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Preface

This volume contains selected papers presented at the 12th International Conference on vibrations in rotating machines, SIRM, which took place February 15-17, 2017 at the campus of the Graz University of Technology.

By all meaningful measures, SIRM was a great success, attracting about 120 participants (ranging from senior colleagues to graduate students) from 14 countries. Latest trends in theoretical research, development, design and machine maintenance have been discussed between machine manufacturers, machine operators and scientific representatives in the field of rotor dynamics.

SIRM 2017 included thematic sessions on the following topics: Rotordynamics, Stability, Friction, Monitoring, Electrical Machines, Torsional Vibrations, Blade Vibrations, Balancing, Parametric Excitation, and Bearings.
The papers struck an admirable balance between theory, analysis, computation and experiment, thus contributing a richly diverse set of perspectives and methods to the audience of the conference.

All participants were invited to submit full-length papers to a special issue of Technische Mechanik. The contributed papers were peer-reviewed and appear in this volume.

Horst Ecker, Katrin Ellermann, Franz Heitmeir, Elmar Woschke
October 2017

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On Dynamics and Stability of the Automotive Engine Turbocharger Rotor Supported by the Electrodynamic Passive Magnetic Bearings

T. Szolc

In the paper dynamic investigations on the automotive turbocharger rotor-shaft supported on the electro-dynamic passive magnetic bearings (EDPMB) and on the traditional floating-ring journal bearings have been carried out using a computer model. The results of computations obtained for the two mutually compared kinds of suspension are demonstrated in the form of Campbell diagrams and amplitude-frequency characteristics. Here, the main attention is focused on resonant-free operation ability assured by the support on the EDPMBs. Moreover, conditions of stability for the support on the journal bearings and on the EDPMBs have been investigated by means of the eigenvalue analysis. There is studied an influence of skew-symmetrical dynamic properties of the both kinds of rotor-shaft suspensions caused by the bearing stiffness negative cross-coupling terms as well as by the gyroscopic effects which are particularly severe at turbocharger high rotational speeds.

1 Introduction

The turbocharger is now viewed as feasible option when looking for an output power increase in a broad variety of internal combustion reciprocating petrol- and diesel engines. At present, this device is frequently applied in passenger car engines subjected to the downsizing tendency. Despite of the fact that the modern turbochargers have reached a relatively high level of robustness and operational excellence, there are still observed various problems with floating-ring journal bearings commonly supporting rotor-shafts of these devices. Namely, the turbocharger rotor-shafts supported by this kind of bearings often indicate a tendency to self-excitations and sub-synchronous oscillations leading to instability (Schweizer (2009), Kamesh (2011), Koutsovasilis and Driot (2015), Göbel et al. (2015)). Turbocharger rotors with foil-air bearing show analogous problems like self-excitations which lead to instability, what is carried out in the studies of Bonello (2016). Since maximal rotational speeds of the relatively light-weight turbocharger rotors reach 200,000 rpm and more, their suspension on another touch-less and lubricant-free bearings seems to be very required. For this purpose passive magnetic bearings (PMB) could be particularly promising. Actually, during the last 10-15 years, by means of the newest achievements of electrical engineering, electronics and material technology, various kinds of passive magnetic bearings (PMB) have been developed to give a chance for broad applications for numerous cases of the high-speed rotating machines. These are: the permanent magnet magnetic bearings (PMMB), superconductor passive magnetic bearings (SCPMB) and the electrodynamic passive magnetic bearings (EDPMB) which seem to be very advantageous here. In order to generate levitation forces these kinds of passive magnetic bearings use conductors mounted on the shaft and rotating in a magnetic field created by permanent magnets built-in the stators embedded in bearing housings. Then, eddy-currents are induced in the conductors, which generate the Lorenz forces levitating the rotor-shaft. The physical fundamentals for a passive magnetic levitation together with conditions of its stable operation can be found in Filatov et al. (2002). In Lempke (2005), Amati et al. (2008) and in Falkowski (2016) the concept of radial EDPMBs has been developed theoretically and by Lempke (2005) tested experimentally for the small-size high-speed rotating systems. The results of comparative rotordynamic analyses performed for the industrial centrifugal compressor and for the single-spool gas turbine supported on the EDPMBs as well as on the classical oil-film journal- and rolling-element bearings were demonstrated in Szolc and Falkowski (2014). The passive magnetic suspensions have got more important advantages then the active magnetic bearings (AMB). Namely, the EDPMBs do not need any power supply, enable us resonant-free operation, have a simple structure and they are cheaper than the AMBs. According to the above, in this paper in order to indicate mechanical advantages of the turbocharger rotor-shaft support on the EDPMBs, a comparative rotor-dynamic analysis has been performed, where the obtained results of computations were confronted with the analogous findings determined for the same rotor-shaft suspended by the floating-ring journal bearings.

2 Modelling of the Turbocharger-Rotor-Shaft System

The object of considerations is a typical automotive engine turbocharger rotor-shaft shown in Figure 1a. This rotating element is originally supported on two floating-ring oil journal bearings (#1 and #2). Its full geometry, material constants, inertial parameters as well as the journal bearing dimensions are taken from Göbel et al. (2015). This rotating system is expected to operate within the rotational speed range of 0-210,000 rpm. It consists of the flexible stepped shaft of the total length 0.16 m with attached two heavy disks corresponding respectively to the turbine and compressor rotor. The considered rotor-shaft is characterized by the entire mass of 0.564 kg and by the bearing span equal to 0.05 m. Such a structure can be very representative for a broad variety of turbochargers applied in internal combustion piston engines. Thus, this rotor-shaft system was theoretically slightly adopted to run on the electrodynamic passive magnetic bearings regarded here as a perspective alternative.

2.1 Modelling of the Rotor-Shaft

In order to obtain sufficiently reliable results of theoretical calculations for the considered rotor-shaft system, the dynamic analysis will be performed by means of the one-dimensional hybrid structural model consisting of finite beam elements and discrete oscillators. The flexural beam elements represent successive cylindrical segments of the stepped rotor-shaft. With an accuracy that is sufficient for practical purposes the heavy turbine and compressor rotors are substituted by rigid bodies attached to the respective beam-element extreme cross-sections. Using such a model, the rotor-shaft geometry as well as its material properties can be described in an identical way as in an analogous classical finite element model of the same structure. However, in the hybrid model inertial-visco-elastic properties of its beam elements are not discretized, but they have been left as naturally distributed in a continuous way. Such a hybrid model of the turbocharger rotor-shaft is presented in Figure 1b.

