Effects of Residual Imbalance on the Rotordynamic Performance of Variable-Speed Turbo Blower

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Abstract

In the variable-speed turbo machinery, it is important to ensure the reliability in operation speed of the large area unlike the constant speed rotating machine. In particular, as a study on the performance maintenance and minimal vibration, the high-speed balancing and field balancing are necessary to remove vibrations due to the unbalanced mass. In using a dedicated balancing device, to add or remove the weight is often not obtained satisfactory results for expanded the range of variable rotating speed. Thus, in order to improve the performance and driving stability of variable speed machine, it has to make the residual imbalance to a minimum, through the field balancing.

This study presents the experimental results to apply the two-plane multi-speed balancing techniques, and ensure the driving stability by reducing the residual imbalance. This paper also shows rotordynamic simulation results of the rigid rotor and bearing system, including damped critical speed and imbalance responses. The operating maximum speed of the turbo blower is up to 20,000 rpm, usually it works normally in the 4,000 ~ 17,000 rpm as the variable-speed region. And this study was tried to reduce the residual imbalance vibration in all operating speed regions. First, by utilizing an influence coefficient method the rotor was adjusted to the ISO 1940 standard grade G1. Additional, as the least square method, the experiment was carried out to minimize the residual imbalance by applying the multi-speed balancing in the field. The first rigid mode was predicted in the near 17,500 rpm, the experimental results showed that rigid mode is consistent. Accordingly, the field balancing was performed at several points of the rigid mode speed region. This study was an effort to reduce excessive vibration due to the rigid mode by minimizing the residual imbalance. As a result, the vibration caused by the residual unbalance can be reduced to 70%, and experimentally verified that this can effectively improve the rotordynamic performance.

Keywords

Variable-speed turbo machinery — Residual imbalance — Influence coefficient method — Least square method — Rotordynamic performance

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INTRODUCTION

According to high-speed rotating machine market increase, recently there is required a variable-speed turbo machine for energy saving and efficient operation [1]. Basically the rotor bearing systems should be designed to avoid the critical mode from the operation region. In addition, performing the high-speed balancing is an essential requirement for satisfying durability and allowable vibration range of the turbo machine [2]. Even though the design of the rotor completely, in actual industrial applications, imbalance vibration of the system can be increased due to the misalignment of the rotor assembly. Especially, the variable-speed turbo machine must be ensure the stability of the system with high precision balancing of the various speed. This balancing technology is largely divided into a modal technique and influence coefficient method. Modal technique has the effect of reducing the amplitude of the residual vibration of imbalance occurring in each mode by using the orthogonality of the modes [3-5]. However, there are some shortcomings to know the critical speed of the rotating shaft and the corresponding mode in advance. On the other hand, influence coefficient method does not require prior information of the rotor mode. It is a method of measuring magnitude and phase as installed a trial mass in sequence to each balancing plane. It can obtain the correction mass and angle from calculation of the residual imbalance at the desired speed [6]. The influence coefficient method is also sensitive to other system parameters which can result in an ill-conditioned influence coefficient matrix thereby producing erroneous corrections weights [7]. Further, when utilizing least square method proposed by Goodman [8], this approach allowed vibration at multiple locations and multiple speed conditions. Palazzolo [9] also presents to be effective in reducing the overall vibration level by minimizing the sum of squares of the residual vibration. Moreover, multiple balance planes can be accommodated by expanding this equation which sets to zero the vibration magnitude at as many locations or speed conditions as balance planes are available. It is known as the exact point balancing procedure by Tessarzik and Badgley [10-13]. This method has been widely applied to balancing of large power generation turbines with many rotor critical speeds and bearing points to minimize vibration [14, 15]. It provides good results as long as the vibration effects at all points and speeds. As a case of practical study, Gunter [16] was provided the practical aspects of multi-plane field balancing 70 MW gas turbine generators. The optimum balance was determined by means of a constrained least squared error multi-plane program using vibration data at the various speeds and power levels.

