SIMULATION of SURGE in TURBOCHARGER COMPRESSION SYSTEMS

THESIS

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The goal of the present study is to predict the stable operating limit of centrifugal compression systems and the transition from mild to deep surge, including discrete sound peaks at low frequencies and their amplitudes at key locations throughout the system. Two different compression system configurations were studied. One configuration incorporated a plenum (large volume), which produced surge as the flow rate was reduced. The other configuration consisted of the smallest possible downstream volume, allowing compressor performance measurements left of the surge line of large volume. A set of small and large volume configurations installed on the turbocharger facility at The Ohio State University (OSU) were studied along with additional experimental results available in the literature (Fink, 1988). A compressor map preprocessor was developed in the present study to extrapolate and interpolate the data from the small volume systems. To facilitate surge predictions, each map was incorporated into a one-dimensional (1-D) model of the corresponding large volume configuration, including an upstream compressor duct, a centrifugal compressor, a downstream compressor duct, a plenum, and a throttle.

The model of Fink’s large volume compression system was used to simulate a mild surge operating point, with the frequency, amplitudes, and time-averaged values of nondimensional flow rate, pressure ratio, and rotational speed nearly reproducing the
corresponding experimental observations from the literature. The frequency of measured and predicted mild surge oscillations occurred at the theoretical Helmholtz resonator frequency. As the flow rate was reduced, the compression system entered deep surge and the frequency decreased below the Helmholtz resonator value. The predictions captured the dominant cycle period and the four distinct phases (quiet, instability growth, blowdown, and recovery) reported in the literature. The model also correctly predicted a decrease in the cycle period and the elimination of the quiet phase as the flow rate was further reduced. During the instability growth phase, the amplitude of the secondary Helmholtz resonator oscillations was smaller in predictions relative to the measurements.

The turbocharger facility at OSU incorporated a variable volume plenum to change the surge frequency. The predicted stable operating limit occurred just left of the peak pressure ratio and was nearly constant for the plenum volumes studied, which agreed closely with the measurements. As the flow rate was reduced, the compression system entered mild surge, with the frequency of simulated and experimental oscillations occurring near the theoretical Helmholtz resonator frequency. With a further reduction in flow rate, the compression system entered deep surge. The dominant deep surge frequencies at full (9.2 L) plenum volume occurred at 63 and 50% of the mild surge frequency for the experiment and simulation, respectively. As the plenum volume was decreased to one-eighth (1.15 L), the dominant deep surge frequencies increased to 82 and 88% of the mild surge frequency for the experiment and simulation, respectively. In agreement with the experimental trends, the predicted deep surge line moved to lower mass flow rates as the plenum volume was decreased, however, the predicted flow rates were larger relative to the measurements. During mild and deep surge, a frequency
domain analysis reveals that the dominant predicted sound pressure levels agree reasonably well with the measurements.
DEDICATION

My family and Teri Laub
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**NOMENCLATURE**

\( a \)  Speed of sound

\( A \)  Cross-sectional area

\( A_s \)  Surface area for heat transfer

\( b \)  Diffuser width (exducer blade height)

\( B \)  Nondimensional Greitzer number

\( c_p \)  Specific heat of air at constant pressure

\( C \)  Compressor pressure rise characteristic

\( C' \)  Local slope of compressor pressure rise characteristic

\( C_0 - C_2 \)  Curve fit constants

\( C_x \)  Axial velocity

\( C_t \)  Multiplier for reverse flow nondimensional torque

\( D \)  Equivalent diameter

\( dx \)  Length of spatial discretization

\( e \)  Total internal energy per unit mass

\( f \)  Frequency; Skin friction coefficient

\( G \)  Throttle pressure drop characteristic

\( G' \)  Local slope of throttle pressure drop characteristic
\( h \)    Specific enthalpy
\( h_c \)    Heat transfer coefficient
\( l \)    Rotational inertia of turbocharger spool
\( K \)    Pressure loss coefficient
\( L, l \)    Length
\( m \)    Mass
\( \dot{m} \)    Mass flow rate
\( Ma_{t,0} \)    Mach number of impeller tip
\( n \)    Number of revolutions
\( N \)    Turbocharger shaft rotational speed in [rev/min]
\( p \)    Pressure
\( P \)    Power
\( PR \)    Pressure ratio (total-to-total)
\( r \)    Radius
\( R \)    Gas constant for air
\( S \)    Source term
\( t \)    Time
\( T \)    Temperature
\( T_{DS} \)    Period of deep surge cycle
\( U \)    Velocity of the impeller tip
\( V \)    Volume
Greek Symbols

\( \gamma \) Ratio of specific heats
\( \Gamma \) Nondimensional torque
\( \Delta \) Change in parameter
\( \eta \) Isentropic efficiency
\( \Pi \) Nondimensional pressure
\( \rho \) Density
\( \tau \) Torque; Time constant
\( \phi \) Flow coefficient
\( \psi \) Isentropic head coefficient
\( \omega \) Angular velocity of the shaft

Subscripts

0 Ambient; Total property; Shutoff (\( \dot{m} = 0 \))
1–8 Location
c Compressor; Convection
C Equivalent compressor duct
cor Corrected
d Drive; Diffuser
deep Deep surge
ED Exit duct
EID Equivalent inlet duct
f  Fluid

GTP  Simulation

h  Hub

H  Helmholtz resonator

ID  Inlet duct

ii  Impeller inlet

im  Impeller

in  Into control volume

m  Mean

max  Maximum

mild  Mild surge

min  Minimum

norm  Normalized

out  Out of control volume

p  Plenum

ref  Reference

rev  Reverse flow

SS  Steady-state

t  Tip; Meridional flow path

T  Throttle

v  Volute

w  Wall
CHAPTER 1
INTRODUCTION

1.1 Background

The low-flow instability in axial and centrifugal compressors has been a challenge of great concern for as long as these turbomachines have existed. During stable operation, a compressor operates on the negatively sloped portion of its characteristic and the pressure ratio increases as the mass flow rate is decreased. 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 (Cumpsty, 1989), leading to either stall or surge. The compression systems of turbocharged internal combustion engines are susceptible to such instabilities while operating particularly at high-load, low-speed conditions along with transients resulting in a sudden reduction in mass flow (tip-out or up-shift). When stall and surge instabilities occur, they may result in degraded performance, unacceptable levels of noise, and mechanical failure.

The stall involves regions of low flow velocity within the compressor, resulting in a lower pressure ratio as the mass flow rate is decreased below the peak pressure rise of the characteristic. The “rotating stall,” for example, involves circumferential mass flow variations as the rotating cell passes through the observation point, while the annulus-averaged mass flow remains constant. The number of stall cells can vary from a single to
numerous cells, and they span radially from the blade tips to as far as the hub. Inside the stall cells, the through-flow velocity is significantly reduced relative to the unstalled flow. Additional forms of stall include axisymmetric stall near the inducer tips and stationary non-axisymmetric stall produced by downstream asymmetry of the volute (Cumpsty, 1989).

The stalling behavior of a compressor can dramatically influence the shape of its characteristics, as shown in Fig. 1.1. The pressure rise of centrifugal compressors is typically continuous, for a fixed rotational speed, with the slope of the pressure rise vs. mass flow characteristic becoming positive due to the presence of a “progressive stall” (Cumpsty, 1989), as shown in Fig. 1.1a. This type of stall results in a slight decrease in performance and can be detected by a change in noise or with fast response pressure transducers. 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. Multi-stage axial compressors may exhibit discontinuous characteristics due to the development of a full-span rotating stall, frequently referred to as “abrupt stall” (Cumpsty, 1989), as shown in Fig. 1.1b. When stall is initiated in this machine, the operating point will move rapidly down the throttle line from the unstalled characteristic to the stalled characteristic, resulting in a substantial decrease in pressure rise and mass flow.
Figure 1.1: Compressor characteristics exhibiting: (a) progressive stall and (b) abrupt stall, from Cumpsty (1989).

The surge is the low-flow behavior of greatest concern when operating a centrifugal compressor at mass flows below the peak pressure rise of a characteristic. It 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 annulus average mass flow oscillates but remains in the forward direction at all times. Such oscillations are dictated by the Helmholtz frequency

$$f_H = \frac{a}{2\pi} \sqrt{\frac{A_C}{V_p L_C}}$$

(1.1)

of the compression system (Fink et al., 1992), where $a$ is the speed of sound, $V_p$ is the plenum volume, $A_C$ is the equivalent cross-sectional area of the compressor duct, and $L_C$ is the equivalent 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. If the mass flow oscillations are severe and the flow reverses its direction during part of the cycle, the compressor has entered “deep surge.”
The dominant frequency is now 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 (Dehner et al., 2011) during deep surge. These represent severe pressure fluctuations within the compression system comparable to those in the exhaust primary runners of production engines.

Surge is a system phenomenon and its occurrence is dependent upon both the local slope of the compressor characteristic and the ducting in which the compressor is installed. A representative compression system consists of a compressor operating between two circular ducts, as shown in Fig. 1.2. 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, as shown in Fig. 1.2.

![Figure 1.2: Representative compression system.](image)

A linear stability analysis (Greitzer, 1976a) may be used to estimate if a compression system operating point will enter surge. Figure 1.3 shows a representative compressor characteristic $C$, where the compressor pressure rise $\Delta p_c = p_3 - p_2$ (pressure
locations given in Fig. 1.2) is a function of the compressor mass flow rate $\dot{m}_c$, and a throttle characteristic $G$, where the throttle pressure drop $\Delta p_T = p_7 - p_8$ is a function of the throttle mass flow rate $\dot{m}_T$. During steady operation, $\dot{m}_c = \dot{m}_T$, and the compressor operates at the intersection of $C$ and $G$. The slope of the compressor pressure rise is defined as

$$C' = \frac{d(\Delta p_c)}{d\dot{m}_c},$$

(1.2)

and the slope of the throttle pressure drop characteristic as

$$G' = \frac{d(\Delta p_T)}{d\dot{m}_T}. $$

(1.3)

Figure 1.3: Representative compressor and throttle characteristics.
According to the linear stability analysis (Greitzer, 1976a), surge oscillations are possible when

\[ C' > \left( \frac{A_c L_c}{V_p} \right) \left( \frac{4a_p^2}{U^2} \right) \left( \frac{1}{G'} \right), \]  

(1.4)

where \( U \) is the impeller tip speed. The geometry of the compression system ducting (\( L_c, A_c, \) and \( V_p \)) is crucial for determining the surge inception point. For example, as \( V_p \to \infty \), the compression system will enter surge since the right-hand-side of Eq. (1.4) approaches zero, thereby ensuring \( C' \to 0 \). The stability limit of such an infinite volume system is denoted by point 1 in Fig. 1.3, which is located at the peak pressure rise with \( C'_1 = 0 \). As \( V_p \) is reduced, the surge inception point shifts to lower mass flows (higher \( C' \)). The stability limit of such an intermediate volume, for example, occurs at point 2, with \( C'_2 > 0 \).

In the limiting case where \( V_p \to 0 \), Eq. (1.4) suggests that the compression system is unconditionally stable. Additionally, increasing the impeller tip speed is predicted to have the same influence on the stability limit as increasing volume.

Prediction of the compressor operating conditions at surge inception and the subsequent oscillations of mass flow rate and pressure have received significant attention over the last few decades. Early prediction methods employed lumped Helmholtz resonator approximations. These approaches provided insight into the coupling of the compressor, ducting, and throttle, demonstrating that the surge is a system phenomenon. Some techniques also attempted to determine the mode of instability (stall or surge) by means of a dimensionless number. Nearly all modern diesel engines and an increasing number of spark ignited engines are equipped with turbochargers, hence the ability to
accurately predict surge during the engine design process remains as one of the key objectives.

1.2 Literature Survey

One of the first thorough distinctions between stall and surge was made by Emmons et al. in 1955. They presented experimental results illustrating the propagation of rotating stall around the inducer of a centrifugal compressor, demonstrating that the cells travel in the same direction as the impeller at a fraction of the rotational speed. They also provided simultaneous unsteady surge measurements at different compression system locations, indicating large amplitude fluctuations of mass flow rate throughout the entire ducting. As the mass flow rate was decreased experimentally, the compressor transitioned from stable operation on the negatively sloped portion of the characteristic to mild surge as the characteristic slope became positive and further throttling resulted in deep surge. Emmons formulated a linear analysis of the system, which predicted surge inception to occur near the peak of the characteristic close to the Helmholtz frequency. Since their approach was a linear analysis, it did not allow for prediction of the large amplitude fluctuations during surge.

A nonlinear lumped parameter model for prediction of surge in axial compression systems was presented by Greitzer in 1976(a). This analytical method was formulated for the typical system shown in Fig. 1.2. The dimensions of the plenum are typically much smaller than the wavelength of an acoustic wave at the surge frequency, hence the static plenum pressure $p_6$ is assumed to be spatially constant ($p_6=p_4=p_5=p_6$) at any instant in time. Four lumped balance equations characterize the compression system physics during
steady and unsteady operation. The momentum balance for the equivalent compressor duct

\[
\frac{d\dot{m}_c}{dt} \left( \frac{L}{A_c} \right) = p_0 - p_p + C ,
\]

(1.5)
governs the mass flow rate through the compressor by means of the pressure forces acting on the boundaries of the compressor duct, where \(p_0\) is the pressure at ambient conditions and \(C\) is the compressor pressure rise \((p_3-p_2)\). Equation (1.5) assumes that the pressures at the inlet and exit of the compressor duct \((p_0 \text{ and } p_p, \text{ respectively})\) act on equal areas \(A_c\), which is not always the case. The length-to-area ratio of the equivalent compressor duct

\[
\left( \frac{L}{A} \right)_c
\]

may be calculated as

\[
\left( \frac{L}{A} \right)_c = \int_0^l \frac{dl}{A(l)} = \left( \frac{L}{A} \right)_{ID} + \left( \frac{L}{A} \right)_c + \left( \frac{L}{A} \right)_{ED} ,
\]

(1.6)
with \(\left( \frac{L}{A} \right)_{ID}, \left( \frac{L}{A} \right)_c\), and \(\left( \frac{L}{A} \right)_{ED}\) being the length-to-area ratios of the inlet duct, compressor, and exit duct, respectively.

A similar momentum balance for the throttle duct

\[
\frac{d\dot{m}_T}{dt} \left( \frac{L}{A_T} \right) = p_p - p_0 - G ,
\]

(1.7)
governs the mass flow rate through the throttle, where \(\left( \frac{L}{A} \right)_T\) is the length-to-area ratio of the throttle duct. Assuming isentropic acceleration of flow in the valve and that the static pressure at the throttle discharge plane is equal to ambient \((p_8=p_0)\), the pressure
drop across the throttle may be estimated as the dynamic pressure at this location
(Greitzer, 1976a)

\[ G = \frac{\rho_0 u_T^2}{2} = \frac{\dot{m}_T^2}{2\rho_0 A_T^2}, \]  

(1.8)

where \( \rho_0 \) is the density at ambient conditions, \( u_T \) is the velocity at the throttle discharge
plane, and \( A_T \) is the throttle flow area.

Mass balance for the plenum

\[ \frac{dp_p}{dt} = \frac{\gamma p_p}{V_p \rho_p} (\dot{m}_c - \dot{m}_T), \]  

(1.9)

governs the plenum pressure, where \( \gamma \) is the ratio of specific heats and \( \rho_p \) is the density in
the plenum. Greitzer assumes that \( \frac{p_p}{\rho_p} = \frac{p_0}{\rho_0} \) (\( T_p = T_0 \)) in Eq. (1.9), which is a
simplification that may be appropriate for some circumstances but deteriorates as the
compressor efficiency decreases and pressure ratio increases.

