VIBRATION ANAYLYSIS AND INFLUENCING FACTOR STUTY OF HELICOPTER HYDRAULIC PIPELINNE

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Abstract

The helicopter hydraulic pipelines are distributed across various structural sections of the helicopter, which are characterized by“long, messy, scattered and miscellaneous”. Frequent failures of the hydraulic pipeline leakage caused fundamentally by harsh vibration environments seriously affect the system stability and the safety of the helicopter. Therefore, it is essential to analyze the vibration characteristics of the helicopter hydraulic pipelines. In this paper,based on the fluid wave equation and Timoshenko beam model, the 14-equation FSI model of straight pipe and elbow pipe unit is established and modified. In this model, the effects of quasi-steady friction term, unsteady Zielke friction term and Brunone friction term on axial vibration are all considered. In addition, taking general straight pipes as study object, the finite element simulation model is built in ANSYS software, to obtain the vibration modes varing materials, length, support number, fluid velocity respectively on Ansys software. The results show that different materials and fluid velocity have Virtually no effect on the natural frequency, however a significant impact on deformation. On the other hand, length and support number play a very noteworthy role on the adjustment of natural frequency., which provide important references for the design and optimization of helicopter hydraulic pipelines.

Keywords: helicopter hydraulic pipeline; fluid-structure interaction; natural frequency; mode of vibration;

1. Introduction

The helicopter hydraulic system is used for flight control operations, and the stability of the hydraulic pipeline system directly affects the flight safety of the helicopter. According to statistics, hydraulic conduits are the subsystem with the highest failure rate among aircraft components[1]. Regarding hydraulic tubes failure analysis, extensive research has been conducted in military enterprises. Statistics on aircraft conduit ruptures indicate that most ruptures occur after severe vibration, corrosion, or mechanical damage[2] . As the vibration receptor, the helicopter hydraulic pipeline system is subjected to vibration sources including hydraulic pump pressure pulsations, fuselage vibrations, and fluid-structure interaction vibrations. Therefore, the analysis of the vibration characteristics of the helicopter hydraulic pipeline system is particularly important.

Regarding the study of fluid-structure coupling vibration in hydraulic tube system, many scholars have conducted research from different perspectives and emphases. For example, Jiao Zongxia considered the influence of friction terms on conduit vibration response and proposed a method of dual-coordinate transformation to solve the problem of poor accuracy in coupling calculations [3]. Ashley [4]derived the vibration differential equations under fluid-structure coupling using beam models in 1950 and subsequently studied the vibration problems of conduits based on this foundation. Harold [5]validated the mathematical model through theoretical and experimental comparisons. In the theory of vibration of fluid-filled cylindrical shells, research groups led by Fuller [6] used the Dolmell-Mushtari shell equation to establish the equations of motion for fluid-filled straight pipes and formulated the dispersion equations in matrix form. The introduction of shell models has expanded applicability and improved computational accuracy, but has also introduced certain difficulties, leading scholars to extensively explore the application scope of various shell equations and the use of simplified models. For thick-walled pipelines with a large ratio of length to diameter (L/D, where L is the length of the pipe and D is the outer diameter), beam models can be used for analysis; while thin or short pipe systems use shell models. Budny[7] conducted in-depth research on the dynamic characteristics of simple pipelines subjected to water hammer forces. Lesmez[8], Tentarelli[9], Brown [10], Tentarelli, and De Jong [11] conducted in-depth studies on pipeline systems subjected to periodic force excitation, with De Jong focusing on L-shaped pipes, while Tentarelli, Brown, and Tentarelli studied relatively complex spatial three-dimensional pipeline systems. Heinsbroek [12] compared the MOC and MOC-FEM methods. Matthew [13] used FLUENT+ANSYS software to establish a large eddy model and analyzed pressure pulsation pipelines. Sorokin et al. [14] studied the effective parameter range of fluid-filled pipeline vibration in energy transmission processes. Bezborodov et al. considering pipeline vibration, pipe shape, and damping, established a vibration model for pipelines, and finally validated the reliability of the method through experiments.

