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New method of state analysis and diagnostics of power micro-devices

K e y w o r d s

Co-generation, Micro-turbines, Rotor dynamics, nonlinear vibrations, hydrodynamic instability, heuristic problems, computer simulation.

S ł o w a k l u c z o w e

Heurystyka w eksploatacji maszyn, kogeneracja rozproszona, niestabilność hydrodynamiczna, mikroturbiny, zagadnienia heurystyki, symulacje komputerowe.

S u m m a r y

In the small scale, turbines and bearings are a source of specific problems connected with securing stable rotor operation. Two kinds of high-speed micro-turbines of electric power of about 3 KW with multistage axial and radial rotors supported on foil bearings have been accepted. The scope of the present article is limited to the discussion of the 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. Analysing the dynamic state of the domestic power plants requires qualitatively novel research tools. It turned out that the situation is possible that the rotor, after losing its stability, stabilises again when the rotational speed increases. This is a phenomenon determined by the author as "multiple whirls."

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Another topic discussed in this paper is an assessment of the influence of a random character of certain input data, in this case, changes of external excitations of the system. This problem is related to the "heuristic models" often placed in opposition to widely used algorithmic models.

Introduction and research tools

The counterparts of large power turbines in distributed generation are small steam turbines or micro-turbines (CHP) that operate in an organic Rankine cycle (ORC) using low-boiling agents. Micro CHP units dedicated for individual households of total heat capacity up to 20kWt and electric power up to 4kWe are currently being designed at IFFM PAS. Safe operation of these 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**.** This means operating conditions in which a real vibroacoustic threat for these machines can have place. As we can see, analysing the dynamic state of the domestic power plants requires qualitatively novel research tools.

The MESWIR computer code, based on nonlinear models of complex system rotor-bearings, was applied in research. Theoretical models, basic equations, and the system itself have already been presented at several conferences and in publications [1, 2, 3]. Therefore, having a limitation on length and having other aims, the MESWIR series code will not be presented here in detail. However, it is worth mentioning that the most useful feature of this system is the possibility of the description of the rotor machine state both in a linear and nonlinear range by means of the same tool, thereby describing new vibration forms at the transition of the stability limit. The description capability of bearings of a complex geometry of oil clearance including foil bearings is also important. The MESWIR code was experimentally verified both at the research stand and using real systems such as large power turbo-sets [1].

Objects of investigations

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 device is most often coupled with boilers supplied with renewable energy sources.

A concept of such micro-power plant developed in the IFFM PAS in Gdansk is shown in Fig. 1 [4, 5, 6]. Essential elements of the micro turbine constitute slide bearings of special characteristics ensuring a high stability of the system.

The rotors are supported on foil bearings (Fig. 2) 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. 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 [6]. The analysis of the bearing operation took into account the interaction between the medium that lubricates the bearing and the set of foils.

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

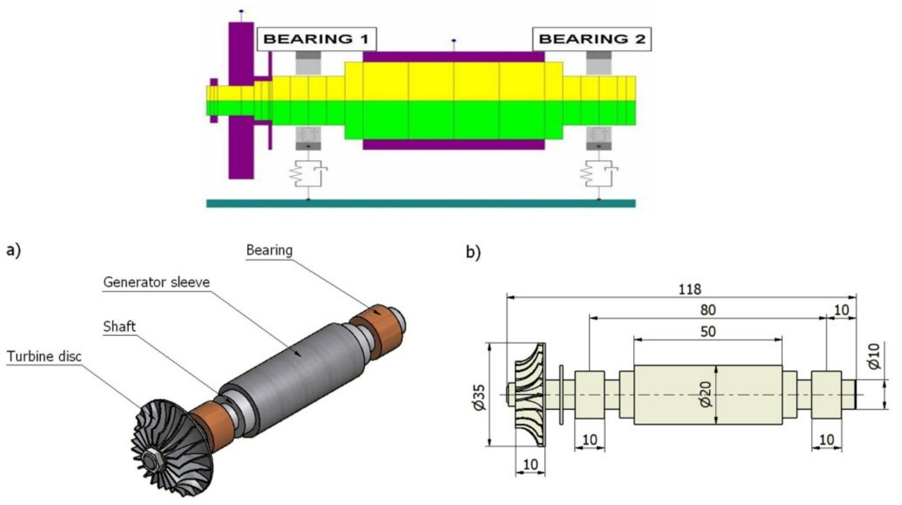

Fig. 1. Object of testing. Micro turbine developed in the IFFM PAS in Gdansk (3 KW of Power) and MES discretisation. Single-stage radial-axial turbine for low-boiling medium ORC and a rotational speed up to 100 000 rpm [4, 5, 6]

