Maintenance Initiation Prediction Incorporating Vibrations and System Availability

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Received 04 October 2021; received in revised form 25 December 2021; accepted 26 December 2021

DOI: https://doi.org/10.46604/aiti.2022.8618

Abstract

As per ISO-10816, electric motors up to 15 kW are classified as Class I machines, and the major reason for their failure is that the vibrations in them are above the alert limit. This study presents a new model for predicting the condition-based maintenance (CBM) initiation points through vibration measurement in a system of Class I machines. The proposed model follows the accelerated life testing (ALT) procedure. ALT includes the formation of an artificial wear environment in bearings to analyze the resultant system vibrations on system availability. The artificial wear environment created is close to the real industrial situation. The results show that the prediction of the CBM initiation points is based on the established relation between the system availability and vibrations. Furthermore, a relation between the available time for maintenance initiation and different vibration velocities is demonstrated.

Keywords: availability, condition-based maintenance, alert limit, alarm limit, acceleration factor

1. Introduction

Reliability and availability management plays a prime role in the success of a company. For the economic performance of an industrial plant, maintaining high availability, reliability, and maintainability for the plant machines and their subsystems is crucial [1-3]. The machines in an industrial process plant can fail because of a wide range of reasons. The major reason behind machine failure is the unchecked magnitude of vibrations above the alert limit. The availability of a system or machine is affected badly when the machine failure occurs. The direct impact of the vibration levels on the system availability and the usability of that influence in condition-based maintenance (CBM) programs are not reported in the literature so far [4-5].

Over the years, maintenance has advanced to adopt the aspect of reliability. Reliability is a design attribute that shows the expected acceptable performance of an item. Reliability-centered maintenance (RCM) is used to decide the maintenance requirements of any physical asset to ensure its satisfactory operation. CBM is a subclass of RCM [3]. Typically, the objective of a CBM program is to devise a maintenance policy that optimizes the system performance according to the criteria such as cost, availability, and reliability [3].

Early failure detection is the responsibility of most CBM programs [4]. For the CBM application, a potential failure-functional failure (P-F) curve can be plotted between the failure resistance or health condition of a machinery system and the time period of its failure [3]. It can be inferred from this curve that as time increases, failure resistance decreases to complete the system failure. This curve can be used to explain various stages of system failure, i.e., failure initiation (I), potential failure (P), and functional failure (F). Along this curve, a point “P” can be identified, where there is a potential to fail,

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and a point “F” beyond “P” can be detected, where the system will not perform as expected. After the point “P,” the health condition decreases rapidly. There is no hard data to define P-F intervals [3]. Therefore, the best strategy is to employ methods that could effectively ascertain the machine condition before the potential failure occurs, and this should permit scheduling the repair activity before the P-F interval. Shin et al. [5] examined the CBM strategy from various perspectives and addressed the data, procedure, and techniques for implementing the CBM approach. Li et al. [4] proposed a CBM model for assuring average system availability and plant safety.

Electrical motors are integral parts of the majority of the machines installed in an industrial plant [3, 6]. The failure of any one of these motors degrades the machine performance, which, in turn, influences the overall plant availability. Manjare et al. [6] reviewed machine learning (ML)-based fault detection techniques and predictive maintenance (PdM)/CBM strategies for electrical motors used in industrial plants. Furthermore, the vibration data acquired from accelerometer sensors are extensively used for data analysis. Kumar et al. [7] presented a comprehensive review of various faults in electric motors, failure causes, and advanced condition monitoring and diagnostic techniques. However, the availability aspects associated with the failure of electric motors and maintenance initiation points are not considered in these studies.

In this experimental study, the deterioration in a system of electric motors with five HP-rated power Class I machines is simulated by deliberately creating the vibrations above the alert limit. An analysis is performed for the values of the alert and alarm limits of vibration velocity (root mean square (RMS)). The alert limit corresponds to the failure initiation, and the alarm limit corresponds to the potential failure for the system [3]. The maintenance initiation points for CBM are defined only in accordance with the shrinkage pattern of system availability with the vibration velocity for a particular system load. The maintenance initiation points of CBM for different system loads are predicted according to this relation. The proposed approach can be applied to accurately predict the value of vibration velocity (RMS) for the maintenance initiation points of Class I machines.

The frequent failure of certain Class I machines in an industrial process plant is mainly due to bearing wear, which gives rise to vibrations in the machines. As the wear increases, the system vibration level increases. In this study, the system failure is simulated by artificially wearing the bearings. The resultant vibration velocity is measured as a time series. The time recorded during the experiment is changed by suitable transformation to the corresponding useful life under normal operating conditions of the machines. This study establishes a relation between the vibration velocity and the projected useful life period of the bearings in the system for different possible system loads. From the value of a maintenance initiation point of vibration velocity for a possible system load, the corresponding life of the bearings in the system can be calculated.

By considering possible system loads, this study establishes a general expression between the available maintenance initiation time for CBM and the vibration velocity for the system under consideration. In the expression, the vibration velocity considered is less than the defined value of maintenance initiation vibration velocity.

The remainder of this study is organized as follows. Section 2 provides a detailed review of the extant literature. Section 3 explains the experimental setup of a Class I machine and the loading arrangement for the same. The experimental procedure of the accelerated life testing (ALT) and vibration measurement is also explained in this section. Section 4 details the experimental data and failure analysis; also, the system availability is modeled in terms of vibration velocity (between alert and alarm limits), and the resulting curves are plotted. The equations for the available maintenance initiation time and the value of maintenance initiation points of the system are established in this section. Finally, Section 5 concludes this study by summarizing the findings.

