International Journal of Latest Technology in Engineering, Management & Applied Science (IJLTEMAS)
Volume VI, Issue IV, April 2017 | ISSN 2278-2540

Modelling and Simulation of Single Rotor System

Amit. Malgol[1], Yogita. Potdar[2]

[1]*M. Tech, Department of Mechanical Engineering, GIT, Belgavi, Karnataka, India.*
[2]*Assistant Professor, Department of Mechanical Engineering, GIT, Belgavi, Karnataka, India*

Abstract: **- This paper presents study of rotor shaft system for three different position of the disk, for a simply supported case and it is important to determine natural frequency, critical speeds and amplitudes of rotor system. This characteristic are found by using ANSYS parametric design tool. Modal, harmonic and transient cases are carried out for the single rotor system. The results obtained for this analysis are useful for design of rotor system. The results obtained from analytical method have close agreement with the results obtained from ANSYS results.**

Keywords: **- Rotor, natural frequency, critical speeds, amplitudes, damping.**

I. INTRODUCTION

Rotor dynamics is branch of physics which deals with study of behaviour of rotating systems under application of dynamic forces. Rotor is part of system i.e. disks, blades or couplings mounted on shaft is called as rotor. Rotor is used to convert one form of energy into another form hence rotational energy must be maximum, so we must reduce vibrational energy as much as possible, so they have wide range of applications in many industries as well as household applications so we need to analyse the system to prevent catastrophic failures. Applications such as centrifugal pumps, generators, motors, compressors, blowers, sewing machine, steam turbines, gas turbines, aero engines, main and tail rotors of helicopters. In rotating system flexural vibrations are main cause as compared to torsional vibration and axial vibration. Unbalance in rotor gives raise to forces and moments in rotor this generates flexural vibration in rotor. The vibrations perpendicular to the axis of rotation such vibrations are known as flexural vibrations. Whirling is one of the main cause for failures of rotating systems due unbalance of rotor i.e. due the manufacturing defects centre of gravity of shaft doesn't coincides with axis of rotation, misalignment of rotor shaft and bearings, due to loose supports or if the machine is operated at critical speeds may lead to catastrophic failure of system.

Generally rotors rotate at high speeds, when the natural frequency of system is equal to the critical speed resonance occur. Resonance is most common problem in rotating systems. In rotating system if there is some percentage of vibration in machine, these vibrations are magnified by resonance. At these critical speeds the amplitudes of vibration goes on increasing this cause rotor to bend and twist so this cause rubs or wear and tear and collide with adjacent parts of system hence excessive force are developed and hence leads to failure. So determination natural frequency, critical speeds and amplitudes of vibrations are very important in rotor dynamics. As a designer by changing mass, stiffness, position of disk etc. such design modification to change critical speeds of system so as to operate in a suitable environment. To reduce whirling amplitudes, we must avoid rotating at critical speeds of the system or squeeze film damper is suitable. By using damper whirling amplitudes are reduced as well as reduces forces on the supports.

II. METHODOLOGY

2.1 ANSYS

The rotor model consist of shaft and disk modelled in ANSYS by considering Beam 2D elastic element and two sets of real constants for shaft and disc are considered, by creating four key points and drawing straight line, as per shaft and disc real constants sets are selected for lines created. To create rotor model table 2.1 and 2.2 data of material properties and dimensions are considered. Figure 2.3 shows the rotor model.

