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Vibration Characteristics of Different Cross-Section Pipes With Different End Conditions

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Abstract

This paper investigates the effect of end conditions on the vibration characteristics of a pipe conveying fluid with different cross sections such as (sudden enlargement and sudden contraction). The governing equation of motion for this system is derived by using beam theory. Three types of end pipe supports (flexible, simply and rigid) were adopted to investigate their effects on the vibration characteristics. Also, the effect of some design parameters like pipe diameter, length, pipe material, and the effect of fluid velocity were investigated.

Two different pipe systems of different diameters were investigated, model-1 [12.7mm, 25.4mm, 12.7mm] and model-2 [6.35mm, 12.7mm, 6.35mm] with length [0.25m, 0.5m, 0.25m] and model-3 with same diameter for model-1 but with length [0.5m, 0.5m, 0.5m]. Three pipe materials were tried, copper, steel and aluminum. The effect of Reynolds number between (500 - 1500) was also investigated. The dynamic behavior of a pipe conveying fluid is described by means of transfer matrix method. A Matlab- R2007 language computer program has been developed in this study to predict the vibration response.

The results of Matlab program were compared with those from ANSYS-11 program and it is found that there is a good agreement between them.

Keywords: Vibration characteristics, End support, End condition.

خصائص الأهتزاز لأنابيب ذات مقاطع وظروف نهاية مختلف

الخلاصة

تم في هذا البحث دراسة تأثير الشروط الحدية على خصائص الأهتزاز لأنبوب ذي مساحة مقطع مختلفة (توسع وتقلص مفاجئ) يستخدم لنقل مائع. معادلة الحركة لهذا النظام أشتقت من نظرية (beam). حيث تم استخدام عدة أنواع من المساند للأنبوب (مرن، بسيط وصلب) لمعرفة تأثيرها على قيم الترددات الطبيعية وطور نسق الأهتزاز بشكل نظري ، كما تم دراسة تأثير تغير بعض المحددات التصميمية مثل أبعاد الأنبوب(القطر، الطول) ، نوع معدن الأنبوب وسرعة جريان المائع، و تم اعتماد أقطار متغيرة. النموذج الأول بأقطار [12.7 mm ,25.4 mm , 12.7 mm] والنموذج الثاني بأقطار[6.35 mm , 12.7 mm , 6.35 mm] وكلاهما بطول [0.25mm , 0.5mm , 0.25mm] .

كما تم دراسة تأثير الطول على قيم الترددات الطبيعية حيث أعتمد النموذج الثالث بنفس أقطار النموذج الأول وبطول [0.5 m , 0.5m , 0.5 m]. ولمعرفة تأثير نوع معدن الأنبوب تم أختيار

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ثلاثة أنواع من المعادن [الفولاذ والنحاس والألمنيوم]. كما تم دراسة تأثير وجود مائع يجري جريانا مستقرا في المنظومة على قيم الترددات الطبيعية للمنظومة لمختلف قيم عدد رينولدز (500 ≤ Re ≤ 1500). اتبعت طريقة المصفوفات الأنتقالية لتوضيح تأثير الأهتزاز علـى الخصـائص الديناميكيـة للأنبوب الذي قسم الى 20 مقطع على طول الأنبوب الناقل للمائع وتم تحديد الخصائص الديناميكيـة متمثلة بـــ (الأنحناء ، الميل ، العزم ، قوى القص ، السرعة ، الضغط) لكل موقع. تم بناء برنـامج بلغة [MATLAB R2007] لأنجاز الجانب النظري وللتأكد من عمل هذا البرنامج تمت مقارنتـه مع برنامج [ANSYS-11] وقد اظهرت النتائج توافقا جيدا بين نتائج كلا البرنامجين .

Nomenclature

A_f	Cross-section area of fluid	m^2
A_p	Cross-sectional area of the pipe	m^2
F_o	External force	N
Fi	Filed matrix	-
F	Fluid force applied to the pipe	N
Do	Outer diameter for the pipe	M
Di	Inner diameter for the pipe	M
E	Modulus of elasticity for pipe	N/m^2
E_m	Mean modulus of elasticity for pipe	N/m^2
f	Friction factor	-
G	Modulus of rigidity	N/m^2
I	Second moment of area for pipe	m^4
I_m	Mean second moment of area for pipe	m^4
L	Length of the pipe	M
L_i	Element length	M
L_m	Mean element length	M
M	Bending moment	N.m
m_f	Mass of fluid per unit length	kg/m
m_p	Mass of pipe per unit length	kg/m
P_f	Fluid pressure	N/m^2
P_1	Inlet pressure to the pipe	N/m^2
P_3	Outlet pressure from the pipe	N/m^2
th	Thickness of the pipe	M
u_1	Inlet fluid velocity to the pipe	m/s
u_2	Fluid velocity after enlargement	m/s
u_3	Fluid velocity after contraction	m/s
Q	Transverse shear force in the pipe	N
W	Coriolis &Compressive forces	N
X	Longitudinal coordinate	-

Y	Transverse displacement of pipe	M
Z_i	State vector	-
Dimensionless Groups		
Re	Reynolds number based on hydrualic diameter and velocity	$u.D_i / u$
C_e	Constant related to losses in enlargement	$\left[1-\left(A_1 / A_2\right)\right]^2$
C_c	Constant related to losses in contraction	$\left[(\frac{A_2}{A_c})-1\right]^2$
Greek Letters		
Φ	Slope of the pipe	rad
μ	Dynamic viscosity of the fluid	kg/m.s
ρ	Density of the fluid	kg/m^3
τ	Shear stress on the internal surface of the pipe	N/m^2
υ	Kinematic viscosity of the fluid	m^2/s
c	Numerical factor	-
ω	Excitation ferquency	rad/sec
ω_n	Natural ferquency	rad/sec
Subscripts		
f	Fluid	
i	State vector	
p	Pipe	
Superscripts		
$\bar{}$	Dimensionless notation	-
L,R	Left and right of the state vector	-
Notations		
/	$\frac{\partial}{\partial x}$	-
•	$\frac{\partial}{\partial t}$	-

Introduction

The subject of vibration deals with the oscillatory motion of dynamic systems. All systems possessing mass and elasticity are capable of vibration. The mass is the inherent of the body, and the elasticity is due to the relative motion of the parts of the body [1].