2. Literature Review and Background Study

The term “reliability” can be defined as the probability that under the stated operating conditions, a system or equipment will perform its intended function satisfactorily for a specified interval [2]. The general expression for reliability with time period t and failure rate λ [2] is given by:

$$R(t) = e^{-\lambda t} \tag{1}$$

where $\lambda = 1/MTBF$. *MTBF* refers to the mean time between failures. From Eq. (1), it can be inferred that as system failure resistance decreases, its reliability decreases.

The term "availability" is used to indicate the probability of a system or equipment being in operating condition at any time *t*, given that it is in operating condition at $t = 0$ [2]. To make a system in operating condition at time period *t*, it must be functional. In other words, it must not fail, and if there is a failure at *t*, it must be repaired. Thus, availability involves the characteristics of both reliability and maintainability. The latter is defined as the probability of repairing a failed system or component in a specified period [2]. Availability features allow the system to stay operational even when faults occur, whereas reliability means it is likely to perform perfectly, and maintainability implies that even if something does go wrong, it can be rectified effortlessly [2-3]. The general expression for the system availability *A* is given as Eq. (2) [2]:

$$A = \frac{MTBF}{MTBF + MTTR} = \frac{1}{1 + \lambda \times MTTR} \tag{2}$$

where *MTTR* refers to the mean time to repair.

Vibration monitoring might be considered the grandfather of CBM/PdM and provides the foundation for the CBM programs of most facilities [3, 8-9]. Bianchini et al. [10] conducted an experimental study for CBM through vibration monitoring on submersible well pumps following vibration severity standards such as ISO 10816-7 (2009). However, in their study, the availability of pumps corresponding to the vibration level was not set. Sulaiman et al. [11] investigated the effect of high vibration (above acceptable limit) in gas turbines installed in Al Ghubra Power & Desalination Company and suggested using online condition monitoring methods to determine the running condition of gas turbines in advance to plan maintenance activities to avoid sudden failure of the turbine.

In industrial situations, the process plants often operate with certain machines, which can be classified as Class I, II, III, and IV on the basis of vibration severity standards ISO 10816 and whose failure is mainly due to severe vibrations. The standard provides a reference for evaluating vibration severity in machines operating over the range of 600 to 12,000 RPM [8-9]. These machines are often subjected to more than the acceptable limits of vibration, which invariably leads to an increase in their failure rate, especially in the case of components with rotating parts. Rotating machinery is extensively used in today's industries, and some of these are extremely critical for the successful operation of the plant. The machine collapse may result in costly downtime of the plant. The faults in rotating machinery, such as machine being out of balance or alignment, gear fault, resonance, bent shafts, bearing failures, mechanical looseness, can be identified by measuring and analyzing the vibration generated by the machine [8-9].

The rolling element bearings in rotating machinery allow a relative movement and bear shaft load [12]. The life of a bearing depends on its use, and its expected life is solely based on experience [13]. The main cause of failure of industrial bearing is rolling contact fatigue (RCF). The RCF wear mechanism involves false brinelling, characterized by plastically formed indentations, which are generally caused by vibration due to overload. The cause of the wear is that lubricant is squeezed out between the contact area of rolling elements and raceways, resulting in direct metal-to-metal contact. Vibration causes wear of the surfaces in contact, and fine abrasive particles are rapidly produced, which results in a characteristic groove with the oxide acting as an abrasive [14].

Failure analysis of the bearing can be investigated by making artificial defects on various elements of the bearing and analyzing it with a vibration signature tool for monitoring its condition [15]. Usually, life testing under normal operating conditions of mechanical parts with high reliability is expensive in terms of both capital and time. Hence, it is desirable to accelerate the testing procedure for gathering the failure data. The chief objective of accelerated testing procedures is to reduce the time required for life testing by strategies such as intensified stress levels.

Physical acceleration or true acceleration means that operating a unit at high stress produces the same failures that would occur at typical use stresses, except that they happen much quicker. Then, by extrapolating the results suitably to the "normal use" conditions, a reasonably accurate estimate of the life of the component under the "normal use" conditions can be obtained [2, 16]. However, the issue of prediction accuracy associated with extrapolating data outside the range of testing has not yet been addressed comprehensively [16]. Acceleration factors show how the time-to-fail at a particular operating stress level (for one failure mode or mechanism) can be used to predict the equivalent time-to-fail at a different operating stress level [17-18].

Thus, an efficient diagnosis system is needed to predict the condition and consistent lead time of the machine. Vibration analysis is a method used for monitoring the condition of the machine. Effective vibration signal extracting techniques, therefore, have a critical part in diagnosing a rotating machine. In detecting early fault generation, frequency domain features in the vibration signals are generally more effective as compared to the time domain features [19]. Tools such as neural networks, hybrid systems, and fuzzy logic are being employed for increasing the effectiveness of fault diagnosis [20].

It is normally accepted that the vibration velocity recorded over the range of 10 Hz (600 RPM) to 1000 Hz (60,000 RPM) provides the finest indication of a vibration's severity on rotating machines [8-9, 11]. As most of the rotating machines operate between 600 RPM and 60,000 RPM, the vibration velocity is the best candidate for vibration measurement and analysis [8, 11], whereas above 60,000 RPM vibration acceleration is only the fine indicator [8, 11]. The RMS value is directly related to the energy content of the vibration and thus its destructive capability [8-9].

The indicators used for the initiation of CBM involve vibration signatures, temperature changes, and process parameters of machining such as spindle speed, depth of cut, and feed rate. By the analysis of these indicators, the failure of the machine can be predicted, and the corrective measures must be performed [21]. The process parameter used in the proposed model is vibration velocity (RMS), and its influence on system availability was analyzed.

