G = diameter at location of gasket load reaction
= mean diameter of gasket contact face, if $b_o \leqslant 6.3$ mm
= outside diameter of gasket contact face less $2b$, if $b_o > 6.3$ mm.
$(2b)$ = effective gasket pressure width,
b = effective gasket seating width, (Table 7.2)
m = gasket factor, (Table 7.1)

7.6.1.2 *Determination of bolt load under bolting-up condition*

For initial gasket seating the minimum bolt load required W_g is given by :

$$W_g = \pi\, Gby \qquad \text{... (7.6.2)}$$

Where, y = minimum gasket seating stress, (Table 7.1)

7.6.2 Determination of minimum bolt area theoretically required, A_m

If A_0 is the bolt area required under operating condition and A_g is the area required under bolting-up condition, then,

$$A_0 = \frac{W_0}{S_0} \qquad \text{... (7.6.3)}$$

$$A_g = \frac{W_g}{S_g} \qquad \text{... (7.6.4)}$$

Where, S_0 = allowable stress for bolting material at design temperature, (Table 7.5)
S_g = allowable stress for bolting material at atmospheric temperature, (Table 7.5)

Theoretically required minimum bolt area, A_m, will be larger of A_0 and A_g. For ideal design A_0 and A_g should be approximated equal.

7.6.2.1 *Determination of actual bolt area, A_b*

Actual bolt area, A_b, will not be less than A_m to satisfy the theoretical requirement. Next, a standard bolt diameter is to be selected ; actual number of bolts should be such that it is a multiple of 4 from practical consideration and also the bolt spacing should not be too large or too small. After satisfying all these requirements A_b becomes usually larger than A_m. It will be economical if the difference is small.

To prevent damage to the gasket during bolting-up, following condition is to be satisfied.

$$\frac{A_b S_g}{\pi\, GN} < 2y \qquad \text{... (7.6.5)}$$

Where, N = actual width of the gasket in contact.

If Eq. 7.6.5 is not satisfied, gasket material should be changed.

It is to be noted that A_b is to be calculated taking root area of the bolts.

7.6.3 Flange stresses

To determine the stresses the flanges are categorized into 3 types, namely, loose-type flanges, integral-type flanges and optional-type flanges.

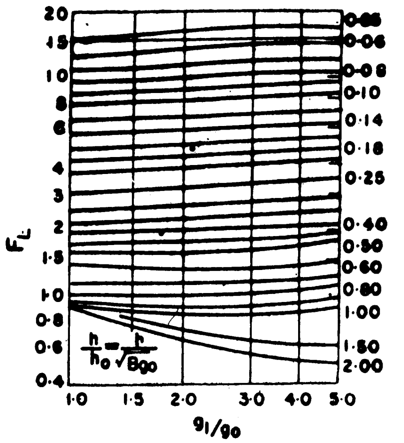

Fig. 7.8a Values of F_L
(loose hub flange factors)

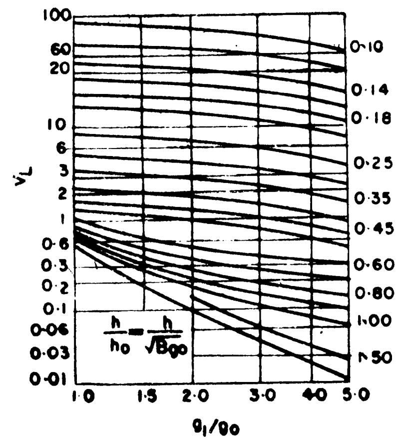

Fig. 7 8b Values of V_L
(loose hub flange factors)

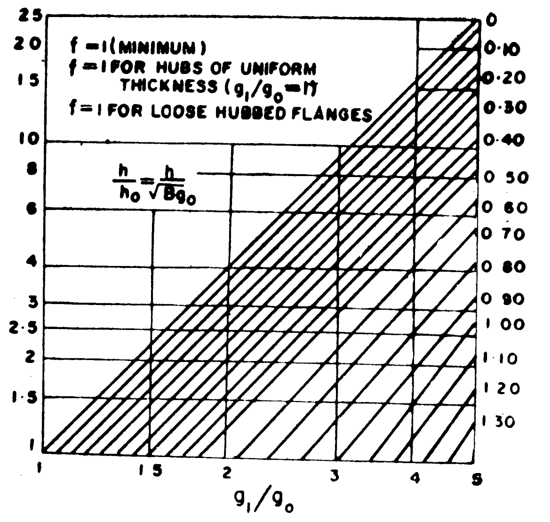

Fig. 7.9 Values of f (hub stress correction factor)

8.1.1 Critical length between stiffeners

If the stiffeners are spaced within critical length, they offer restraint to collapsing of the vessels under external pressure. Under such a condition the vessel with same thickness can sustain higher external pressure.

