Evaluation of a New Balancing Technique for Overhang Impellers through Frequency Response Function

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Abstract: - The main objective of this research paper is to evaluate the new suggested balancing method and compare it to three conventional methods. In this research paper, the details of the test rig facility for the experimental balancing simulator are discussed. Start with describing the experimental test rig components and summarizing the parts of each unit configuration in the investigation of mass unbalance cases. Then, the mechanical waves converted to electrical signal gathered by a set of instruments in balancing and vibration to see the problem and results of corrective action taken. In the end, four balancing methods namely; Conventional software method, four-run methods, mathematical method, and frequency response function method are conducted under different operating conditions. It is found that the new balancing using frequency response function method compared to other conventional methods has some advantages like less downtime required, less risk as no trial weight required, low cost and easy to use.

Keywords: - Frequency response function, Balancing, Conventional software method, four-run method, graphical method.

1. INTRODUCTION

The mass unbalances problem solution theory mainly focused on the addition or removing a certain amount of rotor mass according to the unbalance severity. The process of mass adding or removing occurs along the rotor radius in angular or axial positions to compensate the resultant force of unbalance force which causes the vibration problem for the system.

H. H. Jeffcott [1] shown that the correction of mass unbalance problem needs money and time as the correction consumes much time it costs more from the side of production and maintenance, especially for critical machines. The process partially or depends on balancing efforts.

The ISO standard [2] defined the unbalance as that condition which exists in a rotor when a vibratory force or motion is imparted to its bearings as a result of centrifugal forces. unbalance might be originated during the manufacturing of the rotor where additional mass could be presented or removed at a location of a rotating shaft. Due to erosion between parts, there could be loss of material leading again to an unbalanced condition. They are compensated during commissioning by placing balancing weights. It is not possible to completely balance a rotating system, as there is a small amount of residual unbalance. The system can only be brought to acceptable.

A lot of researches and paper reviews in balancing of rotating machines are available. A wide literature survey introduced by A. Prakinson [3] give a detailed review for more than 70 studies in the previous century on the mass unbalance problem until 1990. Also, W. Foiles et al [4] highlighted a vast amount of studies over 150 references till 1998, where his reviews concentrated on studies of mass unbalance of rotors and covered some of the available correction techniques such as the use of influence coefficient, modal, unified, no phase, and no amplitude methods to balance. The review covers the computational algorithms as well as the physical concepts used in balancing rotating machines.

L. J. Everett [5] provided the influence coefficient model for a rigid rotors which rotates under critical first critical speed and relates measured vibration to unbalance and trial mass magnitude. His studies succeeded to reach for a mean and standard deviation of 19.7% and 12.0%. The results of these statistics and a cumulative normal distribution graph with 96% assert that the optimal method of balancing produces a better residual measure than does the non-optimal technique.

Ambur, R. and S. Rinderknecht [6] suggested a new fault detection method with self-sensing piezoelectric actuators by using the methods of the time domain. A.D.G. Silva et al [7] proposed an alternative balancing methodology for rotating machines presented by. Their work is aiming to overcome the limitations faced by conventionally used methods. The priority is to identify the model of the machine and then solving a typical inverse problem through an optimization method by considering the inherent uncertainties.

B. Xu et al [8] introduced another technique for rotor balancing without test runs which depends on the initial phase point of Holo-spectrum and the balancing objective of the influence coefficient method was developed by. Through measuring original unbalance vibrations and calculating theoretical unbalance responses to optimize the correction masses to minimize residual vibrations at selected measurement locations and balancing speeds which were a new type of intelligent optimization technique

C. Jackson [9] derived the four-run method or four-circle method is considered one of the most common balancing methods today. K. Hopkirk [10] initiated the first steps of using a trial weight in three positions separated by 120o beside the original by. Two plane balancing using only phase technique information. His technique can reach balance by four trial runs beside the initial run. He comprises a two-plane exact-point balance. W. Foiles [11] developed an analytical and graphical solution for single-plane and multi-plane balancing using only phase information.

From the vibration data collected from each end of the machine and used a graphical means, the modal component of the vibration could be determined. J. Lindsey [12] used the sensitivity factor and a high spot number was used to solve static and couple mass unbalance.

