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A study of Vibration Analysis for Gearbox Casing Using Finite Element Analysis

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***Abstract*- This paper contains the study about vibration analysis for gearbox casing using Finite Element Analysis (FEA). The aim of this paper is to apply ANSYS software to determine the natural vibration modes and forced harmonic frequency response for gearbox casing. The important elements in vibration analysis are the modeling of the bolted connections between the upper and lower casing and the modeling of the fixture to the support. This analysis is to find the natural frequency and harmonic frequency response of gearbox casing in order to prevent resonance for gearbox casing. From the result, this analysis can show the range of the frequency that is suitable for gearbox casing which can prevent maximum amplitude.**

I. INTRODUCTION

Gearbox casing is the shell (metal casing) in which a train of gears is sealed.From the movement of the gear it will produce the vibration to the gearbox casing.

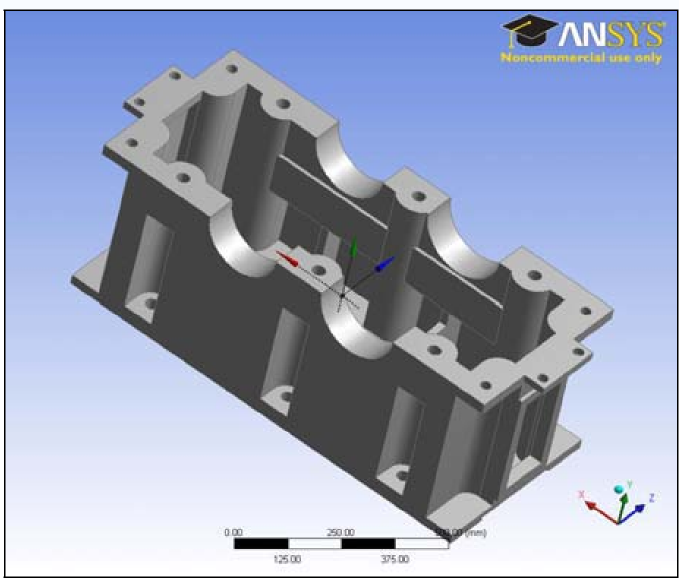

Figure 1. A gearbox casing

Reference [4] show that the study of natural frequency, consider a beam fixed at one end and having a mass attached to the other, this would be a single degree of freedom (SDoF) oscillator. Once set into motion it will oscillate at its natural frequency. For a single degree of freedom oscillator, a system in which the motion can be described by a single coordinate, the natural frequency depends on two system properties; mass and stiffness. The circular natural frequency, ω_n, can be found using the following equation:

$$\omega_n^2 = k/m \quad (1)$$

Where:
k = stiffness of the beam
m = mass of weight
ω_n = circular natural frequency (radians per second)

From the circular frequency, the natural frequency, f_n, can be found by simply dividing ω_n by 2π. Without first finding the circular natural frequency, the natural frequency can be found directly using:

$$f_n = (1/2\pi)(k/m)^{1/2} \quad (2)$$

Where:
f_n = natural frequency in hertz (1/seconds)
k = stiffness of the beam (Newton/Meters or N/m)
m = mass of weight (kg)

For the forced harmonic frequency, the behavior of the spring mass damper model need to add a harmonic force in the form below. A force of this type could, for example, be generated by a rotating imbalance.

$$F = F_0 \cos(2\pi f t). \quad (3)$$

Then, the sum the forces on the mass are calculate using following ordinary differential equation:

$$m\ddot{x} + c\dot{x} + kx = F_0 \cos(2\pi f t). \quad (4)$$

The steady state solution of this problem can be written as:

$$x(t) = X \cos(2\pi f t - \phi). \quad (5)$$

The result states that the mass will oscillate at the same frequency, f, of the applied force, but with a phase shift φ.