The final relationship in Greitzer’s formulation accounts for the deviation of the
instantaneous compressor pressure rise \( C \) from the steady-state characteristic \( C_{ss} \) by
means of a first-order transient model

\[ \frac{dC}{dt} = \frac{1}{\tau}(C_{ss} - C), \]  

(1.10)

to simulate the lag in compressor response, where Greitzer’s time lag constant

\[ \tau = \frac{2\pi n}{\omega}, \]  

(1.11)

is proportional to the time required for some number of rotor revolutions \( n \), where \( \omega \) is
the angular velocity of the shaft. A value of \( n=2 \) was selected based on the number of
rotor revolutions required for a stall cell to develop to approximately 63.2% of its final asymptotic value, as shown in Fig. 1.4.

Nondimensionalization of Eqs. (1.5) and (1.9) revealed a dimensionless number

\[ B = \frac{U}{2a} \sqrt{\frac{V_p}{A_c L_c}}, \]  

(1.12)

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 model was applied to a three-stage axial compression system with varying values of \( B \). All of the simulations started with the compressor operating under stable conditions near the peak of the characteristic, followed by a slight closing of the throttle to reduce the mass flow rate and push the compression system beyond its stability limit. For a relatively low value of \( B=0.45 \), the plenum pressure and mass flow...
decrease continuously, as shown in Fig. 1.5, from the initial (green dot) to a new
operating point (red dot) determined by the intersection of the compressor and throttle
characteristics. This compression system exhibits abrupt *rotating stall*, resulting in
substantially reduced pressure rise compared to the initial point. The parameters in Fig.
1.5 are the nondimensional plenum pressure rise

$$\frac{\Delta p}{\frac{1}{2} \rho_0 U^2} = \frac{p_p - p_0}{\frac{1}{2} \rho_0 U^2},$$  \hspace{1cm} (1.13)

and the compressor flow coefficient, which represents the compressor mass flow rate
nondimensionalized as

$$\phi_c = \frac{C_x}{U} = \frac{\dot{m}_c}{\rho_0 A_c U},$$  \hspace{1cm} (1.14)

where $C_x$ is the axial velocity at the compressor inlet.

Figure 1.5: Transient 3-stage axial compression system behavior for $B=0.45$, from Greitzer (1976a).
Figure 1.6 shows the predicted transient response of the same compressor with \( B=0.70 \). This compression system is operating in \textit{mild surge} \( (C_x/U>0) \) with large amplitude oscillations of pressure and mass flow, and the transient response is markedly different than the stall in Fig. 1.5. The stall/surge boundary in Greitzer’s work was measured and predicted to occur at \( B \) values around 0.8 (Greitzer, 1976b) and 0.7 (Greitzer, 1976a), respectively. Such specific values, however, cannot be generalized to all compressors. For example, measurements demonstrating the occurrence of both mild and deep surge in a centrifugal compression system with \( B=0.38 \) were recently presented by Uhlenhake (2011). This study confirmed that reducing \( B \) allowed the compressor to operate free of surge at lower mass flows, supporting the trend predicted by the linear stability analysis in Eq. (1.4). However, a low \( B \) did not guarantee that the compression system would operate surge-free when the compressor mass flow was further decreased.

Figure 1.6: Transient 3-stage axial compression system behavior for \( B=0.70 \), from Greitzer (1976a).
Greitzer’s predicted transient response of the compression system with a larger value of \( B = 1.58 \) is shown in Fig. 1.7. The compression system is now operating in *deep surge* with reversed flow \((C_x/U < 0)\) during a portion of the cycle.

![Image of transient response](image)

**Figure 1.7:** Transient 3-stage axial compression system behavior for \( B = 1.58 \), from Greitzer (1976a).

The model developed by Greitzer was further refined in subsequent studies to extend and improve its predictive capability. Hansen *et al.* (1981) demonstrated that Greitzer’s lumped parameter approach could be extended to centrifugal compressors. Fink (1988) applied a similar model to a centrifugal compressor with a radial impeller and vaneless diffuser surrounded by a volute. He added the shaft speed dynamics, through the angular momentum balance

\[
\frac{d\omega}{dt} = \frac{\tau_d - \tau_c}{I},
\]

(1.15)

to relax the assumption of constant shaft speed during unsteady surge, where \( \tau_d \) is the drive torque, \( \tau_c \) is the compressor torque, and \( I \) is the rotational inertia of the turbospool.
Fink also eliminated the simplification of $T_p = T_0$ used in Eq. (1.9) and assumed that the throttle duct length was short so that the inertia can be ignored, effectively removing Eq. (1.7) from the analysis. The value of the time lag constant used in Eq. (1.10) was selected as

$$\tau = \left| \frac{L_t}{u_t} \right|,$$  

(1.16)

where $L_t$ is the meridional throughflow length of the impeller and vaneless diffuser, and $u_t$ is the meridional average flow velocity. The time lag constant in Eq. (1.16) corresponds roughly to the convection time through the impeller and diffuser. A method for implementing this lumped approach into an engine simulation code has been outlined by Theotokatos and Kyrtatos (2001).

These lumped parameter approaches have the ability to predict the frequency and amplitude of surge oscillations with reasonable accuracy, 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.

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)-(g),
(2) spatially distributed wave dynamics, and
(3) governing equations common with engine simulation codes.

Galindo et al. (2008) incorporated such 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.

1.3 Objective

The objective of this thesis is 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.
Additionally, the ability of predictions to capture the influence of the compression system
geometry on the instability inception point and the surge frequency is investigated. A 1-
D, unsteady, time-domain approach was selected as the computational tool due to the
inherent advantages over lumped formulations.

The geometries of the small and large $B$ compression systems installed on the
turbocharger test bench at The Ohio State University (OSU), along with the steady-state
measured maps are provided in Chapter 2. The small $B$ configuration is shown to expand
the surge-free operating region to much lower mass flows than the large \( B \) system, allowing a wider map to be obtained. The details of an additional set of small and large \( B \) centrifugal compression systems, available in literature (Fink 1988, Fink et al. 1992), are also provided in the same chapter.

The surge physics within the compression system is investigated using a commercially available engine simulation code GT-Power. Models were constructed for both the large \( B \) compression system at Ohio State and that of Fink, and details describing the model components, along with the boundary conditions are presented in Chapter 3. The governing equations are also presented, defining the 1-D, unsteady, compressible fluid flow within the control volumes of the model.

A MATLAB script was developed in the present work to extrapolate and interpolate the steady compressor data from the small \( B \) systems to cover the entire operating region from choke to reverse flow. This methodology and the implementation into the code are discussed in Chapter 3. The current version of the code treats the compressor as a zero-dimensional (0-D) lookup table, so additional ducting was incorporated into the models to represent the geometry of the compressor. This simplified ducting maintains the physical length and volume of the compressor, without significantly increasing computation time.

Mild and deep surge predictions utilizing the 1-D model (developed in the current work) of Fink’s large \( B \) compression system are compared to the measurements of Fink in Chapter 4. The predicted pressures at key compression system locations along with the rotational speed are compared with the corresponding experimental observations to evaluate the accuracy of simulations.
Predictions and the corresponding measurements from Ohio State’s large $B$ compression system at $U=310$ m/s and three different plenum volumes are presented in Chapter 5. These comparisons are made at the boundary between stable and mild surge operation and at the transition from mild to deep surge. The ability of the model to predict the instability inception point and surge frequency dependence on the downstream volume is evaluated. Finally, Chapter 6 includes a summary of the findings.
CHAPTER 2
COMPRESSION SYSTEMS

The turbocharger test benches of Fink (1988) and that at The Ohio State University (OSU) were utilized to explore how system stability is influenced by the ducting installed downstream of the compressor. Steady-state compressor data was collected with these cold-flow facilities in order to create compressor performance maps and define the surge line for each system. Both benches are also equipped with fast response instrumentation capable of measuring the unsteady fluctuations of pressure, rotational speed, and temperature during surge.

2.1 Fink et al. (1988, 1992)

Fink et al. studied a turbocharger for diesel applications. The compressor was radial with 20 main blades (no splitters) and utilized a vaneless diffuser. The impeller had a tip diameter of 12.8 cm and was designed to operate at rotational speeds up to 70,000 rpm. The dominant form of stall in this machine was the inducer tip stall, which was non-rotating and asymmetric due to a circumferential flow distortion caused by the volute. Therefore, rotating stall was found to be an unimportant contributor to surge initiation for this compressor. They presented steady-state compressor maps for two different compression systems, utilizing the same compressor inlet duct and centrifugal compressor. The volume of compressed air downstream of the compressor was varied in
order to study the influence of the $B$ parameter on stability, giving $B$ values of 0.25 and 2.74 at a rotational speed $N=48,000$ rpm (48 krpm) for the small and large $B$ systems, respectively.

### 2.1.1 Large $B$ System

The large $B$ compression system of Fink (1988) operated with a compressor exit duct connecting the volute exit to a plenum with a volume of 208 L, as shown in Fig. 2.1. The flow then exited the plenum through the throttle duct, which contained the throttle valve.

![Diagram of Large $B$ compression system](image)

**Figure 2.1: Large $B$ compression system of Fink (1988).**

This system was designed with a large volume of compressed air, such that surge would be the low-flow instability mode. A steady-state compressor map was obtained with this compressor at six constant rotational speeds of 25, 33, 39, 45, 48, and 51 krpm, as shown in Fig. 2.2. The total-to-total compressor pressure ratio $PR_c$ is defined as
\[ PR_c = \frac{p_{03}}{p_{02}}, \quad (2.1) \]

where \( p_{02} \) and \( p_{03} \) are the total pressures at the compressor inlet and exit, respectively. All of these constant speed lines are limited at low-flows by the “Large B Surge Line,” which designates the deep surge limit of this compression system. The location of the surge line is near the peak pressure ratio of the compressor, which is the trend predicted by the linear stability analysis of Eq. (1.4) for a large plenum volume. The 25 and 33 krpm speed lines of the compressor nearly extend to choke \((PR_c=1)\), while the power limitation of their facility restricted the maximum obtainable \( \dot{m}_c \) at higher speeds.

Figure 2.2: Large B compressor characteristics from Fink (1988).

The theoretical Helmholtz resonance frequency of Fink’s large B system may be calculated using Eq. (1.1) as
where the speed of sound is based on the mean plenum temperature of 387 K (compressor operating at 48 krpm near the surge line). Therefore, mild surge oscillations are expected to occur with a fundamental frequency near 7.3 Hz. A 1-D model of Fink’s large B system was created in this thesis to compare mild and deep surge predictions with the experimental results of Fink in Chapter 4.

### 2.1.2 Small B System

The small B compression system of Fink (1988) was designed with the throttle valve near the volute exit, to minimize the volume of compressed air, as shown in Fig. 2.3. The small B system has a compressed volume of 1.4 L, which is approximately 0.67% of that in the large B arrangement.

![Small B compression system of Fink (1988).](image)

This system was designed to operate free of surge at mass flow rates to the left of the large B surge line. The compressor characteristics of the small and large B systems are shown in Fig. 2.4 as the red and blue symbols, respectively. To the right of the large B
surge line, the characteristics are identical, indicating 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).

Figure 2.4: Small and large $B$ compressor characteristics of Fink (1988).

The compressor characteristics $PR_c$ vs. $\dot{m}_c$ may, alternatively, be represented as the “nondimensional characteristics” $\psi_c$ vs. $\phi_c$, as shown in Fig. 2.5. The compressor flow coefficient $\phi_c$ is the nondimensional mass flow rate given in Eq. (1.14), and the compressor isentropic head coefficient

$$\psi_c = \frac{\Pi_c^{(\gamma-1)/\gamma} - 1}{(\gamma - 1) Ma_t^{2}}.$$  

(2.2)
is related to the compressor nondimensional pressure

$$\Pi_c = \frac{p_{03}}{p_0},$$  \hspace{1cm} (2.3)\

where $p_{03}$ is the total pressure at the compressor exit and

$$Ma_{t,0} = \frac{U}{a_0}$$  \hspace{1cm} (2.4)\

is the compressor exit tip Mach number, with $a_0$ being the speed of sound at ambient conditions. Note that the compressor characteristics at all six rotational speeds nearly collapse onto a single curve when plotted as $\psi_c$ vs. $\phi_c$, partially removing the speed dependence, as shown in Fig. 2.5. The compressor nondimensional torque

$$\Gamma_c = \frac{\tau_c}{\rho_0 A_c r_2 U^2} = \frac{\dot{m}_c c_p T_{02}}{\rho_0 A_c r_2 U^2} \left( \frac{\gamma - 1}{PR_c^\gamma - 1} \right),$$  \hspace{1cm} (2.5)\

may be calculated from map data, where $r_2$ is the radius of the impeller tip, $c_p$ is specific heat of air at constant pressure, $\eta_c$ is the total-to-total isentropic efficiency of the compressor, and $T_{02}$ is the total inlet temperature. The data at different rotational speeds nearly collapses onto a single curve when plotted as $\Gamma_c$ vs. $\phi_c$, as shown in Fig. 2.5.
Figure 2.5: Nondimensional compressor parameters for the small and large $B$ systems of Fink (1992).

The small $B$ compressor map data in Figs. 2.4 and 2.5 is utilized in Chapter 3 to create a map that covers the entire forward and reverse flow regions of operation. This map is then used within the 1-D model to predict the unsteady compression system behavior in Chapter 4.

2.2 Turbocharger Facility at The Ohio State University

Two different compression systems with a small and large $B$ are studied here experimentally, where $B=0.18$ and 0.63 (at $U=310$ m/s), respectively. 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 ducting immediately downstream of the compressor is the difference between the two configurations. The large $B$ system (Fig. 2.6) includes a plenum before the flow reaches the pneumatically controlled throttle valve, while the small $B$ system (Fig. 2.7) does not. The valve provides a flow restriction which creates a pressure drop from the compressor delivery 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 define 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 (Uhlenhake, 2010).
Figure 2.6: Large $B$ compression system at OSU.

Figure 2.7: Small $B$ compression system at OSU.
2.2.1 Large B System

The large B system has a duct at the compressor exit that flows into a plenum of larger cross-sectional area along with an additional duct connecting the plenum to the control valve, as shown in Fig. 2.6. 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 adjusted in discrete increments to allow 1/8 (1.15 L), 1/2 (4.6 L), and the full (9.2 L) plenum volume.

Compressor performance maps are created for the full plenum volume large B system at three constant, corrected rotational speeds of $U_{cor} = 230$, 310, and 370 m/s, in the form of $PR_c$ vs. $\dot{m}_{c,cor}$ in Fig. 2.8 and $\eta_c$ vs. $\dot{m}_{c,cor}$ in Fig. 2.9. Note that the location of the surge line is near the peak pressure ratio of the compressor in Fig. 2.8. The impeller tip speed $U$ is corrected for total compressor inlet temperature $T_{02}$ (SAE J922, 1995) as

$$U_{cor} = \frac{U}{\sqrt{T_{02}/T_{ref}}} ,$$  \hspace{1cm} (2.6)

where $T_{ref} = 298$ K the standard reference temperature. The compressor mass flow rate $\dot{m}_c$ is corrected for $T_{02}$ and total compressor inlet pressure $p_{02}$ (SAE J922, 1995) as

$$\dot{m}_{c,cor} = \dot{m}_c \frac{\sqrt{T_{02}/T_{ref}}}{p_{02}/p_{ref}} .$$  \hspace{1cm} (2.7)

where $p_{ref} = 100$ kPa the standard reference pressure.
Figure 2.8: Full plenum volume large $B$ compressor characteristics of the BW 1880 DCF.

Figure 2.9: Full plenum volume large $B$ compressor efficiency for BW 1880 DCF.
The geometry of the large $B$ compression system components, with the plenum piston at the mid length (half volume) of the housing is given in Table 2.1. 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 2.1: Geometry of the large $B$ compression system at OSU with half plenum volume.

