Investigation on the Dynamic Characteristics of a Rotor Suffering Impact Foundation External Excitation

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Abstract. High speed rotating machines usually include such components as rotors, bearings, casings, foundations and are widely used in many industries. However their rotors may all face foundation external excitation problems during machine operation and service. Therefore design engineers are interested in accurate rotor response prediction when support structure is subjected to sudden impact excitation in order to set sufficient clearances and ensure machine safe operation. The problem is relevant for marine engines (gas and steam turbines) for evaluation of their reliable operation for the case when a hull of the ship is exposed to impact from giant sea waves. The paper describes methodology for creation of rotor-bearing-support system for HP steam turbine rotor of a transport marine engine whose support structure was subjected to impact excitation. The impact phenomenon was further studied on the base of developed experimental test rig with a simplified rotor structure mounted on foundation with a flexible suspension system. Two numerical models were used for verification of the experimental results: the model with a simplified rotor representation (massless shaft) on a lumped mass foundation structure and a model with beam type rotor on a lumped mass foundation. Proposed numerical models showed adequate results for rotor response prediction, what was confirmed by similarity of obtained curves for impact excitation coefficients and comparability of rotor disk orbits for experiment and simulation. Experimental testing confirmed that external foundation impact excitation may significantly influence on maximum deviations of rotor disk orbits in comparison with the case of rotor normal operation without excitation. Simulation and experiment results revealed that impact excitation coefficients within the tested range of amplitudes and rotor speeds increased almost linear and were proportional to maximum displacement amplitude measured on foundation. For subcritical and supercritical speeds impact excitation coefficients were close in values and increased faster in comparison with excitation performed for the speeds close to rotor critical speed. Proposed method for model creation and analysis could be further used for rotordynamic simulations of more complicated machines e.g. marine power engines.

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Impact excitation · Marine engines

1 Introduction

High speed rotation machines usually include such components as rotors, bearings, casings, foundations and are widely used in many industries. However their rotors may all face foundation external excitation problems during operation and service. Therefore design engineers are interested in accurate rotor response prediction when support structure is subjected to sudden impact excitation in order to set sufficient clearances and ensure machine safe operation. Problem of foundation structure modeling over the years was a subject of fundamental studies. Different types of rotor-foundation models with varying degree of complexity for ground power machines could be found in works of Kramer [1], Kostyk [2], Savinov [3], Shulzhenko [4], Runov [5]. Examples of foundation structure simplified representation with lumped concentrated masses could be found in works of Nicholas et al. [6, 7], Han et al. [8], Ma et al. [9], Kang et al. [10]. The recent trend is focused on improvement of simulation accuracy and requires to represent a foundation structure as a multi-degree of freedom system. Models with foundation structure represented as a base plate could be found in works of Vasquez et al. [11], Lees et al. [12], Cavalca et al. [13–15], Bonello et al. [16], Feng et al. [17]. Examples of models with foundation structures created for real power plants and industrial power machines could be found in papers [18–23]. Recent tendencies and most common techniques for rotor-foundation structures modeling and testing were also observed by Hong et al. [24]. Attempts to solve problem of vessel hulls external excitation and its influence on dynamics of rotating equipment both theoretically and experimentally were performed by American naval engineers [25–28]. Influence of kinematic excitation on dynamics of typical rotor-foundation system was investigated by Shatohin et al. [29] and Zhu et al. [30]. Problem of foundation base excitation with brief description of rotor orbits could be found in works of Duchemin et al. [31] Driot et al. [32, 33], Cole et al. [34], El-Saeidy et al. [35], Das et al. [36].

For many years variety of papers from different industries focused on study of rotor-foundation structure interaction were accumulated, but still this problem is not completely solved. Number of works pointed on description of rotordynamics for rotors subjected to foundation impact excitation is limited. In recent times the question is relevant for marine engines (gas and steam turbines) for evaluation of their safe and reliable operation for the case when the hull of the ship is subjected to impact from the sea waves. Experimental testing performed by marine engineers had shown that impact force of the ocean waves on fixed hull wall is significant and can easily reach values of 10, 20 and even 30 tons on square meter [37, 38]. The energy carried by a wave is proportional to the square of its height. As mentioned in [38] if the vessel goes with the speed at 20 knots and stumbles with the wave, which travels at 35 knots, the resulting slamming force has potential to seriously damage bow of the ship. In physical oceanography phenomenon of giant “freak waves” (rogue waves, monster waves, killer waves) is well known. Rogue waves are large and spontaneous surface ocean waves that can be extremely dangerous even to large marine vessels like container ships and ocean cruise liners. Numerical reasoning for sudden emergence of freak waves on the ocean surface was demonstrated in paper [39]. As indisputable evidence of their existence, giant rogue wave was photographed from the deck of Esso Languedoc tanker ship near the shores of South Africa in 1980s, Fig. 1.

