Transient Analysis of Rotor System

D. S. Megharaj and Amit Malgol

Abstract This paper introduces a transient vibration investigation of horizontal rotor system with three distinctive disk positions mounted on a shaft, for the simply supported case. The material properties of the shaft and disk are the same. A transient analysis is performed to obtain the amplitudes for three distinctive disk positions. A force is applied on the disk for a small time frame. Design a dynamic rotor structure; it is essential to decide the vibration parameter, i.e., natural frequency, critical speeds, and amplitudes. The transient analysis was performed by "ANSYS" parametric design tool. The results obtained from the analysis are helpful for the design of the rotor system.

Keywords Rotor dynamics · Transient · ANSYS · Amplitudes

1 Introduction

Transient vibration is non-occasional motion occurs in a rotating system. Transient vibration is non-periodic motion in actual circumstances. When a vehicle experiences a pot opening, the amplitudes diminish gradually because of the stiffness, damping, and force acting on the system. The amplitude of the system decreases depending on the system parameters and time. In the transient system, a load is applied for a small period, and hence, due to this time-dependent loading, the vibration decay gradually. The vibration energy must be less than the rotational energy of a system. The parameters of rotating systems play a vital role in maintaining the stability of a system. The systems are classified as a linear system and nonlinear system. All systems, in reality, are a nonlinear system, but a system with small oscillations or amplitudes

D. S. Megharaj
Department of Mechanical Engineering, Islamiah Institute of Technology, Bangalore, Karnataka, India
e-mail: megharajds1@gmail.com

A. Malgol (✉)
Department of Mechanical Engineering, National Institute of Technology, Calicut, Kerala, India
e-mail: amitmalgol123@gmail.com

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are considered as a linear system. It is essential to decide the dynamic parameters of a rotor, i.e., natural frequency, whirling speeds, and amplitudes. In a system, there can be nonlinearity due to geometry, misalignments, loose supports, unbalance mass, etc. In the design of a rotating system, the disk mass, bearing stiffness, and damping are important parameters. For a dynamic system, the oscillation gradually reduces and ultimately vanishes, and such a phenomenon of a dynamic system is known as transient vibration. Transient vibration may involve free or forced vibrations or both. The time-dependent transient solution briefs about a variety of sufficiency regarding time under an action a force for a small period.

The investigation of the transient system is progressively imperative in spinning machinery generally for high-speed rotors—the two important elements of study in a dynamic rotor system. The first component is the time-based analysis. The physical components are crucial, but that cannot be particularly legitimated as an element of recurrence precedents that are conducted of bearings, seals, and dampers. The second component characterizes the time-subordinate reaction instances of transient reaction framework comprise of synchronous engines and blowers and motors. Generally, a disk, coupling or a disk with blades attached to a shaft such a rotating system is called as a rotor. Applications of a rotating system are pumps, generators, auto engines, blowers, steam turbines, etc. Instability in a rotating system is the primary cause of failure. In a rotating system, changing some parameters, i.e., disk mass, stiffness, disk position, and different parameter of the rotor system, the amplitudes of the rotating system can be diminished, and such design modification helps us to operate in suitable and safe conditions.

The modal analysis and harmonic analysis of the rotor for three distinct positions of the rotor disk system are performed. In the analysis, natural frequency and critical speeds of the rotating system increase as the disk position moves toward the support, as well the amplitudes decrease presented by Malgol and Potdar [1]. Sharama [2] considered the random form of vibration due to loss of blade and the vibration frequency and demonstrated that nearly at all the working velocities, three-component control law gives more negative eigenvalues signifying better transient properties.