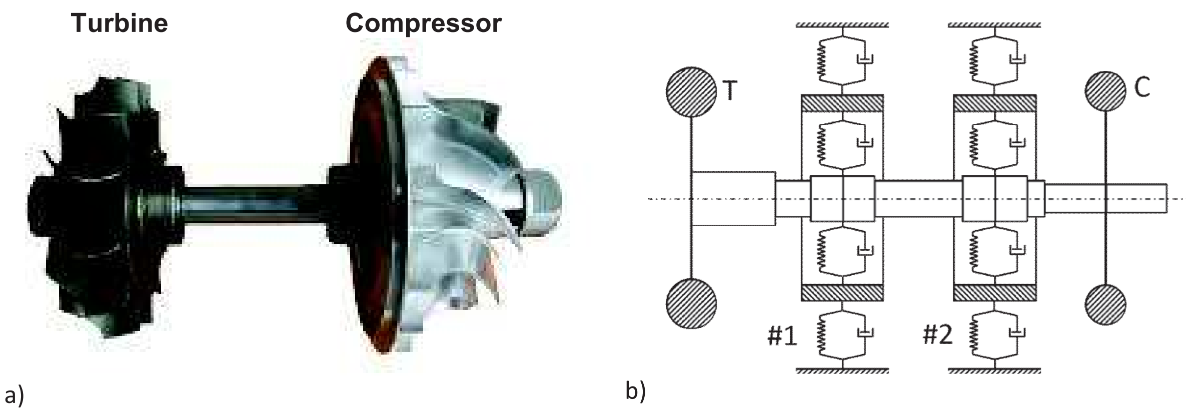

Figure 1: The automotive engine turbocharger rotor-shaft: the real object (a) and the hybrid model (b)

Here, each bearing support is represented by a dynamic oscillator of two degrees of freedom, where its rigid mass represents an inertia of the floating ring in the case of the journal bearing or an inertia of the stator elements in the case of the EDPMBs. Using such a model, apart from the magnetic field or oil-film interaction, also the visco-elastic properties of an embedding of both kinds of bearings in the turbocharger housings are taken into consideration. This bearing model makes possible to represent with a relatively high accuracy kinetostatic and dynamic anisotropic and anti-symmetric properties in the form of constant or variable stiffness and damping coefficients. The obtained in this way mutual combination of continuous finite elements together with discrete oscillators and rigid bodies according to the structure of the real object results in the hybrid mechanical model of the automotive turbocharger rotor-shaft system.

2.2 Modelling of the Electrodynamic Passive Magnetic Bearing

Here, dynamic modelling of the electrodynamic passive magnetic bearings reduces to determinations of their electromagnetic stiffness- and damping characteristics. For this purpose it is necessary to calculate the bearing global levitation force or the global radial 'in-plane' stiffness K regarded as a derivative of the Lorenz force with respect of the conductor-to-stator radial proximity (Lempke, 2005, Szolc and Falkowski, 2014, Falkowski, 2016). This target is usually achieved by computations carried out by means of advanced 3D finite element models of the electrodynamic bearing for various rotational speeds Ω or using the analytical-numerical method based on the Kirchhoff electrical circuit theory (Amati et al., 2008). All these approaches result in magnetic force and stiffness characteristics respectively very similar qualitatively to each other. Then, according to Lempke (2005), the main (rotor)- and cross-coupling stiffness components of the EDPMB can be determined for selected successive values of the rotor-shaft rotational speed by means of the following formulae:

$$k_{xx}(\Omega) = k_{yy}(\Omega) = K(\Omega)\cos\theta\,,$$
$$k_{xy}(\Omega) = K(\Omega)\sin\theta = -k_{yx}(\Omega), \tag{1}$$

where θ = arc tan($R/\Omega\ L$) is the so called 'force angle' expressed as a function of the magnetic bearing coil resistance R and inductance L. Nevertheless, for the need of numerical simulations and qualitative analyses the bearing main (rotor)- and cross-coupling stiffness values obtained for several successive rotational speed values can be approximated using the following continuous analytical functions:

$$k_{xx}(\Omega) = k_{yy}(\Omega) = K_R \cdot f^{\alpha}\cos\left(\mathrm{arc\,tan}\left(\frac{\beta}{f}\right)\right),$$
$$k_{xy}(\Omega) = -k_{yx}(\Omega) = K_C \cdot f^{\delta}\sin\left(\mathrm{arc\,tan}\left(\frac{\gamma}{f}\right)\right), \tag{2}$$
$$f = \frac{\Omega}{2\pi},$$

where K_R, K_C, α, β, δ and γ are the proper fitting coefficients. In Figure 2a there are presented the rotor main- and cross-coupling stiffness characteristics determined using formulae (2) for the EDPMBs suspending the considered turbocharger rotor-shaft system within its entire operating rotational speed range. These plots have been obtained for K_R=3.5·10^4 N/m, K_C=6.32·10^3 N/m, α=0.05, β=270.0, δ=0.3 and γ=270.0 and they are

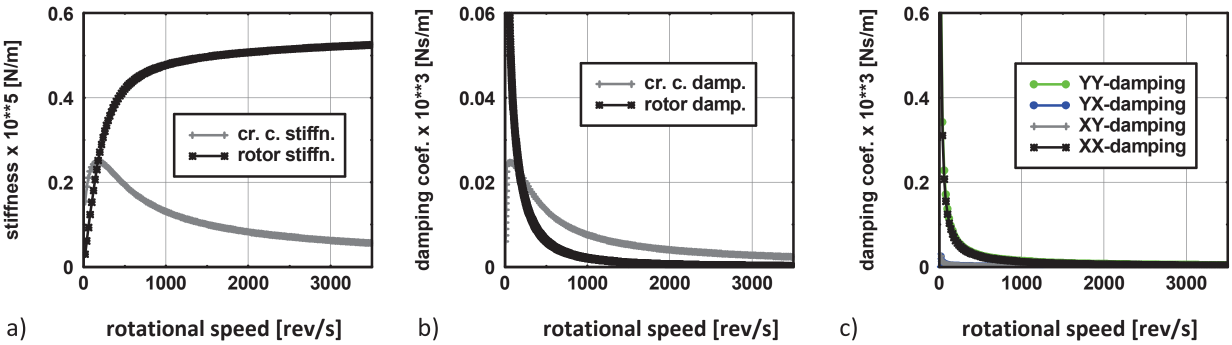

Figure 2: Stiffness (a) and damping (b) characteristics of the EDPMB, damping characteristics of the inner floating-ring journal bearing (c)

qualitatively identical with those obtained in Lempke (2005), Amati et al. (2008), Szolc and Falkowski (2014), Falkowski (2016) for high-speed rotor machines different than investigated here. For the listed fitting coefficient values the static turbocharger rotor-shaft vertical displacement off-set in the bearing stators due to the gravitational forces does not exceed 0.11 mm. Then, according to Lempke (2005), Szolc and Falkowski (2014), damping coefficients of the EDPMB have been calculated by means of the following formulae:

$$d_{xx}(\Omega) = d_{yy}(\Omega) = k_{xy}(\Omega)/\Omega,$$
$$d_{xy}(\Omega) = -d_{yx}(\Omega) = k_{xx}(\Omega)/\Omega. \qquad (3)$$

The plots of the corresponding main- and cross-coupling damping coefficients for the considered EDPMB are presented in Figure 2b. It is to remark that the mutual skew-symmetry of the cross-coupling stiffness and damping coefficient components characterizing the EDPMBs has an essential influence on the levitation stability.

2.3 Modelling of the Floating-Ring Oil-Journal Bearing

A mathematical modelling of the oil-film interaction in the floating-ring journal bearings commonly reduces to a numerical solving of the Reynolds equations and determination of the oil pressure distributions (Schweizer (2009), Kamesh (2011), Koutsovasilis and Driot (2015), Göbel et al. (2015)). Since a character of this interaction is usually non-linear, several associated phenomena as a self-excitation by oil whirl and whip or sub-synchronous oscillations can be taken into considerations. As it follows e.g. from Schweizer (2009), Kamesh (2011), Koutsovasilis and Driot (2015), Göbel et al. (2015), such advanced investigations are often extensive enough to become their own separate studies. But here, the dynamic properties of the turbocharger suspension by the floating-ring journal bearings are going to be regarded as an approximate qualitative reference to the corresponding support on the EDPMBs. Thus, a linearized model of the floating-ring journal bearing has been assumed, where the stiffness and damping coefficients of the inner and outer bearing were determined according to Someya (1989) using the solutions of the Reynolds equations, as well. For great rotational speeds of the turbocharger rotor the respective Sommerfeld numbers of the inner and outer bearing are appropriately high. Therefore, for the admissible journal-to-bushing clearance values the resulting main- and cross-coupling stiffness coefficients are big enough to substitute them by the proper average values. Here, it turned out that the main stiffness components became ca. 13 times larger than the mean levitation stiffness realized by the assumed EDPMBs. According to Someya (1989), the corresponding coefficients of damping in the journal bearings hyperbolically decay with the shaft rotational speed Ω, as demonstrated by the plots in Figure 2c.

