INTERNATIONAL JOURNAL OF
ENERGY AND ENVIRONMENT

Volume 10, Issue 6, 2019 pp.329-344
Journal homepage: www.IJEE.IEEFoundation.org

Active vibration control analysis of pipes conveying fluid rested on different supports using state-space method

Dhurgham Salah Hussein, Muhannad Al-Waily

Department of Mechanical Engineering, Faculty of Engineering, University of Kufa, Iraq.

Received 20 Aug. 2019; Received in revised form 05 Nov. 2019; Accepted 10 Nov. 2019; Available online 30 Dec. 2019

Abstract
The problem of vibrations cause the collapse of the systems 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, in addition to the study of reducing the vibrations of these pipes. The ideal control is designed based on, the state space theory and according to the linear behavior of time, because the stability of the pipes are found from linear change. Different factors were studied on the controller's performance and then the optimal factors were obtained to obtain the best performance. The research included the study of the response and the natural frequency and the study of the dynamic behavior of different types of stabilization of the pipes in the presence with no hydraulic damping (active control) and monitoring the response and stability of each case of stabilization. Where, the control investigation include using the analytical solution of general equation of motion for pipe by using state space technique. There, the parameters studied for active control pipe vibration included the effect of hydraulic damper position, base width of hydraulic damper, damping, and the flow Pressure, on the pipe vibration response, with various pipe boundary conditions supported.

Keywords: Active vibration control analysis; Pipe conveying fluid; State space method; Phase plane; linear time invariant; Multiple input; Multiple output systems.

1. Introduction

The main purpose of the pipes is to transmit energy or energy flow, mass liquid flow. 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. 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. 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 [1]. 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. In general, the behavior of dynamic for fluid conveying pipes are more complex than the structure without fluid.

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 [2]. The flow of fluid inside the pipes may produce the following effects, some or all influences can be achieved reliance on the variable of state [3].

- Effect of pressure, the pressure difference is generated by the fluid inside the pipes. This effect can be displayed as a force static compressive transverse work on the pipe.
- Effects Coriolis force, the effect is produced when the liquid moves horizontally and causes the fluid elements to rotate, this effect changes the movement of the pipes. This is called effect Coriolis force the speed of liquid damping is positive within certain limits of speed and this damping leads to reduced vibrations due to damping technology within specified limits as mentioned. But when the pipe loses its stability and the damping becomes negative and increases the oscillation that leads to the growth of the vibration capacity, this occurs at high velocities. This is another form of sensitivity called the flutter.
- Change the momentum of the fluid that causes the bending of the pipe, this is called the effect of centrifugal force.

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 Active Vibration Control of pipes conveying fluid and vibrations of flexible beam structure.

At (1995) C.H.Yau et al. [4] Designed An active control vibration system and this has been verified numerically, to suppression the unwanted disordered vibration in a restricted flexible fluid conveying pipe, that show regions of chaotic and flutter motions at enough raise velocity of flow, analytical model of the four-dimensional got than the continuous system by Galerkin's manner, which that has been already verified to appear enough the dynamics, is used for designing of the control law. Different methodologies of control design are investigated for various conditions. Primarily, the theory of optimal regulator is applied to finding feedback gains to system stability with Information full state. The effects of the length and location of the piezoelectric actuators on the activity such as a vibration damper are tested theoretically in this status. Secondly, a case observer is added to evaluation the signals of required state. Then, at (2003) Z. Jiao et al. [5], in this study provides a new active vibration control method to reduce the vibrations of fluid energy supply efficiently. Note that passive hydraulic absorption is traditionally used to decrease vibrations of fluid energy supply systems. It is very complicated to adjust the situation of variable operating, like loads of variable and Asymmetric vibration frequency. And explained in his study the theory. Test the active control of the vibration in the fluid supply system with detailed water supply. After this, at (2004) K. Hiramoto and H. Doki [6], Studied the problem of improving the shape for a pipe cantilevered. The critical velocity for the system of closed-loop is maximized by the location of the actuator and the sensor and the distribution outer diameter for the pipe. The external diameter distribution has been improved so that the total size of the pipe does not change with the distribution of primary diameter. The flow velocity that cannot be stabilized by active control System with the amount of energy predefined is the definition of the critical flow velocity for the closed-loop system. Through this definition, it's physically credible comparing the stature of sundry design selectors since all selectors are actively controlled with the energy exhaustion same.

At (2011) Y. Hwang and C. Liang [7] Studied active control the vibration for pipes of Timoshenko to fluid conveying. Excrescent vibration at this flow-induced problem of vibration is subdued through an active feedback control system. Put in this search a strategy of the active vibration deactivation mechanism also formulated a dynamic system. Applied technique independent modal optimal space control for the highest efficiency of the system can be obtained. Also at (2013) Y. Sun et al. [1] Use principle of Hamilton in the notice the equations of the axial and lateral motion of the pipe fluid conveying, where he took into consideration the condition of fluid-structure response, friction coupled and coupling of poison, and to address the research control of vibration for pipes fluid conveying and presented. Thence the theoretical for the foundation is upgraded for more calculation and analysis. Also at (2013) Y. Hwang et al. [8] Studied analysis of numerical stability for freelance configurable vibration deactivation of a pipe conveying fluid employing a piezoelectric inertia actuator (PIA). The stability case of the approach as suggested by the pioneer improved is self-addressed. The approach uses infinite weight control for one component of the configurable input control and leads to an intense control extension problem for the complicated mode there is a possibility controlled, simply resulting in closed-loop system instability unto for stable of open loop systems.

At, (2014) Zhen Wang et. al. [9], the main causes of pipe collapse in ships and submarines are due to vibrations, where vibrations cause stress fatigue and noise. When the engine's speed is changed and the frequency increases, these vibrations will decrease. Active control is used to find the algorithm of adaptive analysis and frequency in the digital signal processing system. The researcher conducted the experiment on the closed-pipe pump, where the acceleration was recorded with control and without control when the system of the pump and pipes at different frequencies. Then, at (2015) M. J. Jweeg and T.J. Ntayeesh, [10], studied a problem about active control of Factors resonance of pipes conveying for pinned-pinned. The active controllers are prepared in view of the Factors resonance. Piezoelectric ceramics are utilized as mode transducer and actuators in control. The most important part of this study is the design of controllers. 1st, on the fundamental of linear quadratic for optimal control theory, designed the optimal controller It is according to with the linear time fixed portion of the system, as a result of the Stabilization of pipes are principally determined by their linear portion and also the velocity of the fluid varies in little range. 2nd, on the fundamental of the optimal controller. Also at same year, same authors, [2], submitted a study to investigate the closed and open loop time responses to active vibration concerning smart cantilever pipe fluid conveying as simulation and experimental study of active vibrations. A program has been written to simulate the lowering of active vibration of pipe stiffened with actuators and piezoelectric sensors in ANSYS and language of parametric design (APDL). This use makes the ability of finite element for the ANSYS program and it includes an amount based on the optimal control of linear quadratic named (LQR) schemes to investigate closed and open loop time responses. Charts to investigate the closed and open loop time responses. In addition, also at 2015, same authors, [11], 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.

At (2017) Y. Beiming et. al. [12] Took the difference of the dimensionless of the frequency complex of the two manners lowest of the cantilevered fluid-conveying pipe is derived theoretically relatively velocity of the dimensionless flow. Further, some methods of stabilization are validity and proposed is empirically examined. In this study presented equations of motion dimensionless and an analytical model are introduced at first. So a difference of complex frequency dimensionless relatively velocity of the dimensionless flow and the mathematical reason that causes vibration of self-excited and discussed it in this study. Also at (2017) X. Dekui and N. Songlin [13] 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. In addition, at same year, M. Al-Waily et. al., [14], investigated the effect of of crack angle nd velocity through the pipe on the vibration behavior for pipe, in addition to, investigated its effect on the flow characterization on the flow of pipe. Where, the investigate included using experimental and numerical technique to calculate vibration and flow characterizations of pipe. These, the numerical technique include using of finite element method, by using CFD technique, to calculate the natural frequency of pipe with crack angle effect and flow behavior with same effect. Also, the experimental technique included calculated for natural frequency of pipe with crack angle, and then, comparison the results together with numerical results calculated.

Also, at same year, M. A. Al-Baghdadi et. al., [15], investigated the effect of sand transportation on the erosion behavior of elbow for pipe. Where, the investigation included calculated the erosion behavior for elbow by using CFD technique by using three erosion models, and then, comparsion the results calculated by experimental results. Finally, at 2018, R. H. Al-Khayat, [16], studied the effect of sand particles on the erosion behavior for elbow of pipe with various pipe velocity effect. Where, the investigation included calculated the erosion behavior for 3-D elbow by using CFD program. There, the investigation included evaluated for erosion position for elbow with various fluid characterizations.

Therefore, at this work presenting the analytical analysis for general equation of motion for pipe with flow induce vibration, to calculate the control for pipe vibrated by active control with using state space technique. In addition to, the analytical solution included using the expression of Ritz-Galerkin technique to solvation the general equation of motion for different pipe supported condition. Where, the results investigation the effect of hydraulic damper position and the base width of hydraulic damper, in addition to flow characterizations, on the vibration response for pipe.

2. Analytical investigation

The analytical solution for engineering problem including drive for general equation of motion for system, [17-20], and then, solution for its equation by using exact solution, [21-24], or numerical technique, [25, 26]. Usually, the analytical solution using to presenting the agreement for experiment, and numerical results, [27-29], by comparison the experimental or numerical results by analytical results, [30-32]. When, the solution for pipe vibration problem, for this work, included drive the general equation of motion and solution by using Ritz-Galerkin technique to calculate the behavior of pipe with flow effect as a function of x and time. In addition to, using the exact solution, with state space technique, to calculate the control behavior of pipe with various flow and damper effect.

