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Article

A speed-variant balancing method for flexible rotary machines based on acoustic responses

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Abstract: As rotary machines have become more complicated, balancing processes have been classified as a vital step in condition monitoring to ensure machines operate both reliably and safely. This is especially important for flexible machines which normally work at rotations speeds above critical limits. Imbalance is a common problem in flexible rotating machinery that can lead to extreme vibration and noise levels. This is one of the major reasons for studying various balancing methods applied to the vibration response of rotating machines. Recently, the relation between acoustic and vibration response during a rotary machine balancing process based on the Four-Run method has been presented for constant speed machines. This method cannot be applied to machines in start-up or shut-off. Hence, by considering the acoustic and vibration responses of a machine between its critical speeds, this research presents a new innovative speed-variant balancing method based on the original Four-Run method, named as "Peak to Peak for Critical Speeds (PPCS)". The proposed method consists of two major types of application: the first is in the Run-up of the machine and the second is in Shut-down. Experimental laboratory results show that the PPCS method can be implemented for speed-variant and flexible rotary machines during run-up or shut-down transient processes based on acoustic and vibration measurements. As a phase-less and a contactless method, the PPCS can be employed as an innovative and readily available method for condition monitoring in the future.

Keywords: flexible rotating machinery; balancing method; speed-variant; acoustic feedback

1. Introduction

Dynamic stresses that are observed as overloads within a rotating machine are detected as unwanted quantities; these could affect machine elements like bearings, shafts, structural joints and foundations and in addition can produce undesirable noise. The most likely reason for these periodic forces to show up is due to unbalanced rotating components inside the rotating part of the machine. These unbalances could arise from manufacturing inaccuracies or operational corrosions. In many rotating machines such as electrical motors, car engines, power turbines, pumps etc. it is essential that these need to be balanced to work properly. It has been found that a balanced rotation will keep the device and its elements such as bearings in ideal working condition which leads ultimately to a longer life-time of the whole system [1] - [3]. The methods of diagnosis, especially in the field of balancing, are defined in a wide range where different methods are used for various types of rotary machines based on their specifications. For instance, in a device with high rotational speed the balancing properties of the device is more complicated, and it should be balanced over both low and high-speed ranges separately.

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Most of the conventional methods in the field of condition monitoring use methodical steps for determining any slightly unbalanced element to prevent further malfunctions. However, in order to achieve the best performance of any device and to keep it safe from damaging vibrations, balancing methods are placed on a high level of importance, so there must be a place for innovations and increase in effectiveness; a sample of these are outlined within these studies [4]- [7]. The standard classic methods such as vector balancing method and the Four-Run or four-run balancing methods are widely used in industrial cases. Some of these methods require additional sensors like tachometers for calculations. A few works have considered reduced data from accelerometers.

Goodman presented the method of least squares for prediction of correction mass and the installation specifications [8] and this method was developed further by Lund and Tessarzik [9],[10]. These studies can be considered as the first step for presenting various methods to balance rotary machines and this has been continued. This research has been followed by Pennacchi et al. [11] to develop the method of identifying unbalance in rotating machines and has resulted in the development of the M-estimators method, [12]. By using run-up and shutdown process, Sinha et al. [13] proposed a method for speed-variant machines to calculate unbalance and misalignment of a flexible rotary machine. The various kinds of modal based balancing methods have also been developed during the last few years, [14]-[17]. The application of these methods haves been studied for industrial equipment, [18]- [24]. By using transient vibrational data, the methods have been used for speed variant rotating machines, [25][26]. Presenting a reliable balancing method to minimize vibrations of the rotating devices was concerned in most of these research works. As an illustration, Shamsah et al. [27] have worked on a new method of balancing which requires a reduced number of vibration sensors to reduce the computational efforts in the complex signal processing. So, the machine down time could be reduced significantly as a result.

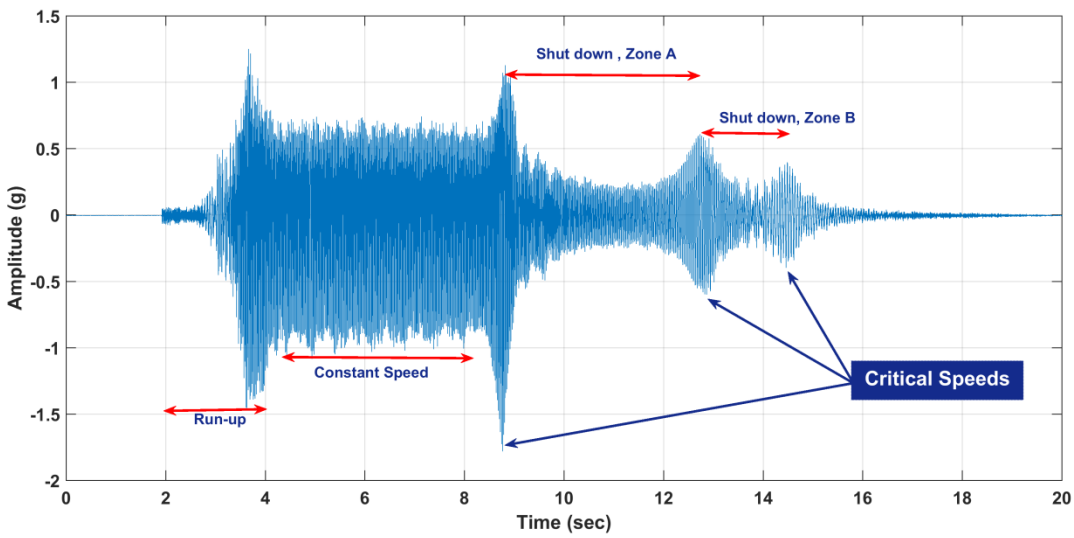

Figure 1: The complete time-sample displaying the vibration response of the machine. Annotations indicate the various mode states of the machine.

Although many vibration-based balancing methods have been presented in the last few decades [28]- [30], a new approach proposes the possibility of using acoustic responses to identify the unbalance of a rotating device. Mao et al. proposed an acoustic transfer matrix to calculate acoustic pressure from a plate which is vibrating, [31] and Isavand et al. [32] showed that there is a possibility to utilize acoustic responses for diagnosis problems within buildings. These authors compared the acoustic and vibration response of a rotating machine balancing process at a constant speed and load, [33]. The main objective in this work was the use of an acoustic response to calculate the properties of a corrective mass for a constant rotational speed device. Although the research showed that

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the trend of the vibration and acoustic response were similar, the study was isolated to the structural health for a constant rotation speed machine.

