TURBOCHARGER TURBINES: AN EXPERIMENTAL STUDY on the EFFECTS of WASTEGATE SIZE and FLOW PASSAGE DESIGN

THESIS

Presented in Partial Fulfillment of the Requirements for the Degree Master of Science in the Graduate School of The Ohio State University

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2013
The present study experimentally investigates three different automotive turbochargers of varying turbine/wastegate combination: two of similar turbine housing (BorgWarner) but different bypass throat diameter (20 and 26 mm), and a third of both different housing (Honeywell) and throat size from the former two (21 mm). The effects of turbine housing flow passage design and bypass throat size on open wastegate turbine performance and wastegate flow efficiency were examined at discrete wastegate valve openings: 5°, 10°, 20°, and 40°. The study also analyzes the effects of these geometrical differences on change in flow rate through the wastegate when examined independent of or in parallel with the rotor. Steady flow experiments were performed on a flow bench and cold-flow turbocharger experimental stand. A semi-empirical physical model has also been developed in this study for 1-D engine simulation codes to better characterize the parallel flow paths through the turbines and thus improve predictive accuracy of open wastegate performance. Measured turbine characteristics with closed wastegate were extrapolated and interpolated with a custom preprocessor and input to the code for the rotor, while wastegate flow was simulated in two ways: as an effective orifice area applied at the rotor (default approach) or as a physical parallel path (proposed alternative).

As the wastegates were opened under parallel flow, total turbine mass flow parameter ($MFP$) increased in proportion to wastegate size at 5° and 10° positions for
similar total-to-static expansion ratio ($ER_{ts}$) and speed parameter. At larger openings, the combined effect of housing design and wastegate flow efficiency are of greater effect; the Honeywell turbine exhibited a similar increase in total MFP from 20° to 40° as from 10° to 20°, while the BorgWarner turbines showed a substantially diminished gain in total MFP from 20° to 40° for a given $ER_{ts}$ and speed parameter.

Wastegate-alone flow efficiency experiments revealed that the Honeywell wastegate discharge coefficient ($C_D$) is higher than both BorgWarner wastegates at 40° for fixed $ER_{ts}$, and it was either equivalent to or slightly greater than the 26 mm BorgWarner wastegate at smaller openings. The $C_D$ for the 20 mm BorgWarner wastegate was greater than the 26 mm variant for all fixed openings and $ER_{ts}$. All wastegates exhibited a slight increase in $C_D$ with $ER_{ts}$. Estimation of combined rotor-and-wastegate flow by adding wastegate-alone and rotor-alone flows resulted in significant over-estimation of total MFP for the BorgWarner turbines. The error increased with bypass opening and, with few exceptions, was greater for the 26 mm than the 20 mm wastegate turbine for fixed opening degree and $ER_{ts}$. For the Honeywell turbine, estimated total turbine MFP was generally within ±5% of measured quantities for all openings and $ER_{ts}$.

Using rotor-alone MFP characteristics and wastegate-alone flow efficiency measurements, the predictive accuracy of open wastegate turbine flow in a 1-D code was improved by physically modeling the parallel bypass path. The rotor-and-wastegate flow predictions for the 26 mm wastegate BorgWarner turbine were particularly more accurate, most significantly at $ER_{ts} > 1.4$: error ranged from -0.50 to 22.94% under the
default approach, whereas in the proposed model the error was confined to -4.62 to 3.21%.
Dedicated to my family
I would like to thank my advisor, Prof. Ahmet Selamet, for his patient guidance and support throughout my Master’s studies and for thoroughly reviewing this thesis. Through his keen insight, penetrating questions, and unwavering dedication to finding the truth behind technical problems, Prof. Selamet taught me much more than an improved understanding of engineering fundamentals. I would also like to thank Prof. Shaurya Prakash for his time in serving as a member of my examination committee and in reviewing this thesis. Additionally, I would like to express my gratitude to Dr. Kevin Tallio, Robert Wade, Keith Miazgowicz, and Kevin Payne of Ford Motor Company for supporting this research project and providing the turbochargers and technical knowledge essential for the study.

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Last, but not least, I would like to thank my family for their constant support throughout my academic and professional career.
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TABLE of CONTENTS

ABSTRACT .......................................................................................................................... ii

ACKNOWLEDGMENTS ........................................................................................................ vi

VITA ........................................................................................................................................ vii

LIST of TABLES .................................................................................................................... xi

LIST of FIGURES ................................................................................................................... xii

NOMENCLATURE ............................................................................................................... xviii

1. INTRODUCTION ............................................................................................................. 1

1.1 Background ..................................................................................................................... 1

1.2 Literature Survey .......................................................................................................... 3

1.3 Objective ....................................................................................................................... 24

2. EXPERIMENTAL SETUPS .............................................................................................. 26

2.1 Turbines Investigated .................................................................................................. 26

2.2 Turbocharger Experimental Stand ............................................................................. 27

2.2.1 Wastegate Actuator and Position Measurement .................................................... 31

2.2.2 Instrumentation, Data Acquisition, and System Control ....................................... 34

2.3 Flow Bench ................................................................................................................... 38

2.4 Turbine Flow Configurations ....................................................................................... 43

3. TURBOCHARGER STAND EXPERIMENTAL RESULTS ........................................... 45

3.1 Closed Wastegate Turbine Performance ................................................................... 46

3.2 Open Wastegate Turbine Performance ..................................................................... 56

3.3 Wastegate Flow Efficiency .......................................................................................... 67

3.3.1 Rotor-Alone and Wastegate-Alone Flow Capacity .................................................. 67

3.3.2 Wastegate Discharge Coefficient .......................................................................... 70
LIST of TABLES

Table 2.1: Details of Turbochargers ................................................................. 27
Table 2.2: Validyne Diaphragm Full-Scale Pressures, Turbocharger Stand ............ 37
Table 2.3: Validyne Diaphragm Full-Scale Pressures, Flow Bench ......................... 42
Table 5.1: Validyne “Thin” Diaphragm Full-Scale Pressures, Turbocharger Stand ..... 114
Table A.1: Curve Fit Coefficients for BorgWarner 20 mm Wastegate ..................... 186
Table A.2: Curve Fit Coefficients for Honeywell 21 mm Wastegate ....................... 187
Table A.3: Curve Fit Coefficients for BorgWarner 26 mm Wastegate ..................... 187
Table B.1: Non-dimensional BorgWarner Turbine Map Curve Fit Variables .......... 188
Table B.2: Non-dimensional Honeywell Turbine Map Curve Fit Variables ............. 189
LIST of FIGURES

Figure 1.1: Turbocharging Configuration; C: Compressor, E: Engine, I: Intercooler, T: Turbine (Heywood, 1988) .................................................................................................................. 2

Figure 1.2: Cutaway View of Internally Wastegated Automotive Turbocharger (Turbochargers NZ, 2012) ................................................................................................................................. 4

Figure 1.3: Radial Turbine Components (Watson, 1982) .................................................................................................................. 5

Figure 1.4: Typical Turbine Performance Map (Heywood, 1988) .................................................................................................................. 7

Figure 1.5: Compressor Power Required vs. Turbine Power Developed When Operating Below Matched Engine Speed (Baines, 2005) .............................................................................. 9

Figure 1.6: Effect of Progressively Reducing Turbine Area (Watson, 1982) .............................................. 10

Figure 1.7: Wastegate Apparatus Components (Fraser, 2010) .................................................................................................................. 13

Figure 1.8: Wastegate Actuator Canister (McEwen, 2007) .................................................................................................................. 14

Figure 1.9: Position of Casing Static Pressure Taps, from Capobianco and Polidori (2008) .................................................................................................................. 19

Figure 1.10: Static Pressure Distribution around the Turbine Casing for Different Wastegate Openings ($\alpha_{WG}$), Rotor-Alone and Rotor-and-Wastegate Flow, from Capobianco and Polidori (2008) .................................................................................................................. 20

Figure 1.11: Rotor and Wastegate Mass Flow Contributions during Individual and Combined Flow Configurations, from Capobianco and Polidori (2008); Wastegate Openings of (a) 8°, (b) 20°, (c) 35°, (d) 60° .................................................................................................................. 22

Figure 2.1: Schematic of Turbocharger Experimental Facility (Uhlenhake, 2010) ......... 29

Figure 2.2: Turbocharger Experimental Stand in Hemi-Anechoic Chamber ......................... 31

Figure 2.3: Electric Linear Wastegate Actuator ......................................................................................... 32

Figure 2.4: Wastegate Position Sensor Setup; (a) Honeywell, (b) BorgWarner ......................... 34

Figure 2.5: Triple-T Piezometric Ring ................................................................................................................. 36
Figure 2.6: Flow Bench Facility Schematic (Christian, 2003) .......................................................... 39
Figure 2.7: Flow Bench Experimental Setup ......................................................................................... 41
Figure 2.8: Turbine Flow Configuration Diagrams; (a) Rotor-Alone Flow, (b) Wastegate-Alone Flow, (c) Rotor and Wastegate Parallel Flow ................................................................. 44
Figure 2.9: Wastegate-Alone Flow Configuration with Plugged Rotor Volute ................................. 44
Figure 3.1: Manufacturer Turbine MFP Map with Experimental Data Superimposed, BorgWarner ................................................................................................................................. 49
Figure 3.2: Manufacturer Turbine Corrected Mass Flow Rate Map with Experimental Data Superimposed, Honeywell ............................................................................................. 50
Figure 3.3: Manufacturer Combined Turbine Efficiency Map with Experimental Data Superimposed, BorgWarner ........................................................................................................ 54
Figure 3.4: Manufacturer Combined Turbine Efficiency Map with Experimental Data Superimposed, Honeywell ........................................................................................................ 55
Figure 3.5: Measured Turbine MFP Characteristics, BorgWarner, 20 mm Wastegate .... 59
Figure 3.6: Measured Turbine MFP Characteristics, BorgWarner, 26 mm Wastegate; *Red Arrows Discussed in Text .......................................................... 60
Figure 3.7: Measured Turbine MFP Characteristics, Honeywell, 21 mm Wastegate ...... 61
Figure 3.8: Total Mass Flow Parameter Sensitivity to Wastegate Opening Angle; HW: Honeywell, BW: BorgWarner .......................................................... 62
Figure 3.9: Measured Turbine Efficiency Characteristics, BorgWarner, 20 mm Wastegate ................................................................................................................................. 64
Figure 3.10: Measured Turbine Efficiency Characteristics, BorgWarner, 26 mm Wastegate ................................................................................................................................. 65
Figure 3.11: Measured Turbine Efficiency Characteristics, Honeywell, 21 mm Wastegate ................................................................................................................................. 66
Figure 3.12: R* and WG* Flow, Rotor Fixed, TS; (a) HW, (b) BW ................................................. 69
Figure 3.13: Measured Wastegate Discharge Coefficient, TS .............................................................. 72
Figure 3.14: Measured Wastegate Effective Flow Area vs. Wastegate Opening, TS ...... 74
Figure 3.15: R&WG and R* Flow, Rotor Fixed, TS; (a) HW, (b) BW ................................................. 76
Figure 3.16: R&WG Flow Comparison Among All Turbines, Rotor Fixed, TS .......... 77
Figure 3.17: R*+WG* vs. R&WG Flow, Rotor Fixed, TS; 21 mm WG .................. 79
Figure 3.18: R*+WG* vs. R&WG Flow, Rotor Fixed, TS; (a) 20 mm, (b) 26 mm WG . 80
Figure 3.19: R*+WG* MFP Error, Rotor Fixed, TS ............................................. 81
Figure 3.20: WG Estimate during R&WG Flow, Rotor Fixed; (a) HW, (b) BW ........ 84
Figure 3.21: Estimated Wastegate (a) $C_D'$ and (b) $A_{eff}'$ in R&WG Flow, Rotor Fixed, TS .......................................................... 86
Figure 3.22: R*+WG* vs. R&WG Flow, Rotor at $N_{par}$; 21 mm WG ................... 89
Figure 3.23: R*+WG* vs. R&WG Flow, Rotor at $N_{par}$; (a) 20 mm, (b) 26 mm WG .... 90
Figure 3.24: R*+WG* MFP Error, Rotor at $N_{par}$ ................................................. 91
Figure 3.25: Comparison of R&WG Estimated $C_D'$; 21 mm WG ...................... 92
Figure 3.26: Comparison of R&WG Estimated $C_D'$; (a) 20 mm, (b) 26 mm WG ... 93
Figure 4.1: R* and WG* Flow, Rotor Fixed, FB; Honeywell 21 mm WG .......... 96
Figure 4.2: R* and WG* Flow, Rotor Fixed, FB; BorgWarner (a) 20 mm, (b) 26 mm WG ........................................................................................................ 97
Figure 4.3: Measured Wastegate Discharge Coefficient, FB; 21 mm WG .......... 99
Figure 4.4: Measured Wastegate Discharge Coefficient, FB; (a) 20 mm, (b) 26 mm WG ........................................................................................................ 100
Figure 4.5: Measured Wastegate Discharge Coefficient Comparison, FB .......... 101
Figure 4.6: Measured Wastegate Effective Flow Area vs. Wastegate Opening, FB .. 102
Figure 4.7: R&WG and R* Flow, Rotor Fixed, FB; HW 21 mm WG ................. 103
Figure 4.8: R&WG and R* Flow, Rotor Fixed, FB; BW (a) 20 mm, (b) 26 mm WG ... 104
Figure 4.9: R&WG Flow Comparison Among All Turbines, Rotor Fixed, FB ....... 106
Figure 4.10: R*+WG* vs. R&WG Flow, Rotor Fixed, FB; 21 mm WG ............... 107
Figure 4.11: R*+WG* vs. R&WG Flow, Rotor Fixed, FB; (a) 20 mm, (b) 26 mm WG108
Figure 4.12: R*+WG* MFP Error, Rotor Fixed, FB .............................................. 109
Figure 5.1: Measured Wastegate (a) $C_D$ and (b) $A_{eff}$, Facility Comparison .......................... 113
Figure 5.2: Wastegate $C_D$ Diaphragm Comparison, TS; (a) HW, (b) BW .......................... 116
Figure 5.3: Honeywell 21 mm Wastegate at 20° Opening; (a) $\rho_3$, (b) $C_D$ .................. 117
Figure 5.4: Honeywell 21 mm Wastegate $C_D$ at 20° Opening vs. Reynolds Number ....... 119
Figure 5.5: Measured BorgWarner WG* Data at Low $ER_{ts}$; (a) $C_D$, (b) Total SPL ...... 122
Figure 5.6: R*+WG* $MFP$ Error, Facility Comparison .......................................................... 124
Figure 5.7: BorgWarner Turbine Housing, Exit Flange; (a) 20 mm, (b) 26 mm WG ... 125
Figure 5.8: BorgWarner Wastegate Throat and Valve Seat; (a) 20 mm, (b) 26 mm WG .... 126
Figure 5.9: BorgWarner Turbine Housing, Facing Inlet Flange ........................................... 127
Figure 5.10: BorgWarner Bypass Port Length Estimate at Shortest Part; (a) 20 mm, (b) 26 mm WG .......................................................... 127
Figure 5.11: Bypass Location with Respect to Volute; (a) BorgWarner, (b) Honeywell 129
Figure 5.12: View into BorgWarner Turbine, Facing (a) Inlet Flange, (b) Wastegate Seat .......................................................... 130
Figure 5.13: View into Honeywell Turbine, Facing (a) Inlet Flange, (b) Wastegate Seat .......................................................... 131
Figure 6.1: One-dimensional Model of Internally Wastegated Turbine, GT-Power Default .......................................................... 135
Figure 6.2: One-dimensional Model of Internally Wastegated Turbine, Proposed Alternative; (a) Rotor-Alone and Rotor-and-Wastegate, (b) Wastegate-Alone .......... 137
Figure 6.3: Turbine Housing Schematic, Cutaway View Illustrating Geometry of Proposed Model .......................................................... 138
Figure 6.4: Measured BorgWarner Map Data and Fits at Speed Parameter Maximum Efficiency Point with Wastegate Closed; (a) $N_{par}$ vs. $ER_{ts}$, (b) BSR vs. $ER_{ts}$, (c) $\eta_{T,ts}'$ vs. $N_{par}$, (d) $MFP$ vs. $N_{par}$ .......................................................... 143
Figure 6.5: Extrapolated Non-dimensional BW Characteristics; (a) $MFP_{norm}$, (b) $\eta_{norm}$ 147
Figure 6.6: Extrapolated BW Characteristics; (a) $MFP$, (b) $\eta_{T,ts}'$ ........................................... 148
Figure 6.7: Non-dimensional BW $MFP_{norm}$ Characteristics at Adjacent Speed Parameters ................................................................. 150

Figure 6.8: Interpolation of $MFP_{norm}$ Characteristics at Adjacent Speed Parameters..... 151

Figure 6.9: Interpolated and Extrapolated BW Characteristics; (a) $MFP$, (b) $\eta_{T,ts}'$ ...... 152

Figure 7.1: Measured vs. Predicted R&WG Flow, Default Model with $C_D$; 21 mm WG .................................................................................................................. 155

Figure 7.2: Measured vs. Predicted R&WG Flow, Default Model with $C_D$;
(a) 20 mm, (b) 26 mm WG .......................................................................................................................... 156

Figure 7.3: Error of Predicted R&WG $MFP$, Default Model with $C_D$; (a) HW, (b) BW 157

Figure 7.4: Measured vs. Predicted R&WG Flow, Default Model with $C_D'$; 21 mm WG .................................................................................................................. 159

Figure 7.5: Measured vs. Predicted R&WG Flow, Default Model with $C_D'$;
(a) 20 mm, (b) 26 mm WG .......................................................................................................................... 160

Figure 7.6: Error of Predicted R&WG $MFP$, Default Model with $C_D'$; (a) HW, (b) BW .......................................................................................................................... 161

Figure 7.7: Measured Wastegate $C_D$ vs. Wastegate Opening, 21 mm Honeywell; FB .. 163

Figure 7.8: Angle of Throttle Component in Proposed Model vs. $ER_{ts}$, 21 mm Honeywell .................................................................................................................. 164

Figure 7.9: Measured vs. Predicted R&WG $MFP$, Proposed Alternative Model with Boundary $ER_{ts}$ Throttle Angle Look-up; 21 mm WG ........................................................................... 165

Figure 7.10: Measured vs. Predicted R&WG $MFP$, Proposed Alternative Model with Boundary $ER_{ts}$ Throttle Angle Look-up; (a) 20 mm, (b) 26 mm WG ................. 166

Figure 7.11: Error of R&WG $MFP$ Predictions, Proposed Alternative Model with Boundary $ER_{ts}$ Throttle Angle Look-up; (a) HW, (b) BW .............................................................................. 167

Figure 7.12: Component $ER_{ts}$ from R&WG Predictions, Proposed Alternative Model with Boundary $ER_{ts}$ Throttle Angle Look-up; 21 mm WG ........................................................................... 170

Figure 7.13: Component $ER_{ts}$ from R&WG Predictions, Proposed Alternative Model with Boundary $ER_{ts}$ Throttle Angle Look-up; (a) 20 mm, (b) 26 mm WG ...................... 171

Figure 7.14: Measured vs. Predicted R&WG $MFP$, Proposed Alternative Model with Component $ER_{ts}$ Throttle Angle Look-up; 21 mm WG ........................................................................... 173
Figure 7.15: Measured vs. Predicted R&WG MFP, Proposed Alternative Model with Component $ER_{ts}$ Throttle Angle Look-up; (a) 20 mm, (b) 26 mm WG .......................... 174

Figure 7.16: Error of R&WG MFP Predictions, Proposed Alternative Model with Component $ER_{ts}$ Throttle Angle Look-up; (a) HW, (b) BW ........................................ 175

Figure A.1: Measured Wastegate Discharge Coefficient Curve Fits, TS .......................... 186
### NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>$a$</td>
<td>Speed of sound</td>
</tr>
<tr>
<td>$A$</td>
<td>Cross-sectional area</td>
</tr>
<tr>
<td>$A_s$</td>
<td>Surface area</td>
</tr>
<tr>
<td>$b$</td>
<td>Normalized efficiency curve fit constant (low $BSR_{\text{norm}}$)</td>
</tr>
<tr>
<td>$BSR$</td>
<td>Blade speed ratio</td>
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<tr>
<td>$BW$</td>
<td>BorgWarner</td>
</tr>
<tr>
<td>$c_0 - c_4$</td>
<td>Discharge coefficient curve fit constants</td>
</tr>
<tr>
<td>$C_D$</td>
<td>Wastegate-alone discharge coefficient</td>
</tr>
<tr>
<td>$C_D'$</td>
<td>Estimated wastegate discharge coefficient (rotor-and-wastegate flow)</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Specific heat of air at constant pressure</td>
</tr>
<tr>
<td>$C_s$</td>
<td>Theoretical gas velocity from nozzle isentropic expansion</td>
</tr>
<tr>
<td>$d, D$</td>
<td>Diameter</td>
</tr>
<tr>
<td>$dx$</td>
<td>Length of spatial discretization</td>
</tr>
<tr>
<td>$e$</td>
<td>Total internal energy per unit mass</td>
</tr>
<tr>
<td>$ER_{ts}$</td>
<td>Total-to-static expansion ratio</td>
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<tr>
<td>$f$</td>
<td>Friction coefficient</td>
</tr>
<tr>
<td>$h$</td>
<td>Specific enthalpy</td>
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</table>
\( h_c \)  Heat transfer coefficient

HW  Honeywell

\( K \)  Loss coefficient

\( m \)  Mass

\( \dot{m} \)  Mass flow rate

\( M \)  Mach number

\( MFP \)  Mass flow parameter in \([(\text{kg/s}) \cdot (\sqrt{\text{K}} / \text{kPa})]\)

\( n \)  Normalized efficiency curve fit constant (high \( BSR_{norm} \))

\( N \)  Turbocharger shaft rotational speed in [rev/min]

\( N_{par} \)  Turbocharger shaft rotational speed parameter in [rev/(min \cdot \sqrt{\text{K}})]

\( NS \)  Specific turbocharger shaft rotational speed

\( p \)  Pressure

\( P \)  Power

\( R \)  Gas constant for air

\( \text{Re} \)  Reynolds number

\( \text{RPM} \)  Revolutions-per-minute

\( R^* \)  Rotor-alone flow configuration

\( \text{R & WG} \)  Rotor-and-wastegate flow configuration

\( t \)  Time

\( T \)  Temperature

\( u \)  Velocity

\( U \)  Velocity of turbine wheel tip
\( V \) Volume

\( \dot{V} \) Volumetric flow rate

\( w \) Specific work

WG Wastegate

WG* Wastegate-alone flow configuration

\( x - z \) Normalized MFP curve fit constants

Greek Symbols

\( \gamma \) Ratio of specific heats

\( \Delta \) Change in parameter

\( \eta \) Isentropic efficiency

\( \eta' \) Combined isentropic-mechanical efficiency

\( \rho \) Density

Subscripts

0 Total property; quantity at zero efficiency

1 Compressor inlet

2 Compressor exit

3 Turbine inlet

4 Turbine exit

\( a \) Air
<table>
<thead>
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<th>Definition</th>
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<td>C</td>
<td>Compressor</td>
</tr>
<tr>
<td>e</td>
<td>Exhaust gas</td>
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<td>Turbine</td>
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<tr>
<td>w</td>
<td>Wall</td>
</tr>
<tr>
<td>WG</td>
<td>Wastegate</td>
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CHAPTER 1

INTRODUCTION

1.1 Background

The fuel economy standard set by the U.S. federal government is scheduled to mandate ever more aggressive levels in the near future. To maintain compliance, automobile manufacturers are challenged to rapidly develop and implement technologies to improve or replace the internal combustion engine. One approach that is gaining traction is turbocharging a downsized displacement engine. This solution is advantageous because it involves proven, reliable technology and delivers a reduction in fuel consumption and emissions without sacrificing drivability or adding substantial upfront cost.

A turbocharger consists of a turbine, a compressor linked to and driven by it via common shaft, and a center bearing housing between the two to support and allow for lubrication of the rotating shaft. A typical turbocharging configuration is shown in Fig. 1.1. The turbine extracts energy from the exhaust gas leaving the engine, and this energy is then used by the compressor to increase the density and thus mass of air entering the engine cylinders. The density may be further increased by reducing the air temperature through an intercooler placed between the compressor exit and engine intake manifold.
With more mass trapped in the cylinder, more fuel can be injected and burned, and an increase in torque and power results. A turbocharged engine will thus have a smaller displacement, and reduced weight, than a naturally aspirated engine of same power. Aside from utilizing otherwise squandered exhaust gas energy, a smaller turbocharged engine realizes increased system efficiency from the reduced parasitic losses of rubbing friction from the piston-crank assembly, as well as from some smaller engine-driven accessories. Although turbocharged diesel engines have been around for decades, the push for higher fuel economy and lower emissions has resulted in increasing prevalence of turbocharged spark ignition (SI) gasoline engines in the U.S. market over the past several years. Turbocharged SI engines pose a few challenges beyond those of their diesel counterparts with respect to both turbine and compressor, including higher exhaust
gas temperatures, a wider range of flow rates, and a method of boost pressure control to avert engine knock.

