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Introduction

The analysis of dynamic properties of rotating machinery has for many years been the subject of numerous research studies carried out in many scientific centers. For modern day rotating machinery it is required to work with increasingly difficult operating parameters while maintaining a light and compact design. Increased efficiency, reliability, and precision are also required. Rotating machinery with hydrodynamic bearings is used in many sectors of the economy, e.g. energy, transport, aviation, and military. Very often they are a key element of large technical objects.

In steam turbines used for energy conversion, one of the key components are hydrodynamic plain bearings. These machines are referred to as "critical machinery," i.e. they are required to be extremely reliable. Unplanned downtime due to poor technical condition leads to significant financial losses. They are therefore monitored and thoroughly analyzed.

The starting point for the analysis of hydrodynamic radial bearings are the equations of motion. From the mathematical perspective, we are dealing with non-linear differential equations (motion equations of the entire rotor) related to the system of partial differential equations describing the properties of plain bearings and the supporting structure. This association occurs through the stiffness and damping coefficients of hydrodynamic bearings. These coefficients determine the dynamic properties of the bearings. Linear and non-linear numerical calculation models are available, but the limit from which the much more complex and time-consuming non-linear models must be used is not clearly defined. An experimental (using a linear algorithm) and numerical (using linear and non-linear algorithms) calculation of dynamic coefficients of bearings for a wide range of rotational speeds was carried out in order to elaborate on the issue formulated in the title of this monograph.

1.1 Current State of Knowledge

The values of stiffness and damping coefficients have a decisive impact on the analysis of rotary machine vibrations. During the dynamic analysis of the rotating shaft, it is necessary to build a discrete model, define the boundary conditions, and calculate the values of these coefficients. The stiffness and damping coefficients change with rotational speed. For the majority of bearings in operation, the coefficients are non-linear in nature,

Bearing Dynamic Coefficients in Rotordynamics: Computation Methods and Practical Applications,
First Edition. Łukasz Breńkacz.

Companion website: www.wiley.com/go/brenkacz/bearingdynamiccoefficients

which means that they change in time and are dependent on the driving force (Kiciński 2006). For most jobs, a linearized form of coefficients is used, which means that they have constant values for a given speed and do not change for different values of driving forces. It should be remembered that dynamic coefficients also change with changes in operating temperature, bearing supply pressure, and bearing load (Hamrock et al. 2004).

For many years at the Institute of Fluid Flow Machinery in Gdańsk under the management of Professor Jan Kiciński, programs from the MESWIR series for numerical calculation of dynamic coefficients of hydrodynamic bearings and rotor dynamics together with imperfections have been developed. A key element in the calculation is the appropriate determination of the stiffness and damping coefficients of hydrodynamic bearings. In the NLDW software (one of the programs of the MESWIR series) it is possible to determine the non-linear form of these coefficients. This is the basis for further dynamic analysis of the entire system, which can be conducted on many planes. Unfortunately, only an indirect comparison of the results of numerical calculations with a real model is possible, i.e. by comparing the amplitudes of vibrations measured on the basis of experimental tests and the amplitudes of vibrations calculated on the basis of numerical analyses. This work presents a method which enables dynamic coefficients of bearings to be determined on the basis of experimental research. They can be directly compared with the results of numerical tests.

Numerical determination of stiffness and damping coefficients of bearings is most often performed by solving the Reynolds differential equation. For linear systems with known parameters such calculations are performed with very high accuracy (Duff and Curreri 1960; Giergiel 1990). For systems for which it is necessary to describe using methods with non-linear calculation algorithms (Hayashi 1964), or methods with complex structure with unknown parameters, calculations become more complicated and are often plagued with significant errors (Fertis 2010). During the calculation it is necessary to take into account the relationships between different parts of the system such as rotor, bearings, and supporting structure (Kiciński and Żywica 2014a; Mikielewicz et al. 2005).

The principles of hydrodynamic lubrication of bearings and the principles of formation of a wedge of lubricant are described in many books on the theory and practice of plain bearings (Neyman and Sikora 1999; Wierzcholski 1994). Numerical linear and non-linear algorithms for calculating stiffness and damping coefficients of hydrodynamic radial bearings are presented in an accessible way in the book *Teoria i badania hydrodynamicznych poprzecznych łożysk ślizgowych* (*Theory and Research on Hydrodynamic Radial Plain Bearings*) (Kiciński 1994). The hydrodynamic bearing can be treated as a mechanical system consisting of elements of varied stiffness. The properties of the bearing are determined by the lubricating film that forms between the journal and the bearing housing (Sikora 2009). Bearing properties are defined as the properties of a lubricating film.

