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Investigation of the dynamic behaviour of the vertical rotor of a refrigeration compressor supported by foil bearings

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ABSTRACT

Keywords: Vertical-axis machines Refrigation compressors Foil bearings Rotordynamics Full-spectrum analysis Differential trajectories This experimental study examines the dynamic behaviour of an innovative refrigeration compressor, whose rotor, operating in a vertical position, is supported by foil bearings. The bearing system eliminates the need for external lubrication and demonstrates a high load capacity and rotational speeds of up to 120,000 rpm. The novelty of this solution lies in the rotor's vertical orientation, which introduces unique dynamic challenges. Numerical analyses, including modal and forced vibration studies, identified critical frequencies at 181 Hz, 204 Hz, and 283 Hz, closely matching the experimental results, with differences not exceeding 10%. Experimental tests confirmed stable rotor operation within the range of 60,000–120,000 rpm, with vibration amplitudes of the rotor journals not exceeding 0.028 mm in the critical operating range. Furthermore, misalignments in the bearing journals were observed and attributed to variations in lubrication clearance and foil deflections. The findings demonstrate the feasibility of foil bearings in vertically oriented high-speed compressors and provide novel insights into their dynamic behaviour. During the experimental study, irregular multi-loop vibration trajectories of the bearing journals were observed, necessitating the use of the differential method to determine their direction. This study advances oil-free compressor technology and offers guidance for future design and optimisation.

1. Introduction

In recent years, the machine-building industry has increasingly focused on energy consumption and environmental protection. Today's machines run at higher rotational speeds to consume less electricity and be more efficient. Due to the aforementioned trends, this paper investigates high-speed vertical-axis compressors. Furthermore, bearing technology is evolving to minimise mechanical losses in bearing nodes [1]. Additionally, both the bearings and the lubricants necessary for their trouble-free operation should not pose a threat to the ecosystem [2]. It is worth mentioning at this point that the research carried out in this article is in line with the idea of sustainable development, which does not allow overexploitation of the environment, while at the same time providing an impetus for the creation of new solutions.

Among the currently available bearing technologies that meet these requirements are supplied gas bearings [3], magnetic bearings [4], foil bearings [5] and spiral bearings [6] and it is with the application of a wide range of research bases that measurements and numerical calculations are carried out. Among other things, gas bearings have been experimentally tested for ORC systems where there is a significant difference between the weight of the generator and the turbine [7,8].

In addition, they are also widely used in gas microturbines [9,10], and also in oil-free blowers [11]. In these applications, it is important to introduce the fluid into the bearing at a sufficiently high pressure in the form of a compressed lubrication fluid [12]. When a lubrication film forms, the surfaces of the journal and the shaft separate from each other, resulting in a reduction in the friction torque and, consequently, the temperature of the bearing [13]. This transformation is also taking place in the refrigeration market, and more specifically in refrigeration compressors [14,15]. For both gas and magnetic bearings, it is crucial to test suitable controllers responsible for their operation [16], especially concerning nonlinear dynamics and motion bifurcations [17,18]. In foil bearings [19,20], as the rotor rotates, gas is drawn into the space between the journal and the top foil, leading to dry friction during startup. However, at a certain rotational speed, a lubrication film forms and pushes the top foil apart, enabling the creation of a thick lubrication wedge responsible for bearing capacity. When a lubrication film forms, the surfaces of the journal and the shaft separate from each other, resulting in a reduction in the friction torque and, consequently, the temperature of the bearing. Despite the great potential of foil bearing technology, primarily owing to its lack of need for lubrication and

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Nomenclature	
1X	Synchronous vibration component at the fun-
	damental frequency (equal to the rotational
	frequency) [Hz]
2X	Second harmonic vibration component (twice the
	rotational frequency) [Hz]
3X	Third harmonic vibration component (three times
_	the rotational frequency) [Hz]
α_n, β_n	Vibration phases in the X and Y directions [rad]
μ	Gas viscosity [Pa s]
Ω	Angular speed [rad/s]
ω	Whirl frequency [Hz]
ρ	Gas density [kg/m ³]
Θ	Pad angular coordinate [rad]
Ax_n, Ay_n	Vibration amplitudes in the X and Y directions
	[µm]
BW	Forward precession (rotor precesses in the same
	direction as its rotation) [–]
С	Damping coefficients [N s/m]
F	Reaction forces of the bearing [N]
FW	Backward precession (rotor precesses in the
	opposite direction to its rotation) [–]
h	Radial clearance [mm]
Κ	Stiffness coefficients [N/m]
L	Bearing axial width [mm]
l, t	Pad leading and trailing edges [–]
р	Gas film pressure [Pa]
p_a	Ambient pressure [Pa]
R	Bearing radius [mm]
W	Static load [N]
X, Y, Z	Cartesian coordinate directions [mm]
z	Bearing length [mm]
Ba _n	Amplitude of backward circular precession [µm]
Fa _n	Amplitude of forward circular precession [µm]

the very low friction torque, this type of bearing is not widely used in commercial machines due to the necessity for further technology development. It is precisely because of these advantages that further research is important in this area, where horizontal bearing solutions in particular have so far been investigated [21,22].

In spiral bearing technology, a lubrication film forms as a result of pressure build-up on the spiral geometry, where the force of transportation is inversely proportional to the clearance between the sleeve and the shaft [23]. This is another group of bearings that do not generate losses resulting from the supply, but their precise manufacturing is crucial for their proper functioning. This is a group of bearings that are currently being implemented in high-speed machines to improve their efficiency and reduce operating costs.

This transformation is also taking place in the refrigeration market, and more specifically in refrigeration compressors. The most common way to reduce the energy consumption of these devices is to adapt the current operating parameters to the cooling demand of the consumer. Until recently, refrigeration systems, in which the compressor is one of the main components, were designed for the extreme operating point of the device to ensure thermal comfort under extreme conditions in a given climate zone. Unfortunately, climatic variability results in varying system operations so a flat characterisation of the devices in different operating regimes is essential [24]. In modern refrigeration systems, for example, the rotational speed is controlled to adapt to weather conditions. Such an operating regime requires bearings that are failure-free and do not need to be lubricated or bearings with an environmentally friendly lubricant. There are machines running on magnetic bearings [17], which are, one might say, ideal, but their price is an inhibiting factor in their further development. Machines that use spiral bearings [23,25] are another example of machines with a great potential for development, but their manufacturing technology is so expensive that it is on the verge of profitability.

One of the products that the TURBOCHILL company is trying to introduce to the market is a compressor with self-acting foil bearings, which is very small compared to its power generation capacity. The compressor operates in a vertical position. Similar machine designs are described in the literature [26,27] but their researchers have encountered instability problems within certain rotational speed ranges.

1.1. Literature gaps

The operation of the rotor in a vertical position causes classic gas- or liquid-lubricated bearings to be underloaded (not loaded enough) and the bearing journals are unable to find the point of static equilibrium. In addition, the high speed further increases this effect since the rotor approaches the centre of the bearing sleeve after passing through the so-called semicircular equilibrium state. The insufficient load of the bearings usually leads to instability of the gas film, making the operation of such a system impossible. In vertical operation the effects of gravity are absent, and as is well known from both theory and experience, a centrally operated shaft is liable to produce instability referred to as either a half-frequency whirl or oil whip [28]. For the above reasons, horizontal-axis bearings are rarely used, especially in situations where the design of the entire system requires it especially in the context of water turbines [29,30] and vertical-axis wind turbines [31]. Refrigeration compressors, due to their design sophistication, are analysed in various aspects [32]. The first step is optimisation by means of refinement of thermodynamic parameters at nodal points, where determination of thermal power, compressor power and COP are crucial for further design. These aspects are particularly developed recently in systems such as the Organic Rankine Cycle-Vapor Compression Cycle System because the operation of the refrigeration compressor is extremely important here [33]. Thus, the present approach does not enforce the insertion of vertical-axis bearings in the design, hence their use is rare in this type of engineering.

A number of articles on high-speed turbomachines operating in a horizontal position can be found in the literature [34]. The results described in this article will enrich the currently scarce literature on high-speed turbomachines operating in a vertical position. The use of an innovative bearing method is an added value of this technical solution. The authors made a comprehensive literature review to show the novelties of our study in terms of various bearing configurations. The results of this review are presented in Table 1 (see [35,36]).