Rys. 1. Przedmiot badań. Mikroturbina opracowana w IFFM PAS w Gdańsku (3 kW) oraz dyskretyzacja MES. Jednostopniowa turbina promieniowo-osiowa dla cieczy niskowrzącej ORC obracająca się z prędkością 100 000 obr/min [4, 5, 6]

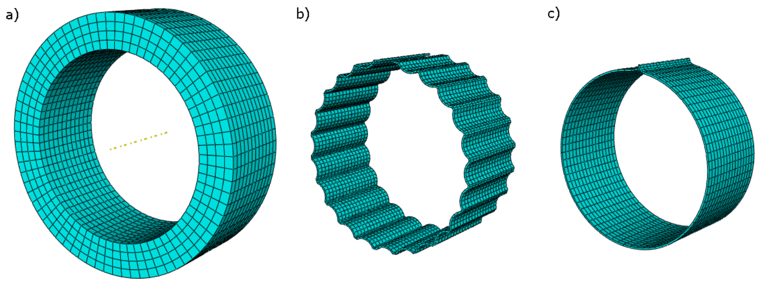

Fig. 2. FEM models of foil bearing elements (a – sleeve, b – bump foil, c – top foil) [6]

Rys. 2. Modele FEM elementów łożyska (a – obudowa, b – folia sprężysta, c – folia ślizgowa [6]

Phenomenon of multiple whirls

The calculation results in the form of amplitude–speed characteristics within the speed range up to 100 000 rpm are presented in Fig. 3.

Attention is focused on the unique 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 the bearing clearance.

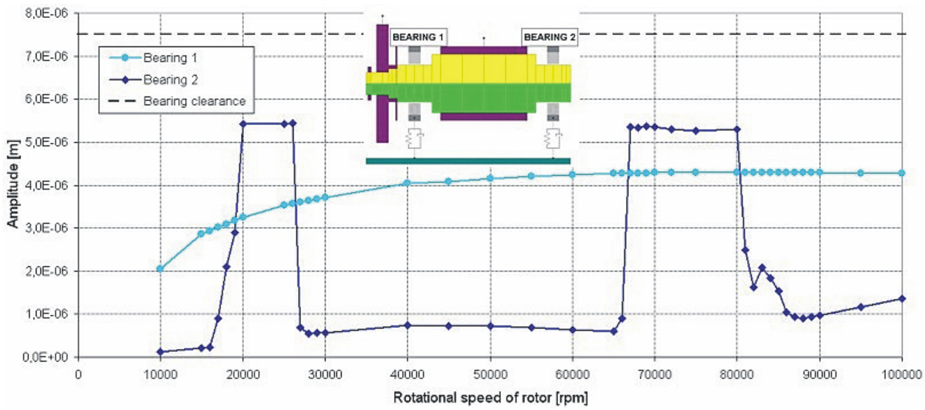

Fig. 3. Amplitude – speed characteristics of the rotor from the micro turbine (shown in Fig. 1) calculated for the relative vibrations of a journal and foil-bearing bush. Illustration of the multiple whirl phenomenon in bearing No. 2 (repeated processes of formation and decaying of high amplitude zones caused by a hydrodynamic instability)

Rys. 3. Charakterystyki amplitudy w funkcji prędkości wirnika dla mikroturbiny pokazanej na rys. 1, obliczone jako względne drgania czopa i folii. Ilustracja „wirów wielokrotnych” w łożysku nr 2 (powtarzające się procesy powstawania i zanikania stref drgań spowodowanych niestabilnością hydrodynamiczną)

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 entire first zone and the transition are presented in Fig. 4.

