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# MODELLING FOR A TURBOCHARGER IN ROTORDYNAMICS

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## ABSTRACT

*Turbochargers are widely used on commercial automotive vehicles, power generation and marine applications. A comprehensive model for a turbocharger would be an effective tool for fault detection. In this paper, research focus on mathematical modelling of turbochargers supported on floating ring bearings in rotordynamics. Nonlinear hydrodynamic forces are considered in order to describe the dynamic behaviour of the rotor system. Following model development, a numerical simulation is implemented. Vibration of turbocharger and floating ring under the influence of rotor imbalance are predicted.*

**Keywords** turbocharger modelling nonlinear

## 1    INTRODUCTION

Turbocharger, an air compressor for an internal combustion engine, comprises an air intake, an air outlet, a compressor and a turbine linked by a shared shaft. It is used to increase the mass of air entering into engine to create more power by its own exhaust gases. The compressor increases the pressure of the air entering the engine, so the fuel/air mixture greatly improves the volumetric efficiency of the engine, the engine power could be increased significantly and meanwhile less fuel would be consumed. Therefore it has been more and more widely used in such machineries as automotives, road vehicles and power generation, etc.

With the development of turbochargers, relevant theoretical research becomes more significant and necessary, since a comprehensive model would be very helpful for us to investigate its working conditions. More significantly a mathematical model could be applied in fault detection, classification and prognosis by comparing simulation results to those experimental data collected from sensors. This kind of model-based approach combining with modern signal processing techniques would obviously be more efficient and accurate than those traditional fault detection methods.

It is the importance of models for turbochargers and other turbomachineries in fault detection that attract more and more researchers to pay attention to this field. Chen, Hakeem and Martinez-Botas (1996) designed a model for a turbocharger turbine under pulsating inlet conditions. One-dimensional unsteady flow method is applied to a mixed flow turbine under steady and unsteady flow conditions. With the new loss model being added, it is shown that this method can predict the instantaneous mass flowrate of the mixflow under pulsating flow conditions. Aretakis and Mathioudakis (1998) apply pattern-recognition techniques to classify radial compressor faults. Three classification methods--geometric, statistical and statistical pattern recognition using optimal directions, are used to process the signals (sound emission and vibration) collected from sensors and the simulation results. Such four kinds of faults as inlet obstruction, obstruction in a diffuser passage, variation of impeller tip clearance and impeller fouling are used to test those methods and it was demonstrated that all of the three approaches can classify faults with high successful rate. Peat, et al (2006) do research on the acoustic effect of the turbine. They present a comprehensive model of the acoustic propagation through the entire exhaust system of a turbocharged engine. The experiment data show that the predicted results for transmission loss given by this model are in good agreement with the experimental results. Katrasnik (2006) designed a model to determine the boundary condition of the turbine and compressor. An innovative algorithm is developed for simulation of a turbocharger gas turbine and compressor combining the extended equations. It makes equations solved considering variations in gas properties in every time step without costing more computational time.

In this paper, research focus on the characteristics of turbochargers in rotordynamics and a model has then been developed for rotor imbalance, the most common fault of turbochargers. The main breakthrough in modelling is nonlinear forces of the two oil films in the floating ring bearings. According to the author's knowledge, most of researchers make such nonlinear linearization by means of Taylor series expansion. Following high order sections ignored, the original nonlinear oil film forces are expressed as the linear combination of the stiffness coefficients and damping coefficients. Such linearized model is generally adequate to calculate the vibration supported by hydrodynamic bearings. However, it is not sufficient to simulate those nonlinear behaviours, for example oil film instabilities. Recently more and more researchers

pay their attention to this nonlinear area. Since the solution of nonlinear equation is restricted by the development of mathematics and computer calculation capability, its development is much slower than the linear field. The latest research achievement in the area of the nonlinear rotordynamics is from Helio Fior de Castro, et al (2008) who obtained the analytical expression of the nonlinear forces of infinite short journal bearing and simulated oil whirl and whip instabilities of the simple Jeffcott rotor supported by hydrodynamic bearings under the influence of rotor imbalance. Unfortunately there is still no nonlinear rotordynamic model for the turbocharger core supported by two floating ring bearings so far.

