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DISTANCE ESTIMATION BETWEEN PIPE SUPPORTS NEAR ROTARY MACHINE BY USING DYNAMIC LOAD FACTOR (DLF)

Dr. Shwan Abdulmuhsin Zainalaabdeen*
* Mechanical Engineer in State Company for Oil Projects (SCOP) / Ministry Of Oil

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ABSTRACT

In this study a distance between pipe supports has been determined near a rotary machine (e.g. compressors) depending on the maximum dynamic load factor ($DLF_{max.}$), where it is advisable to keep this value less than (1.5) as a practice.

A useful tool that may be used to indicate the effect of dynamic response is the natural frequency of the piping system that excited by a rotary machine. The natural frequencies estimated for piping diameters (3", 6", &10") and specifically STD and XS thickness, taking into account the excitation effects that considered from a rotary machine with fixed – fixed case, so the value of lowest natural frequency (w_n) has been utilized with rotary machine run speed (Ω= (3000 RPM) = (50 Hz) = (314.159 rad/sec)) to be substituted in the equation of the dynamic load factor for undamped forced vibration which presented in mathematical model. Furthermore, an effect of ($\frac{\Omega}{Wn}$) ratio has been estimated, exhibited and recommended with respect to the ($DLF_{max.}$). A little consideration will show that the given data for the system have been considered to be compliant with pipe specifications and well-known companies practice, so some points have been concluded accordingly.

INTRODUCTION

Structural analysis is mainly concerned with finding out the behavior of a physical structure when subjected to force (In essence all these loads are dynamic).

The case study of this paper is relevant with the service transportation in oil field by rotary machine, where the pipe is located above ground (specifically on supports), so the excitation of the system represented by the revolution per minutes (rpm) of the rotary, in the other hand the natural frequency has been estimated according to the boundary condition and material properties.

The distinction is made between the dynamic and the static analysis on the basis of whether the applied action has enough acceleration in comparison to the structure's natural frequency. If a load is applied sufficiently slowly, the inertia forces (Newton's second law of motion) can be ignored and the analysis can be simplified as static analysis. Structural dynamics is a type of structural analysis which covers the behavior of structures subjected to dynamic loading (actions having high acceleration), Noting that the dynamic loads include people, wind, waves, traffic, earthquakes, and blasts or any excited sinusoidal effects. Any structure can be subjected to dynamic loading then the dynamic analysis can be used to find the dynamic load factor *(DLF)*, dynamic response, time history, and modal analysis as indicator for a likelihood failure for any case study, where the dynamic load factor is equal to 1.25 means that the dynamic deflection and stresses are 25% above those calculated by the static analysis [1]. Dynamic analysis for simple structures can be carried out manually, but for complex structures finite element analysis may be used to estimate the mode shapes and frequencies. This case study focused on the natural frequencies for piping system related with excitation (as sinusoidal effect) of rotary machine, consequently, the maximum dynamic load factors have been investigated ($DLF_{max.}$) as a powerful tool for design thereby the length between two pipe supports may be determined.

CASE STUDY

The following table arranged depending on the dimensions of the selected pipe which covered by ASME and well known formulas [2]:

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Table (1)

Pipe spec.	Moment of inertia (I) M^4	Weight of pipe (water filled) kg/m	$(I/m)^{1/2}$
3"(STD)	$1.25*10^{-6}$	16.076	$2.788*10^{-4}$
3"(XS)	$1.62*10^{-6}$	19.55	$2.878*10^{-4}$
6"(STD)	$1.17*10^{-5}$	46.94	$4.99*10^{-4}$
6"(XS)	$1.68*10^{-5}$	59.44	$5.31*10^{-4}$
10"(STD)	$6.69*10^{-5}$	111.22	$7.75*10^{-4}$
10"(XS)	$8.82*10^{-5}$	129.8	$8.24*10^{-4}$

Boundary Conditions

Referring to above table, natural frequencies have been evaluated according to boundary condition (fixed-fixed) which is represent more practical situation , so the value of fixation constant (a) that depends on boundary conditions and mode shapes is equal to (22.4) for 1^{st} mode[3] which is represent a most dangerous mode and as indicated hereunder .

