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Vibrations Analysis of 4 Jaw Flexible Coupling Considering Unbalancing in Two Planes

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ABSTRACT

Unbalancing and misalignment are the most possible causes of machine vibrations. An unbalanced rotor always cause more vibration and generates excessive force in the bearing area and reduces the life of the machine. Understanding and practicing the fundamentals of rotating shaft parameters is the first step in reducing unnecessary vibration, reducing maintenance costs and increasing machine uptime. By the term two planes here, we mean that two rotors are used for the analysis of unbalanced vibrations. If only one rotor is used then this system is called a single plane system. In this paper, experimental studies were performed on a 2 rotor dynamic test apparatus to predict the vibration spectrum for rotor unbalance. A 4 Jaw flexible coupling was used in the experiments. The rotor shaft velocities were measured at rotor speed of 30 Hz using accelerometer and a Dual Channel Vibration Analyzer (DCVA) under the balanced (baseline) and unbalanced conditions. The experimental frequency spectrum was also obtained for both baseline and unbalanced condition under different unbalanced forces. The experimental results of balanced and unbalanced rotors are compared at two different rotor locations.

Keywords: *4 Jaw flexible Coupling, Unbalanced rotors, DCVA, Ball Bearings*

1. INTRODUCTION

Rotor unbalance is the most common reason in machine vibrations. Most of the rotating machinery problem can be solved by using the rotor balancing and misalignment. Mass unbalance in a rotating system often produces excessive synchronous forces that reduce the life span of various mechanical elements. A very small amount of unbalance may cause severe problem in high speed rotating machines. Rotors are used in many engineering applications like pumps, fans, propellers and turbo machinery etc. The vibration signature of the overhung rotor is totally different from the midway or intermediate rotors. The vibration caused by unbalance may destroy critical parts of the machine, such as bearings, seals, gears and couplings. Rotor unbalance is a condition in which the centre of mass of a rotating assembly, typically the shaft and its fixed components like disks and blades etc. is not coincident with the centre of rotation. In practice, rotors can never be perfectly balanced because of manufacturing errors such as porosity in casting, non-uniform density of material, manufacturing tolerances and gain or loss of material during operation. As a result of mass unbalance, a centrifugal force is generated and must be reacted against by the bearings and support structures. Analytical methods of lateral response due to torsional excitation of geared rotors were reported by Rao *et al.* [1]. In continuation of this, Shiau *et al.* [2] presented a dynamic behavior of geared rotors. Then, Lee *et al.* [3] studied the rotor-dynamic characteristics of an APU gas turbine rotor–bearing system having a tie shaft. Unbalance response investigations of geared rotor-bearing systems, were carried out by Neriya *et al.* [4] and Kahraman *et al.* [5]. Further, Iida *et al.* [6] and Iwatsubo *et al.* [7] reported on studies utilizing the usual procedure of solving simultaneous equations and Choi and Mau [8] utilizing the frequency branching technique. Further, concerning unbalance response investigations of dual shaft rotor-bearing systems coupled by bearings, based on the transfer matrix modeling, Hibner [9], Li *et al.* [10] and Gupta *et al.* [11] carried out investigations utilizing the usual procedure of solving simultaneous equations. Afterwards Rao [12] published a book on Rotor Dynamics. The study of critical speeds of a continuous rotor was reported by Eshleman and Eubanks [13]. Geared High speed rotors were studied regarding their torsional lateral coupling by Mitchell and Mellen [14]. Further, the coupled lateral-torsional vibration characteristics of a varying speed geared rotor-bearing system were studied by Lee and Ha [15]. Then, Rao *et al.* [16] reported about the coupled bending-torsional vibrations of geared rotors. However all the above investigations resulted in full numerical solutions of the unbalance responses of coupled shaft two rotor-bearing systems.

In this work a general method is presented for obtaining the unbalance response orbit based on the experimentation of a gear-coupled two rotor shaft bearing system, where

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the shaft may rotate at different speeds. In this paper balanced and unbalanced system of 2 overhung rotors and 2 intermediate rotors are considered for the study. Experiments were conducted for two different positions of rotors (2 rotors overhung) & (2 rotors intermediate) for unbalanced weights at rotor speed of 30 Hz (1800 rpm) and results are plotted. However the set up can also be made to take results at different rotor speeds. The rotor unbalance can be detected by spectral analysis. The vibration frequency of rotor unbalance is synchronous that is one times the shaft rotational speed, since the unbalance can be reduced significantly by rotor balancing.

