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DYNAMIC STATE ASSESSMENT OF THE HIGH-SPEED ROTOR BASED ON A STRUCTURAL-FLOW MODEL OF A FOIL BEARING

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Abstract

In the recent years, foil bearings have increasingly been used over a wide variety of different high-speed fluid-flow machinery. Since modern bearings of this type are the complex result of an assembly of several interdependent components, both their theoretical description and their modelling is very problematic. This article provides an analysis of the rotor supported by foil bearings conducted on the basis of an in-house developed numerical model. The model takes into account both structural and flow-related parameters, and makes it possible to determine the characteristics of a foil bearing under various operating conditions. The developed model allows analysing the dynamic performance of different foil bearings, taking into account the lubricating medium's properties as well as the geometry and characteristics of structural materials (top foil and bump foil). It gives an opportunity to shape the bearing freely so as to optimise its structure. An example of such an optimisation, which aims to minimise vibration of the rotor, was presented in the article.

Keywords: rotor dynamics, foil bearing, modelling of bearing

OCENA WŁAŚCIWOŚCI DYNAMICZNYCH WYSOKOOBROTOWEGO WIRNIKA W OPARCIU O STRUKTURALNO-PRZEPŁYWOWY MODEL ŁOŻYSKA FOLIOWEGO

Streszczenie

Łożyska foliowe są w ostatnich latach coraz częściej stosowane w różnego rodzaju wysokoobrotowych maszynach wirnikowych. Ponieważ współczesne łożyska tego typu stanowią wynik połączenia kilku współzależnych części składowych, zarówno ich opis teoretyczny jak i ich modelowanie nastręcza wielu trudności. W artykule przedstawiono przykład analizy wirnika z łożyskami foliowymi, wykonanej w oparciu o własny model numeryczny. W modelu tym uwzględniono zarówno parametry przepływowe jak i strukturalne, co pozwoliło na wyznaczenie charakterystyk układu w zależności od warunków pracy. Opracowany model pozwala na analizę właściwości różnego typu łożysk foliowych z uwzględnieniem właściwości czynnika smarnego oraz geometrii i charakterystyk materiałów konstrukcyjnych (folii ślizgowej i nośnej). Daje to możliwość dowolnego kształtowania właściwości łożyska oraz optymalizacji. Przykład takiej optymalizacji, której celem była minimalizacja drgań wirnika, został przedstawiony w artykule.

Słowa kluczowe: dynamika wirnika, łożysko foliowe, modelowanie łożyska

1. INTRODUCTION

Developing high speed fluid flow machinery requires innovative bearing systems. Foil bearings are ideally suited for such applications as they allow stable and high speed operation of rotors at elevated temperatures [1, 3-5]. They do not require an external lubrication system for proper functioning, even under such conditions. Excellent dynamic properties of foil bearings are achieved by using additional elastic damping elements, which are usually made of thin metal foils in which a properly modified surface layer is of particular significance [7, 15, 16]. Such bearings have many advantages that make them a preferred option for fluid flow machines such as gas and vapour microturbines, compressors or expanders [1, 4]. Because foil bearings do not need an oil lubricating system, by using them we are able to design oil free fluid flow machinery [6, 10, 14]. In their typical applications foil bearings are used to damp vibration, even at high rotational speeds. This results from the fact that the vibration damping element in such bearings is a specially shaped set of foils which, during operation, interact with each other and with the internal surface of the bush [13, 17]. Therefore, it is very important to select the appropriate shape and thickness of the top and bump foils [8, 11, 12].

Obviously, foil bearings also have some disadvantages which render them unsuitable for some types of rotating machinery. As foil bearings are still a long way from widespread use, their availability is limited [6]. They are usually manufactured upon request and have to be adapted to actual requirements (e.g. load capacity, price, etc.). Such bearings are sometimes the origin of operational problems. Chief among these problems are the wear of contact surfaces, high starting torque and significant journal displacements occurring during speed and load changes. Many engineers and scientists all over the world constantly work to eliminate these problems, and new generations of foil bearings are developed [1, 7]. In general, significant attention is focused on the optimisation of bearing designs and tribological issues, including the selection of suitable constructional and functional materials. Sliding layers in foil bearings may be made of different metals including metal-ceramic composites and plastics [5, 15, 16].

