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CRITICAL SPEED ANALYSIS FOR AVOIDANCE PHENOMENON IN HEAVY ROTATIVE MACHINES

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Abstract: *This studies the phenomenon of resonance during operation by using both a theoretical analysis and an experimental one performed directly on the installed equipment. The considered case study is based on the analysis of the dynamic behavior of a 2 MW electric generator from a hydroelectric plant that is driven by a horizontal shaft turbine and a gearbox. The rotating assembly is measured from the point of view of geometric precision and dynamic balancing and mounted on the four supports on the base plate. However, during operation, by increasing the working speed both without load and with, critical speeds appear that lead to vibrations above the permissible ones, which are attributed to the phenomenon of resonance. To determine the sources of vibrations, the assembly is dynamically measured in the area of its bearings in three directions (X, Y and Z). A procedure is applied to determine the vibrations depending on the speed and the load in operation. When the system reaches the critical speeds, the highest level of vibrations is recorded in the X direction in the horizontal plane, exceeding the allowed ISO limits. Vibration measurements are also carried out in other points outside the bearings. Next, considering the theoretical aspects that reveal the fact that vibrations are influenced by stiffness, mass and damping, the stiffness is modified at the level of attachment of the assembly to the supports together with the operation of ensuring the coplanarity of the attachment surfaces with the plane of the base plate. Practically, the tightening force is optimized by tension release in repeated cycles of force adjustment and dynamic measurement. After application of the procedure, the level of vibrations falls within the permissible limits.*

Key words: *generator, vibration, stiffness, critical speed, dynamic test.*

1. INTRODUCTION

Shafts supported on bearings in general and rotary machines in particular are a subject of discussion and analysis because, regardless of the speeds at which they operate (low or high), they must have a behavior in idle and loaded operation as close to optimal as possible. In their operation, vibration problems can occur, the first cause being that of resonance, for which dynamic loads can lead to response levels 100–1000 higher than those produced by static loads [1].

Thus, these systems must be analyzed in the design and commissioning phase, but also in the operation of the equipment or after maintenance or repairs. There are calculation methods, modeling and simulation techniques in the design phase. Mathematical models are relatively well known and frequently applied. In addition, dynamic models (solid body models) [2] are used that load the geometry of the rotating system components and the mass and dynamic characteristics (damping and stiffness). Another possibility is to use models with finite elements and dynamic simulation to determine the natural frequencies and the natural vibration modes.

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Through experimental testing, certain defects of rotating machines can be identified [3]. This can be done by using sensors, making periodic measurements, extracting from the measurements the significant values of some parameters that suggest defects [4] and statistical processing of the measurements and analyzing the results to determine the functional state of the system, the prediction of the evolution of the defects [5] and the life span of the system.

Another way to analyze rotating functional systems is experimental testing in order to have a deeper picture of the characteristics of vibration phenomena [6].

In the analysis of these systems, the critical [7, 8] speeds and their connection with the natural frequencies of the system and the resonance phenomenon that must be avoided in operation are important. Resonance leads to very high vibration amplitudes for an excitation frequency close to one of the system's natural frequencies. In other words, a working frequency of the rotating machine close to the first or second frequency of the system is to be avoided.

It can be stated that what happens at the level of a mass-spring-damper oscillator is also found in the case of systems with several masses, namely the natural frequencies of the system are determined by the mass and stiffness, the damping being able to influence them as well to some extent [9].

For an oscillating system with one degree of freedom, it is considered that the frequency of free vibrations ω depends on stiffness k and mass m, which can be extended to systems that are more complex:

$$\omega = \sqrt{\frac{k}{m}}. \tag{1}$$

The solution of the differential equation of motion for such a system is given by Eq. (2):

$$A = \frac{\frac{F_0}{k}}{\sqrt{\left(1-\frac{m\omega^2}{k}\right)^2+\left(\frac{c\omega}{k}\right)^2}}, \tag{2}$$

where A – amplitude, F_0 – perturbation, ω – frequency, k – stiffness, c – damping.

