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Theoretical Investigation of Flow Induced Vibration for Polymer Matrix Composite Angled Pipe

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ABSTRACT
In this research, the vibration analysis and stability of angled pipe has been investigated; two straight pipes made from composite material connected by elbow; with fixed-fixed ends conveying fluid has been used, using three dimensional finite element model. The characteristic matrices consisting of stiffness, inertia and damping terms which derived by finite element method and the effect of the internal flow velocity, axial force and Coriolis force are considered. Some parameters that influence the dynamic properties have been studied such as pipe angle, diameter ratio and wall pipe thickness. It was found that the composite material increase the properties of the frame pipes; such as fluid velocity and frame frequency; and this increasing continuously with increase the volume fraction of composite material, and the increase in frame angle led to decrease the critical flow velocity of the frame pipes, also the increase in the length pipes ratio decrease the critical velocity of flowing. Finally, the frame frequency decreases as the frame angle increase.

KEYWORDS: curved pipe, composite materials, conveying fluid, angle of spring location, straight pipe, critical velocity, coriolis force.

INTRODUCTION

The dynamic behavior of pipes transporting fluid has been a subject of increasing research interest. As the pipes are used in many application of the industrial fields, an analysis of flow-induced vibration for the pipes carrying fluid include one of the important subjects in structural dynamics applications, such as nuclear fields and aerospace; such as vibrations of heat exchangers, liquid-fuel rocket piping, and nuclear reactor coolant channels. It is well known that piping systems may subject to variation and flutter types of instabilities created by fluid-structure interaction [1].

Reference [2] used the Generalized differential quadrature procedure as a numerical technique to solve this problem. The differential governing equation was convert into a discrete system of mathematical expressions. The stability of the system was reduced to an eigenvalue problem. The convert of trouble instability from one eigenvalue branch to another was investigated and discussed. The loop mass ratios, at which the transmit of the eigenvalue branches related to disturbance, were specified. Reference [3] studied the natural frequency analysis of fluid conveying pipeline with different boundary conditions using eliminated element-Galerkin method. They explained that Galerkin method give good results with ineffective error. While reference [4] used Differential Transformation Method (DTM) to analyze the free vibration problem of pipe conveying fluid. They demonstrated that the DTM method gave good precision as compared with the analytical methods. And reference [5] studied numerically pinned-clamped and clamped-pinned pipes conveying fluid. He found that to predict the dynamical behavior of the clamped-pinned pipe, even 8 significant-figure accuracy was not good enough. The imaginary part of the complex Eigen frequency seemed to be negative, implying unstable behavior

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for any flow velocity greater than zero. Reference [6] analyzed the in-plane and out-of-plane movements of a semi-circular pipe conveying fluid. The mathematical expressions of in-plane and out-of-plane motions were derived according to the extended Hamilton's principle. The derived equations of motion were described by applying the Galerkin procedure. Linearized equations around the equilibrium position were obtained, and then the dynamic characteristics of the pipe were investigated.

Reference [7] presented a general mathematical equations of the dynamic behavior of a clamped-pinned pipeline carrying fluid. The system stability studied by means of the eigenvalues of a Hamiltonian linear system. From this method, characteristic expressions based on material constants had been developed as inequalities, and some specific materials were represented as study cases to compare the mathematical description suggested with the results obtained from specialized software as ANSYS, in order to support the results. Where a simulation of the dynamic system using ANSYS for some different materials used in real cases, such as PVC, Polyethylene, Concrete, Steel and Aluminum, in order to validate the results obtained analytically.

It concluded that the dynamic and stability of pipes conveying fluid not only depends on the boundary conditions, but it is also strongly important the material of the pipe and the pressure produced by the fluid.

In this paper, it will analyze the dynamic vibration due to fluid flow in, a fixed –fixed end conditions, angled pipeline, composed of two straight pipes connected by elbow, considered in three-dimensional space and analyzed by finite element method, also using glass fiber/epoxy composite material instead of metal pipe will discussed to reduce the vibration.

2. Theoretical Equations:

The differential equation of motion in three dimensional coordinate's vibration of a pipe conveying a moving fluid was used to take into account the presence of motion constraints.

The differential equation of motion for three dimensions vibration of a frame pipe carrying a moving fluid is [8]:

$$EI_y \frac{\partial^4 w}{\partial x^4} + EI_z \frac{\partial^4 v}{\partial x^4} + \{MU^2 + (pA_i - Fx)\}[\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 v}{\partial x^2}] + 2MU[\frac{\partial^2 w}{\partial x \partial t} + \frac{\partial^2 v}{\partial x \partial t}]$$
$$+ (m+M)[\frac{\partial^2 w}{\partial t^2} + \frac{\partial^2 v}{\partial t^2}] + EA_p \frac{\partial^2 u}{\partial x^2} + GJ \frac{\partial^2 \theta}{\partial x^2} + (m+M)[\frac{\partial^2 u}{\partial t^2} + r^2 \frac{\partial^2 \theta}{\partial t^2}] = 0 \quad (1)$$

Where u, w, and v are the coordinate axes in the directions of x, y, and z respectively, θ is the pipe angular displacement, E and G are the pipe axial and shear moduli of elasticity respectively, I_y and I_z are moment of inertia of the pipe in y and z directions respectively, J is the pipe polar moment of inertia, m = mass of the pipe per unit length, conveying fluid of mass per unit length (M), U is the steady mean flow velocity of fluid with respect to pipe, x is the coordinate measured along the pipe length, Fx is the tension force in the pipe, A_p, Ai, r are the cross section pipe area, internal pipe area (fluid area), and pipe radius of gyration respectively. **Fig. 1** shows a simple representation, in three dimensional space, of the problem within hand which is consist of two pipe joint at their junction by an elbow.

