STATISTICAL ANALYSIS OF VIBRATION SIGNALS FOR CONDITION MONITORING OF DEFECTS IN EXHAUST FAN BEARINGS

Exhaust fan bearing defects and vibration frequency solutions analysis

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ABSTRACT

Identification of exhaust fan bearing defects has its root foundations in the analysis of frequency modulation and sensing using mostly online techniques.

Centrifugal fans are the most commonly used fans in the industry today. They are preferred to axles because they are cheaper and constructed with simplicity. They are used in wind systems in vehicles and buildings and also in delivering gas or minerals. They are suitable for industrial and air pollution processes and systems.

Fans play a key role in the production of re-circulating applications, in the supply of combustion air and the movement of air through processes and equipment for the protection of the environment. As they are indispensable components in these and other fields, it is essential that they always work reliably and efficiently. Reliability prevents interruptions that result in downtime and product damage. While engine failures or bearings sometimes contribute to fan failure, bearings are the most common failure point. Bearings are the crucial link between the rotating fan shaft and the stationary drive base, two of the strongest components of the fan.

Mechanics form the difference between the direct drive fan and the belt drive. In design, the motor in the direct drive discharge fans is directly connected to the fan board. This system is generally more efficient and has fewer moving parts, which reduces the likelihood of repair. The belt conveyor fans use a belt and motor to control the motor shaft. While the vibration friction of this system might cause inefficiency, the lower price and the quiet operation make this fan a popular choice. Direct drive and belt drive will provide the necessary suction air to heat the heat, smoke and smoke of your commercial kitchen.

The focus of this research study is the identification of exhaust fan bearing defects and requisite solutions to the problems caused by such defects. The use of harmonic and frequency modulations analysis in relation to revolutions per minute is critical in the determination of the extent of such defects. Identification of a clear algorithm to be applied to achieve the underlying objectives of the research study is of crucial importance.

KEYWORDS: Bearings, Exhaust fan defects, harmonics, frequencies.

I. INTRODUCTION

Important is the determination of the defects of rolling elements of the bearing components. Vibration diagnostics is one of the most effective analytical tools for defect analysis in fans, pumps, compressors, turbines, motors, etc.

Sound pattern recognition forms the basis of fault diagnosis. Unlike the time analysis, frequency analysis is able to isolate specific vibrational frequencies. Frequency analysis aids in the easy maintenance and value addition to adding to the life of machines for increased profitability through maintenance cost reduction.

This has been possible with the recent advent and move from preventive measures maintenance to condition based maintenance of bearings. Therefore bearing condition monitoring is preferably carried out through vibration analysis. Condition based maintenance has advanced in health management of fan bearings.

Health assessment and behavior analysis prediction focused on data processing improves the overall performance of data driven frameworks. Provision of better services, cost and risk reduction and ensured smooth and efficient running of the machines are the results of well-focused data frameworks.

Results from the data analysis of the velocity and frequency vibrations are most effective when used in conjunction with a health-based system of evaluating the results. The outcome should be generation of benchmark figures for the comparison of the data in question. Another important paradigm is the classification of the fan bearing vibrations data and defects to the bearings identified to enable the most effective method of systematic maintenance and residual life determination for the exhaust fan bearing.

II. BACKGROUND

Vibration analysis is a technique used to evaluate the operating conditions of the exhaust fan bearing and the deterioration trends, such as maintenance costs and the idle time. The vibration analysis technique consists of the measurement of vibration and its interpretation. Vibration measurement is recording the signal from the machine by measuring the vibrations.

Vibration is the movement of a machine or part of the machine or mechanical vibration. It may be normal, eg pendulum movement or occasional movement if. Vibrations can be expressed in units (m / s2) or gravitational "g" 1 g = 9.81 m / s2. The subject can vibrate in two ways: free vibrations and forced vibrations. Free vibration occurs when the object or structure is moved or shaken and then a natural vibration is obtained.

For example, if you lose a laptop computer, it will ring and will eventually turn off. At natural frequency, it is often the frequency in which a structure wants to "vibrate" after a stroke or change.

Resonance is the tendency of the system to oscillate at certain frequencies of others. Forced oscillation at or near the natural frequency of the object causes the structure of the energy structure. Over time, vibrations can become very important even if the vibration of the input force is very poor. If the structure has natural frequencies suitable for conventional vibrations, the structure loses noise and fails prematurely.

A Fast Fourier Transform Spectrum (FFT) is a very useful tool for machine vibration analysis. If there is a problem, the FFT spectrum will provide information to determine the cause of the problem and how long it will take to verify the problem.

FFT spectra allow the analysis and amplitudes of frequencies and frequencies of different components in the FFT spectrum. This way we can detect and monitor the vibrations that occur at certain frequencies. Since we know that some machine problems generate vibrations at certain frequencies, we can use this information to diagnose the cause of excessive vibration.

The signal is then processed with an FFT analyzer to get the frequency Spectrum. The result is mainly played by parents Frequency measured with relevant causes such as imbalance, Misalignment, bad bearing and resonance.

The overall level is an estimate of the sum vibration amplitude of the entire frequency dimension expanse. The measurement of global vibration, also called broadband, is one relatively simple and inexpensive unique value for collecting, processing, analyzing, setting and trending it.

III. OBJECTIVES

The general objectives of the study are determining the effects of application of frequency analysis in determination of exhaust fan bearing defects.

The specific objectives include;

a) To determine the effect of vibration frequency and speed calculations on the identification of fan bearing faults and defects.

b) To determine the effect of fan bearing defects on the performance of the bearing

c) To determine the impact of the algorithm chosen in solution finding and implementation to solve problems identified

IV. METHODOLOGY

1) ALGORITHM

The research study entailed capturing frequency data from the exhaust fan bearing industry. To be practicable mechanical information and other technical information for exhaust fan bearing vibration defect analysis availed to the industry used in the study.

The information which includes the harmonics and revolutions per minute, ball passing frequency-outer race (BPFO), the ball spin frequency (BSF), fundamental train frequency (FTF) and the ball passing frequency inner race (BPFI) are analyzed using Minitab, a statistical application program for the determination of defect and extent of remaining life of the exhaust fan bearing under investigation.

An inquest was then made to find out the statistical parameters and outputs necessary for determining the health of the fan bearings. The inputs into the computer algorithm aid in the determination the health of exhaust fan bearings[24].

The inputs include the harmonics and revolutions per minute, which are then used to calculate the velocity against which all the frequencies are compared in statistical analysis.

The outputs from the system are statistical frequency graphs and histograms, statistical parameters for vibration data studied and data analysis involving the frequency of vibration signals and velocity (Harmonics multiplied by revolutions per minute).

Further, the industry-collected data analyzed in the algorithm is subjected to health monitoring by comparing it with collected experimental accelerometer frequency and bearing revolutions benchmark standards developed in the algorithm over time.

This being a concise way of determining the possible defects in specific exhaust fan bearings and the relevant life expectancy of the bearing over its remaining useful life [7] .

This is possible by storing analyzed data file and exhaust fan bearing defect analysis information within the system for future reference and analysis.

The nearest neighbor anomaly detector is then applied to detect data, which is placed far from the clustered data, which indicates an anomaly. To decide whether new data X is an anomaly it is compared to its nearest neighbor Y after normalization of the distance between Y and its nearest neighbor Z.