A schematic view of a hydrodynamic radial bearing is shown in Figure 1.1. In the center of the bearing housing there is a rectangular system of coordinates X_p, Y_p with the center marked as O_p. Under conditions of static equilibrium a relationship occurs (1.1), where W_0 is the reaction of the lubricating film to the force P_0, while X_p, Y_p, Y_p determine the position of the center of the journal O_c.

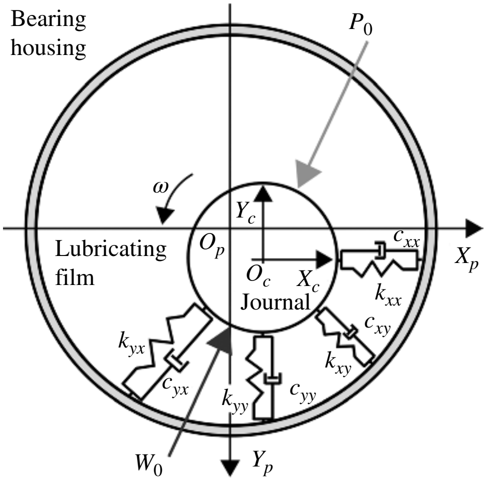

Figure 1.1 Lubrication film model for small journal displacement.

$$W_0 = f\left(x_p, y_p\right) \tag{1.1}$$

It is important to note that the $f(x_p, y_p)$ function, which determines the relationship between the reaction of the film and the displacement of the journal, is non-linear, and this non-linearity increases strongly with the increase of displacements of x_p, y_p. The reaction W_0 can be written as the sum of the reactions W_x and W_y in two perpendicular directions. Definition of the stiffness coefficients of the lubricating film are determined by Eqs. (1.2) and (1.3):

$$k_{xx} \approx \frac{\Delta W_x}{\Delta x} = \frac{\partial W_x}{\partial x}, \qquad k_{xy} \approx \frac{\Delta W_x}{\Delta y} = \frac{\partial W_x}{\partial y} \tag{1.2}$$

$$k_{yy} \approx \frac{\Delta W_y}{\Delta y} = \frac{\partial W_y}{\partial y}, \qquad k_{yx} \approx \frac{\Delta W_y}{\Delta x} = \frac{\partial W_y}{\partial x} \tag{1.3}$$

where ΔW_x, ΔW_y are the changes in the reaction of the film due to a small increase in force ΔP at the point of static equilibrium, and Δx, Δy are the changes in journal displacements due to an increase in force ΔP calculated at the point of static equilibrium. A characteristic feature of radial plain bearings is their inequality of "cross-coupled" reaction and displacement ratios, i.e. inequality of coefficients $k_{xy} \neq k_{yx}$. This is not a property of linear-elastic mechanical systems. It turns out that it is the cause of hydrodynamic instability.

The stiffness of the lubricating film, as an analogy of mechanical systems, often assumes that x_p, y_p displacements are slow. However, in hydrodynamic bearings, when the journal performs small, fast oscillations around the static equilibrium point, significant additional components of the W_x and W_y reaction may arise due to the resistance to motion in the viscous lubricant. In addition to the elastic properties, it is also necessary to determine the damping properties of the lubricating film. They can be formulated using Eqs. (1.4) and (1.5) in the same way as for the stiffness coefficients:

$$c_{xx} = \frac{\partial W_x}{\partial \dot{x}_c}, \qquad c_{xy} = \frac{\partial W_x}{\partial \dot{y}_c} \tag{1.4}$$

$$c_{yy} = \frac{\partial W_y}{\partial \dot{x}_c}, \qquad c_{yx} = \frac{\partial W_y}{\partial \dot{y}_c} \tag{1.5}$$

where $\dot{x}_c$, $\dot{y}_c$ mean the derivative of the displacement in time.

