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Frequency domain analysis by using the bode diagram method of pipes conveying fluid

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Abstract
Fluid conveying pipes are used in all hydraulic systems and enter in all industrial fields, such as water, petroleum products and gases of all kinds. In spite of this widespread and critical importance of fluid conveying pipes, they suffer from major problems and one of these problems is the problem of vibrations that cause the collapse of the systems completely and cause significant economic losses if not avoided. For this reason, the researchers have dealt with this issue in all years, but the problem is not over. In this paper will highlight the problem of controlling the vibrations resulting from fluid flow inside the pipes. The research included the study of the dynamic behaviour of different types of stabilization of the pipes in the presence with no hydraulic damping (active control) and monitoring the stability of each case of stabilization. This study was carried out by deriving differential equations for pipes and for different types of fixation. Special equations with mode response and natural frequency equations were obtained in addition to the stability study of the pipes in the same manner, by using bode diagram technique. This is done by deriving the differential equation of motion and dealing with it for the purpose of reaching the general equation so that it can be easily converted within the method of state space and analysis of equations and find feedback with the change of inputs and outputs. With the help of MATLAB program where the final equations are programmed and the results are found. The main factors of dynamic and stability behaviour were calculated in addition to the main parameters on the pipe such as fluid velocity, pressure and mass ratios.

Keywords: Frequency domain analysis; Pipe conveying fluid; Phase and amplitude; Frequency response; Bode diagram.

1. Introduction
In all industries pipes are used to transport fluids of all kinds despite this proliferation and extreme importance of the pipes these pipes are used in the different fields of the system and the presence of vibrations, which leads to damage to the systems or damage the pipes in addition to the resulting noise therefore, it is necessary to find control systems to reduce the vibrations generated in the pipes. systems pipes of conveying fluid widely used in aircraft power plants, ships, nuclear industry, oil, and energy industry, metallurgy industry, power industry, biological engineering, marine engineering, and in everyday life. The main purpose of the pipes is to transmit energy or energy flow, mass liquid flow. Every year huge economic losses occur in advanced countries is caused due to the vibration of pipes. The annual damage

due to vibrations in pipelines in developed countries is estimated at 10 billion dollars, according to estimates by Canadian experts. Therefore, research on the study of vibrations in fluid pipe conveying fluid systems has engineering and economic value [1].

In general, the behavior of structure with effect of fluid, thermal, heat generation, or wear behavior are more complex than the structure without other effect, [2-5]. Therefore, the behavior of dynamic for fluid conveying pipes are more complex than the structure without fluid. When the tube is without fluid, the stiffness and mass are determined only by the free vibrations of the structure (boundary conditions and degree of freedom). In this structures the Eigen values are attached to the parameters of the structure, subsequently, the natural frequency is singular. The control type is then uncontrolled for vibrations generated. If these structures are exposed to an axial force, Eigen values are affected by the direction and amount of this force, that is, if the axial force is a compressive force, the increase of this force leads to a decrease in natural frequencies. Where the vibration is controlled in this case by this force and it has critical value at that essential frequency recedes drops to zero causing the state of buckling [6]. The researchers presented several studies in the field of controlling the vibrations of pipes conveying fluid. The following studies are similar to the current research, which shows the difference between the current research and published research on the subject of vibration of pipes conveying fluid and vibrations of flexible beam structure.

At 2015, M. J. Jweeg et. al. [7, 8], active control of the resonance parametric, is one of the main problems of active control. To overcome the problems of this control and then design the active control elements and active control units under the resonance barometric designed. The linear part as a controller on the theory of linear quadratic and then examined the work of the numerical control unit. And determined stability of the pipe originally by the fluid velocity, [7]. In addition to investigation the dynamic behavior for pipe conveying fluid by using analytical and experimental technique, where the experimental technique included using of smart materials to calculate the dynamic behavior of pipe, [8]. Also at 2016 T. Zhang et al. [9] in this research dealt with, nonlinear equations of 3-d motion are constructed for straight pipes conveying fluid with general boundary conditions. Primarily, put (10) springs at both ends of the pipes are used to represent the boundary conditions general and expressed about displacements such as the superposition of a cosine Fourier series and four additional functions to satisfy those the boundary conditions. Also at 2017 X. Dekui and N. Songlin [10], the researcher interested in studying the hydraulic transport pipes and found a mechanism for vibration analysis and a mechanism to control these vibrations and proposed a principle to improve the structural system. Proposed an optimum structural principle of MR damper. Designed a type of MR pipeline vibration, and discussed the strategy of the pipe clamp and the algorithm of semi-active control, found during analysis and discussion, the MR damper which works on principle intelligent material it can be modified continuously, with real-time and good controllability. The algorithm of predictive intelligent feedback control, it is a mechanism for semi-effective control in vibrations.

Also at 2017 W. Wang and et al [11] the researchers took a study the vibrations which occur in the structures of submarine pipelines which submerged under the water that lead to dwindling of the existence of it, where (ECTMD) is used , it’s a brief for mass damper tuned eddy current, utilize the force of damping which is created by the relative metal conductive movement non-magnetic (copper, aluminum) Through a magnetism field to control the pipelines when free and forced vibration happen, and to prove the performance of damping in the submerged environment in water for the air environment, and it is a numerical optimization method taken . Then conducted experiments to show the ECTMD effectiveness in the water environment. Also at 2017 M. B. Hunain [12] studied the natural frequency of the pipe simply supported and the velocity of critical for the fluid. And studied the ratio of and thickness of the pipe and the effect of diameters ratio and dynamic behavior of the pipe. This behavior was carried out through the technique of finite elements and developed the Matlab program so that it could predict the vibrations in order to benefit from this program in theoretical work. In this study, found that when the flow rate of the fluid increased, the natural frequency of the pipe would be reduced.

In addition, at 2017, M. Al-Waily, [13], presenting the effect of velocity and crack angle defect on the vibration behavior of pipe conveying fluid. Where, the investigation included calculate the natural frequency and velocity behavior of pipe with different crack angle effect. Where, the experimental technique was used to calculate the natural frequency of pipe, in addition to, using the numerical technique (using finite element method) to calculate the flow behavior and the natural frequency of pipe.

Then, in this study investigation the effect of flow parameters and the damper coefficient on the stability behavior of pipe, with different boundary condition, by using bode diagram technique. Where, the investigation include drive the general equation of motion for pipe conveying fluid, then, solution for its equation by using analytical solution, and finally, using bode diagram for its solution equation to calculate the stability of pipe with various parameters effect.

