RESEARCH ARTICLE OPEN ACCESS

Investigation of Machinery Health Analysing Characteristics of Vibration Using Comprehensive Experimental Setup

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Abstract:

The goal of this paper is to interpret the condition of a machine by investigating the experimental data obtained using CSI 2140 machinery health analyzer from a multi-functioning arrangement. The multi-functioning arrangement consists of pulleys, shafts, ball-bearings, overhung impeller and an electric motor as a power source. These elements generate different forms of vibrational complications. These complications are measured in terms of frequency, amplitude and phase angle and compared with the ISO standard to determine machinery health. The Analyzer helps for the exact determination of the characteristics of the vibration. An experimental setup has been designed and fabricated to create and solve the vibrational complexities. In this research, mass imbalance has been detected and figured out the proper measures such as the polar plot analysis to reduce the severity of vibration and attain the desired level of the vibration according to ISO standard.

Keywords **—Vibration analysis, Mass imbalance, Machinery health, Condition monitoring.**

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I. INTRODUCTION

Condition monitoring (CM)is the process of investigating and analysing the various parameters concerning operational components of a system to identify the pending breakdown [1]-[4]. It is widely known as a predictive or condition-based maintenance technique. Vibration measurement and analysis is considered as one of the key components of the condition monitoring technique. Each component of industrial machinery performs in a certain manner and consumes a certain amount of energy. These components tend to vibrate at specific frequencies [5]-[7]. The measurement of these vibrations in terms of amplitudes and phase angles ascertain numerous aspects of the condition of the system. The analytical data reflect the performance of the components and the entire system which potentially indicates the possibilities of a breakdown or sustainability under the given condition [8], [9]. This paper concerns the fabrication of an arrangement which consists of different elements which are widely used in industrial purposes. These are often subjected to vibrations, the identification and measurement of such vibrations in terms of amplitude and phase angle [10], [11]. By analysing the obtained data to detect the potential fault present in the system, comparing the data to the ISO levels and ultimately mitigating the intensity of vibration by using dampers to achieve satisfactory levels of vibration to ensure a better operational condition of the system [12].

II. THEORETICAL DESCRIPTIONS

The characteristics of vibration data and the form of the spectral value indicates certain faults present in a system. Each of the faults present in the system generates a certain trend of displayed spectral data which is highly useful to detect the problem and take necessary measurements to reduce the generated vibration.

Mechanical systems, especially rotating machinery are widely subjected to mechanical imbalance. The vibration is generated when the centre of mass of the rotating element is not turning on the same axis as the rotating assembly. The amplitude will increase with an increase in speed up to the first critical speed of the rotating element. The spectra generated will display a single frequency vibration whose amplitude is the same in all radial directions. In a pure imbalance, it will be a perfect sinusoidal vibration at the machine running speed. In case of an overhung rotor, an axial component is observed.A vibration analysis system usually consists of four basic parts: a transducer, a signal analyser, analysis software and computer for data analysis and storage. These basic parts can be configured to form a continuous online system, a periodic analysis system using portable equipment that samples a series of transducers at predetermined time intervals.

Mechanical vibration is the measurement of a periodic process of oscillations with respect to an equilibrium point. The vibration amplitude is commonly expressed in one of three units of measure – displacement (mils or microns), velocity (inches per second (ips) or mm/s), and acceleration (ips^2 or mm/s^2). Each type of measurement is used for a specific purpose.The difference between the signal average and the maximum absolute value is called peak value.The root means squares value (RMS)is an indication for the power contained in the signal, or in other words, the effective value. Therefore, it is commonly used in vibration level detection. In case of a set of n values$\{X_1, X_2,.....X_n\}$,

$$Xrms = \sqrt{\left\{\frac{1}{n}(X_1^2 + X_2^2 \dots\dots\dots\dots + X_n^2)\right\}}$$

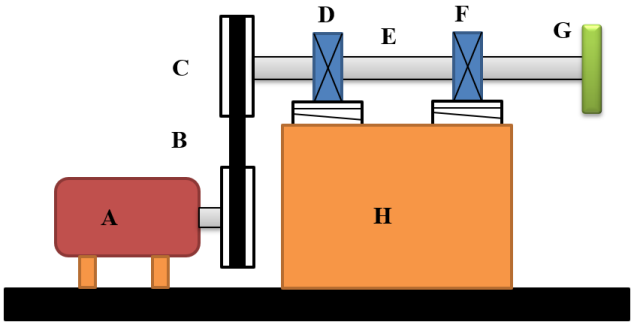
Time waveform is simply displaying the signal in the same manner as the oscilloscope plot. It is the amplitude-time plot. The most common use of time waveform data is to compare the waveform pattern of one machine with another obtained from a machine with similar defects. If necessary, the frequency components of the major events in the waveform pattern can be calculated.

