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Investigation into the Vibration Characteristics and Stability of a Welded Pipe Conveying Fluid

Nabeel K. Abid Al-Sahib [a], Adnan N. Jameel [b], Osamah F. Abdulateef [a,*]

[a] Al-Khawarizmi College of Engineering, University of Baghdad, Jaderyia, Baghdad, Iraq

[b] College of Engineering, University of Baghdad, Jaderyia, Baghdad, Iraq

Abstract

The stability of fluid conveying welded pipe is of practical importance because the welding induced residual stresses which affected on the vibration characteristics and stability. This paper deals with the vibration and stability of straight pipe made of ASTM-214-71 mild steel, conveying turbulent steady water with different velocities and boundary conditions, the pipe was welded on its mid-span by single pass fusion arc welding with an appropriate welding parameters. A new analytical model was derived to investigate the effects of residual stresses at girth welds of a pipe on the vibration characteristics and stability. The reaction components of the residual stresses at a single pass girth weld in a pipe was used in combination with a tensioned Euler-Bernoulli beam and plug flow model to investigate the effect of welding on the vibration characteristics of a pipe. The stability is studied employing a D-decomposition method. A finite element (FE) simulation was presented to evaluate velocity and pressure distributions in a single phase fluid flow. A coupled field fluid-structure analysis was then performed by transferring fluid forces, solid displacements, and velocities across the fluid-solid interface. A prestressed modal analysis was employed to determine the vibration characteristics of a welded pipe conveying fluid. Experimental work was carried out by built a rig which was mainly composed of a different boundary conditions welded pipes conveying fluid and provided with the necessary measurement equipments to fulfill the required investigations. It has been proven theoretically and experimentally that the residual stresses due to welding reducing natural frequencies for both clamped-clamped and clamped-pinned pipe conveying fluid. Also, we proved that for small fluid velocity (sub-critical), the clamped-clamped and clamped-pinned welded pipes conveying fluid are stable. For relatively high fluid velocities (super-critical) the clamped-pinned welded pipe loses stability by divergence.

Keywords: Stability; pipe conveying fluid; welding.

Nomenclature

- A_i Internal cross sectional area of the pipe (m2).
- C_j Amplitudes of vibrations.
- EI Bending stiffness of the pipe (Nm^2).
- $f_{int}(z, t)$ Force acting on the pipe from inside (N).
- g_j Wave numbers.
- K Dimensionless stiffness of the rotational stiffness.
- K_{rs} Stiffness of the rotational spring.
- L Length of the pipe (m).
- m Pipe and Fluid mass per unit length (Kg/m).
- m_f Fluid mass per unit length (Kg/m).
- m_p Pipe mass per unit length (Kg/m).
- P_i Hydrostatic pressure inside the pipe (N/m^2).
- T Axial force due to welding (N).
- $T_{eff.}$ Effective force (N).
- t Time (sec).
- U Fluid velocity (m/sec).
- V Non-dimensional fluid velocity.
- Ω Non-dimensional natural frequency.
- ω Circular frequency of motion (rad/sec).
- λ Eigenvalue of the characteristic equation.
- η Non-dimensional transverse displacement of the pipe y/L.
- ξ Non-dimensional z-coordinate along the pipe z/L.

1. Introduction

The fluid flow and pipes are interactive systems, and their interaction is dynamic. These systems are coupled by the force exerted on the pipe by the fluid. The fluid force causes the pipe to deform. As the pipe deforms it changes its orientation to the flow and the fluid force may change. Mathematical models are generated for the pipe and fluid, the dynamic interaction is described by nonlinear oscillator equations. The stability of fluid conveying welded pipes is of practical importance because the natural frequency of a pipe generally decreases with the increasing velocity of the fluid flow $\omega_1 = \omega_N\ [1-(U/U_c)^2]^{1/2}$, Where ω1 is the fundamental natural frequency of the pinned-pinned pipe conveying fluid, ω_N is the fundamental natural frequency of the pipe in the absence of fluid flow, U_c is the critical velocity of flow for static buckling, and U is the fluid velocity. In certain problems involving very high velocity flows through flexible thin-walled welded pipes, such as

* Corresponding author. ausamalanee@yahoo.com.

those used in the feed lines to rocket motors and water turbines, the decrease in natural frequency can be important. The pipe may become susceptible to resonance or fatigue failure if its natural frequency falls below certain limits. If the fluid velocity becomes large enough, the pipe can become unstable.

