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Vibrations Analysis of Pipe Convoying Pulsating Fluid Flow

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Abstract: ***Non-uniform pipes conveying fluids are widely used in various industrial fields. Also, contraction and expansion pipes are used at inlet and outlet of industrial plants equipment like pumps and compressors. There equipments produce vibrations which badly affect pipes. It is the aim of this proposed research work to study the relative influence of the various parameters on the vibration characteristics of the pipes and their relation to the pipe shape parameters like inlet diameter to outlet diameter, expansion or contraction position along the pipe, and contraction or expansion length. Numerous research works have been put forward to treat the dynamics of pipelines subject to different loading conditions and structural constraints. In this work various pipe shapes are studied and compared to the straight pipe as a reference. The governing equations of motion for a pipe conveying fluid were solved using BVP4C in MATLAB software***

Keywords: Pipes, Vibrations, pulsating, fluid

1. Introduction

Pipes conveying fluids are an important research subject of interest for engineers due to its widely usage in engineering applications. Pipes used in transferring fluids between equipment like in the petrochemicals processes, Fertilizers plant and also transferring fluids for a long distance like LPG pipelines, water pipelines between cities in most practices' pipes are exposed to vibrations caused by rotary equipment like pumps, compressors or by wind. There force cause stresses in pipe sections which in some cases lead the pipe material to fail.

Ismael et al (1981) studied the dynamics of annulus pipe conveying fluid and described it by means of transfer matrix method. They found that the outer and the inner pipes of the annular may vibrate individually in different mode shapes. Wang and Bloom (1999) carried out research topic related directly to the concentric pipe system designs in silo or other mixing units. It has been found that concentric pipe mixers have a long suspended inner pipe. Aldraihem and Baz (2004) investigated the dynamic stability and response of stepped tubes subjected to a stream of moving objects. Ibrahim (2010) carried out dynamic and stability analysis for pipes conveying fluid together with curved and articulated pipes. Different types of modelling, dynamic analyses, and stability of pipes conveying fluid with different boundary conditions have been assessed. Ibrahim (2010) Worked on the problem of fluid elastic instability in single- and two-phase flows and fretting wear in process equipment, such as heat exchangers and steam generators. Tawfik et al (2009) studied the vibrated pipe conveying fluid with sudden enlargement and exposed to heat flux. The governing equations of motion for this system are derived by using beam theory. They found that the fluid forces (Coriolis and Compressive) greatly affect the response of the undamped pipe under vibration. Chen (1975) presented a linear theory to account for the motions of extensible curved pipes conveying fluid. Based on the theory, the flow-induced deformations are obtained in closed form. Olunloyo et al (2007) studied the energy method and they were invoked to derive the governing equations including the effects of external temperature variation along the length of the pre-stressed and pressurized pipe. Simha and Kameswara(2001) developed a finite element program for rotationally restrained long pipes with internal flow and resting on Winkler foundation. They found that in all cases, the natural frequency parameter decreases with increasing flow velocity parameter and increase consistently with increasing foundation stiffness parameter. Reddy and Wang (2004) worked on complete set of equations of motion governing fluid- conveying beams are derived using the dynamic version of the principle of virtual displacements. Equations for both the Euler—Bernoulli and Timoshenko beam theories were developed. Stein and Tobriner(1970) worked on numerical solution to the equation of motion that describes the behavior of an elastically supported pipe of infinite length conveying an ideal pressurized fluid. Baheli(2012) studied the dynamic behaviour of pipe conveying fluid at different cross section. Three kinds of supports are used, which are flexible, simply and rigid supports. He found that the values of the natural frequencies for flexible support are less than those for simply and rigid supports. Fernad et al (1999) carried out simplified method for evaluating the fundamental frequency for the bending vibrations of cracked Euler- Bernoulli beams are presented. Its validity is confirmed by comparison with numerical simulation results. Fengchun, et al (2010) studied the effects of the non-propagating open cracks on the dynamic behaviours of a cantilevered pipe conveying fluid. They concluded that the equations of motion for the cantilevered pipes conveying fluid with an arbitrary number of cracks are developed based on the extended Lagrange equations for systems containing non-material volumes. Yoon, et al

(2007), studied the influence of two open cracks on the dynamic behaviour of a double cracked simply supported beam. Yoon and son (2014) studied the effect of the open crack and the moving mass on the dynamic behaviour of simply supported pipe conveying fluid, they found that When the crack position exists in canter of the pipe conveying fluid, its frequency has the smallest value. Murigendrappa et al (2014) worked on a technique based on measurement of change of natural frequencies to detect multiple cracks in long pipes containing fluid at different pressures. Al-Sahib et al (2010) studied conveying turbulent steady water with different velocities and boundary conditions; the main summarized conclusions are the natural frequencies of a welded pipe with steady flow decreases with increasing the fluid flow velocity in both clamped-clamped and clamped-pinned boundary conditions. Kuiper et al (2004) worked on analytical proof of stability of a clamped-pinned pipe conveying fluid at a low speed is given. The results show this approach could keep stable even for long period of time and is much more rapid than traditional Runge-Kutta method. Shuai-jun et al (2014) considered the effects of pipe wall thickness, fluid pressure and velocity, which describe the fluid–structure interaction behaviour of pipelines. The theoretical results show that the effect of the variation of support position and stiffness is dominant for the lower flexural modes and the higher torsional modes. Tornabene et al (2010) studied the stability of a cantilever pipe conveying fluid by means of the generalized differential quadrature method. Czerwiñski and £uczko(2012) developed on analysis of a model describing the vibrations of simply supported straight pipes conveying periodically pulsatingfluid. They concluded that the considered geometrically non-linear model allows estimating the value of the vibration in the regions of parametric resonance and for flow velocity higher than the critical. Zhang et al (1999) discussed a finite element model in which flowing fluid and moving pipes have been fully coupled using the Eulerian approach and the concept of fictitious loads for the kinematic corrections. Mediano-valiente and Garc´ia-planas, (2014)studied non-linear dynamic model for a pipe conveying fluid. Moreover, a linearization method had been done by approximation of the non-linear system to the linear gyroscopic system. Boiangiu et al (2014) they solved the differential equations for free bending vibrations of straight beams with variable cross section Bessel’s functions. Fresquet et al (2015) studied, the increasing complexity found in onshore and offshore wells demands profound knowledge on the performance to fatigue of threaded connections used in the different stages of hydrocarbons exploitation. The tabulated results obtained by this method were compared against FEA results as well as experimental results obtained during resonant bending tests, showing very good matching. Coşkun et al (2011) solved the vibration problems of uniform and nonuniform Euler-Bernoulli beams analytically or approximately for various end conditions. Al-Hashimy et al (2014) studied the Vibration characteristics of pipe conveying fluid with sudden enlargement-sudden contraction were. they concluded that the natural frequencies for pipe system conveying fluid is less than the natural frequencies for pipe system without fluid. Ritto et al (2014) Studied the problem of a pipe conveying fluid of interest in several engineering applications, such as micro-systems or drill-string dynamics. Collet and Källman(2017) studied pipe vibrations Measurements. They concluded that Each pipe vibration problem is unique and requires a deep understanding of active process events

2. Mathematical Model

Consider a pipe of variable cross-section A(x), length L, modulus of elasticity E, and its second moment of area I(x). A fluid flow through the pipe having a density ρ_f (see Figure1), the pipe is vibrated due to an exciting force f_{ex} (t,x). Figure (2) shows the forces acting on elements of fluid and pipe. Resolving the forces on fluid element along and perpendicular to the tangent to the center line of the deflected element taking into account, $\frac{\partial (AP)}{\partial x} + q\,s = 0$.

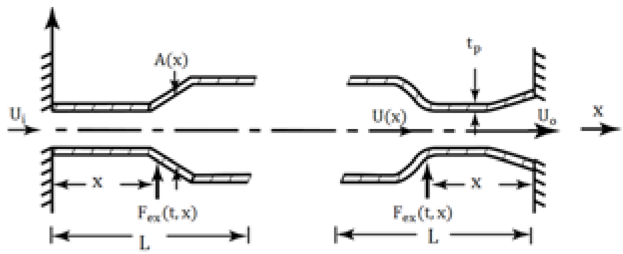


Figure 1: Layout of the pipegeometry and exciting force

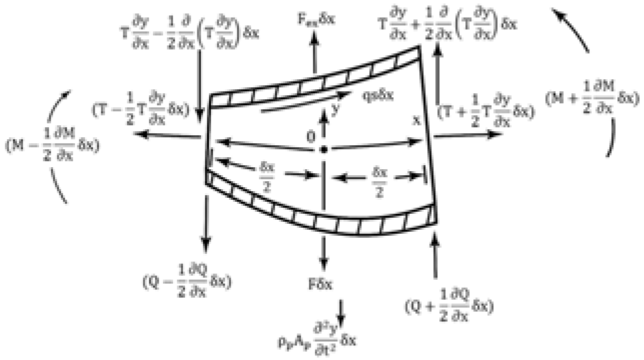
for small deformations and neglect $\frac{\partial y}{\partial x}\frac{\partial (AP)}{\partial x}$. The forces on the element of the pipe normal to the pipe axis accelerate the pipe element“(b) in Figure 2” in the Y direction. For small deformations: $U\frac{\partial y}{\partial x}\frac{\partial U}{\partial x}$ is negligible, where S is the inner perimeter of the pipe, and q is the shear stress on the internal surface of the pipe. The equations governing the force on the tube element are derived as follows: Where ρ_p is the Density of the empty pipe. The bending moment M in the pipe, the transverse shear force Q and the pipe deformation is related by the transverse shear force in the pipe and T is the longitudinal tension in the pipe.