In this study, we establish and refine a 14-equation fluid-structure interaction (FSI) model for straight and elbow pipes using the fluid wave equation and Timoshenko beam model. This model comprehensively incorporates the effects of quasi-steady friction, unsteady Zielke friction, and Brunone friction on axial vibration. Additionally, a finite element simulation model is developed in ANSYS software to analyze the vibration modes under varying parameters such as materials, length, support configurations, and fluid velocity for general straight pipes. The findings reveal that different materials and fluid velocities minimally affect natural frequencies but significantly impact deformation. Conversely, pipe length and support configurations exert a substantial influence on natural frequency adjustment. These insights offer critical guidance for the design and optimization of hydraulic pipelines in helicopter applications

2. Mathematical Model

The hydraulic helicopter pipelines have a large length-to-diameter ratio and approximate their motion in the form of beam modes. There are a total of 7 degrees of freedom shared between the fluid and the pipe body, each of which can be represented by corresponding velocities (or angular velocities) and forces (or torques), ultimately described through 14 variables. The pipeline system consists of basic pipeline units such as straight and curved pipes. Therefore, the analysis of the vibration characteristics of straight and curved pipes is of great significance for studying the overall vibration trends of the pipeline system. Based on the fluid wave equation and the Timoshenko beam model, a 14-equation model for the fluid-structure coupling of hydraulic straight pipes and curved pipes is established. This model includes axial vibration, transverse vibration, and torsional vibration models, providing a theoretical basis for vibration analysis of pipeline systems.

Under bending loads, circular cross-section pipes deform into elliptical shapes, which enhances the elasticity of the curved pipe and leads to a higher and more complex stress distribution in the curved section. On the other hand, at the bending positions of the pipeline, the fluid is forced to change its direction of motion. Fluid pressure waves will generate unbalanced axial and transverse pressures on the inner wall of the curved pipe. Unlike in straight pipes, the axial and transverse movements of the structure are not mutually independent at the junction but form a coupled section.

2.1 Assumptions in Modeling

In the FSI 14 modeling process of straight and curved pipes, the following basic assumptions are applied with certain distinctions:

Both straight and curved pipes assume that the pipe wall material is purely elastic, homogeneous, and isotropic. However, the straight pipe must be assumed to have a uniform circular cross-section.

Both straight and curved pipes ignore the radial fluid movement caused by the radial deformation of the pipe wall, as well as the fluid rotation around the pipeline axis.

Both straight and curved pipes assume that the fluid inside the pipe is a single-phase continuous fluid without cavitation and volume separation phenomena, and the fluid is a Newtonian fluid.

Both straight and curved pipes assume that the bending and torsional vibrations of the pipeline conform to the motion form of the Timoshenko beam model, i.e., the bending force and displacement of the pipeline satisfy the linear relationship described by Hooke's Law.

2.2 Fluid Element Analysis

The axial force diagram of a straight pipe element is shown in Figure 1. The Pipe vibrations in x-z plane is shown in Figure 2.The length of the straight pipe element is *dz*, the diameter is *D*, the cross-sectional area is *A*, and the angle with the horizontal plane is β. The axial forces on the straight pipe element mainly include inertial force, axial internal and external forces on the pipe wall, friction between the pipe wall and the fluid, fluid pressure, and gravity[15]. f_z is the inward force in the z direction on the pipe wall, f_{ez} is the outward force per unit length in the z direction on the pipe wall (unit: N/m), *P* is the pressure of the fluid inside the pipe (Pa), ρ is the density (kg/m³), *g* is the gravitational acceleration, and V_0 is the steady flow velocity. where A_f is the cross-sectional area of the fluid inside the pipe, *V* represents the fluid velocity, including the steady flow velocity V_0 and the average pulsating velocity V_f across the cross-section of the fluid inside the pipe, i.e., $V=V_0+V_f$; *P* represents the fluid pressure, including the steady pressure P_0 and the average pulsating pressure P_f across the cross-section of the fluid inside the pipe, i.e., $P=P_0+P_f$.

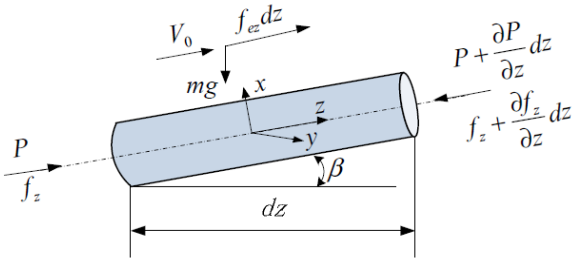


Figure 1 – The force of straight pipe along z direction.

