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Dynamic Behaviour of Meso-Scale Turbines – Application in Dispersed Cogeneration

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Abstract

The article discusses tendencies in the development of turbines, from the macro to micro scale, and their possible application in dispersed cogeneration, i.e. in home power installations. On a small scale, turbines and bearings are a source of specific problems connected with securing stable rotor operation and their susceptibility to some material imperfections. This paper presents results of investigations of the high-speed rotor of a micro turbine being an element of the micro power plant in dispersed power engineering based on renewable energy sources. The basic problem of such devices is to assure stable rotor operation within the entire range of rotational speeds. Foil bearings and special rotor structure were applied. It turned out that the situation, in that the rotor – after loosing its stability – stabilizes again when rotational speed increases, is possible. This is a new phenomenon determined by the author as 'multiple whirls.' Possibilities of stabilizing the situation by the application of hybrid lubrication and siphon pockets also are presented in the paper.

Keywords: rotor dynamics, nonlinear vibrations, hydrodynamic instability

Introductory Remarks

Cogeneration is a process of simultaneous generation of heat and electric current (so called CHP systems). If this process takes place in low-power machines (from a few to a few dozen KW, sometimes up to a few hundred KW) this provides opportunities for so-called small-scale dispersed cogeneration, the idea of building power centres for particular individual receivers. Of high profitability in the dispersed cogeneration is the use of renewable energy sources, biomass and biogas in particular, for feeding electrical appliances. The dispersed cogeneration based on renewable energy sources defines present research trends in the European Union, and also in our country. This is in direct relation to the problems of ecological production of energy and energetic safety of entire regions. That is why this subject matter is one of the strategic research priorities in the country.

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Key elements in the dispersed cogeneration system are micro power plants, i.e. microturbines cooperating with ecological boilers. The term "microturbine" is not very adequate in the context of dispersed cogeneration, although it is in common use. Here we talk about the geometric scale of the turbine of an order of a few to a few dozen millimetres, rotational speed of an order of less than a hundred thousand rev/min and power output from about ten KW to several hundred KW. Perhaps in this case it would be more appropriate to introduce a term of "mesoscale," the more so that the real "micro" scale of the turbine is of an order of a few millimetres. Thus we could define development tendencies as the way from MACRO, through MESO to MICRO scale, as is illustratively shown in Fig. 1 [1-4]. In our further considerations we will use both the term of "mesoscale," and a common definition of a "micro power plant" in reference to the same class of power machines.

Safe operation of those machines brings new challenges for designers, operators, and research workers. The operation of a turbine at rotational speed of an order of several

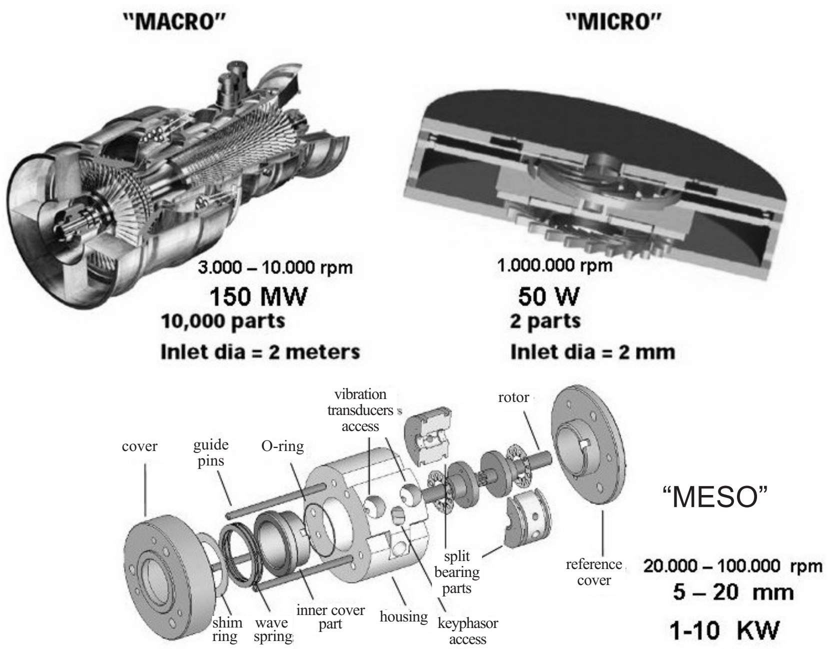

Fig. 1. Tendencies in development of turbines from MACRO through MESO to MICRO scale [1, 2].