2. DESCRIPTION OF 4 JAW FLEXIBLE COUPLING

A 4 Jaw coupling is as shown in Figure 1. It has two flanges. One flange has a pin hole of required number at pitch circle diameter. The other flange will have a number of pins projected outside at pitch circle diameter to accommodate into the first flange holes with rubber bush. The driver and driven shafts are connected to their respective flanges i.e. output and input flanges by means of parallel square key. Mild-steel is considered for the input and output shafts, pins and keys. Over these pins, a circular natural rubber bush is provided and its length is equal to the length of the hole. The diameter of the flange holes is equal to the diameter of pin plus the thickness of rubber bush.

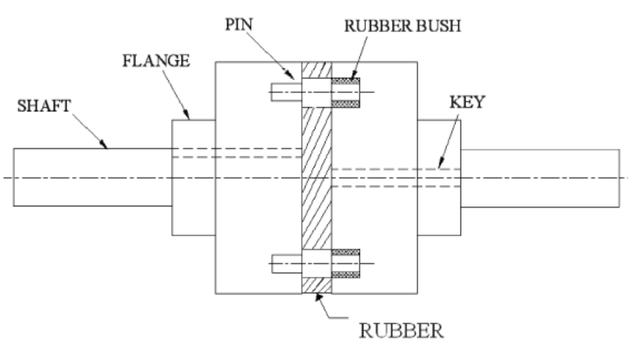

Fig.1 Four Jaw flexible coupling with key and shaft

The cast-iron material is chosen for both left and right flanges and the natural rubber is for bush. There is no nut and bolt to clamp the both input and output flanges. The Figure 1 represent the two dimensional model of 4 Jaw flexible coupling. In between flanges a rubber material is introduced to give the flexibility. Dimensions of the coupling and materials used are given in Table 1 and Table 2 respectively.

3. DESCRIPTION OF THE EXPERIMENTAL SETUP

The Experimental apparatus is shown in Figure 2 (two rotors in overhung position in two planes) and in Figure 3 (two rotors in intermediate position in two planes). It consists of a D.C. motor, a flexible coupling and a double disk rotor. The rotor shaft is supported by two identical ball bearings and has a length of 660 mm with a bearing span of 460 mm. The diameter of the rotor shaft is 20 mm. Two disks of 128 mm in diameter and 10 mm in thickness are mounted on the rotor shaft non drive end. The bearing pedestals are adjustable in vertical direction so that different misalignment conditions can be created. The rotor shaft is driven by 0.75 hp D.C. motor. The D.C. voltage controller known as control panel is used to adjust the power supply so that motor speed can be continuously increased or decreased in the range from 0 to 3000 rpm. The baseline signal has been measured at a rotor speed of 30 Hz. to check the concentricity.

The instruments used in the experiments include accelerometers and a vibration analyzer (DCVA). The accelerometer directly measures the velocity of bearing housing vibrations and displays in the vibration analyzer.

Table 1: Dimensions of shaft and coupling

S.No	Description	Unit
1.	Shaft diameter	20 mm
2.	Length of the Shaft	630 mm
3.	Hub diameter	40 mm
4.	Length of the hub	30 mm
5.	Outside diameter of flange coupling and Rubber pad	80 mm
6.	No. of holes for pin	4
7.	Diameter of pin hole	4.2 mm
8.	Diameter of pin	4 mm
9.	Rubber bush outside diameter	11 mm
	Rubber bush inside diameter	6 mm
10.	**Keyway depth**	
	In shaft	4 mm
	In hub	3 mm
	Keyway X-section	
	Height	6 mm
	Width	6 mm

Table 2: Material Properties

Properties	Cast-iron	Mild steel	Rubber
Young's modulus, (MPa)	1×10^5	2×10^5	30
Poisson ratio	0.23	0.3	0.49
Density, (kg/mm^3)	$7250{\bullet}^{9}$	$7850{\bullet}^{9}$	$1140{\bullet}^{9}$