Table 2.1: Material properties of rotor

S.NO	Parameters	Value
1	Shaft and disk material	Mild steel
2	Young's modulus (E)	$2\times 10^{11}\ N/m^2$
3	Density (ρ)	$7800\ kg/m^3$
4	Poisson's ratio (μ)	0.33

Table 2.2: Dimensions of disk and rotor

S.NO	Parameters	Value
1	Diameter of shaft (d)	0.01 m
2	Diameter of disk (D)	0.15 m
3	Thickness of disk (t)	0.01 m
4	Length of shaft (L)	0.4 m
5	Mass of disk (M)	1.3783kg

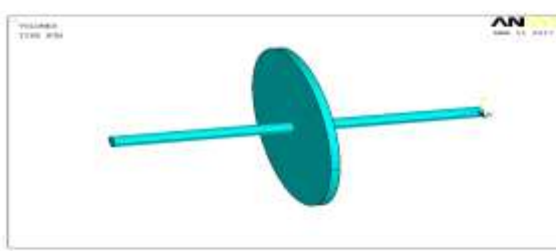

Figure 2.3: Rotor model in ANSYS

2.2 Modal analysis of rotor in ANSYS

In modal analysis, three different disk positions are considered to determine natural frequencies and respective mode shapes at this natural frequency and critical speeds at this three different disk position for a simply supported rotor.