thousand rev/min, small external loads and small dimensions of the entire machine create serious problems with keeping stable operation of the system and securing relevant durability of its particular elements. Of particular importance here are the bearings, which should secure stable and safe operation of the entire machine. Numerous novel solutions are proposed in the form of low-friction polymer bearings, foil bearings, or various types of gas bearings. A concept which becomes more and more attractive takes into account a low-boiling agent that is normally used in the thermal cycle of the microturbine, as a lubricating liquid in the bearings (so-called ORC-based systems). Such an approach provides opportunities for using hermetically tight closed systems. A method for increasing anti-vibrational resistance of slide bearings lubricated by low-boiling liquids, and thus reducing vibroacoustic threats, is increasing its damping abilities by relevant changes of the shape of the lubricating clearance during bearing operation and proper direction of heat flow and bush deformation. To allow studying these problems, bearing characteristics are to be modelled by advanced thermal models and models taking into account thermoelastic deformations of the lubricating clearance.

On the other hand, continuous operation of the system close to the stability limit, or even after exceeding it, requires the use of nonlinear models for analyzing rotor dynamics issues, as only these models allow tracing the process of rotor trajectory transformation and estimating the scale of a potential threat. Moreover, possible faster propagation of rotor cracks, for instance, as a result of significantly increased number of fatigue cycles, forces creating models taking into account this type of material imperfection. This means operating conditions in which the real vibro-acoustic threat for these machines can have a place. Of some importance is also the operation of these machines at low noise emission levels, as, being parts of household equipment, they could disturb the calm of the residents. As we can see, analyzing the dynamic state of the "home" power plants requires qualitatively novel research tools.

The article presents results of the analysis of a microturbine in operation (in the mesoscale), after exceeding the stability limit and taking into account thermoelastic deformations of bearing bushes. The studies were done using models and computer codes developed in the Institute of Fluid Flow Machinery in Gdańsk.

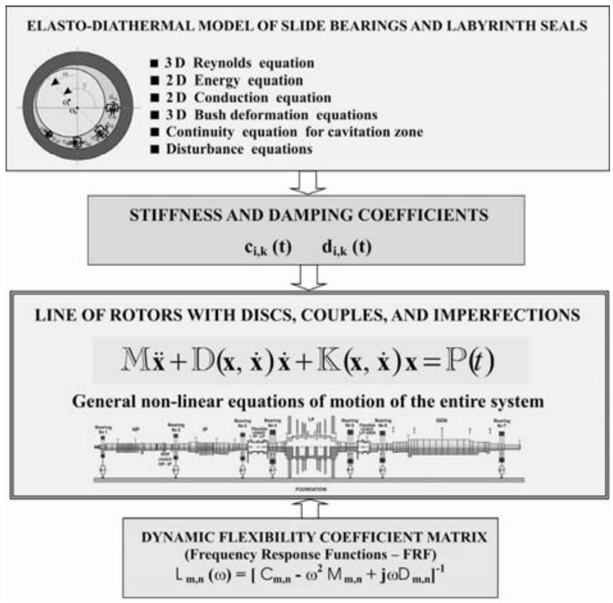

Fig. 2. Time-dependent slide bearing stiffness and damping coefficients, and the supporting structure dynamic flexibility matrices as the elements coupling together the main subsystems of the rotating machine, thus providing opportunities for solving mutually coupled ordinary and partial non-linear differential equations [5].

Research Tools

In the Institute of Fluid Flow Machinery in Gdańsk, modern research tools were developed in the form of a package of computer codes that compose compact environment bearing the name of "MESWIR." Describing particular models and codes composing this package goes far beyond the limits of this article. Many details can be found in the voluminous monograph [5, 6] that also includes complete documentation referring to the verification of the developed research tools, both in laboratory scale and on real objects.

