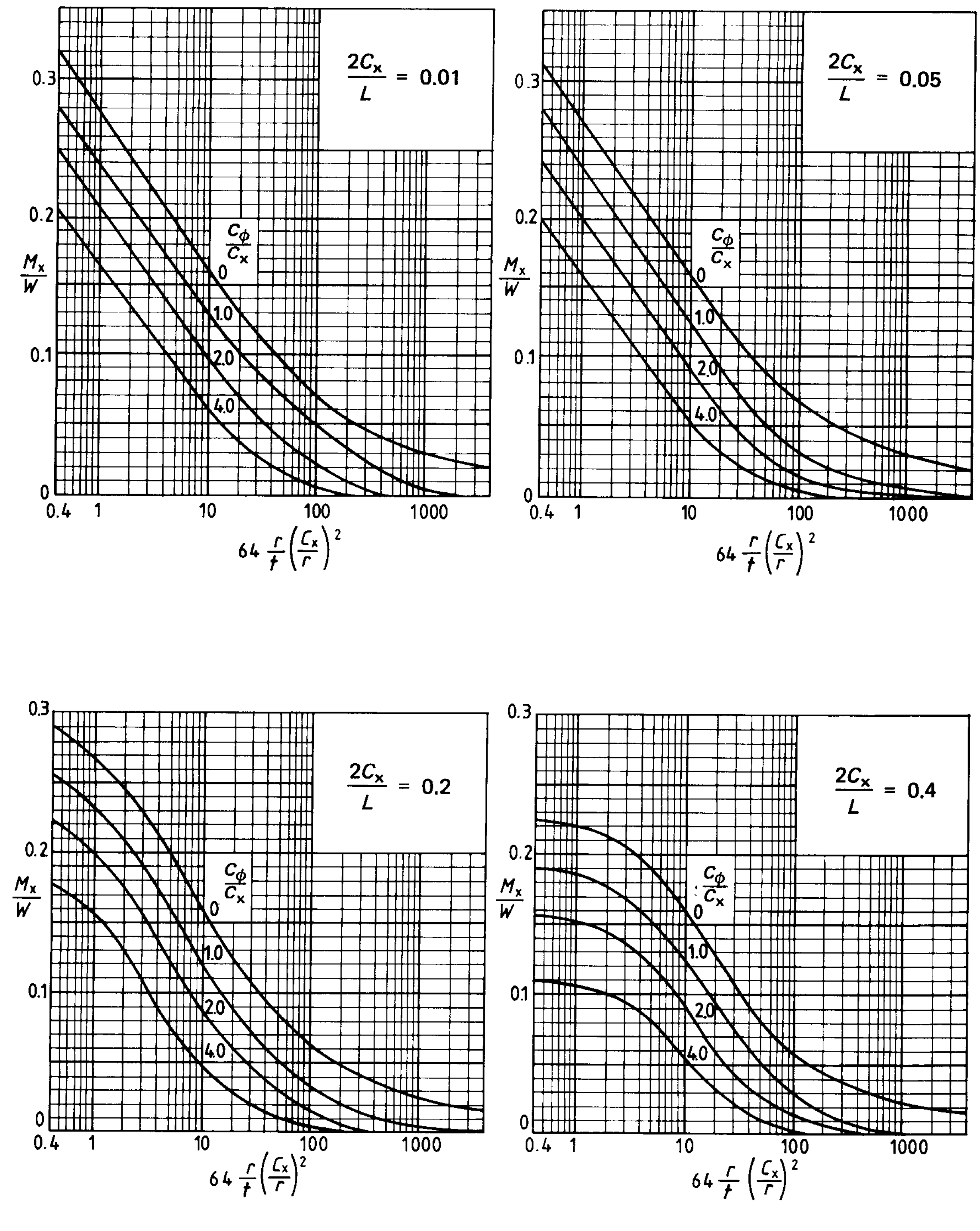



**Figure 47 — Chart for finding $64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$**



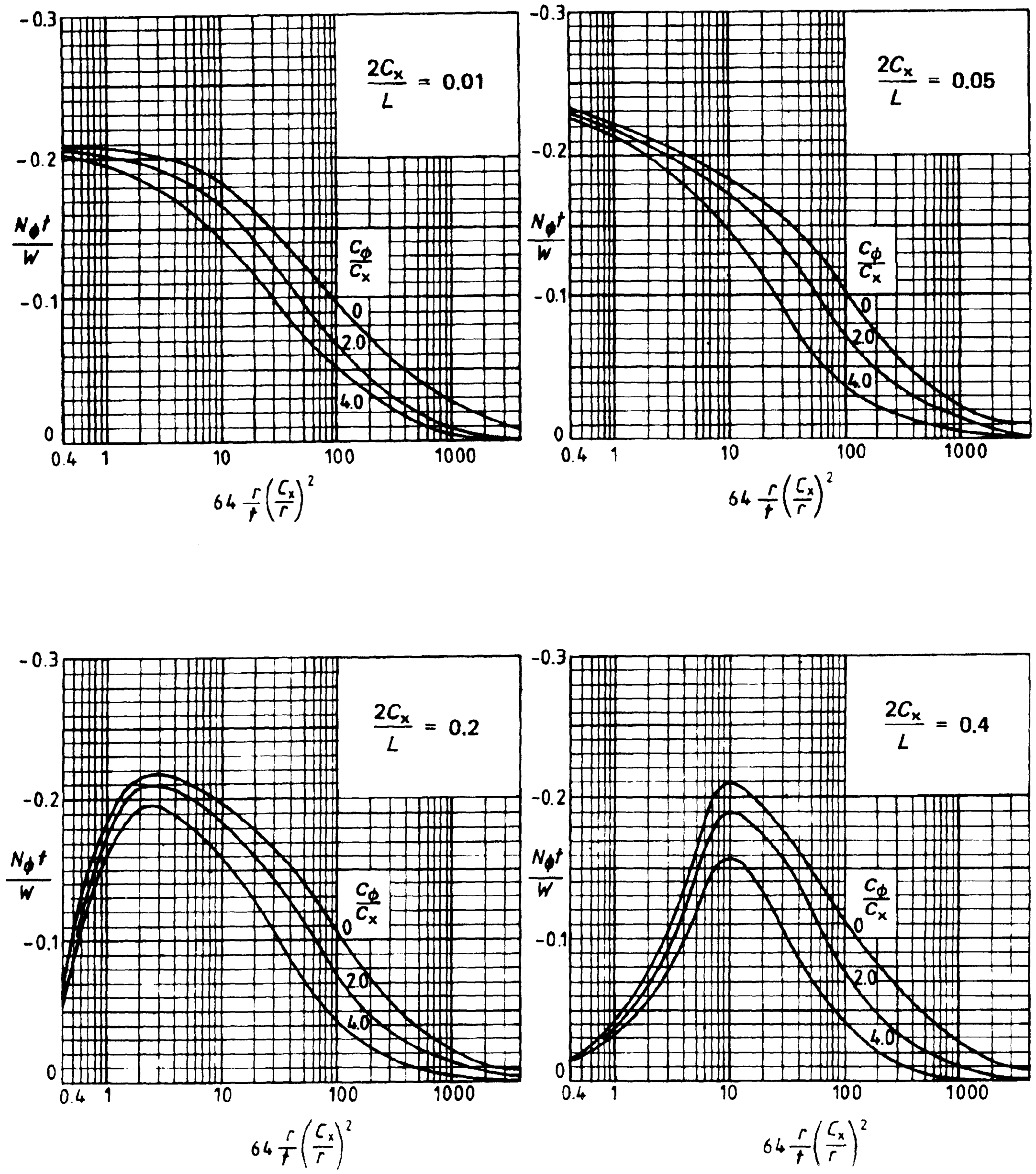

NOTE $64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ is found from Figure 47.

**Figure 48 — Cylindrical shells with radial load: circumferential moment per millimetre width** (see **G.2.2**)



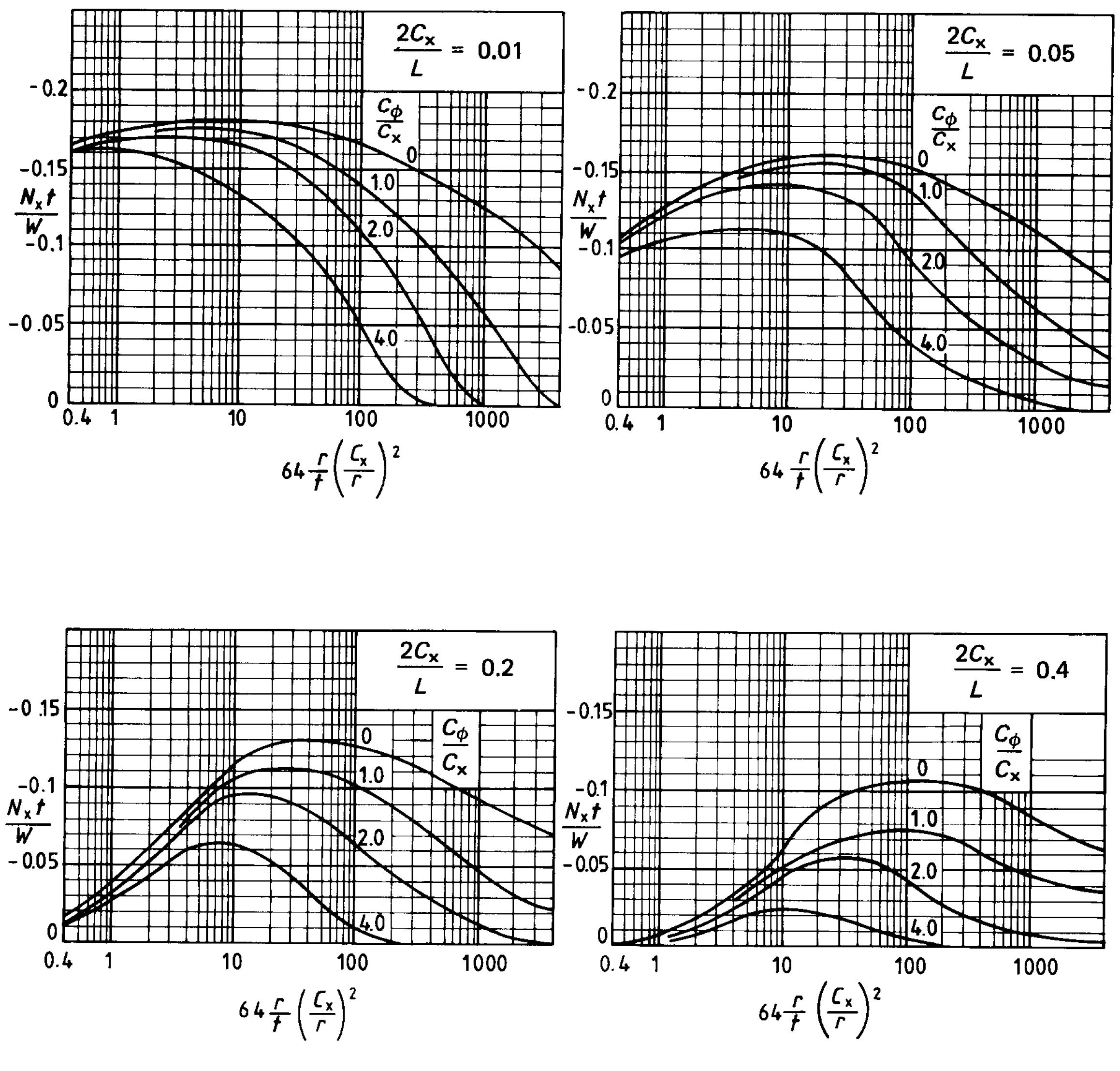

NOTE $64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ is found from Figure 47.

**Figure 49 — Cylindrical shells with radial load: longitudinal moment per millimetre width** (see **G.2.2**)




NOTE $64 \frac{r}{t}\left(\frac{C_x}{r}\right)^2$ is found from Figure 47.

**Figure 50 — Cylindrical shells with radial load: circumferential membrane force per millimetre width** (see **G.2.2**)




NOTE $64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ is found from Figure 47.

**Figure 51 — Cylindrical shells with radial load: longitudinal membrane force per millimetre width** (see **G.2.2**)

**G.2.2.3** *Stresses away from the edge of the loaded area*

**G.2.2.3.1** *General.* Although the maximum stresses occur at the edge of the load, it is necessary to find those at other positions when the effect of one load at the position of another is required.

This happens:

a) when longitudinal or circumferential moments are resolved as in **G.2.3**;

b) when loads are applied close together, e.g. if a bracket is fixed close to a branch.

In general the effect of one load at the position of another can be disregarded when the distance between the centres of the loaded areas is greater than $K_1 C_\phi$ for loads separated circumferentially or $K_2 C_x$ for loads separated longitudinally, where $K_1$ and $K_2$ are found from Table 18 and $C_\phi$ and $C_x$ are for the greater load.

**Table 18 — Values of $K_1$ and $K_2$**

| $64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ | $\frac{2C_x}{L}$ | $K_1$ | $K_2$ |
|---|---|---|---|
| 0.4 | 0.01 | 8 | 8 |
| | 0.05 | 6 | 8 |
| | 0.2 | 3 | 4 |
| | 0.4 | 1.5 | 2 |
| 10 | 0.01 | 3 | 8 |
| | 0.05 | 2.5 | 8 |
| | 0.2 | 1.5 | 3 |
| | 0.4 | 1.5 | 2 |
| 200 | 0.01 | Negligible | 5 |
| | 0.05 | | 4 |
| | 0.2 | | 2.5 |
| | 0.4 | | 1.75 |
| 3 200 | All values | Negligible | 2.5 |

NOTE The value of the non-dimensional factor $64(r/t)\ (C_x/r)^2$ can be found from Figure 47.

**G.2.2.3.2** *Variation of stress round the circumference.* No exact analytical treatment of the variation of stress round the circumference away from the edge of the loaded area is available. The following treatment is an approximation sufficiently accurate for practical purposes. For an experimental verification of it see reference [21].

Consider a radial line load of length $2C_x$, applied at the mid-length of a thin cylinder as shown in Figure 52(a). The maximum stresses due to this load at points away from it are on the circumference passing through its mid-length as A in the figure. The radius through A makes an angle $\phi_1$ with the line of the load.

The moments and membrane forces at A, $M_\phi$, $M_x$, $N_\phi$, $N_x$ can be found from the graphs of Figure 52, Figure 53, Figure 54 and Figure 55 in which the functions $M_\phi/W$, $M_x/W$, $N_\phi t/W$ and $N_x t/W$ are plotted against the non-dimensional group $\phi_1 r/C_x$.

The diagram showing the load and its geometry, as Figure 52(a), is repeated on each chart for convenience.

Line loads are, of course, unusual in practice, and loads distributed over an area having an appreciable circumferential width $2C_\phi$ are treated as follows.

a) Find the value of the function $M_\phi/W$, $M_x/W$, $N_\phi t/W$ or $N_x t/W$ at the edge of the load for the known values of $C_\phi/C_x$ and $2C_x/L$ from the graphs in Figure 48, Figure 49, Figure 50 and Figure 51.

b) Enter the corresponding graph in Figure 52, Figure 53, Figure 54 or Figure 55 at this value.

The intercept on the curve for $2C_x/L$ gives a value of $\phi_1 r/C_x = Z$, e.g. if $64(r/t)\ (C_x/r)^2 = 10$, $2C_x/L = 0.01$ and $C_\phi/C_x = 1$. Figure 48 gives $M_\phi/W = 0.185$. Entering Figure 52 at $M_\phi/W = 0.185$ gives $Z = 0.55$ for $2C_x/L = 0.01$ as indicated by the dotted lines in the left-hand graph of Figure 52.

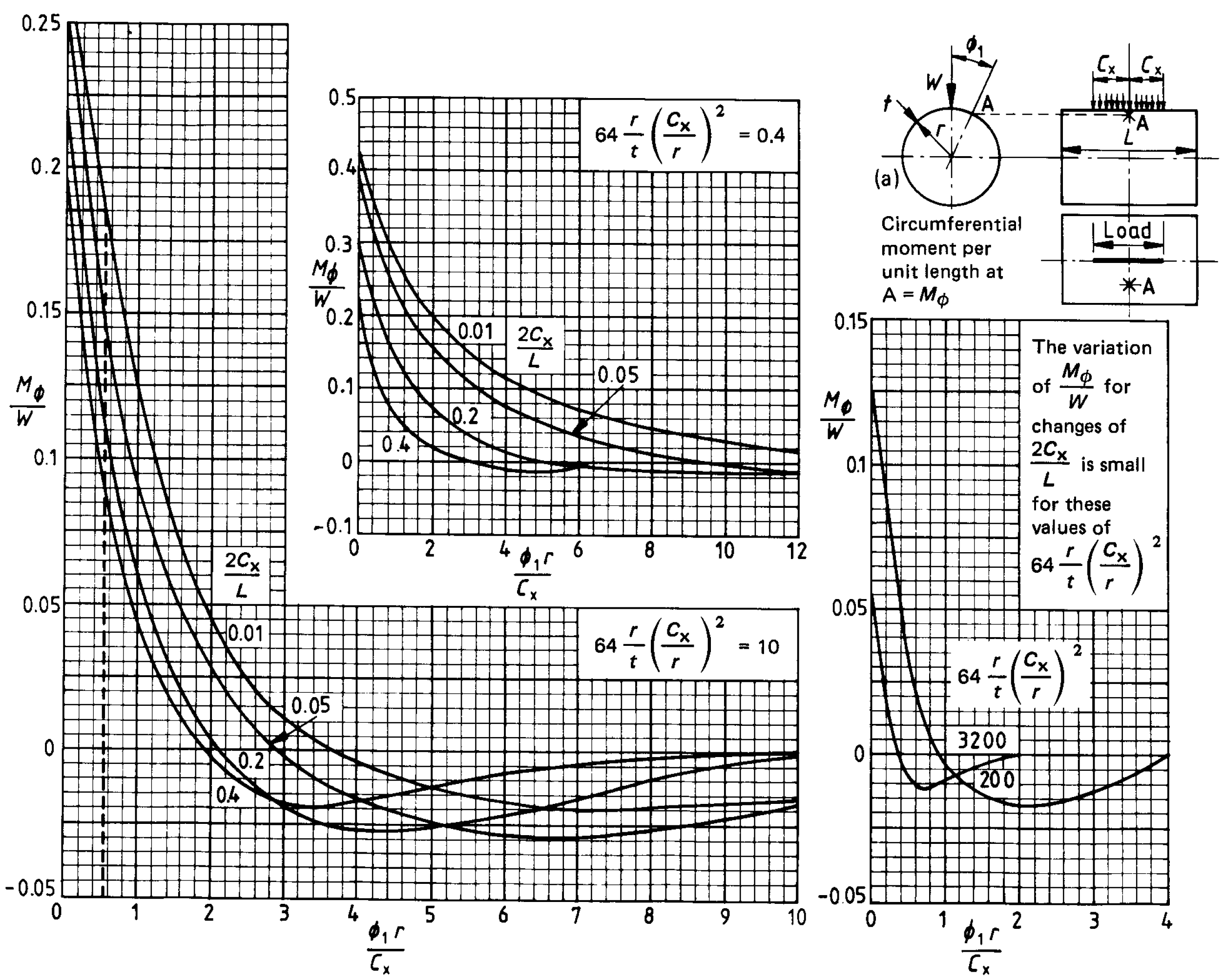
The value of $M_\phi/W$ at A is then found by substituting $(\phi_1 r/C_x + Z - C_\phi/C_x)$ for the actual value of $\phi_1 r/C_x$ in the same graph.

