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Transmissibility Analysis of a Regenerated Rotor System

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Abstract: The aim of this paper is to represents a dynamic behavior of rotor bearing system wirth simply supported beam for three different position disc. rotating machinery such as compressors, turbines, pumps, jet engines, turtobo chargers, etc. are subject to vibrations. rotating machines are operated in very high speed and they are subjected to some unbalance force due to vibration from that machine pass to the foundation of machine.so the analysis of the dynamics parameter of rotor it is important to determine force transmissibility, natural frequency, critical speed and amplitudes of rotor system.
Keywords: force transmissibility, vibration, critical speed, rotor bearing system etc.

I. INTRODUCTION

Dynamics of rotating machinery has been extensively studied in the past by many researchers, mainly due to the numerous applications in industry, such as power generation, large- speed , rotor system, Natural frequency etc. scale manufacturing, automobile engines, aerospace propulsion and home appliances, among others. In all these applications, one of the most common concerns of designers is the long-term exposure of the rotating machines to vibration, which eventually can lead to failures or accidents. This can be the case when e.g. the undesired vibration becomes close to one of the natural frequencies of the machine structure (resonance) or due to dangerous vibration-induced impacts between rotating and stationary components. Modern rotating machines, such as turbines, compressors and generators, are designed for high speed, high flexibility and high efficiency. In order to avoid unstable vibrations at higher operating speeds, more and more attention has been paid to self-excited vibration. The occurrence of rotor lateral self-excited vibration known as “whirl”, “whip” or simply “instability” arises from the presence of nonlinear fluid forces as the threshold speed is exceeded by the rotating speed, which is the phenomenon of dynamic instability resulted from the interaction between the rotor and the siding bearing. The instability is typically sub-synchronous because it would induce excessive vibration at the first or second mode whirl/ whip frequency, and it would contribute to unstable operation of the system, high-level vibration, eventual rubbing between rotor and stator, and potential damage of the rotating machinery.

A.Title and author details-Kalyani athya1, Prof. U.K. joshi [1]Dynamic behavior analysis of crack rotor system is investigated, Experimental studies were performed on a rotor. The vibrations are analyses in terms of velocity, displacement and frequency was measured at different speeds by vibration sensor. The principal components of a rotor-dynamic system are the shaft or rotor with disk, the bearings, and the seals. The shaft or rotor is the rotating component of the system A crack in the rotor will change the dynamic behavior of the system but in practice it has been found that small or medium size cracks make such a small change to the dynamics of machine system that they are virtually undetectable by this means. Amit malgol,yogita Potdar [2]In This 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. 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. 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. mohammad Hadi Jalali ,Mostafa Ghayour, Saeed Ziaei-Rad, Behrooz Shahriari[3]in this full dynamic analysis of a high speed rotor with certain geometrical and mechanical properties is carried out using 3D finite element model, one-dimensional beam-type model and experimental modal test. The, critical speeds, operational deflection shapes, and unbalance response of the rotor are obtained in order to completely investigate the dynamic behavior of the rotating system.

