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THE STATIC AND DYNAMIC PERFORMANCE ANALYSIS OF THE FOIL BEARING STRUCTURE

Grzegorz Zywica*, Jan Kicinski, Pawel Baginski

Institute of Fluid-Flow Machinery, Polish Academy of Sciences, Gdansk, Poland
{gzywica, kic, pbaginski}@imp.gda.pl

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Abstract. *Foil bearings are a variety of slide bearings in which an additional set of foils is applied between the journal and the bush, in order to improve the selected static and dynamic properties. Engineers and researchers from all over the world investigate bearings of this type since many years – both from numerical and experimental point of view. Due to the complexity of construction, the reliable computational models are all the time being searched for. This article discusses the important stages of forming the numerical model of the structural supporting layer in the foil bearing as well as the results of verification tests. The main goal of the conducted study was to assess the reliability of the elaborated numerical model considering static and dynamic properties. This model will be used to develop the numerical model of the whole foil bearing, which will take into account also the phenomena in the fluid-film layer and fluid-structure interactions. These models will be used together to describe the bearing system under operation.*

1   INTRODUCTION

Bearing systems based on foil bearings are frequently used in lightly loaded, high-speed, oil-free rotating machinery. With the use of such bearings a number of benefits are associated. Comparing to conventional aerostatic bearings, foil bearings do not require a stream of compressed air supply, which is advantageous due to the energy balance of the machine. Another advantage is the large dynamic stability of rotors supported on foil bearings, which is achieved mainly thanks to the very good damping properties of the set of foils (bump foil and top foil). Increased stability of the rotor also allows the machine to operate at higher speed. In addition, bearings of this type can operate at very high temperatures reaching several hundred degrees Celsius. Foil bearings are increasingly being used in machines such as: micro-turbines, turbochargers or turboexpanders [1,2].

The main difference in the construction of conventional slide bearings and foil bearings concerns additional element of thin, contoured set of foils having high compliance (Figure 1). This assembly is placed between the mating surfaces of the journal and bush. Set of foils allows changing the stiffness-damping properties of the support system, which has a beneficial effect on the dynamic properties of the rotor-bearing system.
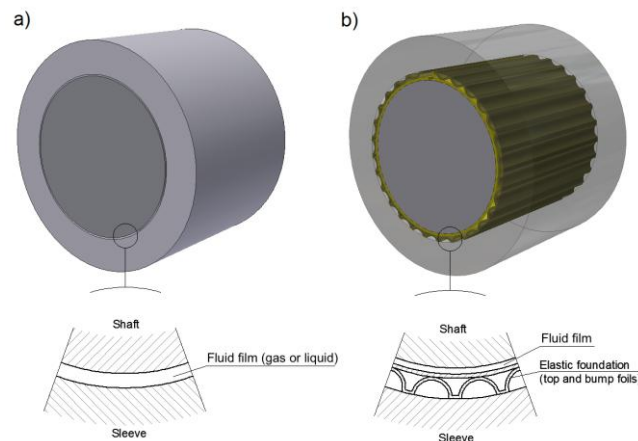

Figure 1: The difference between a slide bearing (hydrodynamic or aerodynamic) (a) and a foil bearing (b).

When discussing foil bearings we should also pay attention to their basic disadvantages. Foil bearings should be uniquely designed for each rotating system. Depending on the load, speed, lubricant, and temperature it is necessary to use a suitable materials and geometry of the compliant elements. Bearings of this kind are installed with a small interference, well adapted for proper operation at higher speed. This results in two consecutive issues. The starting and stopping of the machine causes direct contact between the top foil and the journal, which leads to rapid wearing. The second problem is the relatively high starting torque, required to run the machine. Despite these disadvantages, in some of the aforementioned machines, foil bearings are the most suitable ones.

