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ANALYSIS OF MACHINERY HEALTH CONSIDERING THE PARAMETERS OF VIBRATION IN A MULTI-FUNCTIONING ARRANGEMENT

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Abstract-*This paper aims to interpret the condition of a machine by analyzing the experimental data obtained using CSI 2140 machinery health analyzer from a multi-functioning arrangement. The multi-functioning arrangement consists of different sets of pulleys, shafts, ball-bearings, overhung impeller and an electric motor as 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 the range of the vibration generated in the system. The Analyzer helps for the exact determination of the nature of the vibrational complexities, especially, the mass imbalance present in the system, which facilitates the provision of adequate measures such as the polar plot analysis to reduce the severity of vibration and attain the desired level of the vibration.*

Keywords: Vibration analysis, mass imbalance, polar plot, machinery health, condition monitoring.

1. INTRODUCTION

Condition monitoring concerns the aspects of investigating and analyzing the numerous parameters concerning these operational components of a system in order to identify the pending breakdown [1][2]. It is widely known as predictive or condition-based maintenance technique. Vibration measurement and analysis is considered as one of the key components of condition monitoring technique. Each component of industrial machineries performs in a certain manner and consumes a certain amount of energy. These components tend to vibrate at specific frequencies [5]. 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 as a whole which potentially indicates the possibilities of a breakdown or sustainability under the given condition [8]. 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 [6]. By analyzing 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 [10].

2. THEORETICAL DESCRIPTIONS

The trend 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 in order to detect the problem and take necessary measurements to reduce the generated vibration.

Mechanical systems, specially rotating machineries are widely subjected to mechanical imbalance. The vibration is generated when the center 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 analyzer, 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.

The determination of which configuration would be more practical and suitable depends on the critical nature of the equipment, and also on the importance of continuous or semi-continuous measurement data for that particular

application.
Mechanical vibration: 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.
Peak Value: The difference between the signal average and maximum absolute value.
The root mean squares value (RMS): It is an indication for the power content 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 }, the RMS

$$x_{rms} = \sqrt{\{ \frac{1}{n}(x_1^2 + x_2^2 + ... + x_n^2)\}}$$

Time Waveform: 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 a numbers of harmonically related sine and cosine frequency components. Fourier transform is found to be very efficient and useful tool to analyze 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 proportional 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 analyzers or balancing equipment. The representation of schematic diagram of experimental setup is shown in Fig.1

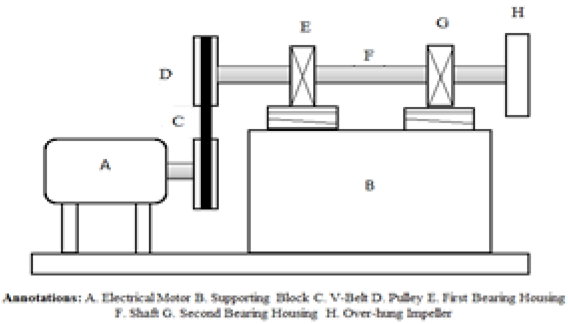


Fig. 1: Schematic diagram of experimental setup.

3. EXPERIMENTAL SETUP

The experimental setup consists of the 0.25hp a single-phase induction motor and an RPM of 1400 works as the power source of the arrangement. 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 center.

4. METHODOLOGY

The three transducers labeled as A, B and C of the CSI 2140 machinery health analyzer were obtained for vertical, horizontal and axial directions respectively. The configuration of the analyzer was shown in Table 1.

Table 1: The configuration of the analyzer

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 aforementioned locations with a separation of 90 degree with each other the power switch of the motor was turned on. The following spectral and waveform data were displayed on the analyzer for each of the components. The peak and phase plot also obtained for the second bearing via the tachometer provided with the device. The representation of close-up view with transducer of experimental setup is shown in Fig. 2

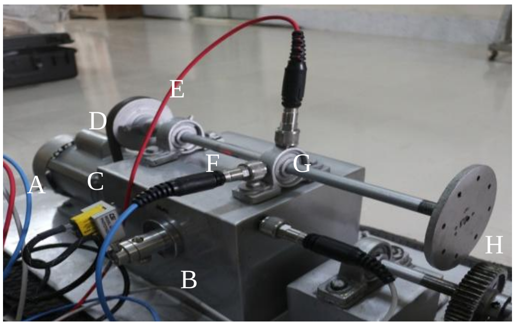


Fig. 2: Close-up view with transducer of experimental setup.

