Mechanics and Mechanical Engineering
Vol. 15, No. 3 (2011) 259–271


# New Method of Analysis of Dynamic State of Power Micro–Devices

Jan Kiciński
*Institute of Fluid Flow Machinery*
*Polish Academy of Sciences (IFFM PAS), Gdask*
*University of Warmia and Mazury in Olsztyn, Poland*
*kic@imp.gda.pl*

Received (16 August 2011)
Revised (17 September 2011)
Accepted (23 September 2011)

Main energy conversion machinery used and to be used in cogeneration systems are schematically described. Some assets of distributed generation are pointed out and small–scale cogeneration systems designed for energy units of distributed cogeneration are described.

In the small scale, turbines and bearings are a source of specific problems connected with securing stable rotor operation. Accepted has been two kind of high speed micro-turbines of electric power about 3 KW with multistage axial and radial rotors supported on foil bearings.

A concept which becomes more and more attractive takes into account low–boiling agent, which is normally used in the thermal cycle of the microturbine, as a lubricating liquid in the bearings (so called ORC based systems). The scope of the present article is limited to the discussion of dynamic characteristics of the selected design. The properties of the rotor combined with slide bearings (foil bearings in this particular case) were taken under investigation. A combination of this type is a certain novelty, since a typical modal analysis of such objects refers to a rotor itself. Analysing the dynamic state of the "home" power plants requires qualitatively novel research tools.

It turned out that the situation, in that the rotor – after losing its stability – stabilizes again when the rotational speed increases, is possible. This is a phenomenon determined by the author as multiple whirls.

*Keywords*: Cogeneration, Micro-turbines, Rotor dynamics, nonlinear vibrations, hydrodynamic instability

## 1. Introduction

Cogeneration is a simultaneous production of electric energy and heat which leads to a more efficient utilisation of primary energy. Thus, cogeneration brings considerable savings in the final energy production and contributes to decrease the level of emissions into the environment, especially $CO_2$. The opportunities for cogeneration are however usually determined by the demand on heat, which can vary for example seasonally and with the daytime. The complex analysis of a cogeneration unit should take into account the characteristics of the heat receiver.

Sample quantitative gains from cogeneration are displayed in Fig. 1. As seen from the picture, in order to produce 21 units of electric energy and 33 units of heat in cogeneration (assuming the theoretical total cogeneration efficiency of 90%) there are 60 units of primary energy required, whereas 97 units of primary energy are needed to produce the same amount of final energies in separate generation.

