International Journal of Mechanical Design and Materials Science

Volume 1, Issue 2 2025

VERTICAL PRESSURE VESSEL DESIGN UNDER APPLIED FORCES AND VIBRATIONAL CONDITIONS

Liyuan ,Wang Xu. Key Laboratory of Mechanics,china

Research Article

Date of Submission: 1-02-2025 Date of Acceptance: 9-03-2025 Date of Publication: 14-04-2025

ABSTRACT

With the exception of the comparatively straightforward scenario of cylindrical shells with standard-type heads and apertures under uniform pressure, pressure-vessel rules do not provide design procedures. When dealing with increasingly complex structures and loading systems, the designer must utilize engineering concepts. The study addresses the vibration of vertical pressure vessels, such as columns or towers, as well as vessels that are exposed to different applied pressures operating in conjunction with internal or external pressure. The vessels under consideration are cylindrical shells with a vertical longitudinal axis. The method for calculating the first mode of vibration period for columns with the same thickness and diameter over their whole length is described. A detailed process for estimating the period of vibration of multi-thickness (stepped shell) vessels and/or vessels built to two or more diameters with conical transitions where the diameter difference is small is described, along with a method for designing vessels that are regarded as dynamic structures. The practical vessel designer faced with the challenge of examining vibration possibilities in vertical pressure vessels will find the described design process helpful.

Keywords: Pressure vessel, dynamic structure, and cylindrical shell

INTRODUCTION

The main loading conditions that the designer should take into account while constructing a vessel are listed in the pressure-vessel regulations. These circumstances may be separated into two categories: applied forces and pressure loadings. The vessel's surface is subjected to internal or external pressures. Applied forces operate either locally or across the vessel's bulk. The following issues need to be thoroughly examined in order to develop this kind of self-supporting vessel:
A vertical vessel is regarded as a dynamic structure when it is essential to depart from the standard procedure of constructing it as a static construction.

1. How may dynamic circumstances be fulfilled most practically?
2. Does the technique provide reliable results and offer more strength to withstand the force brought on by the mass acceleration brought on by the vessel's motion?
3. Does the dynamically designed vessel's vibration period prevent excessive movement due to strong prevailing winds?
4. Do external attachments like platforms, pipes, and ladders that are dispersed throughout the ship contribute to the disruptive vessel system?

These issues will be covered in this paper's sketch of a design process. The design of cylindrical pressure vessels with vertical axes that are susceptible to applied forces in addition to internal or external pressure is discussed in this work. The weight of the vessel and its contents as well as the weight of any attachments to the vessel are taken into account as vertical forces. Piping thrusts, seismic forces, and wind pressures are examples of horizontal forces.
It should be noted that earthquake-induced vessel vibrations are rare, and the focus of this work is on vibrations caused by wind or other forces, which may happen daily or often throughout the day

depending on the location. The design of cylindrical pressure vessels with vertical axes that are susceptible to applied forces in addition to internal or external pressure is discussed in this work. The weight of the vessel and its contents as well as the weight of any attachments to the vessel are taken into account as vertical forces. Piping thrusts, seismic forces, and wind pressures are examples of horizontal forces.

LOADS

The forces resulting from gravity, or weight, make up the majority of the vertical loads. It is also necessary to take into account the vertical component of pipe thrusts. The bottom head and the vessel supports typically carry the liquid contents. However, the weight of the liquid on internal trays is moved into the shell in fractionating columns. Friction transfers a portion of the stored particles' weight into the shell. The design must take into account the bending moments caused by the weights of attachments that are eccentric to the vessel's axis.