1.2 Literature Survey

A depiction of a turbocharger typical for SI light vehicle engine applications is shown in Fig. 1.2. Centrifugal compressors are adopted almost exclusively for this application over axial types since the flow experiences an increasing radius as it turns through the blades, yielding the higher boost pressures desired by modern turbocharged engine design. The performance of such a centrifugal compressor is very sensitive to operating parameters (for instance, rotational speed and mass flow rate), and its design point must balance proximity to regions of peak efficiency and deep surge instability. The compressor design operating point thus takes precedence for the turbocharger. Consequently, turbine power and rotational speed are prescribed by compressor selection.
Additionally, the size and speed of the paired engine dictate turbine mass flow rate and thus effective flow area. For a given inlet area, flow through radial turbines results in greater expansion and more specific work than axial types, owing to the radius change as the gas passes through the rotor. Rotational speed and specific work required by the coupled centrifugal compressor, as well as relatively small flow areas, results in a low specific speed requirement of the turbocharger turbine, which is defined as

\[ NS = \frac{N \dot{V}^{1/2}}{W_s^{3/4}} \]  

(1.1)
where $w_s$ is specific isentropic work, $N$ is rotational speed, and $\dot{V}$ is volumetric flow rate. For acceptable efficiency, this precludes the use of axial machines. The turbines for light vehicle turbochargers are therefore almost exclusively of radial design.

The radial turbine, portrayed in Fig. 1.3, is connected to the engine exhaust manifold by a flange at the entrance to its volute or inlet casing (station 1). After passing through the volute, the exhaust gas moves through the stator (2), which may or may not contain nozzles, followed by a short gap (3) and then into the turbine rotor (4). The gas flow enters the rotor in the radially-inward direction, is turned, and exits in the axial direction (5). As the flow is turned in the meridional plane from the outer radius of the turbine blade to the smaller inner radius, it expands and does work.

![Radial Turbine Components](image)

Figure 1.3: Radial Turbine Components (Watson, 1982)

A typical radial turbine performance map for an automotive turbocharger is shown in Fig. 1.4, where total inlet pressure to static exit pressure ratio, $p_{03}/p_4$, is plotted against mass flow parameter ($MFP$), $\dot{m}_T \sqrt{T_{03}/p_{03}}$, where $\dot{m}_T$ is turbine mass flow rate and $T_{03}$ is...
turbine total inlet temperature. Characteristics of constant speed parameter, \( N/\sqrt{T_{03}} \), are indicated with dashed lines, and turbine total-to-static isentropic efficiency, \( \eta_{T,ts} \), contours are denoted by solid lines. Speed parameter and \( MFP \) are the standard method of representing turbocharger turbine performance (Society of Automotive Engineers (SAE) J1826, J922) since they theoretically compensate for changes in inlet conditions such as temperature. Note however, that though these parameters resemble dimensionless quantities, they have units which must be indicated. Capobianco et al. (1989) empirically confirmed that the same \( MFP \) maps may be used regardless of whether the characteristics were generated from a hot gas stand (900 K turbine inlet temperature) or cold gas stand (400 K). However, turbine efficiency does depend on whether the gas test stand is “hot” or “cold.”
As opposed to marine or power generation applications, light vehicle engines are inherently difficult to match with a turbocharger due to the desirability of engine responsiveness and peak torque over a broad speed range, which is widest for SI engines. To accommodate the larger span of flow rates, radial turbines may employ nozzle-less
stators (recall Fig. 1.3). In the absence of nozzle vanes, the volute alone must accelerate the flow and establish rotor inlet gas angle. Although this is less aerodynamically efficient, the decrease in peak turbine efficiency is made up for by an increased efficiency over a broader range of flow rates than a turbine with fixed nozzles. Even if the turbine can adequately accept a span of flow rates, the difficulty of achieving acceptable engine torque arises because of a mismatch between the two machines; the flow rate in an engine is approximately proportional to its speed, whereas the expansion ratio across a turbine (and thus its work output to the compressor) is approximately proportional to the square of flow rate. Therefore, fixed-geometry/fixed-area turbocharger turbines introduce a conflict when the design operating point must be chosen, since the turbine is ideally matched to the engine at only the design condition. This is shown in Fig. 1.5 where the design point is at engine rated speed. At engine speeds lower than the design point, the flow rate is reduced, but the reduction in turbine expansion ratio is even greater, resulting in inadequate power available to the compressor. The contrasting operating characteristics are further elaborated upon next.

A turbocharger chosen to match an engine at maximum speed and load will be sized to achieve the desired maximum boost pressure at a high turbine mass flow rate (that is, the turbine effective flow area will be sized in proportion to the maximum flow of the engine). At elevated engine speeds such a turbocharger will work most efficiently and thus enable the good engine torque and specific fuel consumption for which it was selected. However, at reduced engine speeds the lower mass flow rate and relatively large turbine flow area will result in low exhaust pressure build-up and thus minimal
energy available for expansion, leading to poor or negligible boost pressure development and unacceptable transient response of the turbocharger. A smaller turbine flow area matched for a lower engine speed will have better transient response and will yield higher boost pressure and thus greater engine torque from increased exhaust pressure available at the turbine inlet for the smaller gas flows. This fact is represented in Fig. 1.6a, where the smaller turbine areas, $A_3$, and correspondingly greater turbine inlet pressures, $p_3$, are indicated by superscripted primes, and the specific enthalpy available to the turbine for expansion at those areas and pressures by $\Delta h_T (p_{01}$ is cylinder total pressure, $p_1$ is static pressure immediately downstream of exhaust valve). Figure 1.6b also shows the elevated turbine inlet pressure in engine crank angle domain.
Figure 1.6: Effect of Progressively Reducing Turbine Area (Watson, 1982)
At higher engine speeds, however, the boost pressure developed may yield in-cylinder pressures and temperatures exceeding the permissible thermal and mechanical loading of the engine design as well as increased knock tendency. Ignition timing retarded from MBT (maximum brake torque), late intake valve closing, and/or fuel enrichment could be employed to offset the increased pressure and temperature. In addition, the increased exhaust back-pressure from the smaller turbine and diminished overall turbocharger efficiency (defined as the product of compressor, turbine, and mechanical efficiencies) would lead to an adverse pressure ratio across the engine cylinders. The foregoing negative ramifications from a small turbine area at high speeds would lead to poor residual gas scavenging and increased pumping work, to the detriment of brake engine power and specific fuel consumption. In summary, the difference in operating characteristics between light vehicle engines and turbochargers necessitates a small turbine at low engine speeds and a large turbine at high engine speeds.

To overcome this trade-off, turbines with variable effective flow areas are typically implemented in passenger car applications to effectively regulate exhaust manifold pressure and thereby control the power produced by the turbine. This is accomplished through either of two technologies: variable-turbine-geometry (VTG) or internal wastegate. VTG produces a variable turbine area to match the exhaust gas flow rate by utilizing nozzle vanes in the stator which pivot to change the throat area. A wastegate is a simple pneumatically-actuated valve device which bypasses a portion of the exhaust gas around the turbine wheel in a parallel path, either internal or external to the turbine housing, and back into the exhaust flow downstream of the turbine wheel. The wastegate thus permits the turbine to be sized for low engine speed operation. As
engine speed increases, the wastegate valve is opened (increasing the turbine effective
flow area) upon attaining the desired maximum boost pressure, which is then safely
maintained at higher engine speeds. As Fig. 1.2 shows, internal wastegates consist of a
short bypass port branching off the main turbine flow passage just downstream of the
inlet flange, leading to a circular opening (the throat) which is covered by a flap valve.
The bypass thus permits direct flow from turbine inlet to turbine exit within the housing,
as controlled by the position of the flap valve. Internal wastegates are almost exclusively
used on light vehicle SI engines instead of VTG to vary the turbine flow area. This is
primarily due to the simplicity and reduced cost of wastegates, and due to the
exceedingly high exhaust gas temperatures of turbocharged SI engines (approaching 1050
°C) being unfavorable to the durability of VTG components.

Figure 1.7 is a rendering of a typical pneumatic wastegate actuation system
installed on an automotive turbocharger with internal wastegate (refer to Fig. 1.2 for
context). Part (1) is the head of the wastegate flap valve which covers the bypass throat
inside the turbine housing. The valve rotates about the axis of a stem which sits in a
bushing (2) where it passes through the turbine housing. The stem is fixed to a
lever/crank arm (3) which is acted upon by the displacement of an actuator pushrod (4).
Movement of the pushrod, and thus rotation of the wastegate valve, is controlled by the
actuator canister subassembly (5), which is mounted to the compressor or center housing
by a bracket (6).
A cutaway view of a common, though different, actuator canister is provided in Fig. 1.8, where part numbering is carried over from Fig. 1.7 (operating principle is identical). Within the canister, the actuator rod extends up to a sliding piston or diaphragm (8), to which it is rigidly connected. A coil spring (9) is placed between this diaphragm and the bottom surface of the canister, coaxial with the actuator rod, such that displacement of the pushrod is governed by compression of the spring. Part (7) in Figs. 1.7 and 1.8 is a small, barbed-end tube for the connection of a boost control signal hose. This signal hose is either directly or indirectly (via solenoid control valve) connected to the boost pressure of the turbocharger compressor, at its volute or exit duct (refer to Fig. 1.2). Consequently, the top surface of the diaphragm is acted upon by the boost pressure.
The wastegate system thus acts according to the following: at a pre-determined, application-specific boost pressure acting on the canister diaphragm, the spring is compressed a certain distance; the pushrod in turn extends this distance, causing a rotation of the valve stem and the opening of the wastegate valve head away from its seat on the bypass throat (in a clockwise manner, with reference to Fig. 1.7).

Prior to tuning wastegate system components such as the stiffness of the spring in the actuator canister, determination of bypass throat cross-sectional area would occur. This area is dictated firstly by its ability to accept the maximum design exhaust gas bypass flow without choking, that is, without the flow reaching sonic velocity. A choked wastegate would result in continued increase in boost pressure and excessive cylinder pressures at higher engine speeds, since the additional flow would not be bypassed and
the exhaust manifold pressure would escalate (the same effect as reducing turbine area in Fig. 1.6). Determining the smallest acceptable throat size to accept sufficient flow requires knowledge of flow efficiency through the wastegate, which is commonly characterized by an empirically determined discharge coefficient. The discharge coefficient is a measure of how close the actual mass flow rate through the wastegate compares to the calculated ideal (measurement methodology and calculation of discharge coefficient is fully described in Chapter 3). The upper limit of throat size is limited by loss of resolution and ease of controlling the boost pressure, as a larger area results in greater flow for the same amount of valve head opening, and only a fraction of the total actuator rod stroke length may be used (Fraser, 2010). It is also limited by premature opening of the wastegate prior to design bypass boost pressure, since the increased area will yield a greater force on the valve head for identical turbine inlet pressure (Dewhirst, 1990). The latter consideration must be balanced with the pre-load of the spring in the actuator canister.

Turbocharger manufacturers (Fraser, 2010) and one-dimensional (1-D) engine simulation software (Gamma Technologies, 2010) treat the internal wastegate and rotor as two nozzles operating in parallel under the same upstream and downstream pressures and temperatures of the turbine. That is, the control volume for both wastegate and rotor is drawn around the boundaries of the physical turbine as a whole, and no inclusion is made of losses due to flow dividing at the inlet or combining at the exit within the turbine housing. Knowledge of the wastegate mass flow rate characteristic as a function of total-to-static expansion ratio across the turbine therefore allows, by the foregoing assumption, for the prediction of total combined mass flow rate through the turbine with wastegate
open. The mass flow rate through the turbine with the wastegate open is taken to be the sum of the rotor mass flow rate (measured with wastegate closed), and the wastegate mass flow rate, predicted by the discharge coefficient (determined with no flow through rotor) and expansion ratio across the turbine. The validity that this practice is an accurate representation of actual flow rates with wastegate open is important for two main reasons. First, the turbocharger manufacturer sizes the wastegate throat and selects the actuator canister under the premise that the quantity of bypass flow at partial and full valve head opening has been accurately predicted. If the wastegate throat prematurely chokes, or the resolution of boost control is poor, engine performance and fuel economy will suffer. Second, automobile manufacturers are only provided a turbine map with wastegate closed, unlike VTG turbine maps which provide MFP and efficiency characteristics at different nozzle positions. To predict engine operation with wastegate open, automobile manufacturers must rely on discharge coefficients measured in their own facilities and applied within engine simulation software, which estimates total turbine mass flow rate in the same manner as turbocharger manufacturers. Until recently, the accuracy of assuming total mass flow capacity of a turbine with wastegate open may be determined by summing independently evaluated rotor and wastegate mass flows had not been experimentally researched.

Capobianco and Marelli (2007) investigated the foregoing premise on a wastegated IHI turbocharger (model RHF3, of undisclosed bypass throat diameter) which had been matched to a downsized 1.4 L SI automotive engine. First, the authors measured the steady flow overall turbine MFP and total-to-static efficiency characteristics with the wastegate closed as well as at a number of openings, the latter
positions referred to the angular rotation of the wastegate crank arm as measured by a “variable resistance transducer.” Though the increase in overall turbine mass flow rate with open wastegate had been examined by Capobianco and different co-authors in the past (1990, 2005, 2006), the work in 2007 expanded on the study by analyzing the individual mass flow rate contributions of rotor and wastegate when the latter was open. An exit duct was installed which had a dividing wall extending 150 mm from within the turbine casing, after which the separated rotor and wastegate flows were re-combined. Within the wastegate section of the divided duct a hot wire probe was installed to measure the corresponding mass flow rate. Three flow configurations were evaluated: flow through rotor alone (wastegate closed), flow through wastegate alone (rotor volute plugged), and combined flow through both rotor and wastegate. These setups were each tested at same total-to-static expansion ratio and identical speed parameter (where applicable) for different fixed wastegate open angles of 8°-60°. Results indicated that mass flow rate through wastegate was lower during combined flow than when operating alone for same total-to-static expansion ratio (equal to 1.20) and wastegate open angle. Also in the combined flow arrangement, mass flow rate through rotor continually decreased as the wastegate open angle increased, for same total-to-static expansion ratio and speed parameter (1.20 and 3000 RPM/$\sqrt{K}$, respectively, as well as 1.35 and 4000 RPM/$\sqrt{K}$, respectively). These changes in individual mass flow rate contributions led to a lower combined mass flow rate (rotor-and-wastegate) as compared to the sum of the flow rates (rotor + wastegate) when the components operated independently. Thus the assumption that the rotor and wastegate operate in parallel under the same expansion ratio, such that their individual flow rates may be summed, was proven invalid.
Capobianco and Marelli found that the error of the assumption in their investigation was 10-25%, with a trend of higher error for larger wastegate openings. The authors concluded that the expansion ratios across rotor and wastegate, individually, during combined flow are effectively lower than that measured across the turbine as a whole.

In an effort to expand on the foregoing theory, Capobianco and Polidori (2008) instrumented the same turbocharger turbine from the Capobianco and Marelli (2007) study with 14 static pressure taps around the circumference of its casing, four of them upstream of the volute tongue where the bypass port was located. Pressure tap locations used in their experiment are shown in Fig. 1.9. The same three flow arrangements (rotor-alone, wastegate-alone, rotor-and-wastegate) were analyzed to relate the results of the previous investigation to casing static pressure distribution. To that end, the three setups were each analyzed for the same overall turbine total-to-static expansion ratio (1.35) and identical speed parameter (4000 RPM/√K) for different fixed wastegate open angles: 0°, 8°, 20°, 35°, and 60°. Results indicated that during combined flow, turbine casing static pressure distribution changed compared to flow with the wastegate closed. These results are provided in Fig. 1.10, where the difference of turbine inlet static pressure (measured upstream of the turbine housing) and casing static pressure is plotted against casing tap location (refer to Fig. 1.9). Static pressure immediately upstream of the bypass port progressively decreased as the wastegate was opened due to the increased mass flow rate for constant total-to-static expansion ratio and speed parameter. Downstream of the bypass (including the part of the volute which delivers gas to the rotor), static pressure was higher with combined flow than rotor-alone flow and it increased with larger wastegate openings. Additionally, static pressure immediately upstream of the bypass
port was higher in the case of wastegate-alone flow (not shown in Fig. 1.10) than with combined flow (total-to-static expansion ratio and wastegate open angle constant). No deviations from these trends were observed for overall turbine total-to-static pressure ratios of 1.25 or 1.45 at the same speed parameter.

<table>
<thead>
<tr>
<th>Tap Number</th>
<th>Azimuth Angle Φ [deg]</th>
</tr>
</thead>
<tbody>
<tr>
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</tr>
<tr>
<td>1</td>
<td>-40</td>
</tr>
<tr>
<td>2</td>
<td>-30</td>
</tr>
<tr>
<td>3</td>
<td>-12</td>
</tr>
<tr>
<td>4</td>
<td>0</td>
</tr>
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<tr>
<td>12</td>
<td>210</td>
</tr>
<tr>
<td>13</td>
<td>235</td>
</tr>
</tbody>
</table>

Figure 1.9: Position of Casing Static Pressure Taps, from Capobianco and Polidori (2008)
The authors indicated that the results suggested a reduced level of the actual expansion ratio experienced by the wastegate during combined flow, compared to the quantity determined from the external measurement planes, which would support, in part, the theory of the prior work. However, the authors only illustrated that the static-to-static expansion ratio, not total-to-static expansion ratio, was reduced at the bypass port. This is because the difference between external and casing static pressure (vertical axis in Fig. 1.10) upstream of the bypass port for fixed wastegate opening was attributed to reduced flow area, and that the difference increased with wastegate opening due to higher mass flow rate. Since no estimation was provided regarding relative contributions of frictional...
losses and/or variation in cross-sectional area on static pressure, the results do not allow for an accurate quantification of the reduction in actual total-to-static expansion ratio at the bypass port (compared to the value calculated at the external measurement planes).

The individual mass flow rate contributions of the rotor and wastegate during combined flow were also reviewed by Capobianco and Polidori, and again the total combined mass flow rate was lower than the sum of the components when operating independently (total-to-static expansion ratio, speed parameter, and wastegate open angle constant). Their results are shown in Fig. 1.11, where R* and WG* refer to rotor-alone and wastegate-alone flows, respectively, and R and WG are the rotor and wastegate contributions, respectively, during combined flow. At smaller bypass openings (8°, 20°) wastegate mass flow rate was found to be similar between combined and wastegate-alone arrangements, but rotor mass flow rate was lower in the case of combined flow than rotor-alone. Large bypass openings (35°, 60°) resulted in a switch of the foregoing trends, as wastegate mass flow rate was lower in the case of combined flow than wastegate-alone, but rotor mass flow rate was similar between combined and rotor-alone configurations.

The authors thus reasoned that the lower wastegate mass flow rate in the combined configuration at large openings was partly due to a lower static pressure level measured in the casing upstream of the bypass port, resulting in a lower expansion ratio across the wastegate than during wastegate-alone flow for same external total-to-static expansion ratio and wastegate opening. However, the reduced inlet static pressure at the bypass was attributed by the authors to increased mass flow rate during combined flow;
that a change in actual total-to-static expansion ratio at the bypass between flow configurations occurs, for same external total-to-static expansion ratio, is inconclusive.

Figure 1.11: Rotor and Wastegate Mass Flow Contributions during Individual and Combined Flow Configurations, from Capobianco and Polidori (2008); Wastegate Openings of (a) 8°, (b) 20°, (c) 35°, (d) 60°

The trend of similar rotor mass flow rate between combined and rotor-alone configurations at large wastegate openings is different from the result in 2007
(Capobianco and Marelli), which observed a continued decrease in rotor mass flow rate with increased bypass opening, the greatest disparity occurring at the largest opening. Moreover, Capobianco and Polidori (2008) indicate the error from summing the individual rotor and wastegate mass flow rates as “ranging between 6 and 13 percent,” whereas Capobianco and Marelli (2007) found that the “differences ranged between 10 and 25 percent.” The two foregoing discrepancies (rotor contribution in combined flow and error of summation) are peculiar in that the same turbocharger was used by a common co-author, and might suggest a change in experimental methodology. Since the rotor mass flow rate in combined flow was measured indirectly as the difference between total turbine mass flow rate (measured via laminar flow meter) and wastegate mass flow rate (measured via hot wire probe), the former inconsistency between the two studies is likely due to issues with hot-wire probe mass flow rate measurement. In fact, Capobianco and Marelli (2007) indicated that they had difficulty achieving repeatable mass flow rates due to slight differences in valve head position for a given wastegate opening angle, resulting from the “typical clearances of the bypass mechanism.” As both investigations used a unique calibration curve for each wastegate position for the hot-wire probe, it can be inferred that the flow field downstream of the wastegate is altered significantly for small changes in wastegate position. Regardless of individual contributions during combined flow, the foregoing research qualitatively demonstrates the error associated with the prevailing assumption that the total mass flow capacity of a turbine with wastegate open may accurately be determined by summing independently evaluated rotor and wastegate mass flows.
However, no work is available in literature which analyzes, on a one-to-one basis, different turbine housing designs – including how the bypass port is incorporated into them – or wastegate throat size. A better comprehension of how housing design and wastegate size affects flow efficiency through the bypass is needed to guide the design of the flow passages, understand the factors which contribute to the magnitude of rotor-alone plus wastegate-alone error, and as a result, improve the physical models which characterize wastegate flow in 1-D engine simulation codes.

1.3 Objective

The main objective of the present study is to experimentally determine the effects of bypass passage/turbine housing design and bypass throat size on (1) open wastegate turbine performance, (2) wastegate flow efficiency, and (3) rotor-alone plus wastegate-alone $MFP$ error with respect to rotor-and-wastegate flow, all at discrete wastegate valve openings. This study was conducted under steady flow for three different turbine/wastegate combinations, two of similar turbine housing but different bypass throat size, and a third of both different housing and throat size from the previous two. Experiments were performed on a flow bench and cold-flow turbocharger experimental stand, and common results are compared. An additional objective of this study is to develop a semi-empirical physical model for 1-D engine simulation codes based on the experimental work to provide alternatives from the code default for improved predictive accuracy of open wastegate turbine performance.

In Chapter 2 the experimental setup is discussed, starting with an introduction of the turbochargers investigated. The turbocharger test stand and flow bench facilities are
then described, including pressure and wastegate position measurement techniques.

Finally, configurations of turbine flow studied throughout the thesis are introduced.

Experimental results pertaining to the turbocharger stand and flow bench are presented in Chapters 3 and 4, respectively, which are then compared and unified in Chapter 5. The 1-D turbine model and engine simulation code are described in Chapter 6, the predictions of which are examined against experimental results in Chapter 7. Finally, concluding remarks are given in Chapter 8 along with recommendations for future work.
CHAPTER 2

EXPERIMENTAL SETUPS

This chapter begins with a detailed description of the three turbochargers, of varying turbine housing design and wastegate throat diameter, which are investigated in the present work. Experimental research for all three turbines was conducted with two distinct facilities: a turbocharger experimental stand and a steady flow bench. A review of pertinent turbocharger facility details is therefore provided, including source of turbine inlet gas, wastegate position and static pressure measurement, and data acquisition. Similar details of the flow bench setup are also discussed. The chapter concludes with an introduction of turbine rotor and wastegate flow configurations which will pervade subsequent chapters of the thesis.

2.1 Turbines Investigated

Experiments were performed for three different internally wastegated radial turbines from turbochargers matched to a variation of a Ford 3.5 L V6 direct-injected and turbocharged SI engine. Two turbines are BorgWarner model 045C from the right bank of the TiVCT version of the engine. The turbines share 45 mm wheel tip diameters, wheel trims of 84, and geometrically similar casings, but have either a 20 mm or 26 mm
diameter wastegate throat. A 26 mm wastegate was used in the final production version of the turbocharger. The cited turbine trim value is calculated by

\[ \text{Trim} = \left( \frac{d_{\text{Exducer}}}{d_{\text{Inducer}}} \right)^2 \times 100, \tag{2.1} \]

where inducer diameter, \( d_{\text{Inducer}} \), is the wheel diameter where gas enters and exducer diameter, \( d_{\text{Exducer}} \), where it exits. The third turbine investigated is from the left bank of the iVCT version of the Ford engine. Manufactured by Honeywell, model NS111 has a wheel of 41 mm tip diameter and 72 trim, and a 21 mm diameter wastegate throat. The details of the turbines and their mated centrifugal compressors are summarized in Table 2.1. Note that compressor trim values are determined by switching the numerator and denominator of Eq. (2.1).

