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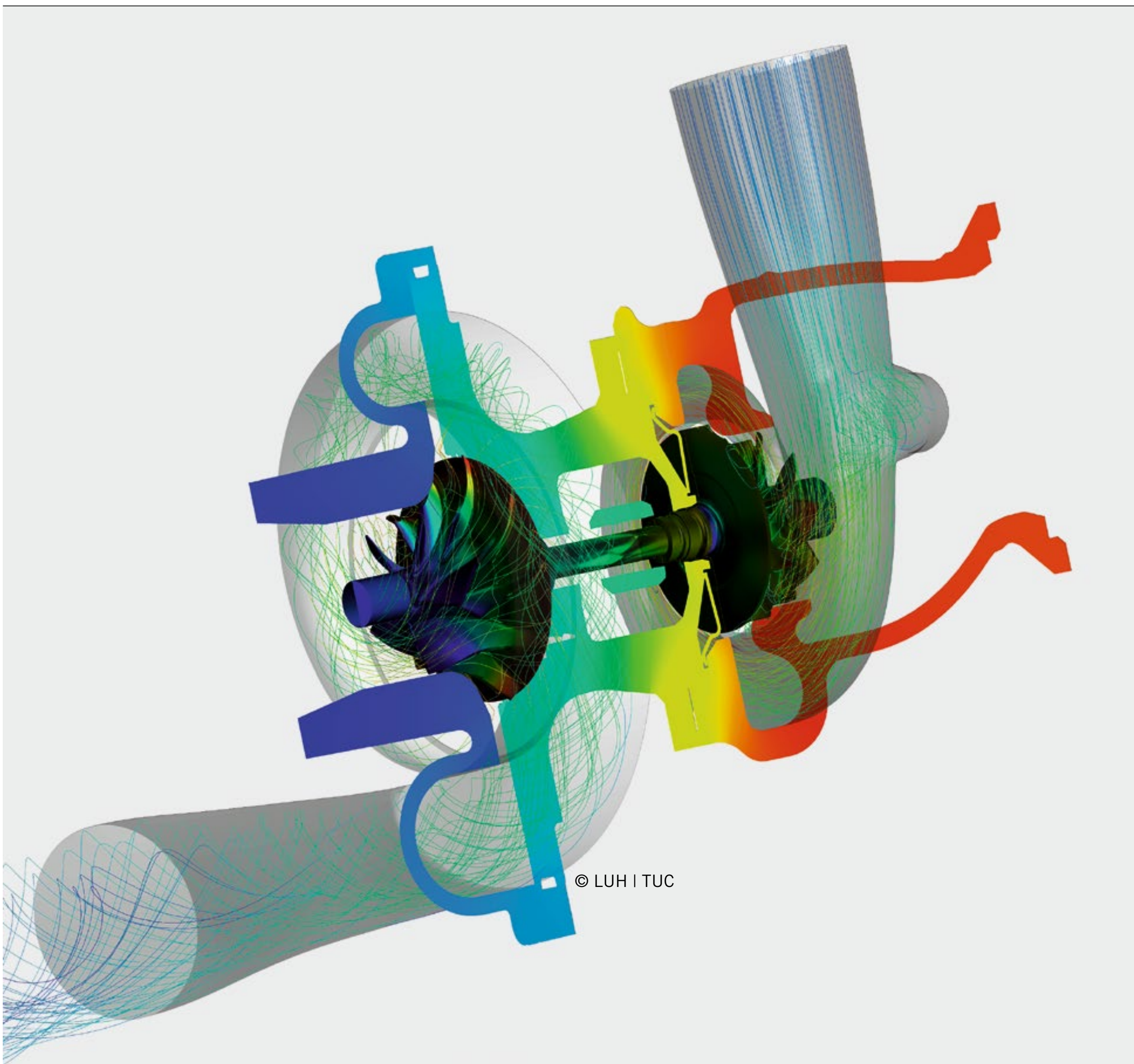
Thermal Influence on the Overall TC System with Consideration of the Coupled Bearings

Model-based calculation tools are for early design evaluation. They can only be used in a meaningful way if thermodynamic processes can be represented realistically, for which a sufficiently deep understanding is required. For this reason, the processes at the plain bearings and the heat flow mechanisms within a passenger car turbocharger were investigated within the framework of the FVV project "Thermally influenced TC bearing friction" (FVV No. 1238) at the Leibniz University Hannover (LUH) and the Clausthal University of Technology (TUC).