Wind Load:

Wind velocity, height above ground, and drag coefficient are some of the variables that affect the force per unit area that the wind exerts. Lastly, the form factor and height-to-diameter ratio are included. The impact of velocity on the drag coefficient is not taken into account in the discussion. The Reynolds number affects the drag coefficient, also known as the friction factor, for a certain form. The coefficient for a circular cylinder is essentially constant when the Reynolds number falls between R = 20,000 and R = 200,000. The coefficient decreases to less than half of its lower range value above R = 500,000. Designers often employ wind pressure figures from purchaser specifications or construction standards. The usage of a drag coefficient for Reynolds number in the range of R = 20,000 to R = 200,000 seems to be the foundation for all such values. A circular cylinder's Reynolds number is 9100 DV, where D is the diameter in feet and V is the wind speed in miles per hour. The drag on the vessel would be less than half of what is stated in the rules if the DV value was higher than 55. As a result, the rules may take into account the need of lowering wind pressure while using circular vessels.

The structure is assumed to be a rigid body that experiences the accelerations of the supporting ground in the standard simplified approach to the issue. The structure's mass times the ground acceleration equals the horizontal force acting on it, which has the same weight-to-gravity ratio as the ground acceleration. Values for this ratio are provided by structural codes and are determined by engineering judgment and experience.

2.2 Seismic loading:

Applying an excitation caused by an earthquake to a structure (or geo-structure) is known as seismic loading. It occurs at a structure's contact surfaces with the earth, nearby structures, or tsunami gravity waves. The capacity of a building to maintain its proper functions, such as its safety and serviceability, both during and after a certain earthquake exposure is known as earthquake or seismic performance. Generally speaking, a building is deemed safe if it doesn't pose a risk to people's lives or well-being by partly or totally collapsing. If a building can do the operating tasks for which it was intended, it may be deemed serviceable. The fundamental ideas of earthquake engineering, which are included into the main pressure vessel regulations, are predicated on the idea that a structure should be able to withstand a rare, very strong earthquake by suffering substantial damage without completely collapsing. However, it should continue to function during more frequent but milder earthquake occurrences. When a particular vessel is directly damaged by a certain amount of ground shaking, engineers must be aware of the quantifiable level of the actual or predicted seismic performance. This kind of evaluation may be carried either analytically or empirically.

Supports and external attachments:

Legs may be used to hold vessels up to 1 m in diameter or 2 m in height. Vessels that experience extreme vibrations due to reciprocating equipment, process flow, etc. cannot be supported on their legs. Typically, vertical vessels must have a support skirt that complies with the following specifications:1. Vessel skirts with a diameter of 2400 mm or more must have two unobstructed access apertures with a minimum diameter of 600 mm, unless there are overriding legislative requirements. One 600 mm diameter unobstructed access port must be provided when the skirt's diameter is less than 2400 mm. To prevent unwanted entry, each aperture may need to have a detachable, locked grill.
2. The skirt must have enough breathability.
3. A skirt's access aperture cannot be used to route pipe work. The skirt must be fastened such that its mean diameter roughly matches the mean diameter of the jar. A shell-to-head weld cannot be covered by attachment welds, which must be continuous. In high temperature reactors, the connection of the skirt to the shell must be constructed to offer sufficient resistance to fatigue from the thermal stresses caused by start-up and shutdown, and an insulated air gap must be provided at the skirt to shell junction to decrease thermal stress.

DESIGN OF PRESSURE VESSEL

Design with Internal Pressure:

The axial stresses set up in the shell may be classified under three types: (a) The longitudinal stress produced by the internal pressure; (b) the uniform compressive stress produced by the sum of the weights assumed to act along the axis of the vessel; (c) the bending stress produced by the horizontal loads and by the resultant weight when eccentric to the axis of the vessel. Literature indicates that a somewhat higher computed stress is required to produce failure under combined bending and compression than under compression alone. Thus we may safely combine compressive stresses due to bending with those due to uniform compression, and design the vessel shell as though these stresses were all due to uniform compression. The tension side of the shell has its highest stress when the vessel is under pressure. On the compression side, the highest stress occurs when the internal pressure is not acting. The stresses set up in the shell for these two conditions are,

Tension $St = \frac{pD}{4t} - \frac{W}{\pi Dt} - \frac{4We}{\pi D^2 t} - \frac{4M}{\pi D^2 t}$ ..... (1)

Compression $Sc = -\frac{W}{\pi Dt} - \frac{4We}{\pi D^2 t} - \frac{4M}{\pi D^2 t}$ ..... (2)

The factor W includes all the vertical loads and the factor M includes all the moments due to horizontal loads for the loading condition under consideration. A value of shell thickness must be selected so that these stresses are not greater than the allowable stress values, taking into account the applicable joint efficiencies.

