BULLETIN OF THE POLISH ACADEMY OF SCIENCES
TECHNICAL SCIENCES
Vol. 52, No. 3, 2004

The flutter effect in rotating machines

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Abstract. The considerations presented in the paper relate to one of the most intriguing phenomena, which is the development of oil whirls and oil whips in rotors with journal bearings. This effect is sometimes referred to as flutter, as its origin is in some relation to self-exciting vibrations of the system. Despite the fact that the flutter has been an object of investigation in numerous research centres all over the world, its nature has not been sufficiently recognized yet. The present paper delivers a description of particular phases of development of the hydrodynamic instability and proposes diagnostic determinants for this state. The object of investigations also included bearings with hybrid lubrication and siphon pockets in the oil gaps. The answer has been received to the question whether the self-exciting vibrations in rotating machines can be avoided, or reduced by means of additional oil supply having the form of siphon oil.

Keywords: flutter, journal bearings, hydrodynamic instability, rotor dynamics.

1. Introductory remarks — research tools

Examining effects connected with self-exciting vibrations in such a system as a large rotating machine with journal bearings requires earlier preparation of very advanced research tools, as the examined effect is of strongly non-linear nature that leads to the generation of new forms of vibrations. The most serious origin of this effect is the system of journal bearings, which under certain conditions generate so called negative damping and intensify vibrations of the system, thus providing good opportunities for possible failure of the entire machine. The self-exciting vibrations of the system that are generated by journal bearings are referred to as oil whirls and whips, or flutter generated by hydrodynamic effects. This latter term is used as an analogy to the aerodynamic flutter of turbine blades or airplane wings. In case of its occurrence, the flutter of an airplane wing may result in total failure of the object within no longer than a few seconds. For large rotating or fluid-flow machines, such as power turbine sets, for instance, the developed flutter in the form of oil whips can destroy bearing joints and the machine rotor as a further result, within the time period between ten and hundred seconds, thus putting power station technical staffs life into jeopardy, not mentioning extremely high economic losses.

For the above mentioned reasons the effect of flutter cannot be examined using linear models, even those most advanced. This leaves aside the commonly used modal analysis and, as a consequence, all widely used commercial programmes such as ABAQUS, NASTRAN and many others. But, if we decide to use a non-linear model, we have to overcome an important obstacle, namely construct a combined model that includes all elements composing the object, as the superposition principle cannot be applied here anymore. The journal bearings and the remaining part of the machine, the rotor in the examined case, must form an integrated whole from the point of view of the adopted model and numerical calculations. Therefore, if we have decided to examine the nature of the hydrodynamic flutter and particular phases of its development, constructing a compact, non-linear model for a complex system including the bearings, the line of rotors and the supporting construction is becoming an absolute necessity.

Such a model and computer system has been constructed in IFFM PAS, Gdansk. Its essential component is a so called elastodiathermic thermal model of a journal bearing (the code DIATER) – Fig. 1 and its further extension in the form of the code IZOSLEW, which allows modelling of hybrid lubrication in the bearing, i.e. lubrication making use of siphon pockets and siphon pressure. The code makes it also possible to include axial dislocations of the bushings, i.e. assembly and/or operating errors resulting from non-parallel position of the journal with respect to the bushing in the axial direction. The diathermic thermal model assumes heat transfer between the oil film, the bushing, and the surroundings, hence it requires simultaneous solution of 3D Reynolds, energy and conduction equations [1,2]. The above bearing codes were coupled with those describing the line of rotors and the supporting construction, thus creating a complex and internally compact computer system, bearing the name of NLDW [3–8]. This system mutually couples complex structures of the entire object, generating, as a result, global equations of motion for the adopted system (i.e. sets of mutually conjugate non-linear differential equations) — Fig. 2. Certainly, the only way leading to the numerical solution is an iterative procedure — Fig. 3. The NLDW system also allows modelling of constructional and operating imperfections, like shaft misalingment or axial dislocation of the line of rotors — Fig. 4. For obvious reasons, details of the above models are not presented here. They can be found in earlier publications [1–8].

The above research tools allow modelling the dynamic behaviour of fluid-flow machines within full range of their operation, including the non-linear range, beyond the

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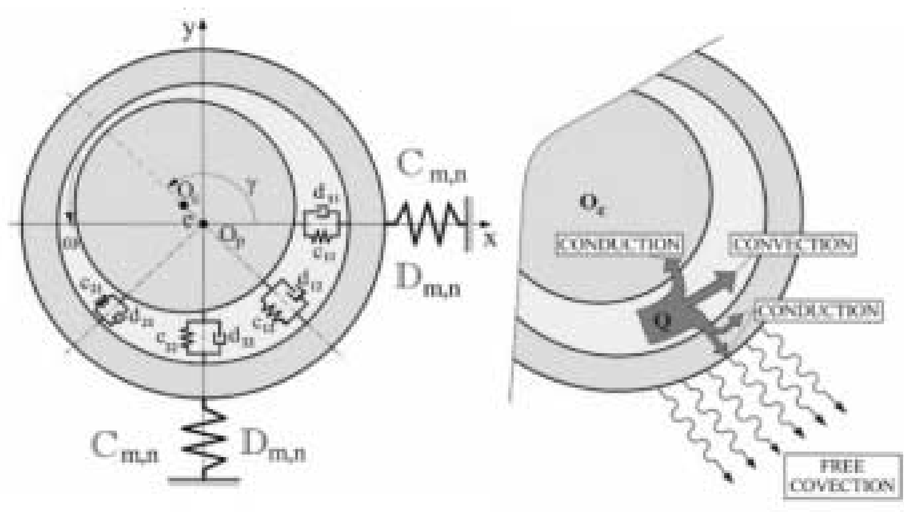