The remaining useful life (RUL) is the life period left on any system of machines at a particular time of operation [3]. From a specific time of operation of the system, the RUL is the time period up to its functional failure. By taking RUL into account, a plant engineer can schedule system maintenance, which can avoid unexpected system downtime. Because of this, the prediction of RUL has the highest priority in CBM programs. The method used to calculate RUL depends on the kind of data available [3]. Coppe et al. [22] proposed the use of a simple crack growth model is proposed to predict the system RUL influenced by the fatigue failure mechanism. Kang et al. [23] implemented an ML-based approach for automating the prediction of RUL of equipment in continuous production lines. Han et al. [24] developed a method for RUL prediction for manufacturing systems using a mission reliability-oriented approach based on the functional dependence of components.

In this study, the alert limit of vibration, alarm limit of vibration, and the system availability are considered for defining the maintenance initiation points of CBM for Class I machines of specific power rating. A relation is established between the system availability and the process parameter vibration velocity (RMS). It is observed that the system availability starts to decline at a point much before the alarm limit of vibration (potential failure), and the maintenance is initiated at this point for the proposed model. From this point onward, the reliability begins to drop rapidly. The time available for maintenance initiation can be predicted from a given level of vibration, which is less than the defined value of maintenance initiation vibration level. The functional failure occurs beyond the potential failure and is not considered in this study [3].

It is evident that even though the relation between vibration velocity and system availability can be deduced, no study has predicted the time available to initiate the CBM from a given level of vibration of the machine [4, 25]. In this study, an attempt is made to ascertain the time available for initiating the CBM by having a measure of the vibration level.

3. Experimental Setup and Design of ALT

According to the vibration severity standards table for ISO 10816 [8-9], Class I industrial machines are individual parts of engines and machines, integrally connected with the complete machine in its normal operating condition (electrical motors of up to 20 HP are typical examples of machines in this category). According to ISO 10816, the speed of rotation of Class I machines lies in the range of 600 to 12,000 RPM.

On the one hand, the upper limit of vibration for a good operating condition, defined as the good operating limit, is 0.71 mm/sec, and for a satisfactory condition, defined as an acceptable or alert limit (failure initiation), is 1.8 mm/sec (Table 1). On the other hand, the corresponding value for the unsatisfactory condition of vibration level, defined as the alarm limit (potential failure), is 4.5 mm/sec. Above the alarm limit, potential failure leads to a functional failure, and some functions of the asset stop working [3]. Therefore, the failure probability of machine components subjected to vibration can be defined such that the failure probability is zero for vibration levels below the alert limit and 100% as it reaches the alarm limit.

Three-phase squirrel-cage induction motors are widely used as industrial drives because they are self-starting, reliable, and economical [8]. Around 70% of the failures that occur in electric motors are mechanical in nature [26]. The bearing failure is the leading cause (~51%) of most mechanical failures [27]. The causes of bearing failure include excessive loads, overheating, true and false brinelling, spalling, contamination, lubricant failure, loose and tight fits, corrosion, and misalignment [9]. The contamination is caused by foreign substances getting into bearing lubricants or cleaning solutions. Examples of such foreign substances include dirt, abrasive grit, dust, and steel chips from contaminated work areas. Solid particle contamination is a serious problem in all industrial sectors, which causes wear in contact surfaces [28].

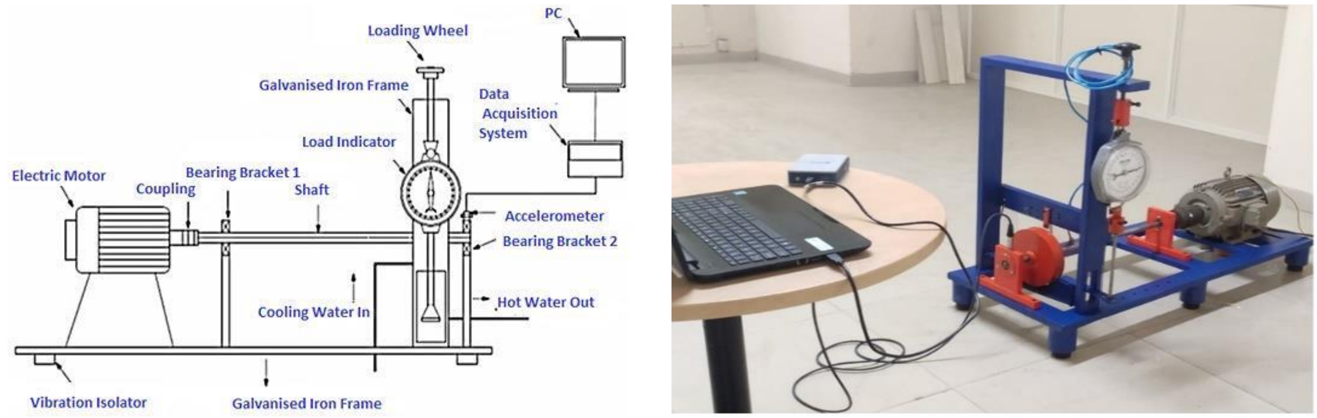
The wear happens in the inner and outer races of the bearing, and when the ball passes over these races, the phenomenon called “ringing” occurs, which is similar to the vibrations in a car moving on an irregular road. As wear increases, vibration increases. The vibration standards ISO 10816 can be used as a reference for evaluating the severity of vibrations. The proposed model creates the wear in the raceways of the bearing of a Class I machine intentionally, and the resultant vibrations above the alert limit can be measured and recorded using vibration measurement systems.

Figs. 1(a)-(b) indicate the line diagram and the photograph of the practical setup used for experimentation. It consists of a delta-connected three-phase induction motor, a long shaft with necessary detachable couplings at the ends, bearings and brackets, and a mechanical loading mechanism with cooling accessories. The rated voltage, current, power, RPM, and frequency of the motor used are 415 V, 7.1 A, 5 HP, 1440 RPM, and 50 Hz, respectively. Similarly, the shaft material used is EN32 grade steel with a diameter of 18 mm and a total length of 75 mm. The shaft is connected to the motor using a flexible element jaw coupling, and a spider rubber bush is placed on the inside of this coupling. The shaft is simply supported at the two bearings, which are placed inside two bearing brackets. The distance between the bearing ends is 65 cm. A brake drum is mounted on the shaft at a distance of 56 cm from the bearing bracket 1.