$$E\frac{\partial^2}{\partial x^2}\left(I\frac{\partial^2 y}{\partial x^2}\right) + (\rho_f A U^2 + PA - T)\frac{\partial^2 y}{\partial x^2} + \left(\rho_p A_p + \rho_f A\right)\frac{\partial^2 y}{\partial t^2} + (2U\rho_f A)\frac{\partial^2 y}{\partial x\,\partial t} = F_{ex.} \qquad (1)$$

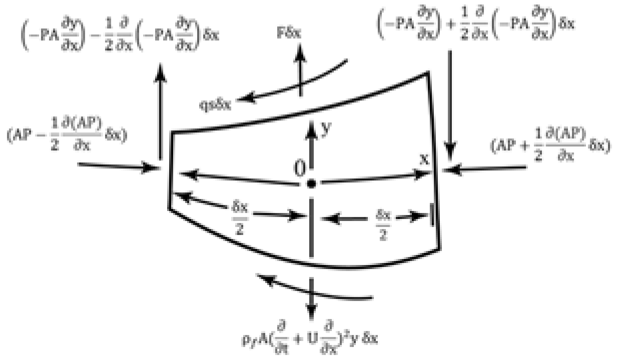
A_P is the area of the pipe, A inner area of pipe at any distance x, I is the second moment of area

$$A_p = \frac{\pi}{4}\left(d + 2t_p\right)^2 - \frac{\pi}{4}d^2 t_p \ll {}^{d_i}/_2$$

$$A_p = A\frac{d_i}{d(x)}\left[\frac{4t_p}{d_i}\right]$$



(a) Pipe Element



(b) Fluid element

Figure 2: Forces and Moments acting on Elements of Fluid and Pipe.

2.1 The Flow in the pipe

Consider in viscid flow along the pipe and Euler momentum equation is to be applied

$\rho_f U \frac{dU}{dx} = -\frac{dp}{dx}$But Q_f is the volume flow rate and that is constant due to continuity consideration, A_i is the area of pipe at inlet ,

$\rho_f \frac{Q_f}{A}\frac{d}{dx}\left(\frac{Q_f}{A}\right) = -\frac{dp}{dx}$, $\int_{P_i}^{P} P = \rho_f {Q_f}^2 \int_{A_i}^{A} \frac{1}{A^3} dA$, therefore,

$P = P_i + \frac{\rho_f}{2} {Q_f}^2 \left(\frac{1}{{A_i}^2} - \frac{1}{A^2}\right)$and,$U = \frac{Q_f}{A}$

$\frac{d}{dx}(PA - T) = 0$, thus $PA - T = constant$,

$$PA - T = P_iA_i - T_i \qquad (2)$$

2.2 Dimensionless variables

$x^* = \frac{x}{L}$, $y^* = \frac{y}{L}$, $t^* = \left[\frac{EI_i}{(\rho_f+\rho_p)A_iL^4}\right]^{1/2} t$,

$P^* = P\Big/\left(\frac{EI_i}{A_iL^2}\right)$, $T^* = T\Big/\left(\frac{EI_i}{L^2}\right)$, $U^* = U\Big/\frac{1}{L}\left(\frac{EI_i}{\rho_fA_i}\right)^{1/2}$,

or $U^* = \frac{{Q_f}^*}{A^*}$, $A^* = \frac{A}{A_i}$, ${t_p}^* = \frac{t_p}{d_i}$, ${A_p}^* = \frac{A_p}{A_i} = 4A^*\frac{{t_p}^*}{d^*}$,

$d^* = \frac{d(x)}{d_i}$, ${d_o}^* = \frac{d_o}{d_i}$, $I^* = \frac{2d.t_p[2d^2+4d.t_p]}{2d_i.t_p[2{d_i}^2+4d_i.t_p]}$

Differentiating I^* twice w.r.t. x^*we have,

$$\frac{dI^*}{dx^*} = \frac{\left(3/2\,{A^*}^{1/2} + 2{t_p}^*\right)}{(1+2{t_p}^*)}\frac{dA^*}{dx^*} ,$$

$$\frac{d^2I^*}{dx^{*2}} = \left\{\left(3/2\,{A^*}^{1/2} + 2{t_p}^*\right)\frac{d^2A^*}{dx^{*2}} + 3/4\,{A^*}^{-1/2}\left(\frac{dA^*}{dx^*}\right)^2\right\}\frac{1}{(1+2{t_p}^*)}$$

${F_{ex}}^* = F_{ex}\Big/\left(\frac{EI_i}{L^3}\right)$, $C = \frac{\rho_p}{\rho_f}$, $\beta = \frac{1}{\rho_p/\rho_f + 1} = \frac{1}{C+1}$, $\alpha = \frac{(4C*{t_p}^*)+1}{C+1}$

Substituting into Eqn. (1),

$$\frac{\partial^2}{\partial x^2}\left(I^*\frac{\partial^2y^*}{\partial x^{*2}}\right) + \left(A^*U^{*2} + P^*A^* - T^*\right)\frac{\partial^2y^*}{\partial x^{*2}} + A^*\alpha\frac{\partial^2y^*}{\partial t^{*2}} + 2U^*A^*\beta^{1/2}\frac{\partial^2y^*}{\partial x^*\partial t^*} = {F_{ex}}^* \qquad (3)$$

The boundary conditions (Fixed-Fixed support) are

$y^*(0,t^*) = y^*(1,t^*) = 0$, $\frac{\partial y^*}{\partial x^*}|_{x^*=0} = \frac{\partial y^*}{\partial x^*}|_{x^*=1} = 0$

The exciting force F*is a function of x*and t*. So, for sinusoidal excitation, the dimensionless exciting force may be put in the following form:

${F_{ex}}^* = f^*(x^*)e^{i\Omega t^*}$, Where Ω is dimensionless circular frequency. $\Omega = \frac{\omega}{\left[\frac{EI_i}{(\rho_f+\rho_p)A_iL^4}\right]^{1/2}}$

Let $y^* = Y(x^*)e^{i\Omega t^*}$,

Where $Y(x^*)$ is the complex dimensionless amplitude.

$$\frac{d^2}{dx^2}\left(I^*\frac{d^2Y}{dx^{*2}}\right) + \left(A^*U^{*2} + P^*A^* - T^*\right)\frac{d^2Y}{dx^{*2}} - \Omega^2A^*\alpha Y + 2i\Omega U^*A^*\beta^{1/2}\frac{dY}{dx^*} = f^* \qquad (4)$$

$Y(0) = Y(1) = 0$, $\frac{dY}{dx^*}|_{x^*=0} = \frac{dY}{dx^*}|_{x^*=1} = 0$

Let $Y = y_r + iy_i$. So,

$$\frac{d^2}{dx^2}\left(I^*\frac{d^2y_r}{dx^{*2}}\right) + \left(A^*U^{*2} + P^*A^* - T^*\right)\frac{d^2y_r}{dx^{*2}} - \Omega^2A^*\alpha y_r - 2\Omega U^*A^*\beta^{1/2}\frac{dy_i}{dx^*} = f^* \qquad (5)$$

$$\frac{d^2}{dx^2}\left(I^*\frac{d^2y_i}{dx^{*2}}\right) + \left(A^*U^{*2} + P^*A^* - T^*\right)\frac{d^2y_i}{dx^{*2}} - \Omega^2A^*\alpha y_i + 2\Omega U^*A^*\beta^{1/2}\frac{dy_r}{dx^*} = 0 \qquad (6)$$

$y_r(0) = y_r(1) = y_i(0) = y_i(1) = 0$,$\frac{dy_r}{dx^*}|_{x^*=0} = \frac{dy_r}{dx^*}|_{x^*=1} = \frac{dy_i}{dx^*}|_{x^*=0} = \frac{dy_i}{dx^*}|_{x^*=1} = 0$

That is, $P^*A^* - T^* = (P_iA_i - T_i)\left(\frac{L^2}{EI_i}\right) = \lambda$

Where λ is a constant.

$$\text{So,}\quad {y_r}'''' + \left(\frac{2}{I^*}\frac{dI^*}{dx^*}\right){y_r}''' + \frac{1}{I^*}\left(A^*U^{*2} + \lambda + \frac{d^2I^*}{dx^{*2}}\right){y_r}'' - \Omega^2\frac{A^*}{I^*}\alpha y_r - 2\Omega U^*\frac{A^*}{I^*}\beta^{1/2}{y_i}' = \frac{f^*}{I^*} \qquad (7)$$

Where the primes donate differential w.r.t. x

$C_1 = \frac{2}{I^*}\frac{dI^*}{dx^*}$, $C_2 = \frac{1}{I^*}\left(A^*U^{*2} + \lambda + \frac{d^2I^*}{dx^{*2}}\right)$,

$C_3 = \Omega^2\frac{A^*}{I^*}\alpha$, $C_4 = 2\Omega U^*\frac{A^*}{I^*}\beta^{1/2}$,$C_5 = \frac{f^*}{I^*}$