Theoretical analysis of the dynamical behavior and vibration of fluid conveying pipes in this section will be offered. For more clarity, the chapter will be separated into two parts, the first study dynamic, and stability of pipe conveying fluid, the second, active vibration control of pipes fluid conveying. So as to conduct an overall investigation into the dynamic behavior of pipes conveying fluid a lot of ways of solutions will be attempted. Based on the returns from former researchers, in the current chapter, the task will focus on the analyses of the stability of the zero balance status of fluid conveying pipes under pulsating flow. Primarily, the motion differential equation of pipes is built, secondly, the differential equation of system state is gained through reduced order treatments, discretization and dimensionless. Then uses the average method to forerun average treatment on the differential equation of system state, based on the averaged independent equation, analyzes of stability of zero balance status done. Therefore, the differential equation of motion for pipes is,

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

where, a: viscoelastic coefficient (minimal value), $\overline{T}$: axial force on pipe unit (cross section), E: elasticity modulus of material, M: fluid mass per unit length, m: pipe mass per unit length, U: velocity of fluid, $\overline{P}$: the pressure exerted on the pipe ends per unit area for the fluid, A: flow cross section area, I: moment of inertia for pipe, v: a Poisson's ratio, $\delta = 1$ for the motion at the lower end of the pipe is bounded and $\delta = o$ for the motion at the lower end of the pipe is free.

So as to get the differential equation of general motion, the equation of motion Eq. (1) 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 Eq. (1) into the dimensionless differential equation. The parameters of dimensionless are,

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

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,

$$\Omega=\omega L^2\sqrt{(m_f+m_p)/EI} \tag{3}$$

Plugging Eq. (2) into Eq. (1), the dimensionless differential equation is established,

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

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,

$$u = u_0(1 + \mu\cos(\omega\tau)),\ u^2 = u_0^2(1 + 2\mu\cos(\omega\tau)),\ \dot{u} = -u_0\omega\mu\sin(\omega\tau) \tag{5}$$

Through that, $\mu << 1$ pointing to micro quantity, u_0 pointing to average velocity. Plugging equation (5) into equation (4) and put nonlinear part and pulsating flow part on the right part of the equation, then the equation is found dimensionless,

$$\begin{pmatrix} \ddot{\eta} + 2M_r u_0\,\dot{\eta}' + \alpha\dot{\eta}^{(4)} + \eta^{(4)} + (u_0^2 - \Gamma + \Pi(1-2\nu\delta) - \bar{g})\eta'' + \bar{g}\xi\eta'' + \bar{g}\eta' = \\ \mu\begin{pmatrix} M_r u_0\omega(1-\xi)\eta''\sin(\omega\tau) - \\ (2u_0^2\eta'' + 2M_r u_0\,\dot{\eta}')\cos(\omega\tau) \end{pmatrix} + 2\alpha\gamma\eta''\int_0^1 \eta'\,\dot{\eta}'d\xi + \gamma\eta''\int_0^1 (\eta')^2 d\xi \end{pmatrix} \tag{6}$$

For the purpose of simplification analysis of the equation of differential motion, high order differential equation dimensionless Eq. (6) 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,

$$\eta(\xi,\tau) = \sum_{i=1}^{\infty} \emptyset_i(\xi)\,.\,q_i(\tau) \tag{7}$$

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

$$\eta(\xi,\tau) = \sum_{i=1}^{2} \emptyset_i(\xi)\,q_i(\tau) = \emptyset_1(\xi)\,q_1(\tau) + \emptyset_2(\xi)\,q_2(\tau) \tag{8}$$

The vibration model function for pined at both ends of pipes is,

$$\emptyset_i = \sqrt{2}\sin(\lambda_i\xi), \quad i = 1,2 \tag{9}$$

The vibration model function for fixed at both ends of pipes is,

$$\Phi_i = \cosh(\lambda_i\xi) - \cos(\lambda_i\xi) + \frac{\cosh(\lambda_i) - \cos(\lambda_i)}{\sinh(\lambda_i) - \sin(\lambda_i)}[\sin(\lambda_i\xi) - \sinh(\lambda_i\xi)],\ i = 1,2 \tag{10}$$

λ_1 and λ_2 are the eigenvalues of pipe, $\lambda_1 = \pi$ and $\lambda_2 = 2\pi$ are for pipes pinned at both ends, $\lambda_1 = 4.73$ and $\lambda_2 = 7.8532$ are for pipes fixed at both ends. The Eq. (8) is turned to matrix type, assuming,

$$\Phi = \begin{Bmatrix} \emptyset_1 \\ \emptyset_2 \end{Bmatrix},\ Q = \begin{Bmatrix} q_1 \\ q_2 \end{Bmatrix}$$

Then,

$$\eta(\xi,\tau) = \Phi^T Q = Q^T\Phi \tag{11}$$

Then, the general equation of motion for pipe, Eq. (6), can be written as,

$$\begin{pmatrix} \phi^T\ddot{Q} + 2M_r u_0\phi'^T\dot{Q} + \alpha\phi^{(4)T}\dot{Q} + (u_0^2 - T - \bar{g})\phi''^T Q + \bar{g}\xi\phi''^T Q + \bar{g}\phi'^T Q + \phi^{(4)T}Q \\ = \begin{pmatrix} \mu M_r u_0(1-\xi)\phi''^T Q\omega\sin(\omega\tau) - \mu\left(2u_0^2\phi''^T Q + 2M_r u_0\phi'^T\dot{Q}\right)\cos(\omega\tau) + \\ 2\alpha\gamma\int_0^1 Q^T\phi'\phi'^T\dot{Q}d\xi\phi''^T Q + \gamma\int_0^1 Q^T\phi'\phi'^T Q d\xi\phi''^T Q \end{pmatrix} \end{pmatrix} \tag{12}$$

Multiply $\Phi = \begin{Bmatrix} \emptyset_1 \\ \emptyset_2 \end{Bmatrix}$ with two sides of (12) and then turned it to the following form,

$$\begin{pmatrix} \phi\phi^T\ddot{Q} + 2M_r u_0 \phi\phi'^T\dot{Q} + \alpha\phi\phi^{(4)T}\dot{Q} + (u_0^2 - T - \bar{g})\phi\phi''^T Q + \bar{g}\xi\phi\phi''^T Q + \bar{g}\phi\phi'^T Q + \phi\phi^{(4)T} Q \\ = \begin{pmatrix} \mu M_r u_0 (1-\xi)\phi\phi''^T Q\omega\sin(\omega\tau) - \mu\left(2u_0^2\phi\phi''^T Q + 2M_r u_0\phi\phi'^T\dot{Q}\right)\cos(\omega\tau) \\ +2\alpha\gamma\int_0^1 Q^T\phi'\phi'^T\dot{Q}d\xi\phi\phi''^T Q + \gamma\int_0^1 Q^T\phi'\phi'^T Q d\xi\phi\phi''^T Q \end{pmatrix} \end{pmatrix} \tag{13}$$

Make ξ integral to Eq. (13) at interval [0,1], and orthogonal substitutions for the trigonometric function,

$$\int_0^1 \emptyset\,\emptyset^T d\xi = I,\ \int_0^1 \phi\phi'^T\,d\xi = B,\ \int_0^1 \phi\,\phi''^T d\xi = C,\ \int_0^1 \xi\,\emptyset\,\emptyset''^T d\xi = D$$
$$\int_0^1 \emptyset\,\emptyset^{(4)T} d\xi = \Lambda = \begin{pmatrix} \lambda_1^4 & \\ & \lambda_2^4 \end{pmatrix},\ E = B + D - C \tag{14}$$

Utilization Eq. (14), after lower order during (13) is,

$$\begin{pmatrix} I\ddot{Q} + (2M_r u_0 B + \alpha\Lambda)\dot{Q} + \\ \left((u_0^2 - T)C + \Lambda\right)Q + \alpha\Lambda\dot{Q} + \bar{g}EQ \end{pmatrix} = \begin{pmatrix} \mu\,M_r u_0\,(C - D)Q\omega\sin(\omega\tau) - \\ \mu\left(2u_0^2 CQ + 2M_r u_0 B\dot{Q}\right)\cos(\omega\tau) \\ -2\alpha\,\gamma Q^T C\dot{Q}CQ - \gamma\,Q^T CQCQ \end{pmatrix} \tag{15}$$

Where,

$$B = \begin{pmatrix} 0 & -b \\ b & 0 \end{pmatrix},\ C = \begin{pmatrix} C_{11} & -b \\ b & C_{22} \end{pmatrix},\ E = \begin{pmatrix} -\frac{1}{2}C_{11} & e \\ e & -\frac{1}{2}C_{11} \end{pmatrix} \tag{16}$$

Where, b, C_{11}, C_{22}, and e are calculation for various boundary conditions as,

1. For pipes (pinned at both ends), this is calculated, $b = \frac{8}{3}$, $C_{11} = -\pi^2$, $C_{22} = -4\pi^2$, $e = \frac{40}{9}$.
2. For pipes (fixed at both ends), this is calculated, $b = 3.342$, $C_{11} = -12.3026$, $C_{22} = -46.0501$, $e = 3.3426$.