From the point of view of goals of the present article, of some importance is the statement that these tools allow continuous modelling of the operation of the system, both in stable regime, and after exceeding the stability limit, i.e. in the nonlinear range. This way we can trace various instability forms, for instance oil whirls and whips, using one research tool. Moreover, the adopted elastodiathermal model of the bearing allows tracing the effect of thermo-elastic bush deformations on the development of various types of instabilities of the entire system.

Fig. 3. Photo of the testing stand operating in the Vibrodiagnostics Laboratory of the Institute of Fluid-Flow Machinery, Polish Academy of Sciences (journal diameter d=0.1 m, discs diameter D=0.4 m and length L=3.2 m) [6, 7].

The most difficult issue in the adopted model is the way in which all subsystems composing the rotating machine system are combined together into one compatible mathematical and numerical system. This is of special importance for nonlinear modelling, when iterative procedures are to be applied. Here, a concept was adopted according to which all slide bearing stiffness and damping coefficients and supporting structure dynamic flexibility matrices are modified at each time step of the iterative procedure. As a result, we obtain a set of tens of hundreds of nonlinear mutually coupled differential equations, composing the equations of motion for the entire system.

This is illustratively shown in Fig. 2. The method used for searching a solution to the equations of motion and obtaining relevant stability of the numerical solutions is given in the already mentioned monograph [5].

The MESWIR code was experimentally verified both at the research stand and with using real objects such as large power turbosets [5].

For the needs of the research, this system was coupled with commercial programs of the ABAQUS type to determine dynamic properties of the supporting structure and of the whole system. Thus, bearing and rotor characteristics (MESWIR system) as well as dynamic characteristics of supporting structure (ABAQUS system) were calculated in one iteration loop. However, this type of coupling requires additional verifying of such research tools. The photo of the testing stand operating in the Vibrodiagnostics Laboratory of the Institute of Fluid-Flow Machinery, Polish Academy of Sciences, is presented in Fig. 3. This is a system of a journal diameter d=0.1 m, discs diameter D=0.4 m and length L=3.2 m. Fig. 4 presents the experimental verification results done by a modal analysis with the application of

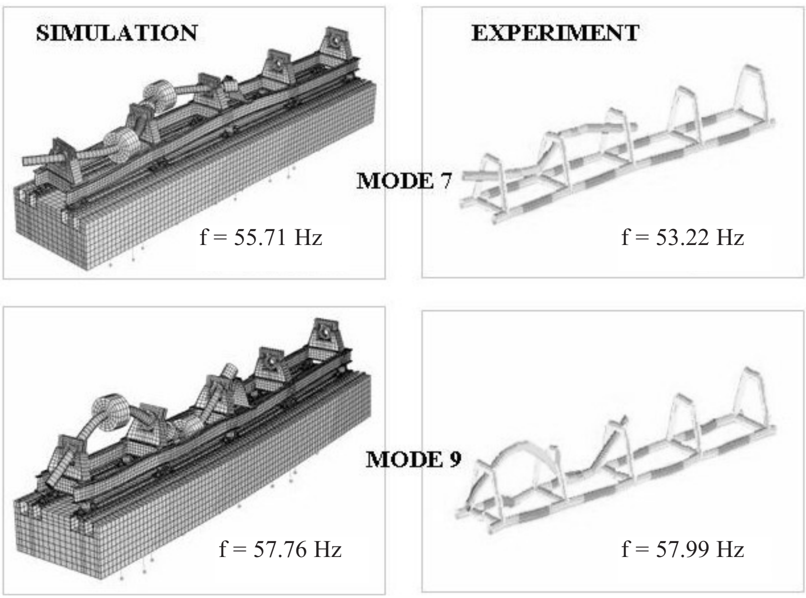

Fig. 4. Results of the experimental verification performed by modal analysis for two main modes (PC CADA system was used in experiments – right-hand side) [6, 7].

PC CADA tools. As can be seen, the compatibility of results is quite satisfactory.

Research tools prepared and verified in such a way were applied in the research constituting the basic contents of this paper.

Stability Testing of High-Speed Rotors

Problems related to ecological energy generation at a small and dispersed scale have become especially actual in recent years. Dispersed power engineering requires building micro power plants which also means micro turbines of power from a few to a few dozen KW. The idea of building micro turbines for low-boiling ORC agents, which ensures small dimensions of devices and facilitation of servicing, has become attractive. Unfortunately, it is obtained at the cost of a high rotational speed of the rotor, approaching 100,000 rpm. Thus, the main problem becomes ensuring stable operation of the device within the entire rotational speed range of the rotor. This type of device is most often coupled with boilers supplied with renewable energy sources.