A foil bearing is a mechanical system the modelling of which is extremely difficult. A model of such a bearing has to take into consideration several physical phenomena such as non linear deformations of thin foils having a complex geometry, friction and wear processes on the contact surfaces, heat exchange, thermal deformations, flow related phenomena taking place within the lubricating gap and fluid structure interactions [17]. Therefore, in this case, simple computational models have poor reliability because they do not take into account all the relevant phenomena. In practice, this means that each manufactured foil bearing goes through a series of tests before application. This is done on specially designed test rigs [2, 5, 14]. An effective implementation of foil bearings requires great experience and long lasting tests during which the bearings operate under various conditions. Therefore, it is worthwhile to develop advanced methods for modelling rotors supported on foil bearings, which facilitate and accelerate the implementation of such innovative bearing systems.

The next part of the article discusses the simulation research on a high speed rotor supported by foil bearings. Computations were conducted using advanced in house developed numerical models. The conducted analysis also covers the optimisation of the bearings' structure, aimed at minimising the vibration level of the rotor in a defined range of the rotational speed.

2. MODEL OF THE ROTATING MACHINE

2.1. Object of investigation

The object of this study is a rotating system mounted on the test rig (Fig. 1) at the laboratory of the Faculty of Technical Sciences UWM. In its basic configuration, the test rig has oil lubricated hydrodynamic bearings. Due to ongoing research projects, there was a need to modernise the test rig. The modernisation consisted in applying a new bearing system. It was decided to develop new foil bearings by means of advanced numerical models. This approach allowed to carefully select the bearing components' geometry and to optimise it as well.

Fig. 1. Test rig with a flexible rotor and two slide bearings.

The test rig was constructed for the purposes of scientific research on small diameter high speed rotors. It is equipped with a drive system that provides rotational speeds of up to 24,000 rpm. The steel shaft and the journals of the bearings have a diameter of 25 mm, and the shaft length is 1,010 mm. The disk, which has a diameter of 200 mm and a width of 20 mm, is equidistant from each bearing support. The total mass of the rotor is approx. 10 kg. The rotor is connected to the drive system using a flexible coupling. The bearing supports were fixed to a thick steel plate that rests on active pneumatic vibro isolators.

2.2. Numerical model of the rotor

The numerical model of the rotor was prepared using programs that are part of the MESWIR system [9], developed at the Institute of Fluid Flow Machinery. During the modelling of the rotor, both geometrical and material properties of the shaft as well as of the disk mounted on it were taken into account. The finite element (FE) model comprised 40 Timoshenko type elements in total and 6 degrees of freedom at each node (Fig. 2).

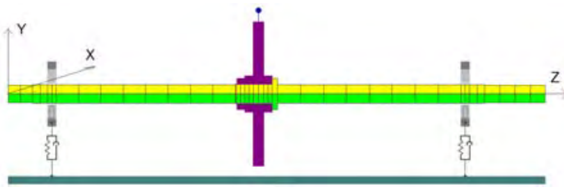

Fig. 2. The numerical model of the rotor.

The rotor model takes into account the disk unbalance with a maximum permissible mass in accordance with ISO 5406 standard. On the supporting points of the shaft, bearing supports with foil bearings were modelled. Their properties were determined on the basis of a structural and flow model discussed in section 2.3 of this article.

2.3. Numerical model of the foil bearing

It was originally considered that the computational analysis of the laboratory rotor would be carried out both for aerodynamic and hydrodynamic foil bearings. Considering a large mass of the rotor and small diameter of the journals it turned out that despite the high rotational speed, unfortunately, the conditions were not appropriate for the application of gas foil bearings. The load capacity of such bearings within the given range of rotational speeds was too low. Therefore, the authors of the article resigned from using gas foil bearings and all computations were conducted with foil bearings lubricated with a liquid lubricant. Since the team from the Institute of Fluid Flow Machinery already had experience in designing microturbines powered by the low boiling medium known under a trade name HFE-7100, it was decided to apply it as a bearing lubricant.