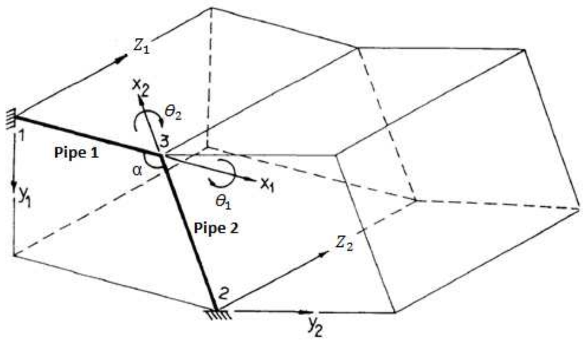


Fig. 1: Angled Pipeline Model.

3. Finite Element Description:

In **Fig. 2**, i and j represent the node points of finite element of length (L). Each node point has 6 degrees of freedom which consist of 3 transitional displacements u, w, v and 3 rotational displacements θ_x , θ_y , θ_z. Therefore the finite element has the total 12 degrees of freedom.

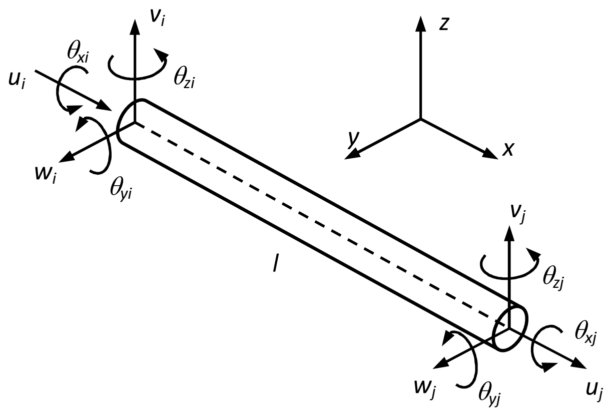


Fig. 2: Degree of freedom of pipe element.

The element displacement vector for a pipe element in space can be written as:

$$\{q\}^T = \{u_1 \quad w_1 \quad v_1 \quad \theta_{x1} \quad \theta_{y1} \quad \theta_{z1} \quad u_2 \quad w_2 \quad v_2 \quad \theta_{x2} \quad \theta_{y2} \quad \theta_{z2}\} \tag{2}$$

For transverse (flexural) pipe vibration, the shape functions are [9]

$$\left.\begin{aligned} N_1 &= \frac{1}{l^3}(2x^3 - 3lx^2 + l^3) \\ N_2 &= \frac{1}{l^2}(x^3 - 2lx^2 + l^2 x) \\ N_3 &= \frac{1}{l^3}(3lx^2 - 2x^3) \\ N_4 &= \frac{1}{l^2}(x^3 - lx^2) \end{aligned}\right\} \tag{3}$$

While, for axial and torsional vibrations the shape functions are

$$\left.\begin{aligned} N_5 &= (1 - \frac{x}{l}) \\ N_6 &= (\frac{x}{l}) \end{aligned}\right\} \tag{4}$$

The displacement models can be expressed as [8]:
-For transverse displacements

$$w(x) = v(x) = \sum_{k=1}^{4} N_k(x) q_k \tag{5}$$

-For axial and torsional displacements

$$u(x) = \theta_x(x) = \sum_{k=5}^{6} N_k(x) q_k \tag{6}$$

The kinetic energy of pipe element is equal to

$$KE = \underbrace{\tfrac{1}{2}\int_0^l (m+M)(\frac{\partial w}{\partial t})^2 dx}_{transverse,\ y-dir.} + \underbrace{\tfrac{1}{2}\int_0^l (m+M)(\frac{\partial v}{\partial t})^2 dx}_{transverse,\ z-dir.} + \underbrace{\tfrac{1}{2}\int_0^l (m+M)(\frac{\partial u}{\partial t})^2 dx}_{axial} + \underbrace{\tfrac{1}{2}\int_0^l (m+M) r^2 (\frac{\partial \theta_x}{\partial t})^2 dx}_{torsional} \tag{7}$$

After finding the individual mass matrices for each term in the above equation, the total arranged mass matrix, according to displacement vector [eq. 2], of pipe element in space has the following form

$$\hat{m}=\frac{(m+M)l}{420}\begin{bmatrix} 140 & 0 & 0 & 0 & 0 & 0 & 70 & 0 & 0 & 0 & 0 & 0 \\ 0 & 156 & 0 & 0 & 0 & 22l & 0 & 54 & 0 & 0 & 0 & -13l \\ 0 & 0 & 156 & 0 & -22l^2 & 0 & 0 & 0 & 54 & 0 & 13l & 0 \\ 0 & 0 & 0 & 140r^2 & 0 & 0 & 0 & 0 & 0 & 70r^2 & 0 & 0 \\ 0 & 0 & -22l^2 & 0 & 4l^2 & 0 & 0 & 0 & -13l & 0 & -3l^2 & 0 \\ 0 & 22l & 0 & 0 & 0 & 4l^2 & 0 & 13l & 0 & 0 & 0 & -3l^2 \\ 70 & 0 & 0 & 0 & 0 & 0 & 140 & 0 & 0 & 0 & 0 & 0 \\ 0 & 54 & 0 & 0 & 0 & 13l & 0 & 156 & 0 & 0 & 0 & -22l \\ 0 & 0 & 54 & 0 & -13l & 0 & 0 & 0 & 156 & 0 & 22l & 0 \\ 0 & 0 & 0 & 70r^2 & 0 & 0 & 0 & 0 & 0 & 140r^2 & 0 & 0 \\ 0 & 0 & 13l & 0 & -3l^2 & 0 & 0 & 0 & 22l & 0 & 4l^2 & 0 \\ 0 & -13l & 0 & 0 & 0 & -3l^2 & 0 & -22l & 0 & 0 & 0 & 4l^2 \end{bmatrix} \quad (8)$$

