Condition Monitoring of Centrifugal Blower Using Vibration Analysis

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Abstract— Condition monitoring is the process used to analyze the operating condition of rotating machines. Vibration analysis is the most commonly used technique. In this paper vibration analysis is used to analyze the operating condition of Centrifugal Blower under various misalignment conditions. The experiments were carried out for various parallel misalignment and angular misalignment conditions between blower shaft and motor shaft. Different levels of misalignments conditions were introduced to study the effect on rotating machinery. Vibration signatures were acquired for normal and faulty conditions at different cut-off speeds of 1250 rpm, 1450 rpm and 1650 rpm in horizontal, in vertical and in axial direction. The acquired vibration signatures were presented and analyzed for these fault conditions. It was observed in the vibration spectrum that the vibration amplitude component at 2X is predominant in both the cases also it was observed that the slight increase in 1X and 3X vibration amplitude component at higher speed with higher level of misalignment. Also the comparison of 1X, 2X and 3X vibration amplitude components for all the three measurement direction for both type of misalignment were made.

Keywords— Condition Monitoring, Vibration Analysis and Shaft Misalignment

I. INTRODUCTION

Criticallity and failure mode analysis techniques are commonly used to identify where improvements in machinery availability and reductions in maintenance costs can be achieved through the integration of condition monitoring techniques [1]. The main common causes of mechanical vibration with rotating shafts are misalignment between driver shaft and driven shaft. Ultimately it causes failures in bearing and shaft couplings. It is practically difficult to make two shaft perfectly aligned each other in other words always a small amount of misalignment is present. There are two types of misalignment problems, first one is Parallel misalignment and the second one is Angular misalignment. Usually in rotating machines misalignment may occurs either only in parallel or only in angular direction or sometimes both combined. Parallel misalignment as shown in Figure 1 is the case when two shafts centerlines are parallel, but not lies on the same centerline. This type of misalignment may arise either in horizontal direction, or in vertical direction or in both the directions [2].



Figure 1: Parallel Misalignment

Angular misalignment as shown in Figure 2 is the case when two shafts centerlines are not parallel to each other but inclined. This type of misalignment takes place either at driver shaft end or at driven shaft end.



Figure 2: Angular Misalignment

II. VIBRATION MONITORING - CONDITION MONITORING TOOL

Vibration analysis is a useful tool for evaluating machines general running conditions. In this work vibration analysis is used to analyze the machine running condition. Guang Zhao [3] studied the meshing forces of misaligned spline coupling and its influence on rotor system, they mentioned in the concluding remarks 2X rotating speed appears when misalignment is present. Estupiflan et al. [4] analyze the

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energy losses due to misalignment in rotating machinery by establishing a correlation between vibration levels, energy consumptions and different degrees of misalignment. They have adopted statistical model based on the Response Surface Methodology (RSM) for data analysis in industrial case studies. Kirankumar et al. [5] studied vibrations analysis to study unbalance and misalignments and imperfect bearings on rotating systems. Dabiri et al. [6] uses vibration analysis to analyze the unbalance effect on blower. Amarnath et al. [7] used different vibration monitoring and analysis techniques such as time domain analysis, frequency domain analysis and spike energy analysis for suitability in identifying different defects in bearings. Monitoring of vibration provides an early warning of impending failures [8].

In this paper an experimental investigation is carried out to investigate the effect of shaft misalignment both in parallel and in angular direction on rotating systems.

III. EXPERIMENTAL SETUP

The Centrifugal Blower experimental setup [9] is used for this work to study the effect of shaft misalignment. The Figure 3 represents the photographic view of Centrifugal Blower Experimental Test Setup. The blower shaft is connected to motor shaft of same diameter through an electromagnetic coupling. The variable speed DC motor is used to vary the speeds. To introduce the horizontal movement of the steel plate, a lead screw is attached at the bottom of the steel plate. The LAB VIEW 7® (Laboratory Virtual Instrument Electronic Workbench NI-National Instrument) software is used to acquire vibration signals with Data Acquisition unit (DAQ) through accelerometer (Model 621B40, IMI sensors, sensitivity is 1.02 mV/m/s^2 and frequency range up to 16 kHz) in horizontal, vertical and axial directions. The software displays vibration spectrum both in time domain and frequency domain and stores data in a data file.

IV. RESULTS AND DISCUSSION

The experimental works for various parallel misalignment and angular misalignment were conducted and the corresponding vibration signatures for each test run were recorded.

