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Diagnosing the Effect of Misalignment on a Rotating System using Simulation and Experimental Study

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Keyword

Diagnosis, FFT Analyzer, Parallel Misalignment Vibration Spectrum, Ansys software.

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Abstract:

Misalignment is one of the common causes of machine vibration. Understanding and practicing the fundamentals of rotating shaft parameters is the first step in reducing unnecessary vibration, reducing maintenance costs, and increasing machine uptime. In the industrial setting, misaligned machines account for 50% of all machine downtime. The most frequent issue with rotating machinery that affects every industry is rotor shaft misalignment. Consequently, misalignment defects can be qualitatively identified using condition monitoring based on vibration measurements. The present study employs vibration measurement method by Building a model that contains suggested defects and then using it in advanced simulation programs such as Ansys. and Validation of the results by comparing them with the results of experimental methods Validation results show that both numerical and experimental data are in good match regarding amplitude and frequency. The verification results proved that the frequencies that were extracted by using finite element techniques (simulation reading) agree with the frequencies that were extracted by the experimental (experimental reading) (100%).

Keywords: Diagnosis, FFT Analyzer, Parallel Misalignment, Vibration Spectrum, Ansys software.

تشخيص تأثير المحاذاة الخاطئة على نظام دوار باستخدام المحاكاة والدراسة التجريبية

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الخلاصة:

يعد الاختلال في المحاذاة أحد الأسباب الشائعة لاهتزاز الماكينة. يعد فهم وممارسة أساسيات معلمات عمود الدوران الخطوة الأولى في تقليل الاهتزاز غير الضروري وتقليل تكاليف الصيانة وزيادة وقت تشغيل الماكينة. في البيئة الصناعية ، تمثل الآلات المنحرفة ٥٠٪ من كل وقت تعطل الماكينة. المشكلة الأكثر شيوعًا في الآلات الدوارة التي تؤثر على كل صناعة هي اختلال محاذاة عمود الدوران. وبالتالي ، يمكن التعرف على عيوب المحاذاة الخاطئة نوعياً باستخدام مراقبة الحالة بناءً على قياس الاهتزاز. تستخدم الدراسة الحالية طريقة قياس الاهتزاز من خلال بناء نموذج يحتوي على عيوب مقترحة ثم استخدامه في برامج المحاكاة المتقدمة مثل ANSYS. والتحقق من صحة النتائج من خلال مقارنتها مع نتائج الطرق التجريبية. أظهرت نتائج التحقق أن كل من البيانات العددية والتجريبية متطابقة بشكل جيد من حيث السعة والتردد. كما أثبتت نتائج التحقق أيضا" أن الترددات التي تم استخلاصها باستخدام تقنيات العناصر المحدودة (قراءة المحاكاة) تتفق مع الترددات التي تم استخلاصها بالقراءة التجريبية بنسبة ١٠٠٪.

الكلمات المفتاحية: تشخيص العيوب، محلل FFT ، اختلال متوازي، طيف الاهتزاز، برنامج محاكاة متقدم.

1. INTRODUCTION:

Rotor systems have been widely used in mechanical engineering. The dynamics of rotor systems have been studied for over a century. With the high-speed demand of today's machinery, it becomes more important than ever. Misalignment is one of the predominant failures of rotating machines driven by induction motors, this leads to economic losses. Also misaligned machinery is more prone to failure due to increased load on bearings, seals, and couplings. Shaft misalignment has major implications for modern day rotating equipment's reliability. Even though many efficient alignment methods have been put into place, alignment nevertheless may deteriorate because of changes in equipment operating conditions. Vibration issues have become more complicated due to the increasing sophistication of rotating machinery. In extreme circumstances presence of shaft misalignment can greatly influence machinery vibration response. All devices, such as motors and turbines, develop unique effects which can be analyzed to improve the design and decrease the possibility of failure [2,3].

Therefore, engineering judgments based on understanding of physical phenomenon are essential to provide diagnosis and method for correcting such faults. Anecdotal evidence suggests that as much as 50% of machine breakdown can be directly attributed to incorrect shaft alignment [1]. A large payback is always seen by regularly aligning the machine. Therefore, machine's operating life increases and as a result, process condition optimization takes place. There is widespread promotion of vibration signatures as a useful tool for studying machine malfunctions. However, the literature on this topic does not provide a clear picture of signature characteristics attributable exclusively to misalignment. Different authors report different signatures for misalignments using different types of couplings. Khot and Khaire [1] studied the effect of misalignment with flexible flange connections with the help of an FFT analyzer to obtain the frequency spectrum using the experimental setup that was developed for their work. The study was done by experimental work distortion work (displacement by an amount of 1.5mm). Between the two rotating shafts. Then simulate the rotary system using the Ansys program. within a specified speed range of (250 rpm to 1500 rpm). The vibration spectrum is gathered by modeling and testing with various stimulation frequencies, and it is observed that they are tightly connected. The results revealed that in second harmonics, or for parallel misalignment, the shaft should operate at 2X its normal speed, and for angular misalignment, at 1X its normal speed. Mohanty [2] presented a work that, the resulting vibration was studied by the presence of two types of problems in the mechanical system prepared for this purpose (unbalance and misalignment). The theoretical part was based on a proposed system consisting of equations that were used in calculating the forces in the three directions (X, Y, Z) resulting from these faults, and the results were identical in both cases. The method may be useful for large systems such as turbine shafts, gearboxes, and the like. Dere and Dhamande [3], proposed a work that uses misalignment of a rotating shaft by using the Ansys program, and good results were obtained by comparing them with the experimental side, and the results obtained by the researcher. Overall vibrational magnitude, as measured by root-mean-square acceleration values, is found to be greater at bearing support-2 (the point away from the motor) than at bearing support-1 (the point near the motor) with respect to both the horizontal and vertical axes for a variety of misalignment conditions, Including parallel or angular misalignment, or both. Babar and Utpat [4], to forecast the vibration spectrum of shaft misalignment, a rotary bearing system was analyzed in their work. An accelerometer and the Fast Fourier Transform (FFT) Analyzers were both employed. To measuring vibration. By modeling the rotating system and identifying the frequencies that signify the existence of the issue, the theoretical findings were discovered using the Ansys software (misalignment). in both the horizontal and vertical directions, angular misalignment is found to be higher at bearing housing2 (the point away from the motor) and lessen at bearings housing1 (the point near the motor). It is also shown that for both bearing supports, the maximum total RMS acceleration for vibration is larger in the horizontal direction and increases as misalignments become more severe. Mogal and Lalwani [5] the problem of unbalance and misalignment was studied experimentally by analyzing orders in a direct examination method for the laboratory apparatus prepared for their work. The results showed that. When there is an unbalance, the total RMS value is higher in the horizontal than in the vertical direction, the amplitude is of the first order and dominates (1X), and the degree of phase changes and ranges between (by 90°±30°) on the drive end bearing and non-drive end bearing. While in the case of angular misalignment, the phase difference is between (180°±30°) and the value of RMS is greater in the axial direction compared to the vertical and horizontal directions. While it was discovered that in the case of parallel misalignment, the phase difference is between (180°±30°) measured via the coupling, and the RMS value is of the second order (2X) and is very large.