1 MOTIVATION

Turbocharging is a key technology in the development of efficient combustion engines. In order to avoid time-consuming and cost-intensive experimental investigations and to match turbocharging system and engine, calculation tools are needed to map the real turbocharger's (TC) operating behavior. In particular, more precise understanding of thermo-physical processes is required for the modeling of plain bearings, since the bearing power loss strongly affects the overall TC system.

In order to gain an understanding of the complex hydrodynamic processes in the TC and their interaction with the bearing locations, sensitivity analyses based on experimental and numerical investigations were carried out at the Institute of Turbomachinery and Fluid Dynamics (TFD) of the Leibniz University Hannover (LUH) and the Institute of Tribology and Friction (ITR) of the Clausthal University of Technology (TUC).

2 INTRODUCTION

In combination with downsizing and downspeeding, turbocharging has become established in the majority of gasoline and diesel engines for commercial and passenger vehicles. In the development and production of efficient exhaust gas TCs and their individual components, the aim is to achieve the useful output by means of higher mean pressure, improved dynamic behavior, lower specific fuel consumption and compliance with statutory emission limits. However, an efficient design evaluation is only possible by mapping the real operating behavior through sufficient understanding of the thermo-physical processes.

In passenger car TCs, usually hydrodynamic thrust and journal bearings with coupled lubricating films and small size are used. The acquisition of the bearing characteristics during operation can only be realized experimentally with great technical effort. Therefore, simplified assumptions or experimental boundary conditions are chosen for the determination of these values [1, 2]. Thermo-hydrodynamic lubrication codes based on the Reynolds equation are used to predict the operating behavior in a time-efficient manner [3]. Individual investigations on thrust and journal bearings show that the determination of the thermal boundary conditions – in particular the shaft and thrust collar temperature – is of decisive importance in order to realistically model the operating behavior of the plain bearings [4, 5]. These boundary conditions are, however, difficult to determine experimentally in TCs of small size, so that due to the complex and coupled hydro-thermodynamic processes there are still considerable uncertainties in the calculation of the essential bearing characteristics.

3 METHODOLOGY

For the sensitivity analyses, a comprehensive instrumentation concept was implemented – in addition to the standard measuring setup for global balancing – which includes more than 60 tem-

