Vibration Spectral Analysis of Air Slider Fan Bearing Pillow

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Abstract— *In recent years, air slider fans are becoming the most important equipment in industry for the transfer of materials and this motivates for their reliable and safe operation. Air slider fan runs at very high speed and this leads to develop the dynamic vibrations in the bearing. These vibration amplitudes are transferred to the bearing housing. If the frequency of vibrations is equal to the natural frequency of the bearing housing the condition of resonance occurs. This leads to failure. Hence it is very significant to find the frequency of resonance The aim of this work is to find the resonance frequency for the bearing housing. This project consist of Preparing a prototype model, identifying the defect type and frequency for which highest amplitudes are obtained using a FFT analyzer machine, 3D model generation of bearing housing using computer aided engineering tool and finding the natural frequencies and mode shapes at the bearing housing using FEM software. The scope of this work is to provide resonance frequency for Air Slider Fan Bearing Pillow. This frequency should be avoided to increase the life of the system and reduce the down time.*

Keywords— Air Slider Fan, Bearing Pillow, Vibrational Analysis, Resonance Frequency, FFT Analyzer

I. INTRODUCTION

Pneumatic conveying system is a conventional material handling system like belt conveyor or chain conveyor. The main advantage of pneumatic conveying system is that material is transferred in close loop, thereby preventing the environmental effect on the material and vice versa. Pneumatic conveying is a practical method for in-plant distribution of large amounts of dry powdered, granular and pelletized materials.[1]

In this research our main focus is to reduce the down time of this pneumatic conveying system. In a research (Lempiäinen, 1995) in Kaukopää paper factory of StoraEnso. During that research around 40 % of all faults were due to electrical systems. Motors represented 24 % of the total shutdown time due to electrical system faults. The number of failures of electric motors is smaller than its relative portion of the total shut-down time because of the longer replacement/clearing time than of man process automation faults, sensor damages and supply faults.[3] The percentage values vary significantly between researches but the same three fault categories are the most important in all faults, those are rotor faults, stator faults and the bearing faults. [4]

Natural frequency is the basic design property like stress or displacement. For some component one of the basic design approval criteria is fundamental frequency or specific mode shape. When external forcing frequency is equal to natural frequency the resonance occurs. At resonance the amplitude of vibration is built up and ultimately results in failure of the component. [5]

A. Problem Statement

The figure 1 shows the Air Slider Fan situated at ACC Limited, Gulberga. The fan motor for the air conveying system were equipped with a variable speed control for better adjustment to frequently changing operating conditions as well as for the sake of saving energy. This leads to develop aerodynamic forces in the air slider fan system.[2]

The fan is used to provide high velocity air with moderate pressure to Pneumatic conveying system in the plant. The early failure of bearings and bearing pillow are the associated problems with the air slider fan. As bearing and bearing pillow are very critical components of conveying system the failure of them cause's huge economic losses due to unscheduled production stop. The production cannot be resumed until the bearing and bearing pillow were replaced. Hence it is necessary to prevent the failure of bearing pillow and to examine vibration spectrum for different bearing defects.

As stated earlier that the Air Slider Fan is very critical equipment in the production line hence it is not possible to conduct the experiments on the actual Air slider fan. To overcome this difficulty a scaled prototype is developed using the scaling rules and the vibration tests were conducted on that prototype.

II. TEST RIG DESIGN

The designers employ most powerful analysis tools using the most elaborate electronic computers even though actual testing is required in order to extract some input design parameters and also to ensure the proper functioning of the designed system [18]. Structural dynamic properties of structures are one of fundamental requirements in designing such structures and evaluating their control systems effectiveness. Finding structural dynamic properties-often including natural frequencies, mode shapes, and damping ratios. Although performing ground vibration tests in case of heavy and huge machines is very difficult to do and also requires advanced and huge test instrumentations and considerable expense and time. But through testing a small scale model of the structure, which simulates the behaviour of its prototype exactly, not only modal parameters can be extracted, but also the reliability and accuracy of assumptions used in numerical and analytical models can be verified. Besides, the designers can access some useful data during designing and pre-manufacturing processes. Through fabricating and testing such small-scale models, design modifications and revisions will be possible without costly and time consuming full-scale fabrication and tests. Scaling laws provide the relationship between a full scale structure and its scale models and can be used to extrapolate the experimental data of a small, inexpensive, and easily tested model into design information for the large prototype [8].