3 Mathematical Solution of the Problem

The complete mathematical formulation and solution for the hybrid models of rotor-shaft systems assumed in the way described above can be found e.g. in Szolc (2000). In these models flexural motion of cross-sections of each visco-elastic macro-element is governed by the partial differential equations derived using the Timoshenko or the Rayleigh rotating beam theory. In such equations there are contained gyroscopic forces mutually coupling rotor-shaft lateral vibrations in the vertical and horizontal plane. The analogous coupling effect caused by the system rotational speed dependent shaft material damping described using the standard body model is also taken into consideration. The solution for the lateral vibration analysis has been obtained using the analytical-computational approach demonstrated in details in Szolc (2000). In the considered case it is to emphasize that since, according to formulae (1), (2) and (3), the visco-elastic bearing support parameters are rotational speed dependent, the fundamental dynamic properties of the rotor-shaft, e.g. its natural frequencies, eigenfunctions, modal masses and others, also depend on the shaft rotational speed value Ω. Then, solving the differential eigenvalue problem for the orthogonal system obtained for the given Ω and an application of the Fourier solutions in the form of fast convergent series in orthogonal eigenfunctions lead to the set of modal equations

$$\mathbf{M}(\Omega)\cdot\ddot{\mathbf{r}}(t) + \mathbf{D}(\Omega)\cdot\dot{\mathbf{r}}(t) + \mathbf{K}(\Omega)\cdot\mathbf{r}(t) = \mathbf{F}(t,\Omega^2), \qquad (4)$$

where:

$$\mathbf{D}(\Omega) = \mathbf{D}_0(\Omega) + \Omega\cdot\mathbf{D}_g(\Omega)$$

and

$$\mathbf{K}(\Omega) = \mathbf{K}_0(\Omega) + \mathbf{K}_b(\Omega) + \Omega\cdot\mathbf{K}_d(\Omega).$$

The symbols $\mathbf{M}(\Omega)$, $\mathbf{K}_0(\Omega)$ denote the diagonal modal mass and stiffness matrix, respectively, $\mathbf{D}_0(\Omega)$ is the non-symmetrical damping matrix and $\mathbf{D}_g(\Omega)$ denotes the skew-symmetrical matrix of gyroscopic effects. Skew- or non-symmetrical elastic properties of the bearings are described by matrix $\mathbf{K}_b(\Omega)$. Anti-symmetrical effects due to the standard body material damping model of the rotating shaft are expressed by the skew-symmetrical matrix $\mathbf{K}_d(\Omega)$ and $\mathbf{F}(t,\Omega^2)$ denotes the external excitation vector due to the unbalance and gravitational forces. On the one hand, for a given value of Ω all matrices are constant. But on the other hand, since visco-elastic properties of the bearing supports are rotational speed dependent, the successive lateral eigenforms and natural frequencies of the turbocharger rotor-shaft are also functions of Ω. Hence, according to the fundamentals of modal analysis, the elements of modal mass, stiffness and damping-gyroscopic matrices become rotational speed dependent, as well. The modal coordinate vector $\mathbf{r}(t)$ consists of the unknown time functions that occur in the Fourier solutions. The number of equations (4) corresponds to the number of lateral eigenmodes taken into consideration in the range of frequency of interest.

Since the main target of the realized study is an investigation of stability of the considered rotating system, its eigenvalue real parts are going to be regarded first as the fundamental measure of the asymptotic stability. In order to determine eigenvalues of the rotor-shaft dynamic model, it is convenient to transform its homogeneous modal motion equations (4) into analogous equations in modal state coordinates. Next, using for them the well-known exponential complex analytical solution one can obtain the standard complex eigenvalue problem. Then, the problem reduces to searching the eigenvalue imaginary and real parts expressed as functions of the shaft rotational speed Ω by means of the following matrix (Szolc et al., 2016):

$$\mathbf{H}(\Omega)=\begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{M}^{-1}(\Omega)\cdot\left(\mathbf{K}_0(\Omega)+\mathbf{K}_b(\Omega)+\Omega\cdot\mathbf{K}_d(\Omega)\right) & -\mathbf{M}^{-1}(\Omega)\cdot\left(\mathbf{D}_0(\Omega)+\Omega\cdot\mathbf{D}_g(\Omega)\right)\end{bmatrix}. \tag{5}$$

where $\mathbf{I}$ is the identity matrix.

It is to remark that the modal submatrix $\mathbf{D}_0(\Omega)$ containing the damping coefficients (see Eq. (3)) of the passive magnetic bearings is non-symmetrical. Moreover, in matrix $\mathbf{H}(\Omega)$ described by Eq. (5), in addition to the skew-symmetrical gyroscopic matrix $\mathbf{D}_g(\Omega)$, also the skew-symmetrical stiffness submatrices $\mathbf{K}_d(\Omega)$ and $\mathbf{K}_b(\Omega)$ occur. This fact can influence very essentially dynamic stability effects of the entire rotor-shaft system. Because matrix $\mathbf{H}(\Omega)$ is a non-symmetrical one, in order to determine effectively the complex eigenvalues it is necessary to reduce it to the Hessenberg form using the Hausholder transformation. Then, the final computation of the eigenvalue real and imaginary parts for each lateral eigenmode of the considered system is achieved by means of the commonly known QR algorithm.

Since the comparison of dynamic behaviours of the turbocharger rotor-shaft suspended by the floating-ring journal bearings and by the electro-dynamic passive magnetic bearings is going to be performed also for forced vibrations at steady-state operating conditions, constant values of the shaft rotational speeds Ω will be assumed. At the constant rotational speed Ω equations (4) are a system of linear ordinary differential equations with constant coefficients and harmonic external excitation due to residual unbalances. For the mentioned above harmonic and gravitational excitation with the respective amplitude modal components $\mathbf{P}$, $\mathbf{Q}$ and $\mathbf{R}$, the induced steady-state vibrations are also harmonic with the same synchronous circular frequency Ω. Thus, the analytical solutions for the successive modal functions contained in vector $\mathbf{r}(t)$ can be assumed in an appropriate harmonic form. Then, by substituting them into (4), derived here for Ω=const, one obtains the following systems of linear algebraic equations:

$$\begin{aligned} &\mathbf{K}(\Omega)\cdot\mathbf{G}=\mathbf{R}, \\ &\left(\mathbf{K}(\Omega)-\Omega^2\mathbf{M}(\Omega)\right)\cdot\mathbf{C}+\Omega\cdot\mathbf{D}(\Omega)\cdot\mathbf{S}=\mathbf{P}(\Omega^2), \\ &\left(\mathbf{K}(\Omega)-\Omega^2\mathbf{M}(\Omega)\right)\cdot\mathbf{S}-\Omega\cdot\mathbf{D}(\Omega)\cdot\mathbf{C}=\mathbf{Q}(\Omega^2), \end{aligned} \tag{6}$$

where vectors $\mathbf{C}$, $\mathbf{S}$ contain respectively the modal cosine- and sine-components of forced vibration amplitudes and vector $\mathbf{G}$ contains the modal components of the rotor-shaft static deflection due to the gravitational force. These equations are very easy to solve with respect of the unknown components of vectors $\mathbf{C}$, $\mathbf{S}$ and $\mathbf{G}$.