2. Theoretical analysis

The theoretical solution is technique using to evaluate the parametric study by low error of results, [14-16], where the theoretical solution can be using analytical or numerical techniques, [17-21]. There, the analytical solution was included exact solution with various parameters effect, [22-24]. Then, in this study using analytical technique to evaluate stability of pipe with various parameters study, where, the analytical solution included drive the general equation of motion for pipe conveying fluid, and then, solution its equation by using analytical technique. Then, after drive and solve the general equation of motion for pipe using bode diagram to calculate the stability behavior of pipe with various flow parameters effect.

2.1 General equation of motion for pipe conveying fluid

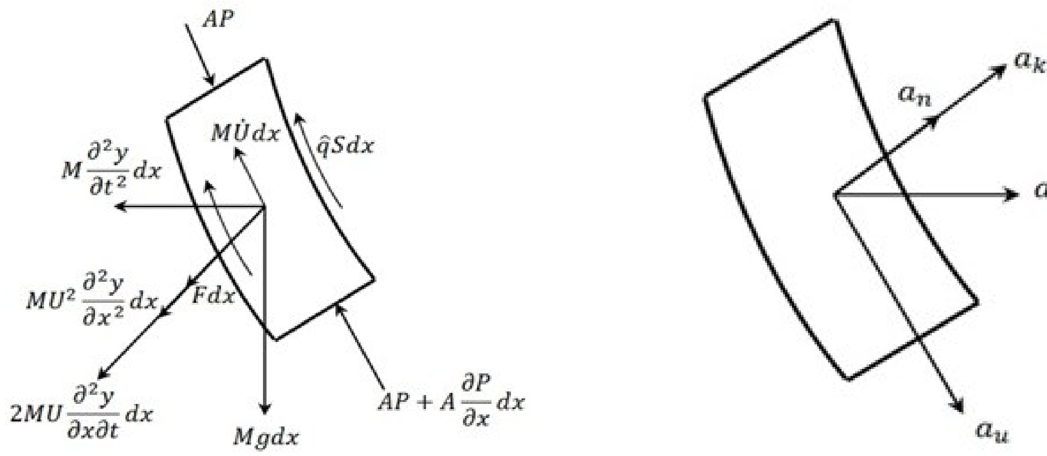
The pipes are placed vertically decreasing the effects of gravitational acceleration on the horizontal movements of the pipes conveying fluid, on this basis, equations were derived, and the research assumes,

1. The Pipe outside damping motion is neglected.
2. The beam model is a pipe. The axis shear stress calculation is neglected.
3. The viscoelastic material pipes, and agreement with Kelvin-Voigt presumption that strain - stress satisfies, as,

$$\sigma = \left(1 + a\frac{\partial}{\partial a}\right)E\varepsilon \tag{1}$$

where, E: elasticity modulus of material, a : viscoelastic coefficient (minimal value).

For analysis of fluid unit stress, dx is fluid unit length. With the inertia force and attention of various external force applied on the fluid unit, where stress diagram and acceleration diagram are showed as Figure 1 a, b.



(a) Stress diagram (b) Acceleration diagram

Figure 1. Fluid unit.

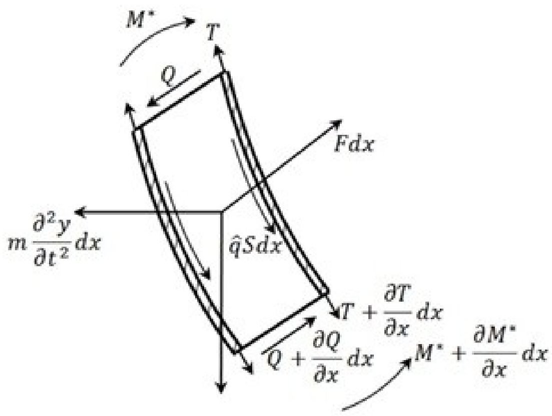
Where, the symbols in the diagrams refer to, U- velocity of fluid , A- flow cross section area, M- fluid mass per unit length, q- a tangential force for the pipe shell on the fluid , S- flow through cross section perimeter, P- pressure of fluid per unit area, F- normal force per unit length pipe shell on fluid. The coinciding relationship between inertia force and acceleration is which a_u indicates to the acceleration of fluid unit with direction pipes, and coincides with $M\dot{U}dx$; a_y indicates to acceleration with direction y and coincides with $M\frac{\partial^2 y}{\partial t^2}dx$; a_n indicates to acceleration centripetal caused by changes the flow speed direction, and coincides with $MU^2\frac{\partial^2 y}{\partial x^2}dx$; a_k indicates to acceleration of Coriolis, and coincides with $2MU\frac{\partial^2 y}{\partial x \partial t}dx$. Assume the angle of deformation is θ (minimal value), and disregard the actions from trace high order, the equation of balancing is calculated the following depend on acceleration diagram and stress diagram of the fluid unit based on suitable equation $\sin\theta = \theta = \partial y/\partial x, \cos\theta = 1$,

$$F\frac{\partial y}{\partial x} - M\dot{U} - A\frac{\partial P}{\partial x} - \hat{q}S + Mg = 0 \tag{2}$$

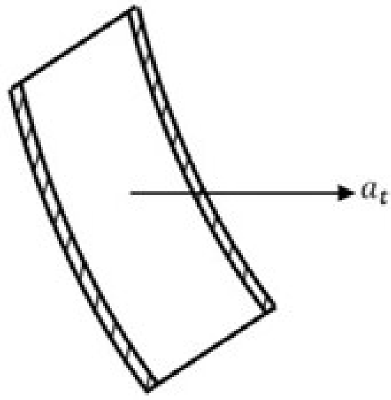
The equation balancing stress at y-direction is:

$$M\left(\frac{\partial}{\partial t} + U\frac{\partial}{\partial x}\right)y + qS\frac{\partial y}{\partial x} + MU\frac{\partial y}{\partial x} + AP\frac{\partial^2 y}{\partial x^2} + A\frac{\partial P}{\partial x}\cdot\frac{\partial y}{\partial x} + F = 0 \tag{3}$$

For analysis stress of pipe unit, dx is pipe unit length. With attach of inertia force and various external force applied on the pipe unit, it is a stress diagram and pipe unit acceleration diagram are showed as such Figure 2 a, b.