<table>
<thead>
<tr>
<th>Manufacturer</th>
<th>Model</th>
<th>Wheel Tip Diameter</th>
<th>Wheel Trim</th>
<th>Wastegate Throat Diameter</th>
<th>Model</th>
<th>Wheel Tip Diameter</th>
<th>Wheel Trim</th>
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<tr>
<td>Borg Warner</td>
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<td>84</td>
<td>26 mm</td>
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<tr>
<td>Honeywell</td>
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<td>72</td>
<td>21 mm</td>
<td>C224</td>
<td>49 mm</td>
<td>55</td>
</tr>
</tbody>
</table>

2.2 Turbocharger Experimental Stand

Turbine performance mapping and part of the flow efficiency analysis was performed on the cold-flow turbocharger experimental stand located within the Center for Automotive Research at The Ohio State University. A schematic of the facility is
provided in Fig. 2.1. The facility includes a Quincy QSI-750 screw compressor capable of outputting a flow range of 0.355 kg/s to 0.52 kg/s at a corresponding pressure range of 11.2 bar to 6 bar. The internal air cooling system of the compressor is bypassed to retain the elevated outlet temperature for eventual use at the turbine downstream. After passing through a Basco Type-TC centrifugal separator to remove entrained lubricating oil and particles, the compressed air is dumped into a 30.02 m$^3$ externally insulated accumulator. Though the screw compressor output is regulated to provide a constant pressure in the accumulator, the large volume storage serves as a buffer and minimizes supply pressure fluctuations prior to passing the compressed air to the facility downstream. Next, the flow passes through a Quincy CPNT500 mist eliminator, which contains a 0.01 micron filter to further remove oil and water entrainment down to 0.01 ppm levels.

Before reaching the turbine, the flow of compressed air is regulated by a 1.5 in segmented ball valve with pneumatic actuator and 32-bit position controller. Next, the flow passes through a calibrated orifice flowmeter for mass flow rate measurement, where calibration was performed against the steady flow bench which will be discussed in Section 2.3. Flowmeter measurement is within $\pm 1.25\%$ of the flow bench as a result of the calibration (Uhlenhake, 2010). Also, seven interchangeable orifice plates of differing size are available to maximize accuracy for the desired range of mass flow rate. After the flowmeter the flow is then heated by a pair of PID-controlled Watlow 7.8 kW semi-cylindrical electric heaters (15.6 kW total) mounted to the outside of a 310 grade stainless steel pipe section. The heaters allow for increased range of turbocharger mapping by elevating turbine inlet temperatures to avoid freezing of condensate after expansion at the turbine exit.
Figure 2.1: Schematic of Turbocharger Experimental Facility (Uhlenhake, 2010)
The physical structure upon which the turbocharger is mounted is housed within a hemi-anechoic chamber to facilitate acoustic measurements, a feature not needed in the present work. A picture of the stand within the hemi-anechoic chamber is shown in Fig. 2.2, where (1) is the turbine inlet duct, (2) is the turbocharger turbine, and (3) is the turbine exit duct. The turbocharger compression system begins with an inlet duct (4) which has a bellmouth entrance from ambient, and downstream of the compressor an exit duct (5) discharges the air into a variable volume plenum (6). The plenum piston was fixed at 1/8 total volume position for all experiments. From the plenum the flow passes through a 2 in segmented ball valve (7) with pneumatic actuator and 32-bit position controller, is cooled by an air-to-water intercooler (8), and is then sent through a calibrated orifice flowmeter for mass flow rate measurement. The facility also includes a lubrication system for the center bearing housing of the turbocharger. A ½ HP oil pump external to the hemi-anechoic chamber, an oil filter, a bypass pressure regulator, and a temperature-controlled 1 kW immersion heater installed within an oil reservoir round out the system. An oil inlet pressure transducer is also installed to prevent operation of the experimental stand without adequate oil flow.
2.2.1 Wastegate Actuator and Position Measurement

A method was sought to safely adjust and hold wastegate position on the turbocharger stand while the rotor was spinning. To that end, the pneumatic actuator canister and pushrod were removed and replaced by an electric linear actuator, Firgelli Automations model FA-PO-150-12-2”. The electric actuator has a 2 in stroke, dynamic and static load capacities of 150 and 300 lbf (667.2 and 1334.5 N), respectively, and a built-in 10 kΩ linear potentiometer for position feedback. A Firgelli Technologies LAC (linear actuator control) board is also installed as an interface between the data acquisition system and actuator. The LAC is a closed-loop PD (proportional-derivative)
controller that adjusts the position of the actuator based on a 4-20 mA position input signal from the DAQ system and the feedback from the potentiometer of the actuator. Actuator speed, position sensitivity, and stroke limit can be set on the board as well. A picture of the electric actuator installed on the Honeywell turbine setup is shown in Fig. 2.3 [part (9), where numbering is carried over from Fig. 2.2]. The angle the actuator makes with respect to the stand mounting surface was chosen to most closely replicate the angle of the pneumatic actuator pushrod, thereby allowing a similar wastegate opening range as available on-engine. A clevis (10) was fabricated to connect the rod end of the electric actuator to the wastegate crank arm.

Figure 2.3: Electric Linear Wastegate Actuator
Instrumentation was also sought to accurately measure the opening position of the wastegate. Since each turbine and wastegate system has a unique set of hardware, including the mounting configuration and dimensions of the pneumatic actuator assembly, it was decided to refer to angular rotation of the wastegate crank arm instead of linear displacement of the actuator rod to most aptly compare the performance of different turbine/wastegate systems. Recall that the crank arm is rigidly connected to the wastegate valve head such that their rotations correspond (Fig. 1.2). Plus, if displacement of the pneumatic actuator pushrod is desired, it can be geometrically determined for any opening angle.

To accomplish the angular measurement, a 10 kΩ linear-taper rotary potentiometer [part (12) in Fig. 2.3] is installed concentric with the rotational axis of the crank arm as a position sensor. The rotating knob of the potentiometer is fastened to the crank arm by a fabricated mounting bracket (11), while the base of the sensor is rigidly held by a second bracket connected to the stand. A close-up picture of this measurement setup is provided in Fig. 2.4. The potentiometer is essentially acting as a variable voltage divider; when a DC voltage is connected across the outer two terminals (source and ground), a rotation of the knob results in a change in electrical resistance and thus a change in output voltage of the center terminal (with respect to the ground terminal). The degrees of rotation per volt was directly measured for the potentiometer and power supply used in the experiments, which permitted the voltage output of the sensor to be read as wastegate opening position by the data acquisition system. Note that the degree-per-volt relationship of the sensor is dependent on the voltage source, and the potentiometer may have a dead band at the extremes of its rotation.
2.2.2 Instrumentation, Data Acquisition, and System Control

To measure rotational speed, the facility utilizes a PicoTurn BMV6 inductive sensor installed in the compressor housing of the turbocharger. The speed sensor was
configured to output a digital pulse signal, though an analog voltage output was also available. Flow temperature measurements were taken at seven locations by use of 1/16 in sheath Type-K thermocouples: turbine and compressor orifice flowmeter inlet, electric heater exit, turbine inlet, turbine exit, compressor inlet, and compressor exit. Two additional specialty 1/8 in Inconel sheath Type-K thermocouples were used within the electric heater chambers for monitoring purposes. At each location, one thermocouple was inserted to 1/2 diameter of the pipe and perpendicular to the pipe wall, except at compressor exit where an additional thermocouple was inserted to 1/4 diameter of the pipe at the same axial location but offset 180° circumferentially from the first. The latter two readings were averaged to improve the accuracy of compressor exit temperature. At each location, thermocouples were installed near the static pressure measurement.

Static pressure measurement was achieved in one of two ways. First, compressor inlet and turbine exit were instrumented with Kistler 4045A2 piezoresistive pressure transducers which have a 2 bar absolute range, 30 kHz natural frequency, and 120 °C maximum compensated operating temperature. The compressor exit was outfitted with a Kistler 4045A5 piezoresistive pressure transducer (5 bar absolute range). For thermal protection, the compressor Kistler transducers were installed within water jackets, and all were flush-mounted perpendicular to the inner diameter of their respective ducts.

Second, static pressure was measured in other locations by four wall taps separated circumferentially by 90° and combined in a piezometric ring arrangement. This method, verified by Blake (1976), is shown schematically in Fig. 2.5. All holes were drilled perpendicular to the pipe wall and 1/8 in diameter, and were de-burred and sanded at the inside surface of the duct. Half-couplings of NPT thread were welded on-center with the
holes, which were connected into a triple-T piezometric ring configuration via Tygon tubing.

![Diagram of Triple-T Piezometric Ring]

Figure 2.5: Triple-T Piezometric Ring

With equal tubing branch lengths, the triple-T setup most accurately yields the average static pressure. This average static pressure from the piezometric rings was measured by Validyne P55D differential pressure transducers. The transducers have an accuracy of ±0.25% of the full-scale range, including non-linearity, hysteresis, and non-repeatability. The pressure range is determined by interchangeable diaphragms, which were selected for each measurement location according to the levels encountered in order to maximize accuracy. The chosen diaphragms are listed in Table 2.2 for the two types of experiments executed. This method of measurement was used at turbine inlet and exit, as well as for static pressure at the inlet to and drop across of the orifice flowmeters. Validyne
transducers used to measure gauge pressure were referenced to barometric pressure, which was measured with a Setra model 270 transducer with 80-110 kPa range. Barometric pressure was then added during post-processing to obtain absolute levels.

Though two turbine exit static pressure measurements were sampled, the Validyne readings were used for all calculations since the average of its four circumferential static pressure locations is more accurate than the single location of the Kistler. Validyne transducer calibrations were performed using a Meriam model 351 smart manometer, with full-scale pressure of 200 in H$_2$O (49.82 kPa) and accuracy of ±0.05 in H$_2$O (12.45 Pa). Calibrations were carried out on all Validyne transducers right before each experiment.

Table 2.2: Validyne Diaphragm Full-Scale Pressures, Turbocharger Stand

<table>
<thead>
<tr>
<th>Location</th>
<th>Diaphragm</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>Performance Mapping</td>
</tr>
<tr>
<td><strong>Turbine</strong></td>
<td><strong>Inlet</strong></td>
</tr>
<tr>
<td>Flowmeter</td>
<td></td>
</tr>
<tr>
<td>Flowmeter</td>
<td></td>
</tr>
<tr>
<td>Differential</td>
<td>N/A</td>
</tr>
<tr>
<td><strong>Exit</strong></td>
<td></td>
</tr>
<tr>
<td><strong>Compressor</strong></td>
<td><strong>Inlet</strong></td>
</tr>
<tr>
<td>Flowmeter</td>
<td></td>
</tr>
</tbody>
</table>

Manipulation of the flow control valves and electric wastegate actuator, as well as sampling of all pertinent turbocharger data is achieved by the data acquisition system. Acquisition hardware includes a National Instruments (NI) cDAQ-9178 chassis with NI
9213 and NI 9265 modules for 24-bit multiplexed thermocouple measurement and 16-bit multiplexed sampling of 0-20 mA valve/actuator control output, respectively. A second chassis, NI PXIe-1073, is used with NI PXIe-6358 and NI PXI-6143S modules for 16-bit simultaneous sampling of analog input voltage from Kistler or Validyne pressure transducers, as well as for digital input from the speed sensor. Connections are made to the PXI modules via NI BNC-2110 connector blocks. The chassis are connected to a PC running Windows 7 operating system and LabVIEW data acquisition software. For turbine performance mapping the signals were sampled for 2.5 s at rates of 30 kHz, 400 Hz, 200 Hz, 100 Hz, and 20 Hz for Kistler, Validyne, thermocouple, speed sensor, and valve/actuator signals, respectively. When flow efficiency experiments were executed, the Validyne sampling rate was increased to 1 kHz and the sample time was lengthened to 10 s.

2.3 Flow Bench

Turbine flow efficiency analysis was also performed on the steady flow bench located within the Center for Automotive Research at The Ohio State University, which had been donated by Ford Scientific Laboratories. A schematic of the facility is provided in Fig. 2.6. As opposed to the turbocharger stand, the flow bench creates a vacuum to draw air through the turbine being investigated by placing it upstream of the intake of a Spencer Turbine Company multi-stage centrifugal compressor, which is driven by an electric motor. The flow bench is capable of pulling a maximum flow rate of 1150 SCFM (0.63 kg/s) with a pressure drop across the turbine of 35 in H₂O (8.72 kPa) and a flow rate of 350 SCFM (0.19 kg/s) at a maximum pressure drop of 90 in H₂O (22.42
kPa). To reduce noise levels in the testing environment, the compressor is located in an adjacent room.

Downstream of the test piece, flow rate is measured by a set of 7 nozzles mounted inside of a plenum tank. The nozzles, used independently or in parallel combination according to flow rate demand, have throat diameters ranging from 5/16 to 3-1/2 in (7.94 to 88.90 mm). Absolute pressure and temperature within the nozzle tank is measured by a Mensor model 11900-402 digital pressure gauge with 0-40 in Hg (0-135.5 kPa) range, and Omega PRX-AP-100 3-wire RTD air probe, respectively. Pressure drop across the nozzles is measured by an inclined manometer which has 12 scales calibrated to read in CFM, where each scale corresponds to a certain nozzle combination. The tank nozzles were calibrated at standard conditions of 300 K and 29.7 in Hg (100.58 kPa) by a set of four NIST traced flow nozzles installed as the upstream test piece. Tank nozzle inlet density compensation to calibrated conditions is accomplished by manually adjusting manometer inclination angle.

Figure 2.6: Flow Bench Facility Schematic (Christian, 2003)
Flow rate through the turbine is regulated by a pair of butterfly valves, 6 and 2 in diameter, for coarse and fine adjustment, respectively. Each valve size pair is connected by a linkage such that the compressor pulls an essentially constant flow rate while the ratio of air which is drawn through the test piece and auxiliary air inlet is varied. Each pair of valves is electrically positioned by a Milwaukee Controls 0A15M-025-96015 actuator. The actuators were controlled by two analog output channels of an NI AT-MIO-16 DAQ board in a PC running Windows XP operating system and LabVIEW data acquisition software.

Turbine inlet and exit ducts were shared between turbocharger stand and flow bench such that axial measurement locations with respect to the turbine were maintained. However, since the turbine inlet duct is the air entrance for the flow bench from ambient, a short bellmouth duct was added at the entrance to improve the flow profile immediately downstream of the inlet opening. A picture of the turbocharger installation on the flow bench is shown in Fig. 2.7.
Inlet and exit static pressure measurements were again accomplished by four wall taps separated circumferentially by 90° and combined in a triple-T piezometric ring arrangement at each location (refer to Section 2.2.2). Average static pressure from the piezometric rings was measured by Validyne P55D differential pressure transducers. The pressure ranges, determined by transducer diaphragms, were selected for each measurement location according to the levels encountered in order to maximize accuracy. The chosen diaphragms are listed in Table 2.3. Note that the Validyne for differential pressure was adjusted by its span control to output full-scale voltage for 41 in H₂O (10.21 kPa). Turbine inlet and exit transducers were referenced to barometric pressure, as
measured by Mensor model 14500 digital barometer with 26-32 in Hg (88.05-108.36 kPa) range. Validyne transducer calibrations were performed using a Meriam model 351 smart manometer, and were carried out on all transducers right before each experiment.

Table 2.3: Validyne Diaphragm Full-Scale Pressures, Flow Bench

<table>
<thead>
<tr>
<th>Location</th>
<th>Diaphragm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine</td>
<td></td>
</tr>
<tr>
<td>Inlet</td>
<td>3.5 or 14 in H₂O (0.872 or 3.49 kPa)</td>
</tr>
<tr>
<td>Differential</td>
<td>35 in H₂O (8.72 kPa)</td>
</tr>
<tr>
<td>Exit</td>
<td>55 or 90 in H₂O (13.70 or 22.42 kPa)</td>
</tr>
</tbody>
</table>

Temperature at turbine inlet and exit was measured with 1/16 in sheath Type-K thermocouples at an axial location near static pressure measurement. At each location, one thermocouple was inserted perpendicular to the pipe wall to pipe centerline depth. Ambient temperature was measured by a mercury thermometer.

The same 10 kΩ rotary potentiometer and associated mounting brackets used on the turbocharger bench as wastegate position sensor was applied on the flow bench setup. A different DC voltage source was utilized, however, and the wastegate degree-per-volt gain was altered proportionally. Additionally, the electric linear actuator was replaced by a turnbuckle to adjust and hold wastegate position, and a screw was thread into the speed sensor hole to prevent the rotor from turning.

Pressure (barometric and static), temperature, and wastegate position signals were sampled by a Keithley model KPCI-3116 DAQ board and collected by LabVIEW data
acquisition software at a rate of 1 kHz. Input signals were averaged for approximately 20 seconds before the experimental values were simultaneously saved and recorded.

### 2.4 Turbine Flow Configurations

Flow efficiency experiments were performed on the turbocharger stand and steady flow bench for flow passing through the turbine in three different configurations, which are introduced here as a reference for subsequent chapters. The schematics representing the flow configurations are presented in Fig. 2.8, where tee-junctions illustrate the flow split within the turbine housing downstream of the inlet flange (the “combined inlet”). The three flow configurations are described as follows:

1. First, flow may pass through the rotor with the wastegate closed; this will be referred to as “rotor-alone” (Fig. 2.8a) and designated in figures by “R*.” The asterisk indicates only that particular component of the turbine had gas moving through it, and the subscript $N_{par}$ indicates if a set of measurements was taken with the turbine wheel spinning ($N_{par}$ stands for “speed parameter” and will be defined in the next chapter).

2. Second, flow may travel through only the wastegate by plugging the rotor volute passage; this will be termed “wastegate-alone” (Fig. 2.8b) and designated by “WG*.” An example of how wastegate-alone flow is achieved is illustrated in Fig. 2.9, where a view from above the Honeywell turbine inlet flange shows the rotor volute plugged with a rubber stopper and sealed with clay.

3. Finally, rotor and wastegate may be subject to flow together “in parallel” (Fig. 2.8c), designated by “R&WG” in figures.
Figure 2.8: Turbine Flow Configuration Diagrams; (a) Rotor-Alone Flow, (b) Wastegate-Alone Flow, (c) Rotor and Wastegate Parallel Flow

Figure 2.9: Wastegate-Alone Flow Configuration with Plugged Rotor Volute
CHAPTER 3

TURBOCHARGER STAND EXPERIMENTAL RESULTS

In this chapter, experimental data acquired on the turbocharger stand outlined in Chapter 2 will be presented and discussed in four main sections, beginning with (1) a comparison of closed wastegate turbine performance maps measured at Ohio State versus those from the respective manufacturers. Turbine housing/bypass passage design (BorgWarner and Honeywell) and wastegate throat size (20, 21, and 26 mm) are then examined at four discrete wastegate valve openings (5°, 10°, 20°, and 40°) with respect to their effects on (2) open wastegate turbine performance with rotor spinning, (3) wastegate flow efficiency, and (4) the accuracy of the assumption of identical rotor and wastegate flow rates during individual and combined rotor-and-wastegate flow configurations, with rotor fixed and with rotor at mapped speed parameters. As a supplement to the last objective, an effort is also made to estimate the wastegate discharge coefficient during combined rotor-and-wastegate flow. Since some analyses were conducted with both experimental facilities, figures in this chapter which involve both will include the acronym TS (Turbocharger Stand) for clarity.
3.1 Closed Wastegate Turbine Performance

Mass flow parameter \((MFP)\) and efficiency performance characteristics for each turbine were initially measured with the wastegate closed to generate maps similar to Fig. 1.4. As \(MFP\), efficiency, and speed parameter \((N_{par})\) make use of turbine total inlet pressure and temperature, these quantities must be computed from the measured static values. Assuming steady, one-dimensional flow of an incompressible, inviscid fluid, total pressure at the turbine inlet measurement plane, \(p_{03}\), is the sum of static, \(p_3\), and dynamic pressures,

\[
p_{03} = p_3 + \frac{\rho_3 u_3^2}{2},
\]

where density, \(\rho_3\), is computed from the ideal gas equation of state, \(\rho = p/RT\) \((R\) being the specific gas constant), and the local velocity, \(u_3\), is found from measured mass flow rate, \(\dot{m}_T\), and the cross-sectional area of the duct at the measurement plane, \(A_3\), as

\[
u_3 = \frac{\dot{m}_T}{\rho_3 A_3}.
\]

Velocity may be normalized by local speed of sound, \(a\), to yield Mach number,

\[
M = \frac{u}{a} = \frac{u}{\sqrt{\gamma RT}},
\]

where \(\gamma\) is the ratio of specific heats. Mach number is used to determine total temperature, \(T_0\), by

\[
T_0 = T \left(1 + \frac{\gamma - 1}{2} M^2\right).
\]

\(MFP\) in the turbine maps can then be determined by

\[
MFP = \dot{m}_T \frac{\sqrt{T_{03}}}{p_{03}},
\]
and the speed parameter by

\[ N_{par} = \frac{N}{\sqrt{T_{03}}}, \]  

(3.6)

where \( N \) is the rotational speed of the turbocharger shaft in revolutions per minute (RPM).

Turbine map characteristics were determined here experimentally by sweeping the same speed parameters as those of the corresponding manufacturer. Since the turbocharger compressor acts as a brake on the turbine, the mass flow rate span that can be mapped for each speed parameter is limited by compressor choke and surge at the high and low end of the flow range, respectively. Figures 3.1 and 3.2 show the manufacturer mass flow rate maps for the BorgWarner and Honeywell turbines, respectively, with measurements taken on the Ohio State turbocharger stand superimposed as orange symbols. The Honeywell map (labeled as Allied Signal in Fig. 3.2) departs from convention by representing mass flow rate as a corrected quantity, as opposed to \( MFP \).

Corrected turbine mass flow rate (\( W^* \) in Fig. 3.2) is calculated as

\[ m_{T,cor} = m_T \frac{\sqrt{T_{03}}}{\sqrt{T_{ref}}} \frac{p_{03}}{p_{ref}}, \]  

(3.7)

where reference temperature, \( T_{ref} \), and pressure, \( p_{ref} \), were chosen by Honeywell to be 519 Rankine (288.33 K) and 29.92 in Hg (101.32 kPa), respectively. Therefore, the corrected mass flow rates in Fig. 3.2 may be converted to \( MFP \) in SI units via multiplication by 0.00756\( \sqrt{T_{ref}}/p_{ref} \), where 0.00756 (kg/s)/(lb/min) is the conversion factor. In each of Figs. 3.1 and 3.2, the manufacturer data symbol shapes have been
converted to match those of the acquired data according to identical speed parameter in order to illustrate the two data sets for a given speed parameter show good alignment, and a legend has been added to indicate these $N_{par}$ quantities. However, for each constant speed parameter the range of mass flow rates swept at Ohio State are at lower total-to-static expansion ratios ($ER_{ts}$) compared to the manufacturer data, and this arises from the difference in turbine inlet temperature used in the mapping process. It may be safely assumed that Honeywell and BorgWarner adhere to SAE turbocharger test code J1826, which stipulates $900 \pm 20 \degree C$ ($1173 \pm 20 K$) turbine total inlet temperature when mapping turbochargers for SI engines, whereas the highest turbine inlet temperature used on the Ohio State turbocharger stand for mapping was approximately $60 \degree C$ ($333 K$). The cooler turbine inlet flow results in reduced enthalpy available for expansion. Moreover, the actual rotational speed of the turbocharger is slower for same speed parameter, and since compressor pressure ratio is approximately proportional to the square of impeller tip speed, less power can be absorbed by the compressor. Once the compressor chokes, its ability to absorb turbine power ends, and any further increase in $ER_{ts}$ that might be available in the gas will only result in a speed increase.
Figure 3.1: Manufacturer Turbine $MFP$ Map with Experimental Data Superimposed, BorgWarner
Figure 3.2: Manufacturer Turbine Corrected Mass Flow Rate Map with Experimental Data Superimposed, Honeywell
Turbine isentropic efficiency is computed on a total-to-static basis since some kinetic energy of the gas remains at the turbine exit. The total-to-static isentropic efficiency is thus defined as the actual work output divided by isentropic expansion work between total inlet and static exit pressures, and is given by

$$\eta_{T,ts} = \frac{h_{03} - h_{04}}{h_{03} - h_{4s}} = \frac{1 - \frac{T_{04}}{T_{03}}}{1 - \left(\frac{p_4}{p_{03}}\right)^{(\gamma e - 1)/\gamma e}}, \quad \text{(3.8)}$$

where $h_{03}$ and $h_{04}$ are total specific enthalpy at turbine inlet and exit, respectively, $h_{4s}$ is specific enthalpy at turbine exit resulting from isentropic expansion, and $T_{04}$ is turbine total exit temperature. However, accurate measurement of turbine exit temperature is made difficult by the flow distribution across the measurement plane of the exit duct. Thus it is common practice to lump turbine total-to-static isentropic efficiency and turbocharger mechanical efficiency together by incorporating the ratio of actual compressor power and actual turbine power:

$$\eta_{T,ts}^{'} = \eta_{T,ts}\eta_{\text{mech}} = \left(\frac{P_T}{\dot{m}_T \Delta h_{s,T}}\right) \left(\frac{P_C}{P_T}\right) = \frac{P_C}{\dot{m}_T \Delta h_{s,T}}, \quad \text{(3.9)}$$

where $\eta_{\text{mech}}$ is the mechanical efficiency of the turbocharger, $P_T$ is turbine power, and $P_C$ is compressor power (SAE J1826, J922; Gamma Technologies, 2010). The combined turbine efficiency of Eq. (3.9) may be calculated in terms of measured variables by

$$\eta_{T,ts}^{'} = \frac{\dot{m}_C C_{pa} (T_{02} - T_{01})}{\dot{m}_T C_{pe} T_{03} \left[1 - \left(\frac{p_4}{p_{03}}\right)^{(\gamma e - 1)/\gamma e}\right]}, \quad \text{(3.10)}$$

where $\dot{m}_C$ is compressor mass flow rate, $C_{pa}$ is constant pressure specific heat of the compressor intake air, $T_{01}$ and $T_{02}$ are compressor inlet and exit total temperature,
respectively, $C_{pe}$ is constant pressure specific heat of the turbine inlet gas, and $\gamma_e$ is the ratio of specific heats of the turbine inlet gas.