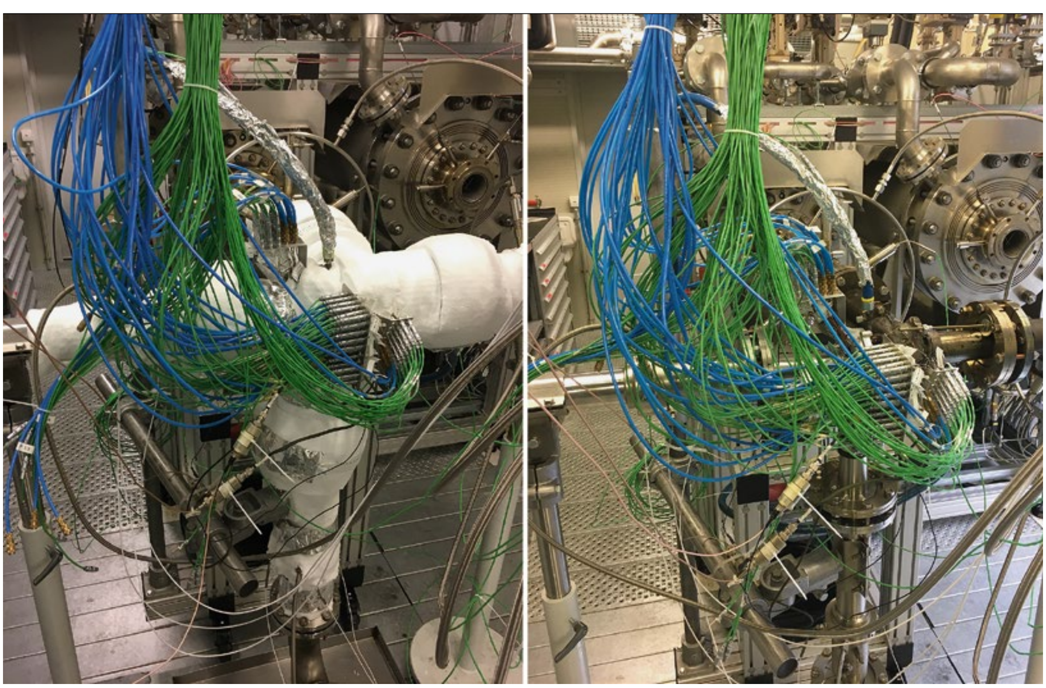


FIGURE 1 Iisolated (left) and non-isolated (right) TC at the TFD test bench (© LUH)

perature and 25 pressure measuring points in the fluid and solid state components including the bearings. The instrumentation was carried out under the aspect of correctly defining the relevant interfaces for the numerical model with regards to the thermal state and to directly determine unknown boundary conditions. In order to minimize the influences of heat exchange by free convection and radiation with the environment, the TC and the inlet and outlet pipes were thermally insulated with high temperature resistant material, **FIGURE 1**. A numerical Conjugate Heat Transfer (CHT) model of the TC – consisting of fluid and solid-state segments – was built and validated on the basis of the experimental data. The thermal boundary conditions were determined by a bidirectional, iterative coupling of the thermo-hydrodynamic lubrication codes (Combros-A and Flobucom) with the CHT model considering the bearing dissipation in the overall heat balance. Since the solid state temperatures have a significant influence on the hydrodynamic processes in the bearing lubrication gaps, plain bearing models were implemented. Thus, the relevant physical effects as well as the energetic, hydraulic and mechanical coupling between the individual fluid films (journal bearings) and non-periodic thermal boundary conditions (thrust bearing) are taken into account. This ensures an optimal detailed bidirectional coupling with the numerical CHT model of the TC [6, 7].

4 MODEL VALIDATION

The validation of the CHT model was performed for both fluid and solid-state regions. The present validation is limited exemplarily to some project-specific Operating Points (OPs), **FIGURE 2**. Due to the used inlet and outlet boundary conditions, the fluid temperature at the compressor and turbine outlet serves as an integral value for the validation of the fluid domain. For validation of the solid domains, axially distributed housing temperature measuring points are used. The deviation from experimentally to numerically determined temperatures amounts to a maximum of 10 K (fluid domain) or 14 K (solid state temperature) for the project-specific reference OP 2. The thermal boundary conditions of the thrust bearing result directly from the coupled analysis of CHT and thermo-hydrodynamic lubrication codes. Therefore, the numerical results do not depend on an estimation of realistic boundary conditions as would be the case with an isolated bearing analysis. The maximum deviation of the thrust bearing temperatures for OP 2 is 5 K, **FIGURE 3**. The indirect validation of the rotor and thrust collar temperature distribution is carried out via the floating ring speed, which is significantly influenced by the rotor temperature, and the TC power loss [6]. On the one hand, the ratio of floating ring speed to rotor speed is formed. The absolute deviation from experimental to numerical results is 0.03 at most, **FIGURE 4**. On the other hand, the oil enthalpy difference is used to determine the TC power loss. The respective contributions to the total power dissipation vary in the investigated range between 55/45 % to 75/25 % and show a maximum percentage deviation of approximately 13 % [6, 7].

5 ANALYSIS OF THE THERMAL INFLUENCE

The established bidirectional coupling between the CHT model and the thermo-hydrodynamic lubrication codes and the subsequent validation allows the qualitative and quantitative evaluation of the heat flow at the interfaces. The analysis of different OPs at higher compressor mass flow rates shows that the surrounding components essentially heat the lubricant. Thus, on the one hand, it serves to compensate for external loads and to limit relative movements of the rotor in the bearings. On the other hand, it acts as a heat sink for the overall TC system and cools the surrounding components. With increasing rotor speed, dissipation within the bearings increases, which in turn leads to a rising lubricant film temperature and thus to a reduced cooling effect. In addition to

