Consulta
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Assunto: ASME VIII
É possível conseguir uma cópia da tabela de tensões admissíveis do código ASME seção VIII div 1.
Existe cópia traduzida para português do código acima?
Vasos que operam sob vácuo podem ser dimensionados pelo código acima?

Resposta
Em resposta às suas questões:

- Tabela de tensões admissíveis do ASME VIII Div 1

O código ASME Sec II Part D - Materials and Properties - estabelece a tensão de tração admissível, em função da temperatura, para os materiais utilizados na fabricação de Vasos de pressão, projetados e construídos conforme o ASME VIII Div 1 de Div 2.
A planilha que é apresentada no petroblog/índice/categoria Análise de Tensões/post: Cálculo automático de tensão admissível conforme código ASME Sec II part D de materiais para Vasos de Pressão é uma forma automática de se conseguir o valor da tensão de tração admissível e a tensão de compressão admissível requeridas para os cálculos.

- Tradução para o português do código ASME VIII Div 1

Infelizmente não.

- Dimensionamento de vasos de pressão sob pressão externa e vácuo

No parágrafo UG-23 MAXIMUM ALLOWABLE STRESS VALUES do ASME Sec VIII Div 1 é apresentada a metodologia para a determinação tensão máxima admissível de compressão longitudinal *maximum allowable longitudinal compressive stress* ou *factor B*, que é básico para o cálculo da máxima pressão externa.
No parágrafo seguinte UG-28 THICKNESS OF SHELLS AND TUBES UNDER EXTERNAL PRESSURE é apresentada a metodologia para o cálculo da máxima pressão externa a que pode ser submetido um determinado vaso de pressão.
No Apêndice *Nonmandatory Appendix L - Examples Illustrating the Application of Code Formulas and Rules* do ASME VIII 1, há um exemplo desse cálculo, copiado a seguir.
Porém, se você analisar com atenção a planilha informada, na resposta da sua questão sobre tensão admissível, há um cálculo automático dessa máxima pressão externa.

VESSELS UNDER EXTERNAL PRESSURE

NOTE: In Subpart 3 of Section II, Part D, the lines in Fig. G express a geometrical relationship between L/D_o and D_o/t for cylindrical shells and tubes which is common for all materials. This chart is used only for determining the factor A when factor A is not obtained by formula in the special case when $D_o/t < 10$.
The remaining charts in Subpart 3 are for specific material or classes of materials and represent pseudo stress–strain diagrams containing suitable factors of safety relative both to plastic flow and elastic collapse.

L-3

L-3.1 Cylindrical Shell Under External Pressure

[An example of the use of the rules in UG-28(c)]

L-3.1.1 Given. Fractionating tower 14 ft. I.D. by 21 ft. long, bend line to bend line, fitted with fractionating trays, and designed for an external design pressure of 15 psi at 700°F. The tower to be constructed of SA-285 Gr. C carbon steel. Design length is 39 in.

L-3.1.2 Required. Shell thickness t

L-3.1.3 Solution
Step 1. Assume a thickness t = 0.3125 in. Assume outside diameter D_o = 168.625 in.

$$\frac{L}{D_o} = \frac{39}{168.625} = 0.231$$

$$\frac{D_o}{t} = \frac{168.625}{0.3125} = 540$$

Steps 2, 3. Enter Fig. G at the value of $L/D_o = 0.231$; move horizontally to the D_o/t line of 540 and read the value A of 0.0005.
Step 4, 5. Enter Fig. CS-2 at the value of $A = 0.0005$ and move vertically to the material line for 700°F. Move horizontally and read B value of 6,100 on ordinate.
Step 6. The maximum allowable external working pressure for the assumed shell thickness of 0.3125 in. is

$$P_a = \frac{4B}{3(D_o/t)} = \frac{4(6,100)}{3(540)} = 15.1 \text{ psi}$$

Since P_a is greater than the external design pressure P of 15 psi, the assumed thickness is satisfactory.

L-3.2 Spherical Shell Under External Pressure

[An example of the use of the rules in UG-28(d)]

L-3.2.1 Given. A spherical vessel having an inside diameter of 72 in., made of an aluminum alloy conforming to SB-209 Alloy 3003-0 to withstand an external design pressure of 20 psi at 100°F.

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L-3.2.2 Required. Shell thickness t

L-3.2.3 Solution

Step 1. Assume a shell thickness $t = 0.50$ in. Then

$$R_o = \frac{72}{2} + 0.5 = 36.5$$

$$A = \frac{0.125}{R_o/t} = \frac{0.125}{36.5/0.50} = 0.00171$$

Steps 2, 3. Enter Fig. NFA-1 at $A = 0.00171$ and move vertically to the material line of 100°F; move horizontally and read B value of 1780.