This study describes the experimental results relating to remove residual imbalance over the entire speed range, including the operation speed. In order to satisfy the condition that minimizes the residual vibration in all driving regions of the variable-speed turbo machine, two-plane multi-speed balancing technology was applied in the field. Since the environment and condition of the turbo system applied to the different industrial field, and large complex turbo system is very difficult to assemble repeatedly. Then, it should be perform the high-speed precision balancing through trial and error in the field directly. The least square method was easily utilized in order to minimize the residual unbalance from the measured vibration. In particular, the speed selection is very important for vibration reduction desired and rotordynamic performance enhancing. Least square method can minimize the sum or average of the squared residual vibration data. However there is a limitation. When the outlier data was a mixed, it can make the wrong result. Thus, multi-speeds selection can be achieved experimentally for field balancing optimization. These can choose in some speed from the rigid mode vibration amplitude peak. By selecting within 10% from the peak of vibration amplitude, when the highest and low of two speed point, this experiment was effective in applying the least square method. Therefore, it could effectively reduced not only the vibration of the corresponding mode speed, also the vibration of the entire operating speed range. On the other hand, applying a lot of data, the balancing effect is decreased. Lastly, the rotordynamics simulation was performed with respect to the residual unbalance to obtain the test. By comparing the dynamic behavior of vibration analysis and balancing a result, it shows that can be minimized of the vibration response with residual imbalance of variable-speed turbo blower.

1. DESIGN AND MODE ANALYSIS OF THE RIGID ROTOR FOR VARIABLE-SPEED

The impeller of overhung structure is driven by a permanent magnet synchronous motor, both ends supported by the ball bearings as shown in Figure. 1. The total length of the rotor is 646.5 mm, weight is 21.25 kg. A finite element model of the rotor was performed base on the natural frequency analysis, the first bending mode is 1,068 Hz (64,071 rpm). In this study, since the maximum operation of the variable speed up to 20,000 rpm, the design margin is sufficient for the bending mode.

Ball bearings in the 7200 series, the inner diameter is 40mm, allowable speed of oil-lubricated is 29,000 rpm. The stiffness of the bearing is 6×10^7 N/m. The quasi-static stiffness is calculated in Ref.[17]. The rotordynamics analysis was performed by using these values, Figure 2 shows the campbell diagram for natural frequency analysis map. The results of nutural frequency analysis, backward is 17,566 rpm and forward is 19,467 rpm of the first rigid mode (cylindrical mode). The second rigid mode (conical mode) backward is 22,548 rpm and forward is 25,979 rpm. On the basis of this analysis result, during variable speed operation, the rotation stability may be concerned that excessive vibration of the bearings in the rigid mode of 17,000 ~ 20,000 rpm. In order to solve this problem, there are several ways. It is a method of raising the bearing mode to more than 20,000 rpm by increasing the bearing stiffness or axial preload. Alternatively, a method of re-design of the rotor bearing systems. But the redesign can be quite a waste of time and money, including the aerodynamic design or motor part. The last option, to apply a high-precision balancing method for reducing vibration of a desired operating speed range. Therefore, despite the limited condition, it is the best choice in a manner to prevent excessive vibration near the rigid modes.

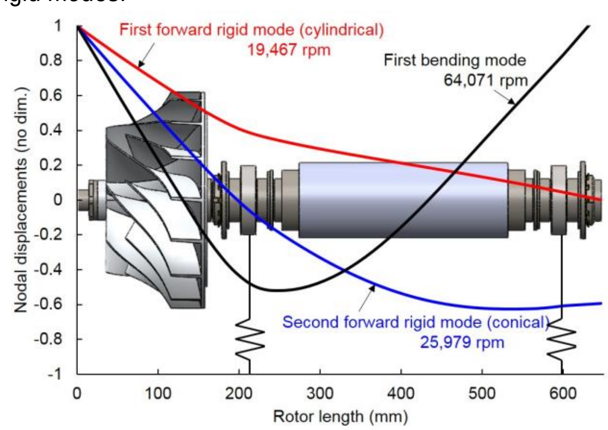

Figure 1. Rigid rotor and mode shape analysis results.