While the potential energy is

$$PE=\underbrace{\tfrac{1}{2}\int_0^l EI_y\left(\frac{\partial^2 w}{\partial x^2}\right)^2 dx}_{transverse,\quad y-dir.}+\underbrace{\tfrac{1}{2}\int_0^l EI_z\left(\frac{\partial^2 v}{\partial x^2}\right)^2 dx}_{transverse,\quad z-dir.}+\underbrace{\tfrac{1}{2}\int_0^l EA_p\left(\frac{\partial u}{\partial x}\right)^2 dx}_{axial}+\underbrace{\tfrac{1}{2}\int_0^l GJ\left(\frac{\partial \theta_x}{\partial x}\right)^2 dx}_{torsional} \quad (9)$$

This will lead to the following symmetry pipe stiffness matrix

$$k_1=\begin{bmatrix} \frac{EA_p}{l} & 0 & 0 & 0 & 0 & 0 & -\frac{EA_p}{l} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{12EI_z}{l^3} & 0 & 0 & 0 & \frac{6EI_z}{l^2} & 0 & -\frac{12EI_z}{l^3} & 0 & 0 & 0 & \frac{6EI_z}{l^2} \\ 0 & 0 & \frac{12EI_y}{l^3} & 0 & -\frac{6EI_y}{l^2} & 0 & 0 & 0 & -\frac{12EI_y}{l^3} & 0 & -\frac{6EI_y}{l^2} & 0 \\ 0 & 0 & 0 & \frac{GJ}{l} & 0 & 0 & 0 & 0 & 0 & -\frac{GJ}{l} & 0 & 0 \\ 0 & 0 & -\frac{6EI_y}{l^2} & 0 & \frac{4EI_y}{l} & 0 & 0 & 0 & \frac{6EI_y}{l^2} & 0 & \frac{2EI_y}{l} & 0 \\ 0 & \frac{6EI_z}{l^2} & 0 & 0 & 0 & \frac{4EI_z}{l} & 0 & -\frac{6EI_z}{l^2} & 0 & 0 & 0 & \frac{2EI_z}{l} \\ -\frac{EA_p}{l} & 0 & 0 & 0 & 0 & 0 & \frac{EA_p}{l} & 0 & 0 & 0 & 0 & 0 \\ 0 & -\frac{12EI_z}{l^3} & 0 & 0 & 0 & -\frac{6EI_z}{l^2} & 0 & \frac{12EI_z}{l^3} & 0 & 0 & 0 & -\frac{6EI_z}{l^2} \\ 0 & 0 & -\frac{12EI_y}{l^3} & 0 & \frac{6EI_y}{l^2} & 0 & 0 & 0 & \frac{12EI_y}{l^3} & 0 & \frac{6EI_y}{l^2} & 0 \\ 0 & 0 & 0 & -\frac{GJ}{l} & 0 & 0 & 0 & 0 & 0 & \frac{GJ}{l} & 0 & 0 \\ 0 & 0 & -\frac{6EI_y}{l^2} & 0 & \frac{2EI_y}{l} & 0 & 0 & 0 & \frac{6EI_y}{l^2} & 0 & \frac{4EI_y}{l} & 0 \\ 0 & \frac{6EI_z}{l^2} & 0 & 0 & 0 & \frac{2EI_z}{l} & 0 & -\frac{6EI_z}{l^2} & 0 & 0 & 0 & \frac{4EI_z}{l} \end{bmatrix} \quad (10)$$

The term ($\{MU^2+(pA_i-Fx)\}[(\frac{\partial^2 w}{\partial x^2})+(\frac{\partial^2 v}{\partial x^2})]$) in eq.1 has a potential energy that can be represented as [9]:

$$PE=\tfrac{1}{2}\int_0^l \{MU^2+(pA_i-Fx)\}\left(\frac{\partial w}{\partial x}\right)\left(\frac{\partial w}{\partial x}\right)dx+\tfrac{1}{2}\int_0^l \{MU^2+(pA_i-Fx)\}\left(\frac{\partial v}{\partial x}\right)\left(\frac{\partial v}{\partial x}\right)dx \quad (11)$$

Leading to symmetry matrix (k_2), which contains the force per unit length (stiffness unit) that conforms the fluid to the pipe (weakening effect) besides the axial tension force (stiffening effect).

$$k_2 = \frac{MU^2 + pA_i - Fx}{30l}\begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 36 & 0 & 0 & 0 & 3l & 0 & -36 & 0 & 0 & 0 & 3l \\ 0 & 0 & 36 & 0 & 3l & 0 & 0 & 0 & -36 & 0 & 3l & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 3l & 0 & 4l^2 & 0 & 0 & 0 & -3l & 0 & -l^2 & 0 \\ 0 & 3l & 0 & 0 & 0 & 4l^2 & 0 & -3l & 0 & 0 & 0 & -l^2 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -36 & 0 & 0 & 0 & -3l & 0 & 36 & 0 & 0 & 0 & -3l \\ 0 & 0 & -36 & 0 & -3l & 0 & 0 & 0 & 36 & 0 & -3l & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 3l & 0 & -l^2 & 0 & 0 & 0 & -3l & 0 & 4l^2 & 0 \\ 0 & 3l & 0 & 0 & 0 & -l^2 & 0 & -3l & 0 & 0 & 0 & 4l^2 \end{bmatrix} \quad (12)$$

Here, we will call the above matrix as a contradictory matrix because it contains two opposite component effects. Where F_x is an axial tension force that caused by the change in fluid's momentum and pressure in a pipe bend (elbow) [10]. **Fig.(3)** shows the induced axial tension forces in the pipe bend.