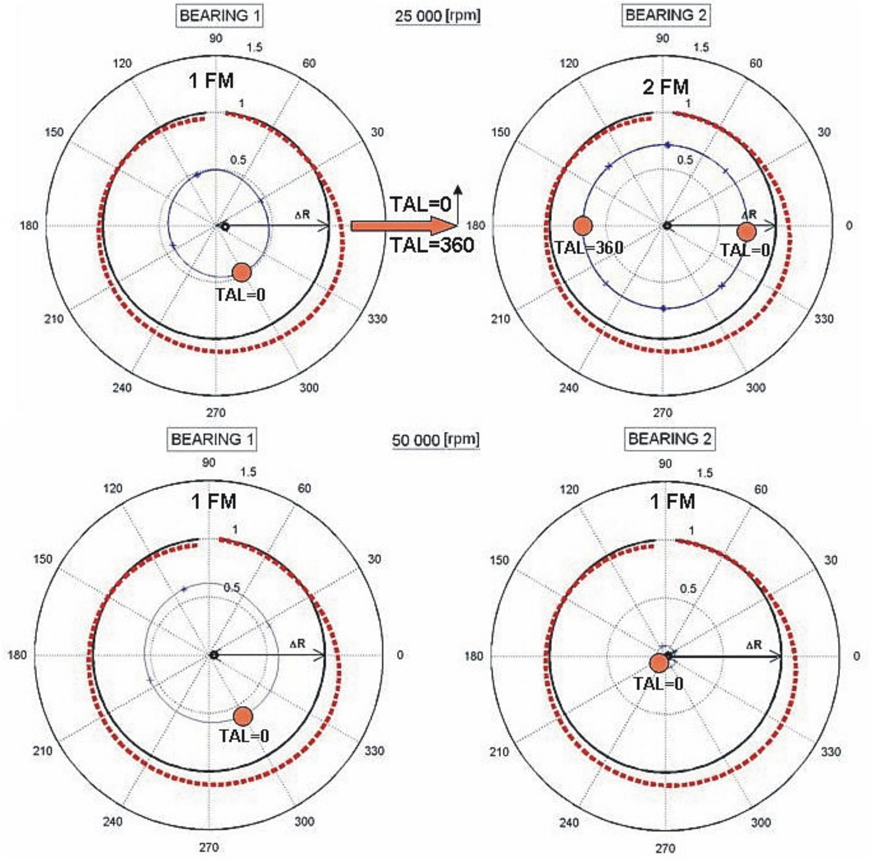

Fig. 4. Displacement trajectories of journal of bearing No. 1 and 2 calculated for the first high amplitudes zone (25 000 rpm) and in the transient period (50 000 rpm). Image of the first hydrodynamic ‘whip’ in the trajectory – 2 phase markers FM (upper right-hand side trajectory). Broken line indicates deformations of the bearing inner foil

Rys. 4. Trajektorie ruchu czopa łożyska nr 1 i 2 obliczone dla pierwszego zakresu wysokich amplitud (25 000 obr/min) i w zakresie przejściowym (50 000 obr/min). Obraz pierwszego hydrodynamicznego „zawirowania” ruchu – 2-fazowe markery FM (prawa górna strona trajektorii). Przerywana linia wskazuje odkształcenie folii ślizgowej łożyska

Analysis of Fig. 4 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, which is called “whip.” In this case, the “whip” means a developed form of whirls of a lubricating medium within the lubrication clearance. This is pointed out by double shaft rotations (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.