The other quantities $M_x/W$, $N_\phi t/W$, and $N_x t/W$ can be found in the same way. This method is used in order to avoid the use of a separate set of four charts for each value of $C_\phi/C_x$ considered.

Diagrams for circumferential bending moments and forces are printed up the page to distinguish them from those for longitudinal moments and forces which are printed across the page.

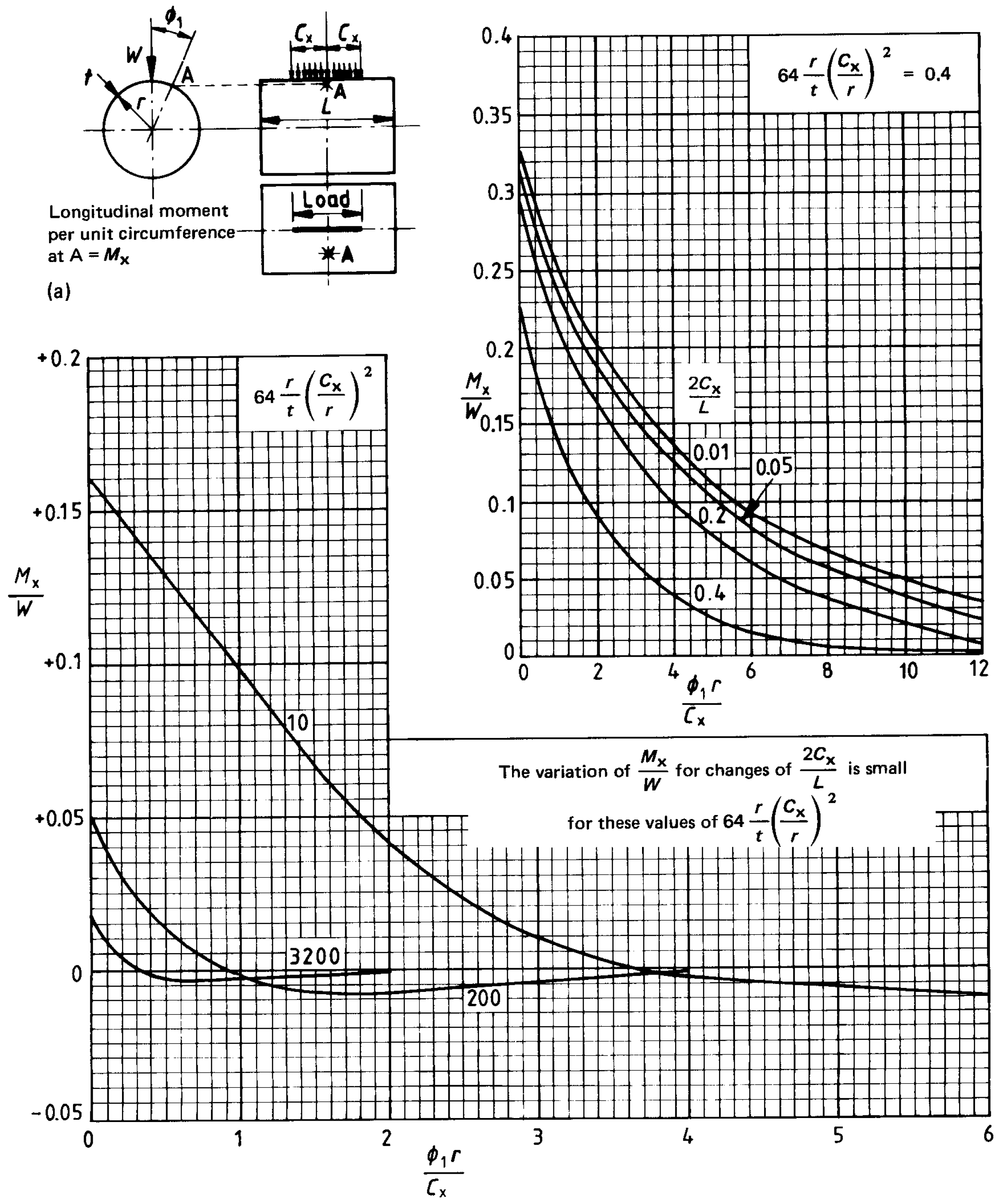
When the centre of the load is away from the mid-length of the cylinder, the equivalent length $L_e$, found as in **G.2.2.2**, should be substituted for $L$ in all cases.

For variations of stress along the cylinder due to radial loading see **G.2.2.3.3**.



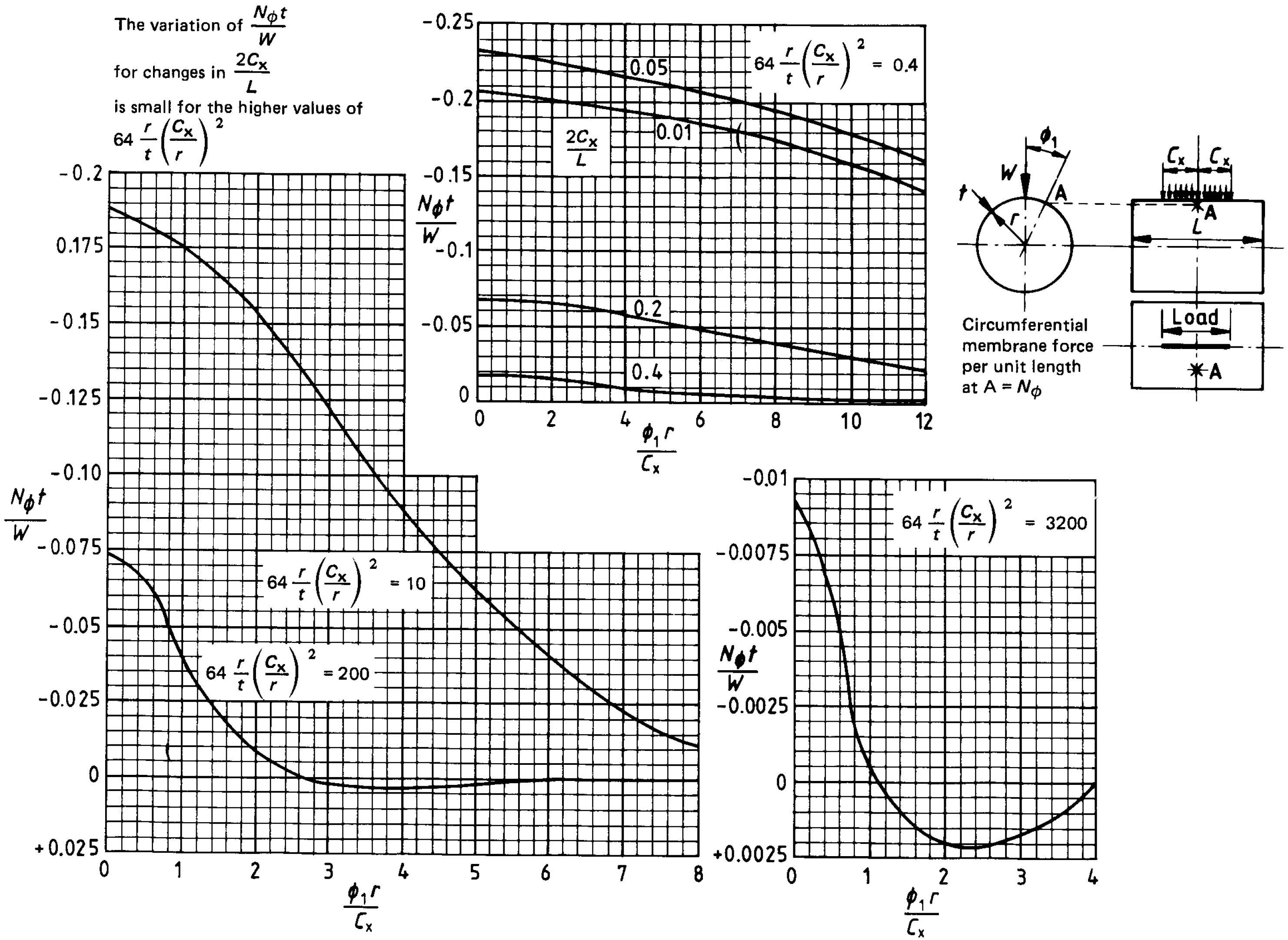

NOTE These charts apply in any consistent system of units.

**Figure 52 — Circumferential bending moment due to a radial line load variation round circumference** (see G.2.2.3.2)



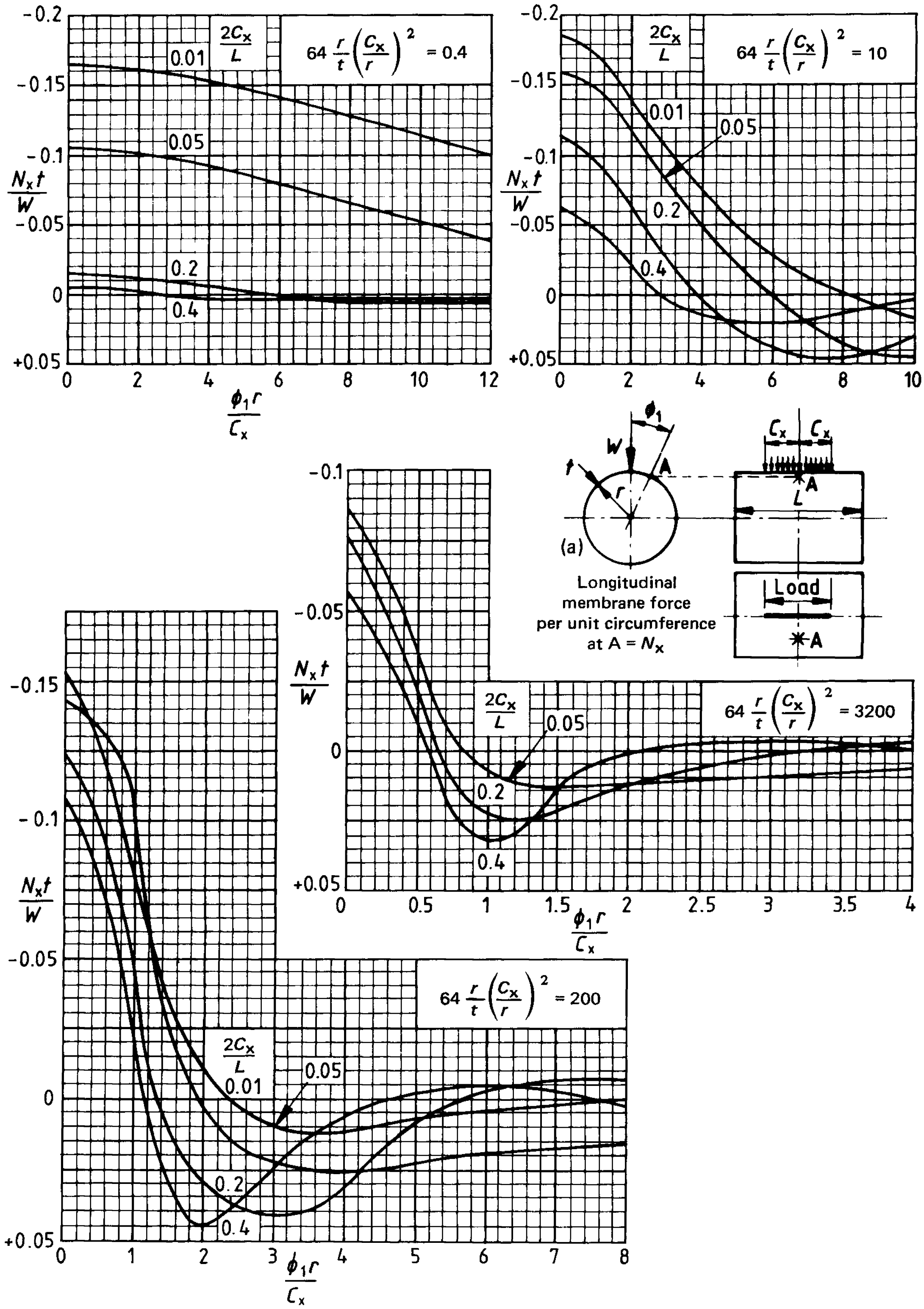

NOTE These charts apply in any consistent system of units.

**Figure 53 — Longitudinal moment from radial line load variation round circumference** (see **G.2.2.3.2**)




NOTE These charts apply in any consistent system of units.

**Figure 54 — Circumferential membrane stress from radial line load variation round circumference** (see **G.2.2.3.2**)




NOTE These charts apply in any consistent system of units.

**Figure 55 — Longitudinal membrane forces from radial line load variation round circumference** (see **G.2.2.3.2**)

**G.2.2.3.3** *Variation of stress along the cylinder.* Consider a radial line load, *W*, distributed over a length $2C_x$ as shown in Figure 56(a).

Values of $M_\phi$, $M_x$, $N_\phi$ and $N_x$ at A can be found from the graphs of Figure 56, Figure 57, Figure 58 and Figure 59, respectively.

In these charts values of $M_\phi/W$, $M_x/W$, $N_\phi t/W$ and $N_x t/W$ are plotted against $x/C_x$ for given values of $64\ (r/t)\ (C_x/r)^2$ and $2C_x/L$.

The resultant stresses in the shell at A are given by:

$$\text{circumferential stress } f_\phi = \frac{N_\phi}{t} \pm \frac{6M_\phi}{t^2}$$

$$\text{longitudinal stress } f_x = \frac{N_x}{t} \pm \frac{6M_x}{t^2}$$

The values for $x/C_x$ less than 1.0, for which no curves are plotted, fall within the loaded lengths, and the curves should not be extended into this region. The values for $x/C_x = 1$ correspond to the maximum stresses found from Figure 48, Figure 49, Figure 50 and Figure 51 for $C_\phi/C_x = 0$.

The loading diagram as Figure 56(a) has been repeated on each chart for convenience.

Diagrams for circumferential bending moments and forces are printed up the page to distinguish them from those for longitudinal moments and forces which are printed across the page.

For a load distributed over an area $2C_x \times 2C_\phi$, the moments and membrane forces at any value of $x/C_x$ are reduced in the same ratio as the corresponding values at the edge of the load found from Figure 48, Figure 49, Figure 50 and Figure 51, e.g. in the ratio:

$$\frac{\text{value for actual } C_\phi/C_x}{\text{value for } C_\phi/C_x = 0}$$

*Example.* A vessel is 2.5 m diameter × 6 m long × 12 mm thick. A radial load *W* is applied to an area 300 mm square at. the mid-length of the shell. Find the circumferential moment at a position 600 mm from the centre of the loaded area measured along the axis of the vessel.

$C_\phi = C_x = 150$ mm; $r = 1250$ mm; $r/t = 104$;

$C_x/r = 0.12$; $2C_x/L = 0.05$; $x/C_x = 4$.

For a line load, interpolating in Figure 56:

$M_\phi/W = 0.054$ at $x/C_x = 4$

From Figure 48 at the ends of a line load when:

$C_\phi/C_x = 0$, $64\ (r/t)\ (C_x/r)^2 = 90$, and $2C_x/L = 0.05$,

$M_\phi/\text{W} = 0.153$

and when $C_\phi/C_x = 1.0$, $M_\phi/W = 0.072$

∴ when the load is distributed over an area 300 mm square:

$$M_\phi/W \text{ at } x = 0.054 \times \frac{0.072}{0.153} = 0.025$$

∴ the circumferential moment at $x = 0.025W$.

**G.2.2.4** *Deflections of cylindrical shells due to radial loads.* The deflections of a cylindrical shell due to a local load are required for:

a) finding the movement of a vessel shell due to the thrust of a pipe connected to it;

b) finding the rotation of a branch due to a moment applied by a pipe connected to it. (See **G.2.3**.)

The deflection of the shell due to a radial load is a function of the non-dimensional parameters $r/t$, $\delta\ Er/W$ and $L/r$ which is given by the full lines in the charts as follows:

Figure 60(a) for values of $r/t$ between 15 and 40;

Figure 60(b) for values of $r/t$ between 40 and 100;

Figure 61 for values of $r/t$ greater than 100.

For a central load $L$ is the actual length of the vessel.

For a load out of centre $L$ is the equivalent length $L_e$ found as in **G.2.2.2**.

For a point load, the value of $\delta\ Er/W$ is given by the full line from the appropriate horizontal $L/r$ line in the top right-hand extension of each diagram as in the example with Figure 60.

For a load distributed over a square of side $2C$, the value of $\delta\ Er/W$ is given by a line joining the intersections of the $L/r$ and $C/r$ lines in the top right-hand and bottom left-hand extensions of each diagram as shown by the dotted line and example on Figure 61.