4 Stability Analysis of the Turbocharger Rotor-Shaft System

Before investigations concerning the proper stability analysis, there is worth studying some fundamental dynamic properties of the considered turbocharger rotor-shaft-bearing systems in the form of the Campbell diagrams in the expected rotational frequency range 0-210,000 rpm which corresponds to 0-3500 rev/s. In Figure 3 there is presented the Campbell diagram for the turbocharger rotor suspended by the floating–ring journal bearings and in Figure 4 the analogous diagram for a support on the EDPMBs is shown. For a better clarity, on the left-hand sides of the both diagrams also the respective lateral eigenfunctions are depicted. Because of the relatively soft suspension by the EDPMBs towards the rotor-shaft flexibility, from among the fundamental lateral eigenforms one can distinguish first two similar to 'rigid-body' ones, see Figure 4. However, the harder support on the journal bearings results in almost all typical bending eigenmodes, as shown in Figure 3. In the considered synchronous external excitation frequency range 0-3500 Hz in the both cases of bearing support four lateral eigenforms of the turbocharger rotor-shaft have been determined, respectively with their backward and forward whirl branches. It is to emphasize that because of an influence of the significant negative cross-coupling stiffness components characterizing both the journal bearings as well as the EDPMBs, the fundamental first natural frequencies appear upon certain shaft rotational speed values, as shown in Figures 3 and 4. Due to his fact, in the case of the turbocharger rotor-shaft support on the journal bearings, the first critical speed coincides with the second eigenmode forward precession. Consequently, the third and the fourth critical speed coincide respectively with the third and fourth eigenmode forward precession, as marked using the small rings in Figure 3. However, from the analogous rings in Figure 4 it follows that in the case of the much "softer" suspension by the EDPMBs only two critical speeds are observed: namely with the forward whirls of the third and the fourth eigenmode. Moreover, in the case of journal bearing support, due to gyroscopic forces the second and the third eigenform tend to mutually coincide with the shaft rotational speed rise. Thus, the natural frequency of the second eigenmode forward whirl as well as the natural frequency of the third eigenmode backward whirl vanish together above ca. 2950 rev/s.

A dynamic stability analysis of the hybrid structural model of the considered turbocharger rotor-shaft system supported on the floating-ring journal bearings as well as on the electrodynamic passive magnetic bearings has been performed within the frequency range 0-6000 Hz containing its 6 lateral eigenforms. The investigations are carried out for the shaft material loss factor 0.002 and for the bearing visco-elastic characteristics depicted in Figure 2. Here, the eigenvalue imaginary and real parts are determined as rotational speed functions using matrix **H** defined by Eq. (5). Because of a commonly low magnitude of steel shaft material damping as well as due to the relatively small damping coefficient values characterizing the both kind of bearings, particularly for bigger rotational speeds, see Figures 2b and 2c, the eigenvalue imaginary parts, playing a role of damped natural frequencies, almost overlay with the corresponding plots of undamped natural frequencies in the Campbell diagrams in Figures 3 and 4. Thus, for a better clarity they have not been presented here in a graphical form. In Figures 5 and 6, respectively for the turbocharger rotor-shaft support on the journal bearings and on the EDPMBs, there are demonstrated eigenvalue real parts. From the plots illustrated in these figures it follows that not every eigenvalue real parts are negative, what means that the both considered rotating systems pain a lack of stability. Namely, in the case of the journal-bearing support with abovementioned significant negative cross-coupling stiffness components the eigenvalue real parts corresponding to backward whirls of the first four eigenmodes are positive, particularly for greater rotational speeds, as shown in Figure 5. The suspension by the EDPMBs also results in the unstable backward whirls of the first four eigenmodes. But this instability is essentially severe for small rotational speeds in the cases of the first and the second eigenmode, see Figure 6, which follows from the visco-elastic properties of this kind of a magnetic support (Lempke, 2005, Amati et al., 2008, Szolc et al., 2016). It is to emphasize that this result has been obtained for a relatively hard structure of the EDPMB stators embedded in the turbocharger housing, i.e. in the so called 'metal-to metal' way. But for an appropriately flexible stator suspension in the housing, e.g. by means of a vulcanized rubber or polymer foil strip, as well as for a properly "soft" visco-elastic structure of the stators, a sufficient amount of passive damping can be introduced. Such damping results in almost complete stabilization of the rotor-shaft support on the EDPMBs, as indicate the system respective eigenvalue real parts presented in Figure 7. It is worth noting that in this way it was possible to stabilize the light-weight and high-speed turbocharger rotor without any specialized dampers, contrary to e.g. Amati et al. (2008) and Szolc et al. (2016), where for other rotors suspended by EDPMBs it had to be made.

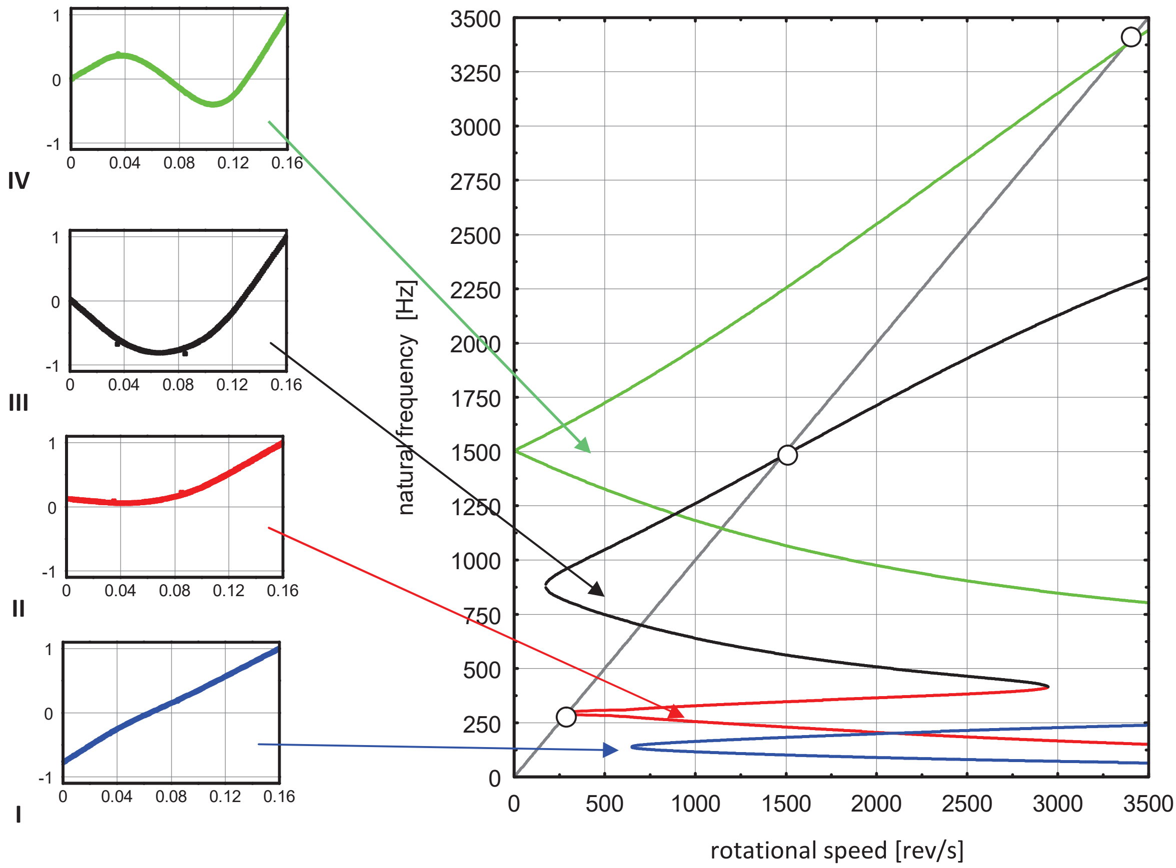

Figure 3: Campbell diagram for the turbocharger rotor-shaft system supported on the journal bearings