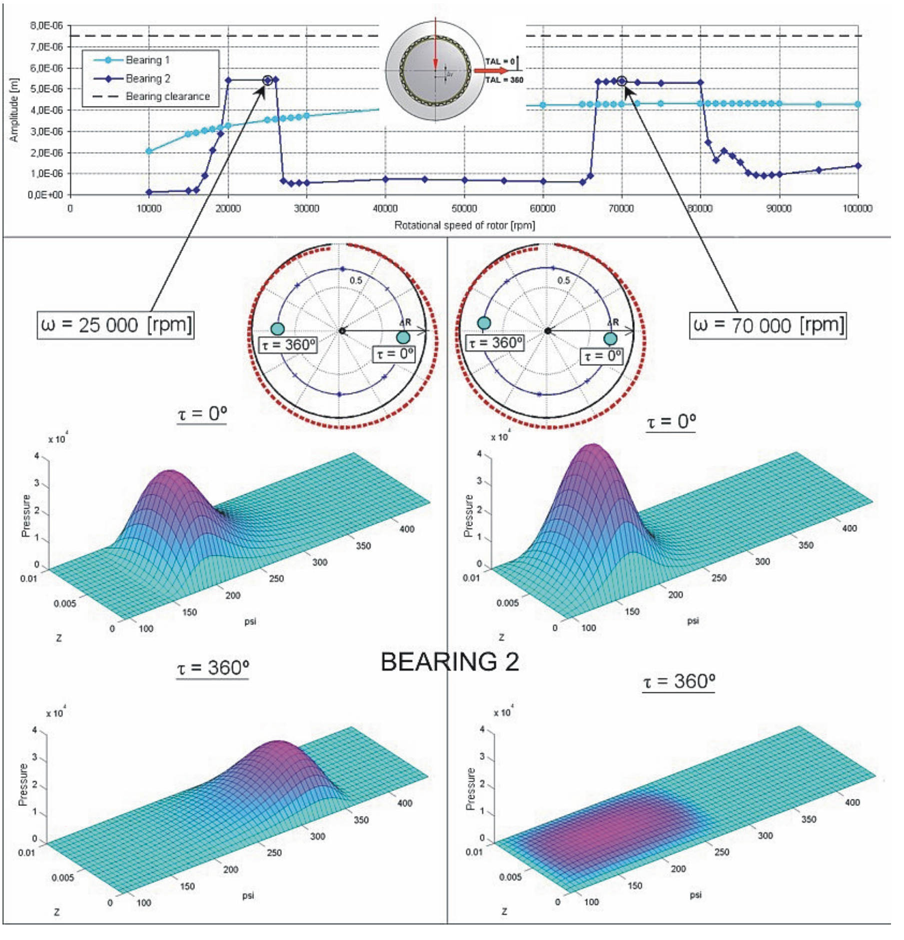

Fig. 5. Pressure distribution calculated for various positions of phase markers within a zone of multiple whirls for the same position of the excitation force vector (horizontally to the right, it means for TAL = 0 and 360 degrees)
Rys. 5. Rozkład ciśnienia obliczony dla różnych pozycji znaczników fazy w zakresie występowania zawirowań dla tych samych pozycji wektora siły nacisku (pozycja pozioma w prawo oznacza TAL = 0 lub 360 stopni)

However, the most unexpected observation is that, after the system has exceeded the first zone of hydrodynamic instability (the first ‘whip’), the system returns to a stable operation of bearing No. 2 (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.

If we assume a stiff bearing bushing, we are unable to model the phenomenon. This allows to assume that variable deformations of the foil-bearing bushing (corresponding to the turbine rotational speed increase) are responsible for this 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 disappeared 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. 5.