(a) Stress diagram



(b) Acceleration diagram

Figure 2. Pipe unit

Where, m- pipe mass per unit length; T- axial force on pipe unit (cross section); M- bending moment applied on the cross section of pipe unit; Q- shear force on the cross section of pipe unit. Acceleration a_1 coincides with the force of inertia $m\frac{\partial^2 y}{\partial t^2}dx$ that pointing to acceleration pipe at direction-y .for same manner with fluid unit, the equation of balancing is wrote as follows,
Equation balancing stress at direction-x,

$$\frac{\partial T}{\partial x} - F\frac{\partial y}{\partial x} + \hat{q}S + mg = 0 \tag{4}$$

Equilibrium equation for stress at direction-y,

$$F + \frac{\partial Q}{\partial x} + T\frac{\partial^2 y}{\partial x^2} + \frac{\partial T}{\partial x}\cdot\frac{\partial y}{\partial x} + \hat{q}S\frac{\partial y}{\partial x} - m\frac{\partial^2 y}{\partial x^2} = 0 \tag{5}$$

The equation of force moment equilibrium,

$$Q = -\frac{\partial M}{\partial x} \tag{6}$$

Depend on knowledge mechanics of materials of $M^* = EI\frac{\partial^2 y}{\partial x^2}$, I is moment of inertia for pipe, it's obtained with strain- stress relation Eq. 1,

$$Q = -\left(1 + a\frac{\partial}{\partial t}\right)EI\frac{\partial^3 y}{\partial x^3} \tag{7}$$

By using Eq. 3 and 4, and plugging into it, E. 7, then,

$$\left(1 + a\frac{\partial}{\partial t}\right) EI\frac{\partial^4 y}{\partial x^4} + (M+m)\frac{\partial^2 y}{\partial x^2} + 2MU\frac{\partial^2 y}{\partial x\,\partial t} + (MU^2 - T + AP)\frac{\partial^2 y}{\partial x^2} + A\frac{\partial P}{\partial x}\frac{\partial y}{\partial x} - \frac{\partial T}{\partial x}\frac{\partial y}{\partial x} + M\dot{U}\frac{\partial y}{\partial x} = 0 \quad (8)$$

Then, Eq. 2 plus Eq. 4 will lead,

$$\frac{\partial}{\partial x}(T - AP) = M\dot{U} - (M+m)g \quad (9)$$

Integral for Eq. 9 for interval (x, L), result,

$$(T - AP)_{x=L} = (T - AP) + (M\dot{U} - (M+m)g) \quad (10)$$

If assume $T = \overline{T}$ at the pipe end ,any $x = L$, stress of fluid $PA = \overline{P}A(1 - 2\nu\delta)$, ν is a poisson's ratio, $\overline{P}$ is the pressure exerted on the pipe ends per unit area for the fluid , and regarding $-2\nu\delta\overline{P}A$ is the extra tension caused by the internal pressure of the pipe. When the motion at the lower end of the pipe is bounded, $\delta = 1$, and when the motion at the lower end of the pipe is free $\delta = o$ thus , Eq. 10 can reformulate it as follows,

$$(T - AP) = \overline{T} - A\overline{P}(1 - 2\nu\delta) + (M\dot{U} - (M+m)g)(x - L) \quad (11)$$

And after added extra axial force, Eq. 11 will be rewritten as follows,

$$(T - AP) = \overline{T} - A\overline{P}(1 - 2\nu\delta) + \big(M\dot{U} - (M+m)g\big)(x - L) + \left(1 + a\frac{\partial}{\partial t}\right)\frac{E\tilde{A}}{2L}\int_0^L \acute{y}^2\,dx \quad (12)$$

Plugging Eq. 12 in Eq. 8, the differential equation of motion for pipes is obtained after arranging,

$$\begin{pmatrix} \left(1 + a\frac{\partial}{\partial t}\right) EI\frac{\partial^4 y}{\partial x^4} + \left(MU^2 - \overline{T} + A\overline{P}(1 - 2\nu\delta) - \left((M+m)g - M\dot{U}\right)(L - x)\right)\frac{\partial^2 y}{\partial x^2} \\ -\left(\left(1 + a\frac{\partial}{\partial t}\right)\frac{E\tilde{A}}{2L}\int_0^L (y')^2\,dx\right)\frac{\partial^2 y}{\partial x^2} + (M+m)\frac{\partial^2 y}{\partial t^2} + 2MU\frac{\partial^2 y}{\partial x\,\partial t} + (M+m)g\frac{\partial y}{\partial x} \end{pmatrix} = 0 \quad (13)$$

So as to get the differential equation of general motion, and obviate unit effects, the equation of motion necessary for it the (dimensionless). Through that, to simplify the analysis of issues, the parameter of dimensionless is submitted and changes are taking place to turn partial differential equation into the dimensionless differential equation, [25]. The parameters of dimensionless are,

$$\eta = \frac{y}{L}, \xi = \frac{x}{L}, \overline{g} = \frac{M+m}{EI}L^3 g, \tau = \left(\frac{EI}{M+m}\right)^{\frac{1}{2}}\frac{t}{L^2}, \Gamma = \frac{\overline{T}L^2}{EI},$$
$$\Pi = \frac{\overline{P}AL^2}{EI}, \gamma = \frac{\tilde{A}L^2}{2I}, u = \left(\frac{M}{EI}\right)^{\frac{1}{2}} LU, M_r = \left(\frac{M}{M+m}\right)^{\frac{1}{2}}, \alpha = \left(\frac{EI}{M+m}\right)^{\frac{1}{2}}\frac{\mu}{L^2} \quad (14)$$

If the movements are as follows $\eta = H(\xi)\exp(i\omega\tau)$ are considered, the frequency (dimensionless) Ω is related with frequency (dimensional radian) ω by,

$$\Omega = \omega L^2\sqrt{(m_f + m_p)/EI} \quad (15)$$

Plugging Eq. 14 into Eq. 13, the dimensionless differential equation is established,

$$\eta^{(4)} + \alpha\dot{\eta}^{(4)} + \begin{pmatrix} u^2 - \Gamma + \Pi(1 - 2\nu\delta) + (M_r\dot{u} - \overline{g})(1 - \xi) \\ -\gamma\int_0^1 (\eta')^2 d\xi - 2\alpha\gamma\int_0^1 \eta'\,\dot{\eta}' d\xi \end{pmatrix}\eta'' + \ddot{\eta} + 2M_r u\,\dot{\eta}' + \overline{g}\eta' = 0 \quad (16)$$

In the equation, $(\ )'$ refers $\frac{\partial(\ )}{\partial\xi}$, $(\dot{\ })$ refers $\frac{\partial(\ )}{\partial\tau}$.
Because of the minimal pulsating flow of system in this research, assume,

$$u = u_0(1 + \mu\cos(\omega\tau)) \ , \ u^2 = u_0^2(1 + 2\mu\cos(\omega\tau)) \ , \ \dot{u} = -u_0\omega\mu\sin(\omega\tau) \tag{17}$$