Distance = $\| X - Y \| / \| Y - Z \|$ …………(1)

2) MODELLING

The methodology of inquest employed was online vibration monitoring which is about constantly buying vibration signals and the use of this reduced data as indicators of machine health in near real time.

Factory and field test and vibration data collection can be greatly undermined by the level of grease and oil in the fan bearing being tested. Several precautions are undertaken to ensure these factors are reduced significantly.

IMPLEMENT APPROPRIATE MODELING ALGORITHMS DATA SET OF YOUR CHOICE.

For the purposes of this research study, the initial data set will comprise of relevant harmonics multiplied with revolutions per minute to determine velocity, fundamental train frequency inner race (FTFI), ball passing frequency-outer race (BPFO), the ball spin frequency (BSF), fundamental train frequency (FTF) and the ball passing frequency inner race (BPFI).

A maximum of twenty-five data set will be analyzed using Minitab software for statistical analysis. This will enable the calculation of various test statistics for analysis of exhaust fan bearing defects emanating from the velocity and frequencies data set implemented.

RESEARCH ANDANALYTICAL TOOLS USED IN THE RESEARCH STUDY

The statistical tools used in the research study varied in all the stages of data identification, collection, analysis and reporting of findings. One very prominent tool used in research for identifying information relevant to the study was the internet. Vast amounts of files downloaded from many internet source websites and most, referenced in this paper.

One important source of raw data necessary for the research study was the bearing expert international source index, Inc. which website can be found at www.sourceindex.com.

The process of generating information on the website requires that the researcher first source a bearing manufacturer part number to input into the website. Once the website recognizes the bearing part number, then the researcher goes on further to generate vibration harmonics and frequencies at various revolutions per minute and bearing contact degrees from the websites database.

The website proved to be a crucial resource to this research study as it provided the raw data and diagrams about the vibration detecting frequencies necessary for analytical calculations needed to draw inferences and make conclusions about the topic of discourse.

Other numerable websites where resource papers were found which are relevant to the topic of research in this study were also used.

Another important tool is the statistical software Minitab that was used to make the calculations and analysis on the exhaust fan bearing data obtained from the bearing expert, source index.

SETTING UP FACTORY TEST EQUIPMENT REQUIREMENTS

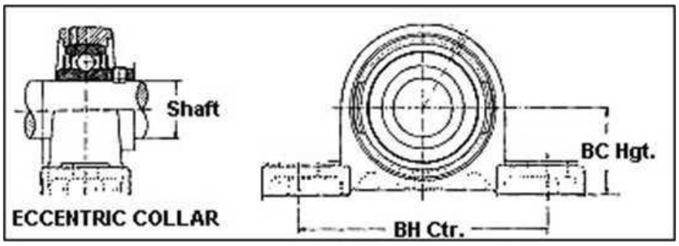


Figure 1 (a) Ball bearing pillow blocks

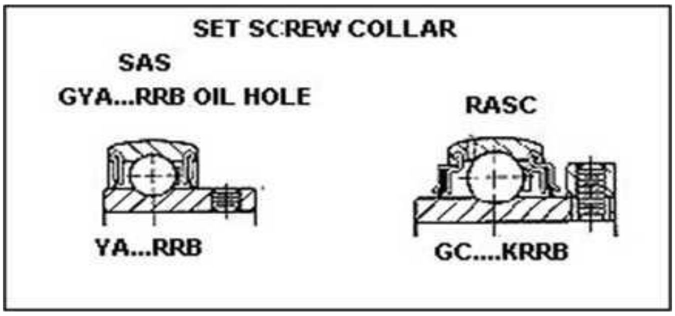
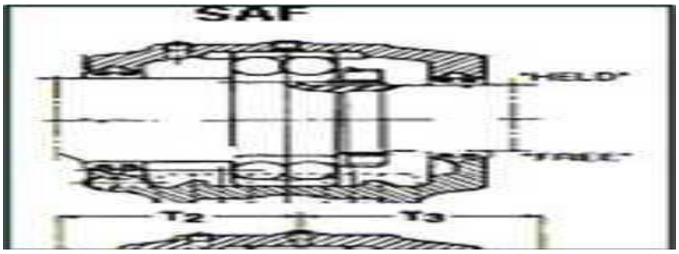


Figure 2 (b) Ball bearing pillow blocks set screw



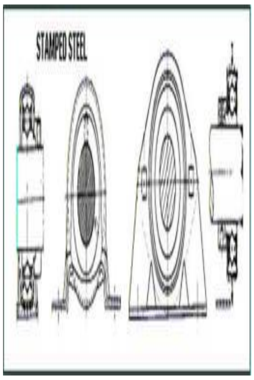


Figure 3 (c) Ball bearing pillow G....KRRB LUB HOLE GRA...RRB LUB HOLE G....KLLB OIL HOLE G....KPPB3(2,4) OIL HOLE

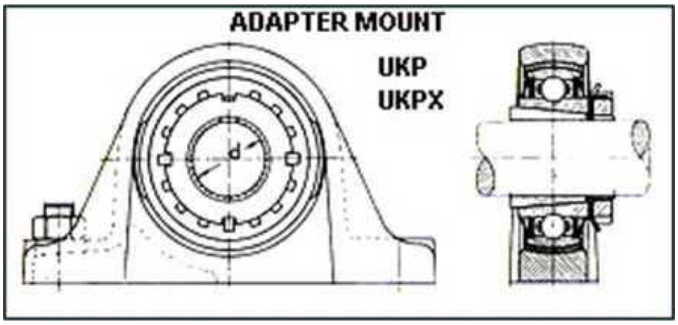


Figure 3 adapter mount the exhaust fan bearing (pillow block)

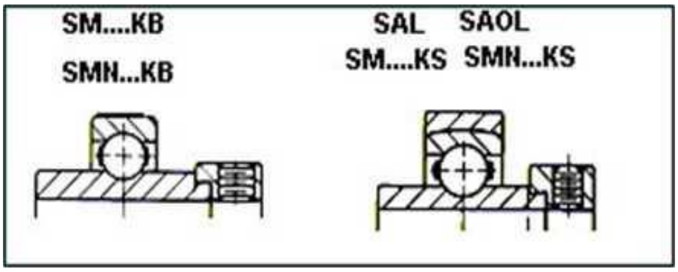
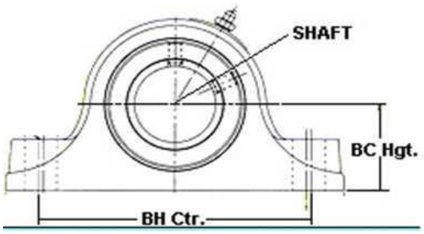
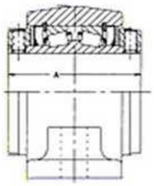