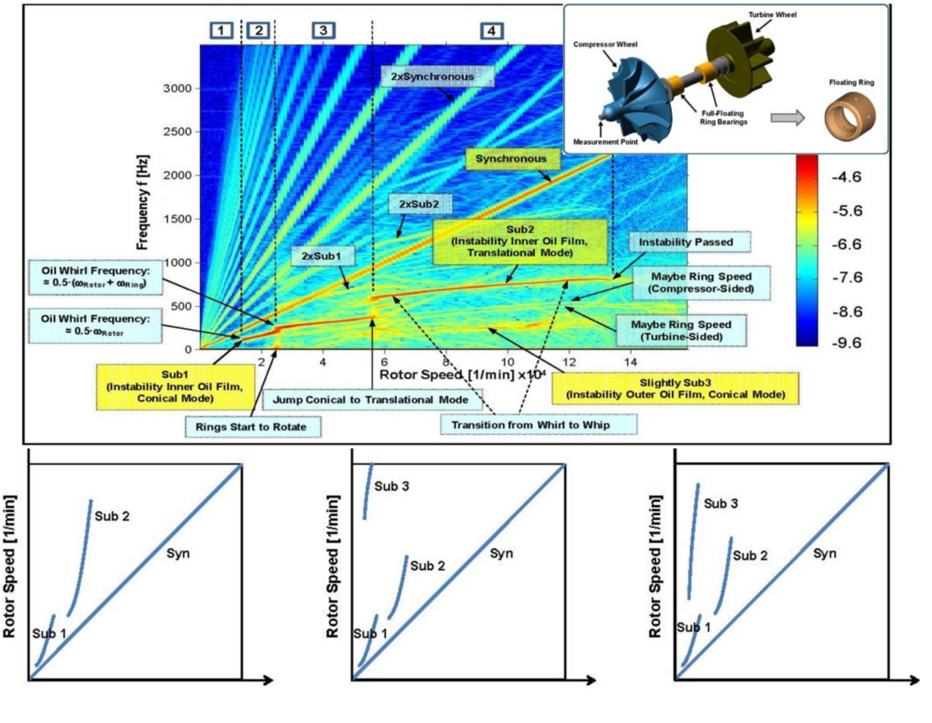

Fig. 6. Examples of experimental verification: bifurcation of turbocharger rotor. The interrupted sub-harmonic frequencies are closed to the multiple whirl phenomena (from Bernard Schweizer, University of Kassel, Germany)

Rys. 6. Przykłady weryfikacji: bifurkacje wirnika turbosprężarki. Nieciągłości częstotliwości harmonicznych są zbliżone do częstotliwości występowania zjawiska zawirowań (z Bernard Schweizer, University of Kassel, Germany)

In Fig. 6, we can find an example of experimental verification of multiple whirl phenomena taken from investigations of a small turbocharger rotor. We can consider these phenomena as very close to the bifurcation process well known in physics.

Fig. 7 presents the summary of considerations regarding the multiple whirls phenomena in the form of schematic drawing.

Stochastic variability of input data in heuristic modelling of rotors

A classic, traditionally applied, approach to the state modelling of various kinds of machines is the **algorithmic** approach, i.e. the one in which, for the known set of input data, we obtain the same, precisely repeatable, set of output data (results). This is the obvious consequence of the calculation capability of computers and the applied programmes. However, this type of 'traditional' research tools, often highly advanced and applicable in practice, are not able to correct the already introduced data nor to modify the assumed model depending on external conditions during the calculation procedure in progress.

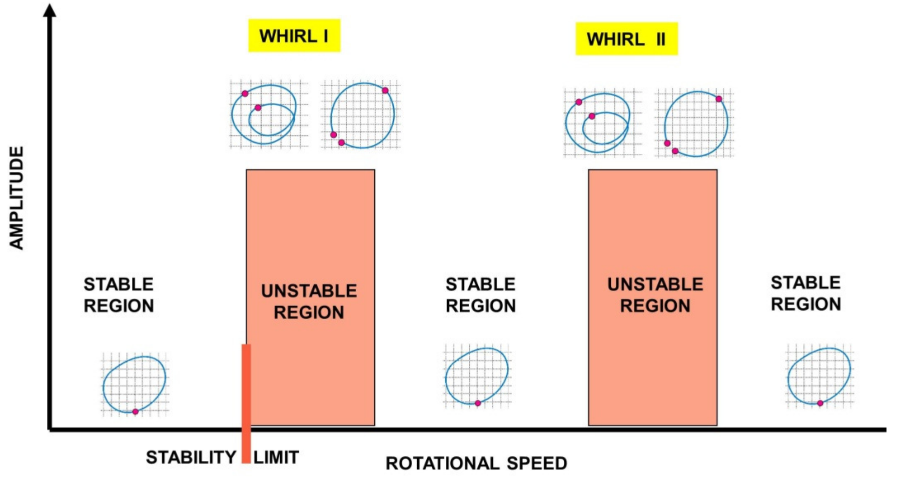

Fig. 7. Multiple whirls phenomena in the form of schematic drawing.
Rys. 7. Schematyczne przedstawienie zjawisk wielokrotnych zawirowań

Meanwhile, natural phenomena and a human nature (and thereby objects created by it) are of a **heuristic** character, which means possible feedback occurance in processes, intrinsic data, and the previously assumed methodology of state assessment corrections. It also means the necessity of taking into account influences of various errors and the uncertainty of input data, which is often done intuitively – Fig. 8.

It is worth mentioning that a trial of heuristic modelling means the necessity of having highly advanced ‘traditional’ research tools. The “**nonlinear description”** is extremely important, since heuristic models are nonlinear by nature. Another substantial feature is the possibility of a smooth transition from the linear to nonlinear description while applying the same research tools (the Superposition Principle cannot be used in this case) – Fig. 9.