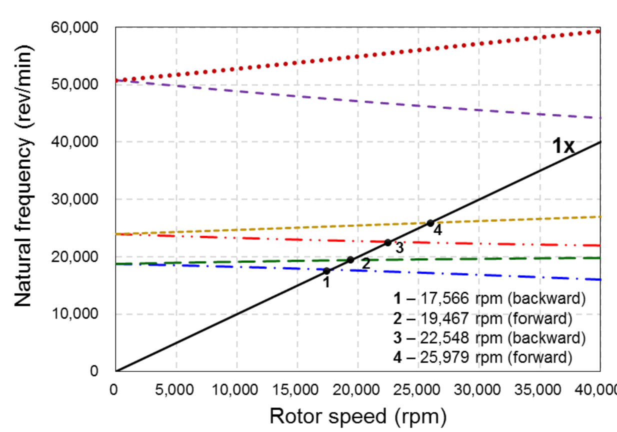

Figure 2. Campbell diagram for natural frequency analysis map.

2. INFLUENCE COEFFICIENT METHOD

The two plane balance corrections were calculated by peak to peak vibrations and phase measurements. This procedure is still being used with success and it is in essence the fundamental concept in which the influence coefficient method was derived for multi-speed rotor systems. An influence coefficient method with an optimal correction balancing algorithm is developed for balancing a high-speed rotor system.

The influence coefficient procedure [18] for final balancing of a rotor at operating speed is as follows:

1. We choose two correction planes A and B, two measurement planes a and b. To obtain phase data, we need to install keyphasor.
2. Take the zero rotor vibration data, the rotor frequency vibration amplitudes and phase angles of all the planes for the original condition of the rotor.
3. Obtain the trial mass data of A plane and B plane. To get the trial mass data, a trial mass is inserted in one balance plane, and the resulting rotor-frequency vibration amplitudes and phase angles of all the planes are measured. This procedure is repeated for the other balance plane.
4. Calculate the response coefficients. The equations of influence coefficient (α) is bellow. In this study, we selected the balancing planes and measurement points as shown in Figure 3.

   A1, A2, A3 are measured vibration data of zero rotor, first trial data, second trial data in A plane. And B1, B2, B3 are measured vibration data of zero rotor, first trial data, second trial data in B plane. Also, TA and TB are first trial mass and second trial mass. The $[\alpha]$ and $[w]$ are influence coefficient and corrected balance mass, respectively.

$$\begin{Bmatrix}\bar{B}_1\\ \bar{A}_1\end{Bmatrix} = -\begin{bmatrix}\bar{\alpha}_{bB} & \bar{\alpha}_{bA}\\ \bar{\alpha}_{aB} & \bar{\alpha}_{aA}\end{bmatrix}\begin{Bmatrix}\bar{w}_B\\ \bar{w}_A\end{Bmatrix} = -[\alpha]\begin{Bmatrix}\bar{w}_B\\ \bar{w}_A\end{Bmatrix} \tag{1}$$

Where,

$$\begin{aligned}\bar{\alpha}_{bB} &= (\bar{B}_2 - \bar{B}_1)/\bar{T}_B\\ \bar{\alpha}_{aB} &= (\bar{A}_2 - \bar{A}_1)/\bar{T}_B\\ \bar{\alpha}_{bA} &= (\bar{B}_3 - \bar{B}_1)/\bar{T}_A\\ \bar{\alpha}_{aA} &= (\bar{A}_3 - \bar{A}_1)/\bar{T}_A\end{aligned} \tag{2}$$

5. By using the influence coefficients to obtain a set of correction masses in the two balance planes that will reduce the residual vibration of the rotor.