The amplitude of the vibration "X" is defined by the following formula.

$$X = \frac{F_0}{k} \frac{1}{\sqrt{(1-r^2)^2 + (2\zeta r)^2}}. \quad (6)$$

Where "r" is defined as the ratio of the harmonic force frequency over the undamped natural frequency of the mass–spring–damper model.

$$r = \frac{f}{f_n}. \quad (7)$$

The phase shift , φ, is defined by following formula. the base.

$$\phi = \arctan\left(\frac{2\zeta r}{1-r^2}\right). \quad (8)$$

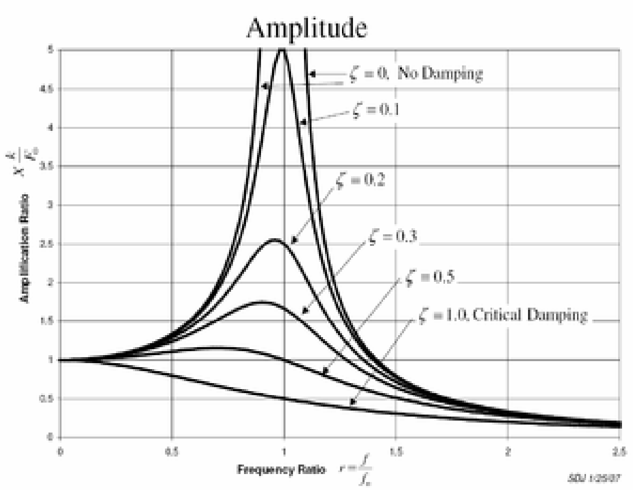

Figure 2. The frequency response of the system

The plot of these functions, called "the frequency response of the system", presents one of the most important features in forced vibration. In a lightly damped system when the forcing frequency nears the natural frequency ($r \approx 1$) the amplitude of the vibration can get extremely high. This phenomenon is called resonance (subsequently the natural frequency of a system is often referred to as the resonant frequency). In rotor bearing systems any rotational speed that excites a resonant frequency is referred to as a critical speed.

If resonance occurs in a mechanical system it can be very harmful – leading to eventual failure of the system. Consequently, one of the major reasons for vibration analysis is to predict when this type of resonance may occur and then to determine what steps to take to prevent it from occurring. As the amplitude plot shows, adding damping can significantly reduce the magnitude of the vibration. Also, the magnitude can be reduced if the natural frequency can be shifted away from the forcing frequency by changing the stiffness or mass of the system. If the system cannot be changed, perhaps the forcing frequency can be shifted (for example, changing the speed of the machine generating the force).

The following are some other points in regards to the forced vibration shown in the frequency response plots.

- At a given frequency ratio, the amplitude of the vibration, X, is directly proportional to the amplitude of the force F_0 (e.g. if double the force, the vibration doubles)
- With little or no damping, the vibration is in phase with the forcing frequency when the frequency ratio $r < 1$ and 180 degrees out of phase when the frequency ratio $r > 1$
- When r □ 1 the amplitude is just the deflection of the spring under the static force F_0. This deflection is called the static deflection δ_{st}. Hence, when r □ 1 the effects of the damper and the mass are minimal.
- When r □ 1 the amplitude of the vibration is actually less than the static deflection δ_{st}. In this region the force generated by the mass ($F = ma$) is dominating because the acceleration seen by the mass increases with the frequency. Since the deflection seen in the spring, X, is reduced in this region, the force transmitted by the spring ($F = kx$) to the base is reduced. Therefore the mass–spring–damper system is isolating the harmonic force from the mounting base – referred to as vibration isolation. Interestingly, more damping actually reduces the effects of vibration isolation when r □ 1 because the damping force ($F = cv$) is also transmitted to the base.

This analysis is to find the natural frequency and harmonic frequency response of gearbox casing in order to prevent resonance for gearbox casing. From the result, this analysis can show the range of the frequency that is suitable for gearbox casing which can prevent maximum amplitude.

II. DESIGN OF GEARBOX CASING

A. Joint Design

Equivalent bolt radius for bolts connecting gearbox halves is = 3r

When r = 16.5mm (inside radius)
=3×16.5
=49.5mm (outside radius)
When r = 13mm (inside radius)
=3×13
=39mm (outside radius)
Thickness is 1mm.

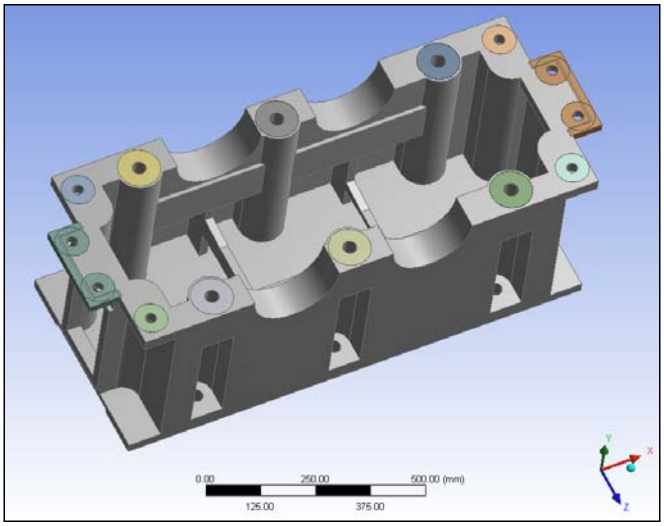

Figure 3: Bolts connecting gearbox halves

B. Supports Design

Equivalent bolt radius to support is
=1.25r
When r = 16.5mm (inside radius)
=1.25×16.5
=20.625 mm (outside radius)
Thickness is1mm

