

Simulation of Mild Surge in a Turbocharger Compression System

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ABSTRACT

The behavior of the compression system in turbochargers is studied with a one-dimensional engine simulation code. The system consists of an upstream compressor duct open to ambient, a centrifugal compressor, a downstream compressor duct, a plenum, and a throttle valve exhausting to ambient. The compression system is designed such that surge is the low mass flow rate instability mode, as opposed to stall. The compressor performance is represented through an extrapolated steady-state map. Instead of incorporating a turbine into the model, a drive torque is applied to the turbocharger shaft for simplification. Unsteady compression system mild surge physics is then examined computationally by reducing the throttle valve diameter from a stable operating point. Such an increasing resistance decreases the mass flow rate through the compression system and promotes surge. Mild surge is predicted as the mass flow rate is decreased below the stability limit, with oscillations of mass flow rate and pressure exhibited at the Helmholtz resonance frequency of the compression system. The computational results are shown to be able to reproduce the experimental observations available in the literature.

1. INTRODUCTION

Away from (to the right of) the surge line, the pressure ratio of the compressor increases in a stable mode as the mass flow rate is reduced. As the operating condition approaches the surge line, the pressure rise reaches its maximum value and a further reduction in mass flow causes a sudden change in the flow pattern of the compressor [1], leading to either stall or surge. The stall in centrifugal compressors manifests itself in different modes, including (1) full- and part-span rotating

stall, (2) axisymmetric stall near the inducer tips, and (3) stationary non-axisymmetric stall produced by downstream asymmetry of the volute [1]. These types of stall result in a redistribution of the flow field and a lower pressure ratio as the annulus average mass flow rate is reduced beyond the peak pressure rise. Stalling is often tolerated in centrifugal compressors without a drastic decrease in performance since the majority of pressure rise is attributed to the centrifugal effects which occur in the presence of stall cells and separated flow.

Surge, on the other hand, is the low-flow behavior of greatest concern when operating a centrifugal compressor near the surge line. It is an axisymmetric, 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 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 compression system plenum, A_c is the equivalent cross-sectional compressor duct area (inducer eye area), and L_c is the equivalent length of the duct. One manifestation of surge is the change in noise. 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

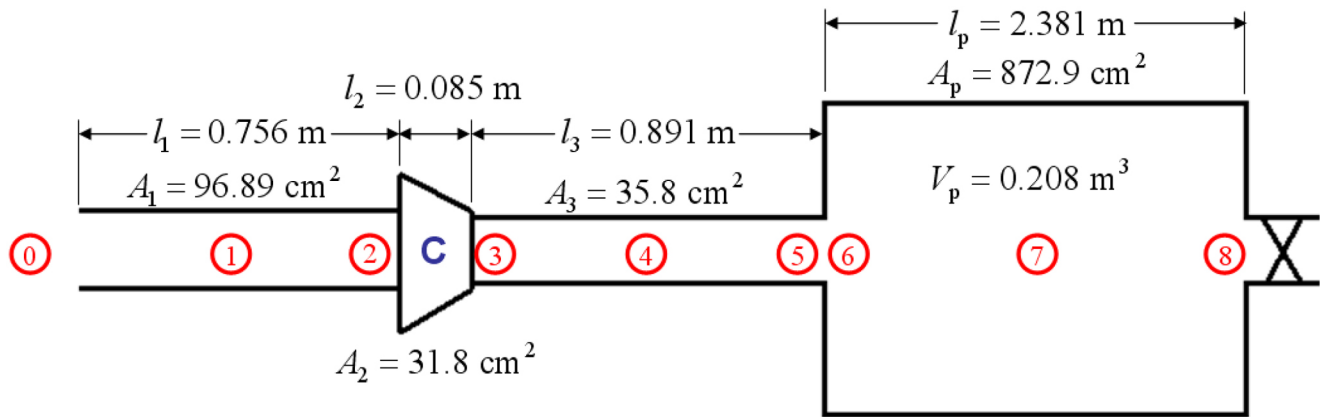


Figure 1. Fink's large B compression system.

by the plenum emptying and filling times and is typically well below the Helmholtz resonance of the compression system.

