chapter ten

Design of vessel supports

Contents

10.1 Introduction

The vessel support is intended to support the pressure vessel on the support base. The support has to be designed to withstand the dead weight and seismic loadings from the pressure vessel and to limit the heat flow from the vessel wall to the base. The pressure vessel support structure should be able to withstand the dead weight of the vessel and internals and the contained fluid without experiencing permanent deformation. The metal temperature of the pressure vessel is usually different to the ambient conditions during its installation. The differential displacement between the supports due to the temperature change should be considered in design. In a large number of cases the design of support requires adequacy to operate in a severe thermal environment during normal operation as well as to sustain some thermal transients. The other source of thermal loading arises from the thermal expansion of the piping attached to the vessel. The design must therefore consider the various combination of piping loads on the vessel to determine the most severe load combinations. In addition the vessel is also subject to mechanical loads due to the action of seismic accelerations on the attached piping. In large vessels containing liquids, the so-called ‘‘sloshing’’ effects must also be considered. Finally, loads due to handling during installation should be carefully considered in design.

The supports for pressure vessels can be of various types including lug support, support skirts, and saddle supports.

10.2 Lug support

This is a common means of support for vertical vessels that are mounted on I-beams. Such a support is shown in Figure 10.1. If the vessel is made of carbon steel, the lugs may be directly welded to the vessel. Bijlaard's classic assessment of local stresses in shells due to loadings on an attached lug is particularly noteworthy.[1] That analysis forms the basis for the Welding Research Council Bulletin 107[2] which has been used extensively for the design of lug attachments to pressure vessels. The method consists of determining the stresses in the vicinity of a support lug of height $2C_1$ and width $2C_2$ as shown in Figure 10.2. The maximum primary plus secondary stress in the shell wall is given as a combination of direct stress due to the thrust, W, bending stress due to longitudinal moment, M_L, bending stress due to circumferential moment, M_C, and the torsional shear stress due to the twisting moment, M_T, with appropriate coefficients. The earlier work by Bijlaard[1] involves representing M_L, M_C, and M_T, by double Fourier series, which enables one to obtain the stresses and deformations in the form of the series. In other words, the series is capable of representing a load with dimensions in both the circumferential, φ, and the longitudinal, x, directions:

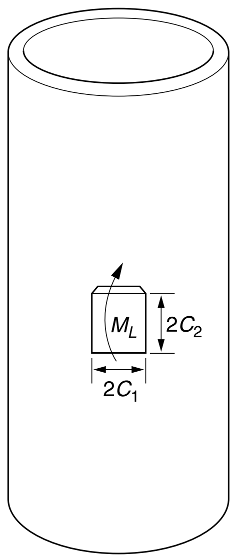

Figure 10.1 Lug supports.

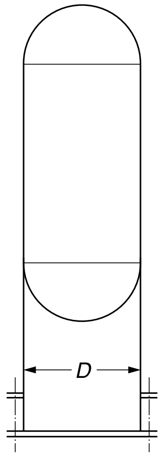

Figure 10.2 Support skirts.

$$P_r = \sum_{n=0,1,2}^{\infty} \sum_{m=0,1,3}^{\infty} P_{n,m} \cos n\phi \, \cos\left(\frac{m\pi x}{L}\right) \tag{10.1}$$

where L is the length of the shell, the term $P_{n,m}$ is the loading term. This representation is used for different forms of vessel loadings, where the direct and moment loadings are expressed as double Fourier series and introduced into the shell equations to obtain the values of stress resultants and displacements. In order to represent the patch load from the lug it is often necessary to have a large number of terms (typically about 200) in both the circumferential and axial directions. This approach has been used to draw up the curves presented in WRC 107.[2] When the attachment contact face is not rectangular, but maybe circular or elliptical, the design codes attempt to resolve the geometry into a rectangular patch. Mirza and Gupgupoglu[3] have studied stresses and displacements in circular cylindrical shells having square and rectangular lugs separated 90° apart along the circumferential direction. They have utilized a finite-element technique using 17-node doubly curved shell elements. For values of $C_1 = C_2 = 0.1\,D$, and $D/t \leq 40$ (where D is the mean diameter of the shell, and t the thickness) there seems to be a good agreement of results between References 2, 3, and 4. However, for smaller values of C_1 and C_2 (typically less than $0.05\,D$), large variations occur between the finite-element results[3] and closed-form predictions of References 2 and 4. However in spite of the variations in predicting the magnitude of the maximum stresses the methods agree on the direction of the maximum stress.[3]

The maximum stress is located at the upper end of the lug-vessel interface and occurs at the outside surface of the shell.