The bearings used are deep groove ball bearing SKF 6202 Z. The inside diameter of the bearing is 15 mm, and accordingly, the diameter of the shaft when it passes through the bearing portion is reduced to suit this value. As per the data sheet of SKF 6202 Z bearing, the fatigue load limit (maximum radial load) is 0.16 kN. For analysis purposes, the bearings fitted inside the bearing brackets 1 and 2 are referred to as bearing 1 and bearing 2, respectively. During experiments, the failure of bearing 2 is created by developing the wear in it by adding C10 coarse (10-micron grain size) silicon carbide paste between its inner and outer races.

Table 1 ISO 10816 vibration severity levels for Class I machines

Operating condition of Class I machines	Vibration velocity in mm/sec (RMS value)
Good	0.28 to 0.71
Satisfactory (acceptable)	1.12 to 1.80
Unsatisfactory (monitored closely)	2.80 to 4.50
Unacceptable	7.10 to 45.90



(a) Line diagram of the experimental setup (b) Practical setup

Fig. 1 Experimental setup for vibration measurement in Class I machines

The shaft design is based on the combined torque and bending moment (static analysis). On the basis of the design, the appropriate shaft diameter is calculated as 18 mm. A modal analysis of the combined system involving the shaft and the brake drum is performed using ANSYS software. The analysis revealed “n” modes of vibration and corresponding natural frequencies. The natural frequencies of the system obtained for the first and second modes are 166 Hz (9,960 RPM) and 332 Hz (19,920 RPM), respectively. The natural frequencies are observed to have an increasing trend for further modes of vibration. In this study, a variable frequency drive (VFD) is used between the motor and input three-phase supply to fix the motor speed at 1,465 RPM. Also, the ratio of the motor’s applied voltage and applied frequency is set as 8.3 V/Hz. As the natural frequency values obtained for all the modes of vibration for the designed system are noted to be much higher than 1,465 RPM, the resonance condition is avoided to make the design safe.

The loading arrangement used is illustrated in Fig. 2 and consists of a brake drum, brake shoe, and loading wheel. The material of the brake shoe is compressed asbestos, and that of the brake drum is EN8 grade steel. The brake drum is connected to the shaft at a distance of 56 cm from the bearing bracket 1. The brake drum has a diameter of 150 mm and a thickness of 40 mm. The loading of the motor is achieved by turning the loading wheel, which makes the brake shoe inside the brake shoe bracket to press against or pull away from the rotating hollow brake drum. The system load applied is noted from the load indicator dial. The hinge, brake shoe bracket, and bearing bracket are made of mild steel. The electric motor, bearing brackets, brake shoe bracket, load indicator, hinge, and cooling water arrangement of the brake drum are mounted on a frame made using a light gauge rectangular hollow cross section galvanized iron tube (IS1239). Rubber bush dampers are used for vibration isolation from the floor.

During loading, because of the friction between brake drum and brake shoe, heat is generated inside the hollow brake drum. It is removed by constantly circulating cooling water through the hollow brake drum. The water flow is controlled by a ball valve. The temperature of water at the inlet and exit are measured using K-type thermocouples, and the steady readings obtained are 27℃ and 63℃, respectively.

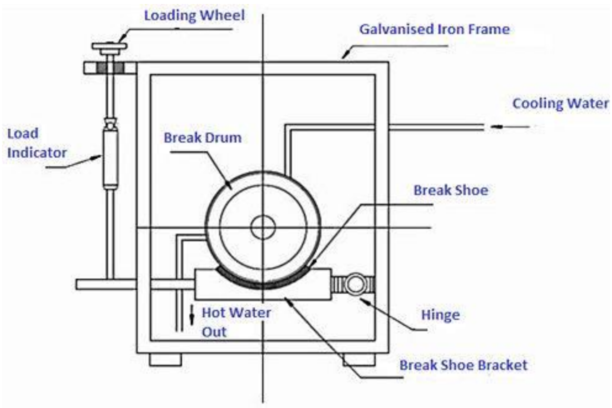


Fig. 2 Line diagram for loading arrangements

The ALT conducted in this experiment follows the procedure detailed in Regattieri's work [29]. The experimental setup is designed such that the failure of bearing 2 is the reason for the failure of the system. Experimental trials are conducted by continuous measurement of the RMS vibration values of the system from the moment the silicon carbide paste is added to the final failure state of the system. During each trial, under a constant radial load, the failure of the system is created by developing the wear in bearing 2. The failure so created is assessed in terms of the induced vibrations of the order of the magnitude above alarm limit vibrations (glut vibrations) in the system by comparing with the vibration severity standards (ISO 10816).

The trials are repeated by changing the radial load on bearing 2 from 14 kg to 18 kg in steps of 1 kg. Eight sets of experiments are conducted under each trial to ensure repeatability. These radial loads are selected such that the fatigue limit of the bearing is never exceeded. The loads on bearing 2 are calculated considering equilibrium conditions of the shaft and assuming a simply supported configuration at bearings 1 and 2.

During the experiments, after the motor starts, the system steadies itself in a few seconds and begins to operate in a stable condition. The vibration level is measured to ensure that there is no vibration interference, such as the vibration of the mounting of the load indicator in the system. Thereafter, 10 g silicon carbide paste is added in bearing 2 between its inner and outer races from the outside to develop the wear in it. The paste is added thrice at the intervals of 5 minutes for every set of the experiment. This results in the failure of the system as observed from the recorded RMS vibration values (Table 1) when the alarm limit is reached.