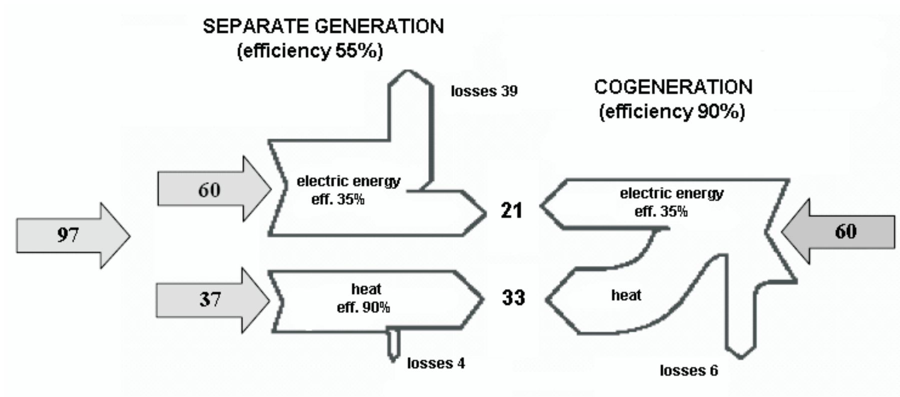


**Figure 1** Production of electric energy and heat in a separate mode and in cogeneration

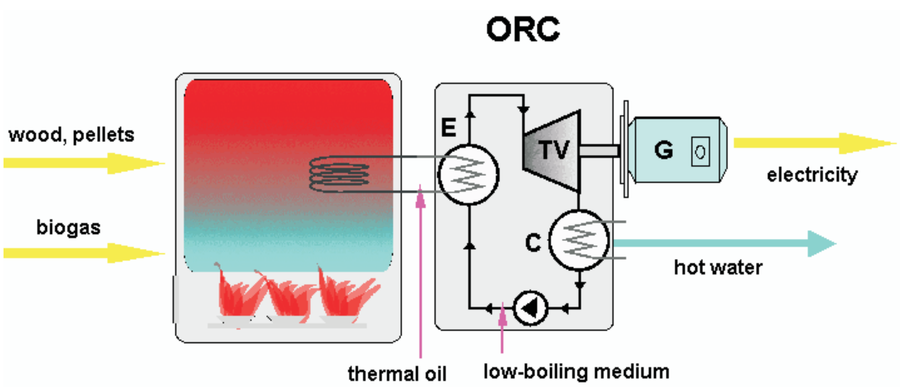


**Figure 2** Cogeneration unit with ORC; E – evaporator, TV – steam turbine, C – condenser, G – generator

The counterparts of large power turbines in distributed generation are small steam turbines or microturbines that operate in an organic Rankine cycle (ORC) whose schematic is presented in Fig. 2. Main components of this CHP station are ecological boiler fit to combust different kinds of biomass or biofuels, intermediate heat cycle to extract heat from flue gases to thermal oil as a heat carrier, evaporator, turbine with a low boiling liquid as a working medium, generator, condenser and circulating pumps for the working medium and thermal oil [1–5]. In the presented heat cycle, electric energy is a by–product and forms only about 10–20% of the total heat. Micro CHP units dedicated for individual households of total heat capacity up to 20kWt and electric power up to 4kWe are currently being elaborated at IFFM PAS.

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 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 low–boiling agent, which is normally used in the thermal cycle of the microturbine, as a lubricating liquid in the bearings (so called ORC based systems). This means operating conditions, in which real vibro–acoustic threat for these machines can have place. Of some importance is also the operation of these machines at low noise emission level, as, being parts of household equipment they could disturb the calm of the residents. As we can see, analysing the dynamic state of the "home" power plants requires qualitatively novel research tools.

## 2. Stability testing of high–speed rotors

Two versions of microturbines – axial and radial, both of 3 kW power but with different rotational speeds, were suggested [6–8]. Different speeds derive from the necessity to avoid supersonic speeds at the ends of blades (ie. speeds higher than 1 Mach). High speeds at the ends of blades lead to noise and possible flow perturbations, which result in an unstable operation of the system. The main aim of the research was to find out whether suggested system of rotors and foil bearings will not exceed the system stability limit, given the assumed range of rotational speeds, and if it will not generate dangerous resonances.

A general concept of the microturbine rotors is shown in Fig. 3 and Fig. 4 whereas the model of accepted journal bearings is shown in Fig. 5. The scope of the present article is limited to the discussion of dynamic characteristics of the selected design.

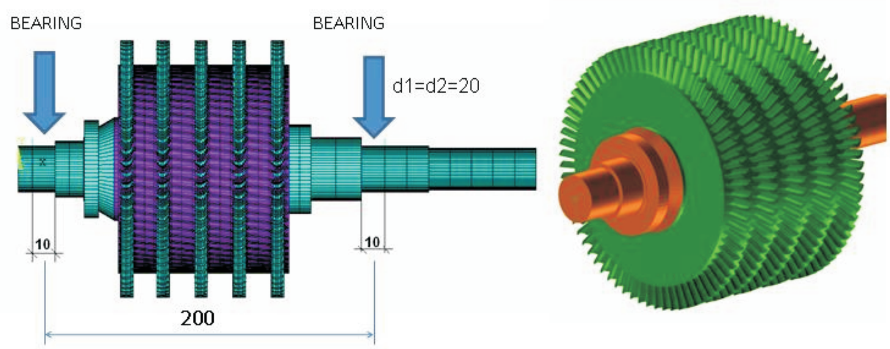


**Figure 3** Object of testing. Five – stage axial microturbine rotor of electric power of 3 KW and rotor speed 8 000 rpm (for low–boiling agents ORC). Bearing journal diameters: d1=d2=20 mm. Model MES: 380 000 DOF (MSC Patran) [7]

