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A Study of Single and Two-Plane Balancing Using Influence Coefficient Method

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ABSTRACT

Unbalance effect on rotating element is a condition resulted from an off centered mass concentration about a rotor's rotating centerline which will then generate centrifugal forces with increasing running speeds. This phenomenon will bring about excessive noise and vibration which in turn causing damages to the rotating elements and its support mechanism. To overcome this problem, a correction is required so as to minimize the excessive vibration as low as possible. In this study, unbalance problems including single and two-plane balancing are studied extensively by using theoretical influence coefficient method followed by the experimental verifications. Experimental procedure is performed by using trial mass to calculate the influence coefficient and the corrected mass values as well as the phase angles. The studies cover the static and dynamics balancing for the single plane and two-plane balancing respectively. Finally, the vibration reduction of the rotor is compared theoretically and experimentally. Based on the results, the improved vibration reduction could be obtained reasonably for both single and two-plane balancing by using influence coefficient method.

Keywords: *Vibration, rotor balancing, influent coefficient method, static and dynamic balancing*

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Nomenclature

Y	*Displacement vector due to unbalance force*
F	*Unbalance force vector*
a	*Vibration response in term of velocity vector (mm/s)*
α	*Influence coefficient Vector*
T	*Trial Mass Vector*
m	*Correction Mass*
Y_{11}	*Displacement Vector at point 1 due to force F_1*
Y_{21}	*Displacement Vector at point 2 due to force F_1*
Y_{12}	*Displacement Vector at point 1 due to force F_2*
Y_{22}	*Displacement Vector at point 2 due to force F_2*
L_1	*Vector of Initial vibration readings at left plane*
R_1	*Vector of Initial vibration readings at right plane*
T_L	*Vector of Trial mass at left plane*
T_R	*Vector of Trial mass at Right plane*
L_2	*Vector of vibration reading at left plane due to T_L*
R_2	*Vector of vibration reading at right plane due to T_L*
L_3	*Vector of vibration reading at left plane due to T_R*
R_3	*Vector of vibration reading at right plane due to T_R*
α_{aR}	*Influence coefficient vector of left plane due to T_R*
α_{bR}	*Influence coefficient vector of right plane due to T_R*
α_{aL}	*Influence coefficient vector of left plane due to T_L*
α_{bL}	*Influence coefficient vector of right plane due to T_L*
W_1	*Correction Mass 1*
W_2	*Correction Mass 1*
W_R	*Correction Mass on Right Plane*
W_L	*Correction Mass on Left Plane*
β_1, β_2	*Predetermined angle on rotors*

Introduction

Balancing is a technique of correcting or eliminating unwanted inertia forces or moments in rotating or reciprocating masses and it can be achieved by changing the location of the mass centers. The objectives of balancing are to ensure the center of gravity of the system remains stationery during a complete revolution of the crank shaft and the couples involved in acceleration of the different moving parts balance each other [5]. Unbalance problem is the most common phenomenon occurring in rotating machinery. The implications are quite substantial which may lead to failure such as bearing failures, crack rotors, bent shafts and unpleasant noises [5, 6, 8 and 13]. Consequently, the machines' reliability reduces considerably, hence affecting the machine lifespan [9]. Apart from that, vibrations set up highly undesirable alternating stresses in structures which may eventually lead to structural failure.

Unbalance can happen due to some residual unbalance left in the finished part of the rotating elements caused by slight variation in the density of the material, inaccuracies in the casting and inaccuracies in machining of the parts. Apart from that, it occurs due to the deterioration of one of the part in the rotating machinery such as blades.

In rotating mass system unbalance occurs when the center of gravity *CG* drifted off from the rotational axis, *O* as depicted in Figure 1 [6]. Hence, with rotational speed ω, it creates a centrifugal force, *F*, [5].

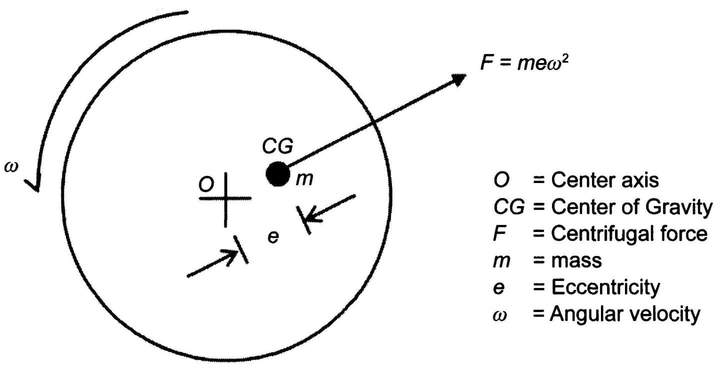


Figure 1: Centrifugal force and acted direction

Centrifugal force acts radially outward away from the rotational axis. The magnitude is affected by three quantities i.e. mass, radius and angular velocity, as in Equation (1). In order to ensure that the rotating mass system is always in balanced state, the centrifugal force should be reduced or possibly eliminated. Thus, balancing is utterly significant for the rotating system as the balancing field accounts for approximately 80% of the rotating equipment problems [7].