<table>
<thead>
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<td>9.372</td>
<td>403.1</td>
<td></td>
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<td>5.562</td>
<td>7.808</td>
<td>71.23</td>
<td></td>
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<td>0.1030</td>
</tr>
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<td>598.6</td>
<td>0.2979</td>
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<tr>
<td>Plenum</td>
<td>42.86</td>
<td>107.8</td>
<td>39.76</td>
<td>4.620</td>
</tr>
<tr>
<td>Plenum Exit Duct</td>
<td>25.40</td>
<td>15.47</td>
<td>164.2</td>
<td>0.3929</td>
</tr>
<tr>
<td>Total</td>
<td>2437</td>
<td>5.414</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

For example, the theoretical Helmholtz resonance frequency $f_{H}$ of this compression system at half plenum volume and $U=310$ m/s is then calculated using a modified version of Eq. (1.1) and the geometry in Table 2.1 as

$$f_{H} = \frac{a}{2\pi} \sqrt{\frac{A}{VL}} = \frac{379 \text{ m/s}}{2\pi} \frac{1}{\sqrt{(0.005414 \text{ m}^3)(2437 \text{ m}^{-1})}} = 16.6 \text{ Hz}, \quad (2.8)$$

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. While Eqs. (2.8) and (1.1) are identical in form, they include different definitions for $V$ and $\frac{L}{A}$. The
5.414 L from Table 2.1 is the volume of compressed air in the system (from diffuser to control valve), whereas Eq. (1.1) includes only the contribution from the plenum. The 2437 m$^{-1}$ from Table 2.1 is the length-to-area ratio of the compression system (from inlet duct to control valve), while Eq. (1.1) includes only the compressor and its upstream and downstream ducts. Therefore, when operating at half plenum volume and 310 m/s, mild surge oscillations are expected to occur with a fundamental frequency near 16.6 Hz. One-dimensional models of the Ohio State large $B$ system with full, 1/2, and 1/8 plenum volumes were created in this thesis to compare mild and deep surge predictions with the experimental results in Chapter 5.

### 2.2.2 Small $B$ System

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. 2.7. This system was designed to operate free of surge at mass flow rates to the left of the large $B$ surge line. Consistent with Fink’s earlier observations, the compressor characteristics and efficiency of the small and large $B$ systems here appear to be identical over the range that steady large $B$ data was obtainable, as shown in Figs. 2.10 and 2.11, respectively.
Figure 2.10: Small and large $B$ compressor characteristics for BW 1880 DCF.

Figure 2.11: Small and large $B$ compressor efficiency for BW 1880 DCF.
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.
CHAPTER 3
COMPUTATIONAL PREDICTION METHOD

A commercially available 1-D, unsteady, time-domain code (GT-Power, 2009) was utilized to create models of the large $B$ compression experimental setup at OSU and the large $B$ system of Fink. A MATLAB script was also developed in the present work to extrapolate and interpolate the steady-state small $B$ compressor data, and put it into the format required by the code.

3.1 Engine Simulation Code

The numerics to treat the balance equations for unsteady, compressible fluid flows have been incorporated into 1-D engine simulation codes since the late 1970’s. The commercial code used here for the current study solves the non-linear balance equations of mass, momentum and energy, 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 1-D mass conservation is represented by

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

where $m$ is the mass of the discrete volume, and $\dot{\bar{m}}$ is the mass flow rate across the control volume boundaries. The 1-D momentum conservation may be expressed as
\[
\frac{dm}{dt} dx = Adp + \sum_{in} (\dot{m}u) - \sum_{out} (\dot{m}u) - 4f \frac{\rho u |u|}{2} \frac{Adx}{D} - K \left( \frac{1}{2} \rho u |u| \right) A ,
\]

where \( p \) is the 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 (pressure) loss coefficient. The 1-D energy conservation is expressed as

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

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, \( T_f \) is the fluid temperature, and \( T_w \) is the wall temperature. The solutions of Eqs. (3.1)-(3.3) in combination with the equation of state yield the scalar fluid properties within each control volume (pressure, temperature, density, internal energy, and enthalpy) and the vector properties at the boundaries (mass flow rate and velocity). Additionally, the code has the capability to solve for combustion species, which is a feature that is unnecessary in the present models.

### 3.2 Ducting

The 1-D model of the ducting installed on the large \( B \) compression system at OSU (recall Fig. 2.6) is discussed in this section, while a model of the large \( B \) system of Fink (Fig. 2.1) is constructed in a similar fashion. The inlet boundary condition to the compression system is defined as a constant pressure, temperature, and composition, which is set to the ambient state of air in the test chamber. The compressor inlet duct is
connected to the ambient boundary condition using a bellmouth, as shown in Fig. 3.1, with both forward and reverse discharge coefficients set equal to 1.

Figure 3.1: One-dimensional model of the large $B$ compression system at OSU.

The inlet duct is modeled as a straight pipe with a constant, circular cross-section, and its downstream end is connected to the compressor. The compressor model consists of the actuator disk, which provides the compressor performance by means of a lookup table, and two additional ducts representing the equivalent geometry of the compressor, as will be further elaborated in Section 3.3. The downstream end of the compressor is connected to a plenum by a straight exit duct with constant circular cross-section. The end of the plenum where both the compressor exit and plenum exit ducts are connected is modeled using a “flowsplit” component, as shown in Fig. 3.1. This flowsplit represents the first control volume of the plenum and its surface area and volume are defined by the plenum diameter and the discretization length $dx$, as shown in Fig. 3.2. The remainder of the plenum is modeled as a pipe of length $L_p \cdot dx$, with the closed end representing the piston surface.
Figure 3.2: Plenum geometry from the large B compression system at OSU.

The plenum exit duct connects the plenum to the control valve, which provides a pressure drop and represents the end of the compression system. This valve is modeled as a circular orifice and the diameter is adjusted to change the compressor operating point.

The ducting downstream of the valve consists of the orifice flow meter and the exhaust piping leading out of the hemi-anechoic test chamber. The ducting and sharp-edged orifice plate of the flow meter are modeled in order to incorporate the flow restriction, with the discharge coefficient calculated from the geometry of the mating ducts and the diameter of the orifice. The flow rate through the orifice is calculated from the discharge coefficient, flow area, and the fluid properties in the adjacent control volumes. The ducting leading the flow out of the hemi-anechoic chamber is included in the model to incorporate the pressure drop due to wall friction and flow separation. Finally, the system exhausts to an ambient identical to that at the inlet.

The rotational inertia of the turbospool is input to the turbo shaft component of the model, which is connected to both the turbine and compressor objects, as shown in Fig. 3.1. The compressor actuator disk determines the amount of torque absorbed at the
current operating point, and this value is applied to the turbo shaft. Instead of incorporating a turbine into the model, a drive torque is applied to the turbo shaft for simplification. The rotational inertia and torques are utilized by an angular momentum balance (recall Eq. 1.15) to determine the change in shaft speed.

3.3 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 the code by providing performance data in the form of corrected mass flow rate, pressure ratio, and efficiency at constant, corrected rotational speeds. This information is either entered into the ‘CompressorMap’ (actuator disk) object or referenced as an external text file (“.cmp” file). Both methods use corrected rotational speed and compressor pressure ratio as the inputs to the lookup map, where $PR_c$ is calculated from the pressures in the adjacent up- and downstream control volumes of the actuator disk. The outputs from the lookup table are the compressor corrected mass flow rate and isentropic efficiency. The actual compressor mass flow is calculated from the corrected mass flow rate along with $T_{02}$ and $p_{02}$ using Eq. (2.7), and it is imposed as a flow boundary on the volumes adjacent to the actuator disk. 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 format required by the code. For further information regarding the “.cmp” compressor map format the User’s Manual may be referred (Gamma, 2009).

The actuator disk also imposes an enthalpy change to the flow passing through it. During forward flow, the total enthalpy $h_{03}$ of the flow exiting the compressor is calculated as

$$h_{03} = h_{02} + \frac{c_p T_{02}}{\eta_c} \left( \frac{\gamma - 1}{PR_c^{\gamma} - 1} \right) ,$$

where $h_{02}$ is the total enthalpy of air at the compressor inlet. When the flow reverses, the air in the compressor exit duct expands through the compressor and enters the compressor inlet duct. The code assumes that this process is isenthalpic

$$h_{02} = h_{03} .$$

An alternative approach for implementing nonzero enthalpy change is discussed in Section 3.3.3.

The torque absorbed by the compressor

$$\tau_c = \frac{\dot{m} (h_{03} - h_{02})}{\omega} ,$$

is calculated from the enthalpy change. Since the code assumes that the enthalpy change is zero during flow reversal, zero compressor torque is imposed, and this aspect is also addressed in Section 3.3.3. The change in angular velocity $d\omega$ of the turbo shaft is calculated at each time step by utilizing the angular momentum balance of Eq. (1.15), the net shaft torque $(\tau_d - \tau_c)$, and the rotational inertia $I$. 

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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 1-D 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. 3.1. A series of calculations specifies the geometry of the equivalent ducts to preserve both the length-to-area ratio and volume of the original components. These properties influence both the Helmholtz frequency of the compression system and the surge stability limit.

The length-to-area ratio \( \left( \frac{L}{A} \right)_{\text{EID}} \) of the equivalent inlet duct (impeller inlet and impeller) is calculated from the geometry of the individual compressor components as

\[
\left( \frac{L}{A} \right)_{\text{EID}} = \int_{0}^{L_{\text{EID}}} \frac{dl}{A(l)} = \left( \frac{L}{A} \right)_{\text{ii}} + \left( \frac{L}{A} \right)_{\text{im}},
\]

where the subscripts ii and im denote the impeller inlet and impeller, respectively. The impeller inlet is the space in the compressor housing between the end of the compressor inlet duct and the impeller, as shown in Fig. 3.3.
The impeller inlet is a straight duct with constant cross-sectional area $A_{ii}$ and length $L_{ii}$, hence the equivalent length-to-area ratio of the impeller inlet $\left(\frac{L}{A}\right)_{ii}$ is calculated as

$$\left(\frac{L}{A}\right)_{ii} = \frac{L_{ii}}{A_{ii}}.$$  \hspace{1cm} (3.8)

Next, we will estimate the equivalent length-to-area ratio of the impeller. The annular flow (eye) area at the inlet of the impeller $A_{im,1}$ is calculated as

$$A_{im,1} = \pi \left( r_{t,1}^2 - r_{h,1}^2 \right),$$  \hspace{1cm} (3.9)
where \( r_{t,1} \) and \( r_{h,1} \) are the radii of the inducer tip and hub, respectively. The flow area at the impeller exit \( A_{im,2} \) is calculated as

\[
A_{im,2} = 2\pi r_{t,2}b ,
\]

(3.10)

where \( r_{t,2} \) is the impeller exit tip radius and \( b \) is the blade height at the impeller exit (diffuser width). The variation of impeller flow area with meridional length is a complex function that is not easily obtainable. In order to simplify the calculation, it is assumed that the impeller has circular cross-sectional area that varies linearly with meridional length \( l \) (a converging circular duct)

\[
A_{im}(l) = A_{im,1} + \frac{l}{L_{im}}(A_{im,2} - A_{im,1}) ,
\]

(3.11)

where \( L_{im} \) is the full length of the impeller mean flow path in the meridional plane. This flow path (a) starts at the mean geometric radius \( r_{m,1} \) of the impeller eye at the inducer, calculated as

\[
r_{m,1} = \sqrt{\frac{r_{t,1}^2 + r_{h,1}^2}{2}} ,
\]

(3.12)

which divides the flow area into two equal sections, and (b) ends at the mean \((b/2)\) exducer blade height. The equivalent length-to-area ratio of the impeller \( \left( \frac{L}{A} \right)_{im} \) may now be calculated as

\[
\left( \frac{L}{A} \right)_{im} = \int_0^{L_{im}} \frac{dl}{A_{im}(l)} = \int_0^{L_{im}} \frac{dl}{A_{im,1} + \frac{l}{L_{im}}(A_{im,2} - A_{im,1})} .
\]

(3.13)

The volume of the equivalent inlet duct

\[
V_{EID} = V_{ii} + V_{im} = L_{ii}A_{ii} + \int_0^{L_{im}} A_{im}(l) \cdot dl ,
\]

(3.14)
should be conserved through this geometry simplification, where \( V_{ii} \) and \( V_{im} \) are the volumes of the impeller inlet and impeller, respectively. For a straight, constant, circular cross-sectional area duct representing the combined impeller inlet and impeller geometry, the equivalent length-to-area ratio is defined as

\[
\left( \frac{L}{A} \right)_{EID} = \frac{L_{EID}}{A_{EID}},
\]

and the volume as

\[
V_{EID} = L_{EID}A_{EID}.
\]

The set of Eqs. (3.15) and (3.16) allows for calculation of the geometry (\( L_{EID} \) and \( A_{EID} \)) of the equivalent inlet duct.

The geometry of the equivalent exit duct is calculated using a similar procedure. The diffuser is treated as diverging duct with inlet area \( A_{d,1} \) equal to the impeller exit area \( (A_{d,1}=A_{im,2}) \) and exit area \( A_{d,2} \) equal to the area at the exit of the vaneless diffuser, as shown in Fig. 3.3. The volute is split into two parts, as shown in Fig. 3.4. The curved scroll portion is approximated by assuming the volute has a cross-sectional area \( A_{v,1} \) that varies linearly with length \( L_{v,1} \) (a diverging circular duct). The section at the volute exit is treated as a straight, circular duct with length \( L_{v,2} \) and constant cross-section area \( A_{v,2} \).
The equivalent inlet and exit ducts maintain the length-to-area ratio and volume of the compressor. These are key parameters, as the length influences wave propagation and the volume determines the mass storage within the components.

3.3.1 Compressor Map

The compressor maps used for simulations were created from experimental data using a MATLAB preprocessor that was developed in the present study. The performance map of the BorgWarner 1880 DCF compressor (installed on the turbocharger bench at OSU) is discussed in this section, while a map of the compressor utilized by Fink (1988) is created in a similar fashion. 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 $\eta_c$ vs. $\dot{m}_{c,cor}$, as shown in Figs. 2.10 and 2.11, respectively. The
compressor characteristics in Fig. 2.10 are transformed into nondimensional form using Eqs. (1.14) and (2.2), as shown in Fig. 3.5. These nondimensional characteristics are extrapolated to zero flow coefficient (mass flow rate) by forcing the fits to pass through the isentropic head coefficient at zero flow coefficient $\psi_0$, which is calculated from radial equilibrium theory assuming an isentropic process

$$\psi_0 = \frac{1 - \left( \frac{r_{m,1}}{r_{i,2}} \right)^2}{2}.$$  \hspace{1cm} (3.17)

Note that Eq. (3.17) is only a function of the impeller geometry, so the model fits to the nondimensional data at all three speeds converge to a single point at zero flow coefficient, as shown in Fig. 3.5. The nondimensional compressor characteristics are also extrapolated to choke using curve fits to the experimental data, where the choked flow coefficient $\phi_{c,\text{choke}}$ for each speed line occurs where $\psi_c=0$. In order to predict deep surge, the compressor characteristics must be extrapolated to cover the reverse flow region. Since the steady-state reverse flow characteristics cannot be measured with the current experimental setup, a quadratic relationship, similar to that in (Theotokatos and Kyrtatos, 2001), is utilized.
Figure 3.5: Extrapolated nondimensional BW 1880 DCF compressor characteristics.

The fits to the nondimensional characteristics are transformed to the more familiar form of $PR_c$ vs. $m_{c,\text{cor}}$, as shown in Fig. 3.6.
The compressor nondimensional torque $\Gamma_c$ is calculated using Eq. (2.5) and the measured data in Figs. 2.10 and 2.11. When the experimental data is represented as $\Gamma_c$ vs. $\phi_c$, as shown in Fig. 3.7, a quadratic fit can be applied at each constant rotational speed in the form

$$\Gamma_c = C_0 + C_1 \phi_c + C_2 \phi_c^2,$$

with $C_0$-$C_2$ being the fit constants.
Figure 3.7: BW 1880 DCF compressor nondimensional torque as a function of flow coefficient.