Equations5 and 6 becomes,

$${y_r}'''' + C_1{y_r}''' + C_2{y_r}'' - C_3y_r - C_4{y_i}' = C_5 \qquad (8)$$

$${y_i}'''' + C_1{y_i}''' + C_2{y_i}'' - C_3y_i + C_4{y_r}' = 0 \qquad (9)$$

With boundary conditions

$y_r(0) = y_r(1) = y_i(0) = y_i(1) = 0$,

${y_r}'(0) = {y_r}'(1) = {y_i}'(0) = {y_i}'(1) = 0$

The solution to equation (8) and (9) may be put in the following form

$y^* = Re\left[Y(x^*)e^{i\Omega t^*}\right]$,

$$y^* = y_r\cos\Omega t^* + y_i\sin\Omega t^* \qquad (10)$$

$y^* = \left(\sqrt{{y_r}^2 + {y_i}^2}\right)(\cos\Phi\cos\Omega t^* - \sin\Phi\sin\Omega t^*)$,

$y^* = Z\cos(\Omega t^* + \Phi)$ (11)

$Z = \sqrt{y_r^2 + y_i^2}$, $\Phi = \tan^{-1}\frac{y_i}{y_r}$

Where Z is the amplitude of oscillation and Φ is the phase shift,

2.3 Pipe geometry

Let the variation of pipe diameter be given by the following polynomial

$$d^* = b_5 x^{*5} + b_4 x^{*4} + b_3 x^{*3} + b_2 x^{*2} + b_1 x^* + b_0 \quad (12)$$

Conditions

$x^* = 0,\ d^* = \frac{d_i^*}{2}, \frac{d(d^*)}{dx} = 0, \qquad \frac{d^2(d^*)}{dx^2} = 0$

$x^* = L,\ d^* = \frac{d_o^*}{2}, \qquad \frac{d(d^*)}{dx} = 0, \qquad \frac{d^2(d^*)}{dx^2} = 0$

Thus, $b_0 = \frac{d_i^*}{2}, b_1 = 0, \quad b_2 = 0$

Applying those boundary conditions that we have:

The first derivative of d^*

$$\frac{d(d^*)}{dx^*} = 5b_5(x^* - L_i^*)^4 + 4b_4(x^* - L_i^*)^3 + 3b_3(x^* - L_i^*)^2 \quad (13)$$

$$d^* = b_5(x^* - L_i^*)^5 + b_4(x^* - L_i^*)^4 + b_3(x^* - L_i^*)^3 + \frac{d_i^*}{2} \quad (14)$$

The second Derivative of d^*

$$\frac{d^2(d^*)}{dx^{*2}} = 20(x^* - L_i^*)^3 + 12b_4(x^* - L_i^*)^2 + 6b_3(x^* - L_i^*) \quad (15)$$

2.4 Exciting force acting on the pipe

Assuming the dimensionless exciting force f^* to take the following form

$$f^*/f^*_{max} = n + (1 - n)[\sin(\pi x^*)]^{2m} \quad (16)$$

Where n and m are the force control parameters

2.5 Numerical solution

The fourth order equations 8 and 9 are to be transformed to eight first-order equations prior to a numerical solution using the MATLAB code BVP4C for solving the boundary value problem. And using he coefficient C_1, C_2, C_3 and C_4 may be rewritten as.

$$C_1 = \frac{2}{I^*}\frac{dI^*}{dx^*}, \qquad C_2 = \frac{1}{I^*}\left(\frac{Q^{*2}}{A^*} + \lambda + \frac{d^2 I^*}{dx^{*2}}\right), \qquad C_3 = \mathbf{\Omega}^2 \frac{A^*}{I^*}\alpha,$$

$$C_4 = 2\frac{Q^*}{I^*}\beta^{1/2}\mathbf{\Omega}$$

The parameters that influence the pipe vibration response are $\frac{d_o}{d_i}$, $\mathbf{\Omega}$, Q^* and λ.

3. Results and Discussion

The results are based on a pipe material of steel with a modulus of elasticity of 210GPa and density of 7850 kg/m^3. Also, the pipe length L is 10 m and the pipe thickness are 12 mm. Moreover, the fluid used in this study in all cases is water having a density equal to 1000 kg/m^3, pressure at entrance of 2 bar and the flow velocity is .48 m/s. The exciting force has a maximum amplitude in the middle of the pipe and it has a maximum of 1000 N for all of the cases. Cases having different pipe shapes are classified as pipes with expansion cross-section and pipes with contraction cross-section. The expansion or contraction area having a varying length L_{ci} which has three values .01, 0.1 and 0.2 of the pipe lengths. The position of expansion/contraction cross section along the pipe L_i was varied using three values 0.2, 0.5 and 0.8 of the pipe length L. Moreover, the pipe diameter ratio was studied using three ratios. These pipe shape parameters variation allowed to study of 27 expansion pipe cases and the same number of cases for contraction pipe shape to be carried out. All cases compared with straight pipe results as a reference case. The frequency changing up to 300 Hz was considered.

3.1. Pipe with expansion cross section area

The cases to be considered are those having a constant outlet diameter of pipe d_o equal 400 mm and the entry diameter of pipe d_i is varied to get three pipe diameter ratios d_i/d_o=.4,.6and .8,The expansion region also has three positions 0.2, 0.5 and 0.8 of the pipe length and dimensionless expansion length L_{ci} as 0.01, 0.1 and 0.2. cases are compared with straight pipe as reference case.

Figures (3), (7) and (11) shows vibration Amplitude Y* versus dimensionless frequency $\mathbf{\Omega}$ at different entry and expansion length having diameter ratio d_i/d_o=0.4, 0 .6 and 0.8 respectively.

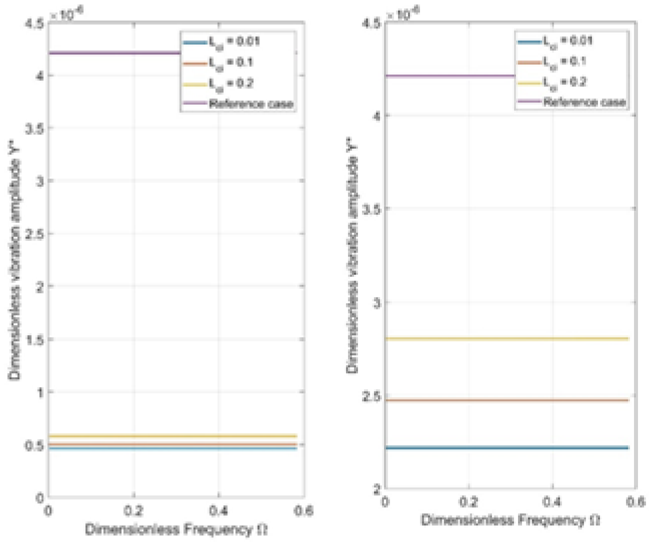
It is noticed that the results are consistent and increasing dimensionless frequency $\mathbf{\Omega}$ has no effect on dimensionless vibration amplitude Y*. vibration amplitude Y* increases with the increase in expansion length L_{ci} for all values of expansion entry length L_i. Moreover, all cases have lower vibration amplitude than straight pipe. While the entry length L_i and the expansion dimensionless length L_{ci} increases the vibration amplitude value get close to the straight pipe value.

Figure (4), (8) and (12) Show pressure gradient variation along pipe length having diameter ratio d_i/d_o=0.4,0 .6 and 0.8 respectively. It is notice that pressure gradient duration increases with increasing the expansion dimensionless length L_{ci}. An adverse pressure gradient take place over the entry expansion dimensionless length L_{ci} and it is probably the reason for increasing vibration amplitude with the expansion dimensionless length L_{ci}. changing the entry length L_i doesn't have any similar effect on the pressure gradient.

Figure (5), (9) and (13) show Vibration velocity versus frequency at different entry length and expansion length having a diameter ratio d_i/d_o= 0.4, 0.6 and 0.8 respectively. It's noties that these cases having vibration velocity within the acceptable range at all frequencies value. And all cases have vibration velocity lower that the straight pipe value.

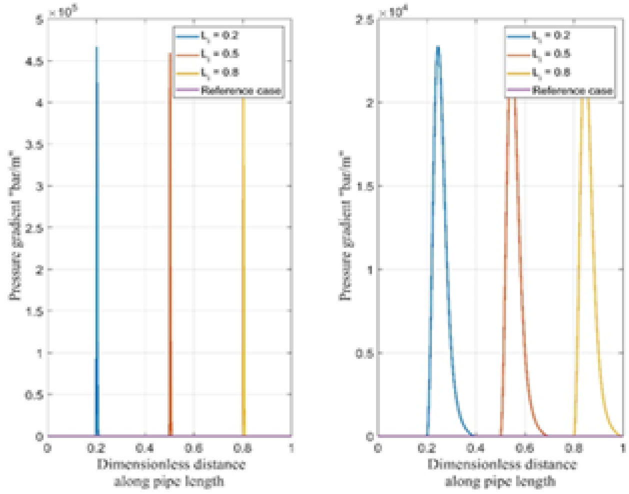
Figure (6), (10) and (14) show the effect of changing expansion length L_{ci} on vibration amplitude Y*at diameter ratio d_i/d_o=.4, .6 and .8 respectively. it is notices that at expansion length L_{ci}=0 it expresses about straight pipe. So, the expansion length L_{ci} decreases the vibration amplitude Y*

Generally, changing the entry L_i length and expansion length L_{ci} have a significant effect on dimensionless vibration amplitude Y*. Increasing the diameter ratio come upon increasing in vibration amplitude and vibration velocity. Increasing frequency has no effect on dimensionless vibration amplitude Y* or vibration velocity. also, expansion cases have lower vibration amplitude and vibration velocity than the straight pipe.