A concept of such a micro power plant developed in the Institute of Fluid Flow Machinery in Gdańsk is shown in Fig. 5 [8]. Essential elements of the micro turbine constitute slide bearings of special characteristics, ensuring high stability of the system. The foil bearings were chosen since – due to facilitation of changing the oil clearance geometry (deformations of a membrane part of a bearing bush) – they can stabilize system operation as the rotational speed increases.

Fig. 6 presents a sketch of bearings applied in the object shown in Fig. 5 and the FEM discretization of the foil system in order to calculate the static substitute stiffness of a bearing, while Fig. 7 presents a verification of the obtained results. Compatibility of theoretical and experimental results is considered to be satisfactory. The notion of the substitute stiffness is essential for the calculation capabilities of the MESWIR code (multiple calculations in one iteration loop).

The system assumed for testing consisted of a rotor (with one disc and a generator) placed on two foil bearings of dimensions shown in Fig. 5. A low-boiling agent ORC was used as bearing lubricant, which significantly simplifies construction of the whole micro power plant.

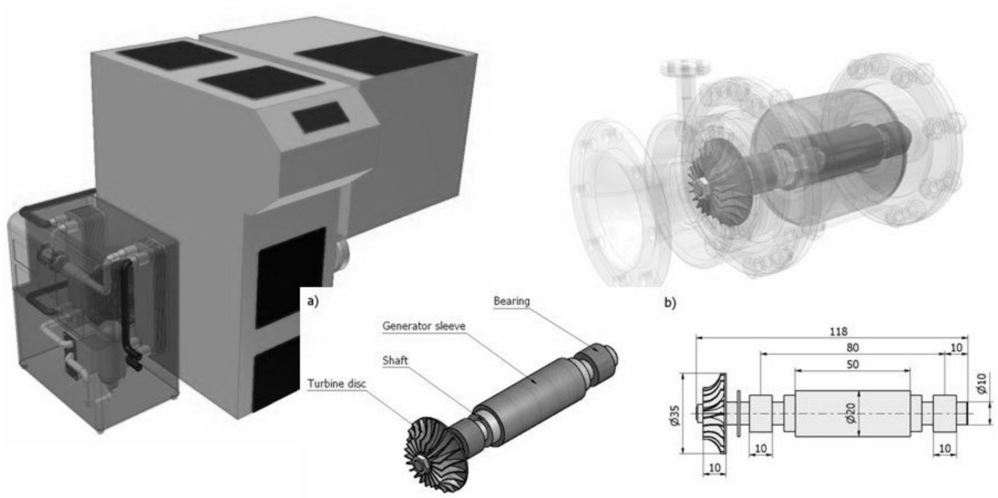

Fig. 5. Object of testing. Micro power plant (a boiler and turbine set) developed in the Institute of Fluid Flow Machinery in Gdańsk of a thermal and electric power of 20 KW and 3 KW, respectively. Single-stage radial-axial turbine for low-boiling agents ORC and a rotational speed up to 100,000 rpm [8].

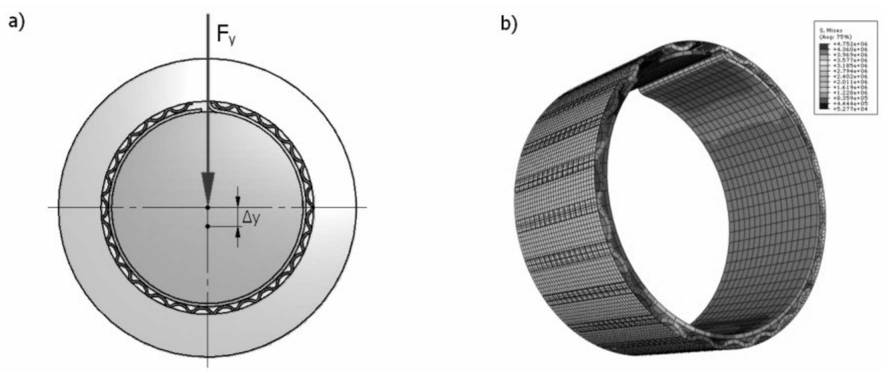

Fig. 6. Sketch of foil bearings applied in a micro turbine from the object shown in Fig. 5 (of a journal diameter 10 mm) and the FEM discretisation of an inner foil, membrane and outer foil – in order to calculate the substitute stiffness of the bearing bush [8].