A. Air Slider Fan Specification

Motor		**Bearing**		**Bearing Pillow**	
RPM	6000-10000	Type	Cylindrical roller	Length	316mm
Power	3Ph-15KW (20HP)	Inner Diameter	158mm		
Efficiency	-	Outer Diameter	204mm	Width	86mm
Output Shaft Diameter	156mm	Roller Dia.	32mm		
		Roller Length	56mm		

Table: 1 Air Slider Fan At ACC Ltd

B. Similarity Condition For Vibration In Prototype And Model

The frequency equation may be written for model and prototype. By defining scale factor λ, the variables of the prototype can be written as

$f_{prototype} = \lambda f_{model}$ [10]

The similarity conditions between model and prototype are determined by substitution of λf_{model} into the frequency equation of the prototype and by requiring that the result be the frequency equation of the model (complete similarity)[7]. It means that if a model is Geometrically scaled by a specific scale factor denoted by λ , then the ratio of the frequency of any particular mode shape of the model to that of corresponding mode shape of the prototype will be equal to the inverse of geometrical scale factor 1/λ.

The above system is scaled down with a scaling factor (λ) 6 to produce a test rig as shown in figure 2. The Test Rig is equivalent to the actual Air slide fan used in Air conveying systems in industry. The test rig consists of a shaft with fan at the end of it, which is supported on a bearing. The design incorporated a bearing, damage bearing at driven head of electromotor and a coupling disk system. The bench type model is about 0.42 m in length. This permits the damaged and undamaged bearing signals to be observed simultaneously. The roller bearing is tested at constant speed of 1600 rpm with Cylindrical roller bearing type (with outer race and inner race defects), have been used for analysis.

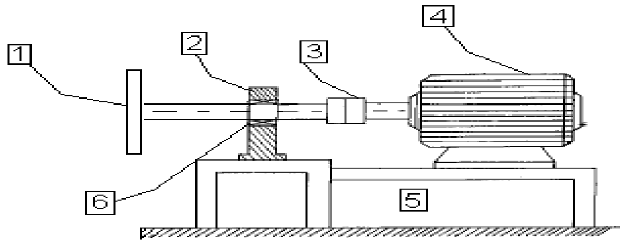

Fig. 2: Layout of Prototype Model

1-Fan
2-Bearing Pillow
3-Shaft Coupling
4-Electro-motor
5-Steel frame Bed
6-Roller Bearing

Fig. 3: Photo of Prototype

The coupling disk can be adjusted to overcome an angular misalignment. All experiment has been done for 6 & 8 blade to apply different loads on the bearing but the result of 6 blade fan was shown in this study.

The test rig assembly used for the experimentation consisted of an electromotor have been coupled to a blade fan. The power of the electromotor was 1.5 K.W (2hp), three phase variable rpm. Details of the prototype are shown in the below table 2

Motor		**Bearing**		**Bearing Pillow**	
RPM	1600	Type	Cylindrical roller	Length	150mm
Power	3Ph-1.5KW (2HP)	Inner Diameter	27mm		
Efficiency	77%	Outer Diameter	52mm	Width	60mm
Output Shaft Diameter	25mm	Roller Dia.	8mm		
		Roller Length	14mm		

Table 2: Prototype Specification

Vibration data was collected on the regular basis after the run in period. The experimental procedure for the vibration analysis consisted of taking vibration readings at two selected locations over the electromotor.

They were taken on the drive end and non-drive end of the electromotor. Vibration measurements were taken on the DE and NDE of the electromotor using a 4 Channel FFT Analyser.

III. Experimental Testing

Firstly the upper portion of bearing cap is made available free from dirt and grease by cleaning it. Then an accelerometer is placed on it. By pushing a start button, vibration spectrums are captured. As the desired vibration spectrum is captured, same procedure is adopted for the next spectrums.

Photo 4: FFT analyser connections

A. *Characteristics of the measurement system*

1) Displacement Sensor

Eddy-current transducer (Shinkawa® VK-202A) with over 2000µm linear range,787mV/100µm sensitivity, and DC to 25 kHz frequency response.

