Simulation of surge in the air induction system of turbocharged internal combustion engines

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ABSTRACT
A computational approach has been developed to accurately predict compression system surge instabilities within the induction system of turbocharged internal combustion engines by employing one-dimensional, nonlinear gas dynamics. This capability was first developed for a compression system installed on a turbocharger gas stand, in order to isolate the surge physics from the airborne pulsations of engine and simplify the ducting geometry. Findings from the turbocharger stand study were then utilized to create a new model of a twin, parallel turbocharged engine. Extensive development was carried out to accurately characterize the wave dynamics within key induction system components in terms of transmission loss and flow losses for the individual compressor inlet and outlet ducts. The engine was instrumented to obtain time-resolved measurements for model validation during surge instabilities, and simulation results agree well with the experimental data, in terms of both the amplitude and frequency. The present quasi-one-dimensional approach relaxes many of the assumptions inherent to earlier lumped parameter surge models, therefore it provides the flexibility to model advanced boosting systems with multiple turbochargers and complex ducting geometry.

Keywords: Surge, Centrifugal Compressor, Turbocharger, Simulation, Internal Combustion Engine
I-INCE Classification of Subjects Number(s): 11.6.2 (Noise Generating Devices: Compressors), 11.6.4 (Noise Generating Devices: Engines), 21.4 (Resonance, standing waves, and normal modes)

1. INTRODUCTION
The combination of turbocharging and downsizing is a proven method to improve the fuel efficiency of spark ignition (SI) internal combustion (IC) engines by increasing the brake mean effective pressure (BMEP). However, even a properly sized contemporary, centrifugal compressor imposes limits on the maximum engine torque at both extremes of the engine speed range, and therefore, the level of downsizing that is deemed acceptable. The flow range of centrifugal compressors places restraints on engine performance due to the high-flow choke limitation and low-flow surge instabilities. At elevated engine speeds and loads, the compressor flow rate is high and approaches the choked (sonic) condition, which requires the compressor to be sufficiently large to provide adequate flow rate and pressure to the intake manifold of the engine during operation at peak power. Additionally, deterioration of the compressor flow field at low flow rates results in compression system instabilities that limit the maximum obtainable boost pressure, therefore torque at low engine speeds.

Unfavorable flow fields are often encountered in the inducer (inlet) of the impeller and the diffuser, due to the presence of adverse pressure gradients. Stall and flow separation develop as the compressor flow rate is reduced, and at low flow rates, the severity of these flow losses increases sufficiently to diminish the ability of the compressor to further increase the delivery pressure. As a result, the (outlet-to-inlet) compressor pressure

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ratio vs. mass flow rate constant speed characteristics reach a peak (zero slope), and the potentially unstable positively sloped region occurs at flow rates below this peak. A linearized stability analysis of the lumped parameter (zero-dimensional or 0D) formulation for unsteady fluid flow in the simplified duct-plenum compression system configuration in Fig. 1 was presented by Greitzer [1]. This analysis revealed that surge instabilities are possible when the local slope of the constant speed compressor characteristic is positive. Surge instabilities are a compression system phenomenon, so the ducting (including plenum) attached to the compressor plays a vital role in determining the location of the surge line, along with the amplitude and frequency of surge oscillations as the instability locks onto the natural frequency of the system.

Figure 1: Simplified compression system.

