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MICROTURBINES IN DISPERSED COGENERATION – VIBROACOUSTIC THREATS

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The article discusses tendencies in development of turbines, from the macro to micro scale, and their possible application in dispersed cogeneration, i.e. in home power installations. In the small scale, turbines and bearings are a source of specific problems connected with securing stable rotor operation and their susceptibility to some material imperfections, such as shaft cracks. New research tools composing the computer system MESWIR are presented, and results of system stability investigations taking into account thermoelastic deformations of bearing bushes and various variants of their fixing are discussed, along with the results of investigations of the effect of crack depth propagation on the dynamic state of the entire system. The obtained results are rather surprising and frequently in opposition to a so-called engineering intuition. The machine selected for investigations is a three-support rotating machine with two discs, in operation in conditions specific for microturbines (low Sommerfeld numbers). The developed research tools have turned out extremely applicable for assessing vibroacoustic threats generated by this type of machines.

Keywords: microturbines, microbearings, dispersed, cogeneration.

1. 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 (between several and several tens of KW, sometimes up to several hundreds of kW), this provides opportunities for so-called small-scale dispersed cogeneration, the idea of which consists in 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 with the problems of ecological production of energy and energetic safety of the entire regions. That is why this subject matter is one of strategic research priorities in the country.

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 the dispersed cogeneration, although it is in common use. Here we talk about the geometric scale of the turbine of an order of several to several tens of millimetres, rotational speed of an order of less than hundred thousand rev/min and power output from about ten KW to several hundreds 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 [2–4]. 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.

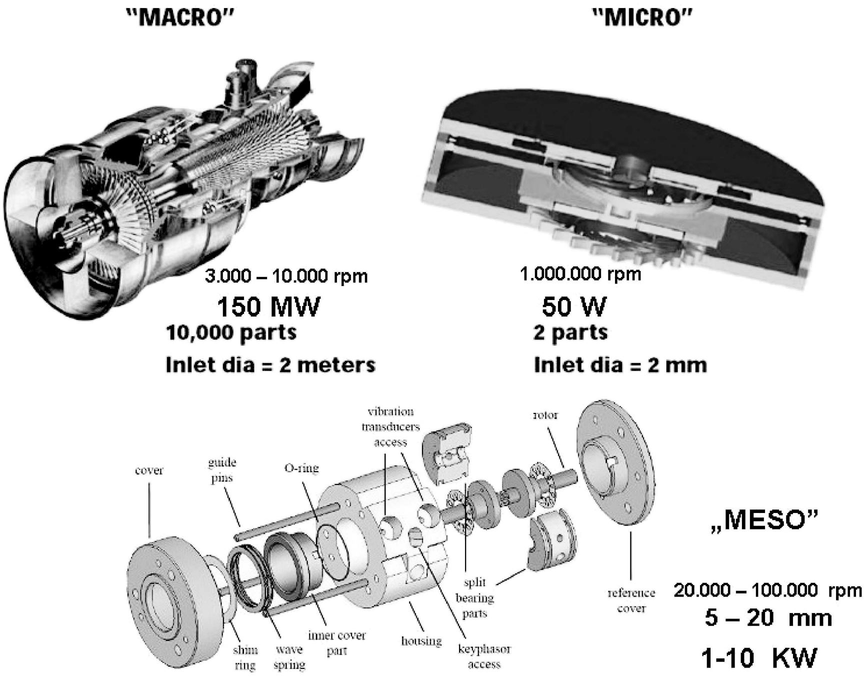

Fig. 1. Tendencies in development of turbines from MACRO through MESO to MICRO scale [2, 3].

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).* 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 analysing rotor dynamics issues, as only these models allow tracing the process of rotor trajectory transformation and estimating the scale of 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 *imperfections*. 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.*

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 analysis also includes the effect of possible rotor cracks. The studies were done using models and computer codes developed in IF-FM PAN, Gdańsk.

2. Research tools

In IF-FM PAN, 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 [1], which also includes complete documentation referring to the verification of the developed research tools, both in the 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 thermoelastic bush deformations on the development of various types of instabilities of the entire system.

What is interesting and novel in the adopted model is a concept of taking into account "breathing" of the shaft crack, depending on the scale of kinetostatic rotor deflections and instantaneous dynamic positions. This is illustratively shown in Figs. 2 and 3.