According to mentioned above, a comprehensively nonlinear rotordynamic model of turbocharger is developed for nonlinear system vibration simulation under the influence of rotor imbalance.

## 2  FLOATING RING BEARING

Bearings, as the most important part, determine the quality of turbochargers greatly. Most of commercial automotive turbochargers incorporate floating ring bearings to support the rotor because general journal bearings could not bear the extreme high angular velocity. Compared to general journal bearing, floating ring bearings include such advantages as: (a) Two oil films increases the flow of lubricating oil so that the bearing temperature could be decreased without increasing the clearance between journal and the fixed bearing. (b) Rotation of ring in a certain velocity decreases the relative speed of journal against the bearing and thus the friction will be reduced since the value of friction power is proportional to the square of the angular velocity. (c) Floating with two oil film lands is a kind of great elastic support. It can be considered as a good buffer for the system vibration caused by such faults as imbalance, oil whirl and whip, etc. consequently floating ring bearing is applied to the high speed and light loaded turbochargers.

Figure 1 shows the coordinate system of cross section of the floating ring bearing. The coordinate origin is fixed at the centre of the bearing for all coordinate system unified. Although obviously the bearings of automotive turbochargers might move as the whole vehicles vibrate, in this paper bearing is assumed to be fixed for analysis simplification. Therefore vibration of other parts is actually the relative vibration in terms of the 'fixed' bearing.

Unlike ordinary journal bearing, floating ring bearing has two oil films (inner oil film between journal and floating ring, outer oil film between floating ring and bearing) and oil film pressure is determined by the relative motion of such solid parts as journal, floating ring and bearing. The lubricating oil is fed into outer oil film by an axial groove from centre to near to the edge of the bearing. As the radial gradient of oil film pressure has been ignored, it can be considered that oil supply pressure of two oil films are approximately the same (Clarke et al (1992)). Hence apart from the middle of the bearing, the oil film pressure of both outer and inner oil film could be estimated by Reynolds equation:

$$\frac{\partial}{\partial \Psi}\left(\frac{h}{12\mu}\frac{\partial P}{\partial \Psi}\right) + \frac{\partial}{\partial z}\left(\frac{h}{12\mu}\frac{\partial P}{\partial z}\right) = \frac{V}{2}\frac{\partial h}{\partial \Psi} + \frac{\partial h}{\partial t} \tag{1}$$

For outer oil film, $h$ represent the gap of floating ring and bearing, $V$ is the linear velocity of floating ring, $\Psi$ and $z$ are the circumference and axial coordinate of outer oil film; for inner oil film, $h$ represent the gap of journal and floating ring, $V$ is relative linear velocity of floating ring and journal, $\Psi$ and $z$ are the circumference and axial coordinate of inner oil film respectively.

So far the attempts to obtain the effective analytical expression of the whole Reynolds equation are still unsuccessful. Although the method of infinite series could express the oil film pressure, it is not suited to the complicated calculation later. The numerical approach is therefore generally introduced here to deal with it. Finite difference method is adopted in this paper to solve Reynolds equation of both outer and inner oil films. The output two dimensional oil film pressure is necessary for the analysis of characteristics of the bearings.

The floating ring rotational speed largely determines the steady state and dynamic force response of the turbocharger core system which is usually estimated by isothermal fluid flow analysis so that a fixed ring speed ratio could be obtained, but the relevant experimental data demonstrate that it decreases rapidly with bearing temperature increases (Andres et al (2004)). In order to obtain a more accurate ring rotational velocity, thermal effect is considered in this paper. The temperature increases of such solid parts as journal, ring, and bearing due to friction of oil film surfaces are firstly calculated according to the L San Andres' lumped thermal model (Andres et al (2004)) and the thermal expansions are then estimated. The new gaps $h$ in the Reynolds equation (1) give rise to the change of oil film pressure distribution as well as oil film reaction forces calculation and thus a new predicted floating ring rotational speed will be obtained. Here thermal effect is based on a steady state calculation rather than transient state which means the temperature increases of those solid parts are considered as average values for a certain rotational speed.