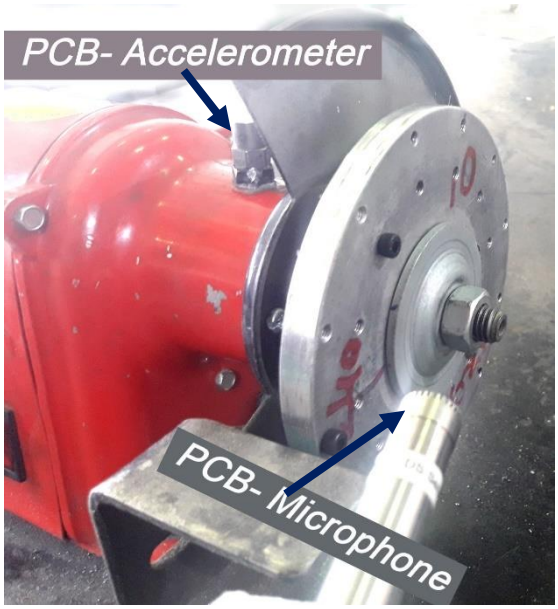

Figure 2: Test setup showing the rotating machine with accelerometer placed on the top and microphone placed approximately 100 mm from the grinder surface.

The research presented in this paper proposes an innovative method which balances an electrical rotating machine and provides comparison of acoustic and vibration response data during "run-up" and "shut-down" as shown in Fig.1 and in Fig.2 the prepared experimental test setup is shown. This article presents the theory of the proposed method, the experimental procedure, and results. Finally, some detail of the results will be discussed.

2. Materials and Methods

The proposed speed-variant balancing method is based on the concept of the Four-Run Balancing Method. In this section, the main concept and the proposed methodology are illustrated. Since the proposed method needs the modal balancing process, this procedure is outlined.

2.1- Balancing Theory

At the first step the general balancing equilibrium, modal balancing and, Four-Run balancing method are clearly illustrated.

2.1.1 General Balancing Equilibrium

Since the machine can be supposed as a typical overhung configuration, the main motion equation can be considered as presented by Fu et al. [21].

$$\boldsymbol{M\ddot{U}(t) + (D + G(t))\dot{U}(t) + K(t)U(t) = F(t)} \quad (1)$$

Where $\boldsymbol{M}$, $\boldsymbol{D}$, $\boldsymbol{G(t)}$ and $\boldsymbol{K(t)}$ are the mass, damping, gyroscopic and stiffness matrices of the rotating system. Due to unbalance on the mass-disk and gravity, the excitation forces term can be written according to reference [21]:

$$\boldsymbol{F = me(\omega^2 cos(\omega t + \varphi) + \dot{\omega}\sin(\omega t + \varphi)) + mg} \quad (2)$$

where $\boldsymbol{\omega}$ and $\boldsymbol{\dot{\omega}}$ denote the rotational speed and the angular acceleration of the machine, and $\boldsymbol{m}$ is the mass of the disc. It should be noted that $\boldsymbol{e}$ (the eccentricity) and $\boldsymbol{\varphi}$ are referenced by the initial unbalanced angle. According to equation (2) the balancing

process can reduce the eccentricity parameter resulting in an acceptable decrease in an undesirable unbalance force. Thus, by decreasing the dynamic unbalance forces, the unwanted vibration character of the machine reduces respectively.

2.1.2 Modal Balancing

To obtain an accurate mass, the basic theory of modal balancing [25],[34] and [35] is necessary. Consider an isotropic flexible rotor system with arbitrary distributed mass $\boldsymbol{m(z)}$, the characteristic modal function can be obtained when the geometrical features and physical parameters of the system are given. The orthogonality condition between two modes can be defined as in the references:

$$\int \boldsymbol{m(z)\Psi_n(z)\Psi_m(z)dz} = \begin{cases} \boldsymbol{0} & \boldsymbol{n \neq k} \\ \boldsymbol{N_k} & \boldsymbol{n = k} \end{cases} \tag{3}$$

where $\boldsymbol{\Psi_n(z)}$ is the nth characteristic modal function and $\boldsymbol{z}$ is the location of the horizontal axis of the rotor bearing system and $\boldsymbol{N_k}$ is the kth distributed mass. The distributions of initial unbalance mass eccentricity of shaft can be expressed as

$$\boldsymbol{u(z)} = \sum_{\boldsymbol{n=1}}^{\infty} \boldsymbol{u_n e^{i\delta_n} \Psi_n(z)}. \tag{4}$$

The dynamic deflection $\boldsymbol{\mathcal{R}(z,t)}$ can be stated as

$$\boldsymbol{\mathcal{R}(z,t)} = \boldsymbol{e^{i\omega t}} \sum_{\boldsymbol{n=1}}^{\infty} \frac{\boldsymbol{\zeta_n^2}}{\boldsymbol{1-\zeta_n^2}} \boldsymbol{u_n e^{i\delta_n} \Psi_n(z)} \tag{5}$$

$$\boldsymbol{\zeta_n} = \frac{\boldsymbol{\omega}}{\boldsymbol{\omega_{cn}}} \tag{6}$$

where $\boldsymbol{\omega_{cn}}$ is the nth c critical angular speed. The critical rotation speeds in revolutions per minute RPM, Fig. 1, of the device are given by $\boldsymbol{60\ \omega_{cn}/2\pi}$.

2.2- Experimental Procedure

The case study comprises a 750-Watt double plate grinding machine where the grindstone is replaced by a graded aluminum disk (aluminum series 5000). The maximum rotating speed for the machine is 3000 RPM. The disk has a thickness of 10 mm, an outer diameter of 150 mm, and inner diameter of 35 mm. The plate is drilled at two different radii of 50 and 100 mm, each 30 degrees from the 0-degree line. The accelerometer sensor was PCB-352C33 type (103.7 mV/g), the 1/2" microphone PCB-HT378B20 (50 mV/Pa) and the data acquisition (DAQ) system is a NI-PXI-4462 unit. The data acquisition rate was set to 2000 samples per second (2000 Hz).

The original Four-Run balancing method is also known as the common phase-less method, [29]. Based on the fundamentals of regular Four-Run balancing method, the "Peak to Peak of different Critical Speeds" (PPCS) balancing method is proposed. Therefore, the PPCS method is simply based on the root mean square of the amplitude of vibration or acoustic responses of the device which are then recorded during "running-up" or free "shutting down" operating processes in presence of critical speeds. It means that the PPCS method can be applied to different types of disc data: the "Run-up" test and the "Shut-down" test. These provide enough information to calculate the position and properties of a required correction mass. The PPCS method requires several steps that, described briefly here, is important to restate that these steps also apply to acoustic data which can be obtained from a single microphone or an array during an operational sample test. This point is considered as the novelty of the method which brings more simplicity for balancing procedure.

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The overall steps of the method are illustrated in Fig.3.