The developed model of a foil bearing took into account both the flow and structural supporting layer. Both models, a structural one and a flow one, had successfully passed the experimental verification process [10, 17] and were used for designing and analysing rotors supported by foil bearings but only the ones that were lubricated with gas [2].

In order to decide on the most appropriate bearing system, our simulation research programme consisted in analysing the same rotor with five different bearing variants. The variants differed substantially, due to differences in geometry and material properties. The basic parameters of all the bearing variants were shown in Table 1. All bearings had the same journal and bush diameters, respectively 25 mm and 26 mm. They also had the same nominal radial clearance and width of 0.025 mm and 20 mm, respectively. The coefficient of friction between bearing components was 0.2.

Table 1. Properties of test foil bearings.

Properties	Foil bearing variants				
	1	2	3	4	5
Generation	I	I	II	II	I
Foil thickness [mm]	0.1	0.1	0.1	0.1	0.15
Number of bump foils	1	1	5	5	1
Number of bumps	35	35	30	30	35
Top foil material	steel	steel	steel	steel	steel
Bump foil material	steel	brass	steel	brass	steel

The FEM model of the bearing (variant no. 1), with a visible mesh, is shown in Fig. 3. This is the generation I foil bearing with foils made of alloy steel ($E = 2.1 \cdot 10^{11}$ Pa; $\nu = 0.3$; $\rho = 7860$ kg/m^3). In the variant no. 2, the bump foil is made of brass. The third variant denotes the generation II bearing, which has five bump foils. The geometry of the bearing in the variant no. 4 is the same as in the variant no. 3, the only difference being that the bump foils are made of different materials. In the last variant (no. 5), the top and bump foils are made of the material that has a greater thickness. Changes in foil thickness aimed to increase the stiffness of the structural supporting layer. Since the bearing's geometry in the variants no. 2 and no. 5 is almost the same as in the variant no. 1, the next figure (Fig. 4) presents the FEM model of the bearing corresponding to the variant no. 3.

The model in question makes it possible to determine characteristics of a structural supporting layer of foil bearings subjected to static and dynamic loads [17]. It has a complex non linear geometry and takes into consideration contact phenomena that can occur between: a journal, top and bump foils and a bush. The characteristics of a structural supporting layer and its deformations were included in the analysis of a flow supporting layer. This approach allows identifying characteristics of complete bearings [10], which is the necessary information for effective rotor analysis.

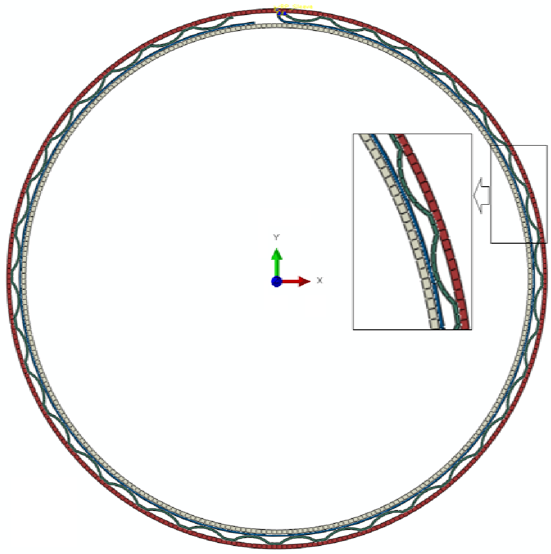

Fig. 3. FEM model of the foil bearing structure with one bump foil (variant no. 1).

