Mode Shapes and Natural frequency of multi Rotor system with ANSYS 14

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Abstract:

Rotating systems at running speed near the natural frequencies are dangerous to excessive angular deflections and hence more stresses which can cause rotating components failure or due to fatigue failures. The objective of the present work is to study the torsional vibration characteristics of multi-rotor using finite element method. Finite element method (FEM) is a numerical technique based on principle of discretization to find approximate solutions to engineering problems. The information about the natural frequencies for rotating systems can help to avoid system failure by giving the safe operating speed range. In the present work, finite element method has been used to find these natural frequencies for different possible cases of multi-rotor systems. The various mode shapes for several cases are also shown to illustrate the state of the system at natural frequencies. The results obtained have been compared with Holzer's method and Ansys 14 and ansys 14.5 software version to establish the effectiveness of finite element method for such systems.

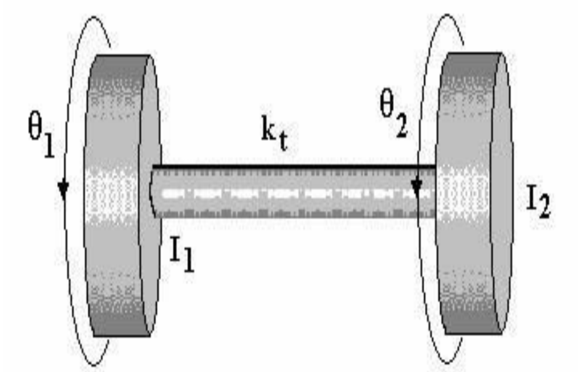
Introduction

Most of the machines used in industries are rotary in nature and hence are subjected to torsional vibrations. These vibrations can damage the machine components. Torsional vibrations are harder to detect and hence are more dangerous.significantly if the system speed is close to the natural frequencies of the system, hence it's important to find the natural frequencies for rotating systems for safe operations. Excessive strain at speeds near the natural frequencies causes excessive stress which causes component failures . Holzer's method-The natural frequency and mode shapes of a multimass lumped parameter system can be determined by this iteration method as devised by Holzer. It's applicable to forced , free, damped, undamped and semi-definite systems as well.

TWO ROTOR SYSTEM

Consider a shaft with torsional stiffness k_t connecting two inertias I_1 andI_2. If the shaft is free to rotate the torsional oscillation will take the form of both ends twisting but some point in between will not be twisting. This is a node. The shaft must of course be supported in at least two bearings.

The natural frequency can be derived from the previous work. For two rotors, T_2 =0



$$\theta_2 = \theta_1 - \frac{\omega^2 I_1 \theta_1}{k_{t1}} \qquad T_1 = \omega^2 I_1 \theta_1 \qquad T_2 ='$$

$$T_2 = 0 = T_1 + \omega^2 I_2 \theta_2 = \omega^2 I_1 \theta_1 + \omega^2 I_2 \theta_2$$

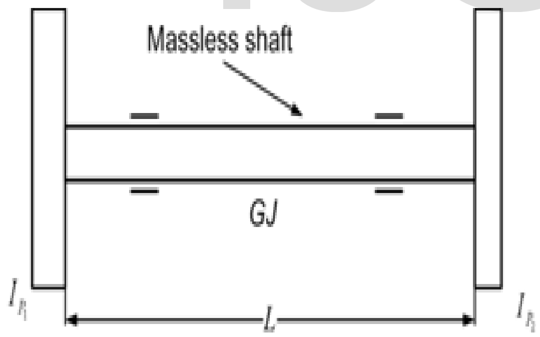
$$0 = \omega^2 I_1 \theta_1 + \omega^2 I_2 \left(\theta_1 - \frac{\omega^2 I_1 \theta_1}{k_{t1}} \right) \text{ simpli}$$

$$\omega_n^2 = k_t \frac{I_1 + I_2}{I_1 I_2}$$

The node will be somewhere between tl

WORKED EXAMPLE

Determine natural frequencies and mode shapes for a rotor system as shown in Figure 6.8. Neglect the mass of the shaft and assume that discs as lumped masses. The shaft is 1 m of length, 0.015m of diameter, and 0.8×1011 N/m of modulus of rigidity. Discs have polar mass moment of inertia as I1 = 0.01 kg-m2 and I2= 0.015 kg-m2.



$$K_t = \frac{GJ}{L}$$

$$= \frac{0.73 \times 10^{11} \times \pi \times (0.015)^4}{32 \times 1}$$

$$= 394.61 \text{Nm/rad}$$

Natural frequency given as :

$$\omega_{nf1} = 0$$

$$\omega_{nf2} = \sqrt{(0.01 + 0.015) \times 397.61/(0.01 \times 0.015)}$$

$$= 257.43 \text{ rad/sec}$$

$$\frac{\Phi z1}{\Phi z2} = -\frac{Ip1}{Ip2} = -\frac{0.015}{0.01} = -1.5$$

Which means disc 1 would 1.5 times angular displacement amplitude as compared to disc 2 ,however ,in opposite direction. The node position can be obtain as

$$\frac{l1}{l2} = \frac{Ip2}{Ip1} = \frac{0.015}{0.010} = 1.5$$

$$.l_1 + l_2 = 1$$

Hence ,we get the node location as L1 =0.6m from disc 1.It can be verified that equivalent two single mass cantilever rotor will have same natural frequency as

