

Instabilities at the Low-Flow Range of a Turbocharger Compressor

Rick Dehner, Neil Figurella and Ahmet Selamet
 The Ohio State University

Philip Keller and Michael Becker
 BorgWarner Inc.

Kevin Tallio, Keith Miazgowicz and Robert Wade
 Ford Motor Co.

ABSTRACT

The acoustic and performance characteristics of an automotive centrifugal compressor are studied on a steady-flow turbocharger test bench, with the goal of advancing the current understanding of compression system instabilities at the low-flow range. Two different ducting configurations were utilized downstream of the compressor, one with a well-defined plenum (large volume) and the other with minimized (small) volume of compressed air. The present study measured time-resolved oscillations of in-duct and external pressure, along with rotational speed. An orifice flow meter was incorporated to obtain time-averaged mass flow rate. In addition, fast-response thermocouples captured temperature fluctuations in the compressor inlet and exit ducts along with a location near the inducer tips. At low flow rates, both the large and small volume systems exhibited mild surge as the slope of the compressor characteristics became positive, but the small volume configuration impeded the growth of instabilities and was able to operate without deep surge at mass flow rates well below that of the large volume system. The fundamental mild surge frequency is shown to be dominated by the geometry of the upstream and downstream ducting along with that of the compressor, with in-duct pressure measurements revealing frequencies well below 100 Hz for the configurations studied. For the large volume compression system, the fundamental surge frequency is observed to occur near the Helmholtz resonance of the system during mild surge and decreases with deep surge.

CITATION: Dehner, R., Figurella, N., Selamet, A., Keller, P. et al., "Instabilities at the Low-Flow Range of a Turbocharger Compressor," *SAE Int. J. Engines* 6(2):2013, doi:10.4271/2013-01-1886.

1. INTRODUCTION

The low-flow instability of axial and centrifugal compressors has long been a source of great concern. Early research in this field has primarily concentrated on measuring and characterizing surge and rotating stall in axial compressors for aviation and radial compressors for diesel engines [1,2,3,4]. A clear distinction between compressor stall and surge was first presented by Emmons *et al.* [1] for axial and radial compressors. Greitzer [2] developed a lumped (zero-dimensional) surge model for an axial compression system, which revealed a dimensionless number

$$B = \frac{U_{t,2}}{2a_p} \sqrt{\frac{V_p}{A_c L_c}}, \quad (1)$$

providing a qualitative description of the susceptibility of a compression system to surge, where $U_{t,2}$ is the blade tip speed at the exducer, a_p is the speed of sound in the plenum, V_p is the volume of compressed air, A_c is the equivalent cross-sectional compressor duct area (inducer eye area), and L_c is the equivalent length of the duct. A compression system is most vulnerable to surge instabilities at large values of B , where, for example, the rotational speed and/or the volume of compressed air are high.

A simplified linear stability analysis of Greitzer's model reveals that the existence of surge is a function of both B and the local slope $dPR_{c,ts} / d\dot{m}_c$ of the total-to-static pressure ratio $PR_{c,ts}$ vs. mass flow rate $\dot{m}_c$ characteristic of the compressor at constant rotational speed. At mid- to high-flow rates,

$dPR_{c,ts} / d\dot{m}_c < 0$, and the compressor operates in a stable mode. As the flow rate is reduced, the slope of the characteristic increases and becomes positive due to progressive stalling of the compressor [5]. The limiting case for a system with $B=\infty$ may enter surge when $dPR_{c,ts} / d\dot{m}_c \geq 0$ (at the peak pressure ratio), and a configuration with $B=0$ is unconditionally stable (will not surge). For a physically realizable system with $0 < B < \infty$, the compression system will enter surge on the positively sloped portion of the constant speed line.

Surge is a low-flow compressor instability resulting in self-excited system oscillations which can be categorized as “mild” or “deep” depending on the degree of mass flow fluctuation. “Mild surge” represents the conditions where the annulus average mass flow oscillates but remains in the forward direction at all times. Such oscillations are characterized by the Helmholtz frequency

$$f_H = \frac{a_p}{2\pi} \sqrt{\frac{A_c}{V_p L_c}}, \quad (2)$$

of the compression system [6]. If the mass flow oscillations are severe and the mean flow reverses its direction during part of the cycle, the compressor has entered “deep surge” which is detrimental to both the turbocharger and engine. The dominant frequency is dictated by the plenum emptying and filling times and is typically below the Helmholtz resonance of the compression system. The accompanying pressure fluctuations upstream and downstream of the compressor manifest themselves as discrete low frequency sound peaks at sound pressure levels (SPL) exceeding 170 dB during deep surge.

