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Effect of Geometric Characteristics of Empty Metal Tanks on the Critical Dynamic Buckling Load

Mokhtar TOUATI
University of Science and Technology

Mohamed CHABAAT
University of Science and Technology

Abstract: We investigate a parametric study on dynamic buckling of empty steel tanks, anchored at the bottom and with open top. The study attempts to estimate the critical load (Pcr), which induces the elastic buckling at the top of the cylindrical shell under a suddenly applied concentrated load with infinite duration in the horizontal direction through transient dynamics analysis (including geometric non-linearity) using the finite element shell of the library of commercial software ANSYS while applying the criterion of Budiansky-Roth and checking by the plan-phase, and subsequently obtain the stabilization level of the critical buckling load versus the geometric characteristics of the tanks in question which led to their design. This study deals three types of tanks with heights (H) of 10m, 20m and 30m, each type has height-radius ratio (H/R) of 1/3, 2/3, 3/3, 4/3 and 5/3, giving fifteen tanks of the same thickness (t). It is reported that the effects of imperfections and damping was not considered. The investigation showed that the studied parameters have a pronounced effect on the buckling load of the tanks and the results are discussed in this study.

Keywords: Dynamic buckling, Critical load, Tanks

Introduction

Damage of metal tanks are usually recorded (Hamdan, 2000 and Seisme du 21 MAI, 2003) ALGERIE, with the existence of liquid, but rarely for the tanks empty, most of the work done in this context consider the instability of the tanks caused by major earthquakes (Fukuyama et al., 2001). A strong wind and turbulent may also lead to instability of the tanks and cause damage in its transitional phase (during its execution).

The area of instability "dynamic" structures is the subject of numerous studies over the past 40 years (Sahu and Datta, 2007) after a good mastery of the static analysis of thin shells (Touati, 2008). Similarly to the analysis of non-linearity of the problem (Moussaoui and Benamar, 2002). The choice of stability criteria is an essential element.

Lagrange in (1788) proposed a criterion, called from "energy criterion", which presents a potential energy minimum for a stable equilibrium. If this energy is maximum, the equilibrium is unstable. As for Mathieu-Hill (Budiansky, 1965), his theorem was used to identify areas of instability. The first observation of parametric resonance is attributed to Faraday in (1831), studies were conducted on the parametric resonance of cylindrical thin shells by Yao (1963, 1965).

A second type of dynamic instability for structures with post-critical behavior unstable type snap-trough, subjected to loads very fast (explosion, crash ...). The dynamic instability appears when realizes a slight disturbance on a structure under dynamic loading, the latter initiates a major shift from the original position undisturbed, this phenomenon is characterized by a jump more or less important from an initial state stable and a stable final state for a finite jump or even unstable for an infinite jump.

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The application of the finite element method in the analysis of dynamic buckling of shells began in the 70s of last century. Experimentally, Ari-Gur et al. (1982) and Yaffe and Abramovich (2003) have undertaken a series of Work at the Haifa Institute of Aerospace, which struck first simple structures (columns) before spreading then more complex structures. Nagazawa et al. (1995) studied experimentally the behavior of thin cylindrical shells under seismic loading biaxial. Ren et al. (1983) conducted an experiment on the influence of impact velocities on the dynamic buckling of cylindrical shell. Michel, Limam and Jullien (2000) and Michel, Combescure and Jullien (2000) have designed an experiment to study the dynamic buckling under shear loading, then validated numerically.

Lundquist (1935) was the first to investigate the buckling of circular cylindrical shells under transverse shear. Clough and Wilson (1971) addressed the case of thin shells in the broader context of nonlinear dynamics. Recently Karagiozov and Jones (2000, 2001, 2002, 2004) have examined the dynamic elastic buckling of elastoplastic cylindrical shells (tubes) using a discrete method called “Buckward Differentiation Formula BDF”. The effect of certain phenomena on the dynamic buckling were investigated but still so disparate, thus Petry and Fahlbush (2000) and others have studied the effect of imperfections, Simitis (1983) showed the effect of a static preload on the critical force.

Parametric studies were also held in this area, Virella et al. (2003) conducted a comparison of natural frequencies of cylindrical shells under the effect of different roof shapes, they have treated also three types of tanks with differents height-diameter ratios (H/D) by comparing modes of fundamental tank-liquid systems (Virella et al., 2006) and peak ground accelerations (PGA) relating to each type of tank (Virella et al., 2006). Regarding recommendations for the calculation of the efforts of various codes, a study was conducted by Barros (2008) by comparing the basal moments and basal shear of the various cylindrical tanks by varying the ratio of height-radius (H/R) one hand and the radius-thickness ratio (R/t) on the other hand. By cons Cao (2010) has treated the buckling of various tanks varying H/R and R/t under harmonic loading, and a recent parametric study of vibrations of different cylindrical shells under the effect of geometrics deformations took place (Kochurov and Avramov, 2011).

