INFORMATION TO USERS

This manuscript has been reproduced from the microfilm master. UMI films the text directly from the original or copy submitted. Thus, some thesis and dissertation copies are in typewriter face, while others may be from any type of computer printer.

The quality of this reproduction is dependent upon the quality of the copy submitted. Broken or indistinct print, colored or poor quality illustrations and photographs, print bleedthrough, substandard margins, and improper alignment can adversely affect reproduction.

In the unlikely event that the author did not send UMI a complete manuscript and there are missing pages, these will be noted. Also, if unauthorized copyright material had to be removed, a note will indicate the deletion.

Oversize materials (e.g., maps, drawings, charts) are reproduced by sectioning the original, beginning at the upper left-hand corner and continuing from left to right in equal sections with small overlaps.

ProQuest Information and Learning
300 North Zeeb Road, Ann Arbor, MI 48106-1346 USA
800-521-0600

UMI[®]

NOTE TO USERS

This reproduction is the best copy available.

UMI[®]

**EFFECTS OF FLUID PROPERTIES ON THE AERODYNAMIC
PERFORMANCE OF TURBOMACHINERY FOR SEMI-CLOSED
CYCLE GAS TURBINE ENGINES USING O₂/CO₂ COMBUSTION**

by

STEPHEN KEIR ROBERTS, B.ENG. (MECHANICAL)

A Thesis submitted to
the Faculty of Graduate Studies and Research
in partial fulfilment of
the requirements for the degree of
Master of Applied Science (Mechanical Engineering)

Department Of Mechanical and Aerospace Engineering
Ottawa-Carleton Institute For Mechanical
and Aerospace Engineering

Carleton University
Ottawa, Ontario, Canada
December 5th, 2001

© 2001

Stephen Keir Roberts

**National Library
of Canada**

**Acquisitions and
Bibliographic Services**

**395 Wellington Street
Ottawa ON K1A 0N4
Canada**

**Bibliothèque nationale
du Canada**

**Acquisitions et
services bibliographiques**

**395, rue Wellington
Ottawa ON K1A 0N4
Canada**

Your file Votre référence

Our file Notre référence

The author has granted a non-exclusive licence allowing the National Library of Canada to reproduce, loan, distribute or sell copies of this thesis in microform, paper or electronic formats.

The author retains ownership of the copyright in this thesis. Neither the thesis nor substantial extracts from it may be printed or otherwise reproduced without the author's permission.

L'auteur a accordé une licence non exclusive permettant à la Bibliothèque nationale du Canada de reproduire, prêter, distribuer ou vendre des copies de cette thèse sous la forme de microfiche/film, de reproduction sur papier ou sur format électronique.

L'auteur conserve la propriété du droit d'auteur qui protège cette thèse. Ni la thèse ni des extraits substantiels de celle-ci ne doivent être imprimés ou autrement reproduits sans son autorisation.

0-612-68291-9

Canada

**The undersigned recommend to
the Faculty of Graduate Studies and Research
acceptance of the thesis**

**Effects of Fluid Properties on the Aerodynamic Performance of Turbomachinery
for Semi-Closed Cycle Gas Turbine Engines Using O₂/CO₂ Combustion**

submitted by

**Stephen Keir Roberts, B.Eng.
in partial fulfilment for the degree of
Master of Applied Science (Mechanical Engineering)**

Thesis Supervisor

Chair, Department of Mechanical and Aerospace Engineering

**Carleton University
December 2001**

ABSTRACT

Many gases, including carbon dioxide and argon, have been considered as alternative working fluids to air in a number of design studies for closed and semi-closed gas turbine cycles. In many of these studies, it has been assumed that if the gas constant R and isentropic exponent γ are included in the speed and flow parameters, the compressor map is applicable to other working fluids. However, similarity arguments show that the isentropic exponent itself is a criterion of similarity and that the turbomachinery characteristics, even when appropriately non-dimensionalized, will in principle vary as the isentropic exponent of the working fluid varies.

A rotating compressor test stand was constructed to measure the aerodynamic performance of a centrifugal compressor. The apparatus operates in closed cycle arrangement to allow for testing of alternative working fluids. The performance of the compressor was measured using air ($\gamma = 1.4$), CO_2 ($\gamma = 1.29$), and argon ($\gamma = 1.67$) as working fluids to determine the influence of the isentropic exponent on the aerodynamic performance of turbomachinery.

For the same value of non-dimensional speed, the pressure ratio, efficiency, and choked flow are significantly different for the three test gases, confirming that the isentropic exponent is an important criterion of similarity. The experimental results have been found to be consistent with a CFD analysis of the impeller, and a meanline analysis of the compressor stage. The changes in performance seen can be predicted reasonably well with simple arguments based mainly on one-dimensional isentropic flow. These arguments form the basis for correction procedures that can be used to project compressor characteristics measured for one value of γ to those for a gas with a different value.

ACKNOWLEDGEMENTS

I would first and foremost like to thank my supervisor, Professor S. Sjolander, without whose support and guidance, this project would not have been possible. I am also grateful to the staff of the Department of Mechanical and Aerospace Engineering, and in particular Terry Goodwin, Alex Proctor, and Jim Sliwka, for their help in manufacturing components of the experimental apparatus, and in all other technical matters.

I would also like to thank my colleagues George Zhang, and Daniel Corriveau, for their assistance and friendship, and especially Ali Mahallati, who spent more than one late night with me in the laboratory when I encountered difficulties.

The financial support of Natural Resources Canada (NRCan), and the Natural Science and Engineering Research Council (NSERC) is also gratefully acknowledged.

Finally, I would like to express my gratitude to my family, and particularly to Carol Weller, for their support in all of my endeavours.

TABLE OF CONTENTS

Abstract	iii
Acknowledgements.....	iv
Table of Contents	v
List of Figures	ix
List of Tables	xii
Nomenclature.....	xiii
1.0 Introduction.....	17
2.0 Similarity Considerations.....	19
2.1 Buckingham's Pi Theorem and Dimensional Analysis	19
2.2 Application of Buckingham's Pi Theorem to Compressor Performance	23
2.2.1 Independent Parameters.....	23
2.2.2 Dependent Parameters (Performance Parameters)	26
3.0 Literature Review.....	30
3.1 Motivation.....	30
3.2 Alternative Working Fluids for Gas Turbine Cycles.....	32
4.0 Experimental Apparatus.....	36
4.1 Test Compressor	36
4.2 Test Apparatus	37
4.2.1 Test Circuit	37
4.2.2 Lubrication System.....	41
4.2.3 Heat Exchanger.....	41
4.2.4 HSWT Facility and Control System	42
4.2.5 Vacuum Pump	44
4.2.6 Gas Sampling.....	45
5.0 Methods of Measurement and Data Reduction.....	46
5.1 Sensors	46
5.1.1 Differential Pressure Transducer	46

5.1.2 Gage Pressure Transducer	48
5.1.3 Thermocouples	49
5.1.4 Tachometer	50
5.2 Probes.....	51
5.2.1 Flow Meter.....	51
5.2.2 Inlet Pressure Probes.....	52
5.2.3 Inlet Temperature Probe	53
5.2.4 Discharge Pressure/Temperature Probe.....	53
5.2.5 Relative Humidity.....	55
5.3 Data Acquisition	56
5.3.1 Pressure and Temperature Measurements	57
5.3.2 Shaft Speed Measurement	59
5.3.3 Gas Composition	60
5.4 Data Reduction.....	61
5.4.1 Fluid Properties of Gases and Gas Mixtures	61
5.4.1.1 Air.....	61
5.4.1.2 Carbon Dioxide and Argon.....	63
5.4.2 Measured Test Parameters.....	64
5.4.3 Corrections to Measured Data	67
5.4.3.1 Corrections Due to Deviations in Test Speed Parameter.....	67
5.4.3.2 Reynolds Number Correction	68
5.5 Assessment of Errors in Test Parameters.....	70
5.5.1 Uncertainties in Measured Test Quantities.....	70
5.5.2 Uncertainties in Calculated Test Parameters	73
6.0 Computational methods	76
6.1 Prediction of the Impeller Performance Using CFD.....	77
6.2 Prediction of Compressor Performance Using a Meanline Analysis.....	78
6.2.1 Impeller Analysis.....	79
6.2.1.1 Inlet Flow Field.....	79

6.2.1.2 Discharge Flow Field and Fluid Properties	81
6.2.1.3 Impeller Losses	85
6.2.2 Diffuser Analysis	90
6.2.3 Volute Analysis	95
6.3 Results of the Meanline Analysis Using Air as the Working Fluid.....	98
7.0 Results and Discussion	102
7.1 Commissioning the Test Stand	102
7.2 Results for Carbon Dioxide and Argon as the Working Fluid.....	105
7.2.1 Effects of γ on the Choking Mass Flow Parameter and Pressure Ratio .	106
7.2.1.1 Choking Flow Parameter	111
7.2.1.2 Pressure Ratio	114
7.2.2 Effect of the Isentropic Exponent on Efficiency	121
7.3 Prediction of Compressor Performance for Alternative Working Fluids	132
7.3.1 Prediction of Impeller Performance.....	132
7.3.2 Prediction of Stage Performance	137
7.4 Implications for Turbine Aerodynamics.....	140
8.0 Conclusions and Recommendations for Future Work	142
References.....	144
Appendix A-1: Experimental Data (Air)	149
Appendix A-2: Experimental Data (Carbon Dioxide).....	154
Appendix A-3: Experimental Data (Argon)	161
Appendix A-4: Gas Analysis Test Reports.....	168
Appendix B: Experimental Procedure	184
Pre-Test Checklist	184
Starting the HSWT Compressor.....	185
Using the Rotating Test Stand.....	186
Evacuating and Filling the Rotating Test Stand	187
Data Acquisition and Control	188
Maintenance.....	191

Appendix C: Detailed Drawings of Test Equipment	192
Appendix D-1: Source Codes of Computer Programs.....	202
Appendix D-2: CFX-BladeGen+ Computer Files	203
Appendix E: Curve Fit Equations of Gas Properties	204
Air.....	204
Water Vapour	204
Carbon Dioxide	205
Appendix F: Effect of Heat Loss on Efficiency.....	206

