

The Effect of Ported Shroud Recirculating Casing Treatment on Turbocharger Centrifugal Compressor Acoustics

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ABSTRACT

Ported shroud compressor covers recirculate low momentum air near the inducer blade tips, and the use of these devices has traditionally been confined to extending the low-flow operating region at elevated rotational speeds for compressors on compression-ignition (CI) engines. Implementation of ported shrouds on compressors for spark-ignition (SI) engines has been generally avoided due to operation at pressure ratios below the region where ported shrouds improve low-flow range, the slight efficiency penalty, and the perception of increased noise. The present study provides an experimental investigation of performance and acoustics for a SI engine turbocharger compressor both with a ported shroud and without (baseline). The objective of implementing the ported shroud was to reduce mid-flow range broadband whoosh noise of the baseline compressor over 4-12 kHz. At the compressor inlet, elevated BPF noise of the ported shroud partially offset the mid-flow range (4-12 kHz) whoosh noise suppression and the maximum overall SPL reduction was 6 dB(A), while elevated 4-12 kHz and BPF noise caused an SPL increase of up to 5 dB(A) at low speed and flow. Since the ported shroud did not significantly increase low-flow 4-12 kHz noise at the compressor outlet, overall SPL was reduced over the mid to low flow range by a maximum of 12 dB(A).

CITATION: Dehner, R., Selamet, A., Steiger, M., Miazgowicz, K. et al., "The Effect of Ported Shroud Recirculating Casing Treatment on Turbocharger Centrifugal Compressor Acoustics," *SAE Int. J. Engines* 10(4):2017, doi:10.4271/2017-01-1796.

1. INTRODUCTION

The passive ported shroud design was first introduced by Fisher [1] in 1988 to enhance centrifugal compressor flow range. This device contains a slot in the compressor cover (shroud) which is located over the impeller and provides a secondary flow path (port) connected to the main flow upstream of the impeller. In theory, this design allows for extension of the low-flow operating range by increasing the inducer flow rate, which reduces the incidence angle and mitigates flow separation. In addition, the ported shroud has been utilized to extend the choke flow capacity by allowing additional air to enter the impeller downstream of the aerodynamic throat, when acceleration of the flow has reduced the static pressure relative to that upstream of the impeller. This simple, inexpensive design feature has been widely incorporated into turbocharger compressors for diesel engines, yet the information on its impact on acoustics is rather limited. Ported shroud compressor covers have been generally avoided for SI engine applications, since the compressor typically operates outside of the region where flow range is enhanced, the slight efficiency penalty [2,3], and the perception of increased noise [4,5].

The operation of turbochargers at high speeds and over wide flow ranges makes them susceptible to narrow and broadband noise. Discrete tones are often observed at integer orders n of the shaft speed N at $f_n = n \times N [\text{rpm}] / 60$, due to unbalanced rotating forces ($n=1-3$)

and geometric tolerances of the impeller ($n=1$) [6]. The dominant discrete sound peaks often occur at the synchronous (first rotor-order; $n=1$) and blade-pass (BPF; $n =$ the number of main impeller blades) frequencies. Broadband flow noise, also referred to as "whoosh" or "blow noise," has been observed earlier over a range of 1.5-3.5 kHz [7, 8, 9] (for CI engines) and 5-7 kHz [10] (for SI engines). When this broadband noise was most evident, the engines [7,9,10] were operating at or near full-load and at low- to mid-engine speeds, which places the corresponding compressor operating regime in the vicinity of the surge line or near the peak compressor pressure ratio for the given operating speed. Since these studies were carried out for turbochargers installed on engines, they include engine airborne pulsations within the induction system and the compressor operating range was limited to a fixed (typically full) load engine speed sweep.