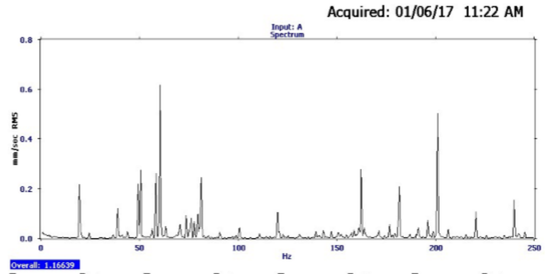
Velocity		Velocity Range Limits and Machine Class			
mm/s RMS	in/s Peak	Up to 15kW Class I	15 to 75kW II	>75 kW(Rigid) Class III	>75kW (Soft) Class IV
0.28	0.02	Good	Good	Good	Good
0.45	0.03				
0.71	0.04				
1.12	0.06	Satisfactory			
1.80	0.10		Satisfactory		
2.80	0.16	Unsatisfactory (Alert)		Satisfactory	
4.50	0.25		Unsatisfactory (Alert)		Satisfactory
7.10	0.40	Unacceptable (Danger)		Unsatisfactory (Alert)	
11.20	0.62		Unacceptable (Danger)		Unsatisfactory (Alert)
18.00	1.00			Unacceptable (Danger)	
28.00	1.56				Unacceptable (Danger)
45.00	2.51				

Fig. 3: ISO standard chart for different levels of vibrations

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. The representation of ISO standard chart for different levels of Vibrations is shown in Fig. 3.

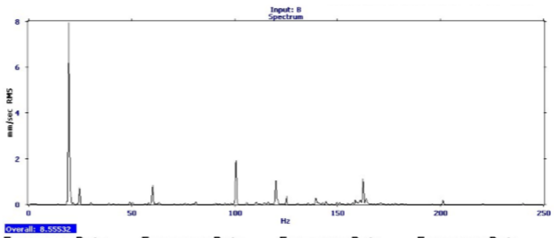
5. EXPERIMENTAL DATA ANALYSIS

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. Then, the obtained data were compared with the ISO standard chart provided in Fig. 3. The detection of the problem that persisted in the system



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



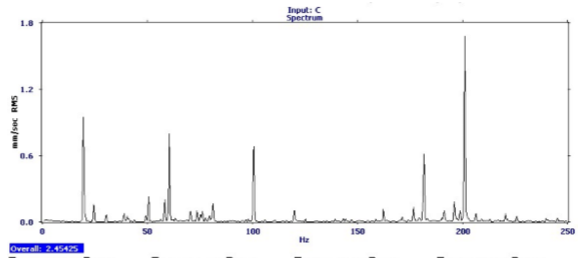
Frequency	Data	Frequency	Data	Frequency	Data	Frequency	Data
19.35	7.93968	24.51	0.80479	49.01	0.11386	60.2	0.9058
63.22	0.10061	81.14	0.13426	100.49	2.15148	110.67	0.11226
116.11	0.11529	119.84	1.20512	125.0	0.35487	139.19	0.31304
139.91	0.10314	144.35	0.13815	158.55	0.21356	159.31	0.12215
161.0	0.20385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

Fig. 5: Frequency and amplitude for input B

was observed too. Then after proper measures to reduce the intensity of vibration due to the problem, the transducers were set up at the same positions and the data were taken for the balanced system. These data were again compared with ISO standard chart. 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 above Fig.3 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.

5.1. Experimental Data for Unbalanced System

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 above 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**.** 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.



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

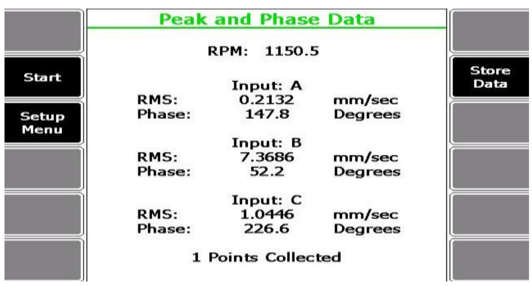


Fig. 7: Peak and phase Data for the second bearing(Unbalanced condition)

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

The motor power which is 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. By comparing this data with primary data that found from experiment, it is seemed that in transducer B's RMS value 7.3686 mm/s should be this range. Moreover, the trend in 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 center of gravity not being on the axis of rotation. It is clear that mass unbalancing issue is responsible for this peak point.