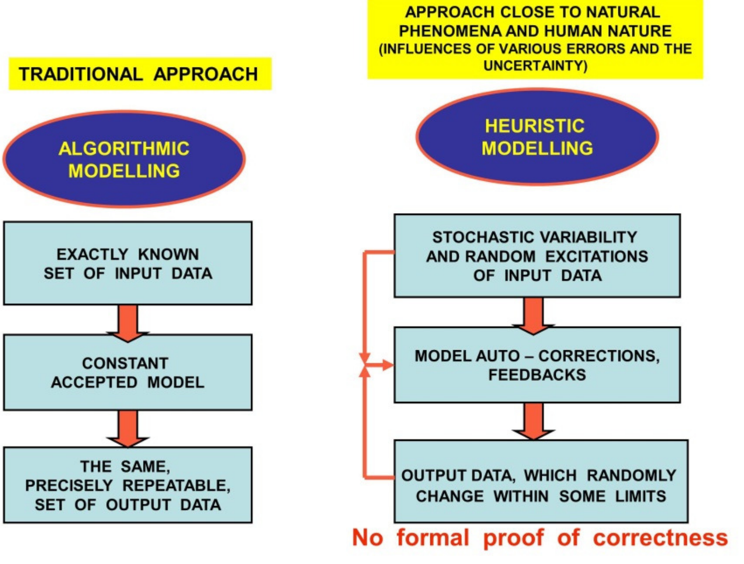

Fig. 8. Heuristic modelling as a contrast to algorithmic one
Rys. 8. Porównanie modelu heurystycznego do modelu algorytmicznego

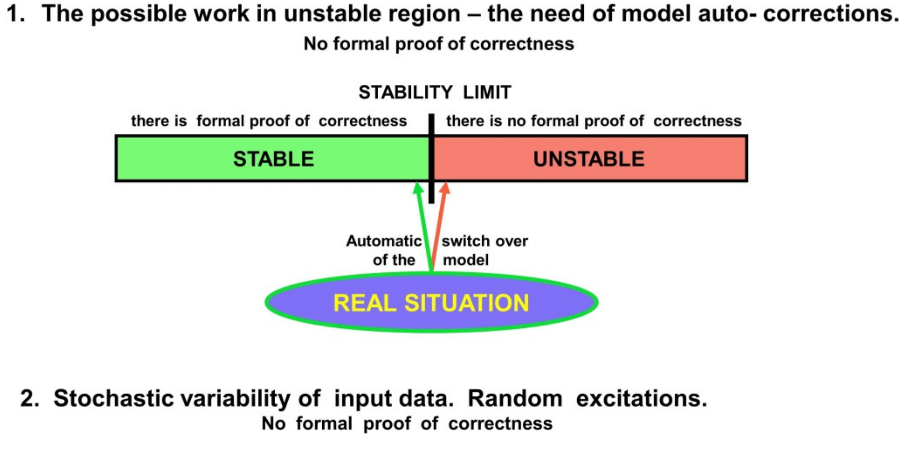

Fig. 9. Heuristic modelling in rotor dynamic. Two main reasons
Rys. 9. Model heurystyczny dynamiki wrzeciona. Dwie główne przyczyny

In consideration of the above, the MESWIR series code was applied in investigations [1]. Figure 10 presents the object used in tests as well as the concept of random changes of external excitation forces acting on a rotor disc. The randomness of changes was assumed (random-number generator was applied) although within limits +/-Δ*P* in proportion to the basic value *P*. Calculations were performed for different Δ*P* values to simulate various possible situations (e.g.: displacement of rotating masses, influence of magnetic fields, etc.).