Figures 3.3 and 3.4 show the manufacturer combined turbine efficiency maps for the BorgWarner and Honeywell turbines, respectively, with measurements taken on the Ohio State turbocharger stand superimposed as orange symbols. Though some characteristics of same speed parameter exhibit good alignment between data sets, the large difference in turbine inlet temperature used to map them precludes the type of direct comparison carried out for $MFP$. This is because turbine efficiency includes mechanical losses, which are a function of actual speed [as opposed to speed parameter, refer to Eq. (3.6)]. Incidence losses (dependent on actual rotor tip speed) and heat transfer effects also significantly differ at elevated turbine inlet temperatures. It is worth noting, however, that a few of the local slope changes at the high $ER_{ts}$ end of the BorgWarner efficiency characteristics (Fig. 3.3) for the three highest speed parameters (5929, 6823, and 7591 RPM/$\sqrt{K}$) are similarly replicated in the OSU-measured characteristics. For example, the 7591 RPM/$\sqrt{K}$ characteristic exhibits a sharp change in slope from positive to negative over a span of 0.1 expansion ratio ($ER_{ts}$=2.15-2.25 for the OSU data, $ER_{ts}$=2.9-3.0 for the BorgWarner data). The behavior duplication may possibly be explained by similar changes in compressor power [numerator of Eq. (3.10)], since the undulations occur in the highest $ER_{ts}$ segment of each respective efficiency characteristic of where the mated compressor would be approaching its choke limit. Also, the data collected at Ohio State is useful in illustrating what combined effect the turbine inlet temperature has on the shape and magnitude of turbine efficiency characteristics. For example, as $ER_{ts}$ decreases at constant speed turbine power development decreases [recall
isentropic power in the denominator of Eq. (3.10)] while bearing friction power is essentially constant. Mechanical losses therefore become a greater fraction of turbine power, which contributes to the steep drop off in combined efficiency at low $ER_{fs}$. 
Figure 3.3: Manufacturer Combined Turbine Efficiency Map with Experimental Data Superimposed, BorgWarner
Figure 3.4: Manufacturer Combined Turbine Efficiency Map with Experimental Data Superimposed, Honeywell
3.2 Open Wastegate Turbine Performance

Next, turbine $MFP$ and combined turbine efficiency ($\eta_{T,ts}'$) characteristics were measured for fixed wastegate openings of 5°, 10°, 20°, and 40° (maximum position for all turbines is approximately 45°, as limited by contact of wastegate crank arm with turbine housing). The same speed parameters measured in the foregoing closed wastegate maps were again swept from compressor choke to surge limits. Measured $MFP$ characteristics for the 20 mm wastegate BorgWarner, 26 mm wastegate BorgWarner, and 21 mm wastegate Honeywell turbines are presented in Figs. 3.5-3.7, respectively. Mass flow parameter quantity indicated in the figures is the total combined flow through turbine, with no distinction made of the relative contributions of rotor or wastegate flow. Also, the rotor-alone $MFP$ characteristics (wastegate closed, or 0°) have been carried over from Figs. 3.1 or 3.2 for reference.

In the $MFP$ maps, all three turbines exhibit an increase in total $MFP$ ($ER_{ts}$ fixed) which is relative to wastegate size from closed to 5° and from closed to 10° opening, the flow for the 26 mm wastegate BorgWarner turbine (Fig. 3.6) increasing the most. However, the Honeywell turbine (Fig. 3.7) behaves differently from that of the two BorgWarner turbines as the wastegate is opened farther. Where the Honeywell turbine results in a $MFP$ change from 20° to 40° opening similar to that from 5° to 10°, for example, the BorgWarner turbines result in a significantly diminished increase in $MFP$ from 20° to 40°. For example, at $ER_{ts}=1.60$, the $MFP$ of the 26 mm wastegate BorgWarner turbine (Fig. 3.6) increases 0.00319 kg·√K/s·kPa, or 18.3%, from 10° to 20°, whereas it only increases 0.00139 kg·√K/s·kPa, or 6.7%, from 20° to 40°. The latter observation may be expressed alternatively by considering that, at $ER_{ts}=1.60$, the $MFP$ at
20° is 93.7% of the MFP at 40°. Since the turbine mass flow rate at 40° may be reasonably approximated as the maximum, this means that roughly 94% of the maximum attainable flow area of the 26 mm wastegate turbine is achieved when the valve is approximately halfway open. Additionally, the lowest ER_{ts} mapped for each speed parameter appears to shift slightly higher for the BorgWarner turbines (Figs. 3.5 and 3.6) at 10° and 20° openings compared to 0°, 5°, or 40°, and the phenomenon is more pronounced for the 26 mm wastegate variant. Considering that the lowest ER_{ts} points for common N_{par} correspond to the same surge operating limit of the compressor, a possible explanation for the shift is that a reduced amount of turbine flow is passing through the rotor at 10° and 20° position for fixed ER_{ts}. To generate enough power for approximately the same map operating point (N and \dot{m}_C) of the mated turbocharger compressor (an example of the corresponding location on the turbine map is indicated by red arrows in Fig. 3.6), the turbine control valve must be opened farther, increasing the turbine inlet pressure (and therefore expansion ratio) and total MFP. As implied by Eq. (3.9),

\[
P_C = \eta_{mech} P_T , \tag{3.11}
\]

and substituting for turbine power,

\[
P_C = \eta_{mech} \dot{m}_T C_p e (T_{03} - T_{04}) . \tag{3.12}
\]

Since turbine inlet temperature, total temperature change across the turbine (\Delta T=T_{03}-T_{04}), and turbocharger speed are only negligibly different for the two indicated points, mass flow rate through the turbine rotor must be approximately equal if the same amount of power is generated, despite different ER_{ts}. At common ER_{ts}, therefore, less flow will be passing through the rotor and less power will be developed at 10° or 20° than at the other openings.
To illustrate the sensitivity of total $MFP$ to wastegate position, the percent increase in total $MFP$ at each wastegate opening with reference to the closed position, defined as

$$Percent \ MFP \ Increase = \frac{(MFP_{x^\circ WG} - MFP_{0^\circ WG})}{MFP_{0^\circ WG}} \times 100,$$

is plotted in Fig. 3.8. The broken lines in Fig. 3.8 are smoothed curves connecting points of common speed parameter and $ER_{ts}$ (3379 RPM/√K and 1.19, respectively, for Honeywell and 3505 RPM/√K and 1.20, respectively, for BorgWarner) and are used to demonstrate the similar behavior among all turbines at small openings ($5^\circ$ and $10^\circ$), as well as the greater gain in flow at $40^\circ$ for the Honeywell turbine. Additionally, for a fixed turbine the two lowest $N_{par}/ER_{ts}$ (circle and square symbols in Fig. 3.8) were generally found to have a larger $MFP$ increase than higher $N_{par}/ER_{ts}$ at each wastegate opening.
Figure 3.5: Measured Turbine $MFP$ Characteristics, BorgWarner, 20 mm Wastegate
Figure 3.6: Measured Turbine MFP Characteristics, BorgWarner, 26 mm Wastegate;
*Red Arrows Discussed in Text
Figure 3.7: Measured Turbine $MFP$ Characteristics, Honeywell, 21 mm Wastegate
Figure 3.8: Total Mass Flow Parameter Sensitivity to Wastegate Opening Angle; HW: Honeywell, BW: BorgWarner

Measured combined turbine efficiency $[\eta_{T,ts}]$ from Eq. (3.10)] characteristics for the 20 mm wastegate BorgWarner, 26 mm wastegate BorgWarner, and 21 mm wastegate Honeywell turbines are presented in Figs. 3.9-3.11, respectively, for fixed wastegate openings of 5°, 10°, 20°, and 40°. As expected, efficiency decreased with increased wastegate opening for most turbines and fixed $N_{par}$ and $ER_{ts}$, since the quantities are calculated with total mass flow rate passing through the turbine, rotor and wastegate. However, efficiency for the BorgWarner turbines (Figs. 3.9 and 3.10) is lower at 20° than 40° opening, which may be explained by a reduced amount of the total turbine flow.
passing through the rotor at 20° than 40° position for fixed $ER_{ts}$, as described previously in this section. Recall that the red arrows in Fig. 3.6 (26 mm wastegate BorgWarner turbine) indicate approximately the same compressor map operating point, and therefore compressor power. If fixed $ER_{ts}$ and $N_{par}=7591 \text{ RPM/√K}$ is considered for the BorgWarner turbine, a point at 40° opening does have slightly higher total turbine mass flow rate than a corresponding point at 20°; however, the mated compressor is operating at a point of higher power [numerator of Eq. (3.10)] at 40° that outpaces the gain in turbine mass flow [denominator of Eq. (3.10)], such that the calculated efficiency is higher than the point at 20°.

Verification of decrease in actual combined turbine efficiency ($\eta_{T,ts}'$) is not possible due to rotor flow rate not being independently measured (efficiency values in Figs. 3.9-3.11 include total flow passing through the turbine, rotor and wastegate). However, Capobianco and Marelli (2007) were able to estimate the flow passing through the rotor and found that the rotor combined efficiency decreased from closed wastegate position to a minimum at 20°-30° opening, after which efficiency increased. Further, Capobianco and Polidori (2008) found that the rotor contribution of total flow passing through the turbine decreased at 8° and 20° opening relative to closed wastegate, but was approximately the same at 35° as with the wastegate closed. Although these results (taken from an IHI turbocharger matched to a 1.4 L SI automotive engine) reinforce the observations of the BorgWarner turbines, the fact that the Honeywell does not exhibit a similar trend suggests this phenomenon is not common to all internally wastegated turbines of small automotive turbochargers.
Figure 3.9: Measured Turbine Efficiency Characteristics, BorgWarner, 20 mm Wastegate
Figure 3.10: Measured Turbine Efficiency Characteristics, BorgWarner, 26 mm Wastegate
Figure 3.11: Measured Turbine Efficiency Characteristics, Honeywell, 21 mm Wastegate
3.3 Wastegate Flow Efficiency

In the next phase of the investigation on the turbocharger stand, flow efficiency through the bypass was examined for the wastegate-alone configuration introduced in Section 2.4, and the flow capacity in this arrangement was compared to that of rotor-alone flow. Although a separate analysis of flow efficiency was also executed on the steady flow bench, the turbocharger stand was additionally utilized due to its ability to reveal the compressibility effects at elevated expansion ratios. To evaluate rotor-alone (wastegate closed) flow without the influence of the centrifugal pressure field created by the spinning turbine wheel, rotation was inhibited by inserting a screw into the hole tapped for the speed sensor on the compressor side of the turbocharger until contact was made with the compressor impeller. Wastegate-alone flow was accomplished by plugging the rotor volute passage immediately downstream of the bypass passage with a rubber stopper and sealing it with clay, as shown in Fig. 2.9.

3.3.1 Rotor-Alone and Wastegate-Alone Flow Capacity

First, rotor-alone and wastegate-alone MFP characteristics were measured for $ER_{ts}=1.05-2.00$ in 0.05 increments. For wastegate-alone flow, fixed wastegate openings of 5°, 10°, 20°, and 40° were again examined. Measured characteristics for the Honeywell (HW) 21 mm wastegate and rotor are presented in Fig. 3.12a, and characteristics for the BorgWarner (BW) 20 mm wastegate, 26 mm wastegate, and rotor are shown in Fig. 3.12b. Measurements confirmed that the rotor-alone MFP characteristic for both BorgWarner turbine housings was identical, and this common curve is represented by “+” symbols in Fig. 3.12b. For the Honeywell, flow through the wastegate is not observed to increase by a similar quantity for each subsequent increase
in opening; this trend is contrary to the one observed in Fig. 3.7 and suggests a change in wastegate flow behavior between configurations. The difference in Fig. 3.12a appears to be reduced $MFP$ levels at $10^\circ$ opening, since the $MFP$ gain from $20^\circ$ to $40^\circ$ is now greater than the $MFP$ increase from $5^\circ$ to $10^\circ$. This decrease is also evident by $MFP$ levels for 21 mm Honeywell wastegate (Fig. 3.12a) and 20 mm BorgWarner wastegate (Fig. 3.12b) which are similar for $5^\circ$ and $20^\circ$ opening but which are lower for the larger Honeywell wastegate at $10^\circ$ for a given expansion ratio. Both 21 mm Honeywell and 20 mm BorgWarner wastegates accept a similar amount of flow as their respective rotor around $40^\circ$, though the 21 mm wastegate $MFP$ levels drop below those of its rotor at $ER_{ts}=1.80$ (as the wastegate throat approaches choke), and the 20 mm wastegate levels drop below those of its rotor by $ER_{ts}=1.25$. The 26 mm BorgWarner wastegate flow, always greater than the 20 mm variant for same opening and expansion ratio, surpasses rotor-alone $MFP$ levels by the $20^\circ$ position.
Figure 3.12: R* and WG* Flow, Rotor Fixed, TS; (a) HW, (b) BW
3.3.2 Wastegate Discharge Coefficient

Introduced in Section 1.2, the discharge coefficient, $C_D$, is a measure of how close the actual mass flow rate through the wastegate compares to the calculated ideal. It is used to determine effective flow area, $A_{eff}$, of the actual bypass throat area by

$$A_{eff} = C_D A_{ref}.$$  \hfill (3.14)

The reference area, $A_{ref}$, is often chosen to be the wastegate throat cross-sectional area (Capobianco et al., 1990; Fraser, 2010), that is,

$$A_{ref} = \frac{\pi d_{wg}^2}{4}.$$ \hfill (3.15)

The discharge coefficient of each wastegate in the present study was calculated under the wastegate-alone flow configuration for the same set of valve head openings considered thus far ($5^\circ$, $10^\circ$, $20^\circ$, and $40^\circ$). Reference area for each wastegate was computed from its respective bypass throat diameter (20, 21, or 26 mm) and was held constant so as to clearly observe the performance of the wastegates in relation to valve position and expansion ratio.

To compute the discharge coefficient, ideal mass flow rate through the wastegate reference area must be determined, which assumes a perfect gas undergoing steady isentropic compressible flow in one dimension. Using the isentropic relation,

$$\frac{p_0}{p} = \left(\frac{T_0}{T}\right)^{\frac{y}{y-1}},$$ \hfill (3.16)

and Eq. (3.4), the ratio of total and static pressure as a function of Mach number, $M$, may be expressed as

$$\frac{p_0}{p} = \left(1 + \frac{y - 1}{2} M^2\right)^{\frac{y}{y-1}}.$$ \hfill (3.17)
where Mach number is calculated by Eq. (3.3). Next, the continuity equation is invoked, along with the perfect gas equation of state, to solve for mass flow rate according to

$$\dot{m} = \rho A_{ref} u = p A_{ref} M \sqrt{\frac{\gamma}{RT}}. \tag{3.18}$$

Equations (3.16-3.18) may then be combined to give unchoked ideal mass flow rate through the wastegate,

$$\dot{m}_{ideal} = A_{ref} \frac{p_{03}}{\sqrt{RT_{03}}} \left( \frac{p_4}{p_{03}} \right)^{1/\gamma} \left\{ \frac{2\gamma}{\gamma - 1} \left[ 1 - \left( \frac{p_4}{p_{03}} \right)^{(\gamma-1)/\gamma} \right] \right\}^{1/2}, \tag{3.19}$$

where static pressure at the bypass throat is assumed to be equal to turbine exit static pressure (that is, $p_4 = p_{throttle}$). If the flow through the wastegate becomes choked (that is sonic, $M = 1$), the ratio of static and total pressure becomes

$$\frac{p_4}{p_{03}} = \left( \frac{2}{\gamma + 1} \right)^{\gamma/(\gamma - 1)}, \tag{3.20}$$

and the choked ideal mass flow rate thus evolves to

$$\dot{m}_{ideal} = A_{ref} \frac{p_{03}}{\sqrt{RT_{03}}} \sqrt{\gamma} \left( \frac{2}{\gamma + 1} \right)^{(\gamma+1)/2(\gamma-1)}. \tag{3.21}$$

Finally, the ideal mass flow rate is compared to the actual mass flow rate, as measured by the turbine orifice flowmeter on the turbocharger stand, yielding the discharge coefficient:

$$C_D = \frac{\dot{m}_{actual}}{\dot{m}_{ideal}}. \tag{3.22}$$

The result of this calculation for all three wastegates is presented in Fig. 3.13. The coefficients, found to be repeatable over multiple experiments, are similar in magnitude to those reported by Capobianco et al. (1990) and Fraser (2010) for internal
turbocharger wastegates. For each opening of 5°, 10°, and 20°, the discharge coefficient is observed to have an inverse relationship with wastegate size, except for relatively similar values at 10° for 21 mm and 26 mm valves. The latter observation is consistent with prior results of the Honeywell turbine at 10°. That the $C_D$ for the 20 mm BorgWarner wastegate is consistently higher than that of the 26 mm wastegate, despite the two sharing similar turbine housings, suggests a possible difference in flow passage geometry which results in greater flow separation for the 26 mm variant (a qualitative analysis of the turbine flow passage geometry will be discussed in Chapter 5).

![Figure 3.13: Measured Wastegate Discharge Coefficient, TS](image)

Figure 3.13: Measured Wastegate Discharge Coefficient, TS
Greater flow separation, and frictional losses, may also be a result of higher flow velocity in the combined turbine inlet passage for the 26 mm wastegate, for same opening and \( ER_{ts} \). The increase in discharge coefficient of the Honeywell wastegate beyond that of both BorgWarner wastegates at 40° is one indication of its different flow passage geometry and wastegate placement, and reinforces the trend of a greater \( MFP \) increase from 20° to 40° for open wastegate turbine flow (recall Fig. 3.8).

Wastegate effective flow area, calculated according to Eq. (3.14), is provided in Fig. 3.14. Because of its inferior discharge coefficient at reduced openings, the 21 mm Honeywell wastegate has a smaller effective area than the 20 mm BorgWarner wastegate at 5° and 10° opening, though it begins to surpass the 20 mm wastegate at 20°. Also, despite its consistently lowest discharge coefficient, the 26 mm wastegate has the largest effective flow area of all sizes examined for wastegate-alone configuration and fixed opening angle. The latter trend is inconsistent with Fig. 3.8 for the 40° wastegate position, where the Honeywell turbine has greater \( MFP \) increase than the 26 mm wastegate BorgWarner turbine for combined rotor-and-wastegate flow. This conflict between flow configurations suggests a reduction in bypass quantity during rotor-and-wastegate parallel flow as compared to wastegate-alone individual flow, and will be analyzed further in Section 3.4. Figures 3.13 and 3.14 also show a slight dependency of discharge coefficient on \( ER_{ts} \) for all wastegates, though for a fixed opening the rate of change is observed to generally be greatest for the smallest throat. This behavior is likely due to increased air compressibility at higher expansion ratios and is consistent with the results of Capobianco et al. (1990) for wastegate valves, as well as the results of Woods and Khan (1966) and Gault et al. (2004) for large lifts of engine intake poppet valves.
(where the minimum geometric flow area is closer to that of the wastegate openings considered). However, increase in $ER_{ts}$ has very little effect on the general shape of $A_{eff}$ versus wastegate opening for a given turbine.

![Diagram showing measured wastegate effective flow area vs. wastegate opening, TS]

**Figure 3.14: Measured Wastegate Effective Flow Area vs. Wastegate Opening, TS**

### 3.4 Rotor and Wastegate “Parallel” Flow

Next, the rotor-and-wastegate parallel flow configuration introduced in Section 2.4 was evaluated. As with rotor-alone flow in Section 3.3.1, the rotor was inhibited from spinning for this set of experiments. $MFP$ characteristics were again measured for $ER_{ts}=1.05$-$2.00$ in 0.05 increments and fixed wastegate openings of $5^\circ$, $10^\circ$, $20^\circ$, and $40^\circ$. 

74
Results of the measurements for the Honeywell (HW) 21 mm wastegate with rotor are presented in Fig. 3.15a with the rotor-alone curve, and results for the BorgWarner (BW) 20 mm wastegate with rotor, 26 mm wastegate with rotor, and rotor-alone measurements are shown in Fig. 3.15b. The Honeywell characteristics exhibit similar behavior with wastegate opening as those in the map in Fig. 3.7, as total $MFP$ increases steadily for a doubling of the valve position. Characteristics for the BorgWarner turbines are also similar to the maps in Figs. 3.5 and 3.6, most notably the meager increase in total $MFP$ from 20° to 40° opening. A comparison of all three turbines under rotor-and-wastegate parallel flow is provided in Fig. 3.16. The 21 mm wastegate Honeywell turbine and 26 mm wastegate BorgWarner turbine are observed to have comparable total $MFP$ levels at 5°, 10°, and 20° openings, with Honeywell magnitudes being slightly higher. At 40° position however, the flow rate through the Honeywell is much greater, which was also observed to lesser extent in Fig. 3.8.
Figure 3.15: R&WG and R* Flow, Rotor Fixed, TS; (a) HW, (b) BW
3.4.1 Error of Predicting R&WG Flow with Wastegate-Alone $C_D$, Rotor Fixed

As fully described in Section 1.2, automobile manufacturers have only the wastegate discharge coefficient (wastegate-alone) and closed wastegate turbine map (rotor-alone flow) at their disposal to estimate turbine mass flow rate with bypass open (rotor-and-wastegate). Current 1-D engine simulation computer software utilize these two sets of data by treating the rotor and wastegate as two nozzles or orifices operating in parallel under the same conditions ($ER_{ts}$ and total inlet pressure and temperature), without modeling any internal flow losses within the turbine housing. Flows through the rotor and the bypass are predicted individually and arithmetically summed to predict total
rotor-and-wastegate mass flow rate. To examine the validity of this approach, wastegate-alone (WG*) and rotor-alone (R*) MFPs at same valve position and ER_{ts} were added and compared to the corresponding total MFP for rotor-and-wastegate flow (R&WG). This analysis was initially carried out for the flow efficiency data of Section 3.3 (zero rotor speed) and is presented first, followed by the same comparison for the map performance data of Section 3.1 (rotor spinning).

The comparison of summed individual flows versus parallel rotor-and-wastegate flow for the fixed-rotor data of Section 3.3 is presented in Figs. 3.17 and 3.18 for Honeywell and BorgWarner turbines, respectively. Summing rotor and wastegate individual MFPs (R^*+WG^*) results in relatively good agreement with rotor-and-wastegate MFP for the Honeywell turbine (Fig. 3.17) at all openings, with a slightly positive difference at 5°, 20°, and 40° positions for the majority of the ER_{ts} range, and a slightly negative difference at 10°. The latter disparity is consistent with diminished wastegate-alone flow reported in earlier figures (for example, Fig. 3.12a). However, summing the individual flows for the two BorgWarner turbines results in significant error compared to rotor-and-wastegate flow. As depicted in Figs. 3.18a and 3.18b (20 mm and 26 mm wastegate, respectively), the summed flows are consistently greater in magnitude, and the difference from rotor-and-wastegate levels increases with wastegate opening. Moreover, the divergence is generally greater for the 26 mm than the 20 mm wastegate, particularly at 20° and 40° openings where the error of the larger wastegate is considerable. This error from adding the individual flows may be expressed as

$$\text{Percent } R^* + WG^* \text{ Error} = \frac{[(MFP_{R^*} + MFP_{WG^*}) - MFP_{RWG^*}]}{MFP_{RWG^*}} \times 100 \text{, (3.23)}$$
where $MFP$s are evaluated at same wastegate opening and $ER_{ts}$. The result of this calculation for all turbines is provided in Fig. 3.19. The wide range of error magnitudes indicates that although some turbine bypass designs are conducive to the isolated treatment of rotor and wastegate flows (such as the Honeywell turbine), the accuracy of this approach is as yet unpredictable and may be substantially poor.