In this research, a study was done investigated the effect of parallel misalignment on a rotor shaft with a rigid as well as flexible flange coupling have 2 and 4, nuts and bolts and rubber coupling. Where it was noted that the use of rubber coupling helps in vibrations reduction by 98%. It is also revealed from the literature review that above, it is important to detect the fault at earlier stage, so that the machine life can be enhanced with less cost. Hence, it is proposed to investigate the effect of parallel misalignment of the rotating machinery by using experimental and simulation study.

2. Faults modeling

2.1 Coupling misalignment

Misalignment of the coupling in modern manufacturing, shaft misalignment in the rotor-bearing system is a typical and primary source of vibration in rotating machinery. In a system with misalignment, the parts are not coaxial because of their individual purposes. When two shafts aren't perfectly aligned, couplings are employed to make up the difference. Coupling is an often overlooked yet crucial aspect of any rotor system despite being relatively inexpensive in comparison to the whole. Complete lineouts may use either rigid or flexible couplings to join the individual shafts. In the case of stiff couplings, the link is modeled as the union of two beam elements. Flexible couplings are frequently employed in rotating machinery because they permit some misalignment between the axis of rotation of any two adjacent shafts. [2].

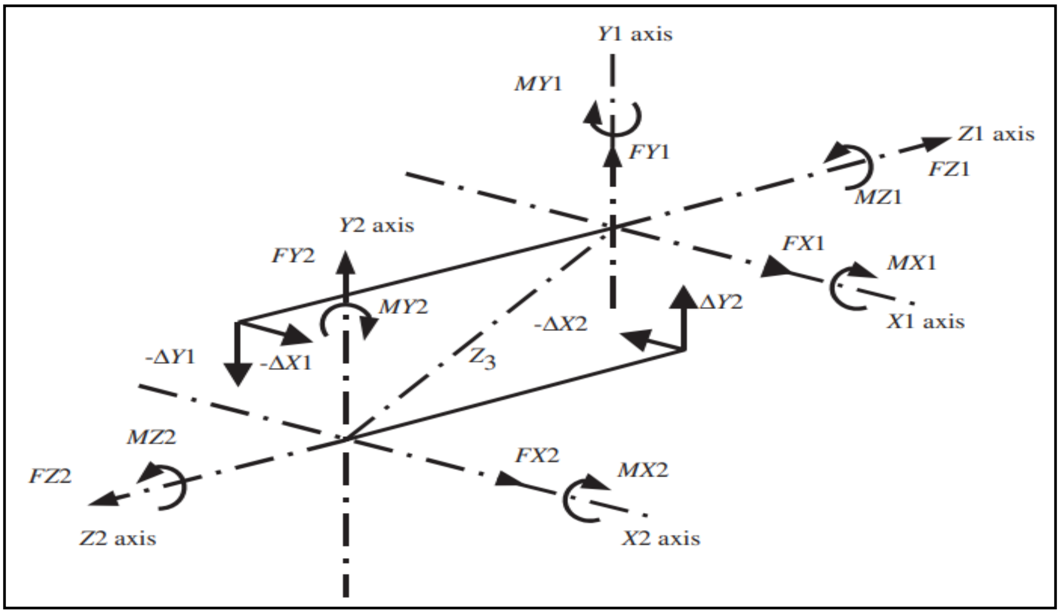

Fig. 1: Coupling coordinate system – parallel misalignment [2].

acts as follows on the shaft of the machine:

For parallel misalignment [2].

$$MX1 = T_q \sin\theta_1 + k_b\phi_1 \quad , \quad MX2 = T_q \sin\theta_2 + k_b\phi_2 \tag{1}$$

$$MY1 = T_q \sin\theta_1 + k_b\phi_1 \quad , \quad MY2 = T_q \sin\theta_2 + k_b\phi_2 \tag{2}$$

$$FX1 = (-MY1 - MY2)/Z3 \quad , \quad FX2 = -FX1 \tag{3}$$

$$FY1 = (MX1 + MX2)/Z3 \quad , \quad FY1 = FY2 \tag{4}$$

3. Experimental details

3.1 Faults Simulator Tests

Faults Simulator Tests To get more robust and approval for this work, many faults have been simulated at the machinery faults simulator. In these tests, The Data Acquisition (IDAC-6C) device was used with two sensor accelerometer type (B&K 4338) of a serial No. (442068) & (B&K 4338) of serial No. (540553) to transfer the data and then analyzed in the MATLAB program to detect fault cases misalignment. The machinery faults simulator can be shown in Figure (2) below. The main experimental part used in the generation of vibration signals are:

1. AC-motor controller
2. Tachometer

3. The Motor
4. Flexible coupling
5. Experimental bearing
6. Disc
7. Accelerometer.

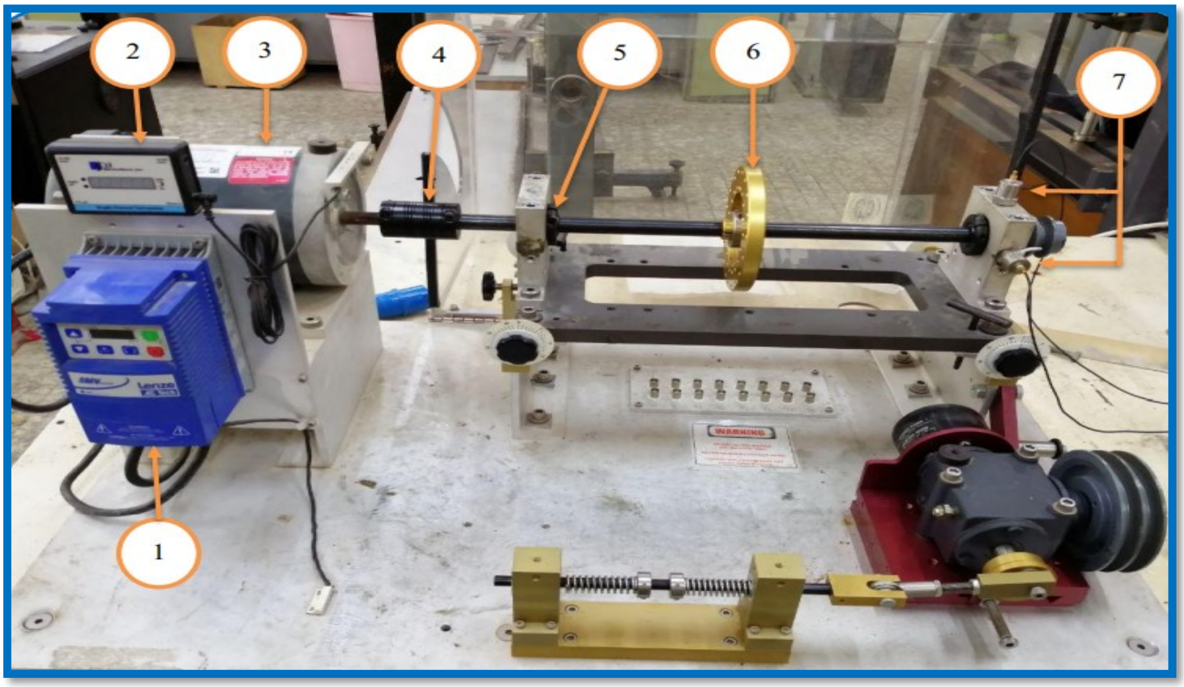

Fig.2: Machinery faults simulator

In these tests, the used rotor was already balanced. It has been used to show its vibration data, vibration spectrum, and phase difference at certain points on it. The shaft of the rotor has a diameter of (0.75 inches or 19.05 mm), and the distance between the supports was (407 mm). fault cases have been tested.