Present-day technologies for machines with rotating components and machines with various rolling, gas or magnetic bearings pose many challenges both to engineers and scientists. They are not only characterised by high rotational speeds but also by specific requirements concerning stability and operational reliability. There are various vibration problems in rotating machinery, especially in vertical pumps using plain bearings [45]. In the case of vertical centrifugal pumps used in nuclear power plants, it is crucial to prevent lubricating oil leakage, as this can result in a long-term reactor shutdown [37]. Modern technological machines increasingly utilise mechanical vibration monitoring systems, enabling early detection of rolling bearing defects [39,42]. The approach to diagnosing and monitoring rotating systems is based on a variety of methods, ranging from theoretical modelling and prototyping to experimental vibration testing [39,40]. One of the key components of many modern systems is magnetic bearings, which are characterised by contactless operation and do not necessitate a lubrication system [41,43]. In specific applications, such

Overview of Literature Gaps on high-speed turbomachines operating in a vertical position.								
Title	Bearing type	Rotational speed	Machine type					
Development of Active Magnetic Bearings for a Vertical Centrifugal Pump Rotor [37]	Active magnetic bearing	2900 rpm	Centrifugal sodium pump					
Application of Foil Bearings to Turbomachinery Including Vertical Operation [36]	Hydrodynamic foil bearing	58,000 rpm	Simulator for the cryogenic device					
Influence of cross-coupling stiffness in tilting pad journal bearings for vertical machines [38]	Tilting pad bearing	3000 rpm	Test rig					
The comparison of diagnostic features between the vertical and horizontal axis rotors [39]	Ball bearing	3000 rpm	Test rig					
Experimental and performance analyses of a turbomolecular pump rotor system [40]	Ceramic bearing	52,000 rpm	Vertical turbomolecular vacuum pump					
Experimental methodology for determining turbomachinery blade Damping using magnetic bearing excitation and non-contacting optical measurements [41]	Magnetic bearing	4000 rpm	Test rig					
Influence of imbalance force angular position to vertical and horizontal rotors rolling bearings defects diagnostics [42]	Ball bearing	3000 rpm	Test rig					
Dynamic and Thermal Investigations of the Forward Dry-Friction Whirl/Whip of a Vertical Rotor-AMB System during Touchdowns [43]	Active magnetic bearing	36,000 rpm	Test rig					
Operation of a Mesoscopic Gas Turbine Simulator at Speeds in Excess of 700,000 rpm on Foil Bearings [44]	Foil bearing	700,000 rpm	Test rig					
Nonlinear Analysis and Characteristic Variation of Selfexcited Vibration in the Vertical Rotor System due to the Flexible Support of the Journal Bearing [45]	Journal bearing	1800 rpm	Test rig					
Numerical simulation and full scale landing test of A12.5 MW vertical motorcompressor levitated byactive magnetic bearings [35]	Ball bearing with damper ribbon	10,160 rpm	Motorcompressor					
Unsteady characteristics of lubricating oil in thrust bearing tank under different rotational speeds in pumped storage power station [30]	Thrust bearing	500 rpm	Hydropower generator					
This work - Investigation of the dynamic behaviour of the vertical rotor of a refrigeration compressor supported by foil bearings	Foil bearings	120,000 rpm	Vertical refrigeration compressor					

Table 1

as turbomolecular vacuum pumps or turbine engines, the bearings must operate at extremely high speeds, reaching up to 700,000 rpm [44]. Under these harsh operating conditions, such as extremely low or high temperatures and extraordinary rotational speeds, the stability and reliability of the bearings' performance are of utmost importance [38,43]. The work of Zhang et al. [30] examined the operation of a vertical-axis rotor installed in a hydropower generator. The thrust bearings used provided a speed of 500 rpm, a considerable outlier to those studied in this paper. In contrast, the work of Shi et al. [46] has already used foil bearings operating at 12,000 rpm, but the location of the rotor is horizontal. In addition, the present work is a broad study comparing flow-type numerical calculations with experiment, but dynamic tests showing the stability of rotor operation are not included. Multidirectional research conducted at various research centres around the world provides valuable information on the operation, performance, and potential problems associated with modern bearing technologies. The green colour in Table 1 indicates the common features of the mentioned works and this article. From Table 1, it can be seen that only the present work analyses the solution of horizontal axis shafts with foil bearings and at such significant rotations of 120,000 rpm. This is an innovative work because it brings possibilities for further development of refrigeration systems working mostly with reciprocating compressors [47]. Based on the literature gaps, there is a lack of scientific work analysing the vertical rotor of a refrigeration compressor supported by foil bearings.

1.2. Research questions, novelties, and contributions

This article focuses on analysing the dynamics of a rotor that operates vertically at high speeds on foil bearings. Foil bearings, with their characteristic structure that allows them to adapt to conditions

such as rotational speed and load, can become an excellent choice for this type of design. It is also important to note that designing foil bearings can be challenging due to their nonlinear properties and the difficulties associated with their manufacturing. Using these bearings may lead to larger blade clearances, potentially negatively impacting the compressor's efficiency. On the one hand, the flexibility and damping of the foil assembly is an advantage over other gas bearings, but this flexibility can lead to foil deflection, potentially causing the rotor disk to come into contact with the housing. Another problem is the dry friction that occurs during the initial run-up phase and during the final run-down phase, which must be minimised.

Taking into account the above and the literature gaps indicated earlier in the previous section, this paper answers the following research questions:

- · How high can the rotational speeds of the presented system be?
- · Can the natural frequencies of vibration in the new innovative solution of a refrigeration compressor with a vertical shaft be effectively determined by a simple mathematical model and experimental studies?
- Do the journals of the rotor operating in a vertical position move within the lubrication clearance circle in parallel according to the theory of rotor dynamics?
- · Is it possible to determine the semicircle of equilibrium along which the journals should move? If not possible, what shape will the trajectories take?

This paper answers the research questions posed and also fills in the literature gaps. Thus, the fundamental novelty of this article is to show new results on a global scale concerning an experiment with foil bearings operating in a vertical position, in a machine rotating at a much higher speed than has been reported in the literature to date.



Fig. 1. (a) Refrigeration compressor diagram, (b) Compressor mounted on the test rig.

In summary, the work carried out brings the following contibutions: They determine the high rotational speed, up to 120,000 rpm, of the presented system without external lube at relatively high load capacity. They determine, by means of a simple mathematical model and experimental studies, the natural frequencies of vibration in a new innovative solution of a vertical-shaft refrigeration compressor. They discover that the journals of the rotor operating in a vertical position move within the lubrication clearance circle do not move parallel to each other. Thus, this is not consistent with the theory of rotor dynamics. The trajectories of the journals resembled the movement of the journal bouncing off the foil.

1.3. Hypothesis and organisation of the article

The hypothesis of the work is that vertical-axis bearings can be used in technological solutions with refrigeration compressors. In order to test its veracity, the next steps of the work presented in the following sections were performed: in Section 2, the description of the refrigeration compressor and its bearings is included. Section 3 discusses the dynamic analysis of the rotor operating in a vertical position. Experimental studies using air are first presented in Section 4 and then discussed in Section 5. The summary and conclusions are provided in Sections 6 and 7, respectively.

2. Description of the refrigeration compressor and its bearings

The compressor is designed for vertical operation. The maximum rotational speed of this machine is 120,000 rpm. The motor power is 10 kW. The estimated compression of R134a fluid [48] is 3.7. The rotor, which weighs 1.23 kg, is supported by two radial foil bearings. At the bottom part of the machine, there is a starting bearing and in the upper part, there is an aerodynamic thrust gas bearing (see Fig. 1). Two rotor disks were used for gas compression. In the vertical position, the machine operates in such a way that during startup, the rotor rests on the lower starting bearing up to a certain speed. Once the axial force generated during gas compression is greater than the weight of the rotor, the rotor is lifted. The shaft begins to levitate, and with increasing speed and increasing axial force, the keep plate approaches the upper thrust bearing. The compressor is designed to operate at speeds between 60,000 and 120,000 rpm. A balance quality grade of G1 was selected for balancing the rotor.

Air was used for the first tests of the compressor. Optical displacement sensors were used in both radial directions (X and Y) as well as in the axial direction (Z), and two accelerometers were mounted at bearing locations (Fig. 2). The position of the top foil lock and the direction of rotation of the rotor are also shown in the figure below to better understand the results obtained.

Three-segment foil bearings were used, which are described in paper [19] and have already been tested with air in previous papers [49,50]. The axial bearing used is a spiral bearing, which was designed using commercially available CFD software and mathematical models [51]. The properties of the bearings were selected for the target fluid, but the possibility of operation with air (of course, with limited operating parameters) was taken into account.