The fluid flow and structure are interactive systems, and their interaction is dynamic. These systems are coupled by the forces exerted on the structure by the fluid. The fluid force causes the structure to deform. As the structure deforms it changes its orientation and hence affects the characteristics of the flow (pressure and velocity). A mathem-atical model has been developed for the structure and for the fluid. The dynamic interaction of the structure and the fluid models is described by linear oscillatory equation [2].

The flow through a pipe with sudden enlargement and contraction occurs in many industrial applications and is characterized by increased pressure losses caused by flow separation close to the change in the cross sectional area. This increasing in pressure losses will increase the erosion rates and the heat in the regions where separated flow occurs [3]. Also, the fluid flowing through the pipe may impose pressures on the pipe walls which deflect the pipe, where at a high velocity flow through a thin wall pipe it can either buckle the pipe or cause it to fail. In certain applications involving very high velocity flows through flexible thin wall pipes combined with vibration such as (the feed lines to rockets and water turbines) the pipe may become susceptible to resonance and fatigue failure if its natural frequency falls below certain limits [4].

The complete set of equations of motion for pipe conveying fluid by using Timoshenko beam theories derived by [5]. They also used Lagrange strain-displacement theory but taking the longitudinal strain only and neglected all other strains, then using extended Hamilton's principle to drive the equations of motion for the longitudinal and transverse displacements. The output included.

1. Forces due to the flowing fluid in the beam.
2. Kinetic energy of the flowing fluid.
3. Finite element models of the governing equations.

The vibration system consisted of a rotating cantilever pipe conveying fluid and a tip mass studied by [6]. The equation of motion was derived by using the Lagrange's equation. This paper included:

1. Studying the influences of the rotating angular velocity and the velocity of fluid flow on the dynamic behavior of a cantilever pipe.
2. Studying the effects of a tip mass on the dynamic behavior of a rotating cantilever pipe.

They found that the natural frequencies of a cantilever pipe conveying fluid are proportional to the angular velocity of the pipe and the tip mass.

1-GoverningEquation of Motion

Consider a straight pipe conveying uniform internal flow as shown in figure (1). The straight pipe, clamped at both ends, has dimensions given by the length (L), the cross-sectional outer diameter (D) and the thickness (th). It is assumed that the pipe is sufficiently slender, that is, (D/L) << 0.1, this ratio makes it considered as a beam. Moreover, the fluid in the pipe is assumed to be incompressible so that its velocity is uniform inside the pipe. This means that the so-called "Laminar flow" is assumed, where the secondary flow effects are negligible.

The equation of motion for free vibration of pipe conveying fluid derived and may be written as:

$$EI\frac{\partial^4 y}{\partial x^4} + \left(m_f u_1 u_2 + P_f A_p\right)\frac{\partial^2 y}{\partial x^2} + 2m_f u_1\frac{\partial^2 y}{\partial x \partial t} + \left(m_f + m_p\right)\frac{\partial^2 y}{\partial t^2} = 0$$

Where:

$EI\frac{\partial^4 y}{\partial x^4}$: Stiffness term

$\left(m_f u_1 u_2 + P_f A_p\right)\frac{\partial^2 y}{\partial x^2}$: Curvature term

$2m_f u_1\frac{\partial^2 y}{\partial x \partial t}$: Coriolis force term

$\left(m_f + m_p\right)\frac{\partial^2 y}{\partial t^2}$: Inertia force term

The equation of motion for forced vibration of pipe conveying fluid which derived may be written as:

$$E\cdot I\cdot y'''' + \left(P_f\cdot A_P + m_f\cdot u_1\cdot u_2\right)y'' + 2\cdot m_f\cdot u_1\cdot\dot{y}' + \left(m_f + m_p\right)\ddot{y} = F(x,t)$$

Where:

F(X,t) is the non-dimensional external force applied normal to the pip axis in (y-direction).

The dimensionless variables are:

$$\bar{X} = \frac{x}{L_m},\ \bar{Y} = \frac{y}{L_m}$$

$$\bar{U} = \left(\frac{m_f}{E\cdot I}\right)^{1/2}\cdot u_f\cdot L_m$$

$$b = \left(\frac{m_f}{m_f + m_p}\right)^{1/2}$$

$$g = \left(\frac{L_m^{\ 2}}{E\cdot I}\right)\cdot P\cdot A_P$$

$$t = \left(\frac{E\cdot I}{m_f + m_p}\right)^{1/2}\left(\frac{t}{L_m^{\ 2}}\right)$$

Then the equations for free vibration become:

$$\bar{Y}'''' + \left(g + \bar{U}_1\cdot\bar{U}_2\right)\bar{Y}'' + 2\cdot\bar{U}\cdot\frac{b}{t}\cdot\dot{\bar{Y}}' + \frac{1}{t^2}\ddot{\bar{Y}} = 0$$

And the equations for forced vibration become :

$$\overline{Y}^{////} + \left(g + \overline{U}_1 \cdot \overline{U}_2\right)\overline{Y}'' + 2 \cdot \overline{U} \cdot \frac{b}{t} \cdot \overset{\bullet /}{Y} + \frac{1}{t^2}\overset{\bullet\bullet}{\overline{Y}} = F\left(\overline{X}, t\right)$$

2-Investigation of the flow stream

The value of inlet velocity (u_1) can be found from inlet Reynolds number where:

$$u_1 = \frac{m.\mathrm{Re}}{r.D_i}$$

While the velocity through the enlargement (u_2) and through contraction (u_3) of the pipe can be determined by using the following formula:

$$u_2 = h_1 u_1 \quad \& \quad u_3 = h_2 u_2$$

Where:

h_1= Area ratio for sudden enlargement pipe.

h_2=Area ratio for sudden contraction pipe.

The pressure change due to friction for flow in pipe for any uniform cross section is given as follows [7].

$$\Delta P = f \cdot \frac{L}{D_i} \cdot \frac{\rho . u^2}{2}$$

Where: (f) Is the friction factor for laminar flow in pipe given by

$$f = \frac{64}{\mathrm{Re}}$$

Re = Reynolds Number $= \frac{u \cdot D_i}{\upsilon}$

D_i = inner diameter of pipe (flow diameter).