In numerical calculations, the following relationship is used, i.e. in case of damping there is no anisotropy of "cross-coupled" coefficients, i.e. $c_{xy} = c_{yx}$ (Kiciński 1994). Assuming small displacements x_c, y_c, elastic and damping properties of the lubricating film can be described by four stiffness coefficients and four damping coefficients. In numerical analyses three damping coefficients can be obtained. The components of the reaction of the lubricating film – W_x, W_y – can be represented by linear relationships (1.6) and (1.7):

$$W_x = k_{xx}x_c + k_{xy}y_c + c_{xx}\dot{x}_c + c_{xy}\dot{y}_c \tag{1.6}$$

$$W_y = k_{yx}x_c + k_{yy}y_c + c_{xy}\dot{x}_c + c_{xx}\dot{y}_c \tag{1.7}$$

The dynamic components W_x,W_y depend on the momentary position of the center of the journal x_c, y_c. Assuming that k_{xx}, k_{xy}, k_{yx}, $k_{yy} = k_{i,k}$ and c_{xx}, c_{xy}, c_{yx}, $c_{yy} = c_{i,k}$ these coefficients can be described by Eq. (1.8), where the functions F_1 and F_2 are non-linear. W_x and W_y relationships form the basis for a linear description of dynamic properties of the lubricating film and rotor–bearing systems (Dąbrowski 2013). They can be used to quickly and easily determine a number of very important characteristics of hydrodynamic bearings. Due to the non-linear nature of bearings, a linear description can only reflect the actual bearing properties and the system associated with them to a limited extent. In this monograph the results of calculations using linear algorithms are shown on a specific example of a laboratory test rig operating with parameters which should be described using methods with non-linear algorithms.

$$k_{i,k} = F_1\left(x_p, y_p\right), \; c_{i,k} = F_2\left(x_p, y_p\right) \tag{1.8}$$

The non-linear description is much more complicated than the linear one (Batko et al. 2008; Minorsky 1967; Skup 2010), but it allows (by determining the trajectory of the journal) to analyze e.g. self-excited vibrations of the system or to determine the influence of various types of external forces and damage. It provides much greater opportunities for theoretical analysis of the bearing's operation and the system associated with it. However, its use may be limited by the time of numerical calculations and computing capabilities of computers. As the computing power of computers increases, these limitations become less and less significant.

The methods most frequently used in non-linear analysis consist of the use of non-linear equations of motion and the principle of superposition to solve the Reynolds equation, where at each time point the part of the equation referring to the so-called combined effect of the rotational speed of the journal (the "kinetostatic" part) and to the so-called extrusion effect (the "dynamic" part) is solved separately. The above principle is the basis for the vast majority of numerical methods of determining the trajectory of a journal in the case of large displacement, e.g. methods developed by Booker, Block, and other authors (Blok 1975; Booker 1965). They have one common flaw, i.e. when the solutions are compiled different boundary conditions are used in the "kinetostatic" part and in the "dynamic" part of the Reynolds equation (Kiciński 1994).

It is possible to formulate a non-linear description without the above-mentioned flaw (Kiciński 1994). This description can be based on four predefined stiffness coefficients and four damping coefficients and is used in the MESWIR environment. If the whole interval of variation of the journal trajectory is divided into a sufficiently large number of successive subintervals at small intervals of Δt, it may be assumed that there are areas of the circle of backlash corresponding to those subintervals in which the stiffness and damping coefficients are approximately constant. Each such area will have constant coefficients which need to be determined. The characteristic feature of this method of determining the stiffness and damping coefficients is the fact that for each such area the full Reynolds equation is solved with only one boundary conditions (and thus without the use of superposition solutions). The dynamic components W_x, W_y depend not only on the momentary position of the center of the journal x_c, y_c, but also on the speed of change of this position $\dot{x}_c$, $\dot{y}_c$, according to Eq. (1.9) (Buchacz et al. 2013).

$$k_{i,k} = F_1\left(x_p, y_p, \dot{x}_p, \dot{y}_p\right),\ c_{i,k} = F_2\left(x_p, y_p, \dot{x}_p, \dot{y}_p\right) \tag{1.9}$$

The functions F_1 and F_2 are non-linear, as in the case of small vibrations around the static equilibrium point. Based on the above at each time point of the trajectory t_{k-1}, t_k, t_{k+1}, etc., and based on the determined stiffness and damping coefficients, it is possible to determine the increments of displacement Δx and Δy in a stepwise procedure. The accuracy of the calculation will increase as the time step decreases, i.e. as the areas in which the stiffness and damping coefficients can be assumed to be constant decrease. This relationship will be true until a certain point, and with too small time steps the accuracy of the calculations will be limited. Figure 1.2 illustrates the difference between linear and non-linear analysis based on stiffness and damping coefficients. The hatched fields show the areas where the coefficients are approximately constant.