3. Dynamic analysis of the rotor operating in a vertical position

The numerical modelling did not fully analyse the fluid flow in the foil bearings, but instead used previously determined stiffness and damping coefficients for both the lubricating film and the supporting structure. This significantly simplified the calculations and avoided the need for computationally expensive CFD simulations coupled with dynamic analyses. In addition, the model assumes a constant temperature across the entire operating range, meaning that the effect of temperature changes on gas viscosity and bearing load capacities was not considered. In fact, an increase in temperature affects the viscosity and density of the lubricant, which can lead to changes in the dynamic characteristics of the system. The stiffness and damping coefficients were assumed to be linear, meaning that their non-linear variation as a function of rotational speed and load was not included in the model. The effect of deformation of the bump foil, which can actually affect the dynamic characteristics of the bearings, has also been neglected. A further simplification was the assumption of perfect alignment between the rotor and bearings, meaning that the calculations did not account for possible shaft misalignments that could cause additional vibrations and changes in journal movement trajectories. The machine casing has been modelled as a rigid structure, meaning that no account has been taken of possible structural deformations that may affect the dynamics of the system under real conditions. In addition, the axial bearing model was simplified and modelled as an asymmetrical spring - in the direction of gravity, it was given a very high stiffness, while in the opposite direction it was fully extended. This means that, at low rotational speeds, the bearing did not generate any force to support the rotor and only began to perform its function once the corresponding axial force from the compressor was reached. This simplification allowed efficient modelling within the operational range of the machine, but may have affected the differences between the numerical



Fig. 2. (a) Measurement diagram (top view), (b) Foil bearing diagram.

and experimental results in transient states. All these simplifications allowed the results to be obtained within a reasonable computational time, maintaining a balance between the accuracy and complexity of the model. In future studies, it is planned to take into account the effects of temperature, misalignment, and the non-linear effects of stiffness and damping in order to further improve the representation of the actual system behaviour.

Properties such as stiffness and damping of the foil bearings have a significant impact on the compressor rotor, so a simple analysis of the rotor dynamics depending on just these parameters was carried out. A determination was made that the rotor must have different support stiffness values, however, these differences could not be large. In this case, the rotor did not have a form of natural vibration that could damage the axial bearing (a conical form with much larger movements at the top of the rotor). The damping of the support had an impact on the reduction of the vibration amplitude but not on the value of the frequency of the modal form of the rotor. The upper bearing should have higher compliance than the lower bearing. Otherwise, at low rotational frequencies (up to 500 Hz), the upper part of the rotor will exhibit vibration modes that can destroy the rotor disks. It was possible to achieve a change in stiffness by optimising the thickness of the lubrication film by adjusting the assembly preload of the bearing [52]. However, when the lubrication clearance is reduced, the resistance to motion increases, as well as the friction that occurs at low speeds. In addition, lubrication clearance and lubrication film thickness are difficult to measure. Based on the information provided in article [49], the same lubrication clearance value was chosen for both bearings, and modifications were made to the shape of the bump foil. The shape and number of bumps can affect the stiffness of the rotor support. On the other hand, the number of bumps increases the damping of the bump foil assembly.

In the numerical simulations conducted in the ANSYS environment, air was used as the working medium. This decision was based on the need to validate the model by comparing the results of the modal analysis and forced vibration analysis with experimental data. Further research is currently underway to analyse the impact of the target working medium, R134a, on the dynamics of the rotor system. In future stages of numerical work, the model will be adjusted to account for the thermophysical properties of this medium, enabling an assessment of its impact on the load-carrying capacity of the foil bearings and the stability of the rotor system. By employing an appropriate numerical approach and in-house developed foil bearing manufacturing technology, it will be possible to fine-tune the bearing geometry to specific operating conditions, considering the properties of both air and other working media.

3.1. Selection of bearing parameters

First, for the rigid sleeve, the parameters of the gas film were estimated for its different thicknesses, depending on the rotational speed (Figs. 3–4). For this purpose, the GAZBEAR program was used, the principle of operation of which is described in the article [53]. GAZBEAR is an original program used to calculate the characteristics of

an aerodynamic gas bearing. The operation of the application includes static analysis and dynamic analysis. Static analysis involves calculating the reaction force of the gas film to a given static excitation, which is generated as a result of the rotation of the journal relative to the stationary sleeve to the viscosity of the lubricating medium. The program can be used for any gaseous medium if its dynamic viscosity is known under given ambient conditions. Dynamic analysis involves calculating the stiffness and damping coefficients at the considered bearing operating point. The operating point is understood to mean constant rotational speed and static load. The characteristics of the bearings were determined using an isothermal model of the aerodynamic gas bearing. Static analysis involves determining the reactive force of the gas film for a given static load. The following equations are solved during the calculations: the Navier-Stokes equation that expresses the conservation of momentum (assuming that the pressure is constant in the direction corresponding to the thickness of the lubrication film) and the continuity equation that represents the conservation of mass [9]. The pressure distribution of the lubrication film is determined from the Reynolds equation (1) for compressible fluids:

$$\frac{\partial}{\partial x}(\rho h^3 \frac{\partial p}{\partial x}) + \frac{\partial}{\partial z}(\rho h^3 \frac{\partial p}{\partial z}) = 6\mu \Omega R \frac{\partial(\rho h)}{\partial x} + 12\mu \frac{\partial(\rho h)}{\partial t}$$
(1)

where, p - gas film pressure, ρ - gas density, h - radial clearance, R - radial and angular coordinates of the polar coordinate system, μ - gas viscosity, z - bearing length, Ω - angular speed [9].

The following Eqs. (2)–(5) were used to determine fluid film forces and dynamic force coefficients:

$$F_{X_0} = \int_0^L \int_{\Theta_l}^{\Theta_l} (p - p_a) \cos \Theta \ R \ d\Theta \ dz \tag{2}$$

$$F_{Y_0} = \int_0^L \int_{\Theta_l}^{\Theta_l} (p - p_a) \sin \Theta \ R \ d\Theta \ dz$$
(3)

$$K_{XY} + i\omega C_{XY} = \int_0^L \int_{\Theta_l}^{\Theta_l} p_Y \cos\Theta \ R \ d\Theta \ dz \tag{4}$$

$$W = \sqrt{F_{X_0}^2 + F_{Y_0}^2} \tag{5}$$

where, p_a - ambient pressure, p - gas film pressure, R - bearing radius, L - axial width of the bearing, l, t - pad leading and trailing edges, X, Y - directions of perturbation for first-order pressure fields, Θ - pad angular coordinate, ω - whirl frequency, F - bearing reaction forces, K, C - bearing stiffness and damping coefficients, W - static load [9].

The rotor was balanced using a balance quality grade of G1, which corresponded to a residual unbalance of 0.136 gmm. At a speed of about 65,000 rpm, which is the speed at which the rotor should theoretically detach from the support bearing, the value of the centrifugal force was approximately 6 N. Analyses were carried out for such a bearing load value. Four different thicknesses of the lubrication film and the damping are marked with different colours. The thin line refers to the parameter located on the right side of the figures. In addition, lines of different thicknesses indicate the stiffness in two directions. As can be seen, the greater the clearance, the greater the differences in stiffness at the beginning of the speed range values depending on the direction. In addition, the rotational speed at which a lubricating film forms depends



Fig. 3. Stiffness (a) and damping (b) characteristics as functions of rotational speed for various air film thicknesses.



Fig. 4. Cross-coupled stiffness (a) and damping (b) coefficients as functions of rotational speed for various air film thicknesses.

on the thickness of the gas lubricating film. As the machine is designed for vertical operation and its bearings may therefore be underloaded, it is necessary to adopt a lubrication clearance as small as possible. Under these assumptions, the lubrication film would form rapidly. However, as the speed increases, the resistance to the rotor's motion resulting from friction in the lubrication gap would also increase [54].

Choosing the correct lubrication clearance value for foil bearings is crucial for the stability of rotor operation, vibration damping efficiency, and temperature control during operation. The value of 20 μ m was chosen based on both previous experimental studies [19,49] and numerical simulations carried out using the GAZBEAR software. Preliminary experimental studies indicated that the first critical speed of the rotor was around 10,000 rpm, meaning the bearing should establish a lubricating film close to this value to prevent contact between the journal and the foil at resonance and to limit vibration amplitudes. Analysis of the impact of lubrication clearance (Fig. 3) showed that:

- For a clearance of 15 μm , the lubricating film formed at 4000 rpm, ensuring stable operation and minimising the risk of the journal rubbing against the foil, although its effect on the operating temperature could be significant.
- For a clearance of 30 μm , the lubricating film did not form until 18,000 rpm, which could lead to damage to the bearing foil within the normal operating speed range of the compressor.
- For a clearance of 20 μm, the lubricating film formed around 8000 rpm, ensuring stable system operation and a reduced temperature rise compared to smaller clearances.