LIST OF FIGURES

Figure 3.1: Schematic Representation of a Closed Cycle O ₂ /CO ₂ Combustion Gas Turbine Power Generator.....	31
Figure 4.1: Schematic View of Impeller (Side View)	36
Figure 4.2: Turbonetics T04E 50 trim Compressor Map.....	38
Figure 4.3: Schematic Diagram of Experimental Apparatus	39
Figure 4.4: Fully Assembled Test Apparatus (Closed Loop Operation)	40
Figure 4.5: Schematic of Lubrication System.....	41
Figure 4.6: Secespol JAD-X 6.50 Heat Exchanger (Secespol, Inc.).....	42
Figure 4.7: High Speed Wind Tunnel Facility (Jeffries)	43
Figure 4.8: Signal Flow Chart of Wind Tunnel Control System	44
Figure 5.1: Sample Calibration Data - Omega PX-139 Differential Pressure Transducer.....	47
Figure 5.2: Sample Calibration Data – Omega PX-613 Gage Pressure Transducer	48
Figure 5.3: Monarch Instrument IRS5-W infrared tachometer sensor	50
Figure 5.4: ISA Short Radius Flow Nozzle (Bore: 1.5610 in., β : 0.7552)	51
Figure 5.5: Inlet Total Temperature Probe Tip (United Sensor TC-8-E-8-W-F)	53
Figure 5.6: Discharge Total Temperature/Total Pressure Kiel Probe Tip (United Sensor KT-8-E-12-E).....	54
Figure 5.7: Discharge Pipe Traverse Total Pressure Measurements	55
Figure 5.8: Discharge Pipe Traverse Total Temperature Measurements	56
Figure 5.9: Signal Flow Chart of Instrumentation and Data Acquisition System	58
Figure 5.10: Sample Transducer Output (PX-613).....	59
Figure 5.11: Sample Transducer Cumulative Average Output (PX-613).....	60
Figure 6.1: Stage Geometry and Station Numbering.....	79
Figure 6.2: Impeller Inlet Velocity Field	80
Figure 6.3: Impeller Discharge Velocity Field Illustrating the Effect of the Slip Factor σ	82
Figure 6.4: Vaneless Diffuser Geometry	91

Figure 6.5: Volute Geometry and Station Numbering.....	95
Figure 6.6: Volute Secondary Flow Illustrating Loss of Radial Kinetic Energy.....	96
Figure 6.7: Pressure Ratio Predicted by the Meanline Analysis – Comparison to Manufacturer’s Results for Air	99
Figure 6.8: Isentropic Efficiency Predicted by the Meanline Analysis – Comparison to Manufacturer’s Results for Air	100
Figure 7.1: Comparison of Measured Pressure Ratio to Manufacturer’s Test Data for Air	103
Figure 7.2: Comparison of Measured Efficiency to Manufacturer’s Data for Air	104
Figure 7.3: Measured Pressure Ratio for Air, CO ₂ and Argon as a Function of Mass Flow Parameter Π_m	107
Figure 7.4: Pressure Ratio for Air, CO ₂ and Argon Predicted Using a Meanline Analysis	108
Figure 7.5: Pressure Ratio for Air, CO ₂ and Argon, Predicted Using CFD	109
Figure 7.6: Variation of Choking Flow Parameter Π_m^* with γ , Based on One- Dimensional Isentropic Flow Analysis (Eq. 7.3).....	113
Figure 7.7: Choking Flow Parameters for CO ₂ and Argon Predicted by One-Dimensional Isentropic Flow Analysis, and by CFD.....	114
Figure 7.8: Deviation of Impeller Work Coefficient for CO ₂ , and Argon Predicted by Euler Analysis.....	118
Figure 7.9: CFD Prediction of Rotor Pressure Ratio as Function of γ , and Pressure Ratio Predicted by Similarity Considerations (Eq. 7.10)	120
Figure 7.10: Measured Isentropic Efficiency for CO ₂ and Air as Function of Mass Flow Parameter	121
Figure 7.11: Measured Isentropic Efficiency for Argon and Air as Function of Mass Flow Parameter	122
Figure 7.12: Measured Peak Efficiency for Air, CO ₂ , and Argon as Function of Speed Parameter	123
Figure 7.13: Isentropic Efficiency for Air, CO ₂ , and Argon Predicted Using CFD	124

Figure 7.14: Isentropic Efficiency for Air, CO ₂ , and Argon Predicted Using a Meanline Analysis.....	125
Figure 7.15: CFD Prediction of Rotor Efficiency as Function of γ , and Efficiency Predicted by Similarity Considerations (Eq. 7.14)	131
Figure 7.16: CFD Prediction of Rotor Efficiency for Argon and CO ₂ and Predicted Impeller Efficiency Using Proposed Efficiency Correction (Eqs. 7.15 & 7.16)	134
Figure 7.17: CFD Prediction of Rotor Pressure Ratio for Argon and CO ₂ , and Predicted Impeller Efficiency Using Proposed Pressure Ratio Correction (Eqs. 7.15 & 7.17).....	135
Figure 7.18: Predicted Stage Peak Efficiencies Using Proposed Efficiency Correction (Eq. 7.16)	136
Figure 7.19: Stage Efficiency Predicted Using Eqs. 7.16 & 7.17 - Comparison with Measured Values for Argon.....	137
Figure 7.20: Stage Efficiency Predicted Using Eqs. 7.16 & 7.17 – Comparison with Measured Values for CO ₂	138
Figure 7.21: Stage Pressure Ratio Predicted Using Eqs. 7.16 & 7.18 – Comparison with Measured Values For Argon and CO ₂	138
Figure 7.22: Revised Stage Pressure Ratio Predicted Using Eqs. 7.16 & 7.18 and Measured Efficiency – Comparison with Measured Values for Argon and CO ₂	139
Figure 7.23: Schematic Representation of Predicted Turbine Characteristics for Different Working Fluids	141

LIST OF TABLES

Table 4.1: Summary of Impeller Geometry	37
Table 5.1: Typical Fluctuations in Single Test Point Measurements	72
Table 5.2: Permissible Fluctuations of Test Readings (Table 3.1, PTC 10, 1997), and Typical Fluctuations Seen During Performance Tests.....	73
Table 5.3: Estimated Total Error in Test Parameters.....	75
Table 7.1: Deviation of Test Reynolds Numbers Re_{b2} from Reference Reynolds Number Extracted from Fig. 5.2	106

NOMENCLATURE

<u>Symbols:</u>		<u>Units</u>
a_0	Stagnation Speed of sound	m/s, ft/s
A	Area	m ² , in ²
b	Impeller blade width (Fig. 4.1), Diffuser Width (Fig. 6.4)	m
B	Blockage Factor (Eq. 6.34)	(none)
C	Absolute Velocity (Fig. 6.2, 6.3)	m/s
C_D	Flow Nozzle Discharge Coefficient (Eq. 5.2)	(none)
c_f	Diffuser Skin Friction Coefficient (Eq. 6.37)	(none)
c_p	Specific Heat at Constant Pressure (Eq. 2.5)	kJ/kgK
c_v	Specific Heat at Constant Volume (Eq. 2.5)	kJ/kgK
D	Impeller Diameter (Fig. 4.1)	m, in
D	Diffuser Divergence Parameter (Eq. 6.38)	(none)
D_{eq}	Diffusion Factor in Mixing Loss (Eq. 6.28)	(none)
D_f	Diffusion Factor in Blade Loading Loss (Eq. 6.21)	(none)
D_H	Hydraulic Diameter	m, in.
E	Diffuser Effectiveness (Eq. 6.40)	(none)
f_{df}	Friction Factor in Disk Friction Loss (Eq. 6.32)	(none)
h	Static Specific Enthalpy	kJ/kg
h_0	Stagnation Specific Enthalpy	kJ/kg
I	Diffusion Loss Term in Diffuser Analysis (Eq. 6.37)	m ² /s ²
L	Length Dimension	m, in., ft.
L_m	Meanline Meridional Length (Fig. 4.1)	m, ft.
m	Mass Fraction (Eq. 5.7)	(none)
$\dot{m}$	Mass Flow Rate (Eq. 2.2)	kg/s, lb/min
M	Mach Number	(none)
Mw	Molecular Weight (Eq. 5.8)	kg/mol

n	Reynolds Number Correction Exponent (Eq. 7.15)	(none)
N	Rotational Speed (Eq. 2.1)	rpm
P	Static Pressure	Pa, psi
P_0	Stagnation Pressure	Pa, psi
PR	Pressure Ratio (Eq. 2.6)	(none)
ΔP	Flow Nozzle Differential Pressure (Eq. 5.1)	Pa, psi
r	Radius Dimension	m, in., ft.
R	Gas Constant	kJ/kgK
RH	Relative Humidity (Eq. 5.3)	(none)
R_U	Universal Gas Constant (8.314 kJ/kgK)	kJ/kgK
Re	Reynolds Number (Eq. 2.3)	(none)
Re_{b2}	Machine Reynolds Number (Eq. 2.4)	(none)
SP	Volute Sizing Parameter (Eq. 6.42)	(none)
T	Static Temperature	K, °R
T_0	Total Temperature	K, °R
U	Blade Speed (Fig. 6.2, 6.3)	m/s, ft/s
v	Velocity	m/s, ft/s
w	Humidity Ratio (Eq. 5.3)	(none)
W	Relative Flow Velocity (Fig. 6.2, 6.3)	m/s, ft/s
$\dot{W}$	Power Input	W
Z	Number of Blades (Table 4.1)	(none)
α	Absolute Flow Angle (Fig. 6.2, 6.3)	deg.
β	Relative Flow Angle (Fig. 6.2, 6.3)	deg.
β	Flow Nozzle Diameter Ratio (Fig. 5.4)	(none)
δ	Diffuser Boundary Layer Thickness (Eq. 6.36)	(m)
ε	Meanline Radius Ratio r_1/r_2 (Eq. 6.10)	(none)
ϕ	Flow Coefficient (Eq. 7.7)	(none)
γ	Isentropic Exponent (Eq. 2.5)	(none)
η	Isentropic Efficiency (Eq. 2.7)	(none)