T. Majewski [13] using the automatic balancing of a rigid disk mounted on an elastic shaft a study for efficiency and stability of the system was introduced.

D. Rodrigues, et al. [14] presented and analyzed for rigid rotors a two-plane automatic balancing device. D. Rodrigues, et al. [15] in 2011 investigated a single plane automatic balancer investigated experimentally and was fitted to a rigid rotor.

G. Hassan [16] studied a way to minimize the number of trials to reach the accepted balance limits using the analysis to relate the vibration amplitude at one or two rotor bearings to the balancing variables which was named as a black box.

This research paper introduces a new rotor balancing method. Firstly, the experimental apparatus and fault simulation for the study of balancing simulator are discussed. Then, the instruments are used to show the balancing and vibration results through explaining all items. To sum up, the unbalance vibration problem is studied and solved with various conventional balancing methods such as Conventional software method; four run method and mathematical methods these methods were referenced to the new suggested frequency response function (FRF) method.

2. DESCRIPTION OF BALANCING TEST RIG

To evaluate the proposed balancing technique, the experimental studies were executed on a mass unbalancing simulator. The components were three-phase 1.1 Kw induction motor, two rollers bearing (ball bearings), flexible couplings (jaw rubber coupling), rotor dimensions were 250 mm radius and (8, 12, 16 mm) thickness with total weight approximate 4 kg. The rotor machines with 8 balancing locations. The mass unbalance weights (5, 30, 45 and 55 grams), as they appeared in Figure (1) Sketch diagram of the simulator indicates positions of the vibration accelerometer sensors for single plane rotor overhang shown in Figure (2). The test rig was fixed with extra weights on its base (total system weight 65 kg) to increase structure stiffness which transfer the system natural frequency away from normal running frequencies and avoid vibration amplification.

The system was correct for all other vibration sources such misalignment, looseness, soft foot, etc.

The studies of the system vibration response and frequency response function under the follow parameters are effect of unbalance mass value, rotor thickness, running speed.

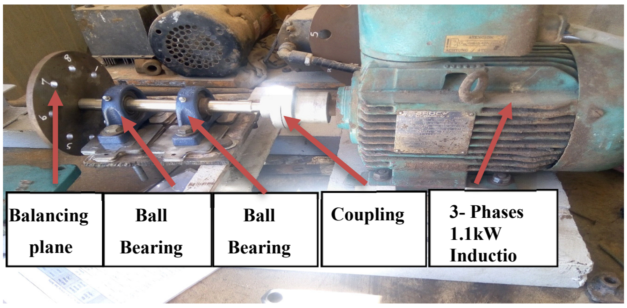


Figure 1. Experimental Test Rig with a Single Overhang Balancing Plane

The balancing rotor was overhang impeller which was widely used in the industry. The structure of the supporting frame was C section under the motor and unbalance plane which were fixed on a heavy multifunction table bed. The mechanical rotation motion of the AC motor was transferred directly to the shaft of the balancing rotor through a flexible coupling.

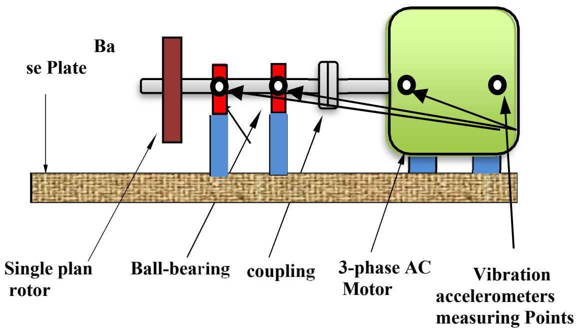


Figure 2. Sketch of the Simulator Indicates the Positions of the Vibration Sensor

3. VIBRATION MEASURING AND ANALYSIS KITS

A piezo-electric tri-axial accelerometer (PCB model 338C04) which was a common transducer in vibration measurements. The main advantages for these sensors are strong signals accuracy in a large frequency ranges to collect accurate vibrations signals from various sources like low-speed shafts which equal fraction of hertz (Hz) to high frequencies in rotor bar pass as several thousand Hz.

Other processes for the transducers used in the vibration measurements were:

- Small output resistance transducers to convert the vibration signals to a voltage signal.
- Avoid the effect of the surface movement through small sensor mass.
- Large frequency ranges (0.5Hz to over 10 kHz).