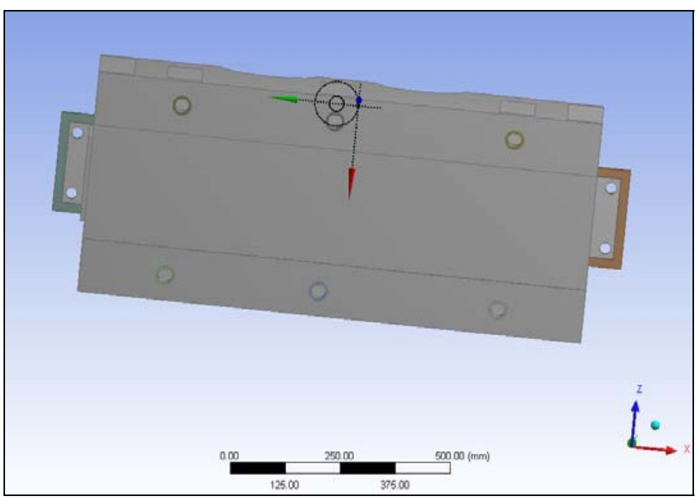

Figure 4: Bolt radius to support (bottom view of gearbox casing)

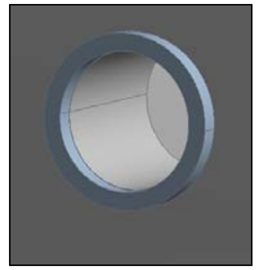

Figure 5: Details of one bolt for support

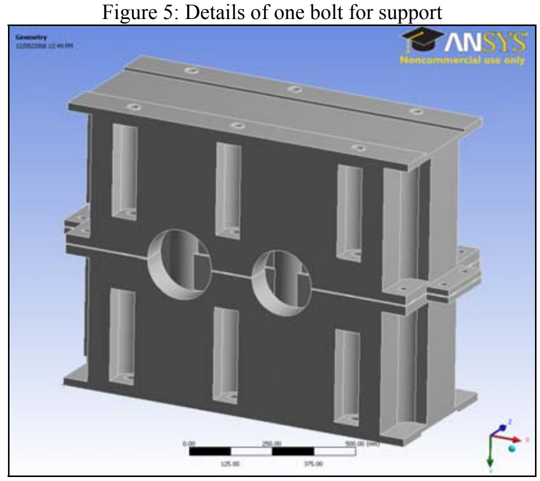

Figure 6: Full box of gearbox casing

III. MESH STRATEGY

The details of mesh strategy are defined in Table 1 and Figure 7.An appropriate mesh is selected to make sure this meshing can solve in 1 hour duration. This mesh is applied to whole object as one body meshing.

Table 1: Details of meshing strategy

Object Name	*Mesh*
State	Solved
Defaults	
Physics Preference	Mechanical
Relevance	0
Advanced	
Relevance Center	Coarse
Element Size	Default
Shape Checking	Standard Mechanical
Solid Element Midside Nodes	Program Controlled
Straight Sided Elements	No
Initial Size Seed	Active Assembly
Smoothing	Low
Transition	Fast
Statistics	
Nodes	71961
Elements	39946

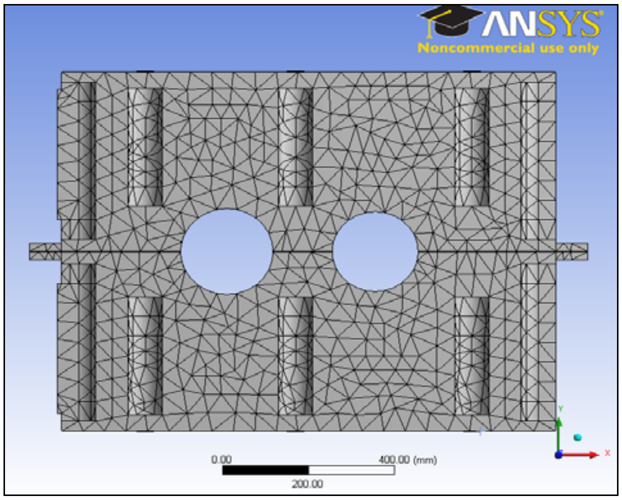

Figure 7: Actual mesh of gearbox casing

IV. BOUNDARY CONDITION AND APPLIED LOAD

This section described the details of applied load and boundary condition of natural vibrations and harmonic analysis.