Recently, an increasing number of spark-ignited internal combustion engines are being downsized and turbocharged to improve fuel economy and emissions. As a result of research and development in the field, the design of centrifugal compressors has progressed over those in earlier publications to include features such as splitter blades and large backsweep angles. In addition, advanced instrumentation and data acquisition systems allow for synchronized, time-resolved measurements of rotational speed along with pressure and temperature at key compression system locations. The objective of the present study is therefore to present detailed measurements and interpretation from a modern centrifugal compressor to further advance the current understanding of surge.

Following this introduction, Section 2 briefly describes the experimental setup, including the facility, instrumentation, and turbocharger. Section 3 discusses the time-averaged compressor performance data. Section 4 presents the acoustic measurements as total sound pressure level in the frequency domain. Time-averaged and time-resolved temperature measurements are provided in Section 5, followed by concluding remarks in Section 6.

2. EXPERIMENTAL FACILITY

The present work includes detailed compressor performance and acoustic measurements from a turbocharger installed on the turbocharger test bench at The Ohio State University [7]. This bench is located in a hemi-anechoic room, which is isolated from external sound sources, including the screw compressor and oil pump. The facility has “warm flow” capability by delivering up to 15 kW of electric heat to the flow upstream of the turbine inlet, while the compressor flow circuit is isolated from the turbine side to maximize flexibility.

Time-averaged compressor performance data is obtained from absolute pressure transducers and thermocouples installed in the compressor inlet and exit ducts along with mass flow rate from a calibrated orifice flow meter and rotational speed from a PICOTURN BMV6 sensor. The absolute pressure measurements are made with Kistler piezoresistive transducers, mounted flush with the internal walls of the compressor inlet (Model 4045A2) and exit (Model 4045A2 or 4045A5) ducts. The compressor upstream pressure transducer is located 90.7 mm from the inducer inlet plane of the impeller and the downstream pressure is measured 49.3 mm from the compressor exit interface. Temperature is measured with in-house built and calibrated (within ± 1 K) type T thermocouples with 0.001" wire diameter, where the junction measures air temperature in the center of the ducts' cross-sections. The compressor inlet thermocouple is installed 84.1 mm from the inducer inlet plane, and the exit temperature is measured 83.3 mm from the volute exit. In-duct instrumentation locations at the compressor inlet and outlet are depicted in Figs. 1 and 2, respectively.

In addition to steady-state compressor performance measurements, the facility is equipped with instrumentation to measure time-resolved pressure and temperature fluctuations. The dynamic pressure upstream of the compressor is measured with a PCB piezoelectric transducer (Model 106B), which is flush-mounted with the internal wall of the compressor inlet duct at the same axial location as the Kistler absolute pressure transducer. The sound propagating from the bellmouth of the compressor inlet duct is measured with a Brüel and Kjær Type 2235 Sound Level Meter (SLM). The SLM is positioned 0.25 m from the bellmouth on the compressor inlet duct, at an angle of 45° from the axis of the duct. Time-resolved temperature fluctuations are measured with the 0.001" thermocouples in the compressor inlet and exit ducts along with an additional location in the compressor housing, which is 14.6 mm upstream of the inducer tips. The time constant for these fast-response thermocouples is approximately 0.003 s, providing adequate frequency response to capture fluctuating temperatures during surge.

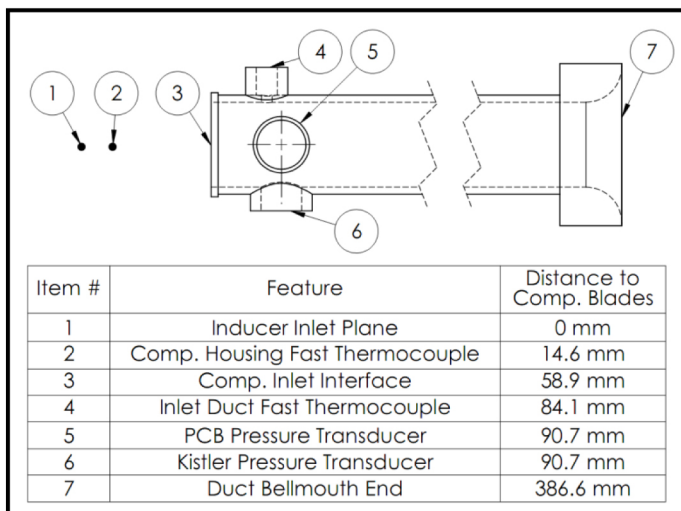


Figure 1. Compressor inlet duct instrumentation locations.