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 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 used extensively in literature to estimate the operating point for surge inception and the subsequent oscillations. This analysis has revealed a dimensionless number defined by

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

where U is the impeller blade tip speed, hence B is a function of the impeller blade tip Mach number as well as the geometry of the compression system. Above a critical value of B , surge becomes the instability mode encountered, as opposed to stall. The measured stall/surge boundary in Greitzer's work was around $B=0.8$. Hansen *et al.* [5] demonstrated that Greitzer's model could be extended to centrifugal compressors. The approach was further advanced by Fink *et al.* [6] to include the shaft speed dynamics for a surging centrifugal compressor (a high-speed radial impeller with vaneless diffuser surrounded by a volute) as an integral part of the cycle. A method for implementing the foregoing lumped model into a zero-dimensional engine simulation code has been outlined by Theotokatos and Kyratatos [7]. Galindo *et al.* [8] incorporated a surge model into their one-dimensional (1-D) in house gas-dynamic 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.

The objective of this study is to reproduce the unsteady physics of mild surge on a turbocharger bench setup with a 1-D engine simulation code. This investigation focuses on predicting the compressor operating point for instability (mild surge) inception along with the amplitude and frequency of the resulting oscillations. The large B compression system ducting from the turbocharger test bench of Fink *et al.* [2] is modeled using a commercially available engine simulation code GT-Power [9]. A schematic of the compression system geometry is shown in Fig. 1. This system is designed with a large plenum such that $B=2.7$ at the speed of interest (48k rpm) and surge is the low-flow instability mode encountered, as opposed to stall. The compressor map data from both the small and large B systems of Fink *et al.* is combined, extrapolated and interpolated using a preprocessor developed in the present study and input as a text file to the code. The oscillations of the nondimensionalized mass flow rate, pressure ratio, and impeller tip speed from the current study are compared with experimental observations of Fink *et al.* to evaluate the accuracy of predictions. A detailed analysis is then presented for pressure, mass flow rate, and temperature at key locations, providing further insight into the physics throughout the compression system.

Following this introduction, Section 2 illustrates the source of the "Helmholtz resonator" frequency. Section 3 describes the compressor map and Section 4 the 1-D compression system model. Simulation results are then compared with the experimental observations of Fink *et al.* [2] in Section 5. Section 6 demonstrates the influence of the compression system geometry on stability, followed by concluding remarks in Section 7.