Many researchers have been carried out on the vibration of a pipe conveying fluid. Amabili, Pellicano, and Paidoussis [1] investigated the non-linear dynamic and stability of simply supported, circular cylindrical shells containing in-viscid incompressible fluid flow. Manabe, Tosaka, and Honma [2] discussed the dynamic stability of a flow conveying pipe with two lumped masses by using domain decomposition boundary element method. Amabili, Pellicano, and Paidoussis [3] investigated the response of a shell conveying fluid to harmonic excitation, in the spectral neighborhood of one of the lowest natural frequencies for different flow velocities. Yih-Hwang and Chih-Liang [4] studied the vibration control of Timoshenko pipes conveying fluid. Excessive vibration in this flow induced vibration problem was suppressed via an active feedback control scheme. Nawaf M. Bou-Rabee [5] examined the stability of a tubular cantilever conveying fluid in a multi-parameter space based on non-linear beam theory. Lee and Chung [6] presented a new non-linear model of a straight pipe conveying fluid for vibration analysis when the pipe is fixed at both ends. Using the Euler-Bernoulli beam theory and the non-linear Lagrange strain theory, from the extended Hamilton's principle the coupled non-linear equations of motion for the longitudinal and transverse displacements are derived. These equations of motion are discretized by using the Galerkin method. Reddy and Wang [7] derived equations of motion governing the deformation of fluid-conveying beams using the kinematic assumptions of the (a) Euler-Bernoulli and (b) Timoshinko beam theories. The formulation accounts for geometric nonlinearity in the Von Karman sense and contributions of fluid velocity to the kinetic energy as well as to the body forces. Finite element models of the resulting non-linear equations of motion were also presented. Kuiper and Metrikine [8] proofed analytically a stability of a clamped-pinned pipe conveying fluid at a low speed. A tensioned Euler-Bernoulli beam in combination with a plug flow model was used as a model. The stability was studied employing a D-decomposition method. Langre and Paidoussis [9] considered the stability of a thin flexible cylinder considered as a beam, when subjected to axial flow and fixed at the upstream end only. A linear stability analysis of transverse motion aims at determining the risk of flutter as a function of the governing control parameters such as the flow velocity or the length of the cylinder. Stability is analyzed applying a finite difference scheme in space to the equation of motion expressed in the frequency domain.

From these papers a considerable shortage of the investigation about the effect of residual stresses at welds in pipes on the vibration characteristics and stability of a pipe conveying fluid.

2. Vibration of A Welded Pipe Conveying Fluid

The welded pipe conveying fluid sketched in Figure. (1) is initially straight, stressed, and finite length. The following assumptions are considered in the analysis of the system under consideration [10]:

- Neglecting the effect of gravity, material damping, shear deformation and rotary inertia.
- The pipe considered to be horizontal.
- The pipe is inextensible.
- The lateral motion y (z, t) is small and of long wavelength as compared to the diameter of the pipe so that the Euler-Bernoulli theory is applicable for description of the pipe dynamic bending.
- Neglecting the velocity distribution through the cross-section of the pipe.

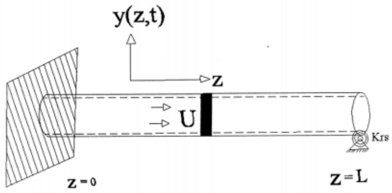


Figure 1. Welded pipe conveying fluid

Derivation of the equation of motion for pre-stressed single-span pipe conveying fluid as a function of the axial distance z and time t, based on beam theory is given by:

$$EI\frac{\partial^4 y}{\partial z^4} - T\frac{\partial^2 y}{\partial z^2} + m_p\frac{\partial^2 y}{\partial t^2} = f_{\text{int}}(z,t) \qquad (1)$$

Where: EI is the bending stiffness of the pipe, mp is the mass of the pipe per unit length, T is a prescribe axial force due to welding, f_{int} (z, t) is a force acting on the pipe from inside.

The axial force due to welding T is obtained by multiplying the axial bending stress obtained from welding on the inner and outer surfaces at any section inside or outside the tensile zone σ_{zb} by the cross sectional area of the pipe. The axial bending stresses are a function of the curvature [11]:

$$\sigma_{zb} = \mp\frac{Et}{2(1 \quad \upsilon^2)}\frac{d^2y}{dz^2} \qquad (2)$$

The internal fluid flow is approximated as a plug flow, i.e., as if it was an infinitely flexible rod traveling through the pipe, all points of the fluid having a velocity U relative to the pipe. This is a reasonable approximation for a fully developed turbulent flow profile [12]. The inertia force exerted by the internal plug flow on the pipe, can be written as:

$$f_{\text{int}} = -m_f\frac{d^2y}{dt^2}\Big|_{z=U.t} \qquad (3)$$

Where m_f is the fluid mass per unit length, U is the flow velocity. The total acceleration of the fluid mass can be decomposed into a local, carioles and centrifugal acceleration:

$$m_f \frac{d^2y}{dt^2}\bigg|_{z=U.t} = m_f\left\{\left\{\frac{d}{dt}\left(\left(\frac{\partial y}{\partial t}+\frac{\partial y}{\partial z}\frac{dz}{dt}\right)\right)\Big|_{z=U.t}\right\}\right\}$$
$$= m_f\left\{\left\{\frac{d}{dt}\left(\left(\frac{\partial y}{\partial t}+U\frac{\partial y}{\partial z}\right)\right)\Big|_{z=U.t}\right\}\right\}$$
$$= m_f\left\{\left\{\frac{\partial^2 y}{\partial t^2}+2U\frac{\partial^2 y}{\partial z\,\partial t}+U^2\frac{\partial^2 y}{\partial z^2}\right\}\right\} \quad (4)$$

The internal fluid causes a hydrostatic pressure on the pipe wall. This can easily be incorporated by changing the true axial force due to welding into a so-called effective force.

$$T_{eff.} = T - A_i P_i \quad (5)$$

Where A_i is the internal cross sectional area of the pipe, and P_i is the hydrostatic pressure inside the pipe. Combining Eqs. (1) ~ (5) the resulting equation of motion for a pre-stressed pipe conveying fluid can be written as:

$$EI\frac{\partial^4 y}{\partial z^4}+\left(m_f U^2 - T_{eff.}\right)\frac{\partial^2 y}{\partial z^2}+2m_f U\frac{\partial^2 y}{\partial z\,\partial t}+m\frac{\partial^2 y}{\partial t^2}=0 \quad (6)$$

