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UNI STUTTGART

Fachverband Fluidtechnik im VDMA e. V. Lyoner Str. 18
50628 Frankfurt am Main
Germany

Phone +49 69 6603-1513
E-Mail maximilian.baxmann@vdma.eu

Internet www.vdma.eu/fluid

Digital Sealing Development for Turbochargers

Björn Kunzelmann

1 Introduction

Turbocharger seals are subjected to particularly harsh conditions. High circumferential speeds, elevated temperature levels and contamination make the use of contacting radial shaft seals with rubber components almost impossible. For large axial turbochargers used e.g. on marine engines, therefore a combination of piston rings and sealing air systems are employed to achieve the necessary sealing functions. In particular, the blow-by level entering the bearing casing and the oil leaking out of the bearing casing has to be controlled for every possible operation condition to prevent downtime and operational impact on customer engines.

Traditional sealing development relies heavily on experimental testing resulting in costly and time-consuming development processes. Thus, this paper will present a combined numerical and analytical approach to predict/assess the above-mentioned sealing functions already during the development stage for a newly developed high-pressure turbocharger. With a detailed CFD study the sealing air system could be optimized and boundary conditions for the piston ring seals could be predicted for the subsequent mechanical design of the piston ring seal. With an analytical model employing a Monte Carlo approach, a robust piston ring seal design could be derived considering production tolerances and variation of friction coefficients of the tribological interfaces.

This consequent digital approach allows to derive key parameters like pressure differences across the seals, temperatures, blow-by magnitude and sealing air consumption before the first validation tests. Subsequently those predicted key parameters are then compared to experimental results to judge the accuracy of the employed digital approach. It can be shown that with use of the digital approach the sealing performance can be accurately predicted and consequently its use ensures an efficient and cost-effective approach for product development of turbocharger sealing systems.

2 Problem Description

The investigated sealing system is employed in a 2-stage turbocharging system on a 4-stroke medium speed engine used for e.g. marine propulsion or power generation as shown in Figure 1. The sealing system is used in the high-pressure stage which exhibits higher pressure and temperature levels than the low-pressure stage. Strict and challenging blow-by and lifetime requirements have to be met as failure of the sealing systems often implies an engine malfunction with potential downtime for the engine operator.

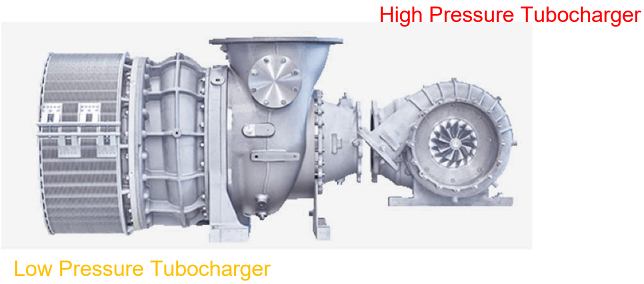


Figure 1: Schematic View: 2-stage turbocharging system for 4-stroke medium speed engines

The significantly smaller framesize of the high-pressure unit automatically leads to higher material temperatures increasing the oil coking risk within the bearing casing and sealing system. To decrease the development costs and time effort a digital approach was used to speed up time to product release.

2.1 Overview Sealing System

The sealing system of the high-pressure turbocharger consists out of a piston ring set of 2 rings. The piston ring pack is positioned on turbine and compressor side respectively, see Figure 2.

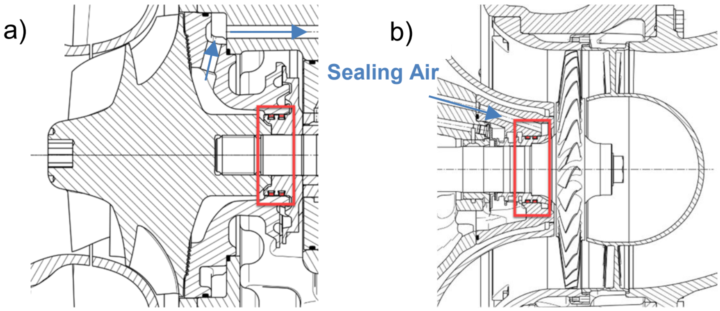


Figure 2: Sealing Arrangement a) compressor side b) turbine side

Purpose of the piston ring seals is to ensure a controlled and limited flow of blow-by gas into the bearing casing. The clamping force of the piston ring pack is designed so that the gas force acting on the piston ring is always smaller than the clamping force. The axial clearance of the rotor is bigger than axial clearance of the piston ring in its groove. Thus, in operation e.g. during run-up/down the position of the piston ring is changed by the shaft influencing the size of the sealing gap.