The system used in the measurement, preprocessing, storage, and postprocessing of the vibration signals consists of an accelerometer sensor, CompactDAQ, a personal computer (PC), and the LabVIEW software (version 2017). CompactDAQ, a data acquisition platform built by National Instruments, includes a broad set of compatible hardware and software; it is a USB-powered plug-and-play type and requires no data card. CompactDAQ integrates hardware for data I/O with LabVIEW software to enable engineers to collect, process, and analyze the sensor vibration data. The accelerometer sensor used is a uniaxial integrated electronic piezoelectric (IEPE; made by PCB Piezotronics). The setup, used to sense the vibrations in a radial direction perpendicular to the axis of the rotating shaft, is mounted magnetically above the bearing bracket 2, where the maximum level of the vibration is obtained. The sensor has the sensitivity of 100 mV/g, measurement range ±50g, resonant frequency 25 kHz, and frequency range 0.5 Hz to 10 kHz.

CompactDAQ is used to acquire and process the analog signal coming from the uniaxial accelerometer sensor. The signal conditioner, which is inbuilt in the data acquisition (DAQ) module, removes noise from the signal and supplies a constant current excitation of 2.1 mA to the accelerometer sensor. CompactDAQ connects to the accelerometer sensor via a wired I/O DAQ module. A shielded twisted pair (STP) cable is used to carry the analog signal from the accelerometer sensor to the DAQ module connected to the DAQ chassis, which, in turn, connects a PC where LabVIEW software is installed (Fig. 3). The function of the DAQ chassis includes the synchronization and transfer of digital signals from the DAQ module into the computer. The synchronization involves determining or enforcing and ordering events on signals. The digital signals from the DAQ chassis can be stored as binary values in the computer. A military standard connector is used to connect the accelerometer sensor to the STP cable. The signals from the DAQ chassis are transmitted through a shielded USB cable to the computer.

In this experiment, the LabVIEW software band-pass filter passes the frequencies of vibration data in the range of 200 Hz to 25 kHz. This software is used to convert the measured value of instantaneous vibration accelerations into instantaneous vibration velocities by integration, and from these instantaneous values, RMS values of vibration velocities are calculated. The RMS value of 51.2k samples of vibration velocity is calculated at each second.

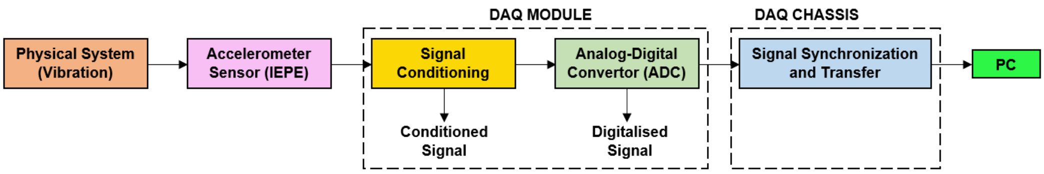


Fig. 3 Block diagram for vibration measurement setup in the proposed system

The DAQ module used in this experiment is NI 9234 with NI 9171. NI 9171 is the USB interface for powering NI 9234. The DAQ module NI 9234 has the maximum sampling rate of 51.2 kilo samples/sec(f_s) with an inbuilt hardware-based anti-aliasing filter (low pass) with a cut-off frequency of 0.45 × f_s (= 23.04 kilo samples/sec). The specifications of the DAQ module are enumerated as follows:

(1) Four analog input channels

(2) ±5V input range (from accelerometer sensor to DAQ module)

(3) Simultaneous sampling

(4) AC/DC coupling

(5) Operating temperature: -40℃ to 70℃

(6) Maximum allowed vibration and shock are 5 g and 50 g, respectively.

(7) Analog-to-digital converter (ADC) amplitude resolution: 24-bit

4. Failure Analysis, Discussion, and Results

The following assumption is made during failure analysis:

(1) Failure and repair rates for each subsystem are constant and statistically independent [1].

(2) Failure of machines with moving parts is caused by vibrations alone and occurs when the vibration velocity is above the acceptable or alert limit of vibration.

While loading, a lot of heat is created inside the hollow brake drum because of the friction between brake drum and brake shoe. This heat is removed using a cooling water arrangement. The cooling water from the tank enters inside the hollow brake drum through a PVC pipe, and its flow is controlled by a ball valve. The centripetal force produced by the rotation of the brake drum pushes the hot water inside the drum to exit through a copper tube. The temperature of hot water coming from the copper tube is measured as 63℃.

(3) The parameter for measuring mechanical vibration is taken as a velocity of vibration. The velocity is the best representation of true energy generated by a machine when the relative or bearing cap data are used [9].

(4) Availability under consideration is inherent availability or steady-state availability.

The steady-state availability represents the long-term availability after the system settles. At the early stages of operation or before the system settles, it may be wobbly because of the issues in (i) employee training, (ii) determining a genuine/good spare part keeping policy, (iii) determining the number of maintenance personnel, and (iv) optimizing the effectiveness/efficiency of maintenance.

(5) The life of the bearing is assumed to be 12,000 hours and is taken as the reference life in the analysis [13]. This reference life corresponds to the maximum radial load acting on the bearings used in the experiment.

(6) The transformations used are linear, i.e., the time-to-fail at high stress is multiplied by a constant (the acceleration factor) to obtain the equivalent time-to-fail at use stress [17-18].

(7) The bearing failure (potential) probability is 0% below the alert limit of vibration, and it is 100% at and above the alarm limit.

