11th International Conference on Vibration Problems
Z. Dimitrovová *et al.* (eds.)
Lisbon, Portugal, 9-12 September 2013

THEORETICAL AND EXPERIMENTAL INVESTIGATIONS OF OIL-FREE SUPPORT SYSTEMS TO PREDICT HIGH-SPEED ROTOR BEARING DYNAMICS

Zbigniew Kozanecki[1], Dorota Kozanecka[1], Jakub Łagodziński*[1], Eliza Tkacz[1]

[1]Institute of Turbomachinery, Lodz University of Technology, Poland
zbigniew.kozanecki@p.lodz.pl
dorota.kozanecka@p.lodz.pl
jakub.lagodzinski@p.lodz.p
eliza.tkacz@p.lodz.pl

Keywords: Oil-free support, High-speed turbomachinery, Gas bearings, Foil bearings.

Abstract. *In modern low-power generation systems, microturbines of the power output ranging between 1 and 20 kW are applied. The development of reliable bearing technology for high-speed small turbomachinery could be essential to these power-generating devices. In order to introduce this technology to common use, a selection of the optimal design from the viewpoint of machine reliability must be conducted. Therefore, one should analyze thoroughly the dynamics of the rotor-bearing-casing system in the whole operating range of the machine. The rotating system presented in the paper is supported in oil-free, airfoil bearings. Compliant surface foil gas bearings are a class of hydrodynamic bearings that use the ambient gas as their working fluid and, thus, require no dedicated lubrication systems, which makes their design much simpler.*

Correctly operating foil bearings are design solutions that have wide possibilities of applications, unavailable for rolling or oil bearings [1]. Nowadays in many scientific centers in the world, the main research on foil bearings is devoted to elimination of their basic disadvantages such as high start-up moment and wear of component surfaces. The second important goal is to predict and design properly the behavior of a complex support system consisting of the gas film and the compliant – damping foil structure. This can be achieved by building a trustworthy foil bearing dynamic model and evaluating it experimentally. The complexity of the theoretical model of such a bearing is deepened by the issue of the relative motion of both foils and the friction of the bump and cylindrical foils that takes place.

The paper presents the development and experimental verification of a theoretical numerical model of the foil bearing for analysis of its dynamic characteristics (a sum of properties: two elastic elements connected in a series and their relative motion, friction that is connected to this motion with respect to the elastic and cylindrical foils subjected to deformation) that will be the part of the investigations referring to numerical analyses oriented on developing a model of a high-speed rotor supported in bump-foil bearings.

1. INTRODUCTION

The micro power system which is used in a low-power source, with the power output ranging between 0.05 and 20 kW, is based on the Brayton or Rankine cycle and consists of a turbine (gas turbine) and a generator. The system requires high revolutions to generate sufficient power in the small-size turbine (microturbine). The term “microturbine” is often used for turbine systems characterized by the power output from a few watts to hundreds of kW.

In this field, we can distinguish:

- a power-MEMS turbine – a few millimeter-diameter microturbine rotating even up to one million rpm,
- a microturbine to produce electricity in the range from a few kW to a few hundreds of kW at the speed of approximately 10000÷100000 rpm.

The existing conventional oil-lubricated slide or ball bearings reveal performance limits at these revolutions, especially when a power or stability loss of the bearing is taken into account. More and more common applications of non-conventional materials in the machine design provide opportunities for considering an idea of applying the working medium used in a small high-speed turbomachine as a lubricating medium for its bearings very realistic. In machines in which a low-viscosity working gas or liquid is used, an application of bearings lubricated by that working medium makes it possible:

- to increase the total efficiency of the machine, which is attained by decreasing friction losses in the bearings and by eliminating the oil system and seals connected with it;
- to simplify the design of the shaft and to reduce its length;
- to maintain absolute purity of the working medium;
- to build a ”hermetic” machine without a rotating shaft end protruding outside the casing, to eliminate the mechanical gear and ”working medium - atmosphere” seals in an electric generator or motor integrated with the shaft.

The materials used in such bearings have to secure:

- conditions for short, non-destructive contact of the rotating bearing journal with the bush during machine start-ups and shut-downs;
- a low friction coefficient between the journal and the bush;
- corrosion resistance during the contact with the lubricating medium.