$$F = me\omega^2 \tag{1}$$

The primary purpose of balancing is to ensure that the machinery is safe and reliable [5]. Its reduction or elimination will minimize the structural stresses and hence, the machine will operate within the acceptable vibration level [5, 6]. Regular balancing practices could minimize possibility of fatigue failure of the machine's components and also minimize the power loss [6].

There are many types of unbalance practices found in rotordynamics such as static unbalance, couple unbalance, dynamic unbalance and quasistatic unbalance [4, 5, and 6]. In this study, the static unbalance in single plane and dynamic unbalance in two planes will be analyzed theoretically and experimentally. There are two types of balancing category namely on-line and off-line balancing [6]. This study focuses on the rigid rotor off-line balancing, in particular.

With regards to the methods of balancing, there are several methods that have been in practice in rotordynamics including vector method with phase, four-run method, static-couple method, modal balancing, influence coefficient method and unified balancing method [3]. Vector method is a technique of balancing by plotting vibration readings including magnitude and phase angle to a polar graph paper [9]. Then, balance corrections are computed by determining the difference between baseline and trial weight vectors and scaling the resultant to obtain the corrective mass and its phase angle [3]. When the data of phase angle is unavailable, the balancing can still be performed by using four-run method [3]. Contrary to that, in modal balancing the knowledge of lateral critical speed of the rotor and mode shape are of important criteria to be known before further balancing steps could be implemented [5,12]. In this research, the influence coefficient method will be used in order to theoretically calculate the corrected mass required to the rotor at a specified phase angle.

The influent coefficient method has become a powerful tool in solving the balancing problem either in off-line or on-line balancing [10]. This method requires the least number of trials mass compared to modal balancing method. Influence coefficient balancing is the determination of how a unit of unbalance at various mass locations along the lateral axis is reduced by selecting discrete correction masses [6]. In single plane balancing procedure, initially, the rotor will rotate with certain speed without trial mass and the reading of the vibration is recorded. A trial mass is then placed at any angle of the disc and the vibration reading is recorded. The trial mass then is removed and the correction mass can be obtained from the theoretical calculation. On the other hand, the additional vibration measurement is needed in two-plane balancing by virtue of two trial masses are to be attached at two different planes alternately. The ultimate goal of the study is to verify experimentally the theoretical results of influence coefficient method for both single and two-plane rotor system.

Influence Coefficient Method

a. Single Plane Balancing

The rotor of the single-plane balancing is located in the mid of two bearings. This can be represented as in Figure 2.

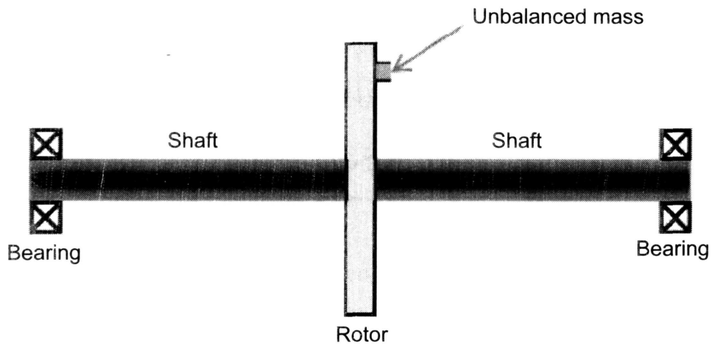


Figure 2: Position of a disc in a single-plane balancing

When the unbalance disc rotates, it will produce centrifugal force. Centrifugal force will always act out of the center of rotation. Static balancing involves resolving primary forces into one plane and adding correction mass in that plane only. Figure 3 depicts the displacement of shaft under a force *F*.