Fourier theorem states that any time waveform can be reconstructed from several harmonically related sine and cosine frequency components. Fourier transform is found to be a very efficient and useful tool to analyse vibration signals and to detect most of the common vibration problems. Fourier spectrum is simply the amplitude-frequency plot and can be done through different techniques.The polar or Nyquist plot is also a representation of the same three variables as considered in a Bode plot. The variables are plotted on a single circular chart instead of Cartesian axes. The overhung rotor is the dominant source of the mass imbalance generated in the system due to its non-uniform geometry. For unbalance of overhung rotors, the FFT spectrum displays a single 1X RPM peak, and the amplitude again varies proportionally to the square of the shaft speed. It may cause high axial and radial vibrations. The axial phase on the bearings will seem to be in phase whereas the radial phase tends to be unsteady. Overhung rotors can have both static and couple unbalance and must be tested and fixed using analysers or balancing equipment.

III. EXPERIMENTAL SETUP

The representation of the schematic diagram of the experimental setup is shown in Fig.1.The experimental setup consists of a single-phase induction motor of 0.25hpwith RPM of 1400 works as the power source of the experimental setup. The motor shaft is connected to a set of pulleys with 100 mm and 75 mm diameters respectively with a V-belt of 16 mm of width and 560 mm length. A mild steel shaft of 17mm diameter is connected to the upper pulley whose axis is at a height of 165 mm from the axis of the motor shaft. Two bearings housings which house ball bearings and support the shaft at a distance of 143 mm. The bearing houses are bolted onto a supporting block made of mild steel with a width of 222 mm and length of 254 mm. The shaft extends to a length of 156 mm from the outermost bearing housing. At the end of the shaft, an overhung impeller is attached. The impeller is of

127 mm diameter and 8 mm thickness with equidistant 8 holes residing 51 mm from the centre.The representation of close-up view with transducer of experimental setup is shown in Fig. 2



A: Electric motor, B: V belt, C: Pulley, D: Bearing housing, E: Shaft, F: Bearing housing, G: Over hung impeller, H: Supporting block

Fig. 1 Schematic diagram of experimental setup

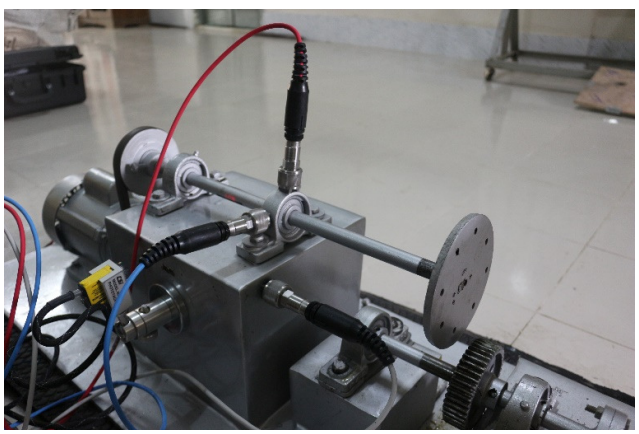


Fig. 2 Close-up view with transducer of experimental setup

IV. METHODOLOGY

The three transducers labelled as A, B and C of the CSI 2140 machinery health analyser were obtained for vertical, horizontal and axial directions respectively. The configuration of the analyser was shown in Table I.

TABLE I
THE CONFIGURATION OF THE ANALYSER

Parameter	Specifications
Settings	Spectra Parameters
Maximum Frequency	10 X actual CPM
Maximum Frequency	0
Lines	800
Resolution	Default
Windows	Hanning
Sensitivity	100mV/G
Sensor setup	Single axis accel.