Figure 4 stamped steel ensures stability and vibration data accuracy

ROLLER BEARING PILLOW BLOCKS





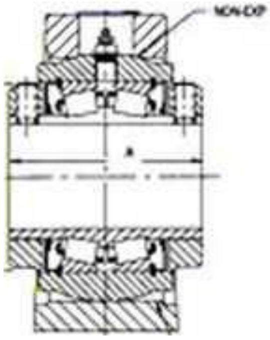


Figure 5TYPEE 2 BOLT 4 BOLT

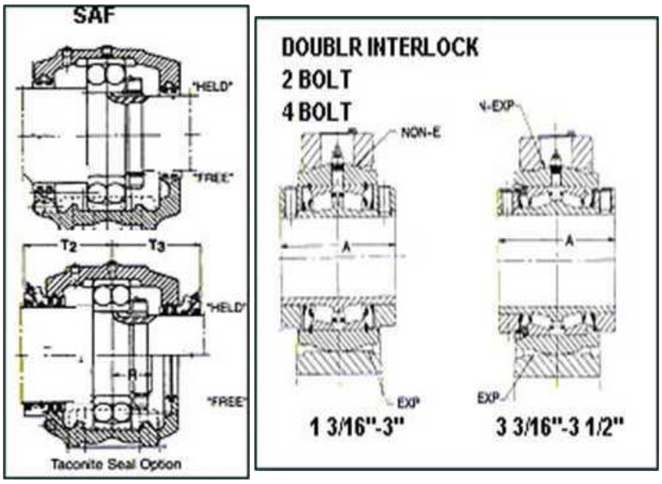


Figure 6 SAF AND DOUBLE INTERLOCK 2 BOLT 4 BOLT

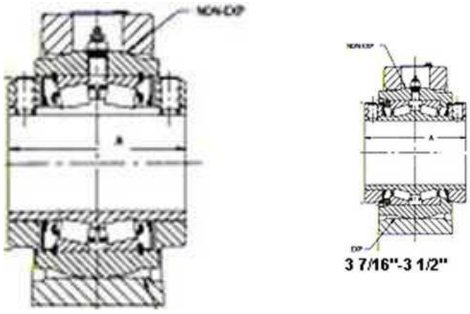


Figure 8 TAF 2 BOLT 4 BOLT

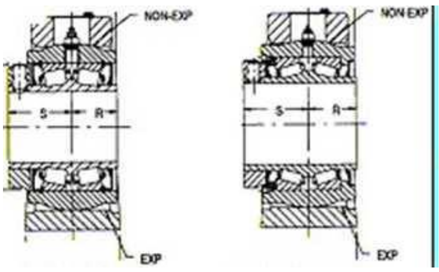


Figure 7 TYPEK 2 BOLT 4 BOLT

SOURCE OF DIAGRAMS
WWW.SOURCEINDEX.COM
INTERNATIONAL SOURCE INDEX, Inc.
The bearing expert TM

Grease hardening and insufficient lubrication can be caused by mixing the grease with different bases and thus should be completely avoided. To change from the lithium-based grease used in factory manufacture of exhaust fan bearings, continually add the new grease to the bearing until only the new grease built with another base such as sodium or synthetic base completely replaces the factory lithium based grease.

The galvanized steel base bracket of the exhaust fan is welded to provide the strength and support needed for it's cleaning, preventing spillage and loss into the building. A factory-covered aluminum grease plug is welded to the fan housing to ensure that it does not spill oil and grease.

Low oil levels do not provide adequate lubrication. The level meters are used to control the oil level must be adjusted to read correctly. Since it has been indicated when the fan is on, it's better to check the oil levels when the fan is off.

Maintenance of the bearing while ensuring proper lubrication ensures the long life of the bearing. Thrusts in the centrifugal fan help to preserve bearing life [12]. To prevent damage to the bearing a sufficient supply of lubrication should be ensured to avoid metal-to-metal contact between the bearings with the fan.

Static or circulating oil should only be used with separation High speed or high temperature cushion bearings applications where grease is inadequate. Lubricated with oil bearings require more rigorous maintenance.

During testing air inside the chamber should be avoided at all costs and at all times. High oil levels can prevent the bearing from venting the air inside rolling while the bearing warms up in revolutions. The accumulated pressure can blow all the oil on landing in a few minutes.

3) *CONDITION MONITORING*

Identification of fan bearing defaults involved the collection of fault frequencies, which include the harmonics and revolutions per minute, fundamental train frequency (FTF), the ball spin frequency (BSF), ball passing frequency-outer race (BPFO), and the ball passing frequency inner race (BPFI).

Data analysis of the ball passing frequency-outer race (BPFO), and the ball passing frequency inner race (BPFI) in conjunction with the revolutions per minute was applied to identify the fan-bearing defect.

4) *EXHAUST FAN BEARING AND ITS USES*

The fan bearing chosen for this research study is the

UCP204-12-AH-SP4 – exhaust fan bearing three quarter inch pillow block bearing with cast housing and grease fitting. Serrated Tip Set Screw With Loctite Patch. Set Screw Torque equal to five-inch lbs. Grease fitting on opposite side of standard.

The UCP204-12-AH-SP4 bearing has been used in many applications including those mentioned below:

	Used in	Quantity Used
DD7FA	Centrifugal down blast belt drive exhaust fan with a 11.75 inch wheel.	2
DD7HPFA	HP Centrifugal down blast belt drive exhaust fan with a 11.75 inch wheel.	2
DD8FA	Centrifugal down blast belt drive exhaust fan with an 11.75 inch wheel.	2
DD8HPFA	HP Centrifugal down blast belt drive exhaust fan with an 11.75 inch wheel.	2
DD9FA	Centrifugal down blast belt drive exhaust fan with a 11.75 inch wheel.	2
DD9HPFA	HP Centrifugal down-blast belt drive exhaust fan with a 11.75 inch wheel.	2
DD11FA	Centrifugal down-blast belt drive exhaust fan with an 11.75 inch wheel.	2
DD11HPFA	HP Centrifugal down-blast belt drive exhaust fan with an 11.75 inch wheel.	2

DD13FA	Centrifugal down-blast belt drive exhaust fan with a 13.75" wheel.	2
DD13HPFA	HP Centrifugal down-blast belt drive exhaust fan with a 13.75" wheel.	2
DD15FA	Centrifugal down-blast belt drive exhaust fan with a 15.75" wheel.	2
DD15HPFA	HP Centrifugal down-blast belt drive exhaust fan with a 15.75" wheel.	2
NCA8FA	Belt Drive Centrifugal Up-blast Exhaust Fan with 11.75 inch wheel	2
NCA8HPFA	HP Belt Drive Centrifugal Up-blast Exhaust Fan with 11.75 inch wheel	2
NCA10FA	Belt Drive Centrifugal Up-blast Exhaust Fan with 13.75 inch wheel	2
NCA10HPFA	HP Belt Drive Centrifugal Up-blast Exhaust Fan with 13.75 inch wheel	2
NCA14FA	Belt Drive Centrifugal Up-blast Exhaust Fan with 15.75 inch wheel	2
NCA14HPFA	High Pressure Belt Drive Centrifugal Up-blast Exhaust Fan with 15.75 inch wheel	2

5) *DETECTION*

Exhaust fans are built to work efficiently. Evident failure in new and old exhaust fan bearings is testimony enough that manufacture and reliance on long life usage is nowhere near perfection. Detecting the causes of such failures or the problems with the bearing is important in predicting, gauging and guarding against eminent loss in usage of the exhaust fan bearing.

A series of high-energy pulses at a rate equal to the frequency of passing the ball in relation to the inner raceway result from a small fault on the inner raceway. the defect will enter and leave the load area as a result of the rotation of the inner ring causing a variation in contact raceway-bearing raceway force, so deflections will be detected[9].