The dynamic characteristics of a high speed rotor with certain geometrical and mechanical properties were evaluated at rest and under operating conditions. The modal analysis of the rotor at rest under free–free boundary conditions was done with beam finite element model, 3D finite element model and modal test and comparison of the results indicated satisfactory agreement between them. Kotikala Rajasekha, Ballepalli Kailash[4]Dynamic response analysis of a cracked rotor is attempted. The breathing of crack is accounted using the response dependent breathing crack model. Transient response of an accelerating cracked rotor is analyzed. The influence of the presence of transverse cracks in a rotating shaft is analyzed. The crack has opening and closing on dynamic response during operation in the rotor. Initially a simple Jeffcott rotor is analyzed considering the lateral vibration. The dynamic response of the rotor with a breathing crack is evaluated by expanding the changing stiffness of the crack as a truncated using FEM. and then approach is based on the fact that the presence of a crack in rotating shaft reduces the stiffness of the structure, hence reducing the natural frequencies of the original untracked shaft. Vibration response analysis of cracked rotor has been carried out. Coupled vibration response has been studied with objective of obtaining crack specific features in the response. Some of the analytical results are reported for a cracked rotor. Qinkai Han, Fulei Chu[5] Dynamic response of cracked rotor-bearing system under time-dependent base movements is studied in this paper. Three base angular motions, including the rolling, pitching and yawing motions, are assumed to be sinusoidal perturbations superimposed upon constant terms. Both the open and breathing transverse cracks are considered in the analysis. The finite element model is established for the base excited rotor-bearing system with open or breathing crack Considering the time-varying base movements and transverse cracks, the second-order differential equations of the system will not only have time-periodic gyroscopic and stiffness coefficients, but also the multi frequency external excitations. An improved harmonic balance method is introduced to obtain the steady-state response of the system under both base and unbalance excitations. The dynamic behaviors of cracked rotor-bearing system subjected to harmonic base angular motions are studied utilizing the harmonic balance method.

II. EXPERIMENTAL SETUP

The experimental setup used in this experiment is shown in fig 1. The motor with belt driven system drives a rotor shaft system. The setup consist a rotor, disk, vibration sensor, and bearing. The rotor can be run at different speeds below and above critical speed.

Fig: 1 Setup of Rotor system with crack disc

1.1 Rotor bearing system details

S.NO	Parameters	Value
1	Rotor Diameter	23mm
2	Length	560mm
3	Mass	1.81kg
4	Bearing span	420mm
5	Density	7800 kg/m3

1.2 Dimensions of disc

S.NO.	Parameters	Value
1.	External diameter	120mm
2.	Internal diameter	25mm
3.	Thickness	8mm
4.	mass	0.6kg
5	Density	7800 kg/m3

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III. METHODOLOGY

We have used a rotor bearing system with disc. We have take observation in three conditions first with disc without crack, second disc with crack and then with regenerated disc. we observe the deflection and frequency in different speed and then calculated the transmissibility by mathematical approach.

Observation Table 2.1 Deflection without crack, with crack and regenerated disc

s.no	speed(rpm)	Deflection (mm)		
		Without crack	With crack	regenerated disc
1	800	1020	1040	1030
2	1000	1400	1440	1405
3	1200	1800	1820	1830
4	1400	2100	2480	2190
5	1600	2650	2900	2690
6	1800	3100	3300	3130
7	2000	3500	4440	3600
8	2200	4050	4620	4090
9	2400	4570	4750	4600
10	2500	4800	5220	4910

Observation Table 2.2 Frequency (Hertz) without crack, with crack and regenerated disc

s.no	speed(rpm)	Frequency(Hertz))		
		Without crack	With crack	regenerated disc
1	800	3.9	4	3.9
2	1000	3.9	4.1	4
3	1200	4.28	4.3	4.28
4	1400	4.3	4.8	4.3
5	1600	4.62	5.1	4.7
6	1800	5.3	5.9	5.4
7	2000	5.72	6.8	5.9
8	2200	6.1	7.4	6.8
9	2400	6.5	8	6.6
10	2500	6.8	8.3	7.3

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IV. NUMERICAL ANALYSIS

Let us consider a single rotor shaft system,in which Force Transmission that is The force transmitted to the foundation of the system is

$$T=\frac{1+2\xi\left(\frac{\omega}{\omega n}{}^{2}\right)}{\sqrt{\left(1-\frac{\omega^2}{\omega n^2}\right)^2+[2\xi(\omega/\omega n)^2]^2}}$$

Where the transmissibility T in which the frequency ratio $\frac{\omega}{\omega n}$ and for several values of the fraction of critical damping ξ.