3.2 Experimental Procedure

Initially, the accelerometer is attached to the bearing housing and connected to the FFT analyzer. An FFT analyzer and a computer are used to record the vibration data. Bearings are subjected to a typical vibration spectrum. Vibration frequency spectrum behavior was investigated at a speed of (1500) rpm in the experiment. the laboratory workflow diagram is shown in **Figure (3)**. Data acquisition device is used to collect the raw vibration signal generated by MFS at speed value (1500 rpm), take new data for diagnosis **[9]**.

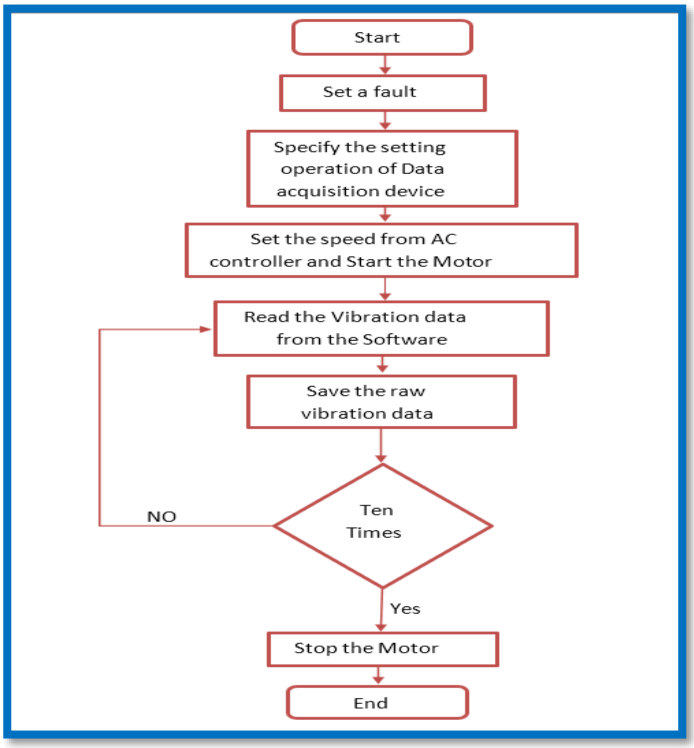

Fig. 3: Experimental Lap Procedure [9]

The setting operation for the IDAC-6C software interface for the speed(1500rpm) are:
• for speed (1500 rpm or 25 Hz) the sample rate is 2048 sample/sec, and the number of samples is 8192 and the low-pass filter is 800 sample/sec, and the high-pass filter is 0.3[9].

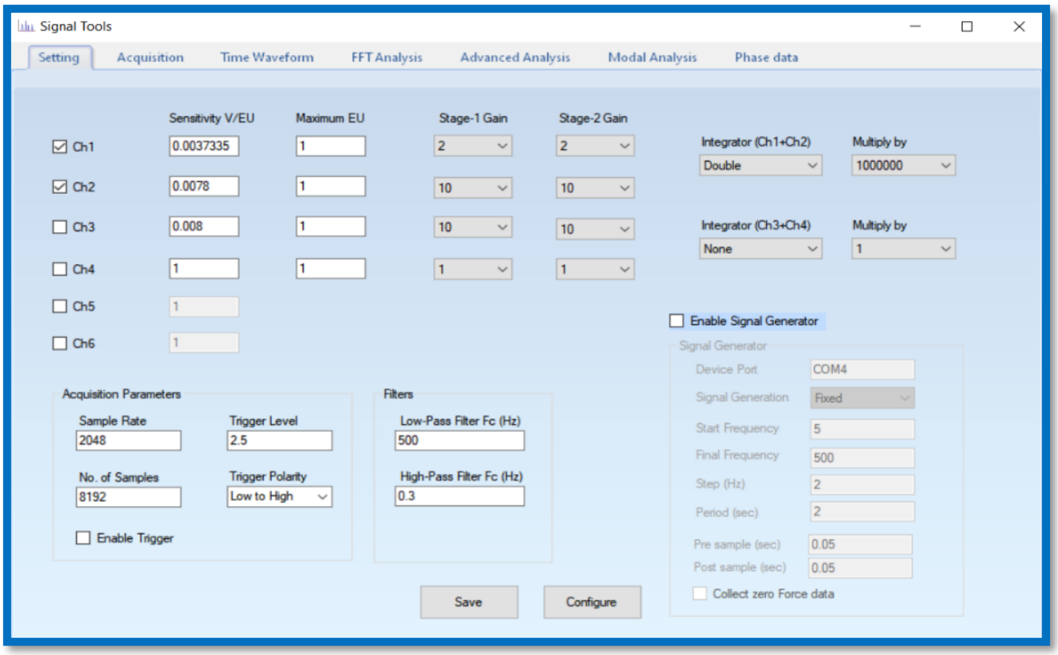

Fig. 4: setting operation for IDAC-6c software interface at speed (1500 rpm).

By Using MATLAB programming on the data that we obtained from the data acquisition device. This is done by writing a code in MATLAB, is possible to obtain vibration spectrum FFT with time waveform diagrams.

3.3 fault tested experimentally parallel Misalignment.

(1) Misalignment using Flange coupling at two-pins

The misalignment fault has been done Where the two shafts are connected using a Flange coupling at two-pin. And then it was given by giving a displacement of (1mm) to each of the main supports of the rotating shaft. The speed of (1500 r.p.m. or 25 HZ). was as shown in **Figure (5)**.