Contamination is a frequent source of bearing damage and premature failure and is caused by the entry of strange element either because of improper treatment in lab testing or when running on the exhaust fan. Naturally the size of the vibration resulting from contamination varies and in the beginning steps is not easy to recognize, but relies largely on the type and nature of impurities. Contamination is a major source of depreciation and damage to the bearing contact surfaces and emanate vibrations over a variation of frequencies.

Other causes of failure exhaust fan bearings include rusting of the bearing result from existence of moisture, broken bearings, low quality wrapping, acidic wear and tear and high temperatures. Buildup of metal in front of the rollers resulting in flaking, contamination due to defective bearing seals, arcing and burning at the point between the races and the rolling elements in electric currency damage of the bearing, mis-alignment and improper mounting and existence of foreign matter in the exhaust fan bearing are reasons, which also cause bearing damage and defects.

Accelerometers placed at keys positions on the motor and fan bearings are used to measure vibrations caused by the exhaust fan bearing during live factory testing and diagnosis. Because the bearings are the support part, mechanical train accelerometers should be placed in the entrance and exit of rolling bearings for vibration measurement levels.

Permanent accelerometers as exhaust fan bearings vibration sensors must be placed in a vertical and horizontal radial and axial positions on the motor. This provides the ultimate recognition to all vibration components, including bearing vibration, imbalance failure, electric disturbance, wings flow disturbances of aeronautical equipment and frequency bands[17].

The piezoelectric accelerometer is widely regarded as a standard vibration sensor for vibration measurement of the machine. System configuration the piezoelectric crystal and the seismic mass depend on the frequency range of the sensor desired.

The prufteck vibroexpert does not only offers frequency information, but also general amplitudes for data analysis. It is essential for tendency or comparing measurements on machines of same type but also for diagnosis of defects on machines. Similar large vibration instruments are tunable filters and FFT - Fast Fourier Transform analysis. Analyzer is a default because analysis is a human function. An electronic analyzer "analyzed" can only measures electrical signals display.

The electrical signals of Accelerometers and speed transducers called millivolts are very small AC voltages. Therefore, the tunable filters and FFT instruments are no more than fancy AC voltmeters with a frequency display axis. These large vibration-measuring instruments, of any one manufacturer, can be used effectively to diagnose machine vibrations.

The balancing and realignment method can be used to correct the blower vibration problems. This entails disassembly, manual sight inspection and the reassembling the bearing. It might result in replacement of the fan bearing or the defective parts, balancing and realignment and lubrication.

Clear and repeatable vibration measurements and data capture keys are not difficult to understand and set up. They are based on sound reason, care, organization and consistency. Precise vibration measurement requires the selection of the best measuring stations fan assembly. It also requires the best vibration converter specific frequencies or frequencies of general interest.

Last years, piezoelectric accelerometers have become the most widely used sensor type because of their general frequency characteristics, size, reliability and overall stability. Unless low frequency vibration measurement is required, the unusually low amplitude accelerometer is usually the best choice for accurate data.

The latest vibration measuring devices have the option of selecting a transition, speed or acceleration vibration measurement parameters, regardless of the type of sensor used. It is desirable, for the sake of accuracy, to use the parameter that provides the best (lowest) interest rate.

It turns out that speed is usually the best choice parameter for measuring vibration of the machine. The most common exception to this rule is using higher frequency accelerations to detect problems with certain components such as roller bearings.

How to use vibration converters to the measuring point is critical aspect in vibration measurement accuracy. Sensors properly connected to the machine achieve this measurement effectively. The different categories of machine vibration discussed are better evaluated to gain confidence about determining defects and possible solutions..

6) CLASSIFICATION

There are four phases of classifying exhaust fan bearing defects. These phases include the phase one of wear and tear of the fan bearing where the possibility of complete failure of the bearing is quite low. At this phase, further monitoring for defect extents can be carried out on the fan bearing.

In the second phase most damage to the fan bearing can be seen with the naked eye and at the same time it can be sensed in the increased quantity and amplitude of fault frequency harmonics. In this second phase, outer race fault frequency is detected in the first bearing band. Inner race fault frequency is detected in second bearing band. Repair should continue at scheduled time intervals

In the third phase three, the ball spin frequencies appear in the outer bearing band. Harmonics of bearing fault frequencies should be easy to detect in a velocity spectrum. The possibility of catastrophic failure is imminent and repair should be more periodical.

In the fourth phase, the discrete frequency indicator will shows the extent of fundamental train frequency FTF (cage frequency) indicates that the bearings life is almost over or already over.

There are four major classifications of belt drives including flat belts, V-belts, cogged V-belts and synchronous belts. This research study focuses on analyzing the fan bearing defects vibrations used mostly in V-belts.

V. RESULTS

Analytical tests conducted using the chosen exhaust fan bearing data yielded various results, which enabled the process of decision-making. The determination of defect and lifetime remaining for a fan bearing enabled the compilation of specific benchmark predictors. In this section of the report follows the discussion of the results from data analysis.

TABLE 1 : HARMONIC AND FREQUNCY TABLE AT 179.99O WITH 50,000 REVOLUTIONS PER MINUTE FOR UCP204-12-AH-SP4 - Bearing three quarter inch pillow block bearing

SOURCE OF TABLES

WWW.SOURCEINDEX.COM

INTERNATIONAL SOURCE INDEX, Inc.

The bearing expert TM

Harmoni	FTFI	FTFO
1	306.564.8047	193.435.1953
2	613,129.6093	386.870.3907
3	919.694.4140	580.305.5860
4	1.226.259.2187	773.740.7813
5	1,532.824.0233	967,175.9767
6	1.839.388.8280	1,160,611.1720
7	2.145.953.6327	1,354.046.3673
8	2.452,518.4373	1.547.481.5627
9	2,759.083.2420	1,740.916.7580
10	3.065.648.0466	1.934.351.9534
11	3,372.212.8513	2.127.787.1487
12	3.678.777.6560	2.321.222.3440
13	3.985.342.4606	2.514.657.5394
14	4.291.907.2653	2,708.092.7347
15	4.598.472.0700	2.901.527.9300
16	4.905,036.8746	3.094.963.1254
17	5,211.601.6793	3.288.398.3207
18	5,518.166.4840	3.481.833.5160
19	5.824.731.2886	3.675,268.7114
20	6.131.296.0933	3.868.703.9067
21	6.437.860.8980	4.062,139.1020
22	6.744.425.7026	4.255,574.2974
23	7,050.990.5073	4.449.009.4927
24	7,357,555.3120	4.642.444.6880
25	7.664.120.1166	4.835.879.8834
26	7.970.684.9213	5.029.315.0787
27	8.277,249.7259	5.222.750.2741
28	8.583.814.5306	5.416.185.4694

TABLE 2 HARMONIC AND FREQUNCY TABLE AT 179.99O WITH 50,000 REVOLUTIONS PER MINUTE FOR UCP204-12-AH-SP4 - Bearing 3/4" pillow block bearing