For a particular rotational speed, the compressor efficiency is extrapolated to zero flow coefficient (mass flow rate) by using the $\Gamma_c$ vs. $\phi_c$ fit over the range bounded by $\phi_c=0$ and $\phi_c$ at the maximum efficiency $\eta_{c,max}$ (for the given speed). Equation (3.18) is substituted into Eq. (2.5) and discrete values of $\dot{m}_c$ and $PR_c$ are utilized from the extrapolated characteristics. During flow reversal, the compressor nondimensional torque

$$\Gamma_{c,rev} = C_\Gamma \left( C_0 - C_1 \phi_c + C_2 \phi_c^2 \right), \tag{3.19}$$

is estimated from the forward flow fits as a fraction of the mirror image over $\phi_c=0$, where $C_\Gamma$ is the nondimensional torque multiplier for reverse flow. This relationship between the forward and reverse flow nondimensional torque is utilized since the steady-state reverse flow characteristics cannot be measured with the current experimental setup. For
varying values of $C_T$, the predicted compressor inlet and exit temperatures during deep surge are compared with measurements in Section 5.4, to illustrate the influence of $C_T$ and the process used for selecting a value of 0.16 for the current compressor. The efficiency is also extrapolated to choke by fitting the $\eta_c$ vs. $\phi_c$ data at flow coefficients greater than $\phi_c$ at $\eta_{c,\text{max}}$ and forcing the efficiency to zero at the choked flow coefficient, as shown in Figs. 3.8 and 3.5, respectively.

![Figure 3.8: BW 1880 DCF compressor isentropic efficiency as a function of flow coefficient.](image)

The compressor data was interpolated at intermediate map speeds using the following methodology (Dehner et al., 2010):
1. Each of the extrapolated constant speed line fits in the $PR_c$ vs. $\dot{m}_{c,\text{cor}}$ space are evaluated with an equal number of points, as shown in Fig. 3.9. These points are distributed at the same normalized pressure ratio

$$PR_{\text{norm}} = \frac{PR_c - PR_{c,\text{min}}}{PR_{c,\text{max}} - PR_{c,\text{min}}}$$

(3.20)

for each constant speed line, where $PR_{c,\text{min}}$ and $PR_{c,\text{max}}$ are the minimum and maximum pressure ratios on a given speed line, respectively.

![Figure 3.9: Extrapolated constant speed lines for BW 1880 DCF compressor with points at equal values of normalized $PR_c$.](image)

2. The points from two adjacent constant speed lines are nondimensionalized, as shown in Fig. 3.10.
3. A speed weighted linear interpolation

\[
\phi_c = \left( \frac{N_{c,2} - N_c}{N_{c,2} - N_{c,1}} \right) \phi_{c,1} + \left( \frac{N_c - N_{c,1}}{N_{c,2} - N_{c,1}} \right) \phi_{c,2},
\]

is performed between corresponding points on adjacent speed lines and at the same normalized pressure ratio, where the nondimensional characteristic ($\psi_c$ vs. $\phi_c$) at the intermediate speed $N_c$ is calculated from those at a lower ($N_{c,1}$) and higher ($N_{c,2}$) speeds, as shown in Fig. 3.11. The head coefficient and efficiency are also interpolated using the corresponding version of Eq. (3.21).
4. The nondimensional compressor characteristic at the lowest measured rotational speed \((U=230 \text{ m/s})\) is extrapolated to zero speed by utilizing Eq. (3.21) and assuming that \(\phi_c=\psi_c=\eta_c=0\) when \(U=0\). The extrapolated and interpolated characteristics are dimensionalized as shown in Fig. 3.12.
Figure 3.12: Extrapolated and interpolated constant speed lines for the BorgWarner 1880 DCF compressor.

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, as shown in Fig. 3.13. The compressor power

$$P_c = \frac{m_c c_p T_{02}}{\eta_c} \left( \frac{\gamma-1}{PR_c^{\gamma}} - 1 \right),$$

(3.22)
is then calculated at each extrapolated and interpolated map point in order to cover the entire operating range, as shown in Fig. 3.14.
Figure 3.13: Final BW 1880 DCF compressor map used for the model.

Figure 3.14: BW 1880 DCF compressor power.
3.3.2 Reverse Flow Implementation

The current implementation of the code (GT-Power V7.0 Build 1) imposes constant enthalpy expansion through the compressor during flow reversal, hence zero compressor power. In order to incorporate nonzero compressor power during flow reversal into the code, the current study applies a power to the turbocharger shaft. The magnitude of the applied power is determined as a function of the corrected mass flow rate and pressure ratio by means of a lookup table utilizing data from the reverse flow portion of Fig. 3.14. In order to satisfy energy conservation, the energy which is transferred from the impeller to the air during reverse flow is accounted for as a source term $S_{\text{rev}}$ which is applied to the energy conservation equation for the first control volume in the compressor equivalent exit duct neighboring the ‘Compressor’ as

$$
\frac{d (me)}{dt} = p \frac{dV}{dt} + \sum_{\text{in}} (\dot{mh}) - \sum_{\text{out}} (\dot{mh}) - h_c A_s (T_T - T_w) + S_{\text{rev}}, \quad (3.23)
$$

The source term is only applied when the compressor flow is reversed and the magnitude is equal to the power which is applied to the turbocharger shaft.

3.3.3 Model Settings

Among the options GT-Power offers for a time constant to dampen changes in compressor mass flow rate, the present study has chosen

$$
\tau_{\text{GTP}} = \frac{n}{\omega}. \quad (3.24)
$$

This time constant is proportional to the time required for a number of rotor revolutions $n$, and a value of $n=2$ is used in the current study. If the damping is not present, a small change in pressure ratio at the peak of the characteristic can cause an unrealistically large
change in mass flow rate in a single time-step. This damping also allows the compressor to deviate from the steady-state map during unsteady surge cycles.

The turbocharger rotational inertia is often artificially manipulated by use of a multiplier to speed up convergence of steady-state simulations. It is important that the inertia multiplier is maintained equal to unity throughout simulations near the surge line, as unsteady surge oscillations may be encountered. This is especially important for turbocharged engine models where pressure pulsations in the exhaust and induction systems cause speed fluctuations.

The simulations in the present study are stopped when the cycle-to-cycle change in mass flow rate amplitude is extremely small (less than $1 \times 10^{-5}$ kg/s here) to ensure that the compression system has either converged to a stable operating point or a repeating surge cycle is fully developed.

### 3.4 Discretization

The spatial discretization length of the compression system models was defined as $dx=40$ mm, requiring roughly 15 minutes of computational time to complete a simulation. The fundamental frequency of a surge cycle is on the order of 10 Hz, hence this spacing provides approximately 900 cells per acoustic wavelength $\lambda$. The numerical solution involves Taylor series expansions to approximate the differential governing equations. When higher order terms are ignored, numerical dissipation and dispersion occur and the error is related to $dx/\lambda$ (Dickey et al., 2003). Theoretically, a smaller grid size ($dx$) will allow for reduced errors and more accurate numerical solutions. In order to ensure that the selected discretization length ($dx=40$ mm) is sufficiently small, identical mild surge simulations are compared for the large $B$ system of Fink using $dx$ values of 40 and 4 mm.
The predicted pressures for the two discretization lengths are compared at the mid length of the compressor inlet duct, compressor inlet, compressor exit, and mid length of the plenum, corresponding to locations 1, 2, 3, and 5 in Fig. 1.2, respectively. The predicted pressures are essentially identical at all locations, as shown in Fig. 3.15.

Figure 3.15: Comparison of 40 and 4 mm discretization lengths for a mild surge prediction with a model of the large B compression system of Fink (1988).

When the frequency analyses of the predicted pressures at the mid length of the plenum are compared for $dx$ values of 40 and 4 mm, the sound pressure level (SPL) peaks are nearly identical, as shown in Fig. 3.16. The fundamental frequency is predicted to occur at 7.3 Hz for both $dx$ values, with a SPL increase of 1 dB for the 4 mm discretization length relative to the 40 mm.
Figure 3.16: Predicted SPL at the mid length of the plenum (Location 5) for 40 and 4 mm discretization lengths.

A $dx$ of 40 mm proves to be sufficient for minimizing numerical error in the frequency range of interest, considering that the computation time increased by a factor of roughly 100 when $dx$ is decreased by a factor of 10. The 4 mm discretization length has ten times the number of control volumes as the 40 mm length, resulting in an increase in computational time by a factor of approximately ten. Additionally, the explicit time integrator restricts the maximum time step $dt$ to ensure numerical stability by satisfying the Courant condition

$$a \frac{dt}{dx} \leq 0.8 .$$

(3.25)
When $dx$ is decreased by a factor of ten, Eq. (3.25) requires a reduction in the maximum $dt$ by the same factor, increasing the computational time by an additional order of magnitude.
CHAPTER 4

COMPARISON of PREDICTIONS with MEASUREMENTS from the LARGE B COMPRESSION SYSTEM of FINK (1988, 1992)

The simulations in the present section are aimed at reproducing the experimental observations of Fink’s (1988) large B compression system. The surge oscillations are presented at a time averaged rotational speed of 48 krpm \((Ma_{t,0}=0.92)\) in terms of the nondimensional parameters: \(\phi_c, Ma_{t,0}\), and plenum head coefficient

\[
\psi_p = \frac{\Pi_p^{(\gamma-1)/\gamma} - 1}{(\gamma-1)Ma_{t,0}^2}, \tag{4.1}
\]

where \(\Pi_p\) is the ratio of total plenum pressure to ambient pressure. All simulations are started at a stable operating point near the surge line at 48 krpm. Then, the flow area of the throttle valve at the exit of the compression system is closed slightly to reduce the mass flow rate through the compressor, therefore promoting surge.

The present simulations of the large B compression system of Fink (1988) utilize a map (.cmp’ input file) that was created from Fink’s small B data. The preprocessor developed in the current work was used to extrapolate and interpolate the experimental data using the methodology presented in Section 3.3.1, leading to Fig. 4.1. The \(\Gamma_c\) vs. \(\phi_c\) data (recall Fig. 2.5) of Fink (1988) was assumed to be independent of rotational speed,
and as a result, the efficiency is independent of speed between 25 and 51 krpm. The power absorbed by the compressor is shown in Fig. 4.2.

Figure 4.1: Final version of Fink’s (1988) compressor map used for the model.

Figure 4.2: Power absorbed by the compressor of Fink (1988).
The turbine torque is not presented by Fink (1988) during his mild and deep surge measurements. As a result, the drive torque for the present predictions was adjusted to obtain the desired time-averaged rotational speed. The selected drive torque was held constant for the duration of the simulations, just as it was experimentally, such that it does not influence the speed fluctuations. Additionally, Fink (1988) did not provide the flow area and discharge coefficients for the throttle valve at the exit of his compressions system. Therefore, an orifice plate was used to model the valve and the diameter (flow area) was adjusted to obtain the desired mass flow rate. Although the diameter of the throttle valve changes for each simulation, it is maintained constant throughout each individual simulation in the same manner as Fink’s throttle position.

4.1 Inherent Helmholtz Resonance

One-dimensional engine simulation codes have the ability to predict the Helmholtz resonator type oscillations that are observed during mild surge. 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 the model of Fink’s system (recall Fig. 2.1) with a duct of equivalent length. A flow is generated by applying a pressure gradient across the compression system 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. 4.3. 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. (1.1).

Figure 4.3: Pressure and mass flow rate oscillations in a 1-D model of Fink’s compression system with the compressor removed.

4.2 Mild Surge

Fink et al. (1992) present experimental mild surge observations at a single time averaged compressor operating point, as shown in Fig. 4.4a. The $\phi_c$ data fluctuates with amplitude 0.046 (20% of the mean flow) about a time mean value of 0.23, and the dominant frequency of oscillations is reported as 7.3 Hz.
Figure 4.4: Mild surge results: (a) Experimental data of Fink (1992) and (b) Simulation from the present study.
The compressor operating conditions on the surge line at 48 krpm are $\dot{m}_c = 0.30$ kg/s ($\phi_c = 0.225$) and $PR_c = 1.98$ ($\psi_c = 0.637$), and this marks the deep surge boundary of Fink’s experimental results. For the simulations performed here, low amplitude mild surge oscillations appear when the stability limit of the compression system is crossed and grow to the converged results presented in Fig. 4.4b for comparison with the experimental results of Fink. For ease of such comparisons, Fig. 4.4b has retained both the horizontal and vertical axis scales the same as in Fig. 4.4a. The simulation is stopped when the cycle-to-cycle change in mass flow rate amplitude is extremely small (less than $1 \times 10^{-5}$ kg/s here) to ensure that the compression system does not enter deep surge. The oscillations of $\phi_c$ are predicted to occur at a mean value of 0.22 with an amplitude of 0.046 (21% of the mean flow) and appear to be nearly identical to the results of Fink. Similarly, the $Ma_{t,0}$ and $\psi_p$ predictions almost exactly reproduce the amplitude, frequency, and the time averaged operating point of the corresponding experimental results. These comparisons demonstrate the ability of nonlinear 1-D time-domain approaches to predict the mild compressor surge. Frequency-domain analyses of the computed $\phi_c$ and $\psi_p$ oscillations are shown in Fig. 4.5. The dominant fundamental frequency is predicted to be 7.3 Hz, which is identical to the measured result reported by Fink. The somewhat steepened wave forms in Fig. 4.4b lead to additional frequency content at harmonics of the fundamental frequency, which are also observed experimentally on the large $B$ compression system at OSU, as will be discussed in Chapter 5.
Figure 4.5: Frequency domain analysis of $\phi_c$ and $\psi_p$ from mild surge simulation result of Fink’s large $B$ system.

The operating points of the compressor and plenum during a single mild surge cycle are shown on the compressor map in Fig. 4.6. Pressure ratios for both the compressor ($PR_c$) and plenum ($PR_p$) are defined relative to the total pressure at the compressor inlet. Mild surge operating points for the compressor ($PR_c$ vs. $\dot{m}_c$ in green symbols) oscillate in the clockwise direction on the map and the plenum ($PR_p$ vs. $\dot{m}_c$ in blue symbols) operates along a curved path. At Point 1 of the compressor surge cycle, the operating point is nearly on the peak of the 48 krpm characteristic. As $\dot{m}_c$ and $PR_c$ decrease, the operating points follow the constant speed line and the surge line is crossed at Point 2. The mass flow rate and pressure ratio reach a minimum at Point 3, and the compressor speed starts increasing. The compressor flow then accelerates, back across the surge line, until the peak pressure ratio of the cycle is reached at Point 4 and the
maximum flow rate at Point 5. Then, the speed, mass flow rate, and pressure ratio decrease to Point 1 and the cycle is repeated. The compressor and plenum operating points have uniformly spaced time intervals of 0.5 ms, making, for example, the rate of change of $\dot{m}_c$ and $PR_c$ clearly visible for the compressor. Note that the discrete operating points are spaced further apart where the compressor mass flow is accelerating (between Points 3 and 5), indicating that the compressor moves through this portion of the cycle at a rate faster than when the mass flow rate is decelerating (between Points 5 and 3).