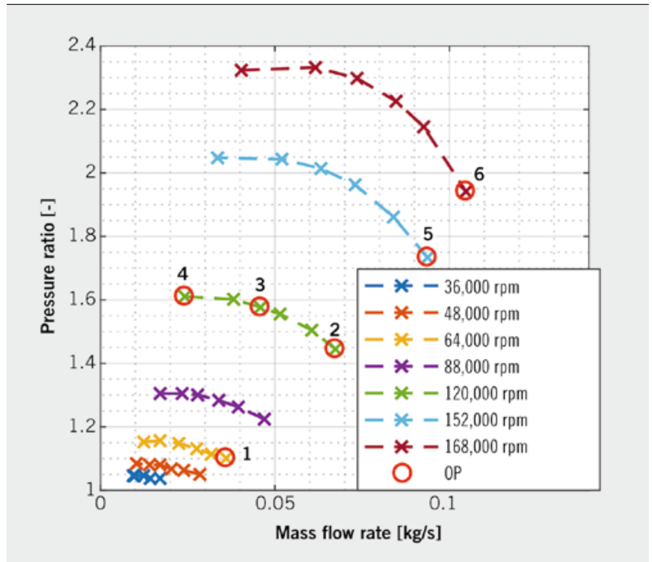


FIGURE 2 Compressor map of the TC with project-specific OP 1–6 [6] (© LUH | TUC)

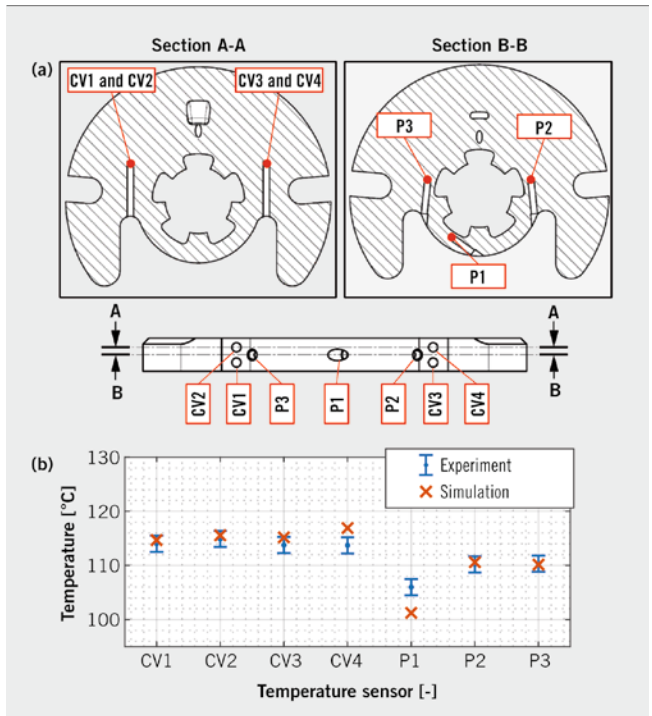


FIGURE 3 Measuring points of the instrumented thrust bearing (top) and experimentally and numerically determined temperatures for OP 2 (bottom) [6] (© LUH | TUC)

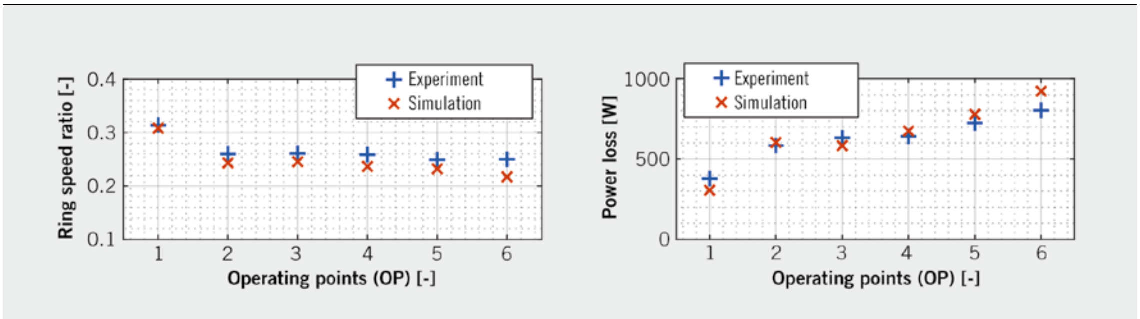


FIGURE 4 Experimentally and numerically determined ring speed ratio of floating ring to rotor of the compressor-sided journal bearing (left) and the TC power loss (right) for OP 1–6 [6] (© LUH | TUC)