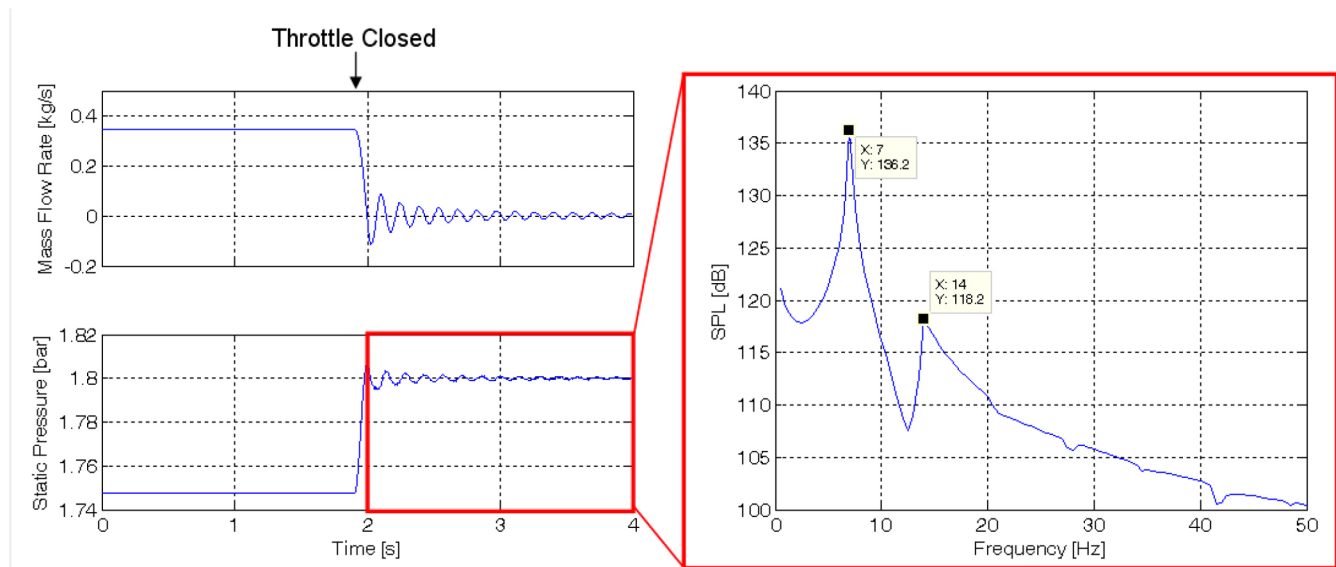


Figure 2. Pressure and mass flow rate oscillations in a 1-D model of Fink's compression system with the compressor removed.

2. INHERENT HELMHOLTZ RESONANCE

A compression system on a turbocharger test bench typically consists of a compressor operating between two circular ducts. The duct upstream of the compressor is open to ambient and the downstream duct exits into a plenum (such as an intercooler for centrifugal and burner for axial compressors) of larger cross-sectional flow area. The flow exiting the plenum then passes through a restriction (such as a throttle or intake valve for centrifugal and turbine for axial compressors) with a flow area considerably smaller than the plenum. This configuration has inspired a lumped parameter approach that models the surge oscillations in a manner analogous to those of a Helmholtz resonator [2,3,10]. 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. As expected, oscillations of pressure and mass flow in the compression system occur at the inherent Helmholtz resonator frequency (even) in the absence of the compressor. This unsteady behavior is demonstrated using a 1-D engine simulation code by replacing the compressor in Fig. 1 with a duct of equivalent length. A flow is generated by applying a pressure gradient across the boundaries. After the flow stabilizes, the throttle valve is fully closed to provide a perturbation leading to oscillations of mass flow and pressure, as shown in Fig. 2. A frequency domain analysis of pressure oscillations shows the dominant frequency to be 7.0 Hz, which is close to the theoretical Helmholtz resonator frequency of 7.3 Hz calculated using Eq. (1).

3. COMPRESSOR

A representation of the compressor, by user provided performance maps, needs to be integrated into the ducting of the air flow system prior to surge predictions. This is accomplished in GT-Power (hereafter referred to as the “code”) by providing performance data in the form of mass flow rate, pressure ratio, and efficiency at constant rotational speeds. This information is either entered into the ‘CompressorMap’ object or referenced as an external text file (“.cmp” file). Both methods use rotational speed and pressure ratio as the inputs to the lookup map, and mass flow rate and efficiency are the outputs. When data is input to the ‘CompressorMap’ object (the first approach), the preprocessor of the code extrapolates the data to choke, zero speed, higher speeds (optional), and reverse flow (optional). Then, the map is interpolated between the user provided data points. The current study employs the alternative method (the second approach) of providing 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. A MATLAB script was also developed in the present study to perform all of the preprocessing and to formulate the compressor information into a text file with the “.cmp” format required by the code. For further information regarding the “.cmp” compressor map format the User's Manual may be referred [11].

The compressor map used in the present study was re-created from Fink's [2,6] small and large B compressor characteristics and Fink's forward flow nondimensional torque Γ_c as a function of flow coefficient ϕ_c as shown in Figs. 3 and 4, respectively. The published compressor map contains data at six constant rotational speeds in the range of 25k to 51k rpm with the large B system data in blue and the small B data in

red, as shown in Fig. 3. The “Large B Surge Line” (hereafter simply referred to as the “surge line”) designates the *deep surge* boundary for the large B compression system shown in Fig. 1. To the right of the surge line, the small and large B data are identical, illustrating 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. The compressor nondimensional torque Γ_c is calculated from map data as

$$\Gamma_c = \frac{\tau_c}{\rho_0 A_c r_2 U^2} = \frac{\dot{m}_c c_p T_{1t}}{\omega \eta_c} \left(\frac{PR_c^{\frac{\gamma-1}{\gamma}} - 1}{\rho_0 A_c r_2 U^2} \right), \quad (3)$$

