

Prediction of Surge in a Turbocharger Compression System vs. Measurements

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ABSTRACT

The unsteady surge behavior of a turbocharger compression system is studied computationally by employing a one-dimensional engine simulation code. The system modeled represents a new turbocharger test stand consisting of a compressor inlet duct breathing from ambient, a centrifugal compressor, an exit duct connected to an adjustable-volume plenum, followed by another duct which incorporates a control valve and an orifice flow meter before exhausting to ambient. Characteristics of mild and deep surge are captured as the mass flow rate is reduced below the stability limit, including discrete sound peaks at low frequencies along with their amplitudes in the compressor (downstream) duct and plenum. The predictions are then compared with the experimental results obtained from the cold stand placed in a hemi-anechoic room. The computational results are shown to be capable of reproducing a number of key experimental observations, including the details of low-frequency pressure fluctuations in the compressor ducts and plenum; and the transition from stable operation to oscillating mild surge.

1. INTRODUCTION

It is desirable to raise the compressor pressure ratio in order to increase the torque of downsized turbocharged engines. At low engine speeds, however, elevated boost pressures may cause the compression system to operate beyond its stability limit, resulting in unsteady flow phenomena and unacceptable noise levels. When the slope of the compressor characteristic is negative, the pressure ratio of the compressor increases in a stable mode as the mass flow rate is reduced. As the slope approaches zero, the pressure rise reaches its maximum value

(for a given turbocharger speed) and a further reduction in mass flow causes a change in the flow pattern [1], leading to either stall or surge. Surge is the low-flow behavior of greatest concern, resulting in axisymmetric, 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

$$f_H = \frac{a}{2\pi} \sqrt{\frac{A_c}{V_p L_c}} \quad (1)$$

of the compression system [2], where a is the speed of sound, V_p is the volume of the compressed air, A_c is the equivalent cross-sectional compressor duct area, and L_c is the equivalent length of the duct. 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 can be 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. One manifestation of surge is discrete low frequency sound peaks at sound pressure levels exceeding 170 dB during deep surge. These represent severe pressure fluctuations within the compression system comparable to those in the exhaust primary runners of production engines.

Zero-dimensional (lumped parameter) models have been developed to predict the surge behavior of both axial [3] and

centrifugal [4] compression systems with reasonable accuracy. These models have formulated a set of nonlinear equations to estimate the system dynamics in an upstream compressor duct open to ambient, a compressor, a downstream compressor duct, a plenum, and a throttle valve exhausting to ambient. Greitzer's approach [3] has been frequently used to estimate the operating point for surge inception and the subsequent oscillations. This analysis has revealed a dimensionless number

$$B = \frac{U}{2a} \sqrt{\frac{V_p}{A_c L_c}}, \quad (2)$$

where U is the impeller blade tip speed. Greitzer has suggested that, above a critical value of B , surge becomes the instability mode encountered. Hence the onset of instability is a function of the impeller blade tip Mach number as well as the geometry of the compression system. The measured stall/surge boundary in Greitzer's work was around $B=0.8$. Such a specific value, however, cannot be generalized to all compressors. Hansen *et al.* [5] demonstrated that Greitzer's lumped parameter formulation could be extended to centrifugal compressors. Fink [6] added the shaft speed dynamics for a surging centrifugal compressor with a radial impeller and vaneless diffuser surrounded by a volute. A method for implementing this lumped approach into an engine simulation code has been outlined by Theotokatos and Kyratos [7]. These lumped parameter approaches involve, however, a number of limiting assumptions [2], including: (a) incompressible flow in the compressor duct, (b) isentropic plenum expansion or compression, (c) choked throttle valve, (d) short throttle duct length so that the inertia can be ignored, (e) negligible velocity in the plenum, (f) discontinuity of pressure and density across the compressor, which is modeled as an actuator disk, and (g) negligible gas angular momentum in the compressor passages compared to the impeller angular momentum. Incorporating surge prediction capabilities into a one-dimensional (1-D) computational approach has numerous advantages over such a lumped formulation, including: (1) elimination of the foregoing assumptions (a)-(e), (2) spatially distributed wave dynamics, and (3) governing equations common with engine simulation codes. Galindo *et al.* [8] incorporated a surge model into their own 1-D gas-dynamics code. They were able to predict the amplitude and dominant frequency of fluctuations in the compressor exit pressure during deep surge with reasonable accuracy. However, their mild surge predictions underestimated the dominant frequency and overestimated the amplitude of pressure oscillations. A methodology for accurately predicting unsteady mild surge physics in the compression system of a turbocharger using a nonlinear 1-D time-domain computational approach employed in engine simulations was recently provided [9].

