Health Condition Monitoring and Control of Vibrations of a Rotating System through Vibration Analysis

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This paper presents condition monitoring, fault detection, and control of vibration of a rotating system using vibration analysis. The system includes ball bearings, pulleys, impellers, shafts, and an electric motor. The experimental approach generated different levels and forms of vibration that were measured using a vibration analyzer (CSI 2140). A vibration analyzer is used for determining the characteristics and nature of vibration of the experimental system. System vibrations were measured in terms of amplitude, phase angle, and frequency during the experimental operating conditions. After that, the experimental results were compared with the ideal working conditions mentioned in the ISO standard for the determination of machine health and condition monitoring. This paper also presents a passive control technique to enhance the machine’s health condition. The generated faults of the system have been detected and solved by using an exact measurement of polar plot analysis. This approach of condition monitoring, fault detection, and improving operating conditions can be used to visualize and analyze vibration data in any rotating system. Furthermore, this technique would be a feasible and effective way of finding strategies for the maintenance of rotary machines.

1. Introduction

Vibration is a common phenomenon in rotating machines. A rotating machine operating without any vibration or noise is imaginary in a working environment. Additionally, rotary and rotating machines are used in different industries, and they are an important part of large industries. As a consequence, the majority of the industrial purposes use rotating machines, and they are considered significant assets for industry [1]. Rotating machines with reciprocating or rotating segments generate vibrations when disturbances are provided from different sources, such as motors, engines, pulleys, gears, and bearings [2–4]. These vibrations are usually generated when applying forces to produce free oscillatory movement [3, 5, 6]. Machine vibrations are directly related to conditions and problems of rotating systems. These vibrations are generated from the rotating or reciprocating parts of a system in any operating conditions. Therefore, machine vibrations at the outer surface contain vital information to monitor the functioning and running conditions of any system. As a consequence, extracting information from machine vibration to analyze the running state of any machine is a growing interest now.

However, unnecessary vibration can also damage machine structure and create performance and operation problems [1, 7]. Vibration occurs due to misalignment of shafts, and they can even be found in the magnetic levitation system where there is no contact between the levitated objects and actuators.
Therefore, continuous monitoring of rotating machines in operating conditions is essential for safe, reliable, and high-quality performance.

There are many reasons for the generation of vibrations, and approximately ninety percent of vibrations are generated due to misalignment and unbalance of masses in the shafts of different rotating machines [12]. However, it is difficult to maintain the exact alignment of any rotating shafts and masses during practical operations due to the thermal distortion of ball bearings, thermal growth of machine parts, movement of structures, generation of forces due to change of pressure, and high operating temperatures [13–15]. In the case of unbalancing of rotating masses, the center of mass can be changed over time which may also generate vibration [16, 17]. Hence, unbalance and misalignment of shafts in rotating masses are the two significant sources of machine vibration [1, 18]. However, it is challenging to balance rotating mass entirely due to the uniform density of rotational materials and loss of mass during operation [19]. Misalignment of the shaft as well as unbalance of mass is always present in rotating machines, and in the majority of practical applications, they are taken place under any operating conditions. As a consequence, vibration is common in rotating machines that are directly related to the working or running state of any rotating system. However, unnecessary vibration is annoying as humans can even sense small displacement/disturbances that may create unbalanced systems [20]. Therefore, proper balancing of rotary mass is needed that can reduce the amplitude of vibration in rotating machines [21].

On the other hand, the determination of machine vibrations in terms of frequency, amplitudes, and phase angles indicates different aspects of the running state of a rotating system. The sampled data show the operating performance of different components of an entire system that can potentially indicate the possibility of failure or breakdown under different conditions [22, 23]. Moreover, the level of unnecessary vibrations in rotating machines indicates wear, friction, unbalance, and looseness of rotating parts, such as bearings, shafts, screws, and so on [2]. All these factors reflect an increase in machine vibrations and create extra loads on rotating parts. Therefore, machine vibration can be used to monitor the operating conditions in any working environment. It can also be used to make a strategic decision for the proper maintenance of machines. As a consequence, it is considered an indicator of inaccuracy during the operation of rotating machines. In addition, it is now used to analyze and monitor the running operating conditions of any rotary machines [24–26].

