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DEVISING METHODS TO AVOID FORMATION OF DEFECTS IN A BALL BEARING THROUGH FFT ANALYZER

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ABSTRACT

Bearings are the most essential parts in rotating machinery. The function of bearings is to permit constrained relative rotation or linear motion between two parts. During the operation, the bearings are often subjected to high speed and severe conditions. Under these severe operating conditions, defects are often developed in the bearing components. If no corrective measures are taken, the machine could halt or be seriously damaged. Each of the different causes of bearing failure produces its own characteristic damage. Such damage, known as primary damage, gives rise to secondary, failure-inducing damage - flaking and cracks. Even the primary damage may necessitate scrapping the bearings on account of excessive internal clearance, vibration, noise, and so on. A failed bearing frequently displays a combination of primary and secondary damage.

Keywords: *- spall defect, bearing, FFT analyser, vibration spectrum, BPFO, Rolling element.*

I. INTRODUCTION

This template, Rolling element bearings are the most essential parts in rotating machinery. Rolling element bearings are widely used in industry from home appliances to helicopter gear boxes. Proper functioning of these machine elements is extremely important in industry in order to prevent long term, costly catastrophic downtimes. It is obvious that more attention must be paid to the condition of a rolling element bearing if the human life is in question. Rolling element bearings are manufactured by assembling different components: The outer ring, the inner ring and the rolling elements which are in contact under heavy dynamic loads and relatively high speeds. When ball bearings are operated, they generate vibration. Even a geometrically perfect bearing may generate vibration due to varying compliance or time varying contact forces which exist between the various components of the bearing. The nature of vibration response changes with the presence of defect in bearing components. The function of bearings is to permit constrained relative rotation or linear motion between two parts. During the operation, the bearings are often subjected to high loading and severe conditions. Under this severe operating condition, defects are often developed on the bearing components. If no action is taken, the machine could be seriously damaged. Therefore, it is of prime importance to detect accurately the presence of faults, especially at their early stages, in bearings to prevent the subsequent damage and reduce the costly downtime. The vibration analysis is the most commonly used technique for monitoring of the bearings. This technique can provide early information about progressing malfunctions and forms base line for future monitoring purpose. The radial clearance in rolling bearing systems, required to compensate for dimensional changes associated with thermal expansion of the various parts during operation, cause dimensional attrition and comprise bearing life, if unloaded operation occurs and balls skid. Also it can cause jumps in the response to unbalance excitation. These undesirable effects may be eliminated by introducing two or more loops into one of the bearing races so that at least two points of the ring circumference provide a positive zero clearance. The deviation of the outer ring with two loops, known as ovality, is one of bearing distributed defects. Although this class of imperfections has received much work, none of the available studies has simulated the effect of outer ring ovality on the dynamic behaviour of rotating machinery under rotating unbalance with consideration of ball bearing nonlinearities, shaft elasticity, and speed of rotation. It is found that with ideal bearings (no ovality), the vibration spectrum is qualitatively and quantitatively the same in both the horizontal and vertical directions. PC, provides authors with most of the formatting specifications needed for preparing electronic versions of their papers. All standard paper components have been specified for three reasons: (1) ease of use when formatting individual papers, (2) automatic compliance to electronic requirements that facilitate the concurrent or later production of electronic products, and (3) conformity of style throughout a conference proceedings. Margins, column widths, line spacing, and type styles are built-in; examples of the type styles are provided throughout this document and are identified in italic type, within parentheses, following the example. Some components, such as multi-leveled equations, graphics, and tables are not prescribed, although the various table text styles are provided. The formatter will need to create these components, incorporating the applicable criteria that follow.

TYPES OF DEFECTS IN BEARING

Rolling bearing defects may be categorized as localized or distributed.

A] A localized defect includes cracks, pits and spalls in the rolling surfaces, as well as particle contamination of the bearing lubricant. The ultimate failure mode of a correctly installed and operated bearing generates defects of this sort (i.e., fatigue spalling of the rolling surfaces). Defects of this sort manifest themselves in the bearing's vibration signal as vibratory transients which result from discontinuities in the contact forces as the defect undergoes rolling (or, perhaps, sliding) contact. A number of techniques have been developed to detect these transients in the vibration signal.