A. Natural Vibration Analysis

A modal analysis is performed with number of modes is 10.The details of the support is in Table 2 and Figure 8.

Table 2: Details of boundary condition

Object Name	*Fixed Support*
State	Fully Defined
Scope	
Scoping Method	Geometry Selection
Geometry	6 Faces
Definition	
Type	Fixed Support
Suppressed	No

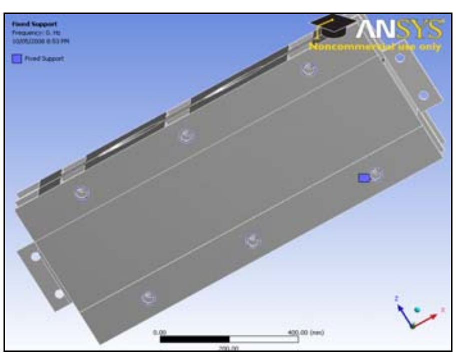

Figure 8: Actual fixed support on bottom created circle surface

B. Harmonic Frequency Response Analysis

In the harmonic frequency response analysis, the fixed support is exactly same condition in Figure 8.

In this analysis, 1MPa pressures is applied to the upper half of the bearings on one side of the gearbox and to the lower half of the other side for a frequency range from zero to 1.2 times the frequency of the tenth vibration mode.

This 1MPa pressure is applied normal to the surface according to the Table 3 and Figure 9.

Table 3: Details of applied pressure and fixed support

Object Name	*Fixed Support*	*Pressure*	*Pressure 2*	*Pressure 3*	*Pressure 4*
State	Fully Defined				
Scope					
Scoping Method	Geometry Selection				
Geometry	6 Faces	1 Face			
Definition					
Type	Fixed Support	Pressure			
Suppressed	No				
Define By		Normal To			
Magnitude		1. MPa			
Phase Angle		0. °			

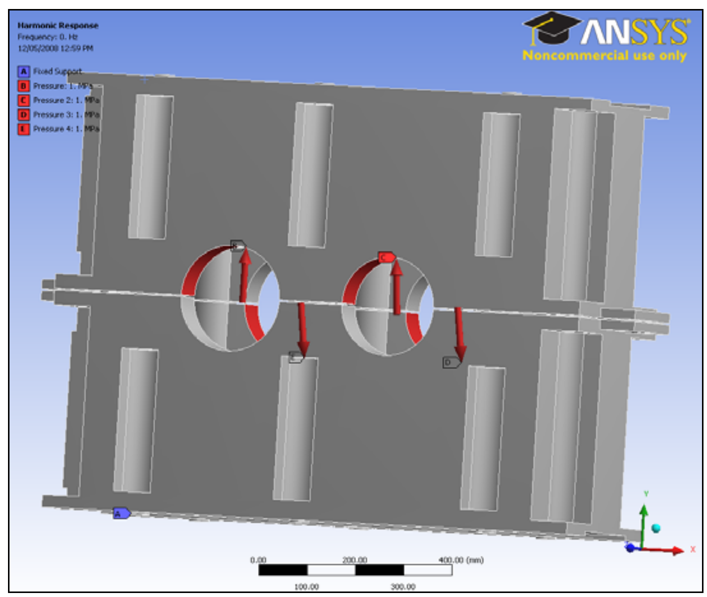

Figure 9: The actual applied load in gearbox casing.