It could be said that during the design and development of marine engines, dynamic

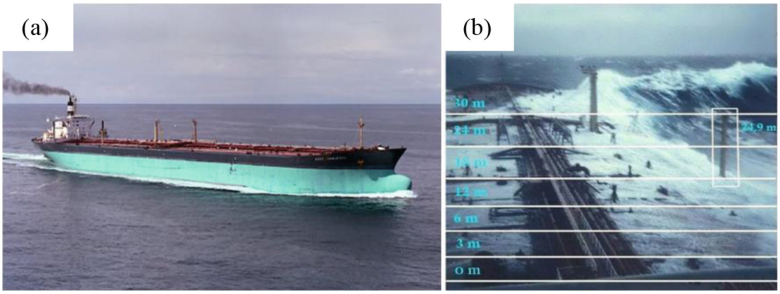


Fig. 1. (a) Esso Languedoc tanker ship (b) Photo of giant rogue wave made from the Esso Languedoc tanker ship

forces from wave impact should be also taken into account in rotordynamic simulations. Previously authors of [40] had already studied the influence of harmonic kinematic excitation applied on foundation on rotordynamics using numerical modeling and experimental testing with example of a simple rotor model. The aim of the current paper is to continue this study and to evaluate the influence of foundation impact excitation on the dynamics of rotor system both numerically and experimentally.

2 Theoretical Development

In mechanics, impact is a short duration excitation, which induces transient dynamic stresses in structure. From kinematic point of view this event can be considered as a short moment event when the points of the full system experience sudden velocity changes. At the same time impact can be also described as emergence and then vanishing of significant impact forces. In such a way equations of motion for rotor-foundation system with centrifugal force from residual unbalance and sudden impact excitation force applied on foundation in general form can be written as system:

$$\begin{cases} [M]\ddot{q} + [C]\dot{q} + [K]q = \{F_{unb}\}, & t \neq t_{imp} \\ [M]\ddot{q} + [C]\dot{q} + [K]q = \{F_{unb}\} + \left\{F_{imp}\right\}, & t = t_{imp} \end{cases}, \tag{1}$$

where [M], [C], [K] - are general mass, damping and stiffness matrixes for the full rotor-foundation system; q - d.o.f. vector in general coordinates; $\{F_{unb}\} = \{U\omega^2\sin(\omega t); U\omega^2\cos(\omega t)\}$ - vector of residual unbalance; U = me- unbalance; $\{F_{imp}\}$- load vector for the impact event; t_{imp}- is the time corresponding to impact event.

To investigate how the rotor of the marine power engine will respond when its support structure is subjected to impact excitation from the ocean wave, simplified beam model was built for high pressure (HP) steam turbine rotor of TS-2 engine, Fig. 2, in XLRotor rotordynamic software based on available in open sources information [41]. TS-2 is a marine power engine with a rated power of 13 970 kW which was used at tanker ship types “Prague” and “Sofia”. Exact rotor mass, mass and number of blades, rotor material were unknown, hence obtained model is only assumption used for paper study. Alloy steel with E = 2.07E + 11 Pa and density ρ = 7800 kg/m^3 were used for rotor model.

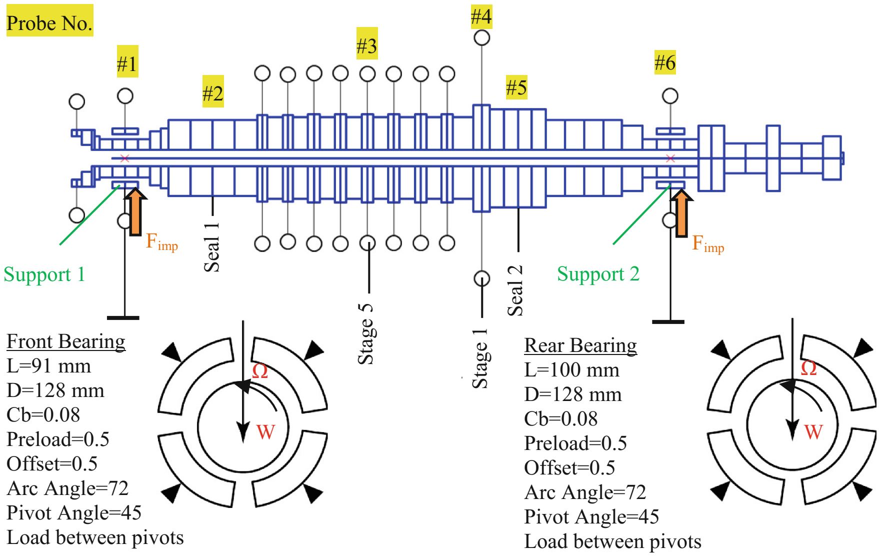


Fig. 2. HP steam turbine rotor model for marine power engine

The mass of foundation structure was also unknown, and simplified representation of the support structure in form of lumped masses (equal to bearing bushings) was used: $M_{support}$ = 50 kg, $K_{support}$ = 1.0E + 06 N/mm. Supports were assumed to be made of steel and modal damping for them was set ξ = 0.01. Exact bearing geometry parameters were unknown and to simplify simulation and to avoid problems with instability 4 pad tilting pad bearings with load between pads were used for front and rear bearing. Unique feature of latter is that horizontal and vertical bearing stiffness and damping coefficients are equal. Typical SAE 10W oil was used for simulation with oil inlet temperature T = 35 °C. Stiffness and damping coefficients were obtained for both bearings by solution of Reynolds equations. Overlap of vertical direct bearing stiffness coefficients on critical speed map helped to estimate that rotor first undamped critical speed will be around 3900 rpm, Fig. 3. Rotor operating speed was known to be 5500 rpm. Evaluation of critical speed map had shown that rotor operates below rotor bending mode and can be considered as rigid. Rotor damped critical speed for mode 1 was identified to be 4250–4300 rpm measured at the bearings. Separation margin for mode 1 critical speed