Then, by using Eq. (16) into Eq. (15), and after arrangements, get,

$$\begin{pmatrix} \ddot{q}_1 \\ \ddot{q}_2 \end{pmatrix} + \begin{pmatrix} \alpha\,\lambda_1^4 & -M_r u_0 \\ 2M_r u_0 b & \alpha\,\lambda_2^4 \end{pmatrix}\begin{pmatrix} \dot{q}_1 \\ \dot{q}_2 \end{pmatrix} + \begin{pmatrix} \left(u_0^2 - T - \frac{1}{2}\bar{g}\right)C_{11} + \lambda_1^4 & \bar{g}e \\ \bar{g}e & \left(u_0^2 - T - \frac{1}{2}\bar{g}\right)C_{22} + \lambda_2^4 \end{pmatrix}\begin{pmatrix} q_1 \\ q_2 \end{pmatrix}$$
$$= \mu\begin{pmatrix} H_1 \\ H_2 \end{pmatrix} - \begin{pmatrix} Q_1 \\ Q_2 \end{pmatrix} \tag{17}$$

For, H, pulsating flow part, Q, nonlinear part. Then the specific terms are,

$$\begin{pmatrix} H_1 \\ H_2 \end{pmatrix} = \begin{pmatrix} \frac{1}{2}c_{11} & -b - e \\ b - e & \frac{1}{2}c_{22} \end{pmatrix}\begin{pmatrix} q_1 \\ q_2 \end{pmatrix}\omega\sin(\omega\tau) - \begin{bmatrix} \begin{pmatrix} 2u_0^2 c_{11} & 0 \\ 0 & 2u_0^2 c_{22} \end{pmatrix}\begin{pmatrix} q_1 \\ q_2 \end{pmatrix} + \\ \begin{pmatrix} 0 & -2M_r u_0 b \\ 2M_r u_0 b & 0 \end{pmatrix}\begin{pmatrix} \dot{q}_1 \\ \dot{q}_2 \end{pmatrix} \end{bmatrix}\cos(\omega\tau) \tag{18}$$

$$\begin{pmatrix} Q_1 \\ Q_2 \end{pmatrix} = \begin{pmatrix} 2\alpha\,\gamma(q_1 \quad q_2)\begin{pmatrix} c_{11} & 0 \\ 0 & c_{22} \end{pmatrix}\begin{pmatrix} \dot{q}_1 \\ \dot{q}_2 \end{pmatrix}\begin{pmatrix} c_{11} & 0 \\ 0 & c_{22} \end{pmatrix}\begin{pmatrix} q_1 \\ q_2 \end{pmatrix} + \\ \gamma(q_1 \quad q_2)\begin{pmatrix} c_{11} & 0 \\ 0 & c_{22} \end{pmatrix}\begin{pmatrix} q_1 \\ q_2 \end{pmatrix}\begin{pmatrix} c_{11} & 0 \\ 0 & c_{22} \end{pmatrix}\begin{pmatrix} q_1 \\ q_2 \end{pmatrix} \end{pmatrix} \tag{19}$$

Then, Eq. (17) it is possible to change into first mode state differential equation, as,

$$\begin{pmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\\ \dot{x}_4\end{pmatrix}=\begin{pmatrix}0&0&1&0\\0&0&0&1\\a_1&a_2&a_3&a_4\\a_2&b_2&-a_4&b_4\end{pmatrix}\begin{pmatrix}x_1\\x_2\\x_3\\x_4\end{pmatrix}+\mu\begin{pmatrix}0\\0\\H_1\\H_2\end{pmatrix}-\begin{pmatrix}0\\0\\Q_1\\Q_2\end{pmatrix} \tag{20}$$

where, $a_1=-\left(u_0^2-T-\frac{1}{2}\bar{g}\right)c_{11}+\lambda_1^4$, $a_2=-\bar{g}e$, $a_3=-\alpha\,\lambda_1^4$,

$a_4=2M_r u_0 b$, $b_2=-\left(u_0^2-T-\frac{1}{2}\bar{g}\right)c_{22}+\lambda_2^4$, $b_4=-\alpha\,\lambda_2^4$

The simulation response of system motion will depend on this equation. The Eq.(20) after adjustment can write as follows,

$$\begin{pmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\\ \dot{x}_4\end{pmatrix}=\begin{pmatrix}0&0&1&0\\0&0&0&1\\a_1&a_2&a_3&a_4\\a_2&b_2&-a_4&b_4\end{pmatrix}\begin{pmatrix}x_1\\x_2\\x_3\\x_4\end{pmatrix}+\mu\begin{pmatrix}0\\0\\H_1-\bar{\alpha}\,\lambda_1^4\,q_3\\H_2-\bar{\alpha}\,\lambda_2^4\,q_4\end{pmatrix}-\begin{pmatrix}0\\0\\Q_1\\Q_2\end{pmatrix} \tag{21}$$

Because of the damping factor (a) is a small amount, therefore assume, $\alpha=\mu\bar{\alpha}$. After simplification of a matrix of Eq. (21),

$$\dot{X}=SX+\mu[\omega\sin(\omega\tau)B_1-\cos(\omega\tau)B_2-\bar{\alpha}B_3]X-\gamma NX \tag{22}$$

Where,

$\dot{X}=[x_1\quad x_2\quad x_3\quad x_4]^T$

$$S=\begin{pmatrix}0&0&1&0\\0&0&0&1\\-\lambda_1^4-\beta c_{11}&-\bar{g}e&0&2M_r u_0 b\\-\bar{g}e&-\lambda_2^4-\beta c_{22}&-2M_r u_0 b&0\end{pmatrix},\ B_1=\begin{pmatrix}0&0&0&0\\0&0&0&0\\\frac{1}{2}c_{11}&-b-e&0&0\\b-e&\frac{1}{2}c_{11}&0&0\end{pmatrix}$$

$$B_2=\begin{pmatrix}0&0&0&0\\0&0&0&0\\2u_0^2c_{11}&0&0&-2M_r u_0 b\\-\bar{g}e&2u_0^2c_{11}&2M_r u_0 b&0\end{pmatrix},\ B_3=\begin{pmatrix}0&0&0&0\\0&0&0&0\\0&0&\lambda_1^4&0\\0&0&0&\lambda_2^4\end{pmatrix}$$

$$N=\begin{pmatrix}0&0&0&0\\0&0&0&0\\x_1^2c_{11}^2&x_1x_2c_{11}c_{22}&2\alpha x_1^2c_{11}^2&2\alpha x_1x_2c_{11}c_{22}\\x_1x_2c_{11}c_{22}&x_2^2c_{22}^2&2\alpha x_1x_2c_{11}c_{22}&2\alpha x_2^2c_{22}^2\end{pmatrix}$$

Above modification just contain system linear parts, the nonlinear portion influences not taken into consideration for system stability through the state of zero balance (excluding critical attitudes),

$$\dot{X}=SX+\mu[\omega\sin(\omega\tau)B_1-\cos(\omega\tau)B_2-\bar{\alpha}B_3]X \tag{23}$$

When critical velocity is bigger than average velocity, or $u_{cr}>u_0$, own matrix S two pair from eigenvalues (pure imaginary). Note that u_{cr}, critical velocity, with term,

$$u_{cr}=\left[T+\frac{1}{2}\bar{g}+\frac{-(c_{11}\,\lambda_2^4+c_{22}\,\lambda_1^4)+\sqrt{(c_{11}\,\lambda_2^4-c_{22}\,\lambda_1^4)^2+4c_{11}c_{22}\bar{g}^2e^2}}{2c_{11}c_{22}}\right]^{1/2} \tag{24}$$

In case of negligence the preload and gravity u_{cr} will be simplified as follows,

$$u_{cr}=\sqrt{-\frac{\lambda_1^4}{c_{11}}}=\begin{cases}\pi i & p-p\\ 6.38i & c-c\end{cases} \tag{25}$$

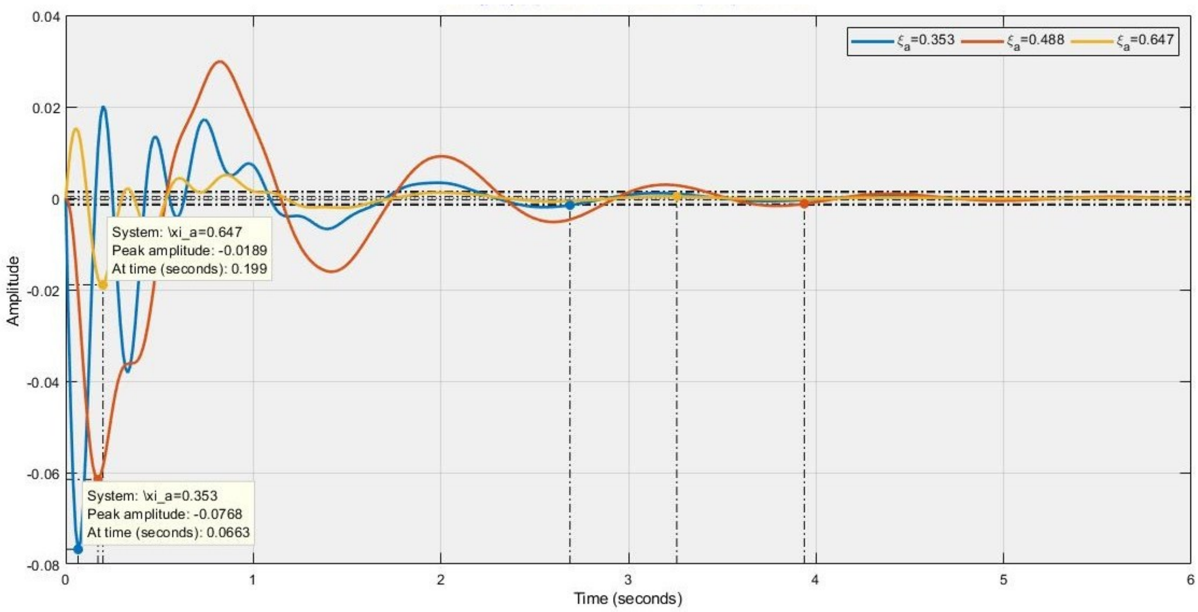
3. Results and discussion

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.

