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Vibration Analysis Of Flywheel Balancing Test Rig

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Abstract— ***Flywheel balancing test rigs are used in several industries for studying Noise, vibration, and harshness (NVH), specifically automotive, aerospace, etc. Project work includes design and manufacturing of flywheel balancing test rig using CAD and ANSYS software. Basic calculation for AVM (Anti Vibration Mount) will be done and suitable design will be selected for absorbing or damping exam vibration. Experimental vibration analysis will be carried out using FFT Analyzer and data acquisition software. Order and signature analysis will be done for studying vibration characteristic of structure. Results and conclusion of various test conducted will be plotted and discussed.***

Keywords— NVH, Vibration, FEA, ANSYS.

I. Introduction

NOISE is perceived by the human ear for both loudness and sharpness after the air pressure fluctuations are converted to the mechanical motion of the cochlea which responds at different locations depending upon the excitation frequency of the sound. characteristics of Sound can be described by sound pressure level and frequency with equal loudness contours. The sound pressure level has a unit of decibel (i.e., one tenth of a Bel), which in fact is a unit of power. It can be acclaimed that the pressure of sound level is not a measure of the loudness of a sound because the human ear is not equally sensitive to all frequencies. If pressure of the sound one noise is concentrated in a higher frequency range where the ear is more sensitive, it may be perceived louder than another noise with equal sound pressure level but concentrated in a lower frequency region that is less sensitive to the ear. In order to obtain the levels which, bear a closer relationship to loudness judgment than the sound pressure levels, three different networks of frequency weighting (A, B, and C) were incorporated in sound level meters.

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The A-weighting most closely mimics the human ear, and it is most widely used in noise control work. The A-weighted level of sound is expressed in the unit of dB(A) or dBA. A human

buzzing may be around 1 kHz and 30 dB(A). A normal conversation is at 0.5 kHz and 65 dB(A). A machine operator

can be exposed to sounds around 0.5–5 kHz and 90–100 dB(A). The noise level in residential areas is around 50 dB(A). A vehicle noise can be 85 dB(A). Generally, a sound pressure level above 75 dB(A) is regarded as noisy, and a noise level above 90 dB(A) is considered very high. The threshold of pain is around 120 dB(A). In order to decrease or prevent hearing damage, standards organizations (OSHA) and national labor laws stipulate the limits of noise and impose hearing protection measures. The audible frequency range (20 Hz–20 kHz) can be divided into bands with one octave wide in the frequency analysis of noise signals, the narrow band spectrum data.

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II. Literature Review

Yue Meng [1] Feature extraction is crucial to rotating machinery prognosis, which is an important aspect of condition monitoring as well as maintenance program, since the quality of feature will impact the result significantly. Vibration signals are commonly used as the source for feature extraction during the prognosis process, especially the energy feature of fundamental frequency (which is written as 1X), 2X, 3X, 1/2X, etc. Yet this kind of feature shows insufficiency for identifying stages of performance degradation and classifying the type of early fault, therefore researchers focused mainly on improving the methods of feature extraction to solve this problem. However, features extracted from vibration signals always ignore some fault information such as kinematics information and phase information, thus other source of feature is needed to provide supplement or even substitute for higher efficiency and sharpness of separation in rotating machinery prognosis, which are strongly demanded by today's complex and advanced machines. This paper introduced one kind of classic feature source: shaft orbit, which is widely used in traditional diagnosis for failure classification, into prognosis, and its effectiveness is verified in rotor early unbalance fault identification using features extracted from it, compared with energy features of frequency band extracted from vibration signals. Result shows that shaft orbit feature can be used in identifying different early fault stages of rotor unbalance, which indicates that utilizing shaft orbit as source of feature extraction can provide a new approach of getting early fault features in rotating machinery prognosis.