As shown in Figure 3, The top picture for Comm test Vb8 is a four-channel vibration analysis system is a uniquely sophisticated and feature-packed instrument that remains intuitive in operation and flexible enough to suit every level of vibration analysis, from novice through to expert. The bottom picture in Figure 3 shows the used impact force hammer which helps in measure magnitude of the injected excitation force to the system. The impact force mainly used to develop system or structure mode shape and define the system natural frequency. To distinguish between unbalance types and solve unbalance faults, it is important to measure the Instantaneous Angular Speed (IAS) and relate it to shaft positions for calibration vibration frequencies, a laser tachometer is fitted, as shown in Figure 4. The Assent 2015 software is suitable for accurate and efficient vibration measurements, data import/export, data analysis and reporting of the results.

Figure 3. Comm Test Vibration Meter Device and impact hammer

Figure 4. Mounting and Reading of the Tachometer

4.TEST PROCEDURES

Various aspects related to efficient balance strategies concern the size of the balance correction weights, modal type weights, the optimal balance planes, angular positions, and the number of balance planes required.

To Solve the unbalance problem and take suitable weight addition or removing to reach balancing limits. As the balancing solution needs a synchronous vibration amplitude and phase has to be measured. Moreover, one or more trial weights have to be attached to the rotating part. The machine should shut-down for several times until the balancing limits reached to attach and remove the trial weights till the final correction weights.

Sometimes it is not a problem for small machines which were a less critical machine. On the other side, it's not easy for large machines, especially with critical duties that because a high cost in form, loses of production time, risk of machine startup stresses and overtime costs.

The targets of this study are machines that run under the shaft critical speeds, that are conventionally named the rotor as rigid. For example, there is a technique for balancing machines use one shot for any number of balance planes which commercially named "one-shot balancing". The current suggested method uses a three-channel vibration analyzer, an instrumented force hammer, one vibration-meter and a photo tach. Experimental tests are used to evaluate the use of FRF as a proposed balancing technique to reduce vibration levels within specific limits. Single and two planes balancing Static or couple or dynamic unbalance for a rotor that rotates under critical speeds.

To reach the balancing acceptance limits the principle is easy. If the unbalance resultant force is known in each correction plane and their orientation concerns the rotating part. An approximate force can be generated with the same effect but in the opposite direction by adding weight in a position named light spot or remove weight (material) from a position named heavy spot that to compensate the unbalance force which causes vibration. There is a number of common techniques to find the suitable amount of mass addition/ removing and the most common are:

- Conventional balancing methods which aim at achieving that by using the number of trials run with known trial weights and its fixation positions that aim at determining the machine sensitivity to imbalance loads. With this value, the machine is balanced without repeating all the balancing steps and correct any future unbalance faults as long as no significant change occurs on the machine in the form of operating conditions or machine design.
- This research proposed balancing method using Frequency Response Function (FRF) method, the initial runs using trial weights are exchanged with measured entered force using controlled loading on the machine typically by impacting the machine rotor plane (the impact location in radial direction the vertical and horizontal give the same results measured while the machine in shutdown mode) with an instrumented force hammer and monitor the result at the transducer location in our case the bearing and direction of high vibration and show symptoms of unbalance problem. The required FRF value will be located at the running frequency which is the unbalance problem frequency.

The theory of the suggested method depends mainly on that the analyzer can measure and compare the input load (force) and the output response (vibration) at the same time using cross channel analysis functions. The result of these measurements gives magnitude and phase relationships between the input force and output vibration over a wide range of frequencies including the synchronous component of the machine. From this information introduce an accurate evaluation of the unbalance loads which cause vibration during operation and actual phase lag/lead between input and output. The impact carried out when the machine will shut down for applying the correction weights and thus the machine stopped only once. In this study, Dynamic compliance (μm/N) with displacement vibration unit is used to avoid phase correction conflicts.

The Conventional balancing methods such as balancing program, four-run methods and static couple balancing method balancing results will be compared to frequency response function method. The parameters affect the balancing weight are:

1. Speed of the machine.
2. The radius of weight placement.
3. Rotor mode shape relative to balance plane selected.
4. Proximity to Rotor Balance Resonance (Critical Speed).