where τ_c is the torque driving the compressor (Baines [12]), ρ_0 is the density of air at ambient conditions, r_2 is the radius of the impeller tip, c_p is specific heat of air at constant pressure, $\dot{m}_c$ is the mass flow rate through the compressor, T_{1t} is the total inlet temperature, ω is the angular velocity of the impeller, PR_c is the total-to-total pressure ratio of the compressor, and γ is the ratio of specific heats. The compressor flow coefficient ϕ_c represents the mass flow rate non-dimensionalized as

$$\phi_c = \frac{\dot{m}_c}{\rho A_c U}, \quad (4)$$

where ρ is the density at the compressor inlet (for forward flow). The Γ_c vs. ϕ_c data shown in Fig. 4 provides a convenient means for extrapolating compressor isentropic efficiency η_c to zero mass flow rate since the data at all rotational speeds nearly collapses onto a single curve.

The compressor characteristics of Fink are extrapolated here to cover the entire forward flow operating region of the compressor including down to zero speed, up to 60k rpm, to zero flow, and to choke. The pressure ratio of each constant speed line at zero mass flow rate PR_0 is calculated using radial equilibrium theory and assuming an isentropic process [7]

$$PR_0 = C_{PR} \left[1 + \frac{\gamma-1}{2\gamma R T_1} \omega^2 (r_2^2 - r_1^2) \right]^{\frac{\gamma}{\gamma-1}}, \quad (5)$$

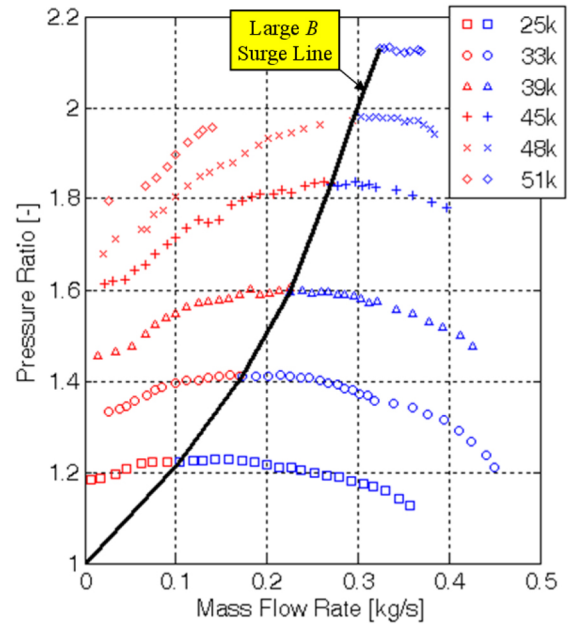


Figure 3. Fink's small and large B compressor characteristics.

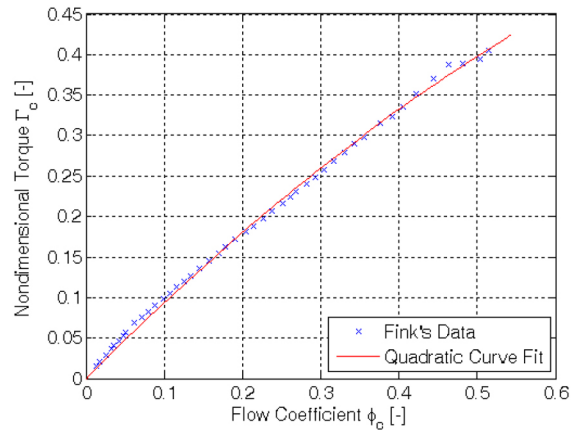


Figure 4. Fink's forward flow nondimensional compressor torque as a function of flow coefficient.

where C_{PR} is a constant multiplier, R is the gas constant for air, and T_1 is the compressor inlet temperature. r_1 is the mean geometric radius of the impeller eye, which divides the impeller eye area into two sections with the same area and is calculated as

$$r_1 = \sqrt{\frac{r_{1,t}^2 + r_{1,h}^2}{2}}, \quad (6)$$

where $r_{1,t}$ and $r_{1,h}$ are the radii of the inducer tip and hub, respectively. For the present study, a C_{PR} value slightly larger than unity was applied to PR_0 in Eq. (5) to match the

experimental data. This multiplier was 1.04 for the lowest speed (25k rpm) and 1.09 for the highest compressor speed (51k rpm).