Based on these results, 20 μ m was considered the optimal compromise between dynamic stability, vibration damping (Fig. 4), temperature control, and rotor start reliability. In addition, the numerical simulations confirmed that the most favourable dynamic characteristics are achieved within this clearance range, both in terms of vibration amplitude reduction and the load-carrying efficiency of the lubrication film. Of course, an even smaller clearance, such as 10 μ m, could be used in numerical analyses, as this would theoretically result in increased bearing stiffness and improved vibration damping. However, our experience with actual rotor systems shows that this solution is not optimal for the operation of the machine, as it can lead to excessive temperature rises and issues during rotor start-up.

Many foil bearing models were then created, where, using FEM, the flexibility of the bump foils was estimated based on their shape for a clearance of 20 µm. Then, several variants of such bearings were manufactured, on which experimental tests were carried out using a known method which consists of cyclically loading the bearing up to a certain value (50 N in this case), recording the displacement [55]. Fig. 5 (side a) shows the series of measurements carried out for two selected types of bearings that differ in the number of bumps and the distance between them. The tests were carried out for three angular positions of the bearings, where the angle of 270 degrees is the location of the top foil lock (no tests were carried out in this direction). In the case studied, one of the bump foils had 11 bumps and their ratio to the flat part (the bump foot) was 0.75. The second tested foil had 7 bumps and their ratio to the flat part was 0.8. It can be seen that the results obtained for the different angles are not similar, but this is due to the strong nonlinearity of the foil assembly [56].

In the model, the starting bearing was presented as a spring with peculiar characteristics. We used a body-to-ground spring located at the lower part of the rotor, as shown in Fig. 6. In the direction of the gravitational force, the spring had a very high stiffness value (1e10 N/m), whereas in the opposite direction, its stiffness was zero. The damping value for this spring was also 0 [N s/m]. Thrust bearing: The stiffness of this bearing was omitted in the analysis because at lower rotational speeds this bearing did not perform its function. Under such conditions, the rotor rests on the starting bearing and the distance between the keep plate and the thrust bearing does not allow a lubricating film to form. It is only at higher speeds, above about 60,000 rpm, that the rotor generates sufficient axial force to lift it upwards, bringing it closer to the thrust bearing. Due to the uncertainty of the value of the rotor displacement in this context, authors decided not to consider the stiffness resulting from this bearing.

3.2. Modal analysis of the rotor

The rotor model was created using ANSYS software. The material data for the rotor components was imported from the GRANTA program. The rotor model was discretised using nearly 80,000 TET10 and HEX20 finite elements. The stiffness and damping of the gas film and foil assembly were set separately. A stiffness with varying



Fig. 5. (a) Static stiffness values for foil bearings with different bump. (b) Stiffness characteristics of the starting bearing modelled as a spring foils.

characteristics was set at the location of the starting bearing-very stiff (1E+10 N/m) in the direction of the gravitational force and zero stiffness (0 N/m) in the direction opposite to the gravitational force (side B of Fig. 5). The rotor's rotational speed was set in 12 steps. The Corioliss effect has been turned on. Two points on the rotor were determined for setting the unbalance according to the ISO 20816-1 standard. Dynamic simulations were carried out for the rotor without disks (Fig. 6). Simplified calculations were used which did not take into account the distribution changes in the gas film caused by the deformation of the top foil and the deformation itself. In this model, the gas film is described by means of springs in both the main and oblique directions, as well as springs in the main directions describing the bump foil assembly. As is well known, the two-way coupling between the gas film and the structure is a strongly nonlinear problem, however, the analysis of these interactions was not the aim of the study. The gas film characteristics of the foil bearing (stiffness and damping) were entered into the program based on Fig. 3. The properties of the bump foil, obtained experimentally, were adopted on the basis of an experiment. The highest values obtained for the angle of 90 degrees (the opposite side of the lock) for one of the directions and the highest values obtained for the angle of 180 degrees for the other direction were selected. Values of 1.16E+06 and 1.6E+06 N/m were adopted for the upper bearing and values of 2.05E+06 and 2.8E+06 N/m for the lower bearing. The damping of the foil assembly (due to friction) was adopted on the basis of previous works [49] as the technology and geometry of the bearings were similar. The rotor model takes into account the support of the rotor in the axial direction using a starting bearing and therefore stiffness with non-symmetrical properties has been added. The stiffness is 1E+10 N/m in the direction of the gravitational force and 0.00001 N/m in the opposite direction. The numerical model took Coriolis effects into account and the structural damping was assumed to be 0.2. The choice of a constant damping coefficient of 0.2 was dictated by our experience in performing both simulation and experimental modal analyses. This value, in our case, improved the stability of the calculations and allowed us to achieve good consistency of results in a short time. In the forced vibration analysis, we used a lower value for the damping factor (0.01, i.e. 1%). Its purpose is to control the amplitude of the vibrations so that they are visible and do not approach infinity, according to vibration theory.

In the numerical analysis, the rotor model was discretised using tet10 and hex20 elements, which was necessary due to the hollow shaft. The shaft was hollowed out to accommodate the generator, which is an integral part of the system. In addition, a bolt was screwed into the threaded hole to serve as a starter bearing element, which also influenced the selection of a suitable numerical grid. To better illustrate these design aspects, a cross-sectional diagram of the rotor has been presented (Fig. 6-c), showing the solution used. The grid independence test was conducted as part of a modal analysis to determine how grid density affects the calculated natural frequencies. The first four mode shapes, which have the greatest influence on the system's dynamic



Fig. 6. Photo of the tested rotor (a), its discretised FEM model (b) and cross-sectional diagram of the rotor (c).

behaviour, were selected for comparison. To ensure computational efficiency, the analysis was performed on a 16-core processor. The table below (Table 2) presents the results of the grid independence test:

To better illustrate the effect of grid density on the obtained results, the percentage differences in natural frequencies between successive grid cases are shown below (Table 3):

After analysing the results, the second grid case (130,781 nodes, 98,824 elements) was selected as the optimal compromise between result accuracy and computational efficiency. One of the key reasons for this choice is that the differences between this case and denser grids are less than 2%, suggesting that further grid densification does not significantly improve the accuracy of the results. In addition, for even denser grids, the analysis time increased dramatically and RAM consumption exceeded the available hardware resources, making it impossible to run the simulation within a reasonable time. Analysis of the results shows that, above the chosen discretisation, the values of the natural frequencies do not change significantly, confirming the model's sufficient accuracy. A key reason for choosing this grid was the need to link modal analysis with harmonic analysis. For the selected discretisation of the model, the harmonic analysis took 23 h, 58 min, and 30 s, and the maximum RAM consumption reached 43 GB. Further increasing the discretisation of the model could potentially improve the quality of the mode shapes but would lead to a drastic increase in the computation time for the harmonics. In addition, an increase in the number of elements would result in further RAM usage, which, in the case of a more detailed grid, could exceed the available limit of 52 GB, making a full analysis impossible. Consequently, the chosen

e 2

Results of the grid independence test.

Number of nodes	Number of elements	Computation time [s]	1 [Hz]	2 [Hz]	3 [Hz]	4 [Hz]
87,327	64,574	46	178.19	207.18	270.47	339.96
130,781	98,824	80	179.96	208.33	274.97	344.92
184,747	138,778	126	178.11	207.02	270.63	339.91
424,892	249,932	696	173.73	204.36	262.15	330.14
542,610	367,331	1101	174.73	204.99	263.82	332.23

Table 3

Percentage differences in natural frequencies between successive grid cases.

Number of nodes	Number of elements	Percentage difference [%]				
87,327	64,574	[-]	[-]	[-]	[-]	[-]
130,781	98,824	73.91304	0.993322	0.555073	1.66377	1.458995
184,747	138,778	57.5	-1.02801	-0.62881	-1.57835	-1.45251
424,892	249,932	452.381	-2.45915	-1.2849	-3.13343	-2.87429
542,610	367,331	58.18966	0.575606	0.30828	0.63704	0.633065



Fig. 7. Campbell diagram of the compressor for rotational speeds up to 200,000 rpm.

grid provides an optimal balance between the accuracy of the results and the ability to determine them within an acceptable computational time. To enhance the clarity of the results, a detailed description of the grid test and an illustration of the rotor cross-section have been added to the updated version of the article, providing a better representation of the model's geometry and the simplifications used.

