898. Modeling and simulation of variable inertia rotor

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Abstract. This article presents experimental and theoretical research of a rotor system with variable inertia moment. A mechanical system model was designed using SimMechanics toolbox in Matlab Simulink. Experimental and simulation results of the test rig are presented in the article. In order to compare the obtained results, model parameters were chosen based on actual parameters of the test rig. Results obtained from the analysis of the developed dynamic model enable to propose a rotor design with a variable inertia moment which, under certain initial conditions, would maximally reduce forces exerted on bearings during the operation.

Keywords: variable inertia rotor, dynamics, modeling, simulation, vibration.

Introduction

The major cause of failure of continuous operation technological machines, for example, chain and pendulum rotor mills with variable inertia moment, is breakdown of bearings. The major cause of bearing breakdown is constant or variable unbalance causing high loads during machine operation. Negative effect of unbalance is evident during machine run-up and coast-down as well as when the rotation speed of a rotor system is similar to the resonance speed [1, 5, 6]. A large variety of software can be used for numerical analysis of systems. SimMechanics with Simscape language and Matlab/Simulink environment are often chosen for modeling and simulating. SimMechanics software is a block Simulink Simscape programming environment, designed for modeling of rigid bodies of mechanical systems. SimMechanics block library contains groups of basic mechanic elements, like: bodies, sensors, actuators, joints, springs, dampers and etc. By varying and connecting them and setting parameters and joining with other Simscape or Simulink blocks, it becomes possible to design almost every mechanical system [2, 3, 8].

Research object

In order to validate the numerical model, an experimental test rig with variable inertia moment was designed (Fig. 1). The test rig is composed of: 1 – control unit, 2 – DC motor, 3 – optical tachometer (designed for measuring of speed and phase angle), 4 – rotor, rotating in journal bearings, 5 – disk with chains, imitating a variable inertia moment, one pair of displacement sensors 1X and 1Y, coupling, joining the motor with a shaft, and two supports. The disk contains four chains attached. Each of them has 5 elements (links). The disk is tightly attached to the rotor. In the test rig the rotor system rotates in journal bearings, while the rotational speed of the rotor is varied by the electronic control unit placing acceleration.

Multi-channel vibration signal analyzer unit OROS Mobi-Pack OR-36 and software OROS NVGate V7.00 and OROS ORBIGate V3.00 were used for signal recording and analysis. Other items, such as proximity probes (model PR6423/005-141-BNC, sensitivity 8000 V/m), optical tachometer ORAC-TACO-02 for measurement of a rotation speed and electronic signal converters were used for the analysis of vibration signals. Analysis of simulation results was carried out with the choice of the most informative data formats of absolute vibration displacement signals: trend during run up and coast down, vibration displacements versus order spectrum, Bode diagrams, orbits waterfall diagrams, etc. with the help of which vibration sources and shaft position in a bearing were identified [4].

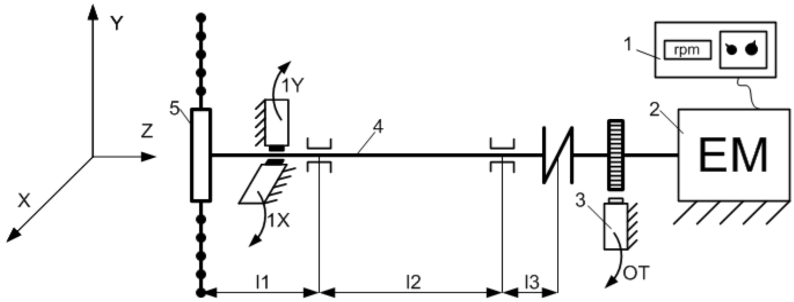

Fig. 1. Test rig for studying dynamics of a rotor system with variable inertia moment

Before commencing the simulation, the assembled test rig was balanced in compliance with ISO 1940/1 requirements corresponding to the highest 0,4 quality class. In compliance with the named class requirements, the residual unbalance of such rotor at 2000 r/min might be up to 1,910 g mm. Having performed the balancing, the system has been balanced by reducing the unbalance from 12 g mm (initial value of unbalance) up to 0,982 g mm, which is twice less than the permissible balance based on the class defined above. The balancing was carried out with OROS Mobi-Pack OR-36 vibration signal analyzer using the OROS Balancing software.