ψ	Work Coefficient (Eq. 7.8)	(none)
λ	Colebrook Friction Factor (Eqs. 5.15, 5.16)	(none)
μ	Fluid Viscosity	Ns/m ²
Π_m	Mass Flow Parameter (Eq. 2.2)	(none)
Π_N	Speed Parameter (Eq. 2.1)	(none)
ρ	Static Density	kg/m ³ , lb/ft ³
ρ_0	Stagnation Density	kg/m ³ , lb/ft ³
σ	Slip Factor (Eq. 6.6)	(none)
ω	Pressure Loss Coefficient	(none)

Subscripts:

<i>1</i>	Condition at Compressor Inlet
<i>2</i>	Condition at Impeller Exit, also Condition at Compressor Discharge
<i>3</i>	Condition at Diffuser Inlet
<i>4</i>	Condition at Diffuser Discharge
<i>5</i>	Condition at Volute Inlet
<i>6</i>	Condition at Volute Discharge
<i>a</i>	Axial Component (of velocity)
<i>c</i>	Compressor
<i>cr</i>	Critical Value
<i>h</i>	Value at Impeller Hub
<i>m</i>	Meridional Component (of Velocity), or Meridional direction
<i>m</i>	Maximum Value (of Divergence Parameter, in Eq. 6.39)
<i>r</i>	Radial Component (of velocity)
<i>s</i>	Isentropic Process, Value at Impeller Shroud, or Specified Condition
<i>sat</i>	Saturation Value
<i>t</i>	Test Condition
<i>u</i>	Tangential Component (of velocity)

Superscripts:

- ' Relative to Blade (velocity)
- * Choking Value (of mass flow parameter), Corrected Value (of Slip Factor)

1.0 INTRODUCTION

While gas turbine engines used in power generation are relatively clean burning, the carbon dioxide emitted by open-cycle gas turbine engines contributes to the greenhouse gas content of the earth's atmosphere. Growing concern about global warming and climate change makes it desirable to reduce the greenhouse gas emissions from all fossil fuel combustion, including that in gas turbine engines.

The use of CO_2 as the primary working fluid in a semi-closed cycle has been proposed as one method for eliminating the emission of CO_2 from gas turbine engines in some applications. A clean hydrocarbon fuel, such as natural gas (CH_4) and the oxygen needed for combustion would be injected into the combustion chamber. The only products of combustion would be water vapour and CO_2 . The water can be condensed and removed, and the excess CO_2 generated by the combustion of the fuel can be bled from the circuit and safely disposed of, or used industrially.

A number of design studies have been conducted to determine the viability of semi-closed O_2/CO_2 combustion gas turbine cycles. As will be discussed in chapter 3, these studies address many important issues about the plant design and performance. However, the authors have generally assumed that if the fluid properties are retained in the speed and flow parameters, then a compressor or turbine performance map is valid for any fluid that can be modelled as an ideal gas. As will be shown in the next chapter, similarity arguments indicate that for an ideal gas, the performance of a turbomachine is in fact a function of four non-dimensional parameters: appropriately non-dimensionalized speed and flow parameters, the Reynolds number, and the specific heat ratio γ . The influence of the Reynolds number is reasonably well understood and standard corrections are available for predicting the performance of a turbomachine at a given Reynolds number from performance data obtained for the same machine at a different Reynolds number. However, there seems to be very little discussion in the literature of the

influence of γ on compressor and turbine aerodynamic characteristics. In a few studies, the authors appear to be aware that matching of the speed and flow parameters for a given compressor using two different working fluids will not necessarily result in the same pressure ratio, but the magnitude of the change in pressure ratio that they predict is very small. In addition, none of the authors seem to be aware that there is also potentially an effect of the isentropic exponent on the efficiency of a compressor or turbine.

The present study was conducted to clarify the influence of γ on turbomachinery characteristics. It is part of a larger study related to semi-closed gas turbines using CO_2 as the primary working fluid. The turbomachinery in such engines will see values of γ significantly different from that in air: About 1.29 for the compressor and as low as 1.18 at the turbine inlet condition (Irvine, 1994).

The study focused mainly on compressor characteristics but the implications for turbines were also briefly considered. Three approaches were used. Firstly, the characteristics of a small centrifugal compressor stage were measured for three different working fluids: Air ($\gamma = 1.4$), CO_2 ($\gamma = 1.29$), and argon ($\gamma = 1.67$). Secondly, computational fluid dynamics (CFD) calculations were conducted for the rotor for the same working fluids. The computations were conducted at a constant Reynolds number, which was not achievable in the experiment. In addition, the computations allowed the choking mass flow rate to be determined, whereas it was not possible to achieve choking in the test rotor. Also, a meanline analysis computer program was developed, and the ability of loss correlations developed for air in predicting the performance of a compressor using a working fluid other than air was evaluated. Finally, an attempt was made to devise simple arguments, based mainly on one-dimensional and isentropic flow, to explain the performance differences observed in both the measurements and the CFD and meanline computations.

2.0 SIMILARITY CONSIDERATIONS

2.1 BUCKINGHAM'S PI THEOREM AND DIMENSIONAL ANALYSIS

In fluid mechanics, as well as in other disciplines, a problem may sometimes be solved directly by solving the equations of motion. However, for problems that are too complex for analytical solutions, engineers and scientists often rely on dimensional analysis as a means of expressing experimental data and physical relationships in a more compact form. Dimensional analysis is a by-product of the principle of dimensional homogeneity, which states that "the additive terms in an equation must have the same dimensions in order for it to truly represent a physical phenomenon" (White, 1994). This principle may seem self-evident, but it is an important concept in the development of the method of dimensional analysis, and led to the conclusion that any dimensionally homogeneous equation can be written in an equivalent and more compact dimensionless form.

Buckingham (1914) presented an algebraic method for reducing the number of dimensional variables to the more compact non-dimensional form. This technique is referred to as Buckingham's Pi theorem, and is based on the following principle:

If the solution to a problem depends on n physical properties, the number of independent parameters $v_1, v_2, \dots, v_n$ can be reduced by the number of primary dimensions k that appear in these physical properties by non-dimensionalization of the variables. The solution, or dependent parameter, will then be a function of these non-dimensionalized independent parameters $\Pi_1, \Pi_2, \dots, \Pi_{n-k}$.

Care must be taken to avoid duplication of the independent variables. For example, if the performance of a prototype is assumed to be dependent on pressure, temperature,

and density, it should be recognized that only two of these variables are independent, and that the third can be obtained from an appropriate equation of state, such as the ideal gas law.

The number of primary dimensions k is between 1 and 4, and in the SI system of units, these dimensions are normally mass, length, time, and temperature (abbreviated $MLT\theta$).¹ All other physical quantities have dimensions that are derived from these four fundamental dimensions. For example, in the SI system, velocity has units of m/s, or length divided by time ($M^0 L^1 T^{-1}$), and entropy has units of J/kgK, or $m^2/s^2 K$ ($L^2 T^{-2} \theta^{-1}$).

To illustrate the concept of non-dimensional variables, let us consider a simple example. Suppose one wished to develop an expression for the area of a rectangle. We assume that the dependent variable, the area (A), is a function of two independent variables: the length L , and width W of the rectangle.

$$A = f(L, W)$$

Since there are two independent variables ($n = 2$) and one primary dimension (length) present in the problem ($k = 1$), Buckingham's Pi theorem implies that the number of independent parameters can be reduced to a single parameter ($n - k = 1$), or criterion of similarity, by non-dimensionalization of the problem. One choice of independent parameter could be the length to width ratio, or aspect ratio, of the rectangle. The dependent parameter (area) must also be expressed non-dimensionally. Since it has dimensions of length squared, we can non-dimensionalize the area by dividing by L^2 .

¹ In some cases, the mass dimension is replaced with a force dimension, but this set of fundamental dimensions ($FLT\theta$) is no longer commonly used.

The theorem then states that the non-dimensionalized dependent parameter should be a function of the single dependent parameter:

$$\frac{A}{L^2} = f\left(\frac{L}{W}\right)$$

By inspection, we determine that the function relating the normalized area to the aspect ratio is: $A/L^2 = W/L$. Therefore, by applying dimensional analysis, we have reduced a two-variable problem to a single variable problem. Note that we could just as easily have chosen to normalize the area by dividing by W^2 instead of L^2 , or defined the independent parameter as the width to length ratio (W/L) and still arrive at a valid solution to the problem.

Often, as in the above example, the non-dimensional parameters are fairly easily determined. However, in more complex problems, it may not be possible to determine the criteria of similarity by inspection. Buckingham provided an algebraic method for determining the independent parameters in these cases. Although this method is not presented here, most textbooks on fluid mechanics provide a discussion of the method. An excellent and very readable summary of Buckingham's Pi theorem, with examples, is provided by White (1994). A more complete treatment of dimensional analysis and similitude is provided by Kline (1965).

In the above example, application of Buckingham's Pi theorem did not help much in solving the problem, since the solution was already known. However, in cases where the relationship between the dependent and independent variables is not known, the use of dimensional analysis can greatly simplify the analysis, by reducing the number of independent parameters that need to be varied experimentally. For instance, if the relationship between the area and the geometry of the rectangle was not already known, by applying Buckingham's Pi theorem, we learn that we do not need to vary both the length and width independently, but that we only need to vary the aspect ratio in order to

arrive at the solution. This greatly reduces the number of experiments that we would need to undertake in order to determine the relationship between the dependent and independent parameters.

An additional benefit of using dimensional analysis is that it provides us with scaling laws whereby one can use a smaller scale model to test the effects of certain parameters on the full scale prototype (White). As an example, two geometrically similar airfoils having the same ratio of span to chord length, (aspect ratio), will have the same appropriately non-dimensionalized performance (e.g. lift and drag coefficient), provided all of the independent parameters, such as the Reynolds number and angle of attack, are the same in the model as in the full-scale prototype. This principle allows us to use smaller and less expensive models in performance tests, and still allows us to predict the actual performance of the prototype.