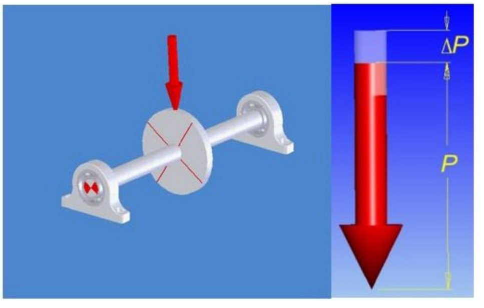

Fig. 10. Object of investigations: twin-support, symmetric rotor of a shaft diameter of 0.1 m, disc diameter 0.4 m, shaft length 1.4 m and a mass of 179 kg. Classic, cylindrical slide bearings lubricated by machine oil were used. The stochastic variability of an external excitation force within limits +/-ΔP = 20% in proportion to the constant (basic) value P was assumed
Rys. 10. Przedmiot badań: podwójne podparcie, symetryczny wirnik wału o średnicy 0,1 m, średnica dysku 0,4 m, długość wału 1,4 m o masie 179 kg. Zastosowano klasyczne, cylindryczne łożysko ślizgowe smarowane olejem. Stochastyczna zmienność wektora siły nacisku zawierała się w przedziale +/-ΔP = 20% w stosunku do stałej wartości przyjętej siły P

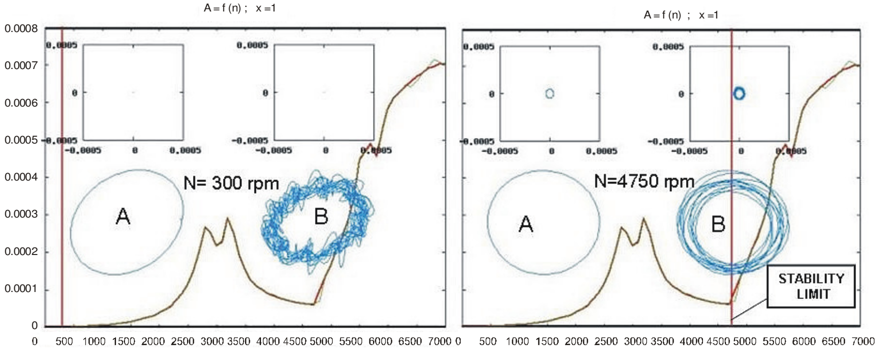

Fig. 11. Displacement trajectories of the rotor centre – within a stable operation range – calculated for the constant excitation force (basic) P (part A) and for the randomly changing – within limits +/-ΔP = 20% (part B) shown at the background of the rotor amplitude-frequency response [7]
Rys. 11. Trajektorie ruchu środka czopa łożyska w warunkach stabilnej pracy obliczone dla stałej siły nacisku P (obszar A) i dla zmieniającej się losowo – w zakresie +/-ΔP = 20% (obszar B) pokazane na tle charakterystyk amplitudowo-częstotliwościowych wirnika [7]

External rotating excitation forces, which can randomly change within limits +/-20% in proportion to the basic value, *P*, was assumed for the analysis, Fig. 10. The calculation results for the rotor shaft rotational speed from 300 rpm to 5550 rpm are shown in Figs. 11 and 12 [7]. The trajectory centre loading of the rotor by a constant force (basic) – rotating synchronously – is shown for the comparison on the left-hand side of each figure; whereas, the trajectory of the rotor loaded by randomly changing force (within limits +/-ΔP = 20% in proportion to the basic force *P*) is shown on the right-hand side of the figure. Images of trajectories in co-ordinate systems related to the maximum value of bearing clearance are placed in the upper part of the figure, while images of trajectories magnified as much as possible to exhibit clearly the phenomena are shown in the lower part of the figure.

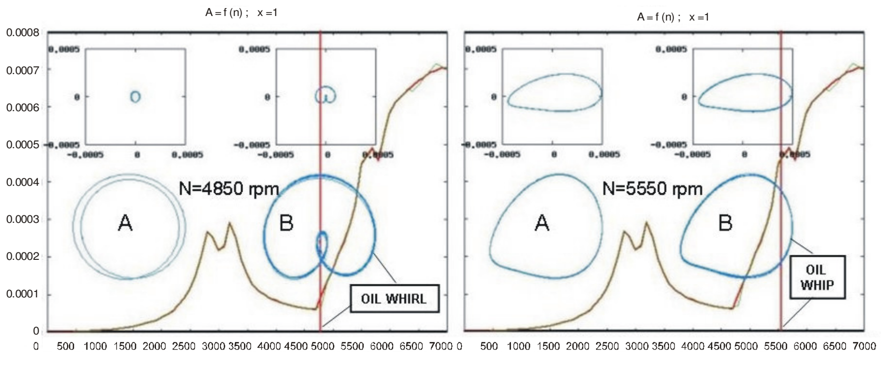

Fig. 12. Displacement trajectories of the rotor centre after the system exceeded the stability limit calculated for the constant basic force P (part A) and for the randomly changing – within limits +/-ΔP = 20% (part B) shown at the background of the rotor amplitude-frequency response [7]
Rys. 12. Trajektorie ruchu środka czopa łożyska w warunkach przekroczenia granicy stabilnej pracy obliczone dla stałej siły nacisku P (obszar A) i dla zmieniającej się losowo – w zakresie +/-ΔP = 20% (obszar B) pokazane na tle charakterystyk amplitudowo-częstotliwościowych wirnika [7]