The deflection due to a load distributed over a circular area of radius $r_o$ is approximately the same as that for a square of side $1.7r_o$.

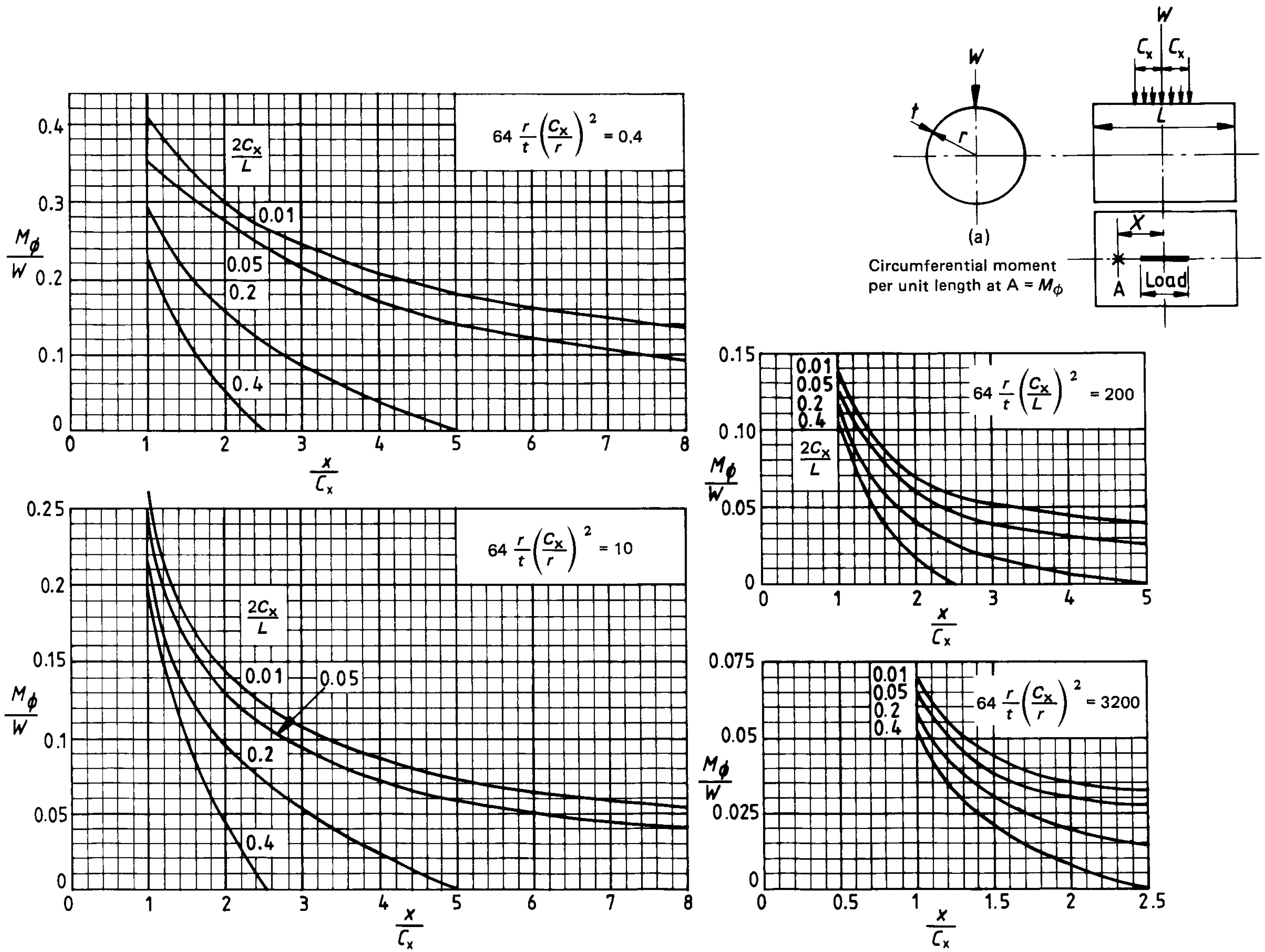
The deflection due to a load distributed over a rectangular area $2C_x \times 2C_\phi$ is approximately the same as that for an equivalent square of side 2 $C_l$, where $C_l$ is obtained as follows:

$$C_l = \sqrt{C_\phi C_x} \text{ when } C_x > C_\phi \quad (81)$$

$$C_l = (C_\phi)^{0.93} \times (C_x)^{0.07} \text{ when } C_\phi > C_x \text{ (or from Figure 62).} \quad (82)$$

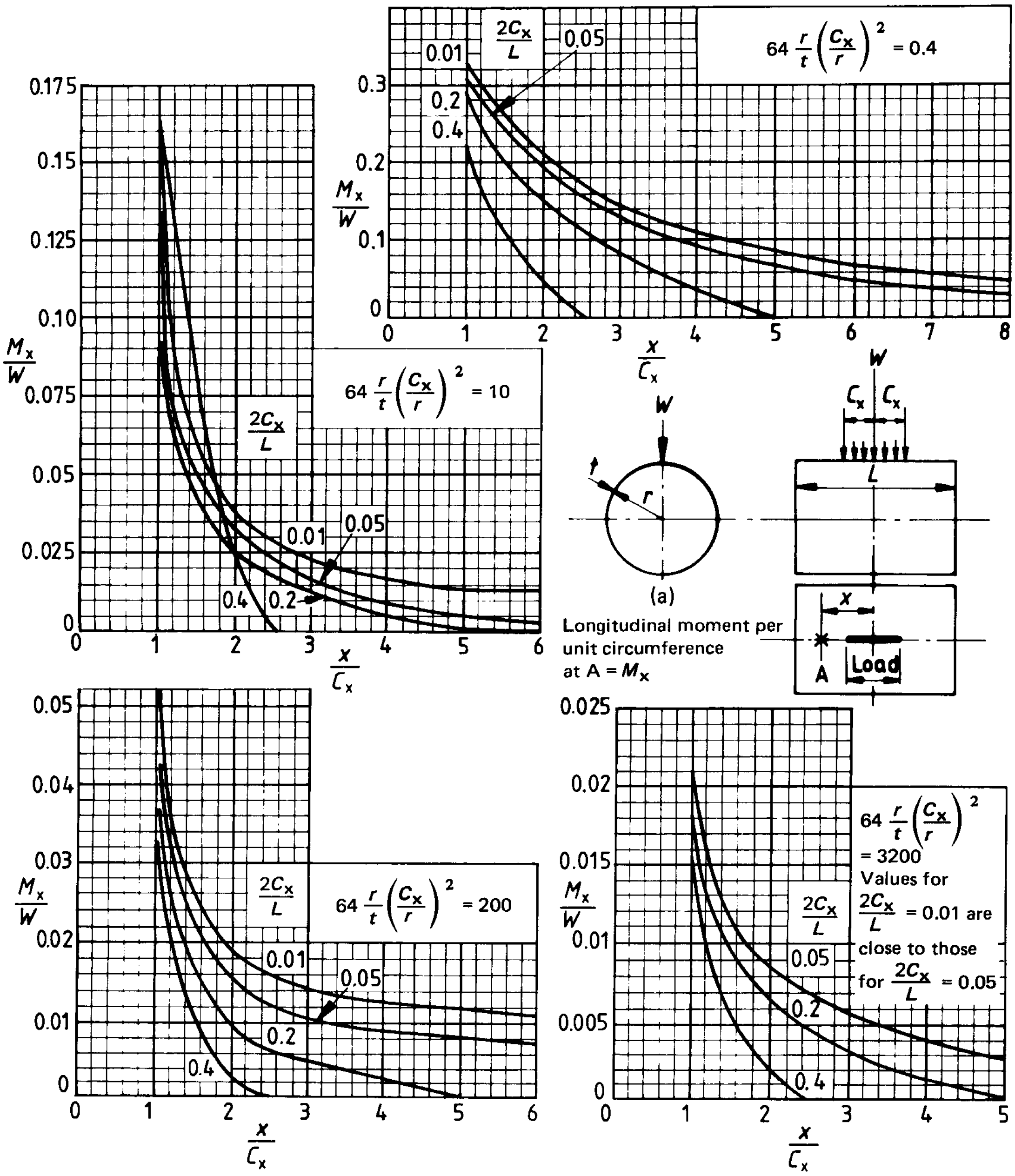
Equation (81) applies to a rectangular area in which the long axis is longitudinal.

Equation (82) applies to a rectangular area in which the long axis is circumferential.



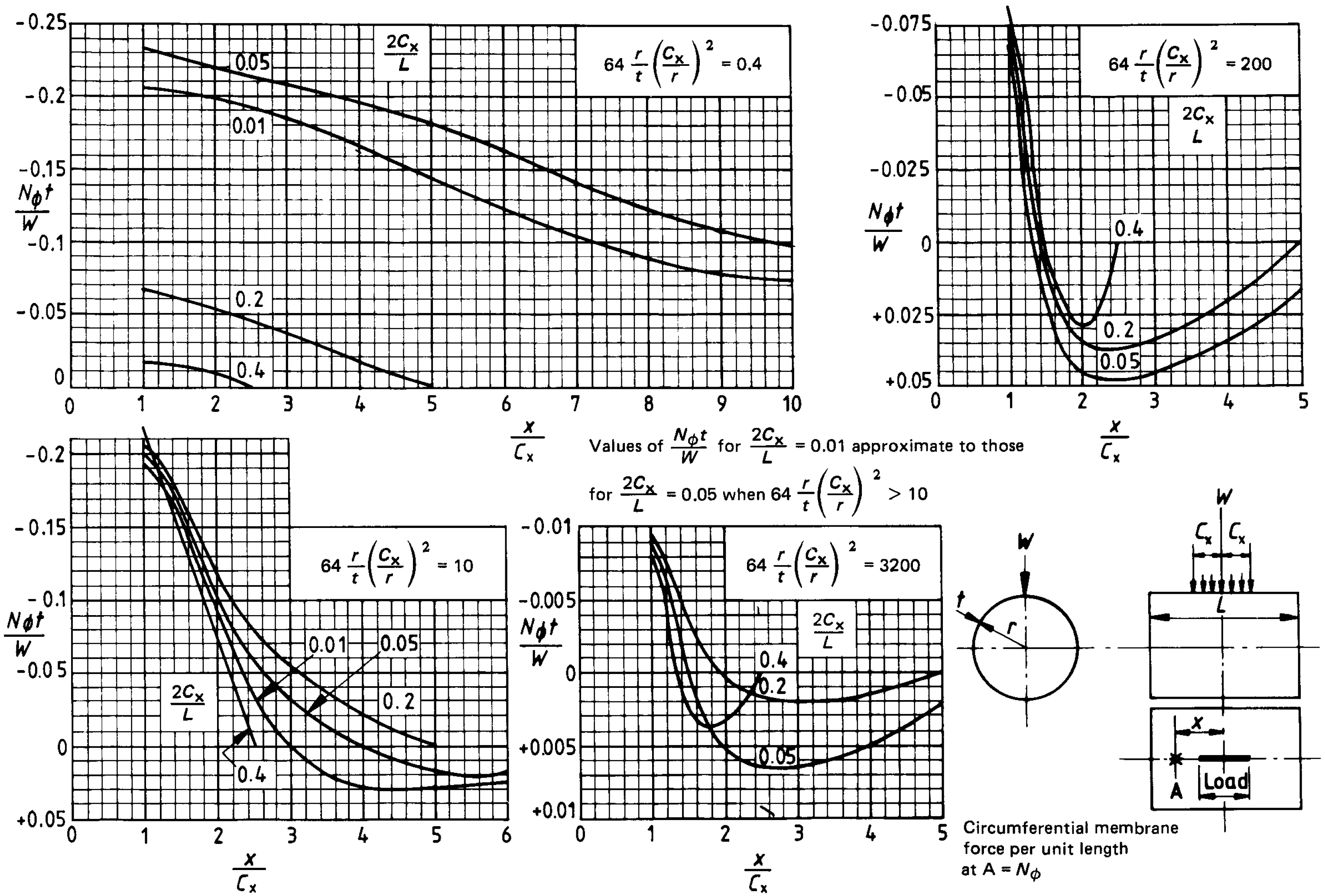

NOTE These charts apply in any consistent system of units.

**Figure 56 — Circumferential bending moment due to a radial line load variation along cylinder** (see **G.2.2.3.3**)



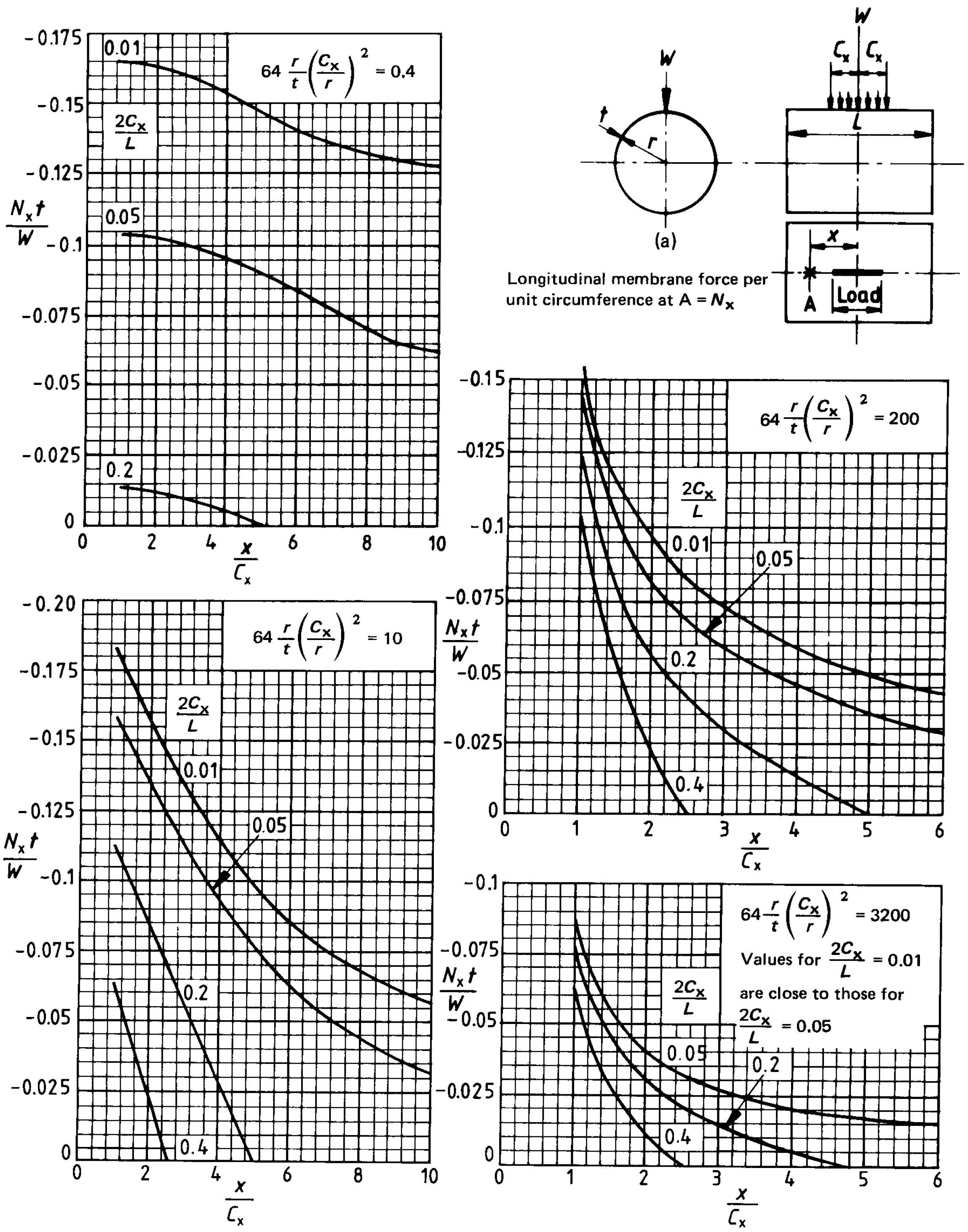

NOTE These charts apply in any consistent system of units.

**Figure 57 — Longitudinal moment due to a radial line load variation along cylinder** (see **G.2.2.3.3**)



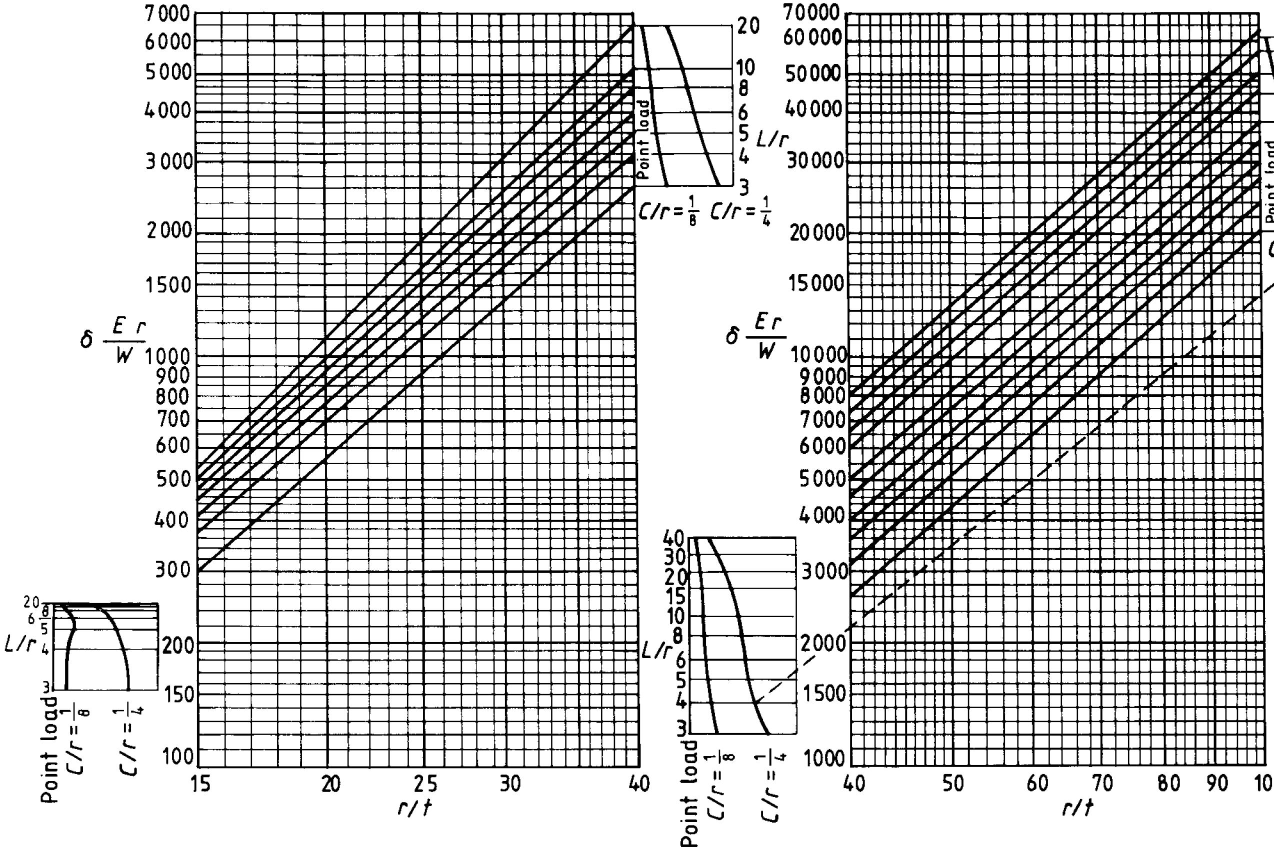

NOTE These charts apply in any consistent system of units.