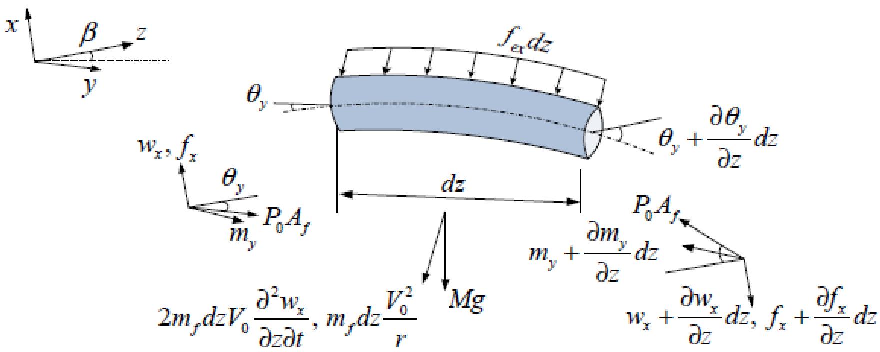


Figure 2 – Pipe vibrations in *x*-*z* plane.

Figure 3 is a diagram of the forces acting on a cross-section of a fluid element within the pipe wall and stress distribution. In the diagram, *R* is the inner radius of the pipe, *e* is the wall thickness, and *P* is the internal pressure of the pipe. Since the pipeline is an axisymmetric structure, it can be assumed that the stress ϕ is only a function of the radial coordinate *r*.

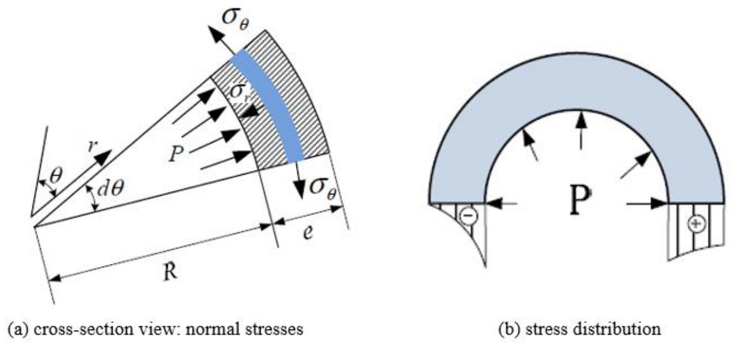


Figure 3 – Stress on element of pipe wall.

2.3 Axial Vibration Equations

$$\frac{\partial V_f}{\partial t} + V\frac{\partial V_f}{\partial z} + \frac{1}{\rho_f}\frac{\partial P}{\partial z} + \frac{4}{\rho_f D}\tau_w + g\sin\beta = 0 \tag{1}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l\ in\ (1)$$

$$\frac{\partial \dot{w}_z}{\partial t} + \frac{1}{\rho_p A_p}\frac{\partial f_z}{\partial z} - \frac{4}{\rho_p A_p \rho_f D}\tau_w + g\sin\beta - \frac{f_{ez}}{\rho_p A_p} = 0 \tag{2}$$

$for\ elbow\ tube,\ replace\ \partial z\ with\ \partial l,\ and\ add\ the\ bend\ relative\ item 1\rho pApfyRw$

$$\partial P\partial t{-}2\nu K'\partial wz\partial z{+}K'\partial Vf\partial z{=}0 \tag{3}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l, and\ add\ the\ bend\ relative\ item\ K'(1-2\nu)\frac{\dot{w}_y}{R_w}$$

$$\frac{\partial f_z}{\partial t} + EA_p + \frac{4\nu^2 K' A_p}{e_R(2+e_R)}\frac{\partial \dot{w}_z}{\partial z} - \frac{2\nu K' A_p}{e_R(2+e_R)}\frac{\partial V_f}{\partial z} = 0 \tag{4}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l, and\ add\ the\ bend\ relative\ item$$

$$EA_p\frac{\partial \dot{w}_z}{\partial l} + [EA_p + \frac{2\nu(1-2\nu)K' A_p}{e_R(2+e_R)}]\frac{\dot{w}_y}{R_p}$$