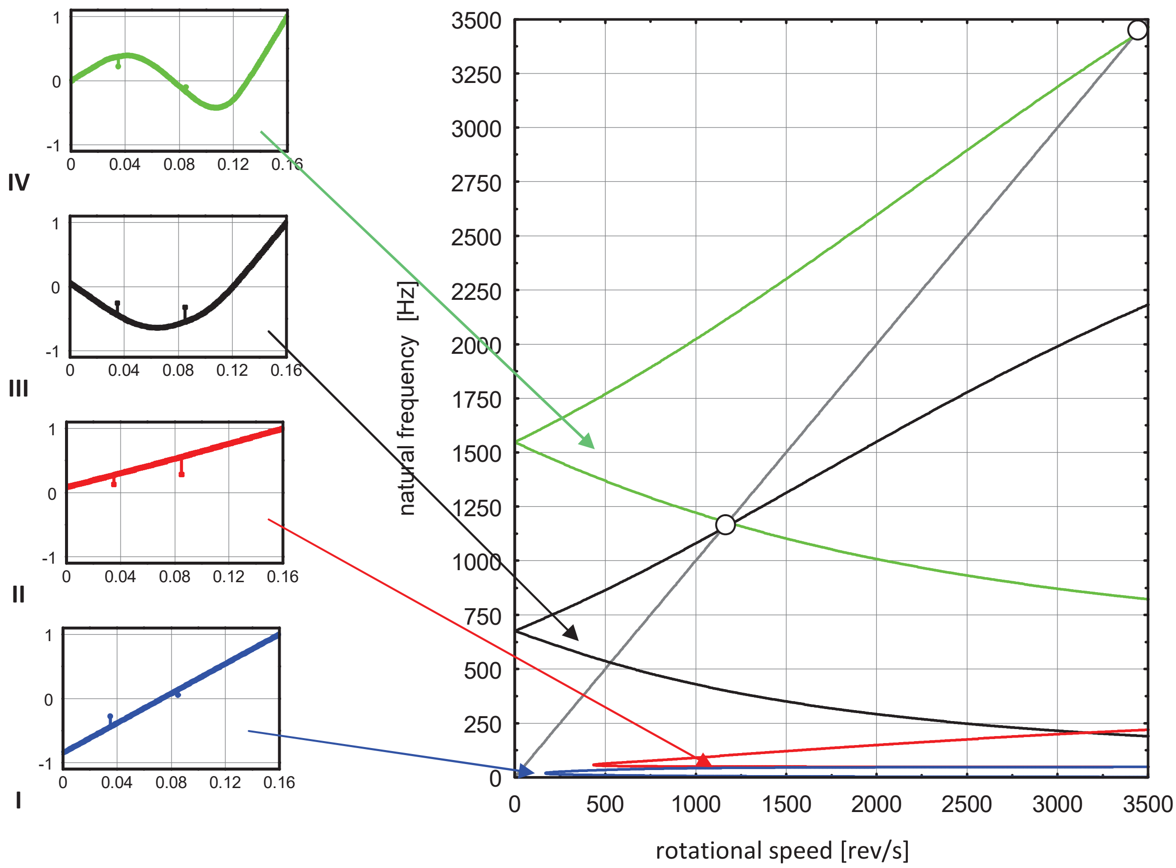

Figure 4: Campbell diagram for the turbocharger rotor-shaft system supported on the EDPMBs

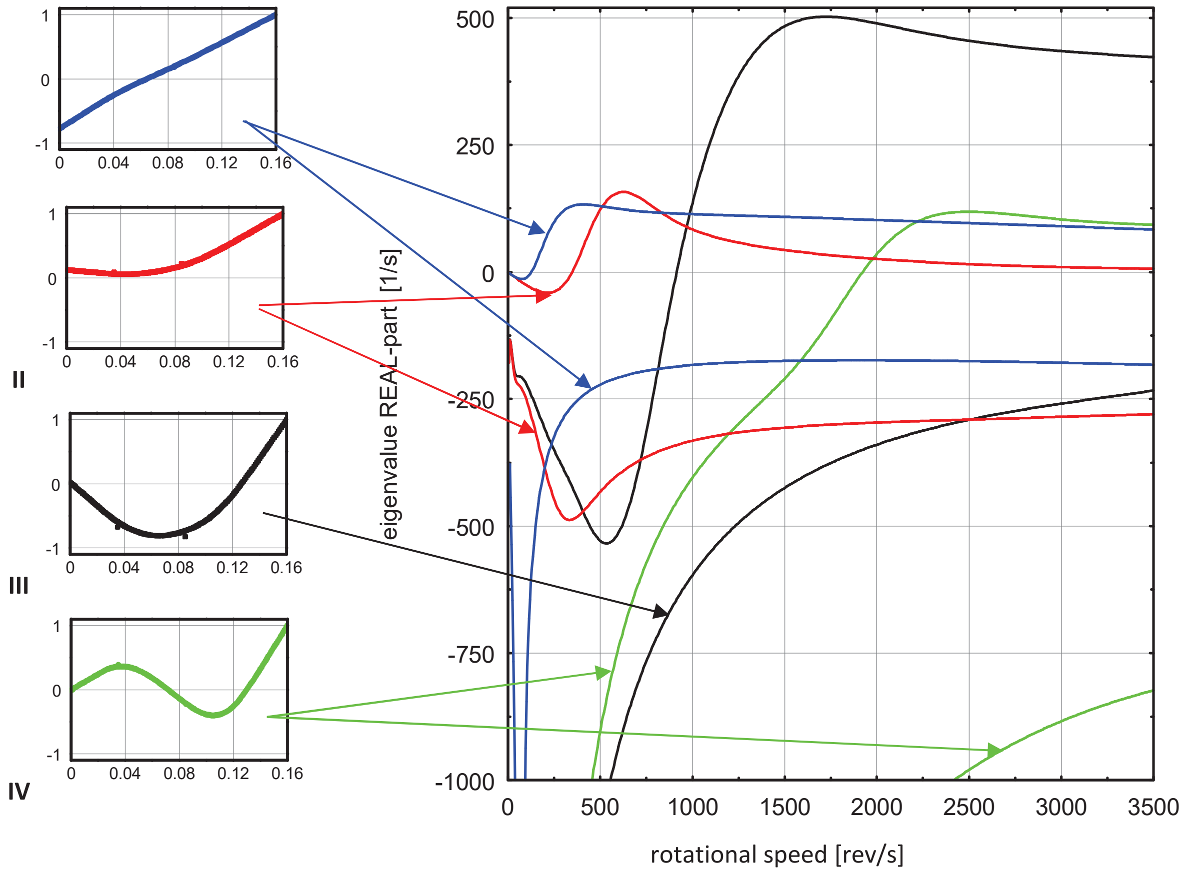

Figure 5: Eigenvalue real parts of the turbocharger rotor-shaft system supported on the journal bearings