V. RESULT

These results for natural vibration analysis and harmonic frequency response analysis is done using ANSYS 11.0

A. Result of Natural Vibration Analysis

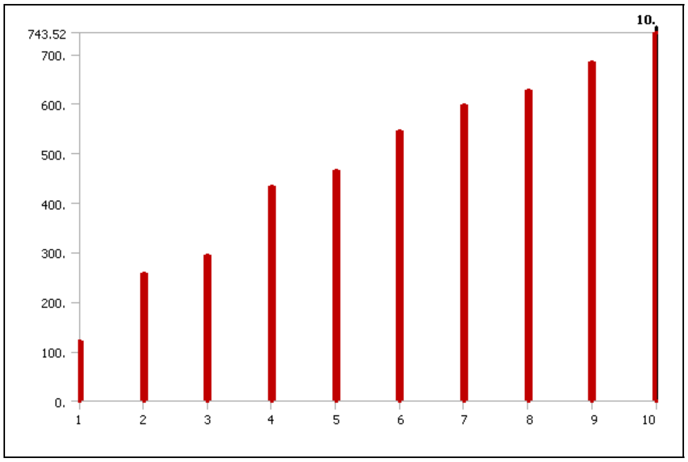

Figure 10: Result of frequency corresponding to 10 modes for normal vibration analysis.

From these result, 10 lowest vibration frequencies are:

Table 4:10 lowest frequencies for natural vibration analysis

Mode	Frequency [Hz]
1.	120.93
2.	256.71
3.	295.27
4.	434.45
5.	464.22
6.	545.23
7.	598.62
8.	627.11
9.	683.95
10.	743.52

B. Result of Harmonic Frequency Response Analysis

In this harmonic frequency response analysis, frequency range need to be set up from zero to 1.2 times the frequency of the tenth vibration mode.In Table 4, tenth vibration mode is 743.52 Hz.

1.2×the frequency of the 10 th vibration mode
= 1.2 × 743.52
= 892.224 Hz
From this result, 0-892 Hz frequency range is applied.

Table 5: Applied frequency in Harmonic Frequency Response Analysis

Object Name	*Analysis Settings*
State	Fully Defined
Options	
Range Minimum	0. Hz
Range Maximum	892. Hz
Solution Intervals	200

All the result is from one vertex as in the Table 5.This point is selected because this point is the maximum total displacement in the Figure 11.

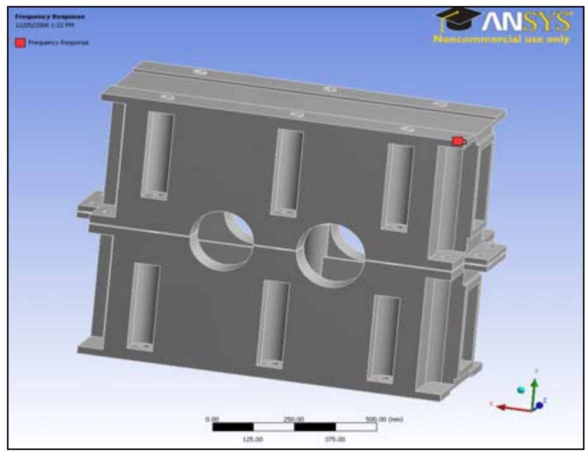

Figure 11: Analysis point

A. Result of Harmonic Frequency Response Analysis

Y-axis result.