(SM = 21.8%) satisfies API requirements [42], hence model can be further used for analysis. Assuming that rotor works with residual unbalance (7.49 kg-mm) which corresponds to G 2.5 balance grade [43] rotor orbits measured at 6 locations at rotor operating speed (5500 rpm) were obtained from simulation in XLRotor by solution of the first equation from system (1), Fig. 4(a). In all planes of measurement rotor orbits were stable and had elliptical shapes. For the next case of study rotor support structure was subjected to foundation impact excitation (t_{imp} = 0.01 s, F_{imp} = 5 tons) applied in horizontal plane. Rotor absolute orbits obtained from solution of second equation from system (1) in all planes of measurement were non-symmetric and different from the case of normal operation, Fig. 4(b). Impact excitation forced them to deviate rapidly in the plane of applied excitation and get sharp distorted shape.

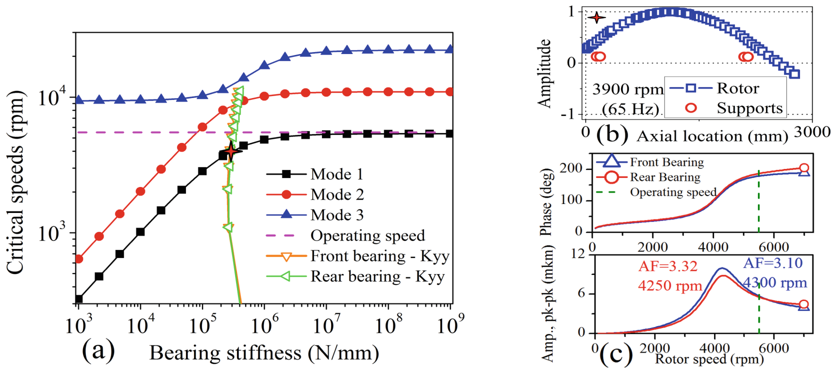


Fig. 3. Marine HP turbine rotor: (a) Undamped critical speed map; (b) Mode 1 – undamped critical speed and mode shape; (c) Bode plot

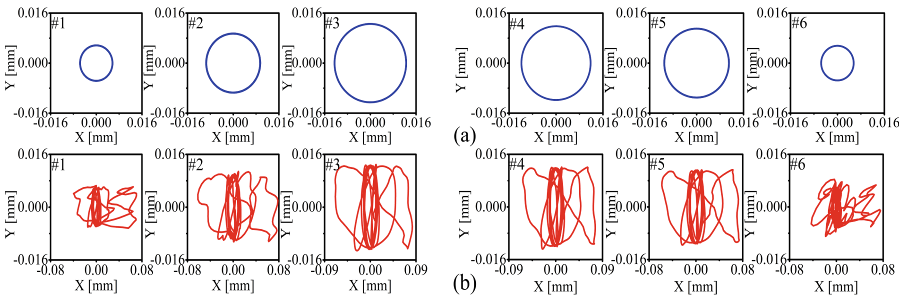


Fig. 4. Absolute rotor orbits for marine HP turbine rotor: (a) Normal operation; (b) With Impact excitation – F_{imp} = 5 tons at each support

To investigate the influence of foundation impact excitation on rotor displacement amplitudes, impact excitation force applied on each foundation support was set to change from 5 to 30 tons. Simulation results revealed that in all planes of measurement rotor displacement maximum amplitudes increased almost linear and were proportional to applied excitation force, Fig. 5. Applied impact excitation of 5 tons was already enough to force rotor exceed its bearing clearance. Assuming that seal clearances were set equal to 0.5 mm, applied impact excitation of 30 tons was sufficient to force rotor exceed clearance limits, especially at rotor end-seals location (#2 at Fig. 5) and rotor midspan stages (#3, #4 at Fig. 5).

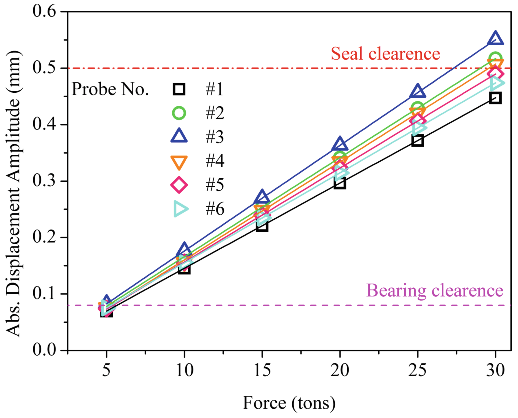


Fig. 5. Simulation results for HP turbine rotor with support structure subjected to excitation from the wave impact

3 Experimental Model

3.1 Experimental Test Rig

The further step of the research was focused on investigation of the influence of impact excitation applied on foundation structure on dynamics of rotor system on example of simple test rotor. For that purpose experimental test rig with flexible foundation system was designed and built in laboratory of vibration testing at School of Energy and Power Engineering in Beihang University, Fig. 6.