Fig. 1. Accepted 3-D elastodiathermic model of journal bearings

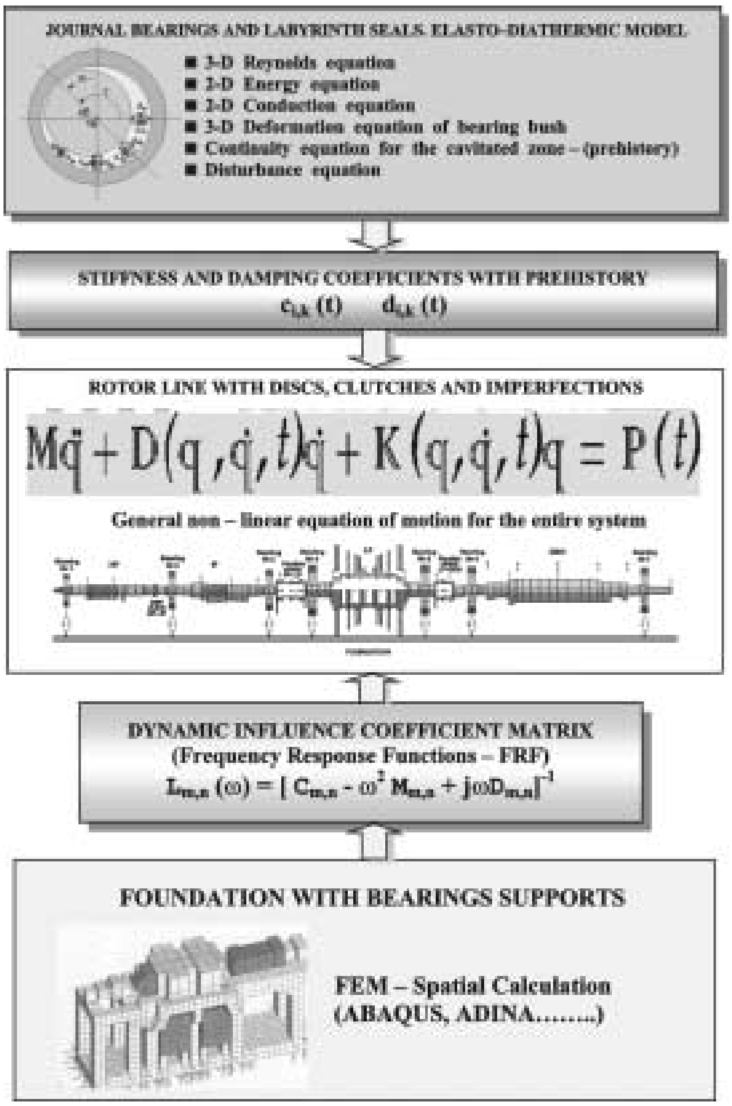


Fig. 2. Basic structural components of a system created by line of rotors, journal bearings, and the supporting construction. Non-linear couplings of particular sub-structures are obtained using journal bearing stiffness and damping coefficients, changing in time, and dynamic flexibility matrices of the supporting construction

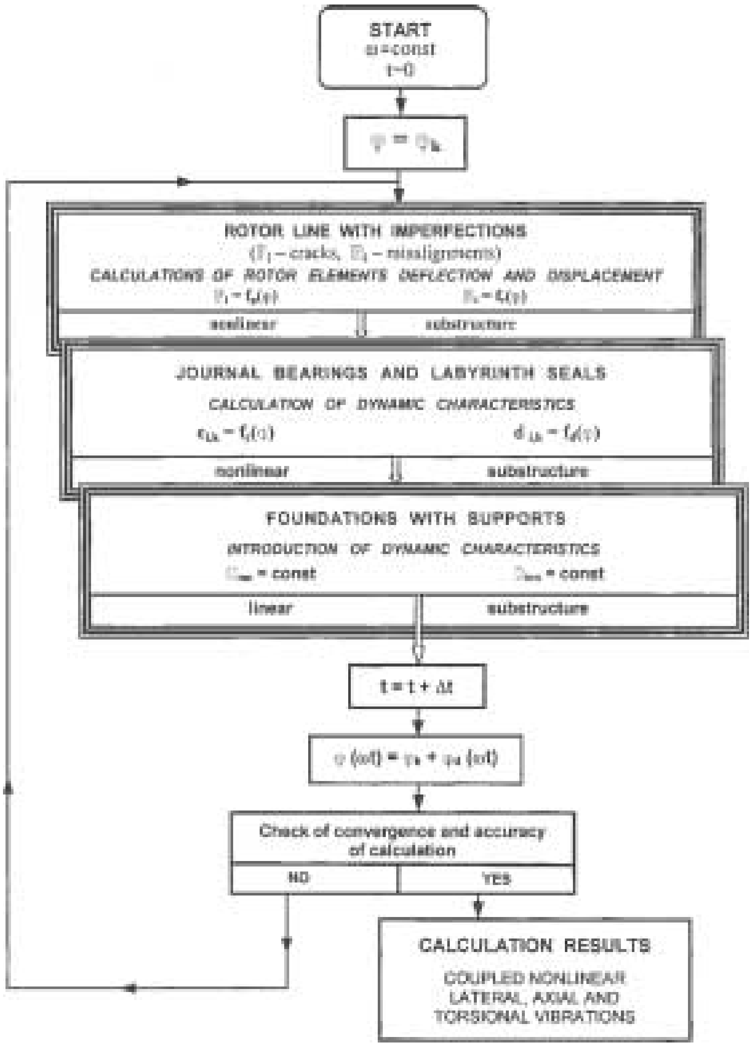
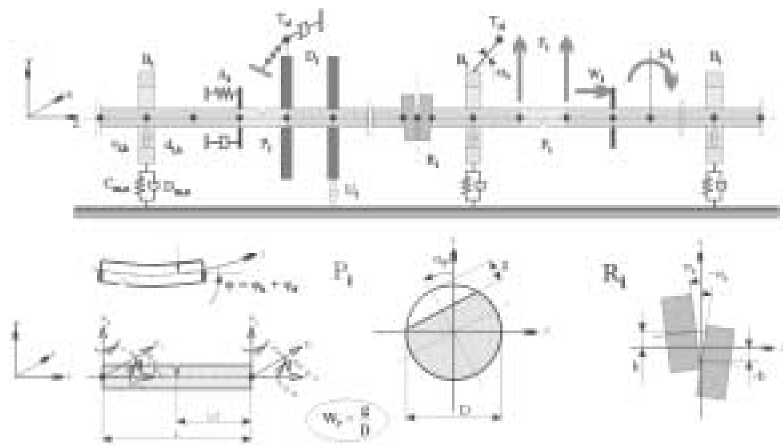


Fig. 3. The adopted algorithm of non-linear calculations



A_i – axial bearings
B_i – journal bearings
D_i – disks
F_i – external lateral forces
M_i – external torsional moments

P_i – rotor cracks
R_i – rotor misalignment
T_{si} – torsion dampers
T_{zi} – external rod dampers (mounted on the bearing bush)

U_i – labyrinth seals
W_i – axial forces
φ_i – kinetostatic rotor deflection angle
φ_i – dynamic rotor deflection angle

Fig. 4. NLDW computer system for non-linear analysis of rotor dynamics with imperfections

stability threshold of the system. This characteristic of the constructed tools (the NLDW system) is of great significance from the point of view of the goal of activities reported in the present paper, which was giving the explanation for the phenomenon of flutter, which in this case means the analysis of propagation of oil whirls and oil whips.

A separate issue is experimental verification of the NLDW system. In IFFM PAS, Gdansk, a multi-stage procedure was employed which included experimental tests performed on a research rig and on real objects (turbine sets of 200 MW power output). The country biggest experimental rig was erected for examining a rotor-bearings system — Fig. 5. Figures 6 and 7 show fragments of verification investigations of the NLDW system. Of special significance are the results obtained in the non-linear range of systems operation, i.e. in the area where the hydrodynamic instability has the form of oil whirls. The effect of phase exchange on internal and external loops has found its experimental confirmation here. The more detailed explanation is in the next sections.