**Figure 58 — Circumferential membrane forces due to a radial line load variation along cylinder** (see **G.2.2.3.3**)



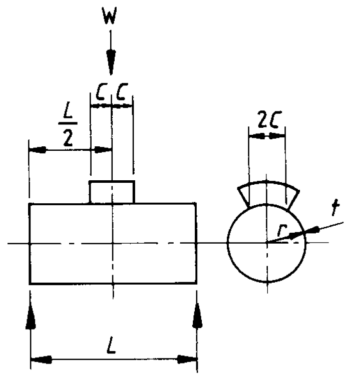

NOTE These charts apply in any consistent system of units.

**Figure 59 — Longitudinal membrane forces due to a radial line load variation along cylinder** (see **G.2.2.3.3**)




(a) For *r*/*t* less than 40

(b) For *r*/*t* between 40 and 100

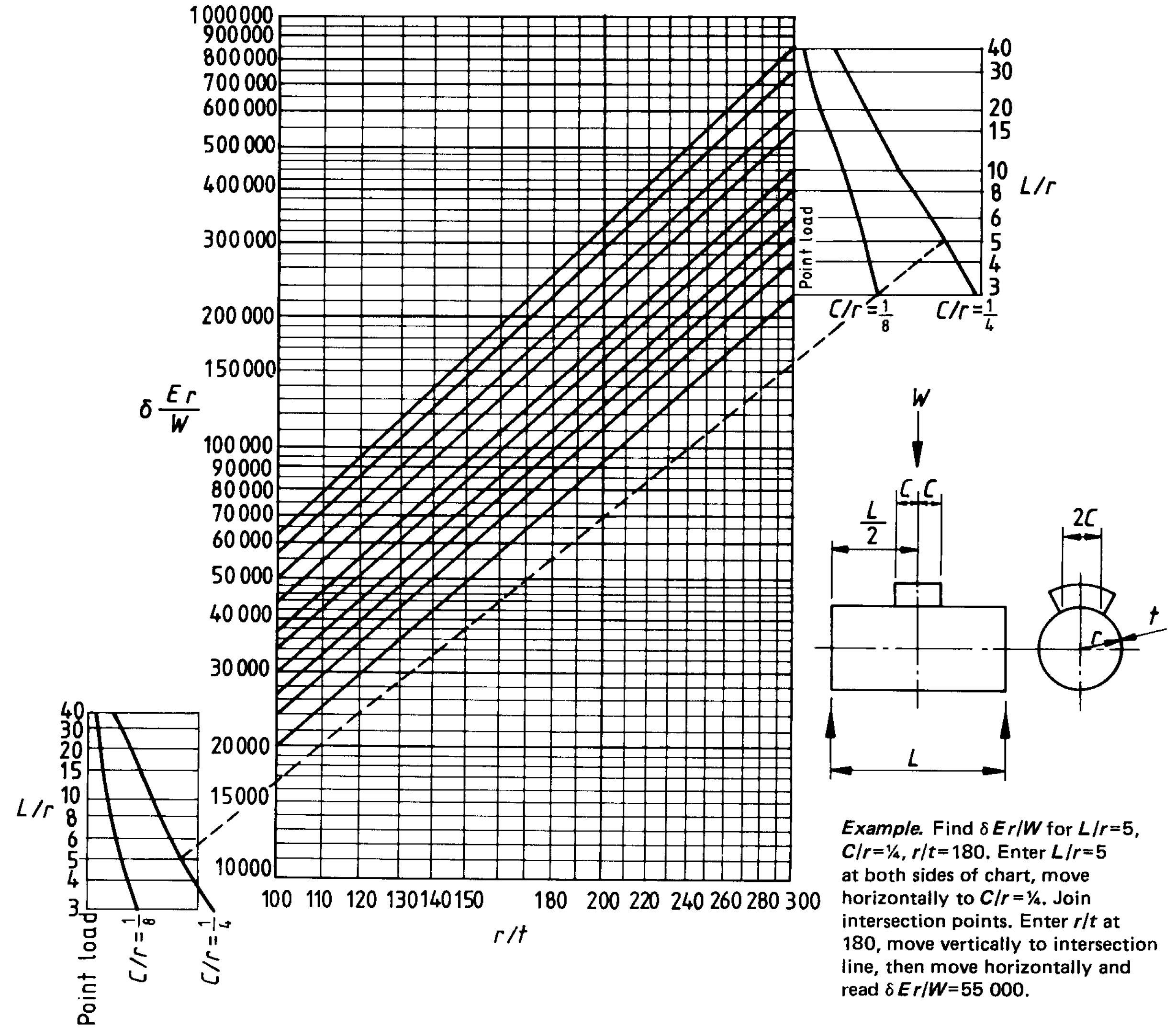



***Example.*** Find δ***E r*/*W*** for ***L*/*r***=10, ***r*/*t***=70, ***C*/*r***=0=point load. Enter ***L*/*r***=10 at right-hand side of chart, move horizontally to ***C*/*r***=0 or point load, proceed down sloping line to meet vertical line for ***r*/*t***=70, then move horizontally and read δ***E r*/*W***=17 000.

NOTE 1 For values of *r*/*t* greater than 100 see Figure 61.

NOTE 2 Values of δ are exclusive of the deflection of the whole shell as a beam.

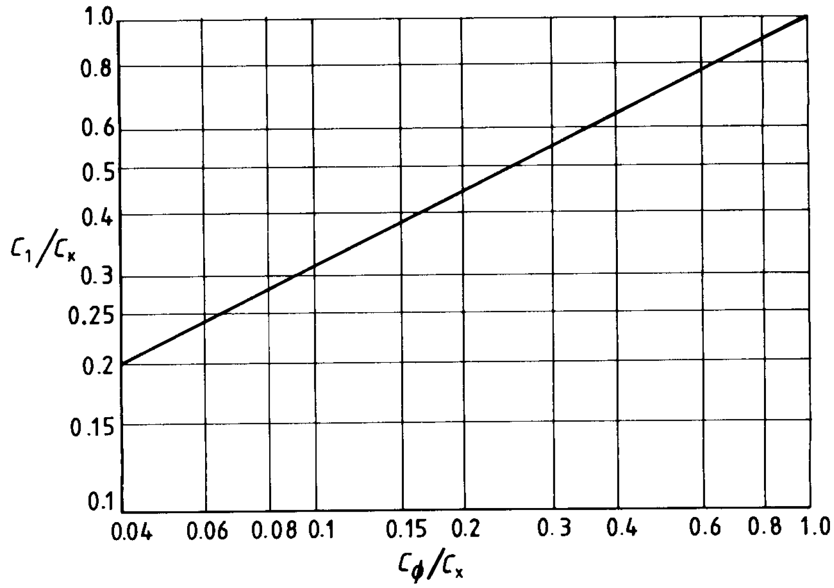
**Figure 60 — Maximum radial deflection of a cylindrical shell subjected to a radial load, *W*, uniformly distributed over a square $2C \times 2C$**



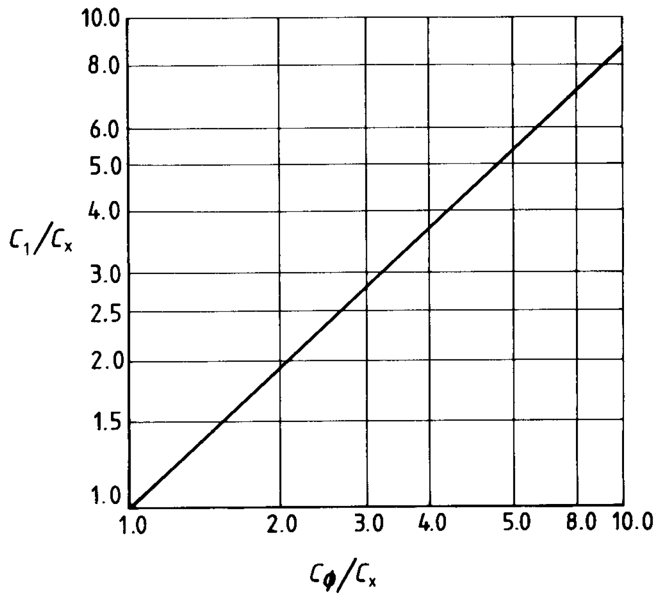

NOTE 1 For values of *r*/*t* less than 100 see Figure 60.

NOTE 2 Values of $\delta$ are exclusive of the deflection of the whole shell as a beam.

**Figure 61 — Maximum radial deflection of a cylindrical shell subjected to a radial load, *W*, uniformly distributed over a square 2*C* × 2*C***




(a) When $C_\phi$ is less than $C_x$




(b) When $C_\phi$ is greater than $C_x$

**Figure 62 — Graphs for finding the square $2C_1 \times 2C_1$ equivalent to a rectangular loading area $2C_x \times 2C_\phi$**

**G.2.3** ***External moments applied to cylindrical shells***

**G.2.3.1** *General.* External moments can be applied to the shell of a vessel by a load on a bracket or by the reaction at a bracket support.

For design purposes external moments are considered as described in **G.2.3.2** to **G.2.3.5**.

The results are not considered applicable in cases where the length of the cylinder $L$ is less than its radius $r$ (see reference [20]). For off-centre attachments the distance from the end of the cylinder to the edge of the attachment should be not less than $r/2$.

In addition, the ratios $C_\theta/2r$ (**G.2.3.2**) and $C_\phi/r$ (**G.2.3.3**) should not exceed those given in Figure 43, depending on the value of $r/t$ for the vessels.

For corresponding values of $C_x/r$ and $C_z/2r > 0.25$ the data should be used with caution (see reference [17]).

These restrictions apply only in relation to the method of analysis in this appendix. They are not intended for practical cases where experimental or other evidence may support the validity of the design falling outside the above restrictions.

In cases where the applicability of the method given in this clause may be in doubt further data may be found in reference [20].

**G.2.3.2** *Circumferential moments.* A circumferential moment applied to a rectangular area $C_\theta \times 2C_x$ (see Figure 63) is resolved into two opposed loads:

$\pm W = \dfrac{1.5M}{C_\theta}$ acting on rectangles of sides $2C_\phi \times 2C_x$,

where $C_\phi = \dfrac{C_\theta}{6}$, which are separated by a distance of $\dfrac{2C_\theta}{3}$ between centres. For a round branch $C_\theta = 1.7r_o = 2C_x$.

**G.2.3.3** *Longitudinal moments.* Similarly, a longitudinal moment, applied to an area $2C_\theta \times C_z$ (see Figure 64) is resolved into two opposed loads:

$\pm W = \dfrac{1.5M}{C_z}$ acting on rectangles of sides $2C\phi \times 2C_x$,

where $C_x = \dfrac{C_z}{6}$, which are separated by a distance of $\dfrac{2C_z}{3}$ between centres. For a round branch $C_z = 1.7r_o = 2C_\phi$.

**G.2.3.4** *Maximum stresses.* The maximum stresses due to the moment occur at the outer edges of the actual loaded area. The circumferential and longitudinal moments and membrane forces are given by:

$M_\phi = M_{\phi 1} - M_{\phi 2}$

$M_x = M_{x1} - M_{x2}$

$N_\phi = N_{\phi 1} - N_{\phi 2}$

$N_x = N_{x1} - N_{x2}$

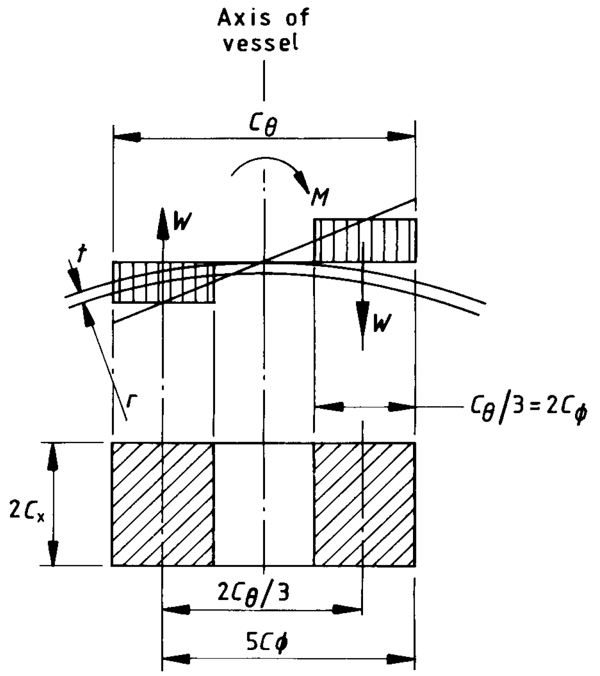



**Figure 63 — Circumferential moment**

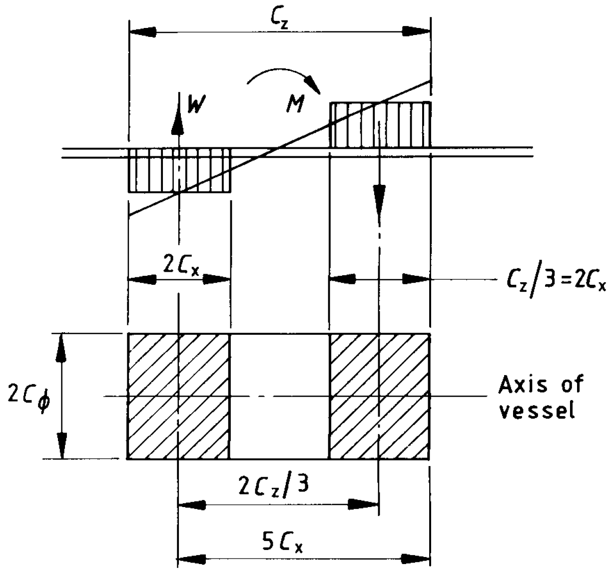



**Figure 64 — Longitudinal moment**

The quantities with subscript 1 are equal to those for a load $W$ distributed over an area of $2C_\phi \times 2C_x$ and are found from Figure 48, Figure 49, Figure 50 and Figure 51.

Quantities with subscript 2 are equal to those due to a similar load at a distance $x = 5C_x$ from the centre of the loaded area for a longitudinal moment or at an angle of $\phi_1 = 5\,\frac{C_\phi}{r}$ from the radius through the centre of the loaded area for a circumferential moment. These can be neglected if the value of $K_2$, from Table 18, corresponding to the value of $2C_x/L$ for a longitudinal moment, or that of $K_1$ corresponding to the value of $2C_x/L$ for a circumferential moment, is less than 5.0. Otherwise they are found as follows.

a) *For a longitudinal moment*

1) Take $x/C_x$ = 5.0 and obtain values for a radial line from Figure 56, Figure 57, Figure 58 and Figure 59. It may be necessary to use different values of $L_e$ (see **G.2.2.2**) for the two resolved loads if the moment is distributed over an area which is *not* small compared with its distance from the nearer end of the vessel.

2) Correct these values for a total circumferential width equal to $2C_\phi$ as in the example in **G.2.2.3.3**.

b) *For a circumferential moment*

1) Find the values at the edge of the loading area $2C_\phi \times 2C_x$ from Figure 48, Figure 49, Figure 50 and Figure 51.

2) Enter the corresponding graph in Figure 52, Figure 53, Figure 54 or Figure 55 at this value.

The intercept on the curve for $2C_x/L$ gives a value of $\frac{\phi_1 r}{C_x} = Z$.

3) the values for quantities with subscript 2 are then given by the ordinate for $\frac{\phi_1 r}{C_x} = \frac{4C_\phi}{C_x} + Z$ from the same graph.

**G.2.3.5** *Rotation due to external moments.* It is sometimes required to find the rotation of a branch or bracket due to a moment applied to it. This is given approximately by

$i = \frac{3\delta_1}{C_\theta}$ for a circumferential moment or

$i = \frac{3\delta_1}{C_z}$ for a longitudinal moment, where $\delta_1$ is the deflection produced by one of the equivalent loads

$W = \frac{1.5M}{C_\theta}$ or $\frac{1.5M}{C_z}$ acting on an area of $2C_\phi \times 2C_x$ as defined in Figure 63 or Figure 64, $\delta_1$ is found from Figure 60 and Figure 61.

*Example.* A vessel is 2.5 m diameter × 4 m long × 12 mm thick; $E = 1.86 \times 10^5$ N/mm² (i.e. steel).

Find the maximum stress due to a longitudinal moment of $1.13 \times 10^6$ N·mm applied to a branch 350 mm diameter at the mid-length, and the slope of the branch.

$$C_\phi = \frac{C_z}{2} = 0.85 \times 175 \approx 150 \text{ mm}$$

$$W = \pm \frac{1.5M}{C_z} = \pm \frac{1.5 \times 1.13 \times 10^6}{2 \times 150} = \pm 5650 \text{ N}$$

$W$ acts on an area $2C_\phi \times 2C_x$, where $C_x = \frac{C_z}{6} = 50$ mm.