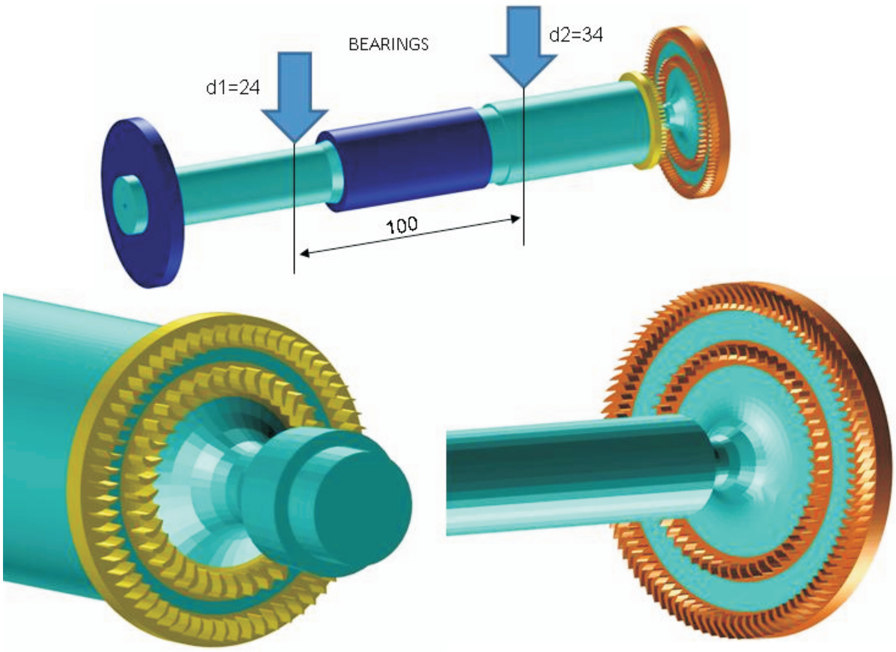


**Figure 4** Object of testing. Four – stage radial microturbine rotor of electric power of 3 KW and rotor speed 23 800 rpm (for low-boiling agents ORC). Bearing journal diameters: d1=24 mm, d2=34 mm. Model MES: 200 000 DOF (MSC Patran)[7]

The investigations were performed using the own MESWIR series codes, which initially had been developed for studying rotor–bearings systems in large power turbosets [9]. After introducing some modifications, now they also make it possible to study small–dimension and high–speed rotors supported on hydrodynamic and hybrid slide bearings, or even on foil bearings. The modal analyses were performed using the known commercial codes like Patran and Abaqus).

The rotors were supported on foil bearings – Fig. 5, in which a low–boiling medium from the micro power plant thermodynamic cycle was used as the lubricating agent. The medium was assumed to be delivered in liquid form to the bearing interspace.

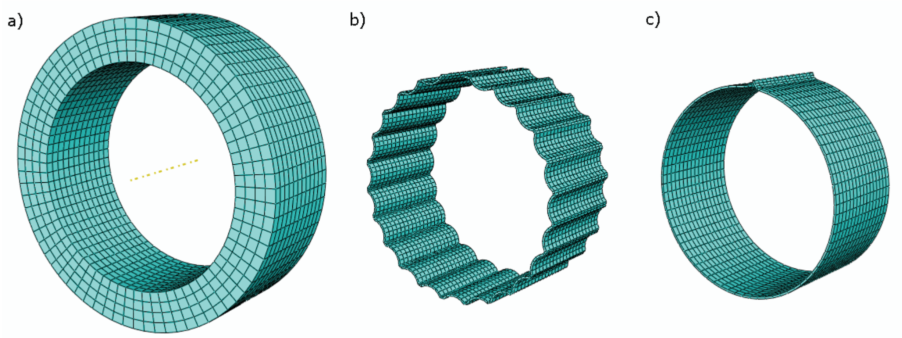


**Figure 5** FEM models of foil bearing elements (a – sleeve, b – bump foil, c – top foil)[8]

The deformation of the set of foils, caused by the pressure in the lubricating space of the bearing, was analysed using the code ABAQUS, along with complementary codes worked out to transfer the data between this code and the MESWIR series codes [4], [5], [9]. The analysis of the bearing operation took into account the interaction between the medium which lubricates the bearing and the set of foils.

The results of investigation were presented in Fig. 6 and Fig. 7 in the form of main stiffness coefficients of low–boiling medium working as lubricant in foil bearing. These coefficients constitute the basis to determine the modal analysis of the accepted rotor – bearings system.

The first two lateral modes calculated for the rotor–foil bearings sets are presented in Fig. 8 for axial microturbine and in Fig. 9 for radial microturbine.

More important, from practical point of view, are the amplitude – frequency responses and vibrations spectra (diagnostic cards) calculated for the two kinds of microturbines and for proper speed regions. The results were presented in Fig. 10 and Fig. 11.