Surge is a self-excited, system oscillation which can be categorized as “mild” or “deep” depending on the degree of mass flow fluctuation. “Mild surge” represents the conditions where the cross-section averaged mass flow oscillates but remains in the forward direction at all times. Such oscillations are dictated by the natural frequency of the compression system, which corresponds to the Helmholtz resonance frequency [2]

\[ f_H = \frac{a_p}{2\pi} \sqrt{\frac{A_C}{V_p L_C}}, \]

for the simplified compression system geometry shown in Fig. 1, where \( a_p \) is the speed of sound in the plenum, \( V_p \) is the volume of compressed air, \( A_C \) is the equivalent cross-sectional area of the compressor duct, and \( L_C \) is the equivalent duct length. The frequency of such oscillations is unique to the duct-plenum (cavity) coupling in terms of inertia in the duct balanced by pressure forces due to compressibility in the cavity. Mild surge instabilities occur as the local slope of the (total-to-static pressure ratio vs. mass flow rate) compressor constant speed characteristic reaches a positive value, and the amplitude of mass flow rate and pressure fluctuations increases with decreasing flow rate. Further reduction in flow rate results in “deep surge,” which is characterized by severe oscillations and flow reversal during part of the cycle. The dominant frequency is now below the Helmholtz resonance of the compression system, since it is influenced by the filling and emptying time of the compressed volume of air. One manifestation of deep surge is discrete low frequency sound peaks at sound pressure levels exceeding 170 dB [3]. It is unacceptable to operate a compressor in deep surge because it is a violent instability, which results in drastically increased levels of noise, loss of pressure rise, and potential mechanical failure. Therefore, the boundary between mild and deep surge operation marks the low-flow operating limit of a compression system.

In addition to the linear stability analysis, a nonlinear lumped parameter model for prediction of surge in axial compression systems was also presented by Greitzer [1] in 1976. This formulation allowed for computational predictions of surge instabilities, by solving a coupled set of nonlinear equations to estimate the system dynamics for the compression system configuration in Fig. 1. This analysis has revealed a dimensionless number

\[ B = \frac{U}{2a_p \sqrt{\frac{V_p}{A_C L_C}}}, \]

providing a qualitative description of the susceptibility of a compression system to surge, where \( U \) is blade tip speed of the impeller. 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. Hansen et al. [4] demonstrated that Greitzer’s model could be extended to centrifugal compressors, and the approach was further advanced by Fink [5] to relax the assumption of constant shaft speed. Fink’s model consists of four lumped balance equations to characterize the compression system physics during steady and unsteady operation, including a momentum balance for the equivalent compressor duct, mass balance for the plenum, a first-order time lag to allow for deviation of the instantaneous compressor pressure rise from the steady-state...
characteristic, and the angular momentum balance for the rotating assembly. A method for implementing the foregoing lumped model into a 0D engine simulation code has been outlined by Theotokatos and Kyrtatos [6].

These lumped parameter approaches have the ability to predict the frequency and amplitude of surge oscillations with reasonable accuracy and quick solution time, however, they involve a number of limiting assumptions, 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) frictionless, (g) adiabatic, (h) discontinuity of pressure and density across the compressor, which is modeled as an actuator disk, and (i) negligible gas angular momentum in the compressor passages compared to the impeller angular momentum.

Incorporation of surge prediction capability into a compressible, unsteady, one-dimensional (1D) gas dynamics code utilized for engine modeling and development has numerous advantages over the traditional 0D approach, including the elimination of foregoing assumptions (a)-(g) and spatially distributed wave dynamics. As the engine air induction system geometry begins to deviate from the simplified case (recall Fig. 1), many of the 0D model assumptions begin to breakdown, hence the need for more detailed 1D treatment.

A 1D surge prediction capability was developed during the early phase [7] of the present work, where the compressor model in a commercially available 1D engine simulation code was appropriately modified. Predictions of both mild and deep surge were carried out with a compression system model of the turbocharger experimental stand at The Ohio State University Center for Automotive Research (OSU-CAR). These predictions agreed well with the time-resolved experimental data from the turbocharger stand in terms of the amplitude and frequency of pressure fluctuations during both mild and deep surge.