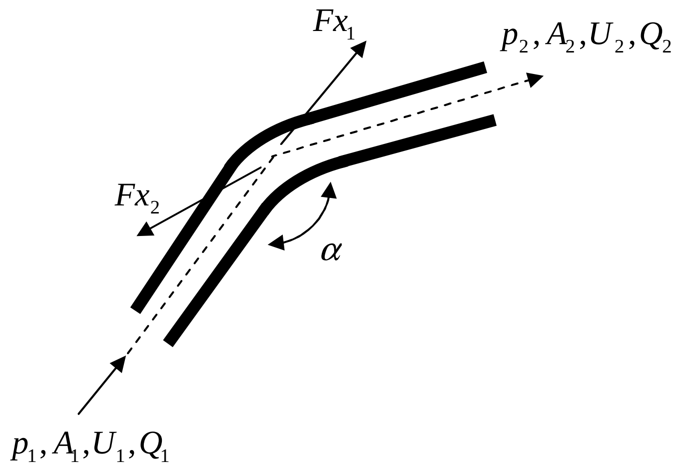


Fig. 3: Tension forces in pipe bend.

From **Fig.(3)**, the axial tension forces in pipe bend are equal to [11]:

$$Fx_1 = p_1 A_1 + p_2 A_2 \cos(\alpha) + \rho_{fluid} Q[U_1 + U_2 \cos(\alpha)] \quad (13)$$

and

$$Fx_2 = R\{\frac{\sin(\psi)}{\sin(\alpha)}\} \quad (14)$$

Where

$$R = [Fx_1^2 + Fy^2]^{\frac{1}{2}}$$

$$Fy = p_2 A_2 \sin(\alpha) + \rho_{fluid} U_2 Q \sin(\alpha)$$

$$Q = U_1 A_1 = U_2 A_2$$

$$p_2 = p_1 + \rho_{fluid} \{\frac{(U_1^2 - U_2^2)}{2}\}$$

$$\psi = \tan^{-1}\{\frac{\sin(\alpha)}{1 + \cos(\alpha)}\}$$

From the mathematical formulation presented above, it is clear that the overall stiffness is composed of two parts, namely the contradictory and pipe structural stiffness matrices.

The term ($2MU[\frac{\partial^2 w}{\partial x \partial t}+\frac{\partial^2 v}{\partial x \partial t}]$) in eq.(1) is the inertial force associated with the Coriolis acceleration arising from the fluid flows with velocity U relative to the pipe. This term has a dissipation energy which is equal to

$$DE = \tfrac{1}{2}\int_0^l 2MU(\frac{\partial w}{\partial x})(\frac{\partial w}{\partial t})dx + \tfrac{1}{2}\int_0^l 2MU(\frac{\partial v}{\partial x})(\frac{\partial v}{\partial t})dx \qquad (15)$$

This gives a skew-symmetry damping matrix

$$C=\frac{MU}{30}\begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -30 & 0 & 0 & 0 & -6l & 0 & -30 & 0 & 0 & 0 & 6l \\ 0 & 0 & -30 & 0 & -6l & 0 & 0 & 0 & -30 & 0 & 6l & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 6l & 0 & 0 & 0 & 0 & 0 & -6l & 0 & l^2 & 0 \\ 0 & 6l & 0 & 0 & 0 & 0 & 0 & -6l & 0 & 0 & 0 & l^2 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 30 & 0 & 0 & 0 & 6l & 0 & 30 & 0 & 0 & 0 & -6l \\ 0 & 0 & 30 & 0 & 6l & 0 & 0 & 0 & 30 & 0 & -6l & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & -6l & 0 & -l^2 & 0 & 0 & 0 & 6l & 0 & 0 & 0 \\ 0 & -6l & 0 & 0 & 0 & -l^2 & 0 & 6l & 0 & 0 & 0 & 0 \end{bmatrix} \qquad (16)$$

It can be seen that the 12 * 12 element matrices given in eqs. (8), (10), (12) and (16) are with respect to the local xyz coordinate system. Since the nodal displacements for the angled pipe are in different local coordinates, thus it must transform the local coordinate to global coordinate system. The transformation matrix, λ, can be identified as [9]

$$\lambda=\begin{bmatrix} l_{ox} & m_{ox} & n_{ox} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ l_{oy} & m_{oy} & n_{oy} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ l_{oz} & m_{oz} & n_{oz} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & l_{ox} & m_{ox} & n_{ox} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & l_{oy} & m_{oy} & n_{oy} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & l_{oz} & m_{oz} & n_{oz} & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & l_{ox} & m_{ox} & n_{ox} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & l_{oy} & m_{oy} & n_{oy} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & l_{oz} & m_{oz} & n_{oz} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & l_{ox} & m_{ox} & n_{ox} \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & l_{oy} & m_{oy} & n_{oy} \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & l_{oz} & m_{oz} & n_{oz} \end{bmatrix} \qquad (17)$$

Here, l_{ox}, m_{ox}, and n_{ox}denote the direction cosines of the x-axis; l_{oy}, ml_{oy}, and n_{oy}represent the direction cosines of the y-axis; and l_{oz}, m_{oz}, and n_{oz}indicate the direction cosines of the z-axis with respect to the global axes. This leads to the following global element matrices [8]: $[\hat{m}]_{Global} = [\lambda]^T[\hat{m}][\lambda]$ (18)

$$[k_{overall}]_{Global} = [\lambda]^T[k_{overall}][\lambda] \qquad (19)$$

$$[C]_{Global} = [\lambda]^T[C][\lambda] \qquad (20)$$

Dynamic Analysis:

The standard equation of motion in the finite element form is [8]:

$$[\hat{m}]_{Global}\{\ddot{q}\}+[C]_{Global}\{\dot{q}\}+[k_{overall}]_{Global}\{q\}=\{0\} \quad (21)$$

Where $k_{overall}=(k_{1,overall}-k_{2,overall})$

Since the above equation has a damping term with skew-symmetric characteristic, thus the solution of eigenvalues problem should be executed to the characteristic matrix [Ω], which is equal to [11]

$$[\Omega]=\begin{bmatrix} [0] & [I] \\ -[\hat{m}]^{-1}[k_{overall}] & -[\hat{m}]^{-1}[C] \end{bmatrix}_{Global} \quad (22)$$

The solution of eigenvalue problem yields complex roots. The imaginary part of these roots represents the natural frequencies of damped system. The real part indicates the rate of decay of the free vibration [8].