Maximum Run speed of rotary machine (excitation frequency):

Excitation frequency has been supposed to be equaled to [(3000 RPM)=(50 Hz) =(314.159 rad/sec)] , where RPM represent rotary machine speed.

MAIN PROCEDURE

Calculate the natural frequencies of the piping, preferably by the following equation, for straight pipes without free-hanging valves and other related weights ; such that the following equation can be used for hand calculation of the piping natural frequencies [4],[5].

$$f_n = \frac{a}{2\pi L^2}\sqrt{\frac{EI}{m}} \quad \text{(Hz)} \ldots\ldots\ldots\ldots\text{(1)}$$

Where $(w_n = 2\pi f_n)$ (rad/sec.)

To achieve the condition that satisfy $(DLF_{max.})$ equaled or less than (1.5) the ratio of $\frac{\Omega}{Wn}$ should be equal to (0.33) that found by iteration on equation (3) and as indicated hereunder:

Table (2)

Ω (rad/sec.)	Wn (rad/sec.)	Ω/Wn
314.15	951.99	0.33

The following figures illustrates the first modes in our study that focused on the fixed – fixed case that is commonly used

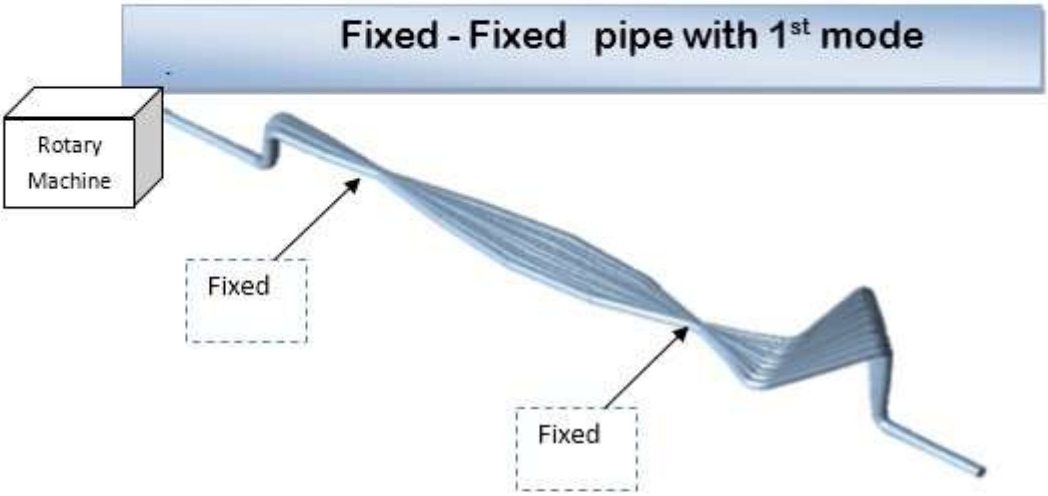
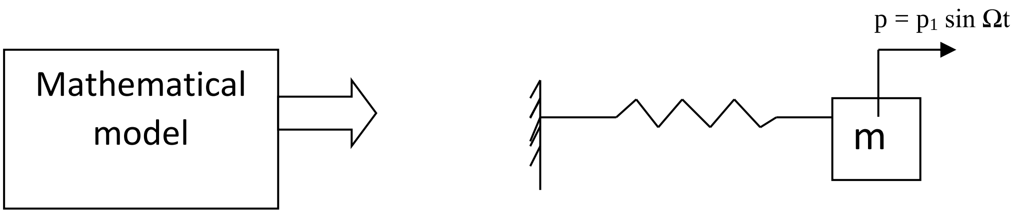


Figure (1) , fixed – fixed case with 1st mode

ASSUMPTIONS (ANALYTICAL ANALYSIS):