Therefore, while studying possible applications of non-conventional lubricating media, one should analyze thoroughly the dynamics of the ”rotor-bearing-casing” system within the whole range of machine operation.

The investigated concept of the micro-machine rotating system is related to the following assumptions:

- power output 2÷5 kW,
- rotational speed up to 50 000 rpm,
- oil-free technology for the bearing system design,
- working medium of the machine – air or dry vapor of the low boiling medium (in the case of ORC cycles),
- a machine shaft integrated with an electrical PM generator or a PM motor.

2. GAS BEARING TECHNOLOGY

The compressibility of gas is an important factor and it has to be included in the analysis of various forms of gas bearings – both aerodynamic and aerostatic (self-acting and externally pressurized). The advantages over liquid-lubricated bearings are well known:

- cleanliness – elimination of the contamination caused by typical lubricants,
- stability of the lubricant – no vaporization, solidification, decomposition under extreme temperatures,
- very low friction (low viscosity).

The main disadvantages of gas-lubricated bearings are recognized as resulting from low viscosity of the gas:

- reduced unit load carrying capacity, especially in aerodynamic (self-acting) bearings,
- closer control of manufacturing tolerances.

Gas bearings are to be met in a variety of applications from small laboratory devices to special high-speed devices. Their development is connected with the fact that they can be often used where the application of well-known traditional design solutions of bearings is troublesome (i.e., high-speed low- or high-temperature applications – cryogenic expanders or gas turbines). Actually, three kinds of gas bearing systems are developed simultaneously for the micro-turbomachine needs:

- aerostatic gas bearings,
- aerodynamic compliant surface (“bump foil”) bearings,
- aerodynamic “tilting pad” gas bearings.

The main problems connected to the reliability of practical applications of aerostatic (externally pressurized) and aerodynamic (self-acting – bump foil and tilting pad) gas bearings are specified in Table 1.

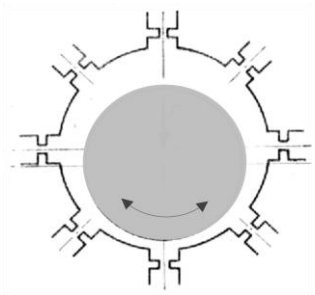
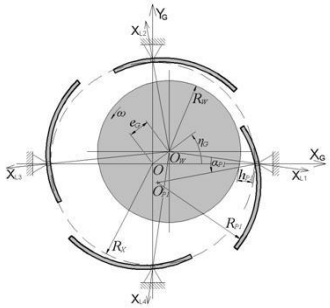
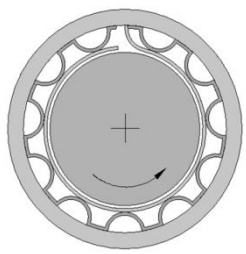
aerostatic circular bearing	aerodynamic “tilting pad” bearing	aerodynamic “bump foil” bearing
		
Reduced unit load carrying capacity		
Additional energy is needed to pressurize the bearing	Dry friction at low rotational speed – a limited number of start-up/shut-down cycles (wear)	
Close manufacturing tolerances	High drive moment for the start-up needed	
Low stability margin	Structural friction in the pivot influences the stability margin	High lateral displacements of the shaft due to the bump foil elasticity
Cleanliness of the lubricant	Special material requirements (high durability and a low friction factor) are imposed	

Table 1: Main problems connected to the reliability of gas bearing applications.

One of the basic problems related to a practical application of the above-mentioned high-speed machine concept equipped with gas or dry vapor lubricated bearings, is the machine operational reliability under various working conditions, which requires an adjustment of the machine design at the early stage of the investigations. The main factors of this adjustment connected to the oil-free technology specificity are as follows:

- possible diminution in thrust and lateral loads of the bearings by a correct design of the turbine flow structure,
- correct selection of the gas bearing type.

A methodology connected to our rotordynamic investigations consists in building up a prototype of the micro turbomachine equipped with a chosen kind of the gas bearing system and an appropriate measuring system, and experimental verification of the theoretical calculations under the conditions near to the real machine operation. In fig.1, a general design of the micro-turbogenerator prototype developed for the ORC cycle and supported in the aerostatic bearing is shown.