The analysis of the figure indicates that the influence of randomly changing values of the external excitation force is significant in the case of small rotational speeds of the rotor. When the speed increases, this influence diminishes, which can be explained by the influence of rotor inertial forces generally attenuating a time-history. At the very stability limit, a certain increase in the trajectory disturbance can be observed. However, disturbances caused by the stochastic variability of input data decay, when the rotor rotational speed increases, is due to the development of hydrodynamic instability (Fig. 12). This is rather a startling result, since it could have been expected that such perturbations – after exceeding the stability limit – would intensify the instability of the entire system, since it is already unstable. Similar conclusions were found when investigations were performed for various ΔP values and various algorithms of random excitations. Thus, a system defect in the form of the hydrodynamic instability attenuates to a certain degree the defect caused by stochastic effects of input data. It is an interesting observation resulting from the performed research.

Final conclusions

The phenomenon of multiple whirls presented in this paper, found by advanced computer simulations and performed by means of the experimentally verified personally designed and commercially available codes, requires further theoretical and experimental investigations. 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 a similar phenomenon was recognised by means of direct measurements of vibrations at large power plants.

Preliminary considerations concerning heuristic modelling of rotors are included in this paper. In such modelling, we took into account the uncertainty and randomness of the calculation input data and mutual couplings. It was found that the influence of the stochastic variability of input data decreases after the system has exceeded the stability limit. This indicates that the defect of the hydrodynamic instability type can to a certain degree attenuate the defect in the form of a random scatter of input data.

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Nowe metody analizy stanu i diagnostyki maszyn na przykładzie mikrourządzeń energetycznych

S t r e s z c z e n i e

W pracy przedstawiony został przykład mikrosiłowni kogeneracyjnej możliwej do szerokiej aplikacji w tzw. mikrowytwarzaniu energii cieplnej i elektrycznej opartym na źródłach odnawialnych. Analizowanym urządzeniem jest wysokoobrotowa mikroturbina o mocy zaledwie kilku KW. Celem pracy nie są jednakże szczegóły konstrukcyjne tak zdefiniowanych urządzeń, lecz nowe metody analizy stanu dynamicznego i pozyskiwania relacji diagnostycznych. Jak się okazało, te małe urządzenia o prędkościach obrotowych wirnika od kilkudziesięciu do kilkuset tysięcy obrotów na minutę stanowią prawdziwe wyzwanie dla badaczy i eksploatatorów. Wiele tradycyjnych pojęć i metod badawczych, z powodzeniem stosowanych w maszynach większych, musiało zostać zweryfikowanych. Przy tak wysokich prędkościach obrotowych układ może przekraczać granicę stabilnej pracy wielokrotnie i dalej bezawaryjnie pracować, może też samoczynnie się stabilizować w wysokich zakresach prędkości. Mamy tu do czynienia z nowym zjawiskiem ***„wirów wielokrotnych”*** o zupełnie innych, nieznanych dotąd w eksploatacji symptommach diagnostycznych.

W pracy zaproponowany też został inny sposób matematycznego opisu stanu maszyny oparty na ***metodach heurystycznych***. Opis heurystyczny jest przeciwieństwem opisu algorytmicznego i uwzględnia m.in. przypadkowe błędy danych czy autokorekcję modelu w trakcie obliczeń. Opis heurystyczny jest bliższy naturze człowieka i warto go szerzej rozpropagować w budowie i eksploatacji maszyn. Czy przyjmie się w praktyce? W pracy podane zostały przykładowe wyniki obliczeń oparte na nowym podejściu odnoszące się do małych urządzeń energetycznych. To pierwszy krok na tej drodze w badaniach prowadzonych w ośrodku gdańskim i olsztyńskim.