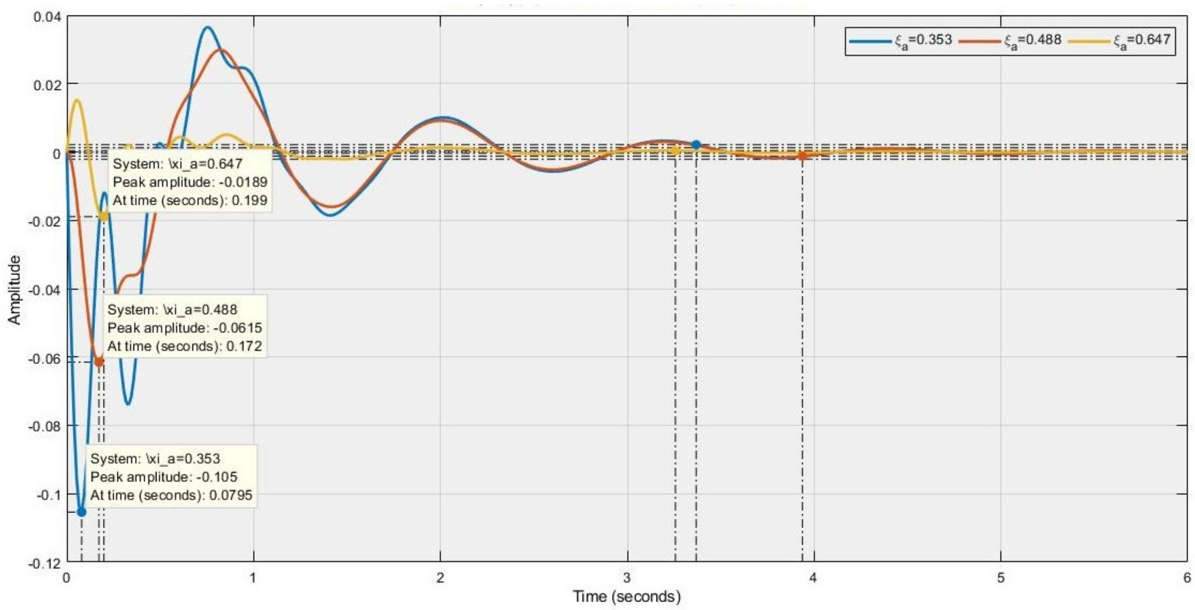
The general equation of the pipe was used in the analytical aspect, where it was derived and applied to each case of fixation after applying the boundary conditions. The results were obtained by using Matlab 2018. In order to justify the work and theory studied, The Matlab simulation was then used to control, in the manner of the servo schemes. Thus, the investigation included various parameters effect and calculated shown in Figures 1 to 4, as,

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

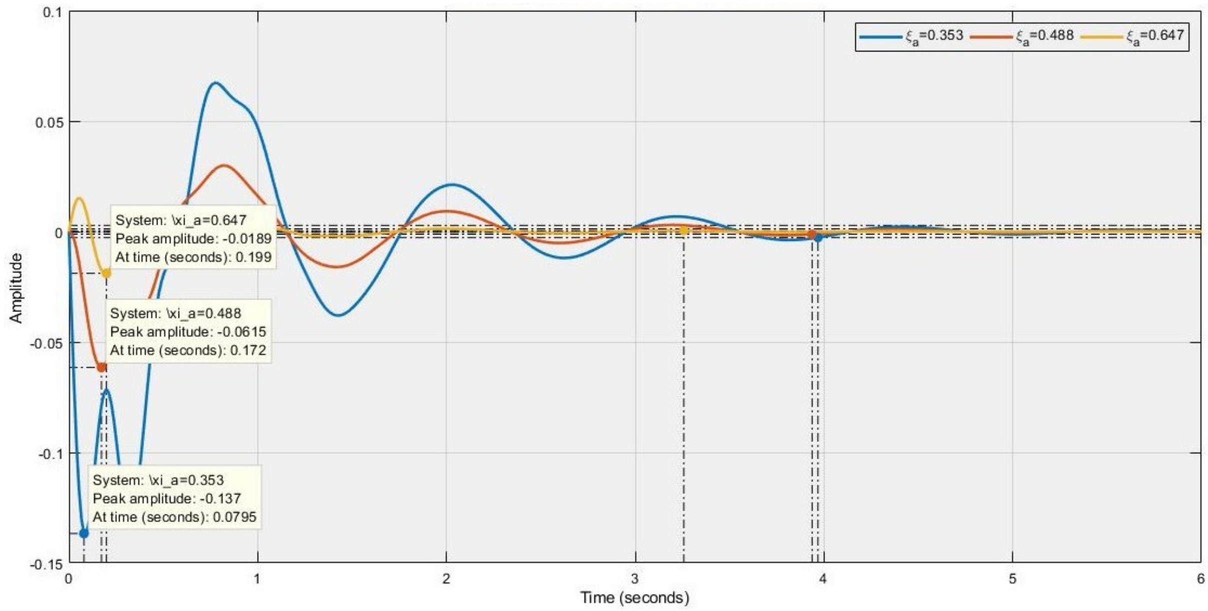
Figure 1 showed the relationship between the damping time and the amplitude and for various damper locations, with $u_o = 1$, $\alpha = 0.01$, and base width of hydraulic damper ($\Delta\xi = 0.1$).



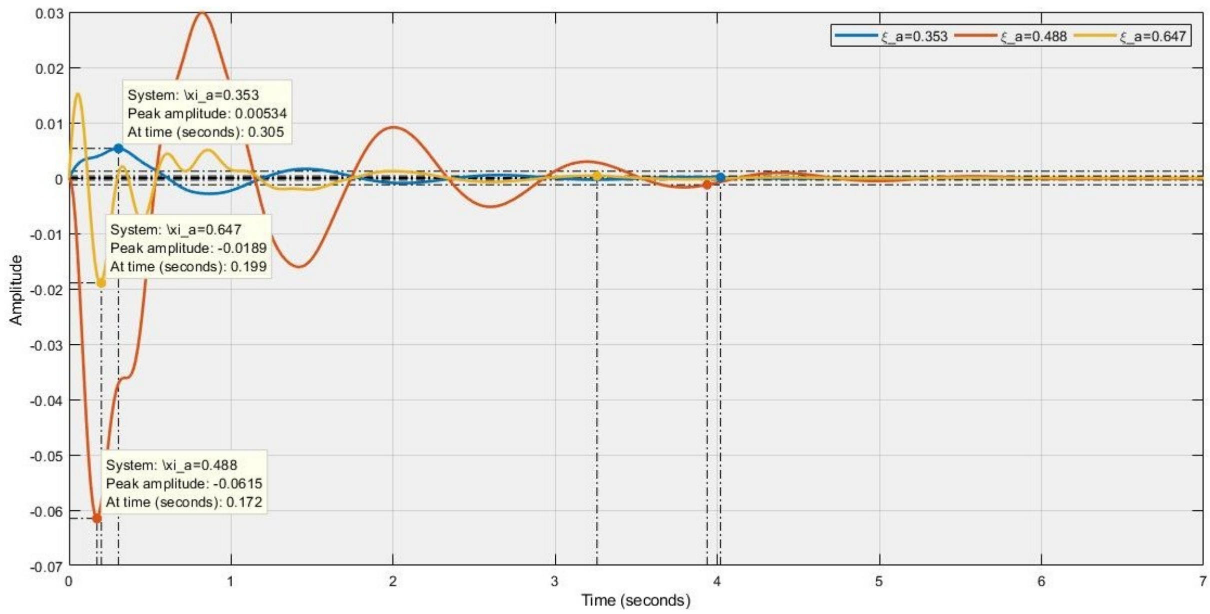
(a) P-P pipes at different damper positions.



(b) C-P pipes at different damper positions.



(c) C-C pipes at different damper positions.



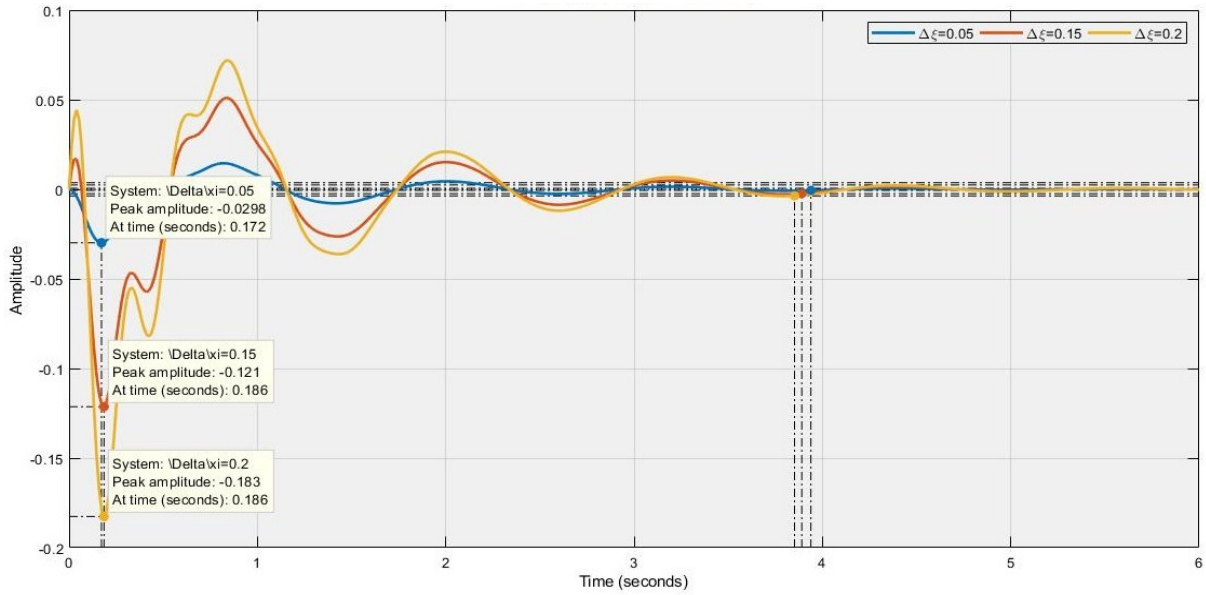
(d) Cantilever pipes at different damper positions.