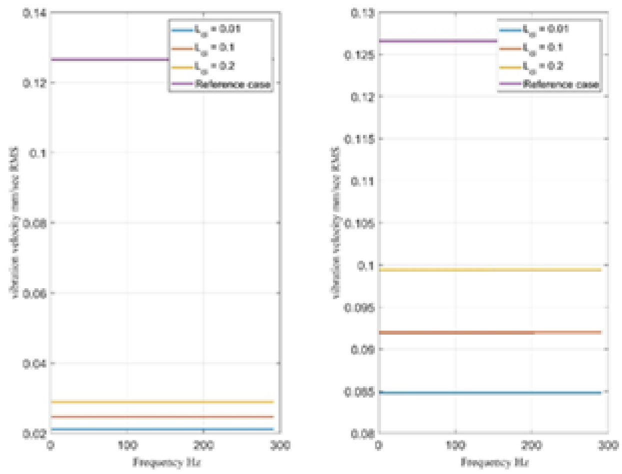
L_i=.2 L_i=.8

Figure 3: Vibration Amplitude Y* versus dimensionless frequency Ω at different entry and expansion length having a diameter ratio di/do=.4



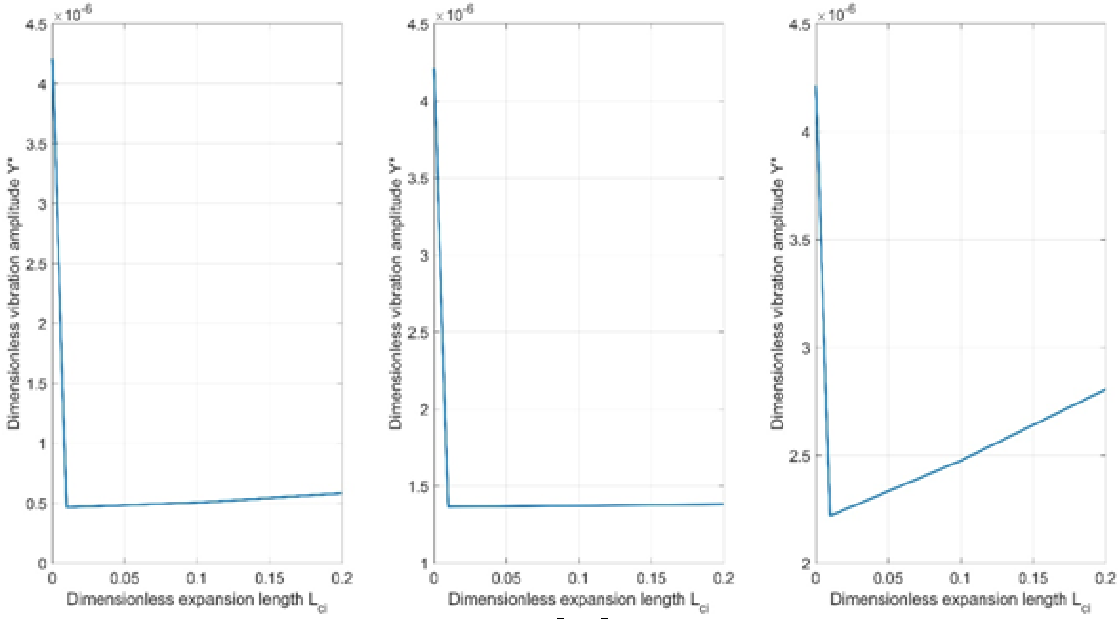
L_{ci}=0.01 L_{ci}=0.2

Figure 4: Pressure gradient variation along pipe length at different entry and expansion lengths, having a diameter ratio di/do=.4



L_i=.2 L_i=.8

Figure 5: Vibration velocity mm\sec RMS versus frequency Hz at different entry and expansion length having a diameter ratio di/do=.4



L_i=.2 L_i=.5 L_i=.8

Figure 6: Effect of dimensionless expansion length L_{ci} on vibration Amplitude Y* for different expansion position having a diameter ratio di/do=.4

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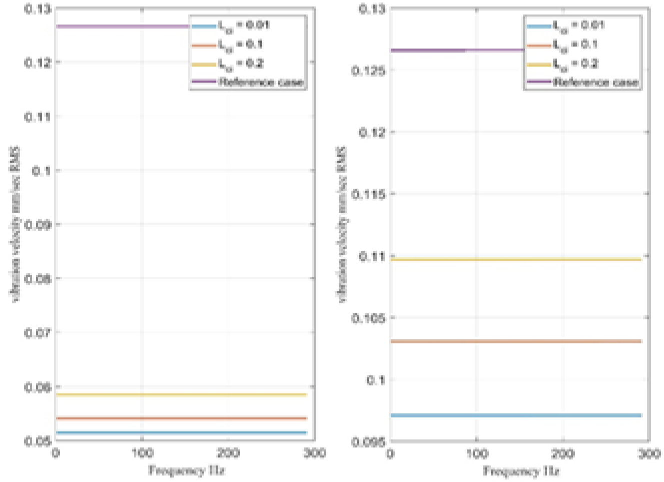
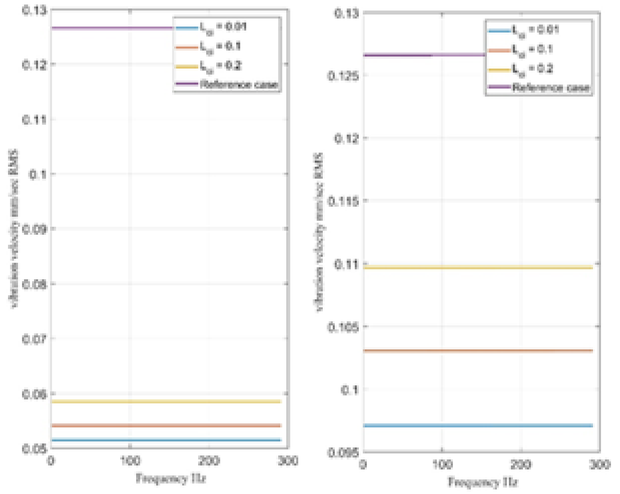


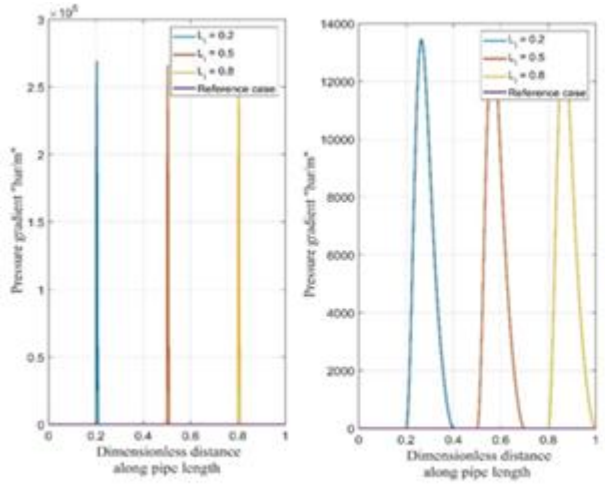
L_i=.2 L_i=.8

Figure 7: Vibration amplitude Y* verses dimensionless Ω at different entry and expansion length having a diameter ratio di/do=.6



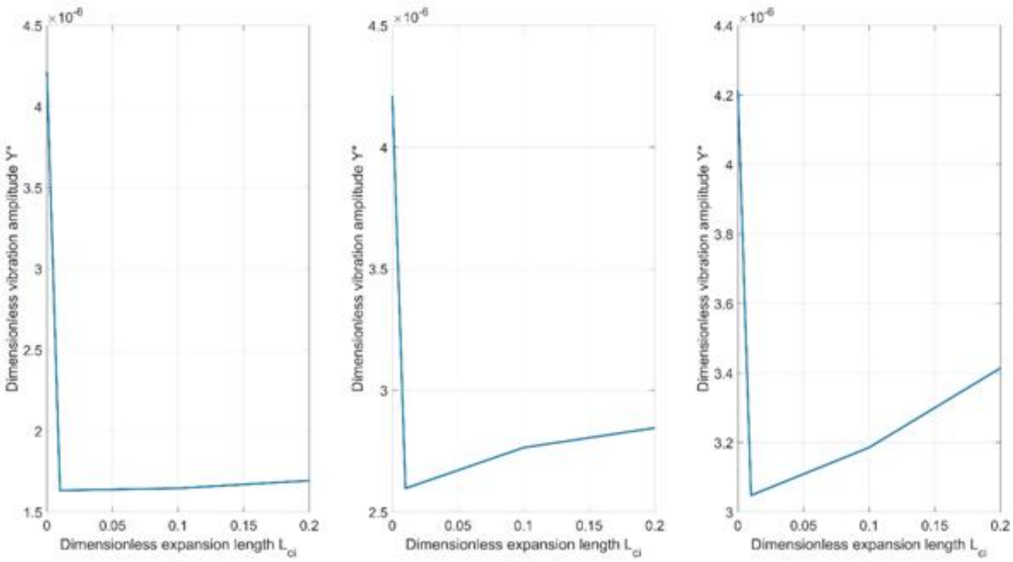
L_i=.2 L_i=.8

Figure 9: Vibration velocity mm\sec RMS versus frequency Hz at different entry and expansion length having a diameter ratio di/do=.6