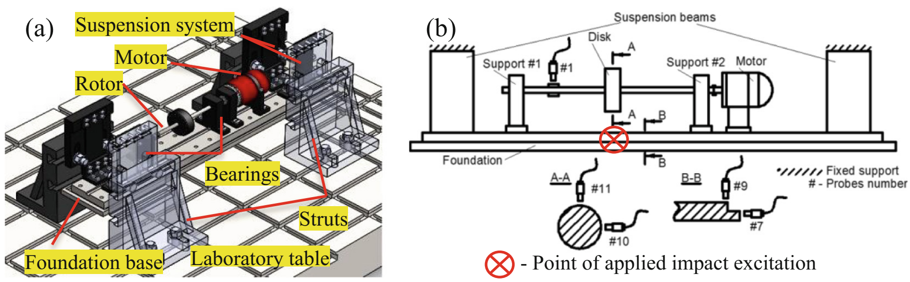


Fig. 6. Experimental test rig: (a) General view; (b) Probes location

Test rig presented a simplified model of a general rotor-foundation system with a single disk rotor mounted on a table-type foundation base. The rotor was rested on two cylindrical oil bearings. Foundation table had a flexible suspension system which consisted from four flexible suspension beams. These suspension beams allowed the foundation base to move in one general direction (horizontal). The beams were connected with heavy struts which were rigidly fixed on the metal-concrete laboratory table. For experimental study the rotor-foundation system was connected with centrally mounted shaker which allowed to perform excitation in horizontal direction. Shaker was connected with an acquisition system and programmed to perform kinematic excitation in form of a pulse signal. The force of applied excitation was not measured, while amperage and frequency of excitation were tuned in order to control displacements of the foundation base. The schematic view of the location of probes is shown in Fig. 6(b). For experimental measurements the test rig was equipped with 6 probes. Standard displacement transducers were used to measure rotor disk absolute vibration (horizontal and vertical – probes #10 and #11) and foundation absolute vibration (horizontal and vertical – probes #7 and #9). All the transducers were mounted on special probe struts which were fixed motionless on laboratory table. Probe #1 was set in front of the ring with the notch which was fixed on the shaft to perform rotor speed and phase measurements. All of the probes were connected through amplifiers with DASP Data Acquisition System to perform on-line experimental measurements and recording stored on PC. Two operation conditions were considered during experiment: normal operation and operation with foundation impact excitation. Schematic view on measurements for the latter one is shown in Fig. 7(a).

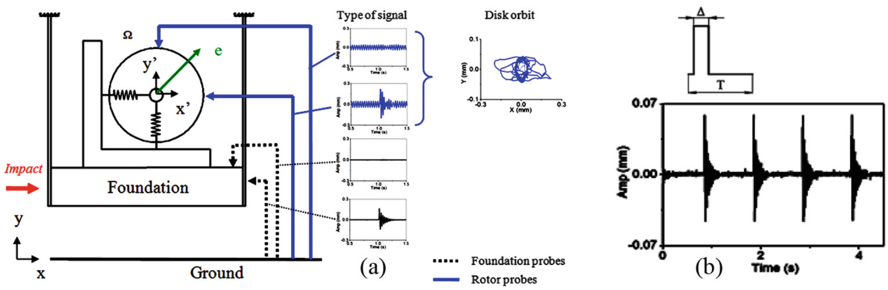


Fig. 7. Principle of experimental measurements: (a) General view; (b) Impact excitation signal

By the term “normal operation” in the current paper was assumed the rotor operating at a constant rotational speed with some small residual imbalance. To perform impact excitation rectangular pulse signal was used for the shaker with pulse width Δ equal to 1% and pulse period T = 1 s, Fig. 7(b). Foundation structure of the test rig was subjected to series of short pulses in horizontal direction. Amperage parameters for the shaker were controlled to ensure that foundation displacement maximum double amplitude was constant for every case of measurement.

3.2 Experimental Measurements – Normal Operation

Test rotor run-up/run-down was performed to get rotor-foundation system dynamic characteristics, Fig. 8. Measurements had shown that foundation structure resonance in horizontal direction was near 1966 rpm (≈33 Hz). First critical speed for the rotor shaft was in vicinity of 6233 rpm (≈104 Hz). It was identified as rotor's first bending mode.

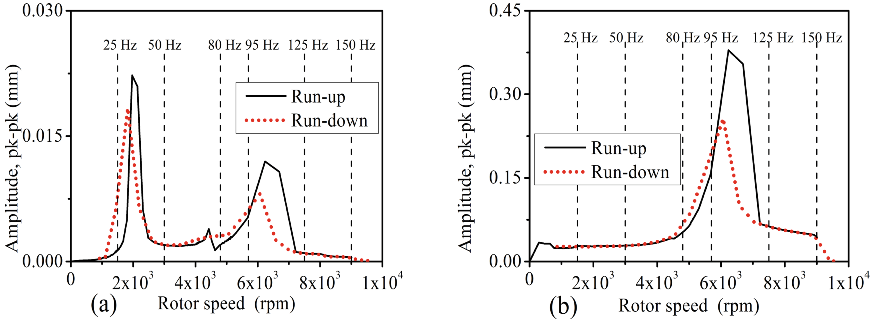
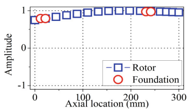
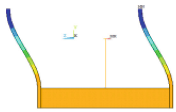
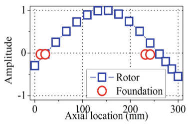
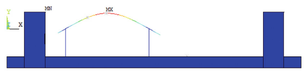


Fig. 8. Rotor-foundation experimental dynamic characteristics: (a) measured on foundation – horizontal direction; (b) measured on the disk – horizontal direction (dashed line – chosen rotor speeds for measurements)

Experimentally obtained critical speeds were in good agreement with simulation results: mode shapes for test rig rotor-foundation structure obtained by FEM modeling were summarized in Table 1. Six rotor speeds were chosen for measurements: three speeds for subcritical operation (Ω = 25, 50 Hz and 80 Hz), one speed close to rotor critical speed (Ω = 95 Hz) and two speeds for rotor supercritical operation (Ω = 125 Hz, 150 Hz). Rotor orbits for all speeds of interest for the case with no excitation were summarized in Fig. 9.