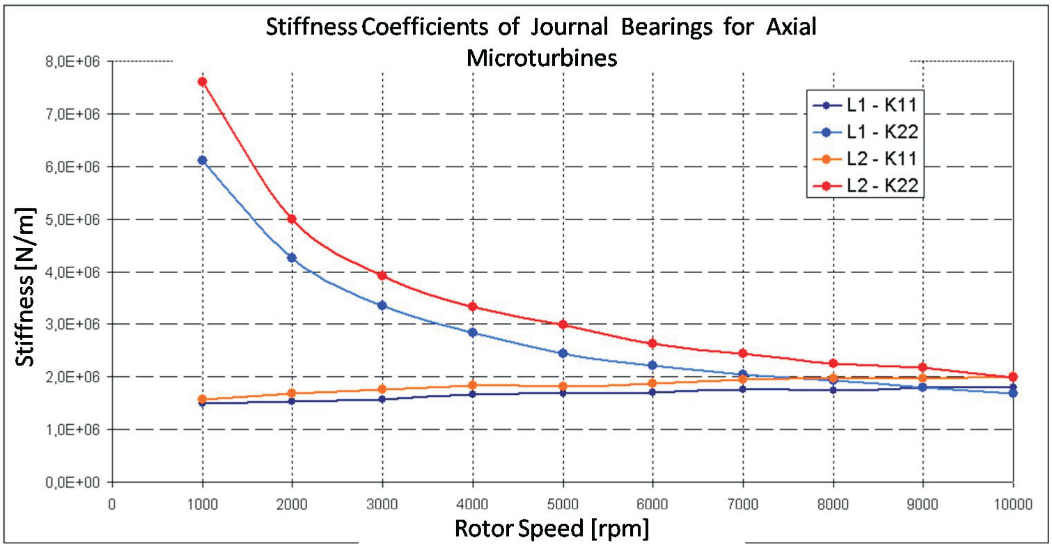


**Figure 6** The main stiffness coefficients (horizontal K11 and vertical K22) calculated for low boiling medium ( lubricant) of foil bearings L1 and L2 of axial microturbines – Fig. 3 [6]

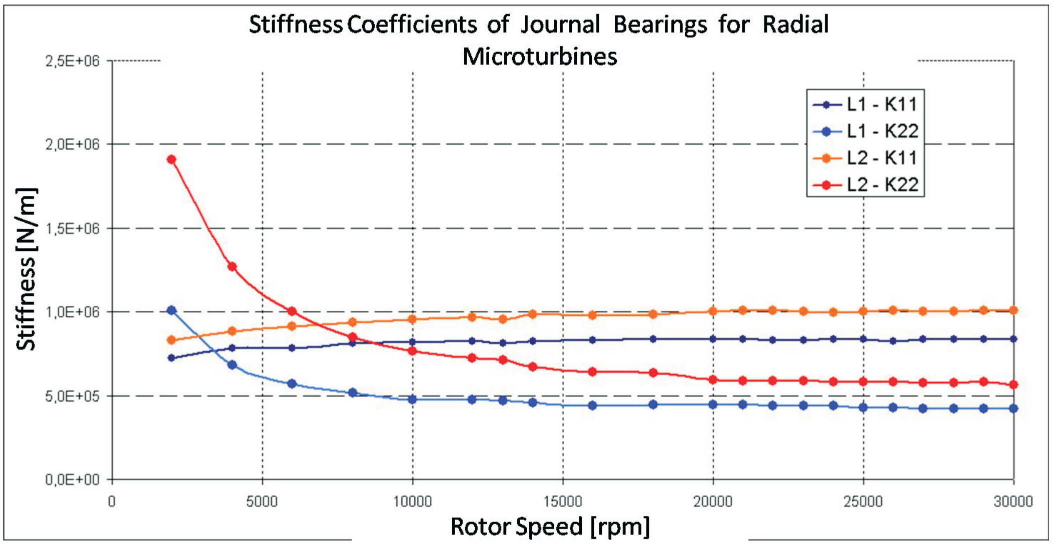


**Figure 7** The main stiffness coefficients (horizontal K11 and vertical K22) calculated for low boiling medium ( lubricant) of foil bearings L1 and L2 of radial microturbines – Fig.4 [6]