In addition to the piston ring seal, a sealing air system is employed to achieve following additional functions:

- Reduce exposure of TS-seal with dirt particles from exhaust gas
- Protection from corrosive components stemming from the exhaust gas
- Cooler sealing area (fewer oil coking deposits in piston ring area)

The sealing air system uses air from the compressor wheel outer diameter which is relatively clean and considerably colder than the exhaust gases driving the turbine. Through a sealing air pipe, the air is guided to a ring channel on turbine side (see Figure 2 a) and b).

2.2 Modelling Approach

The modelling approach consists out of 2 parts:

1. A CFD model of the sealing air system to assess the function of the sealing air system and derive the pressure differentials for both seals to conduct the analytical sealing assessment.

2. Analytical Model: Mechanical and temperature model to assess the piston ring design and predict overall sealing performance.

Both modelling approaches are now described in detail in the following subchapters.

2.2.1 CFD Model

Figure 3 shows a schematic of the employed single phase CFD model (ANYS CFX). Modelled is the sealing air system including the back disk cavity of turbine and compressor wheel. For the correct estimation of the pressure differential across the seals from particular importance is to resolve the velocity profile in those cavities as they are directly adjacent to the piston ring seals. Boundary conditions have been derived for an engine full load case which imposes the most severe conditions onto the sealing system. Pressures and temperatures for the compressor have been derived with the Euler equation assuming a typical flow angle under those operation conditions. This allows to estimate the static pressure at the inlet which is positioned at the outer diameter of the compressor wheel. Boundary conditions at the turbine wheel outlet have been estimated with known pressure loss between turbine outlet flange and turbine outlet position. Subsequently by using matching data, which refers to turbine outlet flange, allows to estimate the static pressure imposed on the turbine outlet position (see Figure 3).

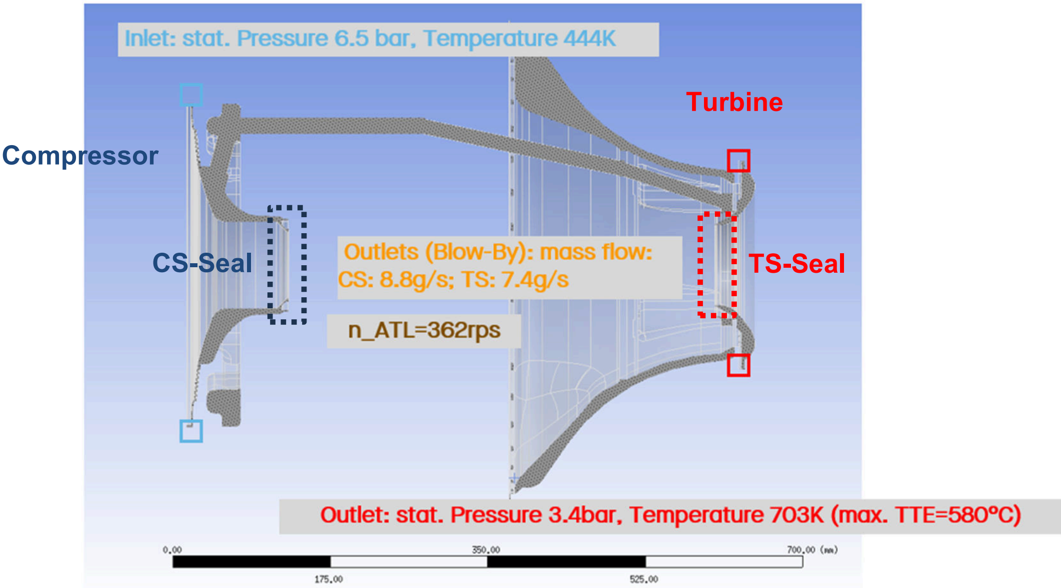


Figure 3: CFD Model Overview

Fluid properties have been chosen as air, governed by the ideal gas law. The air viscosity temperature relationship is described by the Sutherland law. The walls for the whole model have been chosen as being adiabatic with no slip at the wall. Wall velocities for the fluid with contact to rotating parts have been set with a rotating wall boundary condition at 362rps. Figure 4 shows the meshing employed. Figure 4 a) shows the mesh of the compressor wheel labyrinth.

