Ported Shroud Influence on the Aero-Acoustic Properties of Automotive Turbochargers: Quantification by Means of Simulation and Measurement

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Introduction

The development of highly downsized engines is nowadays rather important in order to be able to meet the next emission legislation levels in regards to the future CO2 targets. Downsizing engines refers to turbocharged engines that need to meet the same performance requirements as naturally aspirated larger engines with the same power output and the same torque behavior. Therefore the turbocharger compressor has to operate efficiently over a wide operating range of mass-flow rates, which is limited by the choke and the surge characteristics. The occurrence of the surge phenomenon is difficult to predict, therefore it requires an effort to improve the working range of the centrifugal compressor without compromising the pressure ratio or the compressor efficiency.

There are mainly two methods to increase the width of the compressor map, i.e. using active flow control (e.g. variable vanned diffusers, variable inlet guide vanes [1,2]) or using passive flow control devices (e.g. ported shroud, casing bleed system or internal recirculation or ring groove arrangements) [3, 4].

Passive flow control devices like ported shroud casings are very simple, easy to implement and therefore they are largely used in the centrifugal compressor of turbochargers. With the ported shroud, a part of the low momentum flow blocking the wheel is transferred back to the compressor intake and the blade passage is cleaned from flow disturbances. The ported shroud cavity comes sometimes at the cost of NVH issues, since the recirculation process and turbulent flow field in the cavity can cause acoustical problems. These phenomena were the motivation to acoustically investigate the ported shroud system as a part of the turbocharger [3].

The compressor stage analyzed in this paper is part of a turbocharger compound system with an integrated ported shroud (PS) system to support the functionality of the turbocharger during near surge operation. In order to conduct a thorough investigation two designs (PS, NO PS) are considered in a wide frequency range and within two operating points (one near surge and one near choke line) to assess if and how the ported shroud’s acoustical contribution might be dependent on the selected engine operating points over the frequency range based on the turbocharger’s operation.

Nowadays, with the increasing computer power, it is possible to use high fidelity numerical fluid dynamic techniques, such as large eddy simulations (LES) or detached eddy simulations (DES) to investigate complex engineering problems. The method used in this paper is a Scale Resolving Simulation (SRS) method, with the ability to switch between LES and Reynold Averaged Navier-Stokes (RANS).

In order to evaluate the system acoustically, the computational fluid dynamic (CFD) results are used to calculate the noise sources. The calculation of noise sources is based on acoustic analogies. One of the well-known analogies is the Lighthill analogy [5], which uses the idea to reformulate the Navier Stokes equations to end up with a wave equation

Simulation Method

The aero-acoustic methodology used in this paper consists of two step hybrid approach relying on Möhring’s acoustic analogy [6]. The method assumes the decoupling of noise generation and propagation.

The models are analyzed first using steady state Reynolds Averaged Navier Stokes (RANS) simulations. These results are then used as an initial solution for the unsteady CFD simulations (URANS). The impeller is modelled by the MRF (Moving Reference Frame) method which is based on the assumptions of constant angular velocity and static mesh.

The chosen turbulence modelling is an important CFD parameter for the acoustic evaluation. For the RANS simulation the shear stress transport (SST) k-Omega model is used for its well-known capabilities, but for the URANS calculations a more sophisticated turbulence modelling is used (SAS-SST). This turbulence model is part of the SRS methodology, which dynamically uses RANS and LES for the whole computational domain [7].

In the second step, a modified formulation of Lighthill’s acoustic analogy is discretized by a finite element discretization and solved in the Fourier space. This formulation solves the acoustic wave equation for the radiated noise [8].

This paper investigates two compressor stage designs, which differ from each other only in the aspect of ported shroud. The simulation models and the corresponding boundaries are described in detail in the following.

The boundaries for the CFD setup are chosen based on the testing condition at the gas stand.

Fig. 1 shows a view of the 3D CFD and acoustic model. For the purpose of better CFD convergence inlet and outlet side of the compressor were extended.

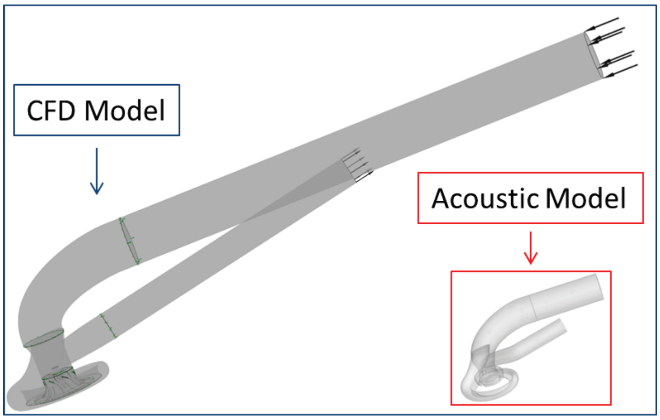

Figure 1: Example of the CFD and acoustic 3D model being investigated within simulations

The geometrical differences between the cases are visible in Fig. 2.