Suppose the mathematical model as undamped forced vibration [3], and assume a sinusoidal dynamic load (p) is applied to the system



$$
\left.
\begin{aligned}
& p = p_1 \sin \Omega t \\
& m\ddot{x} + kx = p_1 \sin\Omega t \\
& \ddot{x} + \frac{k}{m}x = \frac{p_1}{m}\sin \Omega t \\
& x = x_p + x_c
\end{aligned}
\right\} \quad \ldots\ldots\ldots\ldots (2)
$$

After solving particular and complementary part [4] the following equation will be found and as indicated hereunder:

$$
\text{DLF} = \frac{x(t)}{x_s t} = \frac{\sin \Omega t - \left(\frac{\Omega}{w_n}\right)\sin w_n t}{1-\left(\frac{\Omega}{w_n}\right)^2} \quad \ldots\ldots\ldots\ldots (3)
$$

Noting that damping effect not considered for the following reasons:

1. Damping effect is somewhat small.
2. It is preferable to ignore damping effect to study the system in more dangerous state.

RESULTS

The results have been found out and arranged clearly in following table, and all these data focused on the distance estimation as mentioned earlier for the ratio $\frac{\Omega}{Wn}$ that equaled to (0.33) to be recommended value that prevent failure.

Table (3)

Pipe spec.	Moment of inertia (I) M^4	Weight of pipe (water filled) (m)kg/m	(I/m)$^{1/2}$	Ω rad/sec.	Wn rad/sec.	L
3"(STD)	1.25*10^{-6}	16.076	2.788*10^{-4}	314.15	951.99	1.727
3"(XS)	1.62*10^{-6}	19.55	2.878*10^{-4}	314.15	951.99	1.754
6"(STD)	1.17*10^{-5}	46.94	4.99*10^{-4}	314.15	951.99	2.31
6"(XS)	1.68*10^{-5}	59.44	5.31*10^{-4}	314.15	951.99	2.38
10"(STD)	6.69*10^{-5}	111.22	7.75*10^{-4}	314.15	951.99	2.88
10"(XS)	8.82*10^{-5}	129.8	8.24*10^{-4}	314.15	951.99	2.97

Taking into account that the increasing of ratio $\frac{\Omega}{Wn}$ (for any (Ω) and (w) values) causes increasing of $(DLF_{max.})$ and as indicated hereunder:

Table(4)

$\frac{\Omega}{w}$	$(DLF_{max.})$
0.1	1.11
0.2	1.25
0.3	1.42
0.4	1.666
0.5	2
0.6	2.5
0.7	3.33
0.8	5
0.9	10
0.99	99.99

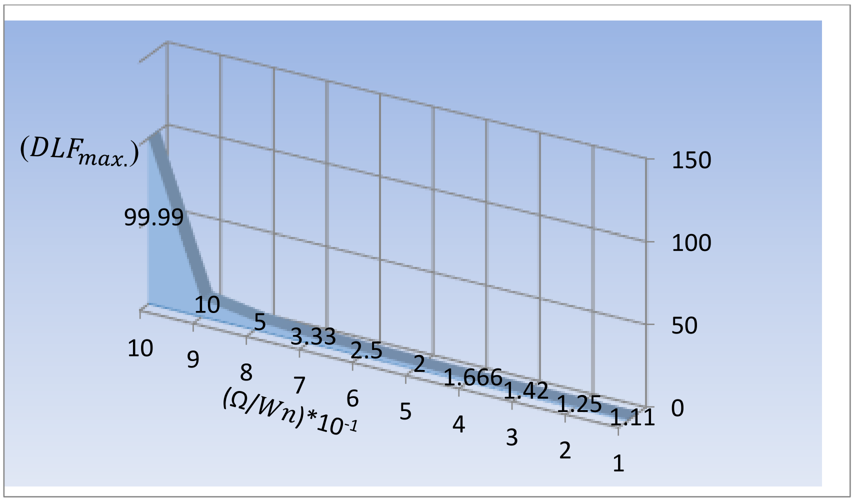


Fig.(2) $(DLF_{max.})$ *against* $\frac{\Omega}{Wn}$

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It is clear from Fig.(2) that the $(DLF_{max.})$ increased to be equaled to (10) at $\frac{\Omega}{Wn} = 0.9$, after that $(DLF_{max.})$ will be increased dramatically, so it is obvious that $\frac{\Omega}{Wn} = 1$ represent resonance state.