Since the fluid discharge to atmosphere; therefore, the out let pressure of the pipe (P_3) = 1 atm and the inlet pressure to the pipe (P_1) can be found from Bernoulli's equation as follows:

$$\frac{P_1}{r \cdot g} + \frac{u_1^2}{2 \cdot g} + Z_1 = \frac{P_3}{r \cdot g} + \frac{u_3^2}{2 \cdot g} + Z_3 + \text{Losses}$$

For horizontal pipe (z_1=z_3=0) substitute in above equation gives:

$$P_1 = P_3 + \left(\frac{u_3^2 - u_1^2}{2} + \text{Losses}\right) * r$$

Where:

Losses = P_{L1}+ P_{Le} +P_{L2}+P_{Lc}+P_{L3}

P_{L1}= losses for the first part of pipe (before enlargement).

P_{L2}= losses for the second part of pipe (after enlargement).

P_{L3}= losses for the third part of pipe (after contraction).

P_{Le}= losses at enlargement

$$= [\frac{1}{2} C_e . r . u_1^2]$$

$$C_e = \text{constant} = \left[1 - \frac{A_1}{A_2}\right]^2$$

P_{Lc}= losses at contraction

$$= [\frac{1}{2} C_c . r . u_3^2]$$

C_c= constant = $\left[\frac{A_2}{A_c} - 1\right]^2$

3-The Transfer matrix method

In this method the system can be converted to a mathematical model consist of number of stations represented by point matrix where the mass concentrated at each station, each station joined with massless element which is represented by field matrix, then it can be found that the equations of deflection (Y), slope (θ), bending moment (M), shear force (Q), velocity (U), Pressure (P) for the vibrated pipe conveying fluid, these equations are:

a-The Equations For field Matrix

$$\overline{Y}_i^L = \overline{Y}_{i-1}^R - \overline{\Phi}_{i-1}^R \frac{L_i}{L_m} - \frac{\overline{M}_{i-1}^R L_i^2}{2(EI)_i d\ L_m} - \overline{Q}_{i-1}^R \cdot \frac{1}{j\ .L_m} \cdot \left[\left(\frac{L_i^3}{6\ (EI\ )_i}\right) - \left(c\ \frac{L_i}{(GA_p)_i}\right)\right] + \frac{\overline{W}_i}{j\ .L_m}\left[\frac{L_i^3}{48(EI)_i} - \frac{c\ .L_i}{(GA_p)_i}\right]$$

$$\overline{\Phi}_i^L = \overline{\Phi}_{i-1}^R + \overline{M}_{i-1}^R \frac{L_i}{(EI)_i d} + \overline{Q}_{i-1}^R \frac{L_i^2}{2(EI)_i j} - \frac{\overline{W}_i L_i^2}{8(EI)_i j}$$

$$\overline{M}_i^L = \overline{M}_{i-1}^R + \overline{Q}_{i-1}^R \frac{L_i d}{j} - \frac{\overline{W}_i L_i d}{2\ .j}$$

$$\overline{Q}_i^L = \overline{Q}_{i-1}^R - \overline{W}_i$$

$$\overline{U}_i = \left[\frac{m_f}{EI}\right]^{1/2} U_i L_m\ ,$$

$$\overline{U}_i^L = \overline{U}_{i-1}^R$$

$$\overline{P}_i^L = \overline{P}_{i-1}^R - \frac{2f_i\ r_f\ u^2\ L_i}{D \cdot P_1}$$

Where:

$$j = \frac{L_m^2}{(EI)_m}\ ,\ d = \frac{L_m}{(EI)_m}\ ,$$

$$L_m = \frac{\sum_{i=1}^{n} L_i}{n-1}\ , I_m = \frac{\sum_{i=1}^{n} I_i}{n-1}\ ,$$

$$E_m = \frac{\sum_{i=1}^{n} E_i}{n-1}$$

b-The Equations for The Particular Node

$$\overline{Y}_i^L = \overline{Y}_i^R\ , \overline{\Phi}_i^L = \overline{\Phi}_i^R\ ,$$

$$\overline{M}_i^L = \overline{M}_i^R\ ,$$

$$\overline{Q}_i^R = \overline{Q}_i^L - \overline{\omega}^2\ \overline{Y}_i - \overline{F}_i\ ,$$

$$\overline{U}_i^L = \overline{U}_i^R\ ,$$

$$\overline{P}_i^L = \overline{P}_i^R$$

C-The Equations for The Supported Node

$$\overline{Y}_i^L = \overline{Y}_i^R , \quad \overline{\Phi}_i^L = \overline{\Phi}_i^R ,$$
$$\overline{M}_i^L = \overline{M}_i^R ,$$
$$\overline{U}_i^L = \overline{U}_i^R , \quad \overline{P}_i^L = \overline{P}_i^R$$

$$Q_i^R = Q_i^L - (m_t.\omega^2 - K).j.L_m.\overline{Y}_i$$

d-The Equations Of The Sudden Enlargements

$$\overline{Y}_i^L = \overline{Y}_i^R , \overline{\Phi}_i^L = \overline{\Phi}_i^R ,$$
$$\overline{M}_i^L = \overline{M}_i^R$$
$$\overline{U}_i^R = h_1.\overline{U}_i^l \quad , \quad \overline{P}_i^R = \overline{P}_i^L + \frac{C_e.r.u^2}{2.P_{inlet}}$$

Where: $h_1 = \frac{A_1}{A_2}$

A_1 =cross-section area of the pipe before enlargement.