For this area: $\frac{C_\phi}{C_x} = \frac{6}{2} = 3$; $\frac{C_x}{r} = \frac{50}{1250} = 0.04$;

$\frac{2C_x}{L} = \frac{2 \times 50}{4000} = 0.025.$

From Figure 47, $64 \frac{r}{t}\left(\frac{C_x}{r}\right)^2 \approx 10.$

The direct effect of each load $W$ is found by interpolating for $C_\phi/C_x$ = 3.0 in the charts of Figure 48, Figure 49, Figure 50 and Figure 51 for $2C_x/L$ = 0.025 which gives:

$M_{\phi 1}/W = 0.09$; $M_{x1}/W = 0.076$; $N_{\phi 1}t/W = -0.155$;

$N_{x1}t/W = -0.14$.

The effect of one load at the outer edge of the other is found by interpolating for $64\,(r/t)\,(C_x/r)^2 = 10$, $x/C_x = 5.0$ and $2C_x/L = 0.025$ in the charts of Figure 56, Figure 57, Figure 58 and Figure 59 for a radial line load, and multiplying the results by a correction factor for the circumferential width of the load as in **G.2.2.3.3**.

The values interpolated from Figure 56 to Figure 59 denoted by subscript 3, are:

$M_{\phi 3}/W = 0.065$; $M_{x3}/W = 0.012$;

$N_{\phi 3}t/W = +0.025$; $N_{x3}t/W = -0.085$.

| Quantity | Values for $C_\phi/C_x = 0$ | Figure | Correction factor = value for $C_\phi/C_x = 3$ value for $C_\phi/C_x = 0$ |
|---|---|---|---|
| $\frac{M_{\phi 3}}{W}$ | 0.255 | Figure 48 | $\frac{0.09}{0.255} = 0.353$ |
| $\frac{M_{x3}}{W}$ | 0.16 | Figure 49 | $\frac{0.076}{0.16} = 0.475$ |
| $\frac{N_{\phi 3^t}}{W}$ | – 0.18 | Figure 50 | $\frac{-0.155}{-0.18} = 0.861$ |
| $\frac{N_{x3^t}}{W}$ | – 0.17 | Figure 51 | $\frac{-0.14}{-0.17} = 0.824$ |

Hence

$$\frac{M_{\phi 2}}{W} = +0.065 \times 0.353 = 0.023$$

$$\frac{N_{\phi 2} t}{W} = +0.025 \times 0.861 = 0.0215$$

$$\frac{M_{x2}}{W} = +0.012 \times 0.475 = 0.005$$

$$\frac{N_{x2} t}{W} = -0.085 \times 0.824 = -0.070$$

$$M_\phi = W\left(\frac{M_{\phi 1}}{W} - \frac{M_{\phi 2}}{W}\right) = 5650(0.09 - 0.023)$$

$$= 5650 \times 0.067 = 379 \text{ N·mm/mm}$$

$$M_x = W\left(\frac{M_{x1}}{W} - \frac{M_{x2}}{W}\right) = 5650(0.076 - 0.0057)$$

$$= 5650 \times 0.0703 = 396 \text{ N·mm/mm}$$

$$N_\phi = \frac{W}{t}\left(\frac{N_{\phi 1} t}{W} - \frac{N_{\phi 2} t}{W}\right)$$

$$= \frac{5650}{12}(-0.155 - 0.0215)$$

$$= 470 \times -0.1765 = -83 \text{ N/mm}$$

$$N_x = \frac{W}{t}\left(\frac{N_{x1} t}{W} - \frac{N_{x2} t}{W}\right) = \frac{5650}{12}(-0.14 + 0.07)$$

$$= 470 \times -0.07 = -33 \text{ N/mm}$$

$$\text{Maximum circumferential stress} = f_\phi = \frac{N_\phi}{t} \pm \frac{6M_\phi}{t^2}$$

$$f_\phi = -\frac{83}{12} \pm \frac{6 \times 379}{144} = -6.92 \pm 15.8$$

Maximum circumferential compressive stress = – 22.72 N/mm$^2$.

Maximum circumferential tensile stress = + 8.88 N/mm$^2$.

$$\text{Maximum longitudinal stress} = \frac{N_x}{t} \pm \frac{6M_x}{t^2}$$

$$f_x = -\frac{33}{12} \pm \frac{6 \times 396}{144} = -2.75 \pm 16.5$$

Maximum longitudinal compressive stress = – 19.25 N/mm$^2$.

Maximum longitudinal tensile stress = + 13.75 N/mm$^2$.

*Slope due to moment.* For this area $C_\phi/C_x = 3$, and from Figure 62(b) the half side of the equivalent square $C_l = 2.8C_x = 140$ mm.

In Figure 60(b):

$C_l/r = 0.112$;

$L/r = 3.2$;

$r/t = 100$;

whence $\delta Er/W = 17\ 000$

$$\therefore \delta_l = \frac{1.7 \times 10^4 \times 5650}{1.86 \times 10^5 \times 1250} = 0.414$$

and from **G.2.3.5**, the slope $i = \dfrac{3\delta_l}{C_z}$

$$= \frac{3 \times 0.414}{300}$$

$$= 0.004\ 14 \text{ radians}$$

**G.2.4** ***Local loads in spherical shells, rigid attachments***

**G.2.4.1** *General.* The methods in this clause are not considered applicable in cases where the ratio $r_o/r$ is larger than 1/3.

**G.2.4.2** *Initial development.* This clause is concerned with the stresses and deflections due to local radial loads or moments on spherical shells. Because these are local in character and die out rapidly with increasing distance from the point of application, the data can be applied to local loads on the spherical parts of pressure vessel ends as well as to complete spheres.

For convenience, the loads are considered as acting on a pipe of radius $r_o$ which is assumed to be a rigid body fixed to the sphere. This is the condition for the majority of practical cases.

Loads applied through square fittings of side $2C_x$ can be treated approximately as distributed over a circle of radius $r_o = C_x$.

Loads applied through rectangular brackets of sides $2C_x$ and $2C_\phi$ can be treated approximately as distributed over a circle of radius $r_o = \sqrt{C_x C_\phi}$.

The following forces and moments are set up in the wall of the vessel by any local load or moment.

a) *Meridional moment* $M_x$: acting per unit width on a normal section, formed by the intersection of shell with a cone of semi-vertex angle:

$\phi = \sin^{-1}(x/r)$ (see Figure 66 and Figure 69).

b) *Circumferential moment* $M_\phi$: acting per unit width on a meridional section passing through the axis of the shell and the axis of the branch.

c) *Meridional membrane force*: acting per unit width on a normal section as for the meridional moment $M_x$.

d) *Circumferential membrane force*: acting per unit width on a meridional section as defined for the circumferential moment $M_\phi$.

A moment is considered as positive if it causes compression at the outside of the vessel.

A membrane force is considered as positive if it causes tension in the vessel wall.

A deflection is considered positive if it is away from the centre, of the sphere.

These forces and moments and the deflection of the shell due to the load can be found in terms of the non-dimensional parameters:

$$s = \frac{1.82x}{\sqrt{rt}}$$

and

$$u = \frac{1.82r_o}{\sqrt{rt}}$$

The parameter *s* defines the position in the shell at which the force, moment or deflection is required.

The parameter *u* defines the area over which the load is distributed.

These two factors can be found quickly from the chart in Figure 65, given *x*, $r_o$ and the ratio *r*/*t*.

The charts in **G.2.4.3** and **G.2.4.4** (Figure 67 to Figure 71) give graphs of non-dimensional functions of the deflection, forces and moments listed in a) to d) plotted against the parameter *s* for given values of *u* which have been derived from references [22] and [23].

The full curves in each set of graphs give conditions at the edge of the loaded area where *u* = *s*. The most unfavourable combination of bending and direct stresses is usually found here.

The dotted curves for particular values of *u* give conditions at points in the shell away from the edge of the loaded area where *x* is greater than $r_o$ and *u* is therefore less than *s*.

Since the charts are non-dimensional they can be used in any consistent system of units.

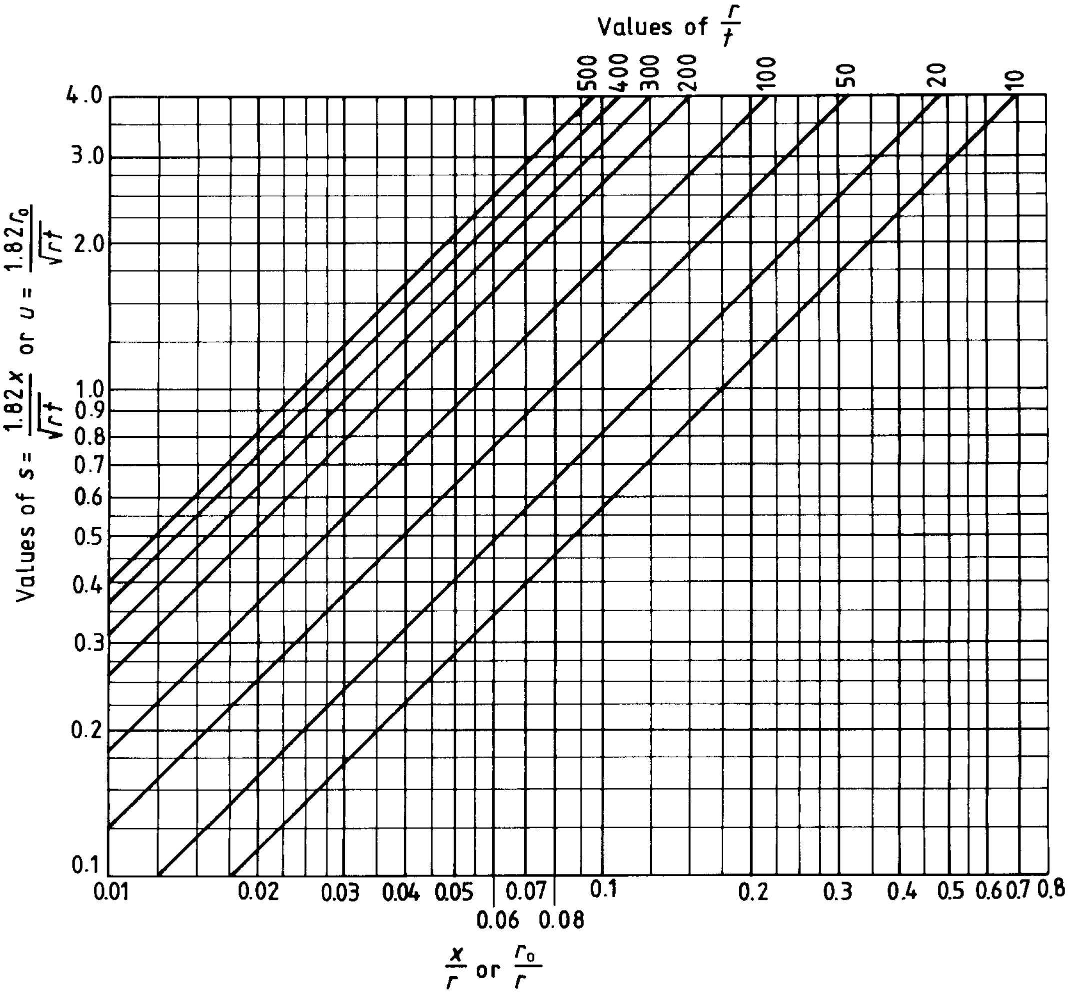



**Figure 65 — Chart for finding *s* and *u***

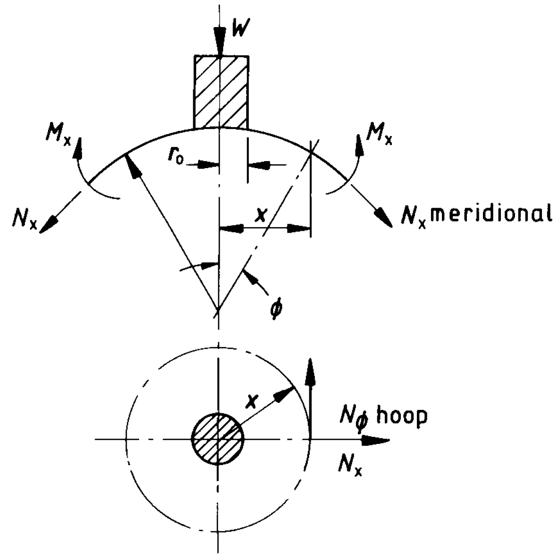



**Figure 66 — Spherical shell subjected to a radial load**

The stresses and deflections found from these charts will be reduced by the effect of internal pressure but this reduction is small and can usually be neglected in practice. (See references [23] and [24].)

**G.2.4.3** *Stresses and deflections due to radial loads*. Figure 66 shows a radial load applied to a spherical shell through a branch of radius $r_o$.

The deflections, moments and membrane forces due to the load $W$ can be found as follows from Figure 67 and Figure 68. For explanation of these curves see **G.2.4.4**. For an example of their use see **G.2.4.5**.

a) Deflection from Figure 67 and the relation:

$$\delta = \text{ordinate of curve} \times \frac{Wr}{Et^2}$$

b) Meridional moment $M_x$ per unit width from Figure 68 and the relation:

$M_x$ = ordinate of $M_x$ curve × $W$.

c) Circumferential moment $M_\phi$ per unit width from Figure 68 and the relation:

$M_\phi$ = ordinate of $M_\phi$ curve × $W$.

d) Meridional membrane force $N_x$ per unit width from Figure 68 and the relation:

$N_x$ = ordinate of $N_x$ curve × $W/t$.

e) Circumferential membrane force $N_\phi$ per unit width from Figure 68 and the relation:

$N_\phi$ = ordinate of $N_\phi$ curve × $W/t$.

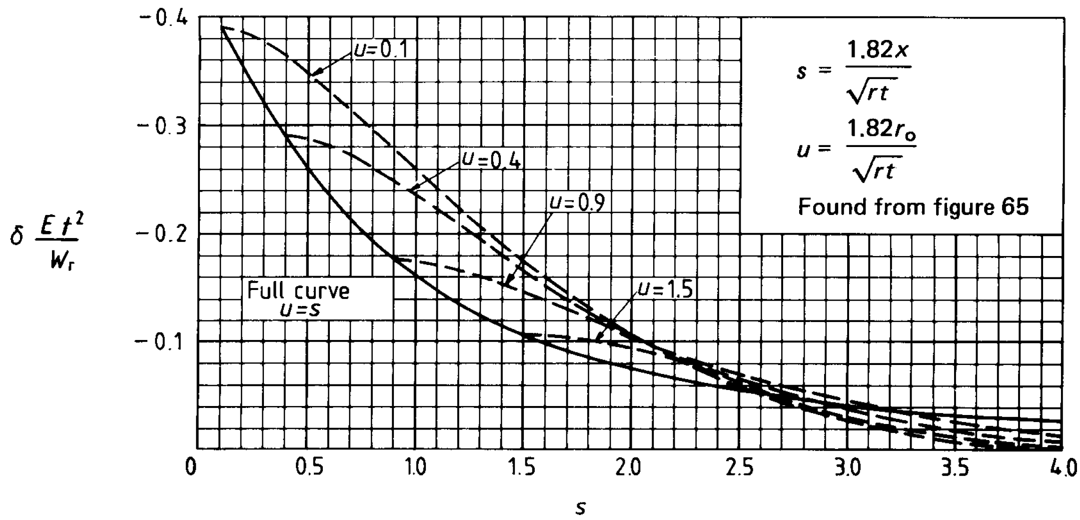



**Figure 67 — Deflections of a spherical shell subjected to a radial load, *W***

**G.2.4.4** *Stresses, deflections and slopes due to an external moment.* Figure 69 shows an external moment applied to a spherical shell through a branch of radius $r_o$.

In this case the deflections, moments and membrane forces depend on the angle $\theta$ as well as on the distance $x$ from the axis of the branch. They can be found as follows from Figure 70 and Figure 71. For explanation of these curves see **G.2.4.2**.

a) Deflections from Figure 67 and the relation:

$$\delta = \text{ordinate of curve} \times \frac{M \cos \sqrt{\frac{r}{t}}}{Et^2}$$

b) Meridional moment $M_x$ per unit width from Figure 71 and the relation:

$$M_x = \text{ordinate of } M_x \text{ curve} \times \frac{M \cos \theta}{\sqrt{rt}}$$

c) Circumferential moment $M_\phi$ per unit width from Figure 68 and the relation:

$$M_\phi = \text{ordinate of } M_\phi \text{ curve} \times \frac{M \cos \theta}{\sqrt{rt}}$$

d) Meridional membrane force $N_x$ per unit width from Figure 71 and the relation:

$$N_x = \text{ordinate of } N_x \text{ curve} \times \frac{M \cos \theta}{t\sqrt{rt}}$$

e) Circumferential membrane force $N_\phi$ per unit width from Figure 71 and the relation:

$$N_\phi = \text{ordinate of } N_\phi \text{ curve} \times \frac{M \cos \theta}{t\sqrt{rt}}$$

Equal and opposite maximum values of all the above quantities occur in the plane of the moment, i.e. where $\theta$ (see Figure 69) = 0° and $\theta$ = 180°.