The stationary and transient motions in rotor systems with higher critical speeds exceed by the operating speeds. The rotor of a turbo-driven pump assembly is used in rocket engines where the rated speed of which exceeds the second basic speed and in which rings serve as sealing. Babukanth and Vimal [3] constructed an FE model of a rotor in which the shaft segments are considered as beam and considering the shear deformation. The blade wheels and the seal rings are considered as rigid disks and elastically attached on a solid bodies. Malcolm [4] described the traditional analysis approach which sights the dynamic performance of a rotor framework as an element of recurrence and investigates rotor elements in the time area and drawing nearer into the non-direct routine. Volokhovskaya and Barmina [6] delineated the underlying redirection in twisting and leftover unbalances of rotor framework to the size of the amplitudes of its transient vibrations at normal frequencies underneath its working turn recurrence on the summary. Szolc et al. [5] study shows a difference between various dynamic and static characteristics of dynamic, asynchronous motor and properties of rotor-shaft system. Balakh and Nikiforov [7] considered the effects

Table 1 Material properties of the rotor

Sl. No	Parameters	Value	Units
1	Shaft and disk material	Mild steel	–
2	Young's modulus (E)	2×10^{11}	N/m^2
3	Density (ρ)	7800	kg/m^3
4	Poisson's ration (μ)	0.33	–

Table 2 Data of the disk and shaft

Sl. No	Parameters	Value	Units
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)	1.3783	kg

of hydrodynamic force due to interaction of rotor and seal lead to disappearance of critical speed with increasing stiffness. Jangde et al. [8] described the dynamics of the single rotor system. Fleming et al. [9] investigated the transient response of rotor system supported with rolling element bearing with internal clearance. The dead band clearance shows a significant effect on the synchronous rotor response. Annentrout and Gunter [10] described the characteristics of rotor system with consideration of squeeze film damper and hydrodynamic effects.

2 Methodology

2.1 ANSYS Modeling

The rotor model consists of a shaft and disk. To develop a model of rotor, Table 1 shows the properties of the rotor, and Table 2 shows the data of disk and rotor which are considered in ANSYS. Figure 1 shows the model of the rotor.

2.2 Transient Analysis in ANSYS

The transient response of the rotor system is performed, and a 100 N time-dependent load is applied on the disk for 1 s. Three different load step files are written, i.e., when there is no load on the rotor system, a load applied for 1 s, and load removed from the rotor system, performed for cases of different disk position. The time history post-processor in ANSYS is utilized to draw amplitude versus time plot for rotor

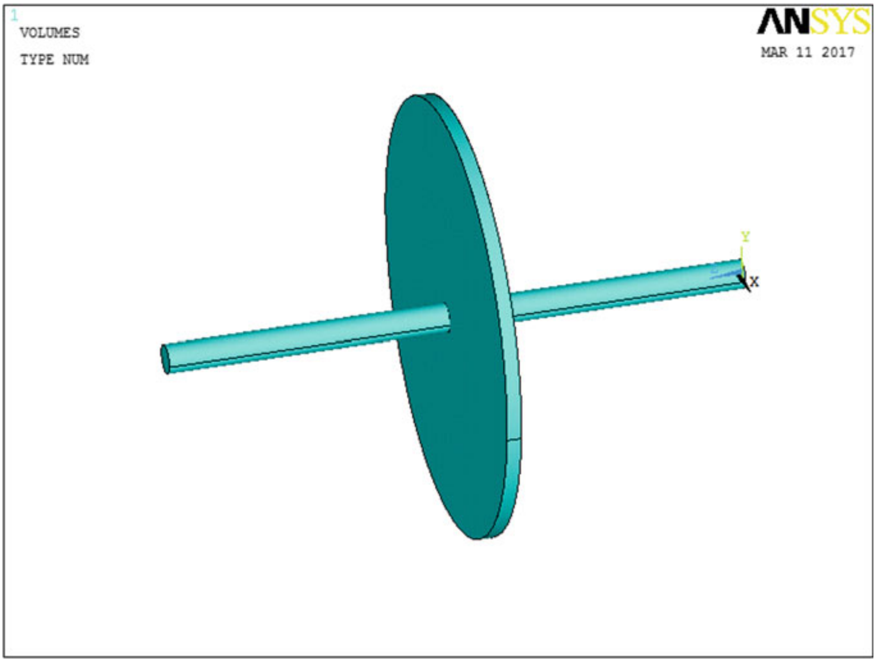

Fig. 1 Rotor model in ANSYS

system by selecting a node on mid of disk, and maximum and minimum amplitudes are obtained at the maximum time (20 s) and minimum time (0.01 s) for the selected node. Figures 2a, 3a, and, 4a show the deviation of amplitude concerning the time of the rotor system for three different positions of a disk mounted on a horizontal shaft for a simply supported case. Figures 2b, 3b, and, 4b show that the amplitudes of vibration increase with the natural frequency, and after the first critical speed, again amplitudes of vibration decrease for three different disk positions for damping ratio = 0.2.