During mild surge, the compressor spends a significant amount of time to the left of the surge line. In order to predict mild surge, compressor performance data is required in this region of the map where steady-state measurements are not obtainable with the large $B$ system. This data must either be collected using a system with a smaller $B$ number (as accomplished by Fink) or estimated through extrapolation. The pressure ratio of the plenum (blue symbols) is lower than that of the compressor because of the pressure losses in the intermediate ducting. The shape of the plenum surge cycles is essentially driven by the pressure wave at the Helmholtz resonator frequency.
Figure 4.6: Predicted mild surge compressor and plenum operating points for a single cycle of Fink’s large $B$ system. Selected constant speed lines used in the present study and Fink’s surge line are shown for reference.
Figure 4.7 shows the nondimensional compressor characteristic used in the present model with the data of Fink. Note that the flow coefficient $\phi_c=0.225$ on the surge line at 48 krpm. Mild surge operating points in the compressor ($\psi_c$ vs. $\phi_c$ in green symbols) and plenum ($\psi_p$ vs. $\phi_c$ in blue symbols) oscillate in a counter-clockwise direction on the nondimensional compressor map. The numbered compressor operating “points” in Fig. 4.7 are maintained identical to those in Fig. 4.6. The change of compressor and plenum fluctuation shapes from dimensional (Fig. 4.6) to nondimensional (Fig. 4.7) maps is partially due to the different definitions of nondimensional pressure and pressure ratio. $\Pi$ is relative to a constant reference pressure and is used for calculating $\psi_c$ and $\psi_p$ in Eqs. (2.2) and (4.1), respectively, while $PR$ is relative to the fluctuating compressor inlet pressure. The other factor contributing to the shape difference is the fluctuating rotational speed, which appears as $U$ in the denominator of Eq. (1.14) and as $Ma_{t,0}$ in the denominator of Eqs. (2.2) and (4.1). The rotational speed is larger when the compressor mass flow rate is increasing (from Point 3 to 5 in Fig. 4.6) than when the mass flow rate is decreasing (from Point 5 to 3 in Fig. 4.6). The higher rotational speed while the flow rate increases (from Point 3 to 5 in Fig. 4.7) reduces $\psi_c$ relative to the period when the speed is lower and flow rate decreases (from Point 5 to 3 in Fig. 4.7). The compressor operating points nearly follow the nondimensional characteristic with decreasing flow, however, they operate below the steady-state map with increasing flow. The nondimensional plenum operating points follow an elliptical path that appears to be nearly symmetrical about the surge line.
To gain further insight into the physics of mild surge, the predicted static pressure, temperature, and mass flow rate are given at four locations within the compression system in Figs. 4.8-4.10, respectively. Locations 1-3 and 5 are specified in Fig. 1.2, corresponding to the mid-length of the compressor inlet duct, compressor inlet, compressor exit, and mid-length of the plenum, respectively. Figure 4.8 illustrates that the pressure fluctuations are most severe at the compressor inlet with an amplitude of approximately 2.2 kPa and least severe at the mid-length of the compressor inlet duct (among the locations studied) with an amplitude of 0.25 kPa. A frequency domain analysis of the pressure traces reveals that the maximum SPL of approximately 160 dB occurs at the compressor inlet, corresponding to the dominant mild surge frequency (7.3 Hz), as shown in Fig. 4.11. The harmonics of the Helmholtz resonances appear throughout the compression system due to non-linearity. It is expected that those harmonics above 20 Hz would become audible. The temperature fluctuations are largest
at the compressor exit during mild surge, as shown in Fig. 4.9, but the amplitude is relatively small at approximately 2.5 K. Mass flow rate fluctuations are severe throughout the compression system, as shown in Fig. 4.10. The large amplitude mass flow rate fluctuations at Locations 1-3 are nearly identical, and the amplitude in the plenum is lower due to smaller oscillations in velocity as a result of the much larger cross-sectional area of the plenum. The amplitude of mass flow rate oscillations at the compressor is approximately 21% of the mean flow rate. Turbocharger rotational speed is shown to fluctuate with an amplitude of 175 rpm, and this is only 0.4% of the mean value, as shown in Fig. 4.12. The compressor efficiency fluctuates between the peak value of 73% and about 67% in the present study, as shown in Fig. 4.13.

![Figure 4.8: Predicted static pressure at compression system locations during mild surge with the large B system of Fink.](image)
Figure 4.9: Predicted temperature at compression system locations during mild surge with the large B system of Fink.

Figure 4.10: Predicted mass flow rate at compression system locations during mild surge with the large B system of Fink.
Figure 4.11: Predicted SPL at compression system locations during mild surge with the large $B$ system of Fink.

Figure 4.12: Predicted shaft rotational speed during mild surge with the large $B$ system of Fink.
4.3 Influence of Compression System Geometry

The compression system in Fig. 2.1 has a plenum length of 2.38 m and a volume of 0.208 m$^3$, resulting in $B=2.74$ (at 48 krpm) and an instability inception point near the peak pressure rise of the compressor. As $B$ decreases, the compression system can operate free of surge at increasingly positive values of compressor characteristic slope (lower mass flow rates). For illustrative purposes, the volume of the plenum in Fig. 2.1 is gradually reduced here to determine a critical $B$ value below which the current compression system operating point does not enter surge with the engine simulation code. The operating conditions here are identical to those of Section 4.2, including compressor drive torque, valve diameter, boundary conditions, and the geometry other than the plenum length. A set of runs was performed by sweeping the value of $B$. The predictions reveal that reducing the plenum volume to 4.8% (or, below) of that in Fig. 2.1 by
essentially decreasing the plenum length to 0.115 m, hence $B$ to 0.70 (or, below) eliminates the surge for this specific operating condition. Note that stability is not guaranteed if the mass flow is further reduced. The stable operating point of the $B=0.70$ compression system is shown by the green dot in Fig. 4.14. Comparison of Figs. 4.6 and 4.14 demonstrates that a compressor operating condition that has originally resulted in mild surge oscillations can be stabilized by adequately reducing the volume of the plenum (therefore $B$). The predictions properly capture how the location of surge line is sensitive to the plenum volume.

Figure 4.14: Simulated stable operating point of Fink’s (1992) compressor with a decreased plenum length and $B=0.70$. 
4.4 Deep Surge

Starting at the mild surge operating point in Section 4.2, a slight reduction in flow rate produced deep surge with a cycle period of $T_{DS}=3.6$ s, as shown in Fig. 4.15 (Fink, 1988). The flow coefficient in Fig. 4.15 is based on velocity measurements from a hotwire anemometer, which is insensitive to flow direction. In order to distinguish between forward and reverse flow, Fink applied a mass balance to the plenum volume

$$\dot{m}_c = V_p \frac{d\rho_p}{dt} + \dot{m}_T,$$

(4.1)

to estimate the mass flow rate and therefore flow coefficient. The flow exiting the plenum ($\dot{m}_T$) was assumed to be choked in Eq. (4.1). Assuming the plenum expansion and compression are isentropic processes, the time derivative of plenum density in Eq. (4.1) was replaced with

$$\frac{d\rho_p}{dt} = \frac{\rho_p}{\gamma \rho_p} \frac{dp_p}{dt}.$$

(4.2)

Fink (1988) applied a moving average to “smooth” the measured plenum pressure before numerically differentiating the term in Eq. (4.2). Then, Eq. (1.14) was used to calculate $\phi_c$ from the estimated $\dot{m}_c$, as shown in Fig. 4.15. During the quiet and instability growth phases, the estimated compressor flow coefficient from the plenum mass balance is nearly identical to that obtained from the hotwire velocity measurement, as shown in Fig. 4.15. However, the maximum hotwire measurement during the recovery phase is lower than the plenum mass balance estimation. The flow coefficient derived from the plenum mass balance in Fig. 4.15 confirms that the flow is indeed reversed during the blowdown phase. A simulated $\phi_c$ with $T_{DS}=3.6$ s is presented in Fig. 4.16 and exhibits a
reasonable agreement with the experimental results in Fig. 4.15. Since the flow direction is clear in the simulations, the mass balance of Eq. (4.1) need not be applied to the plenum in order to estimate the mass flow rate (flow coefficient), which also eliminates the “smoothing” that was applied to the measured plenum pressure. The minimum $\phi_c$ is predicted as -0.158, which appears to agree closely with that estimated by applying the plenum mass balance to measurements. During the recovery phase, the maximum $\phi_c$ is predicted to reach 0.383, which lies between the hotwire and plenum mass balance values from the measurement.

The experimental observations of Fink (1988) demonstrated that $T_{DS}$ decreased and the flat “quiet” period in Fig. 4.15 was eliminated as the throttle flow area (rate) was further reduced. He presented time-resolved fluctuations of $\phi_c$, $\psi_p$, and $Ma_{t0}$ at three deep surge operating points with $T_{DS}$=3.0, 1.24, and 0.70 s. The 1-D model will be used next to simulate each of these operating conditions for comparison with experimental observations.
Figure 4.15: Compressor flow coefficient calculated from the hotwire velocity measurement and plenum mass balance with $T_{DS}=3.6$ s, from Fink (1988).

Figure 4.16: Predicted compressor flow coefficient with $T_{DS}=3.6$ s.
4.4.1 Deep Surge with $T_{DS}=3.0$ s

Starting at the $T_{DS}=3.6$ s operating point (Fig. 4.15), a slight decrease in throttle area initiates deep surge with $T_{DS}=3.0$ s and a time-averaged $\phi_c=0.225$, as shown in Fig. 4.17. There are four regions of flow in the 3.0 s deep surge cycle. The “quiet” region in $\phi_c$ occurs between Points 1 and 2, where $Ma_{t0}$ is increasing and $\phi_c$ is decreasing slightly. During this phase, the compressor operating point is moving left ($\phi_c$ is decreasing) on the nondimensional map and approaching the surge line. After the compressor stability limit is crossed at Point 2, the compressor enters the “instability growth” phase and operates in mild surge with a frequency of approximately 7 Hz. The mild surge oscillations are also observable (to a lesser extent) in the $Ma_{t0}$ and $\psi_p$ traces. The mild surge fluctuations continue to grow until the beginning of the “blowdown” phase at Point 3, where the flow reverses. Once again, the flow coefficient in Fig. 4.17 is based on velocity measurements from the hotwire anemometer, but a corresponding plenum mass balance analysis is not provided by Fink. Since the flow is reversed during the blowdown period and the velocity measurement was transparent to the flow direction, the maximum $\phi_c$ during this phase at Point 4 in fact represents the minimum value. During this period, $Ma_{t0}$ increases at a faster rate due to the lower reverse flow compressor power, and $\psi_p$ decreases rapidly due to flow exiting the plenum through both the compressor and the throttle valve. At Point 5, the flow accelerates rapidly back to the forward direction during the “recovery” period, then $\phi_c$ reaches a maximum value at Point 6. Next, the compressor operating point moves up and to the left on the map towards the quiet period (Point 1) and the cycle is repeated. The corresponding simulation results from the present study are presented in Fig. 4.18.
with the horizontal and vertical axis maintained equal to those in Fig. 4.17 for ease of comparison. The numbering convention used to separate the flow phases in Fig. 4.18 is identical to that in Fig. 4.17, with the time scale of each set of numbers adjusted to fit the corresponding flow periods.
Figure 4.17: Deep surge experimental data of Fink (1988) with $T_{DS}=3.0$ s.
Figure 4.18: Deep surge prediction with $T_{DS}=3.0$ s.
A decrease in the throttle diameter by 0.12 mm (0.84% flow area reduction) is the sole difference between the predictions in Figs. 4.16 and 4.18, resulting in a decrease in $T_{DS}$ from 3.6 to 3.0 s. The predicted nondimensional compressor parameters in Fig. 4.18 compare closely with the measurements of Fink (1988) in Fig. 4.17. The simulation contains the four phases (quiet, instability growth, blowdown, and recovery) of compressor flow coefficient behavior as observed experimentally. However, the length of the instability growth period (between Points 2 and 3) and the maximum amplitude of the secondary Helmholtz resonator oscillations are smaller in predictions relative to the measurements. The predicted $T_{DS}$ reproduces 3.0 s of the experiment. The simulated $Ma_{t0}$ and $\psi_p$ also appear to nearly replicate the corresponding experimental observations.

The time-resolved nondimensional compressor operating points from the experimental observations of Fink (recall Fig. 4.17) along with the time-averaged small $B$ data are shown in Fig. 4.19, where the numbered operating points correspond to those in Fig. 4.17. The compressor flow coefficient $\phi_c$ was estimated from the plenum mass balance, as illustrated in Fig. 4.15. The $\psi_c$ in Fig. 4.19 is estimated from $\Pi_p$ by applying a momentum balance

$$\Pi_c = \Pi_p + \frac{1}{\rho_0} \left( \frac{L}{A_C} \right) \frac{d m_c}{dt}, \tag{4.3}$$

to the compressor exit duct in order to account for the inertia. The nondimensional compressor pressure $\Pi_c$ obtained from Eq. (4.3) is used to estimate the compressor head coefficient $\psi_c$ utilizing Eq. (2.2). The corresponding predicted operating points (recall Fig. 4.18) for both the compressor and plenum along with the time-averaged small $B$ data of Fink are shown in Fig. 4.20, where the numbered operating points coincide with those
in Fig. 4.18. The mass flow rate and pressure are spatially distributed in the predictions, allowing $\phi_c$ to be calculated by substituting the compressor mass flow rate in Eq. (1.14) and $\psi_c$ to be calculated by inserting the compressor exit pressure in Eq. (2.2). The operating points in Figs. 4.19 and 4.20 have uniformly spaced time intervals of 3.2 ms, making, for example, the rate of change of $\phi_c$ and $\psi$ clearly visible. During the beginning of the blowdown phase (between Points 3 and 4), the simulated transition from forward to reverse flow appears to occur at a faster rate than the measurement. Due to the extremely fast transitions between the forward and reverse flow characteristics (from Point 3 to 4) in Fig. 4.20, the predicted plenum head coefficient is nearly unchanged during these times. The predicted nondimensional compressor operating points in Fig. 4.20 agree reasonably well with the estimation from the measurements of Fink (1988) in Fig. 4.19.
Figure 4.19: Experimental time-resolved large $B$ compressor data with $T_{DS}=3.0$ s and small $B$ time-averaged data, from Fink (1992).

Figure 4.20: Predicted time-resolved large $B$ nondimensional compressor and plenum operating points with $T_{DS}=3.0$ s along with small $B$ data of Fink (1992).
The predicted \((PR \text{ vs. } \dot{m}_{c,\text{cor}})\) compressor and plenum operating points are shown in Fig. 4.21 along with the small \(B\) compressor characteristics of Fink. The uniform 3.2 ms interval between operating points has been maintained to make the rate of change clearly visible. Majority of the cycle is spent on the steady-state forward (Point 6 to 2) and reverse (Point 4 to 5) flow characteristics, while operating away from the characteristics during a portion of the instability growth phase (recall the mild surge prediction in Fig. 4.6) and the fast transitions from forward to reverse flow (Point 3 to 4) and reverse to forward flow (Point 5 to 6). The four phases of operation are clearly observable in Fig. 4.21. The quiet period (from Point 1 to 2) begins at the peak of the 45 krpm \((Ma_{\infty}=0.86)\) characteristic and continues until the rotational speed approaches 48 krpm \((Ma_{\infty}=0.92)\), which is also observable in Fig. 4.18. The compressor moves through this region of the deep surge cycle at a slow rate relative to the other phases. Instability growth begins at Point 2 as the speed approaches 48 krpm with mild surge oscillations similar to that in Fig. 4.6. During the blowdown phase, the flow quickly decelerates from Point 3 until it reaches a minimum value of -0.24 kg/s at Point 4. The pressure ratio decreases and the flow rate increases as the compressor operating point moves from Point 4 to 5 along the reverse flow characteristic. At Point 5, the flow rate quickly transitions back to the forward direction, reaching the maximum value of 0.56 kg/s at Point 6. The pressure ratio is nearly unchanged during this transition because the compressor moves through this region at such a fast rate. At Point 6, the predicted power absorbed by the compressor is nearly 47 kW, causing the rotational speed to decrease as the operating point moves towards the beginning of the quiet period (Point 1). The predicted
compressor speed fluctuates between 45.1 and 49.4 krpm with a peak-to-peak amplitude of 4.3 krpm (9.1\% of the mean).