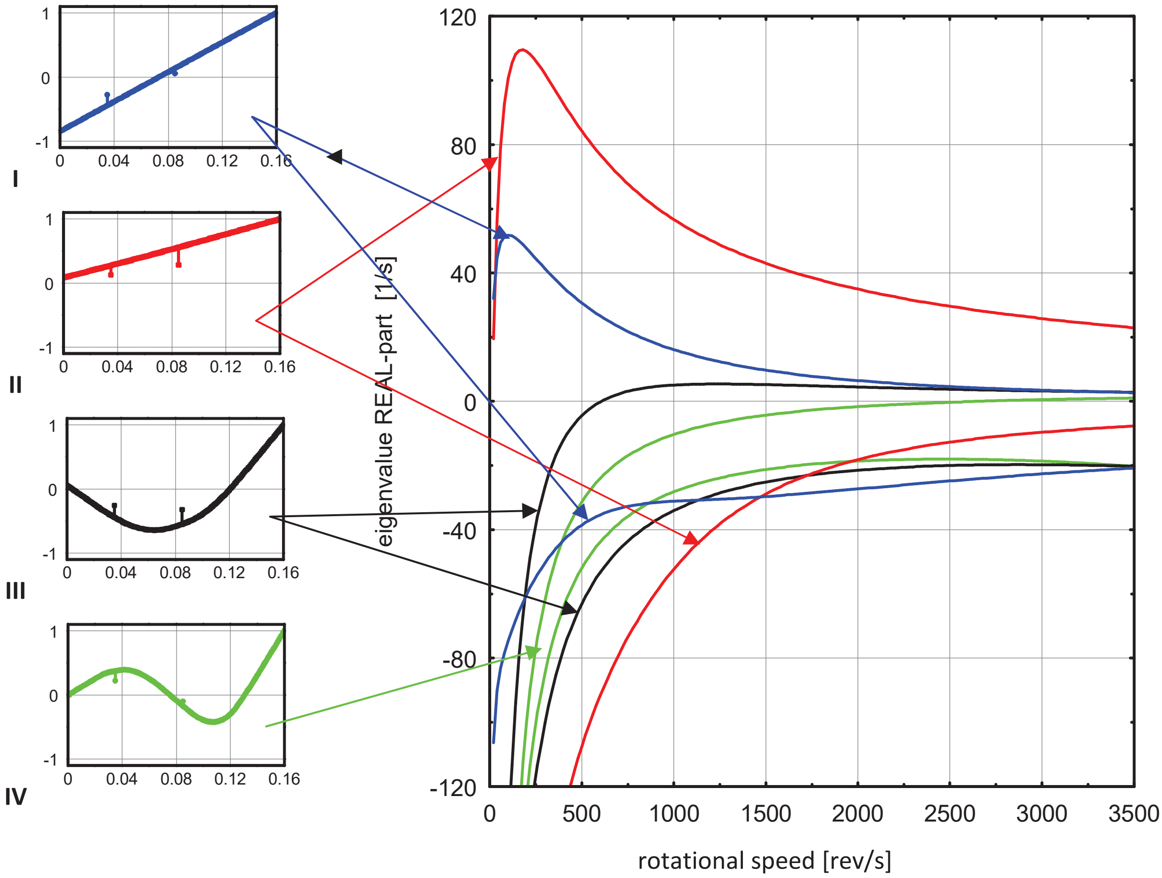

Figure 6: Eigenvalue real parts of the turbocharger rotor-shaft system "hardly" supported on the EDPMBs

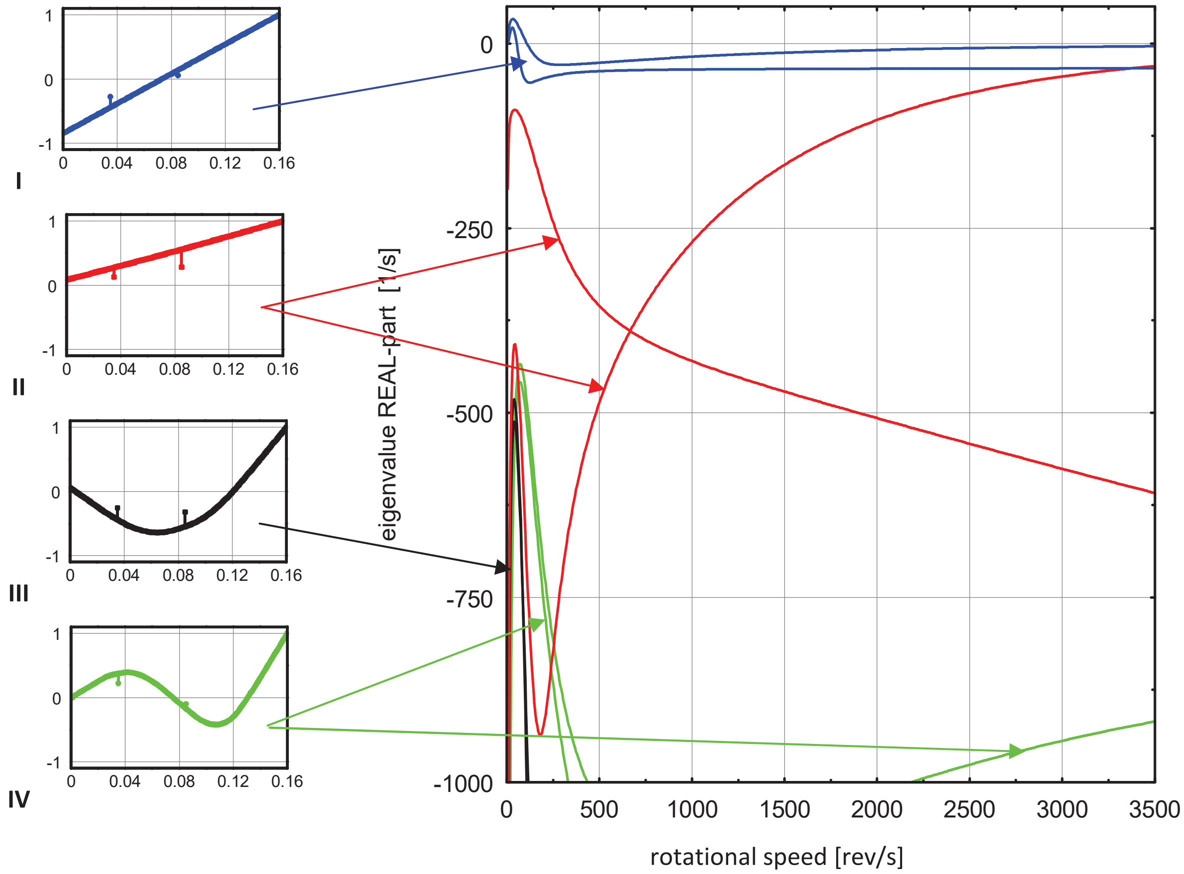

Figure 7: Eigenvalue real parts of the turbocharger rotor-shaft system "softly" supported on the EDPMBs