Yongcun Cui et al. [2] This paper presents nonlinear dynamic differential equations of high-speed cylindrical roller bearing considering the impact of roller dynamic unbalance. The cage vibration characteristics are investigated as a function of roller dynamic unbalance, structure parameters and working conditions. The fast Fourier transformation, cage orbit and vibration acceleration level are used to evaluate the vibration magnitude of the cage. The results show that: (1) Roller dynamic unbalance can affect the state of the cage motion and cause the cage's vibration acceleration level to increase. Once the roller dynamic unbalance exceeds a threshold, the cage will produce a high frequency vibration; (2) The most significant impact on the cage vibration is roller dynamic unbalance in angular direction of roller axis, followed by radial and axial direction; (3) A smaller radial clearance and a reasonable pocket clearance are beneficial to reducing the effect of roller dynamic unbalance on the cage vibration. It is recommended to take a larger cage-race guide clearance within a certain range to lower the cage vibration; (4) The cage vibration due to the roller dynamic unbalance increases with the bearing speed, and is greater than that of the cage with ideal roller. The effect of roller dynamic unbalance on the vibration of cage under high-speed condition is greater than that of the low-speed condition. The increase of the radial load of the bearing, to a certain extent, can reduce the vibration of cage considering the roller dynamic unbalance.

Stephan Rinderknecht [3] Unbalances in rotating machinery cause vibration, noise and wear. Active bearings allow the use of specialized control algorithms which eliminate the bearing forces caused by unbalances. Most algorithms suffer from drawbacks, namely their lack of general stability, the need for exact rotor models, or their unclear rotor dynamic interactions. In this work, we found a closed-form analytical solution for the elimination of un- balance-induced bearing forces on arbitrary, gyroscopic rotors. We demonstrate that the force-free condition is met for any unbalance distribution. Furthermore, we proved that up to two resonances can be fully eliminated. We found a new Lyapunov stability theorem to prove the controller's superior stability properties. The advantage of our approach is that different active bearing technologies are unified in a single, generalized theory. Our theory is not only limited to rotors, but applies to all systems with harmonic excitation that share the same general matrix structure. Finally, we provide evidence that our- oretic assumptions are also satisfied in reality: In an experiment we demonstrate that unbalance-induced bearing forces and rotor resonances can not only be eliminated in theory, but also in practice.

T.S. Morais [4] Classical balancing techniques for rotating machinery consider that these systems are linear.
However, if some nonlinearity appears in the structure, these techniques do not work properly and the results obtained regarding the correction weights and their corresponding angular positions are not satisfactory. This behavior is due to the fact that these techniques, such as the influence coefficients method, consider linear relations involving the unbalance excitations and the resulting vibration. On the other hand, the choice of the number and the repartition of the correction planes depends on the possible accessibility that varies for each machine. In this work, a new method dedicated to the identification of the rotating machinery and the unbalance distribution in linear and nonlinear conditions is realized through pseudorandom optimization methods and the system modeling is performed by using the well-known finite element method. The nonlinearity is introduced by using a frequency dependent bearing. Several computer simulations are performed for different rotor configurations (linear and nonlinear). The methodology is then validated through an experimental test rig. The results obtained demonstrate the effectiveness of the method developed.

Mathias Pfabe [5] Modern turbocharged combustion engines induce high fluctuating torques at the crankshaft. They result in torsional crankshaft vibrations that are transferred both to the gearbox and the auxiliary engine systems. To reduce the torsional crankshaft vibrations, a kinematically driven flywheel (KDF) for compensating fluctuating engine torques has been developed. It comprises a flywheel that is coupled to the crankshaft by means of a non-uniformly transmitting mechanism. The kinematical transfer behavior of the mechanism is synthesized in such a manner that the inertial flywheel torque compensates at least one harmonic of the fluctuating engine torque. The degree of non-uniformity of the mechanism has to be adapted to the actual speed and load of the engine. A novel geared double-crank mechanism with cycloidal-crank input and adjustable crank length is proposed and analyzed. Parameter synthesis is achieved by means of a simplified mechanical model that calculates the required transfer function for a given engine torque. A multibody simulation of a three-cylinder engine equipped with the KDF demonstrates the efficacy of the system. A functional prototype has been built up and tested on an electrically driven test stand. Measurements confirm the simulated behavior of the KDF thus demonstrating the potential of the device.

Jianfei Yao [6] The paper describes the identification and optimization of unbalance parameters in rotor bearing systems. Two methods are proposed for the identification of the unbalance characteristics: the first is based on modal

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expansion combined with the use of optimization algorithms, while the second relates to the use of modal expansion technique applied to the inverse problem. In this work, the modal expansion technique is used to overcome the issue related to the use of a reduced number of measuring points. The equivalent unbalance forces can then be estimated by expanding the modal displacements into generalized coordinates of the equations of motion of the rotor system. An error due to the use of a modal expansion is however inevitable, and to solve the issue we propose the adoption of an inverse problem formulation to avoid the computation of the displacements at each measurement point. The axial location of the unbalance must be however known in advance, if the inverse problem approach is used to identify the unbalance parameters. We therefore propose in this work an integrated modal expansion inverse problem methodology combined with an optimization procedure. The technique allows to identify the axial location of the unbalance, its magnitude and phase. Simulation and experimental investigations are carried out to verify the validity of the proposed methods in a double-disk rotor-bearing system. The results show that identification and optimization procedure for the integrated modal expansion/inverse problem approach provides more accurate predictions than the ones given by the pure modal expansion method.