2.2.1 For the Position of Disk $a = 0.2$ m and $b = 0.2$ m

See Fig. 2.

2.2.2 For the Position of Disk $a = 0.133$ m and $b = 0.267$ m

See Fig. 3.

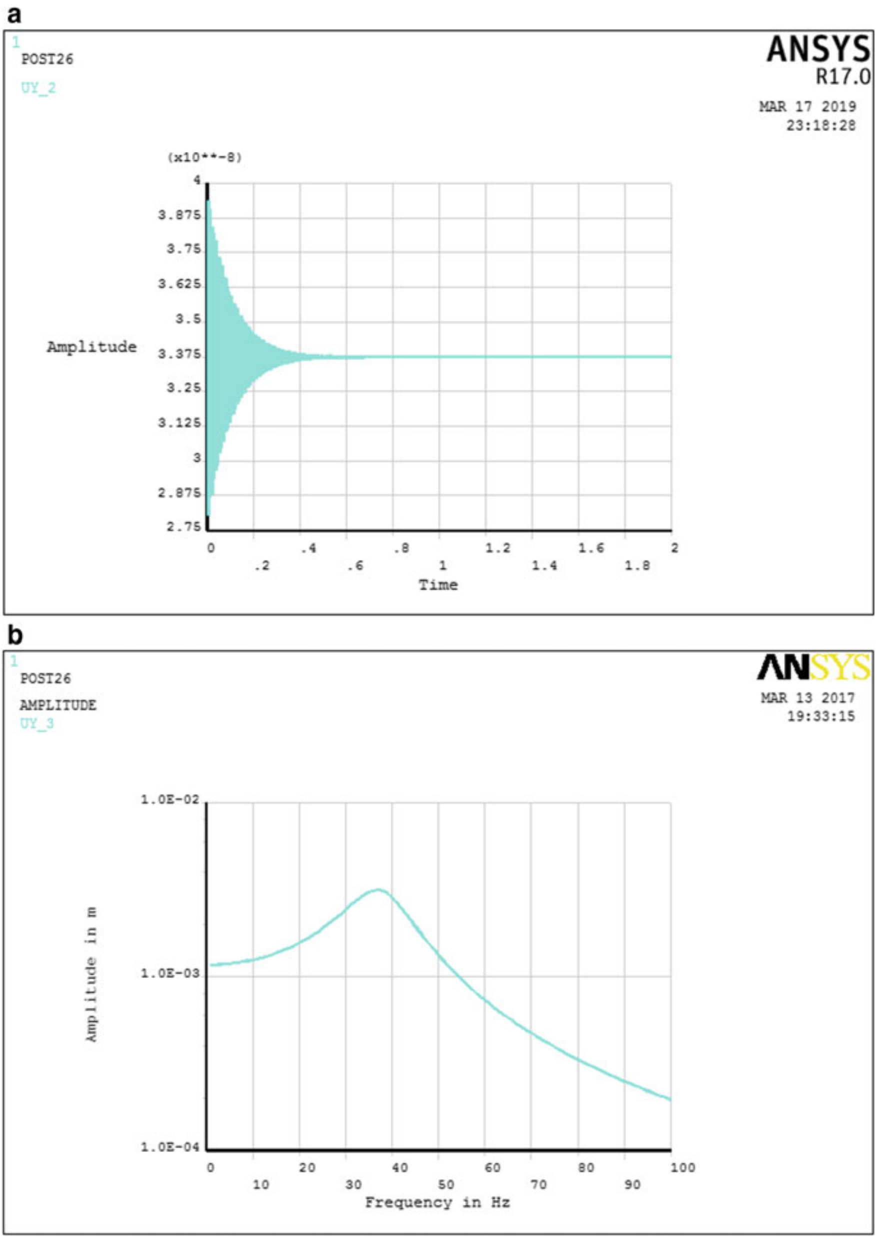

Fig. 2 **a** Amplitude versus time plot for $a = 0.2$ m and $b = 0.2$ m. **b** Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio $= 0.2$