The expression for critical length is available in literature as given below.[1]

For steel vessels ($\mu = 0.3$)

$$L_c = 1.11\, D_o \sqrt{D_o/t} \qquad \text{... (8.1.1)}$$

8.1.2 Out-of-roundness of shells

Out-of-roundness in any form is very much detrimental to the vessel strength under external pressure. Under internal pressure out-of-roundness does not cause so much worries, but this results in increased stress concentration under external pressure. As a result a shell of elliptical shape, or a circular shell, either dented or with flat spots, is less strong under external pressure than a vessel having a true cylindrical shape. Out-of-roundness factor, U, is determined as follows :

(a) For oval shape :

$$U = \frac{2\,(D_{max} - D_{min})}{D_{max} + D_{min}} \times 100, \text{ per cent}$$

(b) For dent or flat spots :

$$U = \frac{4\,a}{D_o} \times 100, \text{ per cent}$$

Where, a = depth of dent or flat spots (maximum value is to be taken).

In case of old vessels, the largest value from (a) and (b) is to be taken for design calculation. For the new vessels, whose out-of-roundness is not known, $U = 15\%$ (minimum) is to be taken. If actually measured, smaller values can be used. In this connection the tolerances on diameter of plate shells prescribed in Indian Standard (IS : 4503 — Specification for shell and tube type heat exchanger) is shown in Table 8.1.

Table 8.1 Tolerances on Diameter of Plate Shells
(All dimensions in mm)

Nominal Diameter	*Permissible Deviation*	
	Grade I	*Grade II*
200 upto 400	± 3	—
Over 400 upto 600	± 3	+ 6 − 3
Over 600 upto 800	± 4	+ 7 − 4
Over 800 upto 1 000	± 5	+ 8 − 5
Over 1 000	± 6	+ 8 − 6

[1] L. E. Brownell and E. H. Young, "Process Equipment Design", John Wiley and Sons, Inc., New York.

From Table 8.2, $K = 9.037$ and $m = 2.62$

Substituting in Eq. 8.2.6

$$0.1 = 9.037 \times 1.67 \times 10^5 (t/D_o)^{2.62}$$

or, $t/D_o = 1.82 \times 10^{-3}$

or $t = 7.28 \times 10^{-3}$ m

Check for plastic deformation (Eq. 8.3.4)

$$p = 2 \times 70 \times 1.82 \times 10^{-3} \times \frac{1}{1 + \frac{1.5 \times 1.5 (1 - 0.2 \times 4)}{0.182}}$$

$$= 0.074 < 0.1$$

The calculated thickness is not sufficient for plastic failure. A new value of t (shell thickness) is to be found out from Eq. 8.3.4.

$$0.1 = 2 \times 70 \times \frac{t}{D_o} \times \frac{1}{1 + \frac{1.5 \times 1.5 (1 - 0.2 \times 4)}{100 \left(\frac{t}{D_o}\right)}}$$

$$= 140 \left(\frac{t}{D_o}\right) \times \frac{1}{1 + \frac{4.5 \times 10^{-3}}{(t/D_o)}}$$

Solution of this involves trial and error. To get the exact value, this can be expressed in quadratic form and the result obtained gives :

$$\frac{t}{D_o} = 2.2 \times 10^{-3}$$

or $t = 8.8 \times 10^{-3}$ m

A standard plate thickness of $t = 10$ mm is selected.

(b) Design of stiffening ring involves first to select a standard structure and then to check for required moment of inertia (Eq. 8 4.5) with the moment of inertia of the structure.

From Eq. 8.4.5

$$I = \frac{D_o^2 L \left(t + \frac{A_s}{L}\right) f}{12 E}$$

where, I = required moment of inertia

D_o = outer diameter of shell = 4 m

L = effective length of tower = 1 m

t = shell thickness = 1.0×10^{-2} m

A_s = cross-sectional area of a stiffening ring

f = allowable compressive stress = 70 MN/m^2