After performing a modal analysis taking into account the rotational speed, a Campbell diagram was obtained (Fig. 7), in which it can be seen which of the first 12 natural frequencies (marked in different colours) intersect with the rotational speed (marked in black). According to the calculations, increased vibration levels can be expected at speeds of 10,883.7 rpm, 12,341.6 rpm, 16,960.8 rpm and 19,352.9 rpm. The purple area indicates the operating range of the machine (up to maximum rotational speed) where no natural vibration modes are present, assuming correct operation (1X). Indeed, there is an inaccuracy in the quoted passage of the text. Harmonic 2X has the potential to excite modes No. 7 and 8 within the nominal operating range of the rotor marked in purple in Fig. 7. On the other hand, when the rotor reaches a speed within the range of 60,000 rpm to 120,000 rpm and the 3X harmonic component is present, the 7th and 8th vibration modes of the rotor can also be excited. At certain shifts of the harmonics, higherorder modes can be excited. Authors also investigated the higher-order modes. However, most of these were associated with the keep plate or the end of the shaft on which the rotor disks were mounted. For the sake of clarity of the Campbell diagram, we have decided not to include these modes.

Fig. 8 shows selected modes of natural vibration (i.e. modes no. 2, 3, 4 and 5) in which the direction and shape of the vibrations can be seen. For the first two modes, the trajectories of the rotor motion were in the shape of very flat ellipses and as the frequency increased the trajectories became more circular. Table 4 shows the exact values of the frequencies and critical speeds (for 1X) as well as the direction of rotation of the rotor (only up to nominal speed).

As can be seen from the table below, the direction of rotation of the rotor changes quite often, which can lead to improper operation of the rotor. In addition, the frequency at which the given natural vibration mode occurs strongly depends on the rotational speed for natural vibration modes no. 5, 7, 8, 9, 11 and 12.

3.3. Rotor dynamics

To verify the dynamic response of the system and the values of vibration amplitudes at a given force, analyses were performed using harmonic excitation (Fig. 9) where a residual unbalance of 0.136 gmm was used, which corresponds to a balance quality grade of G1. The static load and shaft deflection on the bearings, caused by the gravitational force, was 0. The residual unbalance was placed at the centre of the rotor. Fig. 9 shows the course of vibration amplitude and phase over a wide range of frequencies. There are no worrisome natural vibration modes within the operating range of the rotor. In the initial phase (up to 400 Hz), increased vibration levels can be expected. But to compensate, a short rotor acceleration time will be required to reach a given speed.



Fig. 8. Natural vibration modes represented by trajectories; (a) mode 2–181.4 Hz (10,883.7 rpm), (b) mode 3–205.7 Hz (12,341.6 rpm), mode 4–282.7 Hz (16,960.8 rpm), mode 5–322.5 Hz (19,352.9 rpm).

Results of	Results of the modal analysis in which the rotational speed has been taken into account. BW indicates backward precession, FW - forward precession.												
Mode	Precession	Critical	0 rpm	12,000	24,000	36,000	48,000	60,000	72,000	84,000	96,000	108,000	120,000
		speed		rpm	rpm								
1		0 rpm	0.004	0.004	0.003	0.004	0.002	0.008	0.005	0.005	0.005	0.006	0.006
			Hz	Hz									
2	BW	10,884	181.48	181.39	181.17	180.99	180.82	180.64	180.45	180.25	180.04	179.81	179.57
		rpm	Hz	Hz									
3	FW	12,283	203.12	204.7	205.69	205.99	206.18	206.39	206.64	206.93	207.24	207.59	207.96
		rpm	Hz	Hz									
4	BW	16,893	275.87	280.56	283.01	282.68	281.78	280.62	279.34	278 Hz	276.66	275.29	273.96
		rpm	Hz		Hz	Hz	Hz						
5	FW	18,681	286.91	305.02	316.4	319.87	322.55	325.35	328.42	331.81	335.44	339.35	343.51
		rpm	Hz	Hz									
6	FW	30,416	506.93	506.93	506.93	506.94	506.94	506.94	506.94	506.94	506.94	506.94	506.94
		rpm	Hz	Hz									
7	BW	141,830	2501.8	2494.3	2481.7	2469.5	2457.3	2445.1	2433	2420.9	2408.9	2397	2385.1
		rpm	Hz	Hz									
8	FW	160,800	2505.1	2516.4	2531.4	2544.4	2557.1	2569.1	2582.8	2595.7	2608.6	2621.7	2634.7
		rpm	Hz	Hz									
9	BW	0 rpm	6261.7	6259.7	6259.9	6244.7	6233	6219.2	6203.9	6187.4	6170.1	6152	6133.4
			Hz	Hz									
10	BW	0 rpm	6329.9	6329.9	6330	6330.2	6330.4	6330.7	6331	6331.4	6331.9	6332.4	6332.9
			Hz	Hz									
11	FW	0 rpm	6442.6	6444.2	6448.9	6456.1	6456.2	6475.5	6486.6	6498	6509.5	6521	6532.1
			Hz	Hz									
12	FW	0 rpm	6998.4	6997.9	7000.4	7002.9	7006.4	7010.9	7016.4	7022.9	7030.4	7039	7048.6
			Hz	Hz									



Fig. 9. Bode plot showing amplitude and phase angle as a function of frequency for a given unbalance.

After initial analyses, it was concluded that at low speeds certain forms of vibration may occur, but in the operating range of 60,000 to 120,000 rpm, none of them should appear, assuming that the bearing parameters are reached as those selected in the analyses.

3.4. Uncertainty in numerical results

The uncertainty in the numerical results was primarily due to the adopted model simplifications and the accuracy of the calculation methods. In the numerical analysis, the foil bearings were modelled as a system with constant stiffness and damping, simplifying realworld conditions where these parameters can vary dynamically with rotational speed and load. The simplifications used are presented in the points below:

- A linear approximation of the properties of the foil bearing, although, in reality, the stiffness and damping are highly non-linear.
- The assumption of continuous support of the rotor by the lubricating film, whereas, in reality, there may be contact between the journal and the foil, which could have affected the dynamics of the system.
- Disregarding the effect of temperature, which alters the gas viscosity and bearing properties under actual operating conditions.
- The use of a tet10 and hex20 grid, which, while providing good computational efficiency, may limit the accuracy in representing the geometry of the foil bearing.
- Approximation of the gyro damping matrix (Coriolis), which could have affected the accuracy of the results in the higher natural frequency range.
- Assumption of ideal boundary conditions, whereas local deformations may occur in the actual structure, affecting the dynamic response of the system.

The numerical convergence criteria that were applied in the calculations:

- Modal analysis: The Lanczos iterative method was used, with a residual tolerance of 10^{-6} , which ensured the stability of the natural frequency values and the accuracy of the mode shape extraction.
- Harmonic analysis: A convergence criterion of 10^{-5} was adopted, limiting numerical errors to less than 0.01% and ensuring the accuracy of the vibration amplitude results.
- Grid independence test: A study of the effect of grid density was conducted, which showed that further increases in the number of elements resulted in natural frequency changes of less than 2%, confirming the stability of the numerical results.
- Solver stability: An Augmented Lagrange contact formulation was used to ensure the accuracy of the modelling, and gyroscopic damping (Coriolis effects) was taken into account to better represent the actual operating conditions of the rotor system.

Despite these limitations, the numerical model showed good agreement with the experimental results (differences of less than 10%), confirming its suitability for analysing the dynamics of a rotor supported by foil bearings.

4. Experimental studies using air

During the experimental studies, many quantities were measured to assess not only the dynamic state of the machine but also the proper functioning of the bearings. The compressor designed as part of the study was ultimately intended to operate with R134a refrigerant. However, in the initial phase of the research, it was decided to conduct tests using air. This decision was driven by both technical and safety considerations and aimed at verifying the dynamic stability of the compressor before its integration into the complete refrigeration system.

The main focus of the tests was to determine the rotor's behaviour in terms of:

- the operating stability of the foil bearings,
- · axial clearance selection,
- · potential issues with the rotor discs rubbing against the seals,
- temperature control and the effects of friction on bearing components.

There are many examples in the literature indicating that preliminary tests of newly designed rotor systems are conducted with air before being integrated into the final working medium [57]. This decision was made to reduce potential risks and eliminate technical issues before progressing to the testing phase with the actual refrigerant. Refrigerants such as R134a require a sealed, high-pressure system, which significantly increases the complexity of the experiment and introduces additional safety requirements. In addition, if during the tests it was found that any of the components required redesigning, the entire refrigeration system would need to be rebuilt in the case of the pressure system, which would incur high costs and increase the testing time.

The air also allowed for:

- easy identification of potential leaks,
- · control and regulation of pressure within the compressor casing,
- analysis of the effect of pressure on the dynamic behaviour of the system.

In the next stage of the study, the compressor will be integrated into a complete refrigeration system using R134a, which will allow its operational characteristics to be analysed in detail under real conditions.

A precision measurement system, including displacement sensors, accelerometers, and a rotational speed sensor, was used to record the dynamic parameters of the rotor.