Composite material properties:

The mechanical properties (Young's modulus, Shear modulus, Poisson's ratio and density) of the composite system used in this study are theoretically determined depending on the rule of mixture.

To define the fiber volume fraction V_f and the matrix volume fraction V_m consider a composite consisting of fiber and matrix so.[12]

$$V_m=\frac{\upsilon_m}{\upsilon_c}=\frac{A_m}{Ac}\ .....(a) \quad and \quad V_f+V_m=1\ .....(b) \quad (23)$$

The mass fraction (weight fraction) of the fibers (M_f) and the matrix (M_m) are defined as:

$$M_f=\frac{w_f}{w_c}\ ...(a)\ ,\ M_m=\frac{w_m}{w_c}\ ...(b)\ and\ \ M_f+M_m=1\ ...(c) \quad (24)$$

From the definition of the density of a single material in terms of the fiber and matrix volume fractions:

$$M_m=\frac{\rho_m}{\rho_c}V_m\ ...(a)\ \ and\ \ M_f=\frac{\rho_f}{\rho_c}V_f\ ...(b) \quad (25)$$

In terms of individual constituent properties, the mass fractions and volume fractions are related by:

$$M_f=\frac{V_f}{V_f+\frac{\rho_m}{\rho_f}(1-V_f)}...(a),\ \ V_f=\frac{M_f}{M_f+(1-M_f)\frac{\rho_f}{\rho_m}}\ ...(b) \quad (26)$$

The density of the composite ρ_c in terms of the constituents' weight fractions and densities can be defined as:[13,14]

$$\rho_c=\frac{1}{\frac{M_f}{\rho_f}+\frac{M_m}{\rho_m}} \quad (27)$$

To define elastic properties of composite materials as follows:

$$Ec=E_fV_f+E_mV_m \quad (28)$$

$$Gc=\frac{G_mG_f}{G_f-V_f(G_f-V_m)} \quad (29)$$

$$V_c=V_f\nu_f+V_m\nu_m \quad (30)$$

In this study, E-Glass fiber is used; it was obtained in the form of chopped strand mats, with epoxy polymer as matrix.

Different values of volume fraction (30, 40 and 50 %) are used, as listed in the table (1).

Table 1: Mechanical properties of epoxy reinforced by chopped glass fiber:

Volume fraction %		E (GPa)	G (GPa)	Density (g/cm^3)	Poisson ratio
Vf1	30	23.7	9.38	1.673	0.365
Vf2	40	30.6	12.04	1.808	0.348
Vf3	50	37.5	14.7	1.944	0.34

RESULTS AND DISCUSSION

Fig. (4)shows the effect of frame angle on the critical velocity of fluid. It is clear that any increase in frame angle will lead to decrease the critical flow velocity, i.e. accelerate the instability of structure. This behavior is mainly caused by decreasing the axial tension forces in the pipe bend, which play a stiffening role, with increasing the frame angle.

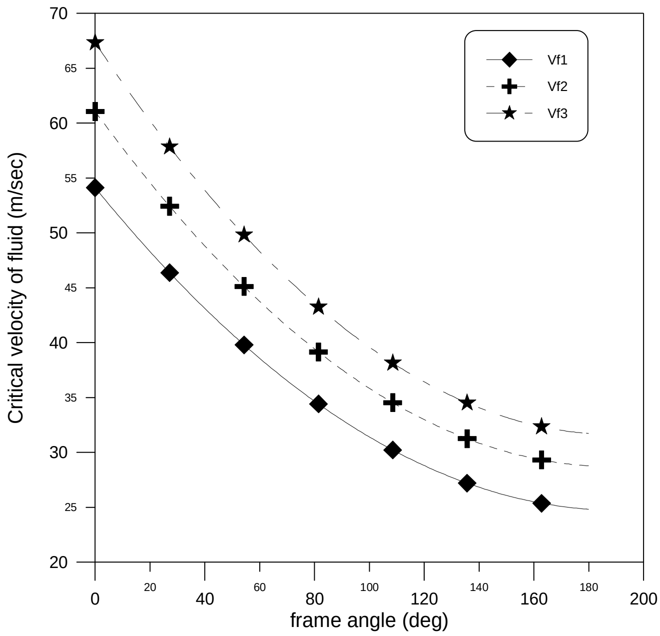
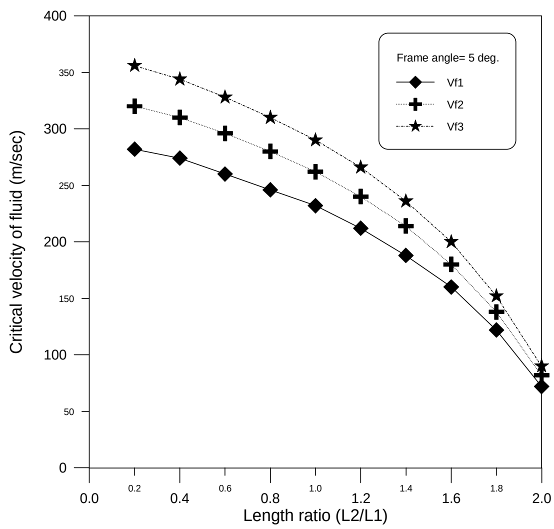
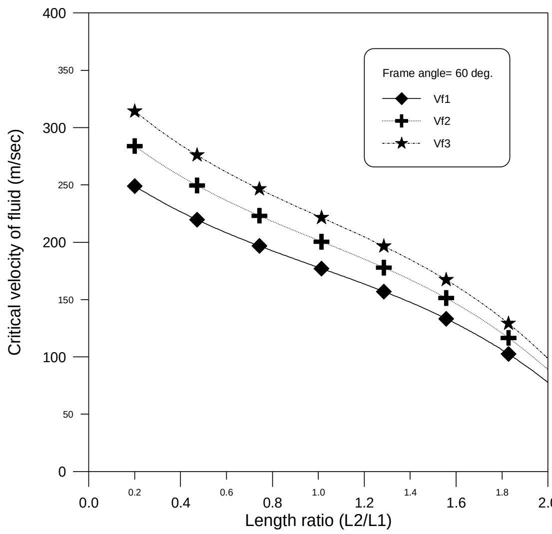