A_2 =cross-section area of the pipe after enlargement.
P_{inlet} = inlet pressure to the pipe.

u = fluid velocity

Ce=constant = $\left[1 - \frac{A_1}{A_2}\right]^2 = 0.5$

e-The Equations Of The Sudden Contraction

$$\overline{Y}_i^L = \overline{Y}_i^R , \overline{\Phi}_i^L = \overline{\Phi}_i^R ,$$
$$\overline{M}_i^L = \overline{M}_i^R$$

$$\overline{U}_i^R = h_2.\overline{U}_i^l \quad , \quad \overline{P}_i^R = \overline{P}_i^L + \frac{C_C.r.u^2}{2.P_{inlet}}$$

$$, h_2 = \frac{A_2}{A_3} ,$$

$$C_c = \left[\frac{A_2}{A_c} - 1\right] = \text{constant}$$

A_2 =cross-section area of the pipe before contraction
A_3 =cross-section area of the pipe after contraction
P_{inlet} = inlet pressure to the pipe.

u = fluid velocity

4-Result and discussion

A suitable MATLAB_R2007 language program has been developed to embrace the theoretical work. The pipe span was discrtized into twenty element and twenty one point station and the forced vibration at different excitation frequencies is imposed at station eleven with represented the mid span of the pipe system shown in figure (2). This program is uses to determine (natural frequencies, mode shape, deflection, slope, bending moment, and shear force) for different [diameter, material, length, supports and fluid velocity]

A-Effect of support

The deflection at mid length of pipe without fluid with various excitation frequencies for different kinds of supports [flexible, simply, rigid] for model-1 are presented in fig. (3). Also, it may be observed the values of the natural frequencies from the peaks of this figure which are given in table (A-1) for different cases of pipe support. Fig (1-a, b, c) show the values of natural frequencies for flexible support are less than that for simply and rigid support, also the values of natural frequencies for simply support are less than that rigid support. This because the flexible support have the ability to move in Y-direction therefore, its flexibility is very high compares with simply and rigid support, that leads to decrease the pipe stiffness and hence its natural frequency. While rigid support is tightly supported more than the other two kinds of supports [Y(0,t)=0 &Y(L,t)=0] also there is no slope at the support's position of pipe[dY/dX(0,t)=0 &dY/dX(L,t)=0] which leads to increase the stiffness of the pipe at support's position and thus decreases the natural frequencies more than the other two kinds of supports.

B-Effect of diameter size

In order to study the effect of the diameter size on the natural frequencies of the pipe system with different types of supports, two different pipe diameters were used with two models. The first model with diameters (12.7mm, 25.4mm, 12.7mm) and the second model with diameters (6.35mm, 12.7mm, 6.35mm). It seems that the first model has the highest natural frequencies values than the second model for all kind of supports. Figures (4) show the three lowest natural frequencies for copper pipe system without fluid with different support for (model-2).

Table (A-2) shows the comparison of natural frequencies values of pipe with different diameters and supported with flexible, simply and rigid supports, respectively. It's obviously seen that the natural frequency is affected by the diameter size for all kind of selected supports. The natural frequencies for (model-1) are twice the natural frequencies for (model-2). So, the increasing in diameter size will cause an increasing in inertia, therefore increasing in stiffness yields increasing in natural frequency. The values of natural frequencies in rigid support case are higher than those in flexible and simply supports, because the overall stiffness of the system is higher.

C-Effect of pipe material

To study the effect of pipe material on the natural frequencies, three pipe materials were selected which are copper, steel and aluminum. Their mechanical properties are listed in Table (A-3). Figure (5), show the (1^{st}, 2^{nd} and 3^{rd}) natural frequencies for the steel pipe material while figure (6), show the (1^{st}, 2^{nd} and 3^{rd}) natural frequencies for aluminum pipe material. These figures indicate that the natural frequencies values of steel pipe are higher than those of copper pipe for all kinds of supports as well as the natural frequencies values of aluminum pipe are higher than those copper and steel with different ratios as listed in Table (A-4). This is because the steel and aluminum have more stiffness than copper because of their physical properties.

D-Effect of Pipe Length

In order to study the effect of pipe length on the natural frequencies of the pipe system with different types of supports, two different pipe lengths were used with the same diameter (12.7mm, 25.4mm, 12.7 mm). Model one is with a length (0.25m, 0.5m, 0.25m) and the other model are with a length (0.5m, 0.5m, 0.5m). It seems that the first model has the highest natural frequencies values than the third model for all kind of supports. Figure (7) show the three lowest natural frequencies for copper pipe (model-1) and (model-3).

Table (A-5) shows the comparison of natural frequencies values of pipe with different pipe lengths and supported with flexible, simply and rigid supports, respectively. It is obviously seen that the natural frequency is affected by the pipe length for all kinds of selected supports. The natural frequencies for (model-1) are higher than the natural frequencies for (model-3).

E-Effect of Fluid Velocity

Table (A-6) shows the effect of increasing the Reynolds number on the natural frequencies.

Where;

$$Re = \frac{u D r}{m}$$

In the present study, at mid length of pipe conveying fluid with various velocities for different kinds of supports (flexible, simply and rigid) the deflections are presented in Figures. (8) and (9). The values of the natural frequencies from the peaks of these figures are given in the above table for different cases of pipe supports. It can be noticed from these figures and tables that the values of the natural frequency for the case of vibrated pipe system conveying fluid remain constant with the increasing of Reynolds number because increasing Reynolds number leads to increasing the fluid velocity and this increase doesn't affect the properties of the pipe system material (stiffness and mass). Table (A-7) shows that the values of the natural frequencies for the case of vibrated pipe system conveying fluid are less than the values of the natural frequencies for the case of vibrated pipe system without fluid. This can be related to the effect of the fluid mass which is added to the mass of the system and it is inversely proportional to the natural frequencies.

F-Results of the comparison between the ANSYS-11 and MATALAB-R2007 program

The comparisons are made for different cases of support pipe model-1 without fluid. Some of the numerical results obtained from transfer matrix method by adopting [MATLAB-R2007] program were compared with finite element method by using [ANSYS -11] program. The comparisons between the results for these two programs show a good agreement with a maximum difference of (2.027 %) and a minimum difference of (0.032 %). The comparisons are presented in the tables (B-1).

G-Conclusions

From the results of the present work, the following conclusions may be deduced:

1-The values of the fundamental natural frequency for the pipe with flexible support are less than those obtained for rigid support for the adopted stiffness values with a percent [49.7%, 41.9%, and 31.5%] for three lowest natural frequencies, respectively. Also, the natural frequencies for simply supported are less than those obtained for rigid support with a percent [47.4%, 36.7%, and 19.4%] for three lowest natural frequencies, respectively.

2- The decreasing in the system pipe diameter will reduce the natural frequencies.

3-The increasing in the system pipe length will reduce the natural frequencies.

4-The natural frequencies values of aluminum system are higher than those for steel and copper system pipe for all kinds of supports.