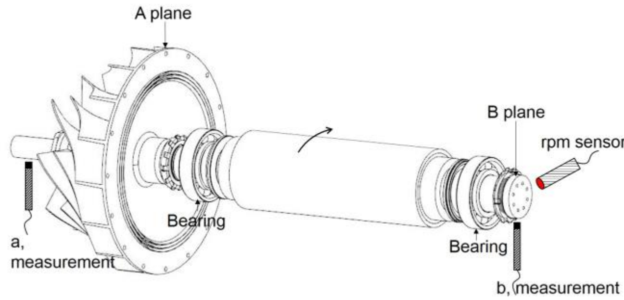

Figure 3. Indicated correction planes and vibration measurement points for balancing.

In addition such as least square balancing algorithm can be satisfied allowable residual vibration levels in overall speed ranges. The number of balancing plane is n, the number of desired balancing speed is l_1, and the number of position measured vibration is l_2. The number of measurement m is below.

$$m = l_1 \times l_2 \tag{3}$$

When m is larger than n, the correction mass vector $\{W\}$ relates to the unbalance vibration vector $\{A\}$ as below.

$$[\alpha]\cdot\{W\} + \{A\} = \{0\} + \{\varepsilon\} \tag{4}$$

Where $[\alpha]$ is the influence matrix of m x n, {0} is a zero vector of nth row and {ε} is the residual vector. As residual vector {ε} is lower, we can obtain the better performance of balancing. Therefore, the least square of residual vibration is as follow.

$$\begin{aligned}&\{\varepsilon\}^T \cdot \{\varepsilon\} \Rightarrow Min.\\ &= \varepsilon_1 \cdot \varepsilon_1^{\ *} + \varepsilon_2 \cdot \varepsilon_2^{\ *} + ...+ \varepsilon_m \cdot \varepsilon_m^{\ *} \Rightarrow Min.\\ &= \sum_{i=1}^{m}\{[Re(\varepsilon_i)]^2 + [Im(\varepsilon_i)]^2\} \Rightarrow Min.\end{aligned} \tag{5}$$

Where $\{\varepsilon\}^T$ is a transpose of {ε} and composed of a conjugation complex numbers. By inserting (5) into (4), we can obtain the following equation.

$$\begin{aligned}&[\alpha]^T[\alpha]\cdot\{W\} + [\alpha]^T\{A\} = \{0\}\\ &[\alpha]^T[\alpha]\cdot\{W\} = -[\alpha]^T\{A\}\end{aligned} \tag{6}$$

As the equation (6) is n square matrix, it is possible to calculate inverse matrix. Thus, the equation (6) is converted to correction mass $\{W\}$.

$$\{W\} = -\{[\alpha]^T[\alpha]\}^{-1}[\alpha]^T\{A\} \tag{7}$$

3. ZERO ROTOR DATA AND BALANCING SPEED CONDITION DETERMINATION

In order to apply influence coefficient the method, two of correction plane was determined as shown in figure 3. In addition, as part of a multi speed balancing technique, to reduce the residual unbalance of the rigid mode the speed point should determine from zero rotor data.

In the A and B plane, a distance of center and trial mass position is 120mm and 15mm respectively as shown in Figure 4. By using balancing dedicated machine, each residual imbalances of A and B plane are 172.44 g-mm at 167.66˚ and 92.25 g-mm at 178.07˚. The imbalance response simulation is performed with above data. This result shows in Figure 5. It indicates that a vibration increases caused by residual imbalance in high speed. The general speed of the variable-speed turbo blower is from 4,000 to 17,000 rpm, and a maximum operating speed is 20,000 rpm. The balancing speeds can choose in some speed from the rigid mode vibration amplitude peak. It was selected within 10% from the peak of vibration amplitude, the speed of highest peak is 19,500 rpm and the speed of low vibration amplitude is 17,000 rpm. In the critical speed range shown in Figure 5, the balancing speed point of four case that the following was optional determined.