B] Distributed defects involve the entire structure of the bearing. Here are included such conditions as misaligned races, eccentric races, off-size rolling elements and out of-round components. These defects may result from manufacturing error and abrasive wear. The major factors for causes of bearing failure are dirt, misalignment, insufficient lubrication, overloading, corrosion, foreign particle or shifted bearing cap. These sorts of defects often give rise to excessive contact forces which in turn result in premature surface fatigue and ultimate failure. The distributed defects are likely to increase the repetitive surface and subsurface stresses to which the bearing races are subjected and which eventually cause their fatigue failure. Therefore, a study of the vibratory response of bearing races due to distributed defects assumes importance. A technique for detecting these sorts of defects by vibration analysis is desirable primarily as an inspection tool to verify the proper operation of newly installed bearings.”.

II. PROBLEM DEFINITION

Bearings are the most essential parts in rotating machinery. The function of bearings is to permit constrained relative rotation or linear motion between two parts. During the operation, the bearings are often subjected to high loading and severe conditions. Under these severe operating conditions, defects are often developed in the bearing components. If no corrective measures are taken, the machine could halt or be seriously damaged. Therefore, it is of prime importance to detect accurately the presence of faults especially in early stages, to prevent the subsequent damage and reduce the downtime. In the past decades, health monitoring of critical machine bearings has attracted significant research efforts and vibration analysis, which has been extensively used in the fault detection and localization of the bearing. Being able to measure the vibration signal, the transducers or accelerometers are mounted as close as possible to the bearings that are being monitored. When we put across the comparison of signals collected from other sensors like acoustic emission or laser, the vibration signals from accelerometers have the advantage of providing a wide dynamic range and wide frequency range.

A bearing is composed of an inner ring, outer ring, rolling elements, and a cage. The inner and outer rings have raceways that form a path for rolling elements. The rolling elements rotate along a path between the inner raceway and outer raceway to provide minimal friction for rotational movement. The radii of rolling elements are slightly smaller than the track to allow the rolling elements to contact the raceways at a single point. The cage maintains an even and consistent spacing of rolling elements to guide them in the raceways while functioning.

The bearing defects may be categorized into two broad classes. The first class being denoted as ‘local defects’, which includes cracks, pits, spalls on the raceways or rolling elements. As mentioned above, local defects result in discontinuous contact forces that generate a specific signature in the vibration signal. The second class of bearing defect, denoted as ‘distributed defects’, which involves the structure/installation of bearing, such as misaligned races, eccentric races, off-size rolling elements, and out-of-round components. Distributed defects also generate specific signatures in vibration signals and increase the chance of 'local defects'.

Hence, the need arises to devise a suitable method with the use of FFT Analyzer for avoiding a local defect, which if occurs, could damage the machine component resulting into downtime.

III. BEARING COMPONENTS

All rolling bearings are composed of four basic parts: inner ring, outer ring, rolling elements, and cage or separator as seen in Figure 1. Some bearings have additional components; the guide ring and seals are used only in some special bearings.

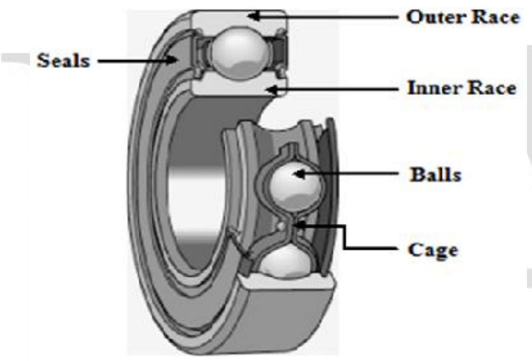

Fig. 1. Components of the rolling bearing

1.1 Inner Ring

The inner ring is mounted on the shaft of the machine and is in most cases the rotating part. The bore can be cylindrical or tapered. The raceways against which the rolling elements run have different forms such as spherical, cylindrical or tapered, depending on the type of rolling elements.

1.2 Outer Ring

The outer ring is mounted in the housing of the machine and in most cases it does not rotate. The raceways against which the rolling elements run have different forms depending on the type of rolling elements. The forms of the raceways may be spherical, cylindrical or tapered.

1.3 Cage

The cage separates the rolling elements to prevent metal-to-metal contact between them during operation that would cause poor lubrication conditions. With many bearing types the cage holds the bearing together during handling. Cages are made from cold rolled steel strip.