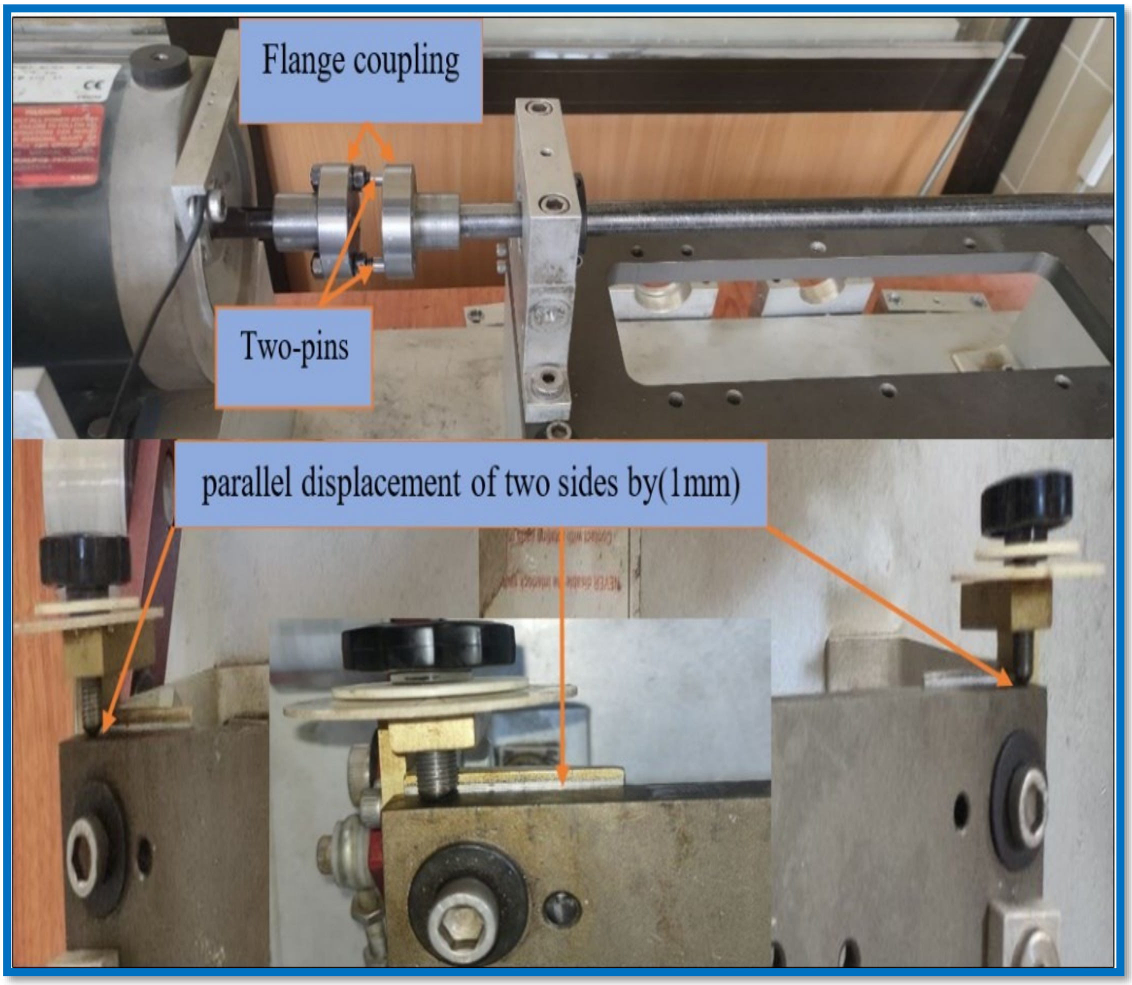

Fig. 5: shows the experimental parallel misalignment (using Flange coupling at two-pins).

(2) Misalignment using Flange coupling at four-pins

It has been accomplished in the same way as in the above case (Misalignment using Flange coupling at two-pin). But it was completed by changing the coupling. using Flange coupling at four-pin. was as shown in the **Figure (6)**.

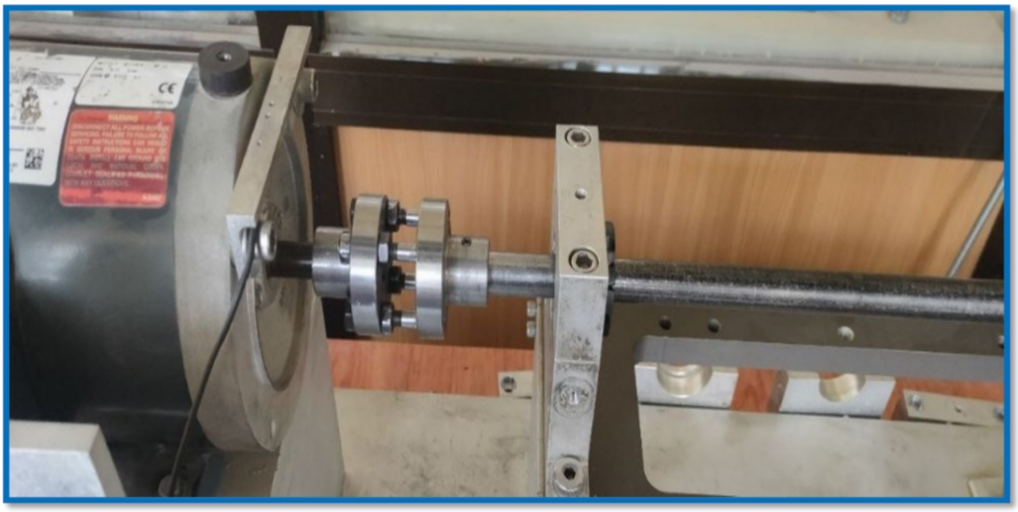

Fig. 6: shows the experimental parallel misalignment (using Flange coupling at four-pins)

(3) Misalignment using Rubber coupling

It has been accomplished in the same way as in the above case (Misalignment using Flange coupling at two-pin). But it was completed by changing the coupling. using Rubber coupling. The two ends of the coupling tool are tied with a rubber band of diameter(50mm) and length (20). With certain specifications. was as shown in the **Figure (7)**.

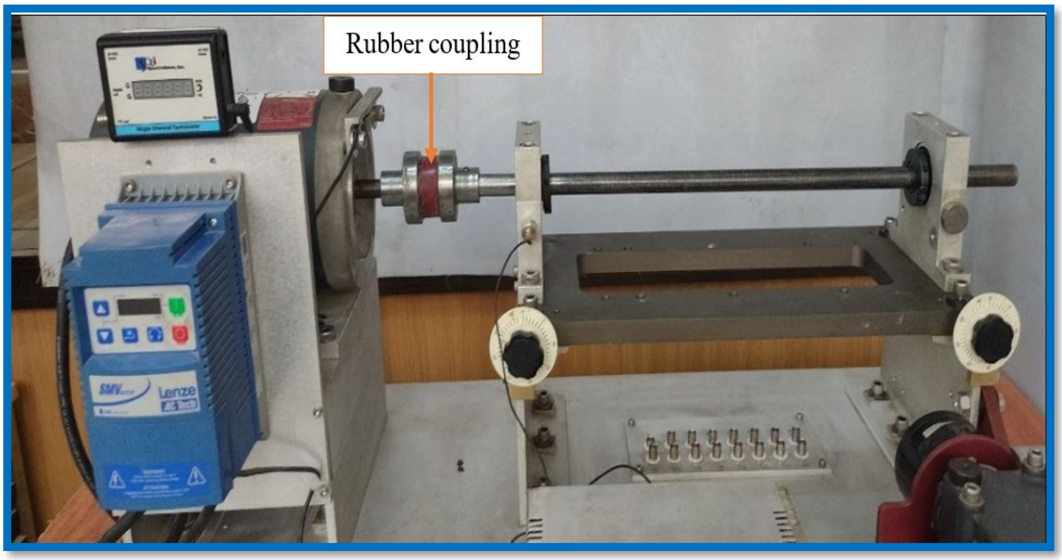

Fig. 7: shows the experimental parallel misalignment (using Rubber coupling)

4. Simulation Study

4.1 Modeling and mesh

Numerical analysis is carried out in this work by using FEA tool-ANSYS (2020R2) to determine the modal frequencies, mode shapes, and steady-state response of the system operation under parallel Misalignment, And the speed of the test was1500rpm (25HZ).