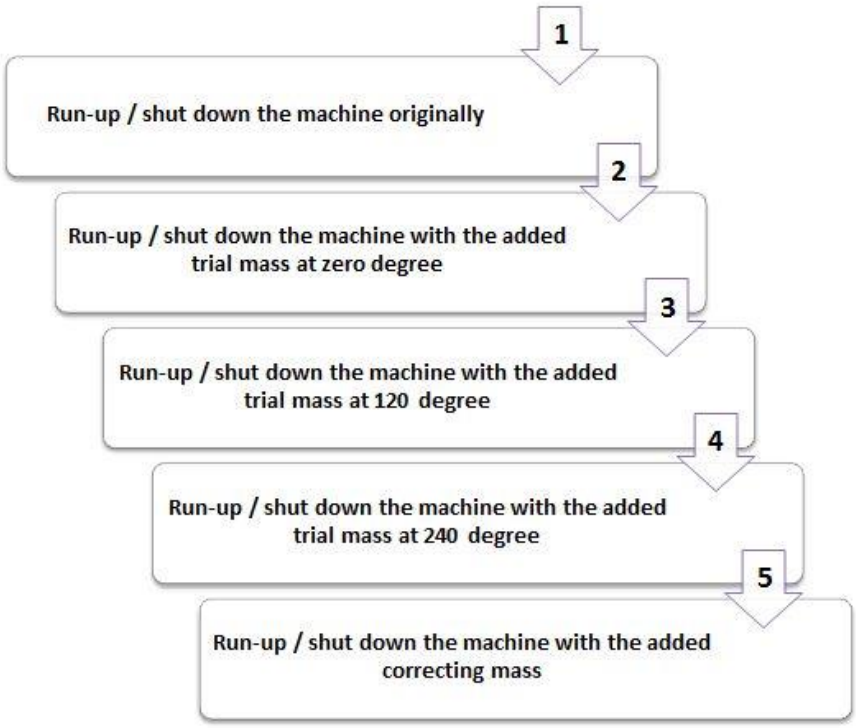

Figure 3 The proposed PPCS balancing method flowchart

It must be stressed that the machine must reach a specific speed higher than the critical speeds during run-up or shut down operation. The acoustic response data of the machine could be considered from the run-up area, zone A or zone B of shut down process, according to Fig.1. The zone A is related to the time from the beginning of the shutdown process to the time when the third and second critical speeds have been excited. The zone B consists of the time between excitation of the first and second critical speeds of the machine. Here, the root mean square (RMS) values have been calculated within the response periods and considered as the representing values for the magnitude of the vibration and acoustic responses.

At the first step, a modal analysis test has been done to clarify the critical speeds of the machine. Then, the (faulty) device of interest is started normally and after reaching a specific speed higher than the critical speeds, the device is then switched off. Then, the root mean square of the amplitude of the machine response during the selected time range is calculated. Here, the acoustic response of the device is recorded via microphone. This type of recording also applies to the following steps. At the second run-up or shut-down test, a trial mass is added to a certain point of the rotating part. The process is repeated, and RMS values are recorded. In the third and fourth run-up or shut-down tests, the same process is repeated but the trial mass is placed at angular positions which respectively differ by 120 and 240 degrees from the original point at the same radius.

For calculating the position of the correcting mass, at the first step a circle with the radius of the RMS value of the vibration/acoustic response is drawn. By connecting the trial mass at zero position and acquiring the response, the next circle is drawn on the perimeter of the first circle at zero degree with the radius equal to the RMS value of the second step. The third and fourth circles are drawn on the perimeter of the first circle at $\frac{2\pi}{3}$ and $\frac{4\pi}{3}$ respectively with the RMS values related to third and fourth run-up or shut-down tests

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It should be noted that if there is no enclosed area between these four circles, the procedure must be done again with a heavier trial mass. The center of this enclosed area between circles is an important point to calculate the position and the final mass of the correcting mass. A schematic of drawn circles is shown in Fig.4 and related information is listed in Table.1. The same format is used to present the results of the experimental tests in this paper for both vibration and acoustic responses.

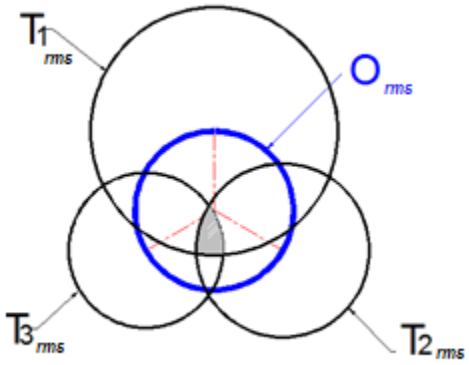

Figure 4: Schematic of drawn circles for vibration or acoustic response. Reference point located at center of grey shaded area

It should be added that, although the position of the trial mass at the second Run-up or Shut-down test is completely arbitrary, after choosing the zero angle the following positions are located based on this zero-angle point.

Table1 : The measured parameters

Parameter	Symbol
The RMS value of the vibration / acoustic response at the first run-up or shut-down test	O rms
The RMS value of the vibration / acoustic response at the presence of trial mass at the zero-degree position	T1 rms
The RMS value of the vibration / acoustic response at the presence of trial mass at the 120-degree position	T2 rms
The RMS value of the vibration / acoustic response at the presence of trial mass at the 240-degree position	T3 rms

3. Results

Before presenting the results of the experimental balancing process, as the first and main assumption, it is critical to ensure the load of the machine during the balancing process is set constant for all phases of the procedure. It should also be noted well that by using proper filtering, the effect of ambient noise should be neglected unless there is a nearby noise source radiating at a critical frequency, in which case it should be switched off.

As shown in Fig.5, the test procedure is divided into different steps. The results of the modal test are presented in section 3.1, and the experimental measurements and results of the run-up are illustrated in section 3.2. Respectively, the results related to shut-down process in zones A and B are highlighted in sections 3.3 and 3.4.

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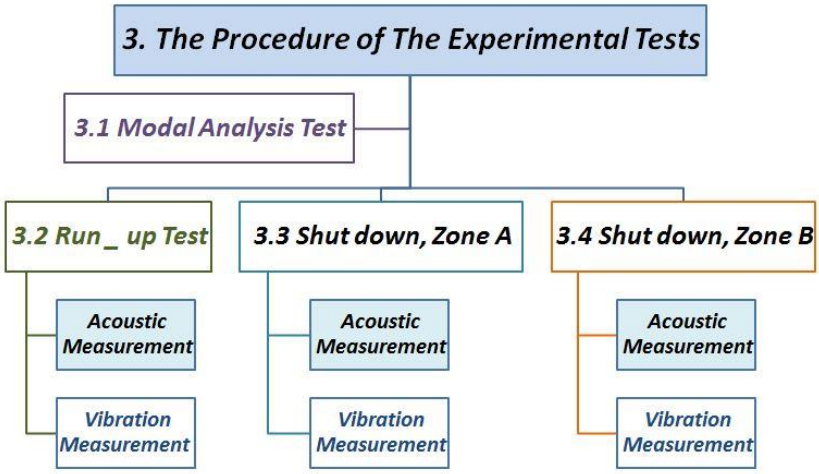

Figure 5: The experimental tests procedure

It should be noticed that these three types of proposed PPCS method including Run-up, Shut-down in zone A and Shut-down in zone B are parallel. It means all variants of the methods would result in the same correcting mass and position.

3.1 Modal Analysis Test

In this section of the paper, an output only Fourier-based modal test has been conducted on the machine where the machine is switched off. Impulse- response method has been employed , [27] and [36]. By using a hard plastic-tip impact hammer, the machine has been excited in the studied frequency range.