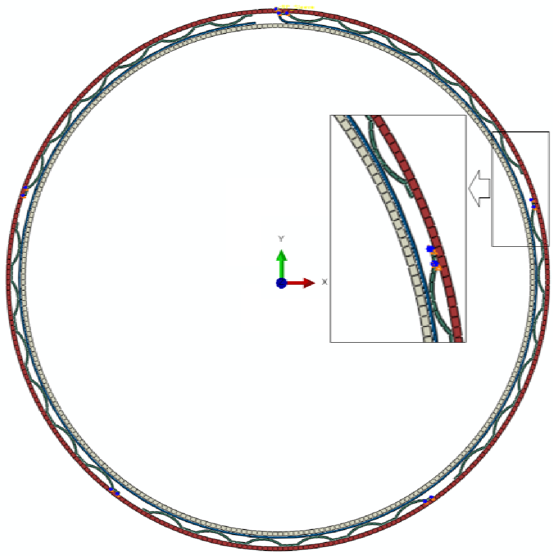

Fig. 4. FEM model of the foil bearing structure with five bump foils (variant no. 3).

3. RESULTS OF THE ANALYSIS

The numerical analysis have been organised in two stages starting with the determination of stiffness of all bearing variants and thereafter a dynamical analysis of the rotor.

3.1. Properties of the foil bearing structure

In the framework of the analysis of the structural supporting layer of foil bearings, their stiffness coefficients were identified using a method presented in the previous publication [17]. The FEM model was applied that has been already discussed in more detail above (see chapter 2.3). The simulation consisted in applying an excitation force acting vertically downwards to the bearing journal and determining the displacements related to it. The load force applied to the journal increased linearly, taking the values within the range from 0 to 100 N. The displacement of the journal was registered at the middle point of the journal. Due to an initial clearance of the bearing, large journal displacements could have been observed at the beginning of the loading period since the force value was very low. This time period is clearly visible in Fig. 5. Further displacement of the journal depended solely on the stiffness of a set of foils. In order to minimise the impact of the initial clearance on computation results, the stiffness was determined within the range starting from the point where the journal and the top foil meet to the load of 50 N. Such a load corresponds to half the value of rotor's mass, i.e. a static load acting on a single bearing during a stable operation of the rotor.

Based on the authors' experiences in numerical analysis and experimental research, the stiffness of the foil bearing structure in the horizontal direction (which was perpendicular to the direction of the acting load) was set to half the value of stiffness in the vertical direction. In the context of earlier research on foil bearings of a similar structure, the damping coefficient of a set of foils was 1000 N·s/m (for two directions). The plot showing the displacements of the journal (occurring in five different bearing constructions) is presented in Fig. 5. The relevant stiffness values are listed in Tab. 2.

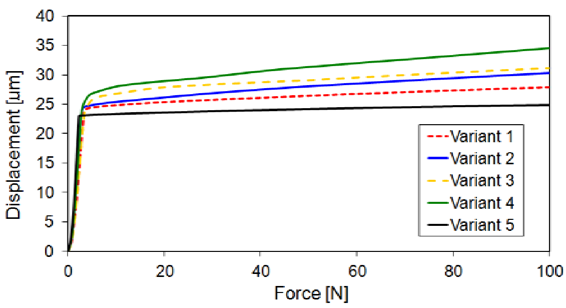

Fig. 5. Structural stiffness of a supporting layer for five different foil bearings.

Table 2. Stiffness of the foil bearings' structure.

Variant number	Stiffness [N/m]	
	vertical (X)	horizontal (Y)
Variant no. 1	$21.4 \cdot 10^6$	$10.7 \cdot 10^6$
Variant no. 2	$13.0 \cdot 10^6$	$6.5 \cdot 10^6$
Variant no. 3	$10.4 \cdot 10^6$	$5.2 \cdot 10^6$
Variant no. 4	$7.7 \cdot 10^6$	$3.9 \cdot 10^6$
Variant no. 5	$42.3 \cdot 10^6$	$21.2 \cdot 10^6$

On the basis of the numerical results obtained, the conclusion can be made that changes in the foils' geometry and thickness as well as in the bump foil's material allowed for the creation of bearing constructions with a different stiffness. By altering the stiffness of a rotor's support it is possible to change the performance of the entire rotating system, as shown in the next chapter of this article.