Through that, $\mu << 1$ pointing to micro quantity, u_0 pointing to average velocity. Plugging Eq. 17 into Eq. 16 and put nonlinear part and pulsating flow part on the right part of the equation, then the equation is found dimensionless,

$$\begin{pmatrix} \ddot{\eta} + 2M_r u_0\,\dot{\eta}' + \alpha\dot{\eta}^{(4)} + \eta^{(4)} + (u_0^2 - \Gamma + \Pi(1 - 2v\delta) - \bar{g})\eta'' + \bar{g}\xi\eta'' + \bar{g}\eta' = \\ \mu(M_r u_0\omega(1-\xi)\eta''\sin(\omega\tau) - (2u_0^2\eta'' + 2M_r u_0\,\dot{\eta}')\cos(\omega\tau)) + 2\alpha\gamma\eta''\int_0^1 \eta'\,\dot{\eta}'d\xi + \gamma\eta''\int_0^1 (\eta')^2 d\xi \end{pmatrix} \tag{18}$$

2.2 Solution for general equation of motion

For the purpose of simplification analysis of the equation of differential motion, high order differential equation dimensionless Eq. 18 is decrease and discretized for a reduced order differential equation through Ritz-Galerkin manner. Assuming displacement η is a function for variables τ and ξ, then the expression of Ritz-Galerkin will be,

$$\eta(\xi,\tau) = \sum_{i=1}^{\infty} \emptyset_i\,(\xi)\ \ q_i\,(\tau) \tag{19}$$

$\emptyset_i\,(\xi)$ is the comparison function, $q_i\,(\tau)$ is the generalized coordinate where satisfies whole the boundary conditions. And take the first two orders to proceed studies that is,

$$\eta(\xi,\tau) = \sum_{i=1}^{2} \emptyset_i\,(\xi)\,q_i\,(\tau) = \emptyset_1\,(\xi)\,q_1\,(\tau) + \emptyset_2\,(\xi)\,q_2\,(\tau) \tag{20}$$

For pipes pinned at both ends, the function of its vibration model is,

$$\phi_i = \sqrt{2}\sin(\lambda_i\xi), \qquad i = 1,2 \tag{21}$$

$\lambda_1 = \pi$, $\lambda_2 = 2\pi$, where λ_1, λ_2 and are pipe eigenvalues. For pipes fixed at both ends, the function of its vibration model is,

$$\phi_i = \cosh(\lambda_i\xi) - \cos(\lambda_i\xi) + \frac{\cosh(\lambda_i)-\cos(\lambda_i)}{\sinh(\lambda_i)-\sin(\lambda_i)}\big(\sin(\lambda_i\xi) - \sinh(\lambda_i\xi)\big), i = 1,2 \tag{22}$$

Where, $\lambda_1 = 4.73, \lambda_2 = 7.8532$. For pipes pinned at one end and fixed at another end, the function of its vibration model is,

$$\phi_i = \cos(\lambda_i\xi) - \cosh(\lambda_i\xi) + \frac{\cos(\lambda_i)-\cosh(\lambda_i)}{\sin(\lambda_i)-\sinh(\lambda_i)}\big(\sin(\lambda_i\xi) - \sinh(\lambda_i\xi)\big), i = 1,2 \tag{23}$$

Note that, $\lambda_1 = 3.9267, \lambda_2 = 7.0686$. For pipes (cantilever), the function of its vibration model is,

$$\phi_i = \cosh(\lambda_i\xi) - \cos(\lambda_i\xi) + \frac{\sinh(\lambda_i)-\sin(\lambda_i)}{\cosh(\lambda_i)+\cos(\lambda_i)}\big(\sin(\lambda_i\xi) - \sinh(\lambda_i\xi)\big), i = 1,2 \tag{24}$$

Note that, $\lambda_1 = 1.87512, \lambda_2 = 4.6941$.

After simplifying the equation and finding the constants to change into first mode state differential equation,, get the final equation then,

$$\begin{Bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \end{Bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ a_1 & a_2 & a_3 & a_4 \\ a_2 & b_2 & -a_4 & b_4 \end{bmatrix} \begin{Bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{Bmatrix} + \mu \begin{Bmatrix} 0 \\ 0 \\ H_1 \\ H_2 \end{Bmatrix} - \begin{Bmatrix} 0 \\ 0 \\ Q_1 \\ Q_2 \end{Bmatrix} \tag{25}$$

The simulation response of system motion will depend on this equation.

Where, $a_1 = -\left(u_0^2 - T - \frac{1}{2}\bar{g}\right)c_{11} + \lambda_1^4$, $a_2 = -\bar{g}e$, $a_3 = -\alpha\,\lambda_1^4$, $a_4 = 2M_r u_0 b$, $b_2 = -\left(u_0^2 - T - \frac{1}{2}\bar{g}\right)c_{22} + \lambda_2^4$, $b_4 = -\alpha\,\lambda_2^4$, H: pulsating flow part, Q: nonlinear part.

Using the function of Galerkin,

$$\dot{X} = AX + Bv = F(X, \tau)$$
$$y = Cx + Du \tag{26}$$

where, $X = (x_1, x_2, x_3, x_4)^T$, $B = (0\ ,0\ ,B_3, B_4)^T$, $C = [\varphi_1(\xi), \varphi_2(\xi)\ ,0\ ,0]$, $D = [0]$, $F(X,\tau) = (0\ ,0\ ,-Q_1 + \mu H_1, -Q_2 + \mu H_2)$, $A = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ a_1 & a_2 & a_3 & a_4 \\ a_2 & b_2 & -a_4 & b_4 \end{bmatrix}$, $B_3 = \varphi_1'(\xi_b) - \varphi_1'(\xi_a)$, $B_4 = \varphi_2'(\xi_b) - \varphi_2'(\xi_a)$

In the equation above, the expression of a_i $(i = 1,2,3,4)$, b_i $(i = 2,4)$, Q_i and H_i $(i = 1,2)$.

2.3 Frequency domain analysis by using the bode diagram

The design of control system in the frequency domain it can be done utilization measurements are taken from the components in the control loop, or alternatively utilization purely theoretical approach. Frequency domain analysis measures or calculates the system output of a stable state when variable frequency sinusoidal input, responding to a constant amplitude. The errors of steady-state, in terms of phase and amplitude relate directly to the characteristics of dynamic (the transfer function of the system). The Bode diagram is the version of logarithmic for the diagrams of frequency response, and consists of,