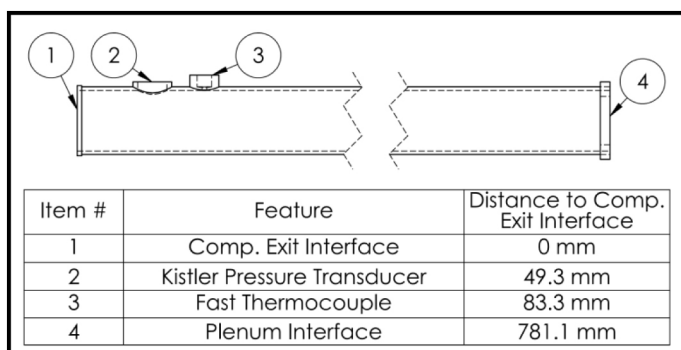


Figure 2. Compressor exit duct instrumentation locations.

The turbocharger studied is from the right-bank of the 3.5 L TiVCT GTDI engine. The compressor is model 046K77, with a 46 mm exducer tip diameter and a trim of 77. The connected compression system geometry was varied in order to study its influence on surge line location and frequency at which the instability occurs. Two distinctly different compression systems were studied, large and small B configurations, with multiple large B variations utilized. The large B compression system incorporates a well-defined plenum, which is connected to the compressor exit duct and an additional duct containing a control valve, as shown in Fig. 3. The plenum of this large B system has a moveable piston, allowing the volume of compressed air to be varied.

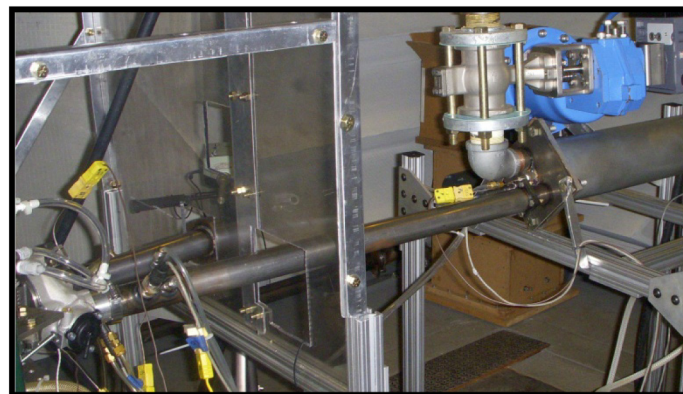


Figure 3. Large B compression system.

This system was studied primarily at 1/8 (1.155 L) of the full plenum volume, but results for full (9.240 L) and 1/2 (4.620 L) plenum volumes are included in Section 4 to demonstrate the influence on deep surge line location and frequency of surge instabilities. In addition to the plenum, compressed air occupies the volume between the diffuser inlet of the compressor and the plenum inlet along with that between the plenum exit and the control valve, where this total volume of compressed air is presented in Table 1 for each large B configuration. B is calculated using Eq. (1) and the time-averaged a_p from the lowest flow rate mild surge operating point and is shown in Table 1 for the mid rotational speed of 120 krpm as an example.

Table 1. Volume of compressed air and B for variations of the large B system.

Piston Position	Volume of Compressed Air, V_p [L]	B at 120 krpm [-]
1/8	2.71	0.48
1/2	6.18	0.72
1	10.80	0.95

Time-averaged compressor data is presented in Sections 3,4,5 at constant, corrected rotational speeds [8]

$$N_{\text{cor}} = \frac{N}{\sqrt{T_{01} / T_{\text{ref}}}}, \quad (3)$$

of 80, 95, 107, 120, 132, 147, and 162 krpm, where N is the actual rotational speed of the compressor, T_{01} is the total air temperature in the compressor inlet duct, and $T_{\text{ref}} = 298$ K is the reference temperature. All pressure (Section 4) and temperature (Section 5) results presented in the time and frequency domains are an example case from the 1/8 volume large B compression system at $N_{\text{cor}} = 120$ krpm, and the trends are similar at the other speeds and large B configurations.

A small B compression system was the second configuration studied. This system minimizes the volume of compressed air to 0.417 L by incorporating a shorter compressor exit duct, which is connected directly to the control valve, as shown in Fig. 4. The reduced V_p of this system, relative to the large B configurations, reduces B to 0.20, impedes the growth of instabilities, and extends the usable range of the compressor, as will be discussed in the following section.