2.4 Transverse Vibration Equations

$$\frac{\partial \dot{w}_x}{\partial t}+\frac{1}{m}\frac{\partial f_x}{\partial z}-\frac{2m_f V}{m}\frac{\partial \dot{w}_x}{\partial z}-\frac{m_f V^2+PA_f}{mEI_p}M_y+g\cos\beta-\frac{1}{m}f_{ex}=0 \tag{5}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l, and\ add\ the\ bend\ relative\ item\ \frac{m_f V^2+PA_f}{m}\frac{\theta_z}{R_p}$$

$$\frac{\partial \dot{w}_x}{\partial t}+\frac{1}{m}\frac{\partial f_x}{\partial z}-\frac{2m_f V}{m}\frac{\partial \dot{w}_x}{\partial z}-\frac{m_f V^2+PA_f}{mEI_p}M_y+g\cos\beta-\frac{1}{m}f_{ex}=0 \tag{6}$$

$$for\ elbow\ tube,\ replace\ \partial z\ with\ \partial l,\ and\ add\ the\ bend\ relative\ item\ \frac{m_f V^2+PA_f}{m}\frac{\theta_z}{R_p}$$

$$\frac{\partial f_x}{\partial t}+\kappa GA_p\frac{\partial \dot{w}_x}{\partial z}-\kappa GA_p\dot{\theta}_y=0 \tag{7}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l$$

$$\frac{\partial \dot{\theta}_y}{\partial t}+\frac{1}{T_y}\frac{\partial M_y}{\partial z}+\frac{1}{T_y}f_x-\frac{1}{T_y}M_{ey}=0 \tag{8}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l, and\ add\ the\ bend\ relative\ item\ -\frac{1}{T_y R_p}M_z$$

$$\frac{\partial M_x}{\partial t}+EI_p\frac{\partial \dot{\theta}_x}{\partial z}=0 \tag{9}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l$$

$$\frac{\partial \dot{w}_y}{\partial t}+\frac{1}{m}\frac{\partial f_y}{\partial z}+\frac{2m_f V}{m}\frac{\partial \dot{w}_y}{\partial z}+\frac{m_f V^2+PA_f}{mEI_p}M_x+g\cos\alpha-\frac{1}{m}f_{ey}=0 \tag{10}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l, and\ add\ the\ bend\ relative\ item$$

$$-\frac{m_f V^2+PA_f}{m}\frac{1}{R_p}-\frac{1}{mR_p}f_z$$

$$\frac{\partial f_y}{\partial t}+\kappa GA_p\frac{\partial \dot{w}_y}{\partial z}+\kappa GA_p\dot{\theta}_x=0 \tag{11}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l, and\ add\ the\ bend\ relative\ item\ -\frac{\kappa GA_p}{R_p}\dot{w}_z$$

$$\frac{\partial \dot{\theta}_x}{\partial t}+\frac{1}{T_x}\frac{\partial M_x}{\partial z}-\frac{1}{T_x}f_y-\frac{1}{T_x}M_{ex}=0 \tag{12}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l, replace\ M_{ex} with\ M_{ey}$$

2.5 Torsional Vibration Equations

$$\frac{\partial \dot{\theta}_z}{\partial t}+\frac{1}{\rho_p J_p}\frac{\partial M_z}{\partial z}-\frac{1}{\rho_p J_p}M_{ez}=0 \tag{13}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l, and\ add\ the\ bend\ relative\ item\ \frac{1}{\rho_p J_p R_p}M_y$$

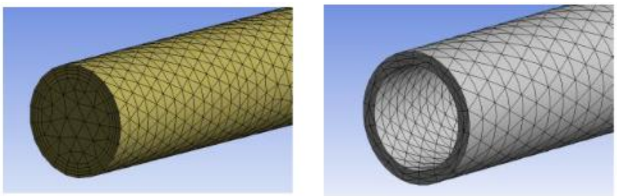
$$\frac{1}{\rho_p J_p R_p} M_y \frac{\partial M_z}{\partial t} + G J_p \frac{\partial \dot{\theta}_z}{\partial z} = 0 \tag{14}$$

$$for\ elbow\ tube, replace\ \partial z\ with\ \partial l, and\ add\ the\ bend\ relative\ item \frac{GJ_p}{R_p}\dot{\theta}_y$$