Figure 3.17: $R^*+WG^*$ vs. R&WG Flow, Rotor Fixed, TS; 21 mm WG
Figure 3.18: R*+WG* vs. R&WG Flow, Rotor Fixed, TS; (a) 20 mm, (b) 26 mm WG
Note that error trends with respect to wastegate opening of the two BorgWarner turbines are not similar. Since the increase in \( MFP \) levels from 20° to 40° position for the smaller BorgWarner is the same for R&WG as for R*+WG* (Fig. 3.18a), the error is approximately the same at those openings. This is disparate from the behavior of the 26 mm wastegate Borg Warner (Fig. 3.18b), which exhibits a marked increase in \( MFP \) from 20° to 40° position for R*+WG* as compared to R&WG quantities, leading to greater error at 40°. Also, there is a ramped increase in error evident for 20 mm wastegate data at 10° opening from approximately \( ER_{ts} = 1.40-1.80 \), which is due to a dip in the rotor-and-wastegate \( MFP \) curve over the same expansion ratio range (refer to Fig. 3.15a or 3.18a).
This dip was found to be repeatable, though the $ER_{ts}$ at which the most sudden shift in flow occurred (1.65 to 1.70 in Fig. 3.18a) was found to vary by ±0.05 depending on the magnitude of the changes in the turbine control valve as the characteristic was swept. Furthermore, when the data in question was being recorded, the sudden shift in $MFP$ was accompanied by a change in character of the sound of the flow as perceived by the turbocharger stand operator. It is thus theorized that the drop in $MFP$ is due to an increased separation of the flow at the bypass port resulting in diminished flow through the wastegate.

The overall higher error for the BorgWarner turbines suggests the efficiency of the bypass passage to accept flow during wastegate-alone configuration was greatly facilitated by the presence of the rubber stopper. Without a plug to help direct the gas to the wastegate, the flow was less inclined to turn into the bypass during rotor-and-wastegate configuration. This artificial increase of flow efficiency through the wastegate was also evident to a lesser extent for the Honeywell turbine. A theory regarding the difference in error behavior is presented in Chapter 5, where geometry of the flow passages and bypass port placement are analyzed.

In order to eliminate the errors indicated in Fig. 3.19 yet still operate within the contemporary wastegate modeling approach of 1-D engine simulation codes, an effort was made to determine the bypass discharge coefficient during rotor-and-wastegate flow. This was accomplished by executing the following three steps:

1. Rotor-alone $MFP$ was converted to mass flow rate with turbine total inlet pressure and temperature ($p_{03}$ and $T_{03}$, respectively) from rotor-and-wastegate data at matched operating condition of $ER_{ts}$.  

82
2. Rotor-alone mass flow rate was subtracted from rotor-and-wastegate mass flow rate at the foregoing matched operating point to estimate the bypass mass flow rate \((\dot{m}_{R&WG} - \dot{m}_R^* \approx \dot{m}_{WG})\). Following current convention of 1-D simulation software, this step assumes mass flow rate through the rotor is unchanged during rotor-and-wastegate flow. The estimated bypass MFP characteristics are shown in Fig. 3.20 with their respective rotor-alone characteristics.

3. Estimated wastegate mass flow rate from Step 2 was divided by the ideal wastegate mass flow rate [recall Eq. (3.22)], the latter being calculated according to Eq. (3.19) or (3.21) with the pressure and temperature data of the rotor-and-wastegate operating point. Again, following the current approach of 1-D codes, this assumes the expansion ratio across the wastegate throat is the same as the quantity measured external to the turbine housing.
Figure 3.20: WG Estimate during R&WG Flow, Rotor Fixed; (a) HW, (b) BW
The estimated discharge coefficient resulting from this exercise, denoted as $C_D'$, is presented in Fig. 3.21a for all three turbines based on the fixed-rotor flow efficiency data. The corresponding wastegate effective flow area, $A_{eff}'$, is included in Fig. 3.21b. As expected, little difference is observed for the Honeywell discharge coefficient, as compared to wastegate-alone configuration of Fig. 3.13. The 26 mm wastegate BorgWarner discharge coefficient, however, changed substantially, most noticeably for the 40° opening where the 0.77-0.79 values of Fig. 3.13 shrunk to approximately 0.48-0.49. As Fig. 3.20b shows, the estimated wastegate $MFP$ characteristics in rotor-and-wastegate configuration for all fixed openings are now below the levels of rotor-alone for common $ER_{ts}>1.10$ (refer to Fig. 3.12b for comparison to wastegate-alone). The drop in rotor-and-wastegate flow for the 20 mm BorgWarner at 10° opening and $ER_{ts}=1.70$ is manifested in Fig. 3.21a as a drop in discharge coefficient, since the rotor contribution is assumed to be unaltered. In the effective flow area estimate of Fig. 3.21b, the reduced 26 mm wastegate discharge coefficient estimate results in the 21 mm wastegate beginning to surpass it at 20° valve position. This trend is more consistent with the mass flow sensitivity curves in Fig. 3.8 than the wastegate-alone effective area of Fig. 3.14, as alluded to in Section 3.3.1. Also, the estimated effective flow areas for the 20 mm and 26 mm BorgWarner wastegates show less spread with $ER_{ts}$ (indicated by color in Fig. 3.21b) than in the wastegate-alone configuration, though this observation may be an artifact of assuming unchanged flow through the rotor. In the next and final analysis of the chapter, it is determined if the $R^*+WG^*$ error and R&WG estimated discharge coefficient with the rotor fixed is comparable to that when the turbine wheel is rotating at the mapped speed parameters.
Figure 3.21: Estimated Wastegate (a) $C_D'$ and (b) $A_{eff}'$ in R&WG Flow, Rotor Fixed, TS
3.4.2 Error of Predicting R&WG Flow with Wastegate-Alone $C_D$, Rotor at $N_{par}$

The examination of predicting total rotor-and-wastegate flow from wastegate discharge coefficient and rotor-alone data was extended to the measured map $MFP$ characteristics in Section 3.1 where the turbine wheel was rotating. Since there are many more operating points on each speed parameter characteristic in Figs. 3.5-3.7 than were measured for wastegate-alone flow for the same range of $ER_{ts}$, a different approach was taken than in the preceding subsection. Instead of adding wastegate-alone $MFP$s from Section 3.3.1 to rotor-alone $MFP$s of Section 3.1 at same $ER_{ts}$, the wastegate-alone discharge coefficients were applied to the closed wastegate operating points to generate the same number of wastegate-alone points. This exercise was accomplished by carrying out the following four steps for each of the turbines:

1. First, a least-squares curve fit of the discharge coefficient data in Fig. 3.13 was determined as a function of $ER_{ts}$ for each opening degree (4 fits per turbine; provided in Appendix A).

2. Next, ideal wastegate mass flow rate was calculated according to Eq. (3.19) or (3.21) for each closed wastegate (rotor-alone) operating point in Figs. 3.5-3.7.

3. The ideal wastegate mass flow rate was then multiplied by the four discharge coefficients for 5°, 10°, 20°, and 40° openings (determined by $ER_{ts}$ of the point and curve fit equations from Step 1) to determine the “actual” wastegate-alone mass flow rate [recall Eq. (3.22)] for each valve position.

4. Finally, the wastegate mass flow rate from Step 3 was converted to $MFP$ and added to the rotor-alone $MFP$ of same $ER_{ts}$ to compute the $R^{+\text{WG}}_{N_{par}}$ quantity.
The comparison of summed individual flows against the measured open wastegate turbine maps (the latter being unaltered from Figs. 3.5-3.7 aside from the scale of the vertical axis) is presented in Figs. 3.22 and 3.23 for Honeywell and BorgWarner turbines, respectively.

As with the R*+WG* characteristics with the rotor fixed, those for the Honeywell turbine (Fig. 3.22) with the rotor spinning are relatively similar to the mapped rotor-and-wastegate curves. A slightly different behavior is observed, however. Though the error of the summed flows is slightly negative at low ERts like the fixed-rotor data, at elevated expansion ratios (>1.70) the error is again negative for all wastegate openings. Moreover, the consistently reduced R*Npar+WG* MFPs for all ERts at 10° opening are even lower than the fixed-rotor levels.

The summed rotor-alone and wastegate-alone flows for the two BorgWarner turbines (Fig. 3.23a and 3.23b for 20 mm and 26 mm wastegate, respectively) again exceed the rotor-and-wastegate parallel flows by a considerable amount. Also consistent with the fixed-rotor data, the error of the summed flows is observed to increase with wastegate opening, as well as with wastegate size. However, since the slope of the closed wastegate characteristics is observed to have a pronounced difference from the open wastegate rotor-and-wastegate characteristics of corresponding speed parameter, the error of the R*Npar+WG* curves generally increases with expansion ratio. This trend is most noticeable for the majority of the speed parameters at 20° and 40° opening, as well as for the highest speed parameter at 5° and 10°. The error of the R*Npar+WG* flows for all turbines, calculated according to Eq. (3.23), is presented in Fig. 3.24 and may be
compared with the fixed-rotor error of Fig. 3.19. Though much of the effects of the spinning rotor compared to the fixed rotor have already been discussed, Fig. 3.24 indicates that the levels of the error have shifted to lower values for nearly all wastegate openings and $ER_{ts}$ of each turbine. It is this error that would be incurred if the current wastegate modeling approach in 1-D engine simulation codes (that is, independently evaluating the wastegate as an effective orifice area subject to the same upstream and downstream pressures and temperatures as the rotor, and summing its predicted flow with the rotor-alone quantity) were followed.

Figure 3.22: $R^*+\text{WG}^*$ vs. R&WG Flow, Rotor at $N_{par}$; 21 mm WG
Figure 3.23: $R^*+WG^*$ vs. R&WG Flow, Rotor at $N_{par}$; (a) 20 mm, (b) 26 mm WG
An effort was again made to determine the bypass discharge coefficient during rotor-and-wastegate flow using the same three steps introduced in the preceding subsection. When matching rotor-and-wastegate and rotor-alone operating points on an $ER_{ts}$ basis, care was taken to limit the maximum difference in expansion ratio to 0.01, though the vast majority of matched points are within 0.005 (for example, 91.5% of the matched points for the 26 mm wastegate have $ER_{ts}$ within 0.005 of each other). The estimated discharge coefficient resulting from this exercise, again denoted as $C_D'$, is compared to that of the fixed-rotor data in Figs. 3.25 and 3.26 for Honeywell and BorgWarner turbines, respectively. Immediately apparent is the fact that discharge
coefficient derived from rotor-and-wastegate flow differs depending on whether or not the turbine wheel is rotating. Aside from a small range of $ER_{ts}$ for the Honeywell turbine at $40^\circ$ opening, the estimated discharge coefficient with a spinning rotor is consistently higher. Therefore, implementation of $C_D'$ values derived from fixed-rotor flow efficiency tests within 1-D engine simulation tools would under-estimate open wastegate total turbine $MFP$. For the BorgWarner turbines considered, however, application of fixed-rotor $C_D'$ would be more accurate than using wastegate-alone $C_D$ and this is explored in the modeling work of Chapter 7.

![Figure 3.25: Comparison of R&WG Estimated $C_D'$; 21 mm WG](image)

92
Figure 3.26: Comparison of R&WG Estimated $C_D'$; (a) 20 mm, (b) 26 mm WG
CHAPTER 4

FLOW BENCH EXPERIMENTAL RESULTS

In this chapter, experimental data acquired on the steady flow bench outlined in Chapter 2 (refer to Figs. 2.6 and 2.7) will be presented and discussed in two main sections: (1) an examination of turbine housing/bypass passage design (BorgWarner and Honeywell) and bypass throat size (20, 21, and 26 mm) on wastegate flow efficiency at eight discrete wastegate valve openings (5°-40° in 5° increments), and (2) a review of the accuracy of summing rotor and wastegate individual flow rates to predict total flow rate in the combined/parallel configuration for the same wastegate sizes and valve openings. An estimation of the wastegate discharge coefficient during combined rotor-and-wastegate flow with rotor fixed is also included as part of the latter section. Figure captions in this chapter may include the acronym FB (Flow Bench) to differentiate analyses here from those also presented in the previous chapter.

4.1 Wastegate Flow Efficiency

Although a flow analysis conducted on the turbocharger stand is already presented in the previous chapter, such a facility at Ohio State is unique in North America outside of turbocharger manufacturers. Automobile manufacturers instead rely on flow efficiency data obtainable with a flow bench. Therefore, a set of results similar to those presented in Sections 3.3 and 3.4 were measured on the flow bench introduced in Chapter
2, with rotor fixed for all experiments. The analysis here begins with the comparison of flow capacity in the wastegate-alone and rotor-alone configurations followed by a quantification of wastegate flow efficiency. Turbine wheel rotation was again inhibited by inserting a screw into the hole tapped for the speed sensor on the compressor side of the turbocharger until contact was made with the compressor impeller, and wastegate-alone flow was accomplished by plugging the rotor volute passage in the same manner as outlined in Chapter 3.

4.1.1 Rotor-Alone and Wastegate-Alone Flow Capacity

First, rotor-alone and wastegate-alone mass flow rate characteristics were measured for differential static pressures across the turbine ($\Delta p = p_3 - p_4$) of 4-40 in H$_2$O (0.996-9.96 kPa) in 4 in H$_2$O (0.996 kPa) increments. For wastegate-alone flow, fixed wastegate openings of 5°-40° in 5° increments were studied. Measured characteristics for the Honeywell (HW) 21 mm wastegate and rotor are presented in Fig. 4.1, and characteristics for the BorgWarner (BW) 20 mm wastegate and 26 mm wastegate, each with rotor, are shown in Fig. 4.2a and 4.2b, respectively. Mass flows have not been nondimensionalized to $MFP$ as in Chapter 3 since turbine inlet total pressure and temperature are equal to ambient conditions in the flow bench for all turbines and flow configurations considered (refer to experimental setup in Section 2.3), and these quantities did not vary appreciably from day to day.

The Honeywell wastegate mass flow rate is observed to increase by a relatively similar quantity for incremental openings from 5° to 25° for a given pressure drop, after which further openings result in only modest gains. Both BorgWarner wastegates exhibit a somewhat similar increase in mass flow rate from 5° to 10° and 10° to 15° opening for
fixed pressure drop, but starting at 20° the flow rate increase for a change in opening quickly diminishes. Measured independently, the wastegate mass flow rate characteristics reach rotor-alone levels at openings in reverse order of their size: 20 mm wastegate at 30°-35°, 21 mm wastegate between 25° and 30°, and 26 mm wastegate between 10° and 15°. Also for the BorgWarner wastegates, mass flow rate quantities for the 26 mm wastegate are always greater than the 20 mm wastegate for fixed opening and pressure drop, as expected.

Figure 4.1: R* and WG* Flow, Rotor Fixed, FB; Honeywell 21 mm WG
Figure 4.2: R* and WG* Flow, Rotor Fixed, FB; BorgWarner (a) 20 mm, (b) 26 mm WG
4.1.2 Wastegate Discharge Coefficient

As a means of quantifying wastegate flow efficiency, the discharge coefficient was introduced in Section 1.2 and its calculation was explained in Section 3.3.2 preceding the results on the turbocharger stand. Discharge coefficient was again determined for the wastegate-alone flow presented in Figs. 4.1 and 4.2, where consistent with the turbocharger bench, reference area chosen for the ideal mass flow rate calculation of Eq. (3.19) was taken to be the bypass throat cross-sectional area of the respective turbine considered and was constant for all wastegate openings. Barometric pressure and ambient temperature measured in the flow laboratory were used for total turbine inlet pressure and temperature, respectively, and turbine exit static pressure was measured in the same manner and at same axial location in the exit duct as on the turbocharger stand. The result of this calculation for Honeywell 21 mm wastegate is presented in Fig. 4.3, and for BorgWarner 20 mm and 26 mm wastegate in Figs. 4.4a and 4.4b, respectively. A comparison of all three wastegates at openings of $5^\circ$, $10^\circ$, $20^\circ$, and $40^\circ$ is also provided in Fig. 4.5.

As remarked in the preceding chapter, the measured discharge coefficients are similar in magnitude to those reported by other authors for internal turbocharger wastegates. Additionally, the coefficients were found to be repeatable over multiple experiments. Figures 4.3 and 4.4 show a trend of minimal change in discharge coefficient with increasing pressure drop at a fixed opening for all wastegates. Based on the isentropic assumption of Eq. (3.17), the maximum Mach number for the flow at each wastegate throat (occurring at $40^\circ$ opening) was calculated to be just over 0.4, such that the foregoing observation is in line with the expectation that air compressibility is of little
effect. For the two smallest openings (5° and 10°), the smallest wastegate is the most efficient, while the discharge coefficients for 26 mm and 21 mm wastegates are very similar, values being marginally greater for 26 mm wastegate. The smallest wastegate continues to be the most efficient up to 25° position, at which point the 21 mm wastegate is approximately equal to it. The 21 mm wastegate becomes the most efficient for the largest three openings (30°-40°). It is worth repeating, however, that the Honeywell turbine has different flow passage geometry and wastegate placement within the housing than the BorgWarner turbine, and the effects of this fact will also be seen in the next subsection for rotor-and-wastegate flow.

Figure 4.3: Measured Wastegate Discharge Coefficient, FB; 21 mm WG
Figure 4.4: Measured Wastegate Discharge Coefficient, FB; (a) 20 mm, (b) 26 mm WG
It is also interesting to note that though $C_D$ values for the smaller 20 mm wastegate are shifted approximately 0.1 higher for a fixed opening and pressure drop, the two BorgWarner wastegates in Figs. 4.4a and 4.4b exhibit nearly identical discharge coefficient behavior as the wastegate is opened, as expected. As remarked in Chapter 3, this apparent contradiction in discharge coefficient level suggests greater flow separation for the larger BorgWarner wastegate and will be explored in Chapter 5.

Next, with the discharge coefficient data presented in Figs. 4.3 and 4.4, wastegate effective flow area was calculated from Eq. (3.14) and is provided in Fig. 4.6. Because of its lower discharge coefficient levels at reduced openings, the 21 mm Honeywell
wastegate has a smaller effective area than the 20 mm BorgWarner wastegate from $5^\circ$-$15^\circ$ openings. With a nearly equivalent discharge coefficient at $25^\circ$ position, the 21 mm wastegate effective area exceeds that of the 20 mm wastegate. The 26 mm wastegate makes up for having the lowest discharge coefficient with its greater actual physical area, and has the largest effective flow area of all sizes studied for fixed opening angle.

Figure 4.6: Measured Wastegate Effective Flow Area vs. Wastegate Opening, FB

4.1.3 Rotor and Wastegate Parallel Flow

Next, the rotor-and-wastegate parallel flow configuration was examined. Mass flow rate characteristics were again measured for differential static pressures across the
turbine of 4-40 in H₂O (0.996-9.96 kPa) in 4 in H₂O (0.996 kPa) increments and for fixed wastegate positions of 5°-40° in 5° increments. Results for the Honeywell (HW) 21 mm wastegate with rotor are presented in Fig. 4.7 along with the rotor-alone curve, and results for the BorgWarner (BW) 20 mm wastegate with rotor, 26 mm wastegate with rotor, and rotor-alone measurements are shown in Figs. 4.8a and 4.8b, respectively. As observed for wastegate-alone measurements, Honeywell characteristics increase steadily as valve position is opened up to 25°.

Figure 4.7: R&WG and R* Flow, Rotor Fixed, FB; HW 21 mm WG
Figure 4.8: R&WG and R* Flow, Rotor Fixed, FB; BW (a) 20 mm, (b) 26 mm WG
Characteristics for the BorgWarner turbines are also similar to the wastegate-alone curves of Figs. 4.2a and 4.2b, including the mass flow rate increases from 5° to 10° and 10° to 15°, followed by meager increases at openings greater than 20°. However, the 26 mm wastegate-and-rotor levels show a behavior slightly different from the 20 mm variant above 15° opening: though the characteristics for both become crowded, from 20°-30° the characteristics nearly collapse for the 26 mm, with levels at 30° only slightly higher for pressure drops of 4 kPa and larger.

The characteristics of all three turbines under rotor-and-wastegate parallel flow are compared in Fig. 4.9. Values in the figure have been nondimensionalized to $MFP$ (vertical axis) and total-to-static expansion ratio (horizontal axis) to enable a fair comparison, and the greatest effect observed is in terms of a lateral shift of the individual points with respect to each other at common pressure drop (turbine inlet total pressure and temperature being approximately equal, relative placement with respect to the vertical axis is largely unaffected). Broken lines are included at 5° and 40° positions, connecting data points of same turbine and common wastegate opening to facilitate observation. Note that $MFP$ levels of the two BorgWarner turbines with respect to each other (opening and $ER_{ts}$ constant) are in order of size, as expected from results shown in Fig. 4.6. However, $MFP$ quantities for the 21 mm wastegate Honeywell turbine are greater than those for both BorgWarner turbines at all openings for fixed $ER_{ts}$. At 40° position in particular, the Honeywell turbine $MFP$ levels are much greater. This result is in direct conflict with Fig. 4.6 and, as noted in the previous chapter, suggests a reduction in bypass quantity during rotor-and-wastegate flow as compared to wastegate-alone flow for the BorgWarner turbines.
Automobile manufacturers typically rely on the wastegate discharge coefficient to estimate turbine mass flow rate with bypass open (rotor-and-wastegate) by predicting the wastegate contribution independently and then adding it to rotor-alone levels. To examine the validity of treating rotor and wastegate as two nozzles or orifices operating in parallel under the same conditions, wastegate-alone (WG*) and rotor-alone (R*) MFPs at same valve position and total-to-static expansion ratio were added and compared to the corresponding total MFP for rotor-and-wastegate flow (R&WG). The comparison of
summed individual flows ($R^* + WG^*$) versus parallel rotor-and-wastegate flow is presented in Figs. 4.10 and 4.11 for Honeywell and BorgWarner turbines, respectively. Summing rotor and wastegate individual $MFP$s ($R^* + WG^*$) results in relatively good agreement with rotor-and-wastegate $MFP$ for the Honeywell turbine at all openings (Fig. 4.10), with a slightly positive difference at 10° and 20° positions for the majority of the total-to-static expansion ratio range. However, summing the individual flows for the two BorgWarner turbines results in significant error compared to rotor-and-wastegate flow.

Figure 4.10: $R^* + WG^*$ vs. R&WG Flow, Rotor Fixed, FB; 21 mm WG
Figure 4.11: R*+WG* vs. R&WG Flow, Rotor Fixed, FB; (a) 20 mm, (b) 26 mm WG
As depicted in Figs. 4.11a and 4.11b (20 mm and 26 mm wastegate, respectively), the summed flows are consistently greater in magnitude, and the difference from rotor-and-wastegate levels increases with wastegate opening. Further, the divergence is generally greater for the 26 mm than the 20 mm wastegate, particularly at 20° and 40° openings where the error of the larger wastegate is considerable. The error from adding the individual flows, evaluated according to Eq. (3.23) for MFPs at same wastegate opening and total-to-static expansion ratio, is provided in Fig. 4.12 for all turbines.

Figure 4.12: R*+WG* MFP Error, Rotor Fixed, FB
Although the Honeywell turbine is conducive to the isolated treatment of rotor and wastegate flows, the accuracy of this approach is shown to vary greatly, even between the two BorgWarner wastegate throat sizes which reside within similar turbine housings. Though the two BorgWarner turbines yield similar error at 5° opening, the error trends with respect to wastegate opening are different; the error at 20° and 40° position for the smaller BorgWarner is very similar, while the error for the larger wastegate variant increases significantly from 20° to 40°. The higher error for the BorgWarner turbines at all openings may suggest the flow efficiency of the bypass passage during wastegate-alone configuration was improved by the presence of the rubber stopper. With no plug to help direct the gas to the wastegate, the flow was evidently less inclined to turn into the bypass during rotor-and-wastegate configuration. This artificial increase of flow efficiency through the wastegate was also evident to a much lesser extent for the Honeywell turbine. The behavior was also observed for similar measurements on the turbocharger stand, and a theory regarding the change in bypass flow between configurations is presented in Chapter 5.
The primary purpose of this chapter is to compare the fixed-rotor experimental results presented in Chapters 3 and 4 as measured on the turbocharger stand and steady flow bench, respectively. Measured wastegate discharge coefficient is considered first, followed by a discussion of the error introduced by using the coefficient to predict open wastegate turbine flow capacity. Common trends between the two facilities on the effect of wastegate size for the BorgWarner turbine are also reviewed. The chapter will conclude with a comparison of bypass port design and its observed effects.

5.1 Wastegate Discharge Coefficient

The discharge coefficients for each of the three wastegates studied, as measured on the turbocharger stand (Section 3.3.2) and flow bench (Section 4.1.2), are shown in Fig. 5.1a against turbine total-to-static expansion ratio as a function of wastegate opening. Open symbols represent turbocharger stand (TS) measurements, while the filled/solid symbols indicate flow bench (FB) measurements. The data point of highest $ER_{ts}$ for each wastegate size and opening degree for the flow bench results corresponds to a turbine pressure drop of 40 in H$_2$O (9.96 kPa), the highest pressure drop attainable with that facility. Also, recalling from the previous two chapters, all discharge coefficients were calculated using Eq. (3.19) or (3.21) and Eq. (3.22) under steady flow, where
reference area was fixed at the respective wastegate throat cross-sectional area.