Fig. 4: Effect of frame angle on critical velocity of fluid. Pipe lengths $L_1= L_2=$ 1m, fluid density is (1000 kg/m3), thicknesses of pipes are (0.001 m) and outer diameters of the pipes are (0.03 m).

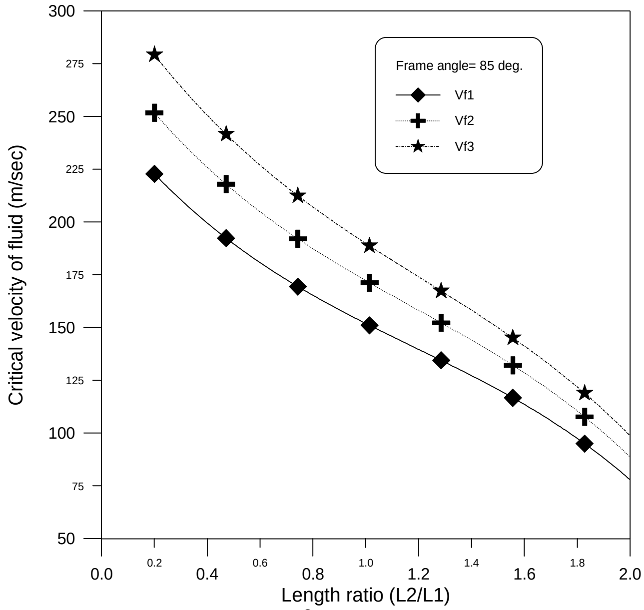
Fig. (5) presents the effect of pipe lengths ratio on the critical flow velocity with different frame angles. The increase of the pipe lengths ratio leads to decreases the critical velocity of flowing. The main reason of this behavior is that the fame structure becomes heavier in weight and weaker in its stiffness with increasing frame length ratio. Moreover, the curves behave to converge when the value of this ratio reaching one and will continue in compactness after this value. Also, Fig. 5 shows that as the volume fraction (Vf) of the composite material of the pipe increase the critical velocity of the fluid will increase too, this is attributed to the increasing the modulus of the elasticity as the volume fraction increase, as shown in table (1).



a) frame angle of (5^o).



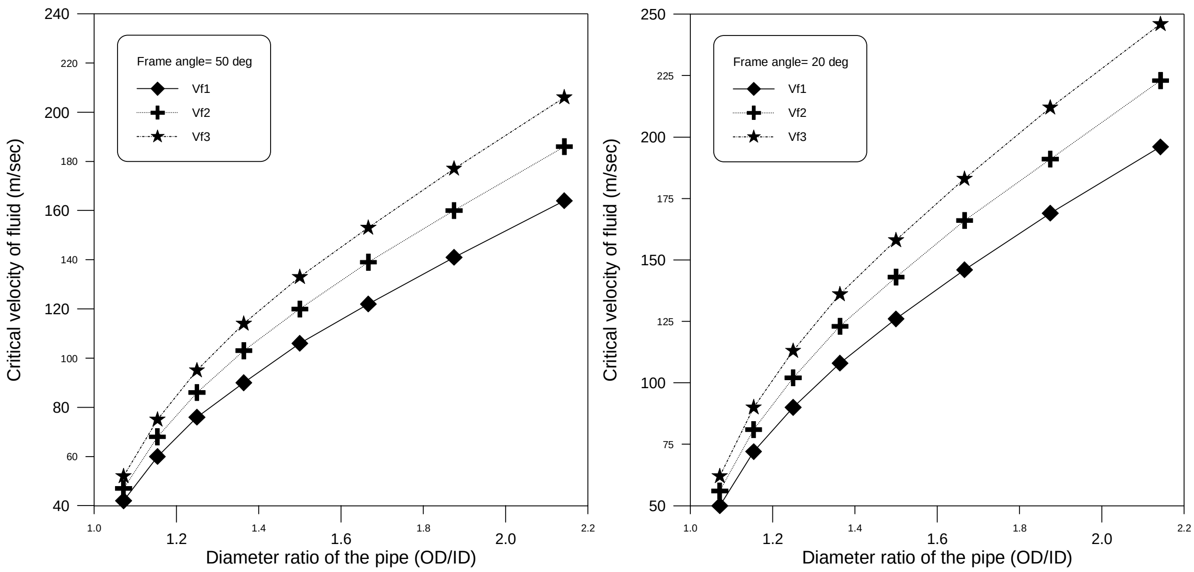
b) frame angle of (60^o).



(c) frame angle of (85^o).

Fig. 5: Effect of frame lengths ratio on the critical fluid velocity with different frame angles. Pipe length $L_1$is (1 m), fluid density is (1000 kg/m3), thicknesses of pipes are (0.008 m) and outer diameters of the pipes are (0.05 m).