In which $m=m_f+m_p$. The left end of the pipe is rigidly support, whereas the right end is assumed to allow no lateral displacement but to provide a restoring moment proportional to the rotation angle of the pipe. The clamped-clamped or clamped-pinned pipe is obtained from this formulation in the limit of the restoring rotational moment going to infinity or zero respectively. Thus, the boundary conditions at ends of the pipe are given as:

$$y(0,t)=0 \quad (7)$$
$$\frac{\partial y(0,t)}{\partial z}=0 \quad (8)$$
$$EI\frac{\partial^2 y(L,t)}{\partial z^2} = K_{rs}\frac{\partial y(L,t)}{\partial z} \quad (9)$$
$$y(L,t)=0 \quad (10)$$

Where K_{rs} is the stiffness of the rotational spring at the right end. Introducing the following dimensionless variables and parameters:

$$\eta = y/L,\ \xi = z/L,\ \tau = t\sqrt{EI/m}/L^2,$$
$$V = U\sqrt{m_f/EI},\ \ K = K_{rs}L/EI,$$
$$\alpha = L\sqrt{T_{eff.}m_f/(mEI)},\ \ \beta = L^2 T_{eff.}/EI.$$

The statement of the problem Eqs. (6) ~ (10) is rewritten as:

$$\frac{\partial^4\eta}{\partial\xi^4}+\beta\left(V^2-1\right)\frac{\partial^2\eta}{\partial\xi^2}+2\alpha V\frac{\partial^2\eta}{\partial\xi\partial\tau}+\frac{\partial^2\eta}{\partial\tau^2}=0 \quad (11)$$
$$\eta(0,\tau)=0 \quad (12)$$
$$\frac{\partial\eta(0,\tau)}{\partial\xi}=0 \quad (13)$$
$$\frac{\partial^2\eta(1,\tau)}{\partial^2\xi}=-K\frac{\partial\eta(1,\tau)}{\partial\xi} \quad (14)$$
$$\eta(1,\tau)=0 \quad (15)$$

To find the eigenvalues of the problem (11) ~ (15), which determine the stability of the pipe, the displacement $\eta(\xi,\tau)$ is to be sought in the following form:

$$\eta(\xi,\tau)=w(\xi)e^{\lambda\tau} \quad (16)$$

The pipe is unstable if one of the eigenvalues λ has a positive real part. Substituting Eq. (16) into the equation of motion Eq. (11), the following ordinary differential equation is obtained:

$$\frac{d^4w}{d\xi^4}+\beta\left(V^2-1\right)\frac{d^2w}{d\xi^2}+2\alpha V\lambda\frac{dw}{d\xi}+\lambda^2 w=0 \quad (17)$$

The general solution to this equation is given by:

$$w(\xi)=\sum_{j=1}^{4}C_j e^{ig_j\xi} \quad (18)$$

Where C_j are amplitudes of vibrations and g_j are the wave numbers. Substituting Eq. (18) into Eq. (17), a relationship is obtained between the wave numbers g_j and the eigenvalues λ:

$$g^4{}_j - \beta\left(V^2-1\right)g^2{}_j + 2\alpha V\lambda_i g_j + \lambda^2 = 0 \quad (19)$$

From these relationships four wave numbers g_j can be determined as functions of λ and the pipe parameters.

3. Natural Frequency

In order to evaluate the natural frequency for the system under consideration, substituting Eq. (17) into the boundary conditions Eqs. (12)~ (15). This yields the following system of four linear algebraic equations which can be written in matrix form as follows:

$$[H_{i,j}]\{C_j\}=0 \quad (20)$$

Where i,j=1,2,3,4. This system of equations has a non-trivial solution if, and only if its determinant Δ is equal to zero, which leads to the characteristic equation Δ =0. Trial and error procedure, the value of Ω (where λ=i Ω) that makes the determinant vanished can be found which will represent the non-dimension natural frequency. The non-dimensional frequency Ω related to the circular frequency of motion ω by the following equation:

$$\Omega=(m/EI)^{1/2}L^2\omega \quad (21)$$

4. Stability of A Welded Pipe Conveying Fluid

Linear structures containing flow inside them are found commonly in the wide range of applications. The macroscopic example is a pipeline for the oil industry. Intuitively, it seems that such a tube-like structure can be unstable and will rush about widely for the powerful flow inside it, but in the context of physics it is not obvious. The D-decomposition method, which is used in this paper, was developed for stability analyses of linear dynamical systems [13, 14].

The D-decomposition method has both advantages and disadvantages. The main advantage becomes apparent if a parametric study of stability should be performed. A D-decomposed plane of a parameter P contains information on stability of the system for all values of P, whereas the Argand diagram, for example, indicates the stability for a specific value of this parameter, only (if this parameter is not the fluid speed). Another advantage of the D-decomposition method is that it is equally applicable to pipes with any (linear) boundary conditions. For applying this method, no introduction of comparison functions is necessary (which takes quite an experience), like in the case of Galerkin method. The major disadvantage of the D-decomposition method is that having decomposed the plane of a parameter, it is still necessary to know the number of "unstable" eigenvalues for a specific, though arbitrary, value of this parameter. To find this number is not necessarily an easy task, although it can always be done by using the principle of the argument [15]. The other possibilities are to combine the D-decomposition method with the classical one or (the best, if possible) to use a degenerate value of the parameter, for which the system stability is known either from physical considerations or previous research.