Characteristics of tanks and modeling of walls

The tanks to study are empty (under construction), with one end clamped and the other one free, schematically by elastic, thin, circular cylindrical shells. The metallic walled material is assumed to be homogeneous, isotropic, and linearly elastic with physical properties supposed to be Young's modulus $E= 210.10^6$ *KPa*, Poisson's ratio $\nu=0.3$ and density $\rho = 7.7\ t/m^3$.Fifteen tanks are studied, of same thickness (*t=1cm*) and different heights *H=10m, H=20m* and *H=30m* (Figure 1), each height contains five types of height-radius ratio (H/R) *1/3, 2/3*, *3/3*, *4/3* and *5/3* (Barros, 2010).

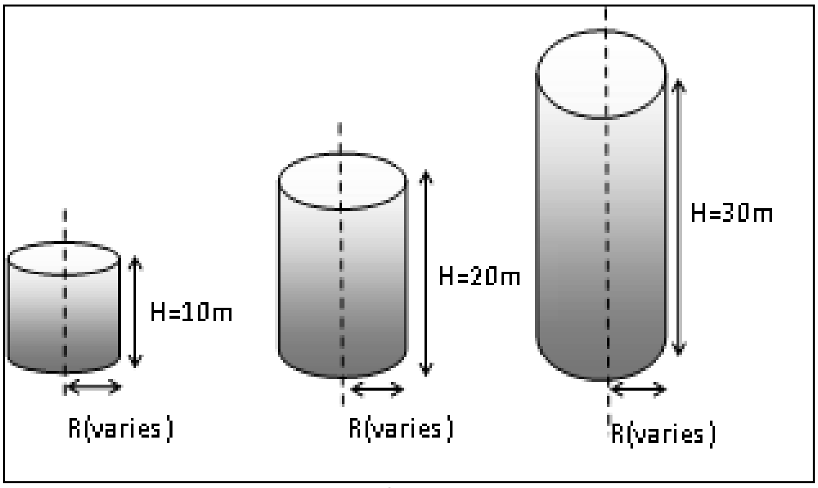

Figure 1. Characteristics of tanks with varying radius (R).

For modeling the thin-walled structures, a quadrilateral shell element, SHELL63 from the ANSYS element library (ANSYS 11 ,2004, Structural Analysis Guide), were used. These are four nodes, 3D elastic shells with both bending and membrane capabilities. The SHELL63 element has 6 DOFs at each node: 3 translations in the nodal x, y and z directions and 3 rotations about the nodal x, y and z axes. Convergence tests were conducted using a series of different meshes: 10, 16, 24, 30 and 60 divisions along the shell axis, until obtaining the values of neighboring loads with deviations of less than 3%.

Methodology

Criteria for Dynamic Critical Conditions

Several approaches are used to calculate the instability conditions for this type of problem (Huyan, 1996). In this paper, the dynamic buckling critical load is determined by the approach of Budiansky-Roth (1962). In this criterion, the response is calculated for different loading parameters from the numerical solution of equations of motion. By drawing the curve of displacement versus time while varying the intensity of the applied load, a jump of the curve is found from the curves drawn for neighbouring values. A particular value of the load causing this remarkable leap corresponds to the critical value of dynamic buckling (Figure 2).

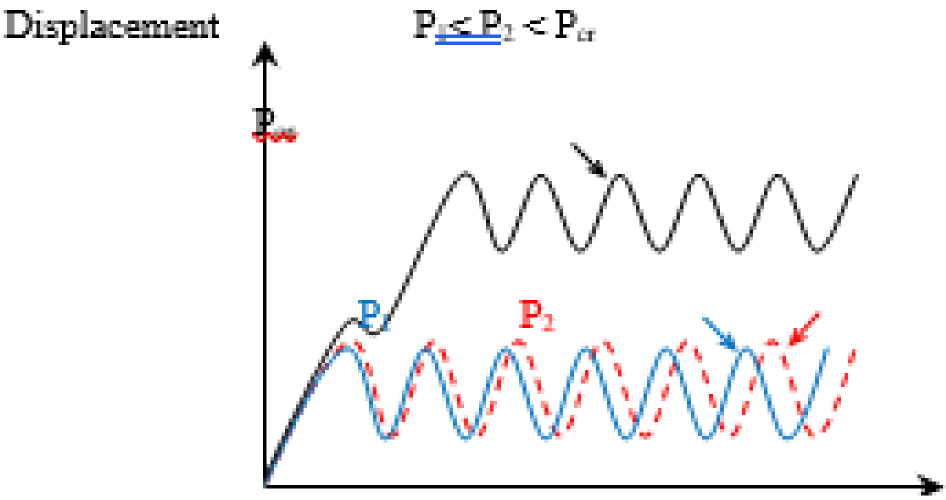

Figure 2. Detection of the critical load by criterion of Budiansky.