Fig. (6), shows the relation between diameters ratio and critical inlet velocity at different frame angles. In this figure, the critical inlet velocities will rise continuously with increasing diameters ratio. It is well known that decreasing the inlet pipe diameter leads to maximize the fluid velocity through the pipe (positive effect on the frame dynamic characteristics).



a) frame angle of (20^o). b) frame angle of (50^o).

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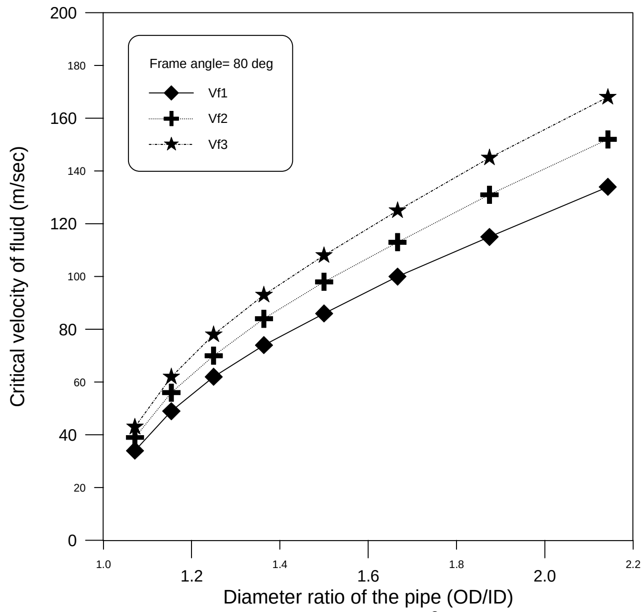
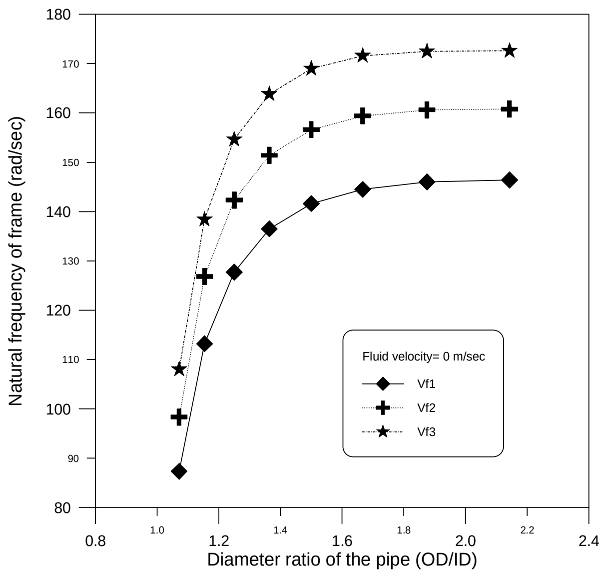


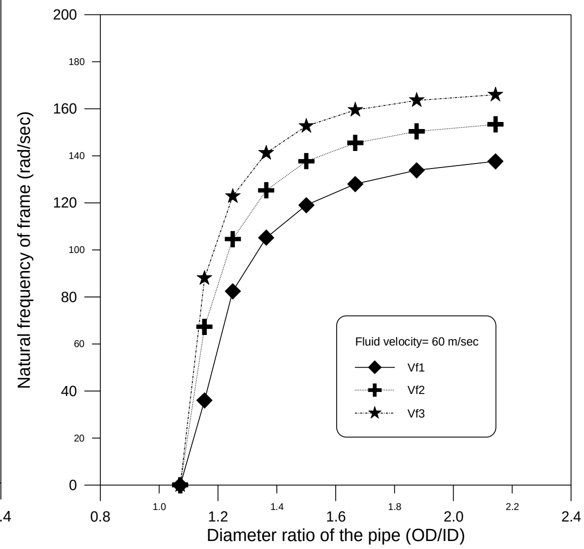
c) frame angle of (80°).

Fig. 6: Effect of diameters ratio on the critical fluid velocity at different frame angles. Pipe lengths L_1= L_2= 1 m, fluid density is (1000 kg/m3) and outer diameter of pipes are (0.03 m).

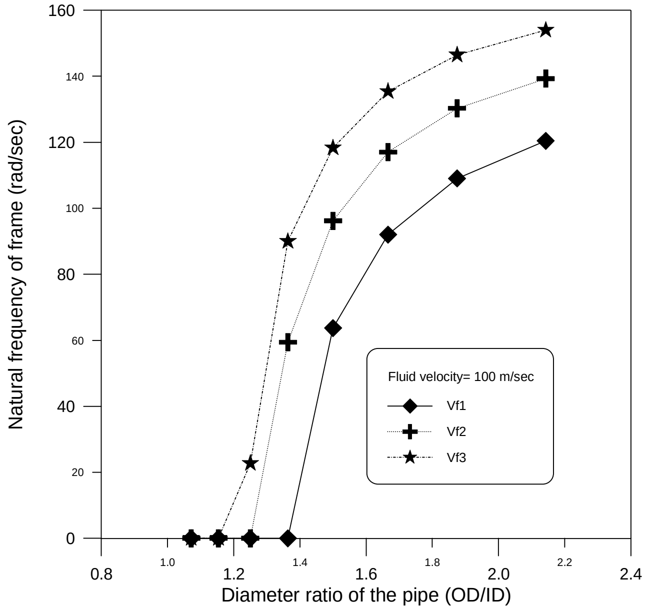
Fig. (7) presents the effect of diameters ratio on the frame frequency at different inlet fluid velocities. When the pipe thickness is relatively increase the frequency of the frame will increase. Increasing pipe thickness to certain values gives the best ever frame frequency for each fluid velocity. This behavior is mainly caused by increasing pipe stiffness and weight with thickness increasing. This means that there is an optimum pipe thickness for each of the flowing velocity that gives best ever frame frequency. The combined effects of these two parameters will control the dynamic behavior of the angled pipeline structures. In other words, Fig. 7 shows that the natural frequency of the frame increases as the volume fraction of the composite material of the pipe increase, this is due to the increasing of the elastic modulus.



a) fluid velocity of (0 m/sec).