4. EXPERIMENTAL PROCEDURE

Experimental facility as shown in Figure 2 (with two rotors in overhung position in two planes) and Figure 3 (with two rotors in intermediate position in two planes) is used for both baseline and unbalance tests. First the setup is run for few minutes to settle down all minor vibrations. Before creating unbalancing, the shaft is checked for any misalignment and unbalance. After that an unbalance has been created by placing a mass of 18 gram in the overhung rotor at a radius of 54 mm. Accelerometer along with the vibration analyzer is used to acquire the vibration signals. The photograph 1 shows the set up for two rotors in overhung position in two planes and photograph 2

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shows the set up for two rotors in intermediate position in two planes The accelerometer is attached with the help of wires to take readings at three positions (Horizontal, Vertical and Axial) at NDE (Non Drive End) and DE (Drive End) for both motor and rotor. Following are the three positions for motor and rotor:

MOTOR
[NDE (H)]$_{M}$ – Horizontal Non Drive End of motor
[NDE (V)]$_{M}$ – Vertical Non Drive End of motor
[NDE (A)]$_{M}$ – Axial Non Drive End of motor
[DE (H)]$_{M}$ – Horizontal Drive End of motor
[DE (V)]$_{M}$ – Vertical Drive End of motor
[DE (A)]$_{M}$ – Axial Drive End of motor
ROTOR
[NDE (H)]$_{R}$ – Horizontal Non Drive End of rotor
[NDE (V)]$_{R}$ – Vertical Non Drive End of rotor
[NDE (A)]$_{R}$ – Axial Non Drive End of rotor
[DE (H)]$_{R}$ – Horizontal Drive End of rotor
[DE (V)]$_{R}$ – Vertical Drive End of rotor
[DE (A)]$_{R}$ – Axial Drive End

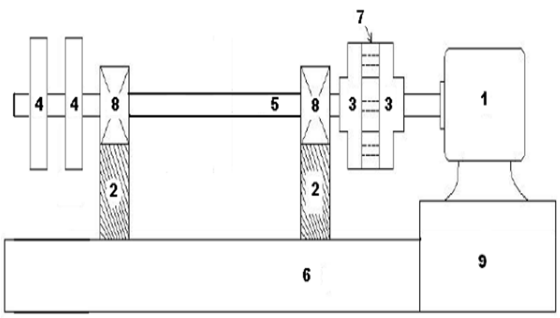
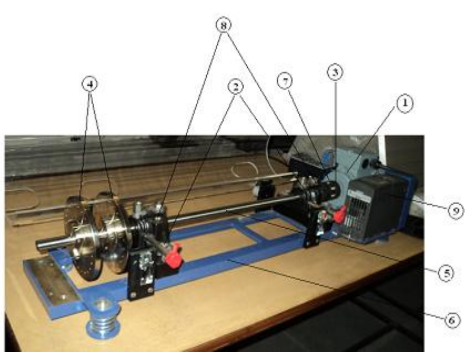


Figure 2: Experimental setup with two rotors at overhung position

1- D.C Motor
2- Bearings supported in Plummer Blocks
3- 4 Jaw Pin type flexible coupling
4- Rotors (5-8 kg each, adjustable)
5- Rotor shaft
6- Base
7- Rubber
8- Ball bearings
9- Control Panel



Photograph 1: Experimental setup with two rotors at overhung position

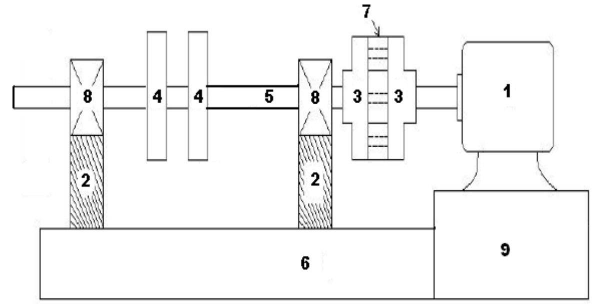
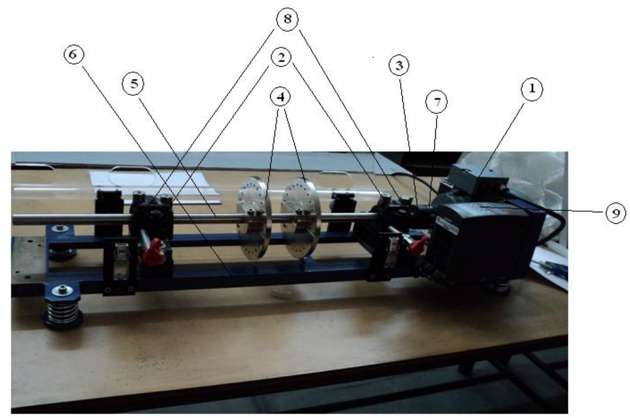