Observation Table 2.3 mathematical result table of Frequency ratio (ω/ωn) without crack, with crack and regenerated disc

s.no	speed(rpm)	Frequency ratio (ω/ωn)		
		(f¹) Without crack	(f²) With crack	(f³) regenerated disc
1	800	31.2	32	31.2
2	1000	31.2	32.8	32
3	1200	34.24	34.4	34.24
4	1400	34.4	38.4	34.4
5	1600	36.96	40.8	37.6
6	1800	42.42	47.2	43.2
7	2000	45.76	54.4	47.2
8	2200	48.8	59.2	54.4
9	2400	52	64.0	52.8
10	2500	54.4	66.4	58.4

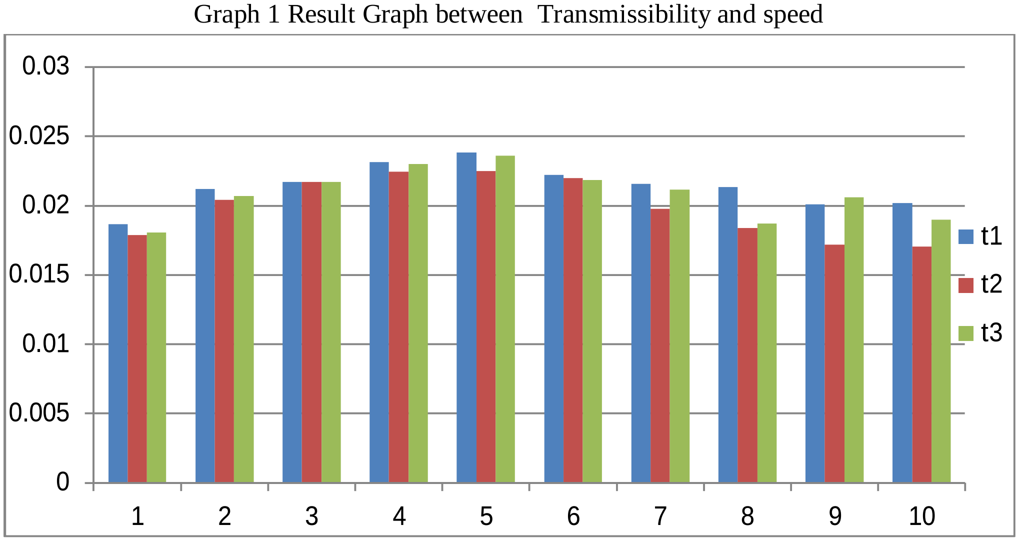
Observation Table 2.1 mathematical result table of damping factor without crack, with crack and regenerated disc

s.no	speed(rpm)	Damping factor		
		Without crack	With crack	regenerated disc
1	800	0.16035	0.1634	0.1618
2	1000	0.2177	0.2236	0.2183
3	1200	0.2755	0.2784	0.2798
4	1400	0.3179	0.0371	0.3293
5	1600	0.3881	0.0420	0.39377
6	1800	0.4427	0.4652	0.44612
7	2000	0.486	0.57910	0.4973
8	2200	0.542	0.59360	0.51375
9	2400	0.593	0.6033	0.5910
10	2500	0.603	0.640	0.61598

Observation Table2.2 mathematical result table of Transmissibility without crack disc , with crack disc and regenerated disc.

s.no	speed(rpm)	transmissiblity		
		(t1) Without crack	(t2) With crack	(t3) regenerated disc
1	800	0.01868	0.01790	0.01807
2	1000	0.02120	0.02042	0.02069
3	1200	0.02172	0.02171	0.02170
4	1400	0.023163	0.022457	0.023022
5	1600	0.023859	0.022480	0.0236
6	1800	0.02221	0.0220	0.02187
7	2000	0.021555	0.019764	0.021138
8	2200	0.02134	0.0184089	0.018707
9	2400	0.02009	0.0171662	0.020590
10	2500	0.02019	0.01704	0.019009

Graph 1 Result Graph between Transmissibility and speed



The graph 1 shows that the graph between Transmissibility and speed in "disc without crack(t1)",disc with crack"(t2), and "regenerated disc"(t3).

V. CONCLUSON

The main conclusions of this mathematical investigation are found that maximum vibration Transmissibility in "crack disc" compare to "disc without crack" and we also found that the maximum vibration Transmissibility in "crack disc" compare to "regenerated disc."

