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PV Newsletter

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Tall Vertical Pressure Vessels

Introduction

Tall pressure vessels are generally defined as those where the height to diameter ratio (H/D) is more than 15. These vessels are built these days as self-supporting structures, i.e., they are supported on cylindrical or conical skirts with a base ring resting on a concrete foundation and firmly fixed to the foundation by anchor bolts embedded in concrete. Basically, they are designed as cantilever beams.

Design Considerations

Tall vessels must be capable of withstanding the following loadings:

1) Internal and external pressure
2) Dead loads including the weight of the vessel itself plus its contents and the weight of the insulation and attached equipment
3) Wind loads acting on the vessel and its attachments
4) Seismic (earthquake) loads on the vessel and its attachments

Tall vessels may also be subjected to applied forces and moments from thermal expansion of piping. The most critical combination of loadings that causes the highest stresses may not occur when all of the loads are applied at the same time. Certain loads may cause critical stresses during the time of erection of vessel, whereas other combinations of loadings may cause critical stresses when the vessel is filled. The proper design of the vessel requires examining several different loading conditions to establish the proper thickness and other requirements for a safe design.

The required thicknesses and other design requirements vary somewhat depending upon the design theory chosen. The maximum stress theory is chosen for design of most tall vessels, and is the theory used in ASME Section VIII, Division 1.

Internal and External Pressure Loading

A cylindrical vessel under internal pressure tends to retain its shape in that any out-of-roundness or dents resulting from shop fabrication or erection tend to be removed when the vessel is placed under internal pressure. Thus any deformation resulting from internal pressure tends to make an imperfect cylindrical more cylindrical.

However, the opposite is true for imperfect cylindrical vessels under external pressure. Any imperfection in the cylindrical vessel will tend to be aggravated with the result of possible collapse of the vessel. For this reason, a given vessel under external pressure in general has a pressure rating only about 60% as high as it would have under internal pressure.

Tensile Stresses Resulting from Internal Pressure

Axial tensile stress due to internal pressure in the shell of a closed vessel is given by the following formula:

$$f_a = \frac{pd}{4t}$$

Circumferential tensile stress due to internal pressure in the shell of a closed vessel is given by the following formula:

$$f_c = \frac{pd}{2t}$$

Compressive Stresses Resulting from External Pressure

External pressure acting upon a cylindrical shell and its heads may result in the failure of the vessel either by yielding or by buckling. If the vessel has relatively thin wall, the stress at which the buckling begins to occur is usually below the yield strength of the material. If the vessel has a relatively thick wall, the stress at which the buckling occurs is the yield point of the material under consideration at the temperature of service.

For vessels operating under external pressure, the design is based upon the critical pressure at which buckling occurs rather than upon the allowable stress for the material.

Dead Loads

Stresses caused by dead loads may be considered in three groups for convenience:

1) Stress induced by shell and insulation

At any distance, X feet, from the top of a vessel having a constant shell thickness,

$$W_{shell} = \frac{\pi}{4}\left(D_o{}^2 - D_i{}^2\right)\rho_s X$$

$$f_{dead\ wt\ shell} = \frac{W_{shell}}{unit\ area}$$

$$= \frac{X\rho_s}{144}$$

$$W_{ins.} = \frac{\pi}{12} D_{ins.} X t_{ins.} \rho_{ins.}$$

$$f_{dead\ wt\ ins.} = \frac{W_{sins.hell}}{unit\ area}$$

$$= \frac{X t_{ins.} \rho_{ins.}}{144 t_s}$$

For steel construction, the density of the shell material is approximately 490 lb/ft^3, and for most insulation, the insulation density is generally taken as 40 lb/ft^3.

2) Stress induced by liquid in the vessel

$$f_{dead\ wt\ liq.} = \frac{\sum liquid\ wt.}{12\pi D_m t_s}$$

3) Stress induced by attached equipment

$$f_{dead\ wt\ att.} = \frac{\sum weight\ of\ attachments}{12\pi D_m t_s}$$

The weight of steel platforms may be estimated at 35 lb/ft^2, and the weight of steel ladders at 25 lb/ft for caged ladders and 10 lb/ft for plain ladders. Trays in distillation columns, including liquid hold-up trays, may be estimated to have a weight of 25 lb/ft^2 of try area.

The total dead load stress acting along the longitudinal axis of the shell is then the sum of the above dead weight stresses. If the vessel does not contain internal attachments, such as trays which support liquid, but consists only of the shell insulation, the heads and minor attachments such as manholes, nozzles and so on, the additional load may be estimated as approximately equal to 18% of the weight of a steel shell.

Tensile and Compressive Stresses Caused by Wind Loads

Two distinctly different kinds of design considerations result from wind loading. First, the static force from the wind-loading pressure against the vessel causes an overturning moment that must be considered in designing tall vessels. The second consideration is the dynamic effect from vortex shedding of wind passing around the vessel.