- The displacement of bearing journals in two directions (X, Y) was measured using fibre-optic displacement sensors (Philtec RC32) with a sensitivity of 2.5 mV/ μ m, an overall range of 2 mm, a linear range of 0.75 mm, and a sampling frequency of 20 kHz.
- The axial displacement of the rotor was measured using a separate Philtec RC32 sensor with the same parameters as the journal displacement sensors.
- Bearing housing vibration was measured using IMI HT622A01 accelerometers with a sensitivity of 100 mV/g, mounted at the bearing attachment points.
- Rotor rotational speed was recorded using an Optel-Texys 152 G7 optical speed sensor, synchronised with the data acquisition system (sampling rate: 260 kHz).
- Data recording was carried out in real time using the Siemens SCADAS system, which enabled high-resolution measurements and signal analysis over a wide frequency range.
- The results were analysed using Test.Lab software, which allowed for a detailed evaluation of the vibration trajectories and rotor dynamics.

All the sensors were calibrated before the measurements commenced. The displacement sensors were calibrated using a calibration instrument, and their exact sensitivity was input into the data acquisition system. Similarly, the accelerometers were calibrated using a sinusoidal excitation method to ensure the accuracy of the readings. After calibration, the initial movement tests were conducted to verify the correct operation of the system before detailed dynamic measurements were performed. The checks included:

- · Verifying that the compressor starts correctly,
- · Ensuring the axial clearance has been properly selected,



Fig. 10. Course of the displacement of the journals in both radial directions and the rotational speed curve.

- Confirming there is no overheating of the foil bearings,
- · Ensuring there is no rubbing between the rotor discs and seals,
- Assessing whether the system is dynamically stable over a wide range of rotational speeds.

The course of the tests involved gradually increasing the rotor's rotational speed and recording the dynamic parameters, as shown in the article. The compressor was tested over a range of rotational speeds from X to Y rpm, passing through the first and second critical rotor speeds. Analysis of the results made it possible to assess the influence of bearing clearances and dynamic rotor loads on vibration amplitudes and system stability.

Fig. 10 shows the displacement of bearing journals in radial directions and the rotational speed as a function of time. The top journal moved about 10 μ m in the X direction in the bearing, while it took much longer for the second journal's similar displacement in the Y direction to stabilise. The bottom journal could not find the point of static equilibrium until a speed of about 40,000 rpm, which was due to the support of the rotor by the starting bearing. At a higher speed, when the rotor separated and began to levitate, the position of the journal changed significantly, then stabilised. The journal moved about 7 µm in the X direction and about 25 µm in the Y direction. Fig. 11 shows the course of the rotor displacement in the axial direction, which, as can be observed, is similar in nature to the displacement of the lower journal in the Y direction. When the speed was 40,000 rpm, the rotor moved in the direction of the axial force. The starting position of the rotor was approximately -0.10 mm as the rotor did not always drop down onto the starting bearing. During free rundowns, without a specified time, it is the foils that clamped the journals faster than the rotor dropped down onto the lower bearing. On the other hand, when the rundown time was defined, the rotor would usually drop down onto the starting bearing. A value of 0 on the ordinate axis indicates contact between the rotor and the starting bearing.

As can be seen in the diagrams above, the static equilibrium position of the journals depends on the axial position of the entire rotor. The starting bearing significantly affects the operational stability of the lower journal (which in turn affects the operation of the upper journal), which can be caused by misalignment of the starter bearing and the lower journal or by the excessive geometric clearance between these two components. The figures below present waterfall plots showing the frequency distribution of the journal vibrations in each direction measured during the start-up and rundown. Fig. 12 shows the amplitude–frequency graphs for the upper journal. As can be seen in this distribution, there are many harmonic components. In the X direction, the 2X, 3X and 4X components had a greater amplitude than the 1X component originating from the unbalance. The 2X harmonic component, which had a value of approximately 0.012 mm, was dominant over the entire rotational speed range. The synchronous component (1X) did not exceed a value of 0.005 mm. In the other direction (Y direction) it was completely different. Higher harmonics were visible but had significantly lower vibration values. The synchronous component (1X) was dominant over the entire rotational speed range and its maximum value was 0.028 mm. In addition, in this direction, the journal could not find static equilibrium for a long time.

As shown in Figs. 12 and 13, multiple harmonics are clearly visible in the vibration spectrum of the second journal. Frequency analysis showed that the X-direction was dominated by the 1X and 3X components, with a maximum amplitude of 0.009 mm. In the Y direction, a clear increase in amplitude to 0.023 mm was observed for the 1X component. Similar effects have also been observed in the literature for gas and contact bearings [39].

Fig. 14 shows the vibration of the rotor in the axial direction. The dominant component of the vibration was 1X with a maximum amplitude of 0.290 mm, which occurred at low rotational speeds. After the lift-off speed was exceeded, the vibration level decreased significantly, but not to the level expected, because the axial force generated by the rotor disks running on air was not high enough. The rotor operated at a distance of about 0.1 mm from the axial bearing. As could be observed, the vibration level of the rotor in the axial direction did not have a significant impact on the vibration values of the journals in the radial directions.

The appearance of harmonic components was probably caused by the excessive clearances of the foil bearings and by the angular misalignment of the bearings. Similar symptoms were observed by researchers of the rotor system that operated with rolling bearings [39], where it was easier to maintain a common axis for the bearings. The researchers identified the problem as being related to bearing clearances, which had a major impact on the vertical-axis rotor because the bearings were not sufficiently loaded. The figures below (Figs. 15-16) show the vibration of the machine recorded by accelerometers installed on the machine. It can be observed that the vibration spectra are much cleaner than those showing the vibration of the rotor. There are not many harmonics visible, which may be due to the damping properties of the foil bearings. In each spectrum, 1X and 2X components are visible, with the latter being dominant. This may indicate a misalignment of the journals caused by the different parameters of the foil bearings used. The lubrication clearance was probably too big or there was a misalignment.

The vibration of the housing, recorded at the location of the upper journal, did not exceed 0.7 g RMS in terms of vibration acceleration, while the vibration velocity was 0.6 mm/s RMS. To refer to the standard in which the vibration velocity values are specified, the graph







Fig. 12. Waterfall plots showing upper journal vibrations in the X and Y directions - (a) and (b), respectively.



Fig. 13. Waterfall plots showing lower journal vibrations in the X and Y directions - (a) and (b), respectively.



Fig. 14. Waterfall plot showing rotor vibrations in the axial direction.

shown on the right has been created. The standard (ISO 20816-1:2016) states that for new machines with a capacity of up to 15 kW, the value of the vibration velocity must not exceed 0.71 mm/s RMS. It is also the case with the location of the lower journal, where the vibration

acceleration values do not exceed 0.170 g RMS, so they are much smaller than those obtained for the upper journal. At maximum speed, frame vibration occurred, in the frequency range of 3500–4800 Hz, which caused the rotor to vibrate. The vibration velocity did not exceed 0.4 mm/s RMS, which according to the standard indicates that the machine can continue to operate. In both cases, there were constant vibration frequencies (approximately 1600 Hz and 2660 Hz) which were associated with the natural vibration modes of the compressor support structure. This structure was not bolted to the foundation.

No instability of the bearing lubrication film (manifesting as vibrations of approximately 0.5X) was observed in the vibration spectra above. The rotor vibrations recorded by the displacement sensors suggest that the vibration trajectories of the bearing journals are neither circles nor ellipses. This is why the vibration trajectories were plotted. 1X filters are commonly used to evaluate the dynamic state of machines. There are many harmonic components in the case described. Therefore, the figures below present the vibration trajectories for different speeds, which were obtained using a 1X filter and a low-pass filter up to 4X. The trajectories obtained using the low-pass filter are plotted



Fig. 15. Vibration spectrum of the upper journal presented as vibration acceleration - (a) and (b), respectively.



Fig. 16. Vibration spectrum of the lower journal presented as vibration acceleration - (a) and (b), respectively.



Fig. 17. Vibration trajectories of both journals for rotational speeds of 12,701 and 22,157 rpm - (a) and (b), respectively.

in black. On the other hand, the vibration trajectories obtained with the help of the 1X filter are plotted in red. Counting from the left, the first and third diagrams show the vibration trajectories of the journal of the first bearing (upper bearing), and the second and fourth diagrams show the vibration trajectories of the journal of the second bearing (lower bearing). Fig. 17 shows the vibration trajectories obtained at speeds of 12,701 rpm and 22,157 rpm during the run-up. At the lower speed, both journals had similar vibration amplitudes. Given the nomenclature of trajectory shapes used in the article [58], the trajectory of the journal of the lower bearing (bearing No. 2) can be classified in the class of "tornado" (T) trajectories with forward precession (1X). In the case of the upper journal (marked as "BEARING 1" in Fig. 17), it is the "heart" (H) and backward precession (1X) occurs there, which may be due to the smaller lubrication clearance that resulted in the absence of the lubrication film or the presence of a very thin lubrication film. These two trajectory shapes are indicative of the misalignment of the journals. At the higher speed (side B of Fig. 17), the direction of precession remained unchanged but the vibration amplitudes decreased. Asymmetrical movement of the journal was observed in the lower bearing, which would confirm the misalignment of the two journals caused by the support in the form of a thrust bearing.