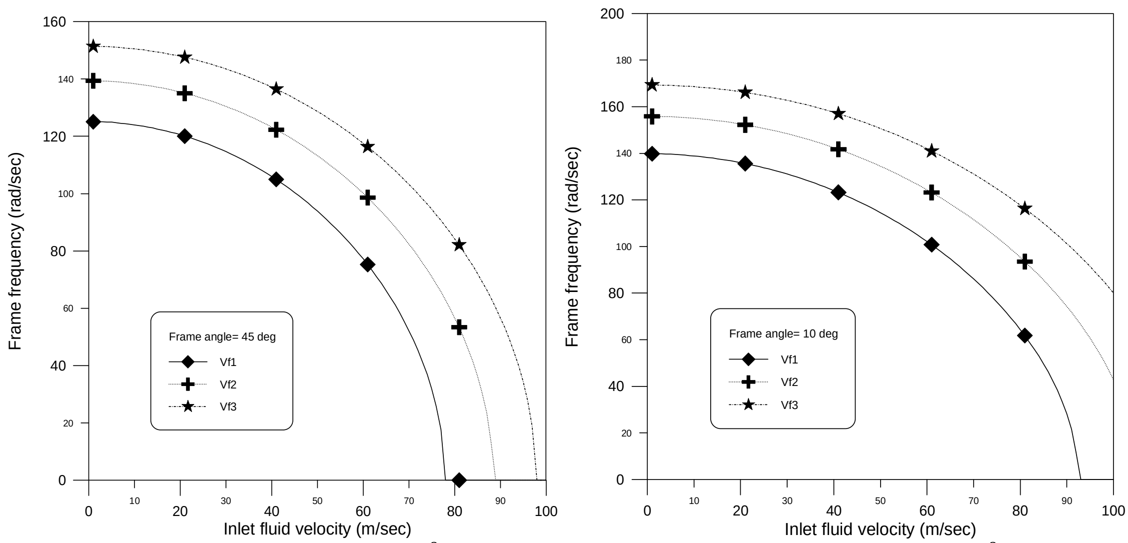
b) fluid velocity of (60 m/sec).



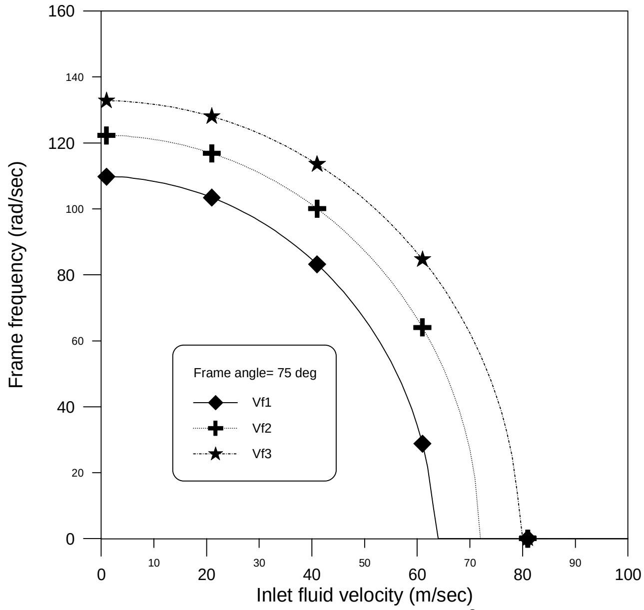
c) fluid velocity of (100 m/sec).

Fig. 7: Effect of diameters ratio on the frame frequency at different inlet fluid velocities. Pipe lengths $L_1 = L_2 = 1$ m, fluid density is (1000 kg/m3), frame angle (40^o) and outer diameter of pipes are (0.03 m).

Fig. (8) shows the effect of inlet fluid velocity on the fundamental frame frequency with different frame angles. In general, the increasing of the inlet fluid velocity leads to decreasing the frame frequency. Further increasing in inlet fluid velocity leads to drop in frequency. This behavior can be depicted as an alternative one. Where, at relatively low inlet velocity, the force conforms fluid (weakening effect) seems to be larger than the axial tension forces (stiffening effect) generated in the pipe frame. The values of axial tension forces components are very sensitive to frame angle and pipes sections (pipe diameters ratio). Also it is shown from Fig. 8, as the frame angle increase the frame frequency will decrease, this because that the pipe stiffness will decrease.



a) frame angle of (10^o). b) frame angle of (45^o).



c) frame angle of (75^{o}).

Fig. 8: Effect of inlet fluid velocity on the frame frequency at different frame angles.

Pipe lengths L_1= L_2= 1 m, fluid density is (1000 kg/m3), thicknesses of pipes are (0.001 m) and outer diameter of pipes are (0.03 m).

Conclusions:

The following conclusions are obtained:

1) The composite material will increase the properties of the frame pipes; such as fluid velocity and frame frequency; and this increasing continuously with increase the volume fraction of composite material.
2) The increase in frame angle will lead to decrease the critical flow velocity of the frame pipes.
3) The increase in the length pipes ratio leads to decrease the critical velocity of flowing.
4) The critical inlet velocities will rise continuously with increasing the diameters ratio of the pipe.
5) When the pipe thickness is relatively increase, the frequency of the frame pipe will increase.
6) The frame frequency tends to reduce with increasing the inlet fluid velocity.
7) The frame frequency decreases as the frame angle increase.

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