Wastegate effective flow area, computed from Eq. (3.14), is compared in Fig. 5.1b, where turbocharger stand data is evaluated at $ER_{ts}=1.10$ to eliminate the effect of $C_D$ discontinuities (discussed in the next section) observed below that expansion ratio. 

The relative order of the wastegates with respect to discharge coefficient is observed to be maintained between facilities for each fixed opening of $10^\circ$, $20^\circ$, and $40^\circ$. However, the $C_D$ values measured on the flow bench are observed to be consistently higher than those measured on the turbocharger stand (aside from the Honeywell wastegate at $5^\circ$ opening), with the discrepancy generally increasing with decreasing opening. As Fig. 5.1b illustrates, effective flow area increases the most from closed to $5^\circ$ and tapers off for subsequent openings, such that mass flow rate sensitivity to wastegate position is greater for smaller openings (recall Fig. 3.8); therefore, the foregoing trend of discrepancy is not entirely unexpected. In fact, the $C_D$ discrepancy may be attributed to a combination of two things: measurement inaccuracy and a difference in the incoming flow field which results in local flow separation change. Regarding the former, note that the $C_D$ disparity is greatest for the smallest wastegate at the smallest openings and expansion ratios, where mass flow rates are lowest and measurement inaccuracies will be of greater effect. The orifice flowmeter of the turbocharger stand is also at the bottom of its calibrated flow range for the $ER_{ts}$ of interest at $5^\circ$ and $10^\circ$, whereas the flow bench manometer scale used has a readability resolution of 0.1 SCFM, or 0.0000552 kg/s, increments. Further, the rotor-alone flow characteristics for both turbines were measured to be slightly lower on the turbocharger stand than on the flow bench for $ER_{ts}<1.10$, after which the curves from the two facilities converged.
Figure 5.1: Measured Wastegate (a) $C_D$ and (b) $A_{eff}$; Facility Comparison
With regard to a difference in flow separation, note that some of the turbocharger stand data exhibits a discontinuity in measured discharge coefficient at $ER_0 \leq 1.15$ (such as 20 mm BorgWarner and 21 mm Honeywell wastegate at $20^\circ$ opening from $ER_0 = 1.10$ to 1.15) not observed for the flow bench data, and that these discontinuities were found to be repeatable (demonstrated in the following analyses).

Therefore, an investigation was carried out to determine if any of the more obvious factors may have contributed to either $C_D$ measurement inaccuracy or an alteration of the incoming flow between facilities. The first such endeavor involved the replacement of Validyne pressure transducer diaphragms on the turbocharger stand with those of reduced full-scale range, in an effort to improve the sensitivity and thus accuracy of static pressure measurements both at the turbine and at the turbine orifice flowmeter. These revised, thinner diaphragms are listed in Table 5.1, and may be compared with the “standard” full-scale ranges (refer to Table 2.2) used thus far.

Table 5.1: Validyne “Thin” Diaphragm Full-Scale Pressures, Turbocharger Stand

<table>
<thead>
<tr>
<th>Location</th>
<th>Diaphragm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flowmeter Inlet</td>
<td>8 psi (55.16 kPa)</td>
</tr>
<tr>
<td>Flowmeter Differential</td>
<td>55 in H$_2$O (13.70 kPa)</td>
</tr>
<tr>
<td>Inlet</td>
<td>90 in H$_2$O (22.42 kPa)</td>
</tr>
<tr>
<td>Differential</td>
<td>90 in H$_2$O (22.42 kPa)</td>
</tr>
<tr>
<td>Exit</td>
<td>3.5 in H$_2$O (0.872 kPa)</td>
</tr>
</tbody>
</table>

Additional measurements were made, and Figs. 5.2a and 5.2b illustrate good discharge coefficient agreement between the two sets of diaphragms for Honeywell (HW) and
BorgWarner (BW) wastegates, respectively, where open symbols represent thin diaphragm values and filled symbols, standard diaphragms. The thinner diaphragms also permitted confirmation that the measured discharge coefficient discontinuity on the turbocharger stand was not an artificial byproduct of the standard Validyne diaphragms at low pressure drop across the orifice of the turbine flowmeter.

The second factor examined regarding measured discharge coefficient disparity between facilities was turbine inlet density. Since the flow bench draws flow by vacuum, with the turbine inlet as the air entrance (refer to Figs. 2.6 and 2.7), turbine inlet static pressure and thus inlet density is very close to ambient conditions, reducing by only slight amounts as vacuum is increased to generate pressure drop. On the other hand, the turbocharger stand supplies compressed air to the turbine inlet such that its inlet static pressure and density continually increase to create greater expansion ratios across the turbine (recall Fig. 2.1). For operating point of same $ER_{in}$, the turbine inlet density will usually therefore be higher on the turbocharger stand, unless the electric heaters are utilized to raise turbine inlet temperature. The latter scenario was carried out to deliberately lower turbine inlet density to flow bench levels for the Honeywell wastegate at 20° opening. Additional discharge coefficient measurements with use of thinner diaphragms, with and without use of electric heaters for the Honeywell 21 mm wastegate at 20° are shown in Fig. 5.3b as red and aqua open symbols, respectively. The corresponding turbine inlet density values are provided in Fig. 5.3a, and in both figures the green symbols indicate the same data presented in Fig. 5.1a (standard diaphragms on flow bench or turbocharger stand).
Figure 5.2: Wastegate $C_D$ Diaphragm Comparison, TS; (a) HW, (b) BW
Figure 5.3: Honeywell 21 mm Wastegate at 20° Opening; (a) $\rho_3$, (b) $C_D$
The discharge coefficients measured with thin diaphragms on the turbocharger stand with and without electric heater usage show very good agreement. However, as Figs. 5.3a and 5.3b illustrate, when turbine inlet density on the turbocharger stand is either just below, equivalent to, or just above the levels measured on the flow bench, the discharge coefficient measured on the turbocharger stand is not observed to approach the levels acquired on the flow bench. Thus for the Honeywell wastegate at 20° position, turbine inlet density and diaphragm sensitivity do not contribute to the measured discharge coefficient discrepancy between facilities.

### 5.1.1 Wastegate Discharge Coefficient Discontinuity on Turbocharger Stand

Measured discharge coefficient discontinuity on the turbocharger stand has been shown to be repeatable with more sensitive diaphragms (Figs. 5.2a and 5.2b), and with turbine inlet density comparable to that of the flow bench data (Fig. 5.3b). Though not shown in Fig. 5.3b, possible hysteresis was also examined for the Honeywell wastegate at 20° with thin diaphragms by incrementing from higher to lower $ER_{ts}$ when taking measurements; no change in discharge coefficient values or expansion ratio of discontinuity was observed. Plus, the discontinuity is not due to an orifice change of the flowmeter. It is worth noting, however, that the ramped increase in discharge coefficient observed on the flow bench and turbocharger stand in Fig. 5.3b occurred at approximately the same turbine inlet Reynolds number, which may be defined as

$$\text{Re}_3 = \frac{4m_r}{\pi d_3 \mu_3}, \quad (5.1)$$

where $d_3$ is the inner diameter of the turbine inlet duct, and $\mu_3$ is the turbine inlet dynamic viscosity calculated according to the Sutherland equation (Christian, 2003),
\[ \mu_3 = \frac{b \sqrt{T_3}}{S + \frac{T_3}{1}} \]  

(5.2)

where \( b = 1.458 \times 10^3 \text{ kg/(m} \cdot \text{s} \cdot \sqrt{\text{K}} \) and \( S = 110.4 \text{ K} \). The foregoing observation is shown in Fig. 5.4, where the horizontal axis of Fig. 5.3b has been converted to turbine inlet Reynolds number, and a gray background indicates where the ramped increase in discharge coefficient roughly begins. The discharge coefficient increase for each facility starts at approximately \( \text{Re}_3 = 4-5 \times 10^4 \), and after an initial discontinuity for the turbocharger stand data, the slope of each is similar. A sudden rise in pressure drop across the orifice of the turbine flowmeter was observed for the turbocharger stand data.

![Graph showing turbine inlet Reynolds number versus discharge coefficient](image)

Figure 5.4: Honeywell 21 mm Wastegate \( C_D \) at 20° Opening vs. Reynolds Number
in Fig. 5.4, which led to a sudden increase in measured mass flow rate and thus a jump in discharge coefficient. The relatively sudden jump in pressure drop across the flowmeter orifice at low $ER_{ts}$ was observed for nearly all wastegate sizes and openings (wherever a measured discharge coefficient discontinuity was observed), and it occurred for different flowmeter orifice sizes and therefore at different pressure drop values. Flowmeter calibration, flowmeter orifice, and Validyne pressure transducer diaphragm can most likely be excluded, therefore, as the cause of the discharge coefficient discontinuity.

The discharge coefficients of Fig. 5.2b (BorgWarner wastegates) measured with thin diaphragms was also plotted against turbine inlet Reynolds number, and the result is shown in Fig. 5.5a. Considering that the two BorgWarner wastegates share a similar turbine housing, and that for a given opening the two wastegates exhibit a discharge coefficient jump at approximately the same turbine inlet Reynolds number, a change in flow separation within the turbine housing may be causing the discontinuity. If a change in flow separation is occurring, there may be a change in sound pressure level. Therefore, a fast Fourier transform (FFT) of turbine exit pressure from the Kistler transducer (located approximately 6.4 cm downstream of the turbine exit flange) was performed to determine root-mean-square pressure, $p_{rms}$, in the frequency domain. Sound pressure level (SPL) in decibel (dB) at each discrete frequency was then calculated according to

$$\text{SPL} = 20 \times \log_{10} \left( \frac{p_{rms}}{p_{ref}} \right), \quad (5.3)$$

where reference pressure, $p_{ref}$, is equal to $2 \times 10^{-5}$ Pa. A computation of total SPL from 0-1 kHz was then executed by
Total SPL_{0-1 kHz} = 10 \times \log_{10} \left( \sum_{i=0 \ Hz}^{1000 \ Hz} 10^{-\frac{\text{SPL}_{i}}{10}} \right), \quad (5.4)

(clearly excluding static pressure at 0 Hz) and the result for BorgWarner wastegates at 20° and 40° position is provided in Fig. 5.5b. Note how SPL levels dip at approximately Re₃=6-7x10⁴, which coincides with Reynolds number of discharge coefficient increase in Fig. 5.5a for same wastegate openings. The foregoing SPL trend may be evidence of reduced flow separation, leading to the abrupt increase in measured discharge coefficient at 20° and 40°. Though the search for the exact cause of the discontinuity is ongoing, experimental data suggests it is likely not an artificiality, and its absence from flow bench measurements reinforces the idea of Section 5.1 that a difference in local flow separation is contributing to the discrepancy in wastegate-alone flow efficiency between facilities. It is worth noting that the $C_D$ discontinuity and difference between facilities occurs at expansion ratios below those of the lowest mapped speed parameter, such that it will not impact open wastegate 1-D flow predictions in Chapter 7. Moreover, it is unlikely the studied wastegates will be operated at low $ER_{ts}$ on engine; since they are passively controlled by compressor boost pressure, the valves will open or close when turbine $ER_{ts}$ is higher.
Figure 5.5: Measured BorgWarner WG* Data at Low $ER_3$; (a) $C_D$, (b) Total SPL.
5.2 Error of Predicting R&WG Flow with Wastegate-Alone $C_D$

As fully explained in prior chapters, predicting turbine flow capacity with bypass open is currently accomplished in 1-D engine simulation codes by applying the wastegate-alone discharge coefficient to the pressure and temperature boundary conditions of the turbine to determine bypass mass flow rate, and then summing that quantity with the turbine map mass flow rate, acquired with wastegate closed. The error introduced by adding the independently evaluated flows was discussed in Sections 3.4.1 and 4.2 for the turbocharger stand and flow bench, respectively. Those results are duplicated together in Fig. 5.6 for comparison. Though the values of the errors are generally not observed to be consistent between facilities, certain trends are observed:

- The error of the 21 mm wastegate Honeywell turbine is the least of the three sizes and, except at the lowest $ER_{ts}$ measured on the turbocharger stand, is within ±4% of measured total $MFP$.
- For a given opening degree and $ER_{ts}$, the 20 mm wastegate BorgWarner turbine is always lower in error than the 26 mm variant.
- For BorgWarner turbines, error increases with wastegate opening degree.
- The error of the 20 mm wastegate BorgWarner turbine at 40° opening is similar to or slightly greater than the error at 20° ($ER_{ts}$ fixed), while for the 26 mm variant the error at 40° is much greater than at 20°.

Since the BorgWarner discharge coefficients measured on the flow bench were found to be generally higher than those on the turbocharger stand for fixed opening and $ER_{ts}$ (Fig. 5.1a), the higher $R^*+WG^*$ error observed in Fig. 5.6 for the BorgWarner turbines logically follows.
Figure 5.6: R*+WG* MFP Error, Facility Comparison

5.3 Wastegate Size Effects

The purpose of this section is to summarize the trends seen with both experimental facilities regarding the two BorgWarner wastegate sizes and provide insight into their possible cause. The 21mm wastegate Honeywell turbine is not included in the analysis because turbine inlet cross-sectional area, turbine flow passages, and wastegate placement within the housing, are not similar among the two manufacturers, precluding direct comparison. However, the foregoing differences in turbine design will be explored in Section 5.4.
It has been remarked previously that the two BorgWarner wastegate sizes have similar turbine housings, including identical inlet and exit cross-sectional area and shape, and to all appearances, the same bypass port location with respect to the turbine inlet flange and to the mean radius of the turbine volute. Since it was concluded in Chapter 3 that rotor-alone ($R^*$) $MFP$ characteristics were the same for the two BorgWarner housings (rotor fixed and rotor spinning), and since a common turbine wheel was employed for both, it may be safely assumed that the $A/r$ ratio (a design parameter equal to cross-sectional area of the volute where it first admits gas to the wheel, divided by the radius from the turbocharger centerline to the centroid of that area) of the housings is also identical. Pictures of the two BorgWarner housings facing the exit flange are provided in Fig. 5.7, and close-up photographs of the wastegate throat and valve head seat are shown in Fig. 5.8. The foregoing physical similarities, combined with the views in these photographs, suggest the only major differences between the two housings may be the diameter of the wastegate throat and the diameter of the valve head.

Figure 5.7: BorgWarner Turbine Housing, Exit Flange; (a) 20 mm, (b) 26 mm WG
Yet, as the preceding sections of this chapter have shown, an additional difference must exist between the two housings, as the following trends are consistently observed on both the turbocharger stand and flow bench:

- Wastegate-alone discharge coefficient is lower for 26 mm than 20 mm diameter bypass throat for fixed opening degree (Fig. 5.1a).
- Error of applying the discharge coefficient and rotor-alone flow characteristics to determine turbine flow capacity with bypass open is higher for 26 mm than 20 mm diameter bypass throat for fixed opening degree (Fig. 5.6).

Therefore, greater flow separation within the housing must be occurring for the 26 mm wastegate BorgWarner turbine, opening degree and $ER_t$ constant. Figure 5.9 shows the BorgWarner turbine housing with the inlet and bypass port, to provide perspective for the inlet flow passages pictured in Fig. 5.10.
Figure 5.9: BorgWarner Turbine Housing, Facing Inlet Flange

Figure 5.10: BorgWarner Bypass Port Length Estimate at Shortest Part; (a) 20 mm, (b) 26 mm WG
In Figs. 5.10a and 5.10b, flow entering the housing would either continue on around the volute or make a right-hand turn into the bypass port and exit out the wastegate throat, the latter highlighted as a green oval in the figures (the 20 mm wastegate throat is hidden from view) and the green arc indicating where the bypass port branches off the main passage. The shortest length of the bypass port (located at the top of its circumference, as shown in Figs. 5.10a and 5.10b) was found to be significantly different between the two wastegate sizes, with this length being longer for the 20 mm wastegate. With a longer length, the port may better facilitate the turning of the flow and thereby reduce the separated flow region behind the near wall of the port; however, the exact mechanism is unknown.

5.4 Bypass Port Design Effects

Where the previous section examined the difference in behavior resulting from wastegate throat size of the BorgWarner turbine, this section will analyze the placement and design of the port between the Honeywell and BorgWarner turbines studied in the present work. The aim of this comparison is to gain understanding into the contributing factors behind the following trends, observed consistently on or between turbocharger stand and flow bench:

- Honeywell turbine has a better wastegate-alone $C_D$ than BorgWarner turbines at the largest opening (Fig. 5.1a).
- Honeywell turbine exhibits a larger change in $MFP$ from $20^\circ$ to $40^\circ$ opening for fixed $ER_{ts}$ compared to BorgWarner turbines for rotor-and-wastegate configuration (Figs. 3.16 and 4.9).
- Honeywell turbine is more conducive to isolated treatment of rotor and wastegate contributions during rotor-and-wastegate parallel flow than BorgWarner turbines, that is, $R^*+WG^*$ error is minimal for all openings (Fig. 5.6).

- With few exceptions, discrepancies in measured values between facilities are typically less for Honeywell turbine than BorgWarner turbines for fixed opening angle (Figs. 5.1 and 5.6)

The difference in bypass port location within the turbine housing for the two manufacturers is illustrated in Fig. 5.11, where the photograph of each turbine has been taken facing the exit flange. Green lines have been drawn to delineate the volute path, and an orange circle the bypass throat.

Figure 5.11: Bypass Location with Respect to Volute; (a) BorgWarner, (b) Honeywell
As shown, the Honeywell bypass port branches off of the straight, converging combined inlet passage near its centerline, whereas the BorgWarner bypass is situated below the mean radius of the curved inlet passage, which is also converging but to a lesser extent than the Honeywell. Pictures of the flow passages within the BorgWarner and Honeywell turbine housings are provided in Figs. 5.12 and 5.13, respectively. The Honeywell turbine passages more closely resemble a dividing type 1 tee junction (Christian, 2003), with differences being a converging through duct and a side branch/bypass port that may not intersect it at precisely 90°. Regardless, this flow split design has been shown to have similar wastegate flow efficiency for both wastegate-alone and rotor-and-wastegate (as estimated) configurations during steady flow, indicating that flow separation at the bypass port is comparable between the foregoing configurations. With very good flow efficiency at a near-maximum position of 40°, the design achieves desired maximum bypass flow with a smaller bypass port and throat (for example, recall the difference in total MFP between 21 mm Honeywell wastegate and 26 mm BorgWarner wastegate in Fig. 3.16).

Figure 5.12: View into BorgWarner Turbine, Facing (a) Inlet Flange, (b) Wastegate Seat
Alternatively, to connect the wastegate throat to the inlet passage the BorgWarner bypass port utilizes a half-bowl shaped depression which is indicated in Fig. 5.12. Under rotor-and-wastegate configuration, axial velocity will increase toward the outer radius of the volute curve due to the momentum of the flow, which combined with bypass port location biased to the low velocity tight inner radius may create a separated flow region within the depression. The higher mean axial velocities resulting from increased mass flow rate at large valve openings may increase the severity of the separation, and could possibly explain the meager increase in total $MFP$ from $20^\circ$ to $40^\circ$ under rotor-and-wastegate flow. Under wastegate-alone flow, the foregoing flow behavior and local flow separation would be altered significantly and would not be representative of the flow pattern in the parallel configuration, therefore leading to large $R^*+WG^*$ errors for the BorgWarner turbines.
A model of each turbine was created with a commercially available 1-D, unsteady, time-domain code (GT-Power, 2010) and is presented in this chapter. Two approaches to modeling internally wastegated turbines are outlined: the 1-D code default, which does not physically model the parallel flow paths through the housing, and an alternative proposed here, which does. Closed wastegate turbine maps measured at Ohio State were curve fit, extrapolated, and interpolated with MATLAB, and input as a text file to the code in place of its data pre-processor. Comparison of Ohio State map fits to the manufacturer MFP map data is also discussed.

6.1 Flow Simulation Code

The computational tool used in the present work is a 1-D engine simulation code which numerically solves the governing equations for unsteady, compressible fluid flows. The code solves the nonlinear conservation equations of mass, momentum, and energy, along with the ideal gas equation of state, using an explicit time integration method. Ducts in the flow system of the model are spatially discretized along their length to control volumes of length $dx$, to which the conservation equations are applied. One-dimensional mass conservation is written as
\[
\frac{dm}{dt} = \sum_{in} m - \sum_{out} \dot{m} ,
\]

where \( m \) is mass in the control volume and \( \dot{m} \) is mass flow rate across the control volume boundaries. One-dimensional momentum conservation may be expressed as

\[
\frac{d\dot{m}}{dt} dx = A dp + \sum_{in} (\dot{m}u) - \sum_{out} (\dot{m}u) - f \frac{4A}{D_h} \left( \frac{1}{2} \rho u |u| \right) dx - K \left( \frac{1}{2} \rho u |u| \right) A ,
\]

where \( A \) is cross-sectional area of flow, \( u \) is velocity at the boundary of the control volume, \( f \) is friction coefficient, \( D_h \) is hydraulic diameter, and \( K \) is loss coefficient.

Finally, one-dimensional energy conservation is represented by

\[
\frac{d(me)}{dt} = -p \frac{dV}{dt} + \sum_{in} (\dot{m}h_0) - \sum_{out} (\dot{m}h_0) - \dot{h}_c A_s (T_f - T_w) ,
\]

where \( e \) is total internal energy (internal energy plus kinetic energy) per unit mass, \( V \) is volume, \( h_0 \) is total specific enthalpy, \( \dot{h}_c \) is heat transfer coefficient, \( A_s \) is heat transfer area, \( T_f \) is fluid temperature, and \( T_w \) is wall temperature. Solution of the foregoing governing equations results in the determination of scalar fluid properties (pressure, temperature, density, enthalpy, and internal energy) in each control volume, and vector properties (mass flow rate and velocity) at the control volume boundaries.

6.2 Ducting and Turbine Housing Flow Passages

The 1-D model of the external ducting and internal housing flow passages of the turbine is described in this section. Two different approaches to model wastegate flow are defined: the GT-Power default option and the alternative proposed here. The former is discussed first.
6.2.1 1-D Code Default Approach

The 1-D code default approach for modeling wastegates built into the housing consists of an option within the turbine component, but this option does not physically model the two parallel paths through the housing. Instead, in order to allow incoming gas to bypass the rotor inside the control volume of the turbine component, the measured effective diameter of the bypass throat is entered in the “Wastegate Diameter” attribute of the turbine component, and this diameter may be set constant or dynamically altered [for example, a controller may change the effective diameter during simulation based on compressor exit pressure (Gamma Technologies, 2010)]. No modeling is therefore done of the physical flow split passages, bypass throat, or cavity within the turbine housing at rotor exit; within the code the two parallel paths through the turbine are only conceptual. With this treatment, the wastegate and rotor experience the same total-to-static expansion ratio as imposed on the turbine component by the adjacent upstream and downstream control volumes. Rotor and wastegate flows are predicted individually and summed to determine total turbine mass flow rate, an approach which has been shown in the present work to be potentially highly erroneous.

The inlet boundary condition of the default model, shown at the far left of Fig. 6.1, is set as a constant total pressure and temperature, specified for each operating point to be the measured quantities [calculated from Eqs. (3.1) and (3.4), respectively] of the incoming air. The boundary condition is connected to the turbine inlet duct by a bellmouth, with forward and reverse discharge coefficients set to 1.0. The inlet duct is modeled as a straight pipe of circular cross-section, its constant inner diameter set equal to the measured dimension of the actual duct (specific to BorgWarner or Honeywell...
turbine), and its length set equal to the model spatial discretization length $dx$ of 5 mm (that is, one control volume). At the downstream end of the inlet duct, the turbine component represents the entire housing and rotor, inlet flange to exit flange, and provides turbine performance by look-up tables (described in Section 6.3). As mentioned in the preceding paragraph, the turbine component handles the prediction of wastegate flow without physically modeling the parallel flow path. Downstream of the turbine, the exit duct is similarly modeled as the inlet duct, its length equal to $dx$ and its constant inner diameter set equal to the measured dimension of the actual duct. The turbine exit duct is connected to the exit boundary condition by a bellmouth, with forward and reverse discharge coefficients equal to 1.0. Finally, exit boundary condition is set as a constant static pressure and temperature, specified for each operating point as the measured quantities. In the foregoing manner, the inlet and exit boundary conditions specified for each operating point are essentially imposed directly at the turbine. If duct lengths were set equal to the actual axial distance between static pressure measurement plane and turbine flange (inlet or exit), frictional losses in the ducts would result in imposition of an expansion ratio across the turbine different from that used to create the
turbine map, and the \( MFP \) output by the code would be inaccurate for the operating point considered.