The means of influencing the natural frequencies are by changing the mass, rigidity and damping. Relation (1) suggests the solutions for changing only the first two parameters. Obviously, the mass can be modified in the design stage, but also for functional systems by adding additional mass. For complex systems located in certain conditions, mass modification becomes difficult. In addition, the constructive modification of the rigidity can be done from the design phase, but also by introducing elements with adjustable rigidity in the existing constructions.

2. RESONANCE PHENOMENON IDENTIFICATION

The resonance phenomenon is one of the most important and frequently encountered dynamic phenomena encountered in rotating machines. In countless industrial applications, engineers and technicians are faced with the problems generated by the resonance phenomenon, the critical speed being the condition for the appearance of vibrations.

The high level of vibrations above the limits of admissibility is caused in many situations of existing resonance. The appearance of resonance is not an easy problem to identify due to the multitude of sources, and for a sustainable solution, it is essential to identify the root cause.

The sources of the resonance can be related to several factors and in operating conditions in the field, the action of these sources can take place through the coupling between them.

In this work, the highlighting of the phenomenon of resonance in operating conditions is considered through the connection between the theoretical and experimental analysis through the determination directly on the installed equipment. The industrial case considered is based on the analysis of the dynamic behavior of an electric generator from a hydroelectric plant, a 2 MW generator driven by means of a reducer, respectively a turbine with a horizontal shaft (Fig. 1).

The equation of motion of the system is given by relation (Fig. 1):

$$m\ddot{x} + c\dot{x} + kx = F, \tag{3}$$

where m is the mass of the entire assembly; c – damping distributed on the four supports; k – stiffness.

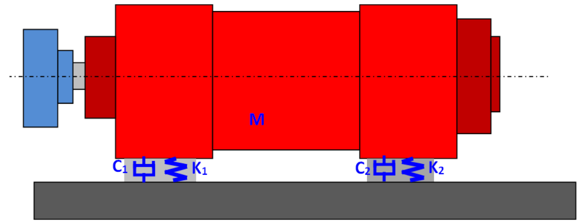


Fig. 1. Dynamic model.

Thus, the total damping (c) and stiffness (k) considering the four fixing points can be expressed as follows,

$$c = c_1 + c_2 + c_3 + c_4, \tag{4}$$

$$k = k_1 + k_2 + k_3 + k_4. \tag{5}$$

The study considers the analysis of a cause generating vibrations in resonant conditions, one whose dynamic sensitivity is very high. In the case of large rotary machines, the main parameter available to engineers is the stiffness parameter modification (Fig. 2). The mass parameter is impossible to implement in the operation phase. When the stiffness cannot be changed or the influence is minimal, we resort to changing the damping parameter, by using different damping solutions or absorbers.

In the present case, the stiffness parameter is coupled in parallel (4), considering the type of 4-point fixation. The main direction of movement of the generator is along the horizontal direction (x) and less along the vertical direction (y) due to the contact pressure and the mass exerted on this direction.

According to the theory, in the resonance area, the amplification of vibrations is very high, leading to instability and the appearance of specific deformation modes.

The main purpose of the research is to highlight the tightening, respectively the stiffness and its influence on the resonance but also on the other types of defects such as misalignment

3. EXPERIMENTAL PROCEDURE

In order to highlight the dynamic characteristic of the generator, an experimental protocol is implemented so that the state of resonance of the generator-fixing solution-foundation assembly can be identified.

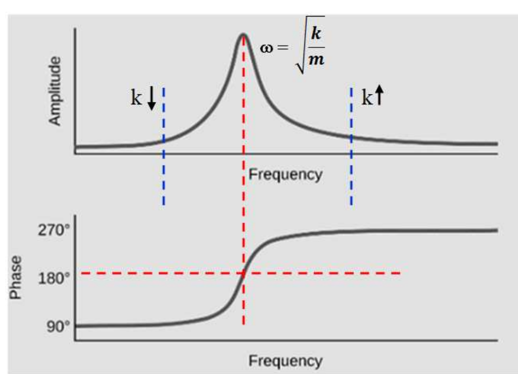


Fig. 2. Dynamic charcaterics for resonance fenomenon.