5.3. Reduction of Flaws of the Arrangement

As the problem encountered 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. The peak and phase data in Fig. 7 referred to reason of such a high level of vibration. So this value is primarily taken to consider the issue and polar graph paper is used to reveal the unbalance issue.

From Fig. 8, it is shown that 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 above polar graph. The radial distance from center to outer ring is considered as 10 unit. The peak RMS value found from the Fig.6 is 7.37 at 52.2^0 phase angle in vertical direction is labeled 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. The representation of Polar plot for reduction of mass imbalance is shown in Fig. 8.However, still then the value in vertical direction is high and exceeds the desired range. The peak RMS value found from the Fig.7 is 4.75 at 27.1^0 phase angle in vertical direction is labeled in polar graph as vector 2. A new vector is drawn from head of vector 1 to head of vector 2 and labeled it as vector 3. A vector 4 is drawn starting at the origin 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.

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

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 was 6cm

The trial weight was 4.1gram

The length of vector 3 was 3 cm

So, the calculated correct weight required for balancing is,

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

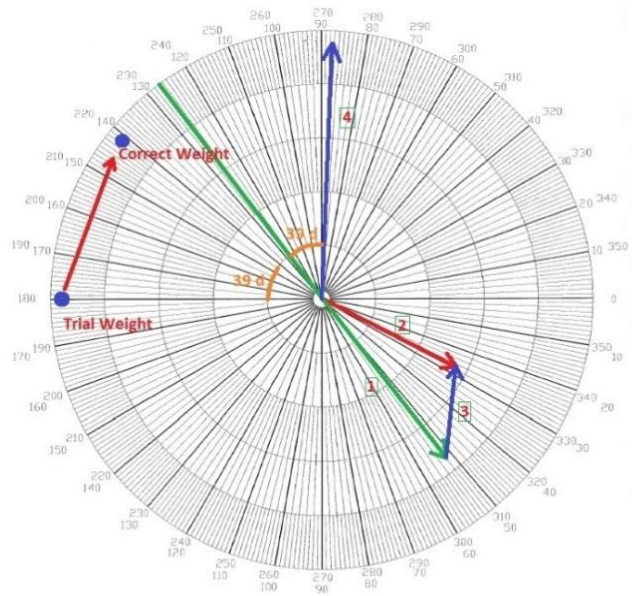


Fig. 8: Polar plot for reduction of mass imbalance

5.4. Experimental Data for Balanced System

The RMS values in all three directions are decreased in Fig. 5 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 following plots were obtained by the same process afterwards attaching the mass. 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 is far less than the ones obtained prior to balancing. The obtained peak and phase data are provided in Fig. 9. The representation of Peak and phase Data for the second bearing (Balanced condition) is shown in Fig.9. The representation of Frequency and amplitude for input A is shown in Fig.10(a). The representation of Frequency and amplitude for input B is shown in Fig.10(b). The representation of Frequency and amplitude for input C is shown in Fig.10(c)

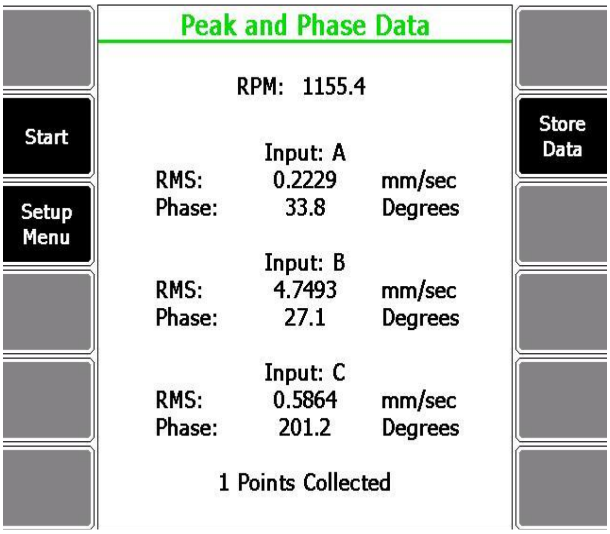


Fig. 9: Peak and phase data for the second bearing (Trial)