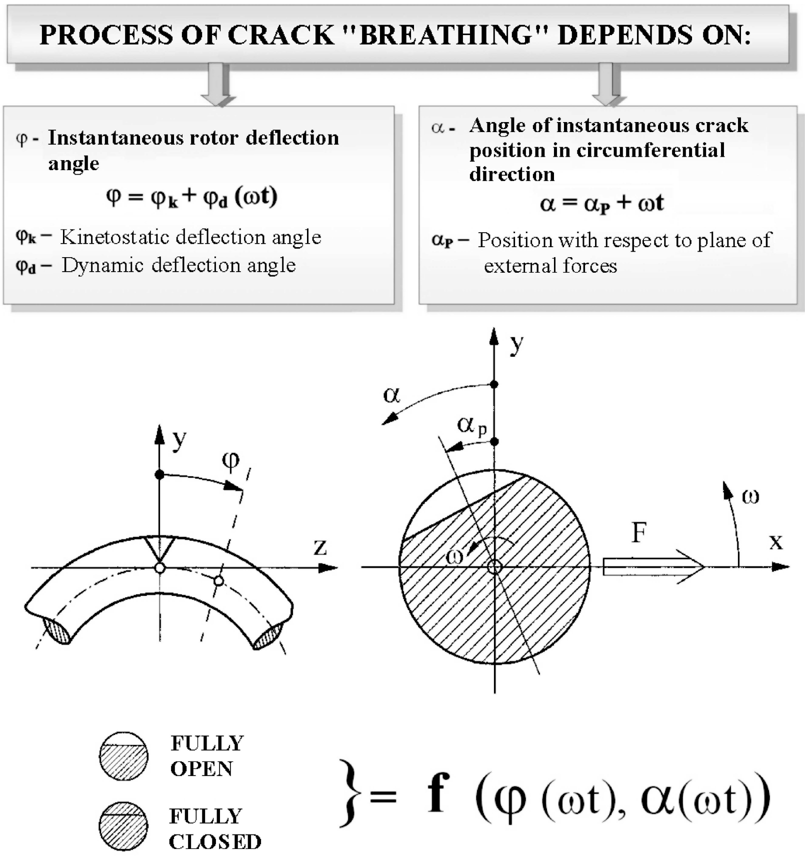

Fig. 2. The adopted concept of rotor shaft crack "breathing" process depending on instantaneous dynamic and kinetostatic positions [1].

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. 4. 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 [1].

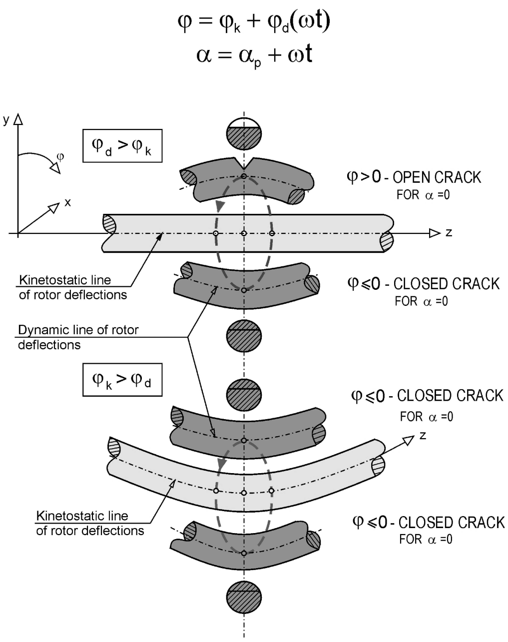

Fig. 3. The effect of kinetostatic rotor deflections on crack "breathing" process [1].

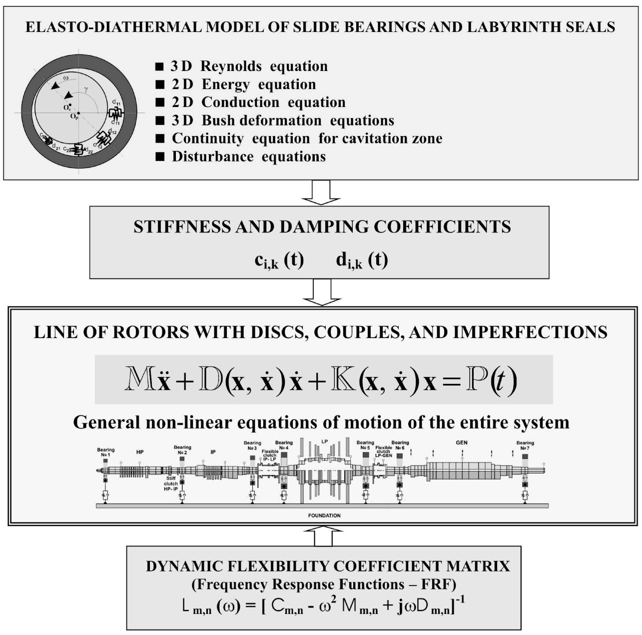

Fig. 4. Time-dependent slide bearing stiffness and damping coefficients, and the supporting structure dynamic flexibility matrices as the elements coupling together the main subsystems of the rotating machine, thus providing opportunities for solving mutually coupled ordinary and partial non-linear differential equations [1].

3. Results of investigations

A typical configuration of the microturbine rotor shaft (in the mesoscale) is composed of two discs supported by two or three bearings (Fig. 1). Of certain importance is the selection of the (low-boiling) agent flowing through the turbine, which could also be used for lubricating bearings. Here, a selection can be made from among R245fa, R134a, HFE7100, n-butane, isopropanol, toluene, and silicon oil. Selecting the agent depends on numerous factors, most important of which include: the effect on the environment, thermal stability, corrosivity, toxicity, explosiveness, and last but not least, rheological properties. Certainly, microbearings lubricated with such agents work in conditions of a very low Sommerfeld number, being the hydrodynamic similarity number and usually labelled as S_o. For the lack of more precise data, at the present stage of discussion let us assume that this number can be approximately equal to $S_o = 0.1$. If we preserve this value in the examination carried out on a research rig with different dimensions, from the hydrodynamic point of view the microturbine bearings and the bearings examined on the rig should behave in the same way.

Making use of this rule, let us select for further examination a two-disc three-support rotating machine, in operation in the vibro-diagnostics laboratory owned and operated by IF-FM PAN, Gdańsk – Fig. 5 (here, the diameter of bearing journals is equal to $d = 0.1$ m). In the operating conditions selected in such a way that the Sommerfeld number is approximately equal to 0.1, from the dynamic point of view the operation of the research rig rotor can be treated as corresponding to typical microturbine systems in the mesoscale (Fig. 1).