The slope of the branch due to the external moment is found from:

$$i_b = \frac{\delta_l}{r_o}$$

where $\delta_l$ is the maximum deflection at the edge of the branch for $\theta = 0$ and $u = s$, i.e.:

$$\delta_l = \frac{M\sqrt{\frac{r}{t}}}{Et^2} \times \text{ordinate of full curve in Figure 70 for } x = r_o.$$

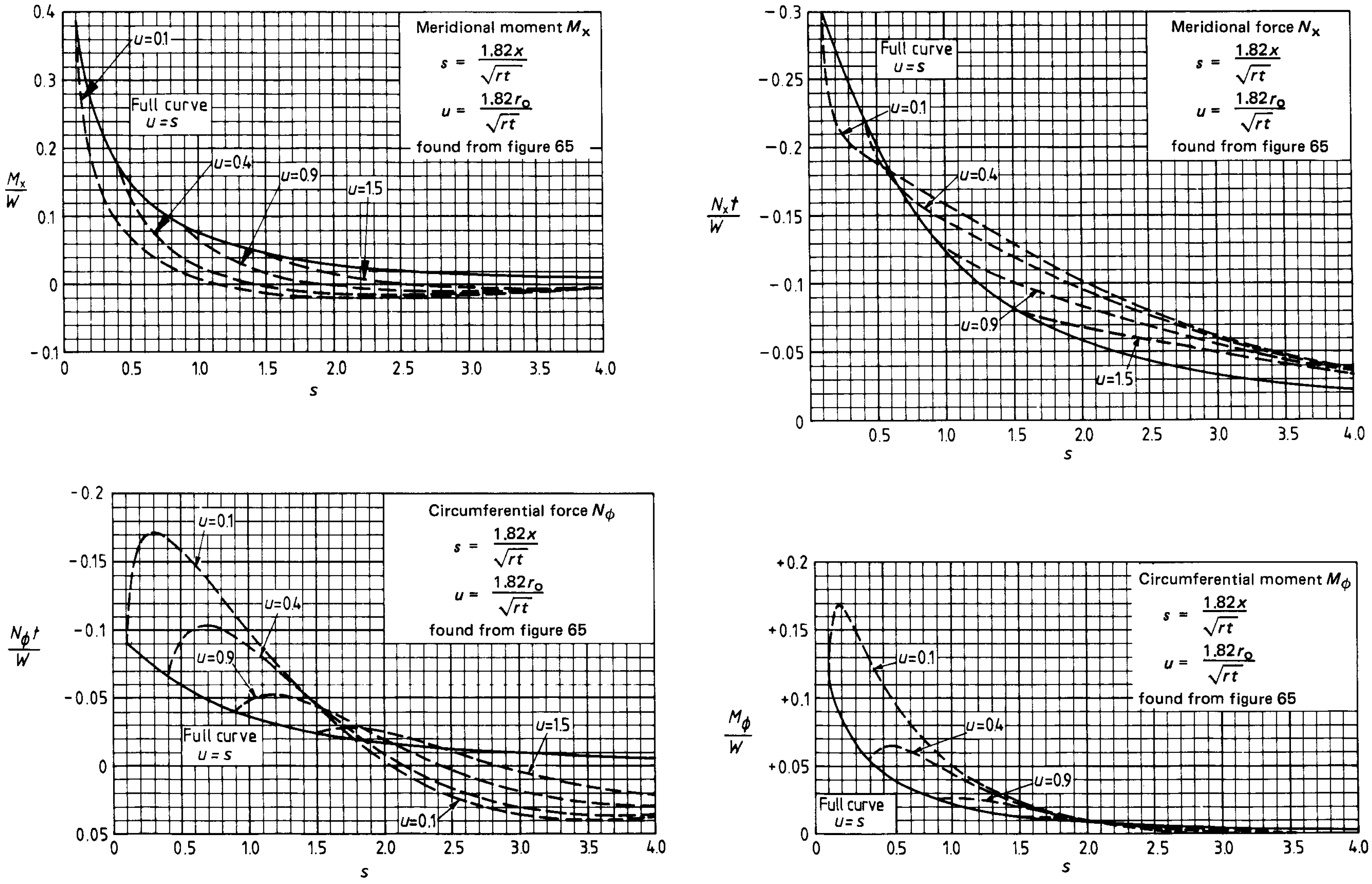



**Figure 68 — Moments and membrane forces in a spherical shell subjected to a radial load, *W***

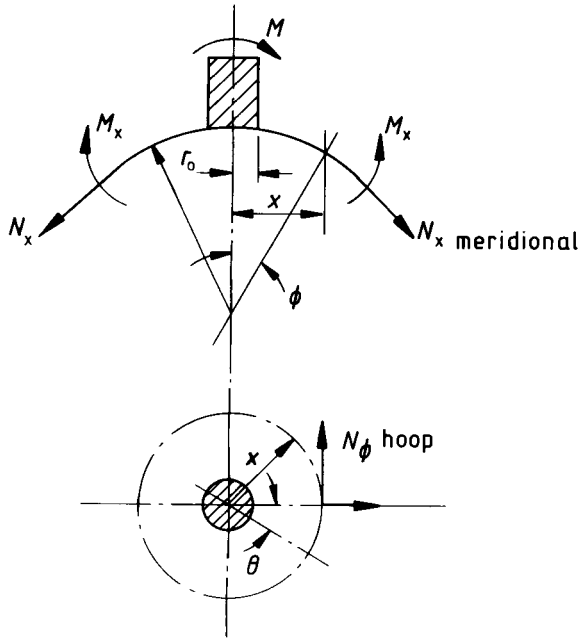



**Figure 69 — Spherical shell subjected to an external moment**

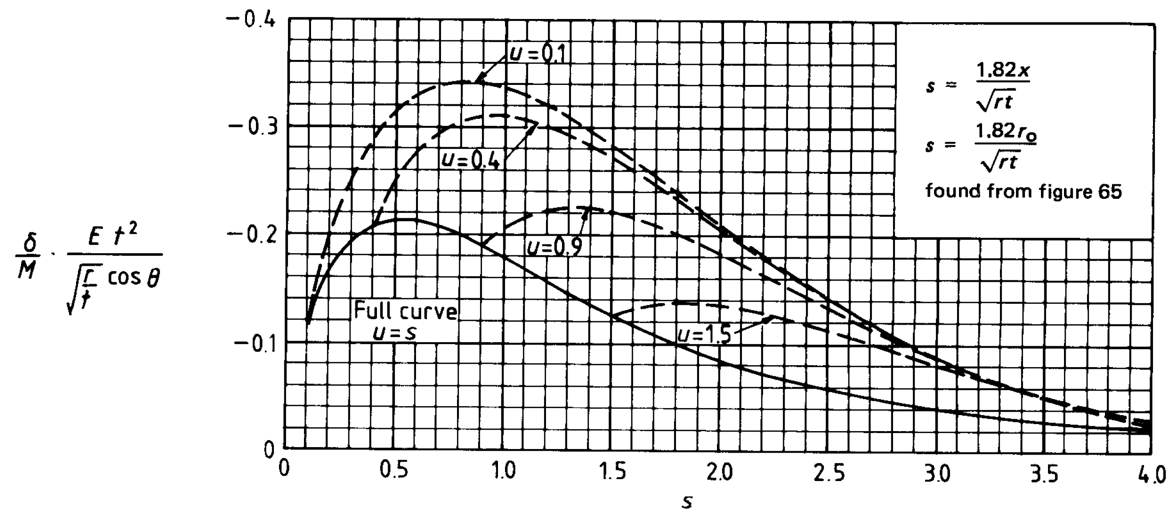



**Figure 70 — Deflections of a spherical shell subjected to an external moment, *M***

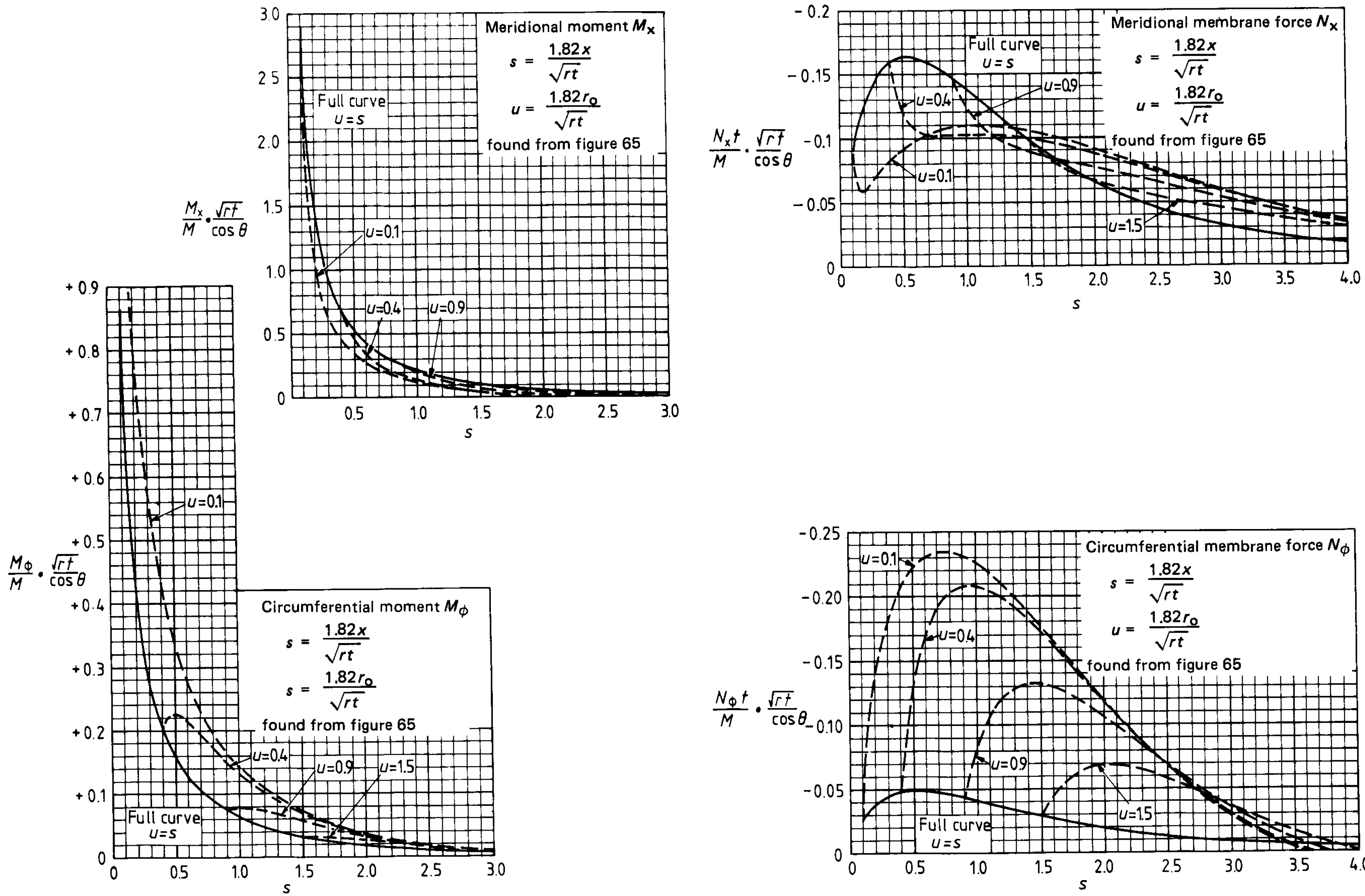



**Figure 71 — Moments and membrane forces in a spherical shell subjected to an external moment, *M***

**G.2.4.5** *Examples*

**G.2.4.5.1** A load of 4 500 N is applied to a sphere 2.5 mm diameter and 12.5 mm thick through a branch 150 mm diameter. ($E = 1.86 \times 10^5$ N/mm², i.e. steel.)

Find the deflection and the stresses:

a) next to the branch;

b) 225 mm from the centre of the branch.

$$\frac{r}{t} = \frac{1250}{12.5} = 100; \quad \frac{r_o}{r} = \frac{75}{1250} = 0.06.$$

a) *Next to the branch*

$s = u = 1.09$ (from Figure 65)

Ordinate of full curve in Figure 67 = – 0.145.

$$\therefore \text{Deflection} = -0.145 \times \frac{Wr}{Et^2}$$

$$= \frac{-0.145 \times 4500 \times 1250}{1.86 \times 10^5 \times (12.5)^2} = 0.0281 \text{ mm}$$

Ordinate of full $M_x$ curve in Figure 68 = + 0.067

∴ Meridional moment $M_x = +0.067W = 301$ N·mm/mm

Ordinate of full $M_\phi$ curve in Figure 68 = + 0.02

∴ Circumferential moment $M_\phi = +0.02W$

$= 90$ N·mm/mm

Ordinate of full $N_x$ curve in Figure 68 = – 0.11

$$\therefore \text{Meridional membrane force } N_x = \frac{-0.11W}{t}$$

$$= \frac{-0.11 \times 4500}{12.5} = -39.6 \text{ N/mm}$$

Ordinate of full $N_\phi$ curve in Figure 68 = – 0.034

$$\therefore N_\phi = \frac{-0.034W}{t} = \frac{-0.034 \times 4500}{12.5} = -12.2 \text{ N/mm}$$

The resulting meridional stresses are given by:

$$f_x = \frac{N_x}{t} \pm \frac{6M_x}{t^2} = \frac{-39.6}{12.5} \pm \frac{6 \times 301}{(12.5)^2}$$

∴ At the outside $f_x = -3.17 - 11.5 = -14.67$ N/mm² (compression)

At the inside $f_x = -3.17 + 11.5 = +8.33$ N/mm² (tension)

The resulting circumferential stresses are given by:

$$f_\phi = \frac{N_\phi}{t} \pm \frac{6M_\phi}{t^2} = \frac{-12.2}{12.5} \pm \frac{6 \times 90}{(12.5)^2}$$

∴ At the outside $f_x = -0.98 - 3.46 = -4.44$ N/mm² (compression)

At the inside $f_\phi = -0.98 + 3.46 = +2.48$ N/mm² (tension)

b) *225 mm from the centre of the branch*

$$u = 1.09 \text{ as before; } \frac{x}{r} = \frac{225}{1250} = 0.18;$$

from Figure 65, $s = 3.25$

Interpolating between the dotted curves in Figure 67 at $u = 1.09$ and $s = 3.25$ gives:

$$\frac{\delta E t^2}{W r} = 0.022$$

When deflection $= -0.022 \dfrac{W r}{E t^2}$

$$= \frac{-0.022 \times 4500 \times 1250}{1.86 \times 10^5 \times (12.5)^2} = -0.004\ 25 \text{ mm}$$

Interpolating similarly in Figure 68 gives:

$$\frac{M_x}{W} - 0.01;\quad \frac{M_\phi}{W} = +0.005;$$

$$\frac{N_x t}{W} = -0.04;\quad \frac{N_\phi t}{W} = +0.015.$$

Whence;

meridional moment $M_x = -45$ N mm/mm

circumferential moment $M_\phi = +22.5$ N mm/mm

meridional membrane force $N_x = -14.4$ N/mm

circumferential membrane force $N_\phi = +6.25$ N/mm

The resulting meridional stresses are:

$$\text{at the outside } f_x = \frac{-14.4}{12.5} + \frac{6 \times 45}{(12.5)^2}$$

$$= -1.15 + 1.73 = +0.58 \text{ N/mm}^2;$$

$$\text{at the inside } f_x = -1.15 - 1.73 = -2.88 \text{ N/mm}^2.$$

The resulting circumferential stresses are:

$$\text{at the outside } f_\phi = \frac{+6.25}{12.5} - \frac{6 \times 22.5}{(12.5)^2}$$

$$= +0.5 - 0.865 = -0.365 \text{ N/mm}^2;$$

$$\text{at the inside } f_\phi = +0.5 + 0.865 = +1.365 \text{ N/mm}^2.$$

Hence the deflection and stresses due to the load are negligible at 225 mm from the centre of the branch, which illustrates the local nature of the stresses.

**G.2.4.5.2** A moment of $1.13 \times 10^5$ N mm is applied to the branch in example **G.2.4.5.1**. Find the maximum deflection, the maximum stresses next to the branch, and the rotation of the branch due to this moment, if $E = 1.86 \times 10^5$ N/mm$^2$ (i.e. steel).

As before $\frac{r}{t} = 100$; $\frac{r_o}{r} = 0.06$, and, next to the branch, $s = u = 1.09$ (from Figure 65).

The maximum stresses and deflection are at $\theta = 0$:

$\therefore \cos\theta = 1$

From Figure 70, $\delta = -0.17 \times \dfrac{M \cos\theta \sqrt{\frac{r}{t}}}{E t^2}$

$$= \frac{-0.17 \times 1.13 \times 10^5 \times 1 \times 10}{1.86 \times 10^5 \times (12.5)^2}$$

$\therefore$ Maximum deflection $= -0.0066$ mm.

The deflection at $\theta = 180°$, on the opposite side of the branch, will be + 0.0066 mm.

From Figure 71:

$$\text{meridional moment } M_x = 0.175 \times \frac{M\cos\theta}{\sqrt{rt}}$$

$$= \frac{0.175 \times 1.13 \times 10^5}{\sqrt{1250 \times 12.5}}$$

$$= 158 \text{ N·mm/mm}$$

$$\text{circumferential moment } M_\phi = 0.055 \times \frac{M\cos\theta}{\sqrt{rt}}$$

$$= 49.6 \text{ N·mm/mm}$$

$$\text{meridional membrane force } N_x = -0.129 \frac{M\cos\theta}{t\sqrt{rt}}$$

$$= -9.3 \text{ N/mm}$$

$$\text{circumferential membrane force } N_\phi = -0.039 \frac{M\cos\theta}{t\sqrt{rt}}$$

$$= -2.81 \text{ N/mm}$$

The maximum stresses are the resulting meridional stresses given by:

$$f_x = \frac{N_x}{t} \pm \frac{6M_x}{t^2} = \frac{-9.3}{12.5} \pm \frac{6 \times 158}{(12.5)^2}$$

∴ At the outside $f_x = -0.74 - 6.04 = -6.78$ N/mm² (compression).

At the inside $f_x = -0.74 + 6.04 = +5.3$ N/mm² (tension).

The slope of the branch due to this moment will be:

$$i_b = \frac{\delta_l}{r_o} = \frac{0.0066}{75} = 8.8 \times 10^{-5} \text{ radians}$$

**G.2.5** ***Local loads on spherical shell nozzle attachments***

**G.2.5.1** *General.* The method of calculating local stress levels at a nozzle junction is based on data given in reference [19]. Using this data it is possible to estimate the maximum stress which can occur at a sphere/nozzle attachment due to the application of internal pressure, thrust, external moment and shear force. The method covers both flush and protruding nozzles. In the original work the nozzle length is treated as semi-infinite without any restriction on its length. It is, however, considered necessary to stipulate a lower limit on the internal protrusion equal to $\sqrt{2rt}$. Nozzles with internal protrusion less than $\sqrt{2rt}$ should be treated as flush nozzles. In this way some additional conservatism will be introduced for those protruding nozzles where the internal projection does not satisfy this restriction.