Displacement of the shaft, *Y* due to the force resulted from the unbalance rotation

$$Y = \alpha F \tag{2}$$

Influence coefficient can be obtained experimentally by formula:

$$\alpha = \frac{Y}{F} \tag{3}$$

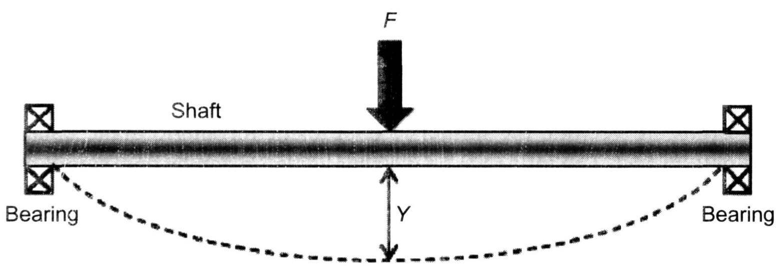
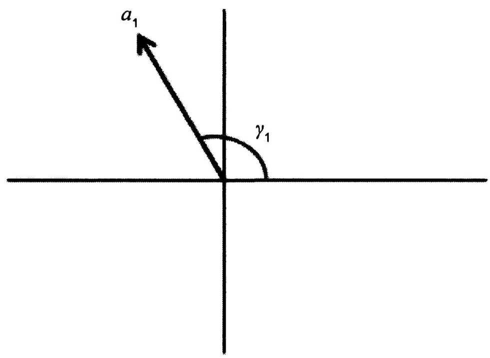
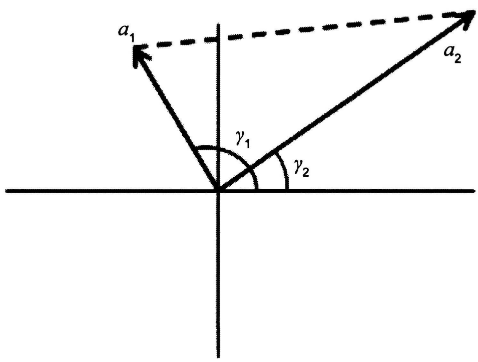


Figure 3: The displacement of a rotor system under a force, *F*

Figure 4 illustrates the vectors involved for the unbalance conditions without and with trial mass.



(a) Initial vibration vector of unbalance rotor (First run)



(b) Vibration vector of unbalance rotor with trial mass (Second run)

Figure 4: Obtaining the influence coefficient from the resultant vector.

From the vectors, the influence coefficient for single plane balancing can be determined. The influence coefficient vector, α can be computed similar to the Equation (3) as:

$$\alpha = \frac{\Delta}{T} \tag{4}$$

In order to balance the unbalanced rotor system, a correction mass, m must be added to the rotor so that the effect of centrifugal force produced from the operation can be eliminated. The placement of the corrected mass must be 180° apart to that of the calculated vibration phase angle. The corrected mass as well as the angle can be determined by the following formula.

$$W = \frac{a_1}{\alpha} \tag{5}$$

Substitution the Eq. 5 into Equation (4) gives rise:

$$W = \frac{a_1}{\Delta} \times T \tag{6}$$

b. Two-Plane Balancing

Figure 5 shows the arrangement of two-plane rotor system.

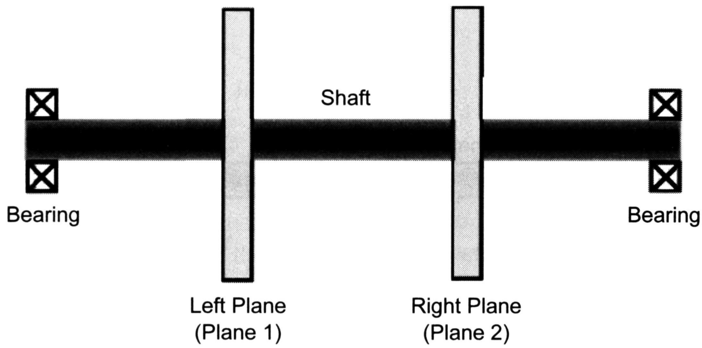


Figure 5: Two-plane rotor system

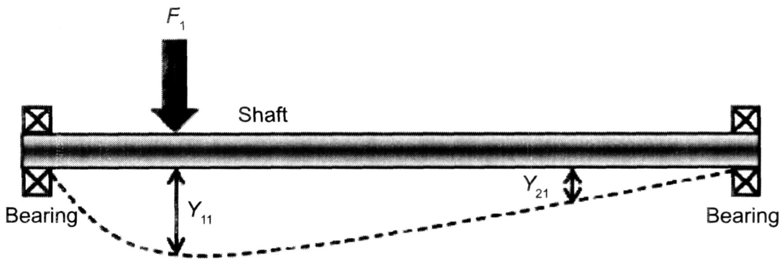
Similar to that of single-plane, the correction of unbalance can be solved by resolving unbalance forces at plane *1* (left plane) and plane *2* (right plane) as shown in Figure 6. In this case, trial masses will be attached to both planes alternately. Initially the plane *1* will be attached by a trial mass, T_L and then followed by plane *2*, denoted by T_R. The effect of this trial mass will give the displacement of shaft as follows.