Some of the more recent studies have provided detailed compressor acoustic measurements for a turbocharger installed on a steady-flow stand (isolated from engine noise), at both in-duct and external measurement locations. Turbocharger stand acoustic measurements from a centrifugal compressor with a vaneless diffuser, splitter blades, and no casing treatment were presented by Figurella *et al.* [11, 12, 13] and Dehner *et al.* [14]. These studies demonstrated that the most severe compressor inlet duct broadband noise occurred in the expanded 4-12 kHz frequency range, and it was most dominant

within the mid-flow operating region of the compressor. At low flow rates (in the vicinity of the surge line), measurements from the same compressor showed that the overall SPL in the 4-12 kHz band reduces and the compressor inlet duct noise becomes dominated by broadband noise below 3 kHz, but the overall levels were significantly reduced relative to the mid-flow range 4-12 kHz noise. An additional compressor acoustics study [15] with two different ported shroud compressors demonstrated that 4-12 kHz noise was not dominant over any portion of the flow range. Rather, sound elevation from these ported shroud compressors occurred primarily due to narrowband 2-3 kHz and BPF noise. Based on observations and comparisons of these studies [11, 12, 13, 14, 15], it is conceivable that the ported shroud recirculating casing treatment reduces 4-12 kHz whoosh noise in the mid flow range.

In order to characterize the impact of a ported shroud casing treatment on a turbocharger centrifugal compressor, the present work utilizes a steady-flow turbocharger bench to provide detailed acoustic and performance measurements from a new, wide-flow range compressor designed for a twin, parallel turbocharged V6 SI engine. The initial (baseline) compressor design consisted of a straight, constant diameter inlet to the inducer of the impeller. Preliminary testing of this baseline design identified elevated (4-12 kHz) whoosh noise in the mid to low flow operating range, which exceeded that of the previous generation design by up to 14 dB at the compressor inlet duct measurement location. A compressor cover design change, incorporating a ported shroud recirculating passage to reduce whoosh noise, was implemented based on previously unpublished measurements from a CI engine turbocharger compressor with a ported shroud tested both with recirculation passage open and blocked. This earlier data revealed a significant reduction in mid to low flow range whoosh noise with the recirculating passage open, relative to the blocked configuration. Therefore, the present work compares compressor performance and acoustics for a turbocharger compressor with two different compressor covers that are identical except for the presence of a ported shroud recirculating flow passage within one of them. The experimental facility is introduced in [Section 2](#), and the compressor designs are described in [Section 3](#). Time-averaged compressor performance data is provided in [Section 4](#), including flow range and efficiency comparisons. Acoustic measurements are presented in [Section 5](#), with the goal of defining the operating regimes associated with elevated narrow and broadband noise and the corresponding frequency ranges for each configuration. Finally, concluding remarks are provided in [Section 6](#).

2. EXPERIMENTAL FACILITY

The turbocharger laboratory [16] at The Ohio State University Center for Automotive Research (OSU-CAR), shown in [Fig. 1](#), is instrumented to measure compressor performance and acoustics. The turbine is driven by compressed air delivered by a screw compressor, and after expanding through the turbine, it is exhausted outside of the lab. Unlike operation on an engine, the compressor flow circuit is isolated from the turbine side, in order to decouple the mass flow rates and improve the operating range over which the compressor may be examined. Computer controlled valves are installed at the

turbine inlet and compressor outlet in order to adjust the compressor and turbine operating points, and a high-speed data acquisition system records time-resolved measurements.

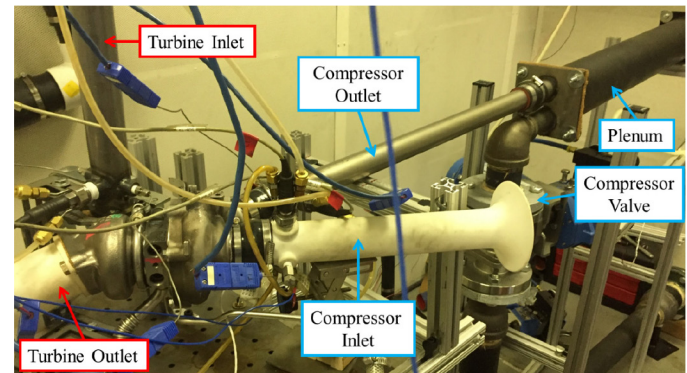


Figure 1. Turbocharger installed on the turbocharger stand at OSU-CAR.