5-The natural frequencies for pipe system conveying fluid is less than the natural frequencies for pipe system without fluid.

6- The natural frequencies for pipe system conveying fluid stay constant with the increasing of Reynolds number (fluid velocity).

7- The results of the transfer matrix method by using (MATLB-R2007) Program and finite element method by using (ANSYS-11) Program, gives a good agreement with percentage for a maximum difference of (2.027%) and a minimum difference of (0.032%).

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Tables (A)

Table(A-1) The natural frequencies values of copper pipe with different types of

Copper pipe (model-1)								
Flexible Support			Simply Support			Rigid Support		
ω_{n1} (rad/sec)	ω_{n2} (rad/sec)	ω_{n3} (rad/sec)	ω_{n1} (rad/sec)	ω_{n2} (rad/sec)	ω_{n3} (rad/sec)	ω_{n1} (rad/sec)	ω_{n2} (rad/sec)	ω_{n3} (rad/sec)
195	585	1634	204	637	1921	388	1007	2386

Table (A-2) Comparison of natural frequencies values with different pipe diameter.

	Copper pipe								
	Flexible Support			Simply Support			Rigid Support		
Diameter (mm)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)
Model-1	195	585	1634	204	637	1921	388	1007	2386
Model-2	93	290	904	94	293	917	180	465	1137

Table (A-3) Properties of pipe material [8].

Material	$E(N/m^2)$	$G(N/m^2)$	$\rho(kg/m^3)$
Copper	$120*10^9$	$40*10^9$	8933
Steel	$200*10^9$	$79*10^9$	7860
Aluminum	$70*10^9$	$26*10^9$	2710

	Flexible Support			Simply Support			Rigid Support		
Material	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)
Copper	195	585	1634	204	637	1921	388	1007	2386
Steel	260	764	2022	281	878	2649	535	1389	3292
Aluminum	275	841	2437	283	884	2667	539	1398	3313

Table (A-5) Comparison of natural frequencies values with different pipe lengths.

	Copper pipe								
	Flexible Support			Simply Support			Rigid Support		
Model	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)
Model-1	195	585	1634	204	637	1921	388	1007	2386
Model-3	163	556	1511	168	590	1850	342	913	2496

Table (A-6) Comparison of natural frequencies values with different Reynolds number

	Flexible Support			Simply Support			Rigid Support		
Re	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)
500	161	502	1430	169	544	1661	320	853	2051
1000	161	502	1430	169	544	1661	320	853	2051
1500	161	502	1430	169	544	1661	320	853	2051

Table (A-7) Comparison of natural frequencies valuetys with and without fluid for different supports.

	Copper pipe [model-1]								
	Flexible Support			Simply Support			Rigid Support		
	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)	ω_{n1} (rad/s)	ω_{n2} (rad/s)	ω_{n3} (rad/s)
Without fluid	195	585	1634	204	637	1921	388	1007	2386
With fluid	161	502	1430	169	544	1661	320	853	2051

Table (B)

Table (B-1) Results of comparison between the T.M.M by using MATLAB-R2007 program and F.E.M by using ANSYS-11 program for copper pipe without fluid.

Copper pipe [model-1] , Without fluid				
Re = 0		w_{n1} (rad/s)	w_{n2} (rad/s)	w_{n3} (rad/s)
Flexible Support	T.M.M	195	585	1634
	F.E.M	195.407	589.564	1634.527
	Error %	0.208	0.774	0.032
Simply Support	T.M.M	204	637	1921
	F.E.M	204.832	642.996	1930.194
	Error %	0.406	0.932	0.476
Rigid Support	T.M.M	388	1007	2386
	F.E.M	389.928	1018.027	2405.832
	Error %	0.494	1.083	0.824

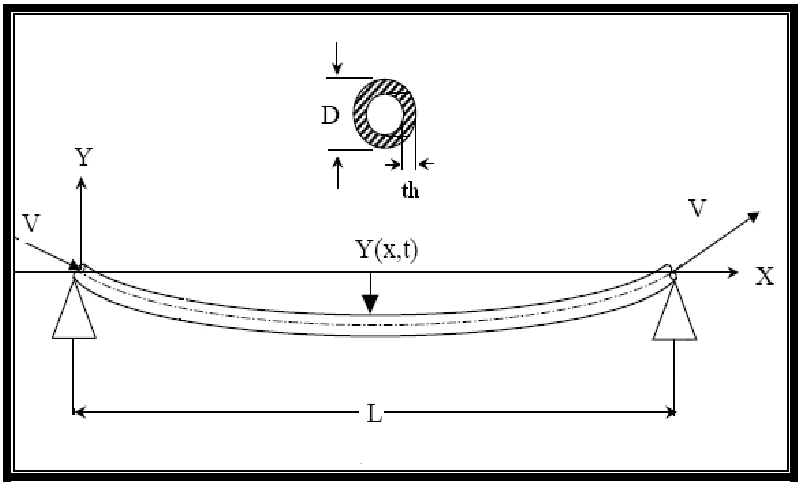

Figure (1) Pipe conveying fluid.

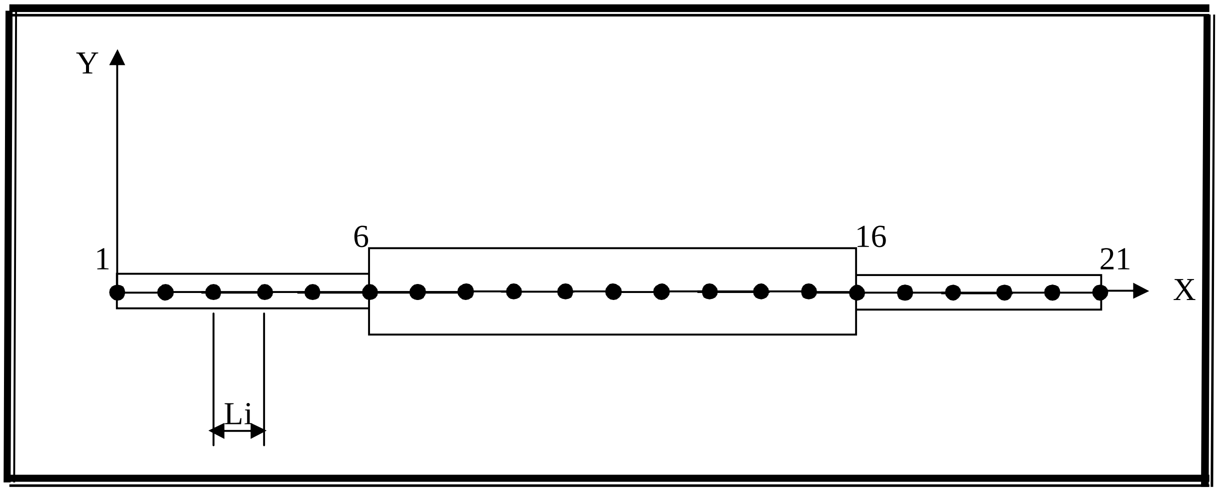

Figure (2) Pipe with discrete elements and masses.