The objective of this study is to further advance the approach in [9] to computationally determine the boundary between stable and mild surge operation and the transition from mild to deep surge, while characterizing the resulting oscillations in terms of amplitude and frequency. The unsteady surge physics within the compression system is investigated using a commercially available 1-D engine simulation code. A parallel experimental effort [10] on a small and large B compression system has also been utilized here. The compressor map data from the small B system is extrapolated and interpolated using a preprocessor developed in the present study and input to the code. The large B system is designed with a large plenum such that surge is the low-flow instability mode. The predicted pressure oscillations at key compression system locations from the current study are then compared with experimental observations to evaluate the accuracy of simulations.

Following this introduction, Section 2 briefly describes the experimental setup. Section 3 discusses the compressor performance map and Section 4 the 1-D compression system model. Simulation results are then compared with the experimental observations in Section 5, followed by concluding remarks in Section 6.

2. EXPERIMENTAL SETUP

Two different compression systems are studied experimentally: a small and large B system with B numbers calculated as 0.18 and 0.63, respectively, at an impeller tip speed $U=310$ m/s. The ducting of the two configurations was designed differently to explore the dependence of the low-flow stability limit on the B number. Both systems utilize the same turbocharger and compressor inlet duct, which has a bellmouth interface with ambient. The compressor is a BorgWarner 1880 DCF with six main and six splitter blades on the impeller, all of them backswept. Flow exiting the impeller is decelerated by a vaneless diffuser and volute. The small B system has a short duct connecting the volute exit directly to the control valve, minimizing the volume of compressed air, as shown in Fig. 1. The large B system has a longer duct at the compressor exit that flows into a plenum of larger cross-sectional area along with an additional duct connecting the plenum and control valve, as shown in Fig. 2. The plenum in the large B configuration has a piston that seals against the inner surface of the plenum housing and acts as a moveable wall to adjust the length and therefore volume. For the present study, the piston position is maintained at the mid length (0.429 m) of the plenum housing, creating a volume of 4.6 L. The pneumatic control valve provides a flow restriction which creates a pressure drop from the plenum pressure to near ambient (plus additional backpressure) and represents the end of the compression system. The angular position of this valve is adjusted through the data acquisition system to obtain the desired compressor mass flow rate, while an additional control valve at the

turbine inlet is used to attain the rotational speed. This mass flow rate and speed defines a unique operating point for the compressor with a corresponding pressure ratio and isentropic efficiency. The flow exiting the control valve passes through an air-to-water intercooler to reduce its temperature well below the acceptable limit for the PVC ducting in the subsequent flow meter. This orifice flow meter allows for measurement of the steady mass flow rate during stable operation and time-averaged values during unsteady surge. Additionally, the turbo bench is instrumented to obtain steady and unsteady measurements of turbocharger rotational speed along with static pressure and temperature at the compressor inlet, exit, and in the plenum (of the large *B* system). Specific details regarding the turbo bench are provided in [10].

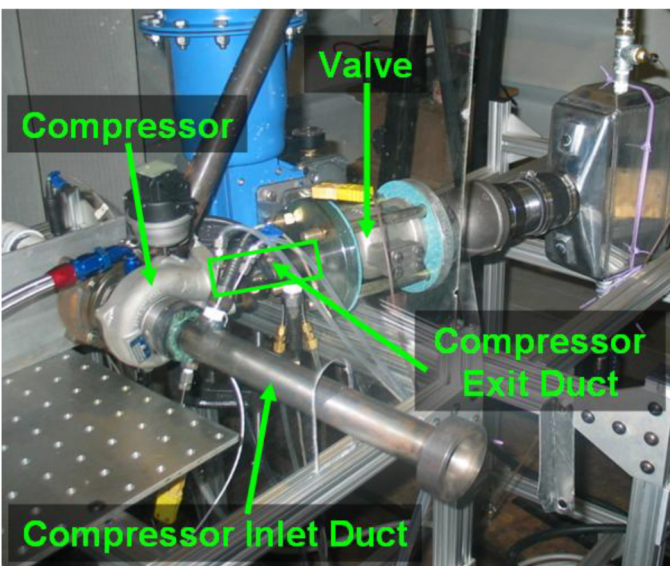


Figure 1. Small *B* compression system.