The portions of the setup, which were subjected to the vibration measurement, are motor outboard, motor inboard, first bearing and second bearing. After mounting the transducers on each of the locations with a separation of 90 degree with each other, the power switch of the motor was turned on. The spectral and waveform data were displayed on the analyser for each of the components. The peak and phase plot also obtained for the second bearing via the tachometer provided with the device. Then, the obtained values were compared with the ISO standard for different levels of vibrations, as provided in the Fig. 3. The trend of the spectral plot indicated the problem that existed in the system. Afterwards, the problem was detected as mass unbalance and measures were taken accordingly. The reduced vibration data were obtained in the same procedure by the device and compared again with the ISO standards for measuring the level of vibration that obtained. After attaining the suitable level, the system apparently generated a far lesser degree of vibration in all three directions.

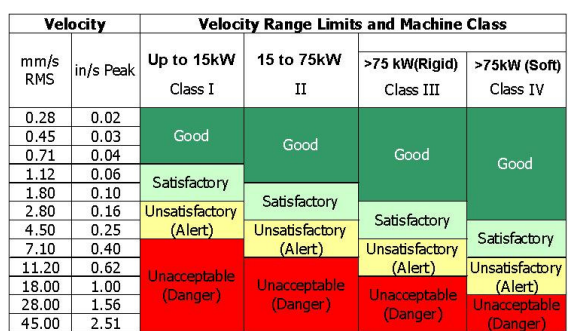


Fig. 3 ISO standard chart

V. EXPERIMENTAL DATA ANALYSIS

At first, the transducers were attached to the specific locations of the second bearing to measure

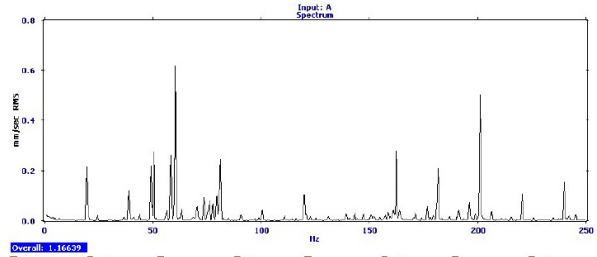
the amplitude of vibration in all three directions while the system remained unbalanced. The values obtained for the second bearing in Fig. 5 extensively exceeded the ISO standards specified to good or satisfactory levels of performance. This was observed by comparing the highlighted data of the amplitudes obtained from the resonant frequencies in the plots with the ISO standard chart. Moreover, the peak and phase value of the second bearing from Fig. 7 depicts that the highest peak RMS value is found at 52.2^0 phase angle in input B (vertical direction). This critical RMS value is responsible for generating the vibration due to mass imbalance. This peak value should be minimized through adding mass.

A. Experimental Data for Unbalanced System

The representation of Frequency and amplitude for input A is shown in Fig. 4, Frequency and amplitude for input B is shown in Fig. 5, Frequency and amplitude for input C is shown in Fig. 6 and Peak and phase Data for the second bearing is shown in Fig. 7 for unbalance condition.

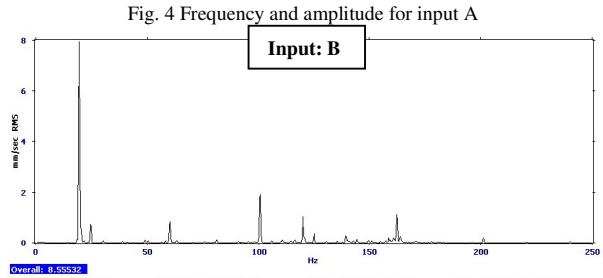
B. Comparison of obtained values with ISO standards and Detection of fault

The motor power which was used in this experiment is 180 W. As per ISO standard table, this should be acceptable for Class I cause of its power is less than 15 KW. To attain the satisfactory level, the RMS value should not cross 1.80 mm/s. Now experimental data will be compared ISO standard. It is seemed that RMS value of transducer input B is 7.3686 mm/s that should be satisfactory level of ISO standard. Moreover, the behaviourof data referred to mass imbalance. This was detected as the displayed plot had a form of 1X which is an indication of mass imbalance. The mass imbalance had to occur in the overhung impeller due to design faults and its centre of gravity not being on the axis of rotation. It is clear that mass unbalancing issue is responsible for this peak point.