Harmonic	BPFO	BPFI
1	2,452,518.4373	1.547.481.5627
2	4.905,036.8746	3,094.963.1254
3	7,357,555.3120	4.642.444.6880
4	9.810,073.7493	6,189.926.2507
5	12,262,592.1866	7,737,407.8134
6	14,715,110.6239	9.284.889.3761
7	17,167,629.0612	10.832,370.9388
8	19.620,147.4985	12,379.852.5015
9	22.072.665.9359	13,927,334.0641
10	24,525,184.3732	15,474.815.6268
11	26.977,702.8105	17,022,297.1895
12	29,430.221.2478	18,569,778.7522
13	31.882,739.6851	20,117,260.3149
14	34,335,258.1225	21.664,741.8775
15	36,787,776.5598	23.212.223.4402
16	39,240,294.9971	24,759,705.0029
17	41,692.813.4344	26,307,186.5656
18	44,145,331.8717	27,854.668.1283
19	46,597,850.3091	29,402,149.6909
20	49,050,368.7464	30.949.631.2536
21	51,502.887.1837	32,497,112.8163
22	53.955,405.6210	34.044,594.3790
23	56,407,924.0583	35,592.075.9417
24	58.860.442.4956	37,139,557.5044
25	61,312.960.9330	38.687,039.0670
26	63,765,479.3703	40.234,520.6297
27	66,217,997.8076	41,782.002.1924
28	68.670.516.2449	43.329.483.7551

SOURCE: www.sourceindex.com

COMPARE AND CONTRAST PERFORMAMCE

The harmonic, inner, and outer race vibrations frequencies garnered in the research study varied according to revolutions per minute and the size of the angle of degree of contact of the exhaust fan bearing. These performance metrics are discussed at length in the following tables.

HARMONICS ANALYSIS

TABLE 3 FREQUECIES ANALYSIS AT 0^{O} 50000 RPM

Shaft

19.050 mm 0.7500 in B-C Height

33.340 mm 1.3126 in B-H Ctr

96.045 mm 3.7813 in Enter a Contact Angle

RECALCULATE |

Click here for Contact Ancle help

Part No.

UCP204-012 Manuf. Rolling Elements:

NTN 8 Element Diameter:

7.9248 Pitch Diameter:

34.4932 Contact Angle:

0.00° Published

Harmonic	FTFI	FTFO
1	192.562.5920	307.437.4080
2	385,125.1841	614.874.8159
3	577.687.7761	922.312.2239
4	770.250.3682	1.229.749.6318
5	962.812.9602	1,537,187.0398
6	1,155,375.5523	1.844.624.4477
7	1.347,938.1443	2.152.061.8557
8	1,540,500.7364	2.459.499.2636
9	1,733.063.3284	2,766.936.6716
10	1.925.625.9205	3,074.374.0795
11	2,118.188.5125	3.381,811.4875
12	2,310,751.1046	3.689.248.8954
13	2.503,313.6966	3.996.686.3034
14	2.695.876.2887	4,304.123.7113
15	2.888.438.8807	4.611,561.1193
16	3.081,001.4728	4.918.998.5272
17	3.273.564.0648	5.226.435.9352
18	3.466.126.6568	5,533.873.3432
19	3.658.689.2489	5.841,310.7511
20	3.851.251.8409	6.148.748.1591
21	4.043.814.4330	6.456.185.5670
22	4.236.377.0250	6.763.622.9750
23	4.428.939.6171	7.071.060.3829
24	4.621,502.2091	7,378.497.7909
25	4.814.064.8012	7.685.935.1988

BPFO	BPFI
1.540.500.7364	2,459.499.2636
3.081,001.4728	4,918.998.5272
4.621,502.2091	7,378.497.7909
6.162.002.9455	9.837,997.0545
7,702.503.6819	12.297,496.3181
9.243.004.4183	14,756,995.5817
10.783.505.1546	17,216.494.8454
12.324.005.8910	19.675,994.1090
13.864.506.6274	22.135,493.3726
15,405,007.3638	24,594.992.6362
16.945.508.1001	27,054.491.8999
18.486.008.8365	29.513.991.1635
20.026.509.5729	31,973,490.4271
21,567.010.3093	34.432.989.6907
23,107,511.0457	36.892.488.9543
24.648.011.7820	39.351.988.2180
26.188.512.5184	41.811,487.4816
27,729.013.2548	44.270.986.7452
29.269.513.9912	46.730.486.0088
30.810.014.7275	49.189.985.2725
32.350.515.4639	51,649.484.5361
33.891.016.2003	54.108.983.7997
35.431,516.9367	56.568.483.0633
36.972.017.6730	59.027.982.3270
38.512.518.4094	61,487,481.5906

The fundamental fault frequency inner race is less than the fundamental fault frequency outer race. The ball passing frequency outer race is less than the ball passing frequency inner race. This shows also that the fundamental fault frequency inner race is less while the ball passing frequency outer race is greater. The defect of the fan bearing is therefore at the initial stages. The ball spin frequency is always at the center of all these defect vibration frequencies at any level of harmonics.

TABLE 4 FREQUECIES ANALYSIS AT 179.99^{O} 50000 RPM

Harmonic	FTFI	FTFO
1	307.437.4079	192.562.5921
2	614.874.8159	385,125.1841
3	922.312.2238	577.687.7762
4	1.229.749.6318	770.250.3682
5	1,537,187.0397	962.812.9603
6	1.844.624.4477	1,155,375.5523
7	2.152.061.8556	1.347.938.1444
8	2.459.499.2636	1.540,500.7364
9	2,766.936.6715	1,733.063.3285
10	3,074.374.0794	1.925.625.9206
11	3.381,811.4874	2,118.188.5126
12	3.689.248.8953	2,310.751.1047
13	3.996.686.3033	2.503.313.6967
14	4,304.123.7112	2.695.876.2888
15	4.611,561.1192	2.888.438.8808
16	4.918.998.5271	3.081.001.4729
17	5.226.435.9351	3.273.564.0649
18	5,533.873.3430	3.466.126.6570
19	5.841,310.7509	3.658.689.2491
20	6.148.748.1589	3.851,251.8411
21	6.456.185.5668	4.043.814.4332
22	6.763.622.9748	4.236.377.0252
23	7.071,060.3827	4.428.939.6173
24	7,378.497.7907	4.621,502.2093
25	7.685.935.1986	4.814.064.8014

BPFO	BPFI
2.459,499.2636	1.540,500.7364
4,918,998.5271	3,081.001.4729
7,378,497.7907	4.621,502.2093
9.837,997.0542	6,162,002.9458
12,297,496.3178	7,702,503.6822
14,756,995.5813	9.243,004.4187
17,216,494.8449	10,783,505.1551
19.675,994.1084	12,324,005.8916
22,135,493.3720	13.864,506.6280
24,594,992.6355	15,405,007.3645
27,054,491.8991	16.945,508.1009
29,513,991.1626	18,486,008.8374
31.973,490.4262	20.026,509.5738
34,432,989.6897	21,567.010.3103
36.892,488.9533	23,107,511.0467
39,351,988.2168	24.648.011.7832
41.811,487.4804	26,188,512.5196
44,270,986.7440	27,729.013.2560
46,730,486.0075	29.269,513.9925
49,189.985.2711	30.810.014.7289
51.649,484.5346	32,350,515.4654
54,108,983.7982	33.891.016.2018
56,568,483.0617	35.431,516.9383
59,027,982.3253	36.972,017.6747
61,487,481.5888	38,512.518.4112

When the angle is adjusted to 179.99^{o}, now the Fundamental train frequency, which shows fundamental damage to the inner race of the bearing, is greater than the fundamental train damage outer race. The ball passing frequency outer race also becomes greater than the ball passing frequency inner race. This hint at continued damage to both the inner race, outer race and ball spin elements of the bearing.