The aim of this paper is to detect faults and monitor the operating conditions of a rotary system through vibration. The rotary system was developed using common industrial elements, such as ball bearings, pulleys, screws, electric motor, and shafts where vibrations were measured by using a vibration analyzer (CSI 2140) for condition monitoring. This vibration analyzer is an accelerometer that has three axes in three different directions including horizontal, vertical, and axial. By adjusting and placing the three accelerometers at three different measurement points, the vibration could be quantified directly from the developed system. In any mechanical system, condition monitoring is significant because it investigates and analyzes different parameters related to the operation of the system that can identify the possible failure and breakdown of machines [27–30]. In this work, the fabricated rotary system was disturbed using different levels of vibrations. After that, they were determined and identified using the vibration analyzer (CSI 2140) in terms of frequency, amplitudes, and phase angle to identify the running condition of the system which is rare in previous research.

This paper also shows that it is possible to identify and monitor the condition of a rotating system by analyzing the sampled data obtained from the vibration analyzer. The experimental results were compared with the standard level of vibration required to ensure a safe, satisfactory, reliable, and quality operation of rotating systems provided in the ISO [12]. Finally, after detecting faults, the developed system was balanced through the balancing of mass technique where an exact amount of required mass was determined by the mass measurement process using the polar plot analysis. It is a technique where polar coordinates (radius r) are represented as angular distance (θ). The outcome of this research shows that this method can improve the overall machine health condition of any rotary machine.

2. Governing Equations

Vibrations are known as a periodic process of free oscillations with respect to an equilibrium body. In any mechanical system, such as a rotating turbomachinery system, they are generated due to external forces, unbalance, and misalignment of rotating shafts. The main reason for the generation of vibration in any condition is the change in the center of gravity of the rotating mass. Therefore, if rotating masses do not turn on the same rotational axis, then vibration and damages occur.

The generated faults of a rotating system create a specific trend of spectral data that can be used to identify the inaccuracy and problems of a rotating system. This data also helps to make necessary strategic measurements to reduce the unnecessary vibrations of the system.

In any mechanical system, mathematical modeling of vibration can be presented by the following equation [31, 32]:

$$m \ddot{x} + c \dot{x} + kx = F,$$  \hspace{1cm} (1)

where $m$ is the mass of the rotating system, $c$ is the damping coefficient, $k$ is the stiffness of the system, $x$ is the displacement of the system, and $F$ is the externally applied force.

The displacement $x$ of Equation (1) can be determined by using the following equation:

$$X_{rms} = \sqrt{\frac{1}{n} \left( X_1^2 + X_2^2 + \cdots + \cdots + X_n^2 \right)},$$  \hspace{1cm} (2)

where $X_1, X_2, \ldots, X_n$ shows the displacements for $n$ values in any rotating system. However, displacement $x$ is usually used to find the level of vibration.
In a rotating system, the root mean square (RMS) values indicate the contained power in the signal of vibration. It is directly interrelated to the contents of energy as well as the destruction capability of vibration that takes the history of time in the form of waves. Therefore, it is the measure of the root of the square of the mean values (squared) of the waveform. These values are proportional with respect to the area that lies under the waveform curve. The overall RMS value of the displacement and velocity of a system can be determined by the following equation.

\[ X_{\text{rms}} = \sqrt{\frac{\text{Sum of mid – ordinate} (d^2)}{\text{Number of mid – Ordinate}}} \]  

(3)

where \( d \) shows the displacements. In a similar way, the RMS value of the velocity was quantified.