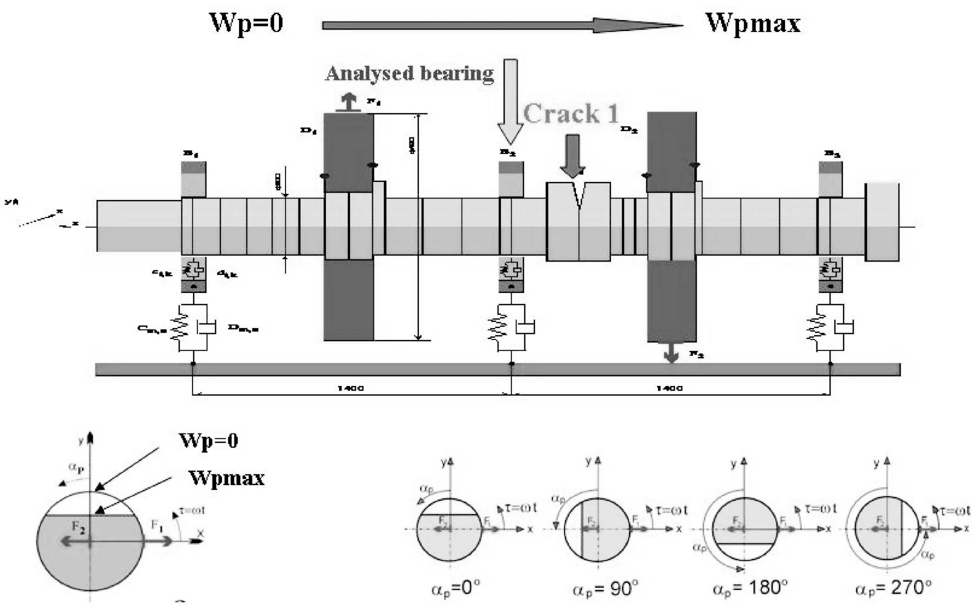

Fig. 5. Object selected for examination: three-support rotating machine with two discs.

A sample case of experimental verification of the computer system MESWIR is shown in Fig. 6. It is noteworthy that this case refers to the verification performed in the situation when the system loses its stability. This type of examination is extremely difficult and dangerous. Taking this into account, the obtained agreement between theoretical and experimental results can be considered fully satisfactory.

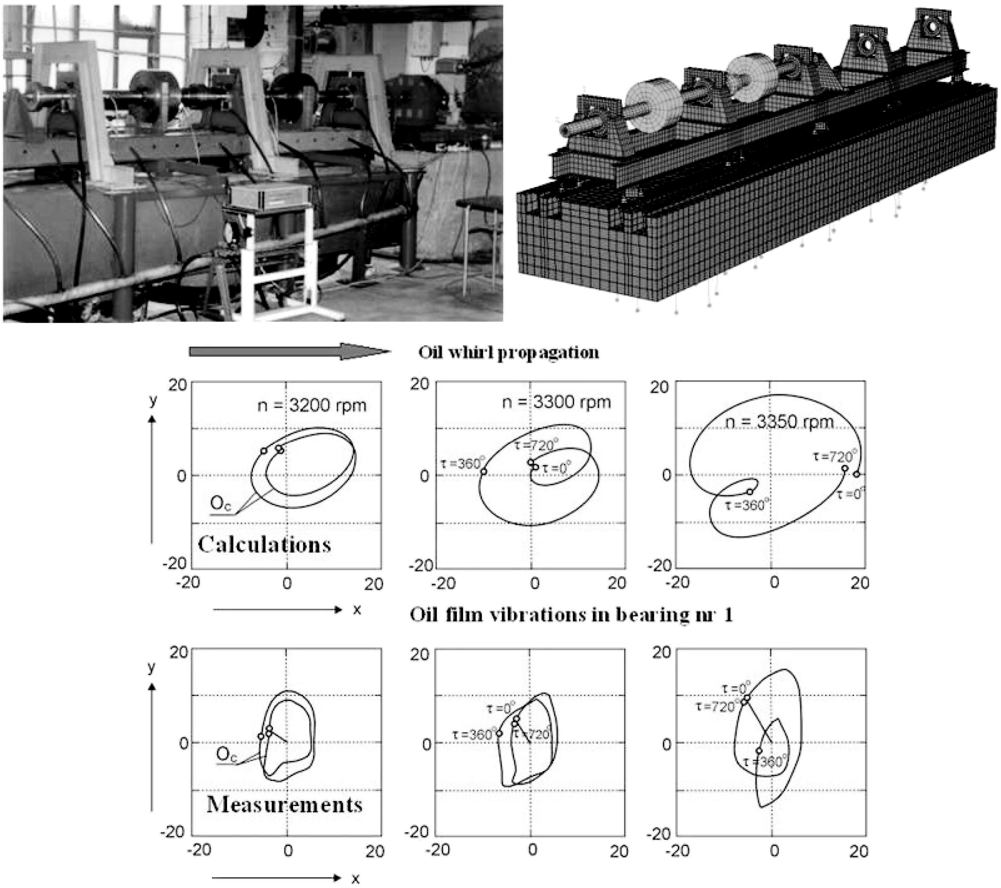

Fig. 6. Sample case of experimental verification of the computer system MESWIR on the research rig (photo taken in the IF-FM PAN vibrodiagnostic laboratory). Case of development of hydrodynamic instability (whirls).