10.3 Support skirts

Most vertical vessels are supported by skirts as shown in Figure 10.2. These supports transfer the loads from the vessel by shear action. They also transfer the loads to the foundation through anchor bolts and bearing plates. A major problem in the design of support skirts involves the consideration of thermal stress introduced by the thermal gradient of the skirt in contact with the vessel (the vessel being at a considerably higher temperature than the cold support base). During heating up of the vessel, the outside of skirt juncture experiences tension, the magnitude of which depends upon the severity of the thermal gradient along the length of the skirt. Fatigue cracks may appear on the tensile surface of the weld due to alternating heating and cooling cycles. Some of the features that enter into the design of such supports aim to avoid attachments with high stress concentrations, to avoid partial penetration welds, and to employ generous fillet radii. In order to reduce the axial thermal gradient, suitable insulation must be adopted.

Skirt construction permits radial growth of pressure vessel due to pressure and temperature through the bending of skirt acting like a beam on an elastic foundation. The choice of proper height of the skirt support ensures that bending takes place safely. Finite-element methods can be effectively used to determine the stresses and deflections due to imposed pressure and temperature distribution.

In order to design a skirt support for mechanical loads alone, consider the vessel deadweight, W, and the bending moment, M, produced by seismic, wind and other mechanical loads. The stress in the skirt is then a combination of axial and bending stresses and is given by

$$\sigma = \frac{-W}{A} \pm \frac{M}{Z} \tag{10.2}$$

where A is the cross-sectional area and approximately equals πDt, with D the diameter and t the thickness. Z is the section modulus and for the circular cross-section is approximately equal to $0.25\pi D^2 t$. Eq. (10.2) then becomes

$$\sigma = \frac{-W}{\pi\, D\, t} \pm \frac{4M}{\pi\, D^2\, t} \tag{10.3}$$

Once the thickness of the skirt, t, is determined, the next task is to design the anchor bolts. If the total number of anchor bolts are N, then the load on a single bolt, P, from the consideration of bending about a neutral axis is given by

$$P = \frac{-W}{N} \pm \frac{4M}{ND} \tag{10.4}$$

10.3.1 Example problem

Design a skirt support for a pressure vessel with a total vertical load of 720 kN, and an overturning moment of 2050 kNm. The bolt circle diameter of the support may be assumed to be 4.5 m. Assume a thickness of 10 mm for the support skirt and the mean diameter of the support as 4.25 m.

10.3.2 Solution

The support is to be designed such that it does not buckle under the compressive load. With the given dimensions, namely $D = 4.25$ m, $t = 10$ mm, $W = 720$ kN, and $M = 2050$ kNm we have

$$\sigma = \frac{-720 \times 10^3}{\pi(4250)(10)} - \frac{4 \times 2050 \times 10^3 \times 10^3}{\pi(4250)^2(10)} = -19.84 \text{ MPa} \tag{10.5}$$

The procedure outlined in Chapter 5 may be used to demonstrate that the skirt does not buckle for the compressive stress of 19.84 MPa, calculated in Eq. (10.5).

The anchor bolts are designed as follows. Let $N = 12$ bolts. The load per bolt is calculated using Eq. (10.4) as

$$P = \frac{720 \times 10^3}{12} + \frac{4 \times 2050 \times 10^6}{12 \times 4250} = 166.8 \text{ kN} \tag{10.6}$$

For class 4.6 metric steel bolts, the proof strength is 225 MPa.[5] The load determined in Eq. (10.6) when divided by this proof strength gives the required bolt area as 741 mm^2.

For ISO metric standard screw threads with a major diameter of 36 mm the stress area is 816.72 mm^2, which is closest to the required bolt stress area.[5] Therefore 12 bolts of 36 mm diameter can be used as the anchor bolts.

10.4 Saddle supports

Horizontal pressure vessels are usually supported on two symmetrically spaced saddle supports (Figure 10.3). Zick[6] presented a semiempirical analysis approach that has traditionally formed the basis of design of saddle supports for horizontal pressure vessels in a number of pressure vessel design codes. In this approach the vessel is assumed to behave as a simply supported beam, and the cross-section is assumed to remain circular under load. The simplified model as represented in Zick's analysis is an overhanging beam subjected to a uniform load due to the weight of the vessel and its contents (Figure 10.4). If the longitudinal bending moment, M_1, at the midspan of the vessel is equal to the longitudinal bending moment, M_2, at the saddle, then an optimum location of the saddle support can be obtained. The bending moments M_1 and M_2 are determined to be