In the worldwide literature more and more space is devoted to the research and modelling of foil bearings [3-6]. A couple of research and industry centers have been involved in examining the subject, mainly in the United States of America. The theoretical description of the operation of foil bearings must take into consideration many physical phenomena occurring in parallel and requires the search for new, reliable models. The main problem relates to an adequate description of the phenomena occurring in the geometrically complex, flexible set of foils, as well as the integration of structural and flow analysis (description of the phenomena occurring in the lubricant wedge). The appropriate description of the processes occurring in

these bearings should take into account: the phenomena of contact, deformations of geometrically complex set of foils, thermal phenomena, the flow of a lubricant in deformed bearing interspace and fluid-structure interactions [7,8].

For several years the research team at the Institute of Fluid-Flow Machinery PASci has been dealing with problematics of application and analysis of foil bearings. The results of previous studies have been included in several earlier publications [9-12]. The ongoing work is aimed at creating a complete model of the foil bearing, taking into account both structural and flow supporting layer. This article discusses the next phase of work on the foil bearing numerical model, which aims to study the properties of statically and dynamically loaded bearing structural layers. Since foil bearing elements during operation are imposed by variable static and dynamic forces, it should be included in a bearing model. In addition to the discussion of the numerical model and simulation studies, the article also presents the results of experimental verification of the model.

2   NUMERICAL MODEL OF THE FOIL BEARING STRUCTURE

The geometry of the foil bearing was prepared in a parametric program called Autodesk Inventor. The parameterization of the model allowed any change in the geometry and quick adaptation of dimensions to current needs. Due to the two-dimensional phenomenon of structural deformation of the foil bearing support layer and low computing performance of 3D models it was decided to use a two-dimensional model. This model allows a very precise analysis of the phenomena occurring in the foil bearings (first and second generation), in which there is no variation of the bump foil geometry in the axial direction. The two-dimensional model is able to reproduce the full geometry of the test bearing in a plane perpendicular to the axis of rotation and the width of the bearing is taken into account by an additional parameter.

The dimensions and parameters of the test bearing are given in Table 1. They were determined on the basis of [5,13], which allowed the comparison of results. Due to the need to preserve the uniqueness of the geometry description, not so much relevant differences occurred only in the case of the output, such as the bump pitch and bump length.

Dimension/Parameter	Value
Inner diameter	38,17 mm
Bearing length	38,10 mm
Nominal journal diameter	38,10 mm
Nominal radial clearance	0,035 mm
Number of bumps	25
Bump pitch	4,57 mm
Bump length	4,06 mm
Bump height	0,38 mm
Foil thickness (top and top foil)	0,1 mm
Young's modulus	$2,1 \cdot 10^{11}$ Pa
Poisson's ratio	0,29
Density of material	7860 kg/m^3

Table 1: Nominal dimensions, parameters and material specification of the foil bearing based on [5,13].

The bump foil of the tested bearing was composed of five identical sectors, equally spaced around the circumference of a bush. The total number of bumps was set to 25. The nominal clearance in the actual bearings is only approximate and, in practice, due to the pre-clamp of the bearing, it is usually determined experimentally on a special test rig [14]. However, it is

given in the case of such bearing models, since it is necessary to draw an unambiguous geometry.

FEA model of the foil bearing was developed using the Abaqus software. Due to the relatively high compliance of the foil set consisting of thin steel plates, bush and journal were treated as rigid bodies in the analysis. Therefore, these elements were simplified and replaced by the rings of a thickness that allows to apply a regular mesh. Developed FEM model is shown in Figure 2.
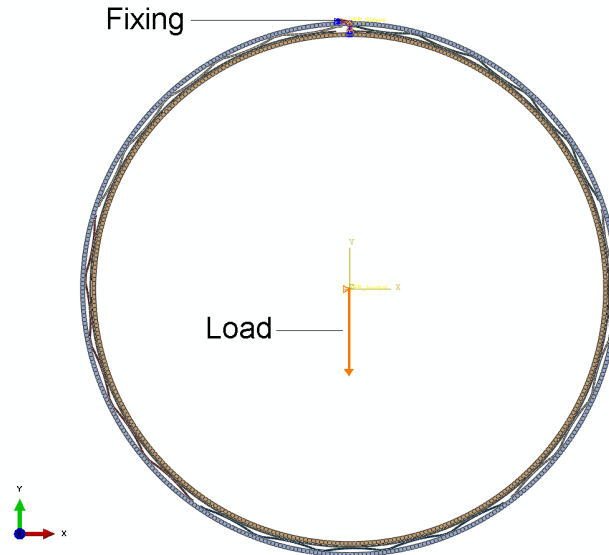

Figure 2: FEM model of the foil bearing structure.