All the stress concentration factors given in Figure 72 to Figure 79 inclusive are based on the maximum principal stress theory.

The stress concentration factors given in **G.2.5.2** to **G.2.5.7** are based on data obtained for a sphere of constant thickness $T'$, whereas in practice $T'$ is looked upon as the local shell thickness adjacent to the nozzle, the main vessel being of a smaller thickness $T$. For these curves to be valid the thickness of the shell should not be reduced to $T$ within a distance $H$, where $H$ is measured from the outer surface of the branch piece or the bore of the opening, if no branch is fitted, and is the smaller of $H = D_b/2$ and $H = \sqrt{DT'}$.

Work in progress shows that when the vessel thickness is reduced from $T'$ to $T$ at a distance $H$ from the nozzle, higher stresses than those given in Figure 72 to Figure 79 inclusive may occur for small values of $\rho$ and high values of $t/T'$. Further guidance cannot be given at the present stage.

This procedure provides a method of computing maximum stresses which occur in the shell rather than in the nozzle. In some instances calculated stresses may be higher in the nozzle wall than in the vessel shell, especially for very thin nozzles. These are not considered for the reasons stated in reference [25].

For the purposes of **G.2.5** to **G.2.6**, which are applicable to radial nozzles only, the following additional notation applies:

$M$ is the external moment applied at a nozzle (in N·mm)

$P$ is the internal pressure (in $N/mm^2$)

$Q$ is the radial thrust applied at a nozzle

$R$ is the mean radius of spherical shell (in mm)

$r$ is the mean radius of nozzle (in mm)

$S$ is the shear load applied at a nozzle (in N)

$T'$ is the local wall thickness of shell, adjacent to nozzle (in mm)

$t$ is the local wall thickness of nozzle (in mm)

$\rho$ is the non-dimensional parameter $= \frac{r}{R}\sqrt{\frac{R}{T'}}$

$\sigma_{max}$ is the maximum stress due to local loading

**G.2.5.2** *Maximum stress at a sphere/nozzle junction due to application of internal pressure.* Figure 72 gives plots of stress concentration factors (SCF) against the non-dimensional parameter $\rho$, for various nozzle/shell wall $t/T'$ ratios for flush nozzles. The maximum stress, $\sigma_{max}$, is then calculated by multiplying the SCF thus obtained by the nominal pressure stress given by $\frac{PR}{2T'}$, i.e.:

$$\sigma_{max} = \text{SCF} \times \frac{PR}{2T'}$$

Figure 73 gives similar plots for protruding nozzles.

Before using Figure 73 a check should be made to ensure that the internal nozzle protrusion is equal to or greater than $\sqrt{2rt}$; if it is not, Figure 72 should be used as for a flush nozzle.

**G.2.5.3** *Maximum stress at a sphere/nozzle junction due to application of radial load or thrust.* Figure 74 gives plots of stress concentration factors against the non-dimensional parameter $\rho$ for flush nozzles. The maximum stress is calculated by multiplying the SCF obtained from Figure 74 by:

$$\frac{Q}{2\pi r T'}\sqrt{\frac{R}{T'}}, \text{ i.e.}$$

$$\sigma_{max} = \text{SCF} \times \frac{Q}{2\pi r T'}\sqrt{\frac{R}{T'}}$$

Figure 75 gives similar plots for protruding nozzles.

Before using Figure 75 a check should be made to ensure that the internal nozzle protrusion is equal to or greater than $\sqrt{2rt}$; if it is not, Figure 74 should be used for obtaining the SCF.

**G.2.5.4** *Maximum stress at a sphere/nozzle junction due to application of external moment.* For flush nozzles the maximum stress at a sphere/nozzle junction can be determined by using Figure 76. The first step is to read off the SCF for the appropriate vessel nozzle geometry. The maximum stress is then obtained by multiplying the SCF thus obtained by the factor:

$$\frac{M}{\pi r^2 T'}\sqrt{\frac{R}{T'}}, \text{ i.e.}$$

$$\sigma_{max} = \text{SCF} \times \frac{M}{\pi r^2 T'}\sqrt{\frac{R}{T'}}$$

For protruding nozzles, the equivalent maximum stress is determined in the same manner by using Figure 77 for obtaining the SCF.

A check should be made to ensure that the length of the internal nozzle protrusion is equal to or greater than $\sqrt{2rt}$; if it is not, the procedure for flush nozzles should be used.

**G.2.5.5** *Maximum stress at a sphere/nozzle junction due to application of shear load.* Figure 78 should be used for determining the SCF for flush nozzles. The maximum stress, $\sigma_{max}$, is then calculated by multiplying the SCF obtained in the first step by the factor $S/\pi r\ T'$, i.e.:

$$\sigma_{max} = SCF \times \frac{S}{2\pi r T'}$$

For protruding nozzles the procedure is identical with the aforementioned procedure except that Figure 79 should be used.

A check should be made to ensure that the length of the internal nozzle protrusion is equal to or greater than $\sqrt{2rt}$.

If this condition is not satisfied the procedure given for flush nozzles should be used.

**G.2.5.6** *Maximum stress at a sphere/nozzle junction under combined loading.* For a conservative estimate of the stresses occurring under the action of combined loading the maximum stresses obtained from each of the individual loadings should be added together. This will always be conservative because the maximum stresses for individual loadings may occur at different locations and different direction ($\sigma_\theta$ and/or $\sigma_z$).

**G.2.5.7** *Stresses away from the loaded area.* The method given in **G.2.5** for calculating local stresses at a sphere/nozzle junction caters for the maximum stress levels only. No information is given on stresses away from the loaded area.

Stress distributions in the vicinity of the sphere/nozzle junction are required in cases where other loaded areas are in the proximity to the one under consideration. It is proposed to use the data already available in **G.2.4** to determine these stresses. The assumption here is that, although the magnitudes of local stresses may differ, the plot of stress level versus distance from loaded area remains basically similar. The stress distribution away from the loaded area can then be calculated by the procedure outlined in **G.2.4** and the values so obtained are to be multiplied by a factor $K$, where $K$ is the ratio of $\sigma_{max}$, as determined in **G.2.5**, to the stress at the edge of the attachment, as calculated in **G.2.4**, where applicable.

This method is conservative, but an alternative approach is available in reference [26].

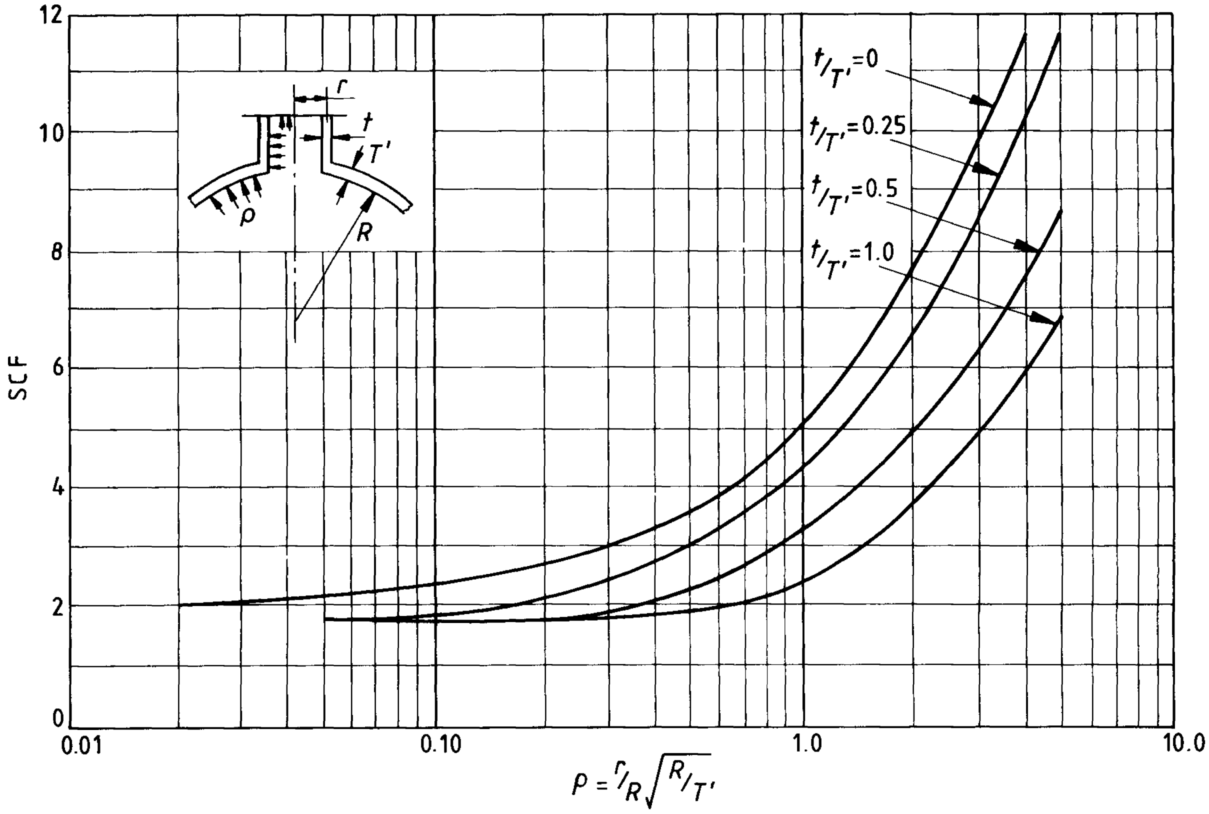



**Figure 72 — Maximum stress in sphere for internal pressure (flush nozzles)**

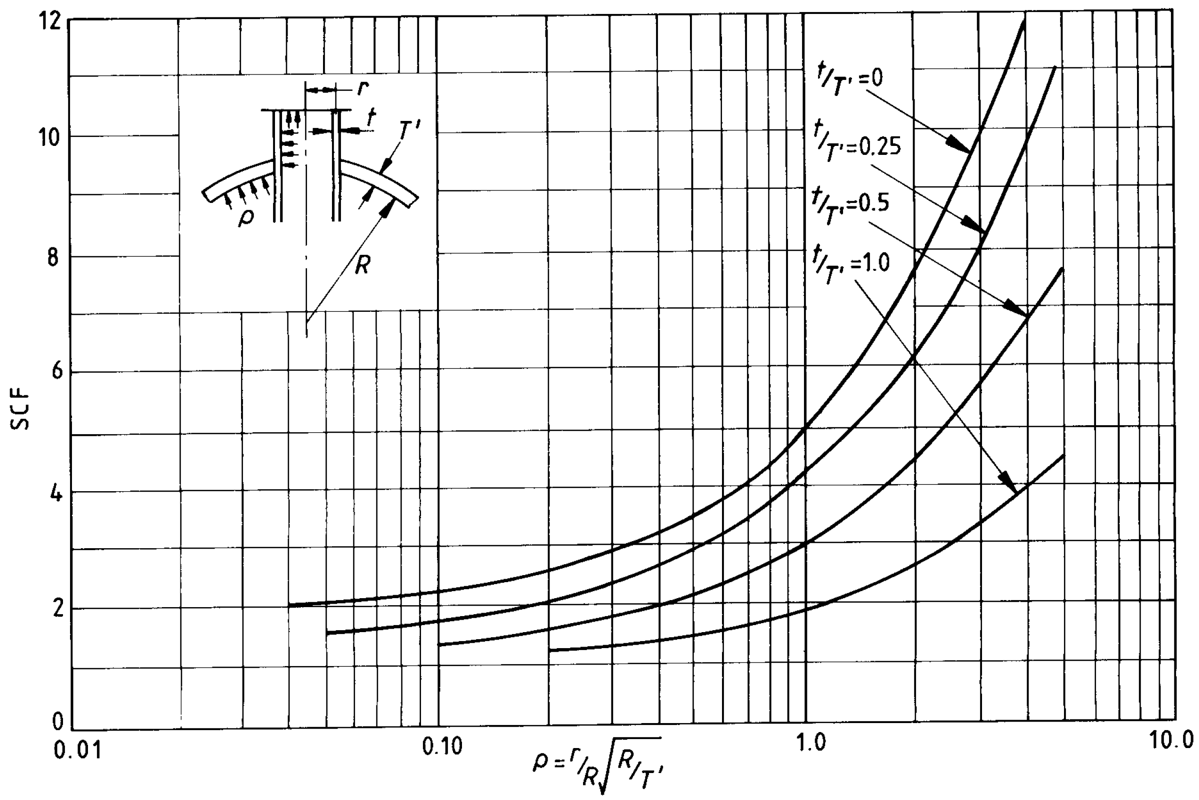



Figure 73 — Maximum stress in sphere for internal pressure (protruding nozzles)

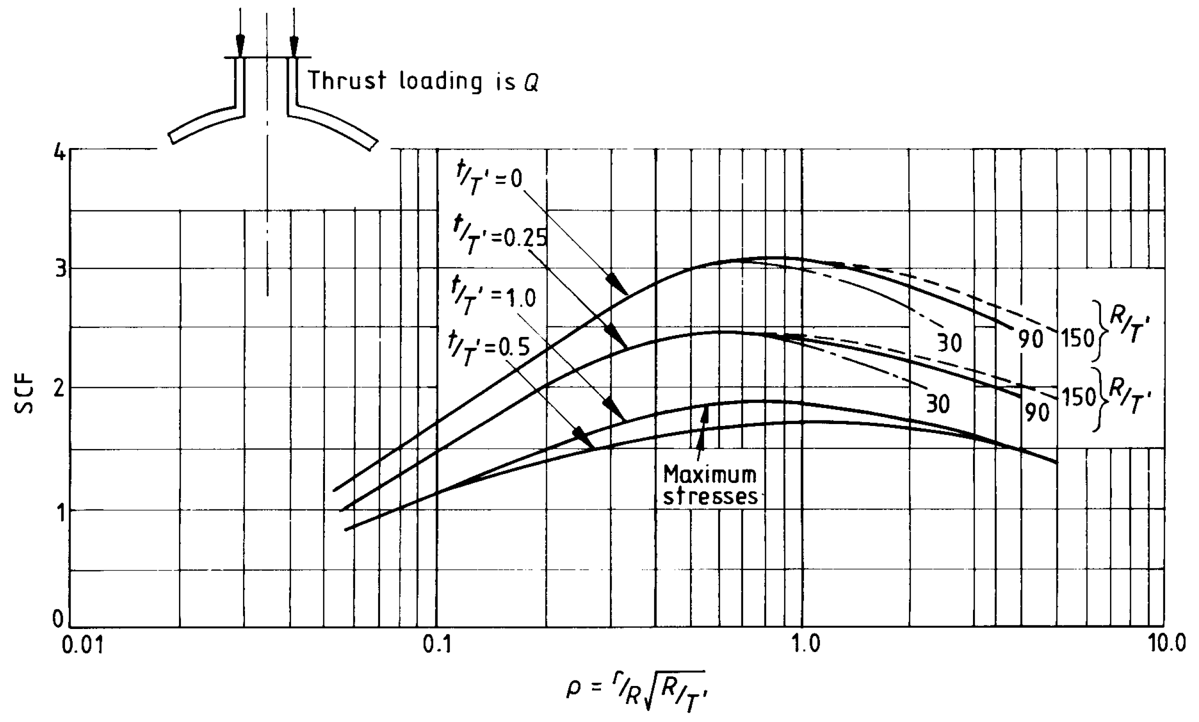



Figure 74 — Maximum stress for thrust loading (flush nozzles)

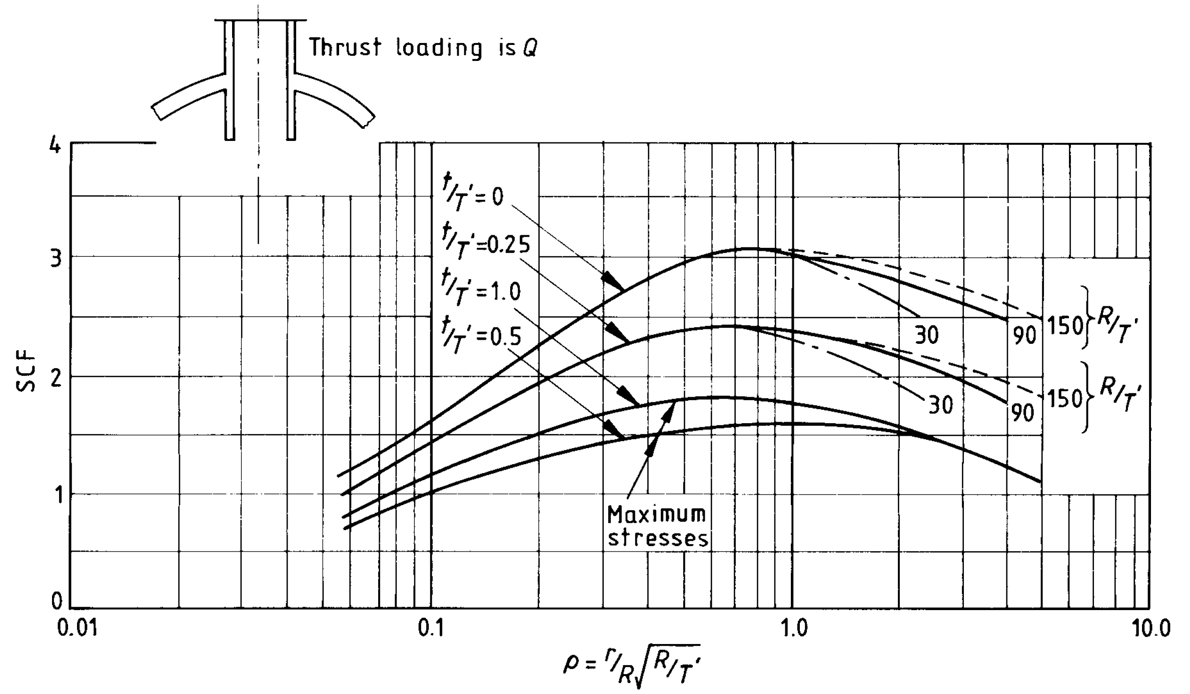



Figure 75 — Maximum stress in sphere for thrust loading (protruding nozzles)