The surge modeling methodology presented in the current work builds upon the capability developed for the idealized turbocharger stand compression system configuration in Fig. 1. The simplified ducting geometry and isolation from engine pulsations was indispensable while developing the model, but ultimately, the goal is to predict compression system surge while operating on an engine. Therefore, the present work incorporates the surge prediction capability into a 1D model of a twin-turbocharged engine, and extensive work was carried out to improve the predicted pressure losses and wave dynamics within the air induction system. This 1D approach provides the flexibility to model any number of compression system duct and volume configurations, along with multiple turbocharger setups in either series or parallel arrangements. This general formulation provides a unique capability that has the potential to markedly improve turbocharged engine models.

Following this introduction, Section 2 describes the turbocharger stand and engine experimental facilities. Section 3 discusses the 1D gas dynamics code utilized for modeling, along with modifications carried out on the compressor model. Surge predictions are compared with experimental data from the engine in Section 4, followed by concluding remarks in Section 5.

2. EXPERIMENTAL FACILITIES

Numerous experimental facilities were utilized throughout the present work, but only the major setups that are directly modeled for surge prediction are elaborated on in the following sections, including the turbocharger and engine laboratories. An impedance tube setup was also employed to determine the transmission loss (TL) of major engine air induction system (AIS) components, including the air cleaner box and charge air cooler (air-to-water heat exchanger). In addition, a flow bench was utilized to calibrate mass air flow (MAF) sensors that were installed within the compressor inlet ducts of the engine, along with characterizing the pressure drop of the individual compressor inlet and outlet ducts.

2.1. Turbocharger laboratory

The turbocharger laboratory at OSU-CAR [8], shown in Fig. 2, is located within a hemi-anechoic chamber and 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 of temperature, absolute pressure, dynamic (acoustic) pressure, mass flow rate, and rotational
speed. This facility is utilized to carry out experiments to obtain extended flow range compressor performance data and time-resolved data during surge instabilities.

Figure 2: Instrumented turbocharger installed on the steady-flow stand at OSU-CAR.

The turbocharger studied on the stand is from the right-bank of a twin, parallel turbocharged Ford 3.5 L Twin Independent Variable Camshaft Timing (TiVCT) Gasoline Turbocharged Direct Injection (GTDI) engine, which is also installed in the engine laboratory. Two distinctly different compression systems are investigated, which are categorized as “large” and “small” B configurations. The large B (compressed air volume) compression system incorporates a well-defined plenum (cross-sectional area ratio of plenum to outlet duct is approximately 15), which is connected to the compressor outlet duct and an additional duct containing a control valve, similar to the configuration shown in Fig. 1. For this large B system, \( V_p = 2.71 \) L, and specific information about the instrumentation and measurement locations is provided by Dehner et al. [9].

Detailed, time-resolved compressor data was collected at constant, corrected rotational speeds [10]

\[
N_{cor} = \frac{N}{\sqrt{T_{01}/T_{ref}}},
\]

of 80, 95, 107, 120, 132, 147, and 162 krpm, where \( N \) is the actual rotational speed of the impeller, \( T_{01} \) is the total air temperature at the compressor inlet, and \( T_{ref} = 298 \) K is the reference temperature. Experimental data from the large B compression system was utilized to provide time-resolved data for direct comparison with 1D surge predictions, as was done for a different turbocharger earlier [3]. The steady-state compressor performance characteristics with the large B system are shown in Fig. 3, which is represented as the ratio of total inlet to static outlet pressure \( PR_{c,ts} \) vs. corrected mass flow rate

\[
\dot{m}_{c,cor} = \frac{\dot{m}_c}{\sqrt{T_{01}/T_{ref}}},
\]

where \( \dot{m}_c \) is the actual mass flow rate, \( p_{01} \) is the total pressure at the compressor inlet, and \( p_{ref} = 100 \) kPa is the reference pressure.

A small B compression system was the second configuration studied, which minimizes the volume of compressed air by incorporating a short, instrumented duct between the compressor outlet and a control valve. This small volume system impedes the growth of instabilities [recall Eq. (2)] and is able to operate without deep surge at mass flow rates well below that of the large volume system, as shown in Fig. 3. The extended
flow range compressor data from the small B system is used to characterize the steady-state compressor performance within 1D models of both the turbocharger stand and the engine.