The numerical model of the foil bearing structure presented above was composed of a total of 9778 degrees of freedom. The size and type of finite elements, FEM seeding of individual pieces of geometry, was selected by the numerical model optimization. Finally, for discretization CPE4RH type elements were chosen which have four nodes and first order shape functions. These elements allowed to obtain the most reliable results. The top foil and each of the five sectors of the bump foil were one-sidedly restrained. The displacements of the rest of the foil were limited by journal and bushing surfaces. Between these elements contact was modelled where friction coefficient was set to 0,1.

3   COMPUTATIONAL ANALYSIS

The basic criterion for assessing the usefulness of numerical models should be its reliability, which is easily determined by comparison with the real system. In the case of this model of foil bearing verification was carried out in two stages. The first stage of the verification concerned the characteristics of structural part of the bearing under static load. In the second stage the statically proven model was verified under a dynamic load operation.

3.1   Static analysis and verification

A simulation was planned in a manner enabling imitation of the conditions of the experiment described in [5]. During the simulation, the journal of the bearing under investigation was loaded with a static force with the maximum value of 224 N. The value of the force was increasing in a linear manner with time of analysis, and it reached its maximum after 1 second. One end of the top foil and one end of the bump foil were fixed to the bush surface. Dis-

placements of free fragments of foil were limited by the surfaces of journal and bush, between which a contact was modelled. The journal of the bearing could be displaced only in a vertical direction (according to the force direction) in the surface perpendicular to the axis of the journal.

In order to better identify the model of structural supporting layer of the foil bearing, investigation enabling the assessment of the model in terms of energy dissipation during the loading and unloading processes was conducted. In the model under study the dissipation of energy occurred as a result of the sliding friction between cooperating elements of the bearing. During the first second of the analysis the system was loaded linearly with a force of maximal value of 224N, and in the second analysis this force was decreasing linearly to 0. The comparison of the obtained characteristics is shown in figure 3. The investigation was done only on the bearing with the journal diameter equal to 38,10 mm. The research carried out for the greater number of cases have been discussed in [12].
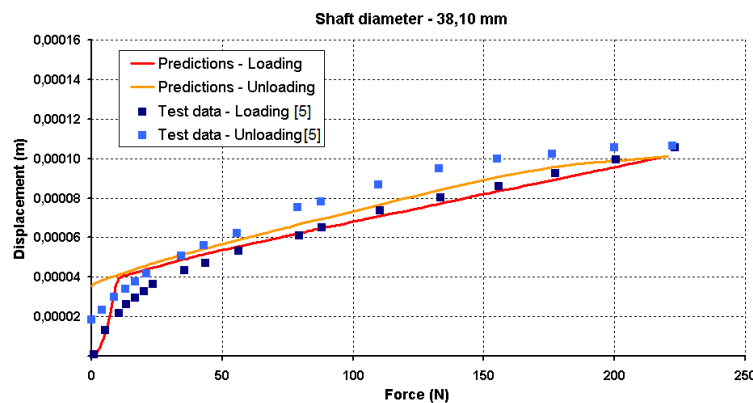

Figure 3: Foil bearing structure deflection versus static load (nominal clearance 0,035 mm).