Example:

Let us substitute the ration $\frac{\Omega}{Wn} = 0.5$ in equation (3) the following result will be obtained:

DLF = 1.33 $\sin(\Omega t)$ - 0.666 $\sin w_n t$, Where: $(DLF_{max.})$ =1.33 +$|-0.66|$ = 2

Forced Part) (Free Part) (

Figure (3)
Forced Part
1.33 sin (Ωt)

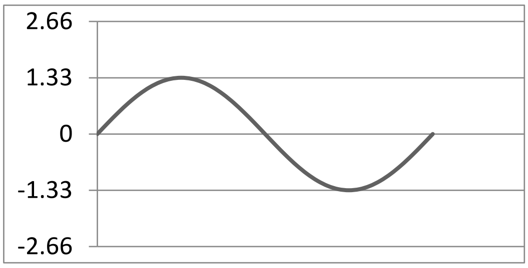


Figure (4)
Free Part
$-0.66\sin(w_n t)$

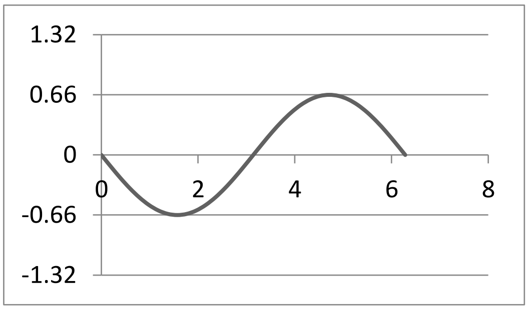
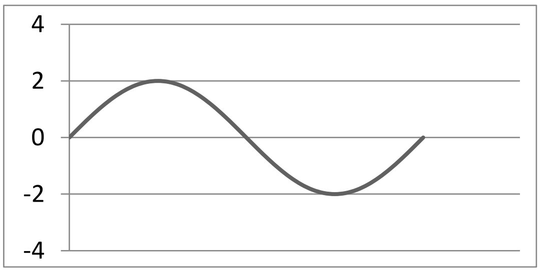


Figure (5)
$(DLF_{max.})$ =2



CONCLUSIONS

1. Since it is advisable DLF should not exceed (1.5), therefore the ratio of $\frac{\Omega}{Wn}$ should be less than or equaled to (0.33) to be recommended and satisfied with execution requirement that avoid the rupture due to cyclic loads and undesirable vibration, where the dynamic deflection and stresses less than or equaled to 50% above those calculated by the static analysis.
2. Distance between pipe supports will be increased when pipe size increased due to the fact that pipe mass and moment of inertia will be increased, see Table (1).
3. The equation $f_n = \frac{a}{2\pi L^2}\sqrt{\frac{EI}{m}}$ exhibit that f_n is reciprocal to L², and to increase the natural frequency (*to be far from excitation frequency to avoid resonance*) you have to decrease the length as much as possible

within reasonable distance and boundary conditions (*i.e. the shorter distance is advisable in terms of safety*).

4. By increasing the ratio of $\frac{\Omega}{Wn}$ the $DLF_{max.}$ increased, adding to this that $\frac{\Omega}{Wn}$ =0.9 related to the sharp behavior of the curve in figure (2) to be ready for resonance state.

SYMBOLS

f_n = natural frequency of the pipe (Hz)
Wn= natural frequency of the pipe (rad./sec.)
Ω =excitation of system from rotary machine (rad./sec.)
$x_s(t)$=static response (m)
$x(t)$=dynamic response (m)
a = fixation constant depending on boundary conditions and mode
Shapes (a = 22.4 for 1st mode & B.Cs. (Fixed- Fixed)).
L = span length between pipe supports (m).
E = young's modulus for pipe material=206.85*10^9 (N/m^2)
I = moment of inertia (m^4)
m = effective mass per unit length of carbon steel pipe (kg/m)
P= force (N)

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