The generator was measured both in idle or loaded operation. The purpose of the measurements was to highlight the vibration-generating phenomenon, to locate and determine the cause of the vibrations. The vibration measurement was carried out both on the Non Drive End bearing (NDE) and on the Drive End bearing (DE) of the generator, measuring the vibrations in the three directions, horizontal (*X* axis), vertical (*Y* axis) and axial (*Z* axial) (Fig. 1). In order to obtain the dynamic characteristics of the generator/fixing assembly and operating regime, all parameters are monitored at the same time. For this purpose, a test procedure was carried out (Figs. 3 and 4) to allow highlighting the vibrations in relation to the speed and the load on the generator.

To measure the vibrations, accelerometers with a sensitivity of 100 mV/g are used, synchronizing the speed with the vibrations being possible by using a speed laser sensor. The signals are acquired through a digital signal analyzer DSA 550 device, using an acquisition board type NI4432, with 5 channels. The signals are monitored and processed through the Fastview software, using a sampling rate of 25 kSamples/s and a buffer size of 32 768 samples or 12 800 spectral lines.

4. ANALYSIS AND RESULTS

The generator was tested under normal operating conditions.

The generator was repaired, with the bearings replacement, seals replacement, stator winding, balancing. The maximum power of generator is 3.2 MW at a speed of 1000 rpm.

When put into operation, the generator was connected to a gearbox and then to the turbine. Testing started from zero speed, to nominal speed, synchronization and load increase.

The main purpose of the analysis is to identify the cause of the vibrations. In this sense, a series of diagrams (trend parameters, waveform, spectrum, polar diagram, etc.) are used to understand the dynamic behavior of the generator.

When passing through the critical speed, the vibration level is much higher, reaching 5 mm/s rms in the horizontal direction (Fig. 5). In the vertical and axial direction, the amplitudes are much lower. At the full speed, no load (FSNL), Fig. 6, the horizontal vibration is 1.8 mm/s·rms, while in vertical direction it is 0.7 mm/s·rms and in axial direction it is 1.2 mm/s·rms.

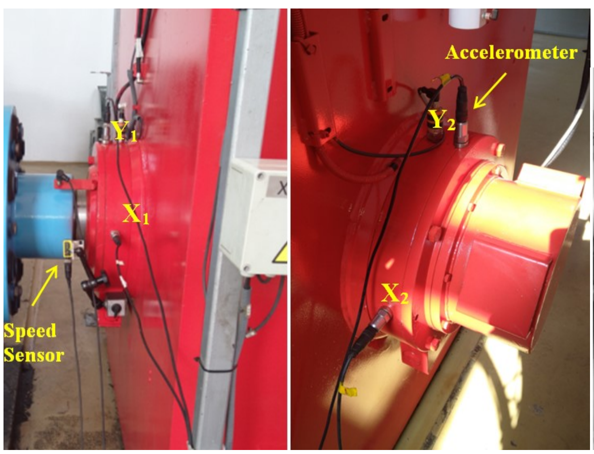


Fig. 3. Sensors position on the bearing housing.

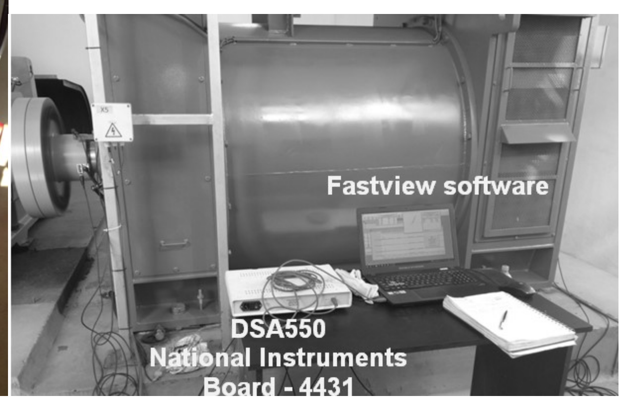


Fig. 4. Experimental device.