3.2. Rotor dynamics analysis

The numerical model of the rotating system takes into account the properties of the rotor and of the complete foil bearings by which it was supported, including structural and flow supporting layer. The fluid structure interactions manifested themselves in such a way that changes in pressure of the lubricant resulted in displacements of a flexible top foil and this, in turn, resulted in subsequent pressure changes [10, 17]. The foil bearing computations were performed iteratively until the desired accuracy was achieved.

The stiffness and damping coefficients determined for a complete foil bearing were used during a kinetostatic analysis and thereafter a dynamic analysis of the rotor. By conducting a forced vibration analysis it was possible to determine vibration levels at different rotational speeds. Computation results obtained for all bearing variants were presented in Fig. 6-15. Due to large disparities in vibration levels of the bearing journals and of the disk, two plots (figures) were presented for each computational variant one showing vibrations of the disk & bearing journals and one showing only vibrations of the bearing journals. Large vibrations of the disk resulted from the fact that the shaft was flexible.

The forced vibration plots, obtained for the tested rotating system, are quite typical and indicative of resonance areas. The maximal peak to peak vibration amplitude within those areas is in the range from 0.14 to 0.15 mm, depending on the computational variant. In variant no. 1, the resonant speed occurs at 2,350 rpm and vibration amplitude of the disk is 0.146 mm (Fig. 6). The vibration amplitudes of the bearing journals were over ten times lower than those of the disk, namely 0.011 mm (Fig. 7). For higher speeds (outside the resonance area), a linear increase in vibration amplitude can be observed. In variant no. 1, the computation results were obtained within the whole range of rotational speeds except the speed of 24,000 rpm.

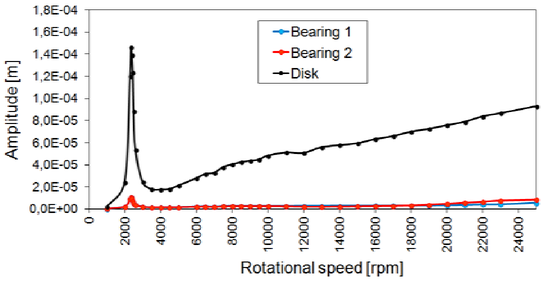

Fig. 6. Vibration amplitudes of the bearing journals and of the disk (variant no. 1).

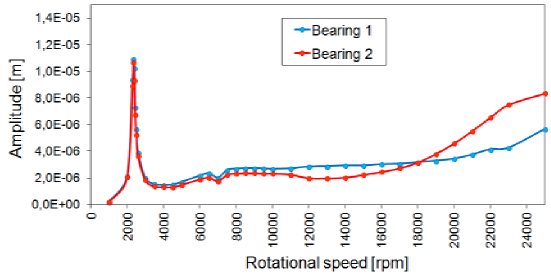

Fig. 7. Vibration amplitudes of the bearing journals (variant no. 1).

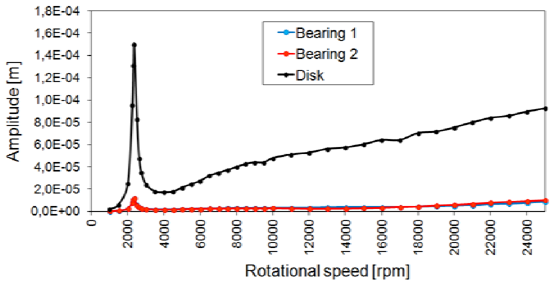

Fig. 8. Vibration amplitudes of the bearing journals and of the disk (variant no. 2).