The characteristics shown in figure 3 confirm the high compatibility of the developed model. A very high consistency of characteristics during the process of the system loading was obtained. Slightly worse matching of characteristics was obtained during the unloading of the system. The results of simulations showed, that during the decreasing load, the values of journal displacement corresponding to the same values of the force were higher than in case of the loading process. It was consistent with the results of the experiments and was connected with the friction phenomenon occurring during the journal displacements inside the bearing. As a result of the dissipation of part of the energy supplied to the system so-called hysteresis loop was created. The surface area of the hysteresis loop obtained as a result of experiments was higher than the one obtained as a result of calculations, which can be explained by the fact that some simplifications of the model were assumed, such as: omission of the internal friction or two-dimensional character. Due to the fact, that the mechanical system under investigation was very complex, and apart from the deformation of elements with complex geometry, the contact phenomena occurred as well - it can be stated that the obtained results are satisfactory.

3.2 Dynamic analysis and verification

In order to determine the dynamic characteristics, based on the discussed numerical model, simulation calculations had been planned to be adapted for experimental tests conducted by Kim et al. [13]. These tests consisted of forcing vibrations on the structural part of the foil bearing using the electromagnetic exciter. During the experiment, the rotor was not rotating,

and its function was done by the shaft rigidly mounted in the spindle lathe. The bearing sleeve was connected only with the tension member, through which harmonic excitation was transmitted from the exciter. The schematic view of the test rig is shown in Figure 4.

The conditions of the experiment were reproduced during the simulations by an appropriate set of boundary conditions and load. The essence of these researches relied on the determination of the relative displacements of the journal and bush caused by the external load, which had been limited by the presence of the set of foils and the inertia of bearing components. The bush was fixed and unable to move in any direction. This approach allows to obtain results that can be compared to the results of the mentioned work [13].
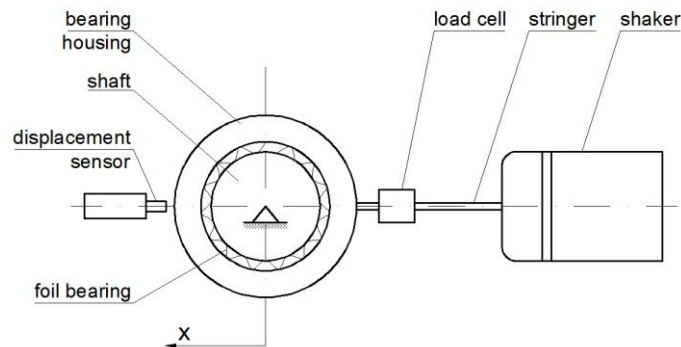

Figure 4: The schematic view of the test rig for dynamic excitation of the foil bearing.

The computational calculations focused on the dynamic response of the system in the horizontal direction (in the direction of the excitation force). The procedure "Dynamic, Implicit" was used along with the active option "Nonlinear geometry". The excitation frequency was 40 Hz, and the amplitude of the exciting force 22,5 N (according to the experiment). The force was changing by the sine function. The total time of the dynamic analysis was 0,2 s, while the harmonic force was active only for 0,1 s (from 0,1 s to 0,2 s). The initial analysis time of 0-0,1 s was necessary to determine the contact conditions. The time course of force acting on the bearing is shown in Figure 5.
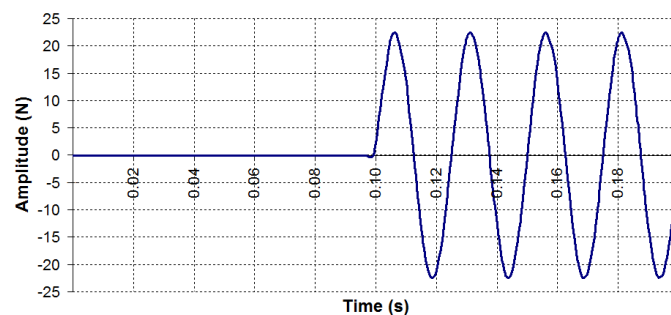

Figure 5: The time course of the excitation force.