Finally, by grouping the independent parameters in non-dimensional form, it gives us some insight into the physical phenomena involved in complex problems, and familiar non-dimensional parameters will often appear in the analysis. This can greatly simplify the analysis, especially when the influence of one or more of the independent parameters is already well known. For instance, in the majority of fluid flows, one of the criteria of similarity will be a Reynolds number in one form or another. The effect of Reynolds number on most types of flows is well understood, and in many cases, as the Reynolds number increases, its influence on the performance parameters decreases. The threshold value beyond which the performance parameters are virtually independent of Reynolds number depends on the nature of the flow, but it can often be estimated by comparison to other similar flows. It is therefore possible to make predictions of the performance of a full scale prototype based on tests conducted on a scale model, even in some cases when not all of the independent parameters can be matched exactly.

2.2 APPLICATION OF BUCKINGHAM'S PI THEOREM TO COMPRESSOR PERFORMANCE

2.2.1 Independent Parameters

In order to determine the independent non-dimensional parameters that govern the performance of a compressor, we must first list the independent dimensional variables that are of importance. Assuming the working fluid is a perfect gas, it requires eight independent variables to fully specify the performance of a particular member of a family of geometrically-similar turbomachines:

$$N, D, \dot{m}, \begin{pmatrix} a_{01} \\ or \\ T_{01} \end{pmatrix}, \begin{pmatrix} P_{01} \\ or \\ \rho_{01} \end{pmatrix}, \mu, R, \gamma$$

where

N = rotational speed of the machine

D = characteristic length of the machine (typically its tip diameter)

$\dot{m}$ = mass flow rate

$a_{01} = \sqrt{\gamma RT_{01}}$ = inlet stagnation speed of sound

T_{01} = inlet stagnation temperature

P_{01} = inlet stagnation pressure

ρ_{01} = inlet stagnation density, $\rho_{01} = P_{01} / RT_{01}$ for a perfect gas

μ = viscosity

R = perfect gas constant for working fluid

$\gamma = c_p/c_v$ = specific heat ratio of working fluid

With eight independent parameters and four basic dimensions (mass, length, time and temperature) Buckingham's Π Theorem indicates that the independent parameters can be grouped into four non-dimensional independent groups (or criteria of similarity). These parameters are presented here without derivation:

(1) Speed parameter:

$$\Pi_v = \frac{N}{\sqrt{T_{01}}} \left(\frac{D}{\sqrt{\gamma} R} \right) \quad \text{Eq. 2.1}$$

The speed parameter is effectively the Mach number based on the rotor tip speed.

(2) Flow parameter:

$$\Pi_m = \frac{\dot{m} \sqrt{T_{01}}}{P_{01}} \left(\frac{1}{D^2} \sqrt{\frac{R}{\gamma}} \right) \quad \text{Eq. 2.2}$$

The flow parameter can also be interpreted as the Mach number based on the axial velocity through the machine.

(3) Reynolds number:

$$Re = \frac{\rho v L}{\mu} \quad \text{Eq. 2.3}$$

For centrifugal compressors, the blade speed at the impeller tip U_2 and the vane height at the tip b_2 are usually used as the velocity and length scales in the Reynolds number. Thus:

$$Re_{b2} = \frac{\rho_{01} U_2 b_2}{\mu} \quad \text{Eq. 2.4}$$

(4) Specific heat ratio:

$$\gamma = \frac{c_p}{c_v} \quad \text{Eq. 2.5}$$

The specific heat ratio relates changes in pressure to ideal changes in density and temperature in a compressible flow and therefore is also referred to as the isentropic exponent.

All other performance parameters are then functions of these four independent parameters:

$$\frac{P_{02}}{P_{01}}, \eta = fns \left(\frac{N}{\sqrt{T_{01}}} \left(\frac{D}{\sqrt{\gamma R}} \right), \frac{\dot{m} \sqrt{T_{01}}}{P_{01}} \left(\frac{1}{D^2} \sqrt{\frac{R}{\gamma}} \right), Re_{b2}, \gamma \right)$$

In practice, turbomachines are often tested using a full scale prototype, using the service gas during the tests. In these cases, the geometric constants and fluid properties are often omitted from these performance parameters, leaving the more familiar quasi-dimensional performance parameters:

$$\frac{P_{02}}{P_{01}}, \eta = fns \left(\frac{N}{\sqrt{T_{01}}}, \frac{\dot{m} \sqrt{T_{01}}}{P_{01}}, Re_{b2} \right)$$

Note that there is one fewer criterion of similarity, since the isentropic exponent is a fluid property that is constant for an ideal gas.

Compressor performance is normally presented on a compressor map, with the pressure ratio plotted against the mass flow parameter, and with lines of constant speed parameter and efficiency also shown. Since compressors are normally designed and tested using the same working fluid, the fluid properties are usually omitted from the speed and flow parameters, and the isentropic exponent is not included as a criterion of similarity. However, such a map is only valid for fixed values of both the Reynolds number and the isentropic exponent. There is not normally a strong effect of the Reynolds number on the performance of the compressor, except at very low values of machine Reynolds number ($Re_{b2} < 90,000$ for centrifugal compressors). Also, empirical corrections to the efficiency are widely available in the literature, and minor corrections can also be made to the pressure ratio and flow parameter (Strub et al., 1987) to account for differences in machine Reynolds number between the test and operating conditions. As will be discussed in the following chapter however, the literature provides virtually no guidance on the influence of γ on the performance map, and most authors have assumed that the performance is at most a weak function of γ .

2.2.2 Dependent Parameters (Performance Parameters)

In compressors, the usual performance parameters of interest are the pressure ratio and efficiency. These parameters can be defined in terms of static or stagnation properties, and the efficiency can be defined as polytropic or isentropic. However, the most widely used definitions of these parameters are the total pressure ratio, and isentropic efficiency.

(1) Total Pressure Ratio:

The compressor total pressure ratio is defined as the ratio of the absolute stagnation pressure at the discharge to the absolute stagnation pressure at the inlet:

$$PR = \frac{P_{02}}{P_{01}}$$

Eq. 2.6

The pressure ratio is a simple means of expressing non-dimensionally the increase in total pressure produced by the compressor.

(2) Isentropic Efficiency:

If there is no heat transfer into or out of the compressor, it can be shown that the minimum compression work is for an isentropic process. The isentropic efficiency for a compressor is thus defined as the ratio of the work input required for an isentropic compression to the actual compressor work input at the same pressure ratio:

$$\eta_c = \frac{\dot{W}_S}{\dot{W}_C} = \frac{h_{02S} - h_{01}}{h_{02} - h_{01}}$$

where the stagnation enthalpy $h_0 = c_p T_0$ for a perfect gas. For an isentropic process involving an ideal gas:

$$\frac{T_{02S}}{T_{01}} = \left(\frac{P_{02}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}}$$

Thus, if the working fluid can be modelled as an ideal gas, the isentropic efficiency can be expressed as a function of the measured inlet and discharge stagnation pressures and temperatures:

$$\eta = \frac{T_{02S} - T_{01}}{T_{02} - T_{01}} = \frac{\left(\frac{P_{02}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{02}}{T_{01}} - 1}$$

Eq. 2.7

The temperatures and pressures in Eq. 2.7 must be expressed in absolute terms.

The above definitions of these performance parameters are used exclusively throughout the text, and future references to pressure ratio and efficiency represent the total pressure ratio and isentropic efficiency.

Real Gas Effects:

One of the assumptions of the ideal gas approximation is that the fluid properties are constant. However, in real gases, the properties c_p and γ vary with temperature, and to a lesser extent with pressure. It was therefore necessary to examine the validity of using the ideal gas approximation for the three test gases.

For the modest pressure ratios measured in the experiments, the fluid properties of all three test gases did not vary significantly with pressure. The greatest temperature changes seen in the experiments were for argon, where the difference between the inlet and discharge temperature was as high as 200°C. For argon, there is virtually no variation in fluid properties over a wide range of temperatures, and like other monatomic gases, argon can be modelled successfully using the ideal gas assumption of constant fluid properties under normal circumstances. For air and CO₂, the greatest change in temperature seen in the experiments was approximately 100°C. Within this range of temperatures, there is only a 0.4% change in the isentropic exponent and specific heat for air. Of the three test gases, the fluid properties of CO₂ vary the most with temperature, and the change in isentropic exponent for the range of temperatures in the experiments was approximately 3% for CO₂.

These variations in fluid properties are considerably lower than the maximum variations allowed by the Performance Test Code (ASME PTC 10, 1997), which recommends the use of ideal gas relations when the difference between the inlet and

discharge isentropic exponent is less than 9% for pressure ratios up to 4. At lower pressure ratios, the difference in fluid properties can be as high as 12%, without any significant error in the efficiency calculation (PTC 10, 1997).

For the temperature range and pressure ratios seen in the experiments, use of the ideal gas approximation results in a deviation in the calculated value of the efficiency of less than 1% for all gases at all pressure ratios, when compared to the equivalent real gas efficiency calculations. In general this deviation is less than the uncertainty in the efficiency measurement (which is discussed in chapter 5). Thus, over the range of temperatures seen in the experiments, the variation in fluid properties for all three test gases was small enough that the fluid properties could be considered constant, and the isentropic efficiency could be calculated using the ideal gas approximation.

In the calculations, the specific heat at constant pressure c_p , and the isentropic exponent γ were evaluated at the average temperature between inlet and discharge. The methods used to evaluate the fluid properties for all test gases is described in section 5.4.

3.0 LITERATURE REVIEW

3.1 MOTIVATION

Environmental concerns have created a need to reduce the emissions of carbon dioxide and other airborne pollutants produced by fossil fuel burning processes, including gas turbine engines. It is estimated that gas turbine power generation will account for 24% of world-wide electricity production in the period from 1995 to 2004 (Stein, 1996). The reduction of CO₂ emissions by these plants therefore represents a significant reduction in the total global greenhouse gas emissions. In the 1996 Kyoto accord, Canada and 37 other developed countries agreed to reduce their emissions of greenhouse by 5.2% compared with their 1990 levels, by the year 2010. Although there is some concern that Canada and some other countries will no longer honour the accord, the Ministry of Natural Resources Canada, as well as other national and international organizations, are currently working to develop economically viable technologies that will allow them to meet the goals of the agreement.