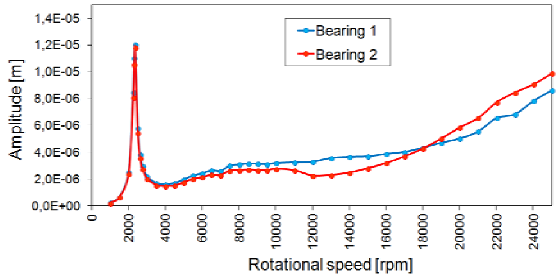

Fig. 9. Vibration amplitudes of the bearing journals (variant no. 2).

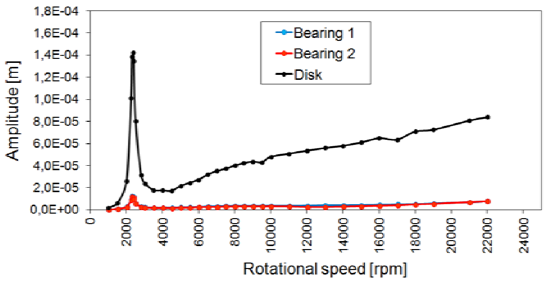

Fig. 10. Vibration amplitudes of the bearing journals and of the disk (variant no. 3).

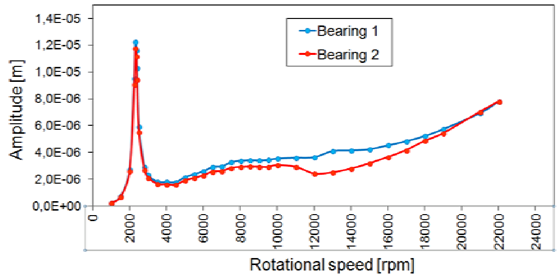

Fig. 11. Vibration amplitudes of the bearing journals (variant no. 3).

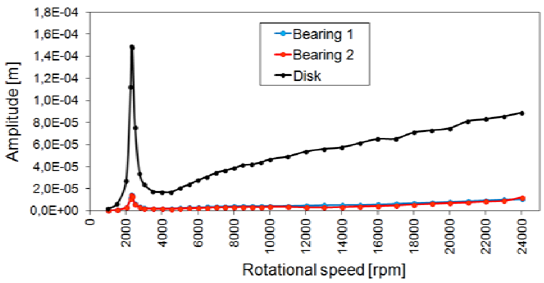

Fig. 12. Vibration amplitudes of the bearing journals and of the disk (variant no. 4).

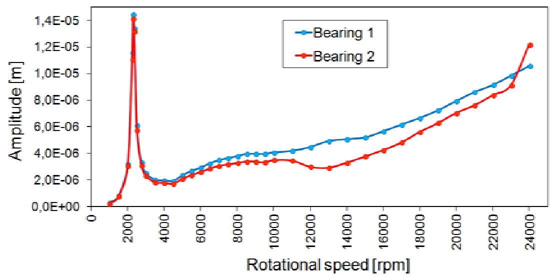

Fig. 13. Vibration amplitudes of the bearing journals (variant no. 4).

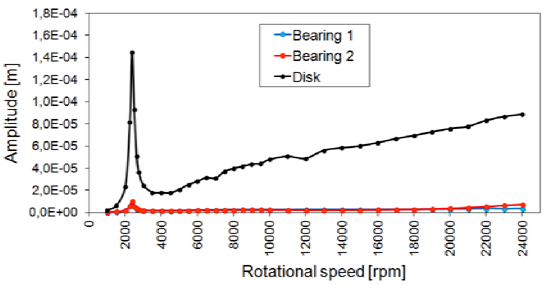

Fig. 14. Vibration amplitudes of the bearing journals and of the disk (variant no. 5).

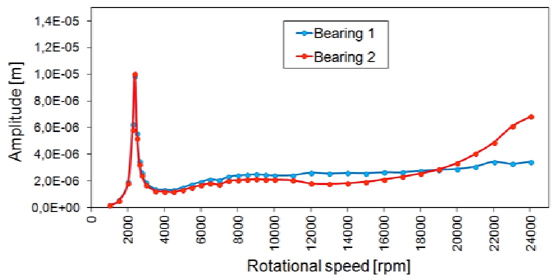

Fig. 15. Vibration amplitudes of the bearing journals (variant no. 5).