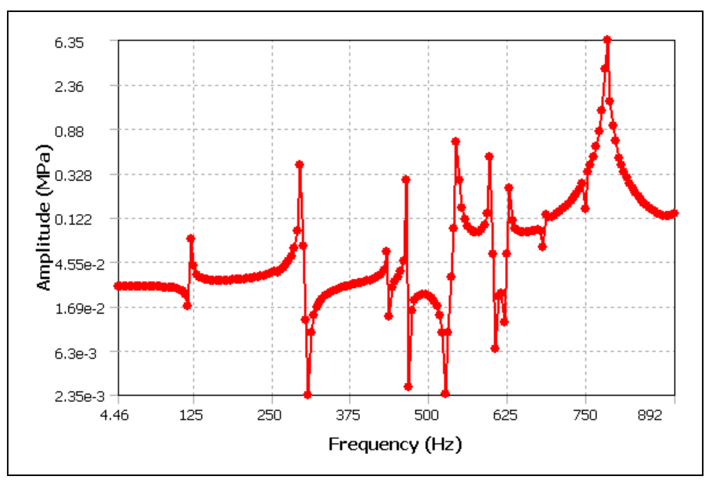

Figure 12: Details of Y-axis result for normal stress

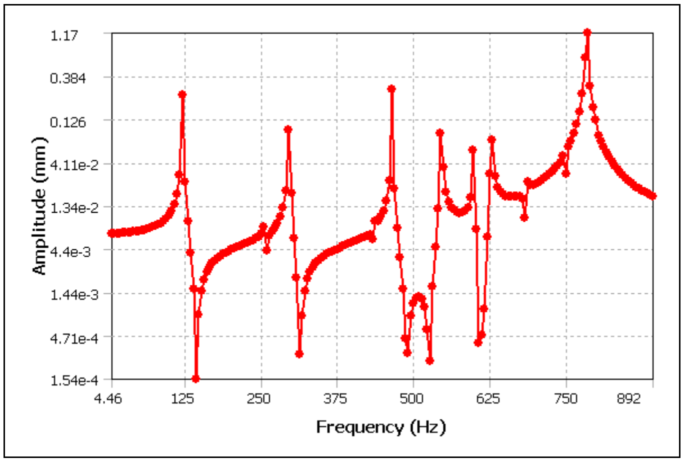

Figure 13: Details of Y-axis result for directional deformation.

X-axis result.

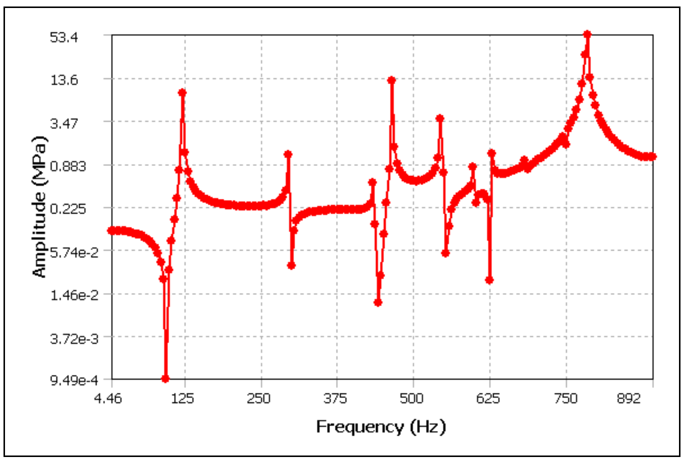

Figure 14: Details of X-axis result for normal stress

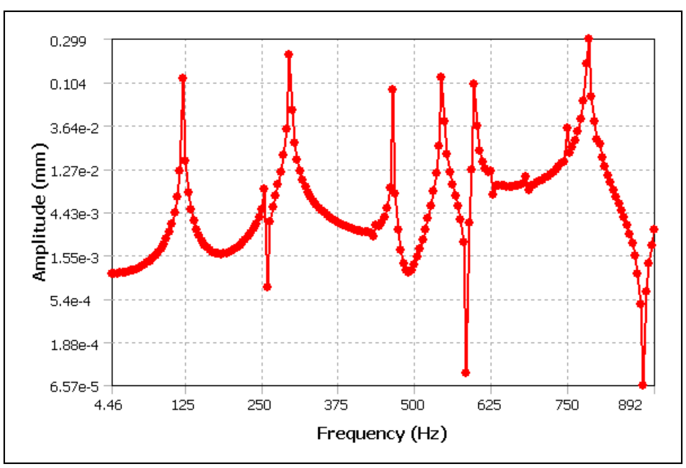

Figure 15: Details of X-axis result for directional deformation

CONCLUSION

From Figure 12 until Figure 15, the conclusion is:

(a)In this analysis, pressure is applied to surface as in Figure 9 as a normal to that surface. This is meaning that force is mainly applied to X-axis and Y-axis. Due to this reason, only result for Y-axis and X-axis is more considerable in this harmonic analysis.

(b) For the Y-axis and X-axis, the first maximum amplitude for normal stress and directional deformation are happen at 124.8 Hz. At this frequency, the resonance is occurred.

(c) In this analysis, first resonance is happen when the ratio of harmonic forced frequency over natural frequency is

r = first resonance in harmonic forced frequency/first modal natural frequency

= 124.8/120.93

= 1.032 ≈ 1

(d) In order to prevent the resonance, frequency ratio need to be setup to be less than 1.When r<<1 the amplitude is just the deflection of the spring under the static force F_0. This deflection is called the static deflection δ_{st}. Hence, when r<<1 the effects of the damper and the mass are minimal. The magnitude can be reduced if the natural frequency can be shifted away from the forcing frequency by changing the stiffness or mass of the system. If the system cannot be changed, perhaps the forcing frequency can be shifted.

(e) In this study, frequency ratio can set to 0.25 from the first modal natural frequency analysis in order to prevent resonance.

Forced frequency = 0.25× natural frequency

= 0.25×120.93 = 30.2325 Hz

Static deflection can be achieved if forced frequency is from 0 Hz to 30.2325 Hz.

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