2.21 For disk position a = 0.2 m and b = 0.2 m

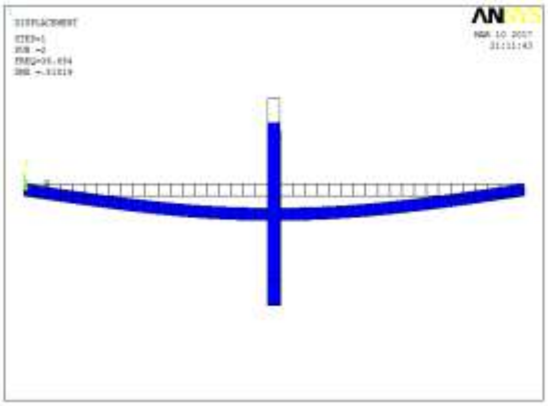

Figure 2.4: Mode 1 for a = 0.2 m and b = 0.2 m

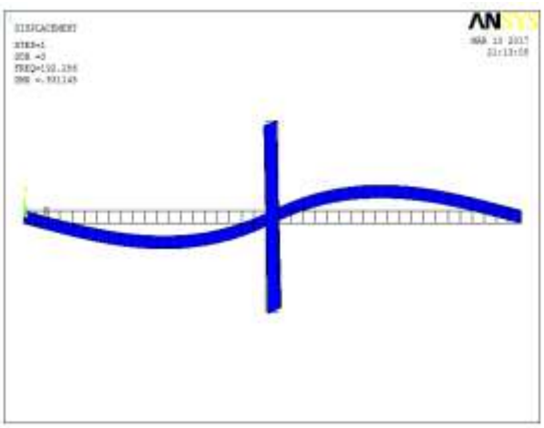

Figure 2.5: Mode 2 for a = 0.2 m and b = 0.2 m

2.22 For disk position a = 0.133 m and b = 0.267 m

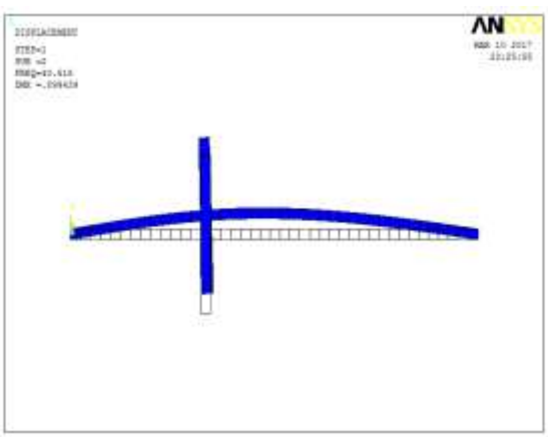

Figure 2.6: Mode 1 for a = 0.133 m and b = 0.267 m

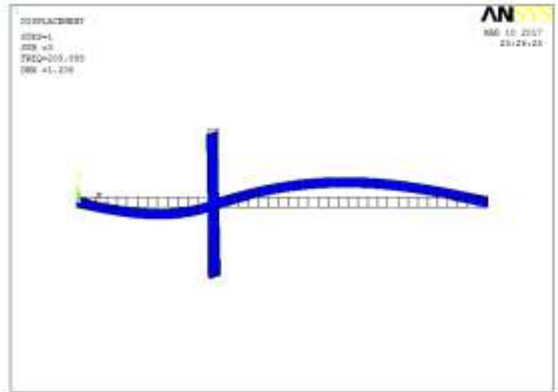

Figure 2.7: Mode 2 for a = 0.133 m and b = 0.267 m

2.23 For disk position a = 0.066 m and b = 0.334 m

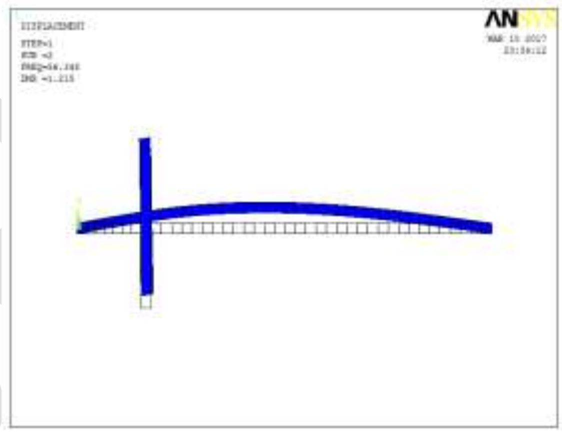

Figure 2.8: Mode 1 for a = 0.066 m and b = 0.334 m

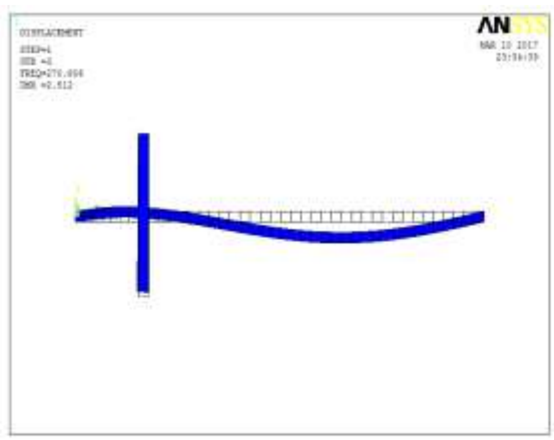

Figure 2.9: Mode 2 for a = 0.066 m and b = 0.334 m

2.3 Frequency responses of rotor in ANSYS

2.31 For disk position a = 0.2 m and b = 0.2

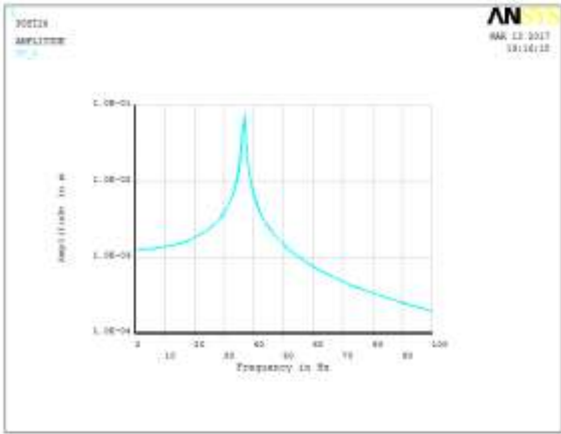

Figure 2.10: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0

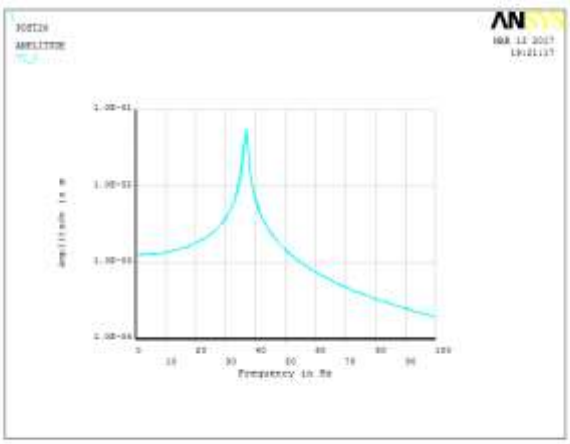

Figure 2.11: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.0073265

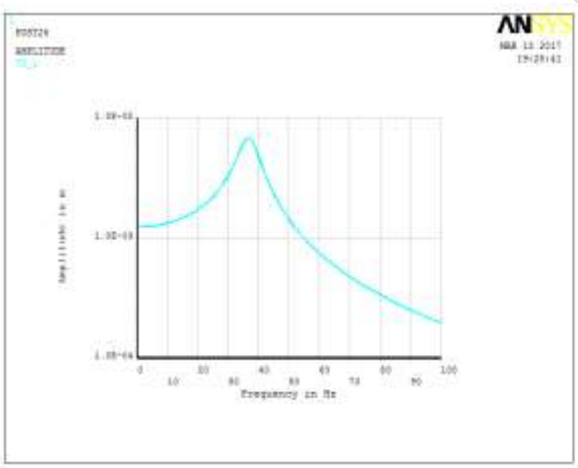