# 3 TURBOCHARGER MODELING IN ROTORDYNAMICS

The scheme of a turbocharger core is shown in figure 2 and figure 3 illustrates its finite element model of seven mass points linked by elastic cylindrical shafts. The seven mass points represent two locknuts (node 1 and 7), two floating ring bearing (node 3 and 5), the middle shaft (node 4), the turbine (node 2) and the compressor (node 6). According to the knowledge of multi-degree of freedom vibration, the system vibration equation without influence of damping is given by:

$$[M]\{\ddot{U}\} + [K]\{U\} = \{F_u\} + \{F_{inner}\} - \{W\} \qquad (2)$$

where $\{U\} = \{x_1, x_2, ..., x_7, y_1, y_2, ..., y_7\}$ is the system displacement vector representing rotor system vibration, [M] and [K] are mass and bending stiffness matrix, {W} is the weight of turbocharger core, $\{F_u\}$ is unbalance centrifugal force vector, $\{F_{inner}\}$ is hydrodynamic force of inner oil film.

The vibration of the ring is determined by vector sum of inner and outer oil film hydrodynamic force (3):

$$m_R \ddot{u}_R = -\vec{F}_{inner} + \vec{F}_{outer} \qquad (3)$$

where $m_R$ is the mass of ring, $u_R$ is the displacement of ring.

In order to find out nonlinear characteristics of hydrodynamic forces, $\{F_{inner}\}$ is kept on the right hand side of the equation instead of considering it as the linear combination of stiffness and damping coefficients. Theoretically vibration of those mass points as well as the rings could be calculated by solving equations (2) and (3) together. However the nonlinear characteristics of oil film forces make the calculation more complicated. Although the analytical expression is obtained of hydrodynamic force of the infinite short cylindrical journal bearing, unfortunately this achievement is not suited to the floating ring bearings of the automotive turbochargers, whose width-radius ratio is about one. Obviously neither infinite long or short bearing theories would meet the demands.

In order to overcome the problems mentioned above, numerical method is implemented in this paper. The whole simulation period is firstly discreticized to a number of time intervals. During one of them hydrodynamic forces are assumed as constant whilst unbalance centrifugal forces still vary as time increases. In each time interval, two oil film pressure distributions and reaction forces are solved numerically, the results of which are then substituted into rotor system equations as well as ring vibration. By integrating those equations in this time interval, displacements and velocities of all the system nodes and rings at the end of this period could be calculated, that will be then substituted back into Reynolds equation as the new initial parameters for the next interval. It can be seen that bending vibration of turbocharger as well as ring orbit could be simulated if only the time interval is small enough.

# 4 SIMULATION RESULTS

In this paper, the rotational velocity of turbocharger is at a constant value 2000rad/s and that of floating ring is then estimated at 832rad/s according to the thermal balance equation (Andres et al (2004)), ignoring the influence of torsional vibration due to exhaust pulses. The pressure of lubricating oil at the supply oil hole is set to $1.38 \times 10^5 Pa$ and $1.00 \times 10^5 Pa$ for outside bearings.

Figure 4 show the orbit of journal and ring under the influence of rotor imbalance $3.00 \times 10^{-6} kgm$ on the turbine surface. Due to nonlinear hydrodynamic forces the turbocharger whirls inside the bearing rather than rotates at the equilibrium position and results predicted also demonstrate that the orbits are approximately the same, which means in terms of ring the eccentricity of journal is very small.