5 Forced Vibration Comparative Analysis

In addition to the comparison of eigenvibration properties of the turbocharger rotor-shaft supported on the floating-ring journal-bearings and on the EDPMBs, there are going to be compared also amplitude-frequency characteristics of the steady-state forced dynamic responses due to synchronous excitations caused by unavoidable residual unbalances. According to Göbel et al. (2015), $0.6 \cdot 10^{-6}$ kgm unbalance of the turbine rotor and $0.4 \cdot 10^{-6}$ kgm unbalance of the compressor rotor have been assumed. Here, two unbalance variants will be investigated, i.e. the commonly called "static unbalance", when the turbine and compressor rotor unbalances are mutually oriented 'in phase', as well as the "dynamic unbalance", for which these two unbalances are mutually oriented 'in anti-phase'. In order to determine the amplitude-frequency characteristics Eqs. (6) had to be solved for the both mentioned above unbalance variants and for the turbocharger rotor-shaft system parameters representing the two considered kinds of bearing support within the expected rotational speed range 0-210,000 rpm corresponding to the harmonic synchronous excitation frequency band 0-3500 Hz.
Figures 8 and 9 demonstrate amplitude-frequency characteristics of the steady-state dynamic responses excited by the static unbalance of the rotor-shaft. In Figure 8 the lateral displacement amplitudes of the turbocharger rotors are depicted and in Figure 9 the bearing vertical reaction force amplitudes are plotted. In these figures the black lines correspond to the rotor-shaft location close to turbine rotor and the grey lines to that of the compressor one. In an identical way Figures 10 and 11 illustrate amplitude-frequency characteristics of the analogous dynamic responses excited by the dynamic unbalance of the rotor-shaft. Figures 8-11a correspond to the support on the EDPMBs and Figures 8-11b to that on the journal bearings regarded here as an approximate reference. From the obtained characteristics it follows that the visco-elastic properties of the rotor-shaft suspension by the EDPMBs result in the much smaller vibration displacements in the critical speed vicinity and in the almost unremarkable bearing force amplitudes in a comparison with the classical, original support on the journal bearings. As it follows from the Campbell diagram in Figure 4, the skew-symmetry of the EDPMBs significantly "shifts" the fundamental first two 'rigid-body' modes far away from a synchronous excitation

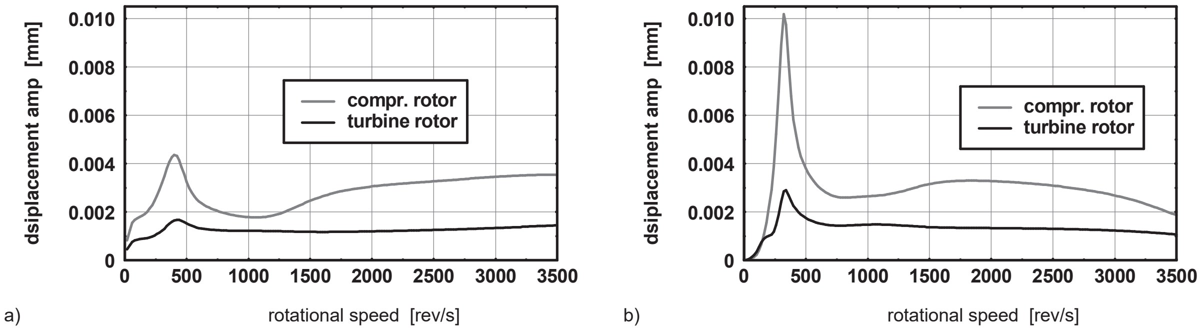

Figure 8: Displacement amplitude characteristics of the turbocharger rotor-shaft supported on the passive magnetic bearings (a) and on the journal bearings (b) obtained for the static residual unbalance

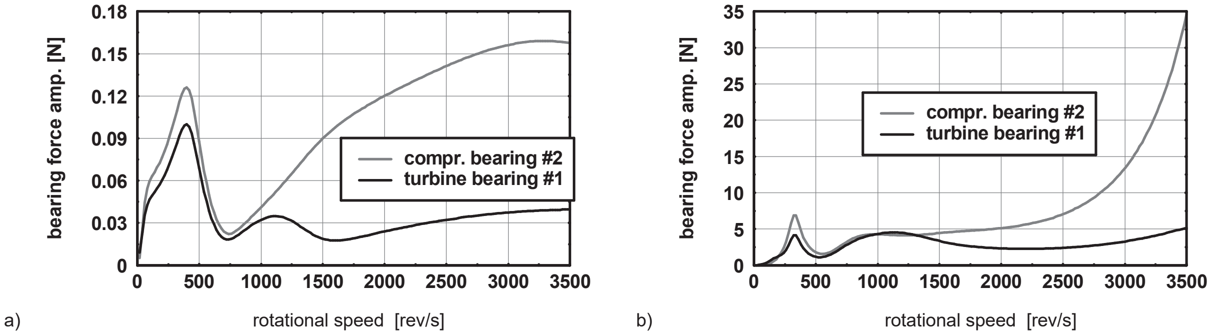

Figure 9: Bearing force amplitude characteristics of the turbocharger rotor-shaft supported on the passive magnetic bearings (a) and on the journal bearings (b) obtained for the static residual unbalance

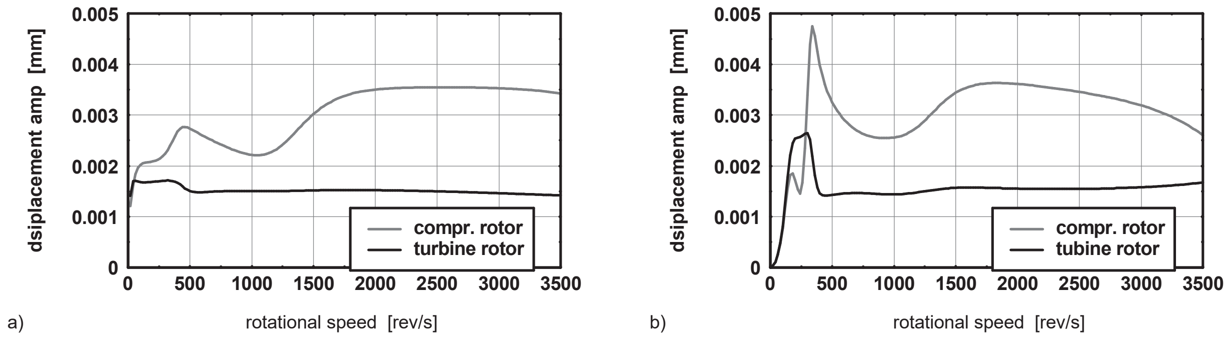

Figure 10: Displacement amplitude characteristics of the turbocharger rotor-shaft supported on the passive magnetic bearings (a) and on the journal bearings (b) obtained for the dynamic residual unbalance