6.2.2 New Alternative Approach

A new alternative approach is now proposed which models the wastegate valve and the flow junctions within the turbine housing: the flow split at the bypass port on the inlet side, and on the exit side where rotor and wastegate flows combine. Whereas the 1-D code default model essentially represents the actual turbine from inlet flange to exit flange (the “turbine” component embodying the entire housing), this alternative model represents the actual turbine from inlet flow split to exit combining flow junction within the housing (the “turbine” component now embodying just the volute and rotor, with the “Wastegate Diameter” attribute fixed at zero). That is, the extent of the ducting in the new model, inlet bellmouth to exit bellmouth, now represents the turbine housing. The short turbine inlet and exit ducts used in the 1-D code default model (Fig. 6.1) are not included here for simplicity, as their length renders their presence inconsequential to the comparison of the two models in Chapter 7. Note that this proposed model allows the expansion ratio across the wastegate to differ from that across the rotor by modeling the dividing and combining flow junctions and bypass flow path.

The proposed model, shown in Fig. 6.2, again begins with an inlet boundary condition set to be the measured total pressure and temperature of the incoming air for a given operating point, as well as a bellmouth of forward and reverse discharge coefficients equal to 1.0. A straight pipe of circular cross-section, with a length set to \( dx \) (5 mm), connects the inlet bellmouth to the inlet flow split.
The constant inner diameter of the pipe was determined based on measurement of the circumference of the actual flow passage immediately preceding the bypass port (specific to BorgWarner or Honeywell turbine), and is indicated as $d_{FS,\text{inlet}}$ in Fig. 6.3. Next is the inlet “flowsplit” component, for which the nominally 1-D code accounts for conservation of momentum in three dimensions (Gamma Technologies, 2010). The flowsplit geometry is defined by $d_{FS,\text{inlet}}$, a length of $d_{WG}$, and with measurement of the circumference of the actual flow passage immediately succeeding the bypass port, $d_{FS,\text{exit}}$, as shown in Fig. 6.3.
The $d_{FS,\text{exit}}$ flowsplit opening, which leads to the rotor, is modeled concentric with the inlet to the flowsplit, and the $d_{WG}$ opening, which leads to the wastegate valve, is modeled at 90° to $d_{FS,\text{inlet}}$ and $d_{FS,\text{exit}}$, as depicted in Figs. 6.2 and 6.3. Based on the foregoing dimensions ($d_{WG}$, $d_{FS,\text{inlet}}$, and $d_{FS,\text{exit}}$), volume and surface area of the inlet flowsplit were calculated and input to the code. Between the flowsplit and turbine component is a straight pipe of circular cross-section, its constant inner diameter set equal to $d_{FS,\text{exit}}$ and its length set equal to $dx$. The curved section of the volute which feeds the rotor is not physically modeled.

The bypass port connects the $d_{WG}$ opening of the inlet flowsplit to the wastegate valve and is modeled with an orifice of forward and reverse discharge coefficients set to 0.97, to model the smoothness of the interface radius of the port, and a straight pipe of
circular cross-section, its constant inner diameter set equal to $d_{WG}$ and its length equal to the measured dimension for each turbine. The wastegate valve is modeled using a “throttle” component, for which an array of forward and reverse discharge coefficients may be entered versus throttle angle. Calibration of the throttle for measured wastegate flow efficiencies is discussed in Chapter 7, where use is made of the wastegate-alone model shown in Fig. 6.2b which displaces the turbine component with end caps at the flowsplits.

A pipe of circular cross-section and length $dx$ connects the downstream end of the turbine component (labeled “Volute and Rotor” in Fig. 6.2a) to the exit flowsplit component, the constant inner diameter of the pipe being equal to the measured diameter common to the actual turbine exit flange and exit duct. This dimension is also used to define the rotor inlet opening and combined outlet opening of the exit flowsplit, and the flowsplit length is set equal to $dx$. The exit flowsplit openings from the throttle component and the rotor are separated by $5^\circ$ between their longitudinal axes to model the approximate physical relationship between the two for all turbines (unlike the $90^\circ$ representation in Fig. 6.2). The exit flowsplit is followed by a pipe of circular cross-section and length $dx$, with the same constant inner diameter of the turbine exit pipe and flowsplit. The model ends with a perfect bellmouth and boundary condition set as the measured static pressure and temperature of the air for a given operating point. Note that all settings or attributes of the model components which did not pertain to measured dimensions were held constant for all three turbines, and that this alternative turbine model may be integrated into a full engine model by connecting turbine inlet and exit ducts to the inlet and exit bellmouths, respectively.
6.3 Turbine

The turbine component ("Volute and Rotor" in Fig. 6.2a) is modeled in the software by user-inputted rotor performance maps which serve as look-up tables for the code. If the modeled turbine utilizes VTG technology, map data is entered for each known “rack” position, corresponding to discrete nozzle vane angle settings. Map operating points include $MFP$, $ER_{ls}$, and efficiency values for a set of constant speed parameters. This efficiency is the combined efficiency, or $\eta_{T,ts}'$, defined in Eqs. (3.9) and (3.10) in accordance with SAE standards J1826 and J922. The foregoing information is either entered into the map reference object or read from an external text file (".trb" file extension). Either way, turbine expansion ratio and speed parameter are used from initialization or the previous time step as inputs to the look-up map, $ER_{ls}$ being calculated from the upstream total pressure and downstream static pressure in the corresponding adjacent control volumes of the turbine component. The outputs of $MFP$ and efficiency are then imposed, where $MFP$ is converted to mass flow rate via Eq. (3.5), with total pressure and temperature from the control volume immediately upstream of the turbine component. An exit temperature is also imposed to the control volume downstream of the turbine by an enthalpy change to the flow. The exit specific total enthalpy is predicted in the code as

$$h_{04} = h_{03} - C_{p3} T_{03} \left[ 1 - \left( \frac{P_4}{P_{03}} \right)^{(y-1)/y} \right] \eta_{T,ts}', \quad (6.4)$$

with which exit temperature is calculated as

$$T_4 = \frac{h_{04} - \frac{1}{2} u_4^2}{C_{p4}}, \quad (6.5)$$

140
where turbine inlet constant pressure specific heat in Eq. (6.4), $C_{p3}$, is equivalent to $C_{pe}$ in the measured turbine efficiency calculation [Eq. (3.10)] described in Section 3.1.

Prior to simulation, however, the turbine map data must be preprocessed. If operating point data is entered into the map reference object, the code uses that data to interpolate between points, to extrapolate to $ER_{ts}=1.0$ and zero speed parameter, and if desired, to expansion ratios and speed parameters higher than those entered. If map data is instead entered as a “.trb” text file, all interpolation and extrapolation is done externally and is contained within the data of the file, bypassing the pre-processor of the code. This latter option was chosen since it affords greater control over extrapolation and interpolation.

Finally, a turbocharger shaft is connected to the turbine in the models, which is evident in Figs. 6.1 and 6.2. Instead of connecting the corresponding compressor in the model and inputting the actual shaft moment of inertia, the latter quantity is entered as a very large value to maintain the speed parameter of the simulation initial state. This is done since only steady turbine operation is considered in the present work and only steady/non-pulsating flow was measured experimentally.

6.3.1 Turbine Maps

Turbine map data measured at Ohio State with wastegate closed was curve fit, extrapolated, and interpolated with a custom MATLAB code, and input as a text file to the 1-D code in place of its data pre-processor for both the GT-Power default model and the new alternative model. The methodology to do so, which is explained next, is based on that of the 1-D code (Gamma Technologies, 2010), but with modifications. First, for each operating point blade speed ratio ($BSR$) is calculated, which non-dimensionalizes
turbine rotor inlet tip velocity, $U_3$, by dividing it by the gas velocity theoretically achievable by isentropic expansion in an ideal nozzle, $C_s$,

$$
\frac{U_3}{C_s} = \frac{U_3}{[2(h_{03} - h_{4s})]^{1/2}} = \frac{U_3}{\left\{2C_{p3} T_{03} \left[1 - \left(\frac{P_4}{P_{03}}\right)^{(y-1)/y}\right]\right\}^{1/2}}.
$$

(6.6)

Next, $BSR$, $MFP$, and efficiency data of all map operating points are normalized by the respective value at the maximum efficiency point for constant $ER_{ts}$, and this is carried out over the following steps:

1. For each constant speed parameter, the measured maximum efficiency point is found and $BSR$, $MFP$, $N_{par}$, $ER_{ts}$, and $\eta_{T,ts}'$ at that point are recorded.

2. From the collection of maximum efficiency point data (step 1), four plots are generated and curve fit, forcing $MFP=\eta_{T,ts}'=0$ and $ER_{ts}=1.0$ when $N_{par}=0$:

   a. $BSR$ at maximum efficiency vs. $ER_{ts}$ at maximum efficiency

   b. $ER_{ts}$ at maximum efficiency vs. $N_{par}$

   c. Maximum $\eta_{T,ts}'$ vs. $N_{par}$

   d. $MFP$ at maximum efficiency vs. $N_{par}$

These four plots are shown in Figs. 6.4a-6.4d, respectively, for map data measured at Ohio State for the BorgWarner turbine with wastegate closed.
1.4: Measured BorgWarner Map Data and Fits at Speed Parameter Maximum Efficiency Point with Wastegate Closed; (a) $N_{par}$ vs. $ER_{ts}$, (b) $BSR$ vs. $ER_{ts}$, (c) $\eta_{T,ts}'$ vs. $N_{par}$, (d) $MFP$ vs. $N_{par}$

Figure 6.4: Measured BorgWarner Map Data and Fits at Speed Parameter Maximum Efficiency Point with Wastegate Closed; (a) $N_{par}$ vs. $ER_{ts}$, (b) $BSR$ vs. $ER_{ts}$, (c) $\eta_{T,ts}'$ vs. $N_{par}$, (d) $MFP$ vs. $N_{par}$

3. For each map operating point, maximum efficiency and $BSR$ and $MFP$ at maximum efficiency are found for the $ER_{ts}$ of the point by utilizing the curve fits in Fig. 6.4. $BSR$ at maximum efficiency is found by plugging the $ER_{ts}$ of the operating point into the curve fit of Fig. 6.4a. However, since the expansion ratios of $MFP$ and $\eta_{T,ts}'$ map characteristics at constant $N_{par}$ may overlap only a little (such as the Honeywell turbine, Figs. 3.2 and 3.4) or not at all (such as the
BorgWarner turbine, Figs. 3.1 and 3.3), interpolation must be done to find their maximum efficiency point data as follows:

a. \( ER_{ts} \) of the operating point is applied to the curve fit in Fig. 6.4b to find \( N_{par} \) where the \( ER_{ts} \) is that of maximum efficiency.

b. The interpolated \( N_{par} \) is applied to the curve fits in Figs. 6.4c and 6.4d to determine maximum \( \eta_{T,ts} \) and \( MFP \) at maximum \( \eta_{T,ts} \), respectively, for the \( ER_{ts} \) in question.

4. Normalization for each operating point is performed by dividing the actual \( BSR \), \( MFP \), and \( \eta_{T,ts} \) by the respective quantities found in step 3.

A least-squares curve fit of normalized \( MFP \) vs. normalized \( BSR \) is then performed for each constant speed parameter of the form

\[
MFP_{\text{norm}} = x + [BSR_{\text{norm}}^y \times (z - x)],
\]

where \( x, y, \) and \( z \) are the variables of the fit. In the code \( z \) is fixed at 1, but the fits for the experimental data required a value which deviated slightly from 1 for each speed parameter. Also, a single least-squares curve fit of normalized efficiency vs. “high” normalized \( BSR \) (greater than 1.0) is performed for all constant speed parameters of the form

\[
\eta_{\text{norm}} = 1 - [n \times \left( BSR_{\text{norm, high}} - 1 \right)^2],
\]

where \( n \) is the coefficient of the fit. Equation (6.8) is also consistent with that used by the code, and \( n \) is a function of the \( BSR_{\text{norm, high}} \) intercept (where efficiency is equal to 0), \( BSR_0 \), according to

\[
n = \left( \frac{1}{BSR_0 - 1} \right)^2.
\]
A single fit for all mapped speed parameters was found to be essential for proper interpolation between speeds, which is discussed later in this section. For “low” normalized BSR (less than 1.0), the normalized efficiency is fit with the equation used by the code,

$$\eta_{norm} = 1 - (1 - BSR_{norm,low})^b,$$

(6.10)

where $b$ is the exponent of the fit (Gamma Technologies, 2010). Since no data was available for the studied turbines at low normalized BSR, the exponent was chosen to be 1.80, the median of the range of values suggested by the code as typical. Fit variables $b$, $n$, and $x$ for each speed parameter are provided in Appendix B for both BorgWarner and Honeywell turbines with wastegate closed.

The resulting extrapolated MFP and $\eta_{T,ts'}$ characteristics for the BorgWarner turbine with wastegate closed are shown in Figs. 6.5 and 6.6 in non-dimensional and dimensional domains, respectively, with Ohio State map data superimposed. In Fig. 6.6a, manufacturer map data is also superimposed to highlight the similarity in measured MFP characteristics, and consequently, that the extrapolated characteristics are largely independent of turbine inlet temperature. The slight disparity observable between fit and manufacturer data for $N_{par} = 4873, 5929, \text{ and } 6823 \text{ RPM}/\sqrt{K}$ is due to a combination of the different behavior of the curves from $N_{par} = 3505 \text{ and } 7591 \text{ RPM}/\sqrt{K}$ (evident in Fig. 6.5a) and the theoretical requirement of maintaining higher MFP at lower rotational speeds for fixed expansion ratio [since the opposition to flow of the centrifugal pressure field of the rotor will be less at reduced speeds (Watson, 1982)]. However, the error of the MFP fits with respect to manufacturer data is less than 1% for all speed parameters, and the error of the MFP fits to Ohio State data is within $\pm 0.79\%$ ($\pm 0.33\%$ for $N_{par} = 3379-6823$).
RPM/$\sqrt{K}$) for the BorgWarner turbine, which is a significant improvement over the error incurred by the single fit for all speed parameters imposed by the base code. The error of the efficiency curve fit to measured data for $N_{par}=3379$ RPM/$\sqrt{K}$ in non-dimensional (Fig. 6.5b) and dimensional (Fig. 6.6b) domains is due to the occlusion of that speed from the maximum $\eta_{T,fs}'$ vs. $N_{par}$ curve fit shown in Fig. 6.4c. This was done because that data point was found to deviate from the trend of higher speed parameters, and may be due to the actual maximum efficiency not closely coinciding with the highest efficiency measured.
Figure 6.5: Extrapolated Non-dimensional BW Characteristics; (a) $MFP_{\text{norm}}$, (b) $\eta_{\text{norm}}$
Figure 6.6: Extrapolated BW Characteristics; (a) $MFP$, (b) $\eta_{T,ls}'$
Interpolation and extrapolation of $MFP$ and $\eta_{T,ts}'$ characteristics was also performed to account for speed parameters between and outside of those measured. Though only constant rotor speed and steady flow will be examined with the computational tool, interpolation is useful in establishing that the curve fitting methodology is sound and in setting the framework for future studies involving pulsating flow. Interpolation and extrapolation was carried out by the following steps:

1. $BSR_{norm}$ was calculated for each mapped constant speed parameter with an equal number of points distributed between $ER_{ts}=1.0$ and 3.0, with a large majority of point allocation between 1.0 and 2.0.

2. Each of the extrapolated $MFP_{norm}$ and $\eta_{norm}$ fits were evaluated with the corresponding $BSR_{norm}$ array from step 1, yielding the example characteristics shown in Fig. 6.7.

3. A speed parameter-weighted linear interpolation was executed between adjacent constant speed parameter $MFP_{norm}$ characteristics at points of identical $ER_{ts}$ according to

$$MFP_{norm} = \left( \frac{N_{par,2} - N_{par}}{N_{par,2} - N_{par,1}} \right) MFP_{norm,1} + \left( \frac{N_{par} - N_{par,1}}{N_{par,2} - N_{par,1}} \right) MFP_{norm,2}, \quad (6.11)$$

where $MFP_{norm}$ at the intermediate speed parameter $N_{par}$ is calculated from those at a lower ($N_{par,1}$) and higher ($N_{par,2}$) speed parameters, as shown in Fig. 6.8. Non-dimensional efficiency characteristics were also interpolated using the corresponding version of Eq. (6.11).
4. Non-dimensional characteristics below those of the lowest measured speed parameter ($N_{par} = 3505 \text{ RPM/}\sqrt{K}$ for BorgWarner turbine) were extrapolated down to $N_{par} = 2000 \text{ RPM/}\sqrt{K}$ by utilizing the curve fit of the lowest measured speed parameter and setting $MFP_{norm} = \eta_{norm} = 0$ when $N_{par} = 0$. Similarly, non-dimensional characteristics above those of the highest measured speed parameter were extrapolated to a slightly higher $N_{par}$ by utilizing the curve fit of the highest measured speed parameter.

Figure 6.7: Non-dimensional BW $MFP_{norm}$ Characteristics at Adjacent Speed Parameters
Figure 6.8: Interpolation of $MFP_{\text{norm}}$ Characteristics at Adjacent Speed Parameters

5. The interpolated and extrapolated characteristics were dimensionalized via the fits in Fig. 6.4, and the result is shown in Fig. 6.9.

The interpolated and extrapolated characteristics shown in Fig. 6.9 were then compiled in a text file for input to the code in place of its map data pre-processor.
Figure 6.9: Interpolated and Extrapolated BW Characteristics; (a) $MFP$, (b) $\eta_{T,ts'}$
CHAPTER 7

COMPARISON BETWEEN MODEL and EXPERIMENTAL RESULTS

In Chapter 6 a model of each turbine and its internal flow passages was created with a 1-D time-domain code, and closed wastegate turbine maps measured at Ohio State were curve fit, extrapolated, and interpolated for input to the code in place of its data pre-processor. The present chapter compares open wastegate turbine performance between the model results and the experimental data measured at Ohio State. Two approaches to modeling wastegate flow were outlined in Section 6.2 – the 1-D code default method of incorporating the wastegate into the turbine component, and a new alternative method of modeling the bypass flow split within the turbine housing as an actual parallel path – and the accuracy of both are reviewed here.

7.1 1-D Code Default Model

First, the 1-D code default approach was examined, which allows incoming gas to bypass the rotor inside the control volume of the turbine component. The effective diameter of the bypass throat, measured from wastegate-alone flow efficiency experiments, is entered as an attribute in the turbine component, and the code assumes the wastegate and rotor experience the same total-to-static expansion ratio as imposed on the turbine component by the adjacent upstream and downstream control volumes. The error of the foregoing assumption was initially reviewed via MATLAB calculations in Section
3.4.2, and it is repeated here to verify the same open wastegate MFP levels would in fact be computed by the 1-D code. Two different discharge coefficients were used to determine effective wastegate diameter for input to the “Wastegate Diameter” attribute of the turbine component: wastegate-alone $C_D$ and rotor-and-wastegate estimated $C_D'$. Application of the former is examined first.

**7.1.1 Application of Wastegate-Alone $C_D$**

For the analysis of the 1-D code default approach, use was made of the turbine model shown in Fig. 6.1 in conjunction with the wastegate-alone $C_D$ versus $ER_{ts}$ relationship used in Section 3.4.2 and listed in Appendix A. Boundary conditions of the model were set as the measured turbine inlet total pressure and temperature ($p_{03}$ and $T_{03}$, respectively) and exit static pressure and temperature ($p_4$ and $T_4$, respectively) of the rotor-alone operating points mapped at Ohio State, and these pressures were used to determine the $ER_{ts}$ for $C_D$ look-up. Simulation results for the open wastegate turbine maps are presented in Figs. 7.1, 7.2a, and 7.2b for 21 mm wastegate Honeywell, 20 mm wastegate BorgWarner, and 26 mm wastegate BorgWarner turbines, respectively, where “measured” quantities have been carried over from Section 3.2. The expansion ratio indicated in the figures, therefore, corresponds to the boundary condition for the model results (“predicted” in Figs. 7.1 and 7.2). The error of the predicted open wastegate MFP with respect to the measured quantities is provided in Fig. 7.3. The direct comparison of Figs. 7.1-7.3 to Figs. 3.22-3.24 in Section 3.4.2 confirms the simplified, independent treatment of rotor and wastegate flow by the 1-D code.
Figure 7.1: Measured vs. Predicted R&WG Flow, Default Model with $C_D$; 21 mm WG
Figure 7.2: Measured vs. Predicted R&WG Flow, Default Model with $C_D$;
(a) 20 mm, (b) 26 mm WG
Figure 7.3: Error of Predicted R&WG MFP, Default Model with $C_D$; (a) HW, (b) BW
7.1.2 Application of Rotor-and-Wastegate Estimated $C_D'$

In an effort to improve the predicted open wastegate turbine $MFP$ and yet retain the default modeling approach of the 1-D code, Section 3.4.1 estimated the wastegate discharge coefficient during rotor-and-wastegate flow by assuming no change in rotor $MFP$ from closed wastegate levels. This approximation was carried out with rotor fixed, since the potential benefit was intended for those with access to only a flow bench. As Section 3.4.2 ultimately showed, however, the estimated discharge coefficient, $C_D'$, with rotor fixed is generally lower than that obtained with the rotor spinning at mapped speed parameters, in part because rotor flow capacity is speed dependent for constant $ER_{ts}$ (refer to Fig. 6.6a). For example, the departure of the BorgWarner closed wastegate $MFP$ characteristic at $N_{par} = 3505$ RPM/$\sqrt{K}$ from the 0 RPM/$\sqrt{K}$ (fixed rotor) $MFP$ characteristic for same $ER_{ts}$ will be less than the difference at 7591 RPM/$\sqrt{K}$, since rotor flow capacity decreases with increasing $N_{par}$ ($ER_{ts}$ constant). Nevertheless, the fixed rotor estimated discharge coefficient was curve fit as a function of $ER_{ts}$ and applied in the 1-D code in the same manner as the wastegate-alone discharge coefficient to determine its accuracy as an alternative. Figures 7.4, 7.5a, and 7.5b show the open wastegate $MFP$ map result of the $C_D'$ simulations for 21 mm wastegate Honeywell, 20 mm wastegate BorgWarner, and 26 mm wastegate BorgWarner turbines, respectively. The error of the predicted open wastegate $MFP$ quantities is provided in Fig. 7.6.

As anticipated, the error of the predicted $MFP$ associated with the Honeywell turbine (Fig. 7.6a) for constant bypass opening and $ER_{ts}$ is largely unchanged from that shown in Fig. 7.3, though the 10° position, and to a lesser extent the 40° position, shows some improvement. Error of $C_D'$ application with the BorgWarner turbines (Fig. 7.6b)
exhibits marked improvement in magnitude with respect to error of wastegate-alone $C_D$ for constant bypass opening and $ER_{ts}$, particularly at $20^\circ$ and $40^\circ$ opening, though the sign of the error is reversed. Both BorgWarner turbines also display a trend of increasing error magnitude with increasing $N_{par}$ for fixed openings of $20^\circ$ and $40^\circ$ in Figs. 7.5 and 7.6b. This trend is a result of the speed dependence of rotor flow capacity described previously in this section combined with similar rotor-and-wastegate total $MFP$ at $N_{par}$ (Section 3.2) and with rotor fixed (Section 3.4) at $20^\circ$ and $40^\circ$. Inclusion of $MFP$ speed dependence would benefit the BorgWarner turbines at $20^\circ$ and $40^\circ$ openings but would increase error for $5^\circ$ and $10^\circ$ openings and for the Honeywell turbine in its entirety.

Figure 7.4: Measured vs. Predicted R&WG Flow, Default Model with $C_D$; 21 mm WG
Figure 7.5: Measured vs. Predicted R&WG Flow, Default Model with $C_D$;
(a) 20 mm, (b) 26 mm WG
Figure 7.6: Error of Predicted R&WG $MFP$, Default Model with $C_D'$; (a) HW, (b) BW
7.2 Proposed Alternative Model

Next, the new alternative approach is examined; the 1-D model of this concept was introduced in Section 6.2.2 and is pictured in Fig. 6.2. In this model the two parallel paths through the turbine are physically modeled such that the relative flow efficiency of the two paths, and the actual expansion ratio existing respectively across them, dictates their individual \( MFP \) and the combined total \( MFP \).

The model shown in Fig. 6.2 satisfies two important criteria in this regard: first, the flow capacity of the rotor path (through the turbine component) with wastegate valve (that is, throttle component) closed is effectively unchanged by the presence of the flowsplit and added pipe components; second, the flow efficiency through the wastegate path with no flow to rotor (utilizing the model in Fig. 6.2b) is the same as measured experimentally. To achieve the latter for each wastegate studied, the following steps were taken with the wastegate-alone model:

1. Wastegate discharge coefficient was curve fit as a function of opening angle for the flow bench data (Section 4.1.1) at \( \Delta p = 40 \) in H\(_2\)O (9.96 kPa) and is shown in Fig. 7.7 for the 21 mm Honeywell wastegate. An array of \( C_D \) was then input into the throttle component of the model for angles 0\(^\circ\) to 45\(^\circ\) in 1\(^\circ\) increments. Flow bench data was used to more accurately capture the \( C_D \) versus opening angle relationship, as the coefficient was measured in 5\(^\circ\) increments with that facility, and it was interpolated and extrapolated in 1\(^\circ\) increments since the code linearly interpolates \( C_D \) for angles not explicitly provided.
2. Next, the measured turbine inlet total pressure and temperature \( p_{03} \) and exit static pressure and temperature \( p_4 \) from the turbocharger stand flow efficiency experiments were imposed as upstream and downstream boundary conditions, respectively, in the model for individual cases of \( ER_{ts} = 1.05 - 2.00 \) in 0.05 increments for each wastegate opening of 5°, 10°, 20°, and 40° (80 individual cases in all for each turbine/wastegate).