4.1. Initial balancing procedures:

1. Initially, eliminate any other problem has the same symptoms of mass unbalance at the machine running speed like resonance, misalignment to deal only with unbalance fault
2. Find a high vibration direction (radial/ axial) to achieve a good balancing job in case of overhang rotors the unbalance problem creates a thrust force in axial direction which appear in high axial vibration beside the radial centrifugal forces.
3. Install a suitable once-per-revolution pulse signal to determine the phase difference between the unbalance fault and the running speed.

4.2. Balancing methods:

When the rotor balancing job is needed, there are two unknown factors which need to be determined to solve a rotor balance problem:

1. The amount of weight required.
2. The angle of weight placement.

These two unknowns are the target of any balancing procedure.

1) Balancing model and procedures for an overhang single-plane rotor:

a) Conventional balancing method using software and procedures:

It depends on the data acquisition system software to solve the mass unbalance problem directly. Add 38 grams into hole number 5 and add 12 grams into hole number 6 will reduce bearing vibration 4 µm p-p (at 25 Hz) which is accepted balancing results.

b) Balancing using influence coefficient method and procedures (mathematical):

1. Measure the Initial Vibration Response A= (437 µm @ 330°) was measured, the trial Weight (TW) was Placed (38 gram @ 225 °)
2. B = Trial Weight Vector = A + Effect of Trial Weight (92.5 µm @ 50°)

C = Trial Weight Effect =B – A C (415 µm @ 150 °) (Draw a line from the head of the A to the head of the B vector. Measure the magnitude of C).

3. Calculate the unbalance influence coefficient:

$$R_{11} = \frac{\text{C (µm @ Angle) response at plane 1}}{\text{Weight at plane 1 TW (Weight @ Angle)}}$$

=11.9 µm /g @ 285 °

4. Calculate the location of the heavy spot:

$$U_{11} = \frac{\text{A (µM @ ANGLE)}}{\text{UNBALANCE AT PLANE 1}}$$

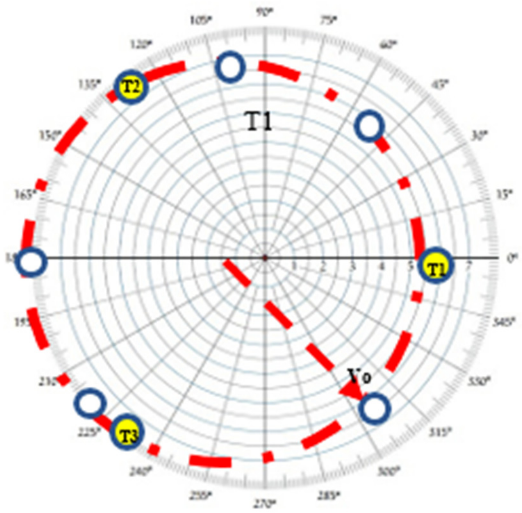
= **7 gram @ 45°**

5. Weight add position = U11 + 180° (Light Spot) = 225 ° (the balance weight of 40 gram installed in hole 5).

c) Four-run balancing method:

Are the simplest balancing method results can easily be derived using polar plot paper and a compass used in balancing single plane static unbalance. The balancing procedures are:

- Measure the original vibration which Vo and draw a circle with radius equal V_o as shown in figure (5) and figure (6) below.
- Impeller hole number 5 is approximately the optimum CW position for the rotor and correction weight was 40 grams, slightly more than the trial weight of 38 grams.



- **Figure 5.** Original Vibration and Trial Weight T Positions T1, T2, T3 on the Rotor

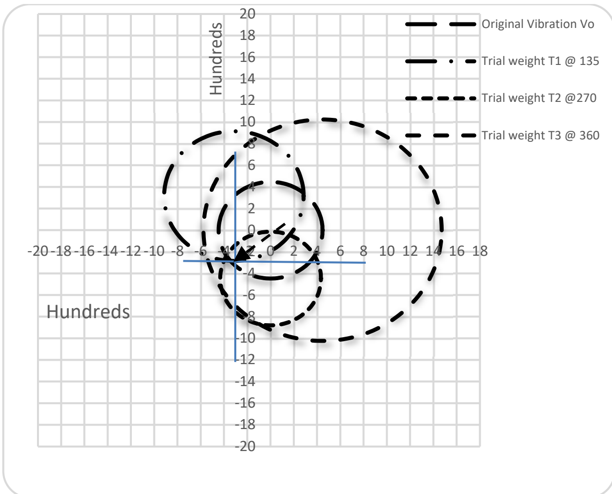


Figure 6. Four-Run Method Results

d) New balancing method

The newly proposed method is very helpful for machines to unbalance fault correction. It depends as mentioned above on the following tools:

1. Two-channel vibration meters one fore vibration transducer and the other for impact force from this combination the frequency response function can be generated as shown in figure 7.
2. The system coherence was checked and it is 92.5 % as shown in figure 8 at the system running frequency 25 Hz (1500 RPM) as coherency is more than 90 % it validates the vibration reading at this frequency for the effective force.
3. From the same coherency check graph, the ratio between the responses in µm to effective force N which is cold dynamic compliance.
4. The phase reading (Ø) on the same graph, represents the phase lag between effective force and vibration response.

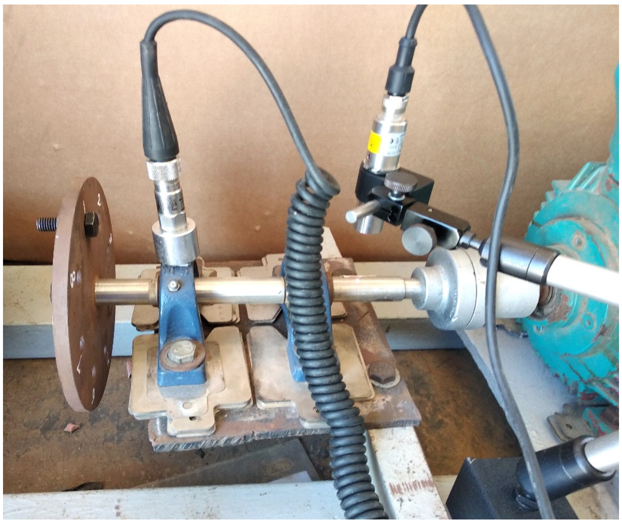


Figure 7. Proposed Balancing System Tools

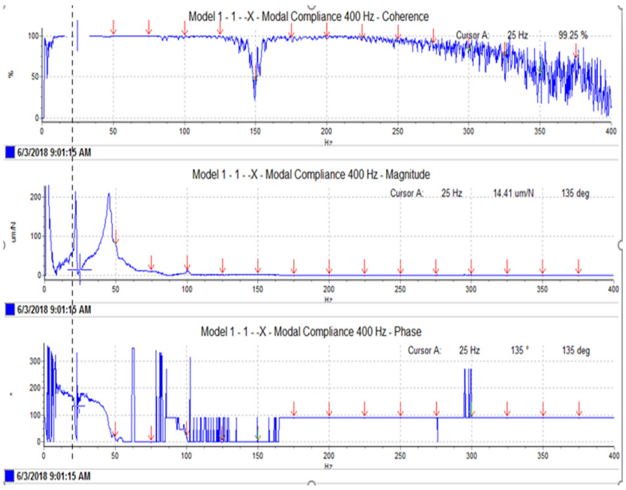


Figure 8. Measured Result for the System

5. Run the system and measure the vibration magnitude Vo and phase (ɵ) by external triggers like tachometer this phase is between the external trigger and vibration response.
6. Multiplying the vibration magnitude Vo in dynamic compliance will determine the unbalance weight value and its location will be the difference between (Ø - ɵ) which is the phase lag between the unbalance force and the vibration response and to remove its unbalance force all we need is to put a similar weight.
7. Impeller hole number 5 is approximately the optimum CW position for the rotor and correction weight was 40 grams, slightly more than the trial weight of 38 grams.

5. THEORETICAL MODEL

5.1. Developing FRF model using analytical functions:

Assume the system is linear.