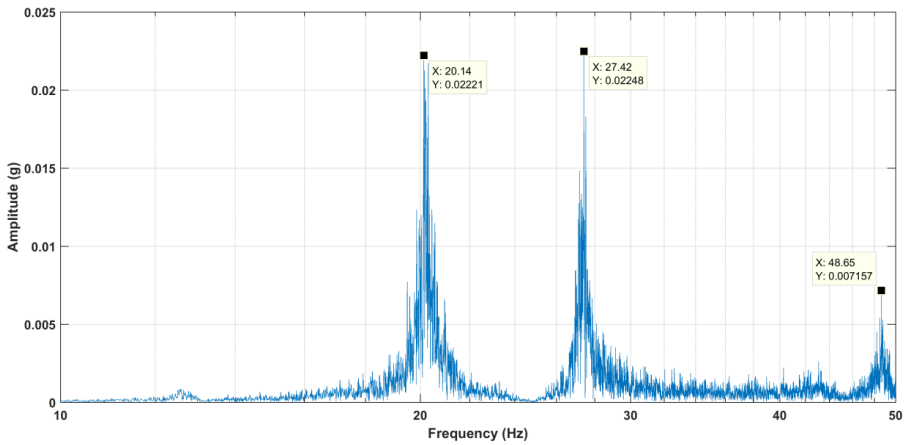

Figure 6: The result of the impact modal test

Figure 6 presents the identified natural frequencies and Table 2 makes a list of these natural frequencies and their related critical speeds.

Table 2: Natural frequencies and related critical speeds

Parameter	Frequency (Hz)	Related Critical Speed (RPM)
The first natural frequency	*20.14*	*1208.4*
The second natural frequency	*27.42*	*1645.2*
The third natural frequency	*48.65*	*2919.0*

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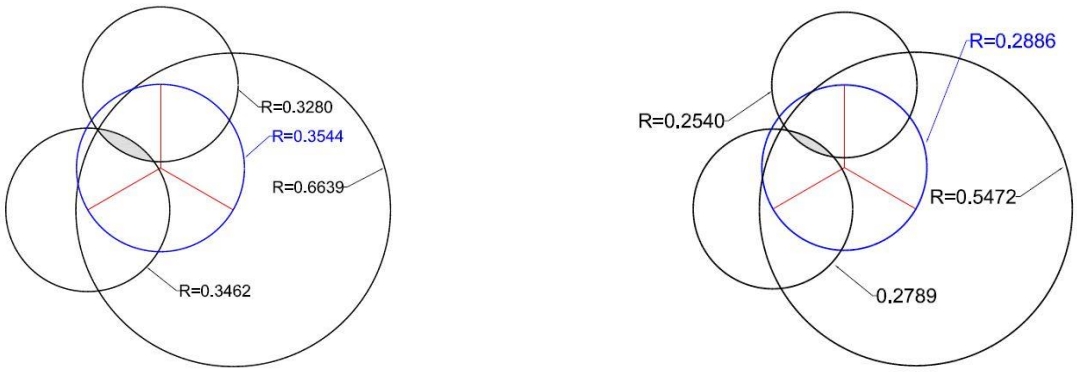
3.2 The Run-up balancing method

In this part of the study, the running-up state of the machine has been concerned where the first three critical speeds have been excited in sequence. According to the proposed PPCS method presented earlier, the responses of the machine between the peaks of the first and third critical speeds have been concerned. At the first step, the rotating machine should be started in original condition and after reaching a specific speed which is higher than the third critical speed is switched off. Then the RMS values of the vibration and acoustic responses can be determined during the selected time. As mentioned earlier, the placement of accelerometer or microphone must be fixed during all steps of the test to ensure valid results. The results of the first run-up in addition to the results of the remaining runs in the presence of a 1.6-gram trial mass installed at three angular positions of 0, $\frac{2\pi}{3}$ and $\frac{4\pi}{3}$ are presented in Table 3.

Table 3: The measured parameter in the Run-Up test

Parameter	Acceleration (g_{rms})	Sound pressure (Pa_{rms})
The RMS value of the first Running-Up	*0.3544*	*0.2886*
The RMS value of the second Running-Up with test mass at 0° angular position	*0.3280*	*0.2540*
The RMS value of the third Running-Up with test mass at 120° angular position	*0.6639*	*0.5472*
The RMS value of the fourth Running-Up with test mass at 240° angular position	*0.3462*	*0.2789*

This table presents the vibrational and acoustical results of the Run-Up test. By running the machine without the trial mass, the RMS value of the vibration is 0.3544 g_{rms} and, it is corresponding to an acoustical pressure of 0.2886 Pa_{rms}. Adding the trial mass in a 0-degree position (12 O'clock) results in the RMS value of the vibrational response decreased to 0.3280g_{rms} equivalent to the acoustic response of 0.2540 Pa_{rms}. By rotating the trial mass to 4 O'clock position, the vibration of the machine increased to 0.6639g_{rms} and the acoustical response rose from 0.2540 Pa_{rms} to 0.5472 Pa_{rms}. In following step, a decreasing trend could be observed in vibrational and acoustical responses. By rotating the trial mass another 120 degrees, the RMS values of the vibrational and acoustical responses acquired 0.3462 g_{rms} and 0.2789 Pa_{rms} respectively. As shown in Fig.7 the vibration and acoustic response circles are drawn in accordance with the results presented at Table.3. By comparing the circles' positions in (a) and (b), it is clear there exists the same trend in vibration and acoustic responses.



b. The response circles based on vibration response

a. The response circles based on acoustic response

Figure 7: The response circles based on acoustic/ vibrational responses for the Run-Up test

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According to the proposed speed-variant PPCS balancing method, the calculated correction mass is 3.2 gram and should be installed at 57 counter-clockwise (CCW) degree based on vibration measurements (3.2gr@570) and, by considering the acoustic data, the mass of the correcting mass is 3.2 gram at the 55 CCW degree position (3.2gr@550). Finally, a 3.4-gram correcting mass added to the machine at 10 O'clock position (3.4gr@600). The concluding results after all vibrational and acoustical responses after installation of the correction mass are shown in Table 4. As presented in the table, the RMS values of the amplitudes of the vibrational and acoustical responses for the machine have been decreased noticeably.

Table4 : The RMS value of the acoustic and vibration response for the balanced case in the Run-up

Acceleration (g_{rms})	Sound Pressure (Pa_{rms})
0.1530	0.1764

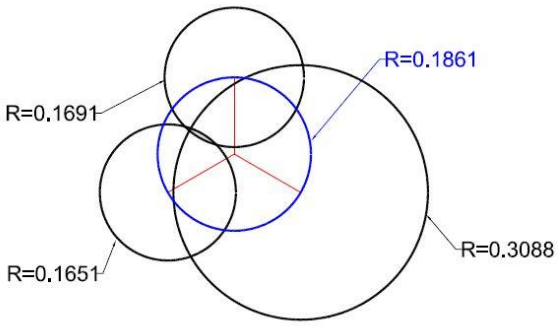
3.3 The Shutting-down balancing method in Zone A

In this part of the study, the shut-down state of the machine has been restricted to zone A as shown in Fig.1. After reaching a speed higher than the third critical speed, the machine is switched off. Then the RMS value of the vibration and acoustic responses should be determined between the peaks of the third and the second critical speeds. The results of the first shut-down in zone A, in addition to the results of the remaining runs in the presence of a 1.6-gram trial mass installed at three angular positions of 0, $\frac{2\pi}{3}$ and $\frac{4\pi}{3}$ are presented in Table 5.