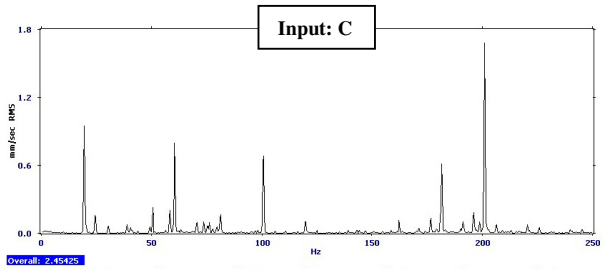
Frequency Data		Frequency Data		Frequency Data		Frequency Data	
19.35	0.2157	38.7	0.12301	49.01	0.22141	50.36	0.278
58.05	0.27023	60.38	0.63296	70.48	0.06468	73.52	0.09718
75.98	0.08159	77.4	0.07259	79.55	0.11266	81.14	0.26325
119.84	0.12166	162.28	0.29395	176.48	0.05893	181.64	0.21375
195.83	0.07728	200.99	0.50741	220.34	0.10643	239.69	0.15495

Fig. 4 Frequency and amplitude for input A



Frequency Data		Frequency Data		Frequency Data		Frequency Data	
19.35	7.93968	24.51	0.80479	49.01	0.11386	60.2	0.9058
63.22	0.10061	81.14	0.13426	100.49	2.15148	110.67	0.11226
116.11	0.11529	119.84	1.20512	125.0	0.35487	139.19	0.31304
139.91	0.10314	144.35	0.13815	158.55	0.21356	159.31	0.12215
161.0	0.20385	162.28	1.19657	163.84	0.2695	200.98	0.2028

Fig. 5 Frequency and amplitude for input B



Frequency Data		Frequency Data		Frequency Data		Frequency Data	
19.35	0.9471	24.51	0.17731	50.36	0.23027	58.05	0.2116
60.39	0.82249	70.48	0.10673	73.52	0.10393	75.16	0.07587
75.98	0.09464	81.14	0.17769	100.49	0.7671	119.84	0.11992
162.28	0.11926	176.48	0.13525	181.64	0.63075	191.18	0.10626
195.84	0.19452	198.52	0.10329	200.99	1.70348	206.14	0.08267

Fig. 6 Frequency and amplitude for input C

C. Reduction of Flaws of the Arrangement

As the problem was detected as mass imbalance in the overhung impeller, by the polar plot analysis a corrected mass was used to balance the rotor and reduce the intensity of vibration to a satisfactory level. So, this value is primarily taken to consider the issue and polar graph paper is used to reveal the unbalance issue.

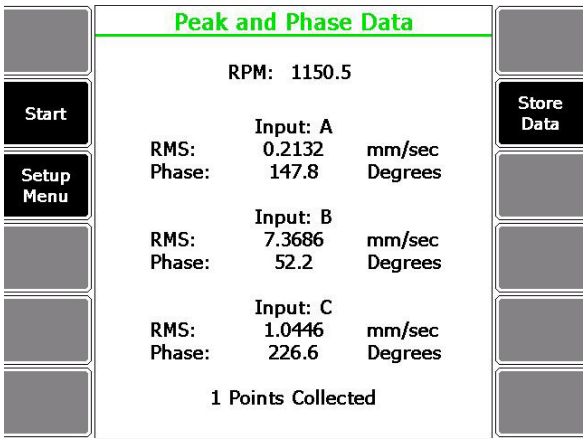


Fig. 7Peak and phase Data for the second bearing (Unbalanced condition)