Instrumentation installed in the compressor inlet and outlet ducts is positioned at an axial distance equal to one internal duct diameter from the connection to the compressor, which are 41.7 and 37.0 mm, respectively. Each location contains an absolute pressure transducer [Kistler model 4045A2 (2 bar) at the compressor inlet and 4045A5 (5 bar) at the outlet], piezoelectric dynamic pressure transducer (PCB model 106B), and a pair of (1/16" diameter sheathed type T) thermocouples positioned 90° apart. All pressure transducers are installed flush with the internal surface of the ducts to minimize flow disturbance, and the thermocouples are inserted a radial distance equal to one-third of the corresponding internal duct diameter. Rotational speed is measured with a non-contact inductive sensor (PICOTURN-2G), which detects impeller blade-passes, and mass flow rate with a flow bench calibrated orifice plate flow meter installed downstream of the compressor control valve. The upstream and downstream compression system ducting is identical for both of the tested configurations, where the downstream plenum volume is set to 1.16 L (1/8 of the full volume).

Simultaneous measurements are obtained over a 3.0 s duration after the compression system has reached steady-state operation, which is defined here by calculating time-averaged efficiency over 0.5 s intervals and requiring a change of less than 0.1% for a one minute duration. Absolute pressure transducers are sampled at 2.5 kHz, mass flow rate (two differential pressure transducers and a slow-response thermocouple) and slow-response thermocouples at 100 Hz, and dynamic pressure at 50 kHz. Rotational speed is calculated from the period between consecutive impeller blade-passes (both main and splitters), and the signal is averaged for the number of impeller blades to improve accuracy. Steady-state compressor performance results in [Section 4](#) are calculated utilizing time-averages of experimental data over the 3.0 s sample period.

The lubrication system for the turbocharger bearings incorporates pressure and temperature control of oil entering the center housing, which is maintained at 2.07 bar (gage) and 355 K, respectively. Additionally, the ambient air temperature within the laboratory is

approximately constant throughout all testing to minimize external, convective heat transfer differences between tested configurations, and therefore the calculated efficiencies.

The accuracy of compressor performance measurements from this facility has been evaluated by comparison with test data from four turbocharger manufacturers and an independent testing lab. Overall, the constant speed compressor pressure ratio vs. flow rate performance measurements from this facility provide excellent agreement with those from other facilities. At mid to high rotational speeds, compressor isentropic efficiency data from the present facility compares favorably with data from manufacturers. At low rotational speeds, the efficiency data from the turbo stand at OSU-CAR is higher than that of other stands, due to heat transfer effects. Within this operating region, the air velocity within the compressor is low (residence time for heat transfer is high), the air temperature difference (driving heat transfer) from the turbine to compressor side is maximum, and the temperature rise through the compressor is low. All of these variables contribute to the non-negligible gas stand turbine to compressor heat transfer effects at low rotational speeds, and the resulting efficiency calculation. The “warm flow” facility of the present work incorporates a 15 kW electric heater to elevate the turbine inlet air temperature to approximately 75°C, while the manufacturers’ turbo stands use natural gas burners to elevate the temperature typically to a range of 600-930°C. As a result, heat transfer from the turbine to compressor is less important for the present “warm” turbo stand, and the efficiency calculated from measurements is closer to the isentropic (adiabatic) case. Acoustic measurements from the turbocharger lab at OSU-CAR have been verified to be repeatable to within 1 dB.

3. COMPRESSOR DESIGNS

The turbocharger of this study is an initial prototype for the left-bank of a twin, parallel turbocharged V6 TiVCT GTDI engine, where the right-bank turbocharger is essentially a mirror image. The compressor is designed for a wide flow range, which features a vaneless diffuser, a 6+6 (main+splitter) blade impeller with inducer and exducer tip diameters of 39.1 and 50.2 mm, respectively. In order to accurately quantify the impact of a ported shroud recirculating flow passage on compressor performance and acoustics, separate compressor covers were manufactured for the baseline (BL, without recirculating casing treatment) and ported shroud (PS), shown as black and brown, respectively in Fig. 2. Therefore, all aspects of the compressor design and internal geometry are identical for the baseline and ported shroud configurations, with the exception of the recirculating casing treatment to isolate its impact on performance and acoustics.