III. PROBLEM STATEMENT

Rotating unbalanced machine with balanced and unbalanced masses needs to investigated and designed considering its dynamic characteristics. Hence, noise free setup design is needed.

IV. OBJECTIVES

- To prepare CAD Model of Flywheel design using Catia V5 Model
- Modal Analysis of setup using ANSYS workbench
- Manufacturing of experimental test rig.
- Perform experimental analysis using FFT/Analyzer and Data Acquisition software.
- Validation of analytical and experimental work

V. METHODOLOGY

Step1: - I started the work of this project with literature survey and gathered many research papers which are relevant to this topic. After going through these papers, I have learnt about FFT test and analysis of Test rig
Step2: - After that the components which are required for my project are decided.
Step3: - After deciding the components, the 3 D Model and drafting will be done with the help of CATIA software.
Step4: - The Model Analysis will be carried out with the help of ANSYS software.
Step5: - Manufacturing of Test specimen in workshop
Step6: - The experimental observations will be taken on FFT.
Step7: - Comparative analysis will be made between simulation and experimental results and then Results and conclusions will be drawn.

VI. VIBRATION ANALYSIS OF NVH TEST RIG

Modal analysis: Every object has an internal frequency (or resonant frequency) at which the object can naturally vibrate. It is also the frequency where the object will allow a transfer of energy from one form to another with minimal loss. As the frequency increases towards the "resonant frequency," the amplitude of response asymptotically increases to infinity. In other words, the results of the modal analysis are these frequencies at which the amplitude increases to infinity.
Every system can be described in terms of a stiffness matrix that connects the displacements and forces. These frequencies are known as natural frequencies of the system and are provided by the eigenvectors of the stiffness matrix. These frequencies are also known as the resonant frequencies.