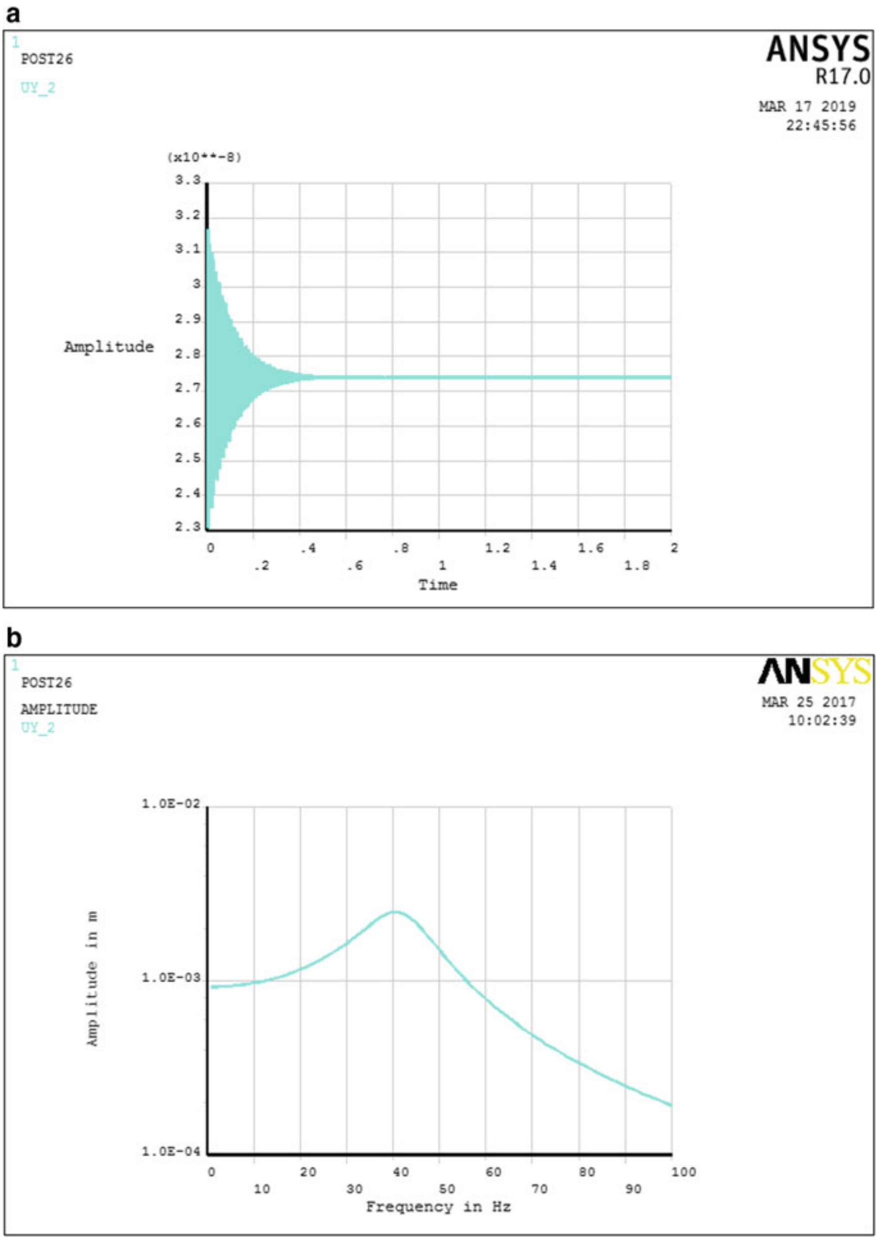

Fig. 3 **a** Amplitude versus time plot for $a = 0.133$ m and $b = 0.267$ m. **b** Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio = 0.2