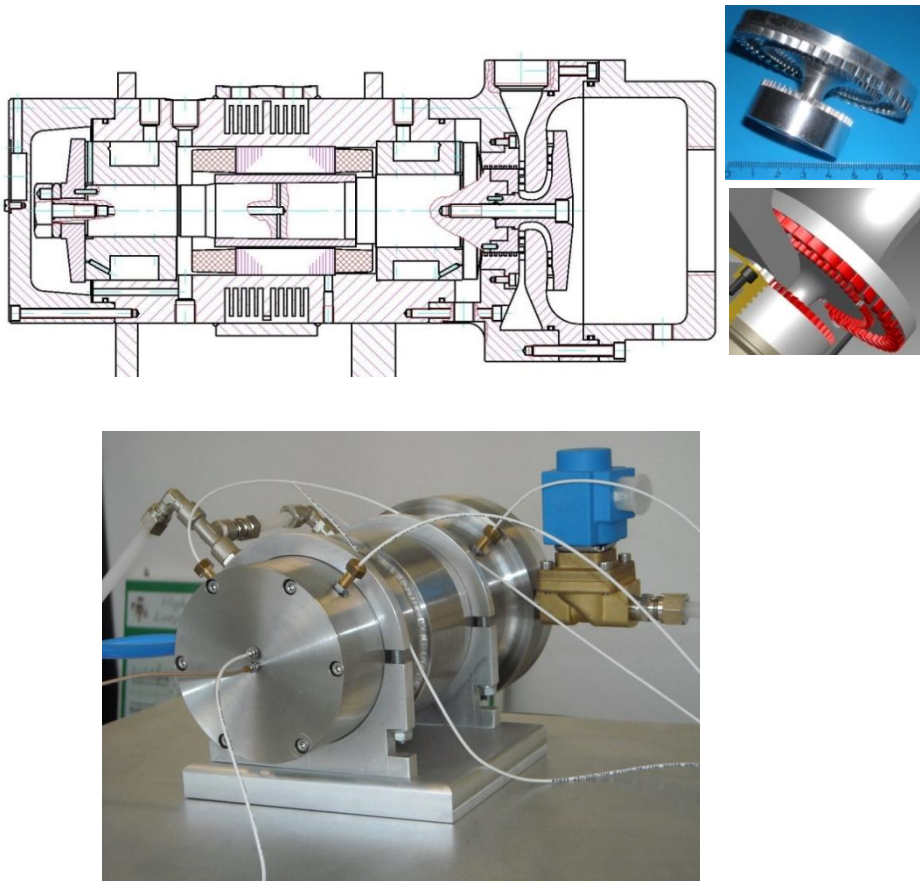


Figure 1: General design and a prototype view of the micro-turbogenerator equipped with an aerostatic gas bearing and a vibration measuring system.

The four-stage turbine concept (2 radial centripetal and 2 radial centrifugal stages) makes it possible to balance the large thrust load connected to the classical turbine operation.

In fig.2, a machine prototype equipped with an aerostatic gas bearing system is shown. The cascade plot obtained during rotordynamic tests confirms excellent stability of the

machine rotating system during the start-up and at the maximal turbine rotational speed amounting to 25 000 rpm.