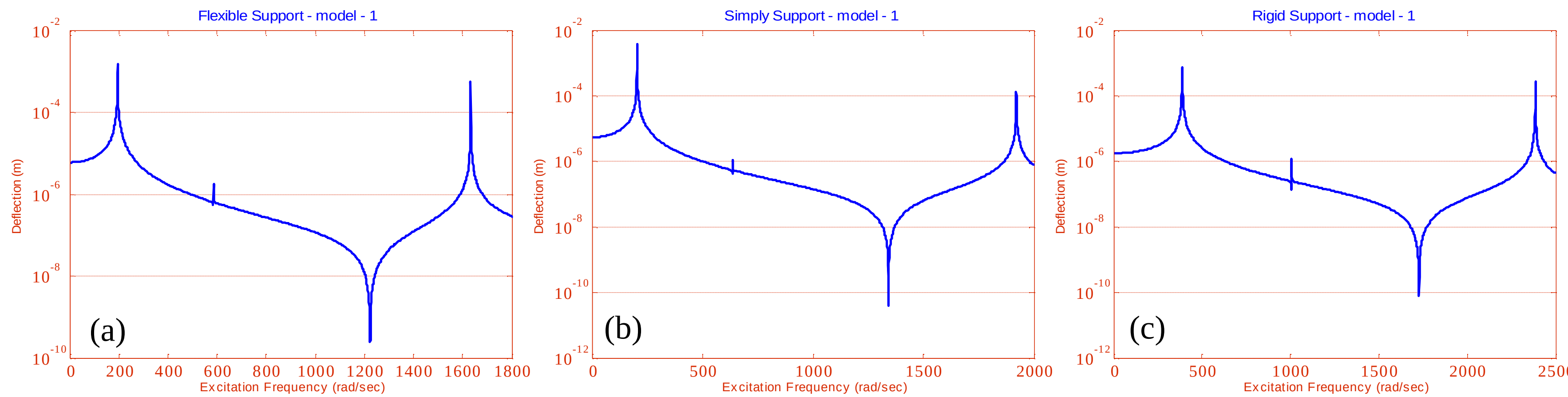

Figure (3) Deflection for (flexible, simply, rigid) supports pipe without fluid with various excitation frequencies a mid span of copper pipe (model-1) represent three lowest natural frequency

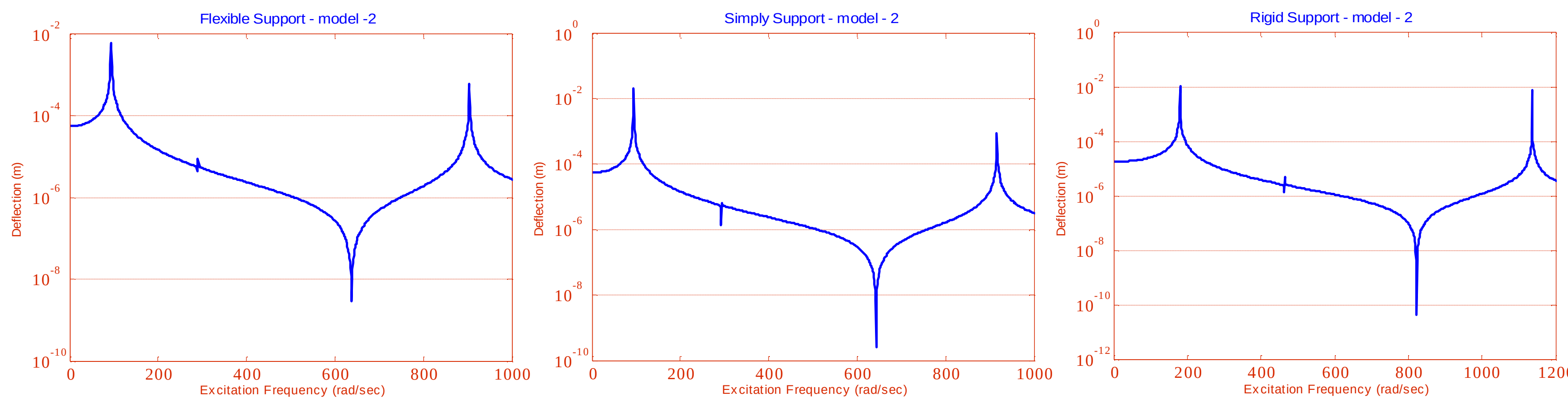

Figure (4) Deflection for (flexible, simply, rigid) supports pipe without fluid with variousxcitation frequencies at mid span of coppe pipe (model-2) represent three lowest natural frequency.