Model analysis of NVH Test

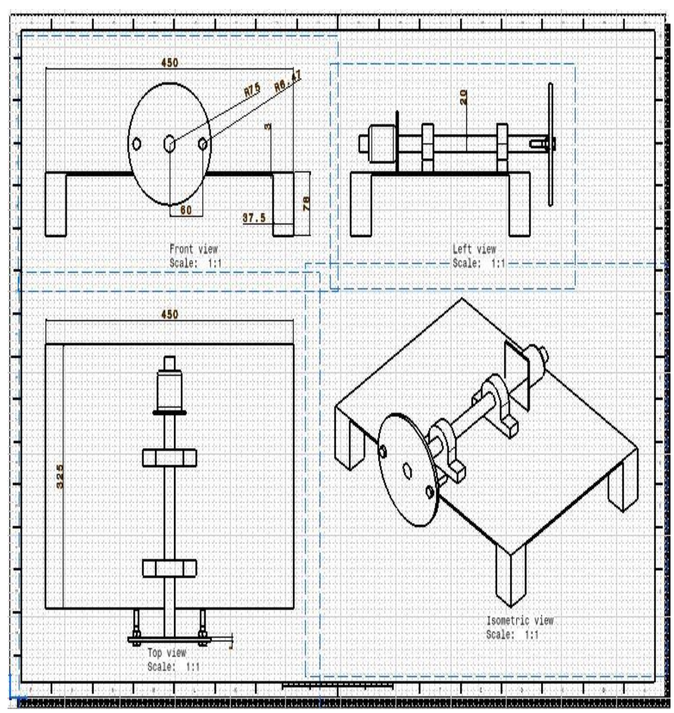

Figure.1 Drafting

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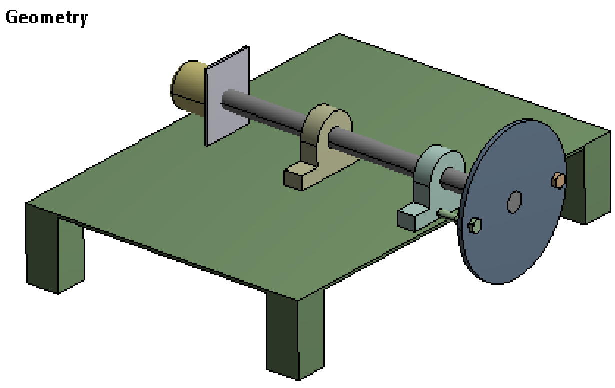
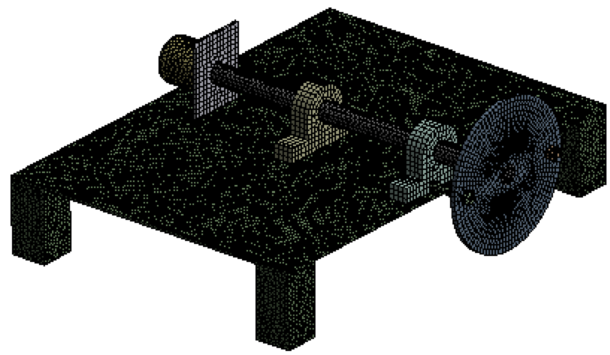


Figure.2 Catia Model



Statistics	
Nodes	142810
Elements	68312

Figure.3 Meshing of NVH Test Rig

Material properties of Test Rig

Properties of Outline Row 3: Structural Steel

	A	B	C
1	Property	Value	Unit
2	Material Field Variables	Table	
3	Density	7850	kg m^-3
4	Isotropic Secant Coefficient of Thermal Expansion		
5	Coefficient of Thermal Expansion	1.2E-05	C^-1
6	Isotropic Elasticity		
7	Derive from	Young's Modu...	
8	Young's Modulus	2E+11	Pa
9	Poisson's Ratio	0.3	
10	Bulk Modulus	1.6667E+11	Pa
11	Shear Modulus	7.6923E+10	Pa

Mild Steel

Mesh: ANSYS Meshing is a general-purpose, intelligent, automated high-performance product. It produces the most appropriate mesh for accurate, efficient metaphysics solutions. A mesh well suited for a specific analysis can be generated with a single mouse click for all parts in a model. Full controls over the options used to generate the mesh are available for the expert user who wants to fine-tune it. The power of parallel processing is automatically used to reduce the time you have to wait for mesh generation. After meshing of Test Rig we get nodes and elements of geometry.

Boundary Condition: A boundary condition for the model is the setting of a known value for a displacement or an associated load. For a particular node you can set either the load or the displacement but not both.

The main types of loading available in FEA include force, pressure and temperature. These can be applied to points, surfaces, edges, nodes and elements or remotely offset from a feature. The way that the model is constrained can significantly affect the results and requires special consideration. Over or under constrained models can give stress that is so inaccurate that it is worthless to the engineer. In an ideal world we could have massive assemblies of components all connected to each other with contact elements but this is beyond the budget and resource of most people. We can however, use the computing hardware we have available to its full potential and this means understanding how to apply realistic boundary conditions.