The numerical analysis of vibrations with regard to large displacements of the journal carried out with the methods discussed above, regardless of whether it uses the principle of superposition or not, which is described as non-linear, is in fact a linear analysis of "pseudostatic" states (Kiciński 1994). In this context, bearing dynamics is a sum of separate "pseudostatic" states, in which time is treated as a parameter. In practice, the state of the bearing at any time t_k is influenced by phenomena occurring at moments preceding the moment under consideration, i.e. t_{k-1}, t_{k-2}, etc. This means that it is necessary to "memorize" the entire history of previous states and a continuous mathematical description in which time is no longer a parameter but is the third (next to geometric coordinates) independent variable.

In the case of a description with "prehistory", the time relationships used ensure that the Reynolds equation will not be integrated in an "empty" lubrication gap. In the MESWIR environment, where numerical analyses were performed, a continuous description with "prehistory" is used.

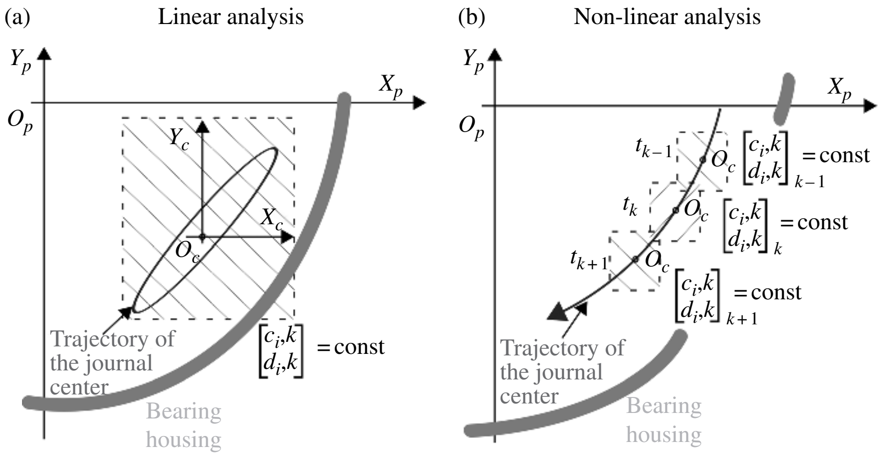

Figure 1.2 Methods of analysis for (a) small and (b) large displacement of the journal.

1.2 Review of the Literature on Numerical Determination of Dynamic Coefficients of Bearings

Attempts at a theoretical description of the operation of rotating machinery have been undertaken since objects with rotating elements began to be used in practice (Chmielniak 1997; Miller 1985; Zieliński 1969). The intensive development of these machines, and with it a more detailed description of them, took place at the turn of the eighteenth and nineteenth century. Significant progress in the theoretical description of rotating machinery began with pioneering works on turbines used for energy conversion (Bannik and Słuczajew 1956; Chodkiewicz 1998; Kiciński and Żywica 2014b; Łączkowski 1974; Nikiel 1956; Smolaga 1959; Wiśniewski 1974). The basis for the theoretical model of hydrodynamic plain bearings is the fundamental equation of hydrodynamic lubrication theory, i.e. Reynolds equation (Reynolds 1886). To this day we encounter some problems, the most important of which are (Barwell 1984): analysis of three-dimensional flow in the bearing lubrication gap, taking into account changes in viscosity, and determining the limits of integration of the pressure distribution curve (which is related to the occurrence of cavitation phenomenon). The impact of the supporting structure (Żywica 2008) is not insignificant either.

The first researcher who suggested applying Reynolds' theory to the calculation of slide bearing dynamics was S. Dunkerley in 1894 (Dunkerley 1894). In 1904 Sommerfeld solved the case of two-dimensional flow in plain bearings on the assumption that the liquid is able to transfer an unlimitedly high pressure (Sommerfeld 1904). Such a simplification caused the obtained result to be incorrect, but a dimensionless parameter determining the hydrodynamic similarity of bearings, the so-called Sommerfeld number, was described. D.G. Christopherson was the first to solve the Reynolds equation using the relaxation method (Christopherson 1941). In his work he stated that assuming that viscosity of oil is constant, it is possible to correctly determine the value of the bearing's load, the amount of oil flowing through the bearing, and temperature changes. The assumption made it impossible to determine the exact friction resistance and load direction. This method (with the application of finite differences) was implemented by C. Cryer in 1971 (Cryer 1971).