Figure 2.12: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.091235

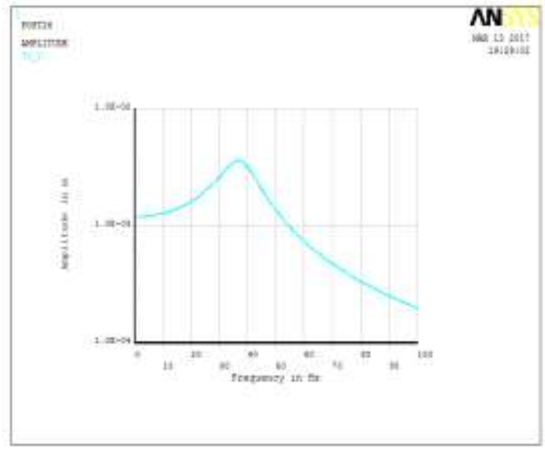

Figure 2.13: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.1748

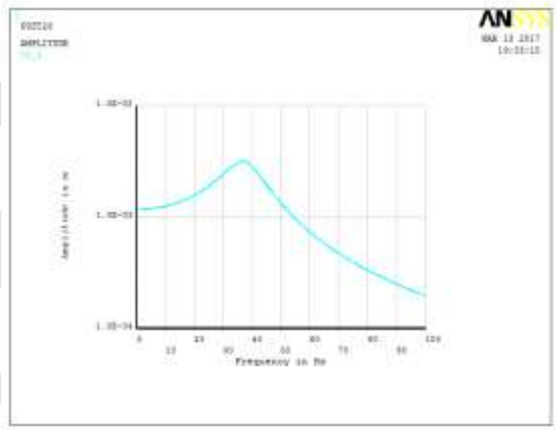

Figure 2.14: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.2

2.32 For disk position a = 0.133 m and b = 0.267 m

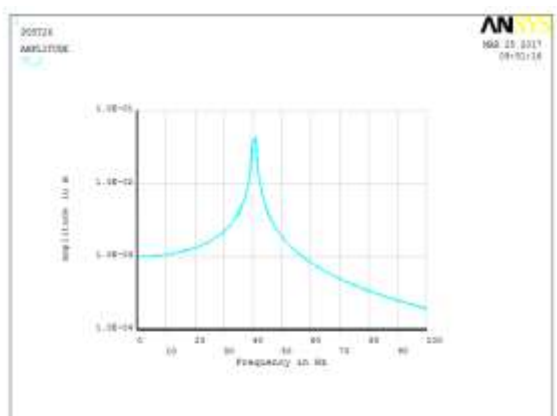

Figure 2.15: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0

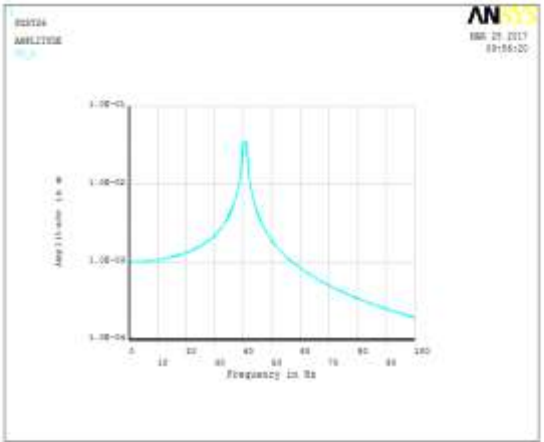

Figure 2.16: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.0073265

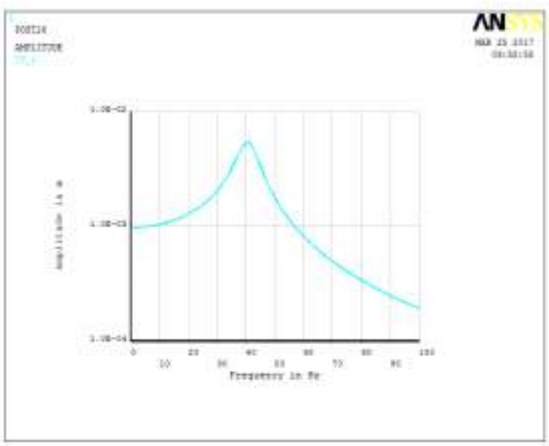

Figure 2.17: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.091325

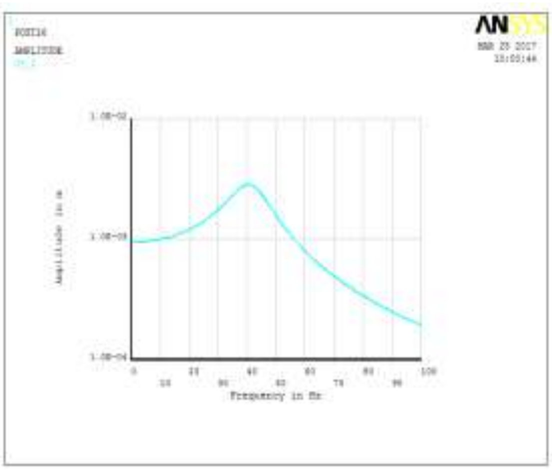

Figure 2.18: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.1748

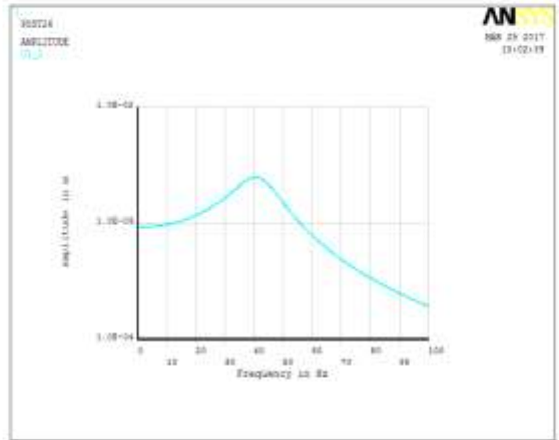

Figure 2.19: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.2

2.33 For disk position a = 0.066 m and b = 0.334 m

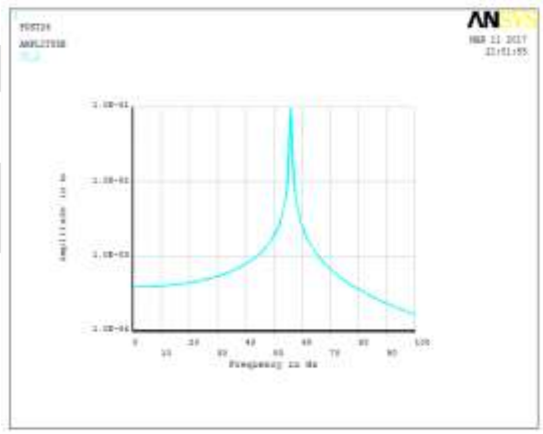

Figure 2.20: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0

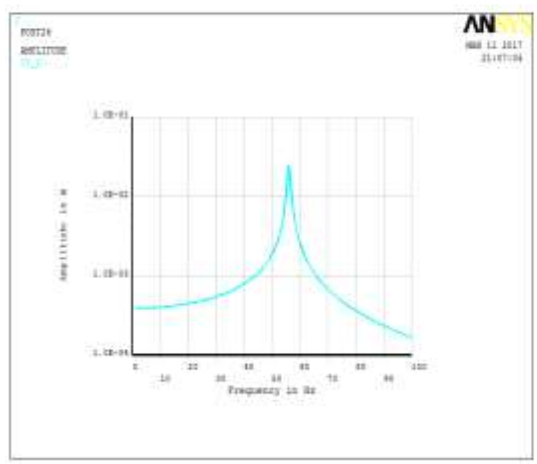