The other two significant parameters of vibration are amplitude and phase angle where the amplitude of vibration generally increases with an increase in the speed of the rotating system. This increase of vibration amplitudes continues till it reaches the first critical speed of the rotating system. In this work, the generated vibration shows only a frequency of single spectra whose amplitude is precisely similar in all radial directions. During the unbalance or misalignment of rotating shafts and masses, the generated vibration spectra display a sinusoidal line at the operating speed. Amplitude and phase angle of vibration of a rotating system can be identified by the following equation [33, 34]:

\[ A = \frac{\text{mrf}^2}{\sqrt{(k - \text{mf}^2)^2 + [c(1 - \lambda)f - k]^2}} \]  

(4)

where \( f \) is the frequency of vibration, \( r \) is the radius, \( A \) is the amplitude of vibration, \( m \) is the rotary mass, \( c \) is the damping coefficient, \( k \) is the stiffness of the system, and \( (1 - \lambda) \) presents the reduction factor of damping of rotating machines.

The Fourier theorem shows that at any time, vibration waveform is possible to construct with several sinusoidal and cosine components of frequency. Fourier transform is a very effective tool to find and analyze signals generated from vibration. The spectrum of the Fourier transform gives an amplitude-frequency relation as well as a plot that can be used to find the amplitude of vibration of any rotating system. Moreover, Nyquist or polar analysis is also effective to present amplitude, frequency, and phase angle of vibration using a Bode plot.

In this work, the amplitude, frequency, and phase angle of vibrations have been plotted on a single circular chart to indicate the condition of the developed rotary system. However, vibration waveform is the summation of sinusoids of several amplitudes, frequencies, and phases. The Fourier transform is usually used to deconstruct the waveform of the vibration signal into its form of sinusoidal waves. By presenting the amplitudes of vibration in terms of the function of frequency, it is possible to show the outputs in the frequency domain.

Fast Fourier transform (FFT) provides a deeper and better understanding of the profile of vibration. In this work, an overhung rotor was used as the primary source of unbalancing of rotating mass, where FFT analysis of the rotor spectrum was used to find the single root mean square (RMS) value 1X peak. If \( N \) is the length of the vibration signal obtained from the system, then frequency lines or numbers are \( N/2 \). The bins of frequency (\( \Delta f \)) are equal to the frequency of vibration (\( f \)) when it is divided by the sample numbers (\( N \)). It indicates the interval between samples in frequency domain analysis. Therefore, the FFT analysis of the generated vibration signal can be expressed by the following equation.

\[ \Delta f = \frac{1}{T} = \frac{f}{N}. \]  

(5)

In this case, amplitudes were varied proportionally to the square of the rotating shaft speed that generated high radial and axial vibrations.

3. Experimental Setup

In this work, the experimental setup consists of a single-phase induction motor, supports, two pulleys with a belt, shafts with five bearings, and gears with coupling, as shown in Figure 1. In this system, an overhung circular disk (annotation J in Figure 1) has been used at the end of the motor shaft so that external mass can be added or removed for balancing of the developed system. It is mainly added to make the system more robust as the system was mainly developed for the generation of vibration through bearings. The overhung circular disk rotates during the operation of the setup. The disk is 127 mm in diameter, while 8 mm in thickness. There are a total of eight holes equally apart from the center, and this distance is 51 mm (marked circular disk in Figure 2). The belt and pulley were used for power transmission in the developed setup.

The specification of the motor that is used as the source of power during the operation of the experiment is 0.25 hp and 1400 rpm. The motor (annotation A in Figure 1) shaft is connected to a set of two pulleys (annotation C in Figure 1 with 75 mm and 100 mm diameters) by using a V-belt (annotation D in Figure 1 with a length of 560 mm and width 16 mm). A shaft (fabricated from mild steel, annotation E in Figure 1 with 17 mm diameter) is attached to a pulley located at the upper side. The axis of the upper pulley is around 165 mm in height from the shaft of the single-phase motor, and the motor shaft is supported by using two ball bearings at a distance of approximately 143 mm that are housed inside the bearing housings (annotation G in Figure 1). The housings of the ball bearing are supported by using a mild steel supporting block (annotation B in Figure 1 with a width of 222 mm and length of 254 mm). In addition, the shaft of the motor is extended up to a length of around 156 mm from the outer side of the ball-bearing housings.