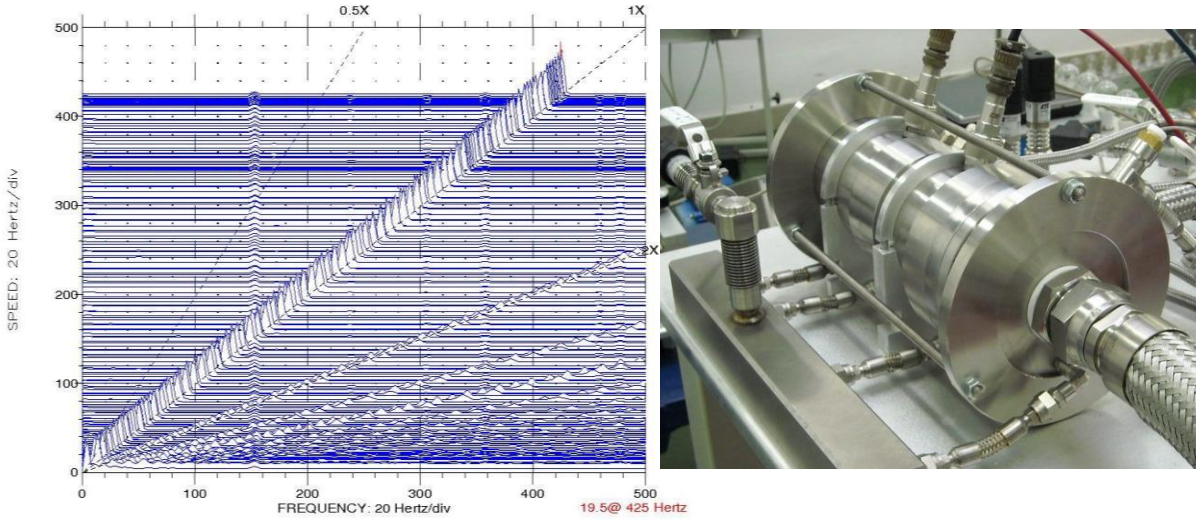


Figure 2: Oil-free ORC turbogenerator prototype during tests and a cascade plot illustrating the stability of the machine rotating system in the whole range of rotational speed.

At the maximal operational speed, the rotating system is still subcritical, and the critical speed for the considered system appears at about 35 000 rpm. It is worth noticing that the precessional mode is the so-called “rigid body mode” and the bearing stiffness and damping are decisive for the nature of the shaft dynamic response.

Despite of the good rotordynamic characteristics of the machine, the main problem connected to the aerostatic bearing application is additional energy needed to pressurize the supply pressure orifices spaced uniformly in two ranges of the bearing sleeve. From the viewpoint of energy consumption, one should analyze thoroughly the total mass flow of the incoming pressurized vapor through the aerostatic bearing system. The bearing mass flow reduces the total mass flow of the turbine and, consequently, decreases the efficiency of the machine. Consequently, from this point of view, aerodynamic compliant surface (“bump foil”) bearings lubricated by ambient gas are much more interesting.

2.1 Aerodynamic foil bearings

The design of the experimental foil bearing as well as the calculated aerodynamic pressure distribution are shown in fig.3. The uniqueness of foil bearing operation results from the fact that the *top foil* is clenched during the bearing operation on the rotating journal by means of the elastic *bump foil*. The aerodynamic film geometry of very low thickness, theoretically close to the cylindrical one, is generated by the viscosity effects. A great advantage of aerodynamic bearings is that they require no external pressurization system for the working fluid.

The important problem of the aerodynamic gas bearing application is connected to the start-up and the shut-down in contact with the shaft surface and thus:

- a high drive moment for the start-up (disputable from the viewpoint of the turbine mechanical characteristics) is needed,
- there is a limited number of start-up/shut-down cycles (wear).

Furthermore, the theoretical analysis of gas foil bearings is difficult due to an interaction between the gas film pressure and the complicated deflection of the top foil and the underlying bump strip support structure. Dynamic properties of aerodynamic gas bearings,

according to the linear theory, are usually represented by a set of eight coupled dynamic coefficients, linearized around the static equilibrium position of the bearing. It is necessary, however, to limit the scale of the excitation forces in order to fulfill the basic condition of small displacements around the equilibrium point. The motion of movable, but non-rotating parts of variable-geometry bearings is inevitably connected with friction. It is evident that the nature of elastically supported bush motion can dramatically affect dynamic characteristics of the bearing. Therefore, nonlinear modeling of dynamic properties, including design characteristics of the support and the generated friction forces, becomes indispensable in high-speed foil bearing applications.