Figure 8 shows the polar plot for reducing mass imbalance where outer numbers of the circle are considered as the shaft rotates counter clock wise direction. Phase angle increases against rotation of the shaft as well the outer ring of polar graph. The radial distance from centre to outer ring is considered as 10 unit. The peak RMS value found from the Fig. 7 is 7.37 at 52.2^0 phase angle in vertical direction is labelled in polar graph as vector 1.A trial weight of 4.1 gram is added at 180 degree of polar graph paper. After adding a trial weight at 180^0, it is shown that the peak RMS value got decreased in three direction. However, still then the value in vertical direction is high and exceeds the desired range. The peak RMS value found from the Fig. 9 is 4.75 at 27.1^0 phase angle in vertical direction is labelled in polar graph as vector 2. A new vector is drawn from head of vector 1 to head of vector 2 and labelled it as vector 3. A vector 4 is drawn starting at the origin of polar plot and parallel to vector 3. The angle between the vector 4 and the extension of vector1 is measured and found it as 39^0. This is the same angle that the correct weight goes with respect to trial weight. Therefore, the correct weight location is fixed at 219^0 of polar graph. The correct weight is calculated using the following formula.

$$\text{Correct weight} = \frac{\text{Length of vector 1} \times \text{Trial weight}}{\text{Length of vector 3}}$$

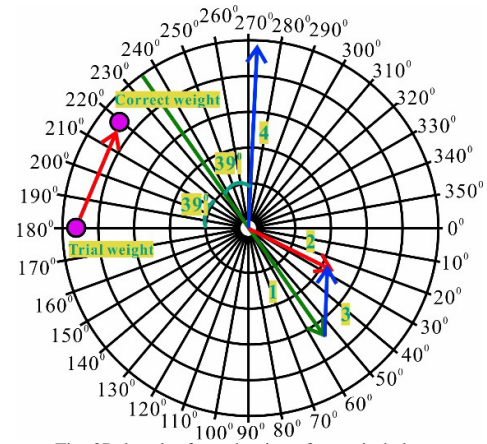


Fig. 8Polar plot for reduction of mass imbalance

The length is measured with a fine scale ruler and weight by a digital weighing scale. The measurements are as follows-

The length of vector 1 is 6cm
The trial weight is 4.1gram
The length of vector 3 is 3 cm
So, the calculated correct weight required for balancing is,

$$\text{Correct weight} = \frac{6\text{ cm} \times 4.1\text{ gm}}{3\text{ cm}} = 8.2\ gram$$

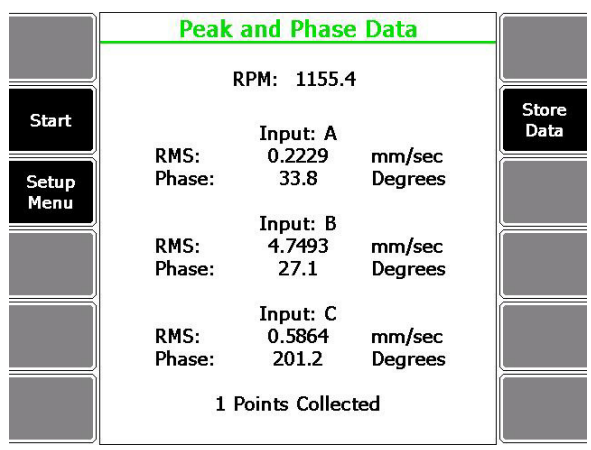


Fig. 9Peak and phase data for the second bearing (Trial).

D. Experimental Data for Balanced System

The representation of Peak and phase Data for the second bearing (Balanced condition) is shown in

Fig. 10. The RMS values in all three directions are decreased after placing the correct weight of 8.2 gram at 219^0. In this stage, the spectrum waveform is taken from the experimental setup. The obtained spectral plots depict that the amplitude of vibration reduced in all directions, significantly in the vertical direction as displayed in Fig. 5.By attaching the correct weight at the calculated phase angle by the polar plot analysis,the intensities of vibration are far less than the ones obtained prior to balancing. The representation of frequency and amplitude for input A is shown in Fig.11. The representation of frequency and amplitude for input B is shown in Fig.12. The representation of frequency and amplitude for input C is shown in Fig.13.