3. With the measured mass flow rate from the turbocharger stand wastegate flow efficiency experiments as a target, a PID controller actuated the throttle angle in the model until the simulated mass flow rate through the wastegate path was equal to the target quantity. Throttle angle was thus recorded as a function of \( ER_{ts} \) for
each nominal opening degree of each wastegate, and this is shown in Fig. 7.8 for the 21 mm Honeywell wastegate.

Figure 7.8: Angle of Throttle Component in Proposed Model vs. $ER_{ts}$, 21 mm Honeywell

The throttle component angle versus expansion ratio relationship was curve fit and imposed in the rotor-and-wastegate model of Fig. 6.2a. Once again, boundary conditions of the model were set as the measured turbine inlet total pressure and temperature and exit static pressure and temperature of the rotor-alone operating points mapped at Ohio State, and these pressures were initially used to determine the $ER_{ts}$ for throttle angle look-up. Note that this alternative modeling approach still makes use of only the rotor-alone turbine map and the measured wastegate-alone $C_D$. Simulation results for the open
wastegate turbine maps are presented in Figs. 7.9, 7.10a, and 7.10b for 21 mm wastegate Honeywell, 20 mm wastegate BorgWarner, and 26 mm wastegate BorgWarner turbines, respectively. For predicted total $MFP$ in the figures, rotor mass flow rate is converted to $MFP$ via total pressure and temperature entering the “Volute and Rotor” component, while wastegate mass flow rate is converted to $MFP$ via total pressure and temperature at the pipe immediately downstream of the inlet bellmouth component (essentially the quantities imposed as the inlet boundary condition). The error of the predicted open wastegate $MFP$ quantities is provided in Fig. 7.11.

Figure 7.9: Measured vs. Predicted R&WG $MFP$, Proposed Alternative Model with Boundary $ER_{ts}$ Throttle Angle Look-up; 21 mm WG
Figure 7.10: Measured vs. Predicted R&WG MFP, Proposed Alternative Model with Boundary $E_{R\omega}$ Throttle Angle Look-up; (a) 20 mm, (b) 26 mm WG
Figure 7.11: Error of R&WG MFP Predictions, Proposed Alternative Model with Boundary $ER_{ts}$ Throttle Angle Look-up; (a) HW, (b) BW
Although the negative error of the predicted total MFP for the Honeywell turbine (Fig. 7.11a) increased slightly with this approach compared to application of $C_D'$ in the default model, most notably at 10° and 40° opening for fixed $ER_{is}$, the error of the BorgWarner turbines (Fig. 7.11b) improved for the three highest speed parameters (5929, 6823, and 7591 RPM/√K) and three largest wastegate positions (10°, 20°, and 40°), for same $ER_{is}$. Additionally, the slope of the predicted characteristic for the 26 mm wastegate BorgWarner turbine at $N_{par}=7591$ RPM/√K more closely matches that of the measured characteristic at all wastegate openings compared to the 1-D code default approach. The trend of the predicted Honeywell MFP characteristics at 10° breaking from the accuracy of the other openings is continued here, and this anomaly may not be able to be accounted for in a 1-D code. The deficit in predicted total MFP for the BorgWarner turbines at lower speed parameters (3505 and 4873 RPM/√K) and expansion ratios (1.18-1.42), particularly at 5° and 10° openings, is primarily due to low wastegate flows; predicted MFP through the rotor is only 1.5-3% less than rotor-alone levels at these conditions due to lower $ER_{is}$ across the turbine component than imposed at the boundary. The low wastegate flows may possibly be explained by the relatively poor wastegate flow efficiencies at low openings that decrease with $ER_{is}$, combined with the increased flow capacity of the rotor at lower speed parameters (a result of the decreased centrifugal pressure field of the rotor opposing flow, recall Fig. 6.6a). To achieve the measured rotor-and-wastegate MFP levels for the foregoing operating conditions, wastegate flow efficiency would have to increase at low $ER_{is}$ and $N_{par}$, contrary to the measured wastegate-alone behavior. This was observed to some extent with the estimation of $C_D'$ with the rotor at mapped speed parameters (Figs. 3.25 and 3.26): $C_D'$
was higher at 3505 and 4873 RPM/√K than at 5929 RPM/√K for the BorgWarner turbines at 5° and 10° openings, and it was higher at 3379 than at 4126 RPM/√K for the Honeywell turbine at 5°, 20°, and 40°. Using flow bench mass flow rate and boundary condition data in calibrating the wastegate flow path (steps 2 and 3 in the foregoing procedure) would therefore improve simulation accuracy at low $ER_{ts}$.

In the literature review of Section 1.2, the work of Capobianco and Marelli (2007) and Capobianco and Polidori (2008) was discussed. The authors postulated that the error associated with predicting rotor-and-wastegate flow from a rotor-alone map and wastegate-alone $C_D$ in the simplified, default treatment of 1-D codes (as in Section 7.1.1) was due to a different $ER_{ts}$ existing individually across rotor and wastegate within the turbine housing as compared to the $ER_{ts}$ determined from measurements taken upstream and downstream of the turbine. Given the improvement in prediction accuracy of the modeling approach of this section for the BorgWarner turbines, the simulation results provide a useful means of estimating how different the $ER_{ts}$ across the rotor and wastegate are with respect to the imposed boundary conditions. Figures 7.12-7.13 show the fraction of the boundary condition $ER_{ts}$ experienced by the rotor (turbine component) and wastegate (throttle component) versus the boundary condition $ER_{ts}$ imposed across the model. The BorgWarner simulation results support the theory that the total-to-static expansion ratio across the wastegate is reduced under rotor-and-wastegate configuration compared to the value determined from measurements upstream and downstream of the turbine (the boundary condition in Figs. 7.12 and 7.13). The reduction in $ER_{ts}$ across the Honeywell wastegate (Fig. 7.12) shows a significant cause of the simulation error indicated in Fig. 7.11a, and along with the 1-D code default model results of Fig. 7.1,
provides further evidence that the design of the Honeywell flow passages yields a similar wastegate $ER_{ts}$ between flow configurations.

Figure 7.12: Component $ER_{ts}$ from R&WG Predictions, Proposed Alternative Model with Boundary $ER_{ts}$ Throttle Angle Look-up; 21 mm WG
Figure 7.13: Component $ER_{ls}$ from R&WG Predictions, Proposed Alternative Model with Boundary $ER_{ls}$ Throttle Angle Look-up; (a) 20 mm, (b) 26 mm WG
Finally, a second option is presented here within the proposed alternative model in an effort to further improve the accuracy of the approach. Instead of using the imposed boundary condition $ER_{ts}$ (which is from rotor-alone measured map data) to look up the throttle angle in the model, the $ER_{ts}$ across the throttle itself during simulation was used for look-up based on the corresponding data from the wastegate-alone model calibration. The conversion of wastegate mass flow rate to $MFP$ via total pressure and temperature at the pipe downstream of the inlet bellmouth was retained. Simulation results for this method are presented in Figs. 7.14, 7.15a, and 7.15b for 21 mm wastegate Honeywell, 20 mm wastegate BorgWarner, and 26 mm wastegate BorgWarner turbines, respectively. The error of the predicted open wastegate $MFP$ quantities is provided in Fig. 7.16. The main benefit of this approach is observed to be an improvement in the prediction accuracy for the BorgWarner turbines at the three highest speed parameters (5929, 6823, and 7591 RPM/√K) and two largest wastegate openings (20° and 40°) compared to using boundary condition $ER_{ts}$ for throttle angle look-up. The error associated with the Honeywell turbine increased slightly at 20° for its three highest speed parameters (5218, 5677, and 6147 RPM/√K) and at 5° for $N_{par}=3379$ RPM/√K, but this cost is outweighed by the benefit gained by the BorgWarner turbines. Whichever of the two methods is chosen for throttle/wastegate angle look-up, the proposed model consistently produces more accurate total $MFP$ predictions for a range of turbine housing and bypass passage designs.
Figure 7.14: Measured vs. Predicted R&WG MFP, Proposed Alternative Model with Component $ER_{ts}$ Throttle Angle Look-up; 21 mm WG
Figure 7.15: Measured vs. Predicted R&WG MFP, Proposed Alternative Model with Component $ER_{ts}$ Throttle Angle Look-up; (a) 20 mm, (b) 26 mm WG
Figure 7.16: Error of R&WG MFP Predictions, Proposed Alternative Model with Component $ER_{ts}$ Throttle Angle Look-up; (a) HW, (b) BW
CHAPTER 8

CONCLUDING REMARKS

The main goal of this study was to experimentally investigate three different automotive turbochargers of varying turbine/wastegate combination: two of similar turbine housing (BorgWarner) but different bypass throat diameter (20 and 26 mm), and a third of both different housing (Honeywell) and throat size from the previous two (21 mm). The effects of turbine housing flow passage design and bypass throat size on open wastegate turbine performance and wastegate flow efficiency were examined at discrete wastegate valve openings: 5°, 10°, 20°, and 40°. The study also analyzed the effects of these geometrical differences on flow rate through the wastegate when examined independent of or in parallel with the rotor. Steady flow experiments were performed on a flow bench and a turbocharger stand. An electric linear actuator was used in place of the standard pneumatic actuator to adjust and hold wastegate position, and a unique bracket allowed a rotary potentiometer to measure wastegate opening angle in an accurate and repeatable manner.

Turbine MFP and combined efficiency characteristics were measured on the cold-flow turbocharger stand as a function of total-to-static expansion ratio between the surge and choke limits of the mated compressor. Characteristics were swept at constant $N_{par}$ with the wastegate closed and with wastegate at fixed angles of opening. Closed
wastegate turbine $MFP$ characteristics measured at Ohio State closely align with the curves provided by the respective turbocharger manufacturer, though at lower $ER_{ts}$ due to the reduced gas temperature entering the turbine. Measured total turbine $MFP$ increases with wastegate opening, with diminishing gain in total $MFP$ from one opening to the next larger position, most notably for the BorgWarner turbine (both wastegate sizes) from $20^\circ$ to $40^\circ$. At just $5^\circ$ wastegate opening and $ER_{ts}=1.2$, the increase in total $MFP$ compared to closed wastegate levels is 36.2%, 33.9%, and 47.1% for the 21 mm wastegate Honeywell turbine and 20 mm and 26 mm wastegate BorgWarner turbines, respectively. Total turbine $MFP$ increased in proportion to wastegate size from closed to $5^\circ$ and from closed to $10^\circ$ position for similar $ER_{ts}$ and $N_{par}$, with the 21 mm wastegate Honeywell turbine eventually surpassing the 26 mm wastegate Borg Warner from closed to $40^\circ$.

Wastegate-alone flow efficiency experiments on the turbocharger stand over $ER_{ts}=1.05-2.00$ revealed that the 21 mm Honeywell wastegate $C_D$ is higher than both BorgWarner wastegates at $40^\circ$ for fixed $ER_{ts}$, and it is either equivalent to (10°) or slightly greater than the 26 mm BorgWarner wastegate ($5^\circ$, $20^\circ$) at other openings. These experiments also showed that the $C_D$ for the 20 mm BorgWarner wastegate is greater than the 26 mm variant for all fixed openings and common $ER_{ts}$, despite seemingly identical turbine housings. All wastegates exhibit a slight increase in $C_D$ with total-to-static expansion ratio, most likely due to air compressibility. Wastegate flow efficiency should therefore be measured in a facility capable of achieving expansion ratios across the turbine representative of engine applications.

Using the wastegate-alone discharge coefficients and rotor-alone map data measured on the turbocharger stand to predict flow in the R&WG configuration results in
significant over-estimation of total $MFP$ for the BorgWarner turbines. This large positive error exists with the rotor both fixed and at mapped speed parameters. The error increases with bypass opening (as high as 22.9% for the 26 mm wastegate at 40°, $N_{par}=3505 \text{ RPM}/\sqrt{K}$) and, with few exceptions, is greater for the 26 mm than the 20 mm wastegate turbine for fixed opening degree and $ER_\omega$. For the Honeywell turbine at mapped speed parameters, the foregoing procedure estimates the open wastegate total turbine $MFP$ to within $\pm 5\%$ of measured quantities for all openings and $ER_\omega$, aside from an anomaly for the three lowest speed parameters at 10° opening.

Wastegate-alone discharge coefficients were also measured for the three wastegates on the flow bench, and the trends discovered on the turbocharger stand (such as the 20 mm BorgWarner wastegate having a greater $C_D$ than the 26 mm variant for all fixed openings and common $ER_\omega$) were found to be consistent for bypass openings of 10°, 20°, and 40°. The measured $C_D$ values show relatively good agreement with those of the turbocharger stand for 20° and 40° openings, with slightly higher values for 5° and 10° observed on the flow bench. Estimation of R&WG total $MFP$ by summing rotor-alone and wastegate-alone flows results in the same error trends for all turbines as those measured on the turbocharger stand for fixed rotor; this includes error within $\pm 4\%$ for the Honeywell turbine across all openings and a growing error with wastegate opening for the two BorgWarner turbines.

The 20 and 26 mm wastegates were examined with the purpose of discovering what effects resulted from bypass throat size for an identical BorgWarner turbine housing. The results of common experiments between turbocharger stand and flow bench show that a slight difference in housing designs likely exists, as the $C_D$ for the 20
mm wastegate is consistently higher; at $ER_{ts}=1.5$ the discharge coefficients for the 20 and 26 mm wastegates are, respectively, 0.285 and 0.210 at $5^\circ$, 0.470 and 0.390 at $10^\circ$, 0.748 and 0.641 at $20^\circ$, and 0.854 and 0.773 at $40^\circ$ opening. The only perceivable difference between the two housings is a longer bypass port for the 20 mm variant (approximately 18 mm compared to 8 mm at the shortest part), as well as a slightly smaller wastegate valve head diameter, though this diameter for both wastegates is approximately 7 mm larger than the respective bypass throat diameter. Despite a lower $C_D$, the 26 mm wastegate has a greater effective bypass area for fixed opening degree and $ER_{ts}$, resulting in wastegate-alone mass flow rate that is 17-58% greater than the 20 mm wastegate, depending on wastegate position and $ER_{ts}$ for the turbocharger stand experiments. The corresponding higher turbine inlet velocity, an effect of the wastegate size, may contribute to greater flow separation for the 26 mm wastegate. As expected, another effect of the larger bypass throat size is an increase in R&WG total $MFP$ for all fixed wastegate openings, $ER_{ts}$ being constant. Due to the lower 26 mm wastegate $C_D$ and design of the turbine flow passages (elaborated upon in the following paragraph), the increase in R&WG (and $R_{Npar,WG}$) total $MFP$ from 20 mm wastegate levels is much less than the 69% increase in bypass throat cross-sectional area; the gain in total $MFP$ from 20 mm wastegate levels at $40^\circ$ opening (corresponding to the approximate maximum attainable flow area for these turbines) varies between 9.16% and 17.15% on the turbocharger stand, depending on $N_{par}$ and $ER_{ts}$. This increase is less than the gain in wastegate-alone $MFP$, and as a result the $R^*+WG^*$ and $R_{Npar}^*+WG^*$ errors increase for the larger bypass throat size.
The experimental results of the Honeywell turbine were compared to those of the
BorgWarner turbines to examine the combined effect of bypass port placement and
turbine flow passage design. Physically, the two turbines are very different. The
Honeywell bypass port branches off of the straight combined inlet passage near its
centerline, whereas the BorgWarner bypass is situated to the inside of the mean radius of
the curved inlet passage. Therefore, the Honeywell turbine passages more closely
resemble a typical tee junction, whereas the BorgWarner bypass port must utilize a half-
bowl shaped depression to connect the wastegate throat to the inlet passage. The effects
of this design difference include greater wastegate-alone flow efficiency for the
Honeywell at the largest opening (40°), greater increase in R&WG and R_{N_{par}}&WG total
MFP from 20° to 40° for the Honeywell at a given ER_{ts}, and significantly lower
R^{*+WG*} and R_{N_{par}}^{*+WG*} errors. The last effect likely indicates that flow behavior and
local flow separation at the bypass port are quite different for the BorgWarner turbines
between wastegate-alone and rotor-and-wastegate configurations, while, for the
Honeywell turbine, they may be relatively similar. Though the BorgWarner design was
constrained by turbine orientation and space limitations on engine, it is important to be
aware of the effect of housing geometry on predictability of open wastegate turbine flow
capacity; by understanding the potential for wastegate flow change between
configurations, significant time could potentially be saved during on-engine validation of
the turbocharger and late hardware changes could be avoided. Such a design, with large
increase in wastegate flow at 5° opening but little increase from 20° to 40° during
R_{N_{par}}&WG configuration, will also be difficult to control. In light of the results of the
present work, it may also be noted that a design which reduces the potential for flow separation change at the bypass port between configurations may be more conducive to the simplified wastegate treatment prevalent in 1-D codes. The design that maximizes the utilization of the bypass throat area, like the Honeywell, will also improve the packaging dimensions of the turbine housing. Therefore, further research on the Honeywell flow passage design is recommended for future work, to investigate such aspects as the effect of relative wastegate flow capacity (with respect to rotor-alone levels) on its minimal $R^*+WG^*$ and $R_{npar}^*+WG^*$ errors.

As an additional goal of this study, a 1-D, unsteady, time-domain code was used to develop a semi-empirical physical model of each turbine, modeling the wastegate as an orifice within the control volume of the turbine (the 1-D code default approach) and as a physical parallel path bypassing the rotor (the proposed alternative here). For the proposed alternative model, wastegate-alone $C_D$ data was used to calibrate the flow efficiency of the bypass passage by angle adjustment of a throttle component. This calibration was carried out with the rotor component absent from the model to simulate the wastegate-alone configuration. The closed wastegate turbine $MFP$ and combined efficiency map data measured at Ohio State was extrapolated and interpolated to cover a wide range of total-to-static expansion ratios and speed parameters with a custom preprocessing script. Very good agreement is shown between the extrapolated $MFP$ curves, the Ohio State $MFP$ map data, and the manufacturer $MFP$ map data. The turbine maps from the preprocessing script – consisting of total-to-static expansion ratio, $MFP$, and combined efficiency data at constant speed parameters – provide rotor performance information to the code as a text file in place of its own map data preprocessor.
Pressure and temperature measurements taken upstream (total quantities) and downstream (static quantities) of each turbine during closed wastegate mapping were imposed as constant boundary conditions on the turbine models for all respective map operating points. The measured rotational speed of each point was additionally held constant. For the 1-D code default model with wastegate-alone $C_D$ specified, the simulations reproduce the estimated $R_{npar}^*+WG^* MFP$ maps with the following error ranges: -7.01 to 5.27% for the 21 mm wastegate Honeywell turbine, -1.42 to 12.02% for the 20 mm wastegate BorgWarner turbine, and -4.16 to 26.73% for the 26 mm wastegate BorgWarner turbine, depending primarily on bypass opening, and to a much lesser degree on $ER_{ts}$. This confirms the assumption that current 1-D codes evaluate the rotor and wastegate as independent entities subject to the exact same boundary conditions. When $C_D'$ (estimated from fixed rotor R&WG measurements) is specified instead, the $MFP$ error magnitudes are improved, most significantly for the BorgWarner turbine with 26 mm wastegate. The simulations predict flows for the BorgWarner turbines that are below measured levels across all wastegate openings, however.

For the new alternative model, two methods were studied for look-up of throttle (that is, wastegate) angle based on model calibration data: as a function of $ER_{ts}$ imposed at the model boundaries, and as a function of $ER_{ts}$ imposed across the throttle component. For the BorgWarner models with throttle angle a function of boundary condition $ER_{ts}$, the simulations improve upon the accuracy of $C_D'$ application in the default model for speed parameters mapped at $ER_{ts} > 1.4$: the $MFP$ error range is -1.12 to 4.19% for the 20 mm wastegate BorgWarner turbine, and -4.62 to 3.21% for the 26 mm wastegate BorgWarner turbine, depending on bypass opening and $ER_{ts}$. However, the error worsens slightly for
the Honeywell turbine compared to applying $C_D$ in default model. The total-to-static expansion ratio across the throttle of the BorgWarner turbine model was found to be significantly reduced compared to that imposed at the model boundaries, while it was only slightly diminished for the Honeywell turbine. When throttle angle is a function of throttle $ER_{ts}$, the simulation $MFP$ error for the BorgWarner turbines at $ER_{ts}>1.4$ generally improves slightly over boundary $ER_{ts}$ look-up and improves the slopes of the predicted constant speed $MFP$ characteristics at elevated $ER_{ts}$ to better match those of the measured curves. The Honeywell turbine predictions are not significantly impacted compared to boundary condition $ER_{ts}$ look-up.

Considering the slight shortfall of the proposed model’s accuracy at $ER_{ts}<1.4$ (though improved for the BorgWarner turbines over the default model with $C_D$), in conjunction with $C_D'$ generally increasing at $ER_{ts}<1.4$ with the rotor spinning, the target for future work should be to measure the individual contributions during rotor-and-wastegate configuration both with the rotor fixed and spinning. These experiments should be performed on the same three turbochargers, and perhaps more, to provide insight into the effect of $ER_{ts}$ and wastegate opening on rotor flow capacity, as well as what roles the flow passage design and wastegate size have on it. Since a partitioned turbine exit duct would be required for these experiments, flow efficiency through the bypass should also be evaluated under wastegate-alone configuration to verify that the partition does not affect the abrupt expansion loss downstream of the wastegate. Trials were initially carried out in this regard within the present work, and the presence of the duct partition was found to have an impact on measured wastegate $C_D$ that was not insignificant. Experimental quantification of rotor and wastegate individual contributions
during R&WG and $R_{N_{\text{par}}}$ &WG flows would provide model validation, such that turbine efficiency (with respect to mass flow passing through only the rotor) could be more accurately predicted for use in a complete engine-turbocharger model. However, the effect of flow perturbations from the open wastegate on rotor incidence and flow capacity (and thus efficiency) may be beyond 1-D solvers. Finally, the proposed alternative model should be validated under pulsating flow, with time-resolved pressures and temperatures from engine experiments imposed as boundary conditions.
APPENDIX A

WASTEGATE DISCHARGE COEFFICIENT CURVE FITS

Least-squares curve fits of wastegate discharge coefficients measured on the turbocharger test stand are used in Chapter 3 to compare the sum of rotor-alone and wastegate-alone mass flow rate to measured open-wastegate turbine flow capacity with the rotor spinning at mapped speed parameters (Section 3.4.2). The fits are functions of turbine total-to-static expansion ratio, $p_{03}/p_4$, and are linear except for 26 mm wastegate BorgWarner and 21 mm wastegate Honeywell at 40° opening, which are second and fourth order, respectively. Thus:

$$C_D = c_0 + c_1 \left( \frac{p_{03}}{p_4} \right) + c_2 \left( \frac{p_{03}}{p_4} \right)^2 + c_3 \left( \frac{p_{03}}{p_4} \right)^3 + c_4 \left( \frac{p_{03}}{p_4} \right)^4 .$$  \hspace{1cm} (A. 1)

Figure A.1 shows the curve fits superimposed on the measured discharge coefficients, and Tables A.1-A.3 list the coefficients $c_0$-$c_4$ in Eq. (A.1) for the three wastegates investigated. Curve fits for the BorgWarner (BW) wastegates are extended in the figure to cover the higher expansion ratios mapped with the rotor spinning.
Figure A.1: Measured Wastegate Discharge Coefficient Curve Fits, TS

Table A.1: Curve Fit Coefficients for BorgWarner 20 mm Wastegate

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Table A.2: Curve Fit Coefficients for Honeywell 21 mm Wastegate

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Table A.3: Curve Fit Coefficients for BorgWarner 26 mm Wastegate

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APPENDIX B

NON-DIMENSIONAL TURBINE MAP CURVE FITS

Least-squares curve fits of $MFP_{norm}$ vs. $BSR_{norm}$ [Eq. (6.7)] and $\eta_{norm}$ vs. $BSR_{norm}$ [Eqs. (6.8-6.10)] are performed in Section 6.3.1 to extrapolate the turbine performance characteristics of measured constant speed parameters to higher and lower expansion ratios, as well as to enable interpolation at intermediate speed parameters. The fits were carried out for BorgWarner and Honeywell closed wastegate maps measured at Ohio State, and the respective curve fit variables are provided in Tables B.1 and B.2.

Table B.1: Non-dimensional BorgWarner Turbine Map Curve Fit Variables

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Table B.2: Non-dimensional Honeywell Turbine Map Curve Fit Variables

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REFERENCES


GT-Power (Version 7.1), 2010, Gamma Technologies Inc., Westmont, IL.