Step 4. The maximum allowable external working pressure for the assumed shell thickness of 0.50 in. is:

$$P_a = \frac{B}{R_o/t} = \frac{1780}{36.5/0.5} = 24.4 \text{ psi}$$

Since P_a is greater than the external design pressure P of 20 psi, the assumed shell thickness of 0.50 in. is satisfactory.

L-3.3 Cone-to-Cylinder Juncture Under External Pressure

(An Example of the Use of the Rules in 1-8)

Determine the required reinforcement of a cone-to-cylinder juncture under external pressure and the design of a stiffening ring at the juncture such that the juncture can be considered as a line of support.

L-3.3.1 Design Data

External design pressure $P = 50$ psi, design temperature $T = 650$°F, $S_s = 17.5$ ksi, $E_1 = 0.85$, $E_s = 25.3 \times 10^6$ psi.

Cylinder at large end of cone
- inside diameter $D = 200$ in.
- minimum required thickness $t = 1.22$ in.
- nominal thickness $t_s = 1.25$ in.

Cylinder at small end of cone
- inside diameter $D = 50$ in.
- minimum required thickness $t = 0.330$ in.
- nominal thickness $t_s = 0.375$ in.

Cone section
- minimum required thickness:
  - $t_r = 1.22$ in. at the large end
  - $t_r = 0.55$ in. at the small end
- nominal thickness $t_c = 1.25$ in.
- axial length $L = 130$ in.
- cone half-angle $\alpha = 30$ deg
- $S_c = 15.0$ ksi, $E_2 = 0.85$, $E_c = 25.3 \times 10^6$ psi

Stiffening ring

$$S_r = 14.5 \text{ ksi}, \quad E_r = 25.3 \times 10^6 \text{ psi}$$

L-3.3.2 Solution

$$D_L = D + 2t_s = 200 + 2(1.25) = 202.5 \text{ in.}$$

$$D_s = D + 2t_s = 50 + 2(0.375) = 50.75 \text{ in.}$$

$$L_c = \sqrt{(130)^2 + (101.25 - 25.375)^2} = 150.5 \text{ in.}$$

$$L_L = 250.0 \text{ in.}, \quad L_S = 75.0 \text{ in.}$$

$f_1 = 250$ lb/in. and $f_2 = 62.5$ lb/in. are in compression

$$y = S_sE_r = 17{,}500 \times 25.3 \times 10^6$$

$$k = y/S_rE_s$$

$$= 17{,}500 \times 25.3 \times 10^6/14{,}500 \times 25.3 \times 10^6$$

$$= 1.21$$

L-3.3.2(a) At Large Cylinder-to-Cone Juncture. Assume $A_s = 0$.

$$A_{TL} = L_Lt_s/2 + L_ct_c/2 + A_s$$

$$= 250\,(1.25)/2 + 150.5(1.25)/2 + 0$$

$$= 250 \text{ in.}^2$$

$$M = -(R_L \tan\alpha)/2$$

$$+ L_L/2 + (R_L^2 - R_S^2)/(3R_L \tan\alpha)$$

$$= -101.25 \times 0.577/2 + 250/2$$

$$+ (101.25^2 - 25.375^2)/(3 \times 101.25 \times 0.577)$$

$$= -29.25 + 125.0 + 54.82$$

$$= 150.6$$

$$F_L = PM + f_1 \tan\alpha$$

$$= 50(150.6) + 250 \times 0.577$$

$$= 7{,}530 + 144.3$$

$$= 7{,}670$$

$$B = {}^3/_4\, F_LD_L/A_{TL} = {}^3/_4\,(7{,}670)\,(202.5)/250 = 4{,}660$$

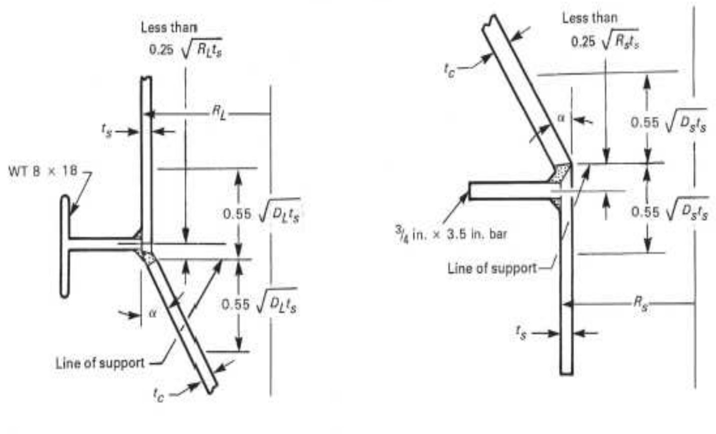
$$A = 0.00037 \text{ from Fig. CS-2}$$

$$I'_s = AD_L^2A_{TL}/10.9 = 0.00037(202.5)^2 \times (250)/10.9$$

$$= 348 \text{ in.}^4$$

Try a WT8 × 18 standard tee with the stem welded to the shell-to-cone juncture on the shell as shown in Fig. L-3.3.2 sketch (a).