The difference \((p_{ce} - p_p)\) between the predicted pressures at the compressor exit \(p_{ce}\) and in the plenum \(p_p\), as shown in Fig. 4.22 (corresponding to \(t=3.4-4.4\) s in Fig. 4.18), represents the net pressure force acting on the air in the compressor exit duct. The green vertical lines in Figs. 4.22 and 4.23 indicate the maximum mass flow rates during the instability growth (left line) and recovery (right line) phases along with the minimum flow rate in the blowdown (middle line) period. Between Points 3 and 4 in Fig. 4.22, the large negative spike of \(p_{ce}-p_p\) provides a (negative) force acting to decelerate the flow until it reaches the minimum value at Point 4, as shown in Fig. 4.23. Immediately
following Point 5 in Fig. 4.23, the flow is then accelerated due to the net (positive) force created by the large (positive) spike of $p_{cc} - p_p$, as shown in Fig. 4.22.
Figure 4.22: Predicted $p_{cc}-p_p$ with $T_{DS}=3.0$ s.

Figure 4.23: Predicted, corrected compressor mass flow rate with $T_{DS}=3.0$ s.
4.4.2 Deep Surge with $T_{DS}=1.24$ s

As the throttle at the exit of the compression system is further closed, the experimental observations of Fink (1988) demonstrate that $T_{DS}$ decreases from 3.0 s to approximately 1.24 s, as shown in Figs. 4.17 and 4.24, respectively. Comparison of Figs. 4.17 and 4.24 reveals that the quiet phase of Fig. 4.17 (from Point 1 to 2) of the deep surge cycles is eliminated when the time-averaged $\phi_c$ is reduced experimentally from 0.225 to 0.19 of Fig. 4.24. During the instability growth phase in Fig. 4.24, the deep surge cycles undergo either 5 or 6 Helmholtz resonator cycles. The two complete deep surge cycles in Fig. 4.24 have periods of approximately 1.30 and 1.18 s, with the difference roughly equal to the Helmholtz resonance period.

As the throttle area is reduced by approximately 6.0% in the simulation, $T_{DS}$ also decreases to 1.22 s, as shown in Fig. 4.25, which agrees closely with the measured 1.24 s. In Figs. 4.25-4.29, the same numbering convention is used for the predicted compressor operating points, allowing the phases of flow to be observed in different spaces. The predicted $\phi_c$, $Ma_{t_0}$, and $\psi_p$ appear to nearly reproduce the measurements during the dominant blowdown and recovery phases. However, a short quiet period (Point 1 to 2) precedes the predicted instability growth phase (Point 2 to 3), which is not observed in the measurements. The predicted amplitudes of Helmholtz resonator oscillations are also lower than the corresponding measurements. Overall, the simulation is capable of predicting the period and amplitudes during the dominant blowdown (Point 3 to 5) and recovery (Point 5 to 1) portions of the deep surge cycles with reasonable accuracy.
Figure 4.24: Deep surge experimental data of Fink (1988) with $T_{DS}=1.24$ s.
Figure 4.25: Deep surge prediction with $T_{DS}=1.22$ s.
The predicted compressor and plenum nondimensional operating points with a deep surge cycle period of 1.22 s are shown in Fig. 4.26, which are nearly identical to those of the 3.0 s period in Fig. 4.20. The numbered compressor operating points in Fig. 4.26 correspond to those in Fig. 4.25. Decreasing the throttle diameter also reduces the time-averaged \( \phi_c \) during the instability growth phase, as shown by comparison of Point 2 in Figs. 4.20 and 4.26. This trend is consistent with the corresponding observations of Fink (1988), where the time-averaged \( \phi_c \) decreased from 0.225 to 0.19 with a decrease in the throttle flow area.

![Figure 4.26: Predicted time-resolved large B nondimensional compressor and plenum operating points with \( T_{DS}=1.22 \) s along with small B data of Fink (1992).](image)

The predicted \((PR \ vs. \ \dot{m}_{c,cor})\) compressor and plenum operating points along with the small B compressor characteristics of Fink are shown in Fig. 4.27. Once again, majority of the cycle is spent on the steady-state forward (from Point 6 to 2) and reverse
(from Point 4 to 5) flow characteristics. The compressor operates away from the characteristics during a portion of the instability growth phase (Point 2 to 3) and the small fraction of time required to transition from forward to reverse flow (Point 3 to 4) and reverse to forward flow (Point 5 to 6). The four phases of operation are clearly observable in Fig. 4.27. The quiet period (Point 1 to 2) of the 1.22 s deep surge cycle is shorter in duration and occurs at lower pressure ratios and rotational speeds than that of the 3.0 s period in Fig. 4.21. Instability growth begins (Point 2) as the speed approaches 45 krpm and continues until it reaches 46 krpm (Point 3). Next, the flow quickly reverses in the blowdown phase until it reaches a minimum value of -0.23 kg/s at Point 4. As the compressor operating point moves along the reverse flow characteristic to Point 5, the pressure ratio decreases and the flow rate increases. Then, the flow rate quickly transitions back to the forward direction, reaching the maximum value of 0.52 kg/s at Point 6. At Point 6, the predicted power absorbed by the compressor is nearly 41 kW, causing the rotational speed to decrease as the operating point moves towards the beginning of the quiet period (Point 1). The predicted compressor speed fluctuates between 43.9 and 47.5 krpm with a peak-to-peak amplitude of 3.7 krpm (8.1% of the mean).
The difference \( p_{ce} - p_p \) between the predicted pressures at the compressor exit \( p_{ce} \) and in the plenum \( p_p \), as shown in Fig. 4.28 (corresponding to \( t=1.4-2.6 \) s in Fig. 4.25), again represents the net pressure force acting on the air in the compressor exit duct. The green vertical lines in Figs. 4.28 and 4.29 indicate the maximum mass flow rates during the instability growth (left line) and recovery (right line) phases along with the minimum flow rate in the blowdown (middle line) period. Between Points 3 and 4 in Fig. 4.28, the large negative spike of \( p_{ce} - p_p \) provides a (negative) force acting to decelerate the flow until it reaches the minimum value at Point 4, as shown in Fig. 4.29. Immediately following Point 5 in Fig. 4.29, the flow is then accelerated due to the net positive force created by the large (positive) spike of \( p_{ce} - p_p \), as shown in Fig. 4.28.
Figure 4.28: Predicted $p_{cc} - p_p$ with $T_{DS}=1.22$ s.

Figure 4.29: Predicted, corrected compressor mass flow rate with $T_{DS}=1.22$ s.
4.4.3 Deep Surge with $T_{DS}=0.70$ s

Experimental observations of Fink (1988) demonstrated that an even further decrease in throttle area reduced $T_{DS}$ to approximately 0.70 s, as shown in Fig. 4.30. The number of Helmholtz resonator periods during the instability growth phase is one or two (varies cycle-to-cycle), and the time-averaged $\phi_c$ is reported as 0.16 (Fink 1988). Starting at the predicted $T_{DS}=1.22$ s case of the preceding section, an 8.2% reduction in throttle area is accompanied by a decrease to $T_{DS}=0.68$ s, which agrees closely with the measurement. A quiet period does not occur in the prediction or the measurement, as shown in Figs. 4.30 and 4.31, respectively. The simulated number of Helmholtz resonator oscillations, during the instability growth period, is one or two, which reproduces the experimental observation of Fink (1988), while the predicted amplitudes of the Helmholtz resonator oscillations are lower than the corresponding measurements. Overall, the simulation is capable of predicting the deep surge cycles with reasonable accuracy.
Figure 4.30: Deep surge experimental data of Fink (1988) with $T_{DS}=0.70$ s.
Figure 4.31: Deep surge prediction with $T_{DS}=0.68$ s.
The predicted compressor and plenum nondimensional operating points with a deep surge cycle period of 0.68 s are shown in Fig. 4.32, which are nearly identical to those with $T_{DS}=3.0$ and 1.22 s periods in Figs. 4.20 and 4.26, respectively. Figures 4.31-4.35 use the same numbering convention for the predicted compressor operating points, allowing the phases of flow to be observed in different spaces.

![Figure 4.32: Predicted time-resolved large $B$ nondimensional compressor and plenum operating points with $T_{DS}=0.68$ s along with small $B$ data of Fink (1992).](image)

The predicted ($PR$ vs. $\dot{m}_{c,cor}$) compressor and plenum operating points are shown in Fig. 4.33 along with the small $B$ compressor characteristics of Fink. The instability growth, blowdown, and recovery phases of operation are clearly observable in Fig. 4.33, while the quiet period is eliminated. Once again, majority of the cycle is spent on the steady-state forward (from Point 6 to 2) and reverse (from Point 4 to 5) flow characteristics, while operating away from the characteristics during a portion of the
instability growth phase (Point 2 to 3) and the small fraction of time required to transition from forward to reverse flow (Point 3 to 4) and reverse to forward flow (Point 5 to 6).

Instability growth begins (Point 2) as the speed approaches 42.6 krpm and continues until the speed reaches 43.8 krpm at Point 3. Next, the flow quickly reverses in the blowdown phase until it reaches a minimum value of -0.21 kg/s at Point 4. As the compressor operating point moves along the reverse flow characteristic to Point 5, the pressure ratio decreases and the flow rate increases. Then, the flow quickly transitions back to the forward direction, reaching the maximum value of 0.48 kg/s at Point 6. At Point 6, the predicted power absorbed by the compressor is nearly 35 kW, causing the rotational speed to decrease as the operating point moves towards the beginning of the instability growth period (Point 2). The predicted compressor speed fluctuates between 42.2 and 45.0 krpm with a peak-to-peak amplitude of 2.8 krpm (6.4% of the mean).
The difference \( p_{ce} - p_p \) between the predicted pressures at the compressor exit \( p_{ce} \) and in the plenum \( p_p \), as shown in Fig. 4.34 (corresponding to \( t = 0.9-1.8 \) s in Fig. 4.31), again represents the net pressure force acting on the air in the compressor exit duct. The green vertical lines in Figs. 4.34 and 4.35 indicate the maximum mass flow rates during the instability growth (left line) and recovery (right line) phases along with the minimum flow rate in the blowdown (middle line) period. Between Points 3 and 4 in Fig. 4.34, the large negative spike of \( p_{ce} - p_p \) provides a (negative) force acting to decelerate the flow until it reaches the minimum value at Point 4, as shown in Fig. 4.35. Immediately following Point 5 in Fig. 4.35, the flow is then accelerated due to the net positive force created by the large (positive) spike of \( p_{ce} - p_p \), as shown in Fig. 4.34.
Figure 4.34: Predicted $p_{cc} - p_p$ with $T_{DS}=0.68$ s.

Figure 4.35: Predicted, corrected compressor mass flow rate with $T_{DS}=0.68$ s.
4.5 Summary

The present study demonstrated the ability to accurately predict mild surge using an unsteady, time-domain approach applied to a 1-D model of Fink’s (1988) large B compression system. The predicted amplitudes, time-averaged values, and frequency of oscillations in $\phi_c$, $Ma_0$, and $\psi_f$ were nearly identical to the corresponding measurements.

As the throttle at the exit of the compression system was further closed, the compression system entered deep surge with four flow phases (quiet, instability growth, blowdown, and recovery). The quiet period was eliminated and the duration of the instability growth period reduced with subsequent reductions in flow area. When the surge cycles are viewed in the $PR_c$ vs. $\dot{m}_{c,cor}$ plane, the time-resolved “surge loops” are noticeably different for each operating point, as shown in Fig. 4.36. A decrease in the throttle diameter shifts the pressure ratios and rotational speeds to lower values. Furthermore, the maximum $\dot{m}_{c,cor}$ decreases and the minimum increases slightly as the duration of the deep surge cycle is reduced. Despite these differences in the $PR_c$ vs. $\dot{m}_{c,cor}$ plane, the time-resolved “surge loops” in the nondimensional $\psi_c$ vs. $\phi_c$ plane are nearly identical for all three deep surge predictions, as shown in Fig. 4.37. The deep surge predictions captured the cycle period and the dominant amplitudes during the blowdown and recovery phases, showing good agreement with measurements. During the instability growth phase, the deep surge predictions underestimated the amplitudes of fluctuations, while the frequency and number of Helmholtz resonator oscillations provided reasonable agreement with measurements. Overall, the surge predictions agreed reasonably well with the experimental observations of Fink (1988).
Figure 4.36: Predicted time-resolved large $B$ compressor operating points with $T_{DS}=3.0$, 1.22, and 0.68 s along with small $B$ data of Fink (1992).

Figure 4.37: Predicted time-resolved large $B$ compressor nondimensional operating points with $T_{DS}=3.0$, 1.22, and 0.68 s along with small $B$ data of Fink (1992).
CHAPTER 5

COMPARISON of PREDICTIONS with MEASUREMENTS from the LARGE B COMPRESSION SYSTEM at THE OHIO STATE UNIVERSITY

The simulations in the present work are aimed at reproducing the experimental observations of the large B compression system at OSU (recall Fig. 2.6). Predictions are presented at a time averaged impeller tip speed of 310 m/s for full, half, and eighth plenum volumes. For each volume, the ability to predict the instability inception point with the 1-D approach used in the present work will be evaluated through comparisons with measurements at a stable operating point on the boundary of mild surge. Furthermore, the ability to predict the transition from the largest obtainable amplitude mild to deep surge will also be assessed against measurements along with the resulting pressure oscillations. All simulations are started at a stable operating point near the surge line at \( U = 310 \) m/s. Then, the flow area of the throttle valve at the exit of the compression system is reduced slightly to decrease the mass flow rate through the compressor, therefore promoting surge.

5.1 Full Plenum Volume

The simulations in this section are aimed at reproducing the experimental results of the large B compression system at OSU at a plenum volume of 9.2 L (full volume) and an impeller tip speed of 310 m/s. Experimentally determined boundary between stable
and mild surge ("Stable Boundary") of this system occurs at a corrected mass flow rate of $\dot{m}_{c,\text{cor}} = 0.0295 \text{ kg/s}$ and a total-to-total pressure ratio of $PR_c = 1.633$, as designated by the green dot in Fig. 5.1. As the mass flow rate is reduced below the stable boundary, pressure and mass flow rate oscillations occur at the Helmholtz resonance frequency of the system and grow as the time average mass flow rate is further decreased. The largest amplitude mild surge (in terms of pressure and mass flow rate fluctuations) occurs at a time averaged corrected mass flow rate of $\dot{m}_{c,\text{cor}} = 0.0262 \text{ kg/s}$ and $PR_c = 1.621$, as shown by the yellow dot in Fig. 5.1. From this mild surge operating point, a slight reduction in mass flow rate leads to deep surge at a time averaged corrected mass flow rate of $\dot{m}_{c,\text{cor}} = 0.0237 \text{ kg/s}$ and $PR_c = 1.517$, as shown by the red dot in Fig. 5.1.

![Figure 5.1](image)

**Figure 5.1:** Experimentally determined stable, mild, and deep surge time averaged operating points for the large $B$ system at OSU with full plenum volume.
5.1.1 Stable

Figure 5.2 compares the predicted (blue dot) and measured (green dot) compressor operating points at the stability limit. This comparison demonstrates the ability of nonlinear 1-D time domain approaches to capture the instability inception point of the compression system. This capability would allow the low-flow compression system delivery pressure to be maximized, while avoiding the instability and acoustic issues associated with mild and especially deep surge.

![Diagram showing predicted and measured compressor operating points at the stability limit for the large B system with full plenum volume.]

Figure 5.2: Predicted and measured compressor operating points at the stability limit for the large B system with full plenum volume.

5.1.2 Mild Surge

The predicted compressor mild surge operating points at the boundary with deep surge are shown as the blue “loop” (blue dot representing its time-average) in Fig. 5.3, along with the corresponding time-averaged measured point (yellow dot). These
operating points represent the largest obtainable amplitude mild surge since any further
decrease in throttle diameter (mass flow rate) would result in deep surge.