Design with External Pressure:

The code charts for determining the required thickness of shells under external pressure have been developed for the condition of a uniform pressure on the cylindrical surface and the heads of the vessel. The longitudinal compressive stresses set up in the shell by weight and lateral forces have an effect similar to that produced by subjecting the heads of the vessel to a higher external pressure than that which acts on the shell. The charts can be used to give an approximate solution for such a load condition by a suitable change in the vertical scale on which the factor B is read. The nature of the approximations will be discussed later. In practice, it is simpler to leave the scale unchanged and make the adjustment in the value of pressure to be used in reading the chart.

In reference (7) Sturm gives a method of dealing with end loads on the heads. This method can be used as the basis for the design of a vessel under external pressure and subjected to applied loads.

Using Sturm's Equation, the ratio of the collapsing pressure W_e and W'_e is equal to

$$r = \frac{W_e'}{W_e} = \frac{F}{F=\frac{\pi^2 R^2}{2L^2}}$$

Or

$$r = \frac{n^2-1}{n^2-1+m} \quad \ldots\ldots\ldots\ldots\ldots (3)$$

Where, $m = \frac{\pi^2 R^2}{2L^2} = \frac{\pi^2 D^2}{8L^2} = \frac{1.23D^2}{L^2}$

and F is given as a approximately equal to $n^2 - 1$, where n is the number of lobes into which the shell may buckle. By Sturm's Equation [46], the ratio of collapsing pressures W_e and W'_e is equal to,

$$r = \frac{W_e'}{W_e} = \frac{F}{F=\frac{\pi^2 R^2}{2L^2}+\frac{2P}{We'R}\frac{\pi^2 R^2}{2L^2}}$$

or

$$r = \frac{n^2-1}{n^2-1+m+m\alpha} \quad \ldots\ldots\ldots\ldots\ldots (4)$$

In this equation $\alpha = 2P/(W'_e R)$, where P is the axial compression per lineal inch due to the externally applied loads, and $W'_e R/2$ may be looked on as the axial compression per lineal inch in the shell, if the collapsing value of the external pressure were acting on the ends of the vessel.

Since the ASME Code charts are made up on the basis of the collapsing pressure W'_e, the ratio of W'_e to W''_e is needed to make use of the charts. This is

$$r'' = \frac{W_e'}{W_e} = \frac{n^2-1+m+m\alpha}{n^2-1+m} \quad \ldots\ldots\ldots\ldots (5)$$

Windenburg and Trilling (9) have developed a chart which gives n as a function of t/D and L/D for pressure on the sides and ends of the vessel. This chart is reproduced in Fig. 1. A comparison of Strurm's Figs. 4 and 8 indicates that the values of n for different values of t/D and L/D change very little between the condition of pressure on the sides only and that of pressure on both sides and ends. Thus Fig. 1 should give satisfactory values of n for external loading conditions for which α is not much greater than one. For large values of α, the shell will buckle in fewer lobes

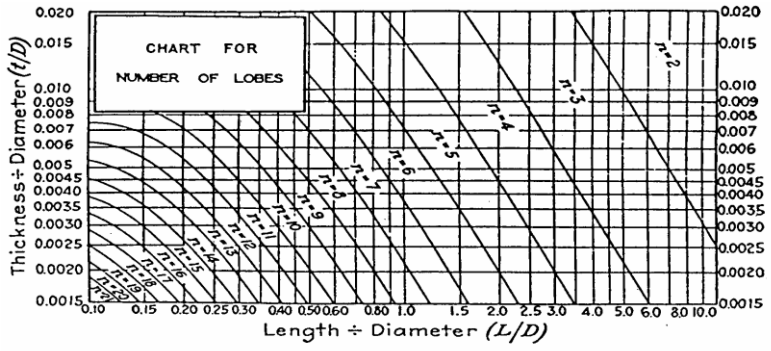


FIG. 1 NUMBER OF LOBES n INTO WHICH A SHELL WILL COLLAPSE WHEN SUBJECTED TO UNIFORM COLLAPSING PRESSURE ON SIDES AND ENDS