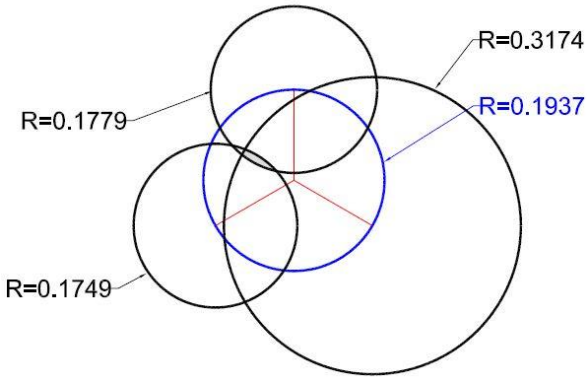
Table 5: The measured parameter at the first Shutting-Down test

Parameter	Acceleration (g_{rms})	Sound pressure (Pa_{rms})
The RMS value of the first Shut-Down	*0.1861*	*0.1937*
The RMS value of the second Shut-Down with test mass at 0° angular position	*0.1691*	*0.1779*
The RMS value of the third Shut- Down with test mass at 120° angular position	*0.3088*	*0.3174*
The RMS value of the fourth Shut-Down with test mass at 240° angular position	*0.1651*	*0.1749*

As presented in Table 5, the RMS value of the vibration response of the rotating machine in the Shut-down test in zone A is 0.1861 g_{rms} originally. The relative value for the acoustic measurements is 0.1937 Pa_{rms}. By adding the 1.6-gram trial mass at a 0-degree position (12 O'clock), the RMS value of the responses of the machine decreased to 0.1691g_{rms} and 0.1779 Pa_{rms}. For the new position of the trial mass at 120 degree (4 O'clock), the vibrational and acoustical responses showed a similar trend and increased to 0.3088 g_{rms} and 0.3174 Pa_{rms}, respectively. Finally, by rotating the trial mass for the second $\frac{2\pi}{3}$ to the new position of 240 degree (8 O'clock), another similar decreasing trend could be observed in the relevant vibration and acoustic responses to 0.1651 g_{rms} and 0.1749 Pa_{rms}. In Fig. 8 the vibration and acoustic response circles are drawn according to the data of the Shut-down test in zone A.



b. The response circles based on vibration response



a. The response circles based on acoustic response

Figure 8: The response circles based on acoustic/ vibrational responses for the Shutting-Down Test in zone A

By using the proposed PPCS method, the correcting mass is calculated 3.2 gram at position 61 degrees CCW based on vibration response and, the results are the same based on acoustic response (3.2gr@ 61^0). Finally, a 3.4-gram was installed at 10 O'clock position (3.4gr@ 60^0) as the correcting mass. The concluding results of the RMS values of the vibrational and acoustical responses after installation of the correction mass are shown in Table 5. It is clear the RMS values of the amplitudes of the vibrational and acoustical responses of the machine have been decreased considerably.

Table6 : The acoustic and vibration response for the balanced case at the Shutting-Down Test in zone A

Acceleration (g_{rms})	Sound Pressure (Pa_{rms})
0.0800	0.1330

3.4 The Shutting-down balancing method Zone B

In this section, the shut-down state of the machine has been restricted to zone B where the first and the second critical speeds are excited. It should be noted again that the machine must be switched off, after reaching a speed higher than the second critical speed. Then the RMS value of the vibration and acoustic responses should be recorded and calculated between the peaks of the second and the first critical speeds. The results of the first shut-down test in zone B, in addition to the results of the remaining runs in the presence of a 1.6-gram trial mass installed at three angular positions of 0, $\frac{2\pi}{3}$ and $\frac{4\pi}{3}$ are presented in Table 7.

Table 7: The measured parameter at the second Shutting-down Test in zone B

Parameter	Acceleration (g_{rms})	Sound pressure (Pa_{rms})
The RMS value of the first Shut-Down	*0.1459*	*0.3058*
The RMS value of the second Shut-Down with test mass at 0° angular position	*0.1368*	*0.2918*
The RMS value of the third Shut- Down with test mass at 120° angular position	*0.2049*	*0.4488*
The RMS value of the fourth Shut-Down with test mass at 240° angular position	*0.1357*	*0.2641*

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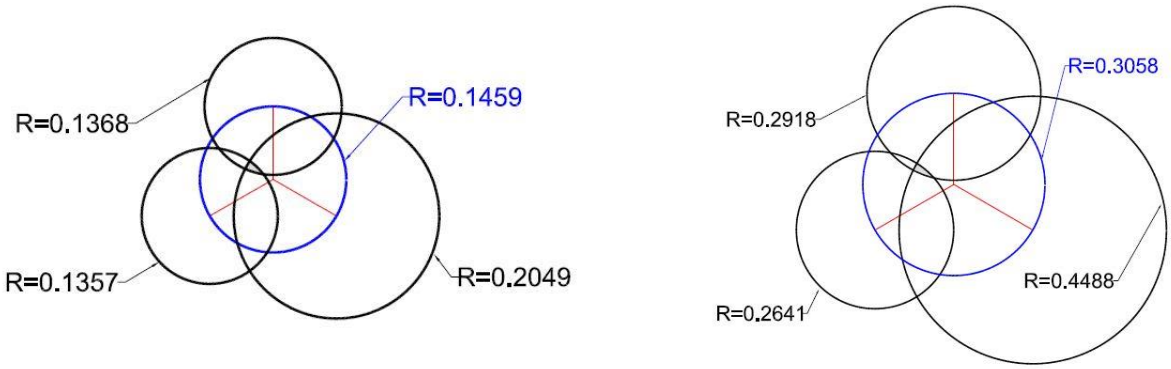
The RMS values of the vibrational and acoustical responses of the rotating machine in the Shut-down test in zone B is 0.2049 g_{rms} and 0.4488 Pa_{rms} which belong to vibration and acoustic responses respectively with adding the trial mass at 4 O'clock position. Also, the minimum of these values belongs to the situation that the trial mass added in 8 O'clock position. The RMS value of the vibration response was 0.1357 g_{rms} and this value of the acoustic response was 0.2641 Pa_{rms}. There are obviously similar trends describing this increase and decrease for both the acoustic and vibration responses. In Fig.9 the vibration and acoustic response circles are drawn according to the data of the Shut-down test in zone B.



b. The response circles based on vibration response

a. The response circles based on acoustic response

Figure 9: The response circles based on acoustic/ vibrational responses for the shutting down test in zone B

It should be noticed that during the test procedure the load of the machine must be kept constant. According to the results presented in Table 7 and Figure 9, based on vibration response, the correcting mass calculated 3.7 grams at 61 degree (3.7gr@61^0) and, based on acoustic response, the correcting mass determined to be 3.8 grams at 66 degree (3.8gr@66^0). The final values of the root mean square values of the vibrational and acoustical responses after installation of the correction mass (3.4gr@60^0) are shown in Table8 . It is clearly demonstrated that the residual unbalance is lower than the original and, the amplitude of the vibrational and acoustical response for the machine have been decreased significantly.