According to vibration severity standards ISO 10816 shown in Table 1, the satisfactory/acceptable vibration level in Class I machines is in the range 1.12 to 1.80 mm/sec (RMS). The unsatisfactory level (monitor closely) of vibration is in the range 2.80 to 4.5 mm/sec (RMS). The vibration level 1.80 mm/sec (RMS) can be defined as alert limit or failure initiation stage as per

P-F curve for Class I machines [3]. The vibration level 4.5 mm/sec (RMS) can be defined as alarm limit or potential failure stage as per P-F curve. Therefore, the system failure probability is zero below the alert limit, and 100% potential failure happens at and above the alarm limit.

In this study, bearing 2 is subjected to ALT by causing the wear in it by adding some silicon carbide paste. The bearing fails because of the excessive wear so created. As the wear in the bearing increases, the vibration in the system also increases. During the experiment, the RMS value of the vibration velocity for a particular system load is recorded after the silicon carbide paste is added. The measurement of RMS vibration velocity continues until the alarm limit of vibration is reached. The availability and reliability of the system are calculated by extrapolating the time corresponding to this recorded RMS value of the vibration velocity to normal-use conditions. The rated life of a ball bearing in working hours can be expressed as [13-14]:

$$L_{10h} = \left(\frac{C}{P}\right)^3 \times \frac{10^6}{60 \times n} \tag{3}$$

where L_{10h} = rated bearing life in hr; C = dynamic load capacity (N; constant for identical bearings); P = load applied on the system (N); n = speed of rotation of the motor (RPM).

In the industrial application of bearings, the speed of rotation is relatively constant, and the desired life is expressed in terms of hours of service. In this study, the bearing speed is kept constant at 1,465 RPM. The expected life for bearings in industrial applications is in the range of 12,000 to 20,000 hr [13]. If two groups of identical ball bearings are tested under loads P_1 and P_2 for respective lives L_1 and L_2, then

$$\frac{L_1}{L_2} = \left(\frac{P_2}{P_1}\right)^3 \tag{4}$$

The data sheet for the SKF 6202 Z bearing (bearing 2) mentions that the maximum radial load that can be applied on it is 160 N. In the experimental setup, the shaft is simply supported at the two bearing ends. When the radial load applied on the system is 18 kg (176.58 N), by using equilibrium equations for the shaft configuration, the load acting on bearing 2 is found to be 15.51 kg (152 N). When L_1 = 12,000 hr and P_1 = 160 N, the life corresponding to the load of P_2 = 15.51 kg (152 N) on bearing 2 is calculated as L_2 = 13996.21 hr. Similarly, the life of the ball bearing for different system radial loads (m kg) can be calculated and is encapsulated in Table 2.

Table 2 Life of bearing calculated for different system loads

Radial load applied on the system (m kg)	Radial load on bearing 2, whose failure due to excess vibration is considered (kg)	Life of bearing 2 based on the reference life of 12,000 hr (hr)
14	12.06	29,771.38
15	12.92	24,212.20
16	13.78	19,959.70
17	14.65	16,609.25
18	15.51	13,996.21

4.1. Experimental data and failure analysis

Eight trials of the experiment are performed for a particular system load to ensure repeatability. Each trial involved the time series vibration level measurement up to the alarm limit. Thus, a total of 40 trials are conducted for the system loads varying from 14 to 18 kg. By performing the error analysis for the data collected, inappropriate trial measurement is removed.

Furthermore, a total of 40 bearings fail during the experiment. The silicon carbide paste is added between the inner and outer races of the bearing to create the wear. The common bearing defect is the indentations in the raceways, and when the ball passes over the raceways, mechanical vibration is induced. Table 3 summarizes the measured failure data of vibration velocity (between alert to alarm limits), the time between failures, and the projected useful life period for bearing 2 under the 14 kg system load obtained for each of the six different trials.

The estimated projected time for the bearing for its useful life period is calculated following assumption 6 for the analysis. For an applied system load of *m* kg, the useful life period of the bearing in the system *L* hr (during its normal operating conditions) can be calculated from the observed failure time of *t* s (during experimentation) as given by Eq. (5) [17-18]:

$$AF^{m} \times t = L \times 3600 \tag{5}$$

where AF^m (the acceleration factor for system load *m* kg) is the proportionality constant. For a system load of 14 kg, the value of the acceleration factor used in the analysis is AF^{14} = 34,786.423. The experiment is repeated for the system loads of 15 kg, 16 kg, 17 kg, and 18 kg in a similar way, and the values are calculated.

Using curve-fitting techniques, the relation between the vibration velocity and the projected time for the system loads from 14 kg to 18 kg is found to be parabolic in nature with goodness of fit, $R^2 \geq 0.95$. The relationship can be written as:

$$V = at_p^2 + b \times t_p + c \tag{6}$$

where *V* represents the vibration velocity recorded in mm/hr between alert to alarm limits for the system load (*m* kg); t_p represents the projected time calculated in hrs using the acceleration factor (AF^m) corresponding to the vibration velocity (*V*). The system load (*m* kg), coefficients (*a*, *b*, and *c*) in Eq. (6), goodness of fit (R^2 value) for Eq. (6), and acceleration factor (AF^m) obtained from this study are summarized in Table 4.

Using curve-fitting techniques, the relation between the vibration velocity *V* (mm/hr) and the average value of projected time for system loads from 14 kg to 18 kg, t_{pa} (hr), is found to be parabolic in nature with the R^2 value 0.9986. The relationship can be expressed as:

$$V = 0.00003 \times t_{pa}^2 + 0.0583 \times t_{pa} + 2169.7, \quad for \;\; 6480\, mm/hr \leq V \leq 16200\, mm/hr \tag{7}$$

For any given vibration velocity, the corresponding projected time can be obtained by solving Eq. (7).