The experiment is conducted in the region of all rotating parts, such as all bearing houses, coupling, belt and pulleys,
and meshed gear. However, in this paper, we have mentioned only the experimental analysis of the ball bearing (F) and bearing housings (G) on top of the support (B). The photograph of the experimental setup shows all the probe locations (axial, vertical, and horizontal), the position of the tachometer, and the circular disk where the experiment is performed, as shown in Figure 2.

4. Experimental Methods

The three transducer probes were labeled as A, B, and C of the CSI 2140 machinery health analyzer for vertical, horizontal, and axial directions, respectively.

In this experiment, bearing housing (G) was used for the measurement of the generated vibration. The power supply of the motor was turned on after mounting all the three transducers to their prescribed locations. The separations of each transducer were around 90° when compared to each other. After that, the spectral data and waveforms were measured and found for the three transducers (A, B, and C). The tachometer was also used for the measurement of the phase and peak of the bearing housing (G).

The measured values were then compared with the standard values provided in the ISO chart for different levels of vibration, as shown in Table 1. By observing the generated pattern of the waveform and spectral plot, the fault and problems of the system were identified [36].

It is possible to estimate the type of vibration sources by analyzing the characteristics or pattern of the FFT values. In the FFT, intervals between samples were quantified using a frequency bin as shown in Equation (5). Therefore, necessary solutions were applied to solve the detected issues depending on the measured values accordingly. After applying the required steps to reduce the generated vibration, the measurement was performed again by following the same steps. After getting the new measured data, it was compared again with the standard values that were shown in the ISO standard chart (Table 1). It was observed that after minimizing the vibration level, the condition of the system became stable at the running operating condition.

4.1. Data Analysis

4.1.1. Investigation on the Initial Condition of the Setup. At first, the transducers were placed to their prescribed
locations on the bearing housing (G) to determine the characteristics of the generated vibration in all three directions. The power of the single-phase motor used in this work was 180 W. Therefore, the system was a class I category as the motor power (180 W) was lower than the reference value, as shown in the standard ISO chart (up to 15 kW for class I, Table 1).

To maintain a balanced system with a good or satisfactory level, the required values of the RMS velocity need to be equal or lower than 1.80 mm/sec, while the peak values must be equal or less than 0.10 in/sec. After the measurement, it was observed that the RMS value was 2.22 mm/sec at the phase of 130.8 degrees in the vertical direction (Probe A) of bearing housing (G) as depicted in Table 2 and Figure 3. This value indicates the unsatisfactory stage of vibration as per Table 1. The maximum RMS value was 7.37 mm/sec at the phase of 52.2 degrees in the horizontal direction (Probe B) of bearing housing (G), as shown in Table 2 and Figure 4. At the phase of 220.4 degrees, the RMS value was 0.45 mm/sec in the axial direction (Probe A) of bearing housing (G), as shown in Table 2 and Figure 5. Only this RMS value exists within the good range of the vibration which is acceptable.

However, the vibration in vertical and horizontal directions exceeds the satisfactory level and reaches the unacceptable stage as mentioned in the standard ISO chart (Table 1). The characteristics of velocity for Probe A and Probe B shows both peak points that generated at the 1X position of frequency. It was estimated that the reason for the generation of vibration was the unbalance condition of rotating parts in the system. Hence, the critical RMS value of Probe A and Probe B was responsible for the unbalanced state of the system that was considered due to the unbalance of mass. This peak value could be reduced by the addition of the required mass in the overhung circular disk (annotation J in Figure 1 and circular disk in Figure 2) in the system due to the faults associated with the design. Additionally, the center of gravity was not on the rotational axis which caused the unbalanced state of the system. Therefore, mass balancing was required to overcome the unbalanced condition of the system.