Figure (8) shows the 3-D model created, for the three misalignment cases with bearings specified in the FEA tool, for the numerical analysis of the system **[10]**. The automatic mesh method was used to mesh the structural model of the system as shown in **Figure (9)**. The material specified is structural steel with properties of elastic modulus E=210GPa, Poisson ratio=0.3, and Density=7850 kg/m^3.

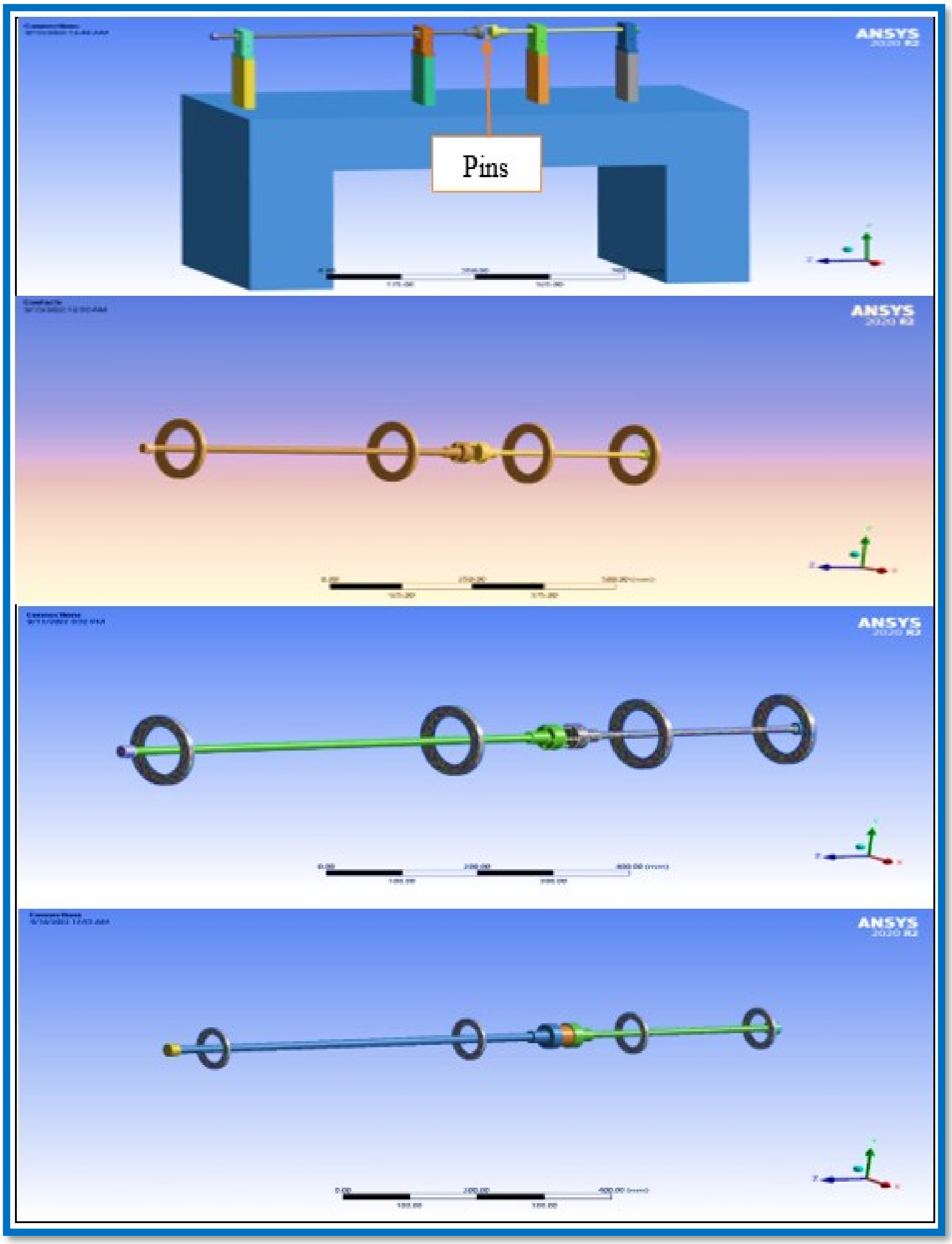

Fig.8: 3-D model for the three misalignment cases

Fig. 9: The automatic mesh using to mesh the structural model of the system.

4.2 Boundary condition and loading

• The bearing stiffness(K) was set to a constant value of 5000 N/mm because the bearing load and influence of rotation speed on bearing stiffness were not considered. Where the value of the velocity was 1500 RPM, and its direction was in the direction of the Z axis.

• The rotational velocity is added by writing commands in APDL in the Ansys program. Where the first command is for the first rotating shaft, which represents driving. While the second command is for the second rotating shaft, which represents the driven shaft. As shown below:

```
ICROTATE, SPOOL1,157,0,0,0,0,0,1,0,0,0,0
ICROTATE, SPOOL2,157,1,0,0,1,0,1,0,0,0,0
```

• Use flexible fixation from the edge of the left shaft to create thrust bearing (elastic support value =10 N/mm^3).

•Type of analysis in ANSYS: Transient Structural.

• A displacement of (1mm) was given in the direction of the axis (X), with the offset of the two axes fixed (Y and Z). The displacement was entered on the form of data. This data was produced by writing code in the MATLAB program. To be converted by the Excel program into a table consisting of two columns, it first represents the displacement, and the second column represents the time. Then they are entered into the Ansys program within the displacement condition and in the direction of the axis X.