The characteristics obtained initially in the form of displacements of the journal in the course of time and displacement versus the value of the load indicated the need to refine the numerical model so that its characteristics were similar to the real system. The modification of the model started from the identification of the most relevant differences between the model and the test rig. The first potential cause could be the neglecting the mass of bearing's elements, which for the static analyses conducted earlier did not matter. Additional concentrated mass was added in the journal model to its centre of gravity, but according to the experiment

it reflected on the weight of the bush, sensors, stringer and thermocouples with the wires. Using the description of the test rig and a series of tests the value of the additional mass was set at 0.5 kg. It was also necessary to introduce to the model additional stiffness-damping elements that were connected to the center of the journal. With the implementation of an additional stiffness-damping element, a visible impact of external components attached to a vibrating bush was noticed. The pictures of the test rig [13] indicated that the wires were of a fairly large diameter, and their stiffness in comparison to the compliant foil set could not be ignored. After many attempts optimal value of damping and stiffness was set at D = 1000 N·s/m and K = 400000 N/m. Journal displacement values depending on time and the load are shown in Figure 6 and 7. The figure 7 provides direct reliability assessment of the elaborated numerical model. A direct comparison of the characteristics shows that by tuning the model, very similar waveforms were obtained. Displacements obtained by computer calculations achieved higher values for the same load, but the shape of the loops obtained was very similar in the whole range of the bearing operation.
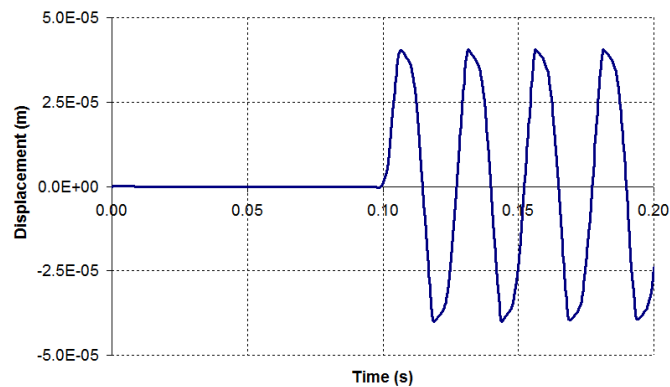

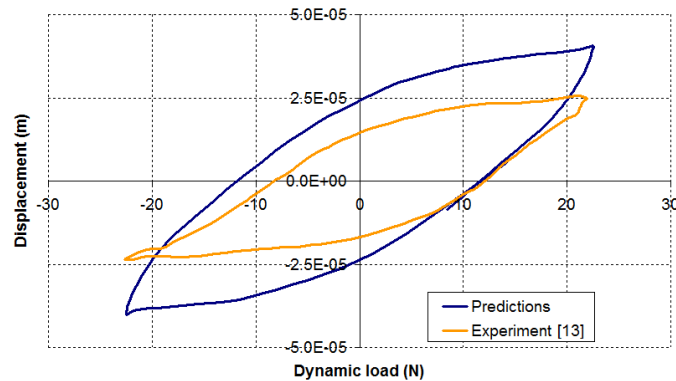Figure 6: Displacement of the shaft in the foil bearing at dynamic load.

Figure 7: Predicted and experimental displacement versus dynamic load for the journal of foil bearing.

Due to the typical low productive precision of foil bearings and a small experimental repeatability, no further attempts were carried out in order to have even better fine-tune of the model. The discrepancies were probably caused by the way the radial clearance was included in the bearing, which does not exist in the actual design due to the initial clamp. Attempts to better tune the model can be taken after an own experimental research, in conditions more similar to those operating. As it turned out, the experimentally obtained performance was affected by several factors also not associated with the bearing, or rather with the test method used.