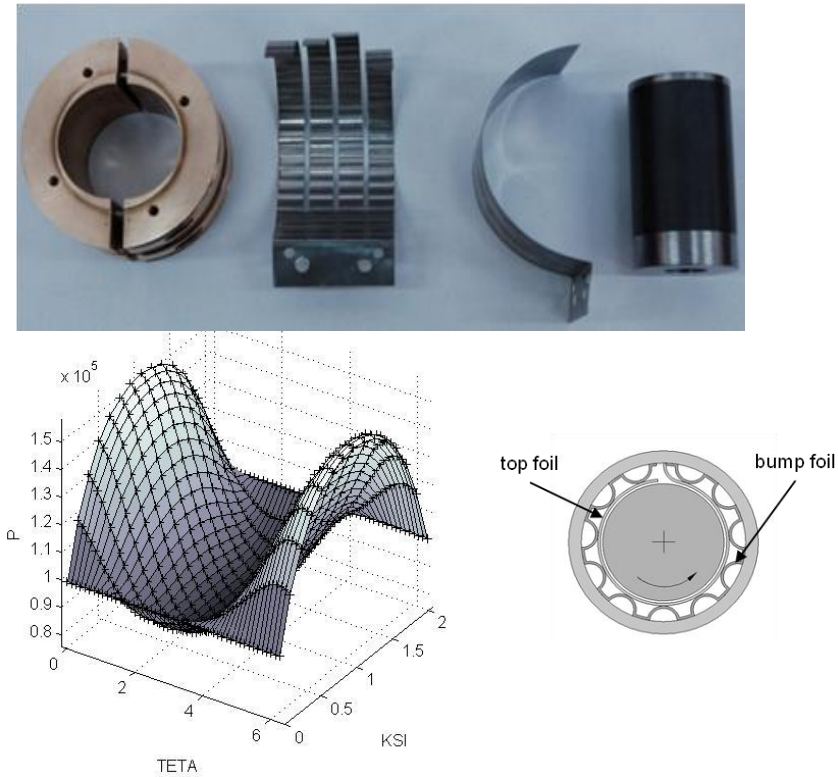


Figure 3: Aerodynamic foil bearing developed for the experimental needs.

The influence of the friction phenomena on characteristics of the aerodynamic bearing at high speed as well as during the start-up and the shut-down requires a careful selection of different materials for the bearing design followed by scrupulous durability tests [3]. The simulation results allow one to formulate the following hypotheses related to the bump foil bearing operation, namely:

- even under relatively high loads of the journal, the “aerodynamical” part of the bearing operates at very small eccentricities,
- high rigidity of the gas film follows from its inconsiderable thicknesses “enforced” by the initial clenching of the elastic bump foil. Therefore, the rigidity of the elastic foil decides in practice about the dynamic properties of the whole bearing system,
- knowledge of elastic and damping properties of the elastic foil will allow one to determine the total dynamic properties of the whole bearing system and consequently, to generate a theoretical model of the whole bearing that will be fully useful for engineering calculations related to the rotating system dynamics of the designed oil-free machine equipped with a bump foil gas bearing system.

2.2 Dynamic properties of the compliant structure of the bump foil bearing

At present, the investigations connected to the optimal design of the bump foil bearing are conducted in two main directions:

- to select the optimum pair of materials (a journal material and a foil coating material) that will ensure its sufficient durability under different machine operating conditions (start-up, nominal speed, shut-down),
- to find the optimum design of the bump foil thrust and radial bearings for the considered application and to determine reliable methodology of prediction of dynamic properties of the compliant surface bump foil bearing.

The results of the investigations related to the material selection are described in [3]. In fig. 4, a simplified physical model of the bump foil bearing is shown.

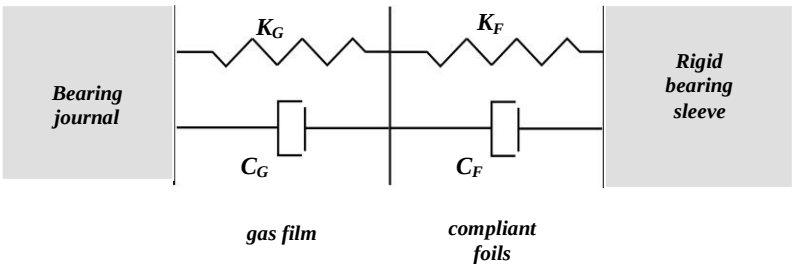


Figure 4: Simplified physical model of the bump foil bearing

Theoretically, static and dynamic characteristics of the foil bearing are the result of elastic combined properties of the two elements serially connected. One of them is a thin gas film of very small thickness and relatively high static and dynamic stiffness. The second elastic element is a pre-tensioned bump foil spring.

It should be noticed that depending on the bump foil pre-tension, a stiff gas film appears at a few to over a dozen krpm. Above this speed limit, a continuous gas film occurs, and the foil bearing operates properly when the rotating journal loses contact with the top foil.