The foil bearing, to which the second variant is devoted, has lower stiffness than the one in the first variant. This is due to the fact that the bump foils are made of different types of materials. Figure 8 shows peak to peak vibration amplitudes of the disk and of the bearing journals, and Fig. 9 depicts only journals' vibrations. The plots obtained in the variant no. 2 are similar to the relevant plots associated with the base variant (variant no. 1). The maximal vibration value was also noted at 2,350 rpm. At this rotational speed, the vibration amplitude of the disk and of the bearing journals is respectively 0.15 mm and 0.012 mm. At higher speeds, a linear increase in vibration amplitude of the bearing journals took place up to the value of 0.01 mm. The computation results were obtained within the entire range of rotational speeds. Compared to the variant no. 1, this variant provided higher vibration levels at the bearing locations. The reason for this can be found in a more flexible bump foil.

Variant no. 3, in comparison with variants no. 1 and no. 2, is characterised by lower structural stiffness. In this variant, a single bump foil is substituted with five short foils. Additionally, the number of bumps, by which the top foil was supported, decreased from 35 to 30. Figure 10 presents vibration amplitudes corresponding to three component parts of the rotor, and Fig. 11 only to the bearing journals. Despite the change in the bearing structure, the obtained vibration amplitude graphs are very similar to the previous variants, except that the resonant speed decreased slightly. The maximal vibration amplitude of the disk is 0.124 mm and occurs at the speed of 2,300 rpm. This amplitude decrease is due to the fact that the resonance appears at a lower rotational speed (this time the rotor was subjected to a lower centrifugal force arising from its residual unbalance). As far as bearing journals are concerned, the maximal vibration amplitude is 0.0123 mm. The computations were conducted for the rotational speed of 22,000 rpm. For higher speeds in this variant, no convergence could be found during computations. This is due to the occurrence of a hydrodynamic instability at the fluid/structure contacts inside the bearings.

In the next foil bearing variant (variant no. 4), steel bump foils were substituted with the brass ones. This caused that the bearing's stiffness decreased even more. Consequently, it also had an impact on the dynamic characteristics of the rotating system itself, as shown in Fig. 12 and Fig. 13. This time, the resonant speed is 2,300 rpm. The maximal vibration amplitude of the disk is as high as 0.148 mm, while this value for the bearing journals is 0.0145 mm. The highest rotational speed for which the computations were successfully conducted is 24,000 rpm. For this speed, the maximum vibration amplitude reached, at one of the bearing journals, the value of 0.012 mm. This foil bearing variant is characterised by the fact that vibrations of the bearing journals are the highest out of all analysed variants, and this is the case within the whole range of rotational speeds.

The bearings used in the last variant (variant no. 5) have the highest structural stiffness. The boost in the stiffness was achieved by an increase in the thickness of the top foil and of the bump foil. Figures 14 and 15 present the computation results that correspond to variant no. 5. The application of bearings of the highest stiffness caused an increase in the resonant speed of the system and a decrease in the vibration level of the bearing journals. The vibration amplitude of the disk and of the bearing journals is 0.144 mm and 0.01 mm, respectively. Having analysed the vibration levels presented in Fig. 15, you can easily notice that in one of the bearings, after having passed the resonance area, they were relatively constant across the speeds unlike in the previous cases.

The vibration levels of the mechanical system were analysed and changes in vibration trajectories of the disk and of the bearing journals were observed during performing computations at different rotational speeds (Fig. 16 and Fig. 17). Vibration trajectories of the bearing journals are of specific importance because they can provide valuable information on the dynamic performance of the rotor and stability of the bearings' operation. One should bear in mind also that the occurrence of the first signs of oil whirl or oil whip in a bearing reflects its unstable operation, and a further increase in rotational speed and/or changes of other operating parameters may result in machine damage [9]. The analysis of changes in the shape of vibration orbits throughout the rotational speeds confirmed that the aforementioned computational problems are related chiefly with the occurrence of the onset of instability of flow in the bearings. The appearance of such an instability is clearly seen in Fig. 17 showing a vibration trajectory at 22,000 rpm. Even within the resonance area, at a far lower speed, the vibration trajectories are regular in shape even despite a large vibration amplitude (Fig. 16).