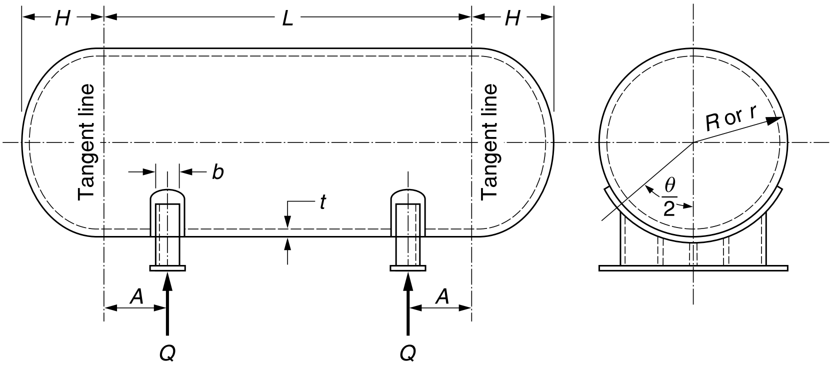

Figure 10.3 Saddle supports.

$$M_1 = q\left[\frac{(L-2A)^2}{8} - \frac{2}{3}HA - \frac{A^2}{2} + \frac{H^2}{4}\right] - M_0 \qquad (10.7)$$

and

$$M_1 = q\left[\frac{2}{3}HA + \frac{A^2}{2} + \frac{H^2}{4}\right] - M_0$$

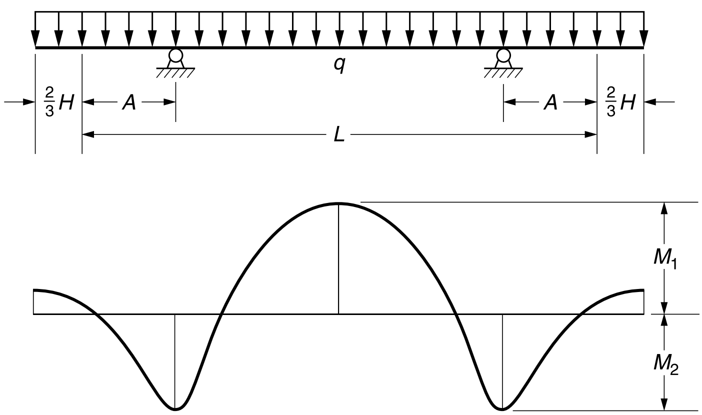

Figure 10.4 Simplified beam model for saddle supports.

Setting $M_1 = M_2$, from Eqs. (10.7) and (10.8) we obtain

$$\frac{(L-2A)^2}{8} - \frac{2}{3}HA - \frac{A^2}{2} = \frac{2}{3}HA + \frac{A^2}{2} \tag{10.9}$$

Furthermore, if we assume hemispherical heads for which $R = H$, and for the particular case of $L/R = L/H = 30$, we obtain the following relationship between A and L:

$$A^2 - 1.08LA - \frac{L^2}{4} = 0 \tag{10.10}$$

This gives:

$$\frac{A}{L} = 0.195 \tag{10.11}$$

Widera *et al.*[7] rightly note that the above optimum location of the saddle results from the consideration of the vessel as a beam, and not as a shell. In order to capture the shell behavior, they analyzed six vessel models using the finite-element method. These models correspond to A/L values of 0.05, 0.10, 0.15, 0.20, 0.25, and 0.30.[7] They concluded that for $A/L = 0.25$, the minimum values of stress intensities occur at the midspan, top line and bottom line of the vessel, as well as the local stresses at the saddle-vessel interface regions. Therefore they suggest that the optimum location of the saddle support is $A/L = 0.25$. Their other conclusions are that the displacements at the saddle and the midspan are in good agreement with the results obtained in Zick's analysis,[6] as far as the distribution and location of the maximum values are concerned; although the maximum values Widera *et al.* obtained are much smaller. Also, in the area of the saddle, high concentrated stresses and large stress gradients exist. Away from the region, the stresses decay very rapidly. In the region of the saddle-vessel interface, the stress distribution is not uniform, and the high localized stress represents a combination of membrane and bending stresses, and should be considered as primary plus secondary stresses.

It should be noted that the Zick's solution[6] is limited to the consideration of the dead weight of the horizontal vessel and contents for a rather specific saddle design configuration. Therefore in a general situation involving specific geometry or loading conditions, recourse must be made to finite-element methods. Horizontal pressure vessels are generally fixed at one saddle and the other is allowed to move to accommodate thermal expansion. The entire seismic load is therefore resisted by the fixed saddle, which can result in very high local bending stress on the shell due to the overturning moment on the saddle. A possible design modification is to add a series of gusset plates to the saddle, to reduce the overturning moment on the shell thereby reducing the stresses.

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