1. A log modus - log frequency plot.
2. A linear phase - log frequency plot.

This method (asymptotes) is used to quickly create frequency response charts by hand. Create charts for high-order systems checked through the simple graphical addition of individual schemes to the elements of separate in the system. On a linear y-axis scale in deciBels the modulus is plotted, where,

$$G(j\omega)|\ dB = 20 \log_{10}|\ G(j\omega)| \tag{27}$$

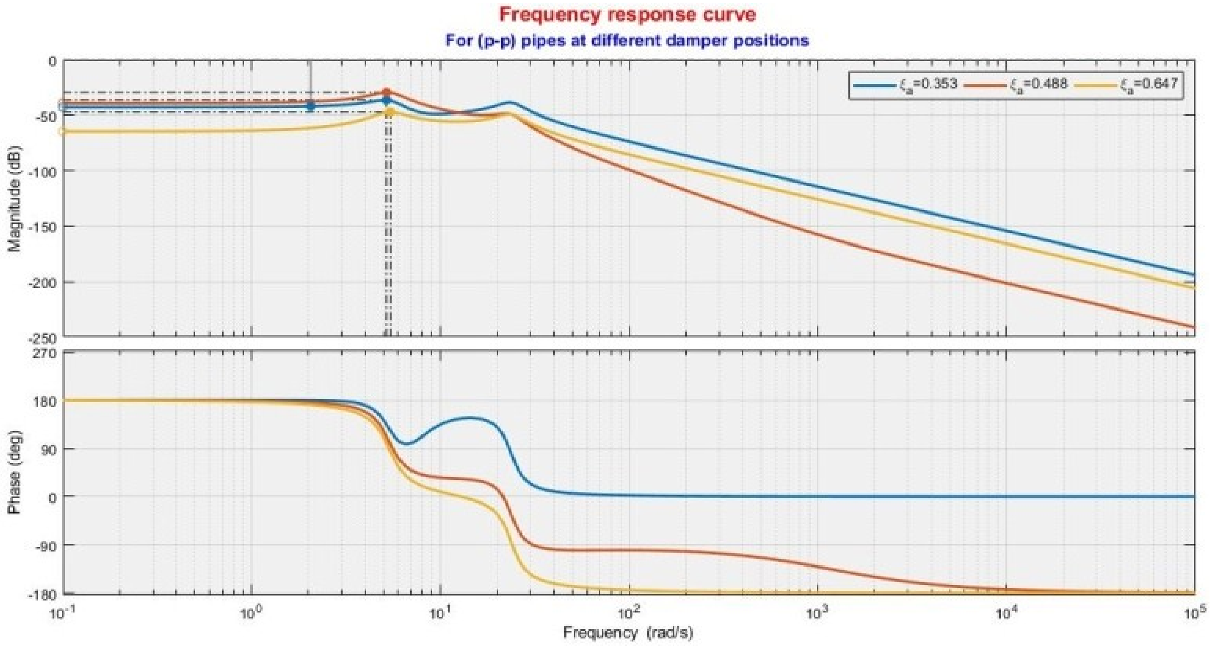
On a logarithmic x-axis scale the frequency is plotted. Will deal with two cases of pipe fixation, pinned for both sides and cantilever pipe fixing case. As the most widely used installation methods in real projects.

3. Results and discussion

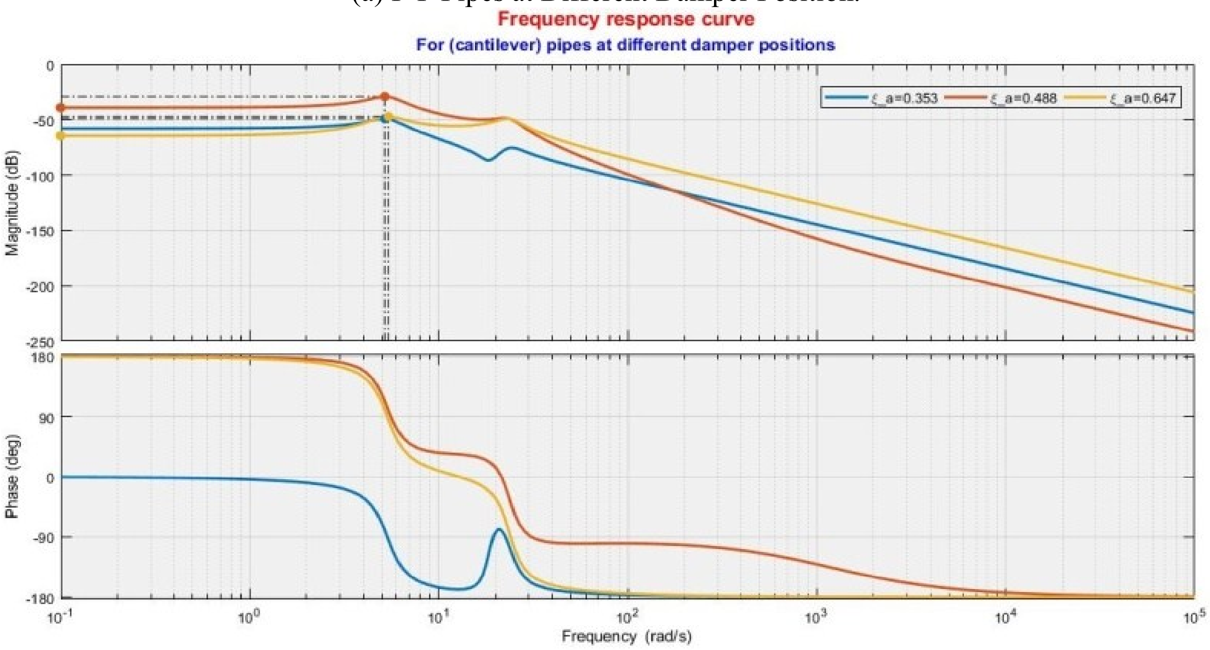
This section deals and discuss theoretical results for vibration and active control vibration of pipes conveying, for the various fixed cases (Simply support, fixed-pinned, fixed- fixed and cantilever) pipes, and different values of pressure (1 bar , 3bar , 4.5 bar). Discussion of the parameters effect of the main pipe as pressure, damping, velocity and mass ratio, on the given dynamic behavior. The general equation of the pipe was used in the analytical aspect, where it was derived and applied to each case of fixation after applying the boundary conditions. The results were obtained by Matlab 2018. In order to justify the work and theory studied, Matlab simulation was then used to control, in the manner of the servo schemes. Thus, the investigation included various parameters effect and calculated as,

1. Effect of the hydraulic damper position on frequency response, as shown in Figure 3.
2. Effect of the base width of hydraulic damper on frequency response, as shown in Figure 4.
3. Effect of the damping on frequency response, as presenting in Figure 5.
4. Effect of pressure on frequency response, as given in Figure 6.

Figure 3 shows the Bode diagram (frequency response), for various damper locations, with $u_o = 1$, $\alpha = 0.01$ and $\Delta\xi = 0.1$. Regarding pinned for both sides, note that the frequency response is fixed at low frequencies and the highest response is achieved at the fixing position $\xi_a = 0.488$ with the approximate maximum response when changing the damping positions but the frequency response decreases significantly at high frequencies. The phase angle between the input and output is high at low frequencies (all positions), but the value of the phase angle or the angle of difference between the input and output decreases at medium frequencies, and in comparison with the different locations of the damper and with the response pattern and phase angle chart. Find that the best result is at the damping position $\xi_a = 0.488$ where get the minimum response and the lowest difference in the phase angle is almost zero at medium frequencies. With respect to the cantilever pipe, the best frequency response and the smallest phase difference angle are achieved at $\xi_a = 0.353$. The reason for this is that the cantilever pipe loses stability at the end of fixation under certain conditions, where the damper is adjacent to the fixation area, get the best frequency response.