Table 1. Predicted natural frequencies and mode shapes for test rig rotor-foundation structure

No.	Lumped mass foundation model		Plate type foundation model	
	Mode description	Mode shape/ frequency, Hz	Mode description	Mode shape/ frequency, Hz
1	Foundation mode	 37.81	Foundation mode	 36.98
2	Rotor first bending mode	 Vert. 102.46 / Hor. 104.72	Rotor first bending mode	 Vert. 98.6/Hor. 99.6

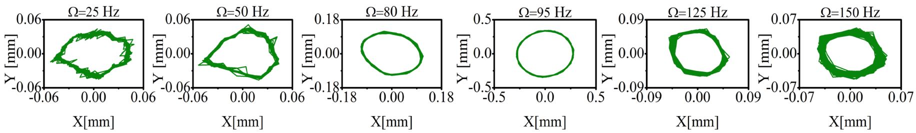


Fig. 9. Rotor disk absolute orbits obtained experimentally – normal operation

Experimentally obtained rotor disk orbits for all speeds of interest were stable and tended to have an elliptical shape, which is common for rotors working on fluid oil bearings.

3.3 Experimental Results – Foundation Impact Excitation

Performed experimental testing helped to identify that test rig foundation structure logarithmic decrement was $\delta = 0.466$. After excitation test rig foundation structure was inherent to have a decaying vibration signal which is common for non-rotating structures subjected to impact excitation. Measurements had shown that through the bearings response from the impact excitation was transmitted to the disk of the rotor and forced it to deviate for a short moment from normal stable operation trajectory. Meanwhile after the vanishing of excitation force, rotor disk orbit was tended to return to a normal level of vibration corresponding to its current speed of rotation. Obtained experimental rotor disk absolute orbits for different cases of impact excitation when foundation displacement double amplitude was changing were summarized in form of Table 2.

Table 2. Experimental rotor absolute orbits after impact excitation - influence of foundation displacement amplitude

Ω	Impact max amplitude measured on foundation, pk-pk				
	0.1 mm	0.2 mm	0.3 mm	0.4 mm	0.5 mm
25 Hz					
95 Hz					
150 Hz					

It could be seen from the obtained results that for subcritical and supercritical rotor operation influence of impact excitation on rotor disk orbit deviations was more severe. Increase of foundation displacement double amplitude had influence on rotor orbits rapid extension in the plane of applied impact excitation. It should be noted that impact event usually force rotor orbit to look non-symmetric, what could be used as diagnostic sign for a similar type of events. In order to estimate the influence of foundation displacement amplitude on maximum deviations of rotor disk, impact excitation coefficients (absolute and relative) for the full system were used:

$$K_{imp} = \frac{A_{imp}}{A_0} \tag{2}$$

$$\overline{K}_{imp} = \frac{\overline{A}_{imp}}{\overline{A}_0}, \tag{3}$$

where A_{imp}, $\overline{A}_{imp}$ - are absolute and relative amplitudes measured on the disk when foundation structure was subjected to impact; A_0, $\overline{A}_0$ - are absolute and relative amplitudes measured on the disk for the case of normal operation. Impact excitation coefficients for all speeds of interest were also summarized in form of plots in Fig. 10.

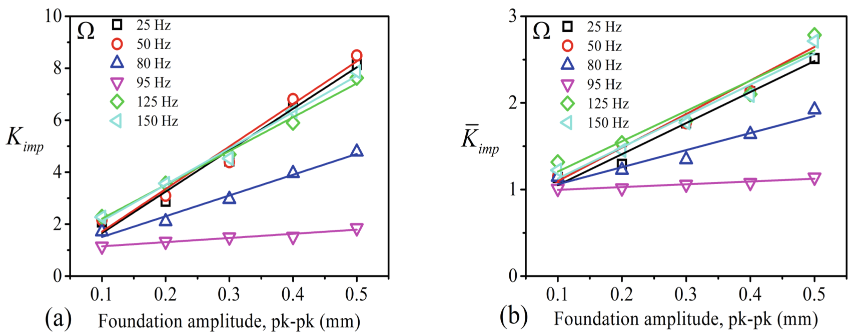


Fig. 10. Experimental impact excitation coefficients: (a) Absolute; (b) Relative

Experimental results revealed that impact excitation coefficients within the tested range of amplitudes and rotor speeds increased almost linear and were proportional to maximum displacement amplitudes measured on foundation. For subcritical and supercritical speeds impact excitation coefficients were close in values and increased faster in comparison with excitation performed for the speeds close to rotor critical speed.