To apply this method efficiently, the characteristic equation should contain a parameter that can be expressed explicitly. Characteristic equations of pipes conveying fluid do not necessarily contain such a parameter. By changing a boundary condition (extra mass, stiffness, dashpot, etc.), the boundary condition element enters the characteristic equation in such a way that it can be expressed explicitly as a function of all other system parameters. For the clamped-clamped and clamped-pinned pipe the rotational spring is introduced for this purpose at right end of the pipe, see Figure. (1) and Eq. (9). If the stiffness of this spring is zero, then the clamped-pinned pipe is retrieved. On the other hand, if this stiffness tends to infinity, the right end of the pipe becomes fixed. The characteristic equation for the pipe at hand consist of two parts, one proportional to the dimensionless rotational stiffness K and the other part independent of this stiffness:

$$\Delta = KA(\Omega) + B(\Omega) = 0 \tag{22}$$

Where A and B are functions of Ω obtained by using the relationship between the wave numbers g_j and the eigenvalues λ (Eq. (19)) and the expression $\lambda = i\Omega$. Expressing the rotational stiffness from Eq. (22), the rule is obtained for mapping the imaginary axis of the λ-plane on to the plane of the parameter K:

$$\boldsymbol{K} = -\frac{\boldsymbol{B}(\Omega)}{\boldsymbol{A}(\Omega)} \tag{23}$$

Note that in complex K-plane the positive part of the real axis only has physical meaning, since K is the stiffness of the rotational spring.

5. Finite Element Modeling Procedure

The FE analysis was carried out to calculate vibration characteristics of a welded pipes conveying fluid with different velocities and boundary conditions using a general purpose FE package ANSYS V9.0. The approach is divided into five parts: thermal analysis, coupled field thermal-structure analysis, computational fluid dynamics (CFD), coupled field fluid-structure analysis, and modal analysis.

5.1. Thermal analysis

A non- linear transient thermal analysis was conducted first to obtain the global temperature history generated during and after welding process(at the weld region). The basis for thermal analysis is a heat balance equation obtained from the principle of conservation of energy. The FE thermal solution employed a nonlinear (material properties depend on temperature) transient thermal analysis using two modes of heat transfer: conduction, and convection, to determine temperatures distributions that vary over time. The applied loads at the region of weld are function of time which described by divided the load-versus-time curve into load steps. For each load step, its need to specify both load and time values, along with other load step options such as stepped or ramped loads, automatic time stepping, etc. It's then written each load step to a file and solves all load steps together. SOLID90 element type is used, which it is a 3-D twenty nodes with a single degree of freedom, temperature, at each node. The 20-node elements have compatible temperature shapes and are well suited to model curved boundaries. It is applicable to a 3-D, steady-state or transient thermal analysis, Figure.(2) shows the model geometry and the mesh of the model

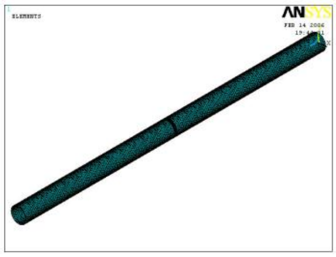


Figure 2. model geometry and the mesh

5.2. Coupled Field Thermal-Structure

A stress analysis was then developed with the temperatures obtained from the thermal analysis used as loading to the stress model. SOLID95 element type was used, which it can tolerate irregular shape without as much loss of accuracy. SOLID95 element has compatible displacement shapes and is well suited to model curved boundaries. It is defined by 20 nodes having six degree of freedom per node. The element may have any spatial orientation. SOLID95 has plasticity, creep, stress stiffening, large deflection, and large strain capabilities

5.3. Computational fluid dynamics (CFD)

The ansys flotran analysis used to solve 3-D flow and pressure distributions in a single phase viscous fluid. For the FLOTRAN CFD elements FLUID142, the velocities are obtained from the conservation of momentum principle, and the pressure is obtained from the conservation of mass principle. A segregated sequential solver algorithm is used; that is, the matrix system derived from the finite element discretization of the governing equation for each degree of freedom is solved separately. The flow problem is nonlinear and the governing equations are coupled together. The sequential solution of all the governing equations, combined with the update of any pressure dependent properties, constitutes a global iteration. The number of global iterations required to achieve a converged solution may vary considerably, depending on the size and stability of the problem.

5.4. Coupled field fluid-structure analysis

The coupled field fluid-structure analysis solved the equations for the fluid and solid domains independently of each other. It transfers fluid forces and solid displacements, velocities across the fluid-solid interface. The algorithm continues to loop through the solid and fluid analyses until convergence is reached for the time step (or until the maximum number of stagger iterations is reached). Convergence in the stagger loop is based on the quantities being transferred at the fluid-solid interface.

5.5. Modal analysis

We used modal analysis to determine the vibration characteristics (natural frequencies and mode shapes) of a welded pipe conveying fluid. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions. The procedure to do a prestressed modal analysis is essentially the same as a regular modal analysis, except that you first need to prestress the structure by doing a static analysis:

Build the model and obtain a static solution with prestress effects turned on from thermal-structure and fluid-structure analyses.