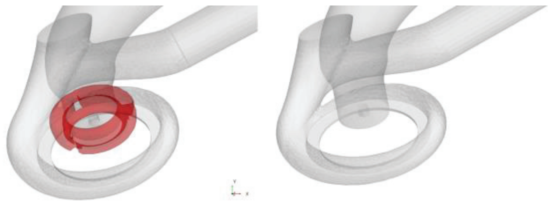

Figure 2: Cases investigated within this paper. The geometrical difference is being highlighted

Fig. 3 shows the compressor map of both designs. Here is the advantage of using a ported shroud from performance perspective obvious.

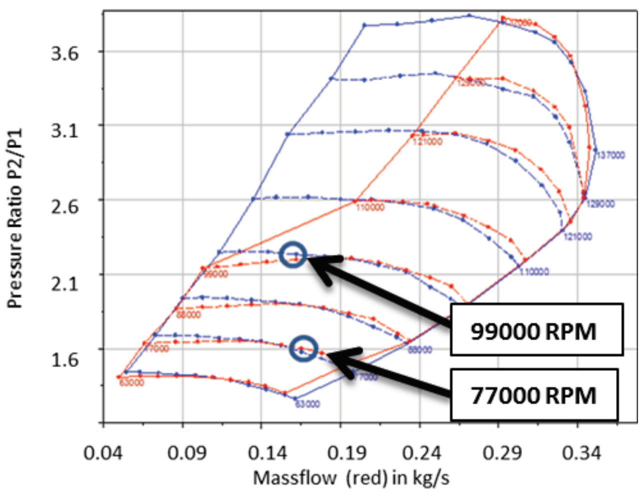

Figure 3: compressor map of two selected designs (red: NO PS, blue: PS) and selected operating points for the acoustic analysis

Flow field results of the first operating point

The following figures depict the flow behavior within different sections of the compressor stage. For this purpose different cut planes are set into the model, to get a deeper look into the flow behavior. Special focus is here set at the ported shrouds.

The first set of figures (Fig. 4) visualize the evolution and distribution of the Turbulence Kinetic Energy (TKE) for both variants within different cuts in the ported shroud section. Turbulence kinetic energy is within the CFD variables a good indication for potential induced turbulence and in-stationarities. If the disturbances visible in the flow field do have a contribution to the noise emitted, should be analyzed with the help of the acoustic analogy.

Top of the ported shroud shows only slight difference in the TKE values between the two variants. The flow behavior for the case PS becomes more non uniform after the flow reaches the more turbulent area of the ported shroud (middle of the PS). Noticeable change in the TKE is visible at the area of impeller eye. PS carries less turbulent energy at impeller eye than NO PS. This difference is mainly due to break up of big turbulent structures in this region (dependent on the selected operating point).

Fig. 5 demonstrates the TKE values for the vertical plane through the PS section and impeller until volute outlet. The differences aren't noticeable in this region.

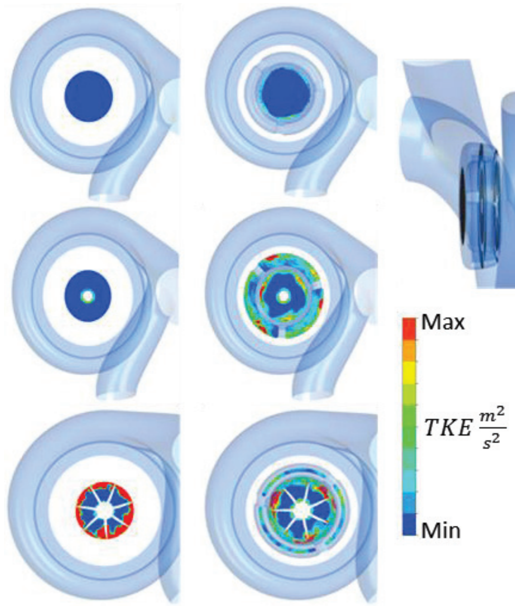

Figure 4: Instantaneous turbulent kinetic energy on different cut planes throughout the compressor stage for the first chosen operating point. Left NO PS, Right PS.