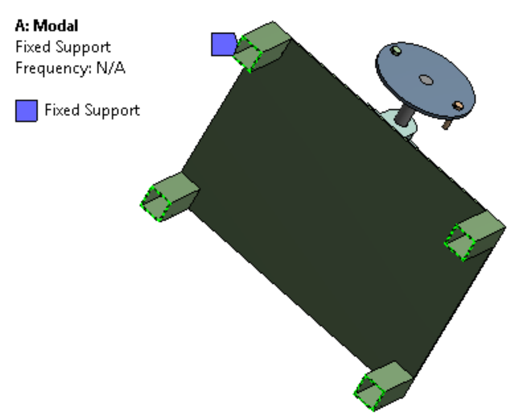


Figure.4 Boundary Condition of NVH Test Rig

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Total Deformation

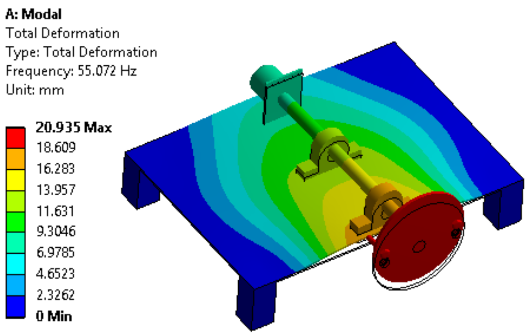

Figure.5 Total deformation of NVH Test Rig mode 1

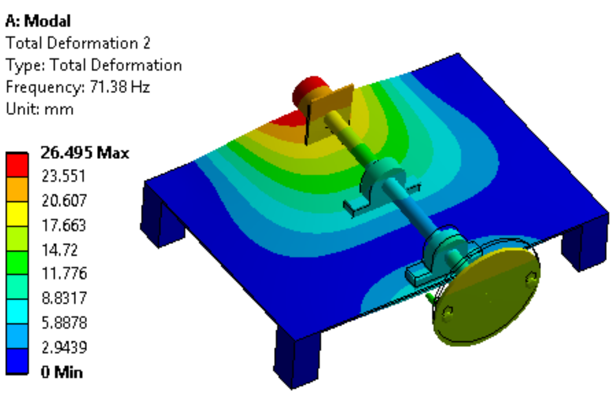

Figure.6 Total deformation of NVH Test Rig mode 2

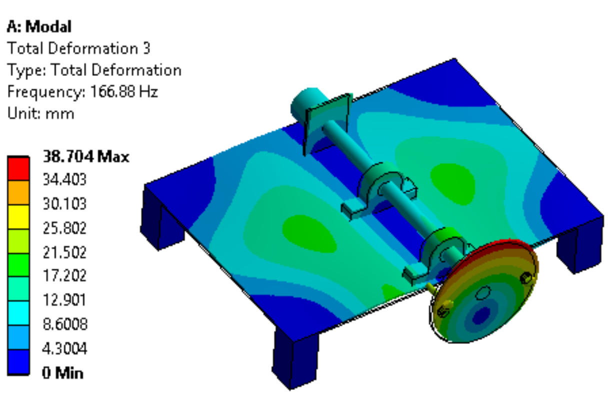

Figure.7 Total deformation of NVH Test Rig mode 3

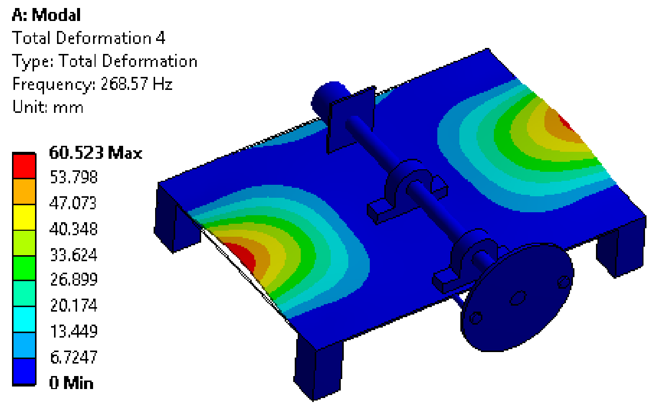

Figure.8 Total deformation of NVH Test Rig mode 4

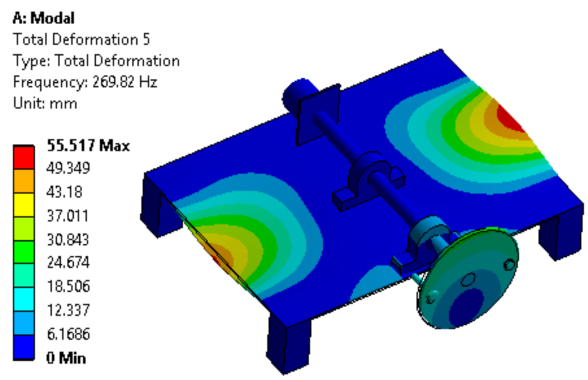

Figure.9 Total deformation of NVH Test Rig mode 5

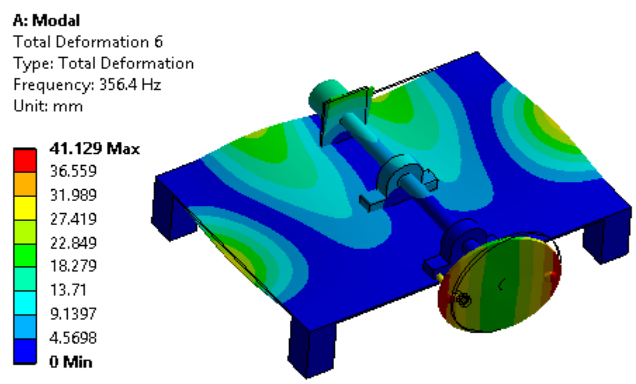

Figure.10 Total deformation of NVH Test Rig mode 6

VII. Experimental Analysis Of NVH Test Rig

A. Fast Fourier Transform(FFT) Analysis

FFT is one main property in any sequence being used in general. To find this property of FFT for any given sequence, many transforms are being used. The major issues to be noticed in finding this property are the time and memory management. Two different algorithms are written for calculating FFT and Autocorrelation of any given sequence.