The accepted physical model of the start-up of aerodynamic foil bearings allows one to formulate the following statements:

- for the journal rotational speed below n_{lim}, where n_{lim} denotes the rotational speed at which the continuous gas film appears, it can be assumed that the dynamic properties of the bearing depend on the stiffness of bump foil springs, because $K_g >> K_f$,
- at the journal rotational speed above n_{lim}, theoretically, the dynamic properties of the support system depend on the combined stiffness of the two elements serially connected.

The complexity of the analysis of the foil bearing theoretical model is caused by friction between the bump foil and the sleeve and between both foils. The friction comes from a relative motion of the foils and a relative motion of the bump foil and the sleeve. This physical phenomenon results in highly nonlinear dynamic properties of the foil bearing support system.

Some experimental attempts were made to identify these properties and a foil bearing test rig (fig. 5) was built for this purpose.

The test rig consisted of a fixed journal, frictionlessly supported sleeve and a modal shaker. The shaker excited the sleeve with a sinusoidal waveform force. It simulated real synchronous

excitation caused by rotor unbalance. During the experiment, the excitation force and the sleeve displacement were measured.

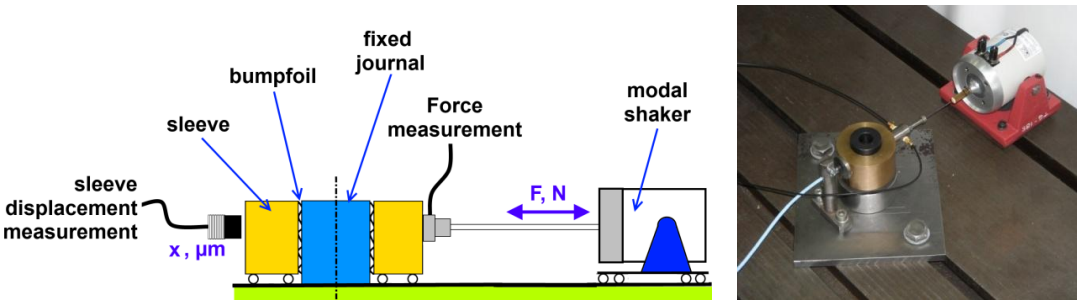


Fig. 5. Functional diagram and a photo of the test rig built for the experimental identification of dynamic properties of the Third+ generation foil bearing support structure.

A typical response of the shaking system is a hysteresis loop, presented in fig. 6. From this image, one can obtain the bump spring overall stiffness k and an area of the hysteresis loop W that represents the energy dissipated in a single motion cycle. The energy dissipation is related to the friction between the bearing foils and can be estimated from Eq.1:

$$W = \oint F dx = \pi \, \omega \, C_{eq} \, X^2 \qquad (1)$$

where: W – area of the hysteresis loop, X – vibration amplitude, ω – circular frequency and C_{eq} – equivalent damping coefficient.

Transforming Eq.(1) to obtain C_{eq}:

$$c_{eq} = \frac{W}{\pi \cdot \omega \cdot X^2} \qquad (2)$$

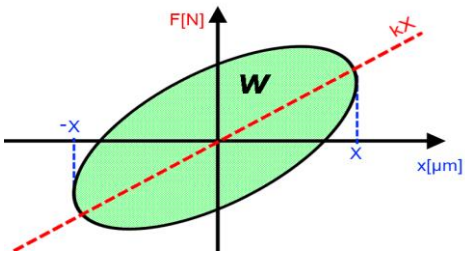


Fig. 6. Image response as a hysteresis loop associated with friction and elasticity of the compliant foil bearing assembly.

In the experiment, the bearing displacement amplitude was measured at the constant excitation amplitude of 10N p-p, for the excitation frequency ranging from 80 to 600Hz. For these test conditions, a few obtained hysteresis loops are presented in Fig 7.

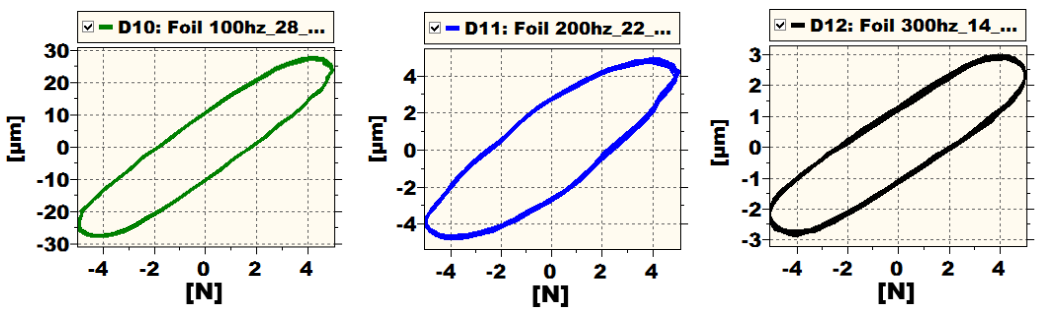


Fig. 7. Exemplary hysteresis loops obtained from the measurements at different excitation frequencies.