2) Accelerometers

PCB® ICP® accelerometer with 10.2 mV/(m/s2) sensitivity (±10%), 0.5Hz to 25 kHzfrequency range (±5%), needs 2-20 mA constant current excitation.

3) Data acquisition system

National Instrument® 4472 dynamic signal acquisition (DSA) board with 45 kHz alias-free bandwidth, 8 simultaneously sampled analog input channels with 24-bit resolution.

4) Encoder

Sick-Stegmann® DGS 20 incremental encoder with 720 PPR resolution.

IV. Iv. Experimental Natural Frequency

The Experimental data is used to find natural frequency of the Bearing Housing with the help of FFT analyser as shown in below table

Sr. No	Natural Frequency(kHz)
1	6.136
2	9.804
3	15.467

Table 3

V. Modeling And Finite Element Analysis Of Bearing Pillow

The 3-D solid model of the Bearing Housing was build using CATIA P3V5R19.

ANSYS workbench 13® was used for pre-processing, solving and post processing. Material properties of Structural steel, grade EN-GJS 600-3 were selected. Solid 3-D model of the Bearing Housing was meshed using Solid 187 element (3-D 10-Node Tetrahedral Structural solid element). The FE model consists of 78,749 elements.

Next, Modal Analysis was used to extract first 10 natural frequencies and mode shapes of the model. These natural frequencies are as follows

A. *Natural Frequency & Mode Shapes*

The below table shows the result of modal analysis of Bearing Housing [natural frequency and mode shape]

Mode	Frequency [Hz]
1	5682
2	5968.1
3	9676.1
4	12189
5	15618
6	20387
7	22227
8	22385
9	24517
10	26275

Table 4: Mode shape and natural frequency

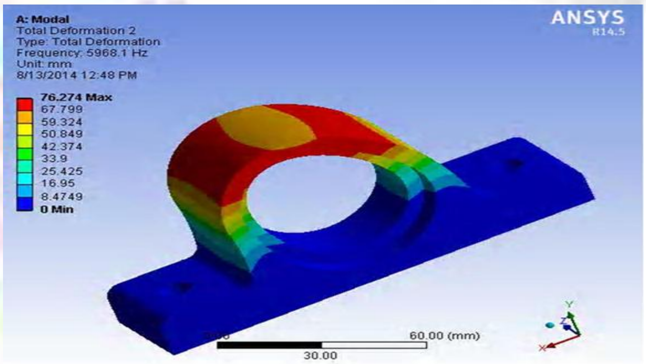

Fig. 5: second mode shape

VI. Result And Discussion

The natural frequency of bearing housing is found using FFT analyser experimentally and then the values are compared with computational techniques (ANSYS) to find resonance frequency and mode shapes. Those are given in below table

Mode No	**Natural Frequency of prototype obtained Experimentally(kHz)**	**Natural Frequency of prototype obtained in ANSYS(kHz**	**Percentage Deviation**

		)	
1	-	5.673	-
2	6.136	5.951	3.0065
3	9.804	9.642	1.649
4	-	12.137	-
5	15.467	15.554	0.557
6	-	20.255	-
7	-	22.227	-
8	-	22.385	-
9	-	24.517	-
10	-	26.275	-

Table 5: Comparison between experimental and FEAs natural frequency

The resonance frequency of air slider fan prototype are shown in table 5, As we have scaled down this prototype with a scaling factor 6, we need to apply the scaling law to find resonance frequency of Air slider bearing pillow.

Mode no.	**Resonance Frequency of Prototype in kHz** **[$f_{prototype}$]**	**Resonance Frequency of Air Slider Fan Bearing Pillow in kHz** **$f_{prototype} = \lambda f_{model}$**
2	5.951	35.706
3	9.642	57.852
5	15.554	93.324

Table 6: Resonance Frequency of Air Slider Fan Bearing Pillow

VII. Conclusion

In this dissertation work, the structural response of Air Slider Fan Bearing Pillow System is done using Finite Element Analysis, which in turn is validated and a comparative study is done using 4-channel FFT analyser. The brief conclusions of the dissertation work are:

- Three different resonance frequency were found these resonance frequency should be avoided to reduce risk of failure.
- The comparative result obtained from Finite Element Analysis and experimental, satisfy the range for resonance condition.

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