![Graph of pressure ratio vs corrected mass flow rate]

Figure 5.3: Predicted and measured compressor operating points for mild surge at the
boundary with deep surge for the large B system with full plenum volume.

The measured and predicted compressor exit pressures during mild surge and the

![Graph of measured and predicted pressures]

corresponding frequency domain analyses are shown in Figs. 5.4 and 5.5, respectively.

The predicted compressor exit pressure in Fig. 5.4 agrees reasonably well with the
measurements, but slight differences exist. The measured compressor exit pressure
fluctuates with an amplitude of approximately 0.43 kPa (0.26% of the mean value).

Figure 5.5 shows the predicted fundamental mild surge frequency is 13.2 Hz, which is
near the theoretical Helmholtz resonator frequency of 12.1 Hz [calculated using Eq.
(2.8)] and the experimental value of 12.0 Hz. The predicted dominant SPL is 140.7 dB,
which compares favorably with 138.9 dB from measurements.
Figure 5.4: Measured and predicted static pressure at the compressor exit during mild surge for the large $B$ system with full plenum volume.

Figure 5.5: Frequency analysis of the measured and predicted compressor exit pressure during mild surge for the large $B$ system with full plenum volume.
The measured and predicted plenum pressures during mild surge and the corresponding frequency domain analyses are shown in Figs. 5.6 and 5.7, respectively. Again, the predicted pressure agrees reasonably well with the measurements, as shown in Fig. 5.6. The observed differences may partially be attributed to a slight cycle-to-cycle variation in the experiments, although the turbocharger bench was given adequate time to reach mild surge cycles with repeating amplitudes. The maximum predicted SPL is 143.1 dB (at 13.2 Hz) which is near the measured 140.4 dB (at 12.0 Hz).

Figure 5.6: Measured and predicted static pressure in the plenum during mild surge for the large B system with full plenum volume.
5.1.3 Deep Surge

The mild surge of the preceding section transitions to deep surge here when the throttle valve area (mass flow rate) is decreased slightly. The predicted compressor operating points during deep surge are shown as the blue “loop” (blue dot representing its time-average) in Fig. 5.8, along with the corresponding time-averaged experimental point (red dot). When the compression system transitions from mild (recall Fig. 5.3) to deep surge (Fig. 5.8), the time-averaged measured compressor pressure ratio decreases by 10.4 kPa (6.4%).
The measured and predicted compressor exit pressures during deep surge and their frequency analyses are shown in Figs. 5.9 and 5.10, respectively. The predicted pressure in Fig. 5.9 shows a reasonable resemblance to the measurement. The measured compressor exit pressure during deep surge fluctuates with an amplitude of approximately 10.8 kPa (7.2% of the mean value). The maximum SPL from the simulation is 171.2 dB (at 6.6 Hz) which is close to the measured 168.2 dB (at 7.6 Hz), as shown in Fig. 5.10. Comparison of Figs. 5.5 and 5.10 demonstrates that (a) both the measured and predicted fundamental deep surge frequencies are well below those of the mild surge; and (b) the peak SPL at the compressor exit increases by approximately 30 dB when the compressor transitions from the largest amplitude mild surge of the preceding section (recall Fig. 5.5) to the deep surge in this section (Fig. 5.10).
Figure 5.9: Measured and predicted static pressure at the compressor exit during deep surge for the large $B$ system with full plenum volume.

Figure 5.10: Frequency analysis of the measured and predicted compressor exit pressure during deep surge for the large $B$ system with full plenum volume.
The measured and predicted *plenum* pressures during deep surge and their frequency analyses are shown in Figs. 5.11 and 5.12, respectively. The predicted pressure continues to agree reasonably well with the measurements, while differences persist, as shown in Fig. 5.11. The maximum SPL from the simulation is 171.7 dB (at 6.6 Hz) which is close to the experimental value of 168.9 dB (at 7.6 Hz). Multiple harmonics of the fundamental frequencies are present in both the experimental and computational results.

![Figure 5.11: Measured and predicted static pressure in the plenum during deep surge for the large B system with full plenum volume.](image)
Figure 5.12: Frequency analysis of the measured and predicted plenum pressure during deep surge for the large B system with full plenum volume.

The difference \((p_{ce} - p_p)\) between the predicted pressures at the compressor exit \(p_{ce}\) and in the plenum \(p_p\), as shown in Fig. 5.13, represents the net pressure force acting on the air in the compressor exit duct. Between the two leftmost vertical bars in Fig. 5.13, the plenum pressure is hence elevated above that of the compressor exit, with the resulting (negative) force acting to decelerate the flow until it reaches the minimum value, as shown in Fig. 5.14. Between the two rightmost vertical bars in Fig. 5.14, the flow is then accelerated due to the net (positive) force created by the higher pressure at the compressor exit relative to the plenum, as shown in Fig. 5.13.
Figure 5.13: Predicted $p_{ce} - p_p$ during deep surge for the large $B$ system with full plenum volume.

Figure 5.14: Predicted compressor mass flow rate during deep surge for the large $B$ system with full plenum volume.
The predicted temperatures at the compressor inlet and exit are shown in Figs. 5.15 and 5.16, respectively, along with the time-averaged measured values for reference. The thermocouples utilized in the present study did not have an adequate response time to capture rapid temperature changes, thus they merely represent a time-averaged value. When the flow reverses (at the left vertical bar), the predicted temperature at the compressor inlet begins to elevate sharply and surpasses the exit. When the flow recovers back to the forward direction (at the right vertical bar), the compressor reingests the hot air at the inlet and further heats it, leading to the spike in compressor exit temperature in Fig. 5.16. The time-averaged predictions at the compressor inlet and exit show reasonable agreement with the (time-averaged) measurements, as shown in Figs. 5.15 and 5.16, respectively.
Figure 5.15: Measured and predicted temperatures at the compressor inlet during deep surge for the large $B$ system with full plenum volume.

Figure 5.16: Measured and predicted temperatures at the compressor exit during deep surge for the large $B$ system with full plenum volume.
5.2 Half Plenum Volume

The simulations in this section are aimed at reproducing the experimental results of the large B compression system at a plenum volume of 4.6 L (half volume) and an impeller tip speed of 310 m/s. Experimentally determined boundary between stable and mild surge (“Stable Boundary”) of this system occurs at a corrected mass flow rate of $\dot{m}_{c,\text{cor}}=0.0294$ kg/s and a total-to-total pressure ratio of $PR_c=1.623$, as designated by the green dot in Fig. 5.17. As the mass flow rate is reduced below the stable boundary, pressure and mass flow rate oscillations occur at the Helmholtz resonance frequency of the system and grow as the time average mass flow rate is further decreased. The largest amplitude mild surge (in terms of pressure and mass flow rate fluctuations) occurs at a time averaged corrected mass flow rate of $\dot{m}_{c,\text{cor}}=0.0253$ kg/s and $PR_c=1.607$, as shown by the yellow dot in Fig. 5.17. From this mild surge operating point, a slight reduction in mass flow rate leads to deep surge at a time averaged corrected mass flow rate of $\dot{m}_{c,\text{cor}}=0.0224$ kg/s and $PR_c=1.507$, as shown by the red dot in Fig. 5.17.
Figure 5.17: Experimentally determined stable, mild, and deep surge time averaged operating points for the large $B$ system with half plenum volume.

5.2.1 Stable

Figure 5.18 compares the predicted (blue dot) and measured (green dot) compressor operating points at the stability limit. This comparison once again demonstrates the ability of nonlinear 1-D time domain approaches to capture the instability inception point of the compression system.
Figure 5.18: Predicted and measured compressor operating points at the stability limit for the large $B$ system with half plenum volume.

### 5.2.2 Mild Surge

The predicted compressor mild surge operating points at the boundary with deep surge are shown as the blue “loop” (blue dot representing its time-average) in Fig. 5.19, along with the corresponding time-averaged measured point (yellow dot). These operating points represent the largest obtainable amplitude mild surge since any further decrease in throttle diameter (mass flow rate) would result in deep surge.
Figure 5.19: Predicted and measured compressor operating points for mild surge at the boundary with deep surge for the large $B$ system with half plenum volume.

The measured and predicted compressor exit pressures during mild surge and the corresponding frequency domain analyses are shown in Figs. 5.20 and 5.21, respectively. The predicted compressor exit pressure in Fig. 5.20 agrees reasonably well with the measurements, with the specific exception of higher frequency components. The measured compressor exit pressure fluctuates with an amplitude of approximately 0.93 kPa (0.58% of the mean value). Figure 5.21 shows the predicted fundamental mild surge frequency is 16.6 Hz, which is identical to the theoretical Helmholtz resonator frequency [calculated using Eq. (2.8)] and lies within the 0.4 Hz frequency resolution (limited here by the selected sample length) of the experimental value of 16.4 Hz. The predicted dominant SPL is 148.1 dB, which nearly reproduces the 148.6 dB from measurements.
Figure 5.20: Measured and predicted static pressure at the compressor exit during mild surge for the large $B$ system with half plenum volume.

Figure 5.21: Frequency analysis of the measured and predicted compressor exit pressure during mild surge for the large $B$ system with half plenum volume.
The measured and predicted *plenum* pressures during mild surge and the corresponding frequency domain analyses are shown in Figs. 5.22 and 5.23, respectively. The predicted pressure agrees reasonably well with the measurements, as shown in Fig. 5.22. The observed differences are partially due to the presence of a slight cycle-to-cycle variation in the experiments, although the turbocharger bench was given adequate time to reach repeating amplitude mild surge cycles. The maximum predicted SPL is 150.3 dB (at 16.6 Hz) which is nearly identical to the experimental value of 150.0 dB (at 16.4 Hz). The harmonic of the fundamental frequency is also captured well computationally with a SPL of 126.3 dB at 33.2 Hz.

Figure 5.22: Measured and predicted static pressure in the plenum during mild surge for the large *B* system with half plenum volume.
Figure 5.23: Frequency analysis of the measured and predicted plenum pressure during mild surge for the large $B$ system with half plenum volume.

The difference $(p_{ce} - p_p)$ between the predicted pressures at the compressor exit $p_{ce}$ and in the plenum $p_p$, as shown in Fig. 5.24, once again represents the net pressure force acting on the air in the compressor exit duct. The pressure in the plenum is greater than that at the compressor exit between the two leftmost vertical bars in Fig. 5.24, creating a net pressure force on the air in the compressor exit duct, acting to decelerate the flow over the same time period, as predicted in Fig. 5.25. Between the two rightmost vertical bars in Fig. 5.24, the pressure in the plenum $p_p$ is less than the pressure at the compressor exit $p_{ce}$, leading to acceleration of the flow in the compressor exit duct during this period, as shown in Fig. 5.25.
Figure 5.24: Predicted $p_{ce}-p_{p}$ during mild surge for the large $B$ system with half plenum volume.

Figure 5.25: Predicted compressor mass flow rate during mild surge for the large $B$ system with half plenum volume.
5.2.3 Deep Surge

The mild surge of the preceding section transitions to deep surge here when the throttle valve area (mass flow rate) is decreased slightly. The predicted compressor operating points during deep surge are shown as the blue “loop” (blue dot representing its time-average) in Fig. 5.26, along with the corresponding time-averaged experimental point (red dot). When the compression system transitions from mild (recall Fig. 5.19) to deep surge (Fig. 5.26), the time-averaged measured compressor pressure ratio decreases by 10.0 kPa (6.2%).

![Figure 5.26: Measured and predicted compressor operating points during deep surge for the large B system with half plenum volume.](image)

The measured and predicted compressor exit pressures during deep surge and their frequency analyses are shown in Figs. 5.27 and 5.28, respectively. The predicted pressure in Fig. 5.27 shows a reasonable resemblance to the measurement. The measured
compressor exit pressure during deep surge fluctuates with an amplitude of approximately 11.5 kPa (7.6% of the mean value). The maximum SPL from the simulation is 171.6 dB (at 10.7 Hz) which nearly reproduces the measured 171.7 dB (at 12.4 Hz), as shown in Fig. 5.28. The peak SPL at the compressor exit increases by approximately 23 dB when the compressor transitions from the largest amplitude mild surge of the preceding section (recall Fig. 5.21) to the deep surge in this section (Fig. 5.28).

![Figure 5.27: Measured and predicted static pressure at the compressor exit during deep surge for the large B system with half plenum volume.](image)
Figure 5.28: Frequency analysis of the measured and predicted compressor exit pressure during deep surge for the large $B$ system with half plenum volume.

The measured and predicted plenum pressures during deep surge and their frequency analyses are shown in Figs. 5.29 and 5.30, respectively. Once again, the predicted pressure resembles the measurements well, with differences observed in the phasing, as shown in Fig. 5.29. The maximum SPL from the simulation is 172.3 dB (at 10.7 Hz) which is close to the experimental value of 172.7 dB (at 12.4 Hz). Multiple harmonics of the fundamental frequencies are present in both the experimental and computational results.
Figure 5.29: Measured and predicted static pressure in the plenum during deep surge for the large \( B \) system with half plenum volume.

Figure 5.30: Frequency analysis of the measured and predicted plenum pressure during deep surge for the large \( B \) system with half plenum volume.
The difference \((p_{ce} - p_p)\) between the predicted pressures at the compressor exit \(p_{ce}\) and in the plenum \(p_p\), as shown in Fig. 5.31, again represents the net pressure force acting on the air in the compressor exit duct. Between the two leftmost vertical bars in Fig. 5.31, the plenum pressure is hence elevated above that of the compressor exit, with the resulting (negative) force acting to decelerate the flow until it reaches the minimum value, as shown in Fig. 5.32. Between the two rightmost vertical bars in Fig. 5.32, the flow is then accelerated due to the net (positive) force created by the higher pressure at the compressor exit relative to the plenum, as shown in Fig. 5.31.
Figure 5.31: Predicted $p_{ce} - p_p$ during deep surge for the large $B$ system with half plenum volume.

Figure 5.32: Predicted compressor mass flow rate during deep surge for the large $B$ system with half plenum volume.
The predicted temperatures at the compressor inlet and exit are shown in Figs. 5.33 and 5.34, respectively, along with the time-averaged measured values for reference. When the flow reverses (at the left vertical bar), the predicted temperature at the compressor inlet begins to elevate sharply and surpasses the exit. When the flow recovers back to the forward direction (at the right vertical bar), the compressor reingests the hot air at the inlet and further heats it, leading to the spike in compressor exit temperature in Fig. 5.34. The time-averaged predictions at the compressor inlet and exit show reasonable agreement with the (time-averaged) measurements, as shown in Figs. 5.33 and 5.34, respectively.
Figure 5.33: Measured and predicted temperatures at the compressor inlet during deep surge for the large $B$ system with half plenum volume.

Figure 5.34: Measured and predicted temperatures at the compressor exit during deep surge for the large $B$ system with half plenum volume.
5.3 Eighth Plenum Volume

The simulations in this section are aimed at reproducing the experimental results of the large \( B \) compression system at a plenum volume of 1.15 L (eighth volume) and an impeller tip speed of 310 m/s. Experimentally determined boundary between stable and mild surge (“Stable Boundary”) of this system occurs at a corrected mass flow rate of \( \dot{m}_{c,\text{cor}}=0.0286 \) kg/s and a total-to-total pressure ratio of \( PR_c=1.628 \), as designated by the green dot in Fig. 5.35. The largest amplitude mild surge occurs at a time averaged corrected mass flow rate of \( \dot{m}_{c,\text{cor}}=0.0235 \) kg/s and \( PR_c=1.600 \), as shown by the yellow dot in Fig. 5.35. From this mild surge operating point, a slight reduction in mass flow rate leads to deep surge at a time averaged corrected mass flow rate of \( \dot{m}_{c,\text{cor}}=0.0219 \) kg/s and \( PR_c=1.507 \), as shown by the red dot in Fig. 5.35.