IV. TYPES OF BEARING FAILURE AND THEIR CAUSES

Rolling element bearings are among the most important and popular components in the vast majority of machines. Additionally, the component most likely to cause machine downtime is the bearing, because all machine forces are transmitted through the bearings. Therefore, rolling element bearings have been the subject of extensive research over the years to improve their reliability. However, since a large number of bearings are associated with any critical process, system failure due to any individual bearing failing can occur in a short period of time. There are many reasons for early failure, such as heavy loading, inadequate lubrication, careless handling, ineffective sealing, or insufficient internal bearing clearance due to tight fits. Each of these factors results in its own particular type of damage and leaves its own special imprint on the bearing.

Rolling bearing damage may result in a complete failure of the rolling bearing at least, however, in a reduction in operating efficiency of the bearing arrangement. Only if operating and environmental conditions as well as the details of the bearing arrangement (bearing surrounding parts, lubrication, sealing) are completely in tune, can the bearing arrangement operate efficiently. Bearing damage does not always originate from the bearing alone. Damage due to bearing defects in material or workmanship is exceptional.

The types of mechanical bearing failure and their frequencies are categorized in Table 5.1. The most frequent bearing failure category is corrosion, which is lubrication related. Chemical reaction occurs between the oil and the surface of the bearing, generally from water or other corrosive materials present in the oil. Dimensional discrepancies of rolling element bearings are a consequence of damage prior to or during service. The causes of dimensional discrepancies could be manufacturing flaws, improper handling or installation, and severe overloading during service. Foreign objects, carried by contaminated lubricant, are trapped inside the bearing between the rolling element and the raceway, and are overloaded. Understanding the underlying reason for the defects and their consequences in terms of failures gives the diagnostic clues to detect early failures.

Reason	Failure percent
Corrosion	35 %
Dimensional Discrepancies	29 %
Foreign Objects	24 %
Other	10 %
Fatigue	2 %

Vibration

When a bearing is exposed to vibration, the forces of inertia may be so great as to cause fatigue cracks to form in the cage material after a time. Sooner or later these cracks lead to cage fracture.

Excessive speed

If the bearing is run at speeds in excess of that for which the cage is designed, the cage is subjected to heavy forces of inertia that may lead to fractures. Frequently, where very high speeds are involved, it is possible to select bearings with cages of special design. To safe guard the occupants from road shocks.

Blockage

Fragments of flaked material or other hard particles may become wedged between the cage and a rolling element, preventing the latter from rotating round its own axis. This leads to cage failure.

Main factor for bearing failure

a) Dirt

b) Corrosion

c) Insufficient Lubrication

d) Overloading

e) Misalignment

f) Misassemble

g) Improper journal finish

V. EQUATIONS

FREQUENCY EQUATIONS REQUIRED

The vibration data collected is analyzed at particular defect frequency. The formulae required for calculation of same are given below. The all frequency values are dependent on the geometry and running speed of the bearing.

Let,

Rotational speed of inner race= W_i

Rotational speed of outer race = W_o

Rotational speed of rolling element or balls=Wm

Contact angle=β

Apart from Wi, Wo, Wm, and β, the relative motion between bearings subcomponents is depends on ball diameter ‘db’ and mean or pitch diameter ‘dm’.

1) Cage speed Wm and Cage frequency fm:

Consider inner race and outer race are rotating at Wi and Wo respectively.

Therefore, the velocity at a point on a rotating body is,

V=r*w (1)

Generally bearing vibration sensors such as accelerometers or proximity probes are pick up the radial component of vibrations. Therefore the equations are developed in terms of radial component of vibration.

The velocity at inner race contact point is,

Vi=Wi*ri

Vi=Wi*(di/2)

Therefore Vi= Wi*1/2(dm-db cos β)

Therefore vi= ½*wi*(dm-db cos β)

Therefore vi= ½*wi*dm[1-(db/dm) cos β (2)

Similarly,

Velocity at outer race contact point is,

Vo= ½*wo*dm[1+(db/dm) cos β] (3)

Put,(db/dm)cos β= µ (4)

Therefore equation 2 and 3 becomes,

vi= ½*wi*dm[1-µ] (5)

Vo= ½*wo*dm[1+µ] (6)

Now the velocity of rolling element or balls is taken as the mean of inner and outer race velocities

Therefore vm=[(vi+vo)/2]

Vm=1/2[1/2*wi*dm*(1-µ)+ ½*wo*dm(1+µ)]

Vm=1/4*dm*[wi* (1-µ)+ wo* (1+µ)] (7)

But vm= ½*(wm*dm) (8)

Put equation 8 in equation 7 we get,

Wm=1/2*[wi* (1-µ)+ wo* (1+µ)] (9)

Since rotational speed is directly proportional to frequency,

Therefore=f

Equation 9 becomes,

Fm=1/2*[[fi* (1-µ)+ fo* (1+µ)] (10)

2) Fundamental train frequency (FTF) or Cage defect frequency(fcd):

If the bearing is too lose or the cage is worn out or cracked, vibration amplitudes at Vm (FTF) and its harmonics may appear. If the defect is severe enough the harmonics of running speed (i.e)multiplies of fi) are modulated by FTF and side bands of FTF will appear in the spectrum.