Many methods have been proposed for capturing the CO₂ emissions of fossil fuel combustion processes using air as the oxidant. However, the capture of CO₂ emissions for gas turbine power generation represents both a significant increase in capital cost, as well as an additional energy usage that reduces the overall efficiency of the plant. The penalty in efficiency is estimated to be between 10% and 32%, and the power generation costs are increased by 30% to 130% when compared to cycles without CO₂ emission control (Zhou & Gauthier, 1998). In addition, CO₂ capture does not necessarily reduce the emissions of other airborne pollutants, such as nitrous oxides (NO_x), and sulphur.

An interesting alternative to CO₂ capture is the use of a closed cycle gas turbine using CO₂ as the working fluid (Fig. 3.1), as proposed by De Ruyck (1992) and Mathieu et al. (1993). Oxygen, and a clean burning fuel such as natural gas, are injected into the

combustor in stoichiometric amounts. Thus the only products of combustion are water vapour and CO_2 . Rather than extracting CO_2 from the exhaust gases, it can be bled from the circuit and safely disposed of, or stored for later use. Other airborne pollutants are eliminated, thus introducing the possibility of a zero-emissions plant (Jackson et al., 2000).

Figure 3.1: Schematic Representation of a Closed Cycle O_2/CO_2 Combustion Gas Turbine Power Generator

Although an additional energy usage is required to produce oxygen for the combustion process, it is generally believed that the additional energy requirements of a closed cycle plant are much lower than those of air-cycle plants employing CO₂ sequestration, and that such cycles may be economically viable (Zhou & Gauthier, 1998, Ulizar & Pilidis, 1999). An additional advantage of the closed cycle gas turbine is that by varying the circuit pressure, the power output of such cycles can be reduced during off-peak periods without changing the operating point of the compressor or turbine, thus allowing the plant to be operated at its peak efficiency at all times.

3.2 ALTERNATIVE WORKING FLUIDS FOR GAS TURBINE CYCLES

The concept of using an alternative working fluid to air in gas turbine engine cycles was first proposed as early as the 1940's (Ferguson, 1963, Shepherd, 1956, Lee, 1954). Various investigators at NACA investigated the advantages of using gases other than air for research into compressor aerodynamics at high Mach number (Shepherd). In particular, these researchers recognized the advantages of using gases with high molecular weights. Since such gases have low sonic velocities, high aerodynamic loading can be achieved at lower rotational speed, and the size of the machine is also reduced, due to the high density of the gas.

These early researchers also recognized the isentropic exponent as a criterion of similarity in the performance of turbomachinery, and there was some research conducted on the effects of changing the working fluid on compressor performance, mostly using Freon 12 as the working fluid ($\gamma = 1.12$ at 1 atm. & 20 °C). von Doenhoff & Braslow (1953) concluded however that the isentropic exponent does not have a significant effect on compressor performance. Although Shepherd expresses some doubt about these findings, it appears that no further research has been conducted to address specifically the effects of γ in terms of compressor performance, and it is widely assumed in the recent literature that the isentropic exponent is not an important criterion of similarity.

Mathieu et al. (1994) investigated the possibility of using existing gas turbine engines for closed cycle CO₂ combustion power generation. The authors noted that for synchronous power generation, the efficiency of such plants would be decreased, since due to the changes in fluid properties, the compressor and turbine would no longer be operated at their respective design points. Their analysis illustrates the effects of the gas constant and specific heat on matching of turbines and compressors, as well as addressing other technical issues with regards to the additional hardware required for these cycles. Although these issues are important to designers, the scope of their work does not include the changes in the aerodynamic behaviour of the turbomachinery, and in particular, whether or not the compressor and turbine characteristics for alternative working fluids can be projected from computed or measured results for air.

Saravanamuttoo et al. (2001) provide a theoretical analysis of the efficiency that might be achieved for a helium cycle. However, in their analysis, the authors make no mention of the fact that the isentropic exponent is a criterion of similarity, and assume that the variation in fluid properties will have no effect on the component efficiencies. Like many other authors (e.g. Hunter, 1994, Navaratnam, 1994, Harman, 1981), Saravanamuttoo et al. believe that gases with low molecular weights, such as helium, are desirable in a working fluid. These gases have high values of the specific heat, and thus are capable of a high work output for a given mass flow. Because it is chemically inert, helium is often used as a coolant gas in nuclear reactors, and there is the possibility of using helium as a bottoming cycle in nuclear power generation. A prototype helium gas turbine cycle has been constructed in Germany (Hunter), but no detailed discussions of the aerodynamic performance of the turbomachinery involved was found in the literature.

The main disadvantage of helium cycles discussed in the literature is that since gases with low molecular weights will also have a lower density at a given pressure, the cycle would need to be operated at high circuit pressure to limit the size of the machinery, thus creating problems with leakage of the working fluid into the atmosphere. However, low molecular weight gases have additional disadvantages that are not addressed by the

authors: Since these gases have high sonic velocities, a high rotational speed is required to achieve the same aerodynamic loading as for air. For example, a turbomachine would need to be operated at approximately four times the rotational speed when using helium as the working fluid, when compared to an equivalent speed parameter using air as the working fluid. The material stress for such rotational speeds would be beyond the limit of conventional materials, and the stage pressure ratio for a given mass flow rate would be reduced, making it necessary to increase the number of compressor and turbine stages.

In addition to the above design studies, the ASME Power Test Code on Compressors and Exhausters (ASME PTC 10, 1997) provides a method for predicting the speed at which a compressor must be operated at to produce the same pressure ratio using a substitute gas, by assuming that the work coefficient of two compressors operating with different working fluids is the same for equivalent non-dimensional speed and flow parameters. ASME PTC 10 acknowledges that the pressure ratio and work coefficient may be different when using a substitute gas, even when the speed and flow parameters are matched, although the test code does not identify the isentropic exponent as the cause of the changes in performance. The method of ASME PTC 10 can be modified to predict the pressure ratio that would result for a compressor operating at the same speed parameter using a substitute working fluid, but the underlying assumption is that there is no effect of the isentropic exponent on the losses. Thus, the results of such an analysis do not necessarily permit engineers to predict compressor performance with sufficient accuracy for design purposes when substitute gases are used.

To date, no studies have been found in the literature that explicitly address the issue of the isentropic exponent on the performance of turbomachinery, and there do not appear to be any recent experimental studies conducted on this topic. Although several design studies have been made recently, there do not appear to be any experimental results published in the open literature with specific reference to the isentropic exponent as a criterion of similarity in turbomachinery. Indeed, there does not appear to be agreement in the gas turbine community on the effect of using different working fluids on gas

turbine engine performance. It has been assumed by most authors that proper non-dimensionalization of the speed and mass flow parameters will result in accurate prediction of the compressor or turbine performance for an alternative working fluid. Most authors (e.g. Hunter, Navaratnam), ignore the isentropic exponent in their discussions of the effects of using an alternative working fluid, and in the cases where it has been recognized as a criterion of similarity (Ulizar & Pilidis, 1999, Shepherd, 1963), the authors assume that the performance is at most a weak function of γ . The present study has been conducted to clarify the effects of the isentropic exponent on aerodynamic performance of turbomachines.

4.0 EXPERIMENTAL APPARATUS

4.1 TEST COMPRESSOR

The centrifugal compressor from a Turbonetics T04E 50 trim turbocharger was selected for the tests. The impeller, shown schematically in Fig. 4.1, is manufactured by Garrett Turbocharger Systems, and consists of 6 full blades and 6 splitter blades. The impeller geometry (Table 4.1) was not available from the manufacturer, so the geometry was obtained from measurements of the actual rotor. Although the efficiency of turbocharger compressors is generally lower than that of compressors found in gas turbine engines, the low cost and input power, and relatively high performance of the turbocharger made it a suitable option for this type of general study. In addition, the compressor map, complete with efficiency contours (Fig. 4.2), was available from the supplier, providing a means of benchmarking the compressor test stand.

Figure 4.1: Schematic View of Impeller (Side View)

Symbol	Description	Dimension
Z	Number of Full Blades	6
Z_s	Number of Splitter Blades	6
D	Impeller Tip Diameter	3.00 in.
b_2	Impeller Tip Blade width	0.200 in.
D_{2H}	Inlet Hub Diameter	2.122 in.
D_{2S}	Inlet Tip Diameter	0.720 in.
β_1	Inlet Meanline Blade Angle	52°
β_2'	Blade Backsweep Angle	35°
L	Axial Length of Impeller	1.00 in.
L_M	Meanline Meridional Length	1.725 in.

Table 4.1: Summary of Impeller Geometry

4.2 TEST APPARATUS

The test apparatus is described below. Although some aspects of the instrumentation and data acquisition are briefly mentioned in this section, a more complete discussion of the instrumentation, data acquisition, and experimental uncertainty is presented in chapter 5.

4.2.1 Test Circuit

The test circuit (Figs. 4.3 & 4.4) was constructed to comply with the guidelines of ASME PTC 10, and can be used in both open-cycle and closed-cycle operation. Since the inlet and discharge of the compressor were not manufactured to fit standard schedule steel pipes, it was necessary to manufacture slip-on flanges with drafted inside diameters to minimize losses due to sudden expansions or contractions. The compressor casing, as well as the inlet and discharge pipes were insulated to minimize any heat loss between

the compressor and the temperature measurement stations. Detailed drawings of some of the main test apparatus components are given in Appendix B.

Figure 4.2: Turbonetics T04E 50 trim Compressor Map (Turbonetics, Inc.)

