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THICKNESS RATIO EFFECT ON THE DYNAMIC RESPONSE OF A LONG CYLINDER TUBE UNDER MOVING PRESSURE

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Abstract. *In this study, the effect of thickness ratio on the dynamic response of a long cylinder tube subjected to an internal moving pressure traveling at a constant velocity has been investigated by using the finite element method. In this paper in each stage the tube wall thickness is considered to be uniform throughout as well as small in comparison with the mean radius of the tube. Various cylinders with different thickness to diameter ratio are assumed and their dynamic behaviors are compared. The thickness to diameter ratio is varied from 0.01 to 0.05. This study shows that the dynamic deflection is about 2.4 to 4.05 times the static displacement depending on cylinder thickness to diameter ratios. In long cylinders, this ratio has a little effect on dynamic to static deflection ratio for thickness to diameter ratio greater than 0.025. The location of maximum dynamic deflection is 0.78L to 0.98L of cylinder in the long cylinders.*

Key words: *Moving Pressure, Radial Dynamic Amplification Factor, Critical Velocity, Thickness-diameter Ratio*

1. INTRODUCTION

Considering potential hazards of moving pressure in pressure vessels and pipes, there is a significant development of reliable methods in design against ensuing sudden or progressive failures. In this respect, a better understanding of the structural response of cylindrical shells to internal moving pressure provides an important step towards reliable design of pressure vessels and pipes. Static analysis of the stresses in the tube wall is valid when the pressure front moves at low velocity. For high velocity the static analysis is not enough and the dynamic analysis must be applied.

The problem of structures subjected to traversing loads is sometimes encountered in modern engineering. A few examples of such structures are: bridges, overhead cranes and airport runways (Fryba) [1]. However, most of the works up to date have modeled these systems using analyses and approximations valid for beams or plate cases (Pesterev

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[2], Gbadeyan [3] and Renard [4]). A few authors have provided the basis for the derivation of the governing dynamic equations of general shells carrying moving loads but have, in practice, implemented beam or plate models only. Hence, the important problem of general shells, in particular cylindrical panels, under traversing pressure has been overlooked or received little attention. Tang [5] presented a model to predict the response of a semi-finite thin cylinder tube under internal moving pressure. Reismann [6] developed a model that includes the effect of pre-stress on the structural response. Also Faria [7] analyzed vibration of a cylinder panel with a moving force or mass by using the finite elements method. He studied the cylinder curvature effect.

Beltman [8] investigated the structural response of a tube to an internal gaseous detonation. Also Mazaheri [9][10] introduced a new analytical model for the transient elasto-dynamic structural response of cylindrical shells with finite length to internal detonation loading. All above researchers worked on a general cylinder without taking into consideration various geometrical parameters such as thickness. In previous work [11] the effect of length to diameter ratio is investigated. According to this study the cylinders with length greater that 3 must be considered as long cylinders. Also, in this study the cylinder thickness is assumed to be constant.

In this paper the effect of thickness to diameter ratio on the dynamic response of a long cylinder subjected to an internal moving pressure is investigated.

In the first part of this study, the dynamic behavior of various cylinder tubes under the action of internal moving pressure is investigated. Various long cylinders with different thickness to diameter ratios are considered and the radial dynamic amplification factor is calculated. The cylinder wall thickness is considered to be uniform throughout as well as small in comparison with mean radius of the tube.

The radial dynamic amplification factor (RDAF) is the ratio of radial dynamic displacement to radial static displacement.

In the second part, the effect of thickness to diameter ratio on the radial dynamic amplification factor is studied.

2. RADIAL DYNAMIC AMPLIFICATION FACTOR

The main objective of this investigation is the determination of radial dynamic amplification factor The RDAF is a convenient way of representing peak loads that can be expected and can be used by the designer to incorporate safety factors into the specification of piping systems that will be subjected to internal moving pressure.

Static pressure vessel design starts by considering the displacements that will be produced by a given internal pressure. Under the dynamic loading conditions, the actual displacements will be further amplified by the response of the structure to a time-dependent load. Therefore, an important factor in designing dynamic loading is the dynamic amplification factor which is defined as the ratio of the maximum dynamic displacement to the static displacement for the same nominal loading pressure. It should be noted that in this study the peak pressure is assumed to be low enough so that the structural behavior is wholly elastic. The amplification factor, also referred to as the dynamic load factor, may be defined as:

$$\text{RDAF} = \frac{W_d}{W_s} \tag{1}$$

Where

W_d = Maximum radial dynamic displacement

W_s = Radial static displacement

3. Finite Element Simulations

The procedure employed in this paper consists of determining the RDAF by using the finite element (FE) method. The RDAF is calculated for a cylinder subjected to a moving pressure (Fig. 1). The pressure front with constant value P_0 begins to move from the left side of cylinder with constant speed V.