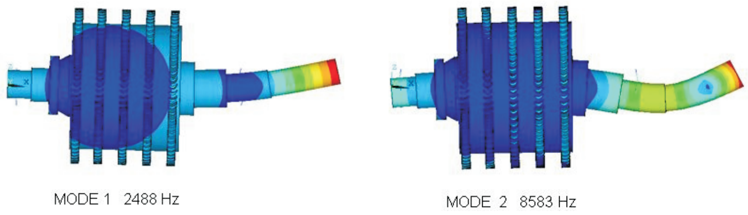


**Figure 8** Modal analysis for axial microturbines (set of rotor – foil bearings). The first two lateral modes[7]

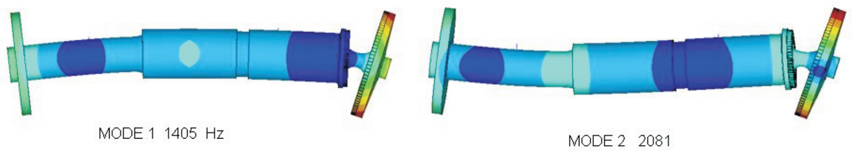


**Figure 9** Modal analysis for radial microturbines (set of rotor – foil bearings). The first two lateral modes [7]

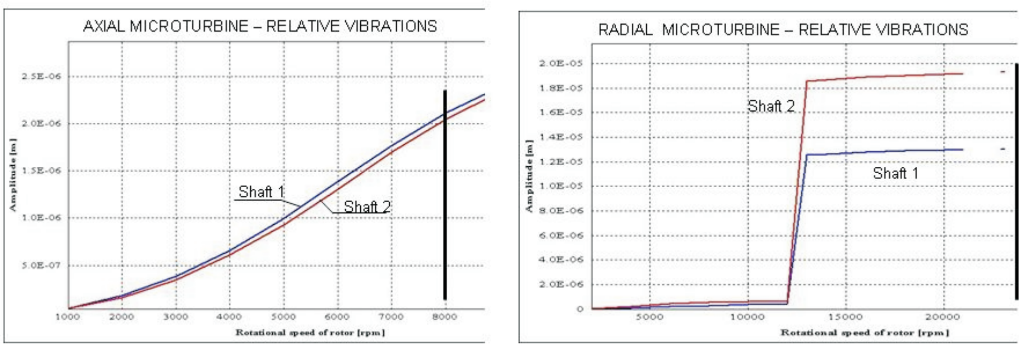


**Figure 10** Amplitude–frequency responses calculated for relative shaft-bush vibrations in bearing 1 and bearing 2 ( axial microturbine left, radial microturbine right) [6]

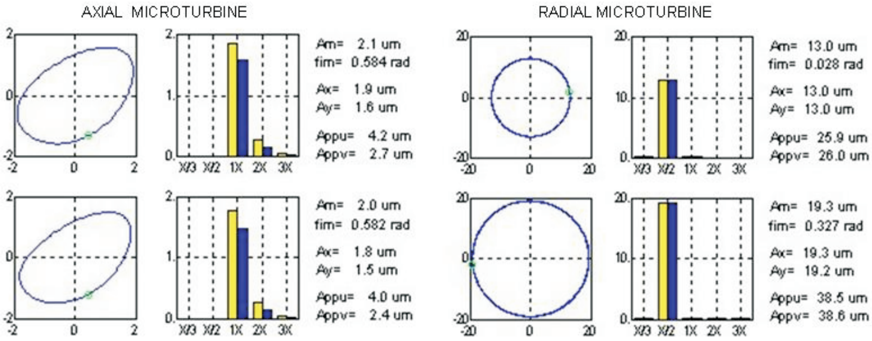


**Figure 11** Displacement trajectories and vibrations spectra calculated for relative shaft–bush vibrations in bearing 1(upper part) and bearing 2 (lower part). Left: axial microturbine at rate speed 8000 rpm, right: radial microturbine at rate speed 23800 rpm[6]

Regarding the axial microturbine (with a rate rotational speed of 8073 rpm that means 134,5 Hz) the research confirmed that the rotors with the suggested construction will operate safely in the whole range of assumed rotational speeds. The first form of eigenvalues (mode 1) occur only at 2488 Hz ( see Fig. 8 ) and in vibration spectra do not appear any subharmonic spectral line (only the 1X synchronous component - see Fig. 11 left part). It is thus exceptionally rigid construction.

The research results in an interesting conclusion concerning the radial microturbine with a rated speed of 23800 rpm (396,6 Hz). Although modal analysis of the rotor shows the lack of possible resonance frequencies (Fig. 9, mode 1 = 1405 Hz) , amplitude - frequency responses (Fig. 10 right) show the sudden increase of amplitude much earlier before it reaches the rated speed. This increase in the amplitude is caused by the exceeding the stability limit and a system operation under a lubricant medium whip condition. It confirms the presence of the subharmonic spectral line close to 0.5X in vibrations spectra (Fig. 11 right). The presence of whip means the hydrodynamic instability of the system. The vibration amplitude is almost one order of magnitude greater than in the case of axial turbine. The above means that in case of radial microturbine (Fig. 4) the suggested foil bearings and rotor construction does not meet the anticipated dynamic properties and have to be reconstructed through, for example - a significant decrease of bearing shaft diameter.

## 3. Phenomenon of multiple whirls

The idea of building micro turbines for low–boiling agents ORC, which ensures small dimensions of devices and easiness 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 the stable operation of the device within the entire rotational speed range of the rotor. This type of devices are most often coupled with boilers supplied with renewable energy sources.