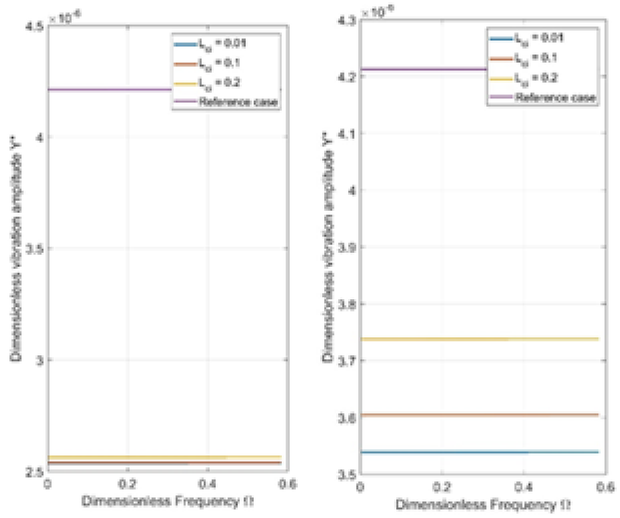
L_i=.01 L_i=.2

Figure 8: Pressure gradient variation along pipelength at different entry and expansion lengths having a diameter ratio di/do=.6



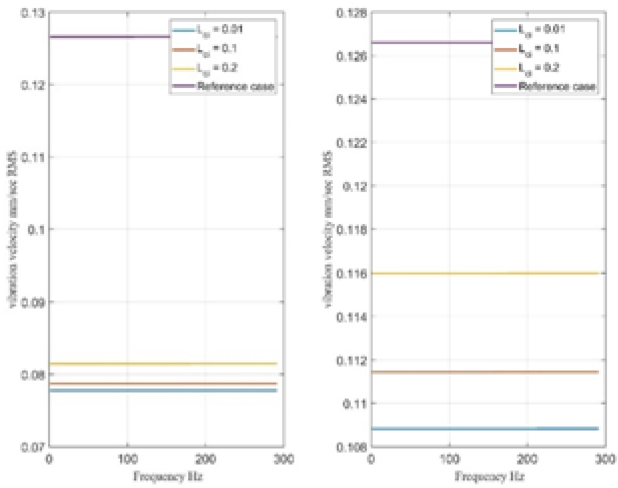
L_i=.2 L_i=.5 L_i=.8

Figure 10: Effect of dimensionless expansion length L_{ci}on vibration Amplitude Y* for different expansion position having a diameter ratio di/do=.6



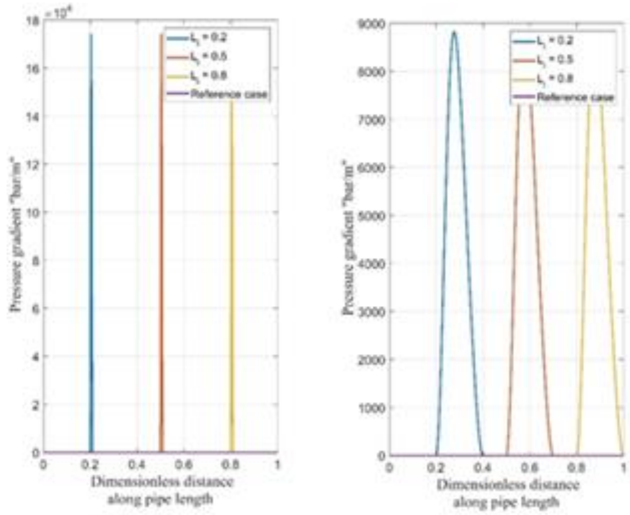
Li=.2 **Li=.8**

Figure 11: Vibration amplitude Y* versus dimensionless frequency Ω at different entry and expansion length having a diameter ratio di/do=.8



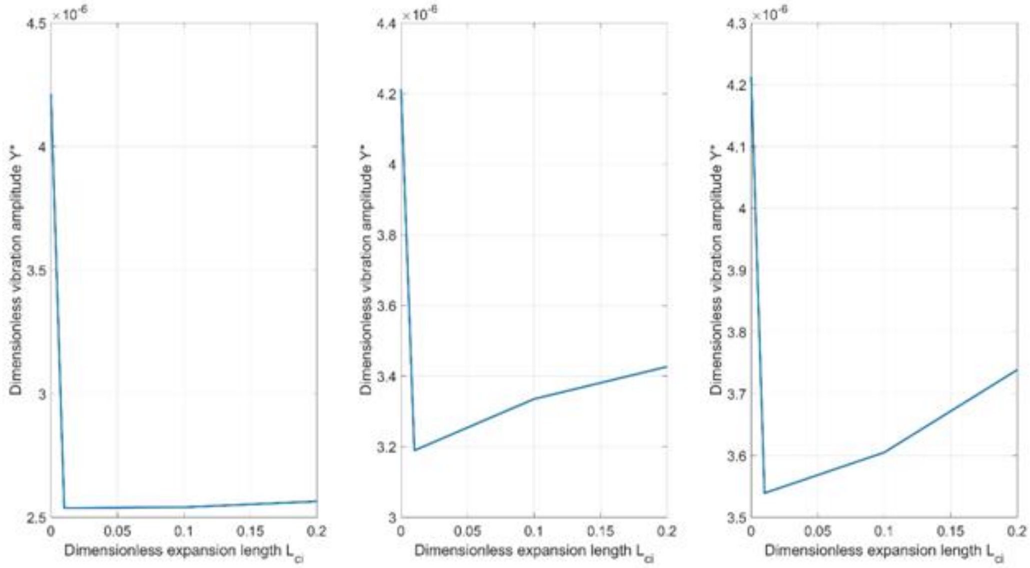
Li=.2 **Li=.8**

Figure 13: Vibration velocity mm\sec RMS versus frequency Hz at different entry and expansion length having a diameter ratio di/do=.8



Lci=0.01 **Lci=0.2**

Figure 12: Pressure gradient variation along pipe length at different entry and Contraction lengths having a diameter ratio di/do=.8



Li=.2 **Li=.5** **Li=.8**

Figure 14: Effect of dimensionless expansion length Lci on vibration Amplitude Y* for different expansion position having a diameter ratio di/do=.8

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3.2. Pipe with contraction cross section area

In this case contraction pipe have a constant inlet diameter of pipe d_i equal 400mm and the outlet diameter of pipe d_o is to be varried to get three pipe diameter ratios d_i/d_o=1.25, 1.5 and 2.5, also the contraction region having three positions L_i 0.2, 0.5 and 0.8 of the pipe length and dimensionless contraction length L_{ci} are 0.01, 0.1 and 0.2, also, in all contraction pipe cases study, frequency changing up to300 Hz too to avoid of vibration Induced Fatigue Failure. All contraction cases compared with straght pipe as a reference.

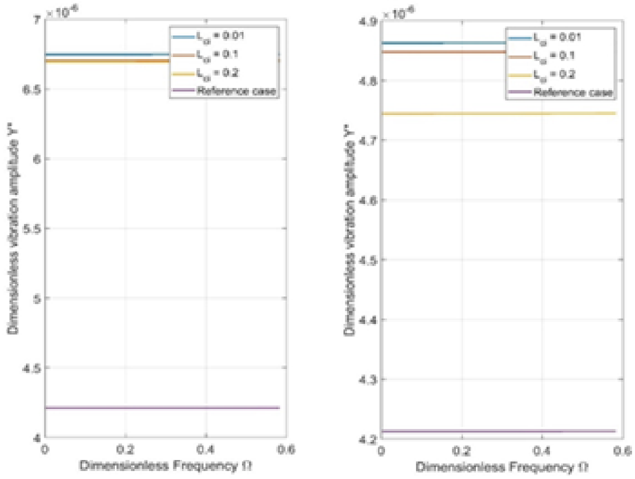
Figures (15), (19) and (23) shows vibration amplitude Y* versus dimensionless frequency **Ω** at different entry and contraction length having diameter ratio d_i/d_o=1.25, 1.5 and 2.5 respectively.

It is noticed that the results are consistent and increasing dimensionless frequency **Ω** has no effect on vibration amplitude Y*. vibration amplitude Y* increases with decreasing contraction length L_{ci} for all values of contraction entry length L_i. Moreover, all cases have higher vibration amplitude than straight pipe. While the entry length L_i and the contraction dimensionless length L_{ci} decreasing the vibration amplitude value get close to the straight pipe value.

Figure (16), (20) and (24) show pressure gradient variation along pipe length having diameter ratio d_i/d_o=1.25, 1.5 and 2.5 respectively. It is notice that pressure gradient duration decreases with increasing the contraction dimensionless length L_{ci}. An adverse pressure gradient take place over the entry contraction dimensionless length L_{ci} and it is probably the reason for decreasing vibration amplitude with the contraction dimensionless length L_{ci}. Changing the entry length L_i doesn't have any similar effect on the pressure gradient.