The open-cycle arrangement was used in the first series of tests, using air as the working fluid, in order to ensure that the test data were in agreement with the performance map supplied by the manufacturer before proceeding with the additional expenditure and construction of the closed loop test arrangement. During these tests, ambient air was drawn into the test circuit through a bellmouth, and the compressor discharge was exhausted to the room. The rotational speed was controlled by controlling the turbine blowing pressure, and the pressure ratio could be adjusted by throttling either the inlet or discharge of the compressor. Although inlet throttling required less power,

allowing for longer test runs, it was not frequently used, since it caused the test Reynolds number to deviate significantly from the implied machine Reynolds number of the manufacturer's map. The results of the commissioning tests described above are presented in section 7.1.

Figure 4.3: Schematic Diagram of Experimental Apparatus (a): Open Cycle Operation, (b): Additional Hardware for Closed Cycle Operation

In the closed loop tests, the discharge from the compressor was cooled and throttled back to the inlet conditions, and reused in the test circuit. Due to a small leakage flow of the test gas through the shaft bearings, the test gas was injected upstream of the flow

nozzle during the run (Fig. 4.3) by using a pressure regulator to maintain a constant inlet pressure of approximately 1 psig. This resulted in a very high concentration of the desired test gas in the test section, and also ensured the machine Reynolds number was constant for a given speed line. Gas samples were taken at random during test runs and analyzed at the Ministry of Natural Resources CANMET Characterization Laboratory using gas chromatography. The majority of the argon samples were found to have in excess of 99.5% molar fraction of the desired test gas, and the majority of the CO₂ samples were found to be in excess of 98.5% pure, also on a molar basis. The chromatography test reports are included in Appendix A-4.

Figure 4.4: Fully Assembled Test Apparatus (Closed Loop Operation)

4.2.2 Lubrication System

A pump was required to supply sufficient oil to the bearing housing to prevent bearing failure. An Albany CEP-93 gear pump was selected, since it was capable of providing the 75 psig oil pressure recommended by the manufacturer. However, this pressure was found to cause excessive friction in the turbocharger rotor, and also caused oil to leak into both the turbine and compressor. To alleviate this problem, an adjustable relief valve and pressure gage were installed. This allowed for adjustment of the oil pressure from 0 to 75 psig, with the excess oil being returned to the reservoir, as shown schematically in Fig. 4.5.

Figure 4.5: Schematic of Lubrication System

4.2.3 Heat Exchanger

For the closed cycle tests, it was necessary to cool the compressor discharge as closely as possible to the ambient temperature. A Secespol JAD-X 6.50 shell and tube heat exchanger (Fig. 4.6) was used, using water as the coolant in a counterflow arrangement. Originally, the gas was passed through the tubes, and the coolant through the shell. However, the pressure losses in the heat exchanger were high when this arrangement was used, and it was not possible to operate the compressor at high mass flow rates. With the gas flowing through the shell instead, the pressure losses were found

to be lower, although the range of flow rate was still limited when compared to the open cycle tests.

Figure 4.6: Secespol JAD-X 6.50 Heat Exchanger (Secespol, Inc.)

4.2.4 HSWT Facility and Control System

The tests were conducted using the High Speed Wind Tunnel facility at Carleton University (Fig. 4.7) to supply the pressurized air for the air turbine. The facility was designed for use as a blow-down cascade wind tunnel for testing the performance of turbine blades, but in the present work, the pressurized air in the storage tanks was ducted to the turbocharger turbine. The capacity of the wind tunnel compressor and storage tanks was sufficient to allow for continuous operation of the compressor at low blowing pressures, and sufficient test duration at high blowing pressures to verify steady-state operation of the compressor.

The turbine blowing pressure was maintained using the existing control system and software of the High Speed Wind Tunnel (Jeffries, 1994 & 2001), shown schematically in Fig. 4.8. The blowing pressure is measured at the turbine inlet, and the control valve is actuated by a signal sent from the wind tunnel control computer. By adjusting the PID controller gain settings, it was possible to maintain acceptable control of the rotor speed. The original control program was modified to allow for adjustment of the blowing

pressure during a test, so the rotor speed can be matched as closely as possible to the desired speed line.

Figure 4.7: High Speed Wind Tunnel Facility (Jeffries)

For further details on the configuration and operation of the High Speed Wind Tunnel Control System, the reader is referred to the work of Jeffries (1994 & 2001). Information about the modified control system software source code is included in Appendix D.

Figure 4.8: Signal Flow Chart of Wind Tunnel Control System

4.2.5 Vacuum Pump

A Cole-Parmer Venturi vacuum pump (P/N 78165-30) was used to evacuate the test circuit of air prior to filling it with another test gas. Shop air at 90 psi was supplied to the pump Venturi tube, which entrained fluid from the test circuit through a small opening in the Venturi throat. Since it was only possible to reduce the circuit pressure to approximately 3 psia using the vacuum pump, it was necessary to flush the remaining air from the circuit by bleeding the test gas into the inlet side of the test circuit and opening a small valve on the compressor discharge side.

4.2.6 Gas Sampling

In order to determine the composition of the test gas, it was necessary to take gas samples during tests. This was done by fitting a small valve to the discharge side of the compressor, downstream of the instrumentation. Airtight CaliBond[®] sample bags were then attached to the valve, and the valve opened briefly to fill the sample bag. The samples were analyzed at the CANMET characterization laboratory.

Due to the high cost of the tests, it was not possible to test the gas composition for each test run. However, in most cases, at least two samples were taken on days when tests were run using CO₂ and argon. One sample was normally taken at the beginning of a series of tests, and another sample was taken during the last of a series of tests, and the results of the gas analyses were assumed to be representative of the concentrations present during the entire series of tests.

5.0 METHODS OF MEASUREMENT AND DATA REDUCTION

The apparatus was instrumented to measure the rotational speed, flow rate, and the inlet and discharge pressure and temperature. These measurements provided sufficient data to evaluate the pressure ratio, mass flow parameter, speed parameter, machine Reynolds number, and the isentropic efficiency, as described in Section 4.2. The locations of the instrumentation, flow device and test compressor are all within the dimensional limits prescribed in section 4.3 of PTC 10, with sufficient straight lengths of pipe both upstream and downstream of the compressor and all instrumentation to avoid measurement errors due to non-uniformity of flow.

5.1 SENSORS

The sensors used in the measurement of the test parameters and the methods of calibrating the sensors are described below. Although some aspects of the measurement uncertainties are briefly discussed in this section, a more thorough assessment of the uncertainties in the measured test parameters is presented in section 5.5.

5.1.1 Differential Pressure Transducer

The flow meter differential pressure was measured using an Omega PX-139 differential pressure transducer. The nominal range of the transducer is ± 5 psid, and the transducer output is 0-5 VDC, with the zero output nominally at 2.25 V.

The differential pressure transducer was calibrated using a water manometer for the range of 0-35 in. WG, and a U-Tube mercury manometer for the range of 35 in. WG to approximately 5 psid (approx. 2.5-10 in Hg). The slope of the calibration curves (Fig. 5.1) was found to be linear and stable, but there was occasionally a drift of up to 20 mV in the zero reading. This zero-shift could result in large measurement errors, especially at low flow rates. The shift was seen to be representative of the offset throughout the

calibration curve, and a zero reading was taken at the beginning and end of each test run to minimize the bias error. In addition, test data was discarded if there was a change in the zero reading of more than 5mV during the test.

A least squares linear fit was applied to the calibration data, and typical values of the correlation coefficient and standard deviation for calibrations of the differential pressure transducer were of the order of 0.999 and 0.01 respectively, relative to a linear fit.

Figure 5.1: Sample Calibration Data - Omega PX-139 Differential Pressure Transducer

5.1.2 Gage Pressure Transducer

In addition to the flow meter differential pressure, it was necessary to obtain the absolute static pressure upstream of the flow nozzle, and the inlet and discharge pressures. This was done by measuring the gage pressure using an Omega PX-613 pressure transducer, with a nominal range of -15 to +60 psig, and nominal output of 0-5 VDC. The gage pressure was then added to the local barometric pressure, measured using an Omega DPI 740 digital barometer.

Figure 5.2: Sample Calibration Data – Omega PX-613 Gage Pressure Transducer

The gage pressure transducer used to measure the inlet and discharge pressure was calibrated using a U-Tube mercury manometer from approximately -8 psig to +20 psig (approximately -17 to +44 in Hg). The calibration curve (Fig. 5.2) was found to be very stable and linear, with very little drift in the zero output. However, a zero output reading was still taken at the beginning and end of each run to ensure there was no significant drift in the zero output reading during the run.

A least squares linear fit was applied to the calibration data, and typical values of the correlation coefficient and standard deviation for calibrations of the gage pressure transducer were of the order of 0.999 and 0.025 respectively, relative to a linear fit.

Since the slopes of both transducers were observed to be stable, it was not necessary to perform slope calibrations very frequently. During periods of testing, three to four point calibrations were performed approximately on a weekly basis, and more extensive calibrations such as those shown in Figs. 5.1 & 5.2 were conducted approximately on a monthly basis.

5.1.3 Thermocouples

Type E thermocouples were chosen for both the inlet and discharge temperature measurements, since they have the highest millivolt output per degree of temperature change of the base metal thermocouples, and a high degree of precision in the temperature measurements was desired in order to accurately measure the efficiency.

Both thermocouple probes were calibrated by using the boiling point and freezing point of water as reference temperatures. In all cases, both probes measured the reference temperatures to within 0.1°C accuracy.

5.1.4 Tachometer

Shaft speed was measured using a Monarch Instruments IRS-5W infrared tachometer sensor (Fig. 5.3). The sensor head consists of an infrared photodiode and phototransistor, and can detect contrasting light and dark surfaces. The sensor was mounted in the turbine discharge, and one of the turbine blades was painted white to reflect the signal from the photodiode. When the phototransistor detects a reflected signal, a 5V TTL pulse is sent to the data acquisition computer.

The tachometer was calibrated by pointing the sensor head at a fluorescent light, and recording the measured frequency for 15 minutes. In all cases, the tachometer measured a steady 7200 rpm (120 Hz) for the entire 15 minute period. In addition, the accuracy of the tachometer at high speed was verified by comparing the output to the waveform seen on an oscilloscope. The output signal was observed to be relatively free of noise, and the signal levels were well beyond the threshold voltage of the frequency meter.