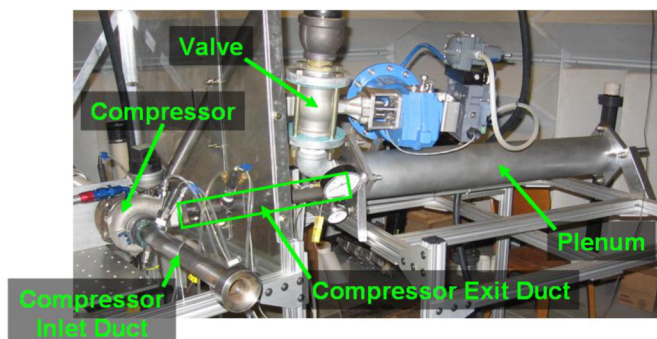


Figure 2. Large *B* compression system.

The geometry of the large *B* compression system components is given in Table 1. The equivalent compressor inlet duct is a 1-D simplification of the impeller inlet and impeller, and the equivalent compressor exit duct is a similar simplification of the diffuser and volute geometry.

Table 1. Compression system geometry.

Component	Length, <i>l</i> [cm]	Area, <i>A</i> [cm ²]	<i>l/A</i> [m ⁻¹]	Volume, <i>V</i> [L]
Comp. Inlet Duct	37.78	9.372	403.1	
Equiv. Comp.	Inlet 5.562 Exit 34.56	7.808 2.980	71.23 1160	
Comp. Exit Duct	42.23	7.055	598.6	0.2979
Plenum	42.86	107.8	39.76	4.620
Plenum Exit Duct	25.40	15.47	164.2	0.3929
Total			2437	5.414

The theoretical Helmholtz resonance frequency f_H of this compression system is then calculated using Eq. (1) and the geometry in Table 1 as

$$f_H = \frac{379 \text{ m/s}}{2\pi} \frac{1}{\sqrt{(2437 \text{ m}^{-1})(0.005414 \text{ m}^3)}} = 16.6 \text{ Hz},$$

where the speed of sound is calculated using the experimental plenum temperature of 358 K from a stable operating point near the surge line at a tip speed of 310 m/s, and 5.414 L is the volume of compressed air in the system (from diffuser to control valve).

3. COMPRESSOR PERFORMANCE

To facilitate surge predictions, the compressor needs to be characterized first by user-provided performance maps. This is accomplished in GT-Power [11] (hereafter referred to as the “code”) by providing performance data in the form of corrected mass flow rate, pressure ratio, and efficiency at constant, corrected rotational speeds. The corrected rotational speed and pressure ratio are the inputs to the lookup map, and the corrected mass flow rate and efficiency are the outputs. The current study provides compressor data as an external text file with a ‘.cmp’ extension, which requires the user to perform all interpolations and extrapolations, and, as a result, the preprocessor of the code is not used. The MATLAB script developed in [9] is expanded for the present study to include reverse flow along with the previously implemented extrapolation and interpolation capabilities to formulate the compressor information into a text file with the format required by the code. For further information regarding this compressor map format the User's Manual may be referred [12].

Compressor performance maps are created for both the small and large *B* compression systems at three constant, corrected rotational speeds of $U_{cor} = 230, 310, \text{ and } 370 \text{ m/s}$, in the form of total-to-total pressure ratio PR_c vs. corrected mass flow rate $\dot{m}_{c,cor}$ in Fig. 3 and total-to-total isentropic efficiency η_c .

vs. $\dot{m}_{c,cor}$ in Fig. 4. The impeller tip speed U is corrected for total compressor inlet temperature T_{01} [13] as

$$U_{cor} = \frac{U}{\sqrt{T_{01}/T_{ref}}}, \quad (3)$$

where $T_{ref} = 298$ K the standard reference temperature. The compressor mass flow rate $\dot{m}_c$ is corrected for T_{01} and total compressor inlet pressure p_{01} [13] as

$$\dot{m}_{c,cor} = \dot{m}_c \frac{\sqrt{T_{01}/T_{ref}}}{p_{01}/p_{ref}}, \quad (4)$$

where $p_{ref} = 100$ kPa the standard reference pressure.