Fig. 5. Multi-scale research rig used for experimental verification of NLDW system

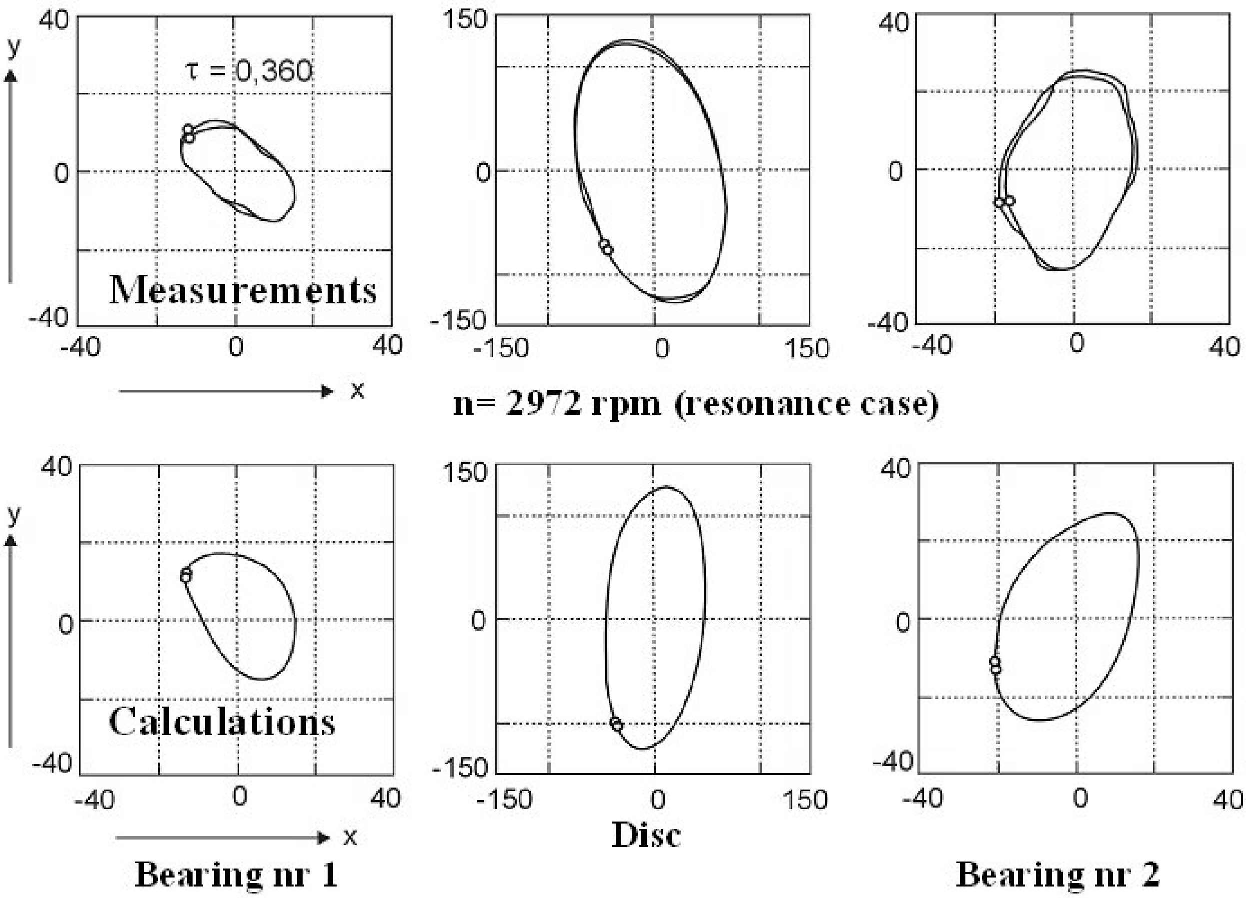


Fig. 6. Example of verification of the NLDW system on the multi-scale research rig within the range of stable operation (in resonance)

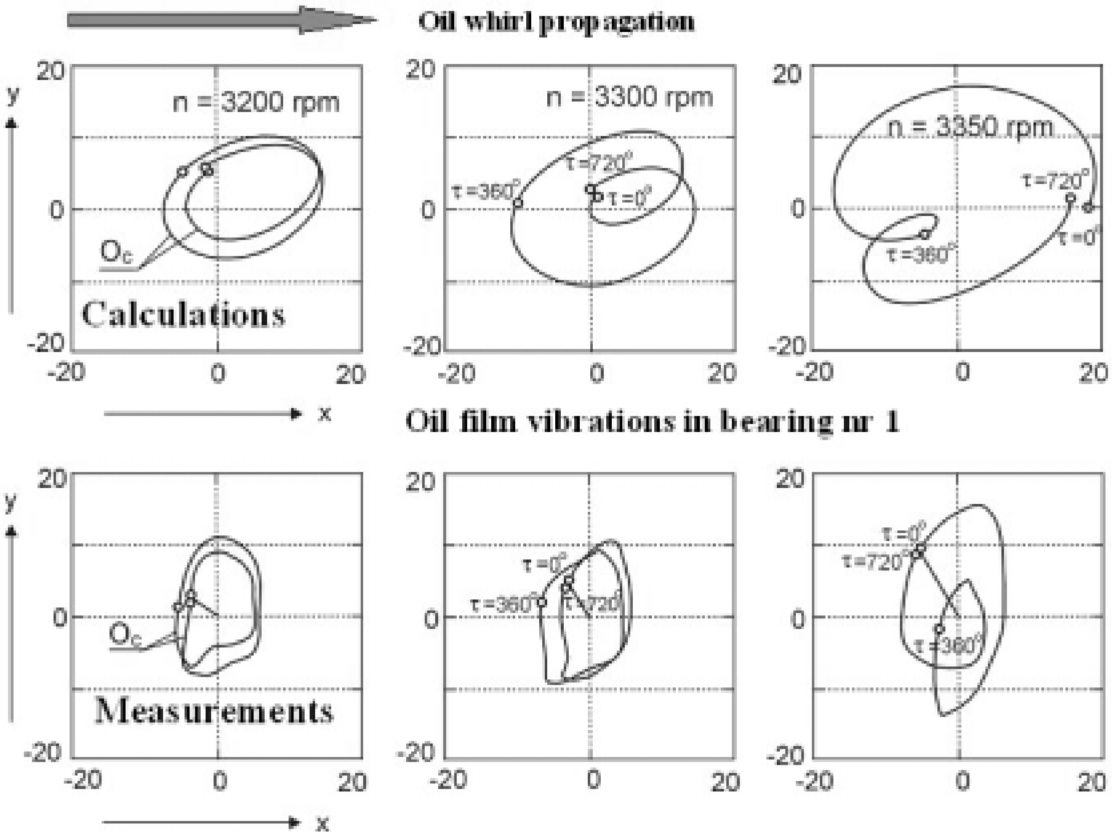


Fig. 7. Example of verification of the NLDW system on the multi-scale research rig after surpassing the stability threshold (within the range of oil whirls)