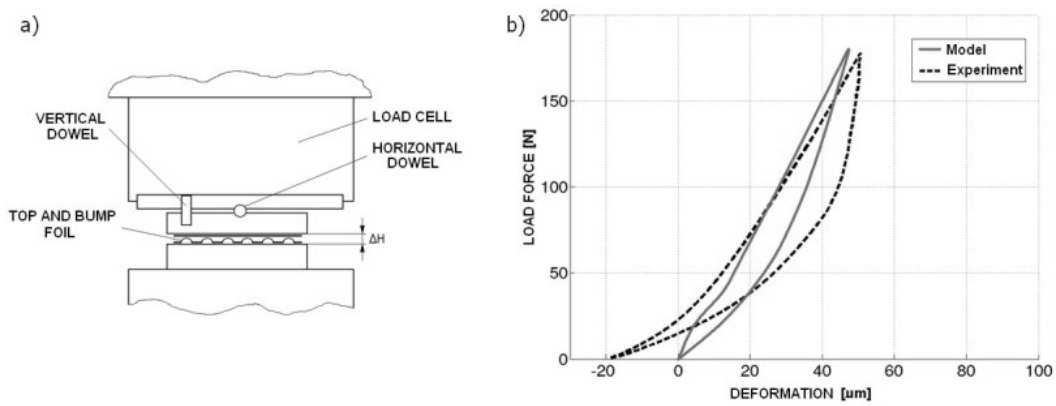

Fig. 7. Verification of a static substitute stiffness of the foil bearing bush. Model and calculations [8], experiment according to [9].

Typical values of the possible unbalancing of a rotor disc (as an excitation force) were assumed as well as some parametric values of damping the foil bearing bush and supporting structure (from the material data sheets).

To determine the hydrodynamic pressure distribution in foil bearing the typical Reynolds boundary conditions has been applied both for static and dynamic external loads. These experimentally verified boundary conditions are also good for high rotor speeds.

The calculation results in the form of amplitude – speed characteristics within the speed range up to 100,000 rpm – are presented in Fig. 8.

Attention is called to the quite different operations of bearing No. 1 (at the disc) and bearing No. 2 (free end). While bearing No. 1 is stable within the entire range of rotational speeds, bearing No. 2 exhibits two characteristic zones of exceptionally high vibration amplitudes exceeding 70% of a bearing clearance. To identify this phenomenon and to exclude common resonance, the shape of relative displacement trajectories of the journal and bearing bush in these zones were analysed. The calculation results for the first zone and after passing through it are presented in Fig. 9.

Analysis of Fig. 9 explicitly indicates that the displacement trajectory of bearing No. 2 at a rotational speed of 25,000 rpm (the first zone of high amplitudes) has features characteristic for the evaluated hydrodynamic instability, the so-called 'whip'. The 'whip' means – in this case – a developed form of whirls of a lubricating agent within the lubrication clearance. This is pointed out by double shaft rotations (it means a vector of external excitations) falling to one full precession, which creates 2 phase markers (FM) on the trajectory. Actually, there are three markers since points for 0 and 720 degrees overlap each other, which does not always happen. This means that the same positions of the excitation force vectors (horizontally to the right: TAL = 0, 360 and 720 degrees) correspond to different positions on the journal trajectory within the bearing clearance.

However, most unexpected is the observation that after the system has exceeded the first zone of hydrodynamic instability (which means the first 'whip') the system returns to a stable operation of bearing No. 2 (i.e. to the typical situation in which one phase marker on the trajectory corresponds to one rotation of the excitation vector). The situation remains a stable one up to the rotational speed of approximately 65,000 rpm. After the system has exceeded this speed a rapid instability ('whip') occurs again, followed by a subsequent calming down. Thus, we are dealing with a multiple 'whip' of a lubricating agent, understood as formation and decaying of the hydrodynamic instability as the rotational speed of the rotor increases.