H.W. Swift (1932) and W. Stieber (1933) proposed a method of modeling cavitation by applying boundary conditions, currently called Reynolds boundary conditions in the literature. The study of hydrodynamic instability was carried out by, among others, A.C. Hagg and P.C. Warner, who proved that the cause of this phenomenon is oil whirls (Hagg and Warner 1953). Even the most complex bearing models enabled the analysis of the rotor–bearing system only in the range of small vibrations. It is a linearization of systems, which in reality often showed strong non-linear properties. A typical approach is to assume constant values for the stiffness and damping coefficients of the lubricating film, which are determined only for the kinetostatic equilibrium point.

A more detailed examination of the properties of hydrodynamic bearings was made possible with the use of numerical calculation methods. The most commonly used methods are the finite volume method and the finite difference method (Lund 1987; Someya 1989; Wasilczuk 2012; Zienkiewicz and Taylor 2000). The finite element method is most frequently used to analyze the bearing journal displacement (Andreau et al. 2007).

While discussing the development of contemporary computational models, one should mention the considerable achievements of the Turbine Dynamics and Diagnostics Department (formerly the Department of Dynamics of Rotors and Slide Bearings), which is a part of the organizational structure of the Institute of Fluid Flow Machinery of the Polish Academy of Sciences in Gdansk. Under the management of Professor Jan Kiciński, new computational models of plain bearings have been developed and experimental research has been carried out for many years (Kiciński 1994, 1988; Kiciński et al. 1997). It is worth mentioning achievements such as the formulation of a non-linear elastodiatermic model of a plain bearing (Kiciński 1989) and participation in the development of the DT-200 diagnostic system. It was installed on one of the units of the power plant in Kozienice. Also a software package called MESWIR was created. It enables an extensive analysis of bearings and complex rotor lines, even after exceeding the stability limit. The MESWIR package includes, among others, the following programs: NLDW, KINWIR, LDW, TRADYN, DIADEF, ADIAB, ISOSLEW, and DYNWIR. The first three programs were also used during the numerical analysis presented in this monograph. The description of the MESWIR system can be found in Kiciński (2005).

In recent years, numerous attempts have been made to calculate stiffness and damping coefficients in various bearing configurations, for various lubricants (Zhang et al. 2015) and geometries (Illner et al. 2015). New types of bearings and their better numerical description are being developed. The impact of boundary cavitation conditions on dynamic systems with hydrodynamic bearings is described in Daniel et al. (2016). Characteristics describing the operation of a floating ring bearing are presented in Chasalevris (2016).

1.3 Review of the Literature on Experimental Determination of Dynamic Coefficients of Bearings

Due to difficulties in numerical calculation of stiffness and damping coefficients of bearings, many experimental methods have been proposed to determine these coefficients (Dimond et al. 2009; Tiwari et al. 2004). The calculation of stiffness and damping coefficients can be performed in the time or frequency domain. Zhang et al. (1992a, 1992b) as well as Chan and White (1991) determined the stiffness and damping coefficients of two symmetrical bearings by adjusting the curves in the frequency response. This approach assumes a rotor with two identical bearings and allows for the calculation of eight dynamic factors (four stiffness coefficients and four damping coefficients). Since many rotor–bearing systems are not symmetrical and their vibration trajectories have more complex shapes than those for symmetrical systems, the limitation associated with the symmetry of the system in many cases cannot be applied. The method of curve matching also consumes a lot of computational power.

The work of Qiu and Tieu (1997) presents a method extending the calculation of eight coefficients of one bearing, allowing the calculation of a total of 16 stiffness and damping coefficients, for two different hydrodynamic bearings (four stiffness coefficients and four damping coefficients for each bearing). A method of determining the linearized form of coefficients (each stiffness and damping coefficient described by means of a single value) was developed. The authors believe that this method is the most effective method for determining stiffness and damping coefficients. In order to experimentally determine the stiffness and damping coefficients of bearings, it is necessary to force the rotor vibrations. This can be achieved in several ways. The three most commonly used are: impulse excitation of the rotating rotor by means of a modal hammer, the use of additional unbalance of the rotor, and the use of vibration inductors. Qiu and Tieu state in their work that the impulse excitation method is the most economical and convenient way to determine these coefficients. Dynamic coefficients of bearings are calculated on the basis of the response signal of the system measured after inducing the rotor in its central part by means of a modal hammer. The signals in the frequency domain are then used for further calculations. The authors suggest that a wider range of identification should be covered in the calculations in order to ensure greater repeatability of the results.