Table8 : The acoustic and vibration response for the balanced case at the Shutting Down Test in zone B

Acceleration (g_{rms})	Sound Pressure (Pa_{rms})
0.0634	0.1384

4. Discussion

In accordance with the similar behavior in both vibration and acoustic responses in Four-Run balancing steps of a constant speed rotary machine observed by Isavand et. al [33], Fig.10 presents the relation between these responses in Run-up, Shut-down in zone A and, Shut-down in Zone B by using the proposed PPCS balancing method.

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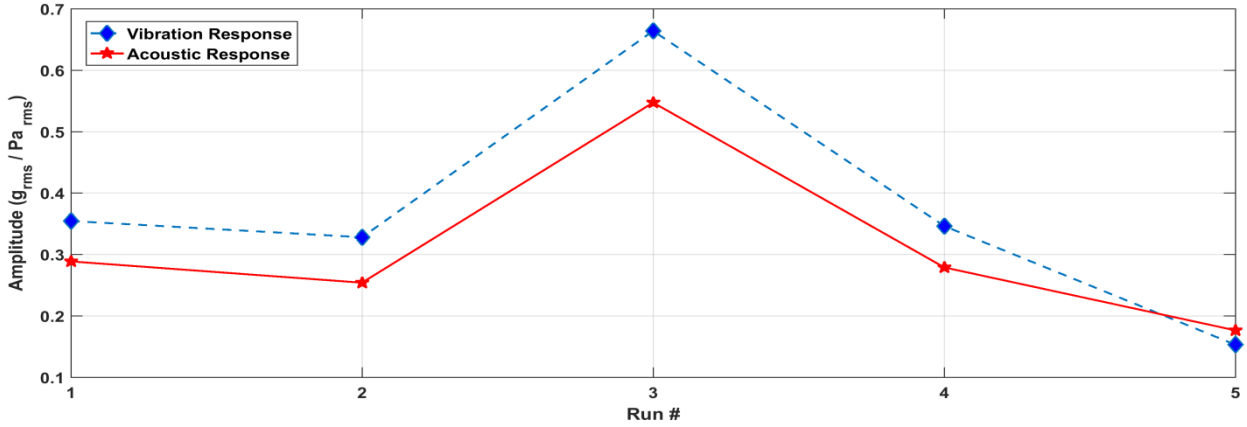
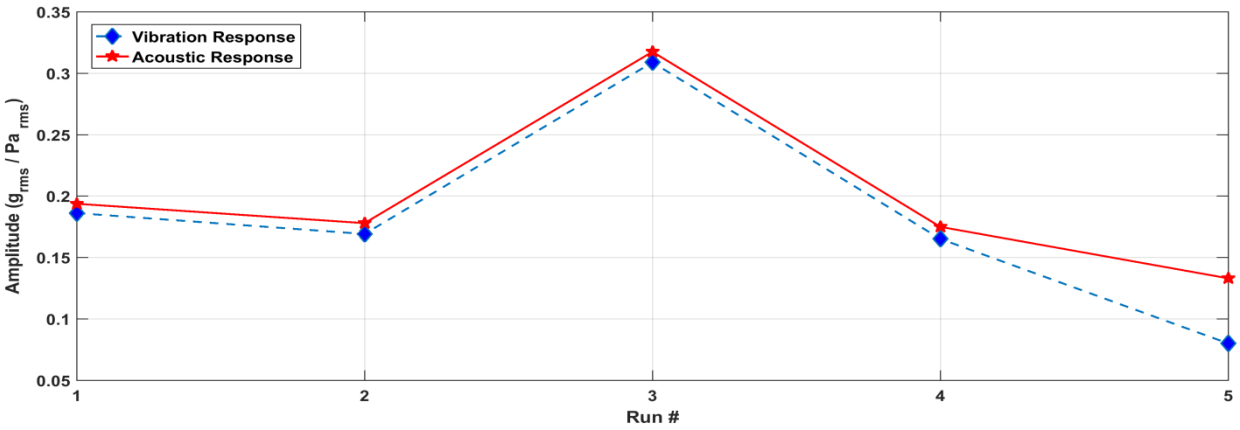
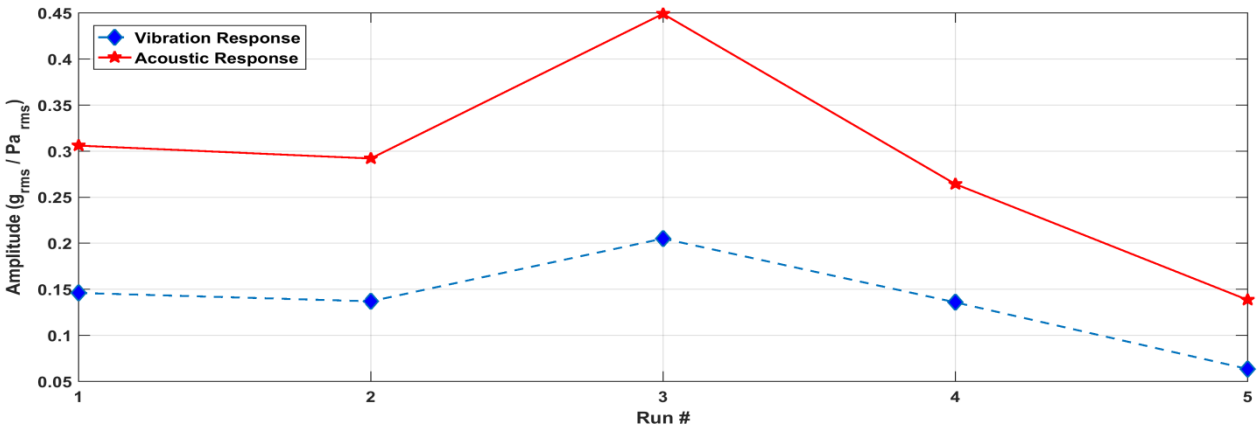


a. Comparison of vibration / acoustic response behavior in Run-up



b. Comparison of vibration / acoustic response behavior in Shut-down in zone A



c. Comparison of vibration / acoustic response behavior in Shut-down in zone B

Figure 10: Comparison of vibration / acoustic response behavior in all types of the PPCS balancing method

The horizontal axis of Fig.10 is about the steps of the proposed method where number 1 means the original working condition. Numbers 2 to 4 are related to the response of the machine after adding the trial mass at positions 12, 4 and 8 O'clock positions, respectively. Finally, number 5 represents the responses of the machine after adding the correction mass to the rotating disk. According to these trends, it clearly appears that vibration