Experiment process: experimental vibration measurement of a rotor system were performed by speeding the rotor system up to 9300 rpm rotation speed, and changing the acceleration of run up and coast down.

Numerical model of the rotor system

This mechanical system was modeled in Simulink environment and represented in Simulink environment with Simulink Simscape and SimMechanics block diagrams in Fig. 2. Each element containing inertia moment is presented with the following parameters: mass, matrix of inertia and body coordinate system. Inertia and mass of a motor shaft was not assessed; only the shaft rotation speed was estimated. Since the motor has not been estimated as well as the coupling, the rotation principle was set directly to the shaft. The disk "weld" is fixed tightly to the shaft by the block element, given in Fig. 2. There are four chains attached to the disc, each of them having 5 elements. Gravitational force is assessed in the model due to which the variable inertia moment occurs upon low shaft speed until centrifugal force of the rotating shaft with a disk and chains do not exceed gravitational force and the chains are not fully stretched. The stiffness of connection joints was assessed using the springs and damper elements by setting them at damper and stiffness modes. Variable rotation speed of a rotor was set using a speed control block. With the help of such, a steady acceleration and a maximum speed limit were set.

Analysis mode was set to forward dynamics. A variable step solver was also chosen for the analysis. Fixed step solver is not suitable in this case as transitional processes take place and in order to ensure precision a short calculation step is necessary, which under the settled balance is not very effective. Variable step solver automatically selects a step, which is significantly better as this way we can reduce calculation period. For the initial calculations under the recommendations, an explicit ode45 solver, which uses Runge-Kutta and Dormand-Prince method [7] was selected, however for this type of task, it was not informative, i.e. did not converge. Then, an implicit ode15s solver which uses numerical differentiation formulas (NDFs) was selected.

Gravity, solver, task solution type, visual simulations and other solution related parameters are assessed with the machine environment in the block diagram.

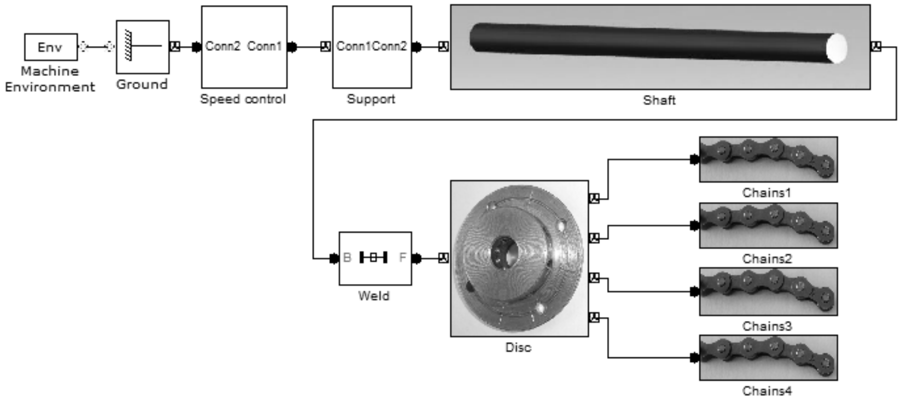

Fig. 2. Model in Simulink environment with SimMechanics block diagrams

The block elements of the model support are presented in Fig. 3. Support consists of a joint, allowing the shaft to move in plane perpendicular to its axis, springs and damper blocks at *x* and *y* directions as well as sensors, enabling observation of the shaft-to-support reaction forces and shaft displacements.

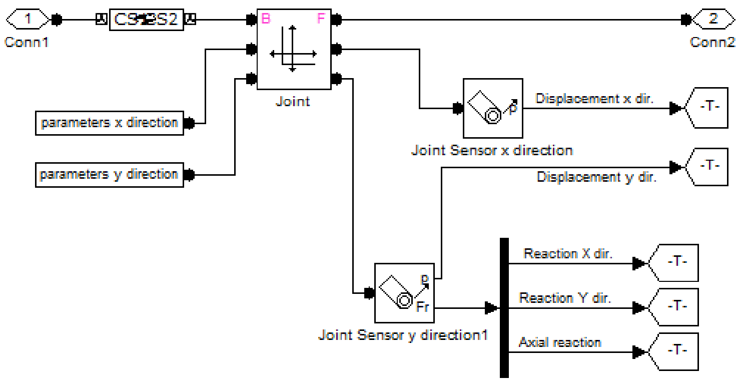

Fig. 3. Block elements of the support

Chain block elements are presented in Fig. 4. The chain is designed of five elements, which are joined by revolute joints allowing rotation towards one direction. Initial chain conditions, i.e. speed and angular position are evaluated in the elements block diagram of revolute joint, given in Fig. 5, damping as well as maximum and minimum chain rotation angles were evaluated there as well.