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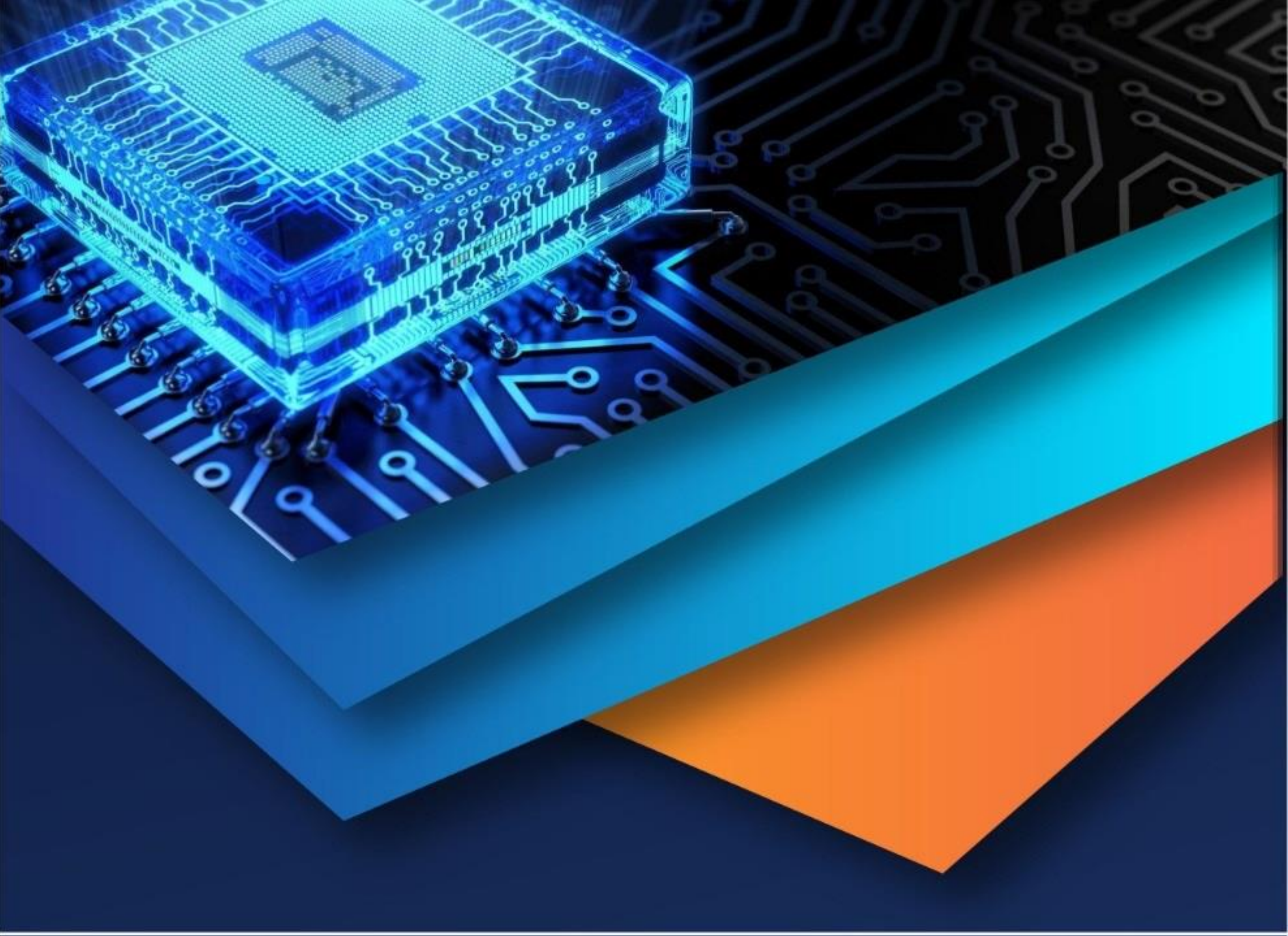

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