![Graph showing flow range compressor data.](image)

Figure 3: Impact of compressed air volume on useable compressor operating range.

2.2. Engine laboratory

A 2011 production version of the Ford 3.5 L TiVCT GTDI engine is installed on a dynamometer in the engine laboratory at OSU-CAR, and the turbochargers on this engine are the same design that was studied on the turbocharger stand. The engine configuration is a V6 with outboard exhaust and parallel turbochargers, such that each turbine is fed from the exhaust gas from one bank of cylinders, and the air flow path is shown in Fig. 4. This engine has been instrumented with the sensors identified in Fig. 4 and connected to a data acquisition system, which is triggered by an encoder to capture time-resolved measurements. In addition, an ATI module allows for communication with the open powertrain control module (PCM) for viewing and recording measurements from standard engine sensors (manifold absolute pressure, total air flow rate, etc.), along with modification of engine operating parameters (air-fuel ratio, spark timing, valve timing, etc.)

![Diagram of engine instrumentation.](image)

Figure 4: Instrumentation installed on the engine.
3. ONE-DIMENSIONAL MODELS

From the outset, the objective of the present work was to accurately predict surge in the compression system of IC engines utilizing a 1D gas dynamics code [11]. To accomplish this goal, the surge model was first developed for the turbocharger stand, which simplifies the duct geometry and isolates the system from engine airborne pulsations. In addition, the engine model was developed and validated in two phases. The first phase focused on accurately capturing full-load performance at low engine speeds, where compressor operation is stable, but in the vicinity of the surge line. This initial phase of engine model development focused on predicting airborne pulsations throughout the compression system, flow rate through the individual compressors, and time-averaged performance. The second phase of engine modeling, which is the focus of the present work, aims to reproduce the deep surge observed experimentally.

3.1. Ducting

The nonlinear balance equations of mass, momentum and energy are solved in the present simulations, along with the equation of state, using an explicit time integration method. These equations are applied to spatially discrete control volumes (of length $dx$) within the air flow system. The 1D mass conservation is represented by

$$\frac{dm}{dt} = \sum_{in} \dot{m} - \sum_{out} \dot{m},$$

where $m$ is the mass of the discrete volume, $t$ is time, and $\dot{m}$ is the mass flow rate across the control volume boundaries. The 1D momentum conservation may be expressed as

$$\frac{dm}{dt} dx = Ap + \sum_{in} \dot{m}u - \sum_{out} \dot{m}u - 4f \frac{p\rho u |u|}{2} \frac{Adx}{D} - K \left( \frac{1}{2} \rho u |u| \right) A,$$

where $p$ is pressure, $A$ is the cross-sectional area of flow, $u$ is the velocity at the boundary of the control volume, $f$ is the friction coefficient, $\rho$ is the density, $D$ is the equivalent diameter, and $K$ is the loss coefficient.

The 1D energy conservation is expressed as

$$\frac{d(me)}{dt} = p \frac{dV}{dt} + \sum_{in} \dot{m}h - \sum_{out} \dot{m}h - h_c A_s (T - T_w),$$

where $e$ is the total internal energy (internal energy plus kinetic energy) per unit mass, $V$ is the volume, $h$ is the specific enthalpy, $h_c$ is the heat transfer coefficient, $A_s$ is the heat transfer area, $T$ is the fluid temperature, and $T_w$ is the wall temperature. The solution of Eqs. (5)–(7) in combination with the equation of state yields the scalar fluid properties within each control volume ($p$, $T$, $\rho$, $e$, and $h$) and the vector properties at the boundaries ($\dot{m}$ and $u$). In addition, the combustion species may also be tracked, which is a feature that is utilized within the engine model.