Characteristic frequency (Hz)	symbol	Equations
Shaft Rotational Frequency	Fs	N/60
Inner race defect frequency	Fid	n/2*fr[1+(bd/pd)*cos β]
Outer race defect frequency	Fod	n/2*fr[1-(bd/pd)*cosβ]
Ball defect frequency	Fbd	Pd/2bd*Fr[1-$(bd/pd)^2*(\cos\beta)^2$]

Therefore

FTF=fm=1/2*fi*(1- µ) (11)

3) Ball pass frequency of outer race (BPFO) of outer race defect frequency(fod):

If there is significant defect on the stationary outer race then each rolling element (ball) produces on impact vibration. The frequency which is generated by this phenomenon is called as outer race defect frequency.

Therefore BPFO=n*FTF

BPFO=n*1/2*fi**(1- µ) (12)

This frequency is generated related to the cage motion relative to a fixed reference frame.

4) Ball pass frequency of inner race (BPFI) or inner race defect frequency (fid):

If there is a defect on the rotating inner raceway then a spike caused by the impacting of each rolling element as it contacts the defect. The frequency which is generated by this phenomenon is called inner race defect frequency.

Therefore BPFI=n*Ffixed

=n*(fi-fm)

=n*(fi-FTF)

=n*[fi-(1/2)*fi(1- µ)

Therefore,

BPFI=n/2*fi(1+ µ) (13)

This frequency is generalated related to the rotational speed of inner race and the cage assembly.

5) Ball spin frequency (BSF) or ball defect frequency(fbd):

A single defect on a rolling element generates a measurable vibrations at the relative spin frequency of the ball relative to the cage.

Fb=1/2*(dm/db)*(1-µ)(1+ µ)*fi

Therefore fb= 1/2*(dm/db)*(1-µ)(1+ µ)*fi

Fb =1/2*(dm/db)*fi*(1-µ2) (14)

Note: Healthy bearing in general do not exhibit significant amplitudes of vibration at the bearing frequencies. An experienced vibration analyst can track progression of working defects through changes in amplitudes and spectral content.

Where,

N:-rotational speed of shaft in RPM

n:-No. of balls,

Fr:-Shaft Rotation Frequency,

bd:-Ball Diameter,

β -Contact angle,

pd:-Pitch Diameter

Outer race defect		
S. No	RPM	FREQUENCY
1	1490	89.12
2	1200	71.56
3	1000	59.6
4	800	47.7

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Inner race defect		
S. No	RPM	FREQUENCY
1	1490	135.05
2	1200	108.3
3	1000	90.32
4	800	72.26

VI. EXPERIMENTAL WORK

EXPERIMENTAL SETUP:

The experiment is carried out on the setup which is already available at COLLEGE. The figure 6 shows the actual setup on which experimental work is carried out. The specifications of the setup are provided below. The setup is designed in order to have the negligible setup vibrations. Here the set-up vibrations term is very important. When the shaft starts running at high speed the all components start vibrating because of small mechanical errors in alignment. The setup vibrations MIX-UP with bearing vibrations and create the inaccuracy occurs in the measurement. The current setup has negligible such structural vibrations. The setup consists of a motor, two bearings, contact type tachometer, SKF FFT Analyzer, single channel accelerometer etc. The whole setup is mounted on a 20 mm thick metal plate with four acrylic pads.

Fig. 2. Actual experimental setup

Motor:-

Power available: - 2.20 kW, 1HP

Voltage Required: - 415+ 10% Volt

Speed: -1490 rpm

Frequency: - 50+ 5% Hz

FFT Analyzer:-

For getting the vibration signatures the SKF single-channel, Fast Fourier Transform Analyzer [FFT], is used. The accelerometer is used to sense the vibrations generated at the bearing surface. The accelerometer is a transducer used to sense the vibrations. The transducer having metallic head which is mounted on the surface of the bearing whose frequencies are to be measured. By using this accelerometer one can measure the frequencies radially and axially

A shaft supported by two bearings will be taken and it will be driven by an electric motor under unloaded conditions and rotating unbalance will be provided and