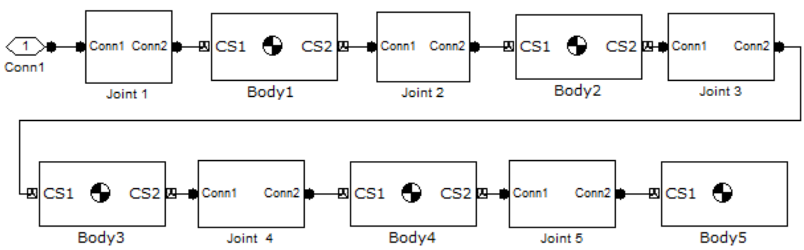

Fig. 4. Flexible chain block elements

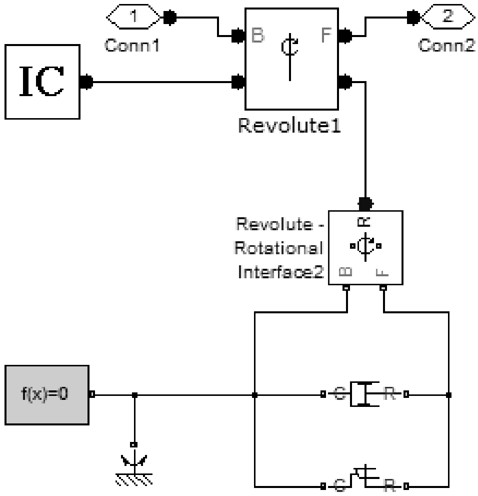

Fig. 5. Revolute joint elements

The parameters of the numerical model elements were selected based on the parameters of an actual mechanical system, given in Fig. 1. The parameters of a shaft, disk and chains used in the numerical model were simulated with the SolidWorks software package. Having designed the necessary elements following the measurements of an actual test rig, and upon entering a material, inertia moments and their mass values used in the numerical model were detected. The moment of inertia used in Simulink has been described in major axes as presented in Eq. (1):

$$I = \begin{bmatrix} I_{xx} & -I_{xy} & -I_{xz} \\ -I_{yx} & I_{yy} & -I_{yz} \\ -I_{zx} & -I_{zy} & I_{zz} \end{bmatrix} \tag{1}$$

Mass values were compared with masses of certain elements upon their weighing. An error did not exceed the value of 2 %. This allows us to receive precise calculations comparing with the experimental data.

The model was simplified - all bodies are completely rigid. Stiffness and damping were assessed in the joints. Aerodynamic forces, friction as well as gyroscopic moment were not included in the model.

Comparison of experimental and simulation results

During the experiment, the rotor system was accelerated from 0 rpm up to 6000 rpm. It took 38 sec. time interval for the rotor to run up. When the rotor reached 700 rpm, the chains fully stretched being effected by the centrifugal force and the mass centre of a rotor synchronized with the pivot. At the same time the mass inertia moment of a disk with chains was the highest. Chain positions of an actual test rig and a numerical model with the different rotation speed are given in Fig. 6.

The numerical model was set to the same rotor rotation mode as the test rig. Upon simulation with the numerical model it is possible to display visualization of a model in real simulation time, as given in Fig. 6b. Visualization is designed based on described inertia moments of the elements by transforming them into ellipsoids.