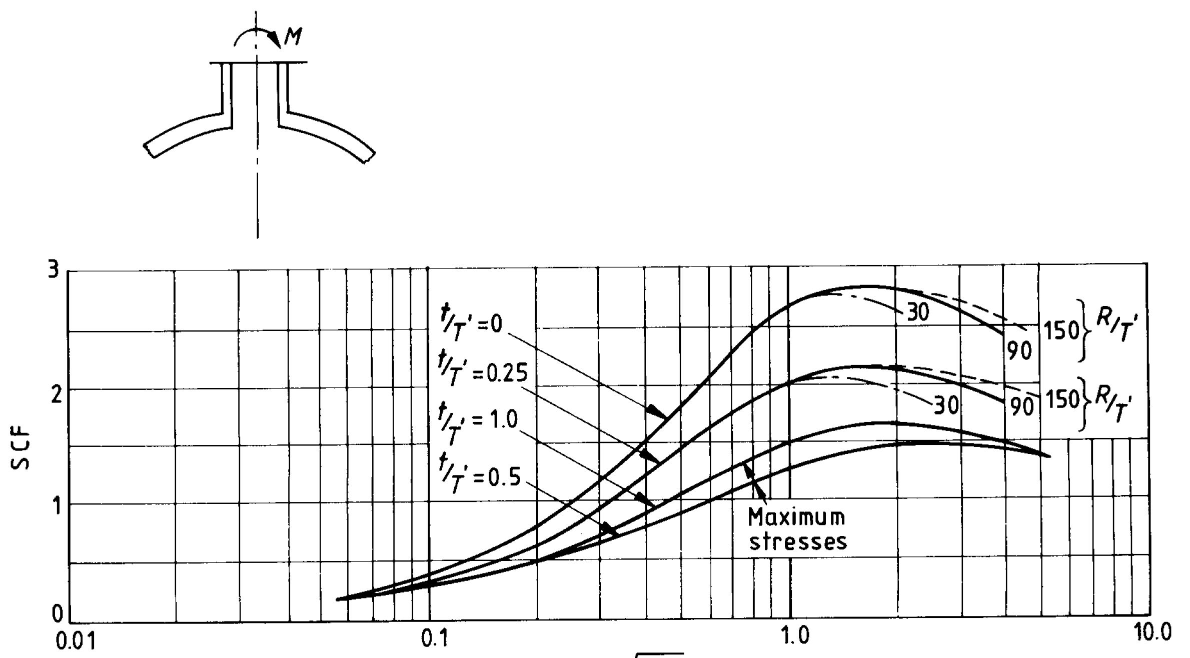



Figure 76 — Maximum stress in sphere for moment loading (flush nozzles)

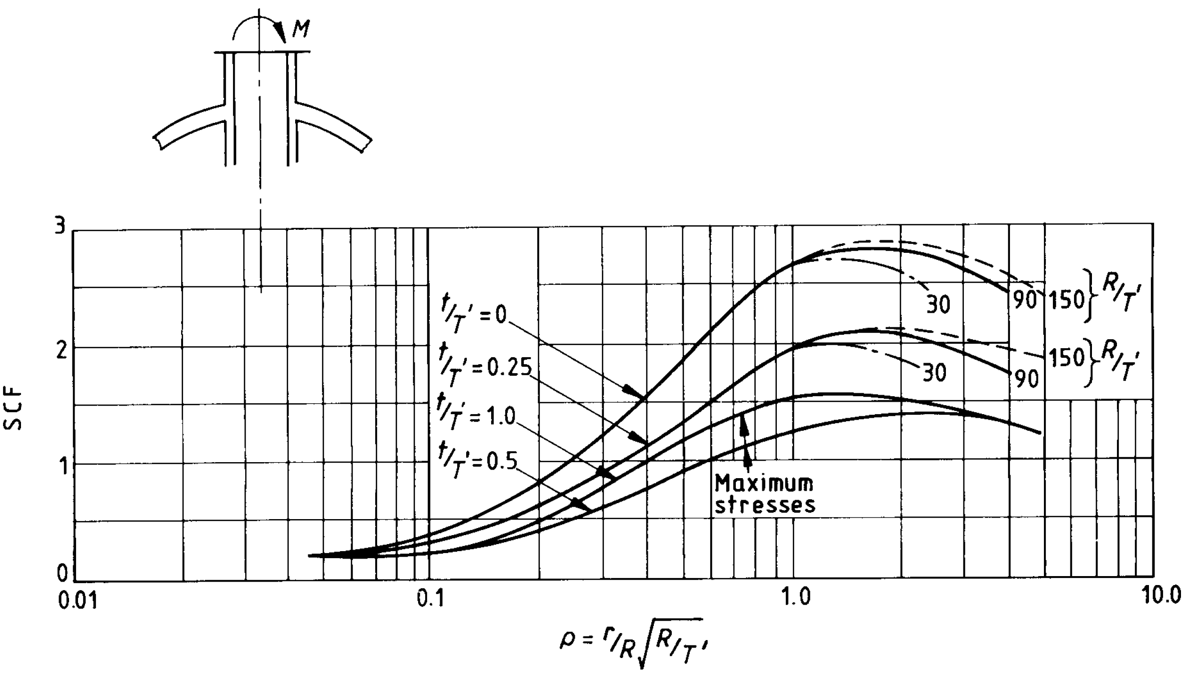



Figure 77 — Maximum stress in sphere for moment loading (protruding nozzles)

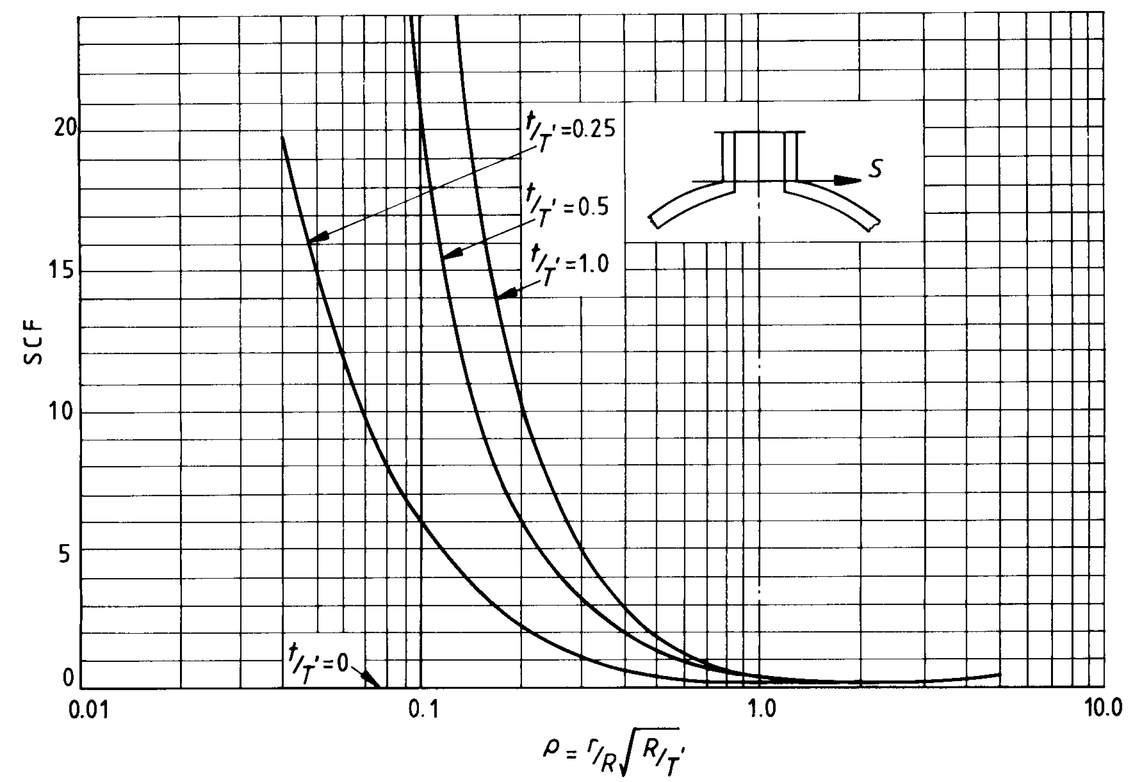



Figure 78 — Maximum stress in sphere for shear loading (flush nozzles)

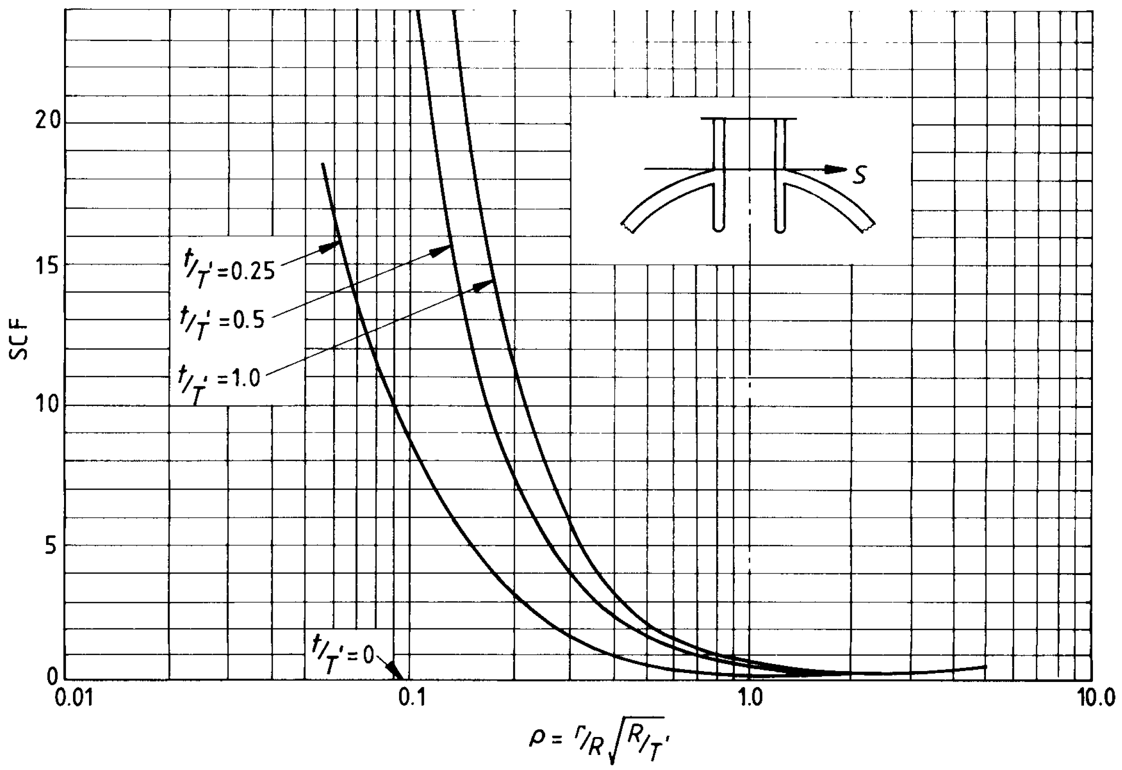



**Figure 79 — Maximum stress in sphere for shear loading (protruding nozzles)**

**G.2.6** ***The effect of external forces and moments at branches***

Large external forces and moments can be applied to the branches of vessels by the thermal movements of pipework.

The stresses due to these are likely to be greatly over-estimated if the forces in the pipe system are determined by assuming that the connection to the vessel is equivalent to an anchor in the pipe system.

More accurate values of the terminal forces and moments can be found if the deflection due to a unit radial load and the slopes due to unit longitudinal and circumferential moments distributed over the area of the branch and its reinforcement are known.

These can be found for a given vessel and branch by the methods given in **G.2.2.4** and **G.2.3** for cylindrical vessels and by methods given in **G.2.4.3** and **G.2.4.4** for spherical vessels. Recent experiments in the USA, discussed in reference [21], have shown that slopes and deflections calculated in this way are sufficiently accurate for practical purposes except that the slope of a branch due to a circumferential moment is about 75 % of the calculated value because of the effect of local stiffening of the branch.

When the loads from the pipework are known, the local stresses in the vessel shell can be found by the methods given in **G.2**, except that, in a branch with an external compensating ring of thickness $t_1$ subject to a circumferential moment there is an additional circumferential moment in the shell at the edge of the reinforcing ring to $N_\phi t_1/4$ and reference [21] recommends that this amount should be added to the value of $M_\phi$ calculated in **G.2.3**.

This correction and that to the slope of the branch given above apply only to circumferential moments and are due to the effect of the rigidity of the attachment of the branch which has little influence on the effect of longitudinal moments.

The tension at the inside of the shell due to the local circumferential bending moment $M_\phi$ is added to the circumferential membrane stress due to internal pressure, but this stress will not be present when the vessel is under hydraulic test.

## Appendix H Recommended supports for vertical vessels and tanks

The recommended supports for vertical vessels and tanks are either:

a) a suitably prepared concrete plinth; or

b) a steel plate supported on an adequate grillage and of sufficient thickness to ensure that local strains in the flat bottom are not greater than the design's strain.

Levelled off earth bases whether or not in-filled with sand or other granular material are not adequate supports and should not be used.

When concrete bases are specified, departure from nominal level should not exceed ± 1 mm/m. Before bedding the tank or vessel, the concrete should be covered with bitumen sand to a thickness of 25 mm.

A suitable bitumen sand can be produced by hot mixing in the following proportions by mass:

| | |
|---|---|
| 9 % ± 0.5 % | non-toxic cut back bitumen (i.e. fluxed with kerosene and not creosote); |
| 10 % ± 1 % | filler either limestone dust passing a sieve of nominal aperture (size 75 µm in accordance with BS 410) or Portland cement; |
| 81 % ± 1.5 % | clean dry washed sand in accordance with Table 2 of BS 882:1983. |

As an alternative to the sand, crushed rock, types 1 to 7, slag or limestone in accordance with Table 53 of BS 4987:1973 may be used.

Loose sand should not be used to bed vessels on concrete bases.

## Appendix J Design of metallic lifting lugs and trunnions

**J.1** Consideration should be given to how the vessel is to be handled during loading for transport, unloading, storage, installation and removal for maintenance. Chain and wire rope should not be in direct contact with GRP.

**J.2** The number, position, orientation and type of lifting fittings, i.e. lug or trunnion, should be determined. The following cases are typical.

a) Tower: a pair of trunnions at the top end, a tailing lug (at 90° to trunnion) at the lower end (see Figure 80).

b) Tall vertical cylindrical tank: a pair of parallel lugs on a diameter on the flat roof (lifted from a spreader beam) and a tailing lug near the base (at 90° to the diameter joining the top lugs) (see Figure 81).

c) Short vertical cylindrical tank: a pair of lugs in a plane on a diameter on the flat roof (see Figure 82) *or* three equispaced radial lugs on the roof periphery (see Figure 83).

d) Horizontal tank: a pair of lugs in a plane on the top centreline (see Figure 84) *or* no lugs, if webbing slings used.

**J.3** The most severe load on each lug or trunnion should be calculated, taking account of load direction.

**J.4** The lug should be designed to match an equivalent load capacity steel shackle (BS 3032).

NOTE 1 A lug design procedure exists in DIN 28086.

The trunnion should be designed to match wire rope sling (BS 1290) or webbing sling (BS 3481-2).

NOTE 2 A trunnion design procedure exists in DIN 28085-1.

**J.5** The substructure should be designed to connect the lug or trunnion to the vessel. Generally this will be a steel ring, pad or framework to spread the load over a large area. Refer to BS 449 for welded steel structures.

**J.6** The attachment of the substructure to the GRP vessel should be checked for adequate strength in peel and shear.

**J.7** Where appropriate, local load strains in the vessel wall should be checked in accordance with Appendix G.

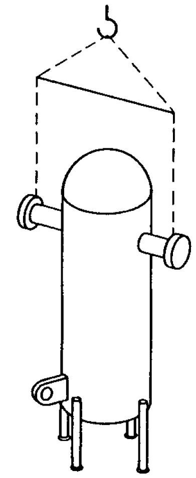


Figure 80 — Trunnions

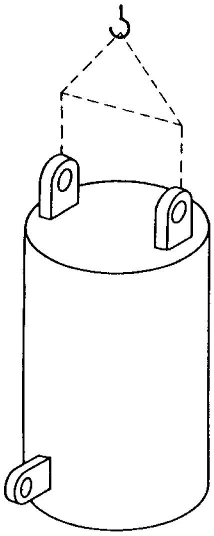


Figure 81 — Parallel lugs

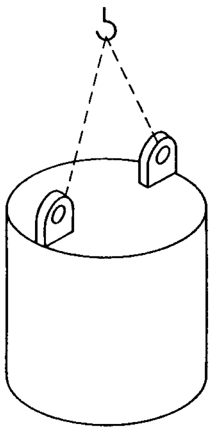


Figure 82 — Lugs in a plane on a diameter

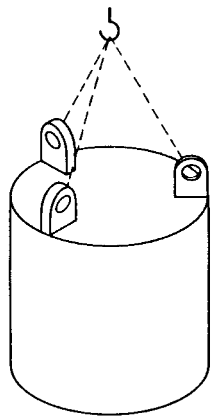


Figure 83 — Equispaced radial lugs

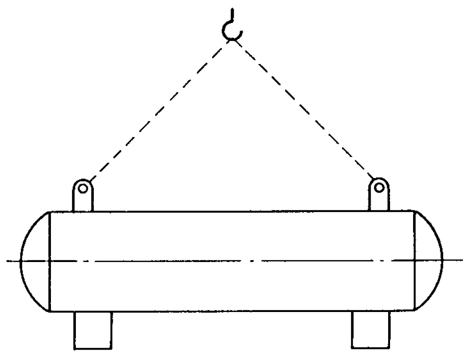


Figure 84 — Lugs in a plane on a horizontal vessel

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