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COMPUTATIONAL ESTIMATION OF SEALING GAS BLOWBY

Kudláček P.[*], Novotný P.[**], Vacula J.[***]

Abstract: *From the design and principle of operation of the sealing system of turbochargers, it is obvious that the seal of the non-contact type cannot perfectly close and separate the central housing from the compressor and turbine housing, thus gas blowby occurs. Approaches to determine the gas flow through a sealing system based on CFD models or direct measurement are often, expensive, time consuming and require some computational power. For rapid analysis of turbocharger blowby, analytical relationships can be used that provide sufficient accuracy. This paper demonstrates the use of these simplified analytical relations and expresses their agreement with measurements made on a real turbocharger under real operating conditions. The results show minor differences between the performed measurement and the analytical calculation. This can be attributed to the many simplifying assumptions in place. In a real machine, due to variable operating conditions, the seal rings move and thus the axial clearances change. A closer analysis of the ring movements could decrease the difference between experiment and calculation results.*

Keywords: Seal, Turbocharger, Gas blowby, Emissions, Side clearance.

1. Introduction

The problem of air pollution caused by internal combustion engines (ICE) is perceived as serious. The most discussed topic is the elimination of fuel consumption and CO_2 production in emissions. However, the operation of ICE also generates significant production of unburned hydrocarbons (HC), particulate matter (PM) and carbon monoxide (CO). Current emission standards penalise the production of these pollutants and thus provide an incentive to gain knowledge of the mechanisms affecting their formation.

Turbochargers (TC) are used to increase specific performance and allow the ICE to reduce fuel consumption. Development trends are pushing for increased ICE efficiency, this can also be achieved by increasing the pressure ratios on the TC. This solution creates problems with the sealing of the TC's central bearing housing. Since a non-contact solution using seal rings is used for gas flow sealing, there is a negative process called gas blowby. That is, the gas leakages through the sealing system, which can occur in any direction depending on the actual pressure conditions and other operating conditions.

The blowby has several negative effects on the powertrain and the surrounding environment. Gas leakage from the turbine or compressor section results in a loss of gas pressure that could have been used for energy conversion. This blowby, which is also called standard blowby, also results in another negative impact, namely engine oil pollution from exhaust gases and carbon particles. In the case of gas flow into the central housing, there is also a change in the properties of the engine oil due to foaming, this reduces the reliability of the ICE, which can be critical for some applications (Furch, 2014). Furthermore, gas flow also affects the generation of emissions, either directly or indirectly. Under certain operating conditions, if there is backflow, i.e., from the central housing to the turbine or compressor casing, engine oil will be entrained with the air and will burn in the turbine. In the event of a leakage into the compressor, the oil mist will be fed into the ICE combustion chamber where it will subsequently be burned. The combustion of this oil mist will increase the production of pollutants, in particular HC, CO and PM. In the case of indirect effects, fouling and clogging of ICE compartments and components occurs, which in turn affect the efficiency and

[*] Ing. Petr Kudláček: Institute of Automotive Engineering, Brno University of Technology, Technická 2896/2; 616 69, Brno, CZ, Petr.Kudlacek@vutbr.cz

[**] Assoc. Prof. Ing. Pavel Novotný, Ph.D.: Institute of Automotive Engineering, Brno University of Technology, Technická 2896/2; 616 69, Brno, CZ, novotny.pa@fme.vutbr.cz

[***] Ing. Jiří Vacula: Institute of Automotive Engineering, Brno University of Technology, Technická 2896/2; 616 69, Brno, CZ, jiri.vacula@vutbr.cz

emissions of engine pollutants, such as the throttle flap, swirl flaps, EGR valve, particulate filter, catalytic converter, etc.

TC gas blowby can be caused by two basic mechanisms. The first one is the gas flow through ring gap (see Figure 1), this blowby in the TC sealing system has a marginal effect from the overall point of view and is only influenced by the geometrical properties of the lock and the pressure gradient. The second mechanism is caused by a gas flow under the seal ring (see Figure 1), i.e., in the space between the rotor and the seal ring. This space is "U" shaped in cross section. In addition to the pressure gradient, the movement of the shaft in the axial direction has a major influence on the blowby mechanism.