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Comparison is done between the two algorithms with respect to the memory and time managements and the better one is pointed. Comparison is between the two algorithms written, considering the time and memory as the only main constraints. Time taken by the two transforms in finding the fundamental frequency is taken. At the same time the memory consumed while using the two algorithms is also checked. Based on these aspects it is decided which algorithm is to be used for better results

B. DEWE-43 Universal Data Acquisition Instrument

When connected to the high speed USB 2.0 interface of any computer the DEWE-43 becomes a powerful measurement instrument for analog, digital, counter and CAN-bus data capture. Eight simultaneous analog inputs sample data at up to 204.8 kS/s and in combination with DEWETRON Modular Smart Interface modules (MSI) a wide range of sensors are supported Voltage Acceleration Pressure Force Temperature Sound Position RPM Torque Frequency Velocity and more The included DEWESoft application software adds powerful measurement and analysis capability, turning the DEWE-43 into a dedicated recorder, scope or FFT analyzer.

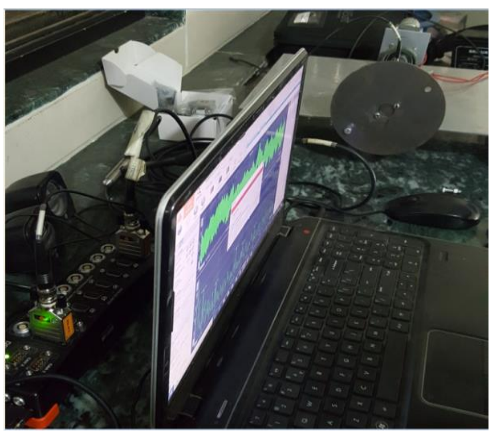

Figure.11 Experimental Setup

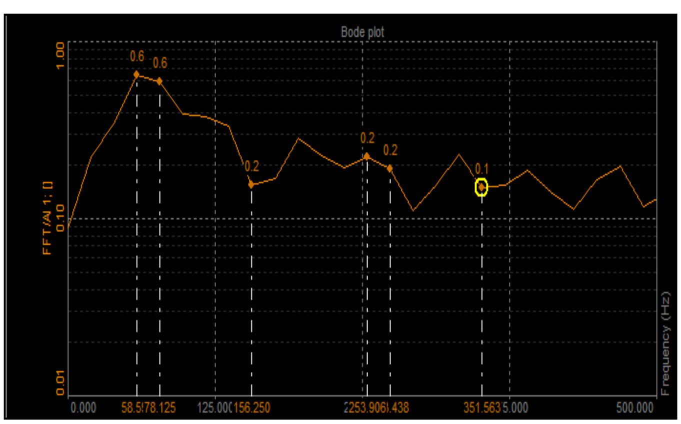

Figure.12 Test FFT Results

MODE NO	FEA (Hz)	TEST FREQUENCY (Hz)
1	55.072	58.5
2	71.38	78.12
3	166.88	156.25
4	268.57	253.9
5	269.82	273.4
6	356.4	351.56

Table.1

Comparison of FEA stimulation done in Ansys & FFT observed Test Frequency have been taken in the table 1.

VIII. NVH Test Rig With Applying Masses

In this experimental testing of disc ring due to rotation of shaft vibration of test rings are unbalance and balance. so we have to check out which masses applied on rig are balance the structure. The mention table 2 shows the frequency at various nodes.