At a speed of 30,218 rpm (side A of Fig. 18), the precession movement of the two journals remained unchanged. However, the vibration amplitude of the lower journal increased and the trajectory shape changed to "heart" (H). After the speed increased to 39,871 rpm, i.e. just before the rotor separated from the starting bearing (side B of Fig. 18), the vibration amplitudes of the upper journal (bearing No. 1) were approximately 50 μ m (p-p) and the component originating from the unbalance (1X) became dominant. Next to the lower bearing, the vibration amplitude was 60 μ m (p-p) and the trajectory changed its shape to T.

After separation of the rotor from the starting bearing (left side of Fig. 19), the vibration amplitudes and the nature of the vibrations were similar to those of the previously described state. It was expected that the upper bearing precession would turn into forward precession but the lubrication film was apparently still too thin for that to happen. After the speed increased by about 10,000 rpm (right side of Fig. 18), the vibration amplitudes of both journals decreased.

On the side A of Fig. 20, the vibration trajectories of both journals for the speed of 60,118 rpm are presented, where it can be seen that the vibration amplitudes have not decreased but that the precession movement of the two journals has become concurrent. In addition, vibrations caused by misalignment are also visible. After the speed exceeded 70,000 rpm (side B of Fig. 20), there was a substantial increase in the share of the 1X component (originating from the unbalance) in the vibration spectrum. The effect of the lower journal on the upper journal can also be observed. This was due to the fact that the rotor had no axial support; its speed did not allow it to generate an axial force large enough for the keep plate to approach the axial bearing. On the other hand, the stiffness of the lower bearing was too low. It is important to remember that the tests were performed with air and the target working fluid has different properties.



Fig. 18. Vibration trajectories of both journals for rotational speeds of 30,218 and 39,871 rpm - (a) and (b), respectively.



Fig. 19. Vibration trajectories of both journals for rotational speeds of 40,980 and 50,005 rpm - (a) and (b), respectively.



Fig. 20. Vibration trajectories of both journals for rotational speeds of 60,118 and 70,048 rpm - (a) and (b), respectively.

After the speed exceeded 60,000 rpm, the rotor continued to exhibit compound movements as if it was bouncing off the top foil, but the direction of precession indicated that no rubbing had occurred. In addition, the share of 1X vibrations in the 4X trajectory became larger with increasing speed. Throughout the speed range, both journals had an elliptical shape and were sometimes strongly flattened. This could have been caused by the reduced stiffness in the Y direction, or an additional excitation force acting in the X direction.

5. Discussion

Let us first analyse the position of the bearing journals. Fig. 21 shows their position in the X-Y plane as a function of rotational speed. The green colour indicates the stable speed period, and the red and blue colours indicate the run-up and coast-down, respectively. The black colour indicates the top foil and its position in a bearing. In addition, a purple dot marks the moment when the rotor separated from the starting bearing. The upper journal changed position within the bearing, regardless of whether the speed was constant or variable. Furthermore, the distance between the start point and the end point is about 13 μ m. The lower bearing journal changed position rapidly when it started to levitate. When the speed was constant, it moved around the point of static equilibrium, unlike the upper bearing journal. On the other hand, at variable speed, it did not move along a curved line in the bearing as it was in the upper bearing. The misalignment of

the bearing journals can also be observed because the nature of their position changes was not similar. It is possible that the natural vibration modes that occurred during speed changes forced such movements of the journals. It is difficult to plot a clearance circle for foil bearings because the foils constantly deflect. A lubrication clearance of 20 μm was assumed in both bearings, but only the total clearance (lubrication clearance + deflection of the foil assembly) could be estimated from the diagram below. For the first bearing, it was 17 μm and for the second 33 μm . These values were determined for the most extreme positions of the journal in the bearing in the Y direction, where rubbing or greater deflection of the top foil could have occurred.

The observed movements of the journals may have been caused by an unknown external excitation force originating, for example, from the heating of the journals when they rubbed against the top foil. It is common in the literature to detect this symptom by analysing the vibration amplitudes of the journals after using a 1X filter. In the studied case, such a possibility existed because the journals operated with high eccentricity. Fig. 22 shows the curses of the run-up, steady state and coast-down. Similar vibration levels were recorded at the beginning and end of the test. Elevated vibration levels were recorded only on the compressor housing and occurred at low speeds (i.e. 13,144 rpm, 20,000 rpm, 27,722 rpm, 24,659 rpm and 16,924 rpm). In the forced vibration analysis (Fig. 9), the vibration amplitudes were very small (about 1 μ m). Three times higher values were obtained for rotor vibrations in the X direction (side B of Fig. 22). However, the values



Fig. 21. Journal positions in the bearings during the test (together with rotational speeds).



Fig. 22. Vibration amplitudes measurement by accelerometers (a) and displacement (b) sensors (1X).

obtained for the other direction were much higher (about 17 μ m). If the bearing clearance were lower than nominal, the resonance response in terms of critical speeds and subsynchronous instability effects would be weaker in the operating speed range [59]. After analysing side B of Fig. 22, no signs of rubbing manifesting as a local increase in vibration amplitude with a flat top were noted. As shown earlier, rubbing may have occurred but did not affect the vibration amplitudes. Therefore, it can be said that the rubbing did not affect the dynamics of the rotor.

After comparing and analysing the characteristic vibration frequencies obtained from the modal analysis and the forced vibration analysis as well as those read from the vibration waveform, it can be said that they are similar to each other. A comparison of the modal analysis and experimental results (Table 5) shows that the differences in natural frequencies do not exceed 10%, confirming the accuracy of the simplified numerical model. It should be noted that similar differences between the calculations and experimental results have been observed in other foil bearing studies [49,50].

The occurrence of differences in results between simulation and experiment is a common phenomenon. This difference is most often due to the fact that the numerical model never fully reflects the experiment, as it cannot take all conditions into account. In addition, the greater the number of variables characterising the model, the longer the calculation time. For the purpose of time optimisation, the model is usually simplified to reflect the experiment to some satisfactory extent. In the case of a result that differs by 8.9%, this is a satisfactory value, considering that the average of all results is estimated to be around 5%. Three characteristic frequencies recorded on the compressor housing were observed during rotor run-up (side A of Fig. 22) and two were observed during rotor rundown. They could also be the natural frequencies of the support structure. On the rotor vibration waveform (side B of Fig. 22), it is not possible to indicate the rotational speeds at which the increased vibration levels occur. The differences between the values obtained for the natural frequencies and vibration amplitudes of the rotor may be due to an erroneous assumption since static stiffness and damping coefficients were adopted for the foil assembly in the calculations. However, these parameters in vertical machines will depend largely on the position of the journal in the bearing. The authors of the article [38] suggest using dynamic coefficients that depend on the position and frequency of the excitation.

In the Campbell diagram, shown in Fig. 7, it can be seen that the rotor's mode shapes can be excited by higher harmonics such as 2X, 3X, and subsequent components. This phenomenon is particularly important for rotor dynamics, as it can influence the stability of the system across a wide range of rotational speeds. Further analysis, based on the waterfall plots shown in Figs. 11–14, demonstrates the amplitude of the rotor displacement vibrations in different directions as a function of rotational speed and frequency. 2X and 3X harmonics, and, in some cases, subsequent harmonics, can be observed in these plots. However, in the analysis of the results, no clear excitation of the natural frequencies by the higher harmonics was observed, and their unambiguous identification proved difficult.

The time taken for the rotor to accelerate to a speed of 70,000 rpm was approximately 10 s, indicating a relatively fast start-up process. This may be the reason for the lack of clearly visible critical speeds in the displacement path of the bearing journals, as the dynamic changes in the system have not had sufficient time to fully develop. However, in the plots from Figs. 15 and 16, where the vibration acceleration for the upper and lower bearings is shown, it can be observed that the 2X component excites the natural frequencies of the rotor or compressor structure at approximately 1500 Hz. This is an important observation, as it suggests that the system may be susceptible to the influence of higher harmonics within a certain frequency range. Further research should involve a more detailed analysis of this effect, particularly with regard to changes in bearing geometry and possible nonlinear

Table 5

Natural frequencies obtained from modal analysis, forced vibration analysis, and the vibration waveform of the housing.