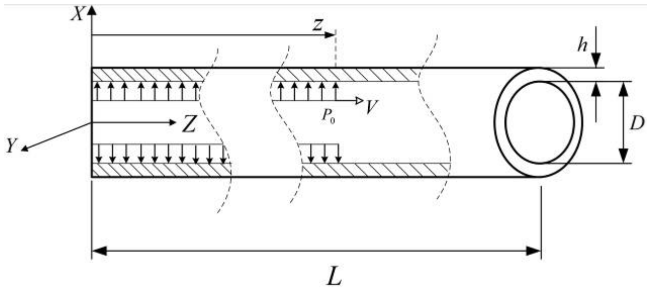


Fig. 1 Cylinder subjected to internal moving pressure [11]

In this paper various long cylinders with different thickness to diameter ratio ($0.01 \le h/D \le 0.05$) are considered. Length L of cylinders is $L = 5D$ so that cylinders are considered as long cylinder. A cylinder with L/D greater than 3 is defined as a long cylinder [11]. For each cylinder the static radial displacement is calculated.

The analysis algorithm that is used is similar to Ref. [11]. This procedure is performed for a wide range of pressure velocities in order to calculate the variation of RDAF of each cylinder with respect to moving pressure speeds.

The dynamic behavior of a simply supported cylinder subjected to an internal moving pressure is predicted using a finite element software Msc.Nastran [12]. The cylinder model is created by using 8-node quadrilateral iso-parametric element with considering shear deformation. The FE method is also used for a non-uniform cross-section beam [13].

Various mesh density in length direction in each cylinder is used to achieve a good mesh density for improving accuracy of results. Densifying is iteratively repeated until the difference between consecutive results becomes less than 0.5%. The radial mesh density is chosen so that elements aspect ratio becomes about 1. Fig. 2 shows the FE model that is used in this study. The FE models consist of 21141 nodes and 7001 elements. Each node has six degrees of freedom.

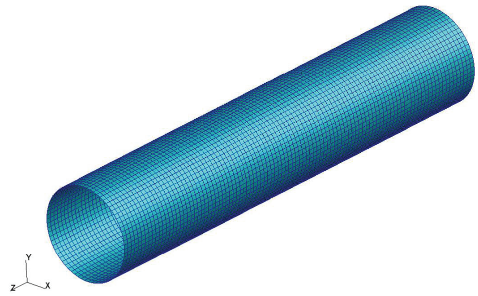


Fig. 2 Finite element model

The most important part of the FE method simulation is the modeling and preconditioning of a moving pressure load with required profile. Transient loading is represented by prescribing pressure as a function of time at each point. Fig. 3 shows the pressure history at point Z_i.

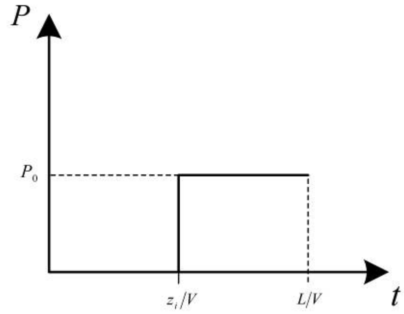


Fig. 3 Pressure distribution [11]

This pressure history for each point of cylinder is defined as

$$P(z,t) = P_0 u(t - z/V) \tag{2}$$

Where

P_0 = Pressure magnitude
u = Unit step function
V = Pressure traveling speed

This pressure is applied to the finite element model with about 1000 segments over a time-period of L/V, equal to the duration of each simulation.

The cylinders are simply supported and can be analyzed by employing the following boundary conditions:

Initial condition:

$$W(z,0) = \dot{W}(z,t) = 0 \tag{3}$$

Boundary condition:

$$W(0,t) = W(l,t) = 0$$
$$M(0,t) = M(l,t) = 0 \tag{4}$$

Where

$W(z,t)$ = The radial deflection at point z and time t

$M(z,t)$ = The bending moment at point z and time t

For transient analysis the Newmark [12] method is used to carry out integration with respect to time where, all parameters for t_n are known, t_{n+1} can be calculated by using the following equation of motion

$$\mathbf{M}\ddot{q}_{n+1} + \mathbf{B}\dot{q}_{n+1} + \mathbf{K}q_{n+1} = \mathbf{F}^{ext}_{n+1}, \tag{5}$$

Where M is the mass matrix, B the damping matrix, K the stiffness matrix, $\mathbf{F}^{ext}_{n+1}$ the external load vector, $\ddot{q}_{n+1}$ the acceleration at t_{n+1}, $\dot{q}_{n+1}$ the velocity at t_{n+1} and q_{n+1} is the displacement at t_{n+1}.