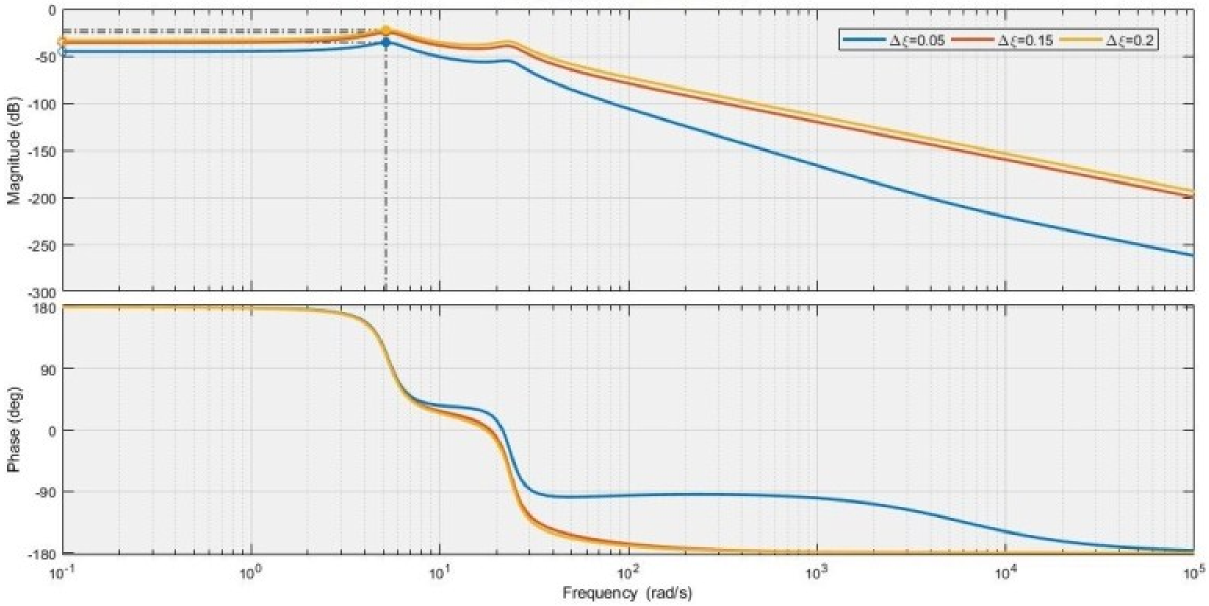
(a) P-P Pipes at Different Damper Position.



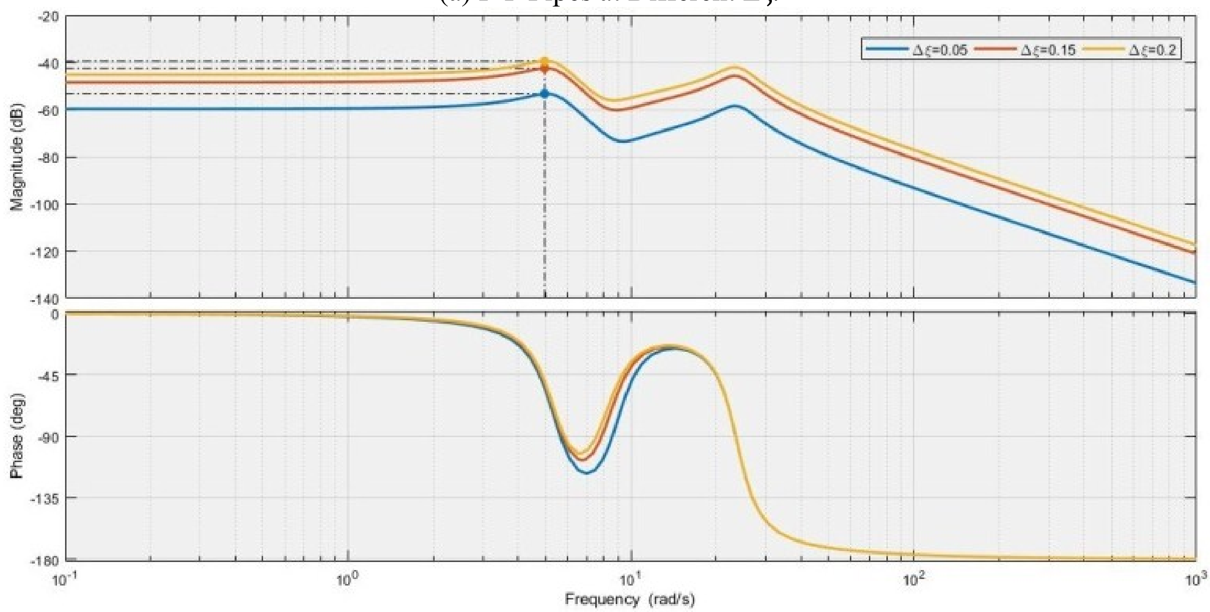
(b) Cantilever Pipes at Different Damper Position.

Figure 3. Frequency response curve at different damper positions.

Figure 4 shows the Bode diagram (frequency response), for the different width of the base of the damper ($\Delta\xi$), with the same survival locations previously selected for the damper, with $u_o = 1$, $\alpha = 0.01$, and damper location $\xi_a = 0.5$. For the pinned of both sides, note that the frequency response is almost fixed at low frequencies and the highest response is achieved at the base width of hydraulic damper $\Delta\xi = 0.2$ with creating the same maximum response when changing $\Delta\xi$ but the frequency response decreases significantly at high frequencies. Note that the phase angle between the input and output is high at low frequencies (all positions), but the value of the phase angle or the angle of difference between the input and output decreases at medium frequencies, however, the difference in phase angle increases when frequencies increase for all $\Delta\xi$, and in comparison with the different $\Delta\xi$ and with the response pattern and phase angle chart. Find that the best result is at the base width of hydraulic damper $\Delta\xi = 0.05$ where get the minimum response and the lowest difference in the phase angle is almost zero at medium frequencies but for a limited frequency. With respect to the cantilever pipe, the best frequency response and the smallest phase difference angle are achieved also at $\Delta\xi = 0.05$ then comes $\Delta\xi = 0.15$. The worst case at the frequency amount is approximately 8 rad/sec, where the frequency response increases and the difference in phase angle increases.