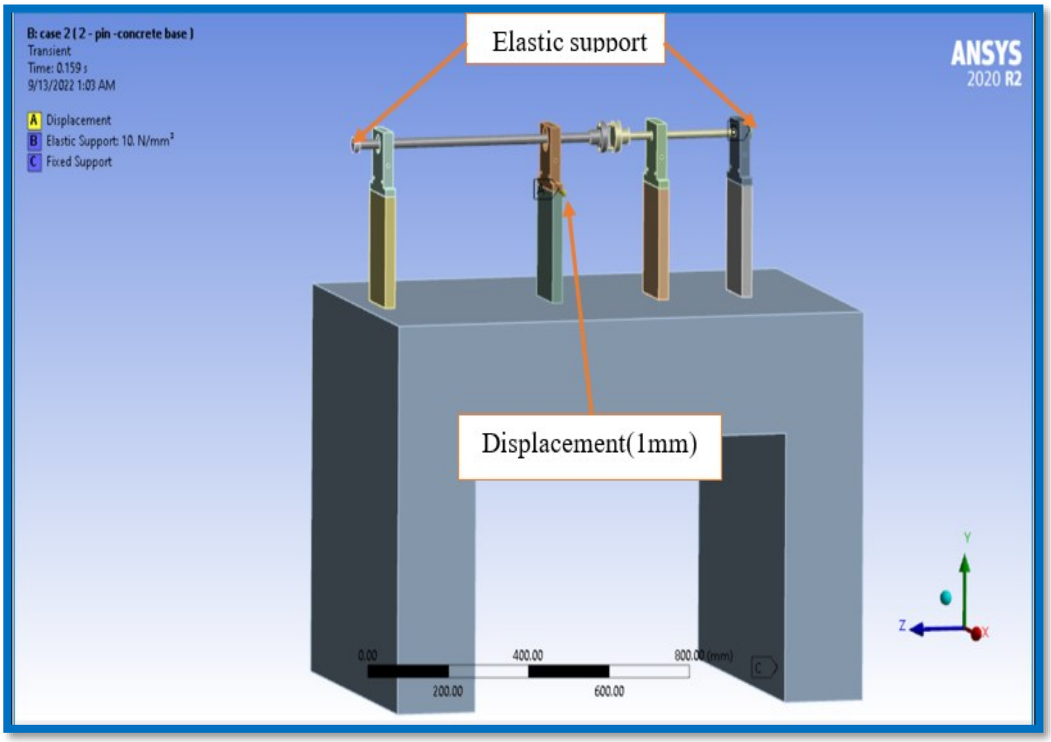

Fig. 10: Use boundary conditions in model of rotor bearing system (the direction of the displacement and elastic support)

• The code used to generate the displacement data using the MATLAB program.

```
t=0:0.001:0.16
y= (1/0.01) *(0:0.001:0.01)
y1=repmat (1, [1,150]);
yt= [y y1]
plot (t, yt)
ylim ([0 1.2])
```

• Convert data from MATLAB to Excel using the formula below.

```
k=table (t', y t','variable Names', {'time','dis'});
>> MS='mytrue.xlsx';
```

```
>> writetable (k, MS);
>> winopen (MS)
```

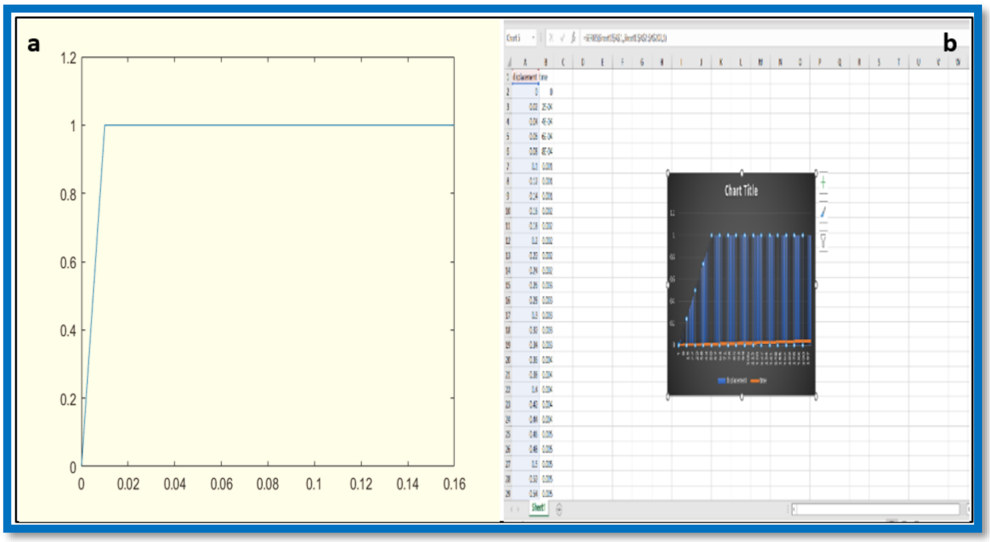

Fig.11: (a) Represents the displacement in MATLAB; (b) Represents the displacement in Excel.

5. Results and discussions

The vibrations are measured at the Drive end (DE) bearing in, horizontal directions, for the misalignment cases The above-mentioned. This work agrees with some of the research that was mentioned in the literature review. Where Some of them used the method of the empirical study to prove the validity of their work, and others used the simulation study (Ansys) and the empirical study to prove the validity of their work as well. Where they both extracted the diagrams of vibration spectrum (FFT) and time waveforms. Therefore, we can refer to some researchers whose research has been helpful to complete the results [1,2].

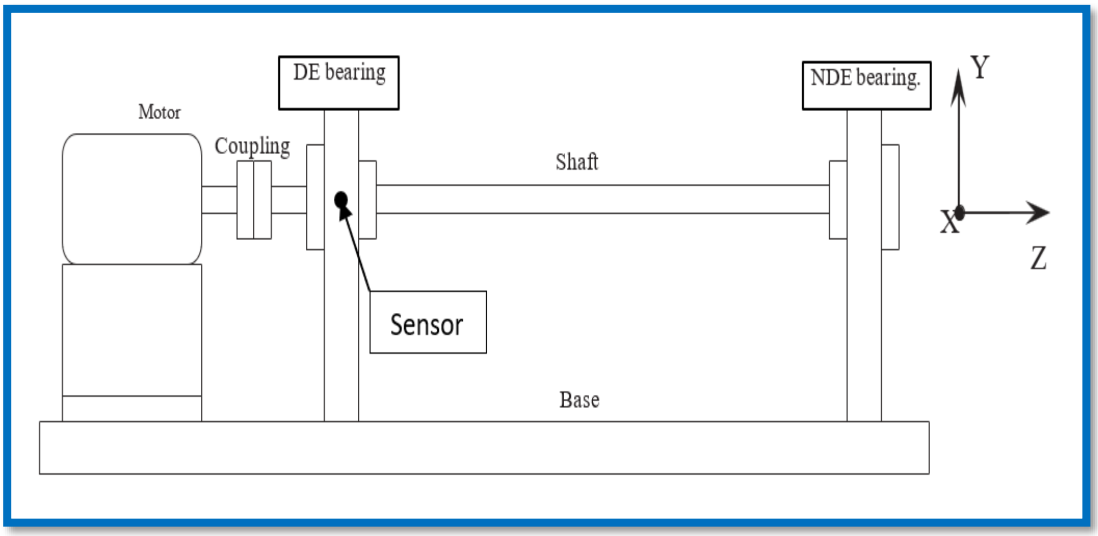

Fig. 12: Rotor–coupling–bearing test rig.