Since the rare small B system data is available for Fink's compressor, the present study uses higher order polynomials to fit the small B constant speed lines down to zero mass flow rate. This method allows the constant speed lines used in the model to obtain a nearly perfect fit to the experimental data. If the rare small B system data were not available, the Moore and Greitzer [13] method would have been used to extrapolate the compressor characteristics from the peak pressure rise to zero mass flow rate, as described in [Appendix A](#).

The compressor map data must also be extrapolated down to zero speed, to choke, and optionally to higher speeds. Extrapolation to both lower and higher speeds is performed on the nondimensional characteristics. The compressor flow coefficient ϕ_c is calculated using [Eq. \(4\)](#), and the nondimensional compressor isentropic head coefficient involves the nondimensional pressure as

$$\psi_c = \frac{\Pi_c^{(\gamma-1)/\gamma} - 1}{(\gamma-1)Ma_{t,0}^2}, \quad (7)$$

where

$$\Pi_c = \frac{P_{3t}}{P_0}, \quad (8)$$

P_{3t} being the total pressure at the compressor exit and P_0 the pressure at ambient conditions, and

$$Ma_{t,0} = \frac{U}{a_0} \quad (9)$$

is the compressor exit tip Mach number, with a_0 being the speed of sound at ambient conditions. Note that the nondimensional compressor characteristics (ψ_c vs. ϕ_c) nearly collapse onto a single curve, partially removing the speed dependence, as shown in [Fig. 5](#). The extrapolation of map data to lower speeds is achieved by performing a speed weighted linear interpolation between the nondimensional form of the lowest constant speed line (25k rpm) and zero speed, assuming zero mass flow rate, pressure rise, and efficiency when the impeller is not rotating. Extrapolation of the compressor data to higher speeds is completed by transforming the nondimensional data of the highest constant speed line (51k rpm) into dimensional form at higher speeds. The compressor data usually requires extrapolation to choke, but this procedure is neglected here since the range of

operation for the compressor in the current study is not near that region of the map.

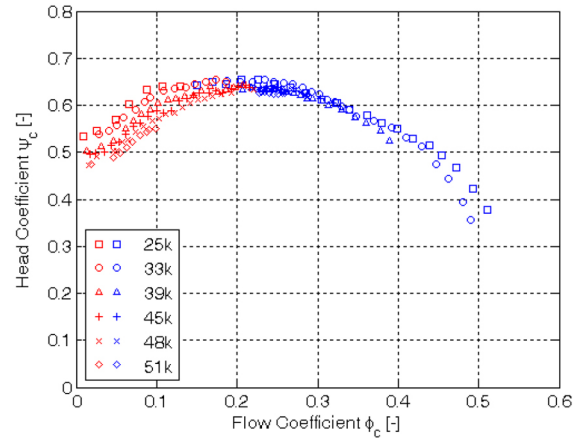


Figure 5. Fink's nondimensional compressor characteristics.

A method similar to Theotokatos and Kyrtatos [7] is used to extrapolate the compressor efficiency down to zero mass flow rate. The nondimensional torque Γ_c vs. flow coefficient ϕ_c data in [Fig. 4](#) is extrapolated by applying a quadratic curve fit in the form

$$\Gamma_c = C_0 + C_1\phi_c + C_2\phi_c^2, \quad (10)$$

with C_0 - C_2 being the fit constants. The extrapolated compressor efficiency is calculated by substituting [Eq. \(10\)](#) into [Eq. \(3\)](#) and using discrete values of $\dot{m}_c$ and PR_c from the extrapolated compressor characteristics. The compressor map usually requires interpolation at intermediate speeds and details regarding the approach used in the current study are included in [Appendix B](#).

The final version of the compressor map used for the code is shown in [Fig. 6](#). The large and small B compression system data of Fink is superimposed on the map along with the corresponding constant speed lines to illustrate the comparison. The MATLAB script writes the compressor map information to a text file with the required ".cmp" format. Since limited data is available for this compressor, the nondimensional torque (used to calculate efficiency) was assumed to be independent of speed. As a result, the compressor efficiency is only a factor of the flow coefficient at speeds above 25k rpm. This results in constant peak efficiency along each constant speed line. If more data were to be available, [Eq. \(10\)](#) could be used to fit the Γ_c vs. ϕ_c data at each constant speed individually and incorporate a speed-dependent efficiency.