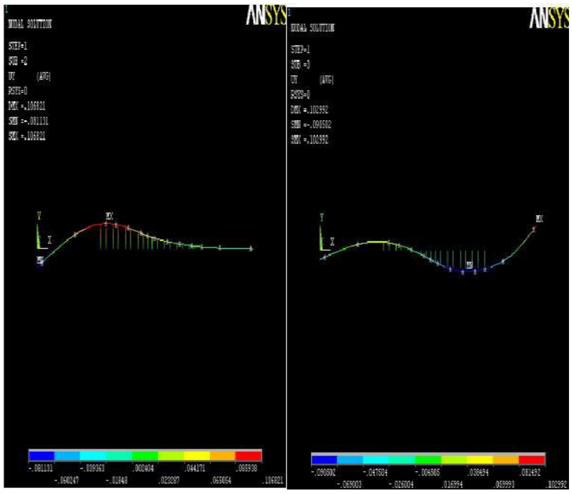
$$K_{t1} = \frac{GJ}{l1} = \frac{0.8 \times 10^{11} \times \pi \ (0.015)^4}{32 \times 0.6} = 662.68 \ Nm/rad$$

$$K_{t2} = \frac{GJ}{l1} = \frac{0.8 \times 10^{11} \times \pi \ (0.015)^4}{32 \times 0.4} = 994.38 \ Nm/rad$$

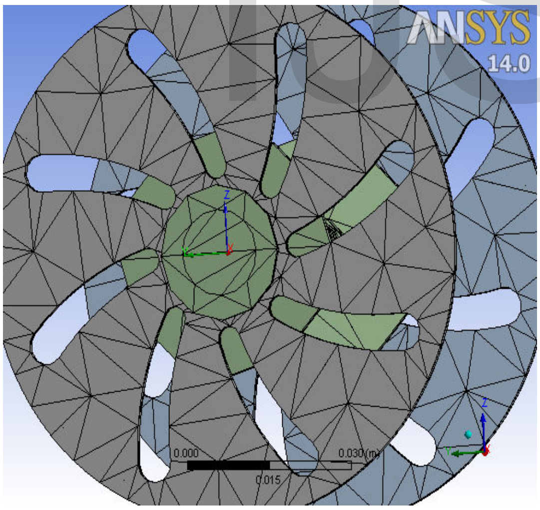
$$\omega_{nf1} = \sqrt{\frac{Kt1}{Ip1}} = \sqrt{\frac{662.68}{0.01}} = 257.42 \ rad/sec$$

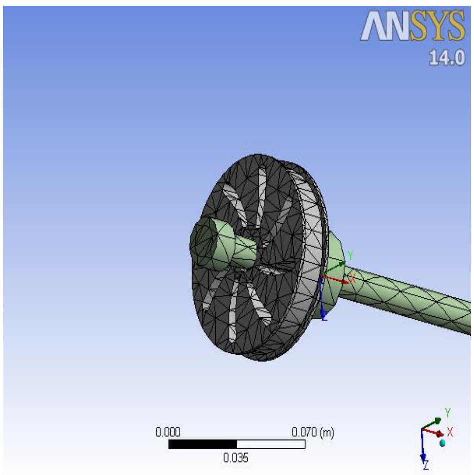
$$\omega_{nf2} = \sqrt{\frac{Kt2}{Ip2}} = \sqrt{\frac{994.03}{0.015}} = 257.42\ rad/sec$$

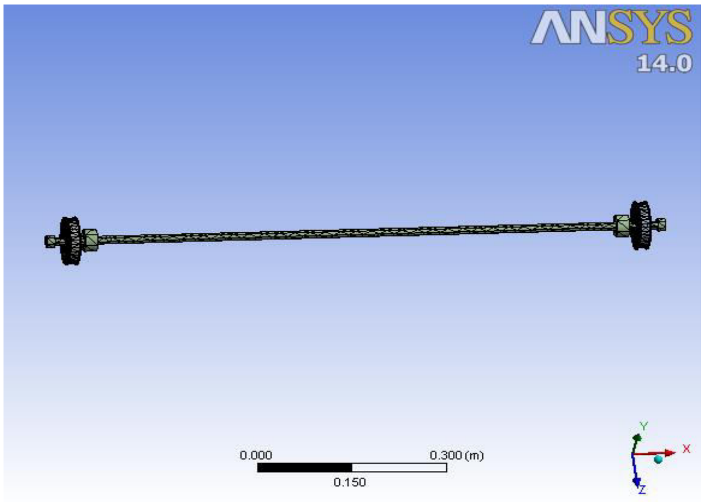
Mode shapes:.



Meshing :







Ansys 14 version Analysis

Mode	Frequency [Hz]
1.	0
2.	48.12
3.	48.18
4.	88.54
5.	159.43
6.	145.23

Conclusion:

The study of torsional vibrations is very important as it can cause component failures or excessive deflections which can lead to fatigue failures if the machine is vibrating at speeds near the natural frequencies of the vibrating system. Different methods have been discussed to find these natural frequencies. FEM is the most effective way as it provides a global mass and stiffness matrix which can be derived by discretizing the system and assembling property matrices of the elements. Holzer's method has specifically been used to have a comparative study between the two methods. It being an iteration method based on hit and trial takes long time for complex system with branches. Several cases of single and branched system have been studied and the results obtained by FEM are in good agreement with those obtained by the Holzer's method. Compare with ANSYS 14 with Different lower mode shapes have been plotted for some cases for a better understanding of the results obtained. The percentage error in results with respect to FEM has been shown as the bar graph for all the cases. Thus we can see that FEM is a very effective method to calculate the natural frequencies and the corresponding mode shapes. Moreover, the ease of its modelling, less time consumption.And there is a difference between HOLZER method and ANSYS 14 software because HOLZER is an approximation method.

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