Figure 2.21: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.0073265

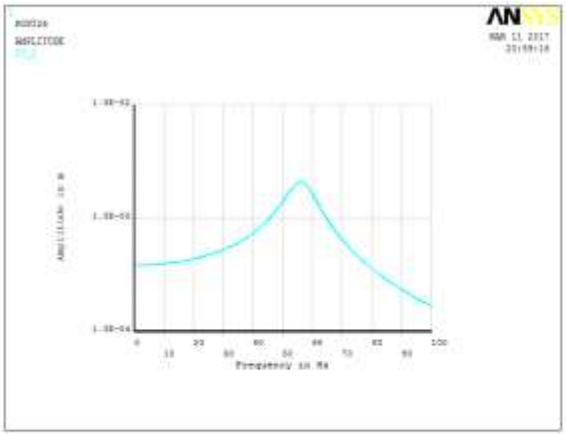

Figure 2.22: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.091325

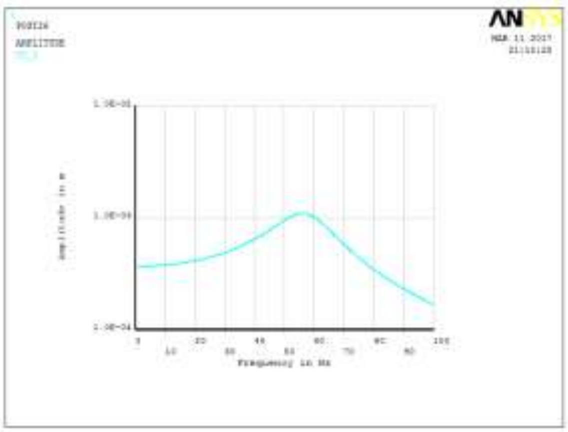

Figure 2.23: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.1748

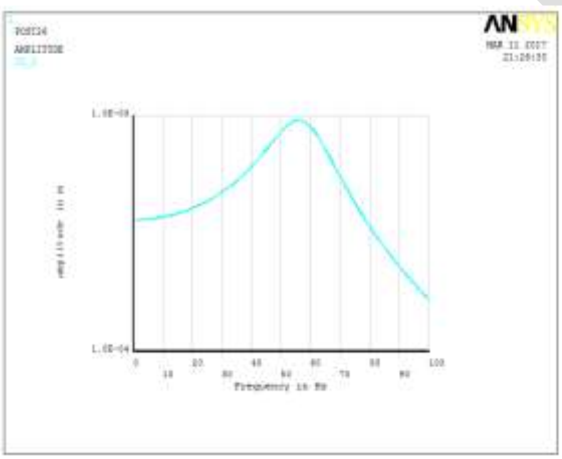

Figure 2.24: Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.2

2.4 Transient response

In transient vibration, the amplitudes of vibration goes on decreasing. Transient response of rotor system in ANSYS for disk position a = 0.066, b = 0.334 is carried out. A time dependent force for is considered, a force of 100N is applied on the disk for 1ms hence time history post processor in ANSYS is used to plot displacement versus time for rotor system is plotted. Figure 2.20 shows the variation of displacement with time of rotor system under a case of time dependent load.

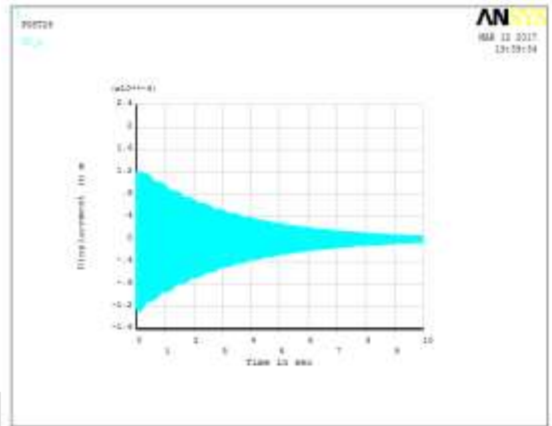

Figure 2.20: Variation of displacement with time of rotor system

III. NUMERICAL METHOD