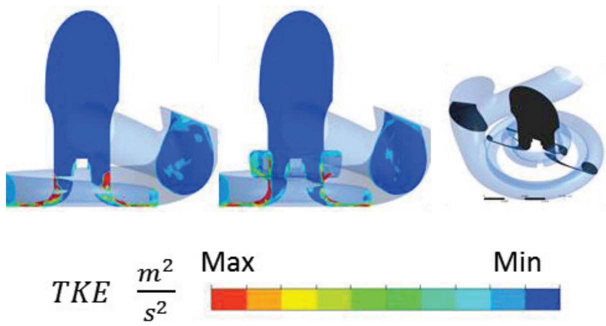

Figure 5: Instantaneous turbulent kinetic energy on vertical cut plane throughout the compressor stage for the first chosen operating point. Left NO PS, Right PS.

Flow field results of the second operating point

Fig. 6 is the depiction of TKE values across the ported shroud for the second chosen operating point. NO PS clearly has higher TKE values across the ported shroud. These differences are consistently visible throughout the whole section. The same difference in TKE distribution is also retractable from Fig. 7 throughout the ported shroud and volute. Here is the distinction between the inlet and outlet side of the compressor stage of special attention (see Fig. 7).

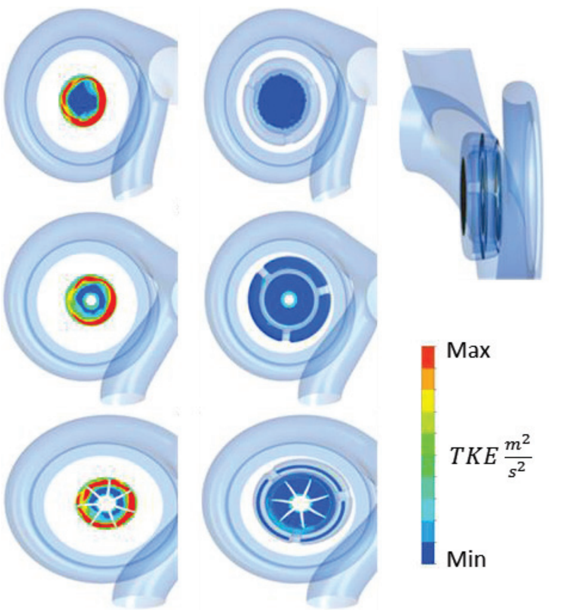

Figure 6: Instantaneous turbulent kinetic energy on different cut planes throughout the compressor stage for the second chosen operating point. Left NO PS, Right PS.

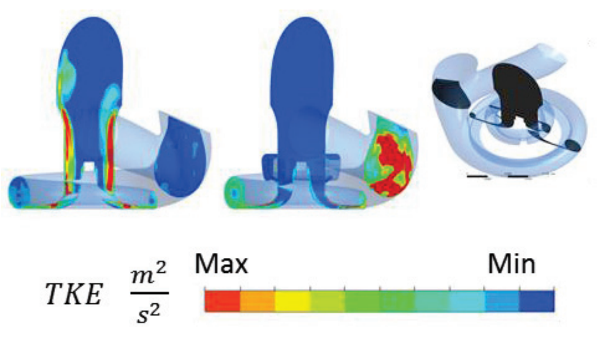

Figure 7: Instantaneous turbulent kinetic energy on vertical cut plane throughout the compressor stage for the second chosen operating point. Left NO PS, Right PS.

Acoustic Results

Near field sound pressure spectrum

One good way to analyze the acoustic results quantitatively, is to compare the sound pressure results at selected virtual microphones over frequency spectrum. This way, the broadband and tonal behavior of the system can be monitored much better, and consequently specific phenomena can be traced back. This method corresponds well to the wall bounded dynamic pressure sensor technique within measurements. The virtual microphones were placed both at suction and pressure side of the compressor stage.

Acoustic results of the first operating point

Comparison between sound pressure level (SPL) of both variants for the first operating point on the outlet surface is visible in Fig. 8. The outlet surface chosen for the microphone position is also depicted below. NO PS case shows at almost all frequencies higher SPL values, but the gap between the SPL level of the two variants is not dominant.

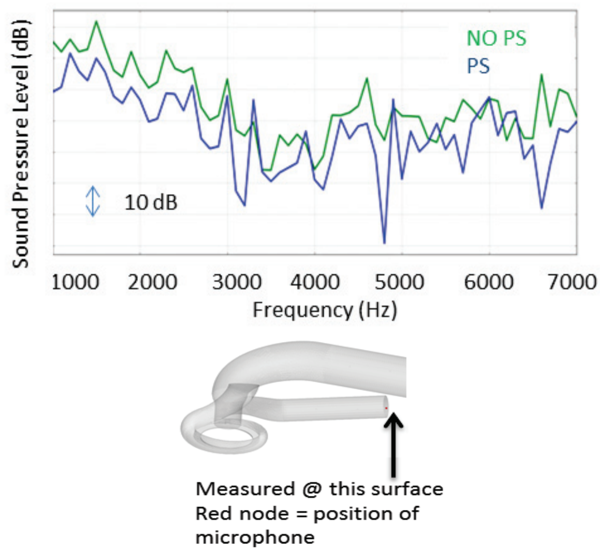

Figure 8: Sound Pressure Level spectrum at outlet surface for both cases. First operating point chosen

The corresponding experimental results for the two cases and selected operating point are presented in Fig. 9. NO PS was created by filling up the ported shroud cavity. There is a good agreement between the experimental and simulated results. NO PS has higher values than PS within this frequency range.