Tiwari and Chakravarthy (2009) described and demonstrated two different identification algorithms for the simultaneous estimation of the residual unbalance and the dynamic parameters of bearings of a rigid rotor–bearing system. The first method uses the impulse response measurements of the journal from bearing housings in the horizontal and vertical directions, for two independent impulses on the rotor in these directions. Time-domain signals of impulse forces and displacement responses are transformed into the frequency domain and are used for the estimation of the residual unbalance and bearing dynamic parameters. The second method employs the unbalance responses from three different unbalance configurations for the estimation of these parameters. The simulated responses were in fairly good agreement with experimental responses in terms of mimicking predominant resonances. The identified unbalance masses matched quite well with the residual masses taken in the dynamically balanced rotor–bearing test rig.

Meruane and Pascual (2008) described a method of numerical determination of non-linear stiffness and damping factors of hydrodynamic bearings. The calculation of fluid film coefficients using a non-linear method was performed for large displacements in the bearing (20–60% of the bearing gap). The non-linear effect was defined by extending the third-order Taylor equations. The non-linear model was created on the basis of a laboratory test stand. It was found that non-linear properties were revealed with oil vortexes that are not taken into account in classical linear models.

Work on modification of experimental methods in order to obtain greater accuracy is currently in progress. Miller and Howard (2009) described a method for identifying stiffness and damping coefficients by using the extended Kalman filter. This filter was developed to estimate the linearized form of stiffness and damping coefficients of bearings in rotor–bearing systems, taking into account noise and unbalance. The system uses impulse excitation. In this method, bearings are modeled as stochastic, random values using the Gauss–Markov model. The noise part is introduced into the system as an estimation error, including modeling and uncertainty of measurement. The system contains two user-defined parameters that can be used to fine tune the operation of the filter. They refer to the covariance of the system and the noise variables. The filter was tested using numerically created data of a system of two identical bearings, reducing the number of unknown coefficients to eight. The method was used to determine the main dynamic coefficients of bearings, while the cross-coupled coefficients were determined with a lower accuracy.

Particular difficulties in numerical determination of dynamic coefficients of bearings may be caused by bearings of complicated construction, e.g. foil bearings (Kiciński and Żywica 2012). The results of experimental identification of dynamic coefficients of a large foil bearing are presented in Wang and Kim (2013). Dynamic characteristics of a hybrid foil bearing (hydrodynamic + hydrostatic), 101.6 mm in diameter and 82.6 mm in length, are presented in that work. The stiffness coefficients were determined using two methods: the quasi-static method by determining deflection curves in the time domain and the impulse method in the frequency domain. The values of damping coefficients were determined using the impulse method only. The values of stiffness coefficients determined using the two above methods were similar, with the differences of around 4–7 MN/m depending on the speed, load, and supply pressure. Frequency calculations were characterized by greater discrepancies in the obtained results. As part of the work, a numerical model was also created, using the linear perturbation method. On the basis of the shaft deflection it was found that the results were similar to those of the experimental research.

A paper by Delgado (2015) presents calculations for a hybrid gas bearing (Kazimierski and Krysiński 1981) of a complex construction, characterized by rigid geometry and complex foil construction. The bearing operates on two lubrication films: hydrostatic and hydrodynamic. Such a procedure ensures the generation of appropriate load-bearing capacity and stiffness of the entire system. The paper presents an experimental verification of stiffness and damping coefficients for a bearing with a diameter of 110 mm. The results were obtained on a specially designed laboratory test rig. The variable parameters during the experiment were: hydrostatic supply pressure, driving force frequency, and rotor speed. Experimental research was aimed at evaluating the application of this bearing type in large-scale energy conversion machines. The dynamic tests showed poor sensitivity of the main stiffness coefficients to most of the test parameters. The frequencies and speeds were an exception: the higher the speed and frequency of the driving force, the lower the value of the calculated stiffness coefficients.

A paper by Kozánek et al. (2009) presents the results of experimental calculations of aerostatic radial bearings on the Bentley Nevada laboratory test rig. Various types of bearings as well as their static and dynamic characteristics were examined. Different methods of identification of dynamic coefficients of bearings were applied. Only the main stiffness and damping coefficients were calculated. The impact of cross-coupled stiffness and damping parameters was examined on the basis of numerical simulations.

During calculations, the mass matrix was defined as a matrix of known parameters, stiffness and damping matrices were determined. The authors recommended that when conducting experimental research the main values of stiffness and damping coefficients should be determined first, followed by cross-coupled ones, which are more susceptible to errors. According to the authors, increasing the bearing supply pressure and frequency of driving force negatively affects the correctness of the obtained values of dynamic coefficients of bearings.