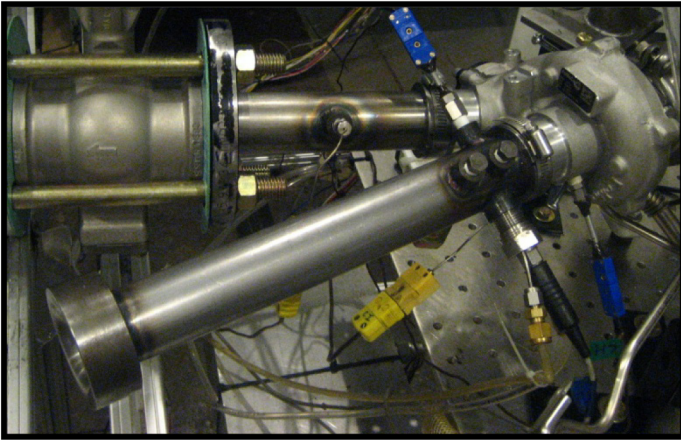


Figure 4. Small B compression system.

3. TIME-AVERAGED COMPRESSOR PERFORMANCE DATA

The compressor characteristics and efficiencies from the studied systems are presented and compared in this section at lines of constant, corrected rotational speed. The characteristic data is presented as total-to-total compressor pressure ratio

$$PR_{c,tt} = \frac{p_{02}}{p_{01}}, \quad (4)$$

versus corrected mass flow rate [8]

$$\dot{m}_{c,cor} = \dot{m}_c \frac{\sqrt{T_{01}/T_{ref}}}{p_{01}/p_{ref}}, \quad (5)$$

where p_{01} and p_{02} are the total pressures at the compressor inlet and exit measurement locations, respectively, $\dot{m}_c$ is the compressor mass flow rate, and $p_{ref}=100$ kPa is the reference pressure. The total-to-total isentropic efficiency $\eta_{c,tt}$ of the compressor is calculated as [9]

$$\eta_{c,tt} = \frac{PR_{c,tt}^{(\gamma-1)/\gamma} - 1}{\frac{T_{02}}{T_{01}} - 1}, \quad (6)$$

where T_{02} is the total air temperature at the compressor exit and γ is the ratio of specific heats for air.

The time-averaged $PR_{c,tt}$ vs. $\dot{m}_{c,cor}$ characteristics for the small B and the large B system at 1/8 plenum volume are shown in Fig. 5, and the $\eta_{c,tt}$ vs. $\dot{m}_{c,cor}$ data is compared in Fig. 6. Where experimental data from both the small and large B systems is available, their compressor characteristics and efficiencies appear to be nearly identical in Figs. 5 and 6, respectively, indicating that the performance is a function of the compressor alone. The large B compression system enters deep surge at flow rates just below the peak pressure ratio on each constant speed line, as shown in Fig. 5, while the stabilizing effect of the small B configuration allows the compressor to operate at much lower flow rates without discernible deep surge. Although deep surge (flow reversal) does not appear to occur for the small B system, mild surge is detected on the positively sloped portion of the characteristics.

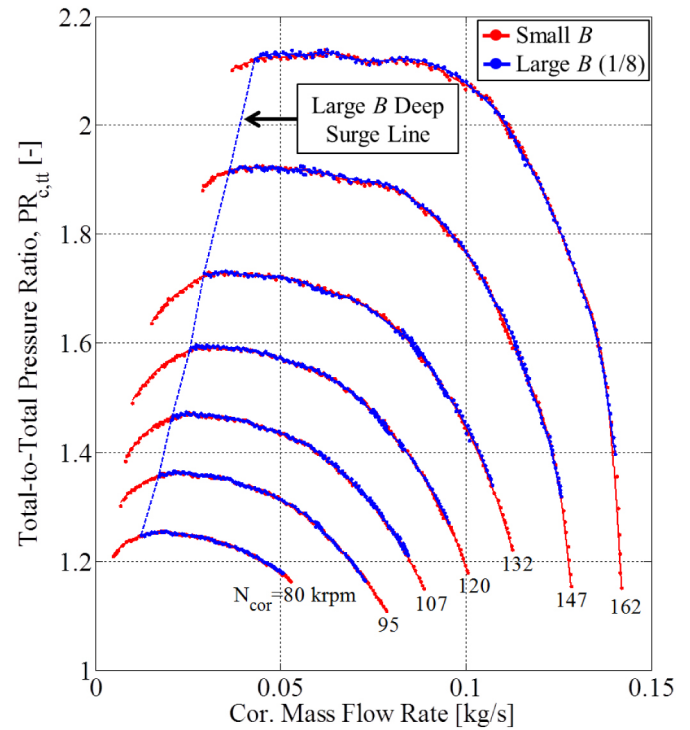


Figure 5. Time-averaged compressor characteristics from the small B and large B with 1/8 plenum volume systems.