Once the buckling load is determined, the latter is verified by the phase plane (Hoff and Bruce, 1954), wich is plotted in a coordinate system (displacement, *x* and velocity, *dx/dt*). As the loading parameters are small, then stable movements describing paths closed are limited and focused around the solution of a static equilibrium in Figure 3a. On the other side, if the loading factor increases, a value is reached at which a movement gets away from the pole without any oscillation around him. In this case, the system is in the condition of instability having a critical value (Figure 3b).

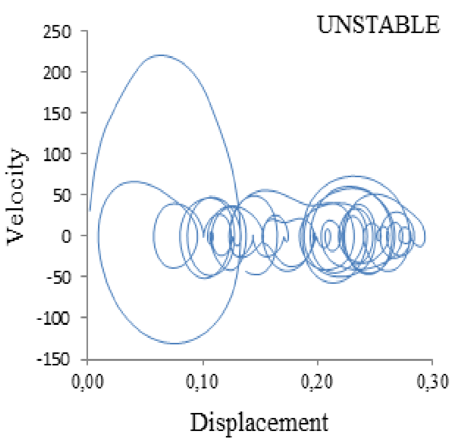

Figure 3.a. Phase-plane, stable motion.

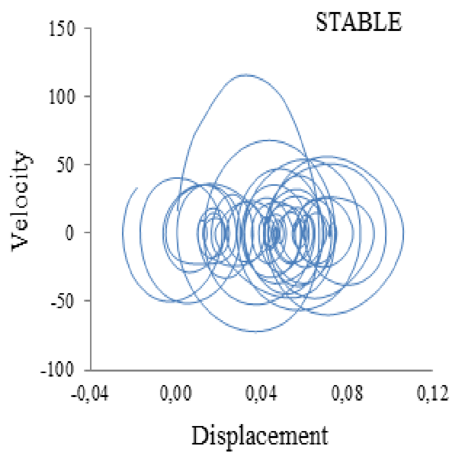

Figure3.b. Phase-plane,unstable motion.

Calculation Procedures

The calculation is achieved by a commercial software ANSYS (2004), solving the general equation of motion obtained by the different principles used in structural dynamics (Lagrange, Hamilton or virtual work)

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = P(t) \tag{1}$$

Where *M* is the mass matrix, *C* is damping matrix, *K* is the stiffness matrix, $\ddot{x}(t)$ is the acceleration vector, $\dot{x}(t)$ is the velocity vector, *x(t)* is the horizontal displacement vector and *P(t)* is the transient external dynamic load vector, concentrated horizontally at the top of the tank (Figure 4). This form of excitation is insensitive to the effects of imperfections relatively to the axial compression. The dynamic load is suddenly applied with constant magnitude and infinite duration which reflects the case of wind.

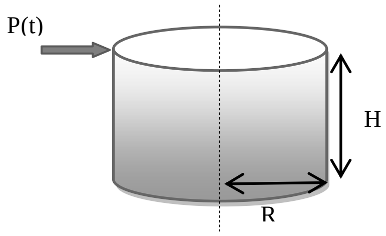

Figure 4. Tanks subjected to a concentrated load at the top.

To obtain answers on the displacement, velocity and accelerations, a transient analysis was used by solving the Equation 1 by direct integration method (step by step) often given by Newmark scheme (Newmark, 1959), displacement and velocity are developed at time n+1 by the Taylor formula (Equations 2 and Equation 3), $\beta_{s\ and}$ γ_s are specific to the Newmark scheme and are chosen to control stability and accuracy.

$$x_{n+1} = x_n + \Delta t\dot{x}_n + \Delta t^2(1 - 2\beta_s)\frac{\ddot{x}_n}{2} \tag{2}$$

$$\dot{x}_{n+1} = \dot{x}_n + \Delta t(1 - \gamma_s)\ddot{x}_n \tag{3}$$

The expressions of Equation 2 and equation 3 are injected into the dynamic equilibrium Equation 1 to calculate the acceleration at n+1, Δt is chosen so that $\Delta t \leq 1/(20.f)$, *f* is the fundamental frequency of the structure.