The main goal of the reported investigations was to assess the dynamic state of the microturbine rotor defined in the above way, and to determine usefulness of the developed research tools composing the MESWIR system – [1]. As was already mentioned in Sec. 1, of highest importance was the problem of assessing machine operation in the conditions close to its stability loss, or after exceeding it, as well as a problem of thermoelastic bush deformation and its effect on the dynamic pattern of the adopted system.

As well as that, the investigations were also oriented on assessing the effect of rotor crack propagation of rotor dynamics before and after exceeding the stability limit.

The abovenamed problems are a real challenge for many research teams worldwide. Analysing the operation of the microturbine after exceeding the stability limit, taking into account bush deformation and rotor cracks, is extremely difficult. In these circumstances it is no wonder that no information can be found on this subject in the literature on microturbines (despite many efforts, the author of the article did not manage to obtain such information).

Let us assume for further examination that the slide bearing bushes were fixed in one of two possible ways:

- (theoretically infinitively) rigid fixing along the entire outer perimeter of the bush (case 1),
- partial fixing of the lower half-bush with simultaneous preliminary clamp of the upper half-bush (case 2).

These ways of fixing can be treated as representing certain limiting cases, between which those observed in real practice can be placed.

Examining dynamic characteristics of the system with the aid of the computer system MESWIR has revealed that for the assumed operating conditions and geometry of the microturbine the stability limits for the bearing fixing case 1 and 2 were approximately equal to 5470 rpm and 5030 rpm, respectively. Figures 7 and 8 present thermoelastic deformations of the inner surfaces of the bush, calculated for the both cases of fixing of outer bearing surfaces. These deformations were confronted with bearing journal displacement trajectories, calculated for rotor rotational speed equal to $N = 6350$ rpm,

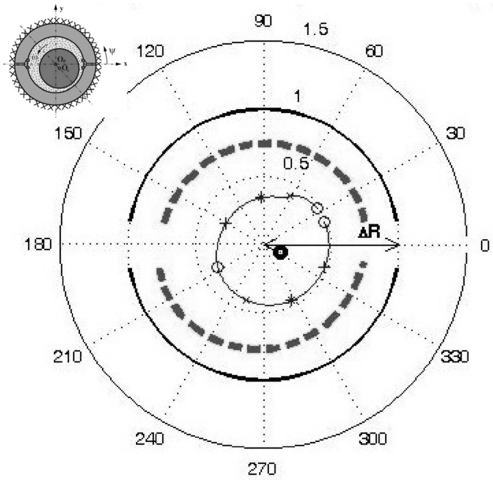

Fig. 7. Bush deformation (broken line) and journal trajectory (after exceeding the stability limit – continuous line – whipping of lubricating liquid – $N = 6350$ rpm) for rigid external bush fixing.

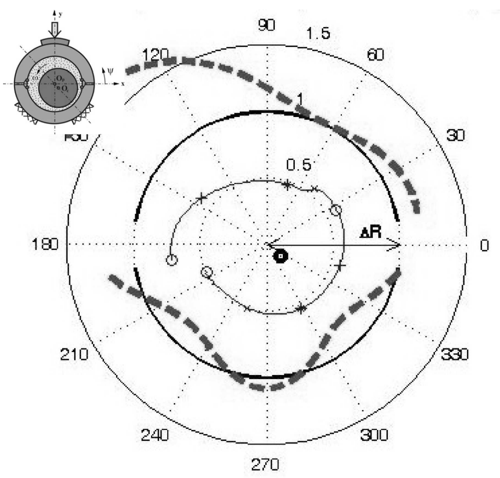

Fig. 8. Bush deformation (broken line) and journal trajectory (after exceeding the stability limit – continuous line – whipping of lubricating liquid – $N = 6350$ rpm) for partial bush fixing with preliminary clamp.

i.e. after the system stability limit has been exceeded. For the both cases of fixing this speed meant that the lubricating liquid was already in a dangerous phase of whipping, i.e. which meant high hydrodynamic instability. Such a situation undoubtedly creates conditions for real vibroacoustic threats for the entire examined system.

It is noteworthy that the way of bush fixing has no real effect on the course of whipping of the lubricating liquid, despite very large thermoelastic deformations of the bush, which in some places exceeded 70% of the nominal design clearance – Fig. 8. For the both cases of bush fixing this state of machine operation is equally dangerous.

But a question can be raised whether a similar situation takes place directly after the stability limit is exceeded, i.e. in the conditions of small hydrodynamic instability. In this case we observe the phase of so-called whirls created in the lubricating liquid. Does the way of bush fixing affect the course of hydrodynamic instability to such a negligible extent as for the whipping phase? Answers to these questions can be found in Figs. 9 and 10.

It has turned out that in this case the effect of the way of bush fixing is definitively much larger. The partial fixing with a clamp is less profitable. The system stability limit appears earlier, and the amplitudes of whirl trajectories are clearly larger.

The above result can seem a bit surprising, as the engineering intuition suggests that the operating shape of the lubricating clearance similar to that shown in Fig. 8 should improve dynamic characteristics of the system, due to the presence of a small pit in the middle part of the lower bush which stabilises vibrations. But in fact, it is quite opposite, which means that the problem of optimal selection of a shape for the lubricating clearance during bearing operation is much more complicated and each time requires individual assessment of the situation using relevant research tools.