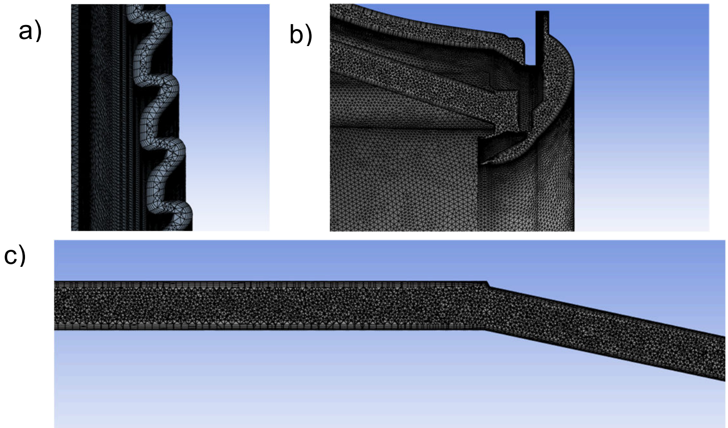


Figure 4: Details from CFD mesh (31.584.612 elements) a) compressor inlet b) turbine wheel backwall c) sealing air channel

Figure 4 b) points out the exit of the sealing air channel with the turbine side ring channel, Figure 4 c) the sealing air channel itself. The wall scale is meshed with 5 prism layers increasing its height with a factor of 1.2 towards the tetra mesh employed for the bulk flow region. The turbulence model was chosen as SST model with automatic wall treatment employing automatically a wall function approach when grid density becomes too coarse for properly resolving the wall scale [1].

Figure 5 shows a representative contour plot for the pressure distribution in the sealing air system. More detailed investigation/optimizations into the performance and validation with experimental results of the sealing air system are shown in chapter 3.

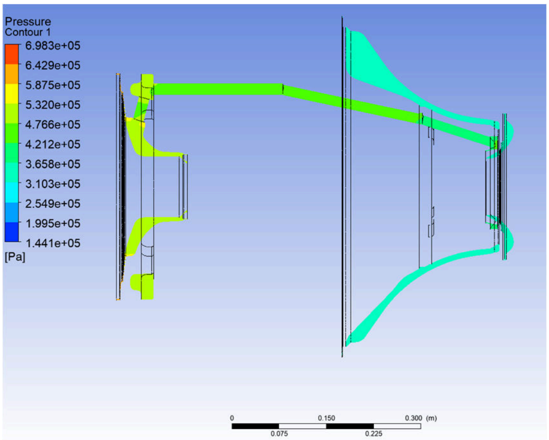


Figure 5: Pressure Contour Plot, Section View

2.2.2 Analytical Piston Ring Model

The mechanical behaviour of the piston ring seals is modelled by an analytical approach. Acting gas and clamping forces are worked out based on piston ring theory available in the engine industry like outlined for example in [2]. Figure 6 shows exemplary the free body diagram of the outer TS piston ring. In addition to the axial gas forces acting on the piston ring, radial gas forces acting both on the inner and outer circumference of the piston ring are considered. The gas forces acting on the outer circumference take into account that due to the rough surface topography of the piston ring still a gas pressure can build up on the outer surface. The pressure decreases towards low pressure side of the piston ring is assumed as linear for the radial contact of the piston ring seal. Similar assumption is made for the axial sealing gap shown in Figure 6. With working out the resulting gas forces on the piston ring a safety factor against slipping of the piston ring is defined. It is intended as explained in chapter 2.1 that the clamping force of the piston ring is always higher than the acting resulting gas force.