(A) Misalignment using Flange coupling at two-pins

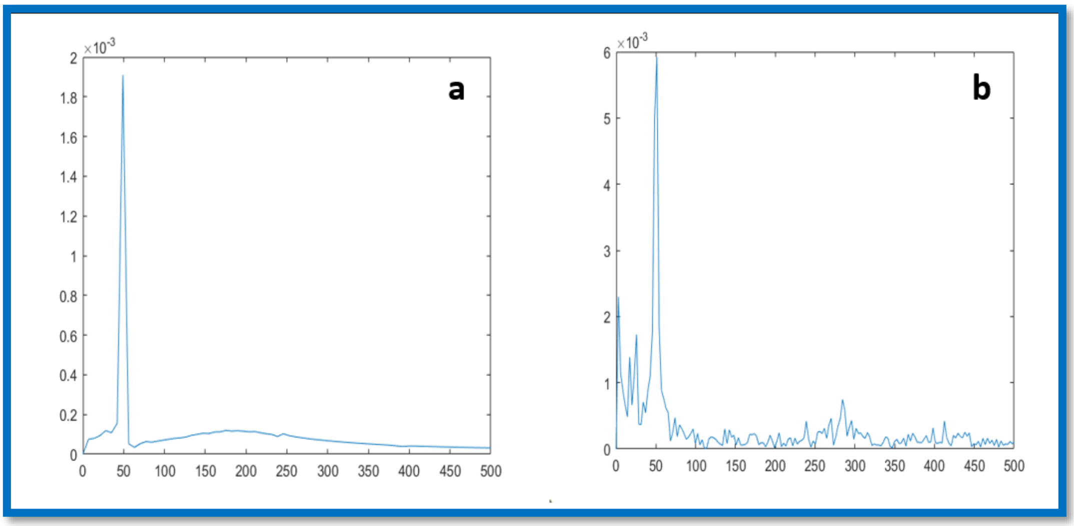

Fig.13: FFT of parallel misalignment (using flange coupling at two-pins for speed 1500rpm) at the Drive-end in the horizontal direction (DE-H). (a) Simulation study; (b) Experimental study.

Table 1: Overall amplitude for parallel Misalignment using Flange coupling at two-pins in horizontal direction at Drive-end (DE) bearings of the rotor.

Speed (RPM)	Amplitude (mm)		
	Bearing end	Simulation study	Experimental study
		Horizontal (H)	Horizontal (H)
1500 RPM (25HZ)	Drive-end (DE)	0.00190	0.00582

Observed through the experiment and simulation the vibration spectrum acquired on end bearings (Drive-end (DE) at a frequency of 25 Hz, it is found that the maximum vibration amplitudes observed are (0.00582mm) and (0.00190mm) for the experimental and simulation respectively. This indicates that the amplitude of vibration in the simulations is less than it is in the experiment. And this is possible because the theoretical model is not completely similar to the practical one in terms of the type of the pin's material and the space between the pin and the coupling, and also because of inaccurate manufacturing and the presence of deviations in the coupling itself that hinder movement more and cause a higher force. Also, other vibration sources contribute to the experimental data. However, the simulation and experimental results are in close agreement regarding dominant frequency since the peak amplitude is seen at 50 Hz which is 2X the shaft speed. As shown above, in **Figure (13)** and **Table (1)**.

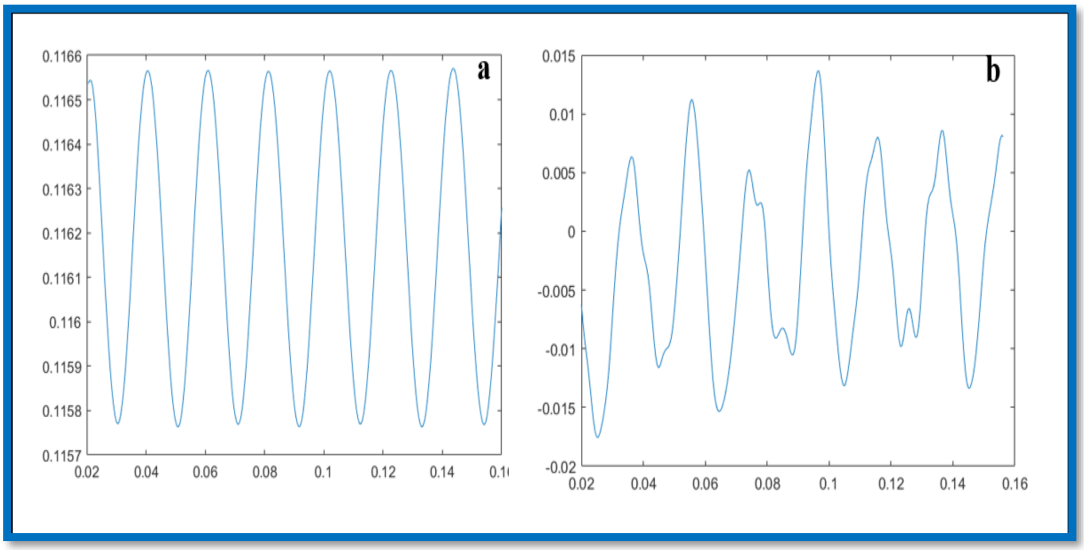

Fig. 14: Parallel misalignment (using Flange coupling at two-pins) responses at the time domine at bearings (DE) in the horizontal direction. [(a) Simulation study; (b) Experimental study at speed 1500rpm].

We note when excitation frequency is increased, the magnitude of vibrational amplitudes increases, **Figure (14)** above shows the responses in the time domain, for machine bearings. for the two studies (Experimental and simulation).

(B) Misalignment using Flange coupling at four-pins.

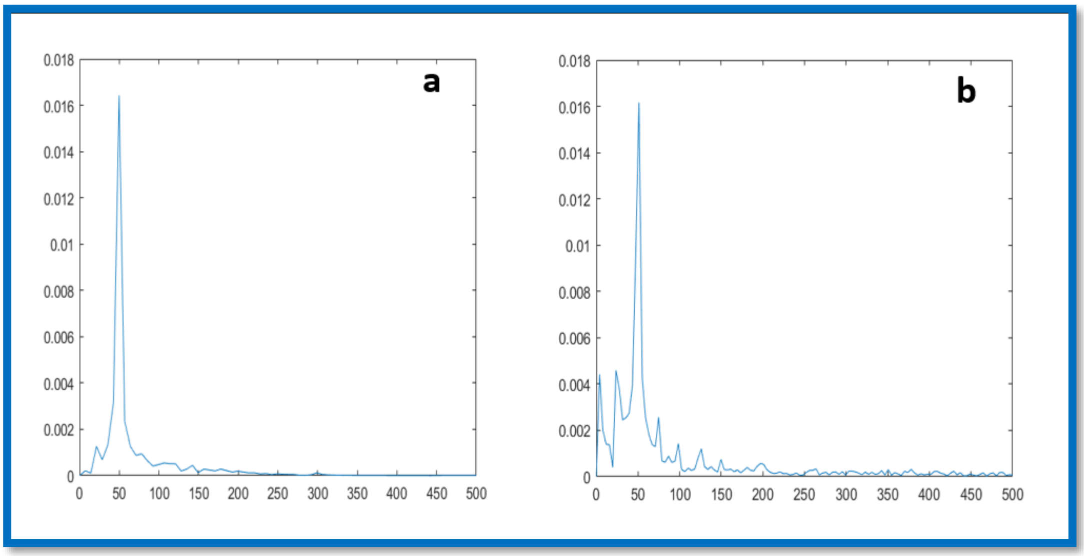

Figure 15: FFT of Parallel misalignment (using flange coupling at four-pins for speed 1500rpm) at the Drive-end in the horizontal direction (DE-H). (a) Simulation study; (b) Experimental study.