Figure 5 displays the vibration of turbocharger on the horizontal plane and Figure 6 illustrates its spectrum for the steady state vibration. Vibration simulated of turbochargers under the influence of imbalance step into steady state after 0.45s while the displacement has not converged before that because of the difference between realistic and boundary condition assumed at first. In the steady state of vibration, it can be seen from the spectrum figure that vibration mainly occurs at the rotational speed around 400rad/s, approximate half rotational speed of the floating ring (832rad/s), whilst the $1\times$ synchronous vibration of the turbocharger at 2000rad/s is much weaker. The simulation results show the existence of the phenomenon 'oil whirl' of

floating ring bearings of turbochargers, similar to the ordinary journal bearing, which is caused by the nonlinear hydrodynamic oil film forces.

## 5   CONCLUSION

The structure and relevant theories about floating ring bearings of turbochargers has been illustrated and finite element method has been introduced to discreticize the turbocharger core system. Following forming the system bending vibration equations considering such exciting forces as rotor imbalance centrifugal forces, nonlinear hydrodynamic oil film forces as well as the rotor's own gravity, numerical approach has then been implemented to deal with the mathematical problem. Nonlinear vibrations under the influence of rotor imbalance of turbocharger and floating ring are simulated and the predicted results show the nonlinear behaviours of floating ring bearings of turbochargers 'oil whirl'.

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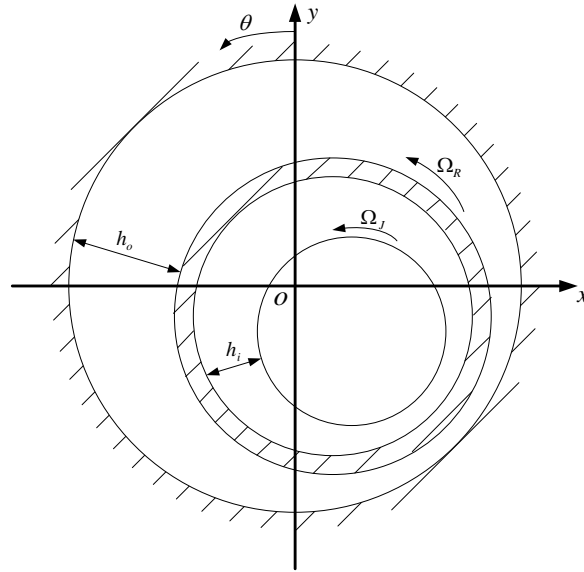
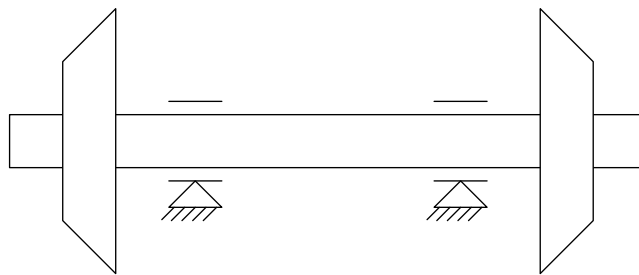**Figure 1: Coordinate system of floating ring bearing**
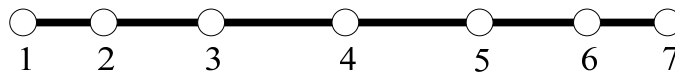