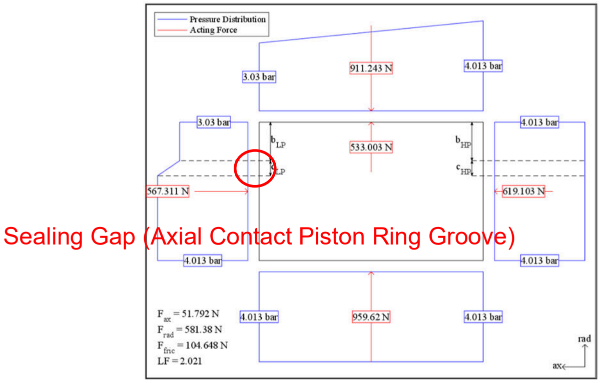


Figure 6: Free Body Diagram Piston Ring

Blow-by is estimated with the Stodola equation [3] as shown in equation 1 considering shown axial sealing gap (see Figure 6):

$$\dot{m}_{BB} = \alpha \cdot \delta^* \cdot \pi \cdot D_1 \cdot \sqrt{\frac{p_{V,T}{}^2 - p_{LAG}^2}{zRT_{BB}}} \tag{1}$$

In this case 2 piston rings have to be considered in the design calculation. This is realized by applying the continuity equation together with the Stodola equation. With this methodology differential pressures can be worked out individually for each of the 2 piston rings. The smallest possible axial sealing gap depends on the piston ring position. Thus, it is essential to also include this effect in the design calculations as the rotor during operation is changing its axial position due to varying thrust forces. This causes a so-called blow-by hysteresis of the sealing system which is shown in Figure 7 for the low pressure, centered and high-pressure position (LP, CENTER, HP).

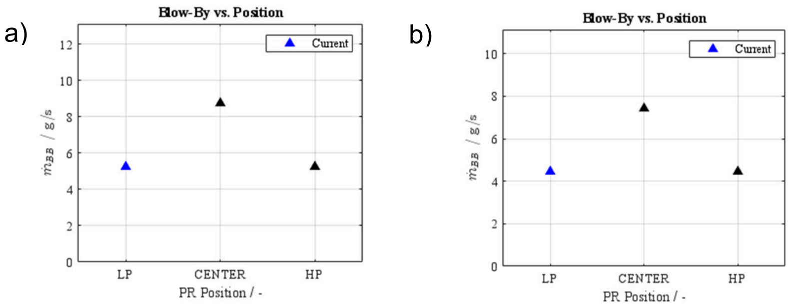


Figure 7: Predicted Blow-By for the a) CS piston rings b) TS piston rings dependent on the position of the piston ring (Blow-By Hysteresis)

According to prediction the max. blow-by magnitude to be expected for the HP turbochargers to be at ~16,2g/s. In addition to the blow-by prediction, a temperature prediction is employed with a radial disc model. For piston ring seals the main heat influx stems from the hot blow-by gas flowing through the sealing gap. The velocity of the blow-by gas is relatively high as the cross-section area of the sealing gap is very small compared to the rest of the flow dimensions. Owed to the high flow velocity the Reynolds number and consequently the heat transfer coefficient of the blow-by gas is therefore from significant magnitude. The average heat transfer for the piston ring surfaces exposed to hot blow-by gas is derived from an analytical correlation derived from experiments with 2 concentric cylinders for which one is rotating [4]. With the correlation outlined in [4] and with the predicted blow-by mass flow from equation 1 subsequently a heat transfer coefficient for the piston ring can be estimated. With a simple thermal resistance model (see Figure 8 a)) the piston ring material temperature can be estimated. Cooling effects of the lubrication oil inside the bearing casing are considered by employing empirical determined heat transfer coefficients and typical oil temperatures. Figure 8 b) shows an example of the predicted radial temperature distribution, showing its maximum at the piston ring seal.