Another example of experimental determination of parameters of aerostatic bearings is presented in Kozánek and Půst (2011). As part of the paper, a numerical model was developed to calculate the stiffness and damping parameters of both radial and thrust bearings.

A new method of identification of stiffness and damping of a bearing, based on phase plane diagrams, is presented in a paper by Jáuregui et al. (2012). The authors emphasize that reliable determination of dynamic coefficients of bearings is a huge challenge, particularly in non-linear systems. They also claim that a single, universal mathematical model does not exist, and the identification of parameters of a system depends on the measured data and the reference model. This model of phase plane diagrams works well when the coefficients are strongly dependent on frequency.

Experimental calculations of stiffness and damping parameters of bearings are not only done for radial bearings, but also for thrust bearings. The experimental determination of the parameters of stiffness and axial damping of foil bearings are presented in a paper by Arora et al. (2011). The paper presents a diagram of the procedure of identification of these coefficients, a description of the laboratory test rig, and a diagram of the values of stiffness coefficients in the function of rotational speed. The Monte Carlo algorithm was used to determine the dynamic parameters of the rotor supported on the magnetorheological layer of the lubricating film (Zapomel et al. 2014).

The method of impulse excitation for the determination of bearing coefficients, which is used in this monograph, is carried out in the frequency domain. Its first basic version was proposed by Nordmann and Schoelhorn (1980). Qiu and Tieu extended the calculation algorithm with the possibility of calculating 16 stiffness and damping coefficients (Qiu and Tieu 1997). This monograph describes a modification of the algorithm developed by Qiu and Tieu to calculate the damping coefficients and stiffness of bearings (using the impulse method and approximation with the least squares method). The algorithm has been extended by the possibility of calculating eight mass coefficients. Determination of damping, stiffness and mass coefficients using a single algorithm enables verification of the results at the initial stage of operation. Since the mass of the shaft is usually a known size, the correctness of the determined dynamic coefficients of bearings can be verified on the basis of mass coefficients. This approach makes it possible to determine all dynamic parameters of the rotor–bearing system through experimental research.

1.4 Purpose and Scope of the Work

The analysis of the literature and research conducted earlier in the Institute of Fluid Flow Machinery of the Polish Academy of Sciences confirm that dynamic coefficients of bearings have a key influence on the dynamic properties of rotating machinery. Since experimental methods for determining the stiffness and damping coefficients of bearings are usually described in the literature as burdened with large calculation errors, and the range of applicability of non-linear numerical methods (Bonet and Wood 2009; Sathyamoorthy 2000) is not clearly defined, the following objectives of work were formulated. They correspond with the subject of my dissertation (Breńkacz 2016) and over a dozen scientific articles.

The main aim of the work is to develop and describe a method of experimental determination of dynamic coefficients of hydrodynamic plain bearings and its verification. An additional objective of the work is to determine dynamic coefficients of bearings on the basis of experimental and numerical studies (linear and non-linear) in a wide range of rotational speeds of the rotor, taking into account resonance and speeds higher than the resonant speeds. Achievement of these two objectives will make it possible to compile data necessary to determine the ranges of linear and non-linear adequacy of methods for determining the dynamic coefficients of hydrodynamic bearings on the example of the rotating machinery under study.

The author participated in a one-year internship (from April 2013), which took place at LMS International in Belgium. The internship was organized as part of the STA-DY-WI-CO (European Commission/CORDIS 2019) project, which is part of the Marie Curie IAPP (Industry Academia Partnerships and Pathways) program. LMS International (www.plm.automation.siemens.com) is a leading manufacturer of instrumentation and software for measuring sound and vibration. This company cooperates with Samtech, the producer of Samcef Rotors software for analyzing the dynamics of rotors (https://blogs.sw.siemens.com/simcenter/). The issues described in this monograph were a common subject of the work of the two aforementioned companies.