The values of vibration velocity in mm/hr and the corresponding values of MTBF and failure rate for the 14 kg system load are encapsulated in Table 5. The MTBF (12,998.2 hr) corresponding to vibration velocity 6,480 mm/hr (1.8 mm/sec) is obtained by deducting the value of average useful life period of the bearing, 16,773.18 hr (corresponding to 6,480 mm/hr) from the useful average life 29,771.38 (corresponding to 16,200 mm/hr). Similarly, the MTBF (11,492.4 hr) corresponding to vibration velocity 7,200 mm/hr is obtained by deducting the value of average useful life period of the bearing, 18,278.98 hr (corresponding to 7,200 mm/hr) from the useful average life 29,771.38 (corresponding to 16,200 mm/hr). Likewise, other values from 10,255.55 hr to 0.00 hr are obtained.

In accordance with Table 5, the variation of failure rate (per hr) of the experimental setup of Class I machine with vibration velocity (mm/hr) for the system load of 14 kg, can be drawn as shown in Fig. 4. The system availability can be calculated using the Eq. (2). The repair corresponding to bearing failure involves (a) changing the bearing and (b) (occasionally) changing the shaft. The MTTR is calculated as 47 min (0.783 hr).

Table 3 The measured failure data and the projected useful life period for bearing 2 under the 14 kg system load

Vibration velocity (mm/hr)	Time (s)							Projected time for bearing for its useful life period (hr)
	Trial 1	Trial 2	Trial 3	Trial 4	Trial 5	Trial 6	Average value	
6,480	1,902	1,734	1,700	1,692	1,694	1,693	1,735.83	16,773.18
7,200	1,988	1,914	1,870	1,858	1,860	1,860	1,891.67	18,278.98
7,920	2,074	2,026	2,008	2,002	2,004	2,004	2,019.67	19,515.83
8,640	2,200	2,176	2,164	2,162	2,162	2,162	2,171.00	20,978.15
9,360	2,268	2,270	2,270	2,270	2,269	2,270	2,269.50	21,929.94
10,080	2,374	2,374	2,374	2,374	2,374	2,373	2,373.83	22,938.10
10,800	2,438	2,448	2,454	2,454	2454	2,454	2,450.33	23,677.31
11,520	2,496	2,520	2,526	2,528	2,528	2,528	2,521.00	24,360.16
12,240	2,570	2,614	2,622	2,624	2,623	2,622	2,612.50	25,244.31
12,960	2,714	2,733	2,738	2,740	2,739	2,738	2,733.67	26,415.13
13,680	2,806	2,838	2,846	2,848	2,848	2,847	2,838.83	27,431.35
14,400	2,896	2,934	2,944	2,946	2,946	2,945	2,935.17	28,362.21
14,760	2,936	2,969	2,980	2,982	2,982	2,982	2,971.83	28,716.51
15,120	2,968	3,004	3,014	3,016	3,015	3,016	3,005.50	29,041.83
15,480	2,998	3,034	3,042	3,044	3,043	3,044	3,034.17	29,318.83
15,840	3,028	3,060	3,068	3,069	3,068	3,069	3,060.33	29,571.68
16,200	3,060	3,086	3,091	3,075	3,090	3,084	3,081.00	29,771.38

Table 4 The system load, coefficients, goodness of fit, and acceleration factor obtained from this study

System load (m kg)	Coefficients			Goodness of fit (R^2 value)	Acceleration factor (AF^m)
	a	b	c		
14	$+1.7491 \times 10^{-5}$	−0.0803	+2828.30	0.997	34,786.423
15	$+2.2053 \times 10^{-5}$	−0.0313	+3975.10	0.999	32,723.243
16	-1.4072×10^{-5}	+1.4825	−7885.70	0.987	32,694.021
17	$+2.2995 \times 10^{-5}$	+0.5762	−294.48	0.989	33,196.922
18	$+2.4260 \times 10^{-4}$	−3.7696	+21356.00	0.997	37,301.122

Table 5 The vibration velocity, MTBF, and failure rate of the system for the 14 kg system load

Vibration velocity (mm/hr)	MTBF (hr)	Failure rate (per hr)
6,480	12,998.20	0.000076934
7,200	11,492.40	0.000087014
7,920	10,255.55	0.000097508
8,640	8,793.23	0.000113724
9,360	7,841.44	0.000127528
10,080	6,833.28	0.000146343
10,800	6,094.07	0.000164094
11,520	5,411.22	0.000184801
12,240	4,527.07	0.000220894
12,960	3,356.25	0.000297952
13,680	2,340.03	0.000427345
14,400	1,409.17	0.000709637
14,760	1,054.87	0.000947988
15,120	729.55	0.001370711
15,480	452.55	0.002209724
15,840	199.70	0.005007524
16,200	0.00	∞

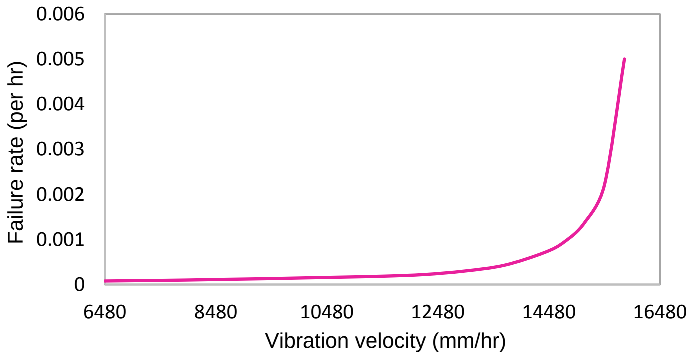


Fig. 4 Variation of failure rate (per hr) with vibration velocity (mm/hr) for the 14 kg system load