4 MODELLING OF HYDRODYNAMIC DEFORMATIONS

In order to determine the characteristics of the foil bearing it is necessary to take into account the phenomena occurring in the flow and structural supporting layer. The deformations occurring in the flexible set of foils are very important and cannot be omitted in determining the static and dynamic characteristics. The stiffness of the set of foils is several times smaller than the stiffness of the bush in conventional slide bearings. The pressure that occurs in the bearing clearance during operation causes such large deformations, that depending on the speed and load, a significant change in the geometry of the clearance is observed. In order to properly analyze the foil bearings, the evaluation of the bearing structure deformation is required and taking it into account in the flow calculation. The foil bearing model must be capable of exchanging data between flow and structural modules (Figure 8), so that the calculations take into account the interaction between the fluid and the structure. The way and frequency of exchanging the appropriate data depend primarily on the type of analysis and selected precision level of the calculations.
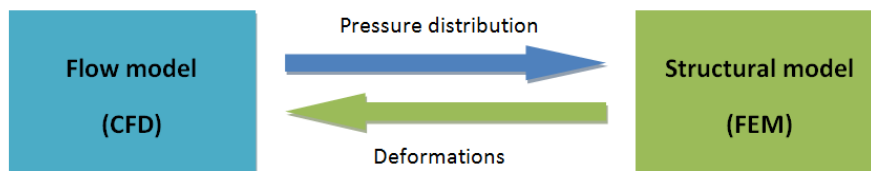

Figure 8: General scheme for data exchange in the numerical analysis of foil bearings.

The most important result obtained from the flow module is a pressure distribution in the bearing interspace. This distribution is determined based on the Reynolds equation, taking into account the viscosity of the lubricant. In the case of gas bearings the Reynolds equation for compressible fluids is used. Designated pressure distribution is then passed to the structural module, in which it is treated as a load applied to the inner surface of a top foil.
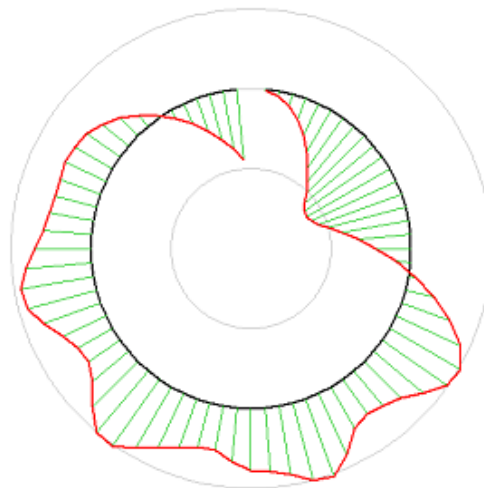

Figure 9: The example of top foil deformations presented in relation to the bearing clearance.

To analyze the foil bearing structure the Abaqus software was used, characterizing by a high potential for calculating the nonlinear systems and contact phenomena. The pressure

distribution obtained from the flow calculation was converted into a set of forces so that they could be applied at the nodes of the top foil FEM model, which formed the inner surface of the bearing.

Based on determined strains of flexible set of foils we can specify a new shape of the bearing interspace, which can be used to re-designate the pressure distribution and to determine the new position the bearing's journal. The shape of the bearing interspace thus depends on the pressure distribution and the local stiffness of the set of foils. The figure 9 shows an example of displacements obtained after applying the forces corresponding to the service load to the bearing structure. The process of determining the pressure distribution and the deformed shape of the bearing interspace must be carried out iteratively until a target level of convergence, at which a stable solution is registered. These foil bearing characteristics can be used to evaluate the work and static and dynamic analysis of the rotor.

5   CONCLUSIONS

- The article discusses the results of the static and dynamic performance analysis of the numerical model of the structural supporting layer in the foil bearing.

- The numerical model of the foil bearing structure was developed using the Abaqus software. The pressure distribution in the bearing interspace was determined using MESWIR system (a series of computer programs) that had been developed in the IFFM PASci in Gdansk.

- The results of the investigation confirmed the validity of assumptions made while developing the model. The developed model can be used as a very useful research tool for investigating the influence of selected parameters in the static and dynamic characteristics of the foil bearing structure.

- The article also presents the method of evaluating the deformations of flexible set of foils caused by the hydrodynamic pressure of lubricant.

- In the next stage of our research, the model of the foil bearing structure will be fully integrated with the fluid-flow model. The concept of a complete model of bearing will be based on the consideration of the fluid-structure interactions and will allow the analysis of the rotor supported on foil bearings.

ACKNOWLEDGMENTS

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