Let us examine now the effect of rotor shaft crack on the dynamic state of the entire system. Let us assume that the crack is situated in the middle part of the rotor, Fig. 5, and propagates deep into the material, from $W_p = 0$ to the maximum value $W_p = \max$, optionally equal to 20% and 30% of shaft diameter. The objects selected for examination are "classical" bearing bushes, without thermoelastic deformations, for which the calculated system stability limit is approximately equal to 4700 rpm. Four cases of circumferential position of the crack with respect to the position of forces making the system vibrate were assumed, Fig. 5, as it turned out in earlier studies that the circumferential position of the crack affects significantly the system vibration pattern. Let us examine the effect of this type of defect in two time instants most dangerous for machine operation, which are: the resonance (here about 2600 rpm) and after the stability limit is exceeded (about 4700 rpm). The results of these examinations are given in Figs. 11–13.

For the crack propagation in the resonance, rapid increase of vibration amplitudes is only observed for W_p close to maximum values, i.e. those threatening with dangerous failure. This characteristic is strongly nonlinear. The trajectory patterns for those maximum crack depths are shown in Figs. 11 and 12. These figures reveal that the effect of circumferential position of the crack is remarkable. For $\alpha_p = 270$ the crack provoked additional coupled forms of axial, lateral, and torsional vibrations, although the sys-

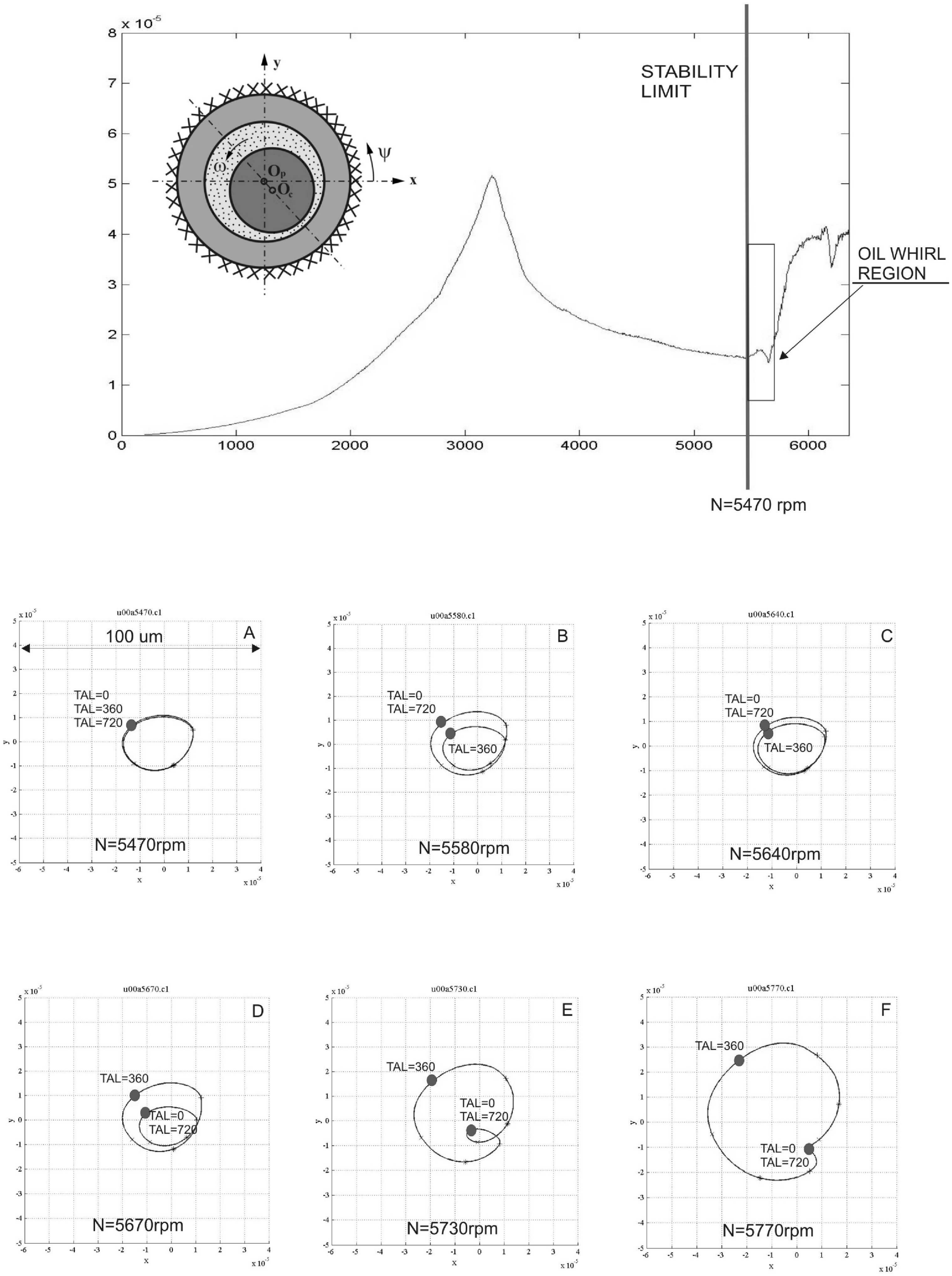

Fig. 9. Development of whirls in lubricating liquid after exceeding stability limit for rigid bush fixing.