4.1.2. Fault Reduction of the System. After detecting the problems associated with the unbalanced system, it was required to fix and reduce the faults. As the fault was estimated as unbalance of mass in the overhung circular disk, hence, it was solved by adding the required amount of mass. It was determined by the polar plot analysis using the trial and error method. In this way, the necessary values of mass were used to balance the overhung circular disk. As a consequence, vibration intensity was reduced to a specific level that balanced the system.

Figure 6 presents the polar plot that was used for the reduction of unbalance of mass. In the plot, the outer layer shows the rotation of the shaft in the anticlockwise direction. Due to the inertial effect and unbalanced mass, the phase angle was increased against the shaft rotation shown in the outer ring of the polar plot graph (Figure 6) [37]. In this case, the distance from the center to the outer ring was considered the radial distance, and it was assumed as 10 units. The RMS peak value for Probe B (horizontal direction) was observed at 7.37 mm/sec at a 52.20 phase angle and mentioned as vector 1 in the polar graph, as shown in Figure 6. Firstly, a mass of 4.5 g was added as trial mass at 180° in the polar plot graph. After the addition of this mass, it was obtained that the peak RMS value was reduced in all three specified directions (vertical, horizontal, and axial).

It was observed that the RMS value for Probe B remained high than the reference value in the ISO standard chart (Table 1), while the values for probes A and C were within the reference values. At the end of the experiment, the RMS value for Probe B was found 4.6 mm/sec at a phase angle of 26.9° (Table 3) that was in the horizontal direction. It was then shown in the polar plot graph as vector 2
After that, a new vector was drawn from the top of vector 1 to the top of vector 2 which was shown as vector 3. Finally, a vector 4 was drawn from the origin of the polar plot graph and parallel to vector 3, as shown in Figure 6. The measured angle between the extension of vector 1 and the new vector 4 was 39° which was the same angle that the correct mass went with respect to the mass used during the trial of the experiment. Therefore, the correct mass location was fixed at 220° in the polar plot graph.

The correct mass can be calculated by the following equation:

\[
Correct\ mass = \frac{\text{Length of vector 1} \times \text{Trial mass}}{\text{Length of vector 3}}. \tag{6}
\]

Experimentally, the length was measured with a fine ruler, and mass was determined by using a digital weighing

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**Figure 3:** Measured FFT spectrum in the vertical direction of the bearing housing.

**Figure 4:** Measured FFT spectrum in the horizontal direction of the bearing housing.
scale. In the polar plot graph (Figure 6), the length of vector 1 was 6.4 cm, the trial mass was 4.5 g, and the length of vector 3 was 3 cm. By inserting all these values in Equation (6), the exact amount of correct mass was 9.6 g.

4.1.3. Experiment with the Balanced System. The intensities of vibration became lower than the unbalanced state after adding the correct mass at the required phase angle in the polar plot. After balancing the system with the required values of mass (9.6 g at 220°), the RMS values and phase angle were measured again, as shown in Table 4. It is seen that the RMS values were reduced in all three directions (vertical, horizontal, and axial) for all the three probes (A, B, and C), and the spectral waveform was taken from the setup of the experiment.

The obtained spectral plots for the balanced condition show that the amplitude of vibration was decreased in all three specified directions. The reduction of the vibration amplitude was the highest in the horizontal direction (Probe B) decreasing the variation of velocity. Hence, it is seen that the variation of velocity in the horizontal direction (Probe B) in the balanced condition was low (Figure 7) compared to the waveform in the vertical (Probe A, Figure 8) and axial direction (Probe C, Figure 9). It is seen from Figure 9 that the peak-to-peak velocity was the highest in the axial direction and lowest in the horizontal direction (Probe B, Figure 7).

5. Results and Discussion

The peak RMS values of the balanced system were maintained below the reference value of 0.45 mm/sec for the good state of vibration in class I (Table 1) in all three directions. Figures 10–12 show the FFT spectrum in vertical (Probe
Figure 7: Waveform of velocity in the time domain in the horizontal direction of the bearing housing.