The displacements of shaft due to forces F_1 and F_2 at both planes can be written as:

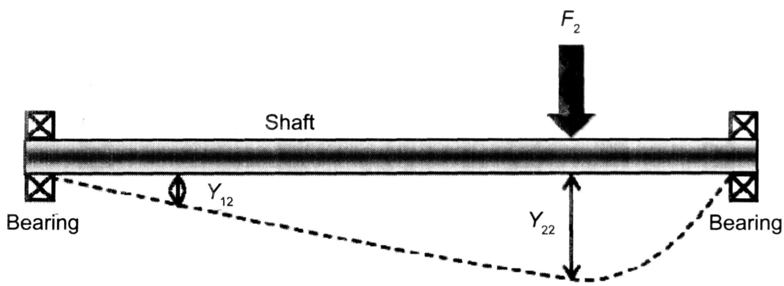
Displacements due to force 1 and force 2:

$$\begin{aligned} Y_{11} &= \alpha_{11}F_1 \\ Y_{21} &= \alpha_{21}F_1 \\ Y_{12} &= \alpha_{12}F_2 \\ Y_{22} &= \alpha_{22}F_2 \end{aligned} \tag{7}$$

By method of superposition, the total displacements at planes *1* and *2* can be written in matrix form:



(a) Displacement of shaft under force F_1



(b) Displacement of shaft under force F_2

Figure 6: Displacements of shaft under two different forces in planes *1* and *2*

$$\begin{bmatrix} Y_1 \\ Y_2 \end{bmatrix} = \begin{bmatrix} \alpha_{11} & \alpha_{12} \\ \alpha_{21} & \alpha_{22} \end{bmatrix} \begin{bmatrix} F_1 \\ F_2 \end{bmatrix} \tag{8}$$

And for the two-plane balancing, the Equation (8) can be specifically written as:

$$\begin{bmatrix} R_1 \\ L_1 \end{bmatrix} = \begin{bmatrix} \alpha_{bR} & \alpha_{bL} \\ \alpha_{aR} & \alpha_{aL} \end{bmatrix} \begin{bmatrix} W_R \\ W_L \end{bmatrix} \tag{9}$$

Where $\alpha_{aR} = \dfrac{L_2 - L_1}{T_R}$, $\alpha_{aL} = \dfrac{L_3 - L_1}{T_L}$, $\alpha_{bR} = \dfrac{R_2 - R_1}{T_R}$ and

$\alpha_{bL} = \dfrac{R_3 - R_1}{T_L}$

Correction mass will produce vibration of equal and opposite to the vibration due to unbalance mass. Hence, the correction mass and appropriate position can be determined by using the formula as follows:

$$-R_1 = W_R\alpha_{bR} + W_L\alpha_{bL} \tag{10}$$

$$-L_1 = W_R\alpha_{aR} + W_L\alpha_{aL} \tag{11}$$

Finally, the corrected masses for two-plane balancing can be computed analytically by the method of vectors as:

$$W_L = \frac{R_1\alpha_{aR} - L_1\alpha_{bR}}{\alpha_{bR}\alpha_{aL} - \alpha_{aR}\alpha_{bL}} \tag{12}$$

$$W_R = \frac{L_1\alpha_{bL} - R_1\alpha_{aL}}{\alpha_{bR}\alpha_{aL} - \alpha_{aR}\alpha_{bL}} \tag{13}$$

The experiment of rotor balancing can be depicted as in Figure 7. The experimental set up consists of rotor equipment, accelerometers, a data collector and a digital balance. Accelerometers are attached at each bearing for the purpose of measuring the vibrations. The data collector then records the measured vibration readings.