- Reenter the solution and obtain the modal solution, also with prestress effects turned on from thermal-structure and fluid-structure analyses.
- Expand the modes and review them in the postprocessor.

in a modal analysis, however, we use the term "expansion" to mean writing mode shapes to the results file. Figure 3 shows a flow chart for solution algorithm of modal analysis.

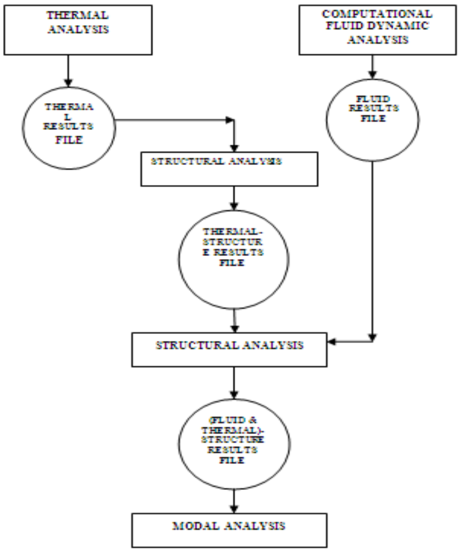


Figure 3. Modal solution algorithm for welded pipe conveying fluid

6. Experimental Work

The experimental procedures for measuring the natural frequencies were carried out with different steps:

- Pipes without flowing fluid

The vibration characteristics measurement was first done on a straight pipe 1m length, 50.8 mm diameter with clamped-pinned support without welding. The natural frequencies were measured by varying the shaker frequency slowly until a sharp increasing in the tube response that was displayed on the oscilloscope occurred. The same procedure was repeated by taking two straight pipe 0.5m length welded together by a single pass fusion arc welding with a current of 30A and voltage equal 460 volt using an electrode type E7010-G, to make a straight pipe 1m length with welding on its mid span. This was performed to investigate the effects of welding on the first few natural frequencies of a pipe.

- Pipes with flowing fluid

The vibration characteristic measurements were performed on straight pipe 1m length (without welding and with welding on its mid span) conveying water at a constant speed. To obtain a steady-state, we operate the centrifugal pump for enough time before began the measurements. This procedure was carried out with different flow velocities, which controlled by the valve. Another models were tested using the same previous procedures but with different lengths, diameters, welding position, and boundary conditions. figureure 4.Shown a schematic diagram for the experimental setup.

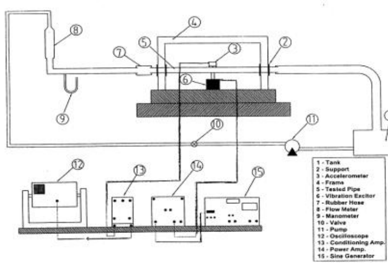


Figure 4. Schematic diagram of experimental setup

7. Results and Discussions

The pressure and velocity distributions in a pipe conveying fluid (water) are shown in figures. (5) and (6). The fluid is assumed water entered a pipe with a non-dimensional velocity (V=2) and a non-dimensional pressure (β=2), and Reynolds number 9.14*106. The outlet pressure is atmospheric pressure. The fluid forces and solid displacements, velocities was transferred across the fluid-solid interface to produce stresses distributed along the pipe geometry.

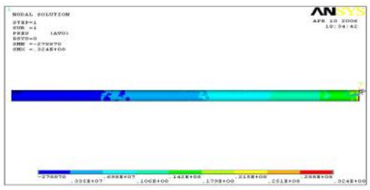
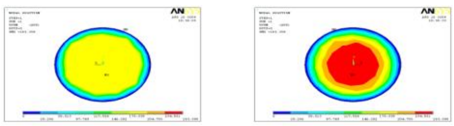


Figure 5. Pressure distribution along a pipe



(a) (b)

Figure 6. Velocity distribution at pipe sections

a- inlet of the pipe b- outlet of the pipe

Figures. (7) and (8) shows the displacement vector sum, Von Mises stress, and Von Mises total strain due to flowing fluid (water) for both clamped-clamped and clamped-pinned boundary conditions respectively. The results of clamped-clamped and clamped-pinned pipes obtained from free vibration analysis under steady flow are studied. The analytical and finite element analyses are applicable to determine natural frequencies and mode shape of a vibrating system before and after welding, and then compared with the experimental results. The sample of calculations was made on a single mild steel pipe with a (1 m) length, (50.8 mm) outer diameter, and a (1.5 mm) thickness, while the welded pipe was formed by joining two (0.5 m) pipes by single pass fusion arc welding with a current of 30 A and voltage equal 460 volt using an electrode type E7010-G to make a straight pipe 1m length with welding on its mid span; the welding procedure was modeled as a single pass in this analysis. Water was supplied to the pipe from an external reservoir; the parameters used in the calculations are listed in table (1). The analytical analysis is performed using Matlab V6.5 software to determine natural frequencies and mode shape. The program was developed to be used for any specified pipe dimensions, length, pipe material stiffness, different flow velocities, and welding specifications