If:

$$S = (M + \Delta tC + \Delta t^2 K) \tag{4}$$

We obtain:

$$S\ddot{x}_{n+1} = P_{n+1} - C\dot{x}_{n+1} - Kx_{n+1} \tag{5}$$

S and the second member are fully known, the resolution of linear system (5) evaluates the acceleration at n+1. The displacement and velocity are finally obtained by injecting the value of this acceleration in expressions 2 and 3.

Numerical Results

The results of tanks of *H=10m* for different ratios of height-radius *H/R=1/3, 2/3, 3/3, 4/3* and *5/3* are shown in Figure 5, those tanks *H=20m* in the Figure 6 and for tanks of *H=30m*, the critical load values are shown in Figure 7.

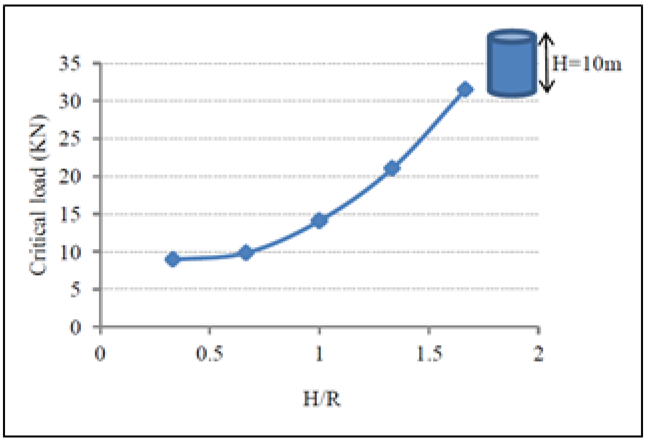

Figure 5. Variation of critical load versus h/r of tank h=10 m.

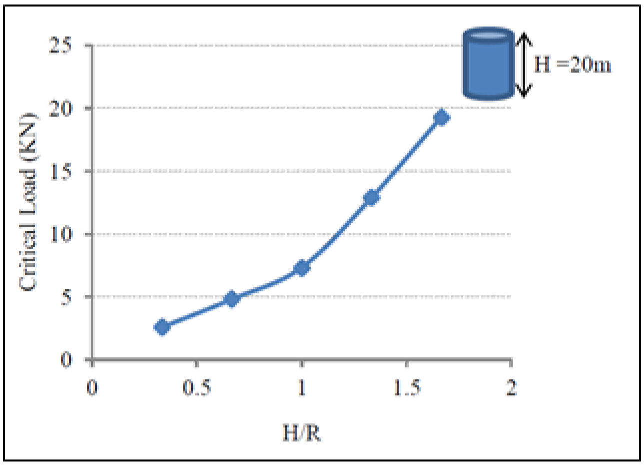

Figure 6. Variation of critical load versus h/r of tank h=20 m.

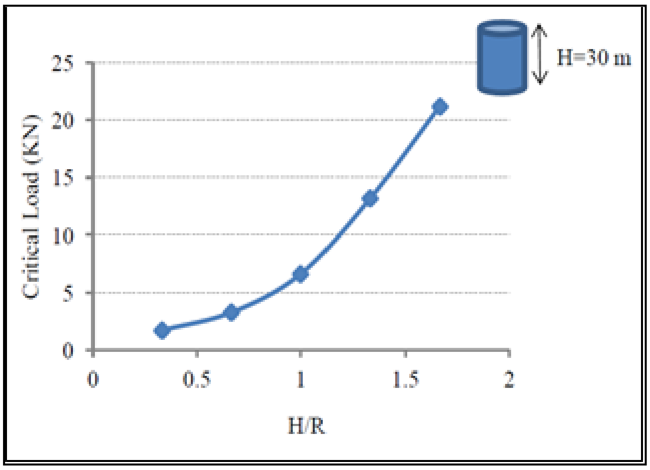

Figure 7. Variation of critical load versus h/r of tank h=30 m.

Conclusion

The results of the tank with constant thickness and radius-height ratios (H/R) variables, the dynamic buckling critical load increases with the increase of the ratio (H/R), this increase is considerable From R=H. while the stabilization of this charge appears to below R=3H, Similarly we can determine the optimum size of heights and radius of these tanks with H/R < 1 / 3.

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Author Information

Mokhtar Touati
University of Science & Technology Houari Boumediene
LBE, Civil Engineering Faculty, University of Science & Technology Houari Boumediene (USTHB)
BP 32 El-Alia / Bab Ezzouar / Algiers, ALGERIA
Contact e-mail: *touatimokh@yahoo.fr*

Mohamed Chabaat
University of Science & Technology Houari Boumediene
LBE, Civil Engineering Faculty, University of Science & Technology Houari Boumediene (USTHB)
BP 32 El-Alia / Bab Ezzouar / Algiers, ALGERIA