3.2. Compressor Model

The compressor portion of the model, along with the attached inlet duct, outlet duct, and shaft is shown in Fig. 5. This compressor model consists of the actuator disk (0D), which provides the compressor performance by means of a lookup table, two additional ducts representing the equivalent geometry of the compressor, along with a torque applied to the shaft and energy equation source term, in order to provide nonzero compressor power during flow reversal. The rotational inertia of the turbospool is input to the turbo shaft component of the model, which utilizes an angular momentum balance to determine the change in shaft speed. Instead of incorporating a turbine into the turbocharger stand model, a drive torque is applied to the turbo shaft for simplification.

Both 0D and 1D surge simulations require knowledge about the steady-state compressor performance under low and reversed flow conditions, which due to surge instabilities, is not accessible using a typical (large $B$) experimental setup. There are two approaches adopted in literature for quantifying the compressor performance in these regions: mathematical extrapolations [6,12] and extended flow range experimental measurements [13]. The approach adopted in the present work involves a combination of these experimental and mathematical techniques, where extended range low-flow rate compressor data is obtained from a small $B$ setup (recall Fig. 3), and the reverse flow performance is estimated by a quadratic expression. 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 addition, the stability of a compression system is extremely sensitive to the local slope of the compressor constant speed line [1], so the present work will utilize extended range small $B$ experimental data for modeling whenever possible.

One-dimensional models utilize steady-state compressor data to characterize performance in the form of corrected mass flow rate, pressure ratio, and efficiency at constant, corrected rotational speeds. Corrected rotational speed and compressor pressure ratio are the inputs to the lookup map, where $PR_c$ is calculated from the pressures in the control volumes immediately up- and downstream of the actuator disk. The outputs from the lookup table are the compressor corrected mass flow rate and isentropic efficiency. Since compressor operation is highly unsteady during surge instabilities, a first order time constant $\tau_{GTP} = n/\omega$ is applied to damp changes in compressor mass flow rate, where $n=2$ shaft revolutions was the duration required for a compressor flow field to adapt to 63.2% of the fully-developed, asymptotic value after a sudden change [1], and $\omega$ is the angular velocity of the shaft. This damping prevents sudden, unrealistically large changes in mass flow rate and allows the compressor operating point to deviate from the steady-state performance map during unsteady operation.

For the current study, a MATLAB [14] script was developed to extrapolate, interpolate, and format this compressor information into a text file with the format required by the code, and as a result, the built-in preprocessor of the software is not used. A plot of the compressor performance map provided to the code is shown in Fig. 6, where steady-state experimental data from the small $B$ turbocharger stand compression system is superimposed for ease of comparison. This experimental data has been extrapolated to $PR_{c,ts} = 1$, to the reverse flow region, and to zero speed. Full details regarding methodology utilized for extrapolation and interpolation of compressor data and treatment of compressor power during flow reversal may be found in Dehner et al. [3,7,15,16].

In order to accurately predict the compression system stability limits and surge, the geometry of the compressor must also be incorporated into the model. A computationally effective method for representing the compressor geometry in a 1D model is to simplify it as straight, constant cross-sectional area ducts. A pipe is placed at the inlet of the actuator disk to represent the combined impeller inlet and impeller geometry, and an additional duct is placed at the exit of the disk to represent the combined diffuser and volute geometry, as shown in Fig. 5. The geometry of the equivalent ducts is defined to preserve both the mean wave propagation length and volume of the original components [7], since these properties influence both the resonance frequency of the compression system and the surge stability limit.
3.3. Twin-Turbocharged Engine Model

The ducting portion of the engine model begins with a constant pressure and temperature boundary condition specified at the opening of the inlet snorkel, which is connected to the air cleaner box (ACB). The ACB is an induction system silencer which targets airborne pulsations due to intake valve events. Downstream of the ACB, the ducting splits into the right-bank and left-bank compressor inlet ducts (RB CI and LB CI, respectively), where MAF sensors are installed. The compressors are the next components in the AIS, and the compressor portion of the 1D turbocharger stand compression system model (recall Fig. 5) is used for both the right- and left-bank compressors within the engine model, including the equivalent ducting geometry and reverse flow implementation. Downstream of the compressors, air flows through the right-bank and left-bank compressor outlet ducts, RB CO and LB CO, respectively. The other end of the compressor outlet ducts connect to a symmetric Y-pipe, which recombines the flows before entering the charge air cooler (CAC).