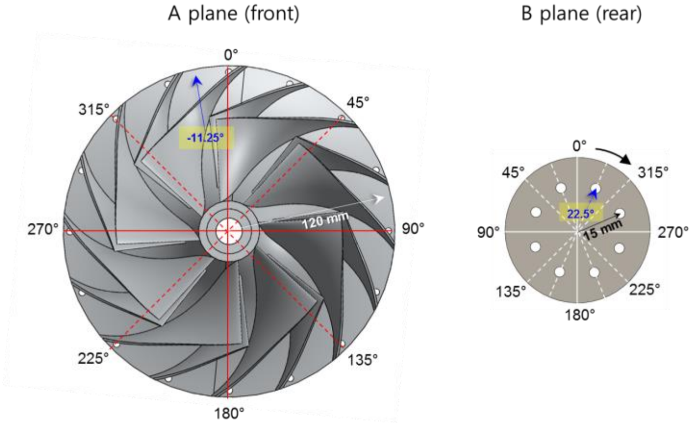

Figure 4. Schematic view of correction A and B plane.

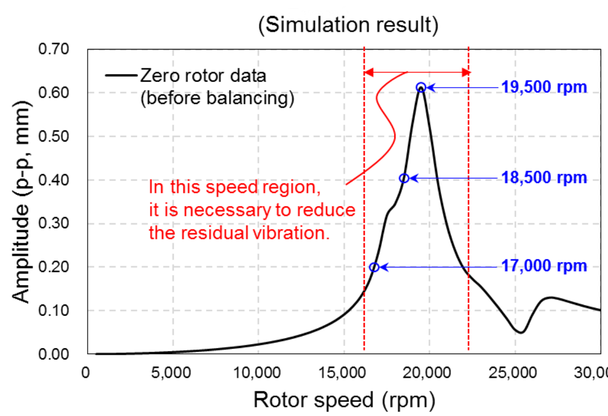

Figure 5. Imbalance response simulation result at 'a' plane.

- Case 1 – 17,000 rpm, 18,500 rpm
- Case 2 – 17,000 rpm, 19,500 rpm
- Case 3 – 17,000 rpm, 18,500 rpm, 19,500 rpm
- Case 4 – 17,000 rpm, 19,000 rpm, 20,000 rpm

4. EXPERIMENT RESULTS

Table 1 shows the data of inserted trial masses accordance with run type. Trial mass of A plane is 1.369 g at -11.25˚ and trial mass of B plane is 2.039 g at -22.5˚. In fact, A plane, trial mass were inserted into 0.697 g at 0˚ and -22.5˚, respectively. This can be seen to try 1.369 g at -11.25˚ by vector calculation.

Table 2 shows the measured imbalance vibration data of each rotating speed. In accordance with the trial mass inserted, both of the vibration and phase angle were measured. Hence, the residual imbalances can be calculated based on the previously determined multi-speed case by utilizing least square method.

Table 3 shows the residual imbalance results of multi-speed balancing cases. The results was obtained the residual imbalance vibration in each multi-speed case by applying the influence coefficient and least square method. This study focused on the vibration of A plane, because the rotor has heavy load on one side, as overhung type. According to the analysis of the imbalance calculation for each multi-speed case, the lowest residual imbalance result can be obtained in the case 2 that 17,000 and 19,500 rpm are selected. On the other hand, when comparing the cases 2 and 3, 18,500 rpm data is being added to the least square calculation. The residual unbalance of case 3 was not significantly reduced as much as case 2. This mean that the data of 18,500 rpm does not help in finding the optimal imbalance reduction. For multi-speed balancing, it is very important to determine the speed because an influence coefficient method is dependent in the speed. The least square method can minimize the sum or average of the squared residual vibration data. However there was an inappropriate data mixed, it cannot make the optimal result. Accordingly, this study was selected for optimum balancing speed as in case 2. In another cases, it did not appear less residual imbalance than case 2, Further, many multi-speed points and influence coefficient are not getting good balancing effect.