the speed dependence, a load dependence of the heat transfer can be proven for the thrust bearing. For the OPs investigated at higher compressor mass flow rates, it applies that on the loaded thrust bearing side, due to the increased dissipation, heat transfer takes place from the lubricant to the surrounding components. On the non-loaded thrust bearing side, heat is transferred from the solids to the fluid. With a balanced axial force and thus an almost concentric thrust bearing to collar, the dissipation is low, so that an increased heat transfer from the surrounding housing components to the lubricant takes place via both bearing sides.

In order to evaluate the influence of the bearing system on the rotor temperature distribution, numerical investigations with adiabatic boundary conditions at the rotor-bearing interfaces were carried out. **FIGURE 5** shows an example of the mean rotor temperature for the diabatic and adiabatic analysis for OP 2. Under adiabatic boundary conditions an approximately linear temperature profile over the rotor shaft is obtained as expected. In contrast, the observation using diabatic boundary conditions shows a significant influence of the heat transfer at the bearing points. For this comparison a maximum temperature difference of the

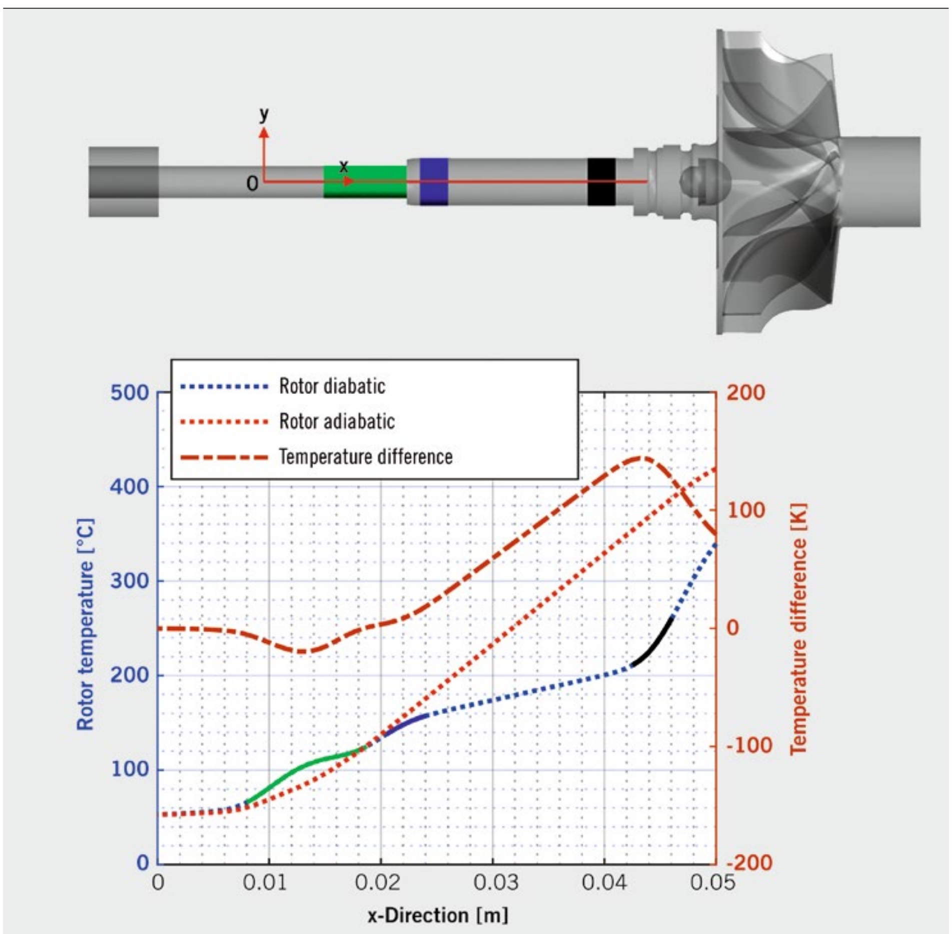


FIGURE 5 Sketch of the temperature distribution at the rotor (top); rotor temperature analysis under diabatic and adiabatic boundary conditions for OP 2 (bottom) [6] (© LUH | TUC)