After the CAC, the air flows through a circular cross-section duct connected to a reduced cross-sectional area orifice plate to represent the flow restriction of the throttle and into the intake manifold. Air flow is then distributed to the individual cylinders through intake runners and ports. All of the engine cylinders are modeled, including (direct) injection of fuel into the combustion chambers and crank-angle resolved burn rates derived from in-cylinder pressure measurements. Flow in and out of the combustion chambers is calculated from discharge coefficients for the intake and exhaust valves in both the forward and reverse flow directions. Experimentally determined discharge coefficients are provided as a function of valve lift (opening), and the lift is specified by cam angle to match the lobe profile. The model also incorporates the ability to phase both intake and exhaust valve lift profiles relative to the base timing.

The high-pressure portion of the exhaust system model is identical for both banks up the turbine outlets, which includes exhaust runners, a log style manifold, collector, and turbine model. The turbine outlet ducts are combined with a Y-pipe before the flow passes through an expansion chamber exhaust silencer and the ducting is terminated with an additional constant pressure and temperature boundary condition, which is identical to that at the inlet.

4. COMPARISON OF ENGINE PREDICTIONS WITH EXPERIMENTAL DATA

The engine experiment was a (low-to-high) load sweep at 1,750 rpm, where the time-averaged compressor operating points are shown in Fig. 7. These experiments were completed under two different types of wastegate (WG) control for the LB turbine, including controlled (normal) operation where both the LB and RB
WGs are opened to limit brake torque at a given load command and a modified configuration which forces the LB WG closed at all times. The green rectangle in Fig. 7 identifies the region where both the RB and LB wastegates are closed with either WG control mode, and as expected, the corresponding compressor operating points for the two control modes are essentially identical within this range. As the load is elevated, the boost pressure must be limited to provide the target engine torque, and the WGs are partially opened to bypass a fraction of exhaust gas around the turbines. During normal (LB WG controlled) operation, both WGs open approximately equal amounts. When the LB WG is forced closed, the RB WG must open more than normal to compensate, and as a result, the mass flow rates of the compressor operating points diverge with further increasing load (RB WG opening), as shown in Fig. 7. With the LB WG closed, the RB compressor enters deep surge at approximately 93% of peak load, and this time-averaged compressor operating point is identified in Fig. 7. Note that during deep surge with the RB compressor, the LB compressor is still stable, since it is operating well within the stable region away from the surge line.

![Figure 7: Time-averaged compressor operating points during a load sweep at 1,750 rpm engine speed, for both LB wastegate controlled (normal operation) and closed.](image)

4.1. Deep Surge

While the engine is operating (1,750 rpm at 93% of peak load) with the RB compressor in deep surge, at the time-averaged point identified in Fig. 7, large amplitude fluctuations occur due to the instability. Predictions from the 1D engine model are compared directly to experimental data at the corresponding measurement locations, with the same sampling rate, and with identical engine operating parameters. The measured pressure at the RB CI location is shown in Fig. 8a, along with the corresponding prediction that closely matches in the time domain. The frequency domain counterpart is shown in Fig. 8b, where the dominant SPL peaks of the measurement and prediction are 152 dB at 3.4 Hz and 151 dB at 3.2 Hz, respectively. During deep surge instabilities, the predicted SPL at the firing frequency (87.5 Hz) is 138 dB, which is 13 dB lower than the value at the fundamental surge frequency. Therefore, the engine pressure fluctuations are dominated by the surge instability when deep surge is present. In addition, the prediction captures harmonics at integer multiples of the dominant peak with reasonable accuracy.