A concept of such micro power plant (with another kind of turbines as shown in Fig. 3 and 4) developed in the IFFM PAS in Gdansk is shown in Fig. 12 [6–9]. Essential elements of the micro turbine constitute slide bearings of special characteristics ensuring a high stability of a system. The foil bearings were chosen since – due to easiness of changing the oil clearance geometry (deformations of a membrane part of a bearing bush) – they can stabilize the system operation as the rotational speed increases.

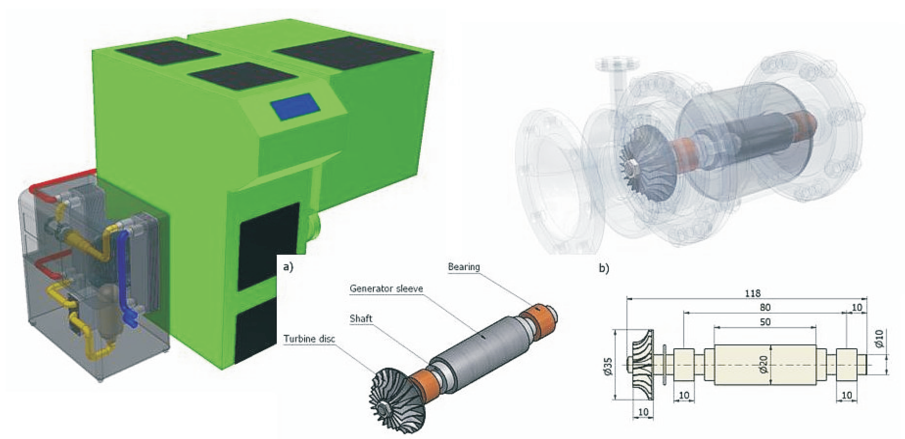


**Figure 12** Object of testing. Micro power plant (a boiler and turbine set) developed in the IFFM PAS in Gdansk of a thermal and electric power of 20 KW and 3 KW, respectively. Single–stage radial–axial turbine for low–boiling medium ORC and a rotational speed up to 100 000 rpm [10]

The notion of the substitute stiffness is essential for the calculation capabilities of the MESWIR code (multiple calculations in one iteration loop) [9].

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. 12 (shaft diameter 10 mm). A low–boiling medium ORC was used as bearings lubricant, which significantly simplifies construction of the whole micro power plant. Typical values of the possible unbalancing of a rotor disc (as an excitation force) were assumed as well as some parametric values of damping of the foil bearing bush and supporting structure (from the material data sheets) [10–12].

The calculation results in the form of amplitude – speed characteristics within the speed range up to 100 000 rpm are presented in Fig. 13.

The attention is called to quite different operation 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.

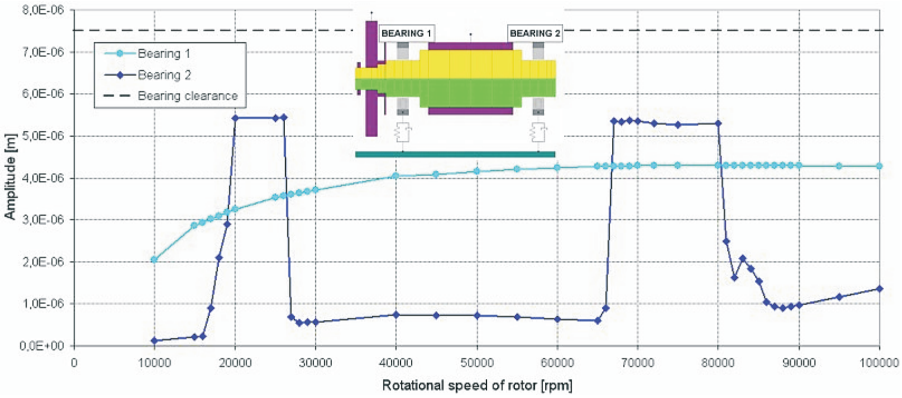


**Figure 13** Amplitude – speed characteristics of the rotor from the micro turbine (shown in Fig. 3) calculated for the relative vibrations of a journal and foil bearing bush. Illustration of the multiple whirls phenomenon in bearing No. 2 (repeated processes of formation and decaying of high amplitude zones caused by a hydrodynamic instability)

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 analyzed. The calculation results for the first zone and after passing through it are presented in Fig. 14.

Analysis of Figure 14 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 expanded hydrodynamic instability, the so–called "whip". The "whip" means – in this case – a developed form of whirls of a lubricating medium 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. 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, the 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 (it means a 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.