2. Development of oil whirls and whips during hydrodynamic lubrication

An example of verification of the NLDW system shown in Fig. 7 refers to experimental measurements and computer simulation of oil whirls forming in the journal bearing on the research rig. It is well known that the oil whirls come into existence when the stability threshold is surpassed by the system, i.e. within the range where traditional method of linear modelling of the phenomenon fails. Here, we present a computer analysis of qualitative transition from classical vibrations with elliptical trajectory shape to those revealing whirl structure. The advanced computer analysis also allows separating characteristic diagnostic determinants of this state and gaining better insight into the physics of the phenomenon. Without this kind of simulations, the evaluation solely based on experimental investigations would be more difficult, often completely impossible, and, last but not least, much more expensive.

Let us carry out a bit more systematic study of propagation of oil whirls. Figure 8 presents an object selected for investigations along with the MES discretisation and node numbering. We observe the development of oil whirls and whips in bearing No. 1 when the revolutions of the rotor increase after surpassing the stability threshold. The system is subject to action of external forces resulting from the residual unbalance of the disk. Basic characteristics of the bearings are the following: journal diameter — 0.1 m, radial cylindrical clearance — 90 μm, bushing width/journal diameter ratio — 0.5, lubricating agent — machine oil Z-26. The results of computer simulation carried out with the aid of the NLDW system are shown in Figs. 9–15.

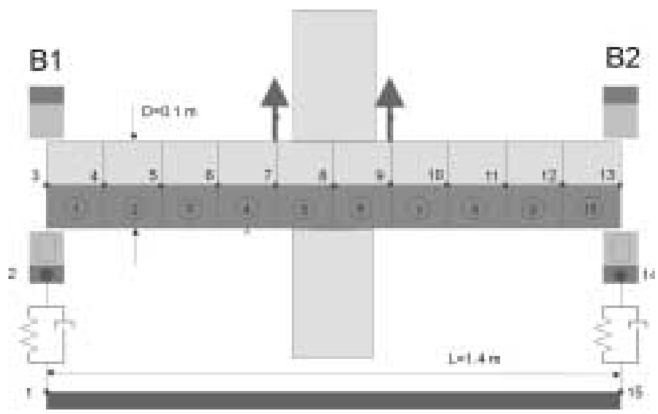


Fig. 8. The object used for investigations, (two-support model of the rotating machine with journal bearings), along with MES discretisation and node numbers

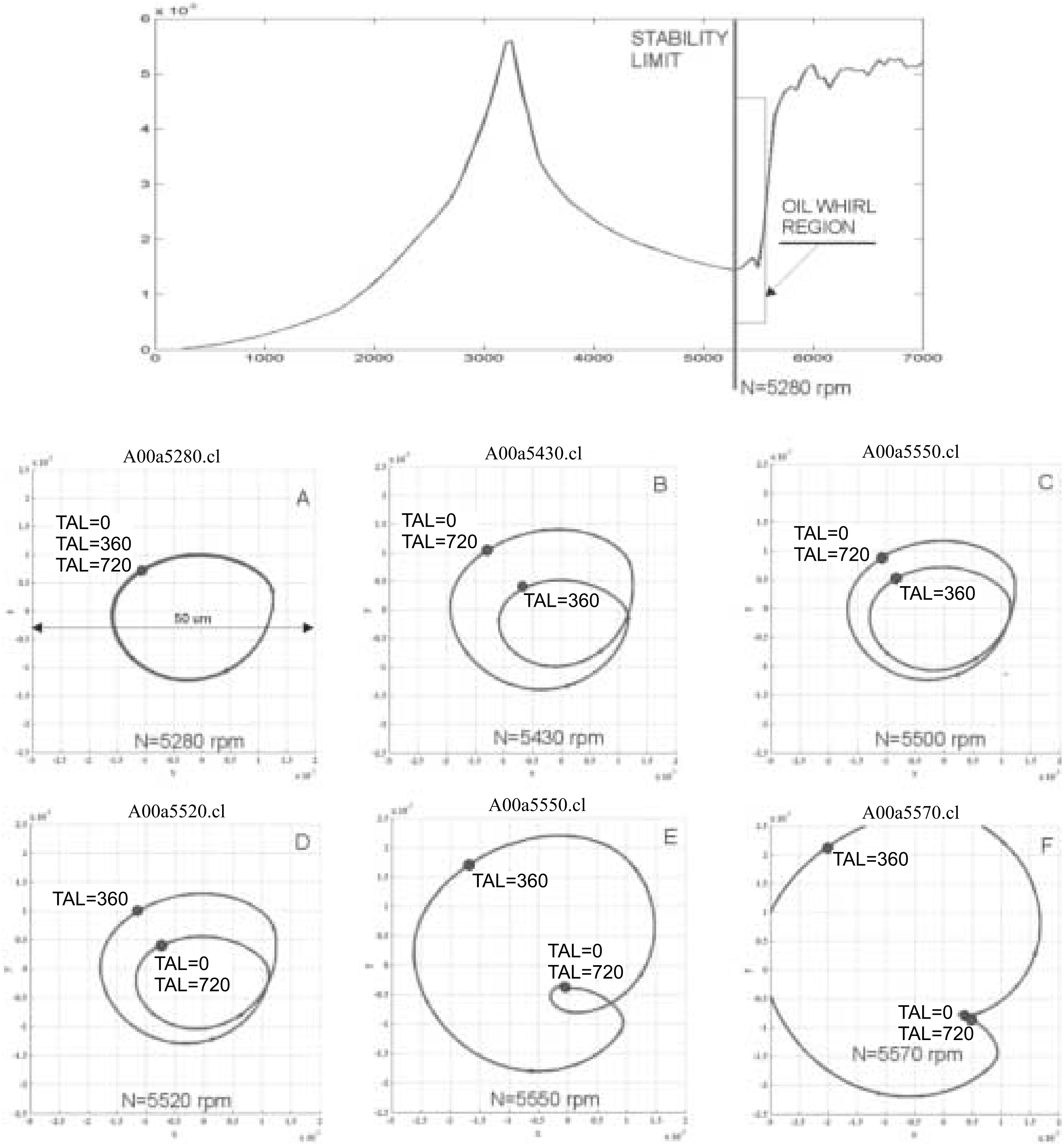


Fig. 9. Computer simulation of the development of oil whirls after surpassing system stability threshold – phase of small oil vibrations. Calculations were carried out using NLDW system

An interesting remark resulting from Fig. 9 is that the oil whirls develop by slow splitting of the elliptical trajectory into two loops: external and internal. During the first phase the internal loop decreases, then it begins to increase again, and finally changes places with the external loop. The initial external loop decays, and in the final whirl phase we have only one trajectory with the shape close to a circle. The whirls move on to the next, much more dangerous phase, which is oil whipping.