**Figure 5.3: Monarch Instrument IRS5-W infrared tachometer sensor
(Monarch Instruments, Inc.)**

5.2 PROBES

5.2.1 Flow Meter

The flow velocity in the inlet section of the apparatus was measured using an ISA short radius flow nozzle with a diameter ratio (β) of 0.75520 (Fig. 5.4). Static taps were placed one pipe diameter upstream of the nozzle inlet, and at the nozzle throat. Although it is usually recommended that the downstream pressure measurement be taken at one-half pipe diameter downstream of the nozzle inlet plane for this type of nozzle, this was not possible due to the thickness of the flanges used to secure the flow nozzle in place. However, the error in measurement when using throat taps instead of $\frac{1}{2}D$ taps is negligible, and the same discharge coefficient can be used for throat taps as for $\frac{1}{2}D$ taps (Beitler et al., 1959).

Figure 5.4: ISA Short Radius Flow Nozzle (Bore: 1.5610 in., β : 0.7552)

Flow velocity in the nozzle throat was calculated from the flow nozzle differential pressure:

$$v = C_D \sqrt{\frac{2\Delta P}{\rho(1-\beta^4)}} \quad \text{Eq. 5.1}$$

The manufacturer provided a discharge coefficient C_D for each gas at an average flow rate. However, since greater accuracy in flow rate measurement was desired over a wide range of flow rates, the discharge coefficient was calculated for each test point using the ISO correlation for ISA short radius nozzles (Uerner, 1997).

$$C_D = \frac{\left(1 - \beta^4\right)^{\frac{2}{\gamma}} \left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} \left(\frac{\gamma}{\gamma - 1}\right) \left(1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma - 1}{\gamma}}\right)}{\left(1 - \beta^4\right)^{\frac{2}{\gamma}} \left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} \left(1 - \frac{P_2}{P_1}\right)} \quad \text{Eq. 5.2}$$

The flow nozzle was fabricated at K-Flo Fluid Meters Inc. of Montreal. Calibration of the nozzle was not necessary, since it was machined to within tolerances of less than 0.001 in.

5.2.2 Inlet Pressure Probes

The inlet pressure was measured by taking the average reading of four static taps spaced at 90 degrees and placed 24 in. upstream of the compressor inlet. Since it was difficult to drill small holes into the pipe, 1/8 in. MPT brass pipe plugs were inserted into the pipe, and 0.018 in. pressure taps were drilled through the pipe plugs, which were then ground and polished to be flush with the inside surface of the pipe.

Since the total pressure ratio was desired, the inlet total pressure was calculated by iteration on the static pressure, total temperature and the flow velocity in the compressor inlet pipe.

5.2.3 Inlet Temperature Probe

The inlet total temperature was measured using a United Sensor T-8-E-12-C Type E (Chromel-Constantan) thermocouple probe, 12 inches upstream of the inlet of the compressor (Fig. 5.5). The sensor consists of a grounded thermocouple junction mounted in a small vented cavity. The sensor was placed in the centre of the inlet pipe and aligned with the oncoming flow.

Figure 5.5: Inlet Total Temperature Probe Tip (United Sensor TC-8-E-8-W-F)

5.2.4 Discharge Pressure/Temperature Probe

The discharge thermodynamic conditions were measured using a United Sensor KT-8-E-12-E type E combination total temperature/total pressure probe (Fig. 5.6). The probe is of the same design as the inlet thermocouple probe, but with the addition of a Kiel probe to measure the stagnation pressure. A Kiel probe was chosen to minimize any measurement errors due to swirling flow. In addition, the discharge probe was traversed across the diameter of the discharge pipe during initial tests to ensure uniformity of the

discharge flow. Since the pressure and temperature at the pipe centre were seen to be representative of the bulk flow values (Figs. 5.7, 5.8), the probe was not traversed during closed loop tests using CO₂ and argon, and the clearance between the probe and the discharge pipe was sealed to eliminate leakage of the test gas out of the test circuit.

**Figure 5.6: Discharge Total Temperature/Total Pressure Kiel Probe
Tip (United Sensor KT-8-E-12-E)**

Figure 5.7: Discharge Pipe Traverse Total Pressure Measurements

5.2.5 Relative Humidity

For tests using air as the working fluid, it was necessary to measure the relative humidity in order to obtain accurate values of the fluid properties, such as the gas constant R , and the isentropic exponent γ . A Mannix PTH-8708 portable thermohygrometer was used to measure the relative humidity with $\pm 5\%$ accuracy. With the ambient temperature and pressure known, it was possible then to calculate the mass fraction of water vapour present in the air, and the fluid properties were then calculated assuming an ideal gas mixture of dry air and water vapour.

Figure 5.8: Discharge Pipe Traverse Total Temperature Measurements

5.3 DATA ACQUISITION

Data acquisition was carried out in accordance with the general guidelines of section 4 of PTC 10. A flow diagram of the instrumentation and data acquisition system is presented in Fig. 5.9, and a summary of the instrumentation and data acquisition hardware is listed in Table 5.2, at the end of this chapter.

Data acquisition was conducted using the High Speed Wind Tunnel's data acquisition mainframe, and the existing data acquisition programs, with minor additions and modifications for the different instrumentation.

The data acquisition system consists of a Hewlett Packard HP3852A data acquisition mainframe, with an interface to a desktop PC, from which the data acquisition program is run. The existing data acquisition program (Jeffries, 1994 & 2001) was modified to accommodate the extra instrumentation of the rotating rig: Instead of a single pressure transducer and thermocouple used in the cascade wind tunnel, a low range differential pressure transducer for the flowmeter, a high range gage pressure transducer for the inlet and discharge pressure measurements, two thermocouples for measuring the inlet and discharge temperature, and a frequency meter for measuring the shaft speed were required. In addition, the data acquisition interface was modified to allow collection of multiple data points in a single test run.

For further details on the configuration and operation of the High Speed Wind Tunnel data acquisition system, the reader is referred to the work of Jeffries (1994 & 2001). Information on the modified data acquisition software source code is given in Appendix D.

5.3.1 Pressure and Temperature Measurements

The output from the differential pressure transducer and from the gage pressure transducer were measured using the existing HP44702B 13-bit High Speed Voltmeter cards installed in the HP3852A mainframe. All gage pressures were measured with the same transducer and voltmeter using a Type-J ScaniValve™, to eliminate any bias error from using separate instruments. Resolution on the voltmeters was set to 312.5 μV , which corresponds approximately to 0.05 psig, and 0.0007 psid.

The inlet and discharge temperatures were measured using the HP44708 20 channel multiplexing thermocouple compensated voltmeter. The voltmeter outputs the

samples up to the point at which a sample is taken, and is used to help determine the step delay and number of samples that should be used in the data acquisition. As shown in Fig. 5.11, the sample average has settled to a stable value after 200 samples. A delay of 0.1s was used when switching ScaniValve™ ports, or when switching instruments, to allow the signal level to settle.

Figure 5.10: Sample Transducer Output (PX-613)

5.3.2 Shaft Speed Measurement

The frequency of the TTL signal from the tachometer was measured using the HP44715 200 kHz 5-Channel Counter/Totalizer. The shaft speed measurements were

taken by counting the tachometer pulses for a duration of 1s. Thus the frequency counter is capable of resolving 1Hz, or 60 rpm. The average of five speed measurements of 1s duration were taken at each data point, to further ensure accuracy of the speed measurement.

Figure 5.11: Sample Transducer Cumulative Average Output (PX-613)

5.3.3 Gas Composition

The gas samples were delivered to the CANMET characterization laboratory within 24 hours of taking a sample, and the analysis was usually performed within hours of delivery of the samples. This was done to minimize contamination of the samples with

air. However, it can be expected that the actual concentration of the test gas was slightly higher than was indicated in the test reports.

Firstly, although the samples were analyzed as quickly as possible, some contamination of the sample can still occur by diffusion of air through the sample bag, and by leakage through the valves in the sample bags. In addition, the sample can be contaminated by incomplete evacuation of the sample bag. Although care was taken to bleed the line to the sample bag prior to taking the sample, it is also inevitable that some contamination will occur while taking the sample. Finally, when the sample is injected into the chromatograph, it is expected that there will be up to 0.2% molar fraction of air drawn into the chromatograph along with the sample (MacDonald, 2001).

Since the process of taking a sample from the test circuit was similar to the process used when injecting the samples into the chromatograph, it was assumed that the levels of contamination for these processes is also similar. The contamination due to diffusion is assumed to be negligible, and the contamination due to incomplete evacuation of the sample bag is also assumed to be fairly small ($<0.1\%$). Thus, the actual concentration of the test gas is estimated to be up to 0.4% higher, on a molar basis, than the values given in the test reports.

5.4 DATA REDUCTION

5.4.1 Fluid Properties of Gases and Gas Mixtures

5.4.1.1 Air

As discussed in section 2.2.2, due to the relatively small variations in temperature, it was possible to model all of the working fluids as ideal gases. For air, the maximum recorded temperature change during the tests was approximately 100°C , and within this range of temperatures, the variations in fluid properties do not have a significant impact

on the results. However, for air, variations in relative humidity were found to have a noticeable effect on the mass flow parameter (due to variations in density), and the efficiency (due to variations in the isentropic exponent). Thus it was necessary to determine the fluid properties by modelling moist air as an ideal gas mixture of dry air and water vapour. Curve fit equations (Irvine, 1984) were used to determine the fluid properties of dry air and water vapour, and the gas constant and isentropic exponent were determined by taking the mass average of the fluid properties for air and water vapour at the average temperature between inlet and discharge, as described below.