3. Simulation Analysis on Ansys

3.1 The Simulation Approches

The 3-D finite element simulation pipeline model with different material, length, fluids and support is built on the Ansys software to calculate the natural frequency and the vibration mode shape. In simulations, the quality, type, and quantity of the mesh determine the computation time and the accuracy of the results. Using tetrahedral meshing is straightforward, resulting in faster computation speeds. Besides, to improve the accuracy of the coupled calculations, local refinement is necessary at the fluid-solid interface during meshing. The fluid mesh model and the pipeline mesh model after meshing are shown in Figure 4. And all the parameters are listed in Table 1.



(a) Local grid of fluid model (b) Local grid of pipe model

Figure 3 – Meshing grid.

Table 1 - The inherent parameters on ANSYS

Prameters	Value	Unit
Inner radius	0.5, 4.8, 4.4	mm
Outer radius	6	mm
length	500,1000,1500,2000,2500	mm
Material	Steel,aluminum alloy,titanium alloy	
Fluid density	875	Kg/m3
Fluid dynamic viscosity	0.012	Pa·s
Fluid bulk modulus	1.95	GPa
Fluid Pressure	0，7，14，21，28，35	MPa
Fluid velocity	0,1, 3, 5, 7，9,11	m/s

The numerical analysis methods for fluid-structure interaction (FSI) can be categorized into one-way and two-way FSI simulations. Due to the high-speed and high-pressure characteristics of the fluid in the helicopter hydraulic system pipelines, and the objective of this study being the analysis of vibration dynamics, the influence of solid deformation on the fluid flow cannot be ignored. Therefore, two-way FSI simulation is required. This paper uses the Fluent, Static Structural, Modal, and System Coupling modules in Workbench to conduct two-way FSI simulations and modal analysis. The main steps include: fluid dynamics simulation in the flow field, static structural simulation in the structural field, and bi-directional transient result transfer. The setup of the simulation modules is shown in Figure 4.

Figure 4 – Two-way fluid-structure interaction - modal simulation analysis steps.

3.2 Vibration Mode of Tubes with Different Materials

To ensure reliable operation and stable performance under harsh conditions， the helicopter's hydraulic system relies on hydraulic tubing for efficient fluid transmission which features by lightweight, high strength and vibration resistance. Aluminum alloy hydraulic tubing and stainless steel hydraulic tubing are most commonly used for lower cost and rich application experience. When taking weight reduction as the primary consideration, titanium alloy hydraulic tubing is preferntially selected especially in high pressure circuit. So the hydraulic tube modal analysis was conducted varing materials on Ansys software. And all of the first 5 orders natural frequency and shape mode are shown from Table 2, Figure 5, Figure 6.

Table 2 - First 5 natural frequencies with different material

N-th order	Aluminum (Hz)	Steel (Hz)	Titanium (Hz)
1	35.196	35.088	31.692
2	96.96	96.663	87.307
3	189.93	189.35	171.02
4	313.64	312.69	282.41
5	467.96	466.54	421.35

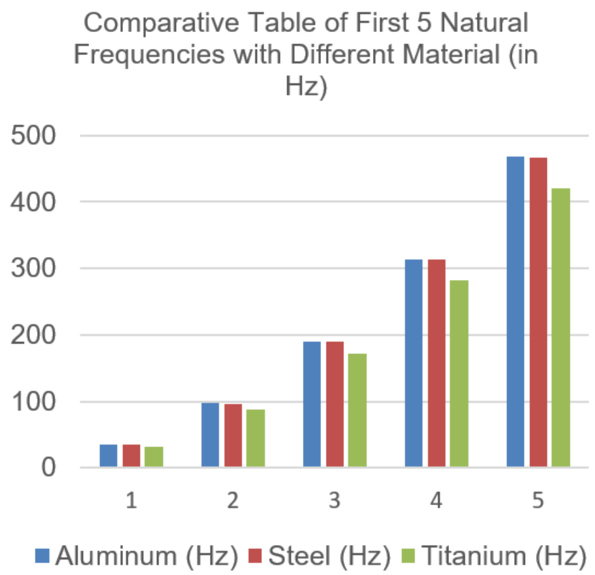


Figure 5 – First 5 natural frequencies with different material.