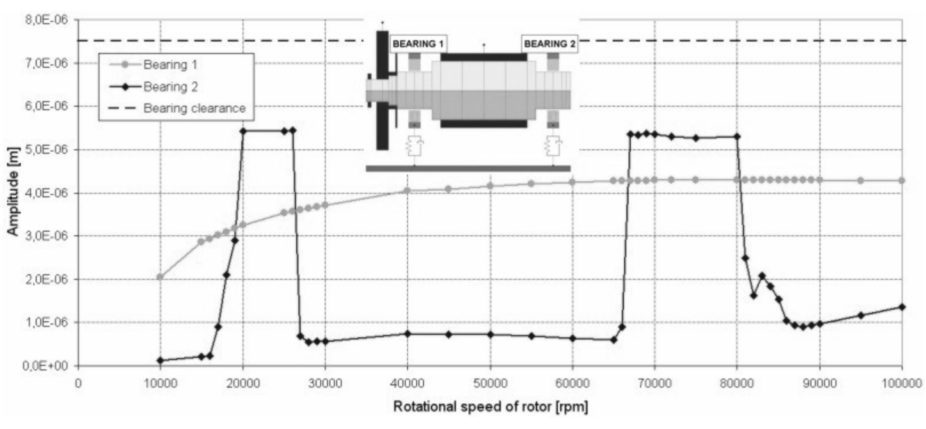

Fig. 8. Amplitude – speed characteristics of the rotor from the micro turbine (shown in Fig. 5) calculated for the relative vibrations of a journal and foil-bearing bush. Illustration of the 'multiple whirls' phenomenon in bearing No. 2 (repeated processes of formation and decaying of high amplitude zones caused by hydrodynamic instability).

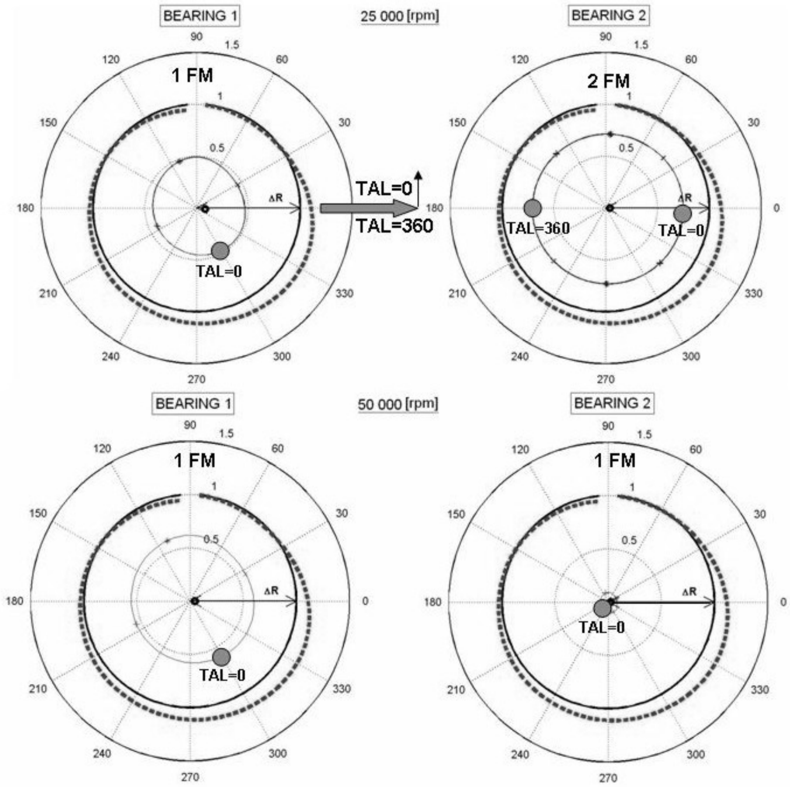

Fig. 9. Displacement trajectories of journal of bearing Nos. 1 and 2 calculated for the first high amplitudes zone (25,000 rpm) and in the transient period (50,000 rpm). Image of the first hydrodynamic 'whip' in the trajectory – 2 phase markers FM (upper right-hand side trajectory). Broken line indicates deformations of the bearings inner foil.