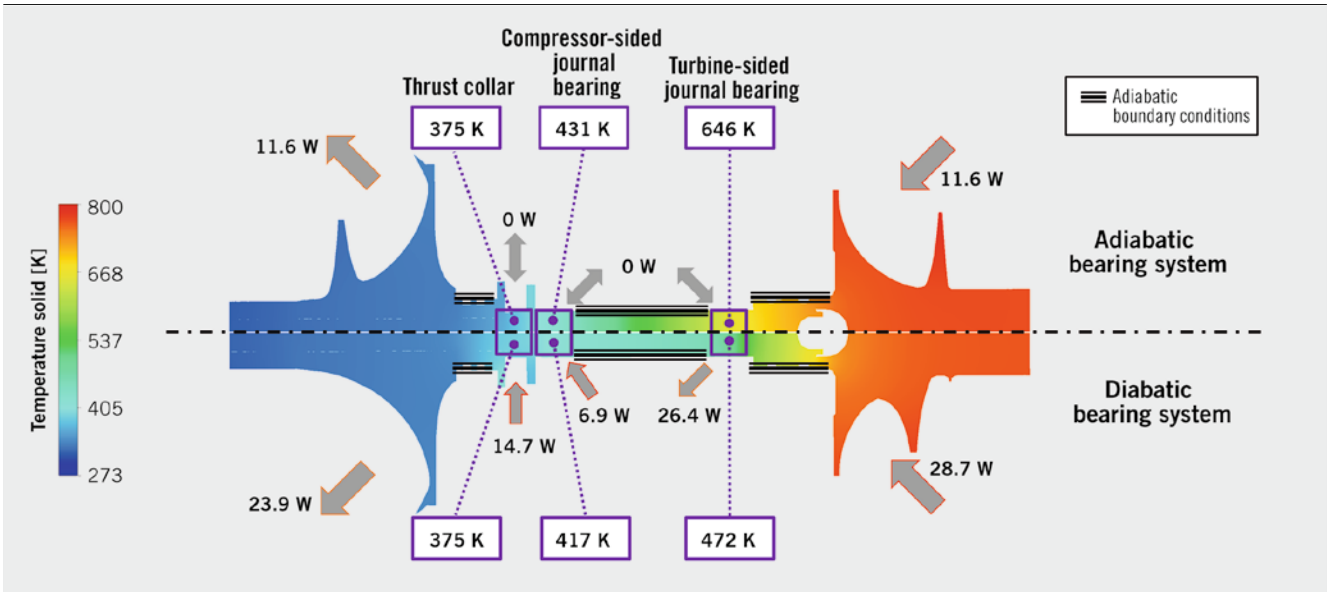


FIGURE 6 Analysis of heat transfer processes under adiabatic (above the axis) and diabatic boundary conditions (below the axis) for OP 2 (© LUH I TUC)

rotor shaft near the turbine radial bearing of approximately 150 K is obtained. This effect is particularly relevant in the area between the axial bearing and the turbine-sided journal bearing. The comparison of both approaches shows positive and negative temperature differences there. The further analysis of the heat transfer processes at the rotor in **FIGURE 6** shows that these change considerably when the bearing influence is taken into account. At the turbine-sided journal bearing a significant amount of heat is dissipated from the rotor through the lubricant film and leaves the system convectively. Furthermore, heat is supplied to the rotor by both the compressor-sided journal bearing and the thrust bearing, since the dissipation in the lubricating films causes corresponding temperature gradients [6, 7].

6 CONCLUSIONS

The high temperature gradients at a hydro-dynamically supported passenger car TC provide complex, locally varying heat flow paths, especially at the bearings, which cannot be predicted in an individual investigation. They can be demonstrated using a diabatic and a bidirectional approach. The thermal influence of the bearing–rotor interaction is considered by the bidirectional coupling of the bearing interfaces. The heat transfer at the interfaces changes depending on the respective operating conditions, the temperature of the system environment and the bearing properties. The varying heat flow paths lead to different temperature profiles of the rotor. Taking into account the influence of the bearings, the rotor temperature changes from an almost linear to a non-linear profile, which leads to high local temperature gradients. In addition, the results show that the different approaches influence the heat transfer between the impellers and the fluid.

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