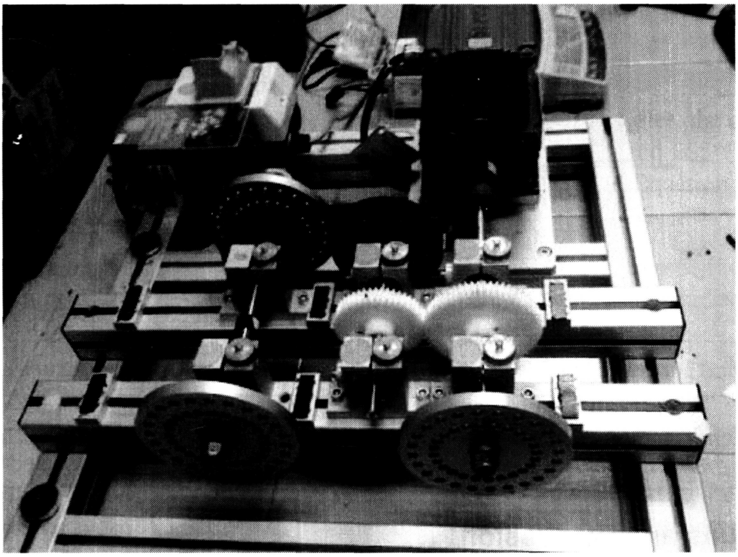


Figure 7: The rotor balancing experiment

Mass Correction Determination

In the experiment, the rotor has a number of drilled holes located at two different radii as shown in Figure 8. Each hole is 15° apart to each other. Should the calculated phase angle obtained from the influence coefficient method does not lie in the designated hole, then the corrected mass at the specified angle, β has to be broke into parts. This can be illustrated as in Figure 9. For this reason, the phase angles (β_1 and β_2) have to be predetermined first and followed by the calculation as below:

$$W_1 = \frac{\sin(\beta - \beta_2)}{\sin(\beta_1 - \beta_2)} \times m \tag{14}$$

$$W_2 = \frac{\sin(\beta - \beta_1)}{\sin(\beta_1 - \beta_2)} \times m \tag{15}$$

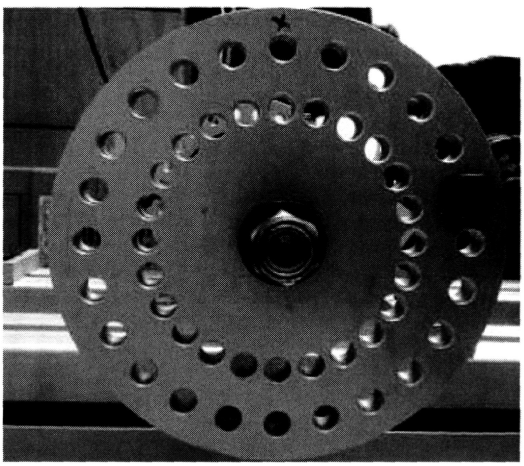


Figure 8: The rotor used in the experiment

The Equations (14) and (15) are also applicable for the two-plane balancing.

Results and Discussion

Both results for single and two-plane balancing are explained at length in the following sections.

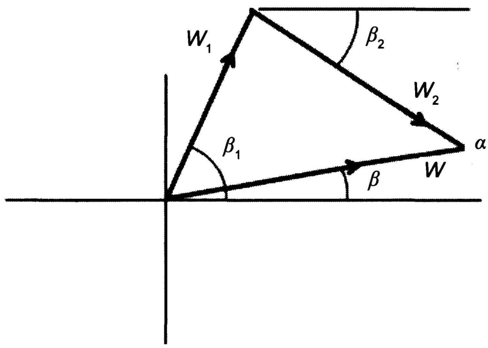


Figure 9: Breaking the correction mass vector into two vectors

Single-plane balancing

The rotor's mass, trial weight radius and rotor's speed for the single-plane balancing are 0.264 kg, 50 mm and 24.72 Hz respectively. Table 1 shows the result of single-plane balancing. In this single-plane balancing experiment, the vibration readings for first and second runs determine the theoretical influence coefficient. From Equation (4), the influence coefficient is:

$$\alpha = 0.1411 \angle -116.8^\circ$$

The trial mass then is calculated based on Equation (5) as:

$$W = 14.37 \angle 227.17^\circ$$

Since the calculated angle does not lie in the designated holes, the mass has to be broke into two parts. The selected angles are 210° and 255°. The respective mass corrections as derived from Equation (14) and Equation (15) are:

Table 1: Results for the single-plane balancing

No. of Run	Weight (g)	Corrected mass (g)	Vibration reading (mm/s)
1	-	-	$2.028\angle 290.37^\circ$
2	$8.17\angle 225^\circ$	-	$1.143\angle 262.28^\circ$
Theoretical	$14.37\angle 227.17^\circ$	$9.49\angle 210.0^\circ$ $6.00\angle 255.0^\circ$	$0.178\angle 336.95^\circ$
Experimental	$14.37\angle 227.19^\circ$	$9.48\angle 210.0^\circ$ $6.01\angle 255.0^\circ$	$0.198\angle 341.25^\circ$

$$W_1 = 9.49\ g \text{ and } W_2 = 6.00\ g$$

Table 1 also shows the experimental results as suggested by the Didactics Test Bench. The initial reading of vibration is 2.028 mm/s and it reduces to 0.178 mm/s and 0.19 mm/s for both theoretical and experimental studies after the balancing is completely done. These figures account for 91.22% and 90.23% vibration reductions for the respective theoretical calculation and experimental identification as tabulated in Table 2.

Table 2: Vibration reductions for the single-plane balancing

	Theoretical	Experimental
% Vibration Reduction	91.22 %	90.23 %

Two-plane balancing

For the two-plane balancing, the rotor's mass, trial weight radius and rotor's speed are 0.5298 kg, 50 mm and 38.12 Hz respectively. The result of experimentation can be tabulated as in Table 3. The initial readings of vibration are found 19.32 mm/s and 4.717 mm/s respectively for the left and right planes. Based on the vibration readings for the first till third runs, the initial readings for the planes *1* and *2* can be computed in terms of influence coefficients and written as:

$$\begin{bmatrix} 4.717\angle 115.71^\circ \\ 19.320\angle 224.26^\circ \end{bmatrix} = \begin{bmatrix} 0.525\angle -45.11^\circ & 1.039\angle 165.40^\circ \\ 0.266\angle 221.22^\circ & 0.786\angle -2.51^\circ \end{bmatrix} \begin{bmatrix} W_R \\ W_L \end{bmatrix}$$

Solving the above matrices give:

$$W_R = 9.16\angle 79.25^\circ \text{ and } W_L = 6.94\angle 89.06^\circ$$

The masses are then broke into two for each plane and the result of the calculated correction masses can be seen in Table 3. The vibrations reductions recorded are 6.13 mm/s and 1.172 mm/s for the respective left and right planes by the ICM, which is by the theoretical determination. On the other hand, the vibration readings reduce to 6.42 mm/s and 1.241 mm/s for the respective left and right planes by the experimental identification. Those figures can be summarized in Table 4 in terms of the vibration reductions. All in all, the theoretical results show better vibration reduction compared to the experimental identification. Though, the discrepancies are slight in percentages.

Table 3: Results for the single-plane balancing

	Trial mass (g)		Correction Mass (g)		Vibration (mm/s)	
No. of Run	*Plane 1*	*Plane 2*	*Plane 1*	*Plane 2*	*Plane 1*	*Plane 2*
1	No	No	-	-	19.32∠224.26°	4.717∠115.71°
2	9.67∠90°	-	-	-	2.85∠240.56°	7.98∠78.81°
3	-	5.27∠135°	-	-	19.63∠212.09°	0.865∠326.85°
Theoretical	9.16∠79.25°	6.94∠89.06°	3.42∠60° 6.04∠90°	3.81∠75° 3.37∠105°	6.13∠320.52°	1.172∠63.23°
Experimental	9.16∠79.26°	6.94∠89.07°	3.41∠60° 6.04∠90°	3.81∠75° 3.38∠105°	6.42∠290.05°	1.241∠29.31°

Table 4: Vibration reductions for the two-plane balancing

	Theoretical		Experimental	
	Plane 1	*Plane 2*	*Plane 1*	*Plane 2*
% Vibration Reduction	68.00 %	75.00 %	67.00 %	74.00 %

Conclusions and Recomendations

The study carried out has successfully verified the experimental results for both single and two-plane balancing. The results of corrected masses and phase angles obtained for this research are best applied for the experimented running speed of the rotor. However, for a rotor that operates at a particular range of speeds, the research has to be done for some various speed conditions. The variation of vibration readings can then be leveraged by the least-squares method [11]. Of course, the percentage of vibration reductions achieved for a specified range of speeds would not be the same and definitely be lowered than that obtained in this research. Furthermore, for more improved vibration reductions, it is suggested that the rotor be operated as close as possible to the natural frequencies of the rotor system [14].

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ABSTRACT

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$$y = A + Bx + Cx^2 \tag{1}$$

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[2] H. Coelho and L. M. Pereira, "Automated reasoning in geometry theorem proving with Prolog," J. Automated Reasoning 2 (3), 329-390 (1986).
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