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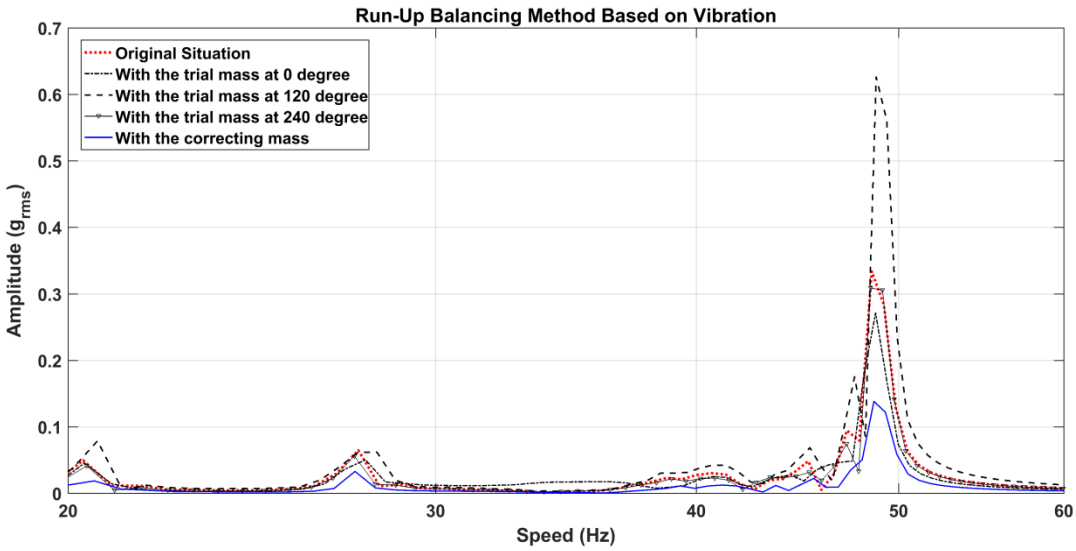
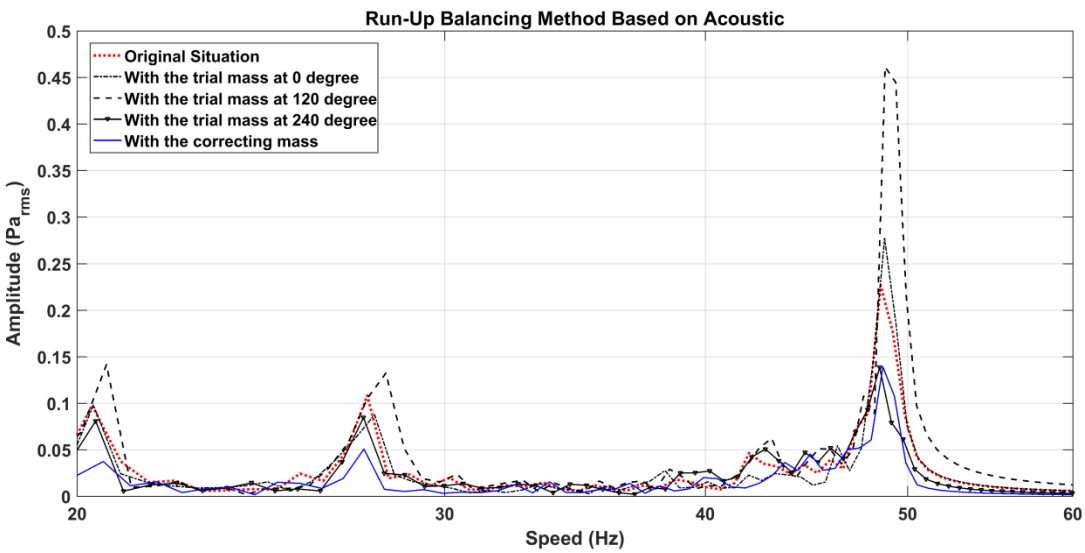
or acoustic responses would present the same solution for the corrected mass for speed-variant balancing process.

By applying the discrete Fourier operation to the vibration and acoustic responses of the machine, the frequency domain of the acoustic and vibration response for all steps of the proposed PPCS balancing process have been presented in Run-up state in Fig.11. All three critical speeds have been shown in the figure. Since the Run-up test considered the peak of the first critical speed to the peak of the third one, in frequency domain diagram all three critical speeds have been found.



a. the discrete Fourier operation applied to the vibration response data



b. the discrete Fourier operation applied to the acoustic response data

Figure 11: The acoustic / vibration response in frequency domain for the Running-Up Test

As shown in Fig.12, the first critical speed is absent in the diagram. It is because that the selected data set, area zone A, is related to the peak of the third critical speed and the peak of the second one. It means that the first critical speed cannot be shown in frequency domain diagram related to the shut-down test in zone A.

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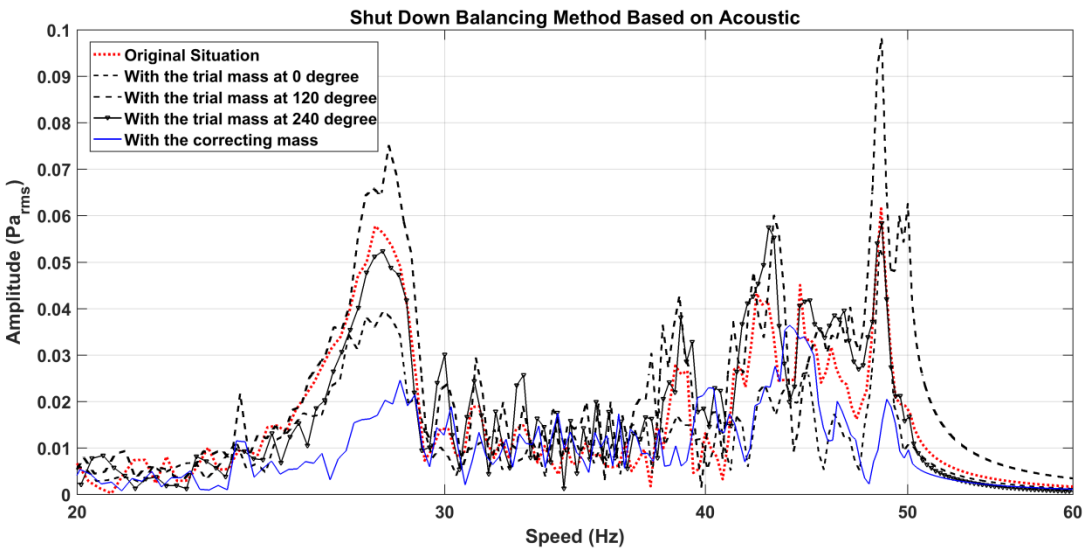
a. the discrete Fourier operation applied to the vibration response data



b. the discrete Fourier operation applied to the acoustic response data

Figure 12: The acoustic / vibration response in frequency domain for the Shutting-Down Test in zone A