The center of the upstream slot of the ported shroud is located an axial distance of 24.3 mm upstream of the leading edge of the impeller main blade tips, and the center of the downstream slot is located an axial distance of 2.9 mm from the leading edge of the impeller main blade tips. The inner annulus of the ported shroud, located between the upstream and downstream slots, is supported by three ribs that pass radially through the cavity. The angular spacing between these ribs is nonuniform at 110, 120, and 130°.

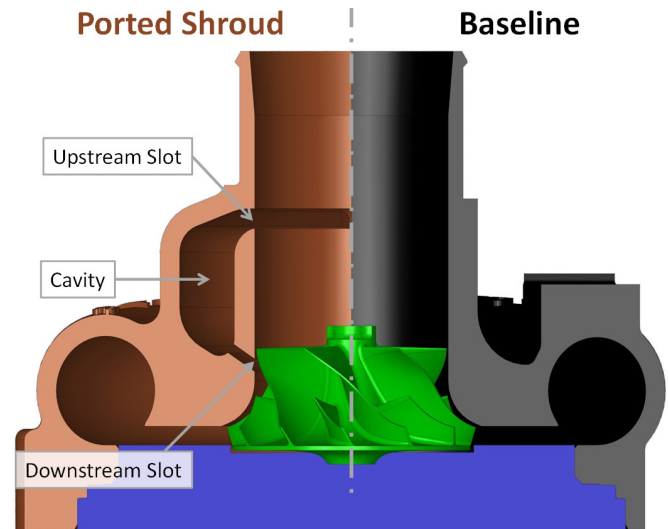


Figure 2. Cross-section comparison of the ported shroud and baseline compressor covers.

4. COMPRESSOR PERFORMANCE

The performance of the compressor with and without the ported shroud is presented as compressor total-to-total pressure ratio $PR_{c,tt}$ vs. corrected mass flow rate $\dot{m}_{c,cor}$ at seven corrected rotational speeds N_{cor} in Fig. 3. The six lowest N_{cor} (80, 100, 115, 126, 142, and 154 krpm) contain data over the full choke (minimum valve restriction) to the last mild surge operating point, and the highest N_{cor} (169 krpm) data only over the low-flow operating range. Therefore, the low-flow boundary of the compressor performance data in Fig. 3 borders the deep surge instabilities (for the studied compression system), which will be referred to simply as the “surge line” in the present work. At elevated flow rates, the performance of the baseline and ported shroud covers is nearly identical at all fully tested rotational speeds. Thus, this combination of casing treatment and impeller design does not modify the choke flow rate of the compressor. As the compressor flow rate is reduced to the mid-flow range, the $PR_{c,tt}$ of the ported shroud configuration begins to deviate from the baseline design at identical N_{cor} and $\dot{m}_{c,cor}$. This $PR_{c,tt}$ deviation increases with further decreasing $\dot{m}_{c,cor}$ until it becomes roughly constant over a significant flow range, before beginning to diminish near the surge boundary. The increased mid-flow pressure rise capability of the ported shroud compressor cover signifies that the inducer flow field is substantially improved by increasing the flow rate (decreasing incidence) near the leading edge blade tips, but the low-flow range is slightly reduced with the ported shroud.

The corresponding total-to-total isentropic efficiency $\eta_{c,tt}$ for the baseline configuration is shown in Fig. 4, where the constant N_{cor} data from adjacent rotational speeds is interpolated to intermediate speeds for this contour plot and all that follow throughout the present work.

The efficiency comparison of the baseline and ported shroud compressor covers is shown in Fig. 5, within the region where the $PR_{c,tt}$ vs. $\dot{m}_{c,cor}$ operating envelopes overlap. The contour quantity in Fig. 5 is the total-to-total isentropic efficiency difference $\Delta\eta_{c,tt} = (\eta_{c,tt})_{BL} - (\eta_{c,tt})_{PS}$ [baseline (BL) minus ported shroud (PS)] between the two configurations. Therefore, $\Delta\eta_{c,tt} > 0$ indicates that the baseline configuration is more efficient, which is the case at particularly low flow rates.