TABLE 5 FREQUECIES ANALYSIS AT 190^{O} 50000 RPM

Harmonic	FTFI	FTFO
1	306.564.8047	193.435.1953
2	613,129.6093	386.870.3907
3	919.694.4140	580.305.5860
4	1.226.259.2187	773.740.7813
5	1,532.824.0233	967,175.9767
6	1.839.388.8280	1,160,611.1720
7	2.145.953.6327	1,354.046.3673
8	2.452,518.4373	1.547.481.5627
9	2,759.083.2420	1,740.916.7580
10	3.065.648.0466	1.934.351.9534
11	3,372.212.8513	2.127.787.1487
12	3.678.777.6560	2.321.222.3440
13	3.985.342.4606	2.514.657.5394
14	4.291.907.2653	2,708.092.7347
15	4.598.472.0700	2.901.527.9300
16	4.905,036.8746	3.094.963.1254
17	5,211.601.6793	3.288.398.3207
18	5,518.166.4840	3.481.833.5160
19	5.824.731.2886	3.675,268.7114
20	6.131.296.0933	3.868.703.9067
21	6.437.860.8980	4.062,139.1020
22	6.744.425.7026	4.255,574.2974
23	7,050.990.5073	4.449.009.4927
24	7,357,555.3120	4.642.444.6880
25	7.664.120.1166	4.835.879.8834

BPFO	BPFI
2,452,518.4373	1.547.481.5627
4.905,036.8746	3,094.963.1254
7,357,555.3120	4.642.444.6880
9.810,073.7493	6,189.926.2507
12,262,592.1866	7,737,407.8134
14,715,110.6239	9.284.889.3761
17,167,629.0612	10.832,370.9388
19.620,147.4985	12,379.852.5015
22.072.665.9359	13,927,334.0641
24,525,184.3732	15,474.815.6268
26.977,702.8105	17,022,297.1895
29,430.221.2478	18,569,778.7522
31.882,739.6851	20,117,260.3149
34,335,258.1225	21.664,741.8775
36,787,776.5598	23.212.223.4402
39,240,294.9971	24,759,705.0029
41,692.813.4344	26,307,186.5656
44,145,331.8717	27,854.668.1283
46,597,850.3091	29,402,149.6909
49,050,368.7464	30.949.631.2536
51,502.887.1837	32,497,112.8163
53.955,405.6210	34.044,594.3790
56,407,924.0583	35,592.075.9417
58.860.442.4956	37,139,557.5044
61,312.960.9330	38.687,039.0670

The comparisons remain the same as in the analysis done in 179.99^{o}. Perhaps an important phenomenon to note is that it was impossible to directly collect the frequencies at 180^{o} to 189.99^{o} by directly inputting the degrees into the system. This may be because at 180^{o} symmetry there may not be any defects detected in the exhaust fan bearing, either because it is properly mounted in the testing equipment or there was no looseness detected.

TABLE 6 FREQUECIES ANALYSIS AT 360^{O} 50000 RPM

Harmonic	FTFI	FTFO
1	192.562.5920	307.437.4080
2	385,125.1841	614.874.8159
3	577.687.7761	922.312.2239
4	770.250.3682	1.229.749.6318
5	962.812.9602	1,537,187.0398
6	1,155,375.5523	1.844.624.4477
7	1.347,938.1443	2.152.061.8557
8	1.540,500.7364	2.459.499.2636
9	1,733.063.3284	2,766.936.6716
10	1.925.625.9205	3,074.374.0795
11	2,118.188.5125	3.381,811.4875
12	2,310,751.1046	3.689.248.8954
13	2.503.313.6966	3.996.686.3034
14	2.695.876.2887	4,304.123.7113
15	2.888.438.8807	4.611,561.1193
16	3.081,001.4728	4.918.998.5272
17	3,273,564.0648	5.226.435.9352
18	3.466.126.6568	5,533.873.3432
19	3.658.689.2489	5.841,310.7511
20	3.851.251.8409	6.148.748.1591
21	4.043.814.4330	6.456.185.5670
22	4.236.377.0250	6.763.622.9750

23	4.428.939.6171	7.071,060.3829
24	4.621,502.2091	7,378.497.7909
25	4.814.064.8012	7.685.935.1988

BPFO	BPFI
1.540,500.7364	2.459,499.2636
3,081,001.4728	4.918.998.5272
4,621,502.2091	7,378.497.7909
6,162,002.9455	9.837.997.0545
7,702,503.6819	12,297,496.3181
9,243,004.4183	14,756.995.5817
10,783,505.1546	17,216,494.8454
12,324,005.8910	19.675.994.1090
13.864,506.6274	22,135,493.3726
15,405,007.3638	24,594.992.6362
16.945,508.1001	27,054,491.8999
18,486,008.8365	29,513.991.1635
20,026,509.5729	31,973,490.4271
21,567,010.3093	34,432.989.6907
23,107,511.0457	36.892,488.9543
24.648.011.7820	39,351.988.2180
26,188,512.5184	41.811,487.4816
27,729,013.2548	44,270.986.7452
29,269,513.9912	46,730,486.0088
30.810,014.7275	49,189.985.2725
32,350,515.4639	51.649,484.5361
33.891,016.2003	54,108.983.7997
35,431,516.9367	56,568,483.0633
36.972,017.6730	59.027.982.3270
38,512,518.4094	61,487.481.5906

The fundamental train damage of the outer race element is now larger the fundamental train damage inner race element although now the damage is much larger.

The ball passing frequency inner race also continues to show a larger deviation in size and magnitude compare to the ball passing frequency outer race. This indicates larger extent of defects detected in the exhaust fan bearing.

The same trends were noted for data at larger and fewer revolutions with the size and magnitude of changes in inner and outer race defects varying with the number of revolutions.

For example the harmonics and vibration frequency data at 0^{o}, RPM × 1 are shown below.