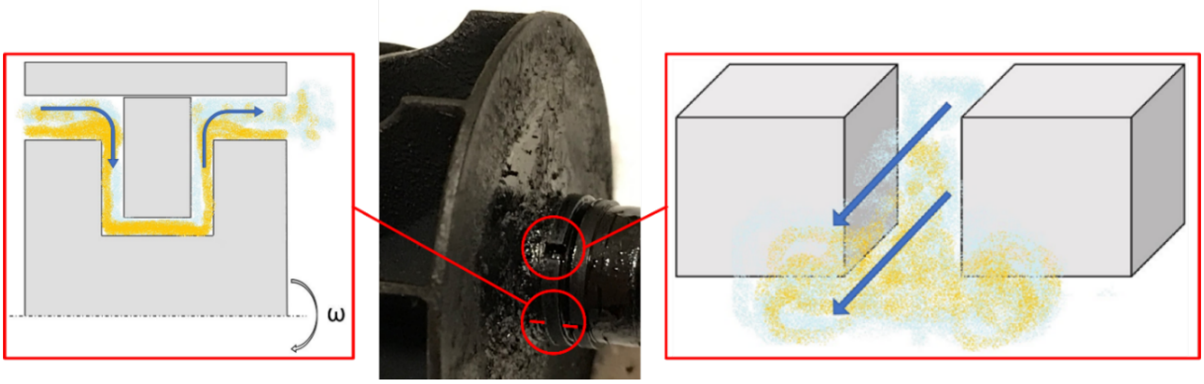


Fig. 1: Diagram of the gas blowby mechanism under the sealing ring (left), ring gap (right).

2. Methods

2.1. Review of state of the art in blowby modelling

The issue of the blowby through the sealing system of TCs is not described in detail in the literature. However, several articles have been written on this subject. However, the issue of TC gas flow is to some extent analogous to the gas flow through the piston seal rings of ICEs. One of the earliest descriptions of the mechanisms of gas blowby in ICE was given by Furuhama (1959). Models for the calculation of gas blowby have gradually evolved from the simplest 1D models described by the Reynolds "lubrication" equation, through the description of various effects such as the influence of ring surface roughness, ring-casing contact, shaft-to-ring eccentricity with the application of flow correction coefficients, to more detailed 2D models described by the Navier-Stokes equations requiring some computational power. The flow of gas through a thin gap modelled as a mixture of air and oil using a 2D model has been described, for example, in (Oliva, 2015). In general, most authors solve the gas flow through the thin gap using the averaged Reynolds equation, such as in (Novotný, 2013). Regarding the solution of the flow through a ring gap, for example, Tian (1998) and Raffai (2015) chose the isentropic labyrinth flow model, as most authors.

The identification and modelling the gas blowby of TC under simplifying assumptions has been presented previously (Hong et al., 2009). In this work, he performed simulation using CFD software and verification by measurements. As a result, there was a discrepancy between the simulations and experiments, which the authors clarified by introducing simplifications for the simulations performed. Among the most significant omissions, Hong neglected of the seal ring motion due to radial and axial rotordynamics, from which he concluded that the gas blowby is strongly dependent on the axial clearance. Similar work was presented by Kondhalkar (2015), but his effort was to model oil leakage from the central housing, particularly problematic on the compressor section of the TC. The author performed CFD simulation and conducted an oil leakage experiment. The results show that oil leakage increases with increasing inlet oil pressure at constant temperature, and with increasing temperature at constant pressure.