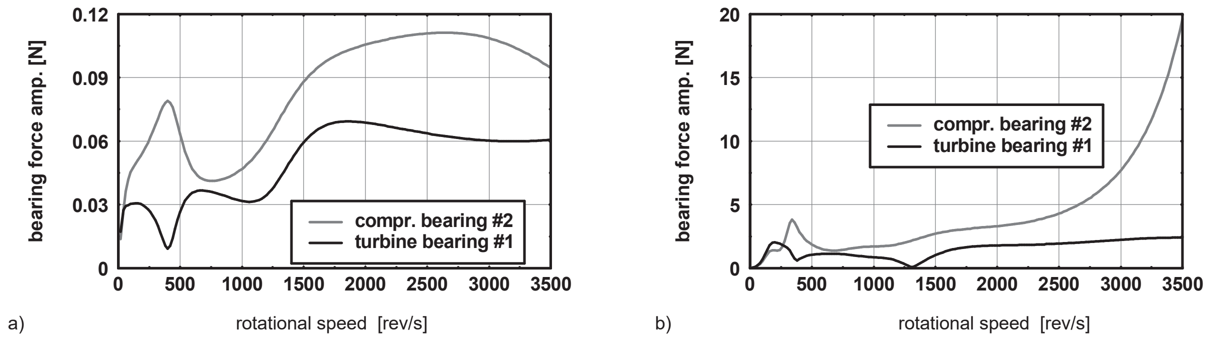

Figure 11: Bearing force amplitude characteristics of the turbocharger rotor-shaft supported on the passive magnetic bearings (a) and on the journal bearings (b) obtained for the dynamic residual unbalance

possibility. However, at the critical speeds marked in this figure resonances with the successive two ‘elastic’ modes can be rather hardly induced. Thus, the turbocharger rotor behaves as a typical overcritical, self-centring shaft, what actually results in the obtained very small dynamic reaction forces transmitted by the bearings. Here, some amplifications of lateral vibration amplitudes are observed only at ca. 450 rev/s, because of predominant damping force activity exceeding an interaction of the elastic forces at low rotational speeds. Such a dynamic behaviour has been indicated also in Amati et al. (2008) for this kind of bearings. The naturally ‘harder’ journal bearings are responsible not only for much greater vibration amplitudes, but at ca. 280 rev/s the severe resonance occurs as a result of the critical speed with the second eigenmode shown in Figure 3. Nevertheless, it is to remember that non-linear properties omitted in the simplified floating-ring journal bearing model assumed here can result in various additional oscillation components, oil whirl and whip effects and in other phenomena (Schweizer (2009), Kamesh (2011), Koutsovasilis and Driot (2015), Göbel et al. (2015)). Hence, all the critical speeds discussed in this study are caused by the synchronous harmonic excitations, i.e. they arise from the linearized inertial-visco-elastic properties of the rotor-shaft-bearing models. Then, possible resonances occurring at these speeds are induced by external excitations due to unbalances, but neither by self-excitations associated with non-linear descriptions of the journal bearings nor with the EDPMBs which actually are linear in character.

6. Final Remarks and Preview

In the paper dynamic properties of the automotive turbocharger rotor-shaft supported on the electro-dynamic passive magnetic bearings (EDPMB) have been investigated. Here, the analogous suspension of this object by the floating-ring journal bearing, commonly applied till present, was regarded as a reference. Such a comparison has indicated essential advantages of the proposed kind of magnetic contact-free and lubrication-free support for the relatively light-weight turbocharger rotor-shafts rotating within very broad speed ranges. Moreover, a properly selected visco-elastic design of the EDPMB stators and their flexible embedding in the turbocharger housing can assure asymptotic stability of the considered rotor-shaft systems. Using such a simple mean, it was possible to introduce a sufficient magnitude of additional external damping into the vibrating rotor-shaft system in order to satisfy the Routh-Hurwitz stability criterion as well as to keep all its eigenvalue real parts always negative. Thus, it turned out that a suspension of relatively small and very quickly rotating automotive turbocharger rotor-shafts by the EDPMBs is particularly advantageous and perspective. Nevertheless, the main target of the introductory theoretical study presented here reduced to lateral vibration- and stability analyses. But in order to realize this idea in an engineering and industrial application many further investigations are required. First of all, this is an axial support in the form of a proper magnetic thrust bearing. Then, it will be necessary to design relatively simple and robust touch-down bearings. It can be expected that an application of the EDPMBs should not cause many problems in design of the turbocharger housing and shaft. Namely, a substitution of the lubricating oil supply system by properly embedded permanent magnets as well as an installation of the touch-down bearings do not seem to be particularly expensive and difficult. Consequently, the rotor-shaft journals should be easily substituted by conductor sleeves. Nevertheless, several thermal loading phenomena as well as many other problems hard to foresee now ought to be solved using an appropriate real prototype of the considered object.

References

Amati, N.; De Lépine, X.; Tonoli, A.: Modeling of electrodynamic bearings, *ASME Journal of Vibration and Acoustics*, Vol. 130, (2008), 061007-1–061007-9.

Bonello, P.: Simulations of the nonlinear dynamic response of a turbocharger on foil-air bearings with focus on unbalance excitation, Proc. of the 11th IMechE Int. Conference *“Vibrations in Rotating Machinery”* VIRM-11, September 2016, Manchester, UK (2016), 419–429.

Falkowski, K.: *Passive Magnetic Suspensions*, Military University of Technology Eds., Warsaw (2016).

Filatov, A. V.; Maslen, E. H.; Gillies, G. T.: Stability of an electrodynamic suspension, *Journal. of Applied. Physics*, Vol. 92, (2002), 3345–3353.

Göbel, S.; Daniel, Ch.; Woschke, E.; Strackeljan, J.: DoE basierte Sensitivitätsanalyse konstruktiver Lagerparameter eines gleitgelagerten Abgasturboladers, Proc. of the 11th Int. Tagung *“Schwingungen in rotierenden Maschinen”* (SiRM), Magdeburg, Germany, February 2015, Paper ID-55 (2015).

Kamesh, P.: *Oil-Whirl Instability in an Automotive Turbocharger*, University of Southampton Research Repository, Ph.D. Thesis (2011), http://eprints.soton.ac.uk.

Koutsovasilis, P.; Driot, N.: Turbocharger rotors with oil-film bearings: sensitivity and optimization analysis in virtual prototyping, Proc. of the 11th Int. Tagung *"Schwingungen in rotierenden Maschinen"* (SiRM), Magdeburg, Germany, February 2015, Paper ID-43 (2015).

Lempke, T. A.: *Design and Analysis of a Novel Low Loss Homopolar Electrodynamic Bearing*, Doctoral Thesis, KTH Electrical Engineering, Stockholm (2005).

Schweizer, B.: Vibrations and bifurcations of turbocharger rotors, Proc. of the 8th Int. Tagung *"Schwingungen in rotierenden Maschinen"* (SiRM), Vienna, Austria, February 2009, Paper ID-23 (2009).

Someya, T.: *Journal-Bearing Databook*, Springer-Verlag Berlin, Heidelberg, New York (1989).

Szolc, T.: On the discrete-continuous modeling of rotor systems for the analysis of coupled lateral-torsional vibrations, *International Journal of Rotating Machinery*, 6(2), (2000), 135–149.

Szolc, T.; Falkowski, K.: Dynamic analysis of the high-speed flexible rotors supported on the electrodynamic passive magnetic bearings, *Mechanisms and Machine Science*, Springer Verlag, Vol**.** 21 Part XI, (2014), 1489–1500.

Szolc, T.; Falkowski, K.; Henzel, M.; Kurnyta-Mazurek, P.: Passive and active stabilization of the electrodynamic magnetic bearings supporting high-speed rotors, Proc. of the 11th IMechE Int. Conf. *"Vibrations in Rotating Machinery"* VIRM-11, September 2016, Manchester, UK, (2016), 721–731.

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