A. Parallel Misalignment Condition

Parallel misalignment in vertical direction is created by inserting shims below the steel plate at all four corners on which blower is mounted. Required amount of parallel misalignment were created by increasing the equal number of shims in all four corners. Experiments were conducted for healthy, and parallel misalignment of 0.40 mm, 0.60 mm and 0.80 mm at shaft speeds of 1250 rpm, 1450 rpm and 1650 rpm respectively. The vibration signals were acquired in horizontal, in vertical and in axial direction for each case of experiments conducted.



Figure 3: Centrifugal Blower Experimental Test Setup

The Figure 4 to Figure 6 represents the vibration spectrums under healthy condition, 0.40 mm, 0.60 mm and 0.80 mm of parallel misalignment conditions at shaft speed of 1250 rpm in horizontal, in vertical and in axial direction. The Figure 7 to Figure 9 represents the vibration spectrums under healthy condition, 0.40 mm, 0.60 mm and 0.80 mm of parallel misalignment conditions at shaft speed of 1450 rpm in horizontal, in vertical and in axial direction. The Figure 10 to Figure 12 represents the vibration spectrums under healthy condition, 0.40 mm, 0.60 mm and 0.80 mm of parallel misalignment conditions at shaft speed of 1650 rpm in horizontal, in vertical and in axial direction.

Here for discussions higher speed is considered since the effect of parallel misalignment is high at higher speeds. As observed from these spectrums, at healthy condition the vibration amplitude is within the specified limits and no predominant frequencies are observed. As parallel misalignment increase to 0.40 mm, 0.60 mm and 0.80 mm, the change in vibration spectrum is observed. In the vibration spectrum it is observed that the 2X vibration amplitude component is predominant in all the spectrum. The 2X vibration amplitude component in horizontal direction is 0.02097 m/s² at 0.4 mm offset, is 0.035484 m/s² at 0.6 mm offset and is 0.05322 m/s² at 0.8 mm. The 2X vibration amplitude component in vertical direction is 0.018326 m/s^2 at 0.4 mm offset, is 0.032219 m/s^2 at 0.6 mm offset and is 0.052819 m/s² at 0.8 mm. The 2X vibration amplitude component in axial direction is 0.064107 m/s^2 at 0.4 mm offset, is 0.099744 m/s² at 0.6 mm offset and is 0.126189 m/s² at 0.8 mm. It is also observed that higher vibration amplitude at higher parallel misalignment. The 2X vibration amplitude components in vertical direction for different parallel misalignment conditions are smaller compared to same values in horizontal direction. It is also observed that the 2X vibration amplitude components in axial direction for different parallel misalignment conditions are greater values compared to same values in both horizontal and vertical directions. These 2X vibration amplitude components are more that 50% in axial direction. This increase in 2X component clearly indicated the presence of misalignment [10].

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Figure 5: Vibration spectrums in vertical direction for various parallel misalignment at 1250 rpm



Figure 6: Vibration spectrums in axial direction for various parallel misalignment at 1250 rpm



Figure 7: Vibration spectrums in horizontal direction for various parallel misalignment at 1450 rpm



Figure 8: Vibration spectrums in vertical direction for various parallel misalignment at 1450 rpm



Figure 9: Vibration spectrums in axial direction for various parallel misalignment at 1450 rpm



Figure 10: Vibration spectrums in horizontal direction for various parallel misalignment at 1650 rpm



Figure 11: Vibration spectrums in vertical direction for various parallel misalignment at 1650 rpm



Figure 12: Vibration spectrums in axial direction for various parallel misalignment at 1650 rpm

B. Angular Misalignment Condition

Angular misalignment between blower shaft and motor shaft were created by inserting shims below the steel plate at two corners on which blower is mounted. Required amount of angular misalignment were created by increasing the equal number of shims in only two corners and respective angles were calculated. Experiments were conducted for healthy, and angular misalignment of 0.0788⁰, 0.1314⁰ and 0.1839⁰ at shaft speeds of 1250 rpm, 1450 rpm and 1650 rpm respectively. The vibration signals were acquired in horizontal, in vertical and in axial direction for each case of experiments conducted.

The Figure 13 to Figure 15 represents the vibration spectrums under healthy condition, 0.0788° , 0.1314° and 0.1839° angular misalignment conditions at shaft speed of 1250 rpm in horizontal, in vertical and in axial direction. The Figure 16 to Figure 18 represents the vibration spectrums under healthy condition, 0.0788° , 0.1314° and 0.1839° angular misalignment conditions at shaft speed of 1450 rpm in horizontal, in vertical and in axial direction. The Figure 21 represents the vibration spectrums under healthy condition spectrums under healthy condition spectrums under healthy condition spectrums under healthy condition spectrums under healthy condition, 0.0788° , 0.1314° and 0.1839° angular misalignment conditions at shaft speed of 1450 rpm in horizontal, in vertical and in axial direction.