4 Simulation Results

4.1 Model Description

Two models were used for verification of the experiment results: simplified rotor model (massless shaft) on lumped mass foundation structure (MS-LMF model) and beam type rotor model on lumped mass foundation (BR-LMF model). In matrix form equation of motion for MS-LMF model could be written as:

$$\begin{bmatrix} M_d & 0 & 0 & 0 \\ 0 & M_d & 0 & 0 \\ 0 & 0 & M_f & 0 \\ 0 & 0 & 0 & M_f \end{bmatrix} \begin{Bmatrix} \ddot{x}_r \\ \ddot{y}_r \\ \ddot{x}_f \\ \ddot{y}_f \end{Bmatrix} + \begin{bmatrix} C_{bx} & 0 & -C_{bx} & 0 \\ 0 & C_{by} & 0 & -C_{by} \\ -C_{bx} & 0 & C_{bx}+C_{fx} & 0 \\ 0 & -C_{by} & 0 & C_{by}+C_{fy} \end{bmatrix} \begin{Bmatrix} \dot{x}_r \\ \dot{y}_r \\ \dot{x}_f \\ \dot{y}_f \end{Bmatrix} + \begin{bmatrix} K_{bx} & 0 & -K_{bx} & 0 \\ 0 & K_{by} & 0 & -K_{by} \\ -K_{bx} & 0 & K_{bx}+K_{fx} & 0 \\ 0 & -K_{by} & 0 & K_{by}+K_{fy} \end{bmatrix} \begin{Bmatrix} x_r \\ y_r \\ x_f \\ y_f \end{Bmatrix} = \{F_{unb}\} + \{F_{imp}\}, \tag{4}$$

where $q_r = \{x_r, y_r\}$ - is the vector of rotor disk displacements; $q_f = \{x_f, y_f\}$ - is the vector of foundation displacements; M_d, M_f – are the masses of rotor disk and foundation; C_b, C_f, K_b, K_f – are damping and stiffness coefficients for bearing and foundation. Equation of motion for the rotor of BR-LMF model could be written as:

$$([M_s] + [M_d])\ddot{z} + ([C_s] + \Omega[G])\dot{z} + [K_s]z = \{F_{unb}\} + \{N_b\} + \{N_d\}, \tag{5}$$

where $[M_s]$, $[C_s]$, $[K_s]$ – are the mass, damping and stiffness matrix for the shaft; $z = \{x,y\}$ - is the vector of shaft displacements for j-th element of the shaft; $[M_d]$, $[G]$ – are the mass and gyroscopic matrix of the disk; Ω - is rotational speed; $\{F_{unb}\}$-is unbalance force; $\{N_b\}$- is the vector of interactive forces in common nodes of the shaft and disk; $\{N_d\}$ – is the vector of interactive forces in common nodes of bearing and foundation suspension. Equation of motion for bearing-foundation system for BR-LMF model could be written as:

$$\begin{bmatrix} [M_{ss}] & 0 \\ 0 & [M_{ff}] \end{bmatrix} \begin{Bmatrix} \ddot{z}_b \\ \ddot{z}_f \end{Bmatrix} + \begin{bmatrix} [C_{bb}] & -[C_{bf}] \\ -[C_{fb}] & [C_{bb}]+[C_{ff}] \end{bmatrix} \begin{Bmatrix} \dot{z}_b \\ \dot{z}_f \end{Bmatrix} + \begin{bmatrix} [K_{bb}] & -[K_{bf}] \\ -[K_{fb}] & [K_{bb}]+[K_{ff}] \end{bmatrix} \begin{Bmatrix} z_b \\ z_f \end{Bmatrix} = -\{N_b\} + \{N_s\} + \{F_{imp}\}, \tag{6}$$

where $\{F_{imp}\}$ – is the vector of foundation impact excitation; $z_b = \{x_b, y_b\}$ – is the vector of the shaft centers displacements corresponding to the bearings locations; $z_f = \{x_f, y_f\}$ – is the vector of foundation displacements; $[M_{ss}]$ – is the mass matrix for the shaft elements in common nodes of bearings and foundations; $[C_{bb}]$, $[K_{bb}]$ – are damping and stiffness matrixes for the bearings; $[M_{ff}]$ – is the mass matrix for foundation; $[C_{ff}]$, $[K_{ff}]$ – are damping and stiffness matrix for foundation; $[C_{bf}]$, $[C_{fb}]$, $[K_{bf}]$, $[K_{fb}]$ – are cross-coupled damping and stiffness matrix for bearings and foundation interaction.

Solution of Eq. (4) was performed in MATLAB Simulink, Fig. 11. General view on BR-LMF model built in XLRotor is shown in Fig. 12. Linearized bearing stiffness and damping coefficients for BR-LMF model were obtained by solution of Reynolds equations. Simulation results are shown in Fig. 12(b). L/D = 0.6, radial clearance $\Delta = 0.02$ mm and Typical SAE 10W oil lubricant were used for both bearings of the model.