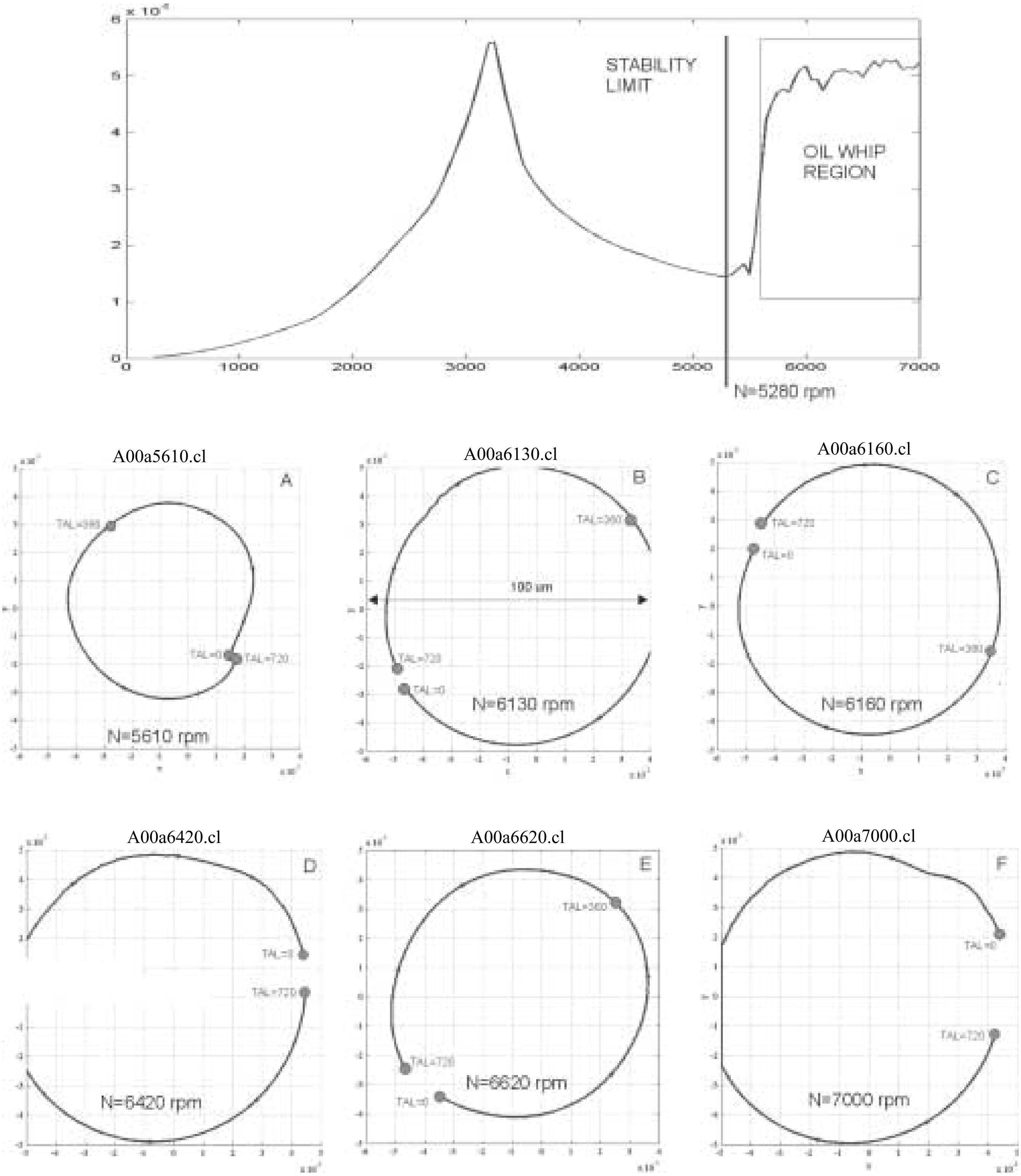


Fig. 10. Simulation of the transition of oil whirls to the oil whipping phase. Highly developed hydrodynamic instability – strong and dangerous oil vibrations

This situation is illustrated in Fig. 10. The observation of phase markers, i.e. positions on the trajectories corresponding to external excitation force vectors directed horizontally right in the assumed reference system (TAL=0, 360, or 720 degrees) delivers practical data on the diagnostic factor referring to the hydrodynamic instability, as shown in Figs. 11 and 12.