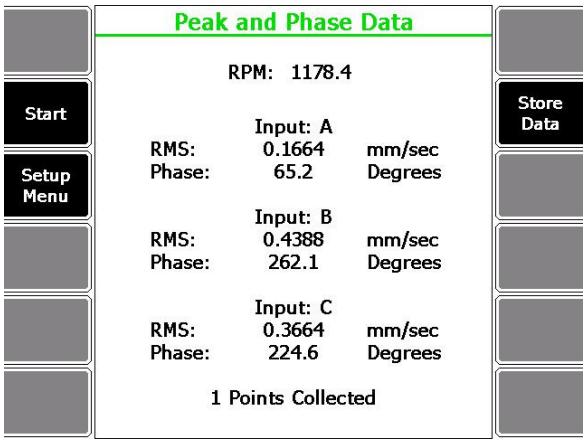
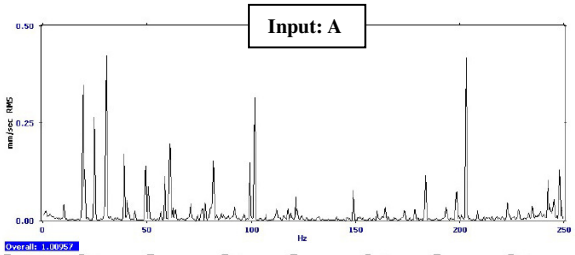


Fig. 10Peak and phase Data for the second bearing (Balanced condition)

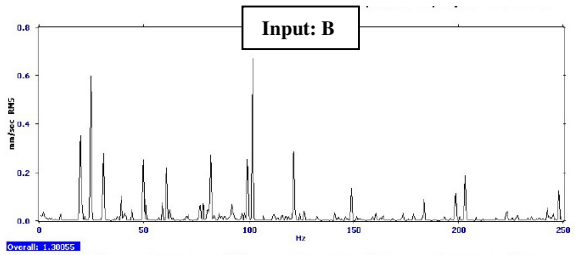
VI. RESULTS AND DISCUSSION

This experiment was aimed to achieve a stable system depending on ISO standard from an unstable system. The main target was to keep the peak value of the system below of 1.8 mm/s. Mass unbalancing issue was detected after observing the characteristics of vibration. Polar graph paper is used here to balance the system with adding mass. Finally, 8.2gram mass has been added at 219^0 of circular overhung impeller. It was found that the system was stable within the satisfactory level of ISO standard. The system can be more stable if some changes in setup foundation would be done. Hence the system is designed only for research only, so permanent fixed foundation is not possible. The mass balancing procedure has been experimented by trial and error method.Two things are very important in this experiment like precision and skill. So precisely mass calculation and addition to exact location were very critical issues. After several attempts, finally more precision solution is introduced in this experiment.



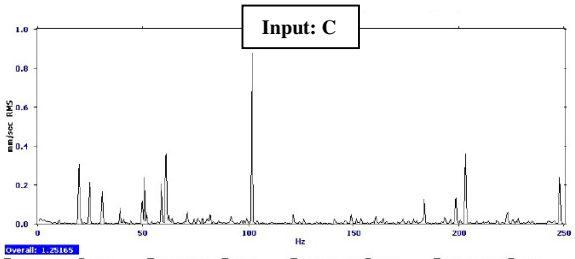
Frequency Data		Frequency Data		Frequency Data		Frequency Data	
19.57	0.38237	20.25	0.13519	24.79	0.28266	30.55	0.43734
39.13	0.17358	49.58	0.1518	50.93	0.08671	58.7	0.1163
61.07	0.22111	81.98	0.16307	99.16	0.15766	101.54	0.31605
121.11	0.06959	148.73	0.07638	183.52	0.1211	198.29	0.08464
203.08	0.42124	242.23	0.1038	244.9	0.05783	247.9	0.13644

Fig. 11Frequency and amplitude for input A



Frequency Data		Frequency Data		Frequency Data		Frequency Data	
19.56	0.3899	24.79	0.63785	30.55	0.28861	39.13	0.10685
49.58	0.27345	50.92	0.08836	58.7	0.07667	60.84	0.23125
76.73	0.07425	78.26	0.07868	81.98	0.29121	91.67	0.07164
99.16	0.26954	101.54	0.67069	121.11	0.32937	148.74	0.13539
183.52	0.09429	198.32	0.12446	203.08	0.19126	247.9	0.12968