1. Vibration analysis of good (healthy) bearing will be carried out by suitable method.

2. Vibration analysis of defective bearings (with distributed defect) will be carried out by suitable method.

BEARING SELECTION

In this project work 6206 deep groove ball bearing is used. Geometry of bearing is shown in fig. and geometrical specification are given in table no. 4.1

The bearings are designated by a number. In general, the number consists of at least three digits. Additional digits or letters are used to indicate special features e.g. deep groove, filling notch etc. The last three digits give the series and the bore of the bearing. The last two digits from 04 onwards, when multiplied by 5, give the bore diameter in millimeters. The third from the last digit designates the series of the bearing. The most common ball bearings are available in four series as follows

The bearing selected is 6206 (2RS), the number “06” indicates the inner bore diameter 30mm, third digit from last i.e. “2” indicates Light Duty, and the first digit from start i.e. “6” indicates deep-groove ball bearing. The term

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"2RS" indicates the Rubber-Seal for the bearing on both sides .

CAD MODELOF SET-UP

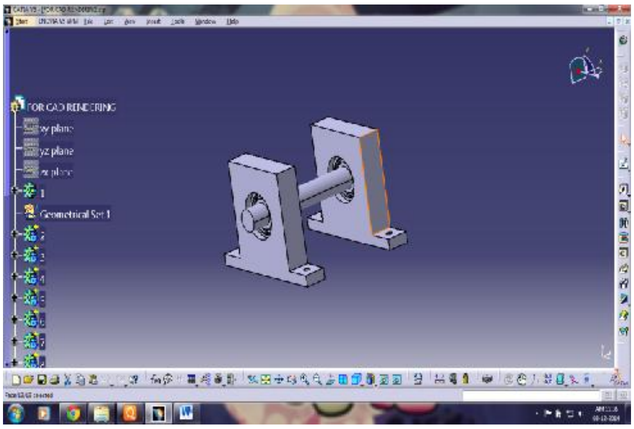

Fig. 3. CAD MODEL

VII. RESULTS AND DISCUSSION

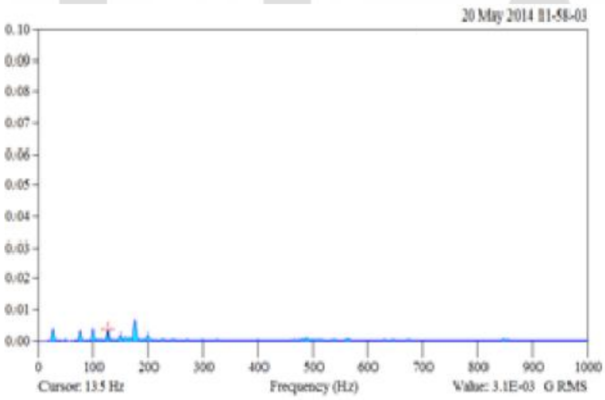

Figure 4. Vibration Signals of Good Bearing (Frequency in Hz vs. Amplitude in G RMS)

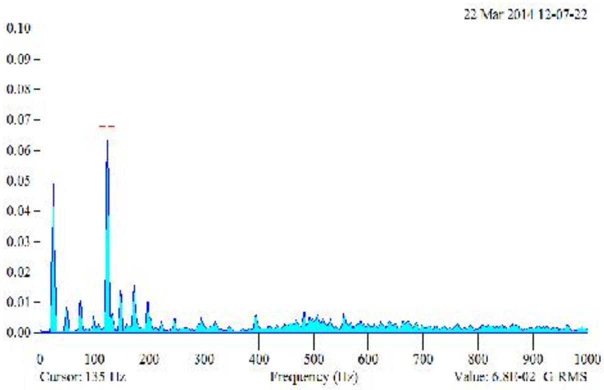

Figure 5. Vibration Signals of Bearing with Inner Race Defect (Frequency in Hz vs. Amplitude in G RMS)

Figure 4 shows that for good bearing the experimental amplitude value shown by cursor is 0.0031G RMS at frequency 135 Hz. In figure 5 the experimental amplitude value shown by cursor for bearing with inner race defect is 0.068 G RMS at same frequency 135 Hz which is more than that of good bearing and thus indicates that it is defective bearing.

If we compare this experimental frequency with the inner race defect frequency which is calculated theoretically by using equation (1) is at 135.05 Hz. It indicates that experimental frequency (135 Hz) is very near to theoretical inner race defect frequency (135.05 Hz).