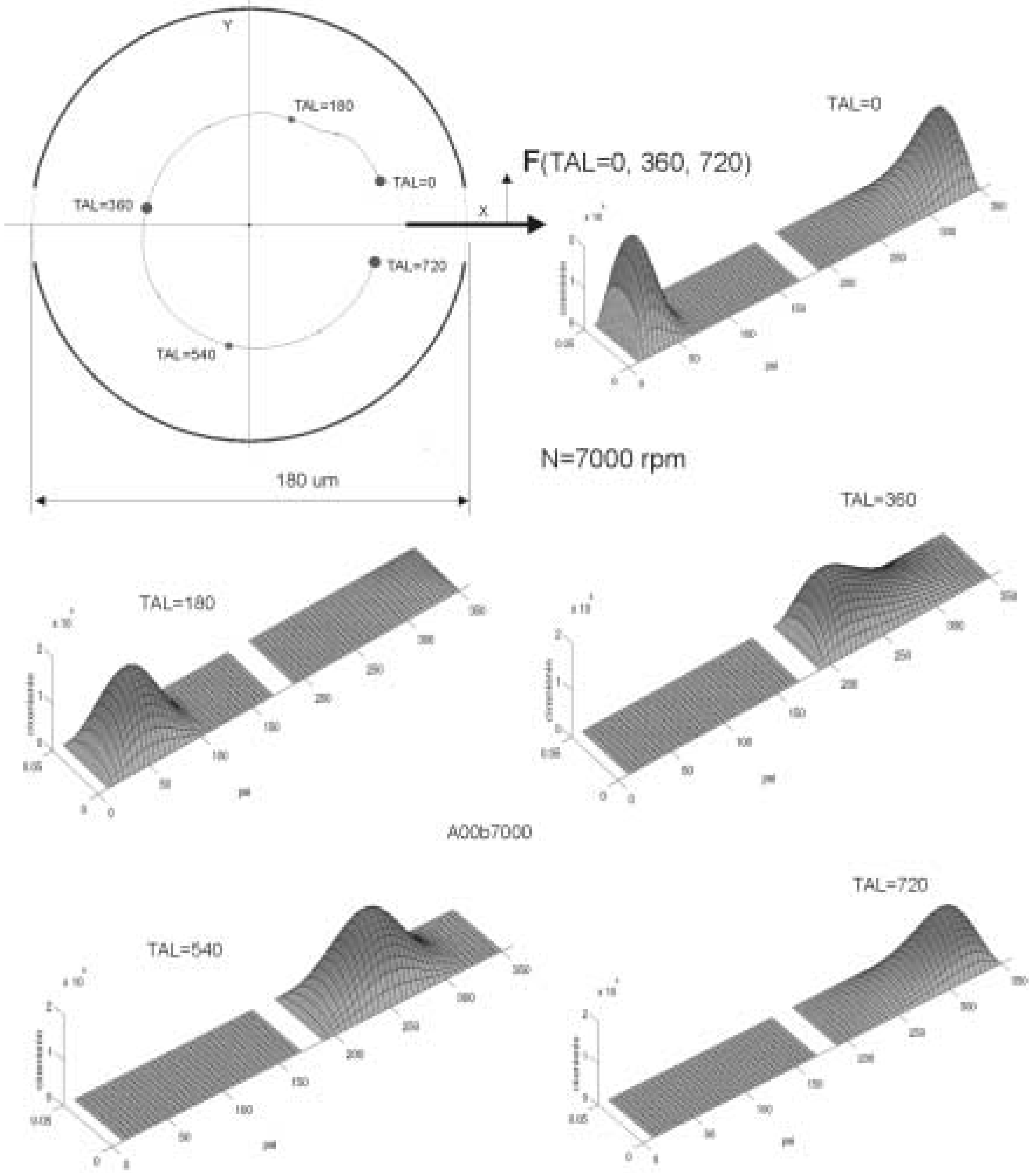


Fig. 11. Instantaneous hydrodynamic pressure distributions in the bearing for selected journal positions on the trajectory within the oil whipping range. The trajectory is presented on the background of the bearing clearance circle

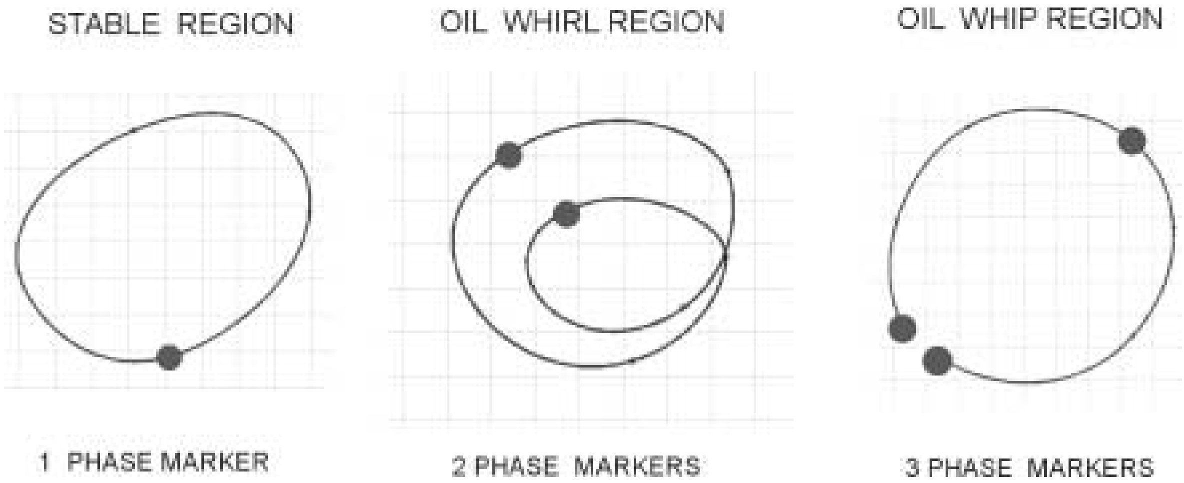


Fig. 12. Proposed classification of hydrodynamic instabilities in the system and introduction of diagnostic determinants

It results from Fig. 11 that within the examined range of recorded rotor revolutions $\langle 0, 720\rangle$, i.e. for TAL=0, 360 and 720, in the case of well developed oil whips, the same position of the exciting force vector, horizontal left, for instance, corresponds to three different pressure distributions and, as a result, three completely different dynamic conditions of bearings operation. That means that this state is characteristic of having as many as three phase markers within the recorded trajectory range between 0 and 720 degrees.

In this convention the oil whirls are represented by two phase markers, while the range of stable operation of the machine — one marker. This situation is illustrated in Fig. 12. The conclusions resulting from the analysis of Fig. 12 can be of great significance for monitoring hydrodynamic instabilities, as they deliver practical measure of this type of states in the form of a number of phase markers. Obviously, other, additional diagnostic factors can be named which are specific for oil whirls and whips (vibration spectra, for instance), but they go beyond the scope of the present paper and cannot be discussed due to its limited volume.

3. Development of oil whirls and whips during hybrid lubrication

We are going to move on now to the research work concerning hybrid lubrication and the description of propagation of oil whirls in this case. The author of the present article is not familiar with any literature publications dealing with this topic. Let us assume that the lubricating gap has three siphon pockets, through which the siphon oil is delivered, or not — Fig. 13 upper left-hand part. It is noteworthy that the presence of siphon pockets themselves can considerably change lubricating conditions, and, as a consequence, affect the development of oil whirls. The research has been carried out for two variants: zero siphon pressure (with the presence of the siphon pockets), and relatively high siphon pressure PLEW=1 MPa. The distributions of the hybrid pressure for those cases, along with actual positions of the journal on the background of the clearance circle, are given in Fig. 13. As can be noticed, the “hydrostatic component” has become dominating. For such a small relative eccentricity of journals position in relation to the bushing EPS=0.02 we can assume that we deal with the hydrostatic bearing. The mentioned above PLEW pressure was purposely assumed at such a high level to allow studying extreme cases. Will the propagation of oil whirls in the hydrostatic bearings be similar? Is it possible to eliminate, or to reduce this phenomenon using siphon pressure? The answer to those questions is given in Figs. 14 and 15.

Figure 14 presents the collections of three development phases of oil whirls for three cases: without siphons (A, B, C) with siphons but without siphon pressure (D, E, F) and with siphons and siphon pressure PLEW=1 MPa (G, H, I). The figure reveals that in the all examined cases the nature of whirl development remains similar, although the amplitude of oil vibrations is visibly reduced in cases of hybrid lubrication. Here, an important conclusion can be formulated that the siphon pressure reduces the amplitude of oil whirls, but does not eliminate them to the full. Figure 15 explains why that happens in such way. While in static bearing load conditions the hydrostatic lubrication has a dominating effect (Fig. 13), in the situation of dynamic load and developed oil whirls the essential hydrodynamic part of pressure in the oil film builds up, due to large and rapid dislocations of the journal, and controls the development of whirls in the already well-known form. The hydrodynamic contribution of pressure is clearly visible when studying pressure distribution changes for selected time instants — Fig. 15.