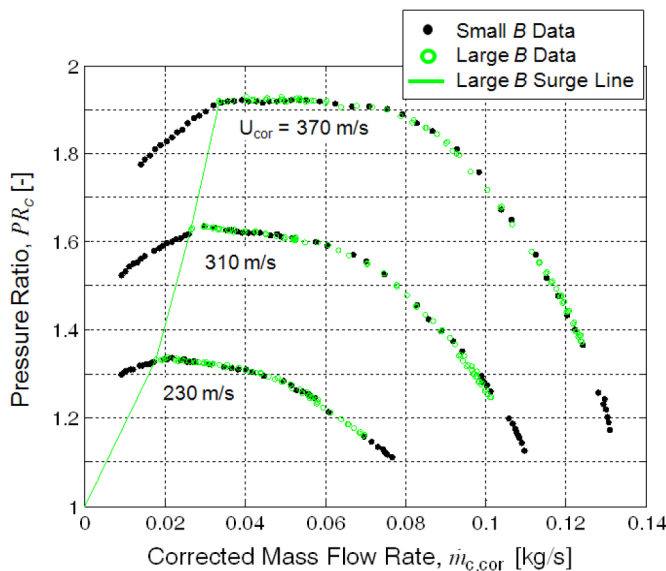


Figure 3. Small and large B compressor characteristics for the BorgWarner 1880 DCF compressor.

The compressor map data from the two systems appears to be identical over the range that steady large B data was obtainable, as shown in Figs. 3 and 4. This illustrates that the compressor characteristics are independent of the compression system ducting (a property of the compressor alone), yet the coupled compression system (compressor, ducting, and throttle) dictates the stable portion of the map available for use (location of the surge line). The small B system is capable of operating free of surge at lower mass flow rates and closer to choke than the large B system, hence it allows a wider compressor map to be obtained. Such an extended map (of the small B system) is desirable for predicting compressor performance because it requires extrapolations over significantly smaller ranges (to choke and zero mass flow rate) in order to cover the entire forward flow

operating region when compared to the large B data. In the present work, the small B system is exclusively utilized for obtaining compressor map data for the model, while the large B system provides steady-state data at the boundary with mild surge along with unsteady mild and deep surge measurements for comparison with predictions.

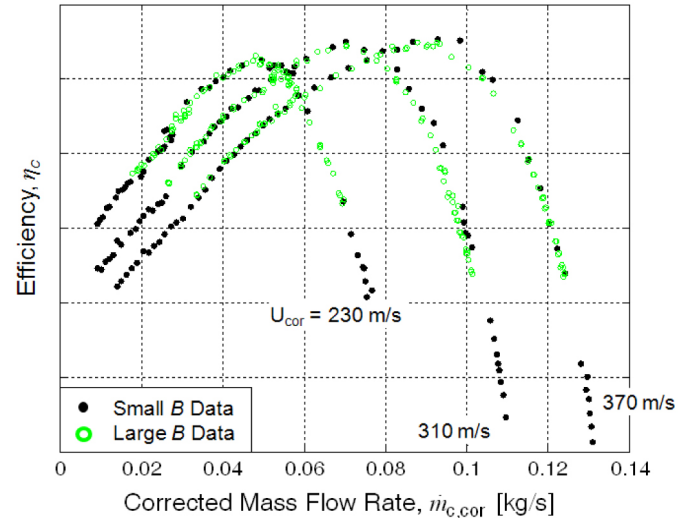


Figure 4. Small and large B compressor efficiency for the BorgWarner 1880 DCF compressor.

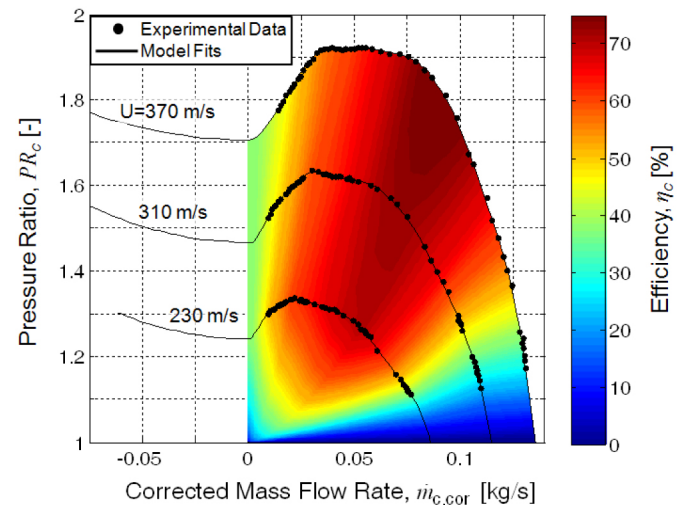


Figure 5. Final compressor map used for the model.

The compressor map used in the present study was created from the small B compressor performance maps in the form of PR_c vs. $\dot{m}_{c,cor}$ and η_c vs. $\dot{m}_{c,cor}$, as shown in Figs. 3 and 4, respectively. The final compressor map that is input to the code consists of the compressor characteristics that are extrapolated and interpolated to cover the entire forward and reverse flow operating region, along with the forward flow efficiency. The model characteristics at tip speeds of 230,