Figure 5.35: Experimentally determined stable, mild, and deep surge time averaged operating points for the large \( B \) system with eighth plenum volume.
5.3.1 Stable

Figure 5.36 compares the predicted (blue dot) and measured (green dot) compressor operating points at the stability limit. This comparison once again demonstrates the ability of nonlinear 1-D time domain approaches to capture the instability inception point of the compression system.

![Figure 5.36: Predicted and measured compressor operating points at the stability limit for the large B system with eighth plenum volume.](image)

5.3.2 Mild Surge

The predicted compressor mild surge operating points at the boundary with deep surge are shown as the blue “loop” (blue dot representing its time-average) in Fig. 5.37, along with the corresponding time-averaged measured point (yellow dot). These operating points represent the largest obtainable amplitude mild surge since any further decrease in throttle diameter (mass flow rate) results in deep surge.
Figure 5.37: Predicted and measured compressor operating points for mild surge at the boundary with deep surge for the large $B$ system with eighth plenum volume.

The measured and predicted compressor exit pressures during mild surge and the corresponding frequency domain analyses are shown in Figs. 5.38 and 5.39, respectively. The predicted compressor exit pressure in Fig. 5.38 agrees reasonably well with the measurements. The measured compressor exit pressure fluctuates with an amplitude of approximately 2.85 kPa (1.8% of the mean value). Figure 5.39 shows 28.4 Hz for the predicted fundamental mild surge frequency which is near the theoretical Helmholtz resonator frequency of 27.9 Hz [calculated using Eq. (2.8)] and the experimental value of 26.8 Hz. The predicted dominant SPL is 160.3 dB, which nearly reproduces the 159.1 dB from measurements.
Figure 5.38: Measured and predicted static pressure at the compressor exit during mild surge for the large $B$ system with eighth plenum volume.

Figure 5.39: Frequency analysis of the measured and predicted compressor exit pressure during mild surge for the large $B$ system with eighth plenum volume.
The measured and predicted *plenum* pressures during mild surge and the corresponding frequency domain analyses are shown in Figs. 5.40 and 5.41, respectively. Once again, the predicted pressure agrees reasonably well with the measurement, as shown in Fig. 5.40. Part of the deviation may be due to the presence of a slight cycle-to-cycle variation in the experiments, although the turbocharger bench was given adequate time to reach repeating amplitude mild surge cycles. The maximum predicted SPL of 162.2 dB (at 28.4 Hz) is near the experimental value of 160.1 dB (at 26.8 Hz).

![Graph showing measured and predicted static pressure in the plenum during mild surge](image)

Figure 5.40: Measured and predicted static pressure in the plenum during mild surge for the large $B$ system with eighth plenum volume.
5.3.3 Deep Surge

The mild surge of the preceding section transitions to deep surge here when the throttle valve area (mass flow rate) is decreased slightly. The predicted compressor operating points during deep surge are shown as the blue “loop” (blue dot representing its time-average) in Fig. 5.42, along with the corresponding time-averaged experimental point (red dot). When the compression system transitions from mild (recall Fig. 5.37) to deep surge (Fig. 5.42), the time-averaged measured compressor pressure ratio decreases by 9.3 kPa (5.8%).
Figure 5.42: Measured and predicted compressor operating points during deep surge for the large B system with eighth plenum volume.

The measured and predicted compressor exit pressures during deep surge and their frequency analyses are shown in Figs. 5.43 and 5.44, respectively. The predicted pressure in Fig. 5.43 shows a reasonable similarity to the experimental observations. The measured compressor exit pressure during deep surge fluctuates with an amplitude of approximately 15.8 kPa (10.4% of the mean value). The maximum SPL from the simulation is 174.1 dB (at 23.4 Hz) which nearly reproduces the measured 174.6 dB (at 23.6 Hz). Additionally, the peak SPL at the compressor exit increases by approximately 15 dB when the compressor transitions from the largest amplitude mild surge of the preceding section (recall Fig. 5.39) to the deep surge in this section (Fig. 5.44).
Figure 5.43: Measured and predicted static pressure at the compressor exit during deep surge for the large $B$ system with eighth plenum volume.

Figure 5.44: Frequency analysis of the measured and predicted compressor exit pressure during deep surge for the large $B$ system with eighth plenum volume.
The measured and predicted \textit{plenum} pressures during deep surge and their frequency analyses are shown in Figs. 5.45 and 5.46, respectively. Again, the predicted pressure agrees well with the measurements, but the mean value of the prediction is slightly higher than the measurement, as shown in Fig. 5.45. The maximum SPL from the simulation is 175.3 dB (at 23.4 Hz) which nearly reproduces the experimental value of 175.6 dB (at 23.6 Hz), as shown in Fig. 5.46. The computational result nearly reproduces the frequencies of the multiple harmonics from the experimental result, while the sound pressure levels are somewhat different.

Figure 5.45: Measured and predicted static pressure in the plenum during deep surge for the large \(B\) system with eighth plenum volume.
Figure 5.46: Frequency analysis of the measured and predicted plenum pressure during deep surge for the large B system with eighth plenum volume.

The difference \((p_{ce} - p_p)\) between the predicted pressures at the compressor exit \(p_{ce}\) and in the plenum \(p_p\), as shown in Fig. 5.47, again represents the net pressure force acting on the air in the compressor exit duct. Between the two leftmost vertical bars in Fig. 5.47, the plenum pressure is hence elevated above that of the compressor exit, with the resulting (negative) force acting to decelerate the flow until it reaches the minimum value, as shown in Fig. 5.48. Between the two rightmost vertical bars in Fig. 5.48, the flow is then accelerated due to the net (positive) force created by the higher pressure at the compressor exit relative to the plenum, as shown in Fig. 5.47.
Figure 5.47: Predicted $p_{ce} - p_p$ during deep surge for the large $B$ system with eighth plenum volume.

Figure 5.48: Predicted compressor mass flow rate during deep surge for the large $B$ system with eighth plenum volume.
The predicted temperatures at the compressor inlet and exit are shown in Figs. 5.49 and 5.50, respectively, along with the time-averaged measured values for reference. When the flow reverses (at the left vertical bar), the predicted temperature at the compressor inlet begins to elevate sharply, but it does not surpass the exit. When the flow recovers back to the forward direction (at the right vertical bar), the compressor reingests the hot air at the inlet and further heats it, leading to the spike in compressor exit temperature in Fig. 5.50. The time-averaged prediction at the compressor exit shows a reasonable agreement with the (time-averaged) measurements, as shown in Fig. 5.50, whereas the time-averaged prediction at the compressor inlet is lower than the measurements, as shown in Fig. 5.49. This discrepancy could possibly be attributed to the reverse flow power estimation.
Figure 5.49: Measured and predicted temperatures at the compressor inlet during deep surge for the large $B$ system with eighth plenum volume.

Figure 5.50: Measured and predicted temperatures at the compressor exit during deep surge for the large $B$ system with eighth plenum volume.
5.4 Summary

The 1-D models of the large $B$ compression system at various plenum volumes provided predictions of the compressor stable boundary that agreed closely with measurements, as shown in Fig. 5.51. The predicted compression system pressures during both mild and deep surge were shown to compare reasonably well with measurements, but the boundary between mild and deep surge was predicted to occur at higher mass flows compared to the measurements, as shown in Fig. 5.51.

![Figure 5.51](image)

Figure 5.51: Compressor operating points at the stable boundary along with time-averaged mild and deep surge from (a) measurements and (b) predictions.

The volume of compressed air $V$ has a strong influence on the theoretical Helmholtz resonator frequency in Eq. (2.8), since

$$f_h \sim \frac{1}{\sqrt{V}}.$$

The fundamental mild surge frequencies from both the computational and experimental results are shown to agree closely with the theoretical Helmholtz resonance in Fig. 5.52.
The predicted mild surge frequencies are slightly higher than the measurements for all plenum volumes. Once the compression system enters deep surge, the fundamental frequencies (of both the computational and experimental results) decrease well below the theoretical Helmholtz resonance frequency, as shown in Fig. 5.53. The predicted deep surge frequencies are slightly lower than measurements for all plenum volumes.
Figure 5.52: Mild surge fundamental frequencies.

Figure 5.53: Deep surge fundamental frequencies.
The ratio of deep to mild fundamental surge frequency varies as a function of compressed air volume, as shown in Fig. 5.54. This ratio also crudely approximates the fraction of the theoretical Helmholtz resonance frequency at which the deep surge occurs, since both the measured and predicted mild surge frequencies were near the theoretical value.

![Figure 5.54: Ratio of deep to mild fundamental surge frequency.](image)

The measured and predicted peak sound pressure levels during mild and deep surge decrease as $V$ increases, as shown in Figs. 5.55 and 5.56. The peak SPL during deep surge appears rather insensitive to changes in $V$ relative to the mild surge. Additionally, as $V$ is reduced, the difference between the peak SPL during deep and mild surge decreases. The computational results provide a good approximation of the measured peak SPL during both mild and deep surge.
Figure 5.55: Measured and predicted peak SPL during mild surge.

Figure 5.56: Measured and predicted peak SPL during deep surge.
5.5 Influence of Reverse Flow Power

Since steady-state reverse flow performance information was not available in the present study, the reverse flow compressor power was estimated using a method similar to Theotokatos and Kyrtatos (2001). The nondimensional reverse flow compressor torque was estimated by taking a fraction \( C_Γ = 0.16 \) of the forward flow mirrored over \( ϕ_c = 0 \) \( (m_c = 0) \), as demonstrated with Eq. (3.19) and Fig. 3.7. The value of \( C_Γ \) influences the dominant frequency, amplitude of speed fluctuations, and the time-averaged compressor inlet temperature. The dominant frequency of the deep surge predictions increased with decreasing values of \( C_Γ \), as shown in Fig. 5.57, with \( C_Γ = 0 \) providing the closest approximation to the experimental observation. During deep surge, the predicted amplitude of rotational speed fluctuations decreases with an increase in \( C_Γ \), as shown in Fig. 5.58, with \( C_Γ = 0.32 \) nearly reproducing the measurements. The predicted time-averaged compressor inlet temperature nearly matches the experimental observation with \( C_Γ = 0.16 \), as shown in Fig. 5.59. The dominant frequency and amplitude of speed fluctuations in Figs. 5.57 and 5.58, respectively, show contradicting trends of \( C_Γ \) values in terms of matching experimental observations suggesting an intermediate value, which reproduces the time-averaged compressor inlet temperature well. Therefore, \( C_Γ = 0.16 \) was selected for the present study since it provided a reasonable compromise among the predicted dominant frequency, amplitude of speed fluctuations, and the time-averaged compressor inlet temperature. The \( C_Γ \) value selected here and the quadratic fit in Eq. (3.19) should be evaluated against steady-state reverse flow compressor data if the experimental capability becomes available to make such measurements.
Figure 5.57: Influence of $C_T$ on the dominant deep surge frequency for the large $B$ system at OSU with half plenum volume and $U=310$ m/s.

Figure 5.58: Influence of $C_T$ on the amplitude of speed fluctuations during deep surge for the large $B$ system at OSU with half plenum volume and $U=310$ m/s.
Figure 5.59: Influence of $C_Γ$ on the time-averaged compressor inlet temperature during deep surge for the large $B$ system at OSU with half plenum volume and $U=310$ m/s.
CHAPTER 6
CONCLUDING REMARKS

The goal of this study was to predict the stable operating limit of centrifugal compression systems and the transition from mild to deep surge, including discrete sound peaks at low frequencies and their amplitudes at key locations throughout the system. The turbocharger facility at OSU and that of Fink (1988) provided steady-state data to characterize the performance of the compressors along with unsteady data during mild and deep surge. Both facilities utilized a small and large $B$ compression system to explore the dependence of the low-flow stability behavior on the $B$ number. Experimental data from the large $B$ configurations was utilized for comparison with predictions. As the mass flow rate was reduced at a constant rotational speed, the large $B$ systems exhibited mild surge as the slope of the compressor characteristics became positive. Further reducing the flow rate caused the amplitude of pressure oscillations to grow until the systems entered deep surge. To facilitate surge predictions, the compressors were characterized by data from the small $B$ systems, which extended far to the left of the corresponding large $B$ surge line. By reducing the pressure drop in the system, the small $B$ system at OSU was also capable of operating closer to choke than the large $B$ configuration. The experimental data of the small $B$ systems was extrapolated and interpolated to cover the entire forward and reverse flow operating range using the preprocessing script developed
in the present work. Steady-state reverse flow compressor measurements were not available for the configuration studied at OSU or that of Fink (1988). As a result, the present study estimated the reverse flow region of the maps utilizing extrapolation techniques for characteristics and nondimensional torque similar to those of Theotokatos and Kyrtatos (2001).

An unsteady, 1-D, time-domain solver was utilized for the computational predictions in this study. The compressor maps from the preprocessing script provided performance information to the code itself as an input (text) file in the form of corrected mass flow rate, pressure ratio, and efficiency at constant, corrected rotational speeds, and as a result, the preprocessor of the code was not used. During flow reversal, nonzero compressor power was implemented as a function of the corrected mass flow rate and pressure ratio by employing a lookup map.

A model of Fink’s large $B$ compression system was used to simulate a mild surge operating point along with three deep surge cases near 48 krpm, with the flow area decreasing between each prediction. The simulation of mild surge nearly reproduced the experimental observations of Fink (1988), including the frequency, amplitudes, and time-averaged values of $\phi_c$, $\psi_p$, and $Ma_0$. As the throttle at the plenum exit was further closed, deep surge was predicted with dominant cycle periods of 3.0, 1.22, and 0.68 s, which closely matched the corresponding measurements of Fink at 3.0, 1.24, and 0.70 s, respectively. During the long 3.0 s surge cycle period, the quiet, instability growth, blowdown, and recovery phases of flow were present in both the simulations and experimental observations, with the quiet period disappearing and the duration of the instability growth phase decreasing when the period was reduced to 0.68 s.
computationally and 0.70 s experimentally. During these deep surge cases, the simulated $\phi_c$, $\psi_p$, and $Ma_{t0}$ nearly replicated the corresponding experimental observations, while the maximum amplitude of the secondary Helmholtz resonator oscillations were smaller in predictions than measurements. The predicted time-resolved operating points in the $\psi_c$ vs. $\phi_c$ plane were nearly identical for all three deep surge cases, while significant differences existed between the cases when viewed in the $PR_c$ vs. $\dot{m}_{c,\text{cor}}$ plane.

Models of the large $B$ system at OSU were used to predict the stability limit and the transition from mild to deep surge at $U=310$ m/s for full, 1/2, and 1/8 plenum volumes. The predictions nearly reproduced the measured stability limit for each plenum volume, which occurred near the peak pressure ratio and was nearly constant for the volumes studied. The simulated mild surge dominant frequencies agreed reasonably well with the experiments, which increased with decreasing plenum volume and were closely approximated by the theoretical Helmholtz resonator calculation. During both mild and deep surge, the dominant predicted sound pressure levels at the compressor exit and in the plenum provide good agreement with experimental observations. The measured and predicted deep surge lines were shown to move to lower flow rates as the plenum volume was decreased, while the flow rates are larger in the predictions relative to the measurements. The dominant deep surge frequencies at full (9.2 L) plenum volume occurred at 63 and 50% of the mild surge frequency for the experiment and simulation, respectively. As the plenum volume was decreased to one-eighth (1.15 L) volume, the dominant deep surge frequencies increased to 82 and 88% of the mild surge frequency for the experiment and simulation, respectively.
The stability of the compression systems is extremely sensitive to the (positive) slope of the characteristics to the left of the peak pressure ratio. The use of data from the small $B$ systems allowed this positively sloped portion of the constant speed lines to be accurately represented in the compressor maps. In order to further refine the deep surge predictions, the steady-state, reverse flow compressor performance information should be measured. This region of the compressor map currently utilizes extrapolations that are based on the measured forward flow data. Overall, the mild and deep surge predictions showed reasonable agreement with the experimental observations.


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