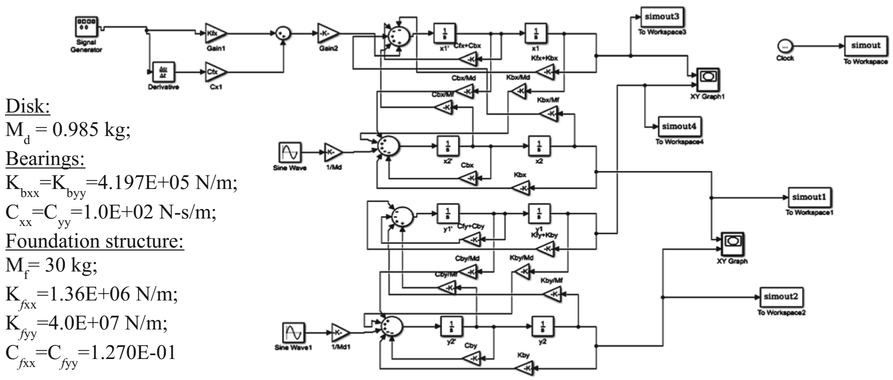


Fig. 11. MS-LMF model general view and model parameters

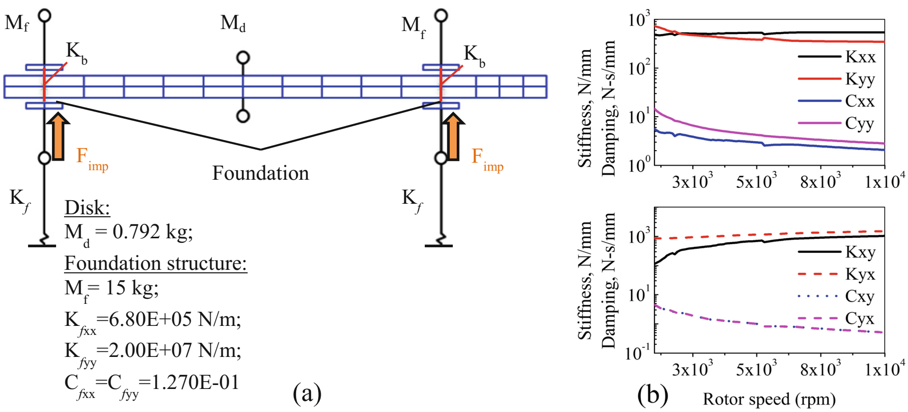


Fig. 12. BR-LMF model general view: (a) Rotor model; (b) Bearing stiffness and damping coefficients

For every model rotor residual unbalance was tuned to get rotor disk absolute vibration for every speed of measurement for the case with no excitation to be close with results obtained experimentally. On the next step foundation structure of both models was subjected to impact external force applied in horizontal plane with t_{imp} = 0.02 s to simulate rectangular pulse signal used in experiment. Applied excitation force was controlled for every case of measurement to ensure that foundation displacement double amplitude was constant and equal to experimental. Simulations for MS-LMF model were performed in MATLAB using ode4 solver based on Runge-Kutta method with the time step 1E−05 s. Meanwhile BR-LMF model was solved using Newmark method with the same time step in XLRotor rotordynamic code.

4.2 Simulation Results – Foundation Impact Excitation

Obtained rotor disk absolute orbits for selected speeds of interest were summarized in form of plots in Fig. 13. For all cases they were quite similar with experimentally measured. Simulation results confirmed that impact excitation forced rotor disk orbits to deviate in the plane of applied excitation from normal elliptical shape. Subjected to excitation rotor orbits were transforming to sharp and non-symmetric. Simulations also proved that influence of impact excitation for subcritical and supercritical rotor speeds was several times stronger than for excitation near rotor critical speed.

Impact excitation coefficients (absolute and relative) for all speeds of interest were summarized in form of plots in Fig. 14. Within tested range of amplitudes and rotor speeds obtained with simulation impact excitation coefficients were similar with experimental, Fig. 10. They also increased almost linear and were proportional to maximum displacement amplitude measured on foundation. In the same manner, obtained from experiment and simulation rotor orbits for simple test rotor were similar in nature with simulation results for more complicated rotor model of marine power engine, what proves relevance of developed numerical models.

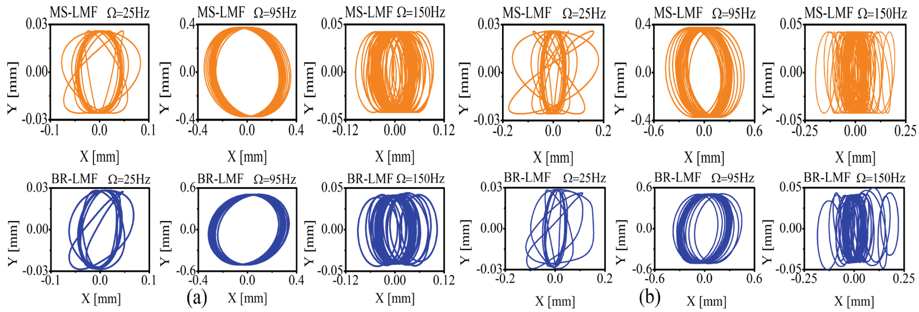


Fig. 13. Rotor disk absolute orbits – simulation results. Impact max amplitude on foundation, pk-pk: (a) 0.1 mm; (b) 0.3 mm

Comparison of modeling and experimental results, Fig. 15, had shown that for subcritical operation (Ω = 25 Hz) model accuracy (for absolute and relative impact coefficients) decreased with increase of foundation displacement double amplitude. For supercritical operation (Ω = 150 Hz) simulation results were quite close to experimental for the absolute values, but still lower in case of relative impact coefficients. Good agreement between experimental and simulation results was achieved for the case of impact excitation near rotor critical speed. It is interesting to point out that developed simplified numerical model in MATLAB was able to follow more complicated beam model with bearing coefficients from Reynolds equations obtained in rotordynamic code.