At the RB CO measurement location, the predicted pressure is extremely close to the measurement in the time domain, as shown in Fig. 9a. The frequency domain counterpart is shown in Fig. 9b, where the dominant SPL peaks of the measurement and prediction are nearly identical, with 168 dB at 3.4 and 169 dB at 3.2 Hz, respectively. Once again, the prediction does reasonably well in capturing the harmonics, and the predicted SPL at the firing frequency (87.5 Hz) is 151 dB, which is 18 dB lower than the fundamental surge frequency.
Figure 8: Pressure at the right-bank compressor inlet location with the right-bank compressor operating in deep surge at 1,750 rpm: (a) time domain and (b) frequency domain.

Figure 9: Pressure at the right-bank compressor outlet location with the right-bank compressor operating in deep surge at 1,750 rpm: (a) time domain and (b) frequency domain.

The mass flow rate through each compressor was measured with the MAF sensors installed in the compressor inlet ducts, as shown in Fig. 10. The measured and predicted LB compressor mass flow rates fluctuate, primarily due the large amplitude pressure disturbances propagating from the connected right-bank compressor ducting, yet the mass flow rate is always well within the stable operating region. In contrast, the predicted mass flow rate of the right-bank compressor fluctuates between approximately -45 and 95 g/s, with rapid time rate of change when the flow direction switches. At the RB CI measurement location, large pressure increase and reductions are observed (recall Fig. 8) during the fast transitions from forward-to-reverse and reverse-to-forward flow direction, respectively. The MAF sensors used for the present study have a stated time-constant of approximately 15 ms, so they are not capable of fully capturing rapid flow rate changes. In addition, the MAF sensor design includes a passage to direct flow over the hot-wire element, which does not allow measurements when flow is reversed. As a result, the MAF sensor reading is approximately zero during flow reversal, but the total air flow (based on intake manifold charge density and volumetric efficiency of the head) through the engine confirms that the flow through the RB compressor is indeed reversed during a portion of the cycle.
With the RB compressor operating in deep surge, the turbocharger rotational speeds fluctuate with a 180° phase difference, as shown in Fig. 11a, which is primarily a result of the compressor mass flow rate phasing in Fig. 10. The power absorbed by the compressors is high when the flow rate is elevated and low for decreased and reversed flow, resulting in rotational speed deceleration and acceleration, respectively. In addition, higher frequency speed fluctuations, observable in Figs. 11a and 11b, occur at $f/2 = 43.75$ Hz due to each turbine being directly exposed to the high pressure exhaust blowdown from one bank (half) of the cylinders.

Figure 11: Turbocharger rotational speed with the right-bank compressor operating in deep surge at 1,750 rpm:
(a) time domain and (b) frequency domain.

5. CONCLUSIONS
A methodology was developed to incorporate compression system surge prediction capability into a 1D gas dynamics model of a turbocharged IC engine. The compressor model was first developed for a 1D model of the compression system from a turbocharger gas stand, which is isolated from engine airborne pulsations along with simplified ducting geometry. This compressor model was then incorporated into the AIS ducting of a twin, parallel turbocharged engine model. In addition, extensive effort was devoted to properly capturing the wave dynamics, ensuring that models of major AIS components closely reproduce the TL and volume of the actual hardware. Predictions from the engine model closely match measurements during normal (stable), full-load operation. The engine operation was then modified to force one of the compressors to operate in deep
surge, and the predictions agree very well with the measurements, in terms of engine performance, along with the amplitude and frequency of pressure and turbocharger rotational speed fluctuations. The flexibility of this modeling approach allows it to be adaptable for different compressor and ducting configurations, and engine simulation codes.

REFERENCES