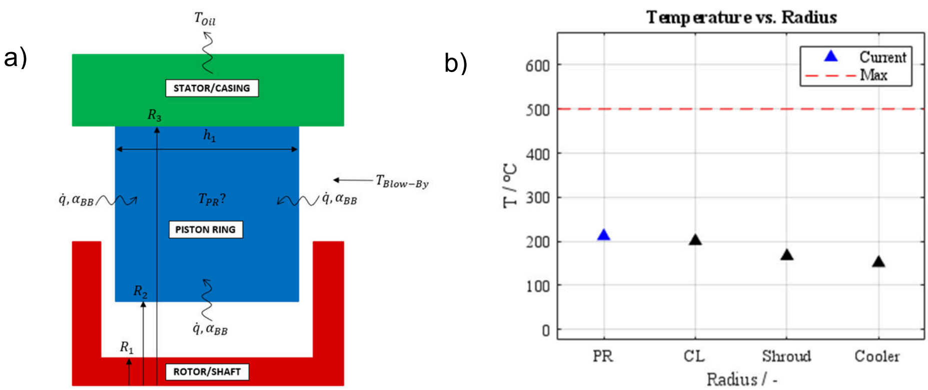


Figure 8: Temperature Prediction a) Thermal Resistance Model b) Example Temperature Distribution

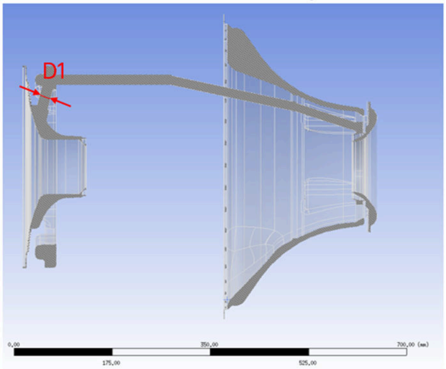
Splash oil present inside the bearing casing contributes to the cooling of the sealing region in addition to heat conducted through the casing itself. Piston ring temperatures are an important parameter as, when kept withing certain limits, the coking risk on the turbine side seal can be controlled on an acceptable level. Severe degree of coking affects sealing performance as contaminants like coked oil in the sealing area accelerate wear and can even lead to sealing failures.

3 Results

3.1 Parameter Study

With the CFD model a parameter study was conducted to optimize the performance of the sealing air system. For the optimization of the sealing air system the diameter of the sealing air pipes shall be modified. With a simplified approach this is only realized by altering the diameter D1 as shown in Figure 9 a) in the simulation model.

a)



b)

D1 [mm]	Cross Section Area [%]
20	100
17	72
14	49
10	25

Figure 9: Parameter Study Sealing Air Channel a) schematic b) chosen parameters

4 different diameters and thus cross section areas have been chosen to assess the effect on the sealing system performance. Relevant parameters for benchmarking the performance of the sealing air system are sealing air mass flow, blow-by temperature (TS seal) and subsequently the sealing temperatures. In addition, the differential pressures across the seals have been evaluated for the investigated cross section areas. As apparent in Figure 10 the 2 smallest cross section areas decrease the sealing air mass flow significantly which is beneficial for the volumetric efficiency but in contrary the blow-by and predicted sealing temperatures rise significantly reducing the seals robustness against oil coking and piston ring relaxation. In addition, with the cross-section areas of 25% and 49% differential pressure for the CS seal would be significantly increased compared to higher cross section areas. Adverse effects are noticed for the TS seal.

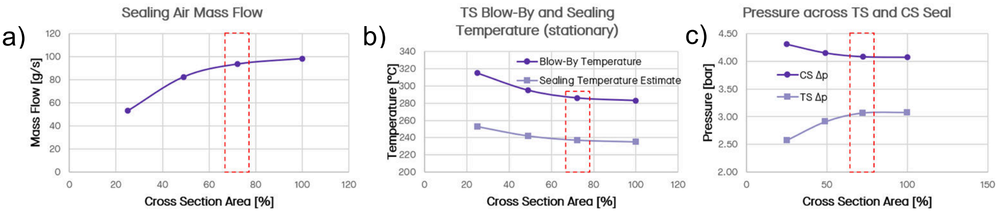


Figure 10: Results Parameter Study: Influence of cross section on a) Sealing Air Mass Flow b) Blow-By and Sealing Temperatures for TS seal c) Differential Pressure CS and TS Seal

In addition to bespoke sealing parameters, the velocity in the ring channel on turbine side has been evaluated (Figure 11), as higher velocities in the ring channel are considered to offer better protection from exhaust gas contamination. Significantly lower velocities into and in the ring channel are observed for 25% and 49% cross section areas.