The stresses produced in a self-supporting vertical vessel by the action of wind are calculated by considering the vessel to be a vertical uniformly loaded cantilever beam. The wind loading is a function of the wind velocity, air density and the shape of the tower and is given by the formula:

$$P_w = 0.004\frac{B}{30}{V_w}^2 F_s \qquad \text{[United States Weather Bureau]}$$

B = Barometric pressure (in. mercury)
P_w = Wind pressure on flat surface (psf)
V_w = Wind velocity (mph)
F_s = Shape factor (1.0 for flat plate at 90° to the wind)

The shape factor for a smooth cylinder is 60% of that of a flat surface normal to the wind and having the same projected area as the cylinder. Projections of auxiliary equipment loaded on the tower will cause turbulence, and the use of value of F_s based on smooth cylinders is questionable. A value between 0.60 and 0.85, depending on the amount and the shape of projections, is generally used.

The appropriate wind velocity is dependent on the location in which the equipment is to be erected. In addition to the shape factor, the wind velocity pressure is also affected by the height factor. This factor is taken as 1.0 for towers having heights from 30 to 49 ft. For other heights, the height factor varies directly as the (height/30) raised to the 1/7 power.

The force P_w acts over the projected area of the tower, and some engineers compensate for the turbulence caused by the projections by using an effective diameter of the tower and allied equipment. This effective diameter is the diameter of the tower plus twice the thickness of the insulation plus an allowance for the projected area of piping and attached equipment. This allowance may be taken as 17 in. if caged ladders are present.

After determining the values of wind loading and the projected upon which it acts, the bending moment any distance X from top of the tower can be expressed as:

$$M_{wx} = P_w X\left(\frac{d_{eff.}}{12}\right)\left(\frac{12X}{2}\right)$$

The stress in the extreme fibre of the shell, due to the wind is obtained as follows:

$$f_{wx} = \frac{M}{Z} = \frac{Mc}{I} = \frac{M_{wx}(r_o)}{I}$$

r_o = Outside radius of shell (inch.)
I = Moment of Inertia perpendicular to and through the longitudinal axis (inch^4)
f_{wx} = Stress at extreme fibre due to wind loads (psi): compressive stress on downwind side, tensile stress on upwind side

Dynamic Analysis of Wind Loads:

When a laminar wind flows by a circular pressure vessel, the wake behind the vessel is no longer smooth. There is a region of pressure instability in which vortices are shed in a regular pattern. These vortices cause an alternating force perpendicular to the wind direction that can make the vessel vibrate. When the frequency of vortex shedding coincides with the natural frequency of the vessel, a resonance is caused with increasing amplitude. To prevent this condition, the natural frequency of the vessel is set higher than the vortex shedding frequency determined by the maximum velocity of laminar wind at the vessel location. The resonant wind velocity is related to the height-to-diameter ratio of a cylindrical vessel.

Analysing tall towers in a plant for vibration would be a time consuming process. The following criteria is used to investigate vibration possibility in a tower:

		$\frac{W}{LDr^2}$	≤	20: Vibration analysis MUST be performed
20	<	$\frac{W}{LDr^2}$	≤	25: Vibration analysis SHOULD be performed
25	<	$\frac{W}{LDr^2}$		Vibration analysis NEED NOT be performed

Wind-induced Deflection of Tall Towers:

A sustained wind pressure will cause a tall tower to deflect with the wind. The magnitude of deflection may seriously influence the performance of the vessel and has to be limited to a certain value. If too small a deflection is specified, the shell thickness may have to be increased and the price of tower may increase. Most engineering specifications limit the tower deflection due to wind to a maximum of 6 in. per 100 ft of the tower height.

Stresses Resulting from Seismic (Earthquake) Forces

In tall vessels, one cause of stresses in the vessel wall is the overturning moment from the lateral force of an earthquake loading. ASME Code requires that the pressure vessels be designed to withstand earthquake forces; however it does not specify rules for the earthquake design of pressure vessels. It is up to the user to specify in the specifications the Code to be used, failing which the designer selects the Code based on his preference.

Two Codes used frequently for earthquake design are ANSI A58.1 Building Code, and Uniform Building Code. Both Codes rely on seismic zone maps to determine the appropriate earthquake factor. These maps indicate seismic zones according to the amount of damage caused by earthquakes. Current practice in designing a vessel for these earthquake forces is empirical and is based on theory of vibration. In both the Codes, the total lateral earthquake force is calculated by the following formula:

$$V = ZIKCSW$$

Z = Coefficient depending upon the earthquake for the location of installation
W = Total dead load of the vessel and contents above plane being considered (lb)
I = Importance factor; assume I = 1.0 for vessels
K = Arrangement factor; assume K = 2.0 for vessels
C = Base shear factor
= $1/(15\sqrt{T}) \leq 0.12$
T = Fundamental period of vibration of the vessel assuming a uniformly loaded cantilever beam fixed at the base
S = Site-structure resonance; assume S = 1.5 unless an exact value is known

We require:

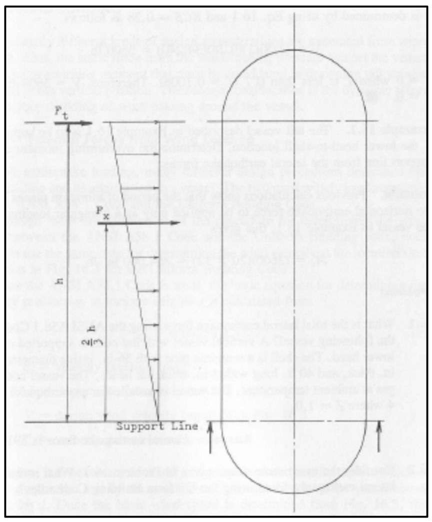
$0.12 \leq KC \leq 0.25$ for UBC
$0.12 \leq KC \leq 0.29$ for ANSI Zones 0, 1, and 2
$0.12 \leq KC \leq 0.23$ for ANSI Zones 3 and 4
$CS \leq 0.14$ for UBC and ANSI in zones 0, 1, and 2
$CS \leq 0.11$ for ANSI in zones 3 and 4 when S = 1.5

KCS need not exceed 0.3

For a cylindrical shell of uniform diameter and thickness, the total lateral force V is distributed as follows:

1. At the upper head-to-shell tangent line, apply a concentrated horizontal force F_t = 0.07TV, where T is the fundamental period of vibration of the vessel. F_t shall not exceed 0.25V and F_t shall be considered zero for T = 0.7 or less.
2. Along the straight length of shell, $F_x = \frac{(V-F_t)w_x h_x}{wh}$

For a shell of uniform diameter and thickness, this gives a triangular load distribution with the apex pointing downward. For calculating the moment, assume a concentrated loading of V-F_t applied at the centroid of the triangle, which is equal to ⅔h above the lower head-to-shell tangent line as shown in the figure below.



Once the values of F_t and F_x are determined and the moment arms to the respective forces are known, the overturning moment is determined as:

$$M_e \quad = \quad F_t(h) + F_x\left(\frac{2}{3}h\right)$$

Combined Stresses in Shell

A controlling combined tensile or compressive stress occurs as a result of combination of stresses. It is important to consider the intended construction, erection and test schedule which is to be followed in erecting the vessel and placing it on stream. With this in mind, the conditions must be determined which establish the controlling stresses. The stresses condition of the vessel may be divided into the following possible cases:

Case 1: Vessel under construction
   a. Empty shell erected
   b. Shell and auxiliary equipment such as trays or packing but no insulation
Case 2: Vessel completed but shut down
Case 3: Vessel under test condition
   a. Hydrostatic test
   b. Pneumatic test
Case 4: Vessel in operation

In the consideration of wind and earthquake loads, it is assumed that the possibility that the most adverse wind and earthquake load will occur simultaneously is remote, and the possibility that these lateral forces will occur in the same direction is even more remote. Therefore, the resulting stresses for wind loads and earthquake loads are computed separately, and the most adverse loading condition is used in the design.

In analyzing the combined stresses, calculations are usually made beginning at the top of the vessel. The shell thickness computations are based on the principal stresses - circumferential stress due to the design pressure or longitudinal stress due to the design pressure plus weight plus wind plus wind or earthquake load. Since the longitudinal stress increases from top to bottom of the vessel, the shell thickness has to be increased below the elevation where summation of the longitudinal stresses becomes larger than the circumferential stress due to design pressure alone.

In designing the shell, it is not necessary to allow for the compressive stress resulting from the weight of the liquid in the hydrostatic test since the bottom head transfers this load directly to the skirt (most tall vessels are supported by skirt). It is essential to check the lower sections for wrinkling failure before the final specifications are fixed.

Sources:

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2. Bednar, Henry H., *Pressure Vessel Design Handbook*: Second Edition
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4. Jawad, Maan H. and James R. Farr, *Structural Analysis and Design of Process Equipment*: Second Edition
5. Mahajan, Kanti K., *Design of Process Equipment*: Second Edition

***** E N D O F T H E A R T I C L E *****

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CoDesign Engineering is involved in providing training and consultancy services as described below:

Training

- Pressure vessel & heat exchanger design (ASME Section VIII, Div. 1)
- Power and process piping design (ASME B31.1 & B31.3)
- Solar PV power plant design

Consultancy

- Engineering solutions related to pressure vessels and heat exchangers
- PMC as well as EPC services for solar PV power plants

We have designed a 3-day training course for ASME BPVC Section VIII, Div. 1 that can be offered at most cities in India. In-house training can also be provided at any location in India or in US upon request. The training is designed as a workshop where the delegates are encouraged to do all calculations using only pencil, paper and calculators. Please contact Ramesh Tiwari at rtiwari123@gmail.com or ramesh.tiwari@codesignengg.com for 2012 training calendar and rates.

Contents of 3-Day Training Course:

- Organization of ASME Pressure Vessel Code
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- Materials of Construction
- Low Temperature Operation
- Joint Efficiencies
- Design of Shell Sections
- Design of Conical Sections
- Design of Formed Heads and Flat Heads
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- Nozzle Loads
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