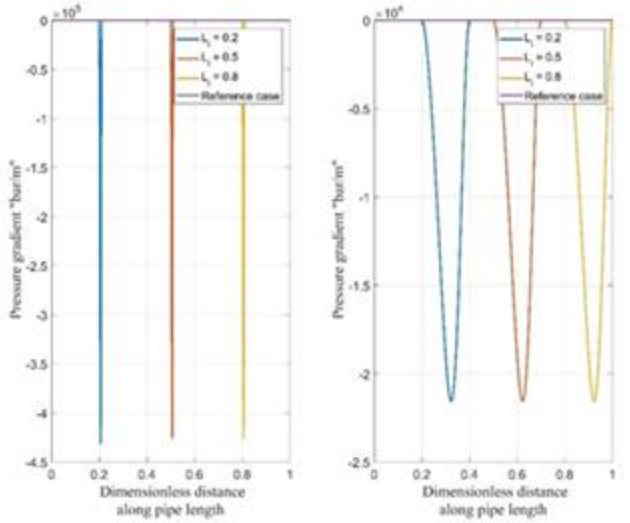
Figure (17), (21) and (25) show Vibration velocity versus frequency at different entry length and contraction length having a diameter ratio d_i/d_o=1.25, 1.5 and 2.5 respectively. It's noties that these cases having vibration velocity withinthe acceptable range at all frequencies value. And all cases have vibration velocity lower that the straight pipe value.

Figure (18), (22) and (26) show the effect of changing contraction length L_{ci} on vibration amplitude Y*at diameter ratio d_i/d_o=.4, .6 and .8 =1.25, 1.5 and 2.5 respectively. it is notices that at contraction length L_{ci}=0 it expresses about straight pipe. So, the contraction length L_{ci} increases the vibration amplitude Y*.



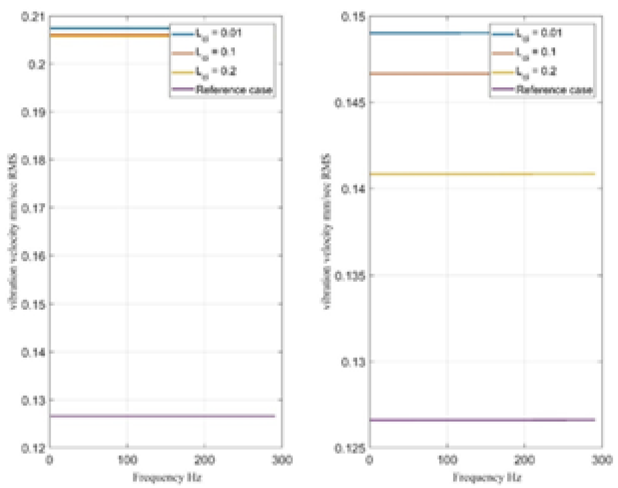
L_{ci}=0.2 L_{ci}=0.8

Figure 15: Vibration amplitude Y* verses dimensionless frequency Ω at different entry and contraction length having a diameter ratio di/do=1.25



L_{ci}=0.01 L_{ci}=0.2

Figure 16: Pressure gradient variation along pipe length at different entry and Contraction lengths having a diameter ratio di/do=1.25



L_i=.2 L_i=.8

Figure 17**:**Vibration velocity mm\sec RMS versus frequency Hz at different entry and contraction length having a diameter ratio di/do=1.25

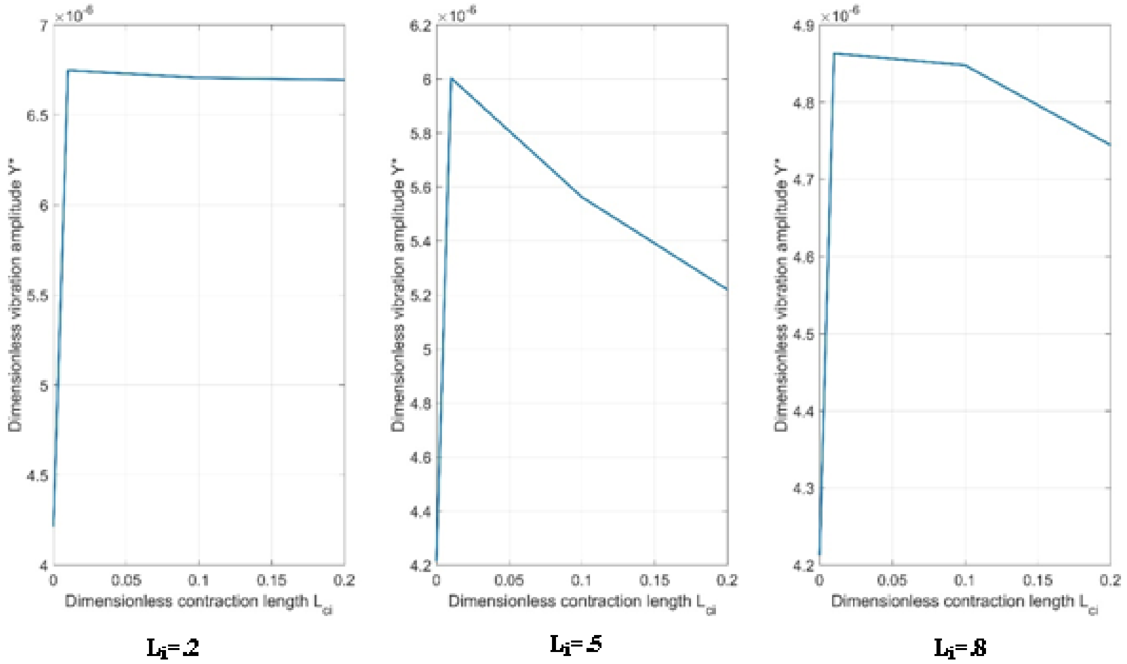


Figure 18: Effect of dimensionless contraction length Lci on vibration Amplitude Y* for different contraction position having a diameter ratio d_i/d_o =1.25

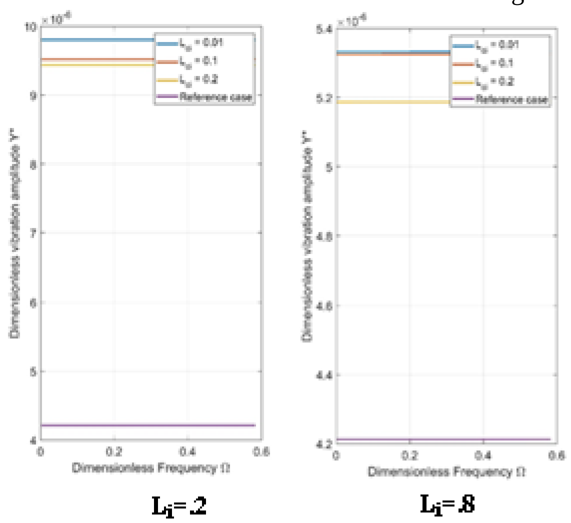


Figure 19: Vibration Amplitude Y* verses dimensionless frequency Ω at different entry and contraction length having a diameter ratio d_i/d_o =1.5

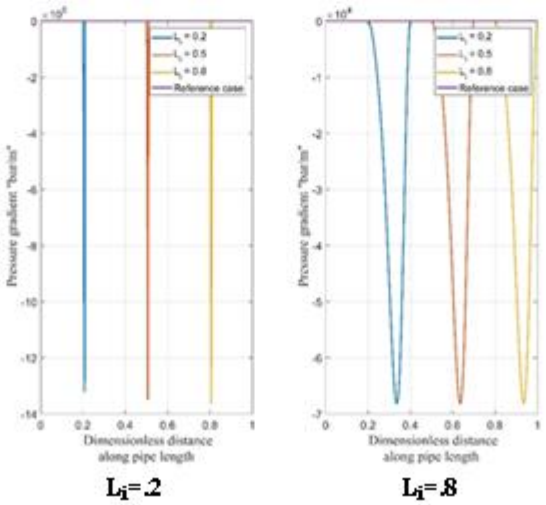


Figure 20: Pressure gradient variation along pipe length at different entry and Contraction lengths, having a diameter ratio di/do =1.5

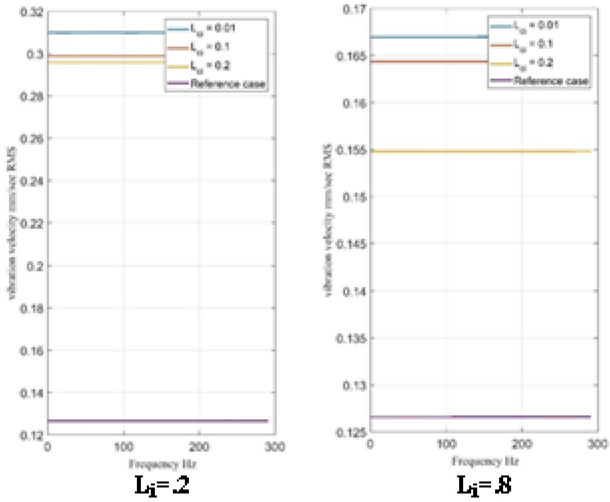
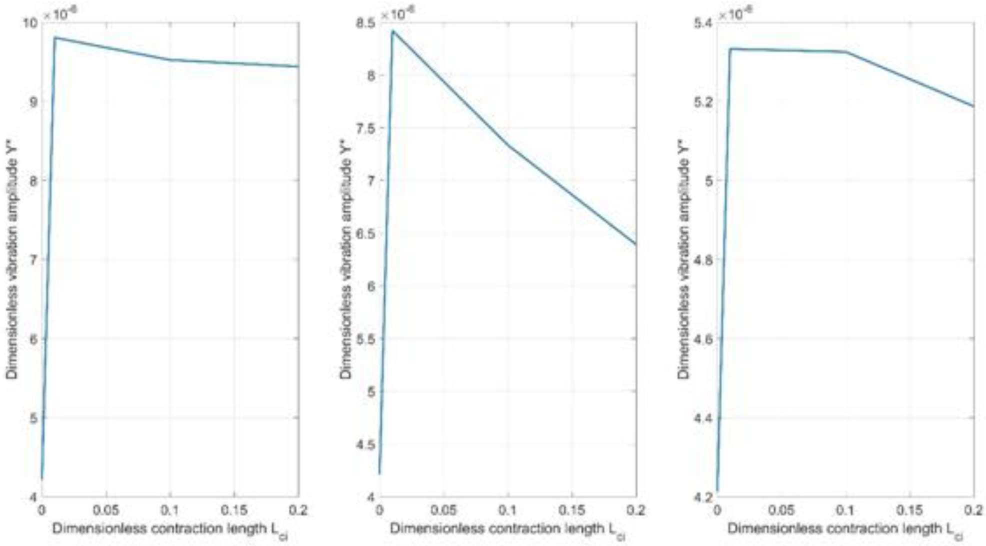
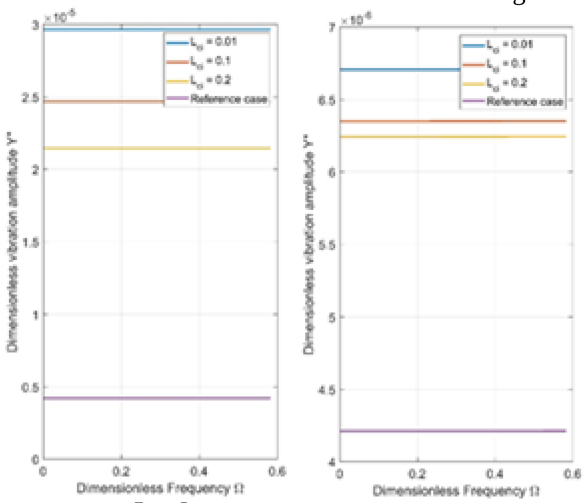