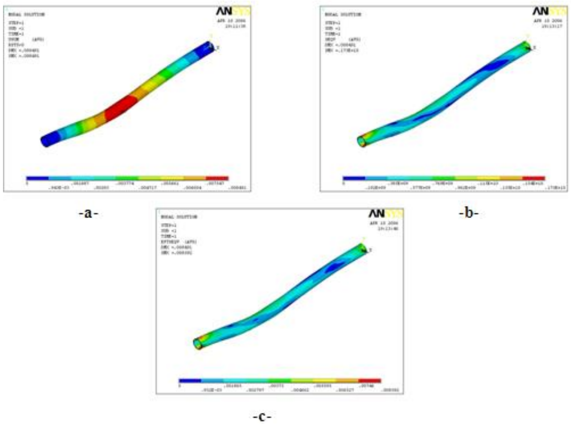


Figure. 7. clamped-clamped boundary conditions

a- Displacement vector sum b- Von Mises stress c- Von Mises total strain

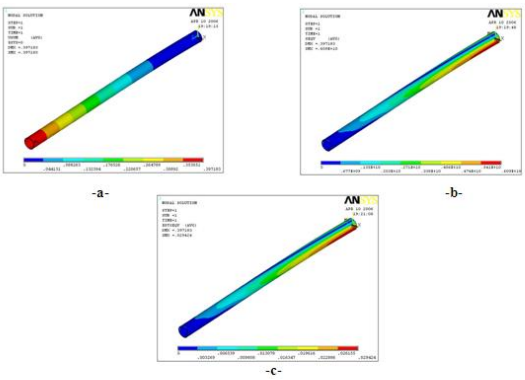


Figure 8.clamped-pinned boundary conditions

a- Displacement vector sum b- Von Mises stress c- Von Mises total strain

Table (1) Parameters used in the calculation

EI	1.4122*104	Nm
mf	1.795	Kg/m
m	3.608	Kg/m
R	25.4	mm
Teff	$3.0243*10^5$	N
L	1	m
ρf	1000	Kg/m^3
β	21.41	
α	3.264	
Reynolds no.	$9.14*10^6$	

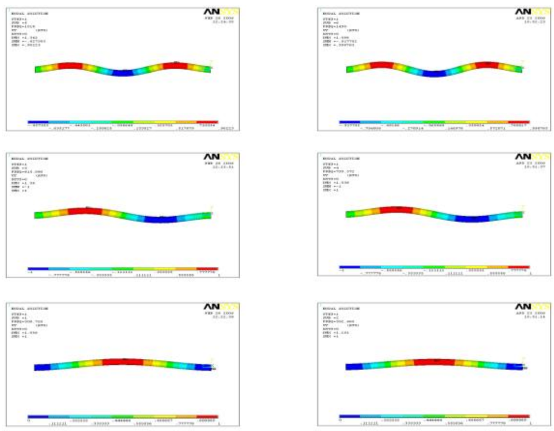
8. -Pipe Without Flowing Fluid

8.1. Clamped-Clamped Pipe

Table (2) shows the natural frequencies of a clamped-clamped pipe with and without welding obtained by finite element analysis and experiments, it shows good agreement between these results. We can see that the welding of a pipe causes reduction in the natural frequencies of it. This result is new and important to explain the effect of welding on the vibration characteristics of a pipe without flowing fluid. Figure. (9) Shows the mode shapes of a clamped-clamped pipe with and without welding.

Table (2) Natural frequencies (Hz) of a clamped-clamped pipe with and without welding

Mode no.	Without welding			With welding		
	FE analysis	Experimental work	% error	FE analysis	Experimental work	% error
1st	308.726	290	6.2	302.47	285	5.69
2nd	815.086	850	-4.1	799.37	840	-4.83
3rd	1518	1440	5.41	1489.6	1400	6.42



(a) *(b)*

Figure 9. Mode shapes of a clamped-clamped pipe with and without welding

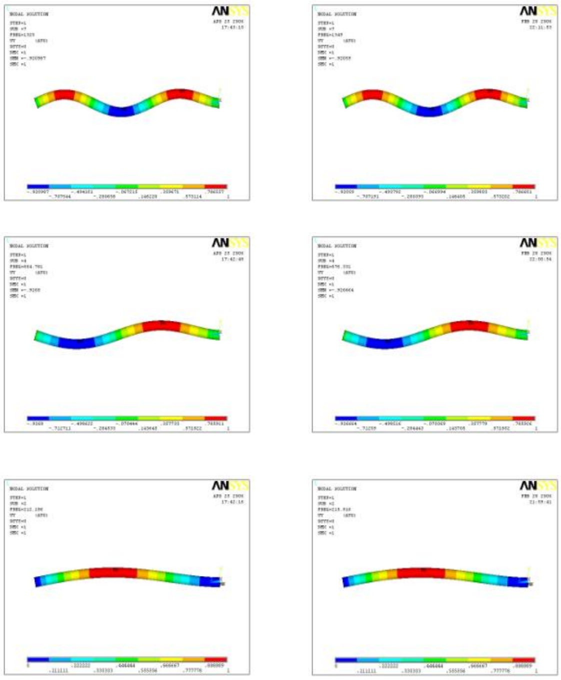
a- without welding b- with welding

8.2. Clamped-Pinned Pipe

Table (3) shows the natural frequencies of a clamped-pinned pipe with and without welding obtained by finite element analysis and experiments, it shows good agreement between these results. Also we can see that the welding of a pipe causes reduction in the natural frequencies of it. This result is also new and important to explain the effect of welding on the vibration characteristics of a pipe without flowing fluid. Figure. (10) Shows the mode shapes of a clamped-pinned pipe with and without welding**.**

Table (3) Natural frequencies (Hz) of a clamped-pinned pipe with and without welding