Table 1. Data of inserted trial masses

Run Type	Plane Containing trial mass	Trial mass [gram]	Trial mass location [deg.]
Zero rotor data	-	-	-
Trial mass data	A	1.369	-11.25
Trial mass data	B	2.039	-22.5

Table 2. Measured imbalance vibration data

Zero rotor	'a' plane		'b' plane	
Rotating speed [rpm]	Magnitude [p-p, mm]	Phase angle [degree]	Magnitude [p-p, mm]	Phase angle [degree]
17,000	0.1750	-179.2	0.0091	134.9
18,000	0.1838	-170.9	0.0313	137.3
18,500	0.2472	179.4	0.0207	118.9
19,000	0.4425	-179.6	0.0243	95.7
19,500	0.6830	-160.7	0.0360	102.6
20,000	0.6905	-150.6	0.0383	99.2
Trial A plane	'a' plane		'b' plane	
Rotating speed [rpm]	Magnitude [p-p, mm]	Phase angle [degree]	Magnitude [p-p, mm]	Phase angle [degree]
17,000	0.1527	-24.7	0.0245	-172.4
18,000	0.1395	-0.1	0.0147	-130.7
18,500	0.1285	-19.8	0.0121	179.8
19,000	0.1882	-18.6	0.0192	178.2
19,500	0.2847	18.2	0.0119	179.9
20,000	0.2705	55.4	0.0234	138.7
Trial B plane	'a' plane		'b' plane	
Rotating speed [rpm]	Magnitude [p-p, mm]	Phase angle [deg.]	Magnitude [p-p, mm]	Phase angle [deg.]
17,000	0.3343	177.6	0.0308	42.0
18,000	0.3668	-173.1	0.0384	81.6
18,500	0.4536	-180.0	0.0385	73.2
19,000	0.6125	-175.8	0.0469	68.7
19,500	0.7713	-154.2	0.0493	79.0
20,000	0.7650	-147.2	0.0601	71.3

Table 3. Residual imbalance data of multi-speed balancing

	Speed [rpm]	A plane		B plane	
		Magnitude [p-p, mm]	Phase [deg.]	Magnitude [p-p, mm]	Phase [deg.]
case1	17,000	0.2137	-23.7	0.1184	179.3
	18,500	0.1881	158.4	0.0933	129.5
case2	17,000	0.0124	79.4	0.0364	-154.4
	19,500	0.0045	-38.8	0.0867	87.3
case3	17,000	0.2375	-25.9	0.092	-163.5
	18,500	0.2016	157.4	0.0514	121.2
	19,500	0.0022	104.3	0.0843	114.8
case4	17,000	0.3141	21.9	0.1256	-113.1
	19,000	0.3668	-137.5	0.0149	157.1
	20,000	0.1423	95.3	0.0932	89.0

Table 4 shows the correction mass and angle calculated by the vibration and the phase measured in each case. This result can be carried out vibration response test by applied correction mass to the corresponding hole. On the other hand, correction's phase angle was not exactly be applied because it can only put mass into correction plane with intervals of 22.5˚. By a vector calculation, the experiment was tried to be as close as possible to apply correction results.

Table 4. Correction mass and phase angle data in each multi-speed balancing cases

	A plane		B plane	
	Correction mass [gram]	Phase angle [deg.]	Correction mass [gram]	Phase angle [deg.]
case1	0.937	-2.29	0.271	-17.80
case2	1.045	-9.50	1.006	-49.92
case3	1.015	-10.56	0.601	-61.15
case4	1.080	-19.92	1.343	-90.75

The results of case 2 and case 3 has the smallest residual imbalance by the residual imbalance calculations. Thus, the operating test of case of 2 and 3 were performed by applying the correction mass, respectively. Figure 6 provides that simulation and experimental result of imbalance vibration responses are compared. There are three cases represented a set of experiments (zero rotor data, correction data of case 2, correction data of case 3). The vibration measurements were performed on the 'a' and 'b' plane as shown in Figure 3, it was expressed as front and rear, respectively.