Figure 21: Vibration velocity mm\sec RMS versus frequency Hz at different entry and contraction length having a diameter ratio d_i/d_o =1.5



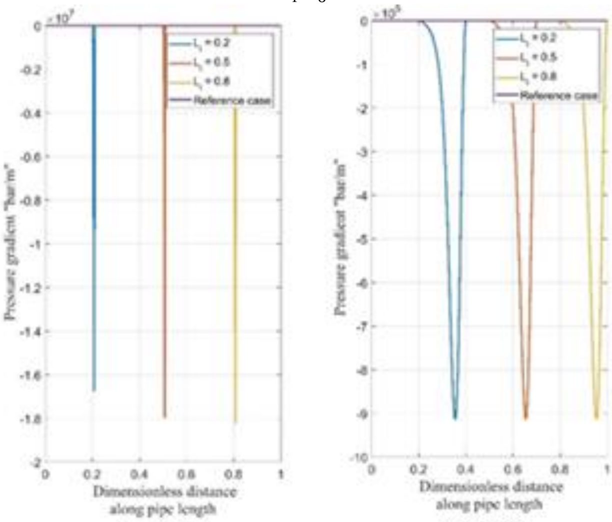
L_i=.2 L_i=.5 L_i=.8

Figure 22: Effect of dimensionless contraction length Lci on vibration Amplitude Y* for different contraction position having a diameter ratio d_i/d_o =1.5



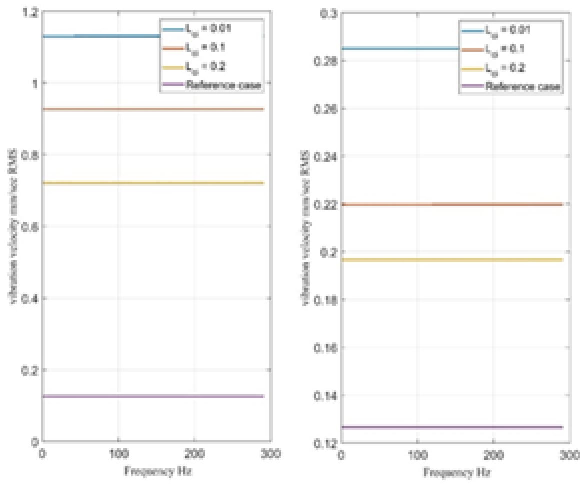
L_i=.2 L_i=.8

Figure 23: Vibration amplitude Y* versus dimensionless Ω at different entry and contraction lengths having a diameter ratio d_i/d_o =2.5



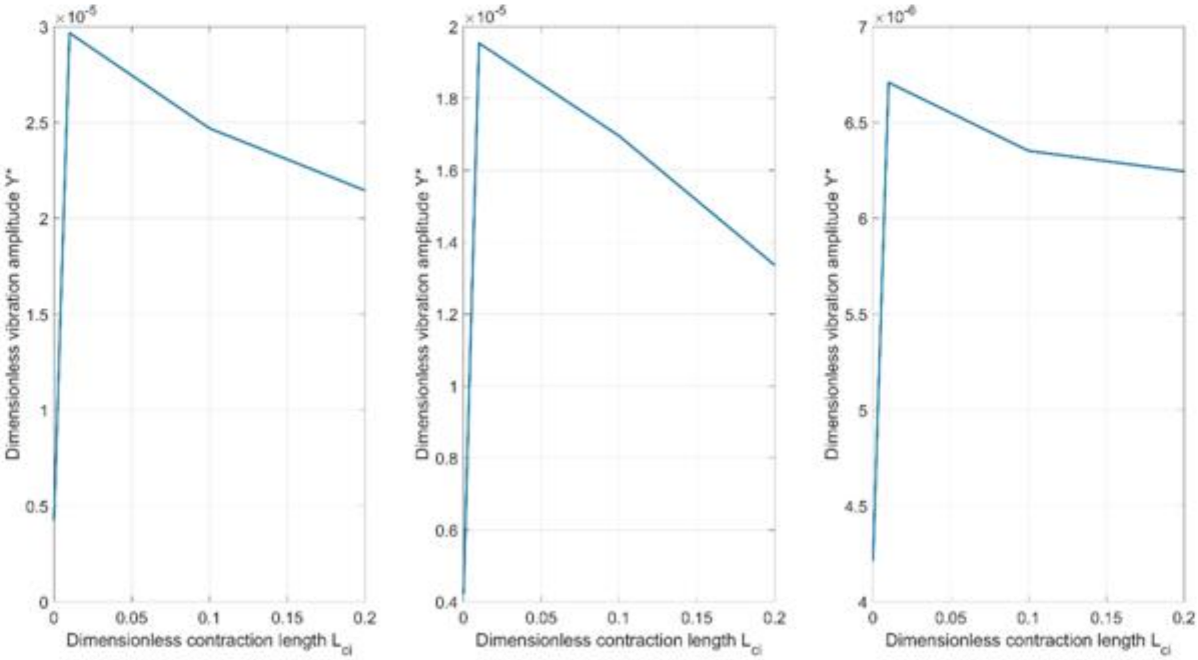
L_{ci}=0.01 L_{ci}=0.2

Figure 24: Pressure gradient variation along pipe length at different entry and contraction lengths having a diameter ratio d_i/d_o =2.5



L_i=.2 L_i=.8

Figure 25: Vibration velocity mm\sec RMS versus frequency Hz at different entry and contraction length having a diameter ratio d_i/d_o =2.5



L_i=.2 L_i=.5 L_i=.8

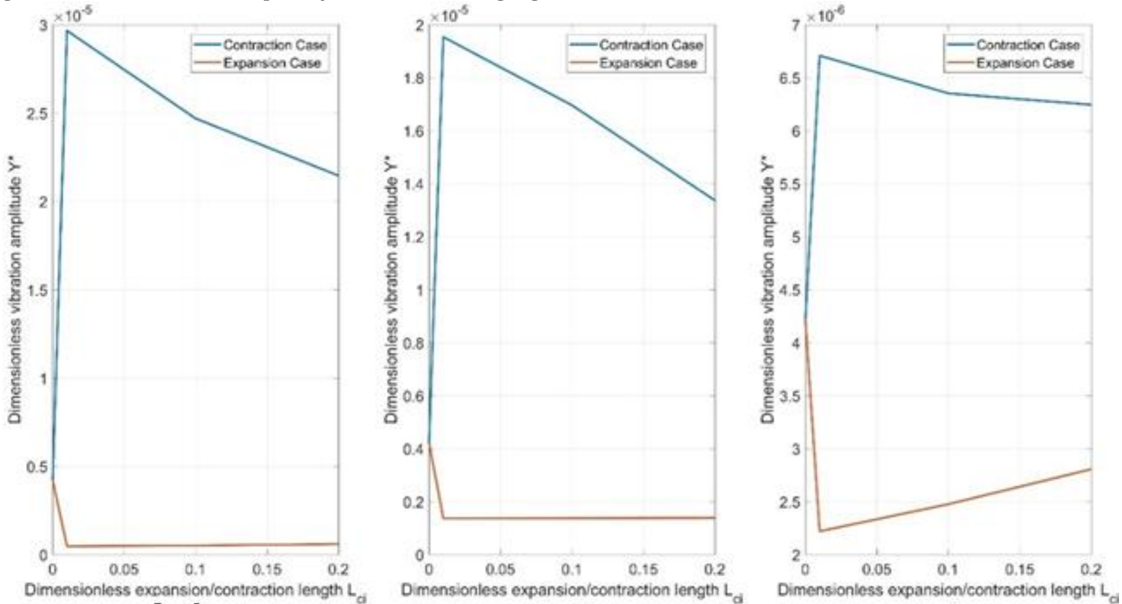
Figure 26: Effect of dimensionless contraction length Lci on vibration Amplitude Y* for different contraction position having a diameter ratio d_i/d_o =1.5

Generally, changing the entry L_i length and Contraction length L_{ci} have a significant effect on dimensionless vibration amplitude Y*. Also, increasing the diameter ratio come upon increasing in vibration amplitude and vibration velocity. Increasing frequency has no effect on dimensionless vibration amplitude Y* or vibration velocity. Moreover, contraction cases have higher vibration amplitude and vibration velocity than the straight pipe.