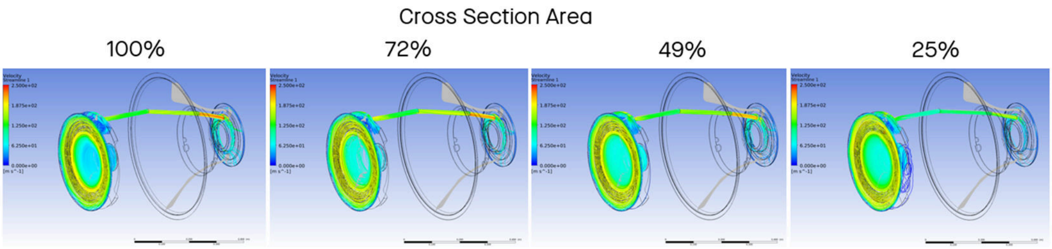


Figure 11: Result Parameter Study: Influence of cross section area on velocity profile in the sealing air system

Thus, therefore a reduction to 72% of the original cross section area size has been chosen to be applied for the sealing air channels connecting compressor and turbine side.

3.2 *Piston Ring Design Monte Carlo Analysis*

For the Monte Carlo analysis 22 geometrical dimensions of the piston ring sealing pack are considered using their production tolerances. In addition, the friction coefficient between the piston ring circumference and seat is considered using a reasonable, partly experimentally confirmed range. The distribution of the relevant parameters outlined above are assumed to be normally distributed for the Monte Carlo analysis within a +/-3σ range for the tolerance band. Besides clamping forces of the piston ring for the varying coefficient of friction, stresses/surface pressures during mounting and in operation as well as operational clearances are assessed. Boundary conditions for the piston ring design have been chosen as derived in chapter 3.1 for the 72% cross section area case. The result of the assessment is shown in Figure 12.

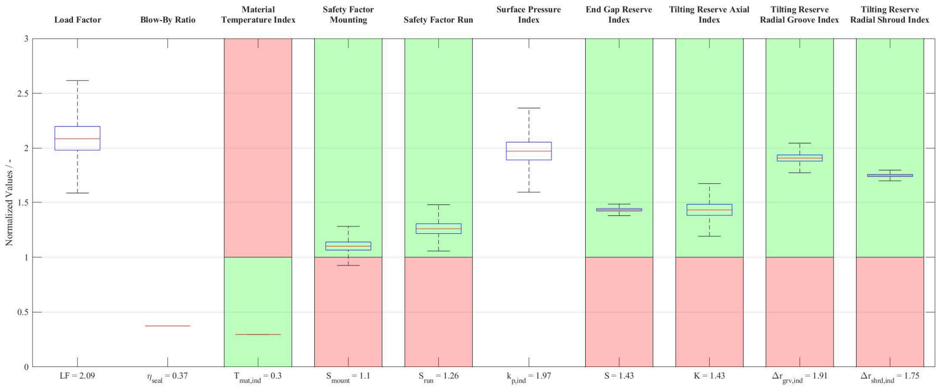


Figure 12: Result Monte Carlo Analysis: Evaluation of Sealing Parameters

Especially for the load factor with this approach, a more robust design could be created. The load factor defines the safety of the piston ring against slipping to acting gas forces. The assessment of the statistical minimal load factors is shown in Figure 13, where the load factors of the original piston ring design have been modified with only slight geometrical modifications like increase of ring thickness, jaw width etc.