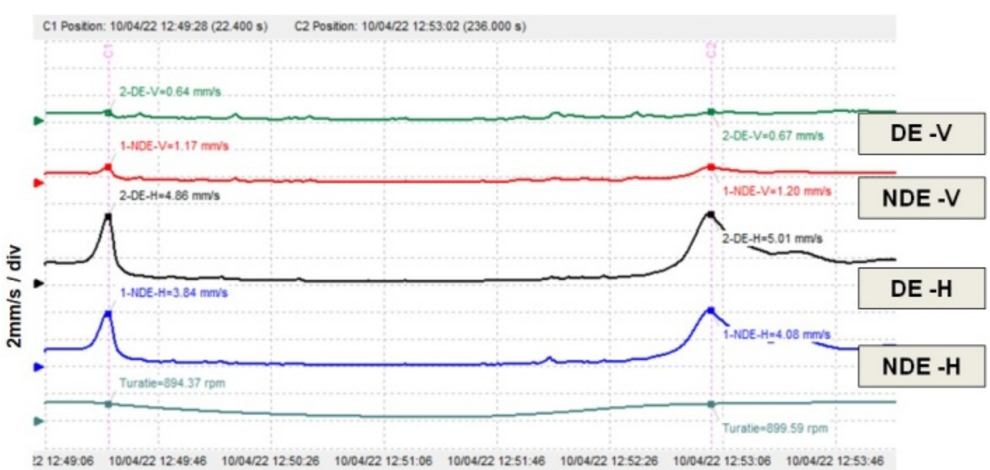


Fig. 5. The evolution of vibrations at start-up.

No.	Parameter	Value	Unit
01	Turatie	999.84	rpm
02	1-NDE-H	1.79	mm/s
03	2-DE-H	1.75	mm/s
04	1-NDE-V	1.17	mm/s
05	2-DE-V	1.24	mm/s

Fig. 6. Vibration parameter at FSNL on horizontal and vertical direction.

No.	Parameter	Value	Unit
01	Turatie	1000.34	rpm
02	1-NDE-H	4.02	mm/s
03	2-DE-H	2.74	mm/s
04	1-NDE-SCUT -H	4.76	mm/s
05	2-DE--SCUT -H	4.56	mm/s
06	H-DE-gE	0.22	gE
07	H-NDE-gE	0.26	gE

Fig. 7. Vibration parameter.

Vibration increases during the time and especially after synchronization, with the increase of the load.

During the synchronization and loading at the 1050 kW, the vibration increases exceeding the ISO limit 10816-5. In addition, for an in-depth analysis regarding the cause of vibration, the acceleration envelope signal was also processed. However, the bearings are in good condition, without defect, the amplitude on both side DE and NDE bearings being reduced (Fig. 7). In order to locate and identify the source of vibration, several measurement points were taken into account. In this sense, it was measured not only on the bearing but also on the generator housing (Fig. 8).

The large amplitude of the vibration is in the horizontal direction, much smaller in the vertical or axial direction.

The vibration spectrum (Fig. 9) presents the harmonic 2X, and for all four positions on horizontal direction, the vibration vector is in phase. The vibration cause is structural one, exciting the second order harmonic, or 33.34 Hz. The 2nd order harmonic governs the dynamic behavior of the generator, the fundamental frequency (1x) being non-existent.

On the entire generator housing, the vibration vectors have the same angular position or phase (Fig. 10).