Table 2: Overall amplitude for Parallel misalignment using Flange coupling at four-pins in horizontal direction at Drive-end (DE) bearings of the rotor.

Speed (RPM)	Bearing end	Amplitude (mm)	
		Simulation study	Experimental study
		Horizontal (H)	Horizontal (H)
1500 RPM (25HZ)	Drive-end (DE)	0.01612	0.01646

It can be seen from the **Figure (15)** and **Table (2)** the vibration spectra are obtained through experimental and simulation for various excitation frequencies and found to be in close agreement. Results from both the experimental and simulated studies showed that the second harmonics (2X), or twice the shaft running speed in the case of parallel misalignment, best characterize the misalignment.

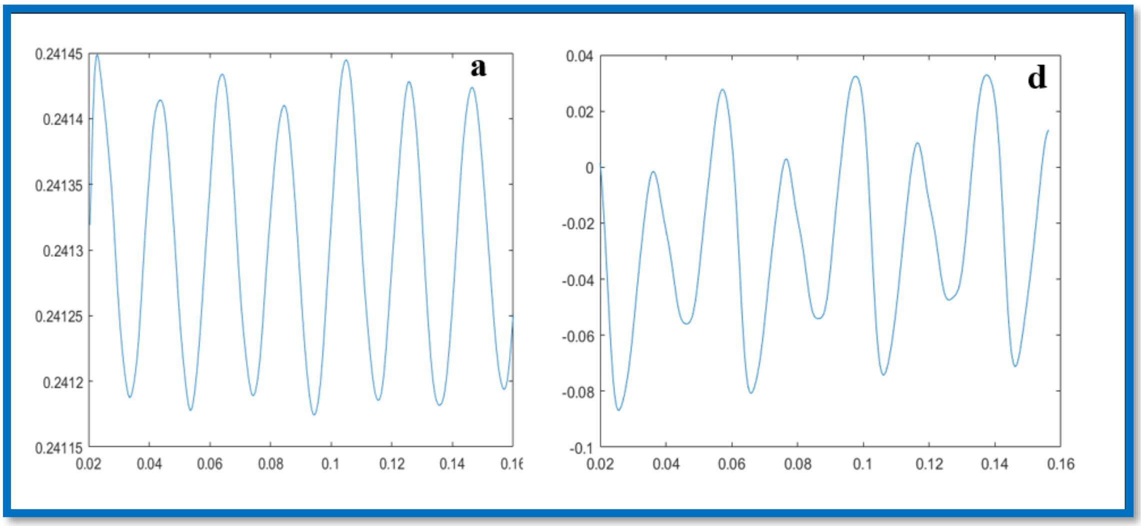

Figure 16: Parallel misalignment responses (using Flange coupling at four-pins) at the time domine at bearings (DE) in the horizontal direction. [(a) Simulation study; (b) Experimental study at speed 1500rpm].

It can also be concluded from the time-domain diagrams shown in **Figure (16)** that the magnitude of the vibrational amplitudes increases with the increase in the excitation frequency. with the prevailing peak remaining (2X). (i.e., 2X the shaft speed).

(C) Misalignment using a Rubber coupling

It can be concluded from the experiment and simulation the vibration spectrum acquired on end bearings (Drive-end (DE)) at a frequency of 25 Hz that the results are in good agreement. Through the FFT diagram shown in the **Figure (17)**, it was found that when using the rubber material in the coupling tool between the drive shafts of the machine, the frequency did not appear clearly. meaning that the absence of the peak prevailing (2X) as in the previous cases of misalignment. And the reason for that goes back to that the rubber material absorbs the forces of vibration and reduces them to the minimum possible.

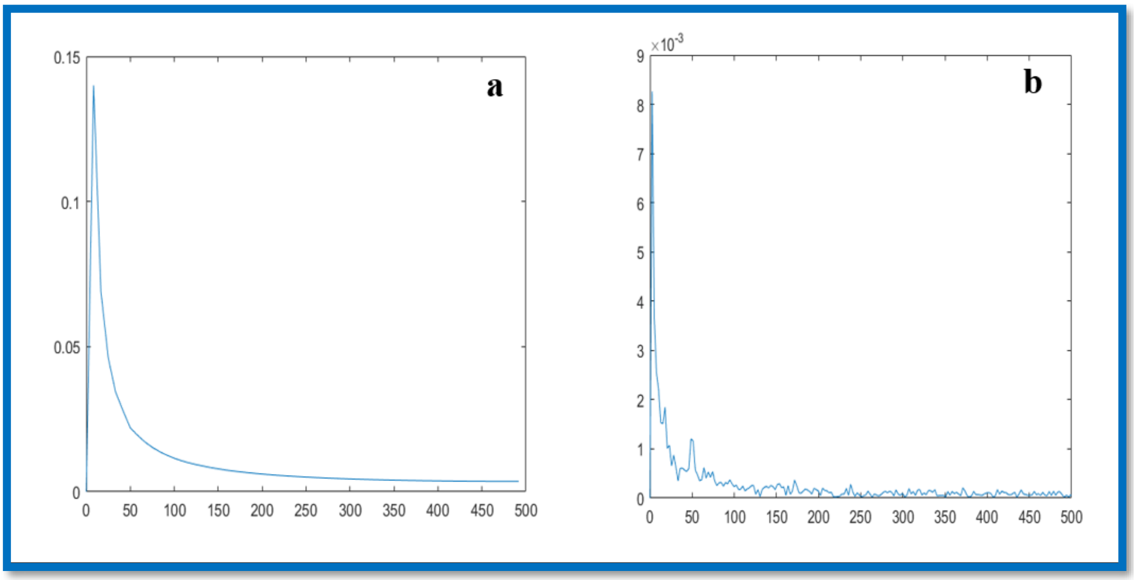

Fig.17: FFT of Parallel misalignment (using a Rubber coupling for speed 1500rpm) at the Drive-end in the horizontal direction (DE-H). (a) Simulation study; (b) Experimental study.

6. Conclusions

the following conclusions can be obtained as a result of the experimental analysis and the theoretical analysis that was used to verify the operating conditions of the vibration characteristics of the rotating mechanical system:

1. For the misalignment case, the vibration spectra are obtained through experimental and simulation for various excitation frequencies and found to be in close agreement. Results from both the experimental and simulated studies showed that the second harmonics (2X), or twice the shaft running speed in the case of parallel misalignment, best characterize the misalignment. In contrast, the amplitude of vibration was much lower in the case of an aligned system. In addition, it can be seen that the amplitude of vibration grows in tandem with the excitation frequency.

2. By conducting the two studies, it was proved that when using rubber material in coupling tool with certain specifications commensurate with the size of the machine and operating conditions, the machine can work without the misalignment defect for a longer period of time

than if the rubber was not used. meaning that the absence of the peak prevailing (2X) as in the previous cases of misalignment.

7. Acknowledgments

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