Figure 3: Experimental setup with two rotors at intermediate position



Photograph 2: Experimental setup with two rotors at intermediate position

5. RESULTS AND DISCUSSIONS

AMPLITUDE TREND FOR TWO ROTORS WITH OVERHUNG POSITION AT BASELINE AND UNBALANCED CONDITIONS

The experimental frequency spectra were obtained to the baseline condition. The perfect alignment and balancing cannot be achieved in practice. Thus, a baseline (well balanced & aligned) case is presented first to show the residual unbalance and misalignment. The measured amplitude trend of vibrations in the form of velocity of a baseline AND an unbalanced system at Drive End (DE) and Non Drive End (NDE) with 4 Jaw flexible coupling at a frequency of 30Hz (1800 RPM) is shown in fig 4. The baseline spectrum is measured experimentally using dual channel vibration analyzer. Table 3 & 4 shows the comparison of experimental vibration amplitudes when two rotors are in overhung position for both baseline and unbalanced signals on MOTOR SIDE and ROTOR SIDE respectively. Figure 4 (a) to (d) shows that the max. amplitude 4.1 mm/sec is observed at [NDE (V)]$_{M}$ in baseline condition and still higher amplitude of 6.1 mm/sec is observed at same location in unbalanced condition. Peak amplitude values of small magnitudes are observed at other ends. This highest amplitude at this location is due to the centrifugal forces acting on the

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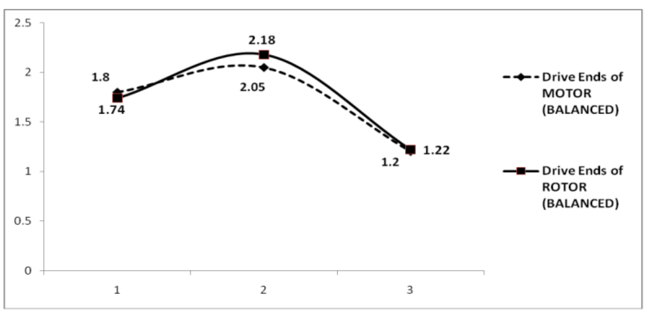
system. Small amount of peaks at harmonics of shaft speed are the indications of manufacturing errors of coupling and other elements like rubber.

AMPLITUDE TREND FOR TWO ROTORS WITH INTERMEDIATE POSITION AT BASELINE AND UNBALANCED CONDITIONS

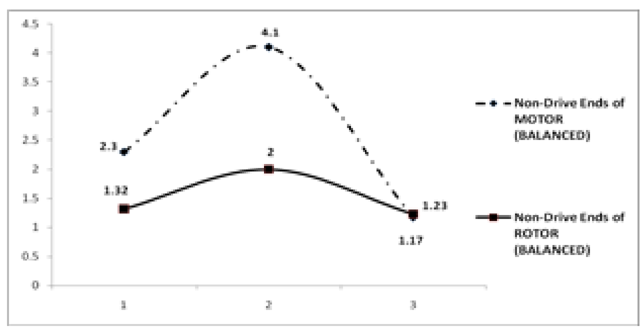
The experimental frequency spectra were obtained to the baseline condition. The perfect alignment and balancing cannot be achieved in practice. Thus, a baseline (well balanced & aligned) case is presented first to show the residual unbalance and misalignment. The measured amplitude trend of vibrations in the form of velocity of a baseline AND an unbalanced system at Drive End (DE) and Non Drive End (NDE) with 4 Jaw flexible coupling at a frequency of 30Hz (1800 RPM) is shown in fig 5. Table 5 & 6 shows the comparison of experimental vibration amplitudes when two rotors are in intermediate position for both baseline and unbalanced signals on MOTOR SIDE and ROTOR SIDES respectively. The baseline spectrum is measured experimentally using dual channel vibration analyzer. Figure 5 (a) to (d) shows that along motor side highest amplitude of 1.2 mm/sec is observed at $[NDE\ (V)]_M$ in baseline condition and amplitude reaching to 6.2 mm/sec is observed at the same location in unbalanced condition. Further along the rotor side, the maximum amplitude of 0.48 mm/sec is observed at $[NDE\ (V)]_R$ in baseline condition and after unbalancing the amplitude increasing to 4.3 mm/sec is observed at $[DE\ (V)]_R$. As discussed earlier the increase in amplitude at this location (vertical location) is due to increase in centrifugal forces acting on the system because of the created unbalance.