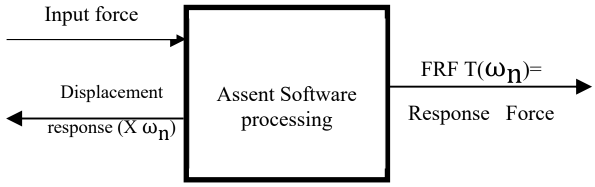


Figure 9. FRF in Signal Processing

where,

$F(\omega_n)$ is the input force as a function of radian frequency;

$X(\omega_n)$ is the displacement response as a function of radian frequency

$T(\omega_n)$ is a transfer function as a function of radian frequency.

(ω_n) is radian frequency

$(\omega_n) = 2\pi f$ (rad/sec)

f is the regular frequency (1/sec).

All these functions represented in terms of magnitude and phase which means each function is a spectral function for simplicity consider each to be a Fourier transform. Representing the relationship in figure (9) by:

$$X(\omega_n) = F(\omega_n) * T(\omega_n)\,, \tag{1}$$

$$T(\omega_n) = \frac{X(\omega_n)}{F(\omega_n)} \left(\frac{m}{N}\right), \tag{2}$$

Equation (2) is the dynamic compliance which is constant as the system is linear for each frequency. In the same way, the other transferred function can be developed according to required response acceleration and velocity considering the phase shift between units as mentioned previously.

5.2. Derivation of theoretical model equations:

Assuming that the system under study is linear and the input and output parameters are measured using previous mentioned sensors equation (1) converted to a state-space model that integrated monitored vectors (measure displacement, velocity, acceleration as shown in figure (8) at the same time the system coherency checked over the measured frequency range to confirm system linearity and result validity.

Consider a single degree of freedom system for simplicity:

$$f_o \sin(\omega t)\frac{\omega_n^{2}}{k} = \ddot{X} + 2\xi\omega_n\dot{X} + \omega_n^{2} X\,, \tag{3}$$

where,

ω_n is the natural frequency in (radians/sec),

ξ is the damping ratio.

The Fourier transformed of each side of the equation (3) taken to drive steady-state transferred function for the absolute response displacement and solved via Laplace transforms as shown in Tom Irvine [17,18] the results of the derivation equation (4) is the required displacement transferred function in equation (2):

$$\frac{X(\omega_n)}{F(\omega_n)} = \left[\frac{1}{k}\right]\left[\frac{\omega_n{}^2}{(\omega_n{}^2-\omega^2)+j(2\xi\omega\omega_n)}\right], \quad (4)$$

From equation (4) to equation (2):

$$T(\omega_n) = \left[\frac{1}{k}\right]\left[\frac{\omega_n{}^2}{(\omega_n{}^2-\omega^2)+j(2\xi\omega\omega_n)}\right](m/N), \quad (5)$$

Equation (5) gives frequency response function in the form of dynamic compliance where displacement divided by force or as mentioned in table (1). The transfer function can represent in terms of magnitude and phase Ø:

$$\left|\frac{X(\omega_n)}{F(\omega_n)}\right| = \left[\frac{1}{k}\right]\left[\frac{\omega_n{}^2}{\sqrt{(\omega_n{}^2-\omega^2)^2+(2\xi\omega\omega_n)^2}}\right], \quad (6)$$

The frequency response function magnitude equal:

$$\left|\frac{X(\omega_n)}{F(\omega_n)}\right| = \left[\frac{1}{m}\right]\left[\frac{1}{\sqrt{(\omega_n{}^2-\omega^2)^2+(2\xi\omega\omega_n)^2}}\right], \quad (7)$$

The frequency response function phase lag between the input force and output response:

$$Tan(\emptyset) = \left[\frac{2\xi\omega\omega_n}{(\omega_n{}^2-\omega^2)}\right]$$
$$\emptyset = Tan^{-1}\left[\frac{2\xi\omega\omega_n}{(\omega_n{}^2-\omega^2)}\right], \quad (8)$$

Equations (7) and (8) are the analytical approach for the proposed method mathematical model. The experimental test rig constants, parameters (such as natural frequency, stiffness, rotor mass, and damping ratio) and measure outputs at a certain frequency (running frequency) were used to validate the theoretical model results out from FRF magnitude and phase lag. After these magnitude and phase lag applied systematically in the proposed balancing method to define the required balancing mass and its location on the rotor.