According to results and their evolution, with the 2x harmonic, it can be considered that the high vibration at the operating speed is based on a misalignment of the generator structure. The appearance of the misalignment is given by the operation near the resonance zone, with a reserve for only 90 rpm.

Considering the cause and the high vibration level (Fig. 11) it was decided to stop and take measures to optimize the generator's stiffness, regarding the four fixing points.

After controlling the tightness and obtaining a relaxation of the structure, it was possible to achieve a better base and implicitly a correct placement on the base plate in rapport with the 4 points of fixture.

The modification of the rigidity, by reducing the contact pressure, allowed a correct placement of the structure and implicitly a reduction of vibrations (Fig. 12).

Parameters

Test: Vibration Analysis and Balancing _Power Gen
Input: oprire.fvs (10/04/22 14:01:06)

No.	Parameter	Value	Unit
01	Turatie	1000.15	rpm
02	1-NDE-H	6.53	mm/s
03	2-DE-H	4.62	mm/s
04	1-NDE-shield-H	7.59	mm/s
05	2-DE-shield-H	6.33	mm/s

Fig. 8. Accelerometer on the generator shield NDE and DE.

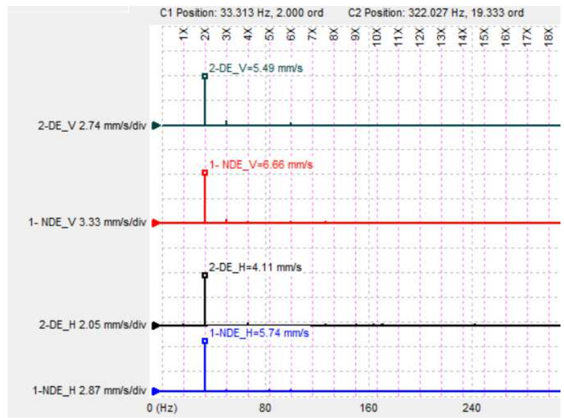


Fig. 9. Vibration spectrum at 1050 kW.

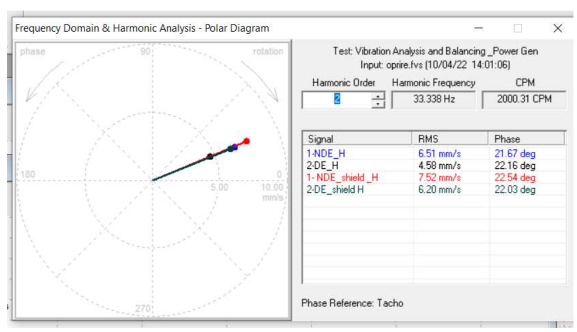


Fig. 10. Polar diagram after one hour and 30 min of operation at 1050 kW.

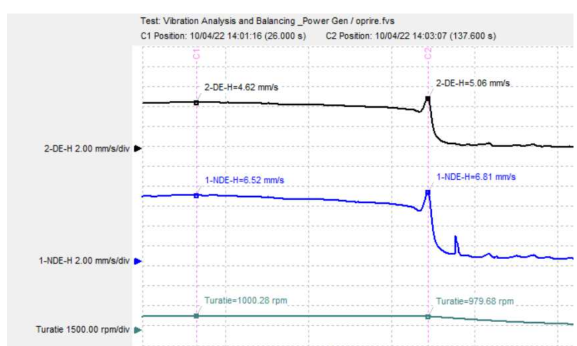
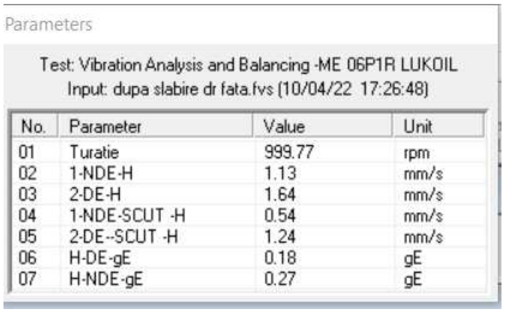


Fig. 11. Trend of vibration parameter after vibration exceeded ISO limits.