**Figure 2: Physical model of turbocharger core**



**Figure 3: Finite element model of turbocharger core**
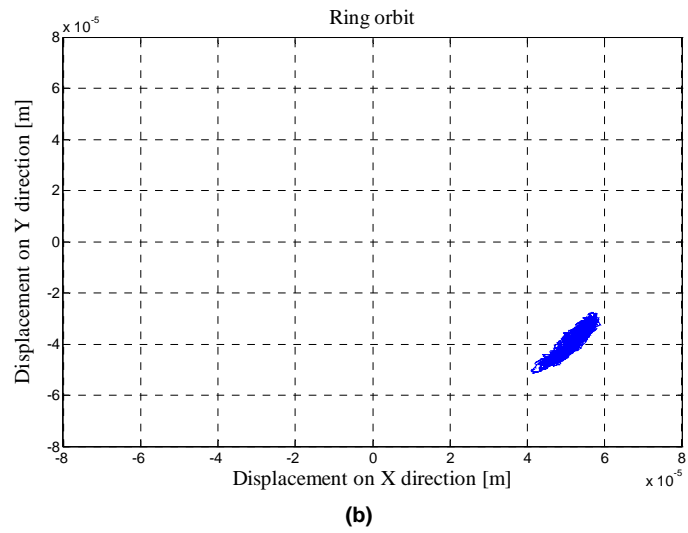


Journal orbit

**(a)**
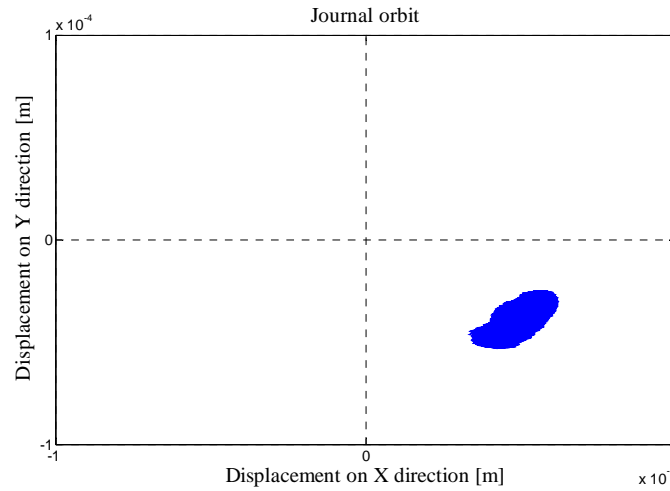
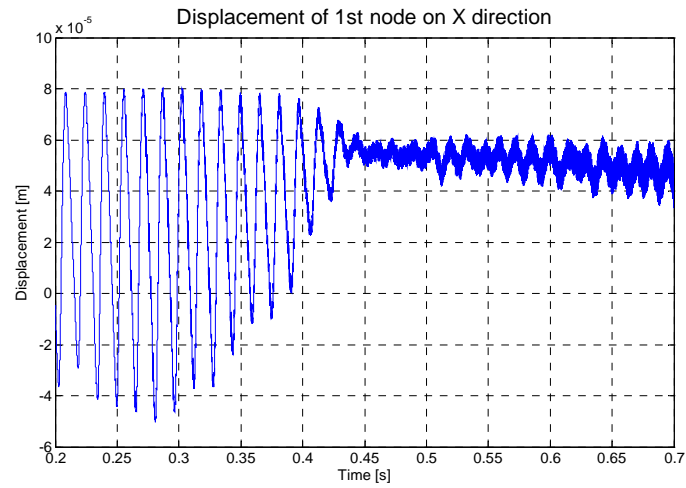**Figure 4: Orbit of turbocharger (a) and floating ring (b)**



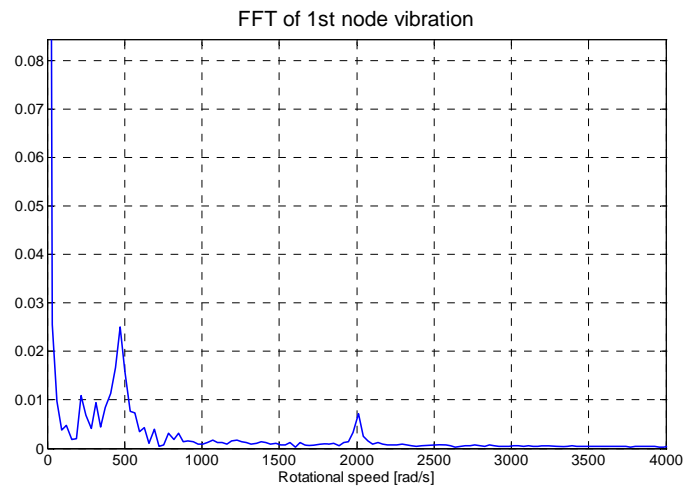**Figure 5: Displacement of turbocharger on horizontal plane**



**Figure 6: Spectrum of turbocharger vibration**

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