Figure 8: Waveform of velocity in the time domain in the vertical direction of the bearing housing.

Figure 9: Waveform of velocity in the time domain in the axial direction of the bearing housing.
A), horizontal (Probe B), and axial (Probe C) in the balanced condition of the system. After analyzing the unbalanced vibration condition, the required mass of 9.6 grams was added to the developed system at the circular overhung disk rotor at an angle of 220°. In fact, it was determined by finding the vibration characteristics in the unbalanced and experimental trial conditions.

It is also seen from Figures 10–12 that the system was stable in the balanced condition and observed that it was within the good and satisfactory level of ISO standard. By comparing the FFT analysis in Figures 3–5 (unbalanced system) with Figures 10–12 (balanced system), it is also clear that the velocity at the balanced system is within the ISO standard range, although during the experiment, the required amount of balanced mass was quantified using trial and error method. In the case of the balanced system, two main factors are very significant including precision and skill. Therefore, precise measurement of the required amount of the balanced mass is crucial for achieving steady operation.
mass is a key factor. Therefore, determination of mass and addition of them to the exact point of location in the overhang circular disk (J, in Figure 1) are very essential for balancing the system.

6. Conclusion

This experiment has been done to monitor the condition of a mechanical system. The system condition was balanced based on the ISO standard reference values. A mechanical system with rotating mass has been developed where the condition and fault of the system have been identified and balanced by applying a mass balancing approach. The system arrangement includes the capacity of vibration that was similar to the vibration produced in different types of machinery. The dominant form of the generated vibration in the developed mechanical system was due to the unbalance of mass in the rotary circular disk that was determined by the nature and behavior of the spectral plot found from the display of the machinery vibration analyzer. In the unbalanced state, the RMS values of probes A, B, and C in the horizontal, vertical, and axial directions were 2.22 mm/sec, 7.37 mm/sec, and 0.45 mm/sec, respectively. The spectral waveform plots along with the peak RMS and phase angle values were determined from the vibration analyzer (CSI 2140). The required values of mass needed for balancing the system have been determined by the analysis of the polar plot graph. A mass of 9.36 g was added to the circular disk at a phase angle of 220°. As a consequence, the intensity and amplitude of vibration were reduced, as well as the system turned into a balanced state. In the balanced condition, the RMS values of probes A, B, and C in the horizontal, vertical, and axial directions were 0.35 mm/sec, 0.38 mm/sec, and 0.30 mm/sec, respectively.

7. Future Recommendations

This paper shows the measurement of vibration in an unbalanced rotation mass (bearings) and the process of balancing the system by adding an external mass to the system. The most important as well as the complex part of the system is to set the vibration analyzer probes at the desired measurement location. Additionally, to make the system balance an external mass was directly added to the circular overhang disk that may damage the system when operating it at a higher speed. A more robust design for the addition of mass in the system would be considered a future recommendation to make the more stable and balanced even at a very higher motor speed. Also, finding the exact amount of required balanced mass with the required angular position of the mass is challenging task for the balanced system. A more automated and system algorithm could be developed and integrated with the system to make the system more feasible and precise would be a future recommendation for future experiments. However, finding the amount of balanced mass on a trial and error basis is a limitation for the current technique as presented in this work.

Data Availability

All data are available in the paper. However, any data will be available to the readers upon reasonable requests to the corresponding author.

Conflicts of Interest

The authors declare no conflict of interest.
Authors’ Contributions

All the authors have contributed substantially to the work reported. Conceptualization was done by M.E.H. Methodology was done by M.E.H., F.A., A.R., and F.R. Data curation was done by A.R. and F.R. Formal analysis was done by M.E.H., F.A., M.M.A., and A.R. Writing—original draft preparation—was done by A.R., M.M.A., and F.R. Writing—review and editing—was done by M.E.H., F.A., A.R., and F.R. All authors have read and agreed to the published version of the manuscript.

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