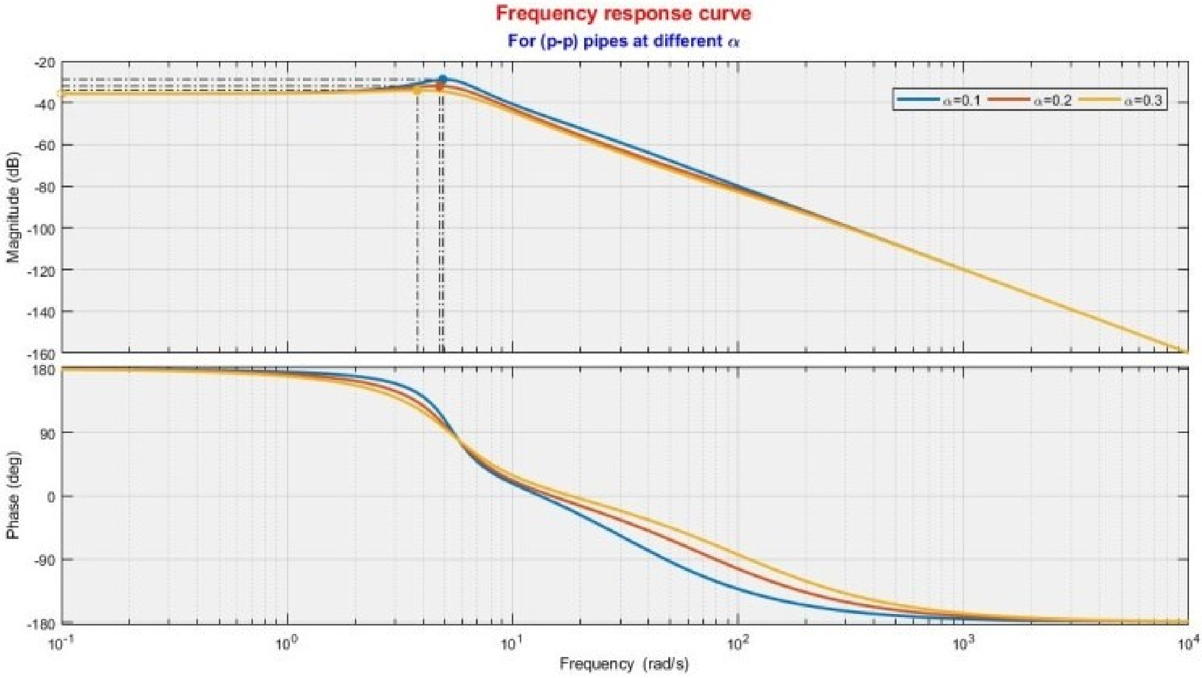
(a) P-P Pipes at Different Δξ.



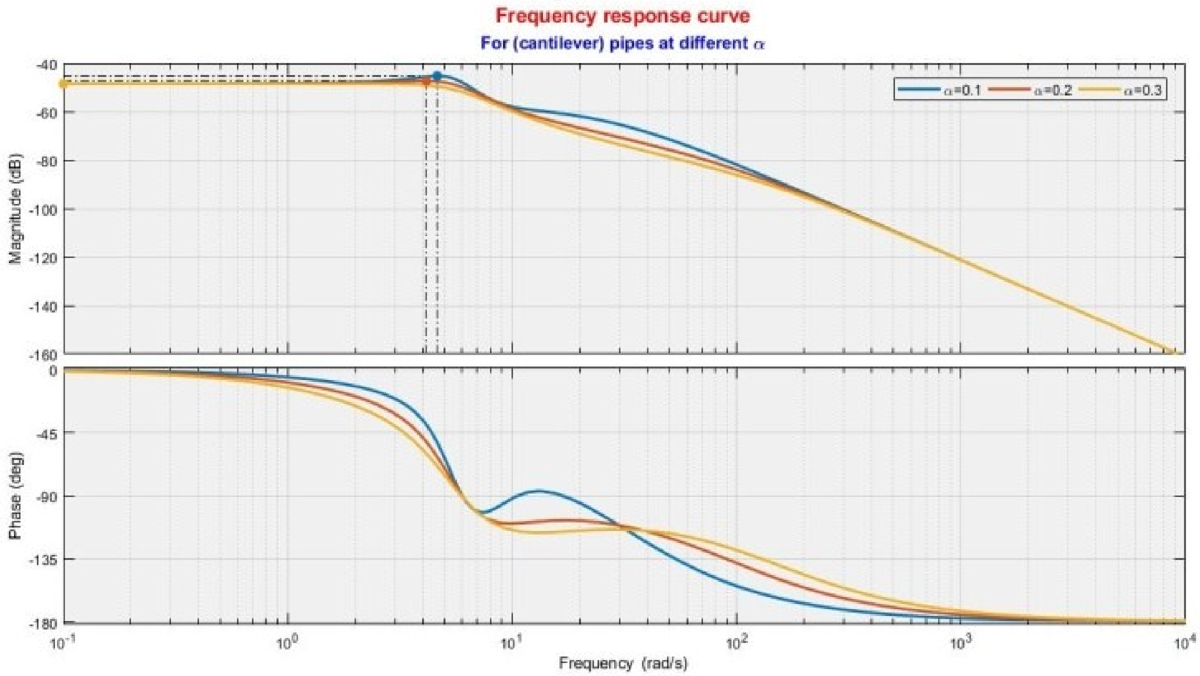
(b) Cantilever Pipes at Different Δξ.

Figure 4. Frequency response curve at different Δξ.

Figure 5 shows the Bode diagram (frequency response), for various coefficient of damping, with $u_o = 1$, $\xi_a = 0.5$ and $\Delta\xi = 0.15$ .For the pinned of both sides, when the damping amount is changed, the frequency response is not significantly affected as the response remains stable at low frequencies, it rises to the highest value at medium frequencies, but only slightly, the response is almost linearly lower than the high frequencies. The maximum response is achieved at $\alpha = 0.1$. As for the phase angle, all the damping values start at 180 degrees to be almost zero at medium frequencies and rise again at high frequencies. In the case of the cantilever pipe, the results are somewhat similar at the frequency response scheme, but the difference at the phase angle, where at low frequencies, the phase angle between input and output is approximately zero and begins to rise with higher frequency value and the best damping $\alpha = 0.3$.