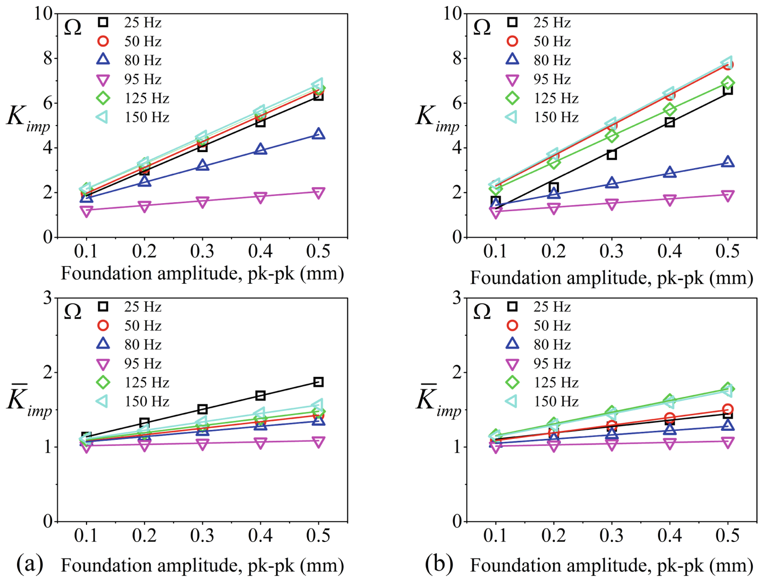


Fig. 14. Impact excitation coefficients obtained from simulation: (a) MS-LMF model; (b) BR-LMF model

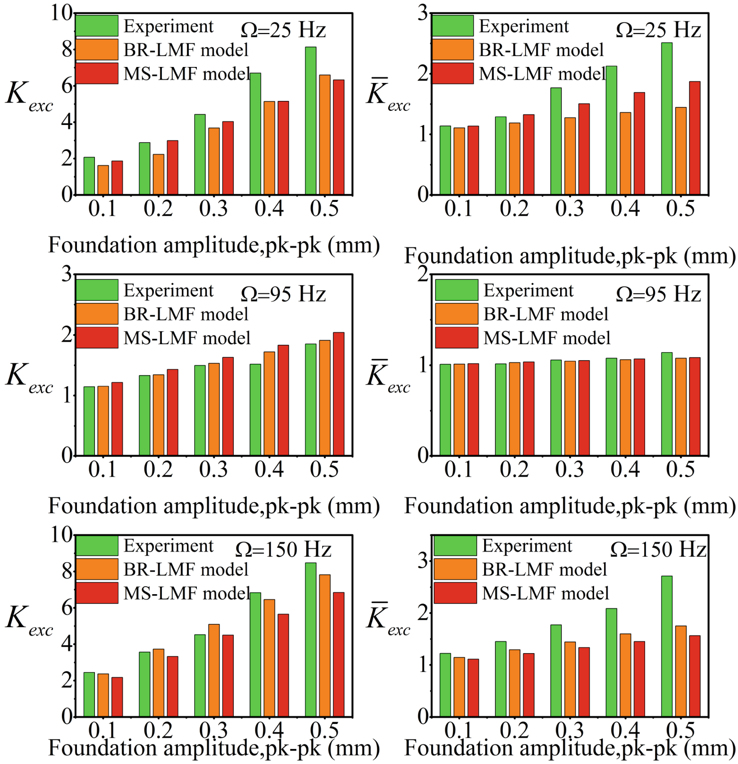


Fig. 15. Comparison of simulation results with experimental for the case of impact excitation

5 Conclusion

Obtained in current paper results can be briefly summarized as:

- Methodology for creation of rotor-bearing-support system for typical transport marine engine whose support structure was subjected to impact excitation was presented. Obtained simulation results had shown that applied excitation from the wave impact was sufficient to force rotor exceed its seals clearance limits. Such forces have potential to seriously damage not only the hull of the ship but also the engine, and thus they should be taken into account during rotordynamic simulations and clearance selection for marine power engines;
- Impact testing performed for rotor-foundation structure of developed experimental test rig indicated that impact excitation coefficients for rotor disk within the tested range of amplitudes and rotor speeds increased almost linear and were proportional to maximum displacement amplitudes measured on foundation;
- Both testing and simulation confirmed that impact excitation forced rotor disk orbits to deviate in the plane of applied excitation from normal elliptical shape. Subjected to excitation rotor orbits were transforming to sharp and non-symmetric. Simulations proved that influence of impact excitation for subcritical and supercritical rotor speeds was several times stronger than for excitation near rotor critical speed;
- Proposed numerical models showed adequate results for rotor response prediction, what was confirmed by similarity of obtained curves for impact excitation coefficients and comparability of rotor disk orbits for experiment and simulation. At the same time for subcritical and supercritical speeds simulation results were lower than experimental for all types of model what could be explained by model simplicity and inaccuracy in its bearing stiffness and damping representation;
- Obtained from experiment and simulation rotor orbits for simple test rotor were similar in nature with simulation results for more complicated rotor model of marine power engine, what proves relevance of developed numerical models.

Finally it can be concluded that the main difficulty in rotordynamics modeling for rotor-foundation structure is in the construction of such models i.e. compilation of adequate mechanical model which can be simple, fast and accurate. The latter is the main difficulty since every model is only assumption which can describe the real phenomenon usually as simplified representation and only in certain range. In such a way the accuracy of the model is usually in direct relation from how well the physics of the process is understood and how accurate the model components (connections, boundary conditions and applied external loads) are represented.

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