than the number given in Fig. 1, or it may even fail by plastic flow without the formation of any lobes. For values of α greater than one, the vessel should also be checked as a cantilever beam, including the axial stress due to external pressure in the computations.

The relation between W_c' and W_e' given by r'' is a ratio. Hence the value of r'' can be used equally well for the relation between allowable external working pressures. Thus the code charts can be used to determine the required thickness with external loads and moments by using an equivalent design extern al pressure W_a' equal to

$$= \frac{n^2-1+m+m\alpha}{W_a' n^2-1+m} \quad \text{..... (6)}$$

X W_a' ..... (a)

Where, W_e' and W_a' are equal to the values of W_e' and W_c' divided by the factor of 4against collapse. The use of Equation [a] in connection with the code charts implies that the pressure on the sides of the vessel is increased in the same ratio that the applied vertical forces increase the axial compression in the shell. Since the applied loads do not increase the circumferential compression in the shell, the use of Equation [a] gives answers that are somewhat on the side of safety.

DISCUSSION OF WIND EFFECTS

Tall, cylindrical structures such as pressure vessels and stacks are subject to being put in oscillatory motion by wind currents. The motion is at right angles or normal to the direction of the wind. The relationship between wind velocity and frequency of eddy shedding is given by the equation.

$$V = \quad = \quad \frac{D}{Tk} \quad \text{..... (7)} \qquad \frac{fD}{k}$$

Since we are primarily concerned with the resonant condition which occurs when the frequency of eddy shedding equals or is in the neighborhood of the natural frequency of vibration for the vessel the symbols for the vessel frequency and period of vibration are shown in equation(7). From this equation, we can estimate the critical wind velocity for most vessels.

The value for *k* was first determined in 1878 by V. Strouhal as 0.185 and is known as the Strouhal number [9]. It is assumed by some authorities to be within 0.18 to 0.27 and dependent upon the velocity of flow. The reproduced graph shown in Fig. 2 gives the variation of the Strouhal number with the Reynolds number as obtained experimentally by Relf and Simmons [9]. Research engineers, employed by the same company as the author, reported the following values for *k* obtained from full size vessels after erection,

k = 0.133, for a 7.67 O.D. insulated vessel at a wind velocity of 39.6 ft/sec (27 mph)

= 0.189, for a 3.0 O.D. vessel at a wind velocity of 32.25 ft/sec (22 mph)

The difference between the values reported from field data and the graph is probably due to the size of cylinders tested and the method of support. When the velocity of the wind is such that the frequency *f* in the equation corresponds to the natural frequency of the vessel, resonance occurs and the vessel will oscillate at excessive amplitude. Since aerodynamic stability theory and calculation methods are beyond the scope of this paper.