The estimates of q_{n+1}, $\dot{q}_{n+1}$ and $\ddot{q}_{n+1}$ are given by

$$\begin{aligned} q_{n+1} &= q^*_n + \beta\ddot{q}_{n+1}\Delta t^2, \\ \dot{q}_{n+1} &= \dot{q}^*_n + \gamma\ddot{q}_{n+1}\Delta t, \\ \ddot{q}_{n+1} &= \mathbf{M}^{*^{-1}}\mathbf{F}^{residual}_{n+1} \end{aligned} \tag{6}$$

Where Δt is the time step and γ and β are constants. The q^*_n, $\dot{q}^*_n$, Matrix M^* and vector $\mathbf{F}^{residual}_{n+1}$ are calculated by these equations.

$$\begin{aligned} \mathbf{F}^{residual}_{n+1} &= \mathbf{F}^{ext}_{n+1} - \mathbf{B}\dot{q}^*_n - \mathbf{K}q^*_n, \\ \mathbf{M}^* &= \mathbf{M} + \mathbf{B}\gamma\Delta t + \mathbf{K}\beta\Delta t^2, \\ \dot{q}^*_n &= \dot{q}_n + (1-\gamma)\ddot{q}_n\Delta t, \\ q^*_n &= q_n + \dot{q}_n\Delta t + \frac{(1-2\beta)\ddot{q}_n\Delta t^2}{2}, \end{aligned} \tag{7}$$

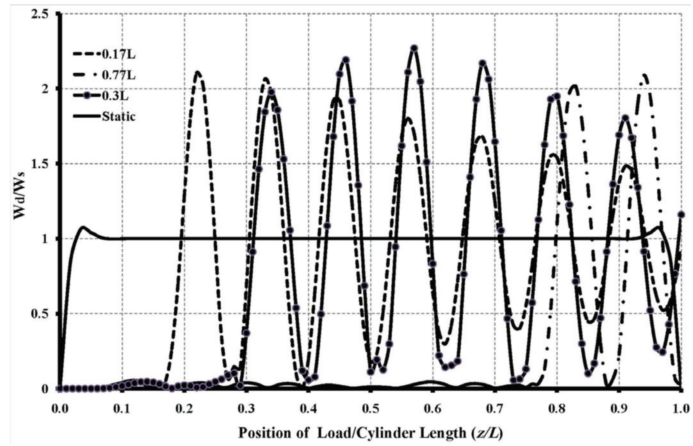
4. Results and Discussion

The cylinders have diameter D = 7 mm, length L = 35 mm. The property of cylinders is elastic modulus $E = 7\times10^{10}$ Pa, Poisson ratio $\upsilon = 0.3$ and mass density ρ = 2300 kg/m^3.

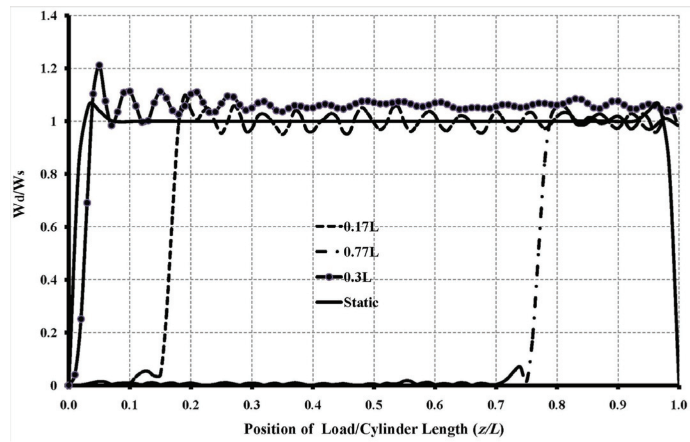
The thickness h of each cylinder varies from $h = 0.01D$ to $h = 0.05D$. The computations are performed for different values of the thickness to diameter (h/D) ratio and a constant length to diameter ratios (L/D = 5). The L/D ratio is chosen so that the cylinder is considered as a long one ((L/D) > 3) [11].