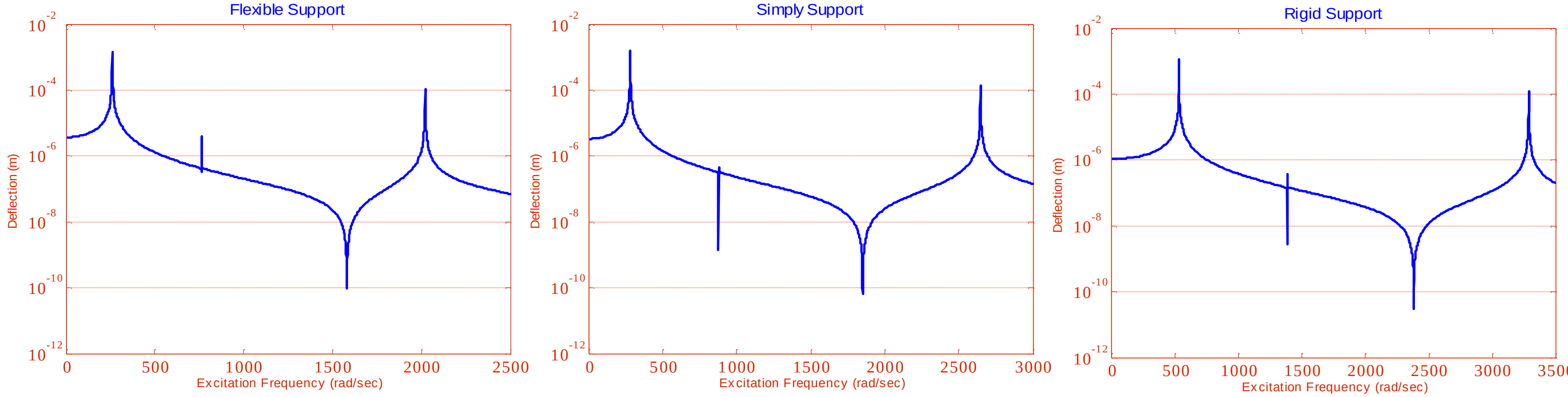

Figure (5) Deflection for (flexible, simply, rigid) supports pipe without fluid with various excitation frequencies at mid span of steel pipe (model-1) represent three lowest natural frequency.

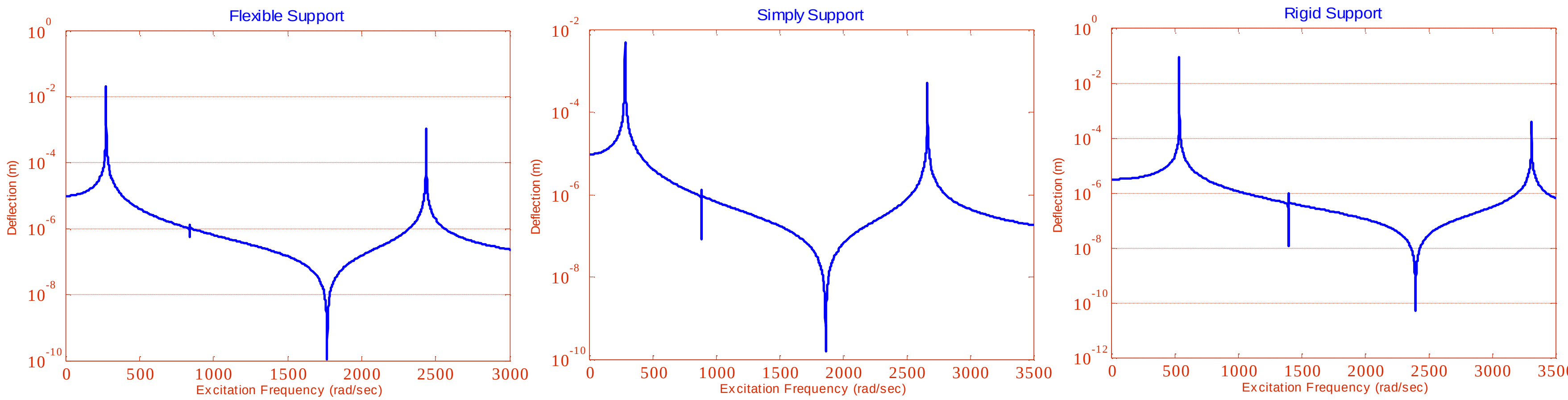

Figure (6) Deflection for (flexible, simply, rigid) supports pipe without fluid with various excitation frequencies at mid span of aluminum pipe (model-1) represent three lowest natural frequency.