2.2. gas blowby computational model

The gas flow through the ring gap can be modeled as an isentropic discharge through a small opening from an infinitely large vessel. The mass flow is expressed by the following analytical relations (Novotný, 2013)

$$\dot{m}_{\mathrm{ori}} = A_{\mathrm{E1}} \rho_1 \sqrt{\kappa R T_1} \qquad \text{for} \qquad \left(\frac{p_1}{p_0}\right) \leq \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}, \tag{1}$$

$$\dot{m}_{\mathrm{ori}} = A_{\mathrm{E1}}\sqrt{\rho_0 p_0}\sqrt{\frac{2\kappa}{\kappa-1}\left(\left(\frac{p_1}{p_0}\right)^{\frac{2}{\kappa}} - \left(\frac{p_1}{p_0}\right)^{\frac{\kappa+1}{\kappa}}\right)} \quad \text{for} \quad \left(\frac{p_1}{p_0}\right) > \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}, \tag{2}$$

where $\dot{m}_{\mathrm{ori}}$ is the mass flow, A_{E1} is the efficient area, R is the gas constant, T is the temperature, p_0 and p_1 are the pressures in the first volume and the second volume respectively, ρ is the gas density and κ is the heat capacity ratio. The flow of gas through the clearance between the ring and the groove can be described by the Reynolds equation as laminar flow under simplifying assumptions. By neglecting centrifugal accelerations, and assuming a logarithmic pressure distribution, the relationship can be written to calculate the mass flow (Stachowiak, 2001) in the form

$$\dot{m}_{\mathrm{clr}} = \frac{\pi\rho h^3 (p_0 - p_1)}{6\eta}\frac{1}{\ln\left(\frac{r_2}{r_1}\right)}, \tag{3}$$

where h is the gap thickness, r_1 and r_2 are the inner and outer radius of the gap and η is the dynamic viscosity. The parameter that varies during the TC operation and has a major influence on the size of the gas blowby according to (Hong, 2010) is the axial clearance h_{sr} between the ring and the groove. This clearance depends on many influences such as mounting position, thermal expansion, rotordynamics and rotor axial load due to which the thickness of the thrust bearing lubrication gap changes. The position of the rings, and therefore the resulting clearance, can be obtained by several approaches such as experimental determination of the ring position or by more detailed simulations. However, these approaches are time, power and cost consuming. Based on established simplifying assumptions, a simple determination of axial clearance relative to motion in the thrust bearing can be made.

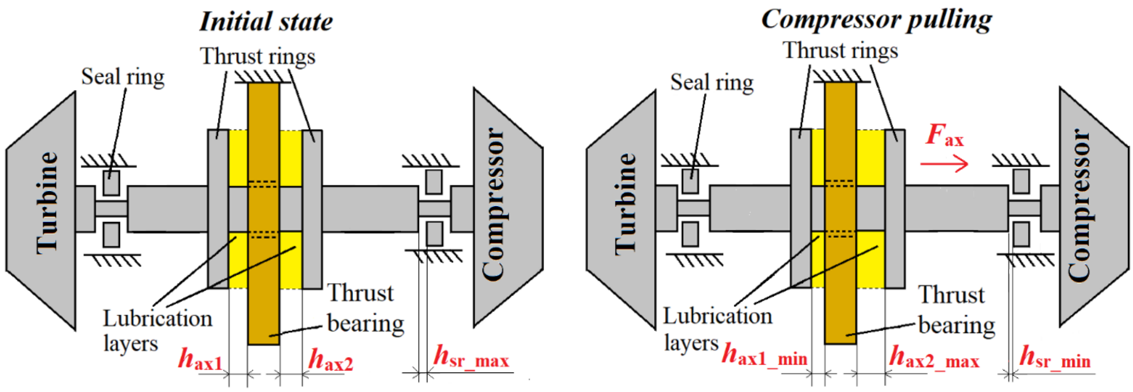
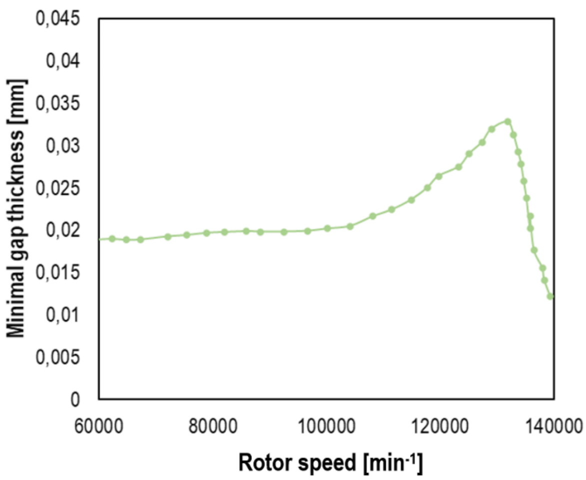


Fig. 2: Different rotor states. Initial state on the left, compressor pulling on the right.