3.3. Comparison between expansion and contraction pipes

For clearing the results. Expansion and contraction results showed together at the same frequency because changing frequency has no effect on vibration amplitude Y*. Also, the comparing has done at the same diameter ratio for expansion and contraction cases. and it's noticed that the dimensionless expansion or contraction length L_{ci}=0 is expresses the straight pipe.

Figure (27), shows the effect of dimensionless expansion or contraction length L_{ci} on vibration amplitude Y* for different position having a diameter ratio d_i/d_o=.4 and 2.5. Also, Figure (28), shows the effect of dimensionless expansion or contraction length L_{ci} on vibration amplitude Y* for different



L_i=.2 L_i=.5 L_i=.8

Figure 27: Effect of dimensionless expansion or contraction length Lci on vibration amplitude Y* for different position having a diameter ratio d_i/d_o =.4 and 2.5

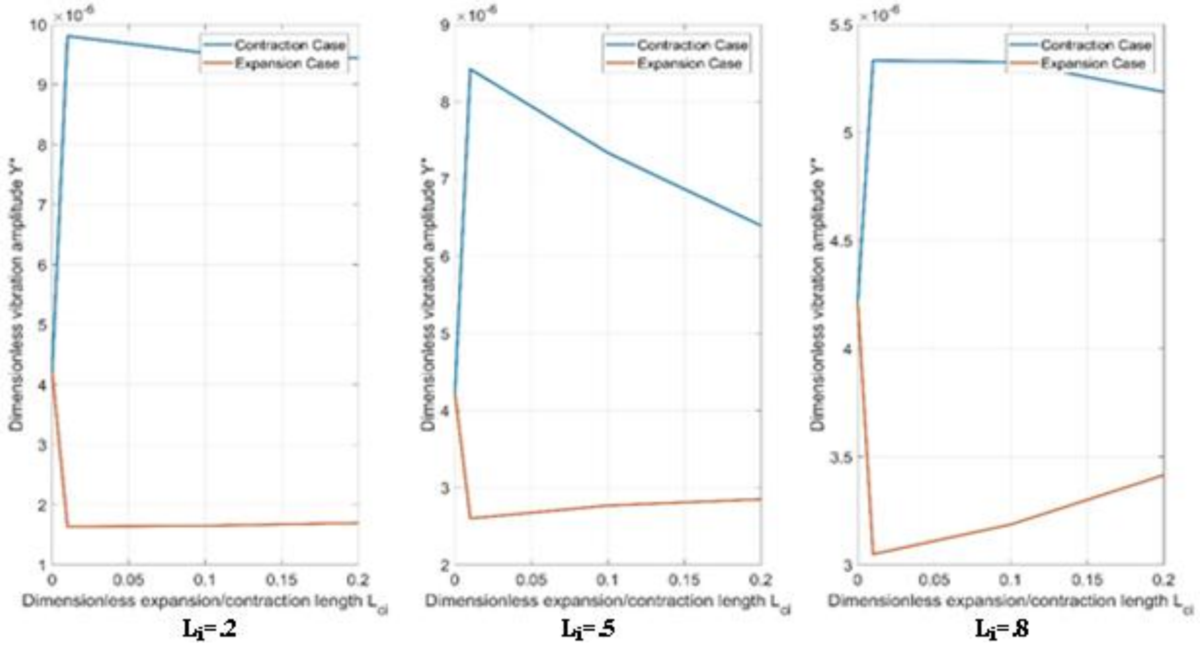


Figure 28: Effect of dimensionless expansion or contraction length L_{ci}on vibration amplitude Y* for different position having a diameter ratio d_i/d_o =.6 and 1.5

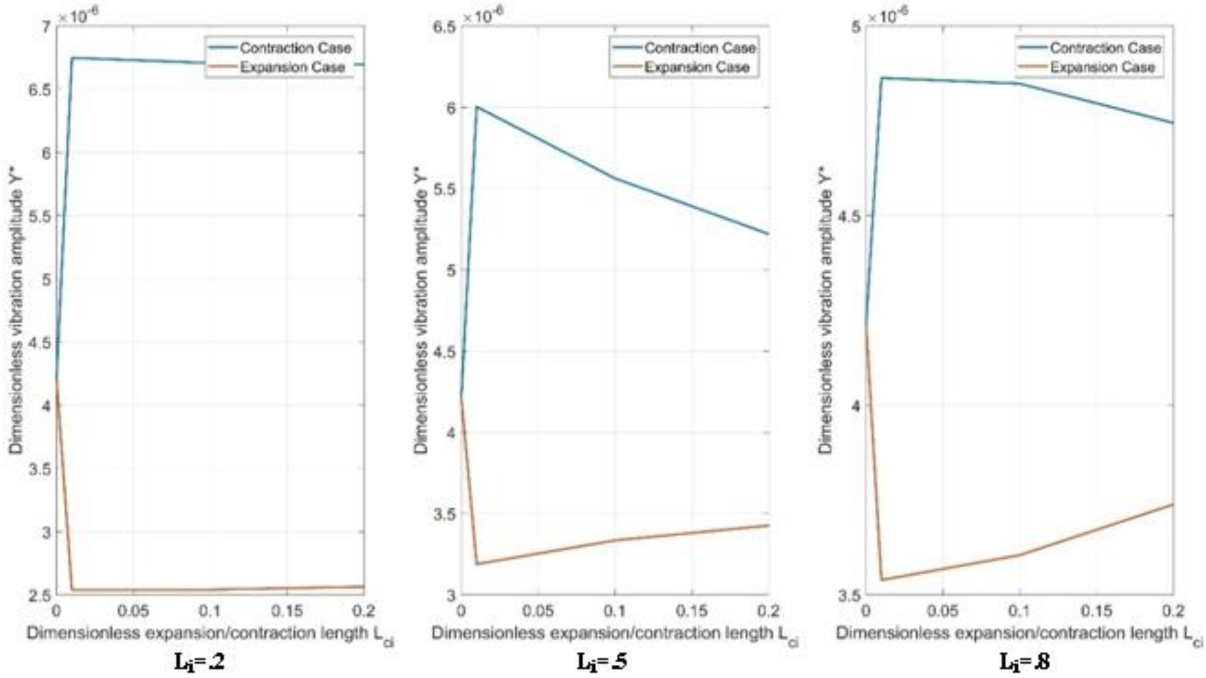


Figure 29: Effect of dimensionless expansion or contraction length L_{ci}on vibration amplitude Y* for different position having a diameter ratio d_i/d_o =.8 and 1.25

position having a diameter ratio d_i/d_o=.6 and 1.5. Moreover, Figure (29), shows the effect of dimensionless expansion or contraction length L_{ci} on vibration amplitude Y* for different position having a diameter ratio d_i/d_o=.8 and 1.25.

There is common result between these figures like the contraction causes an increase in vibration amplitude Y* that's opposite the expansion which causes a decrease in vibration amplitude Y*. Also. Increasing the dimensionless entry length L_i Causes decrease in vibration amplitude Y*. on the contrary, Increasing the dimensionless entry length L_i Causes an increase in vibration amplitude Y*. Moreover, increasing diameter ratio at contraction cases causes a decrease in vibration amplitude Y* on the contrary the expansion cases which increases vibration amplitude Y*.

Generally, changing the entry L_i length and Contraction length L_{ci} have a significant effect on dimensionless vibration amplitude Y*. Also, increasing the diameter ratio come upon

increasing in vibration amplitude and vibration velocity. Increasing frequency has no effect on dimensionless vibration amplitude Y* or vibration velocity. Moreover, contraction cases have higher vibration amplitude and vibration velocity than the straight pipe.

4. Conclusions

Increasing the frequency has no effect on the vibration amplitude in all cases which studied.

While studying expansion cases, the expansion decreasing the vibration amplitude Y*. Also, the vibration amplitude Y* decreased by increasing diameter ratio, increasing the dimensionless entry length L_i or decreasing the dimensionless expansion length Lci. These factors have a simile effect on Vibration velocity. Pressure gradient too decreases by increasing the dimensionless expansion length Lci or increasing the diameter ratio or both. So, it's clear that, the expansion should be sudden and placed at the beginning of the pipe. Moreover.

While studying contraction cases, the contraction increases the vibration amplitude Y*. Also, the vibration amplitude Y* increases by decreasing diameter ratio, decreasing the dimensionless entry length L_i or increasing the dimensionless Contraction length Lci. These factors have a simile effect on Vibration velocity. Pressure gradient too increased by decreasing the dimensionless expansion length Lci or decreasing the diameter ratio or both. In other words, the contraction should be gradually placed at the end of the pipe to get the lowest vibration amplitude.

According to the previous comparison between expansion and contraction cases in straight pipes. Expansion cases are more efficient because they have lower vibration amplitude and vibration velocity. On the other side, contraction cases have higher vibration amplitude and vibration velocity.

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