As shown from Figure 5, Under the same geometric condition, the natural frequency of titanium alloy is slightly lower than that of structural steel and aluminum alloy, while the natural frequency of structural steel is slightly lower than that of aluminum alloy. That can be explained by the following formula：

$$f_n = \frac{\beta_n^2}{2\pi}\sqrt{\frac{EI}{\rho A L^4}} \tag{15}$$

$$f \propto \sqrt{\frac{E}{\rho}} \tag{16}$$

$$\frac{f_{Ti}}{f_{Al}} \approx \sqrt{\frac{E_{Ti}/\rho_{Ti}}{E_{Al}/\rho_{Al}}} = \sqrt{\frac{110/4500}{70/2700}} = 0.97 \tag{17}$$

$$\frac{f_{Steel}}{f_{Al}} \approx \sqrt{\frac{E_{Steel}/\rho_{Steel}}{E_{Al}/\rho_{Al}}} = \sqrt{\frac{200/7850}{70/2700}} = 0.991 \tag{18}$$

Where:

f_n means the n-th natural frequency of a beam with fixed ends;

β_n Empirical Constants Related to the n-th Mode;

Mode Shape	Total Deformation		
	Aluminum	Steel	Titanium
 First Mode Shape(XY-plane)	9.215	5.883	7.726
 Second Mode Shape(XY-plane)	9.868	6.112	8.015
 Third Mode Shape(XY-plane)	10.35	6.956	8.964s
 Fourth Mode Shape(XY-plane)	11.403	7.235	9.462
 Fifth Mode Shape(XY-plane)	12.215	7.951	9.708

Figure 6 – vibration mode shape and deformation with different material.

As shown in the Figure 6, the first mode shape shows a simple bending pattern which has the largest displacement at the center of the pipeline; In addition, the number of nodes on the mode shape increses with the order; Lastly, the vibtation modes are all on the XY-plane and XZ-plane, and certain natuaral frequency is corresponded to two plane vibration modes on the XY-plane and XZ-plane respectively, all of which indicates that the straight pipeline can be regarded as the beam with fixed ends.

Through the simulation， it can be concluded that that there is no particularly significant difference in the natural frequencies of hydraulic pipes made of different materials. From the perspective of vibration modes, aluminum alloy pipes deform the most, followed by titanium alloy pipes, and steel pipes deform the least.

3.3 Vibration Mode of Tubes with Different Length

The pipeline length is set to 500mm, 1000mm, 1500mm, 2000mm, and 2500mm respectively, with 0.5mm wall-thickness, an outer diameter of 6mm and fixed ends. The pipeline is filled with hydraulic oil, which flows at a speed of 2m/s.

The impact of pipeline length on the natural frequency is calculated. The results are shown in Table 3 and Figure 7, which indicating that the pipeline length has a significant effect on the natural frequency of the tube. As the pipeline length increases, the natural frequency of the conduit rapidly decreases. This phenomenon can be partially explained by Equation-1, which shows that the natural frequency is inversely proportional to the fourth power of the length. On the other hand，the conclusion can be get that the fluid force act on the tube has just slight effect to the natural frequency.

Table 3 - First 5 natural frequencies change with different length

n-th order	500mm (Hz)	1000mm (Hz)	1500mm (Hz)	2000mm (Hz)	2500mm (Hz)
1	126.64	35.196	15.645	8.798	5.0714
2	348.24	96.96	43.114	24.248	13.978
3	680.49	189.93	84.49	47.527	27.399
4	1120.3	313.64	139.6	78.545	45.284
5	1665.5	467.96	208.43	117.3	67.634