If we consider the unloaded rotor as the initial state, i.e. the thrust bearing is located between the thrust rings in the centre, and the sealing rings are located in the centre of the rotor grooves, then in this initial state there is maximum gas blowby, as there is also maximum clearance between the seal rings and the rotor. From this position, the rotor will be displaced by Δx after an axial load is applied to the rotor, thus reducing the clearance h_{sr}. The thrust bearing clearances h_{ax1} and h_{ax2} presented in Fig.2 can by calculated by thrust bearing model to match the resulting axial load F_{ax} (Novotný, 2020) or can be determined experimentally. The determination of the axial force is presented, for example, by Nguyen-Schäfer (2015). In the case of this study the thrust bearing clearances are obtained experimentally on the target turbocharger. The resulting loading depends on the geometrical parameters of each TC and the operating conditions.

3. Result discussions

The gas flow calculation was performed for a range of clearances corresponding to the axial loading of the TC in the rotor speed range (60000–140000) min^{-1}. Based on the minimum thrust bearing lubrication gap determined experimentally, the gaps between the seal rings and the rotor were then calculated (see Figure 3). The resulting plot of the calculated gas blowby versus rotor speed can be seen in Fig. 3. At lower and medium TC speeds the calculated gas blowby does not reproduce the measured data. This fact can be mainly attributed to the simplifying assumptions introduced, such as the fixed position of the rings relative to the casing, the neglect of thermal expansion, the occurrence of clearances in the seal uniformly for all rings and the neglect of the effect of radial rotordynamics.



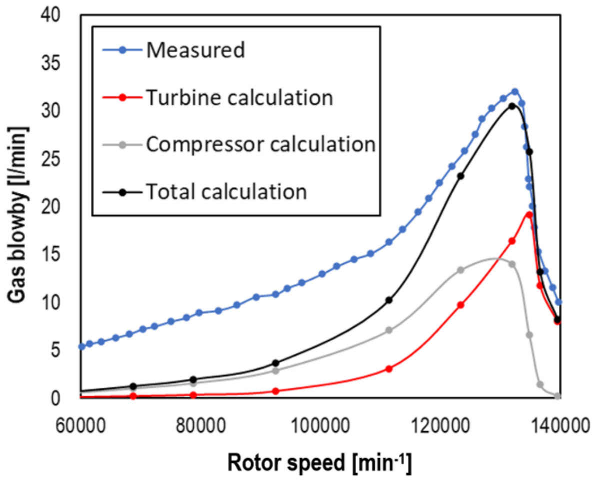


Fig. 3: Minimum thrust bearing gap thickness (left), calculated and measured volumetric gas flow through the TC sealing system (right).

In the medium-high and higher TC rotor speeds, on the other hand, a match between the measured data and the calculated TC gas blowby can be observed. From the course of the total flow in Fig. 3 and the course of the minimum lubrication gap of the thrust bearing plotted in Fig. 3, it is possible to deduce, and therefore confirm, the significant influence of the axial movement of the rotor on the total flow of the TC.

4. Conclusions

The proposed approach for rapid calculations of the TC gas blowby achieves a satisfactory agreement with the experiment, especially under operating conditions with high pressure ratios in the impellers. Nevertheless, this approach to the solution shows some discrepancies with the performed experiment given mainly by simplifying assumptions. From the performed calculations, the determining parameter of the gas blowby is also evident, namely the axial position of the ring relative to the rotor groove and thus the resulting clearance. In order to refine this simple analysis, it would be necessary to carry out a study of the movement of the rings, taking into account more effects, i.e., rotordynamic, thermal expansion and others.

Acknowledgement

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