Hence we can conclude that there is defect on inner race

Analysis of Bearing with Outer Race Defect

The experimental vibration signatures of good (healthy) bearing and defective bearings taken by using FFT analyser for outer race are shown in figure 6.

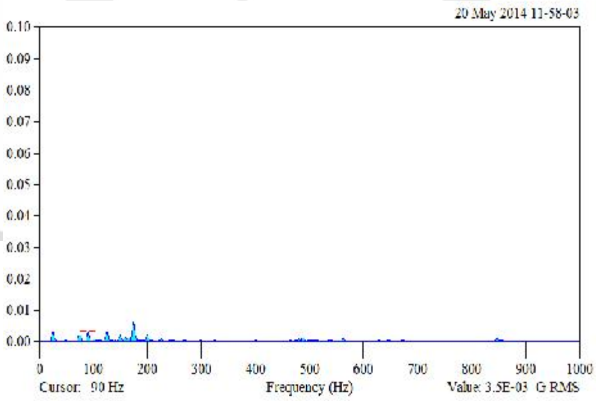

Figure 6. Vibration Signals of Good Bearing (Frequency in Hz vs. Amplitude in G RMS)

Figure 6 shows that for good bearing the experimental amplitude value shown by cursor is 0.0035G RMS at frequency 90 Hz. In figure 7 the experimental amplitude value shown by cursor for bearing with outer race defect is 0.062 G RMS at same frequency 90 Hz which is more than that of good bearing and thus indicates that it is defective bearing.

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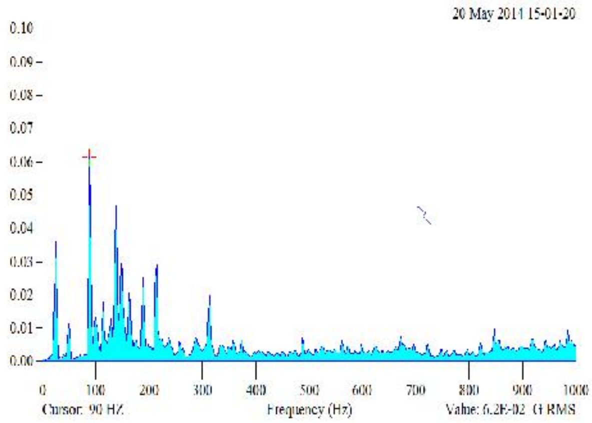

Figure 7. Vibration Signals of Bearing with Outer Race Defect (Frequency in Hz vs. Amplitude in G RMS)

IF WE COMPARE THIS EXPERIMENTAL FREQUENCY WITH THE OUTER RACE DEFECT FREQUENCY WHICH IS CALCULATED THEORETICALLY BY USING EQUATION (2) IS AT 89.14 HZ. IT INDICATES THAT EXPERIMENTAL FREQUENCY (90 HZ) IS VERY NEAR TO THEORETICAL OUTER RACE DEFECT FREQUENCY (89.14).

HENCE WE CAN CONCLUDE THAT THERE IS DEFECT ON OUTER RACE. THE PEAK VALUES (AMPLITUDES) OF GOOD BEARING AND DEFECTIVE BEARING AT ITS DEFECT FREQUENCIES ARE SHOWN IN THE TABLE 2. THE EXPERIMENTAL FREQUENCIES (135 HZ & 90 HZ) OF GOOD AND DEFECTIVE BEARINGS MENTIONED IN THE TABLE ARE VERY NEAR TO THE THEORETICAL DEFECTIVE FREQUENCIES (135.04 & 89.14 HZ) OF BEARINGS.

THE AMPLITUDE VALUES FOR INNER AND OUTER RACE OF GOOD BEARING VARIES WITH RESPECT TO TIME EVEN THERE IS NO DEFECT ON BEARING. THE AMPLITUDE VALUES FOR INNER RACE DEFECTIVE BEARING AND OUTER RACE DEFECTIVE BEARING INCREASES WITH RESPECT TO TIME BECAUSE OF INCREASE IN SEVERITY OF DEFECT.

VIII. CONCLUSION

Vibration analysis by using FFT analyzer is very useful and effective technique for detection of distributed defects on bearings especially at their early stages. Defects can be detected without interrupting manufacturing or production process of rotating machinery. Vibration spectra of good bearing and defective bearings are compared to find out the defect. Because of defect frequency equations it is possible to find out the location of the defect whether it is on outer race, inner race, ball or cage.

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