If we assume a stiff bearing bush we are unable to model the phenomenon. This allows us to assume that variable deformations of the foil bearing bush (corresponding to the turbine rotational speed increase) are responsible for such a process. Since this phenomenon is not known (from references), the author of this paper suggests calling it 'multiple whirls.' The phenomenon of 'multiple whirls' has been quite often observed in practice by exploitation crews of large power plants. Small oil whirls form and then disappear on one of the recorded bearings, but this does not cause any instability in the system. Unfortunately, those results have not been published. In our case, it means in the case of a small power micro turbine that we are dealing with large whirls, i.e. whips. However, it is a different magnitude scale and a different rotational speed range.

A zone of 'multiple whirls' is very interesting from the point of view of the hydrodynamic pressure distribution. The fact of existing 2-phase markers corresponding to the situation, in which for the same position of the exciting force vector (TAL=0 and 360 degrees) we have two different journal positions on the trajectory and two different pressure distributions in the bearing. This is illustrated in Fig. 10.

A question arises: is there any method enabling improvement of the dynamic state of a micro turbine – in this case characteristics of bearing No. 2? Various possibilities were considered. One of them was a concept of a hybrid lubrication with the application of siphon pockets. Several examples have been analyzed. The most favourable was the case with 4 siphon pockets (2 in the upper part and 2 in the lower part of the bearing bush – Fig. 11) and siphon pressure at the level of 0.01 MPa. The calculation results are presented in Fig. 11. Comparing these results with the amplitude-speed characteristics for bearing No. 2 in Fig. 8, we will immediately notice that 'multiple whirls' disappeared and vibration amplitudes are significantly lower in the entire range of rotational speeds. This observation might suggest directions of further research; however, solutions based on hybrid lubrication – in the micro turbine case – significantly complicate the design and exploitation side of the problem.

Conclusions

In this paper we are dealing with an interesting phenomenon, named 'multiple whirls,' by the author of a lubricating agent. We understood it the formation and decay of hydrodynamic instability as the rotational speed of the rotor increases. If we assume a stiff bearing bush we are unable to model the phenomenon. This allows us to assume that variable deformations of the foil bearing bush (corresponding to the turbine rotational speed increase) are responsible for such a process. As stated previously 'multiple whirls' have been observed in large power plants. In our case, it is a different magnitude scale and a different rotational speed range.