Figure 1. Displacement response curve at different damper positions.

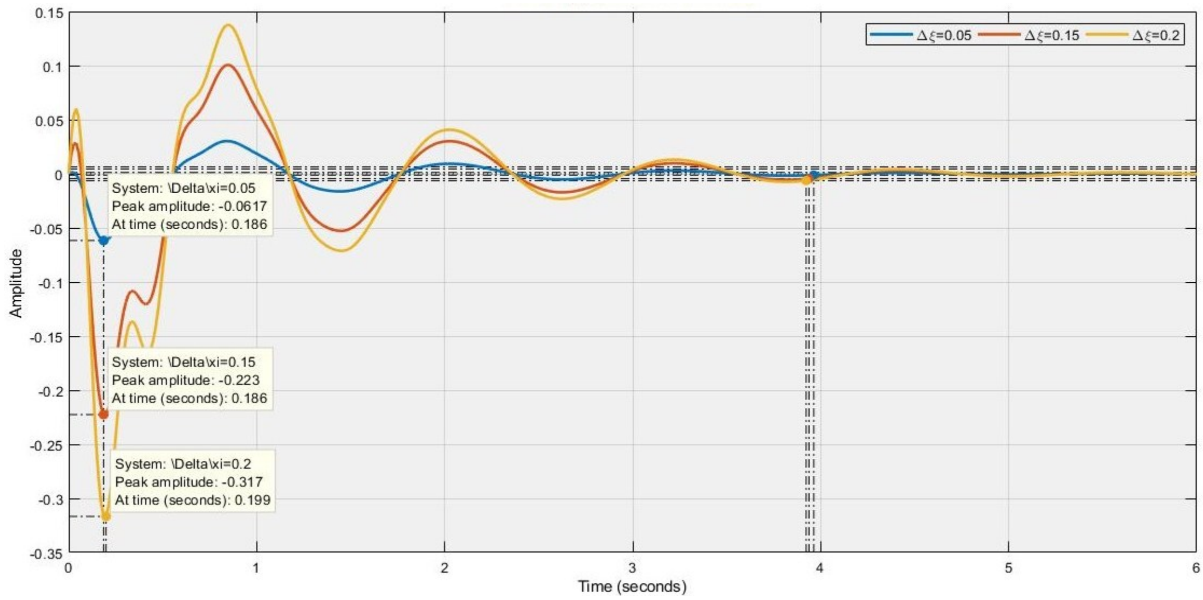
Note that the system tends to stabilize earlier of pinned at both ends of pipes conveying fluid at $\xi_a = 0.353$, but the lowest amplitude at $\xi_a = 0.647$ while at $\xi_a = 0.488$, the amplitude is becoming larger and the system wave frequency is lower and this is logical in such instances of pipe fixation. While the installation (c-p) of the pipes, note that the system tends to stabilize earlier at $\xi_a = 0.647$, and the lowest amplitude at $\xi_a = 0.647$ while at $\xi_a = 0.488$, the amplitude is becoming larger and the system wave frequency is lower and this corresponds to the previous installation state.

When the installation manner of pipe type is fixed for both sides, note that the system tends to stabilize earlier at $\xi_a = 0.647$, and this corresponds to the case (c-p) of the pipe, but it differs from the case (p-p) of the pipe, and the lowest amplitude at $\xi_a = 0.647$ and this corresponds to the cases (p-p), (c-p) of the pipes. As for the last fixation case which is fixation of the cantilever pipe, the lowest amplitude and less disturbance at $\xi_a = 0.353$, this means that the damper is best positioned at the beginning of the cantilever installation despite the increased stabilization time.

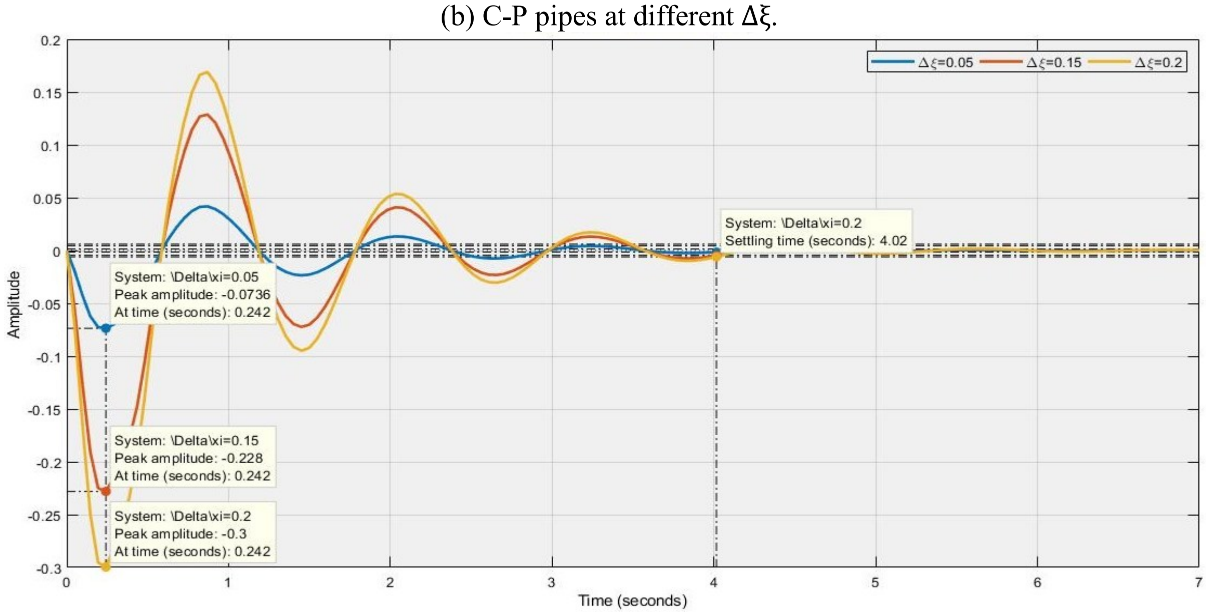
Figure 2 showed the relationship between the damping time and the amplitude and for the different width of the base of the damper ($\Delta\xi$), with the same survival locations previously selected for the damper, with $u_o = 1$, $\alpha = 0.01$, and damper location $\xi_a = 0.5$.



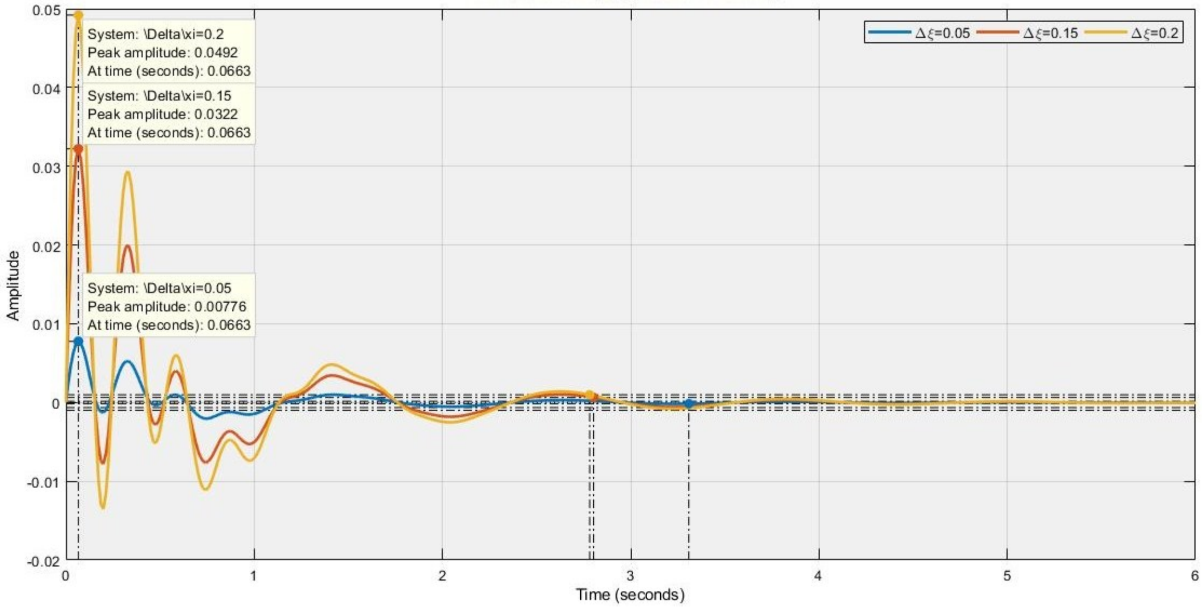
(a) P-P pipes at different Δξ.



(b) C-P pipes at different Δξ.