Harmonic	FTFI	FTFO
1	0.3851	0.6149
2	0.7703	1.2297
3	1.1554	1.8446
4	1.5405	2.4595
5	1.9256	3.0744
6	2.3108	3.6892
7	2.6959	4.3041
8	3.0810	4.9190
9	3.4661	5.5339
10	3.8513	6.1487
11	4.2364	6.7636
12	4.6215	7.3785
13	5.0066	7.9934
14	5.3918	8.6082
15	5.7769	9.2231
16	6.1620	9.8380
17	6.5471	10.4529
18	6.9323	11.0677
19	7.3174	11.6826
20	7.7025	12.2975
21	8.0876	12.9124
22	8.4728	13.5272
23	8.8579	14.1421
24	9.2430	14.7570
25	9.6281	15.3719

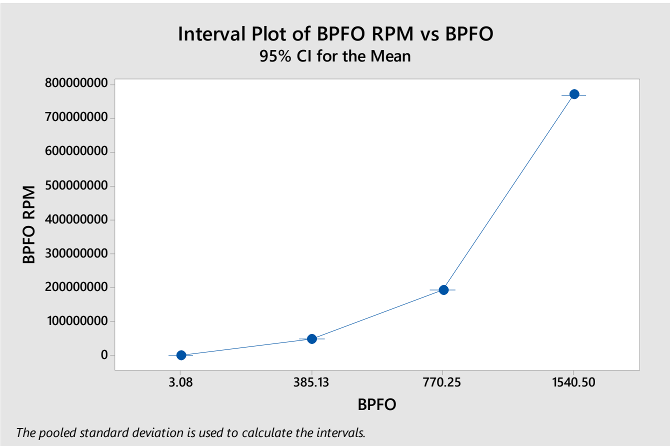
BPFO	BPFI
3.0810	4.9190
6.1620	9.8380
9.2430	14.7570
12.3240	19 6760
15.4050	24.5950
18.4860	29 5140
21.5670	34.4330
24.6480	39 3520
27.7290	44.2710
30.8100	49.1900
33.8910	54.1090
36.9720	59.0280
40.0530	63.9470
43.1340	68 8660
46.2150	73.7850
49.2960	78 7040
52.3770	83.6230

55.4580	88 5420
58.5390	93.4610
61.6200	98.3800
64.7010	103.2990
67.7820	108.2180
70.8630	113.1370
73.9440	118.0560
77.0250	122.9750

Just like in the RPM × 50000, 0o analysis, the fundamental train frequency outer race is greater than fundamental train frequency inner race and the ball pass frequency inner race is greater the ball pass frequency outer race at RPM × 1 with the only difference being the size and magnitude at RPM × 50000, 0o being greater.

VIBRATION STATISTICAL PARAMETER VELOCITY VIBRATION LEVELS

FIGURE 9 INTERVAL PLOT FOR THE CONFIDENCE LEVEL BPFO RPM (VELOCITY) TO FREQUENCY VIBRATION OUTER RACE



SOURCE: Author

TABLE 7 BEARING FAULT FREQUCIES AT VARIOUS REVOLUTIONS PER MINUTE

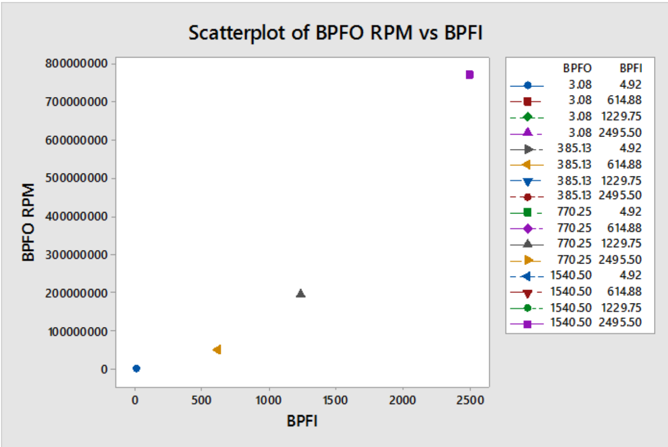
FTFI	FTFO	BSF	BPFO	BPFI	RPM
0.385	0.615	2. 0614	3.08	4.92	1
48.138	76.862	257.6750	385.13	614.88	125000
96.275	153.725	515.3600	770.25	1229.75	250000
192.550	307.450	1030.7000	1540.50	2495.50	500000

Results of multiplying RPM with all frequencies to get velocities

FTFI (RPM)	FTFO (RPM)	BPFO RPM	BPFI RPM
1	1	3	5
6017188	9607813	48140625	76859375
24068750	38431250	192562500	307437500
96275000	153725000	770250000	1247750000

SOURCE: Author

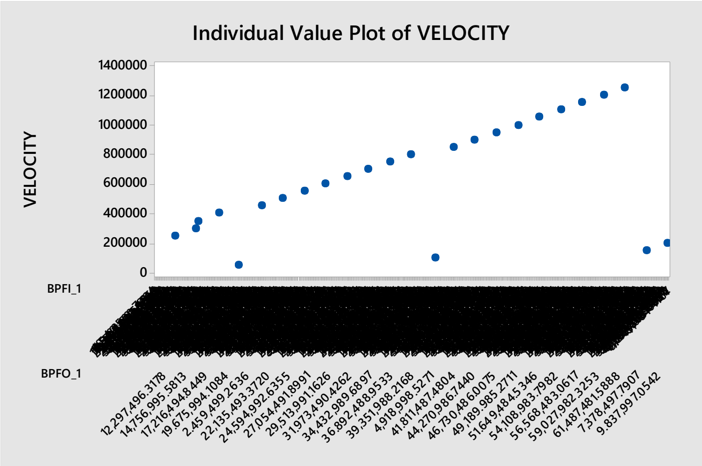
FIGURE 10 SCATTERPLOT OF BPFO RPM **vs** BPFI



The scatter plot for BPFO RPM (or velocity) v/s frequency of vibration outer race defect vibration BPFI shows anomaly in the fan bearing because the data is not linearly co-related, but significant variances are recorded in the data points plotted.

SOURCE: Author

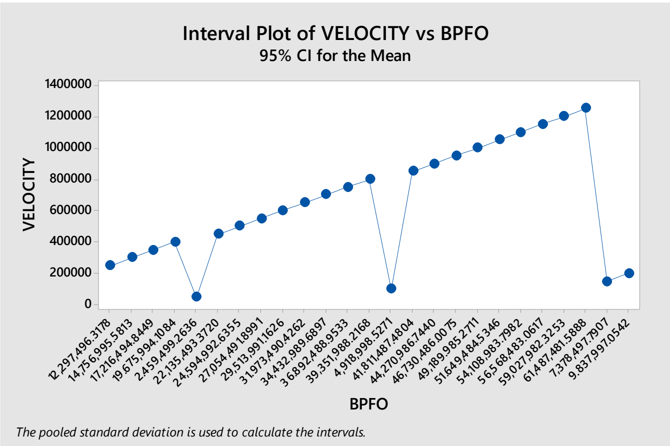
FIGURE 11 INDIVIDUAL VALUE PLOTS VELOCITY AND INNER RACE (BPFI) AND OUTER RACE (BPFO) VIBRATION FERQUENCIES



Source: Author

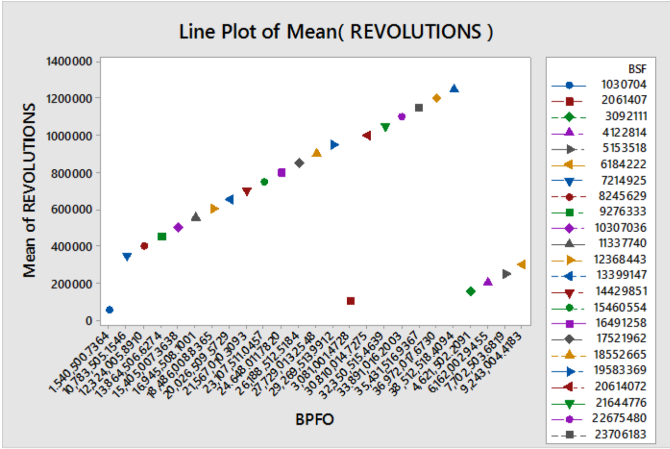
The plot shows deviations from the densely located data thus indicating anomalies in both defect analyzing fault frequencies and therefore possible defects in the exhaust fan bearing. Figure 9 below clearly illustrates the deviations.