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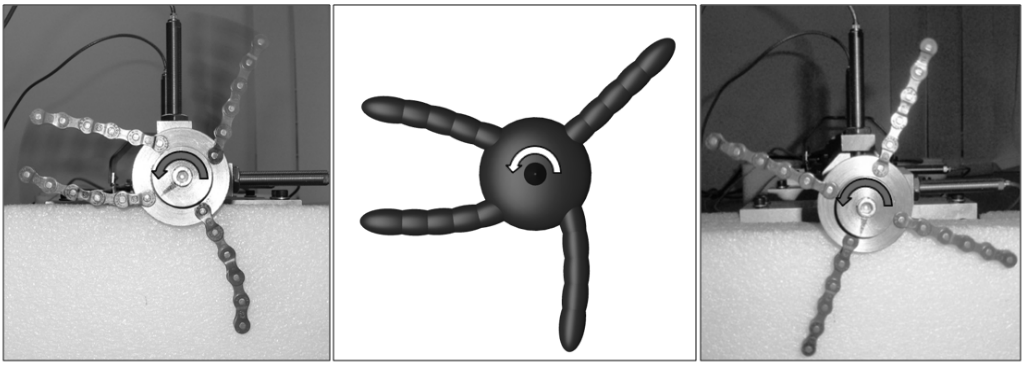
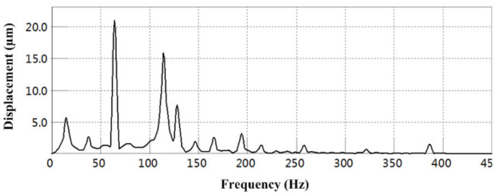


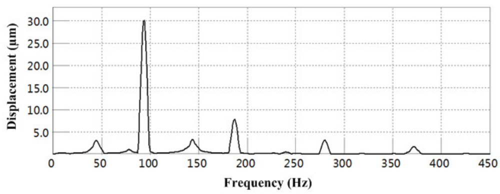
a) b) c)

Fig. 6. Motion of flexible chains under varying rotor speed of rotation:
250 rpm of actual test rig (a), numerical simulation (b), test rig at 700 rpm (c)

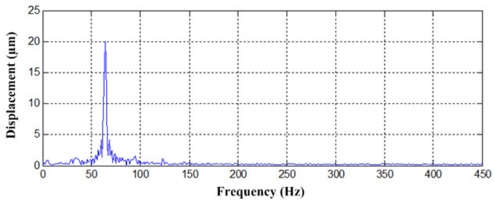
According to visualization, a full stretch of chains is achieved in 4 seconds after a launch of the rotor system. The simulation results were compared with the experimental results. Model conformity with an actual test rig is also demonstrated by the results of rotor vibration displacement spectra as shown in Fig. 7.



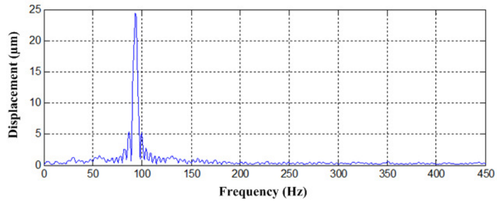
a) Experimental test rig data at 3840 rpm



b) Experimental test rig data at 5600 rpm



c) Numerical model simulation results at 3840 rpm



d) Numerical model simulation results at 5600 rpm

Fig. 7. Rotor amplitude-frequency characteristics of radial displacement

With reference to experimental results, given in Fig. 7a, and calculation results, presented in Fig. 7c, Y values of vibration displacement during the first resonance when rotation speed reaches 3840 rpm correspond with a precision of 95 % (experimental – 21 μm, calculated – 20 μm).

With reference to experimental results, given in Fig. 7b, and calculation results, presented in Fig. 7d, X values of vibration displacement during the second resonance when rotation speed reaches 5600 rpm correspond with a precision of 82 % (experimental – 30 μm, calculated – 24,5 μm). In the actual system it is difficult to approach such parameters to ideal ones, for this reason we receive higher order harmonics and greater values of displacement amplitude, given in Fig. 7(a, b).

Conclusions

1. The numerical model of variable inertia moment was developed and analyzed with the excitation force dynamically changing from 0 Hz up to 100 Hz in acceleration mode.
2. The amplitude-frequency vibration displacement characteristics indicated an amplitude disagreement, because some aspects were not included in a simplified model such as: rotor unbalance, shaft flexibility and aerodynamics forces as well as other factors.
3. The developed original model is beneficial in assessing changes of variable inertia moment in rotor systems. The model may be applied for identifying optimal design parameters of chain and hammer rotor mills, thereby allowing to ensure minimum loads that affect the bearings during operation.

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