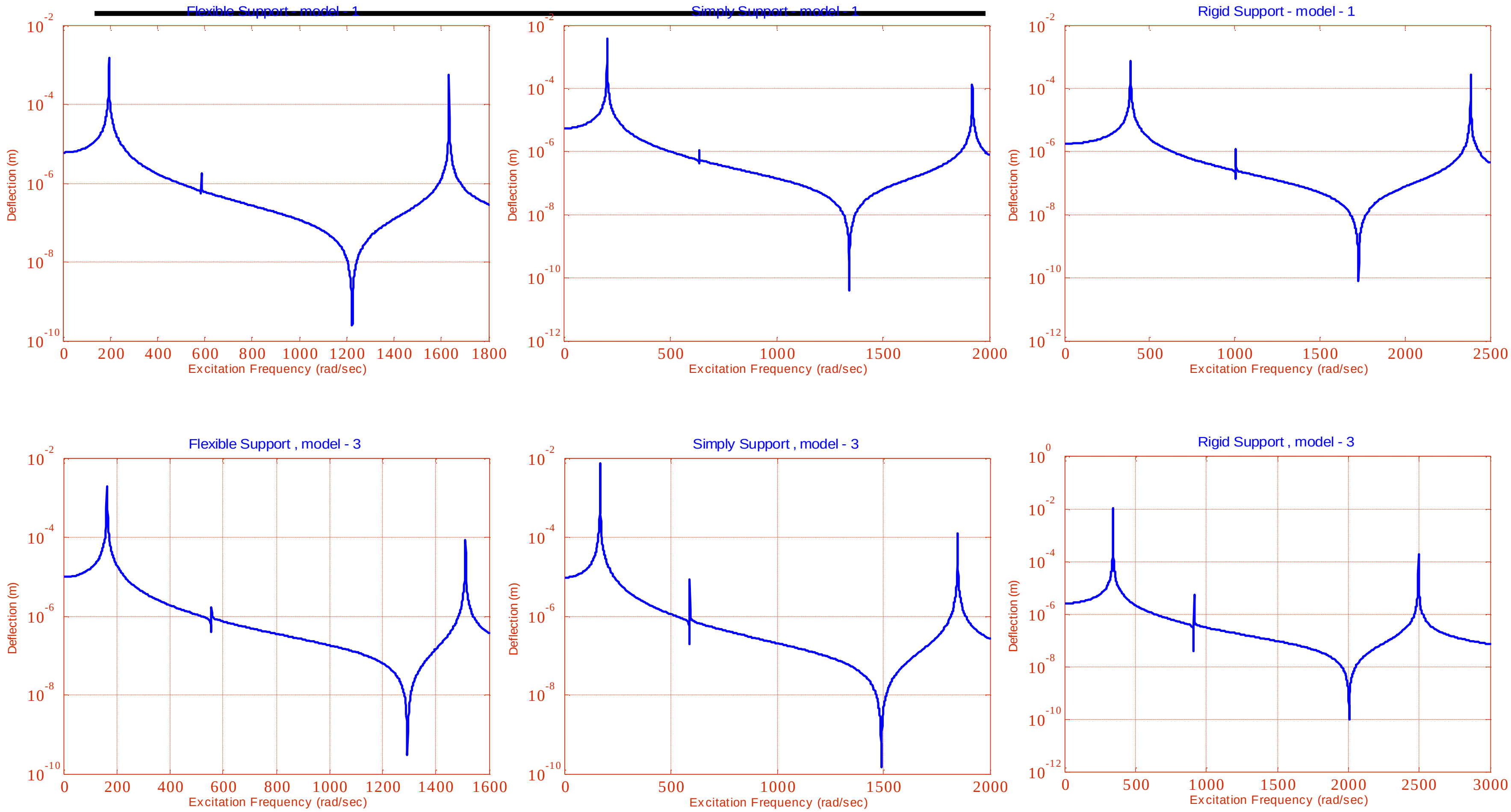

Figure (7) Deflection for (flexible, simply, rigid) supports pipe without fluid with various excitation frequencies at mspan of copper pipe with different lengths (model-1, model-3) represent three lowest natural frequency.

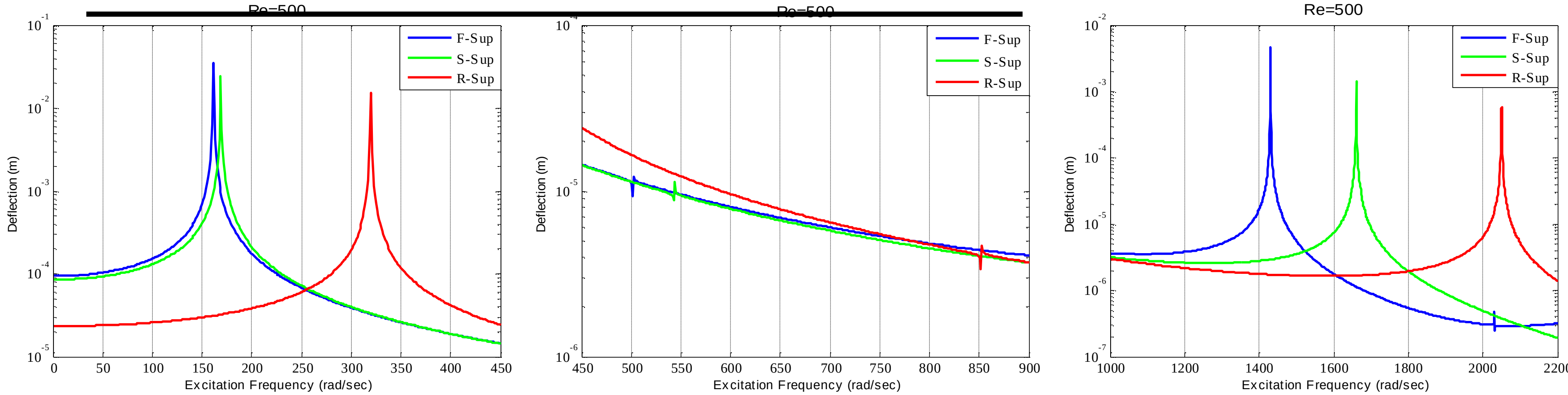

Figure (8) Deflection for (flexible, simply, rigid) supports pipe conveying fluid with various excitation frequencies at mid span of coppe pipe (model-1) represent (1st, 2nd, 3rd) natural frequency for Re=500.

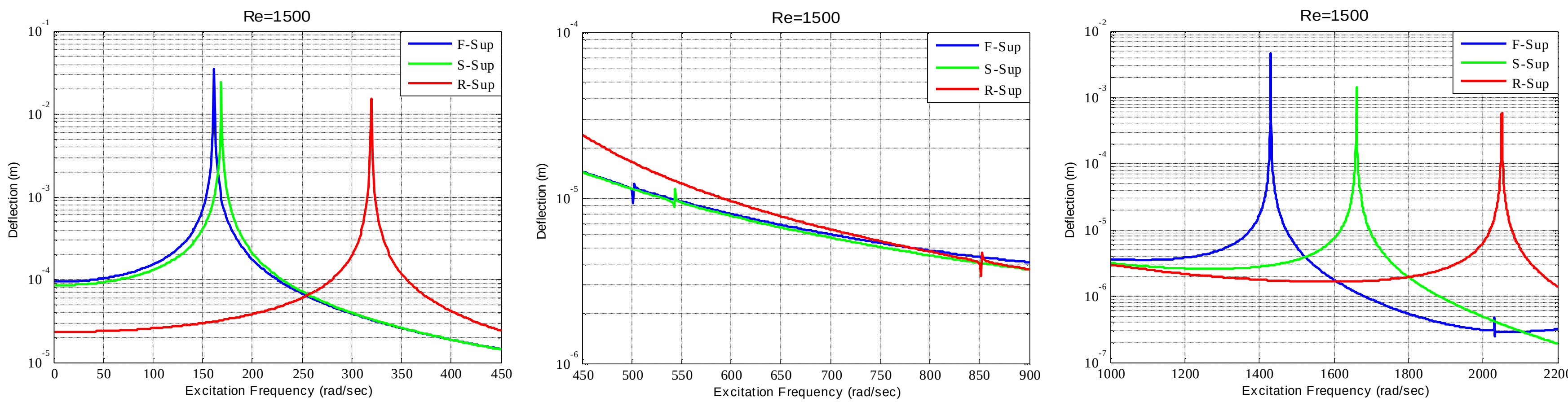

Figure (9) Deflection for (flexible, simply, rigd) supports pipe conveying fluid with various excitation frequencies at mid span of coppe pipe (model-1) represent (1st, 2nd, 3rd) natural frequency for Re=1500.