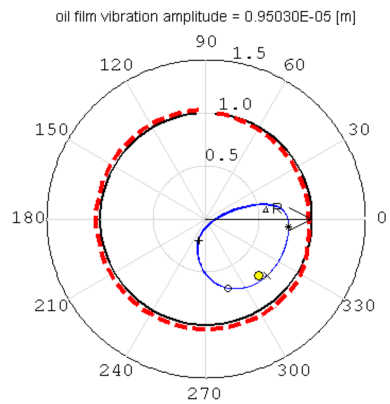

Fig. 16. Vibration trajectory of the foil bearing's journal at 2,350 rpm (variant no. 1).

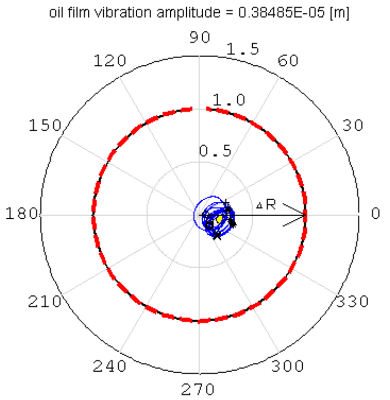

Fig. 17. Vibration trajectory of the foil bearing's journal at 22,000 rpm (variant no. 1).

In spite of the fact that the bearing journals' vibration level was not high at the highest rotational speeds, a non uniform and rapidly changing pressure distribution in the lubricating gap as well as a turbulent flow of the lubricant caused problems in the operational performance of the tested mechanical system.

5. CONCLUSIONS

This article is devoted to a simulation study on the rotor supported by foil bearings, which is aimed to determine the dynamic characteristics of the rotating system in question and choosing the optimal construction of its bearings. The study takes into account several construction variants of foil bearings lubricated with a low boiling liquid known under its trade name HFE-7100. Its most important computation results in terms of operational performance are presented in the form of plots showing vibration amplitudes of the disk and of the bearing journals. The obtained results allowed to draw several conclusions that you can find in the following sentences.

The rotor mounted on a test rig is characterised by a stable operation within a wide range of rotational speeds, and this is the case in each of the foil bearing variants proposed. In each of the five variants, we obtained a resonant speed at which a rapid increase in vibration amplitudes of the disk and of the bearing journals manifested itself. If the increase in the rotational speed continues, starting from a resonant speed, an equally steep drop in the vibration level can be observed. Increasing further the rotational speed, a very wide range of the mechanical system's stable operation can be observed as well as a linear increase in the vibration amplitude until it reaches the maximal speed depending on the bearing's construction variant.

The analysis of the five proposed variants indicates that we can improve the bearing's properties (to a certain extent) by changing the foils' geometry and/or material of the bump foil. This is the way we can alter the dynamic performance of the entire rotating system. In the simulation study in question, parameters such as resonant speed, vibration amplitude and stability threshold of the system have been particularly affected by changes in the bearings' properties. Since we have such an opportunity to shape properties of fluid flow machinery equipped with foil bearings, this makes such a bearing system an important field for innovation.

The main criterion for choosing the appropriate bearing should be a low vibration level. Applying this criterion, it can be concluded that variant no. 5 is the best one out of all analysed variants (up to the speed of 24,000 rpm). But it is necessary to bear in mind that, despite the positive verification of developed computation models, certain simplifying assumptions had to be adopted. For this reason, it is planned to compare obtained results with results of experimental research that can only be undertaken after the slide bearings have been substituted by foil bearings.

The results of this study show that the method consisting in a computational analysis using advanced numerical models of foil bearings is an effective method to assess the dynamic performance of such bearings.

SOURCE OF FUNDING

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