In order for the basic objectives of the work to be achieved, it is necessary to carry out a number of intermediate tasks. The most important include:

- Development of a calculation algorithm which will make it possible to experimentally determine the stiffness, damping and mass coefficients of two hydrodynamic radial bearings.
- Verification of the developed algorithm.
- Evaluation of the sensitivity of the experimental method for the determination of stiffness, damping and mass coefficients.
- Conducting experimental tests in order to determine the basic dynamic characteristics of the tested laboratory test rig.
- Conducting experimental studies on the basis of which dynamic coefficients of two hydrodynamic radial bearings will be determined.
- Verification of the calculated experimental stiffness and damping coefficients based on the numerical model using the Abaqus software.
- Development of numerical models of the rotor and plain bearings with the use of the MESWIR software series. These programs enable the calculation of dynamic coefficients of hydrodynamic bearings using linear and non-linear algorithms.
- Numerical calculation of dynamic coefficients of bearings using linear and non-linear numerical models.
- Verification of the values of dynamic coefficients of bearings obtained with the use of numerical methods, by comparing the changes in journal vibration amplitudes and journal vibration trajectories (calculated on the basis of numerical analyses) with the results of experimental tests.

- Comparison of dynamic coefficients of bearings calculated using three different methods (on the basis of linear algorithm and experimental tests as well as linear and non-linear numerical models) and determination of the ranges of adequacy of their application.

All the above objectives were achieved and are described in this book. This chapter provides an introduction to the dynamics of rotating machinery. It presents the subject matter tackled in this monograph on a background of research carried out worldwide. Fundamental issues concerning dynamic properties of rotor–bearing–supporting structure systems are also discussed and a review of the literature on numerical and experimental methods of determining dynamic properties of rotating machines is presented. The basic research problem is formulated, and the aim of the study is defined.

Chapter 2 shows practical applications of bearing dynamic coefficients. Based on a simple example, it is shown how stiffness and damping coefficients affect dynamical systems. Other examples demonstrate how static and dynamic forces act. Basic equations for the vibrating motion are presented. The following examples show a dynamical system with one and two degrees of freedom. A practical example shows what influence cross-coupling coefficients have and what they mean.

Chapter 3 contains a description of the laboratory test rig and the characteristics drawn up based on experimental research. Rotor vibrations during run-up, acceleration of vibrations of the bearing support and modal analysis enabled the preparation of a complete picture of the dynamics of rotor machines. Vibrations and vibration trajectories of the journal were used to verify the numerical analyses carried out.

Chapter 4 discusses the research tools used during the implementation of the work. Accelerometers were used to measure vibrations of the structure, eddy current sensors were used to measure rotor displacement, and a laser tachometer was used to measure rotational speed. Measurement data were archived with Scadas Mobile and processed with LMS Test. Lab software. The analysis of rotor dynamics was carried out using a Samcef Rotors program and the MESWIR software series. The algorithm for calculating dynamic coefficients of bearings on the basis of experimental tests was developed using the Matlab program. In the same program, operations were performed on signals from numerical and experimental research. Verification of experimental results was carried out using Abaqus software.

Chapter 5 presents the method of experimental determination of dynamic properties of rotating machines. Under one algorithm, mass coefficients were also determined along with stiffness and damping coefficients. The mass coefficients can be interpreted as the mass of the shaft involved in vibrations (Kruszewski et al. 1996; Kruszewski and Wittbrodt 1992). Comparing the known mass of the shaft with the calculated mass coefficients, conclusions can be drawn on the correctness of the calculated stiffness and damping coefficients.

Chapter 6 presents one of the most time-consuming stages of the process related to the experimental determination of stiffness, damping and mass coefficients, i.e. appropriate signal preparation. It is necessary to select only a part of the signal, find a reference signal showing stable operation and subtract these signals. Dynamic coefficients of bearings were calculated on the basis of an appropriately prepared signal.

The sensitivity analysis of the method of experimental determination of dynamic coefficients of bearings is presented in Chapter 7. Using a numerical model, the influence of six parameters on the results of the calculations is presented. The analysis of the impact of these parameters is carried out on the basis that experimental research would not be possible or would be burdened with a significant error.

Chapter 8 presents the experimental tests, on the basis of which the stiffness, damping and mass coefficients were determined. The measured journal vibration signals, driving force signals and calculation results are presented.

Chapter 9 describes the method of linear and non-linear numerical calculations and numerical calculation models. Dynamic coefficients of hydrodynamic bearings were determined on their basis. The results of numerical calculations and their verification are presented.

The penultimate chapter (Chapter 10) summarizes the calculated stiffness and damping coefficients obtained using three different calculation methods. The differences between the results of linear experimental calculations and linear and non-linear numerical calculations are presented. Comments are made on the reasons for the discrepancies obtained.

The final chapter (Chapter 11) contains a summary of the results obtained during the research and conclusions from the work carried out.