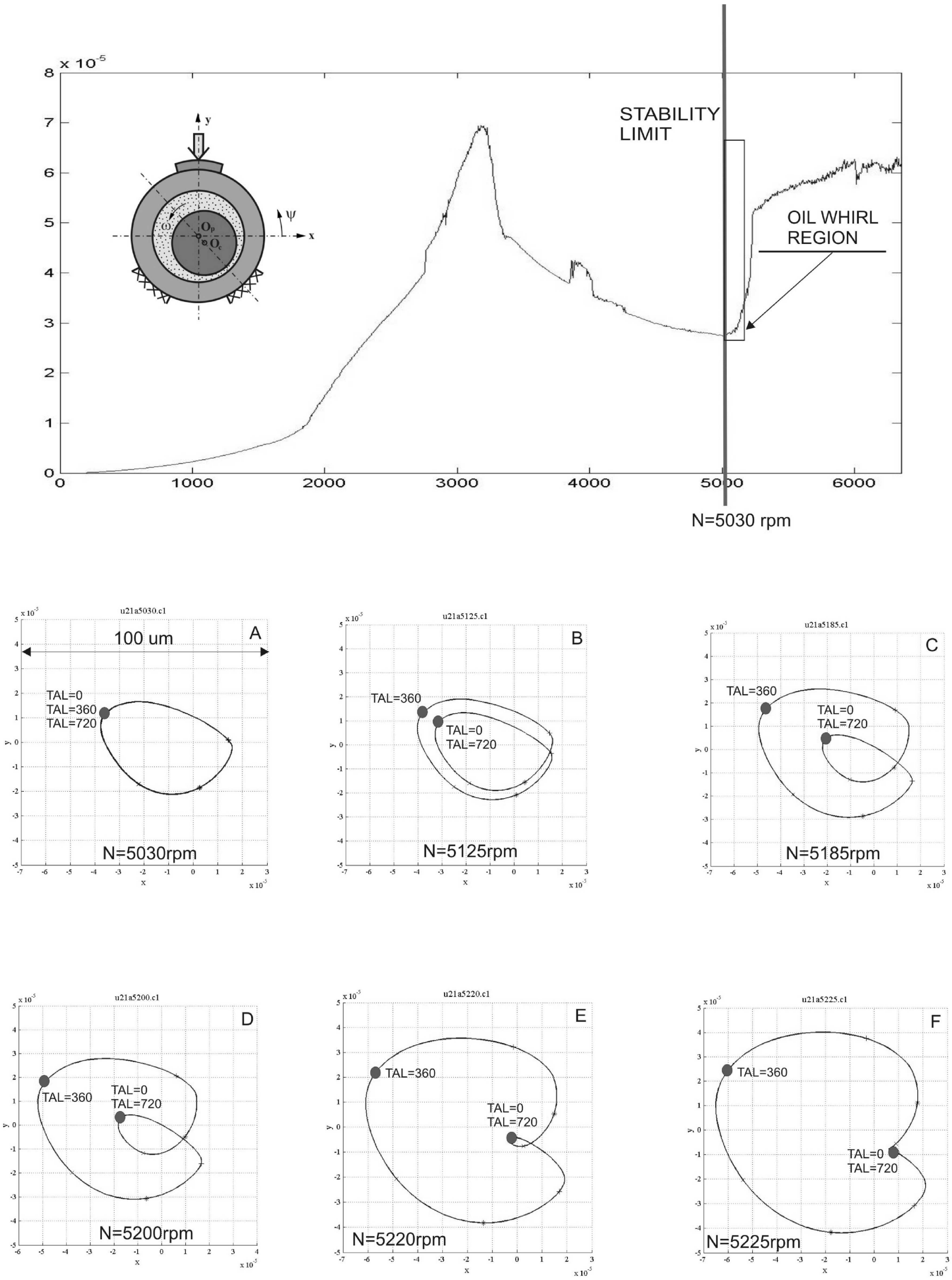

Fig. 10. Development of whirls in lubricating liquid after exceeding stability limit for partial fixing.