6. CONCLUSION

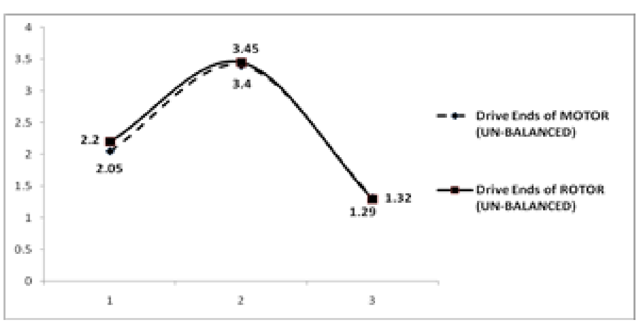
This paper presents mathematical analysis of the results taken from a DCVA both in base line and unbalanced conditions. Then these results were compared and it was found that by creating an unbalance the amplitude of the system increases. This amplitude increase was noticed in all the positions (Horizontal, Vertical and Axial) of the unbalanced system but severe boosting of the amplitudes was observed in the vertical position because of increase in centrifugal forces acting on the system. Based on this paper further work can be initialized for fault analysis and diagnosis (Condition Monitoring) for different machines.



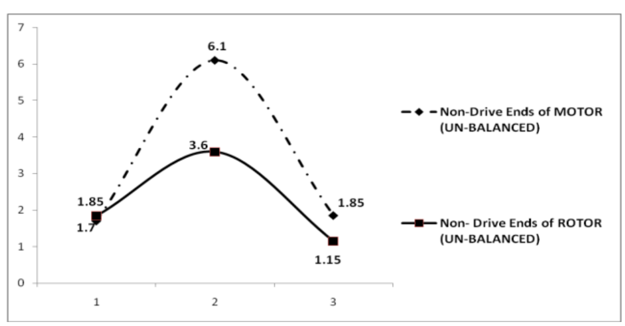
(a)
Baseline Amplitude Trend of Drive Ends (Motor & Rotor) For 2 Rotors Overhung



(b)
Baseline Amplitude Trend of Non Drive Ends (Motor & Rotor) For 2 Rotors Overhung



(c)
Unbalanced Amplitude Trend of Drive Ends (Motor & Rotor) For 2 Rotors Overhung



(d)
Unbalanced Amplitude Trend of Non Drive Ends (Motor & Rotor) For 2 Rotors Overhung

Figure 4: Amplitude trends of NDE & DE of both motor and rotor in baseline and unbalanced conditions with two rotors in OVERHUNG POSITION

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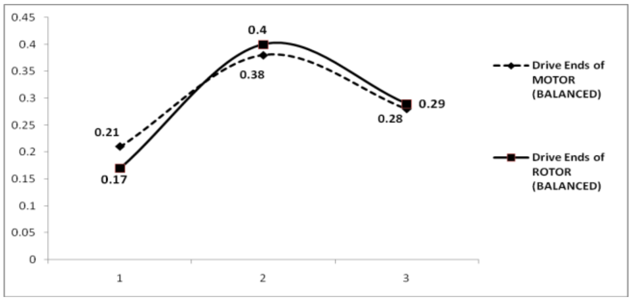
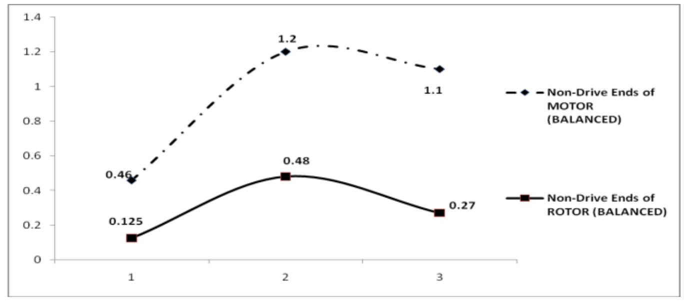
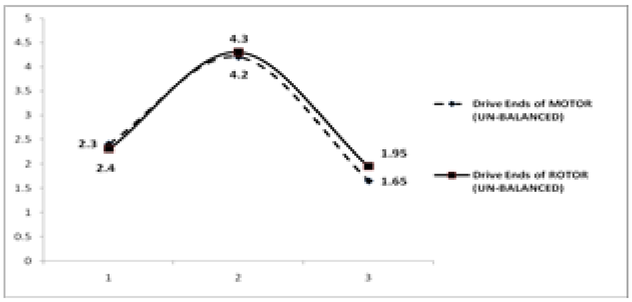