Sr. no	NO MASS ON DISC (Hz)	FFT PLOT WITH SINGLE MASS ON DISC(Hz)	WITH BOTH MASS ON DISC (Hz)
1	100.1	97.66	61.4
2	253.9	246.5	195.3
3	441.9	515.1	427.2
4	720.2	752	515.1
5	810.5	810.5	668.6

Table. 1

FFT PLOT OF TESTING

After placing accelerometer on pedestal bearing we given connection to DC motor of driving shaft on which Test Rig is mounted. reading are taken changing masses which are mounting on test ring. First plot taken when there is no mass on test rig

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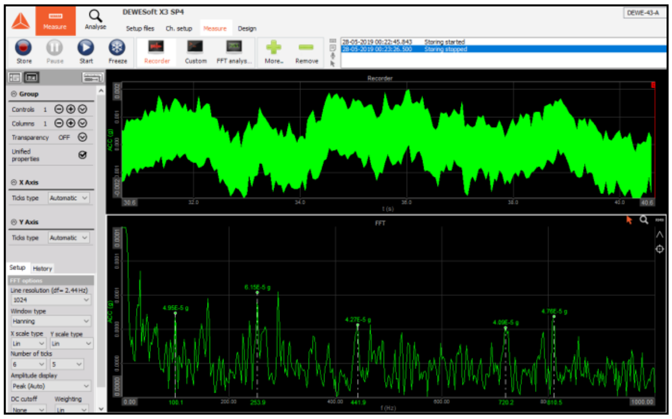

Figure.13 FFT PLOT with no mass on disc

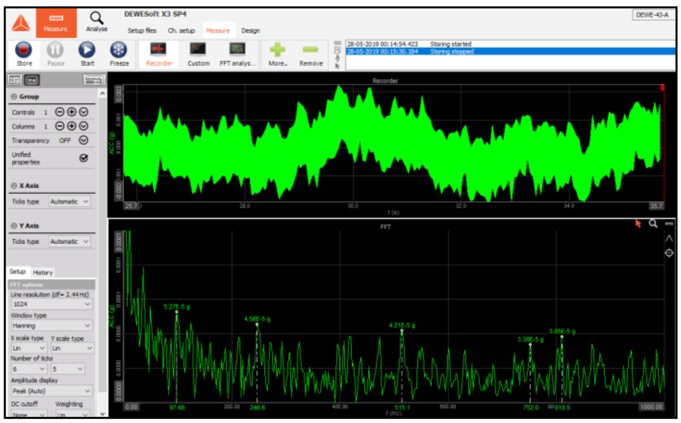

Figure.14 FFT PLOT with single mass on disc

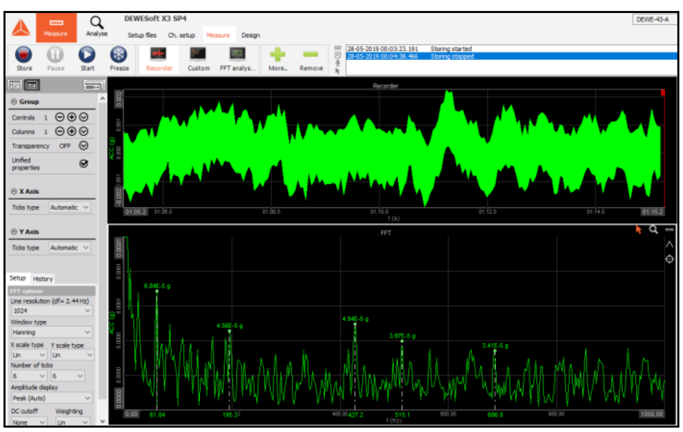

Figure. 15 FFT PLOT with both mass on disc

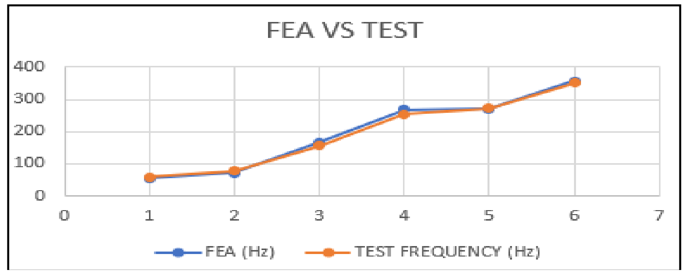

Figure.16 FFT Plot

IX. CONCLUSION

- Model analysis and FFT testing was done for natural frequency of disc
- Natural frequency of FEA results are in good relation with FFT testing as per above graph
- With the help of applying masses on disc maximum frequency obtain at mode 5 with single mass on disc have been seen from table 2.
- Similarly, the above tests can also be done for health monitoring system for other types of applications.

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