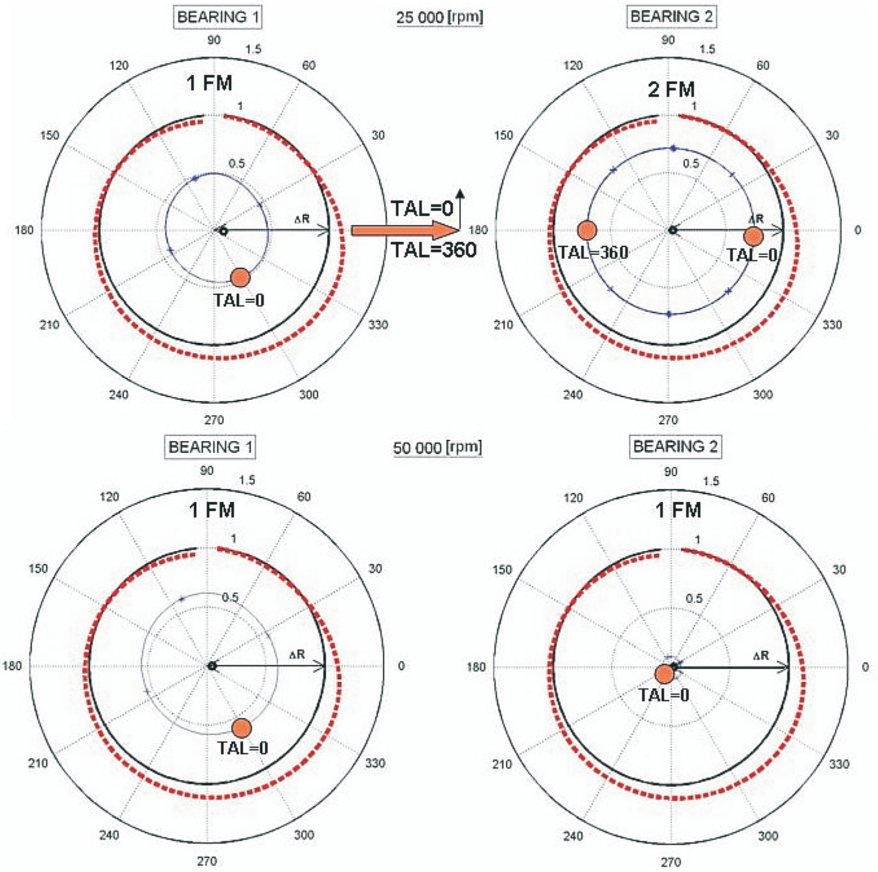


**Figure 14** Displacement trajectories of journal of bearing No. 1 and 2 calculated for the first high amplitudes zone (25 000 rpm ) and in the transient period (50 000 rpm). Image of the first hydrodynamic 'whip' in the trajectory – 2 phase markers FM (upper right-hand side trajectory). Broken red line indicates deformations of the bearing inner foil

If we assume a stiff bearing bush we are unable to model the phenomenon. This allows to assume that variable deformations of the foil bearing bush (corresponding to the turbine rotational speed increase) are responsible for such process.

The phenomenon of multiple whirls has been quite often observed in practice by exploitation service of large power plants. Small oil whirls were formed and then disappearing on one of the recorded bearings and this did not cause any instability of the entire system.

A zone of multiple whirls is very interesting from the point of view of the hydrodynamic pressure distribution. This is illustrated in Fig. 15.