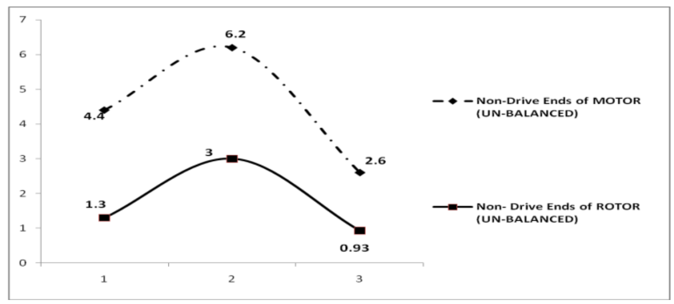
(a)
Baseline Amplitude Trend of Drive Ends
(Motor & Rotor) For 2 Rotors Intermediate



(b)
Baseline Amplitude Trend of Drive Ends
(Motor & Rotor) For 2 Rotors Intermediate



(c)
Unbalanced Amplitude Trend of Drive Ends
(Motor & Rotor) For 2 Rotors Intermediate



(d)
Unbalanced Amplitude Trend of Drive Ends
(Motor & Rotor) For 2 Rotors Intermediate

Figure 5: Amplitude trends of NDE & DE of both motor and rotor in baseline and unbalanced conditions with two rotors in INTERMEDIATE POSITION

Table 3: Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE OVERHUNG for baseline (balanced) and unbalanced signals along MOTOR SIDE

	Two rotors overhung baseline signals (Motor) (Exp. Values in mm/s)			Two rotors overhung unbalanced signals (Motor) (Exp. Values in mm/s)					
							Increase in amplitude		
	H	V	A	H	V	A	**H**	**V**	**A**
Non Drive End (NDE)$_M$	2.3	4.1	1.17	1.7	6.1	1.85	**-0.6**	**2**	**0.05**
Drive End (DE)$_M$	1.8	2.05	1.2	2.05	3.4	1.32	**0.25**	**1.35**	**0.12**

Table 4: Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE OVERHUNG for baseline (balanced) and unbalanced signals along ROTOR SIDE

	Two rotors overhung baseline signals (Rotor) (Exp. Values in mm/s)			Two rotors overhung unbalanced signals (Rotor) (Exp. Values in mm/s)					
							Increase in amplitude		
	H	V	A	H	V	A	**H**	**V**	**A**
Non Drive End (NDE)$_R$	1.32	2	1.23	1.85	3.6	1.15	**0.53**	**1.6**	**-0.08**
Drive End (DE)$_R$	1.74	2.18	1.22	2.2	3.45	1.29	**0.46**	**1.27**	**0.07**

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Table 5: Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE INTERMEDIATE for baseline (balanced) and unbalanced signals along MOTOR SIDE

	Two rotors intermediate baseline signals (Motor) (Exp. Values in mm/s)			Two rotors intermediate unbalanced signals (Motor) (Exp. Values in mm/s)					
							Increase in amplitude		
	H	V	A	H	V	A	**H**	**V**	**A**
Non Drive End $(NDE)_M$	0.46	1.2	1.1	4.4	6.2	2.6	**3.94**	**5**	**1.5**
Drive End $(DE)_M$	0.21	0.38	0.28	2.4	4.2	1.65	**2.19**	**3.82**	**1.37**