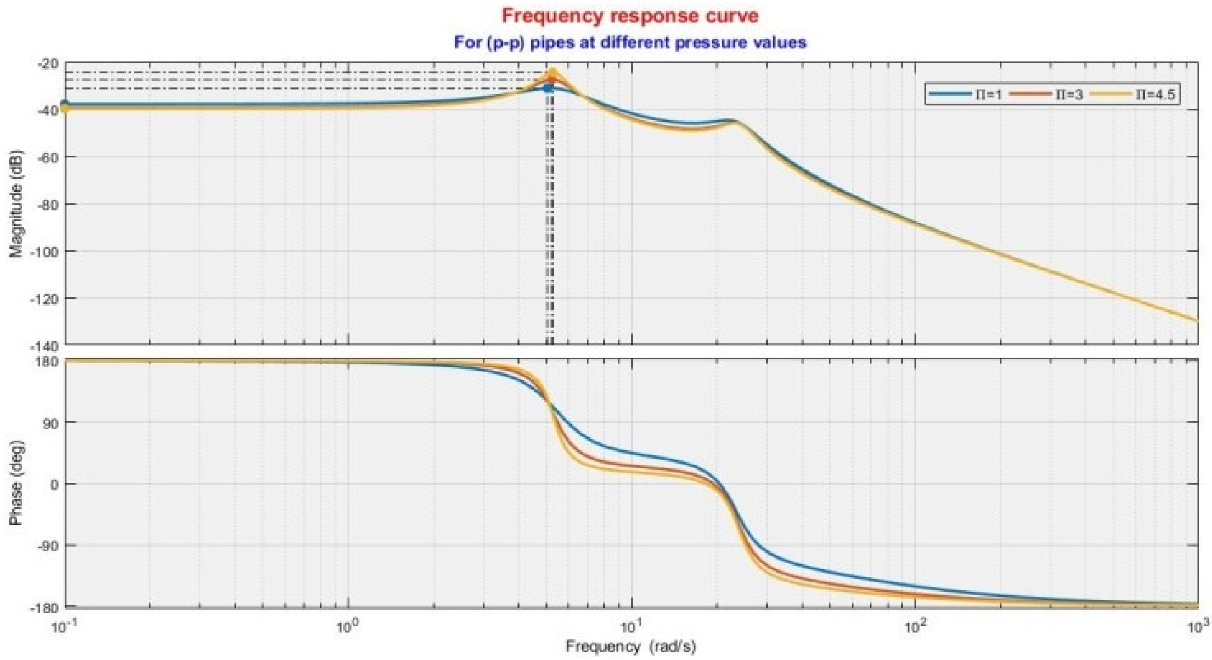
(a) P-P Pipes at Different α.



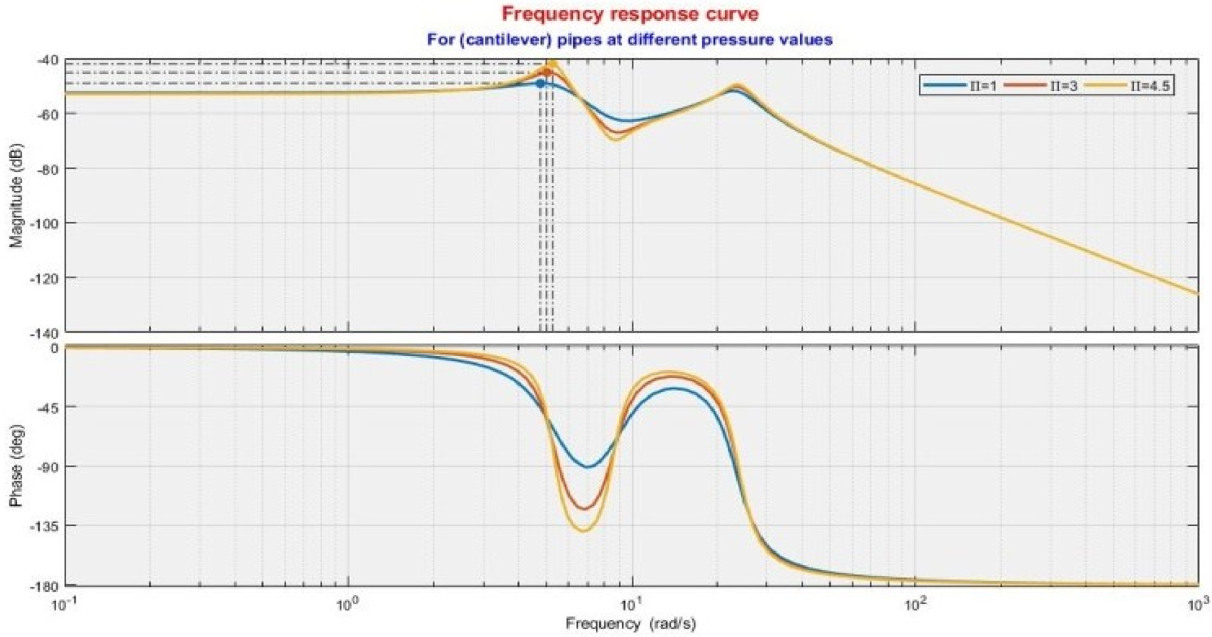
(b) cantilever Pipes at Different α.

Figure 5. Frequency response curve at different α.

Figure 6 shows the Bode diagram (frequency response), for different pressure values, with $\alpha = 0.01$, $\xi_a = 0.5$ and $\Delta\xi = 0.1$.Regarding pinned for both sides, note that the frequency response for almost all pressure amounts is equal at small and medium frequencies, but the response decreases significantly at high frequencies, preceded by a slight rise at the middle of the chart. The minimum frequency response at the frequency value is equal to $\Pi = 1$. For phase angle, all cases have a phase difference of 180 degrees at lower frequencies. But this difference decreases in phase at the average frequency and then due to the increase in phase and for all pressure values. In the case of the cantilever pipe, the highest response at the pressure is $\Pi = 4.5$ and the minimum difference in the phase angle at the pressure is $\Pi = 1$.



(a) P-P Pipes at Different Pressure Values.



(b) Cantilever Pipes at Different Pressure Values.

Figure 6. Frequency response curve at different Π.

4. Conclusion

From discussing the theoretical results and vibrations that occur in each case of pipe fixations and the process of controlling the system and obtaining these results in detail for each case when changing the parameters of the system or changing the structures of the system will be presented conclusions As follows below,

1. The theoretical side was used to find the natural frequencies of different types of pipes and the results are logical. Also the results were satisfactory when changing speeds, pressures, mass ratios and many parameters that govern the system.
2. In all types of fixations of pipes found decreases of natural frequencies when increasing the flow of fluid. In all types of pipe fixations increase the speed increases the pressure and therefore an increase in vibration and therefore a decrease in natural frequencies. When the speed is increased, the control performance decreases due to the increased force of Coriolis.

3. There was a very large convergence in the results of control theories used. The results of the change of the parameters of each fixations, were compared with each control theory used and found a match in stability and response.
4. For critical speeds, the increase in pressure reduces those speeds for all types of fixings for pipes, increase the proportion of mass less critical speeds and all types of fixations for pipes.
5. In the cantilever pipe was extracted the natural frequency, which is two parts, one is a real part and the other is an imaginary case of stability. Increasing this part means an increase in instability. It requires the use of mechanics and changing the governing parameters to reduce this part and thus increase stability and control.

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