Let us consider a single rotor shaft system, the equation of motion and natural frequency of vibration can be derived using newton's method, energy method or Rayleigh's method. The equation of motion of undamped free vibration is given

$$M\frac{d^2x}{dt^2} + Kx = 0$$

Where K is stiffness and M is mass of the disk. Natural frequency of free vibration is given

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

Equivalent mass for simply supported beam carrying disk of mass M at middle where, m is mass of shaft and M is mass of disk.

$$m_{eq} = M + 0.5\,m$$

Equivalent stiffness for simply supported rotor with disk at middle where E is young's modulus of rotor material, I is moment of inertia and L is length of shaft

$$K_{eq} = \frac{48EI}{L^3}$$

Stiffness for disk offset position is given by

$$K = \frac{3EIL}{a^2b^2}$$

For case damped forced vibration,

$$M\frac{d^2x}{dt^2} + C\frac{dx}{dt} + Kx = F_o\sin(wt)$$

On solving equation (3) we get the amplitudes of vibration

$$X = \frac{\frac{F_o}{K}}{\sqrt{(1-\frac{W}{W_n}^2)^2 + (2\xi\frac{W}{W_n})^2}}$$

Where W_n is Natural frequency of the system, $F_o$is force applied on the disk, W is forcing frequency, $\xi$is the damping ratio and X is the amplitudes of vibration.

Deflection of simply supported rotor with disk offset position is given by

$$\delta = \frac{Wa^2b^2}{3EIL}$$

Natural frequency for simply supported rotor with disk offset

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{\delta}}$$

According to dunkerley's method, Natural frequency of free transverse vibration

$$f_n = \frac{0.4987}{\sqrt{\delta}}$$

According to Rayleigh's method lowest natural frequency given by

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g\sum m_i x_i}{\sum m_i {x_i}^2}}$$

Where x_i: total static deflection

$$x_i = mga_{11}$$

Wherea_{11}: influence co-efficient

$$a_{11} = \frac{a^2b^2}{3EIL}$$

IV. RESULT

Table 4.1: Comparison of analytical results and ANSYS results

SI.No	Disk position		Critical speeds N_c (rpm)		Natural frequencies f_n (Hz)	
	a (m)	b (m)	Analytical method	ANSYS	Analytical method	ANSYS
1	0.2	0.2	2208	2201.64	36.80	36.694
2	0.133	0.267	2487	2431.08	41.45	40.518
3	0.066	0.334	3345.6	3383.04	55.76	56.348