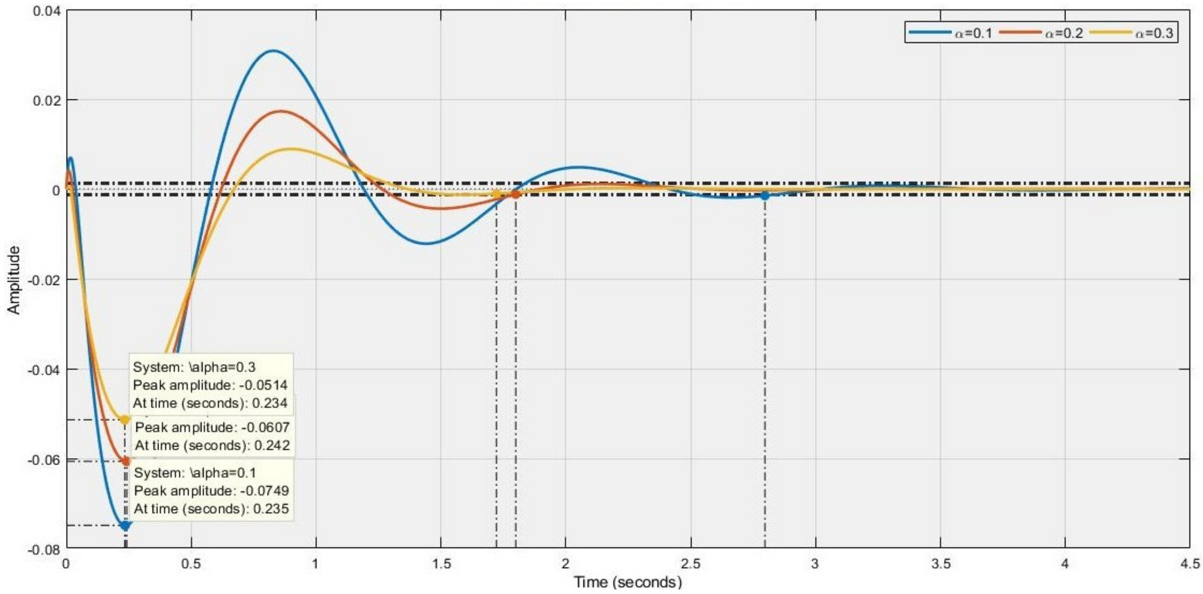
(c) C-C pipes at different Δξ.



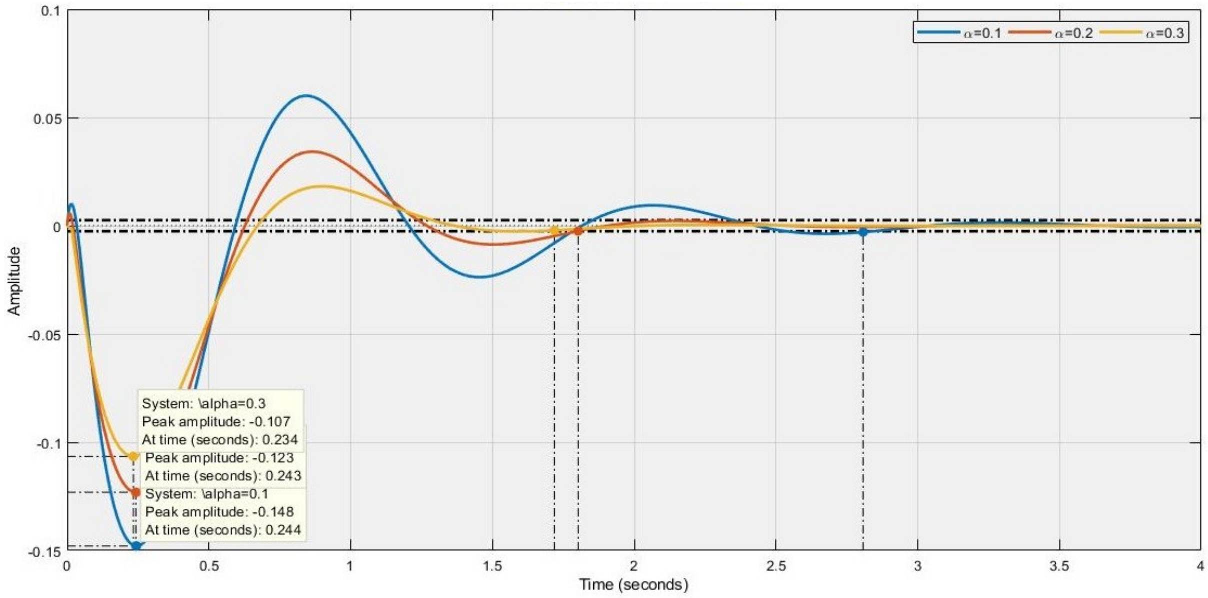
(d) Cantilever pipes at different Δξ.

Figure 2. Displacement response curve at different Δξ.

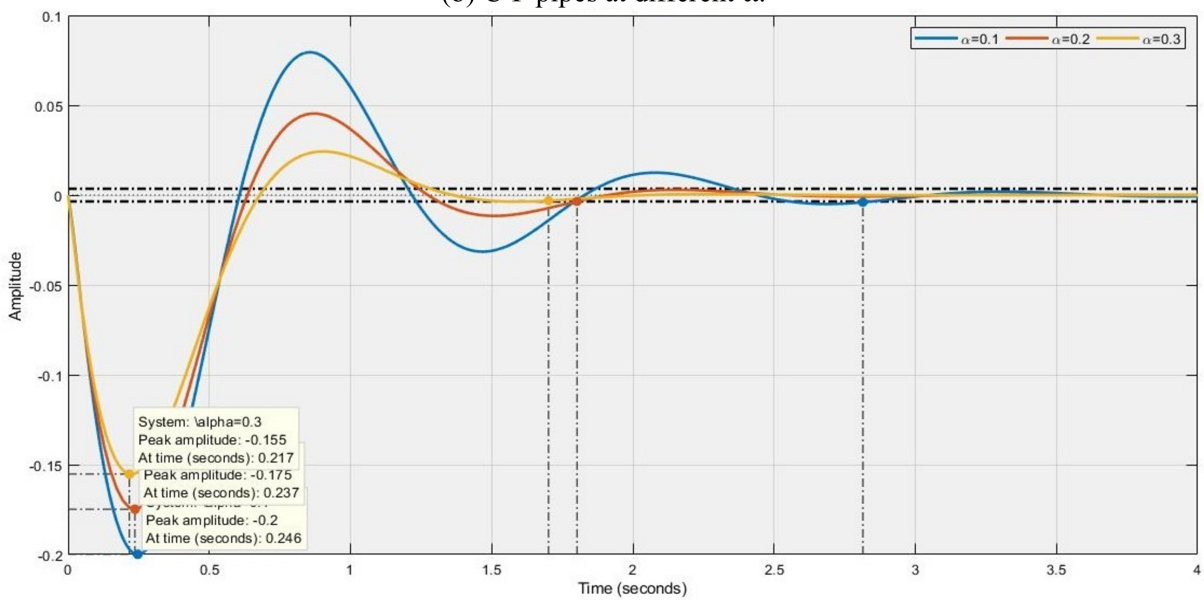
As for the pinned fixation on both sides, notice the stability of the system earlier when the width of the base of the damper is equal to 0.2 and the minimum amplitude at $\Delta\xi = 0.05$ with little disturbance compared to $\Delta\xi = 0.15$ and $\Delta\xi = 0.2$. The highest wave amplitude disturbance occurs at $\Delta\xi = 0.2$, this is almost similar to the results of a pin-fixed installation. In the case of fixation for both sides we note the symmetry of the wave amplitude produced by the oscillation and stability at the fourth second and stability occurs for all cases of $\Delta\xi$ at the same time. As for the cantilever pipe, there is a large wave amplitude disturbance at $\Delta\xi = 0.2$ and $\Delta\xi = 0.15$ .But at these values, early stability occurs compared to $\Delta\xi = 0.05$. Comparing the four charts, we find the fastest stability at the cantilever pipe, at $\Delta\xi = 0.15$ and $\Delta\xi = 0.2$. Then, Figure 3 showed the relationship between the damping time and the amplitude and for various coefficient of damping, with $u_o = 1$, $\xi_a = 0.5$, and base width of hydraulic damper ($\Delta\xi = 0.15$). For the first case we find the fastest stability and the lowest amplitude and turbulence of the amplitude of the wave at $\alpha = 0.3$ with the synchronization of the maximum wave amplitude of all damping values and the results obtained are almost identical to the second and third stabilization, except for a slight difference in capacity and stability time. As for the cantilever pipe, the maximum amplitude at damping is $\alpha = 0.1$ and the minimum amplitude at $\alpha = 0.3$ and a short stabilization time is achieved. Finally, Figure 4 showed the relationship between the damping time and the amplitude and for different pressure values, with $\alpha = 0.01$, and base width of hydraulic damper ($\Delta\xi = 0.1$), $\xi_a = 0.5$.



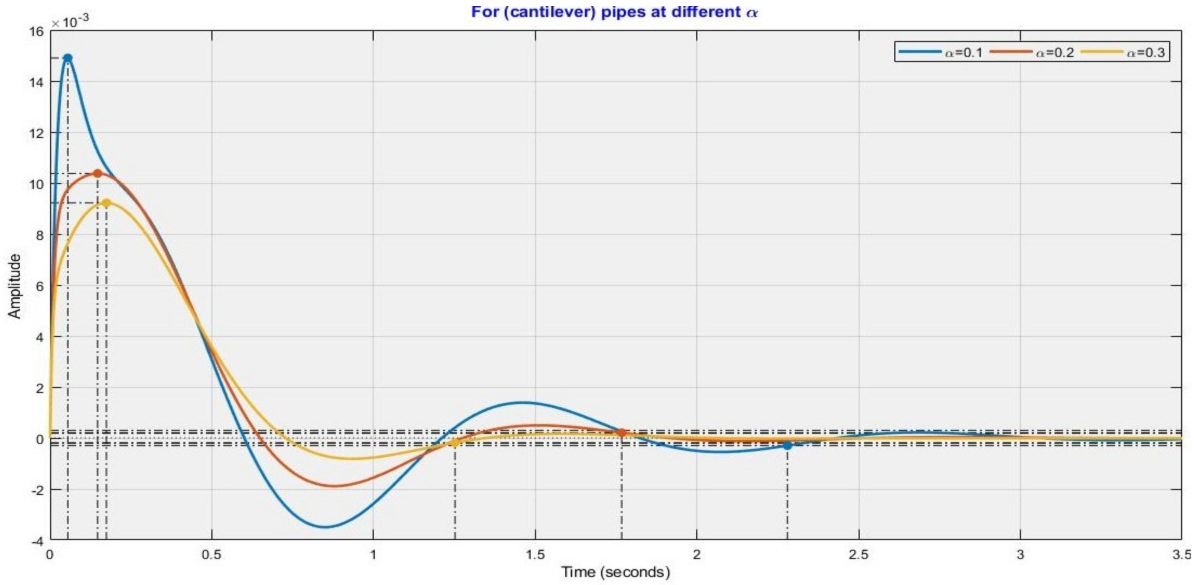
(a) P-P pipes at different α.