Parameters

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No.	Parameter	Value	Unit
01	Turatie	999.77	rpm
02	1-NDE-H	1.13	mm/s
03	2-DE-H	1.64	mm/s
04	1-NDE-SCUT -H	0.54	mm/s
05	2-DE--SCUT -H	1.24	mm/s
06	H-DE-gE	0.18	gE
07	H-NDE-gE	0.27	gE

Fig. 12. Vibration parameters at 1850 kW.

In this new case, the spectrum frequency has several components and not only the second order harmonic (Fig. 13).

The modification of the stiffness also allowed the modification of the critical speed, which is now at a speed of 830 rpm, with a reserve of 170 rpm, leading to stability and an acceptable level of vibrations (Fig .14).

After optimizing the generator tightening on supports, a dynamic behavior was obtained in optimal conditions of the ISO10816-2 standard, the vibration value at 1850 kW, after one hour and 30 minutes, being 1.6 mm/s rms (fig. 15).

In practice, an adjustment of the stiffness *k* of the generator clamps on the base plate was made by adjusting the clamping forces. This operation also allowed the adjustment of the position of the generator by obtaining a planarity of the seating surfaces of the supports in relation to the base plate.

5. CONCLUSIONS

After the repositioning interventions and the optimization of fixing the generator on the pedestal, the vibration level is within the limits of the ISO10816-5 standard. The maximum vibration value in the horizontal direction is 1.6 mm/s·rms at a power of 1850 kW. At the power of 1200 kW, the maximum vibration level is of 1.3 mm/s·rms.

The direction with the highest amplitude is the horizontal one, while the vertical and axial amplitudes are lower.

The cause of the high vibrations is manifested by the existence of the second order harmonic, respectively the frequency of 33.3 Hz, which is based on the existence of a resonance. In other words, the natural frequency of the generator in the case of current fastening is equivalent to twice the rotation frequency. The resonance phenomenon is highlighted both during start-up and during stop, being observed an increase in the amplitude of the second order harmonic at a speed of 900 rpm of 6.7 mm/s·rms.

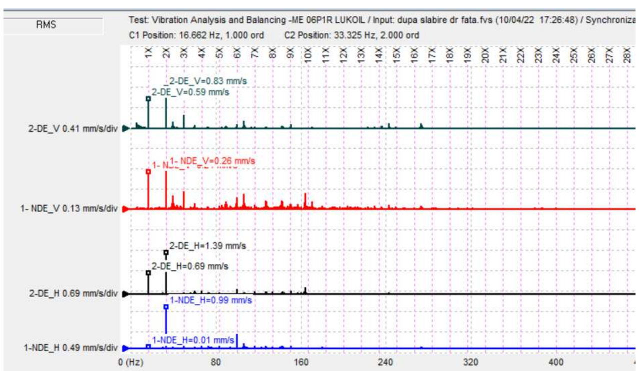

Fig. 13. Frequency spectrum at 1850 kW.

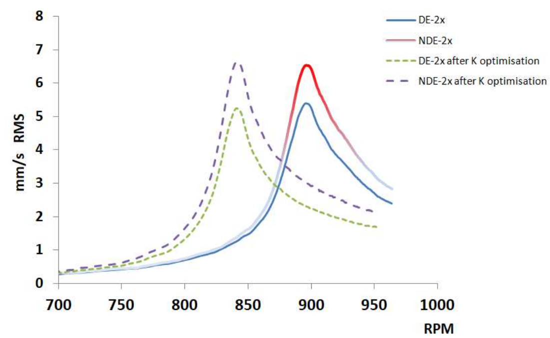

Fig. 14. Critical speed analysis after stiffness optimization.