5.3. Theoretical model run for the same experiment conditions:

System Parameters are:

1. Experimental system damping calculated from Bump test during the machine in the shutdown (static) mode as following:

$$\delta = \frac{1}{n} Ln(\frac{x_o}{x_n}) = 0.66$$

$$\xi = \frac{\delta}{2\pi} = 0.1$$

Where

δ is the amplification factor.

n is the number of selected cycles.

X_o is the magnitude of the first peak.

X_n is the magnitude of the n peak.

ξ is system damping

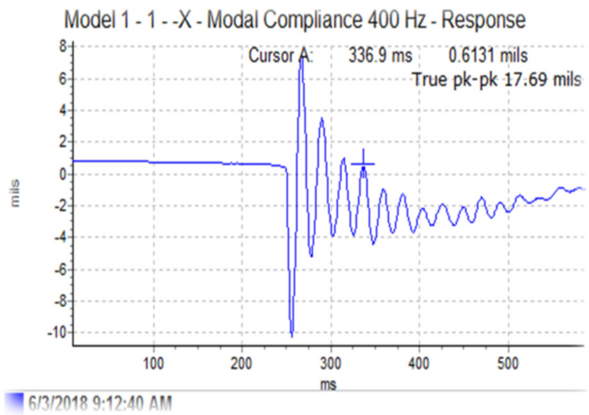


Figure 10. Force Damping over Time

1) System Natural frequency from the Bump test:

ω_n= 2πfn = 281.2 rad/sec

Running frequency (f= 25 Hz) at which frequency response function is required:

ω = 2πf = 157 rad/sec.

2) At rotor mass= 1 kg

Appling equation (7) to define dynamic compliance from the mathematical model:

$$\left|\frac{X(\omega n)}{F(\omega n)}\right| = \left[\frac{1}{\sqrt{(281.2^2-157^2)^2+(2*0.1*281.2*157)^2}}\right]$$

Dynamic compliance magnitude (m/N) = 16 $*10^{-6}$ m/N= 16 μ m/N

To define the phase lag between the input force and output response u degree using equation (8):

$$\emptyset = Tan^{-1}\left[\frac{2*0.1*281.2*157}{(157^2-281.2^2)}\right] \approx \mathbf{150^o}$$

The deviation between theoretical model results and experimental test rig results from the figure (7) shown in table (2) below:

Table 1. Results Deviation:

	Experimental	Theoretical	Deviation %
Dynamic compliance magnitude (μm/N)	14.4	16	11%
Phase lag (degree)	135o	150 o	11%

The Deviation between experimental and theoretical could be related to the next reasons:

1. Theoretical model assumptions such as the system are completely linear.
2. Theoretical equation derivation process.
3. Signal processing noise.
4. Test rig structure not completely linear.
5. Experimental test rig transmitted vibration and noise signal from surroundings.

The experimental results are accurate as it compared previously with other balancing methods and doing well. Regarding the theoretical model results it still need more improvement to meet the expectations.

5.4. Using the theoretical model to study the effect of different damping values on the same system:

The theoretical model applied for a wide frequency ranges from 1 Hz to 1000 Hz system natural frequency at 400 Hz, rotor weight 100 Kg the results graphed for dynamic compliance frequency response function damping value 0.05 on figure 11 and damping value 0.5 on figure 12 phase lag between input force and response on below.

The discussion of the presented results from the theoretical equations indicates that:

1. Increasing rotor weight leads to reduce system dynamic compliance and minimize response to vibration.
2. The effect of changing system damping in figures 11 and 12 from 0.05 to 0.5 look clear around the natural frequency of the system the deviation was not equivalent on both sides. Moreover, the system dynamic compliance differs in two areas on the figure as following before and after natural frequency.
3. The change in system damping has no effect of system phase lag between the input force and response as presented on the figure (13).