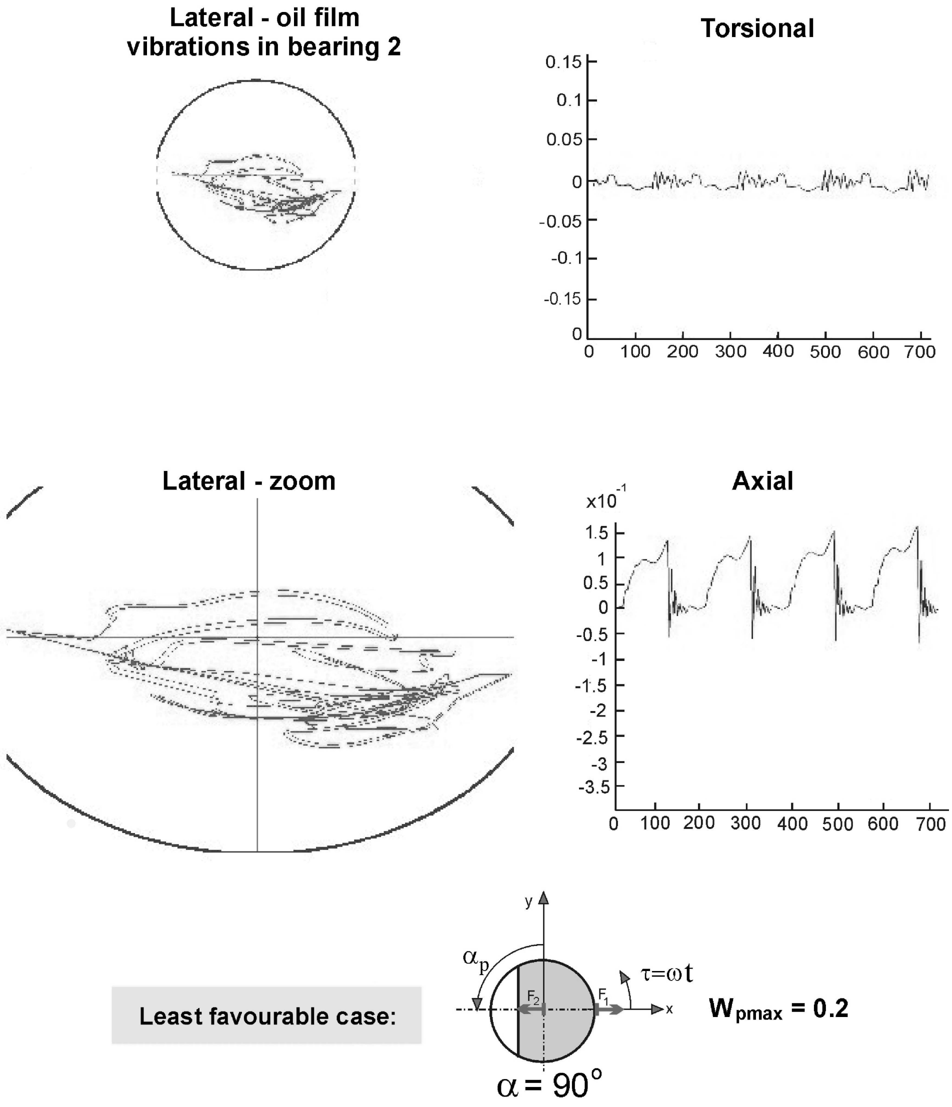

Fig. 11. Coupled vibration forms caused by the crack in resonance range $N = 2600$ rpm (stable range of system operation). Circumferential crack position $\alpha_p = 90$ – Fig. 5.

tem was not subject to any forces in axial and torsional directions. The observed effect is connected with strong structural nonlinearities generated by the crack in resonance conditions.

Will the same effect take place in the second critical case of machine operation, i.e. when the stability limit is exceeded? The results shown in Fig. 13 are surprising. The effect of circumferential position of the crack has practically disappeared, and the increase of whirl trajectory amplitude is almost proportional to the depth of the crack.