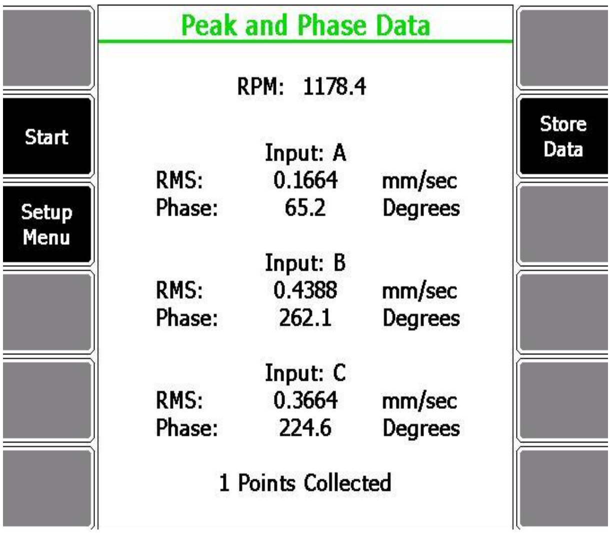
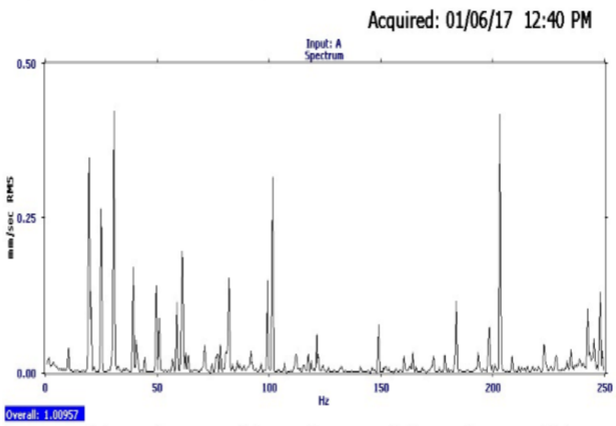
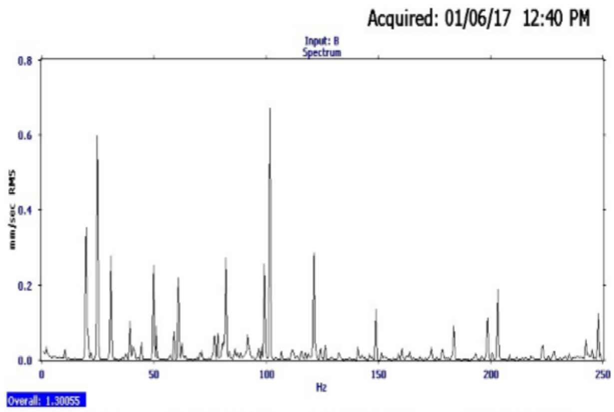


Fig. 10: Peak and phase Data for the second bearing (Balanced condition)



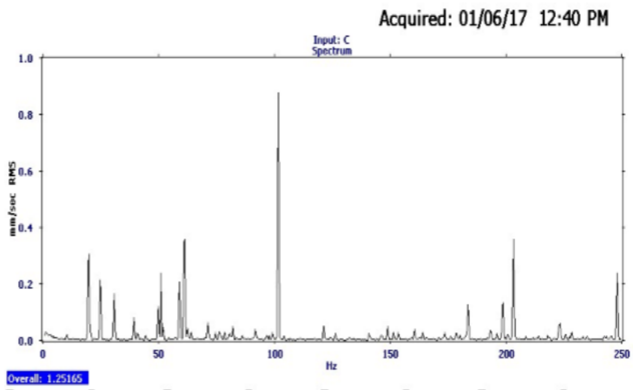
Frequency Data		Frequency Data		Frequency Data		Frequency Data	
19.57	0.38237	20.25	0.13519	24.79	0.28266	30.55	0.43734
39.13	0.17358	49.58	0.1518	50.93	0.08671	58.7	0.1163
61.07	0.22111	81.98	0.16307	99.16	0.15766	101.54	0.31605
121.11	0.06959	148.73	0.07638	183.52	0.1211	198.29	0.08464
203.08	0.42124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

Fig.11: Frequency 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

6. 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 signature of vibration. Polar graph paper is used here to balance the system with adding mass. Finally 8.2 gram 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.

7. 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 trend of the spectral plot obtained from the machinery health analyzer's display. The 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 analyzer 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.

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