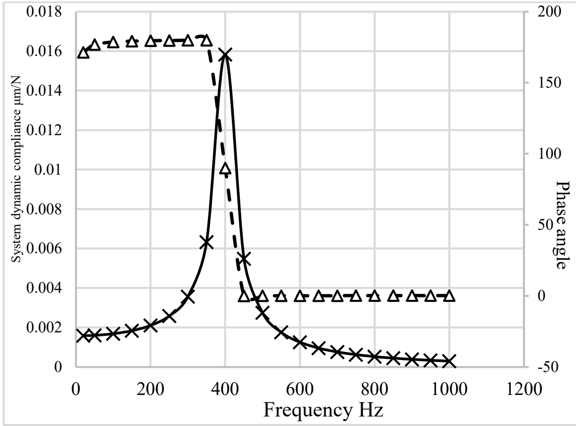


Figure 11. Dynamic Compliance Frequency Response Function and phase angle change for the Frequency Range from 1 to 1000 Hz and Damping 0.05

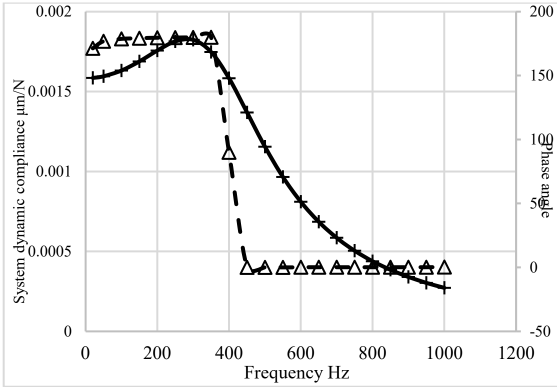


Figure 12. Dynamic Compliance Frequency Response Function and phase angle change for the Frequency Range from 1 to 1000 Hz and Damping 0. 5

6. CONCLUSION

The experimental frequency response function (FRF) results were calculated using impact force hammer as a system excitation input and the response was measured by the accelerometer (piezoelectric sensor model PCB model 338C04). The data was (input and output) delivered to assent software R3, V13.6.7.

Table 2. FRF Display Quantities:

	Response/ Impact force	**Impact force /Response**
Acceleration (g's)	Accelerance (m/s2.N)	Effective mass (s2.N /m)
Velocity (mm/sec)	Mobility (m/ N.s)	Mechanical impedance (N.s/m)
Displacemen t (μm)	Dynamic compliance (μm/N)	Dynamic stiffness (N/ μm)

The theoretical model developed depends on the use of dynamic compliance shown in table (2) where the output response is displacement (m) divided on the input force (N) to avoid the correction of phase reading between response units in balancing process as there is a phase difference between acceleration and displacement 180º and 90º between velocity and displacement.

All the FRF mentioned in table (1) used in vibration analysis and modal testing to identify system natural frequency, damping ratio, mode shape representation and the new suggested use to help in defining the unbalance mass and its location on the rotor using FRF magnitude and phase. The generated FRF is constant for the system at each frequency.

In table 3 below a full comparison made between the balancing methods used in the balancing correction problems for rigid rotors.

Finally, the research work theoretical and experimental results reached to this investigation and analysis of vibration data from the current overhang single plan system configuration. It is possible to conclude that the proposed methods can overcome the other balancing method for overhang rigid rotor with the following benefits:

- The theoretical model supports practical results which means to calculate system dynamic compliance magnitude and phase lag mathematically and use it directly with unbalance measured data original vibration and phase between tacho-meter and running frequency to calculate the balance mass and its location. Even though, it is recommended to execute a practical field measure which is more accurate than theoretical calculations.
- The method takes minimum time in comparison to other balancing methods even theoretical and practical used for the same balancing job.
- Safe and easy to use.
- Low downtime to avoid the effect on production.
- Accurate direct result.
- Low cost.

Table 3. Comparison Between the Four Methods covered in this study:

Item	**Used method** / **Comparison**	**Conventional balancing**	**Influence coefficient balancing**	**4 run balancing**	**Frequency response function (<u>New Proposed method</u>)**
1	**Final vibration after balancing for machine in use**	7 µm p-p (at 25 Hz)	7 µm p-p (at 25 Hz)	7 µm p-p (at 25 Hz)	7 µm p-p (at 25 Hz)
2	**Risk**	Moderate	Moderate	High	Low
3	**Adding trial weight**	Required	Required	Required	Not required
4	**External calculations**	Not required	Required	Required	Not required
5	**Number of shutdowns for the case in study**	Two times	Two times	Four times	One time
6	**The machine test downtime**	60 minutes	60 minutes	180 minutes	30 minutes (Lowest)
7	**Cost of balancing methods in use**	Moderate	Moderate	Low	Moderate

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