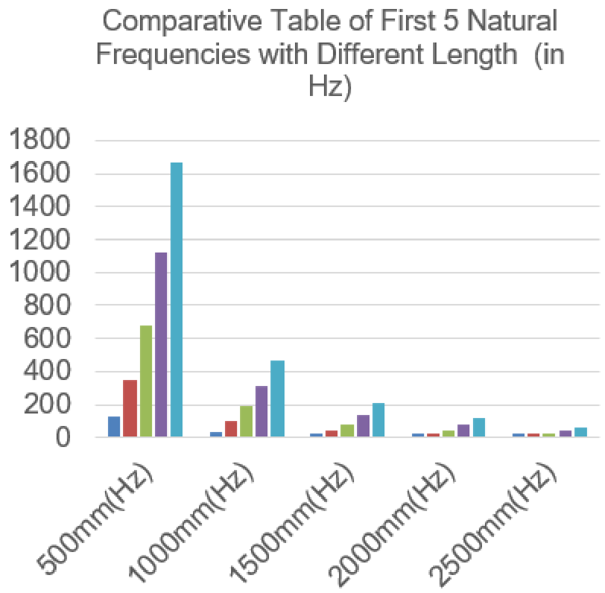


Figure 7 – First 5 natural frequencies with different lenghth.

3.4 Vibration Mode of Tubes with Different Number of Support

The pipeline uses no clamp, 1 clamp，2 clamps, 3 clamps, 4clamps to evenly support respectively, with 1000mm length, 0.5mm wall-thickness, an outer diameter of 6mm and fixed ends. The pipeline is filled with hydraulic oil. Figure 8 shows the clamps mounted evenly on the tube.

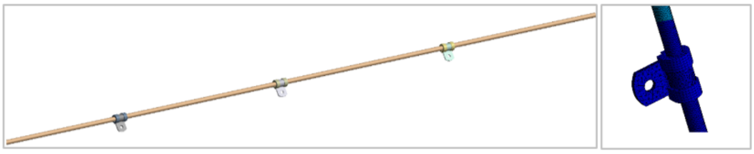


Figure 8 – Evenly distributed clamps supporting a pipe.

The impact of support number on the natural frequency is calculated. The results are shown in Table 4 and Figure 9, which indicating that the support number has a significant effect on the natural frequency of the tube. As the support number increases, the natural frequency of the conduit rapidly increases, which is is highly instructive for engineering applications.

Table 4 - First 5 orders natural frequencies change with support number

n-th order	no clamp (Hz)	1 clamp (Hz)	2 clamps (Hz)	3 clamps (Hz)	4 clamps (Hz)
1	31.692	224.37	383.13	588.22	757.2
2	87.307	356.82	493.15	709.41	972.34
3	171.02	441.99	559.47	953.05	1083.37
4	282.41	522.56	676.8	1124.41	1291.01
5	421.35	768.82	803.37	1334.8	1417.17

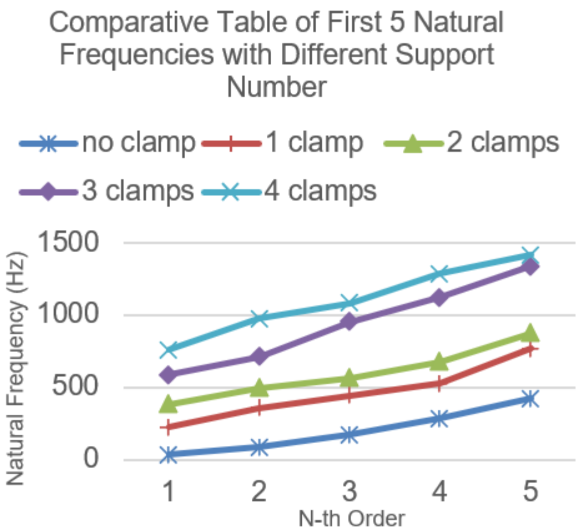


Figure 9 – First 5 orders natural frequencies change with support number.

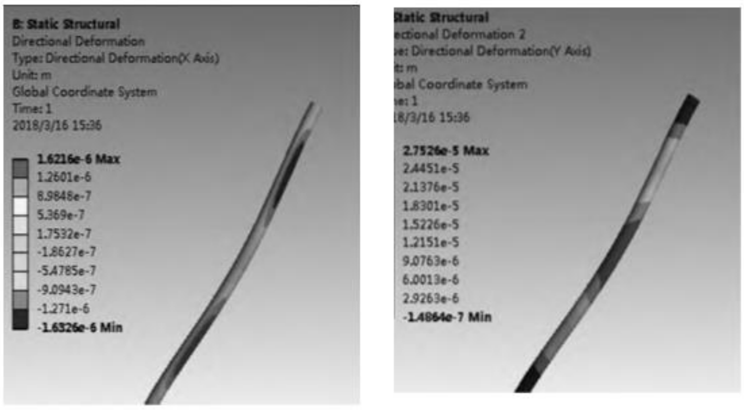
3.5 Vibration Mode of Tubes with Different Number of Support