As it shown in Fig.13, the third critical speed is absent in the diagram. It is because that the selected data set, zone B, is related to the peak of the second critical speed and the peak of the first one. It means that the third critical speed cannot be shown in frequency domain diagram related to the shut-down test in zone B

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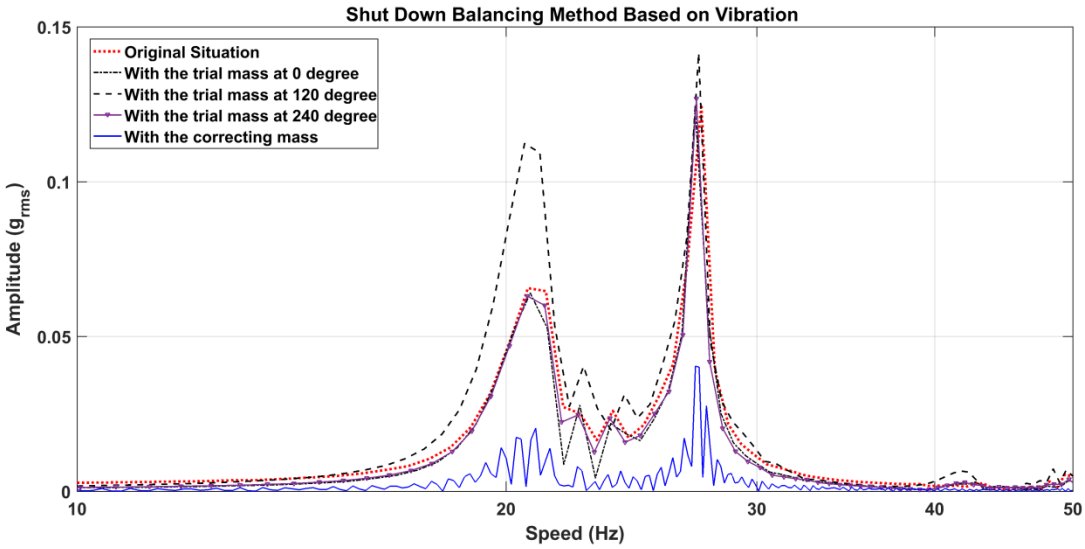
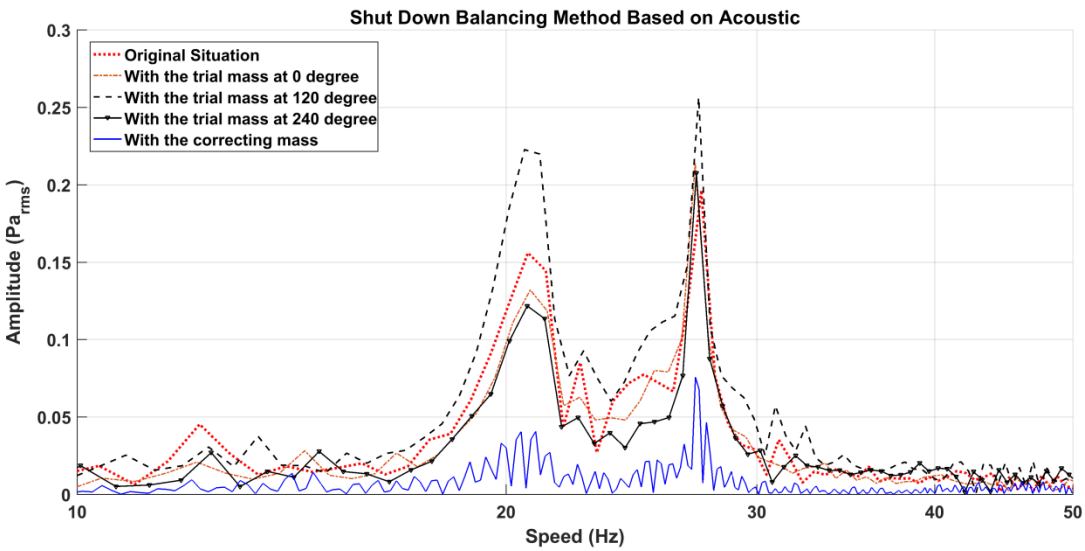


a. the discrete Fourier operation applied to the vibration response data



b. the discrete Fourier operation applied to the acoustic response data

Figure 13: The acoustic / vibration response in frequency domain for the Shutting-Down Test in zone B

It is clearly shown that the residual unbalance of the machine is the lowest graph in all diagrams. The calculated correcting masses and related positions based on using various types of the PPCS balancing method have been presented in Table 9.

Table 9: comparison between the methods in proposed mass and angle

Parameter	Correcting Mass (gram)	Correcting Angle (degree _ccw)	gram@θ^0
Run_up, Vibration based	*3.2*	*57*	*3.2gr @57⁰*
Run_up, Acoustic based	*3.2*	*55*	*3.2gr @55⁰*
Shut down, Zone A, Vibration based	*3.2*	*61*	*3.2gr @61⁰*

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Shut down, Zone A, Acoustic based	*3.2*	*61*	*3.2gr @61⁰*
Shut down, Zone B, Vibration based	*3.7*	*61*	*3.7gr @61⁰*
Shut down, Zone B, Acoustic based	*3.8*	*66*	*3.8gr @66⁰*

On review these results suggest clear evidence that vibration or acoustic responses would reveal the same solution for the corrected mass in each type of the tests. It is worth mentioning that the main objective of the current research is presentation of the PPCS method. Although uncertainties are important enough to be established in further research, they were supposed negligible in this paper.

In addition, it should be noted that there are a few limitations against the use of the PPCS method. The main concern is about acoustical responses. Regarding very low speed machines, they may generate a low frequency response could be located below the microphone frequency threshold. The existence of more than one noise source and room acoustic modes should also be considered for implementation of this method in acoustical mode. Finally, although the PPCS method is presented to balance speed variant rotary machines, this method can be used for flexible constant speed machines.

5. Conclusion

Presenting a phase-less balancing method for flexible speed variant rotary machine has been concerned in this research using acoustic and vibration responses. This paper presents an innovative method which works between different critical speeds. After estimating the critical speeds of the machine, this method calculates the root mean square (RMS) of the amplitude of the acoustic and vibration responses between the different critical speeds. The proposed Peak-to-Peak of different Critical Speeds (PPCS) balancing method was performed on a sample rotating machine in run-up and shut-down states and, the data for vibration and acoustic responses in all steps were compared in detail in time and frequency domains. It should be noted that the machine was considered with constant load during the tests, and ambient noises were controlled, although by using proper filtering, it is possible to reduce the effect of external noises. The results show similarity for both vibration and acoustic responses in all balancing phases and ability to use them separately for balancing the machine. In this way, it can be concluded that we have confidence that the balance process of a speed-variant rotating machine could be performed based on just acoustic responses as an alternative to classical utilization of vibration response data. Being phase-less, contact-less and direct access can be considered as the advantages of the PPCS method based on acoustic response, also the methodology is valid for both constant and variable speed machines.

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