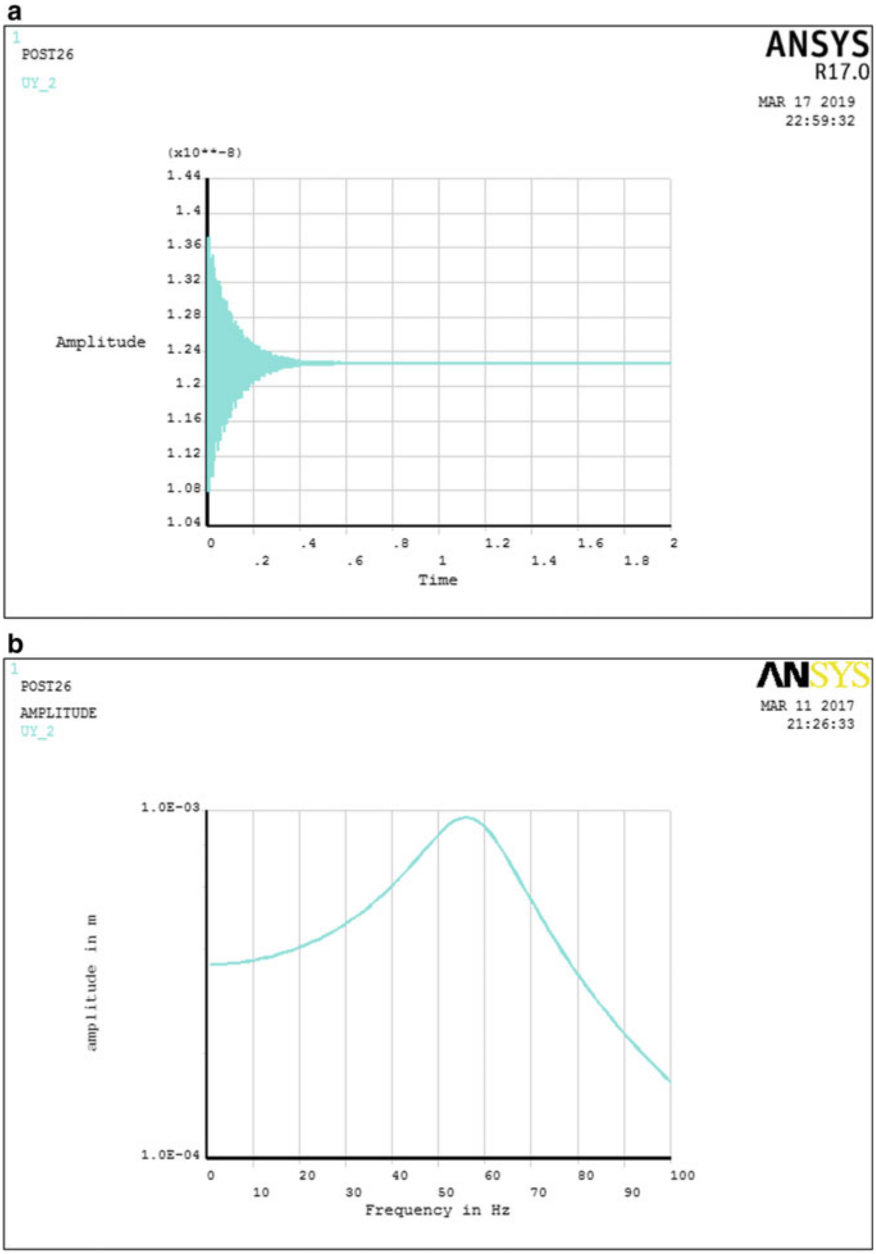

Fig. 4 **a** Amplitude versus time plot for $a = 0.066$ m and $b = 0.334$ m. **b** Variation of amplitudes of vibration with operating frequency of rotor system at damping ratio $= 0.2$

2.2.3 For the Position of Disk $a = 0.066$ m and $b = 0.334$ m

See Fig. 4.