The obtained hysteresis loops allowed for identification of the experimental stiffness (K_F) and damping (C_F) values of the compliant foil bearings built for the test model of the rotating system (see Fig. 7).

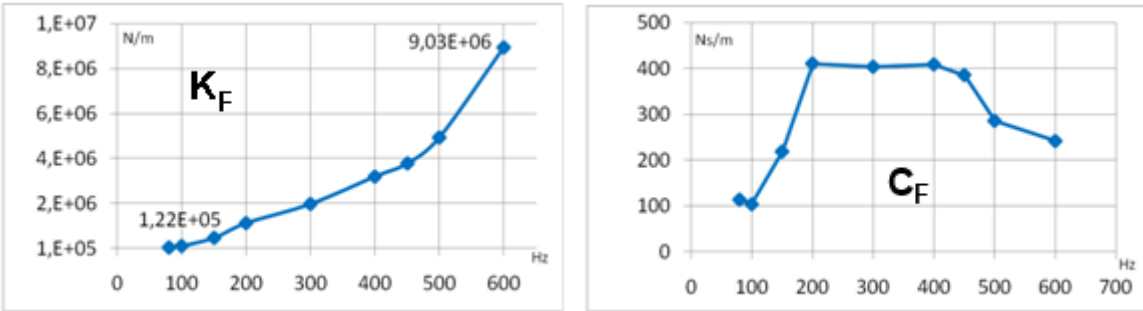


Fig. 7. Stiffness and equivalent damping coefficients of the compliant foil bearing support system vs. excitation frequency.

As can be observed (fig. 7), the bearing stiffness increases exponentially with the excitation frequency. The damping coefficient estimated from Eq.(2) develops fully at 200Hz and weakens noticeably above 400Hz. The damping characteristics of the foil bearing is highly nonlinear.

The obtained stiffness and damping coefficients were used in the model of the turbomachine rotor. In fig. 9, theoretical Bode plots of the investigated rotating system supported in the aerodynamic ”bump foil” bearing are shown. The zone indicated in grey is related to the friction phenomena in the gas film of the aerodynamic bump foil bearing caused by too low rotational speed of the journal. As can be seen, the critical speed is located in this area.

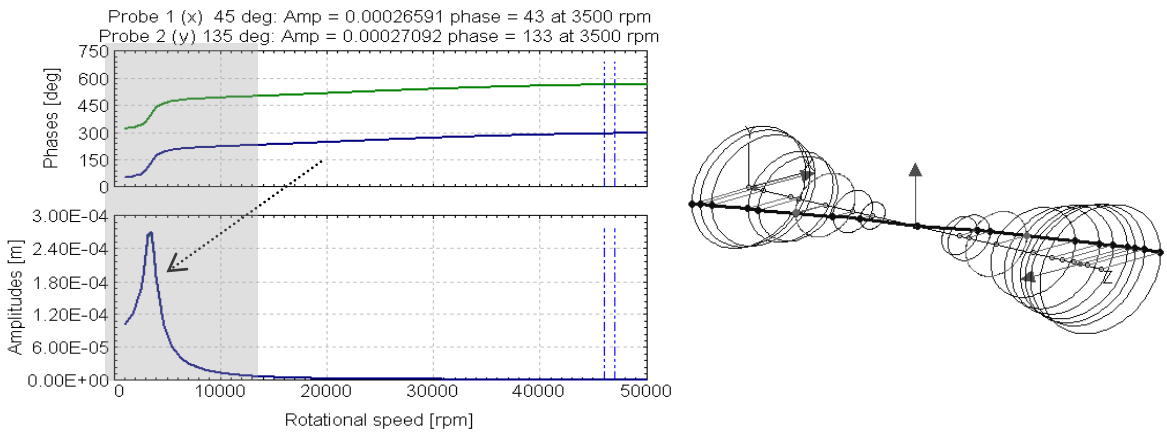


Fig. 9. Theoretical Bode plots of the investigated rotating system and a synchronous precessional response of the rotor at the nominal speed due to the imposed external excitations.