Mode no.	Without welding			With welding		
	FE analysis	Experimental work	% error	FE analysis	Experimental work	% error
1st	215.916	200	8.0	212.19	195	8.8
2nd	676.339	725	-6.71	664.78	710	-6.36
3rd	1349	1250	7.92	1325	1220	8.6



(a) (b)

Figure 10.Mode shapes of a clamped-pinned pipe with and without welding

a- without welding b- with welding

9. Pipe with Flowing Fluid

The fluid parameters (velocity, pressure, and mass ratio) have direct effects on the dynamic characteristics of the system in consideration. The effect of the fluid flow velocity and mass ratio will be discussed. In general the natural frequencies for steady flow decreases with increasing the fluid flow velocity as shown in figure. (11). If the velocity of the flow in the pipe equal zero, then the case will be a normal beam system and when the flow velocity equal the critical velocity the pipe bows out and buckles, because the forces required to make the fluid deform to the pipe curvature is greater than the stiffness of the pipe. Mathematically the buckling instability arises from the mixed derivative term in equation (5) which represent a forces imposed on the pipe by the flowing fluid that always 90o of phase with the displacement of the pipe, and always in phase with the velocity of the pipe. This force is essentially a negative damping mechanism which extracts energy from the fluid flow and inputs energy into the bending pipe to encourage initially, vibration, and ultimately buckling [16].

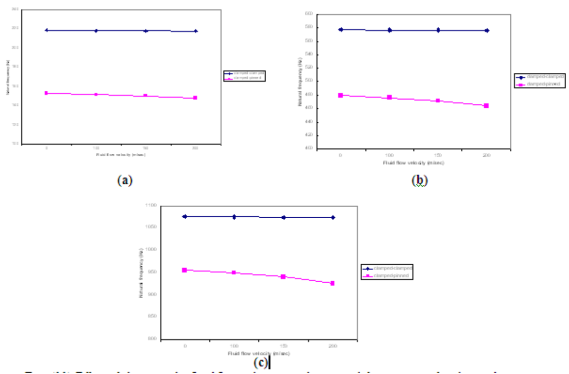


Figure 11. Effect of changing the fluid flow velocity on the natural frequencies of a clamped-clamped and clamped-pinned welded pipe

a-first mode b-second mode c-third mode

9.1. Super-Critical Velocity

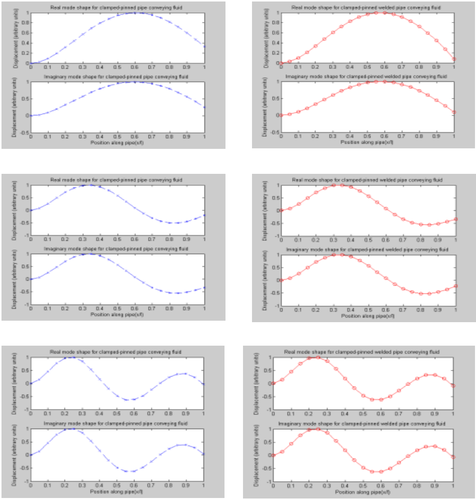
In this case, a comparison is made between the analytical and finite element, because the available pump in the market cannot gives high velocity (maximum fluid velocity = 5 m/sec).

9.2. Clamped-Clamped Pipe

Table (4) shows the natural frequencies of a clamped-clamped pipe with and without welding flowing water as a fluid with a non-dimensional velocity (V=2) obtained by analytical and finite element analysis; it shows good agreement between these results. Figure. (12) Shows the mode shapes of the pipe obtained analytically and by finite element analysis**.**

Table (4) Natural frequencies (Hz) of a clamped-clamped pipe with and without welding flowing water (V=2)

Mode no.	Without welding			With welding		
	Analytical analysis	FE analysis	% error	Analytical analysis	FE analysis	% error
1st	216.62	217.832	0.55	212.7	213.981	0.6
2nd	573.83	576.172	0.4	564.58	566.091	0.26
3rd	1069.3	1074	0.43	1050.55	1055	0.42



(a) (b)

Figure 12. Mode shapes of a clamped-clamped pipe with flowing water (V=2)

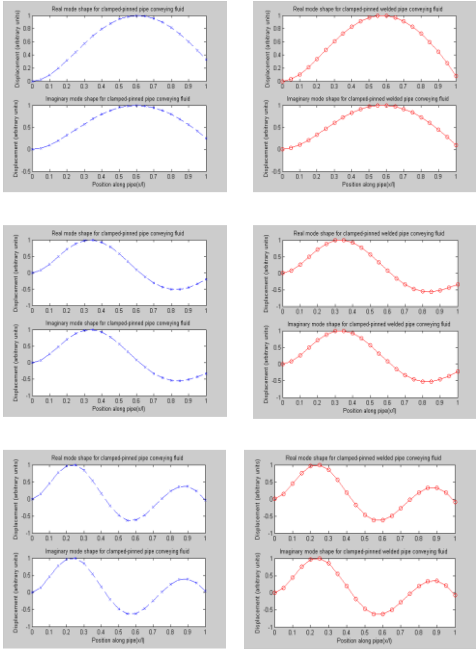
a- without weld b- with weld

10. Clamped-Pinned Pipe

Table (5) shows the natural frequencies of a clamped-pinned pipe with and without welding, flowing water with a non-dimensional velocity (V=2) obtained by analytical and finite element analysis; it shows good agreement between these results. Figure. (13) Shows the mode shapes of the pipe obtained analytically and finite element analysis. We can see that the welding of a pipe leading to reduce the natural frequencies of a pipe conveying fluid for both clamped-clamped and clamped-pinned boundary conditions. This result is new and important to explain the effect of welding on the vibration characteristics of a pipe conveying fluid**.**