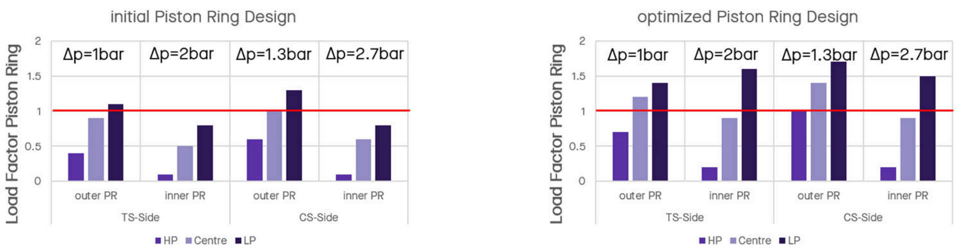


Figure 13: Optimization of Piston Ring Design: Increase of statistical min. load factor for a) initial piston ring design b) optimized piston ring design

The final design was laid out so that the min. statistically possible load factor lies at 1.5 for the LP piston ring position without increasing the overall size of the sealing system.

3.3 Comparison to Measurements

The last 2 chapters outlined the design made with described digital approach. In the final stage of the sealing development for the high-pressure stage validation tests have been conducted on a 2-stage turbocharger test rig. The 2-stage setup is required to mimic the boundary conditions occurring in the engine. In the validation campaign the HP turbocharger was extensively instrumented to measure pressure differential, blow-by and sealing temperatures and to assess the sealing air consumption. These measured boundary conditions have been compared with the predictions made in the design phase. Conditions measured represent a full load point on the engine. Table 1 summarises and compares the differential pressure of compressor and turbine side seal and compares with the measured values in the validation campaign. Sealing differential pressures are slightly overestimated by the CFD model but still within an acceptable range of ~6%. Furthermore, the split of differential pressures between both piston rings can be predicted within fair accuracy.

Table 1: Comparison of Measured differential pressures across the seal with the CFD prediction

	Δp_CS [bar]	Δp_CS _outer-PR [bar]	Δp_CS _inner-PR [bar]	Δp_TS [bar]	Δp_TS _inner-PR [bar]	Δp_TS _outer-PR [bar]
Measured	3.84	1.34	2.5	2.89	2.0	0.89
Predicted (CFD)	4.08	1.29	2.79	3.07	2.07	1.0
Δ prediction - measured [%]	+6.3	-3.7	+11.6	+6.2	+3.5	+12.4

Table 2 compares the predicted and real measured sealing performance which includes sealing air mass flow, total blow-by and sealing temperatures of the turbine side seal.

Table 2: Comparison of Sealing Performance with the Prediction

	Sealing Air Mass Flow [g/s]	Blow-By Mass FLow [bar]	Sealing Temperatures [ºC]
Measured	88	17.4	208
Predicted (CFD)	94	16.2	237
Δ prediction - measured [%]	+6.8	-6.9	+14.0

It is apparent that blow-by is slightly underpredicted whereas sealing temperature and sealing air mass flow are slightly overpredicted. The overprediction of the sealing temperatures are related to the adiabatic wall setup of the CFD model for the sealing air system. In reality the sealing air guided from the compressor towards the turbine side is cooled down in the sealing air channel crossing the bearing casing. Reason for this is the significantly lower bearing casing temperature owed to the lubrication oil cooling the bearing casing internally.

4 Summary and Conclusion

A combined computational and analytical approach was presented predicting the sealing performance of a HP turbocharger. The single phase CFD model was able to model the boundary conditions for a typical engine full load point with sufficient accuracy. Sealing air and blow-by mass flow match well the measured values in the validation tests. With predicted boundary conditions a fast and efficient experimental sealing validation could be successfully performed. The sealing performance of the sealing system matched the requirements as anticipated by the digital development approach.

5 Nomenclature

Variable	*Description*	*Unit*
$\dot{m}_{BB}$	Blow-By Mass Flow	[kg/s]
α	Form Factor	[-]
δ^*	Sealing gap width	[m]
D_1	Piston Ring Nominal Diameter	[m]
$p_{V,T}$	Pressure before Seal	[Pa]
p_{LAG}	Pressure before Seal	[Pa]
z	Number of sealing gaps	[-]
R	Ideal Gas Constant	[J/(mol*K)]
T_{BB}	Blow-By Temperature	[K]
T_{Oil}	Oil Temperature	[°C]

6 References

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7 Authors

Accelleron Industries

Bruggerstrasse 71A

5400 Baden

Dr. Björn Kunzelmann, ORCID 0009-0007-5327-3327
bjoern.kunzelmann@accelleron-industries.com

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