The pressure front begins to move from left side of cylinders with constant intensity (P_0 = 1.3 Mpa). The velocity of the pressure front is uniform and constant in each calculation stage.

Fig. 4 shows the variation in the RDAF of various points of two cylinders with respect to the position of internal moving pressure. According to these figures for low pressure velocity the maximum radial dynamic displacement is close to static displacement (maximum radial dynamic amplification factor, mRDAF, is about 1.0) but with increasing the pressure velocity the maximum radial dynamic displacement becomes higher than static displacement, thus the static analysis of the cylinder is valid only when the pressure front moves with low velocity.



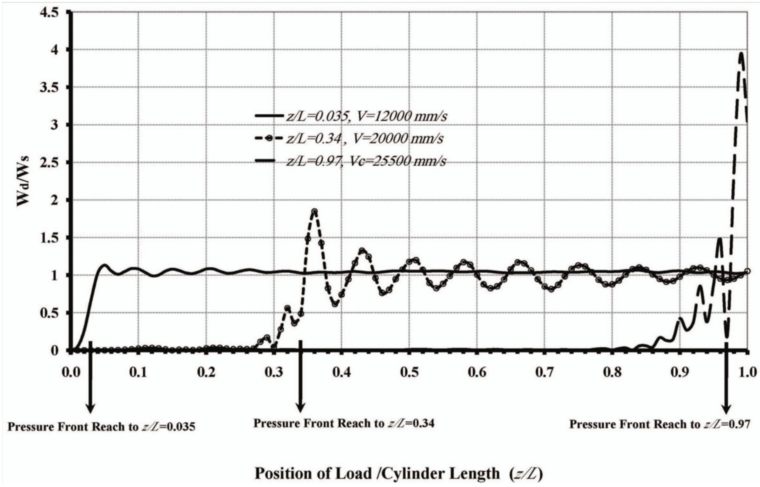
(a)



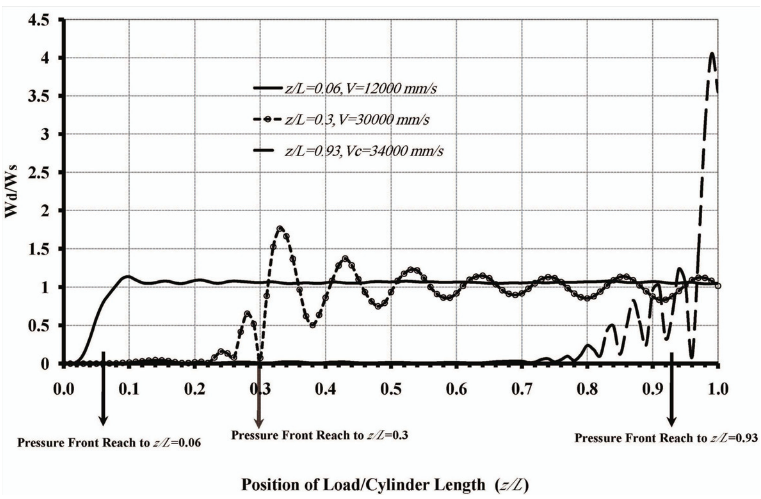
(b)

Fig. 4 Comparison between dynamic and static displacements in different cylinder points for $h/D = 0.03$ at pressure velocity: a) 12000 mm/s, b) 30000 mm/s

In Fig. the variation of the dynamic deflection divided by static deflection for two cylinders is drawn with respect to the position of the moving pressure. In this figure the location of maximum deflection is various velocities is shown. By referring to this figure, it can be concluded that maximum dynamic deflection has a delay with respect to the position of moving pressure. This delay increases by increasing the velocity. The quantity V_c in Fig. is the critical velocity that the mRDAF achieves.



(a)



(b)

Fig. 5 Dynamic to static deflection ratio with respect to the pressure position: a) $h/D = 0.025$, b) $h/D = 0.04$

The location of maximum dynamic deflection of a simply supported beam is at the middle of span [14] but in the long cylinders this location depends on pressure velocity and thickness to diameter ratio. For example, in cylinder with h/D = 0.025 the maximum dynamic deflection occurs at z/L = 0.05 in V = 12000 mm/s when the velocity increases to V_c = 25500 mm/s the location of maximum dynamic deflection moves to z/L = 0.97. In a long cylinder at low pressure velocity the location of maximum dynamic deflection is at the left side of cylinder but once the velocity increases to critical velocity this location moves to the right side of cylinder.