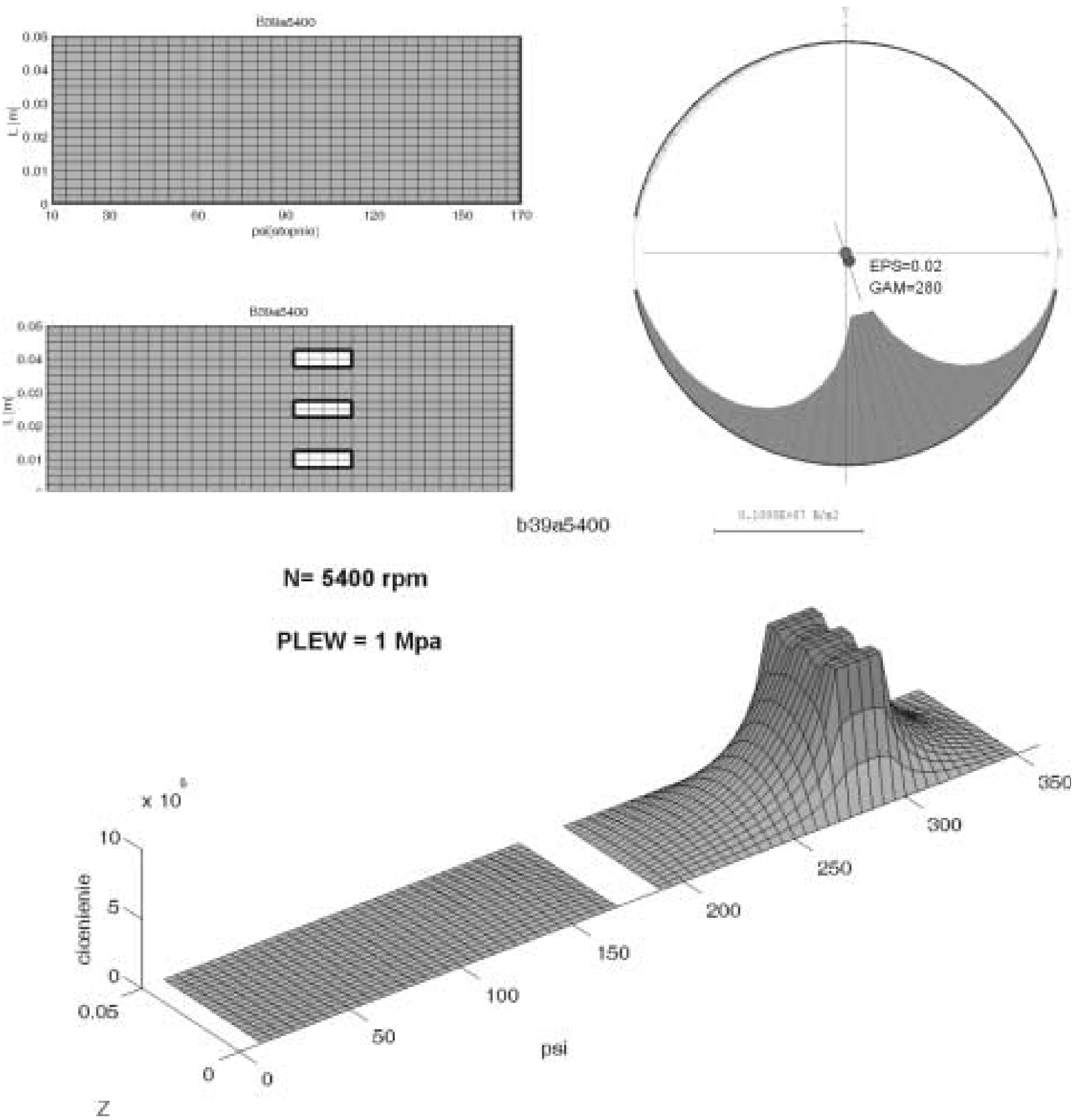


Fig. 13. Examples of hybrid lubrication. The bearing is supplied by three siphon pockets under pressure 1 MPa. Calculations performed using KINWIR code (part of NLDW system) for rotational speed N=5400 rpm