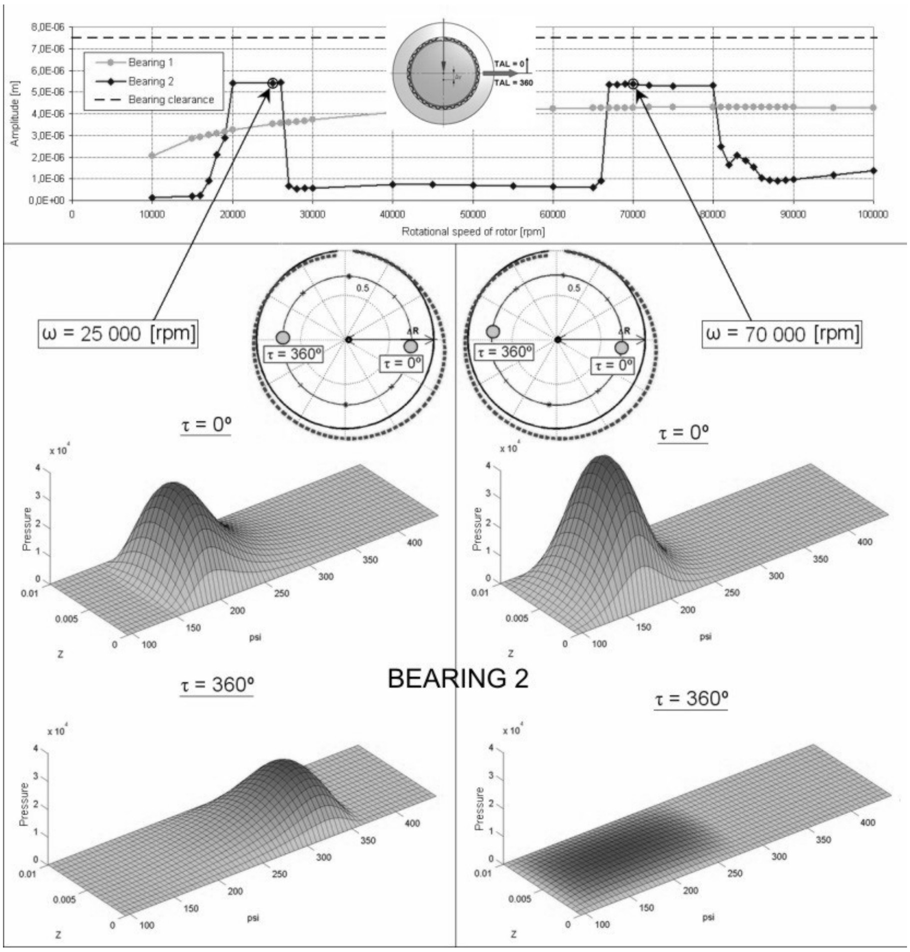

Fig. 10. Pressure distribution calculated for various positions of phase markers within a zone of 'multiple whirls' for the same position of the excitation force vector (horizontally to the right, it means for TAL = 0 and 360 degrees).

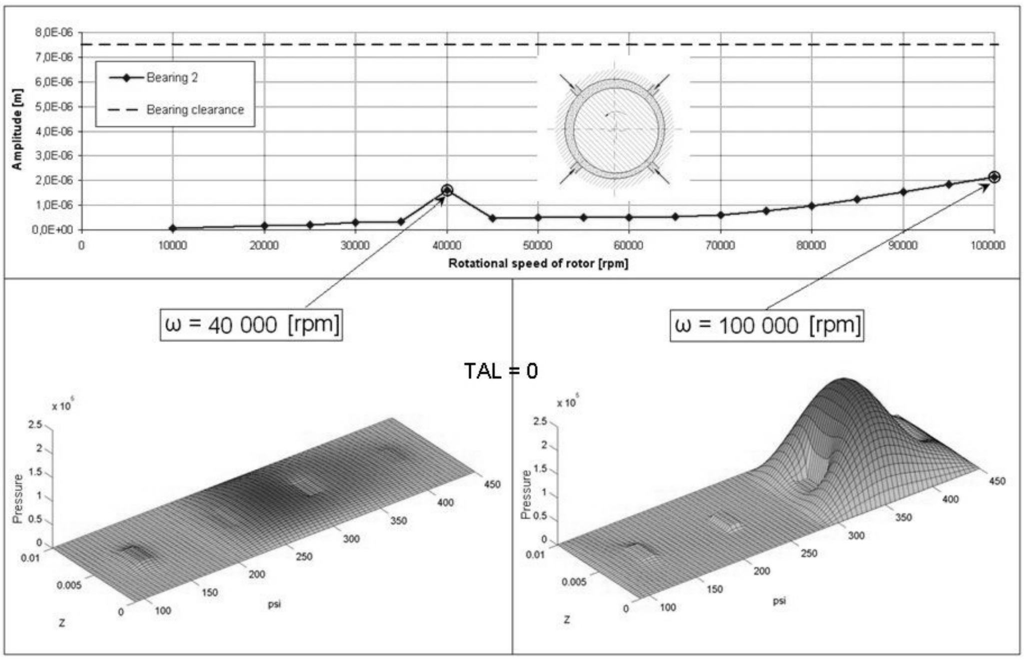

Fig. 11. Influence of siphon pockets and siphon pressure for the time-history of vibration amplitudes and pressure distribution – for bearing No. 2 of the object shown in Fig. 5. 4 siphon pockets were applied (upper part of the Figure) and pressure = 0.01 MPa.

The phenomenon of ‘multiple whirls’ found by advanced computer simulations and performed by means of the experimentally verified codes, requires further investigation. Experimental investigations in this field are planned in the Gdańsk Research Centre. However, they will be put into operation only after building the first prototypes of micro power plants and relevant testing stands. Currently we have only unpublished information that the phenomenon has been recognized by means of direct measurements of vibrations in large power plants [10]. An interesting hint for further investigations constitutes the obtained results of hybrid lubrications. It is possible to stabilize the system by this method but at the cost of significant structural complications.

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