Table (5) Natural frequencies (Hz) of a clamped-pinned pipe with and without welding flowing water (V=2)

Mode no.	Without welding			With welding		
	Analytical analysis	FE analysis	% error	Analytical analysis	FE analysis	% error
1st	148.9	149.182	0.19	147.68	148.224	0.36
2nd	465.04	467.672	0.56	462.82	465.05	0.48
3rd	930.77	935.59	0.51	925.9	932.323	0.69



(a) **(b)**

Figure 13.Mode shapes of a clamped-pinned pipe with flowing water (V=2)

a- without weld b- with weld

11. Sub-Critical Velocity

For low water velocity (V=0.5), the results are summarized in table (6) for clamped-clamped and table (7) for clamped-pinned pipe conveying water (with and without welding). The comparison was made between analytical, finite element, and experimental, it shows good agreement between them and the error percentage is within ranges shown in literatures.

Table (6) Natural frequencies (Hz) of a clamped-clamped pipe with flowing water with a non-dimensional (V=0.5)

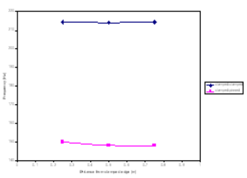
Mode no.	Without welding			With welding		
	analytical	finite element	experimental	analytical	finite element	experimental
1st (error)	218.3 (6.48%)	219.65 (7.14%)	052	217.4 (8.7%)	218.7 (9.35%)	002
2nd (error)	575.21 (4.58%)	577.3 %)4.96(	055	573.26 (6.15%)	574.13 (6.32%)	540
3rd (error)	1070 (-6.7%)	1075.8 (-6.2%)	1147	1060 (-7.01%)	1064.8 (-6.59%)	1140

Table (7) Natural frequencies (Hz) of a clamped-pinned pipe with flowing water with a non-dimensional (V=0.5)

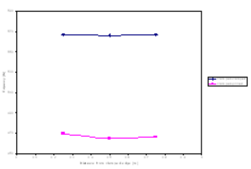
Mode no.	Without welding			With welding		
	analytical	finite element	experimental	analytical	finite element	experimental
1st (error)	151.8 (4.68%)	152.9 (5.44%)	145	149.6 (3.17%)	150.38 (3.7%)	145
2nd (error)	477 (6.0%)	479 (6.4%)	450	465 (5.68%)	470.99 (7.0%)	440
3rd (error)	952.7 (-4.53%)	954.3 (-4.37%)	998	934.36 (-4.94%)	939.21 (-4.45%)	983

12. Effect of Weld Position on the Vibration Characteristics

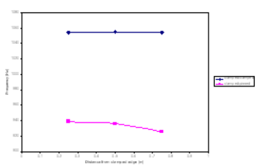
The effect of weld position on the natural frequencies was examined using finite element for clamped-clamped and clamped-pinned welded pipe conveying water with a steady non-dimensional velocity (V=2). The result shows that as the position of welding far from the stationary edge, the natural frequencies reduced more. Figure. (14) Shows the effect of weld position clearly.



-a-



-b-



-c-

Figure 14. Effect of weld position on natural frequencies of welded pipe conveying fluid

a- first mode b- second mode c- third mode

13. Stability of A Welded Pipe Conveying

The stability of welded pipe conveying fluid using the D-decomposition method via Matlab V6.5 software. The result of D-decomposition of the k-plane is shown in Figure. (15) for two fluid speeds V=0.5 (sub-critical) and V=2 (super-critical), respectively. Figure. (15-a) shows that in the sub-critical case, V=0.5, the D-decomposition lines do not cross the positive part of the real axis. This implies that the number of "unstable" eigenvalues does not depend on the rotational stiffness of the pipe's support (the physically admissible values of this stiffness are real and positive). In particular, this implies that the number of unstable roots for the clamped-clamped pipe ($k \rightarrow \infty$) is the same as for the clamped-pinned pipe (k=0). Thus, since it is well known [12] that the clamped-clamped and clamped-pinned pipes are stable at small fluid speeds; we can conclude that the clamped-clamped and clamped-pinned welded pipes are stable at these speeds as well.

The main difference between Figures. (15-a) and (15-b) is that in the former figureure the D-decomposition curves cross the positive part of the real axis. This implies that there is a critical stiffness of the rotational spring (the coordinate of the crossing point) below which the pipe is for sure unstable. Thus, if the flow is super-critical (V=2), the clamped-pinned welded pipe is unstable. The critical velocity corresponds to the situation, when the D-decomposition curves cross the origin of the k-plane.

14. Conclusions

Theoretical analysis and experimental work are used to determine the effect of welding on the vibration characteristics and stability of a welded pipe conveying fluid. 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.
- The welding of a pipe leading to reduce the natural frequencies of a pipe conveying fluid for both clamped-clamped and clamped-pinned boundary conditions.
- The natural frequencies affected by the position of weld. They are reduces more as the position of welding far from the stationary edge for both clamped-clamped and clamped-pinned boundary conditions.
- The welded clamped-clamped and clamped-pinned pipes are stable for small fluid velocities (sub-critical), and the clamped-pinned pipe lose stability by divergence at relatively high fluid velocities (super-critical).

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