Figure 14: Vibration spectrums in vertical direction for various angular misalignment at 1250 rpm



Figure 15: Vibration spectrums in axial direction for various angular misalignment at 1250 rpm



Figure 16: Vibration spectrums in horizontal direction for various angular misalignment at 1450 rpm



Figure 17: Vibration spectrums in vertical direction for various angular misalignment at 1450 rpm



Figure 18: Vibration spectrums in axial direction for various angular misalignment at 1450 rpm



Figure 19: Vibration spectrums in horizontal direction for various angular misalignment at 1650 rpm



Figure 20: Vibration spectrums in vertical direction for various angular misalignment at 1650 rpm



Figure 21: Vibration spectrums in axial direction for various angular misalignment at 1650 rpm

Here for discussions higher speed is considered since the effect of angular misalignment is high at higher speeds. As observed from these spectrums, at healthy condition the vibration amplitude is within the specified limits and no predominant frequencies are observed. As angular misalignment increase to 0.0788⁰, 0.1314⁰ and 0.1839⁰, the change in vibration spectrum is observed. In the vibration spectrum it is observed that the 2X vibration amplitude component is predominant in all the spectrum.

The 2X vibration amplitude component in horizontal direction is 0.035716 m/s² at 0.0788⁰ offset, is 0.053245 m/s² at 0.1314° offset and is 0.069885 m/s^2 at 0.1839° . The 2X vibration amplitude component in vertical direction is 0.031762 m/s^2 at 0.0788^0 offset, is 0.050239 m/s^2 at 0.1314^0 offset and is 0.066579 m/s² at 0.1839⁰. The 2X vibration amplitude component in axial direction is 0.097739 m/s² at 0.0788° offset, is 0.126875 m/s² at 0.1314° offset and is 0.159635 m/s² at 0.1839⁰. It is also observed that higher vibration amplitude at higher angular misalignment. It is also observed that the 2X vibration amplitude components in axial direction for different parallel misalignment conditions are greater values compared to same values in both horizontal and vertical directions. These 2X vibration amplitude components are more that 50% in axial direction. This increase in 2X component clearly indicated the presence of misalignment [10].

C. Comparison of 1X, 2X and 3X vibration amplitude components for Parallel and Angular Misalignment

The comparison of 1X, 2X and 3X vibration amplitude components for parallel and angular misalignment conditions were made. To show the difference the results of misalignment effect in axial direction for both parallel and angular misalignment conditions are presented. The Figure 22 represents the comparison of 1X, 2X and 3X vibration amplitude components for healthy, 0.40 mm, 0.60 mm and 0.80 mm of parallel misalignment at 1250 rpm, 1450 rpm and 1650 rpm speed respectively in axial direction.



Figure 22: Comparison of 1X, 2X and 3X for various parallel misalignments at different speeds in axial direction

The Figure 23 represents the comparison of 1X, 2X and 3X vibration amplitude components for healthy, 0.0788° , 0.1314° and 0.1839° of angular misalignment at 1250 rpm, 1450 rpm and 1650 rpm speed respectively in axial direction.

It is observed from these comparison graphs for parallel misalignment in axial direction at different speeds, the 2X vibration amplitude is the predominant and found increases with speed at higher level of fault. Where as for angular misalignment



at different speeds in axial direction

V. CONCLUSION

Vibration analysis is the best and powerful techniques adopted in condition monitoring process. As noted in the industries which are using rotating machinery most common problem is misalignment. In this project vibration analysis is used to study various misalignments fault conditions such as in parallel and in angular direction. Many experiments were carried out with different level of parallel misalignment and angular misalignment at different selected speeds. In this study it is noticed that the effect of misalignment is very small at lower speed. Whereas the speed increases the effect of misalignment is high on rotating machinery causing more vibrations. As noticed from vibration spectrums for parallel misalignment and angular misalignment, the vibration amplitude at 2X running speed is predominant. This 2X vibration amplitude component is increase with increase in level of misalignment. This increase in 2X component clearly indicated the presence of misalignment. Also at higher speed this components values are more. In addition to this a small increase in 1X and 2X vibration amplitude component are observed at higher speed with higher level of misalignment.

Comparison of these 1X, 2X and 3X vibration amplitude component for various misalignment conditions in all the three measurement directions were also made to show the clear difference between these observed values. Monitoring rotating machinery using vibration analysis provides early warning on these faults with dominant frequency in the vibration spectrum at 2X component. With this indication a corrective action can be initiated to avoid further development of these faults and ultimately to reduce machine breakdown. Hence it is suggested based on the experimental studies the vibration analysis is the best method to detect mechanical faults in rotating machinery.

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