3 Analytical Method

The equation of motion for free undamped vibration is given

$$mx'' + kx = 0 \tag{1}$$

where,

k Stiffness of shaft
m Mass of the disk.

The natural frequency of undamped free vibration

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{2}$$

The equivalent stiffness for a rotor with the disk at middle position simply supported case

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

where,

E Young's modulus (MPa)
I Area moment of inertia (m^4)
L Shaft length (m).

The stiffness for a disk with offset or different position is given by

$$K = \frac{3EIL}{a^2b^2} \tag{4}$$

The equation of motion for damped forced vibration

$$mx'' + cx' + kx = F_o \sin(wt) \tag{5}$$

The amplitudes of vibration (X) are obtained on solving Eq. (5) as

$$X = \frac{\frac{F_o}{K}}{\sqrt{\left(1 - r^2\right)^2 + (2\xi r)^2}}$$

where,

$r = \frac{\omega}{\omega_n}$ Frequency ratio,
f_n Natural frequency of the system (Hz),
F_o Applied force on the disk (N),
ω Excitation frequency (Hz),
ξ Damping ratio, and
X Amplitude of forced vibration (m).

Deflection for a rotor with disk offset position for the simply supported case is given by

$$\delta = \frac{Wa^2b^2}{3EIL} \tag{6}$$

where,

W Weight of the disk (kg).

Natural frequency for a rotor with disk offset position for the simply supported case is given by

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{\delta}} \tag{7}$$

$$f_n = \frac{0.4987}{\sqrt{\delta}} \tag{8}$$

4 Results

See Table 3.

5 Conclusion

The academic finite element ANSYS tool is good to know the dynamic behavior of the rotor. Transient investigation of rotor system for three distinct disk positions is achieved. A load of 100 N is applied on the disk for 1 s. In all the different cases, we can observe that amplitudes are gradually reduced due to the application of a load dependent on time, and plots are obtained for variation amplitudes concerning time, for simply supported rotor system under the action of time-dependent load. For mid disk position, the stiffness of rotor is less compared to disk location near to the support, and hence, comparing mid disk position and disk close to the support,

Table 3 Analytical and ANSYS results

Sl. No.	Position of disk		Node	Result	Results		
					ANSYS results		Analytical results
	a (m)	b (m)			Minimum value	Maximum value	Maximum value
1	0.2	0.2	Mid disk	Time	0.001	2	2
				Y component of displacement	2.080325×10^{-8}	3.98376×10^{-8}	3.9582×10^{-8}
2	0.133	0.267	Mid disk	Time	0.001	2	2
				Y component of displacement	2.30689×10^{-8}	3.1676×10^{-8}	3.1196×10^{-8}
3	0.066	0.332	Mid disk	Time	0.001	2	2
				Y component of displacement	1.07849×10^{-8}	1.37206×10^{-8}	1.2022×10^{-8}

the amplitude of forced vibration is lower for disk closer to the support. Therefore, the higher the stiffness value lowers the vibration amplitudes. As the disk, location closer to the support increases stiffness with decreasing amplitudes of vibration. Further, we can conclude that using a suitable damping ratio, the vibration amplitude of system reduces, and changing disk position the amplitude further reduces to a minimum value. These parameters are useful in the design of the rotor system and finally conclude that the vibration amplitudes depend on the effect of position of the disk, properties, and geometry of the rotor system. This investigation helps us in a steady-state and safe operation of the rotating system. Further, these investigations can be carried out for different materials, multi-disk rotor system, and experimental verification.

The transient analysis with different disk positions can be utilized for design modification of a real rotor system such as textile rotor. The textile rotor is used to wind the band. The gyroscopic effect is considered in the system. The analysis shows that the amplitude of vibration decreases as the disk position nearer to the support compared to the mid position of the disk on the shaft.

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