FIG. L-3.3.2



The calculated I' for the combined ring-shell-cone cross section is

$$I' = 375 \text{ in.}^4$$

Consequently, $I' > I'_s$.

Effective area of reinforcement in the cone and cylinder is:

$$A_{eL} = 0.55\sqrt{D_Lt_s}\,(t_s + t_c/\cos\alpha)$$

$$= 0.55\sqrt{202.5 \times 1.25}$$

$$\times (1.25 + 1.25/\cos 30 \text{ deg})$$

$$= 23.57 \text{ in.}^2$$

Total area available $= A_{eL}$ + area of stiffening ring

$$= 23.57 + 5.28$$

$$= 28.9 \text{ in.}^2$$

$$Q_L = PR_L/2 + f_1 = 2{,}781 \text{ lb/in.}$$

$$P/S_sE_1 = 50/(17{,}500 \times 0.85) = 0.0034$$

From Table 1-8.1, $\Delta = 5.93$.

$$A_{rL} = \frac{kQ_LR_L \tan\alpha}{S_sE_1}\left[1 - {}^1/_4\left(\frac{PR_L - Q_L}{Q_L}\right)\frac{\Delta}{\alpha}\right]$$

$$= \frac{1.21 \times 2{,}781 \times 101.25 \times 0.577}{17{,}500 \times 0.85}$$

$$\times \left[1 - {}^1/_4\left(\frac{50 \times 101.25 - 2{,}781}{2{,}781}\right)\frac{5.93}{30}\right]$$

$$= 12.7 \text{ in.}^2$$

Total area $> A_{rL}$

$$28.9 > 12.7 \text{ in.}^2$$

Since reinforcement area and moment of inertia requirements have been met, use WT8 × 18 as the stiffening ring at the large cylinder-to-cone juncture.

L-3.3.2(b) At Small Cylinder-to-Cone Juncture. Assume $A_s = 0$; calculate

$$A_{TS} = L_st_s/2 + L_ct_c/2 + A_s$$

$$= 75 \times 0.375/2 + 150.5 \times 1.25/2 + 0$$

$$= 108 \text{ in.}^2$$

$$N = R_s \tan\alpha/2 + L_s/2 + (R_L^2 - R_s^2)/(6R_s \tan\alpha)$$

$$= \frac{25.375 \times 0.577}{2} + \frac{75}{2} + \frac{(101.25)^2 - (25.375)^2}{6 \times 25.375 \times 0.577}$$

$$= 154.2$$

$$F_s = PN + f_2 \tan\alpha$$

$$= 50 \times 154.2 + 62.5 \times 0.577$$

$$= 7{,}745$$

$$B = {}^3/_4\ F_e D_s / A_{TS}$$
$$= {}^3/_4 \times 7{,}745 \times 50.75/108$$
$$= 2{,}730$$

$$A = 0.00022 \text{ from Fig. CS-2}$$

Utilizing the combined ring-shell-cone cross section requires $I' \geq I'_s$.

$$I'_s = A D_s^2 A_{TS}/10.9$$
$$= 0.00022 \times (50.75^2) \times 108/10.9$$
$$= 5.61 \text{ in.}^4$$

Try a $\frac{3}{4}$ in. × 3.5 in. bar welded to the shell-to-cone juncture on the shell side as shown in Fig L-3.3.2 sketch (b).

The calculated I' for the combined ring-shell-cone cross section is

$$I' = 7.10 \text{ in.}^4$$

Consequently, $I' > I'_s$.

$$A_{es} = 0.55\sqrt{D_s t_s}[(t_s - t) + (t_c - t_r)\ /\cos\alpha]$$
$$= 0.55\sqrt{50.75 \times 0.375}$$
$$\times [(0.375 - 0.330) + (1.25 - 0.55)/\cos\ 30\ \text{deg}]$$
$$= 2.05 \text{ in.}^2$$

$$\text{Total area available} = A_{es} + \text{area of stiffening ring}$$
$$= 2.05 + 2.63$$
$$= 4.68 \text{ in.}^2$$

$$Q_s = PR_s/2 + f_2$$
$$= 50 \times 25.375/2 + 62.5$$
$$= 696.9 \text{ lb/in.}$$

$$A_{rs} = kQ_s R_s \tan\alpha / S_s E_1$$
$$= 1.21 \times 696.9$$
$$\times 25.375 \times 0.577/(17{,}500 \times 1.0)$$
$$= 0.71 \text{ in.}^2$$

$$\text{Total area} > A_{rs}$$
$$4.68 > 0.71 \text{ in.}^2$$

Since reinforcement area and moment of inertia requirement have been met, use a $\frac{3}{4}$ in. × 3.5 in. bar as the stiffening ring at the small cylinder-to-cone juncture.