(b) C-P pipes at different α.

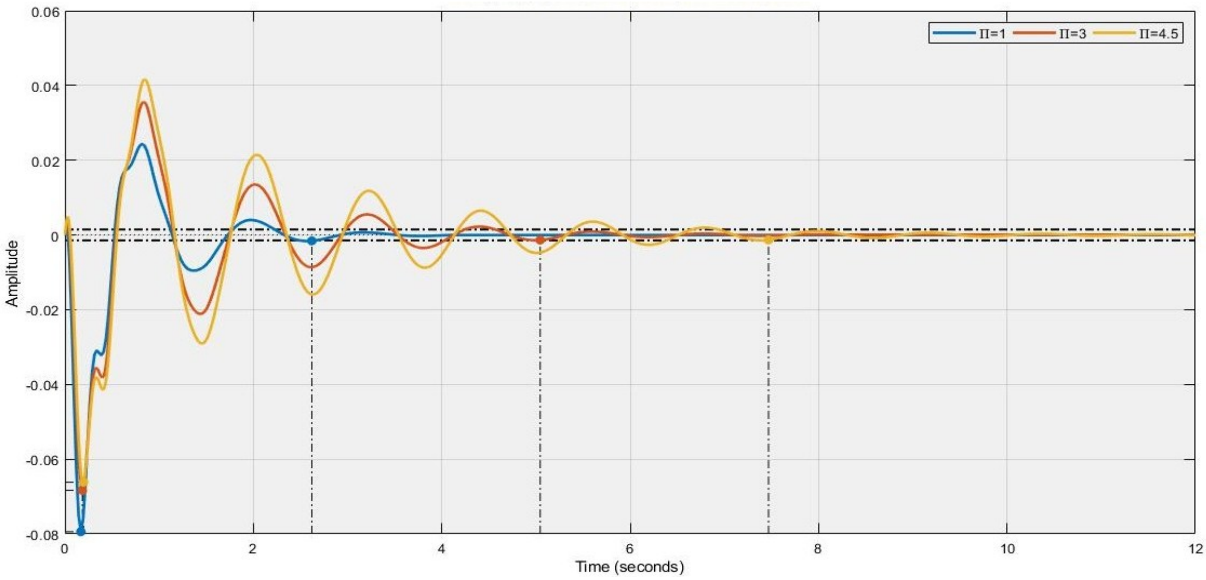


(c) C-C pipes at different α.

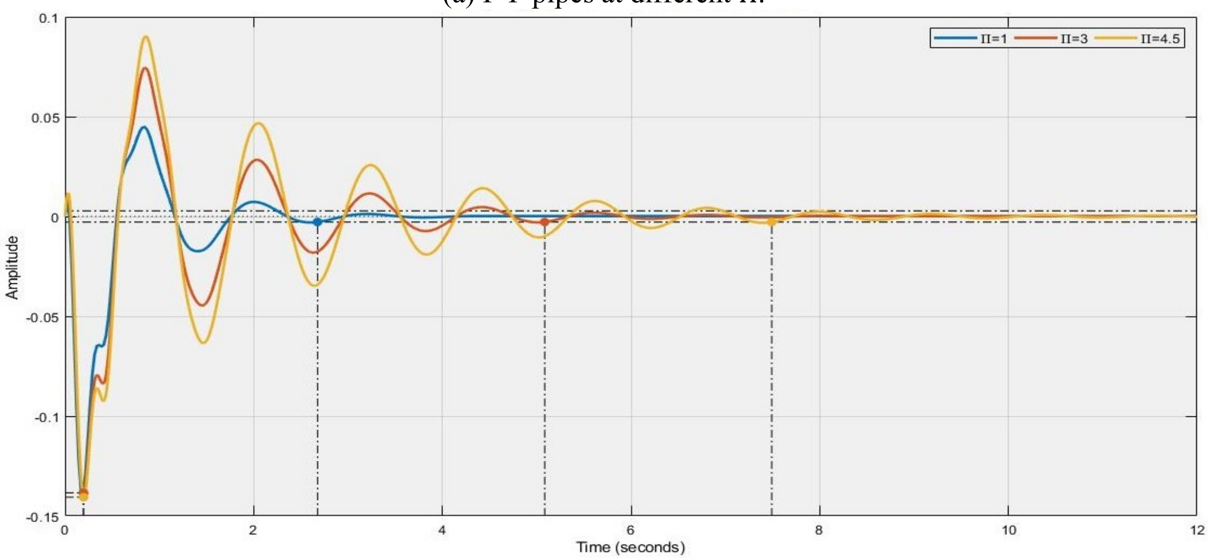


(d) Cantilever pipes at different α.

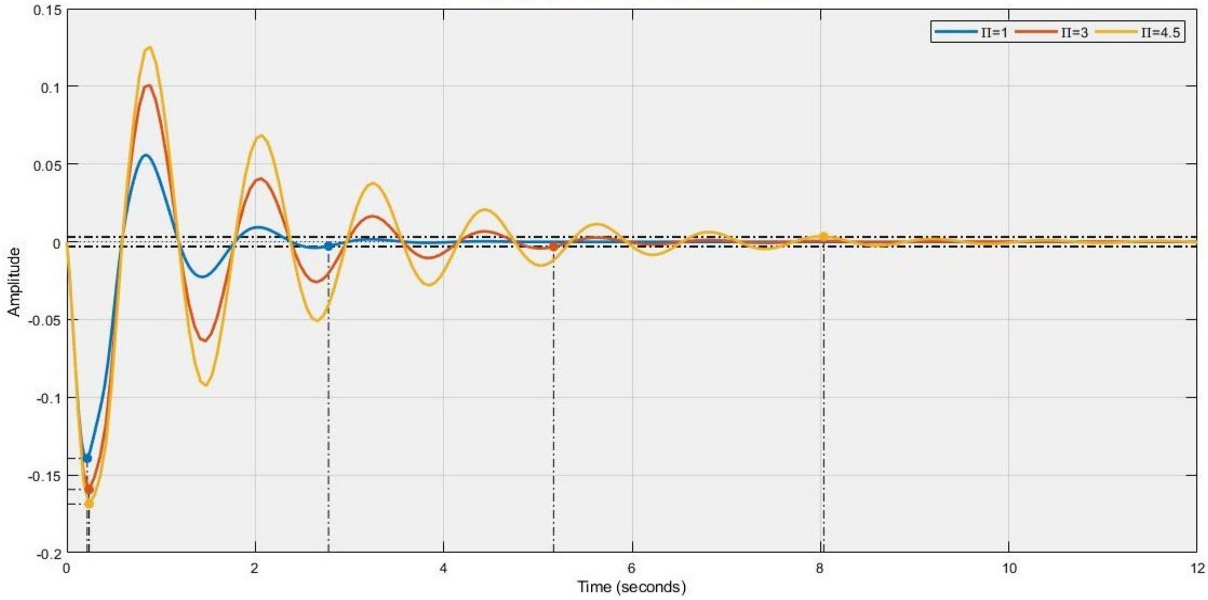
Figure 3. Displacement response curve at different α.



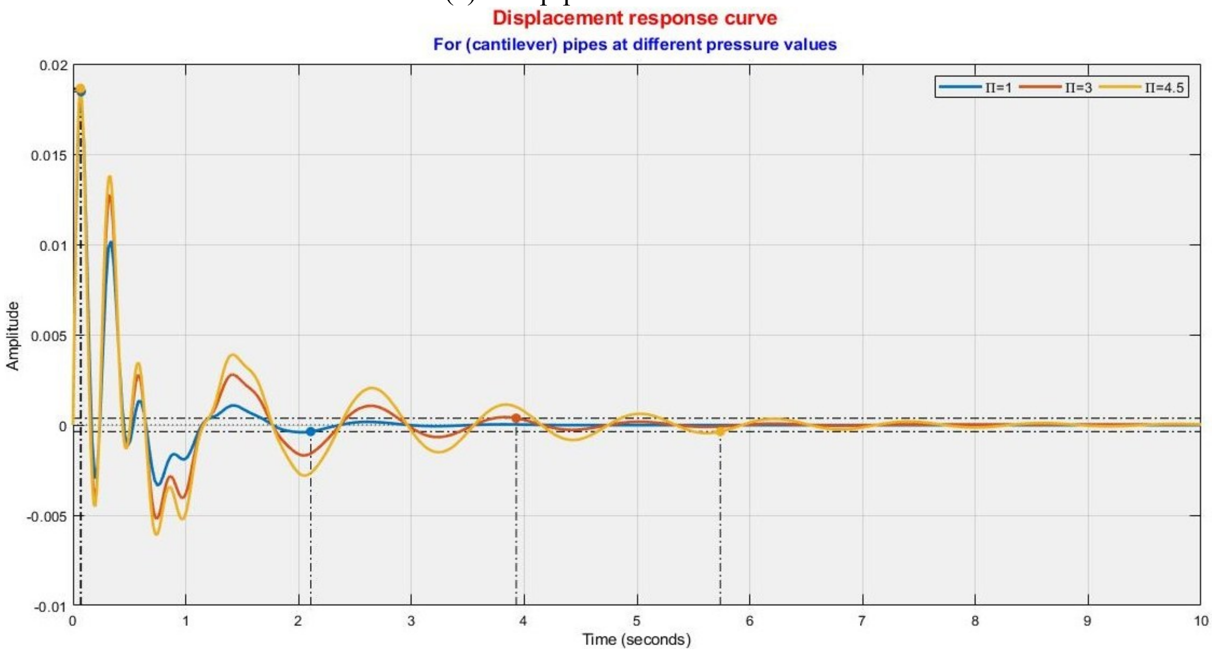
(a) P-P pipes at different Π.