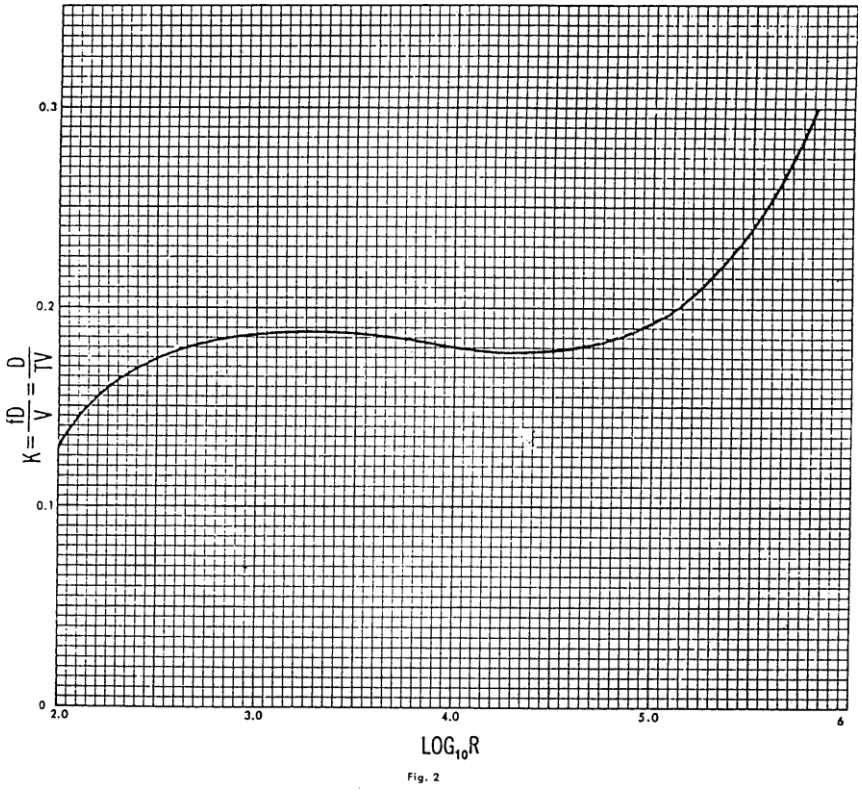


Figure 2 gives the effect of wind loads on vertical pressure vessels

CONCLUSIONS

Whether the longitudinal stress in the shell is tension or compression, as well as whether the vessel is under internal or external pressure, determine which design technique should be used to vertical pressure vessels. The potential behavior of self-supporting vertical pressure vessels under vibrating circumstances should constantly be examined. The distribution of external vessel attachments, the surrounding equipment, and the topography should all be taken into account when evaluating the impacts of wind velocity. Remember that although liquid loading in tray-equipped containers might assist reduce vibration, it shouldn't be used as a panacea.

If vibration problems do arise, it is important to carefully consider any suggested solutions to prevent problems from arising from other sources. When external stresses are applied to vertical pressure vessels, the vessel experiences axial loading and bending moments. To determine the overall longitudinal stress operating in the shell, these axial tensions and compressions must be added to the effects of pressure loading.

REFERENCES

“ASME Boiler and Pressure Vessel Code, Section VIII: Guidelines for Building Unfired Pressure Vessels

2. "API-ASME Code for the Design, Construction, Inspection, and Repair of Unfired Pressure Vessels for Petroleum Liquids and Gases," Fifth Edition, 1951.

3. "Lateral Forces of Earthquake and Wind," Paper 2514 Trans. ASCE, vol. 117, 1952, pp. 716-780

4. W. M. Wilson and N. M. Newmark, "The Strength of Thin Cylindrical Shells as Columns," University of Illinois, Engineering Experiment Station, Bulletin No. 255

5. R. C. Baird, "Wind-Induced Vibration of a Pipe-Line Suspension Bridge, and Its Cure," TRANS. ASME, vol. 77, 1955, pp. 797-804

6. "Factors Influencing the Dynamic Behavior of Tall Stacks Under the Action of Wind," TRANS. ASME, vol. 78, 1956, pp. 1381-1391, M. S. Ozker and J. O. Smith

7. R. G. Sturm, University of Illinois, Engineering Experiment Station, Bulletin No. 329, "A Study of the Collapsing Pressure of Thin-Walled Cylinders"

8. W. M. Wilson and E. D. Olson, "Tests of Cylindrical Shells," University of Illinois, Engineering Experiment Station, Bulletin No. 331, 1941

9. "Collapse by Instability of Thin Cylindrical Shells Under External Pressure," D. F. Windenburg and C. Trilling, Trans. ASME, volume 56, pages 819–825