Mode No.	Natural frequencies obtained in different ways								
	Modal analysis	Forced vibration analysis	Increased vibration level - 1X	Percentage difference					
1	181.48 Hz	188.95 Hz	_	4%					
2	204.70 Hz	216.95 Hz	219 Hz	6.5%					
3	283.01 Hz	304.92 Hz	282 Hz	7.5%					
4	319.87 Hz	-	333 Hz	3.9%					
5	506.94 Hz	-	462 Hz	8.9%					
6	2445.1 Hz	2352.4 Hz	-	3.8%					



Fig. 23. Full spectrum of the amplitudes of the forward and backward circular components of elliptical orbits; (a) upper journal, (b) lower journal.

dynamic interactions that may influence vibration amplitudes and their propagation within the rotor system.

To better illustrate the nature of the operation of the machine, the correlation between the vibrations measured by the two sensors has been visualised. The full spectrum allows us to determine the components of orbital motion and precession (Fig. 22). This is often used in machine diagnostics [60] to detect undesirable symptoms. For this purpose, the following correlations were used, where Fa_n and Ba_n are the components of the amplitudes of the forward and backward precession.

$$Fa_n = \sqrt{Ax_n^2 + Ay_n^2 + 2 \times Ax_n \times Ay_n \times \sin(\alpha_n - \beta_n)}$$
(6)

$$Ba_n = \sqrt{Ax_n^2 + Ay_n^2 - 2 \times Ax_n \times Ay_n \times sin(\alpha_n - \beta_n)}$$
(7)

where, Ax_n and Ay_n stand for vibration amplitudes in the *x* and *y* directions and α_n and β_n stand for vibration phases in the *x* and *y* directions [61].

From Fig. 23 it can be seen that for the upper bearing, during run-up and coast-down, 1X was the dominant component for both precession directions, meaning that the vibration trajectory is a flat, corrugated ellipse throughout the range. After reaching a speed of about 60,000 rpm, the amplitude of the 1X component of the forward precession increased (rounded trajectory). For the upper bearing, no rubbing (1/2X) was observed over the entire range, which was not the case for the lower bearing, where components from the range of 0.32X – 0.47X appeared at low rotational speeds (aerodynamic instability). In this bearing, the motion of the journal was characterised by the forward precession of the rotor axis. Harmonics were visible for both bearings, but for the lower bearing, the 2X and 3X harmonics had significant amplitudes.

In the above figures showing trajectory plots for different rotational speeds (Figs. 17-20) and in Fig. 23, it can be seen that the upper bearing rotated in the opposite direction to the direction of rotation of the rotor and the lower bearing rotated in the same direction as the rotor. This is a rare phenomenon that has been already described [62]. The reasons for such operation may be related to the parameters of the rotating system (rotor stiffness, masses) and/or the bearings. Assuming that the stiffness and mass characteristics are constant, the factors that affect the behaviour of this rotor are damping, excitation force, vibration amplitude and angular orientation of the rotor, mass unbalance. The author of the book [62] also gives a way to check the direction of rotation of the rotor. This involves plotting the difference between the vectors of the trajectories obtained at the same time for the upper and lower journals, for 1X. An example of this process is shown on the side A of Fig. 24, where the trajectories, obtained for a speed of 41,980 rpm, are divided into 10 points which are numbered according to time. The differential trajectory for this speed was determined by connecting the points (black colour on the side B of Fig. 24). Identical operations were carried out for higher rotational speeds, namely 50,005 and 60,118 rpm (side B of Fig. 24). As can be seen, the selected differential trajectories are forward trajectories and, most importantly, the vibration phase does not change significantly. Rotor operation was correct in this speed range, however, there must have been a specific distribution of unbalance along the rotor axis causing these symptoms to appear [61].

As illustrated in Fig. 24, the vibration trajectories took the shape of flattened ellipses, which could be attributed to a reduction in stiffness in the Y direction or the presence of an additional excitation force in the X direction. These effects are consistent with previous studies on the dynamics of rotors supported by foil bearings, which have shown that lubrication clearance and misalignment can lead to characteristic trajectory asymmetries [58]. In the Y direction, that is, in the direction



Fig. 24. (a) Vibration trajectories of the upper and lower journals (backward and forward trajectories respectively) at a speed of 40,980 rpm; (b) Differential trajectories for speeds of 41,980, 50,005 and 60,118 rpm.

of the major axis of the trajectory (Fig. 24), there was a top foil lock, which is always characterised by increased stiffness. Thus, the journal trajectories were flattened due to an increased stiffness value in the X direction (minor axis of the trajectory – Fig. 5) or an external force acting in this direction, coming, for example, from the flow in the compressor disks.

6. Summary

Based on the differential trajectories, it was proved that the rotor vibrations were not disturbing. Even if any rubbing of the journals against the top foil occurred, it did not affect the operation of the rotor. An amplitude–frequency response containing many synchronous components was obtained. The reason for such rotor behaviour was excessively large bearing clearances, which, according to the authors, had a significant impact on the rotor's dynamics.

In addition, different bearing clearances resulted in the misalignment of the journals. The reduction of the lubrication clearance by clamping the foil assembly around the shaft will increase the resistance to the motion of the rotor, which will result in increased damping and thus increased preload, resulting in even better operation of the rotor. No subharmonic components were recorded, which often occur in machines with conventional hydrodynamic bearings, demonstrating that foil bearings are a good choice for oil-free machines that operate vertically. Higher-order harmonics can also have an impact on the high-amplitude modal forms of the rotor associated with partial shaft deformation, increasing the effect of many harmonics.

It has also been observed that appropriately selected foil bearings can operate with a vertical rotor. An interesting and little-known spinning effect was observed when one of the journals rotated in the direction of rotation and the other in the opposite direction. A way was found to analyse such a phenomenon, allowing for the verification of the correct operation of the compressor's rotor.

In the next study of the machine, the run-up time to a speed of 60,000 rpm should be short, which should stabilise the rotor at low rotational frequencies. Much higher vibration levels were observed in the direction of the top foil lock (Y) compared to the direction perpendicular to the lock. This is due to the lack of support at this location (within the lock).

7. Conclusion

In this study, the dynamic performance of the rotor of a high-speed, vertical-axis refrigeration compressor, supported by foil bearings, is analysed . The results of the experimental research confirmed that foil bearings can ensure stable operation over a wide range of rotational speeds, with bearing clearance being a key factor affecting system behaviour . Good agreement was observed between the experimental and numerical results, with differences in natural frequencies not exceeding 10%, confirming the validity of the simplified model used.

It was found that excessive bearing clearance may lead to unstable vibration trajectories and the excitation of higher harmonics, whereas too small a clearance results in increased temperatures and a risk of journal-to-foil contact. In the tested configuration, the optimal clearance was found to be 20 μ m. The Campbell diagram and vibration acceleration signals revealed the presence of 2X and 3X harmonics, with at least one likely exciting the natural frequencies of the structure.

The numerical model did not account for the effect of temperature, which may partly explain the discrepancies observed in comparison with the experimental results. Nevertheless, it demonstrated high numerical stability, supported by a mesh independence test and appropriate convergence criteria. The results obtained indicate that precise adjustment of the foil bearing geometry and control of the clearance are essential for ensuring the dynamic stability of the system. The proposed research methodology can also be applied to the analysis of other turbomachines, such as microturbines, compressors, or expanders.

Further research should focus on extending the numerical model to include the full interaction between the gas film and the support structure, which would allow for a more accurate representation of the actual behaviour of the foil bearings, particularly in terms of their damping and dynamic stiffness.

CRediT authorship contribution statement

Paweł Bagiński: Writing – review & editing, Writing – original draft, Visualization, Validation, Supervision, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Artur Andrearczyk:** Writing – review & editing, Writing – original draft, Visualization, Validation, Software, Investigation, Data curation. **Paweł Ziółkowski:** Writing – review & editing, Project administration, Funding acquisition.

Declaration of competing interest

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: Pawel Ziolkowski reports financial support was provided by Gdansk University of Technology. Artur Andrearczyk reports financial support was provided by National Centre for Research and Development. Pawel Baginski reports financial support was provided by National Centre for Research and Development. Pawel Ziolkowski reports a relationship with Gdansk University of Technology that includes: employment. Artur Andrearcyzk has patent #Pat.242547 licensed to Institute of Fluid-Flow Machinery, Polish Academy of Science. Pawel Baginski has patent #Pat.242547 licensed to Institute of Fluid-Flow Machinery, Polish Academy of Science. If there are other authors, they declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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