As can be seen from Fig. 6 the location of maximum dynamic deflection (Z_{m_dyn}) is with respect to h/D ratio. According to this figure the maximum dynamic deflection for long cylinders is between 0.78 L to 0.98 L of cylinder.

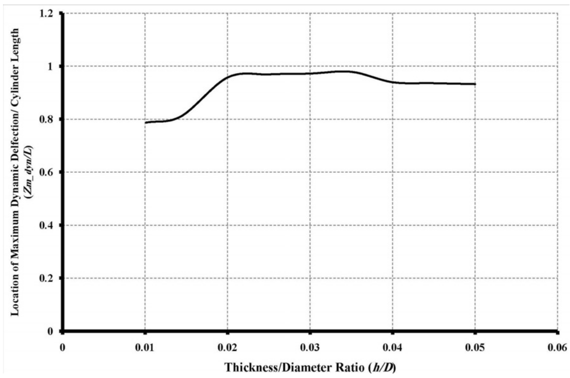


Fig. 6 Location of maximum dynamic deflection in various h/D ratios

Fig. 7 shows the variation of mRDAF respect to h/D ratio; accordingly, the mRDAF increases until h/D = 0.025 and then it become constant. This graph shows that increasing the thickness to diameter ratio for decreasing the mRDAF is the waste of profit especially for (h/D) > 0.025. For all h/D ratios the mRDAF is changed between 2.4 to 4.05 in the long cylinders.

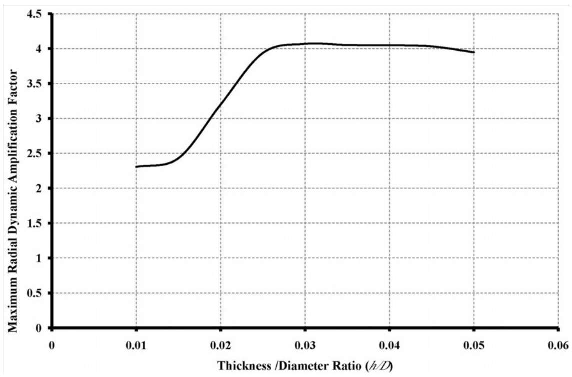


Fig. 7 Maximum radial dynamic amplification factor with respect to h/D various ratios

5. Conclusions

The following conclusions can be drawn from the present investigation:

1. At low pressure velocity, the static analysis is adequate, but with increase in velocity, at high pressure velocity, the transient analysis is essential.
2. The location of mRDAF shifts to the right side of cylinder with increasing velocity.
3. The location of mRDAF for all h/D ratios is between 0.78 L to 0.98 L.
4. The maximum radial dynamic amplification factor is dependent on thickness to diameter ratio.
5. The location of mRDAF has delay with respect to the position of moving pressure. This time delay increases with increasing h/D ratio.
6. The mRDAF for all h/D ratios in long cylinder is between 2.4 to 4.05. This ratio increases with increasing h/D ratio up to $h/D = 0.025$.
7. The mRDAF has a constant behavior for $h/D > 0.025$.

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EFEKAT KOEFICIJENTA GUSTINE NA DINAMIČKU REAKCIJU DUGE CILINDRIČNE CEVI POD POKRETNIM PRITISKOM

Kambiz Bakhshandeh - Bahador Saranjam

U ovom istraživanju se ispituje, uz pomoć metode konačnih elemenata, dejstvo koeficijenta gustine na dinamičku reakciju duge cilindrične cevi izložene unutrašnjem pokretnom pritisku koji se kreće sa konstantnom brzinom. U ovom radu se smatra da je, u svakoj fazi, debljina zida cevi jednaka celom svojom dužinom a mala u poređenju sa srednjim poluprečnikom date cevi. Uzeti su u ispitivanje različiti cilindri sa različitim odnosom gustine prema prečniku a upoređena su njihova dinamička ponašanja. Odnos gustine prema prečniku se menjao od 0.01 do 0.05. Ovo istraživanje pokazuje da je dinamičko skretanje za oko 2,4 do 4,05 puta umnoženo statičko pomeranje i da zavisi od odnosa gustine cilindra prema prečniku. U dužim cilindrima, ovaj odnos ima mali efekat na koeficijent dinamičkog prema statičkom pomeranju za odnos gustine i prečnika viši od 0.025. Mesto maksimalnog dinamičkog pomeranja je 0.78 L prema 0.98 L cilindra u dugim cilindrima.

Ključne reči: *pokretni pritisak, radijalni faktor dinamičkog pojačanja, kritična brzina, odnos gustine i prečnika*