Table 6: Comparison of Experimental vibration amplitudes WHEN TWO ROTORS ARE INTERMEDIATE for baseline (balanced) and unbalanced signals along ROTOR SIDE

	Two rotors intermediate baseline signals (Rotor) (Exp. Values in mm/s)			Two rotors intermediate unbalanced signals (Rotor) (Exp. Values in mm/s)					
							Increase in amplitude		
	H	V	A	H	V	A	**H**	**V**	**A**
Non Drive End $(NDE)_R$	0.125	0.48	0.27	1.3	3	0.93	**1.18**	**2.52**	**0.66**
Drive End $(DE)_R$	0.17	0.4	0.29	2.3	4.3	1.95	**2.13**	**3.9**	**1.66**

REFERENCES

[1] Rao, J.S., Shiau, T.N. and Chang, J.R. (1998), Theoretical analysis of lateral response due to torsional excitation of geared rotor, Mechanism and Machine Theory, Vol. 33, No.6, pp. 761-783.

[2] Shiau, T.N., Rao, J.S., Chang, J.R. and Choi, S.T. (1999), Dynamic behavior of geared rotors, Transactions J. Engineering for Gas Turbine and Power, Vol. 121, pp. 494-503.

[3] Lee, A.S. and Lee, Y.S. (2001), Rotor dynamic characteristics of an APU gas turbine rotor-bearing system having a tie shaft, KSME Intl. J., Vol. 15, No. 2, pp. 152-159.

[4] Neriya, S.V., Bhat, R.B. and Sankar, T.S. (1985), Coupled torsional flexural vibration of a geared shaft system using finite element method, The Shock and Vibration Bulletin (Part 3), Vol. 55, pp. 13-25.

[5] Kahraman, A., Ozguven, H.N., Houser, D.R. and Zakrajsek, J.J. (1992), Dynamic analysis of geared rotors by finite elements, Transactions J. Mechanical Design, Vol. 114, No. 03, pp. 507-514.

[6] Iida, H., Tamura, A., Kikuchi, K. and Agata, H. (1980), Coupled torsional-flexural vibration of a shaft in a geared system of rotors: 1st report, Bulletin of JSME, Vol. 23, No. 186, pp. 2111-2117.

[7] Iwatsubo, T., Arii, S. and Kawai, R. (1984), Coupled lateral-torsional vibration of rotor system trained by gears: Part 1. Analysis by Transfer Matrix Method, Bulletin of JSME, Vol. 27, No. 224, pp. 271-277.

[8] Choi, S.Y. Mau, 1995. “Dynamic analysis of geared rotor-bearing systems by the transfer matrix method”, ASME Design Engineering Technical Conferences 3(Part B), 84: 2967-2976.

[9] Hibner, 1975. “Dynamic response of viscous-damped multi-shaft jet engines”, J. Aircraft, 12(4): 305-312.

[10] Li, L. Yan and J.F. Hamilton, 1986. “Investigation of the steady-state response of a dual-rotor system with inter shaft squeeze film damper”, Transactions J. Engineering for Gas Turbines and Power, 108: 605-612.

[11] Gupta, K.D. Gupta and K. Athre, 1993. “Unbalance response of a dual rotor system: theory and experiment”, Transactions J. Vibration and Acoustics, 115: 427-435.

[12] Rao, J.S., 1996. “Rotor Dynamics, 3rd edition, New Age International Publishers, India.

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[13] Eshleman, R. And A. Eubanks, 1969. "On the critical speeds of a continuous rotor", J. Engineering for Industry, 91: 1180-1188.

[14] Mitchell, L. And D.D.M. Mellen, 1995. "Torsional-lateral coupling in a geared high-speed rotor system", ASME Design Engineering Technical Conferences 3 (Part B), 84(2): 977-989.

[15] Lee, J.W. and D.H. Ha, 2003. "Choi, Coupled lateral and torsional vibration characteristics of a speedincreasing geared rotor-bearing system", J. Sound and Vibration, 263(4): 725-742.

[16] Rao, J.S. Chang and T.N. Shiau, 1995. "Coupled bending-torsion vibration of geared rotors", ASME Design Engineering Technical Conferences 3(Part B), 84-2: 977-989.

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