Fig.12 Frequency and amplitude for input B



Frequency Data		Frequency Data		Frequency Data		Frequency Data	
19.57	0.33675	24.79	0.22936	30.55	0.16949	39.13	0.0835
49.58	0.13014	50.92	0.23686	51.96	0.0461	58.69	0.20967
61.1	0.41236	62.4	0.04332	71.19	0.06319	81.97	0.05295
101.54	0.87865	121.11	0.05776	148.73	0.0484	183.52	0.13098
198.31	0.14843	203.08	0.36037	222.99	0.06706	247.9	0.25009

Fig.13 Frequency and amplitude for input C

VII. CONCLUSIONS

Mechanical systems are widely subjected to vibrations, especially in the industrial sectors. The arrangement undertaken generated the similar form of vibration in numerous mechanical machineries. The dominant form of vibration in the system was due to mass imbalance which was accurately determined by the characteristics of the spectral plot obtained from the machinery health analyser's display. The process of mitigation of this problem required adequate steps to employ specific weight at specific location of the element responsible for the high amplitude of vibration. The spectral plots along with the peak and phase values obtained from the analyser provided the required value to facilitate the balancing of the system by the polar plot analysis procedure. This ultimately mitigated the intensity of vibration to a satisfactory level according to the standard ISO level chart. The system was effectively balanced after the accurate measurement, detection and reduction of the vibration-oriented flaws that prevailed in it prior to balancing the system to the desired level.

ACKNOWLEDGMENT

I would like to express my gratitude and heartiest thanks to PranRfl Ltd. for their support. I also grateful to Saj Engineering & Trading Company for their restless technical support and guidance.

REFERENCES

[1] G. Diwakar, M. R. S. Satyanarayana and P. R. Kumar, "Detection of gear fault using vibration analysis", *International Journal of Emerging Technology and Advanced Engineering*, ISSN 2250-2459, vol. 2, issue 9, 2012.

[2] A. V. Dube, L.S.Dhamande and P.G.Kulkarni, "Vibration based condition assessment of rollingelement bearings with localized defects", *International Journal of Scientific and Technology Research*, vol. 2, issue 4, Apr 2013.

[3] M. Natu, "Bearing fault analysis using frequency analysis and wavelet analysis", *International Journal of Innovation, Management and Technology,* vol. 4, issue 1, Feb 2013.

[4] P. Durkhure and A. Lodwal, "Fault diagnosis of ball bearing using time domain analysis and fast fourier transformation", *International Journal of Engineering Sciences & Research Technology*, vol. 3, pp. 711-715, Jul 2014.

[5] L. Pratyusha, S. Priya and V. P. S. Naidu, "Bearing health condition monitoring: time domain analysis", *International Journal of Advanced Research in Electrical, Electronics and Instrumentation Engineering,* ISSN 2320 – 3765, vol. 3, issue 5, Dec 2014.

[6] P. B. Sonawane and N.K.Kharate, "Fault Diagnosis of Windmill by FFT Analyzer" *International Journal of Innovations in Engineering and Technology*, vol. 4, issue 4, Dec 2014.

[7] G. S. Babu and V. C. Das, "Condition monitoring and vibration analysis of boiler feed pump", *International Journal of Scientific and Research Publications*, vol. 3, issue 6, Jun 2013.

[8] S. S. Kumar and M. S. Kumar "Condition monitoring of rotating machine through vibration analysis", *Journal of Scientific And Industrial Research,* vol. 73, pp. 258-261, Apr 2014.

[9] N. Dileep, K. Anusha, C. Satyaprathik, B. Kartheek and K.Ravikumar, "Condition monitoring of FD-fan using vibration analysis", *International Journal of Emerging Technology and Advanced Engineering,* vol. 3, issue 1, Jan 2013.

[10] V. J. Suryawanshia, "Vibration based condition assessment of rotating cracked shaft using changes in critical speed and rms velocity response functions", *International Journal of Current Engineering and Technology*, ISSN 2277 – 4106, pp. 170-174.

[11] A. Vaziri and M. J. Patil "Vibration analysis of a cracked shaft", *International Journal of Advanced Engineering Technology,* E-ISSN 0976-3945.

[12] A. Tlaisi1, A. Akinturk1, A. S. J. Swamidas and M. R. Haddara, "*Crack detection in shaft using lateral and torsional vibration measurements and analyses*", Canadian Center of Science and Education, vol. 2, issue 2, pp. 52-75,2012.