(b) C-P pipes at different Π.



(c) C-C pipes at different Π.



(d) Cantilever pipes at different Π.

Figure 4. Displacement response curve at different Π.

Note all cases increase the stability time when the pressure increases with the increase in the amplitude, and note that there is a significant disorder at the cantilever pipe but less disturbance when reducing pressure and this applies to the rest of the cases. Conclude from the above that the increase in pressure leads to the failure of pipe systems at certain limits if the results are not calculated with high accuracy.

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 analytical solution for general equation of motion, by using state space control technique, is good tool can be used to calculated the dynamic pipe response induce vibration with various pipe boundary condition, for different parameters effect.

2. Found in all types of fixations of pipes 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. 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. The increase in pressure for critical speeds, 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.

References

[1] Y. Sun, Y. Gu, H. Xiong 'Studied and Their Application of Vibration Control Technologies' International Journal of Computer Science Issues, Vol. 10, Issue 02, No. 2, 2013.

[2] M. J. Jweeg, T. J. Ntayeesh 'Active Vibration Control Analysis of Cantilever Pipe Conveying Fluid Using Smart Material' Innovative Systems Design and Engineering, Vol. 06, No. 12, 2015.

[3] M. R. Ismail 'Evaluating The Dynamical Behavior and Stability of Pipes Conveying Fluid' Ph.D. Thesis Al-Nahrain University, College of Engineering, Mechanical Engineering Department, 2011.

[4] C. H. Yau, A. K. Bajaj, O. D. I. Nwokah 'Active Control of Chaotic Vibration in a Constrained Flexible Pipe Conveying Fluid' Journal of Fluids and Structures, Vol. 09, No. 01, pp. 99-122, 1995.

[5] Z. Jiao, P. Chen, Q. Hua, S. Wang 'Adaptive Vibration Active Control of Fluid Pressure Pulsations' Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering, Vol. 217, No. 04, 2003.

[6] Kazuhiko Hiramoto, Hitoshi Doki 'Simultaneous Optimal Design of Structural and Control Systems for Cantilevered Pipes Conveying Fluid' Journal of Sound and Vibration, Vol. 274, pp. 685-699, 2004.

[7] Yih-Hwang Lin, Chih-Liang Chu 'Active Modal Control of Timoshenko Pipes Conveying Fluid' Journal of the Chinese Institute of Engineers, Vol. 24, No. 01, 2011.

[8] Yih-Hwang Lin, Jui-Lung Chen, Chih-Liang Chu, Yau-Kun Tsai 'Numerical Stability Analysis for Independent Modal Vibration Suppression of A Fluid-Conveying Pipe using A Piezoelectric Inertia Actuator' Journal of Marine Science and Technology, Vol. 21, No. 05, pp. 545-550, 2013.

[9] Zhen Wang, Yudong Sun 'Experimental Research on Active Vibration Control of Pipe by Inertial Actuator and Adaptive Control' 21st International Congress on Sound and Vibration, 13-17 July, Beijing-China, 2014.

[10] Muhsin J. Jweeg , Thaier J. Ntayeesh 'Active Vibration Control of Pipes Conveying Fluid using Piezoelectric Actuators' Journal of Multidisciplinary Engineering Science and Technology, Vol. 02 No. 08, 2015.

[11] M. J. Jweeg, T. J. Ntayeesh 'Dynamic Analysis of Pipe Conveying Fluid using Analytical and Experimental Verification with the aid of Smart Material' International Journal of Science and Research, Vol. 04, No.12, pp.1594-1605, 2015.

[12] Yu Beiming, Chen Xiaoxin, Yabuno Hiroshi, Kiyotaka Yamashita 'Stabilization Control for Self-Excited Oscillation of Cantilevered Fluid-Conveying Pipe' 9th European Nonlinear Dynamics Conference, Budapest, Hungary, June 25-30, 2017.

[13] Xin Dekui, Nie Songlin 'Research on Vibration Control of Hydraulic System Pipeline' 21st World Multi-Conference on Systemic, Cybernetics and Informatics, 8-11 July, Orlando, Florida, USA, 2017.

[14] Muhannad Al-Waily, Maher A. R. Sadiq Al-Baghdadi, Rasha Hayder Al-Khayat 'Flow Velocity and Crack Angle Effect on Vibration and Flow Characterization for Pipe Induce Vibration' International Journal of Mechanical & Mechatronics Engineering IJMME-IJENS, Vol.17, No.5, 2017.

[15] M. A. Al-Baghdadi, K. K. Resan, Muhannad Al-Waily 'CFD investigation of the erosion severity in 3D flow elbow during crude Oil contaminated sand transportation' The 1st International Scientific Conference of Nanotechnology and Advanced Materials in Petroleum & Gas Industry, 3-4 May 2017, Engineering and Technology Journal, Vol. 35, Part A, No. 9, 2017.

[16] Rasha Hayder Al-Khayat, Maher A. R. Sadiq Al-Baghdadi, Ragad Aziz Neama, Muhannad Al-Waily 'Optimization CFD Study of Erosion in 3D Elbow During Transportation of Crude Oil Contaminated with Sand Particles, International Journal of Engineering & Technology, Vol. 07, No. 03, pp. 1420-1428, 2018.

[17] Muhsin J. Jweeg, Ali S. Hammood, Muhannad Al-Waily ‘A Suggested Analytical Solution of Isotropic Composite Plate with Crack Effect’ International Journal of Mechanical & Mechatronics Engineering IJMME / IJENS-Vol. 12, No. 5, 2012.

[18] Muhannad Al-Waily, Zaman Abud Almalik Abud Ali ‘A Suggested Analytical Solution of Powder Reinforcement Effect on Buckling Load for Isotropic Mat and Short Hyper Composite Materials Plate’ International Journal of Mechanical & Mechatronics Engineering IJMME-IJENS, Vol.15, No. 4, 2015.

[19] Muhannad Al-Waily ‘A Suggested Analytical Solution of Oblique Crack Effect on The Beam Vibration’ International Journal of Energy And Environment, Vol. 6, No. 3, 2015.

[20] Muhannad Al-Waily, Alaa Abdulzahra Deli ‘A suggested analytical solution of buckling investigation for beam with different crack depth and location effect’ International Journal of Energy and Environment, Vol. 7, No. 3, 2016.

[21] Muhsin J. Jweeg, Muhannad Al-Waily ‘Analytical Investigation of Time Dependent Tensional Loading of Stiffened and Un-Stiffened Composite Laminated Plates’ International Journal of Mechanical and Production Engineering Research and Development (IJMPERD), (TJPRC), Vol. 3, No. 4, 2013.

[22] Abdulkareem Abdulrazzaq Alhumdany, Muhannad Al-Waily, Mohammed Hussein Kadhim ‘Theoretical analysis of fundamental natural frequency with different boundary conditions of isotropic hyper composite plate’ International Journal of Energy and Environment, Vol. 7, No. 3, 2016.

[23] Mohsin Abdullah Al-Shammari, Muhannad Al-Waily ‘Analytical Investigation of Buckling Behavior of Honeycombs Sandwich Combined Plate Structure’ International Journal of Mechanical and Production Engineering Research and Development (IJMPERD), Vol. 08, No. 04, pp. 771-786, 2018.

[24] Diyaa H.J. Al-Zubaidi, Muhannad Al-Waily, Emad Q. Hussein, Maher A.R. Sadiq Al-Baghdadi ‘A Suggested Analytical Investigation of Heat Generation Inducing into Vibration Beam Subjected to Harmonic Loading’ International Journal of Energy and Environment, Vol. 09, No. 05, pp. 499-514, 2018.

[25] Muhannad Al-Waily ‘Analysis of Stiffened and Unstiffened Composite Plates Subjected to Time Dependent Loading’ M.Sc. Thesis, University of Kufa, Faculty of Engineering, Mechanical Engineering Department, 2005.

[26] Muhannad Al-Waily ‘Theoretical and Numerical Analysis Vibration Study of Isotropic Hyper Composite Plate Structural’ International Journal of Mechanical and Production Engineering Research and Development (IJMPERD), (TJPRC), Vol. 3, No. 5, 2013.

[27] Muhsin J. Jweeg, Muhannad Al-Waily ‘Dynamic Analysis of Stiffened and Unstiffened Composite Plates’ The Iraqi Journal For Mechanical And Material Engineering, Vol. 13, No. 4, 2013.

[28] Muhannad Al-Waily ‘Dynamic Analysis Investigation of Stiffened and Un-Stiffened Composite Laminated Plate Subjected to Transient Loading’ International Energy and Environment Foundation, 2015.

[29] Diyaa H. J. Al-Zubaidi, Muhannad Al-Waily, Emad Q. Hussein ‘Analytical Heat Generation Investigation for Forced Vibration Beam with Different Crack Characterizations Influence’ International Journal of Energy and Environment, Vol. 10, No. 01, pp. 33-48, 2019.

[30] Muhannad Al-Waily ‘Analytical and Numerical Thermal Buckling Analysis Investigation of Unidirectional and Woven Reinforcement Composite Plate Structural’ International Journal of Energy and Environment, Vol. 6, No. 2, 2015.

[31] Muhannad Al-Waily ‘Analytical and Numerical Buckling and Vibration Investigation of Isotropic and Orthotropic Hyper Composite Materials Structures’ International Energy and Environment Foundation, 2015.

[32] Saif M. Abbas, Ayad M. Takhakh, Mohsin Abdullah Al-Shammari, Muhannad Al-Waily ‘Manufacturing and Analysis of Ankle Disarticulation Prosthetic Socket (SYMES)’ International Journal of Mechanical Engineering and Technology (IJMET), Vol. 09, No. 07, pp. 560-569, 2018.