FIGURE 12 INTERVAL PLOT OF VELOCITY V/S BPFO



Source: Author

FIGURE 13 INDIVIDUAL VALUE PLOTS VELOCITY AND OUTER RACE (BPFO) VIBRATION FERQUENCIES LARGE DEFECT AT 360^O BEARING CONTACT



Source: Author

Figure 10 shows that more points vary at wider gaps from the densely distributed data indicating more defect possibilities at a contact angle of fan bearing of 360^O as opposed to the distribution in figure 9, which shows less possibility of defects.

The possible causes would be looseness of the fan bearing due to the extent of defect causing variations in the frequency vibrations or a technical error in mounting the fan bearing during lab testing.

THE K-NEAREST NEIGBOUR (KNN) HEALTH ALGORITHM

The KNN classifier requires a metric *d* and a positive integer K. In KNN Euclidean distance computes distances in multidimensional input space. The Euclidean distance between point *p* and *q* is the length of the line between them. In Cartesian coordinates, if *pi* and *qi* are two points in Euclidean *n*-space, then the distance from *p* to *q* is given by $dE = (pi - qi)^2$. Position of training samples and input sample can be visualized on 2D and 3D Cartesian coordinates.

BALL PASS FREQUENCY OUTER RACE BPFO AND INNER RACE BPFI DATA 179^O

VELOCITY	BSF_1	BPFO_1
50000	1030704	2.459,499.2636
100000	2061407	4,918,998.5271
150000	3092111	7,378,497.7907

200000	4122814	9.837,997.0542
250000	5153518	12,297,496.3178
300000	6184222	14,756,995.5813
350000	7214925	17,216,494.8449
400000	8245629	19.675,994.1084
450000	9276333	22,135,493.3720

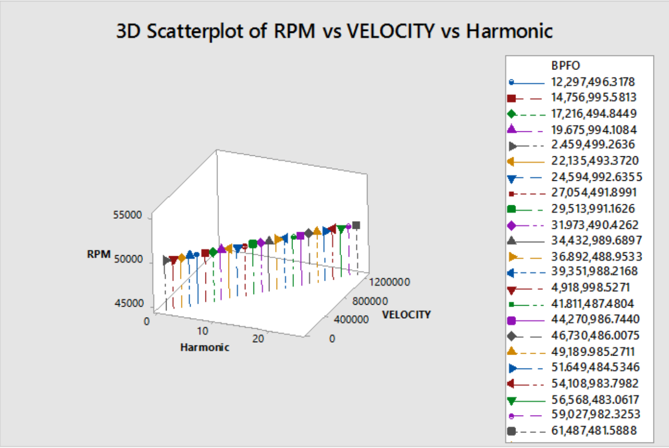
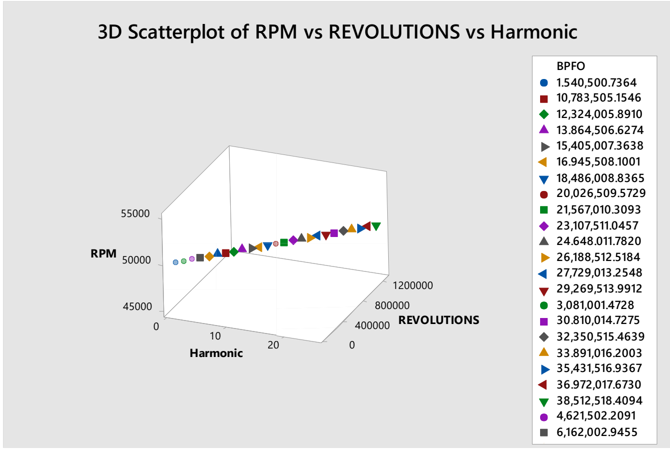


Figure 14 3D SCATTER PLOT OF RPM V/S VELOCITY V/S HARMONICS

Source: Author

BALL PASS FREQUENCY OUTER RACE BPFO AND INNER RACE BPFI DATA 360^{O}

Figure 15 3D SCATTER PLOT OF RPM V/S REVOLUTIONS V/S HARMONICS



Source: Author

BPFO	RPM	REVOLUTIONS
1.540,500.7364	50000	50000
3,081,001.4728	50000	100000
4,621,502.2091	50000	150000
6,162,002.9455	50000	200000
7,702,503.6819	50000	250000
9,243,004.4183	50000	300000
10,783,505.1546	50000	350000
12,324,005.8910	50000	400000
13.864,506.6274	50000	450000
15,405,007.3638	50000	500000
16.945,508.1001	50000	550000
18,486,008.8365	50000	600000
20,026,509.5729	50000	650000
21,567,010.3093	50000	700000
23,107,511.0457	50000	750000
24.648.011.7820	50000	800000
26,188,512.5184	50000	850000
27,729,013.2548	50000	900000
29,269,513.9912	50000	950000
30.810,014.7275	50000	1000000
32,350,515.4639	50000	1050000
33.891,016.2003	50000	1100000
35,431,516.9367	50000	1150000
36.972,017.6730	50000	1200000

The distance between incoming velocities, frequency is compared to the distance between harmonics and revolutions per minute for the two scenarios of Ball Pass frequency Outer BPFO outer race defects.

The distance is greater for the 360 degrees bearing data as compared to the 179.99 degrees frequencies, thus revealing more defects in the 360-degree dimension.

RESEARCH COLLABORATING EVIDENCE

The crux of this research lies in determining defects of exhaust fan bearing from generated vibration frequency. Harmonics data and the calculated velocity from these data assist in determining the level of maintenance needed for the fan bearing and the approximate estimation of the remaining useful life of the bearing by linking the results to the K nearest neighbor health algorithm.

Previous similar research studies exist. Some include studies like Purex canyon exhaust fan bearing temperature monitoring system doric 245 datalogger programming, by Blackbay where a datalogger based on a microprocessor is used to monitor, display and log temperature channels for three electric motor fan assemblies and two bearings monitored on the steam turbine unit [22].

J mathew, in Condition monitoring and fault diagnosis of motor bearings using under sampled vibration signals from a wireless sensor network, conduct similar work to the underlying precepts of this research study [18].

CONCLUSION

The scope of this research study covers the determination of the health and estimation or remaining useful life of exhaust fan bearings using harmonics and frequency vibrations data recorded from the rotating bearing at different speeds and velocities. It is a quest into the strategies, which can be applied to repair and maintain the exhaust fan bearing. This is achieved by evaluating the analysed data using the K nearest neighbor health algorithm.

Vibration frequency analysis is an effective method of ascertaining exhaust fan bearing defects. It is a meticulous technique, which when coupled with the K – nearest neighbor analysis combines to form a formidable force in exhaust fan bearing defect detection through analysis of vibration frequency data and finding of requisite solutions for enhanced bearing health and maintenance.

Felten d, in the paper, understanding vibration frequencies postulates a model which is much more similar to the crux of this research study. Felten discusses the calculation of velocity as a function of multiplying both the ball spin outer race element vibration with the speed or revolutions per minute of the bearing[3].

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APPENDIX

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