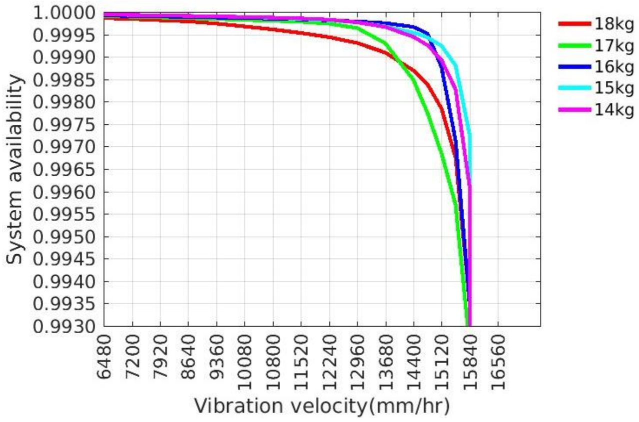


Fig. 5 Variation of system availability with vibration velocity for different system loads

Fig. 5 shows the variation of system availability with vibration velocity (mm/hr) for different system loads from 14 kg to 18 kg. It is found that there is a slight decrease in availability between alert and alarm limit of vibration, and at the latter, there is a sudden decrease in availability to a value of zero for all system loads. It can be noted that at the maintenance initiation point, the system availability begins to decline. This is because, from this point onward, the reliability starts to decrease rapidly.

The value of the maintenance initiation point for possible system loads is obtained as 12,176 mm/hr (3.4 mm/s). The value 3.4 mm/s is considered to be the maintenance initiation point for a system of Class I machines. By considering all the possible system loads, the relation between the time available for maintenance initiation T_a (hr) and the vibration velocity V (mm/hr) is given by:

$$T_a = 0.00008V^2 - 2.5944V + 20196\,, \quad for\ \ 6480\, mm/hr \le V \le 12176\, mm/hr \tag{8}$$

4.2. Error analysis

Keeping the system configuration and applied load unaltered, eight trials of the experiment are conducted, and the RMS vibration velocity values and the corresponding times are recorded. Using Grubbs' test at a significance level (α) = 0.02, the trials which involve outliers are detected and removed for a particular system load. The trials that are removed are two trials for the system loads of 14 kg, 15 kg, and 17 kg, and three trials for the system loads of 16 kg and 18 kg. The remaining trials of time (corresponding to the vibration velocity between alert and alarm limits) for the system loads of 14 kg, 15 kg, 16 kg, 17 kg, and 18 kg are subjected to error analysis to find the margin of error at 95% confidence interval.

The maximum value of percentage margin of error (at 95% confidence interval) in the average time is calculated as 3.5%, which corresponds to the average time of 1735.83 s (corresponding to the vibration velocity 1.8 mm/s) for the system load of 14 kg. The minimum value of the percentage margin of error in the average time is 0.013%, which corresponds to the average time of 2373.83 sec (corresponding to the vibration velocity 2.8 mm/sec) for the system load of 14 kg.

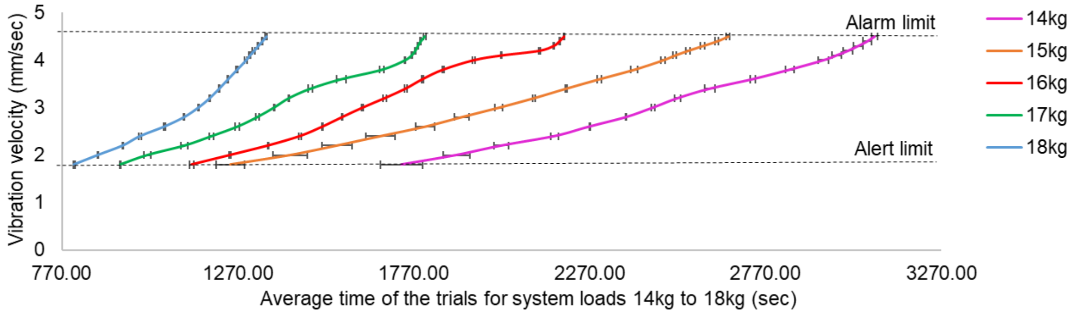


Fig. 6 Vibration velocity vs the average time corresponding to system loads (14 kg to18 kg) with error bars

Fig. 6 depicts the variation of vibration velocity between the alert and alarm limits against the average time of different trials (recorded in ALT of bearing) for system loads from 14 kg to 18 kg. Error bars (corresponding to the margin of error) in the average time are also shown in this figure.

5. Conclusions

In this experimental study, the failure of the machine was assessed by considering vibrations above the alert limit. Accelerated life tests were conducted by creating a severe environment of high stress in the bearing of a Class I machine system for experimentation. By a suitable external mechanism, the wear rate in the bearing was increased rapidly until failure occurred. The resulting vibration data were captured as a time series for analysis. The vibration was recorded as vibration velocity (RMS). Using a linear transformation, the time recorded was converted to the corresponding value in its normal use. A suitable loading mechanism was designed to apply possible magnitudes of system load. The major contributions of the proposed experimental study are summarized below.

(1) It is demonstrated that the CBM initiation point of vibration velocity can be accurately predicted by the analysis of the pattern of system availability with vibrations to ensure the long life of the process plant machines and to enhance the overall performance of the plant.

(2) The proposed model established a relationship between the vibration velocity and the projected useful life period of the bearing for the system of the electric motor with rated power 5 HP (Class I machine) applied with different possible system loads. For a given value of maintenance initiation point of vibration velocity and a possible system load, the corresponding life of the bearing in the system can be computed using the proposed relation to conduct the systematic maintenance of the machines effectively.

(3) In the proposed model, a general equation is developed between the time available for maintenance initiation and vibration velocity for the given system by considering different possible system loads.

(4) The experimental study revealed that in CBM, the vibration parameter can be used as a reliable characteristic to predict the maintenance initiation point.

Conflicts of Interest

The authors declare no conflict of interest.

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