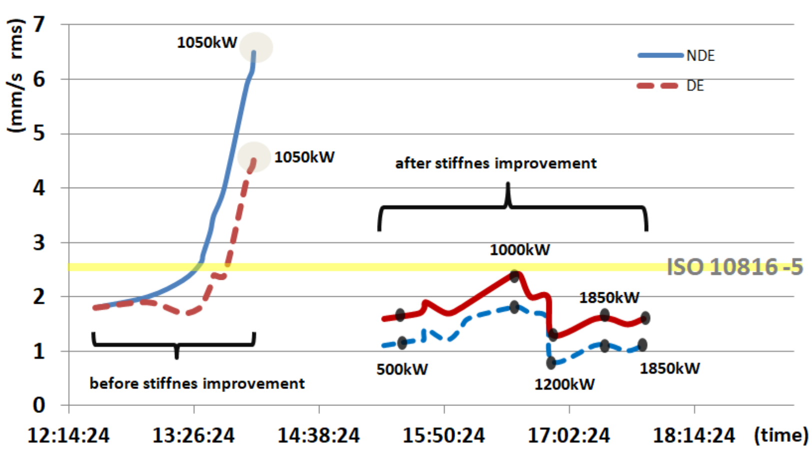

Fig. 15. Vibration evolution before and after improvement.

Because of the reduced contact on the fixing soles, the stiffness decreased leading to a natural frequency very close to the second harmonic of the operating frequency. In this situation, the deformation of the structure manifests itself through the second order harmonic in the form of a local misalignment.

After de-tensioning and relaxing the structure, optimizing the clamping force on the generator fixing points in relation to the pedestal, a significant reduction in vibrations could be obtained. The dynamic behavior of the generator improved and thus the amplitude of the second order harmonic decreased, making possible the appearance of the first order harmonic, which was non-existent before.

REFERENCES

[1] D. Ewins, ‘Control of vibration and resonance in aero engines and rotating machinery–An overview’, *International Journal of Pressure Vessels and Piping*, vol. 87, no. 9, pp. 504–510, 2010.

[2] Z. Huang and B. Han, ‘Effective approach for calculating critical speeds of high-speed permanent magnet motor rotor-shaft assemblies’, *IET Electric Power Applications*, vol. 9, no. 9, pp. 628–633, 2015.

[3] X. Chen, S. Wang, B. Qiao, and Q. Chen, ‘Basic research on machinery fault diagnostics: Past, present, and future trends’, *Frontiers of Mechanical Engineering*, vol. 13, no. 2, pp. 264–291, 2018.

[4] S. Edwards, A. W. Lees, and M. I. Friswell, ‘Fault diagnosis of rotating machinery’, *Shock and vibration digest*, vol. 30, no. 1, pp. 4–13, 1998.

[5] S. M. A. Al-Obaidi, M. S. Leong, R. Hamzah, and A. M. Abdelrhman, ‘A review of acoustic emission technique for machinery condition monitoring: defects detection & diagnostic’, *Applied Mechanics and Materials*, vol. 229, pp. 1476–1480, 2012.

[6] R. B. Randall, ‘State of the art in monitoring rotating machinery-part 1’, *Sound and vibration*, vol. 38, no. 3, pp. 14–21, 2004.

[7] J. Der Hagopian and J. Mahfoud, ‘Rotating machine critical speed suppressing by using electromagnetic actuators’, presented at the Turbo Expo: Power for Land, Sea, and Air, 2011, pp. 315–319.

[8] L. N. Virgin, J. D. Knight, and R. H. Plaut, ‘A new method for predicting critical speeds in rotordynamics’, *Journal of Engineering for Gas Turbines and Power*, vol. 138, no. 2, 2016.

[9] E. Swanson, C. D. Powell, and S. Weissman, ‘A practical review of rotating machinery critical speeds and modes’, *Sound and vibration*, vol. 39, no. 5, pp. 16–17, 2005.

[10] M. I. Friswell, J. E. Penny, S. D. Garvey, and A. W. Lees, *Dynamics of rotating machines*. Cambridge university press, 2010.