The pipeline uses no clamp, with 1000mm length, 0.5mm wall-thickness, an outer diameter of 6mm and fixed ends. The pipeline is filled with hydraulic oil which flows at velocity of 1m/s, 3m/s, 5m/s, 7m/s, 9m/s respectively, under which the stress and displacement distributions of the pipeline along the X, Y, and Z axes are calculated. Results are shown in Table 5, Figure 10 and Figure 11 respectively illustrating the data and contour maps.

From the calculation results, it can be observed that: (1) Under the condition of fixed support at both ends of the straight circular pipe, the main manifestation of the straight pipe under fluid-structure interaction is the displacement of the pipeline structure. The pipeline exhibits significant oscillations in the axial X direction, while in the cross-sectional Y and Z planes, the pipeline shows bending, expansion, and contraction. (2) From the contour map of pipeline displacement, it can be seen that under fluid-structure interaction, the maximum stress of the pipeline is on the inner side of the hydraulic inlet section. (3) From Table 5, it can be seen that with the increase of pipeline flow velocity, the total displacement of the pipeline structure, as well as the displacements in the X, Y, and Z directions, and the maximum stress of the pipeline structure, all show an upward trend.

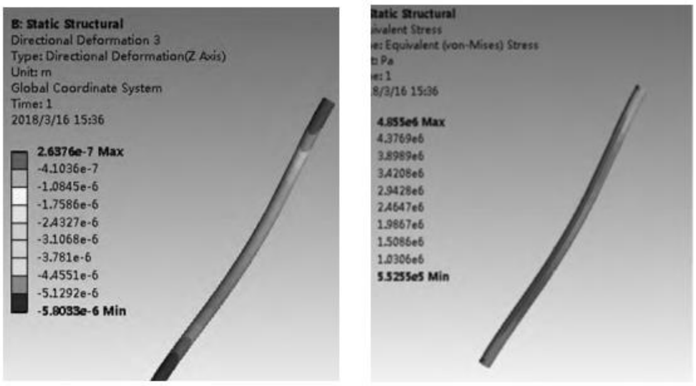
Table 5 - Deformation and stress chage with the fluid velocity

Flow velocity	Max deformation Displacement (mm)	X-axis Displacement (mm)	Y-axis Displacement (mm)	Z-axis Displacement (mm)	Max Stress (Pa)
1	0.1628E-6	0.8824E-8	0.1486E-6	0.1565E-8	0.1525E6
3	0.1526E-5	0.8632E-7	0.1495E-5	0.1558E-7	0.2352E6
5	0.5411E-5	0.3105E-6	0.5315E-5	0.5138E-7	0.9258E6
7	0.1382E-4	0.7406E-6	0.1618E-4	0.1309E-6	0.2387E7
9	0.2813E-4	0.1851E-5	0.2863E-4	0.2743E-6	0.4963E7



(a) X-axis deformation (b) Y-axis deformation

Figure 10 – X-axis & Y-axis deformation contour maps.



(c) Z-axis deformation (d) Stress

Figure 11 – Z-axis Deformation and stress contour maps.

4. Conclusion

This study firstly presents a comprehensive 14-equation fluid-structure interaction (FSI) model, integrating the fluid wave equation and Timoshenko beam model, to analyze straight and elbow pipes. The model accounts for quasi-steady friction, unsteady Zielke friction, and Brunone friction, influencing axial vibration. Furthermore, employing ANSYS software, a finite element simulation model is developed to explore vibration modes considering variations in materials, length, support configurations, and fluid velocity for general straight pipes. Results indicate minimal impact of different materials and fluid velocities on natural frequencies but significant influence on deformation. Conversely, pipe length and support configurations notably affect natural frequency adjustment. These findings provide essential insights for optimizing hydraulic pipeline design, particularly in helicopter applications.

5. Copyright Statement

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