In fig.8, a micro-compressor supported in the compliant foil bearings is shown. This test rig was built for verification of the foil bearing models and design. Both journal and thrust bearings are of a foil-type. The cross-sectional CAD drawing shows the presence of a permanent magnet electric motor and several eddy current probes for rotor vibration measurement.

The preliminary experiments of the microcompressor run-up and operation at the nominal speed have confirmed the earlier assumptions so far. The rotor operation at the nominal speed is stable, with a minor level of vibrations.

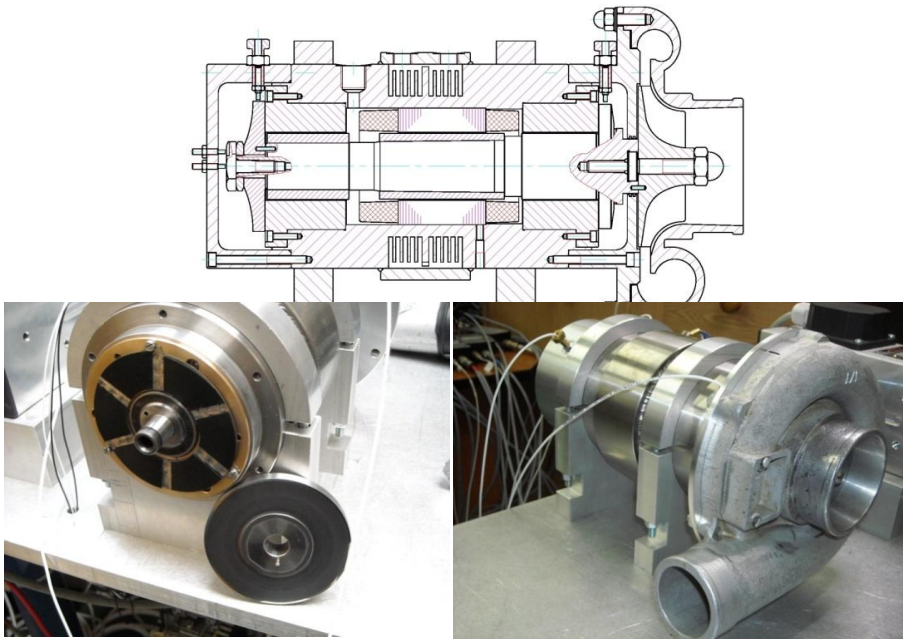


Fig 8. Microcompressor supported in the compliant foil bearings

3. CONCLUSIONS

- During the experimental tests, relevant information about the global stiffness and damping coefficients of the elastic foil bearing structure has been collected. This will enable the development of a reliable model of the compliant bearing support with nonlinear dynamic coefficients.
- From the operating point of view, the tested foil bearing has an optimal damping characteristics. In the useful range of rotational speed, the compliant structure is characterized by a significant damping value, due to the full-fledged phenomenon of dry friction between the top foil and the bump foil.
- A combination of the formerly developed numerical model of the gas film with the model of the compliant foil structure allows one to build a reliable dynamic model of the complete foil bearing.
- The preliminary experiments of micro-compressor run-up and operation at the nominal speed have confirmed the earlier assumptions so far.

4. REFERENCES

[1] Agrawal G.L. (1997) *Foil air/gas bearing technology - an overview.* ASME Int. Gas Turbine & Aeroengine Congress., Orlando (USA), 97-gt-347

[2] Andres L., Rubio D., Kim T. (2006) *Rotordynamic performance of a rotor supported on bump type foil bearings: experiments and predictions*. ASME Turbo Expo, Barcelona (Spain), GT2006-91238

[3] Kozanecki Z., Kozanecka D. (2010) *Theoretical and experimental investigations of oil-free support systems to improve the reliability of industrial turbomachinery*" The 8th IFTOMM International Conference on Rotor Dynamics. September 12-15, 2010/KIST, Seoul, Korea, pp. 686-692