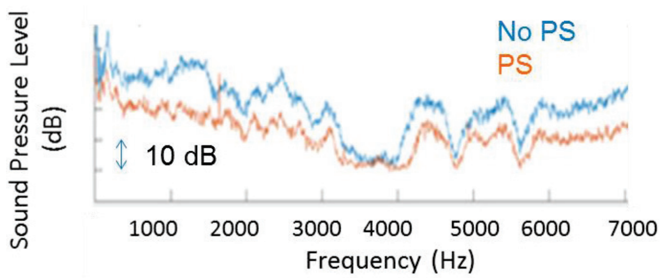

Figure 9: Experimental Sound Pressure Level spectrum for both cases at outlet. First operating point chosen

Acoustic results of the second operating point

In this section the comparison between sound pressure level (SPL) of both variants are presented for the second operating point.

Fig. 11 shows the frequency spectrum of the SPL levels for the virtual microphone positioned on the outlet region. Here there is a change in the behavior of SPL levels compared to the first operating point (see Fig. 8). Ported shroud has higher SPL values in the mid-range frequency (2-5 kHz). The influence of the chosen operating point on the acoustical behavior of the system is visible in this figure. The broadband phenomena visible at the outlet of the compressor stage are stronger in the PS case for the operating point in the vicinity of choke. The SPL value gap between the two variants is not noticeable.
Fig. 11 confirms that the simulation results reproduce the measurement`s tendency pretty good.

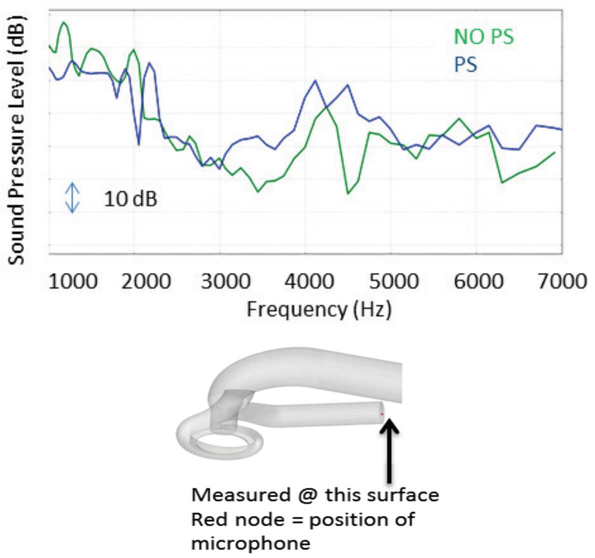

Figure 20: Sound Pressure Level spectrum at outlet surface for both cases. Second operating point chosen

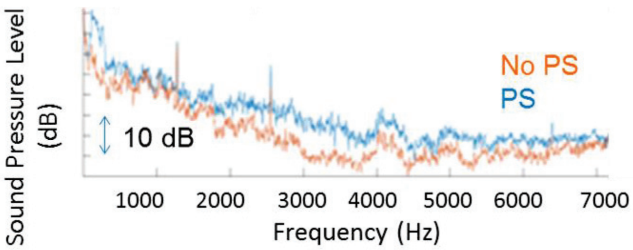

Figure 31: Experimental Sound Pressure Level spectrum for both cases at outlet. Second operating point chosen

Conclusion

The influence of the ported shroud on the acoustic response of the system, in this case the compressor stage of a turbocharger, have been investigated. The acoustic analysis was performed using a hybrid CAA method, based on acoustic analogies. For this purpose two compressor stage designs were chosen. The major difference between these two cases is that the original compressor stage contains a ported shroud, but in the second design the ported shroud was removed from the modelling. In order to quantify the acoustical behavior of the ported shroud more in detail, two operating points were chosen, one near surge line and the second one more in the region of choke line.

The simulations clarify an interesting dependency of the acoustical behavior of the ported shroud on the chosen operating point. Also the position of the microphone does influence the sound pressure level. This paper on one hand shows the validity of the hybrid aeroacoustics method introduced here. On the other hand it emphasizes on the importance of the acoustical investigation of ported shroud in a turbocharger compressor stage.

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