Overall, experimental results and simulation trend have shown that good agreement. In particular, it can be confirmed that large vibration occurred in the rigid mode speed region. This imbalance responses were also found that it is considerably matching simulation and experiment results. Structurally of the rotor, the vibration in the rear just shows runout level due to assembling and production tolerances. Thus, the present research has focused on the front vibration level of the impeller position. Experimental and analytically, excessive vibrations of the near 17,000 rpm were successfully verified that approximately 70% decrease by applying the correction result of the case 2. Further, it showed the effect of the influence coefficient experimentally in the determinant speed, to obtain results that can significantly reduce the variation in the high-speed. It could be reduced not only the vibration of the corresponding mode speed, also the vibration of the entire operating speed range. By comparing the rotordynamic behavior of vibration analysis and balancing result, it shows that can be minimized of the vibration response with residual imbalance of variable-speed turbo blower.

Finally, Figure 7 is presented the orbital vibrations during the speed-down. From 20,000 to 16,000 rpm, vibration obit is shown the effect of residual imbalance on the operating performance in each balancing case. The case 2 can be proven that the good performance of the vibration, also the speed determination is the most important decisions in order to obtain an excellent balancing effect of influence coefficient.

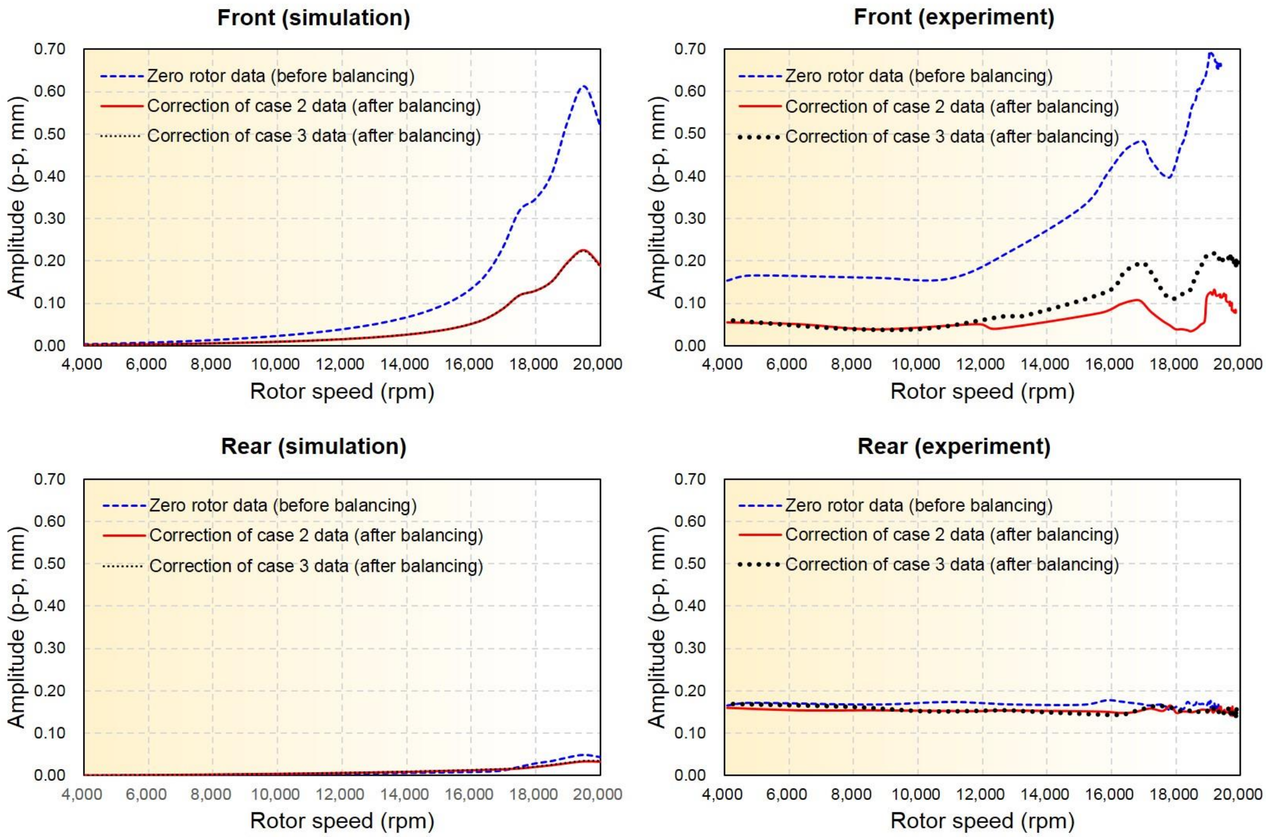

Figure 6. Comparison of simulation and experimental results with balancing (during the speed-up).

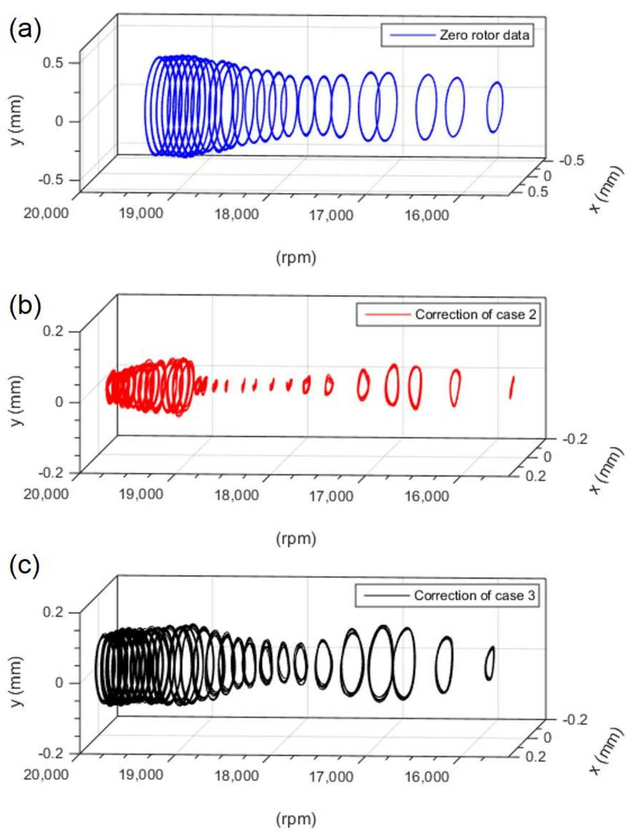

Figure 7. Orbital vibrations in impeller side during the rotor speed-down: (a) zero rotor data, (b) correction of case 2 and (c) correction of case 3.

5. CONCLUSION

This paper has provided a successful experiment and simulation results for verifying residual imbalance effect and rotordynamic performance with two-plane multi-speed balancing by applying influence coefficient method. A rotor bearing system application of a variable turbo blower, experiments were conducted to ensure rotational stability. Base on the least square balancing technique, the experiment was determined the multi-speed point. It was measured peak to peak vibration and phase angle at each speed, according to the multi-speed condition, calculated the residual unbalance and correction value. The rotational speed of the variable speed application turbo machine is up to 17,000 rpm, the maximum rotational speed is 20,000 rpm. Thus, the present study was tried to reduce the vibration near 17,000 rpm in field. This experiment showed the effects that can reduce the vibration more than 70%. In conclusion, for the most effective use of influence coefficient, it is very important to determine the balancing speed. Based on the balancing effect of the residual imbalance was demonstrated that to obtain a rotational stability of the variable-speed turbomachinery.

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