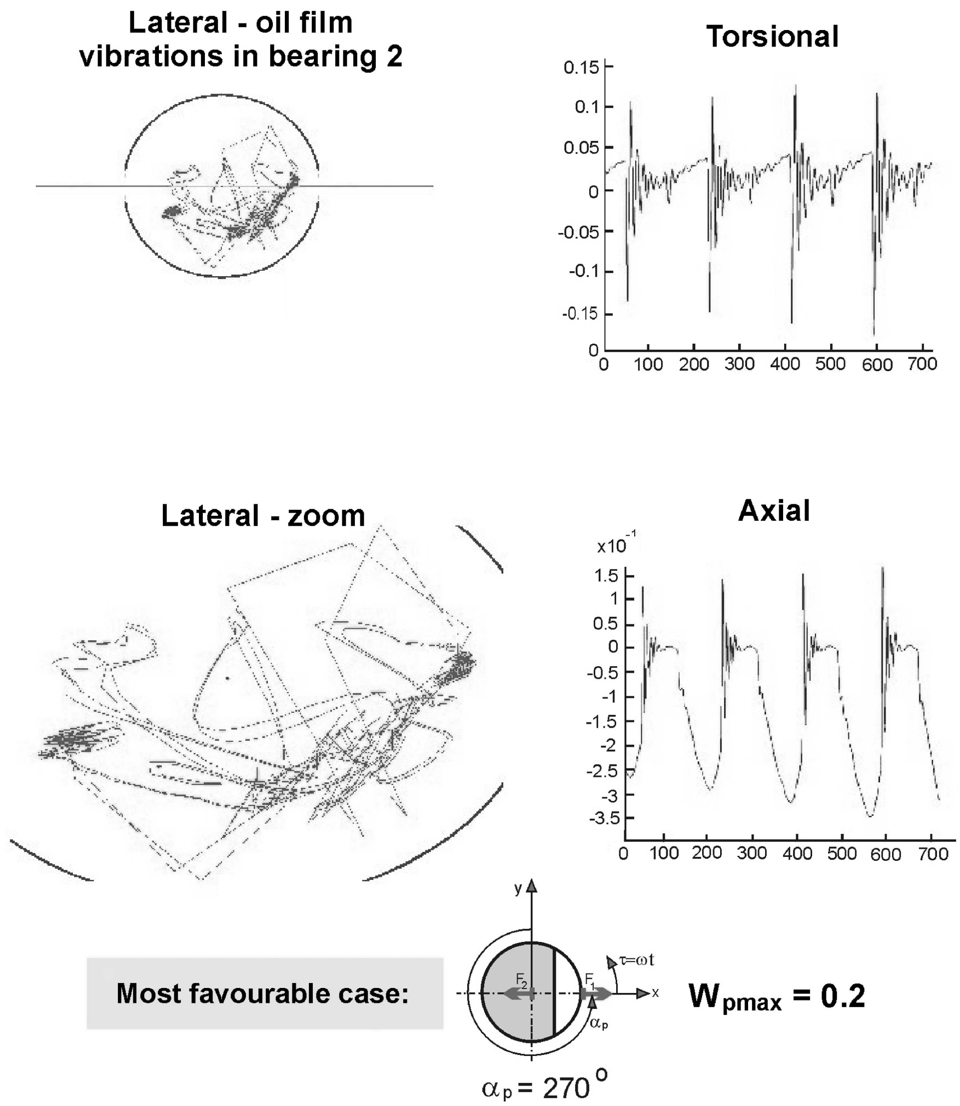

Fig. 12. Coupled vibration forms caused by the crack in resonance range $N = 2600$ rpm (stable range of system operation). Circumferential crack position $\alpha_p = 270$ – Fig. 5.

So the course of crack propagation is smoother than expected in this case. This regularity can be explained by the fact that the hydrodynamic instability (whirls) in initial stages of its development generates and supports relatively small amplitudes of bearing journal displacement trajectories. The situation looks different in the resonance and when the whipping of the lubricating liquid takes place. The effect of crack depth increases with the increasing amplitude of system vibrations, irrelevant of the range of machine operation.

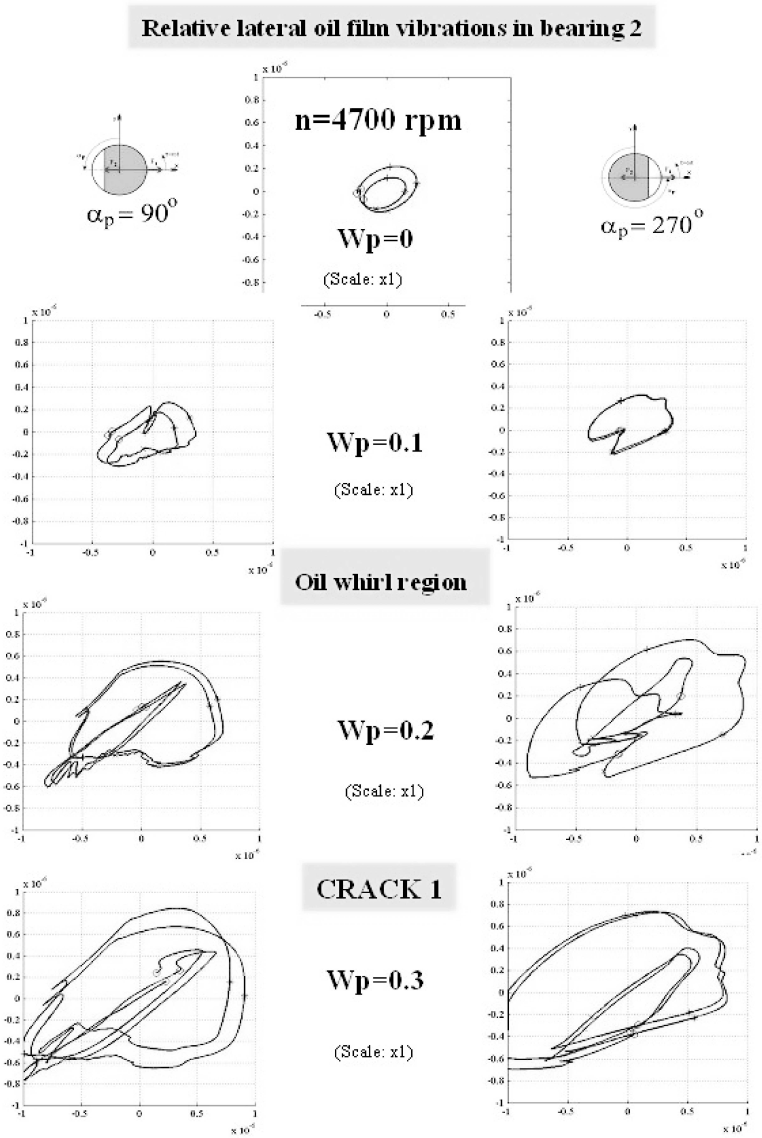

Fig. 13. The effect of rotor shaft crack propagation on the shape of lubricating liquid whirls, for two circumferential crack positions (after exceeding the stability limit $N = 4700$ rpm).

4. Concluding remarks

The performed investigations have proved that the slide bearings in the microturbine strongly react to operating phenomena such as thermoelastic bush deformations and possible material imperfections of rotor shaft crack type, which are typical for those machines. However, their effect is different than that expected by "engineering intuition”. The expected stabilising effect of partial fixing of the lower bush combined with clamp of the upper bush (being the source of a characteristic stabilising pit in the clearance gap) has not found confirmation here.

A similar situation is observed when examining the effect of rotor shaft crack propagation on the dynamic state of the machine. In resonance conditions this effect is so strong that it causes coupled forms of vibrations of the entire system. On the other hand, directly after the stability limit is exceeded, the effect of crack propagation is much smoother, which was difficult to expect.

Generally, the developed research tools composing the computer system MESWIR allow the dynamic state of microturbines to be analysed in extreme conditions of their operation, i.e. in the conditions of system stability loss or strong resonance. Consequently, vibroacoustic threats generated by home power appliances can be assessed using advanced computer analysis.

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