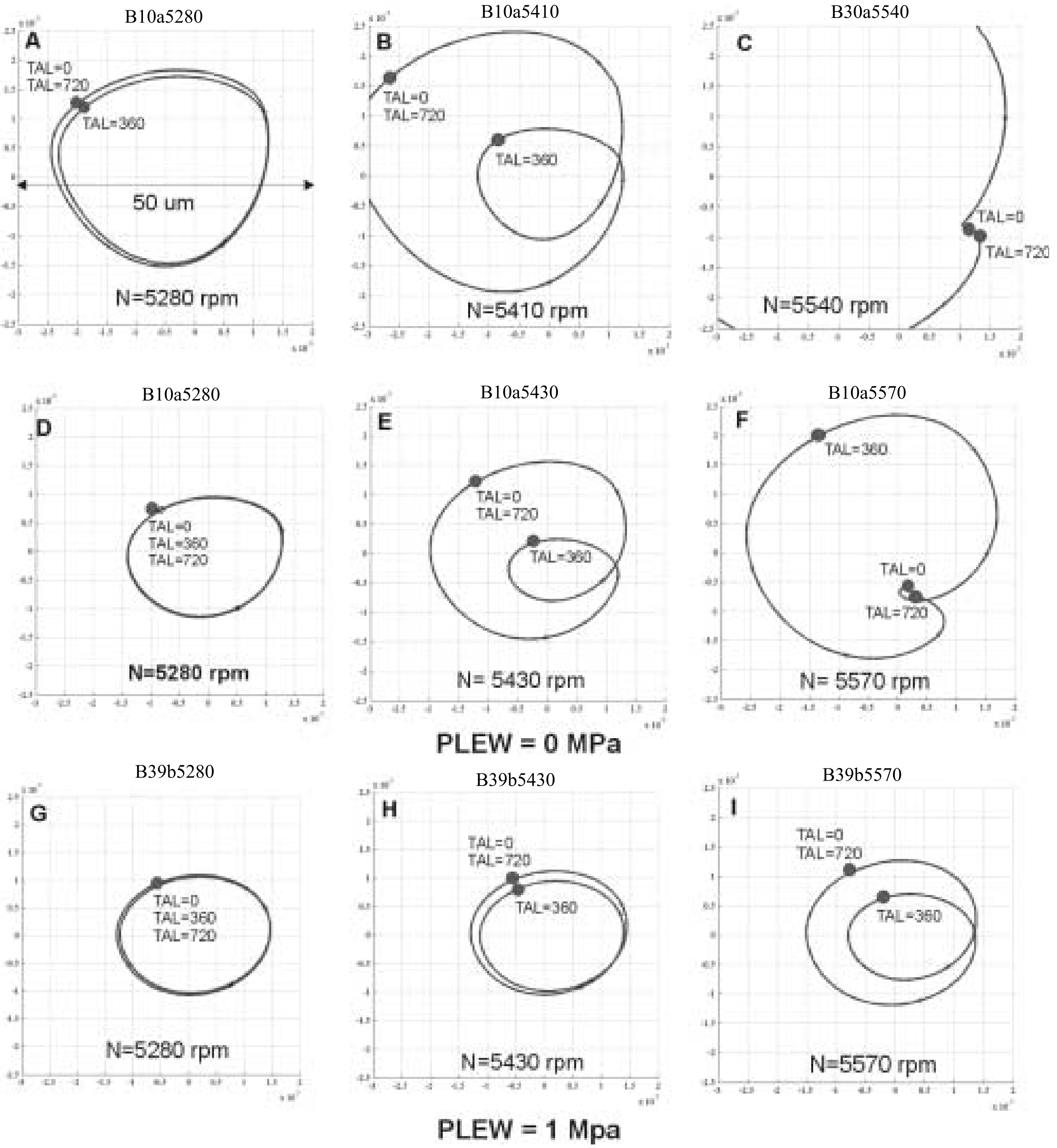


Fig. 14. Oil whirl propagation for three configurations of the oil gap: A, B, C – hydrodynamic lubrication without siphon pockets, D, E, F – hydrodynamic lubrication with three siphon pockets, without siphon pressure (PLEW=0), G, H, I – hybrid lubrication by siphon pressure PLEW=1 MPa

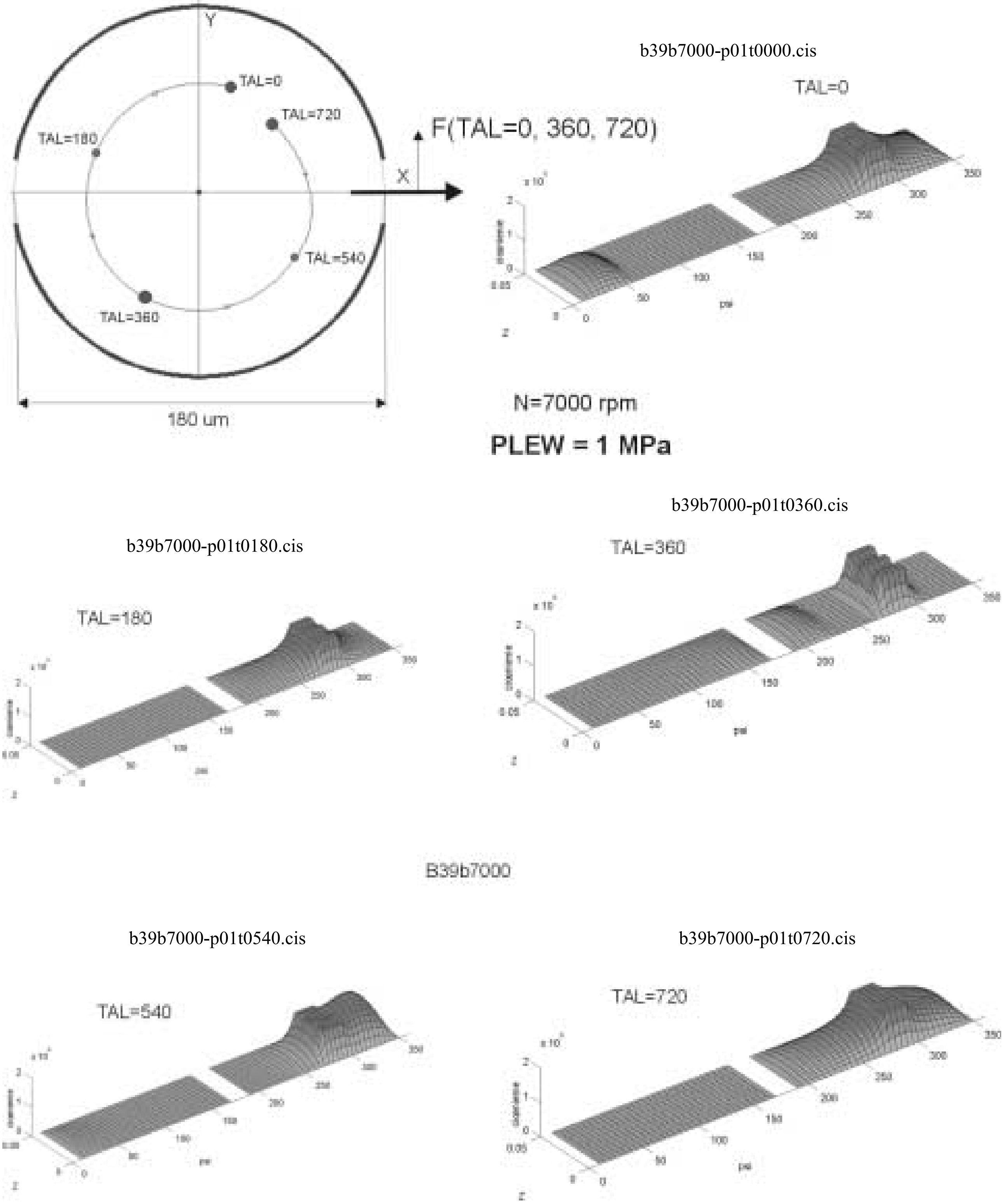


Fig. 15. Simulation of the pressure distribution within the range of oil whips (N=7000 rpm) by hybrid lubrication PLEW=1 MPa. The picture presents pressure distributions for selected positions of the journal

4. Concluding remarks

The presented examples of the analysis of oil whirl and whip propagation make only small part of research activities carried out in IFFM PAS in the area of rotor and journal bearing dynamics. A systematic study of those still intriguing phenomena has led to the conclusion that hydrodynamic instability in the system cannot be easily removed in earlier assumed operating conditions of the machine. Hybrid lubrication, in particular extremely high siphon pressure may reduce the scale of this phenomenon in the aspect concerning the amplitude of vibrations, but not totally eliminate it. The whirls can very fast reach the oil whipping phase, characteristic of high and dangerous amplitudes of vibrations. If we introduce the signal recording system for two last revolutions of the rotor shaft, i.e. within the range of $\langle 0, 720\rangle$ then we will be able to introduce characteristic diagnostic determinants of the hydrodynamic instability, having the form of numbers of revolution markers. Similar advanced analyses of normal and abnormal states recorded in objects of various types are expected to be carried out in the future, thus defining future directions of research development in technical diagnostics. This fact will affect present-day challenges, not only in diagnostics, but also in the entire field of knowledge relating to machine engineering.

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