Table 4.2: ANSYS Results

SI.No	Disk position		$\frac{W}{W_n}$	Amplitudes of vibration (m)				
	a (m)	b (m)		ξ=0	$\xi = 0.0073265$	$\xi = 0.091235$	ξ=0.1748	$\xi = 0.2$
1	0.2	0.2	0.92	8.8977×10^{-3}	8.85034×10^{-3}	5.6236×10^{-3}	3.34862×10^{-3}	3.13305×10^{-3}
			0.94	0.0139523	0.0137719	6.41135×10^{-3}	3.5963×10^{-3}	3.144×10^{-3}
2	0.133	0.267	0.92	5.963×10^{-3}	5.94×10^{-3}	4.53092×10^{-3}	2.77307×10^{-3}	2.43602×10^{-3}
			0.94	8.22429×10^{-3}	8.164×10^{-3}	5.0349×10^{-3}	2.82603×10^{-3}	2.47163×10^{-3}
3	0.066	0.3 44	0.92	2.70538×10^{-3}	2.69096×10^{-3}	1.80246×10^{-3}	1.08735×10^{-3}	4.07223×10^{-4}
			0.94	3.53928×10^{-3}	3.50708×10^{-3}	1.93959×10^{-3}	1.09244×10^{-3}	0.955360×10^{-3}

Table 4.3: Analytical Results

SI.No	Disk position		$\frac{W}{W_n}$	Amplitudes of vibration (m)				
	a (m)	b (m)		ξ=0	$\xi = 0.0073265$	$\xi = 0.091235$	ξ=0.1748	$\xi = 0.2$
1	0.2	0.2	0.92	8.8420×10^{-3}	8.8079×10^{-3}	5.9686×10^{-3}	3.8103×10^{-3}	3.40572×10^{-3}
			0.94	0.01167	0.011586	6.5517×10^{-3}	3.8955×10^{-3}	3.45041×10^{-3}
2	0.133	0.267	0.92	6.9687×10^{-3}	6.9334×10^{-3}	4.7024×10^{-3}	2.77307×10^{-3}	2.43602×10^{-3}
			0.94	9.1924×10^{-3}	9.1287×10^{-3}	5.1618×10^{-3}	3.0691×10^{-3}	2.7184×10^{-3}
3	0.066	0.3 44	0.92	2.68541×10^{-3}	2.6767×10^{-3}	1.81309×10^{-3}	1.15735×10^{-3}	4.01206×10^{-4}
			0.94	3.54364×10^{-3}	3.5194×10^{-3}	1.98996×10^{-3}	1.18324×10^{-3}	1.04796×10^{-3}

V. CONCLUSION

The academic finite element software ANSYS is a good tool to understand the dynamics of rotor system. It is observed that all the verification of analysis that of numerical results for a single rotor system is very good agreement with ANSYS results. Modal analysis of rotor system, for three cases of disk position helps us to understand the variation of rotor dynamic parameters such as natural frequency, critical speed and amplitudes for different damping ratios. The natural frequency and critical speeds of the simply supported rotor system increases as the disk position distance decreases form the support. The resonance effect can be easily understood by plotting modes shapes of the rotor system. The frequency response curves helps us to understand the amplitudes of vibration for three cases of disk position. As the disk position near to the support the stiffness increases so as the amplitudes of vibration decreases. It is also shown in case of transient vibration, the amplitudes gradually goes on decreasing on an application of a time dependent load.

This parameters are helpful in design of rotor dynamic system and finally conclude that the natural frequency, critical speeds and amplitudes of vibration depends on the effect of disk position, material properties and dimensions of rotor system and also this analysis of rotor system helps in safe and stable operation of the rotor system. Further this analysis can be carried out for a different materials for disk and shaft and also for two or more disk rotor system.

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