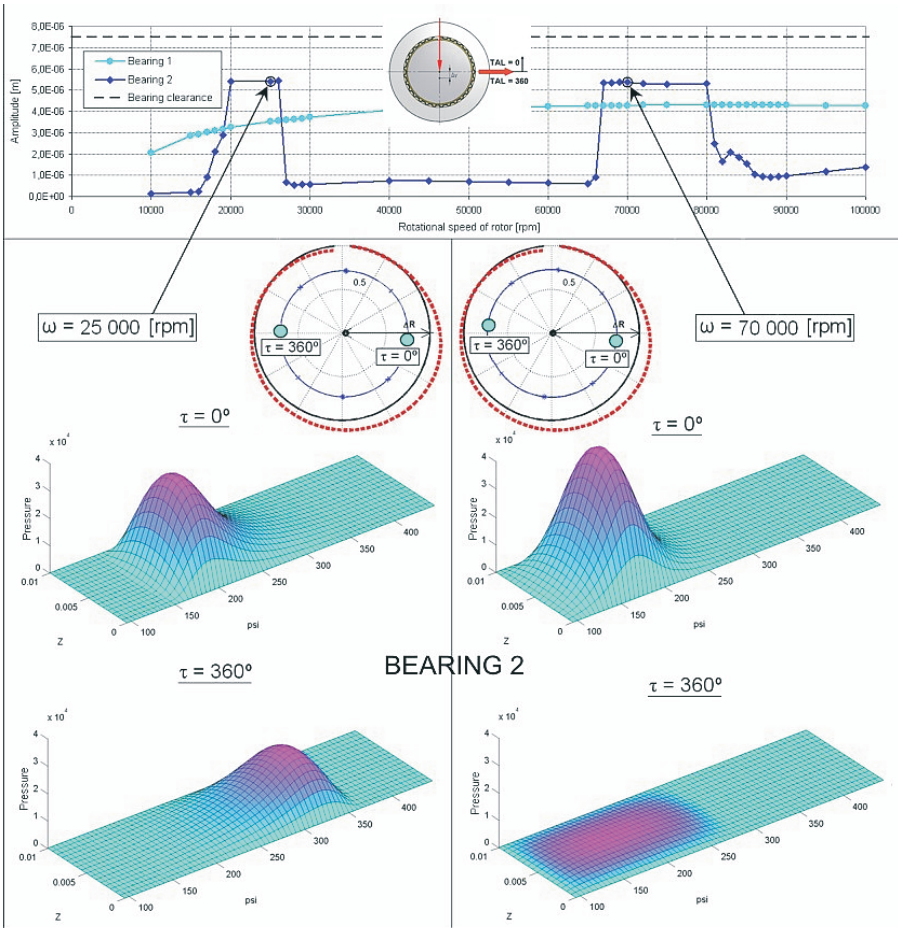


**Figure 15** Pressure distribution calculated for various positions of phase markers within a zone of multiple whirls for the same position of the excitation force vector (horizontally to the right, it means for TAL = 0 and 360 degrees)

## 4. Conclusions

The phenomenon of multiple whirls presented in the hereby paper, found by advanced computer simulations and performed by means of the experimentally verified own and commercial codes, requires further investigations both theoretical and experimental. Experimental investigations in this field are planned in the Gdansk 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 similar phenomenon was recognized by means of direct measurements of vibrations at large power plants.

## References


[1] *Distributed Energy Peer Review*, Darlington, USA, December **2005**.

[2] **Polimeros, G.**: Energy Cogeneration Handbook, *Industrial Press Inc.*, **2002**.

[3] http://www.turboden.it/en/

[4] **Schmidt, M. A.**: Technologies for Microturbomachinery, MIT, Transducers'01, The 11–th *International Conference on Solid–State Sensors and Actuators*, Munich, Germany, June 10–14, **2001**.

[5] **Frechette, L. G. and Jacobson S. A.**: Demonstration of a microfabricated high–speed turbine supported on gas bearings, MIT, *Solid–State Sensor and Actuator Workshop*, Hilton Head Is., SC, June 4–8, **2000**.

[6] **Kiciński, J., Żywica, G., Bogulicz, M., Czoska, B., Banaszek, S. and Bogucka, K.**: Obliczenia sprawdzające dla wytypowanych do dalszej realizacji wariantów wirnika turbogeneratora – analiza własności dynamicznych układu wirnik – łożyska, Opracowanie wewnętrzne *IMP PAN*, Nr arch. 71/2010, Gdańsk, *2010*.

[7] **Rządkowski R., Drewczyski M., Kiciński J. and Żywica G.**: Obliczenia sprawdzające dla wytypowanych do dalszej realizacji wariantów wirnika turbogeneratora – analiza modalna, Opracowanie wewnętrzne *IMP PAN*, Nr arch. 113/2010, Gdańsk, **2010**.

[8] **Kiciński, J., Żywica, G., Banaszek, S., Bogulicz, M. and Czoska, B.**: Modelling of Thermo–Elastic Deformations of the Foil Bearing Bush with the Application of Authors' Own and Commertial Calculation Codes (in polish), Internal Report No 22/08 of the *IF–FM PAS*, Gdańsk, **2008**.

[9] **Kiciński, J.**: Rotor Dynamics, *IFFM Publisher*, ISBN 83–7204–542–9, pp. 539, (www.rotor–dynamics.imp.gda.pl), **2006**.

[10] **Agrawal, G.L.**: Foil Air/Gas Bearing Technology – An Overview, *ASME International Gas Turbine & Aeroengine Congress*, 97–GT–347, Orland, USA, **1997**.

[11] **Andres, L.S., Rubio, D., and Kim, T.H.**: Rotordynamic Performance of a Rotor Supported on Bump Type Foil Bearings: Experiments and Predictions, *ASME Turbo Expo*, GT2006–91238, Barcelona, Spain, **2006**.

[12] **Hou, Y., Zhu, Z.H. and Chen, C.Z.**: Comparative Test on Two Kinds of New Compliant Foil Bearing for Small Cryogenic Turbo–Expander, *CRYOGENICS 44*, 69–72 (2004).