First, the humidity ratio, w , was calculated from the relative humidity:

$$w = 0.62188 RH \frac{P_{sat}}{P - RH \times P_{sat}} \quad \text{Eq. 5.3}$$

where RH is the relative humidity, P is the ambient pressure, and P_{sat} is the saturation pressure of water vapour at the ambient temperature. The following curve fit equation was used to determine the saturation pressure of water vapour (Irvine):

$$P_{sat} = e^{\left(\sum_{n=0}^{n=9} a_n T^n \right) - \left(\frac{b}{T - c} \right)} \quad \text{Eq. 5.4}$$

Equation 5.4 gives the saturation pressure in kPa when the temperature T is in Kelvin. The coefficients and constants for all curve fit equations are given in Appendix E. The specific heat at constant pressure for dry air and water vapour, in kJ/kgK, are then calculated as follows (Irvine):

$$c_{pair} = \sum_{n=0}^{n=4} a_n T^n \quad \text{Eq. 5.5}$$

$$c_{pH_2O} = 237.6 \times 10^{-9} T^2 - 97.9 \times 10^{-6} T + 1.847 \quad \text{Eq. 5.6}$$

The specific heat and gas constant of the mixture is then calculated as the mass average of the values for dry air and water vapour:

$$c_p = m_{air} c_{pair} + m_{H_2O} c_{pH_2O} = (1 - w) c_{pair} + w c_{pH_2O} \quad \text{Eq. 5.7}$$

$$R = m_{air} R_{air} + m_{H_2O} R_{H_2O} = R_U \left(\frac{(1 - w)}{Mw_{air}} + \frac{w}{Mw_{H_2O}} \right) \quad \text{Eq. 5.8}$$

R_U is the universal gas constant (8.314 kJ/kgK), and Mw_{air} and Mw_{H_2O} are the molecular weights of air and water vapour respectively. The isentropic exponent can then be calculated from Eq. 5.9:

$$\gamma = \frac{c_p}{c_v} = \frac{c_p}{c_p - R} \quad \text{Eq. 5.9}$$

5.4.1.2 Carbon Dioxide and Argon

The chromatographic analysis of the CO₂ and argon samples, as discussed previously in section 4.2.1 revealed that essentially pure CO₂ and argon were present in the circuit during tests using these gases. In addition, it was found that accounting for the air content in the samples did not significantly change the values of the fluid properties, and the fluid properties were determined assuming the test gas to be pure and dry.

For argon, as mentioned previously in section 2.2.2, the highest recorded temperature change was approximately 200°C. Within this temperature range, there is no noticeable change in fluid properties. Thus constant values of the gas constant ($R = 208 \text{ J/kgK}$),

specific heat at constant pressure ($c_p = 520.3 \text{ J/kgK}$), and isentropic exponent ($\gamma = 1.667$) were used.

For CO_2 , the highest recorded temperature change was approximately 100°C . Within this range of temperatures, the change in fluid properties was found to have a noticeable impact on the efficiency results. Sixth degree polynomial curve fits (Irvine, 1994) were used to determine the fluid properties for CO_2 as a function of temperature. The polynomial coefficients are presented in Appendix E.

5.4.2 Measured Test Parameters

Speed Parameter:

The speed parameter (Eq. 2.1) was calculated directly from the measured rotational speed in RPM, the fluid properties at the compressor inlet, and the inlet stagnation temperature.

$$\Pi_v = \frac{N}{\sqrt{T_{01}}} \left(\frac{D}{\sqrt{\gamma R}} \right)$$

Flow Parameter:

In order to calculate the flow parameter, it was first necessary to determine the mass flow rate in the compressor, which is obtained from the flow velocity at the nozzle throat, the flow nozzle throat area, and the local density ($\dot{m} = \rho A v$). The flow nozzle velocity was calculated using Eqs. 5.1 & 5.2 from section 5.2.1.

$$v = C_D \sqrt{\frac{2\Delta P}{\rho(1 - \beta^4)}}$$

$$C_D = \frac{(1 - \beta^4) \left(\frac{P_2}{P_1} \right)^{\frac{2}{\gamma}} \left(\frac{\gamma}{\gamma - 1} \right) \left(1 - \left(\frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} \right)}{\left(1 - \beta^4 \left(\frac{P_2}{P_1} \right)^{\frac{2}{\gamma}} \right) \left(1 - \frac{P_2}{P_1} \right)}$$

The density at the nozzle throat was calculated from the measured pressure at that location, and the inlet temperature, using the ideal gas relation.

$$\rho = \frac{P}{RT}$$

The static gage pressure was measured at the nozzle throat, and the absolute pressure was determined by adding the gage pressure to the measured barometric pressure. It was necessary to calculate the flow velocity by iteration, since the inlet stagnation temperature was measured, and the static density is required in Eq. 5.1:

$$T = T_0 - \frac{v^2}{2c_p} \quad \text{Eq. 5.10}$$

The solution for the flow velocity at the nozzle throat in most cases converged to within 0.01% in five iterations. The flow parameter was then calculated directly from Eq. 2.2:

$$\Pi_m = \frac{\dot{m} \sqrt{T_{01}}}{P_{01}} \left(\frac{1}{D^2} \sqrt{\frac{R}{\gamma}} \right)$$

Pressure Ratio:

The pressure ratio was calculated from the measured stagnation gage pressures at the inlet and discharge of the compressor, and the measured barometric pressure:

$$\frac{P_{02}}{P_{01}} = \frac{P_{atm} + P_2}{P_{atm} + P_1} \quad \text{Eq. 5.11}$$

Efficiency:

The isentropic efficiency was calculated from the pressure ratio, and the measured inlet and discharge absolute stagnation temperatures, using Eq. 2.7:

$$\eta = \frac{T_{02s} - T_{01}}{T_{02} - T_{01}} = \frac{\left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{02}}{T_{01}} - 1}$$

Machine Reynolds Number:

The machine Reynolds number was calculated using Eq. 2.4:

$$Re_{b_2} = \frac{\rho_{01} U_2 b_2}{\mu}$$

where the inlet stagnation density is calculated from the inlet stagnation pressure and temperature, using the ideal gas relation. The blade tip speed U_2 is calculated from the rotational speed as follows:

$$U_2 = \frac{\pi N D}{60}$$

Although the viscosity μ varies with temperature, for the range of temperatures seen in the experiments, it can be treated as a constant. The viscosities of air, CO₂ and argon were assumed to be 1.846×10^{-5} Ns/m², 1.39×10^{-5} Ns/m², and 2.1×10^{-5} Ns/m² respectively.

5.4.3 Corrections to Measured Data

Since it was not possible to exactly reproduce the conditions of the manufacturer's map during the experiments, it was necessary to correct the pressure ratio and flow parameter for small deviations in the test speed parameter. In addition, the pressure ratio, flow parameter, and efficiency were corrected for deviations in the test Reynolds number Re_{b2} .

5.4.3.1 Corrections Due to Deviations in Test Speed Parameter

Although it was possible to maintain a fairly constant shaft speed by adjusting the turbine blowing pressure, it was difficult to set the exact speed desired for a given speed line. In general, it was possible to maintain the speed to within 500 rpm of the desired speed for the duration of the test. Since the absolute speeds ranged from approximately 35,000 to 96,000 rpm, this meant that it was possible for the majority of test cases to set the speed to within 1 % of the desired speed, and indeed for most of the test cases, the deviation in the shaft speed from the desired speed was less than 0.5%. In addition, an effort was made to ensure that the data for a given speed line included tests at both higher and lower rotational speed than the desired speed, to ensure that the average of the data matched as closely as possible the desired speed parameter. However, minor corrections to the data were still applied to account for the difference between the desired speed and the actual speed during the test.

If the pressure rise in the compressor is assumed to be proportional to the square of the rotational speed, the pressure ratio can be corrected as follows:

$$\left(\frac{1}{N^2} \left(\frac{P_{02}}{P_{01}} - 1 \right) \right)_i = \left(\frac{1}{N^2} \left(\frac{P_{02}}{P_{01}} - 1 \right) \right)_{sp} \quad \text{Eq. 5.12}$$

The subscripts t and sp are used to denote the conditions at the test conditions, and at the desired speed parameter respectively. The corrected pressure ratio is then obtained by comparison of the actual test speed and the desired test speed:

$$\left(\frac{P_{02}}{P_{01}}\right)_{sp} = 1 + \left(\left(\frac{P_{02}}{P_{01}}\right)_t - 1\right) \left(\frac{N_{sp}}{N_t}\right)^2 \quad \text{Eq. 5.13}$$

Note that Eq. 5.13 is only used to correct for small (e.g. < 1%) deviations in the test speed parameter, and should not be used to predict the pressure ratio for a different speed line.

5.4.3.2 Reynolds Number Correction

The measured performance was corrected to account for deviations in the test Reynolds number using the method of Strub et al. (1987). Although there are many Reynolds number correction methods available in the literature, the method used was deemed to be the most suitable since it has been validated over a wide range of Reynolds number, including the comparatively low machine Reynolds numbers seen experimentally. The Reynolds number correction of ASME PTC 10 for instance is based on the method of Carter et al. (1960), and has not been validated for test Reynolds numbers below 90,000 for centrifugal compressors. Since the Reynolds numbers in some tests were as low as 49,000, this method could not be used. In addition, the method of Strub et al. also provides a means of correcting the flow rate and pressure ratio as well as the efficiency, although these corrections are generally quite small.

Efficiency Correction:

The efficiency correction is of the following form:

$$\frac{1 - \eta_{sp}}{1 - \eta_t} = \frac{0.3 + 0.7 \frac{\lambda_{sp}}{\lambda_{cr}}}{0.3 + 0.7 \frac{\lambda_t}{\lambda_{cr}}} \quad \text{Eq. 5.14}$$

The variable λ is the friction factor in the impeller passage, from Colebrook (1939). The subscript cr , refers to the critical value of the friction factor, calculated as follows:

$$\lambda_{cr} = 1.74 - 2 \log_{10} \left(2 \frac{e}{b_2} \right) \quad \text{Eq. 5.15}$$

Calculation of the friction factors at the test and specified conditions using the Colebrook equation is an iterative procedure. However, Haaland (1983) provides explicit formulae for the Colebrook friction factors at the test and specified Reynolds number:

$$\frac{1}{\sqrt{\lambda}} = -\frac{1.8}{n} \log_{10} \left[\left(\frac{6.9}{\text{Re}_D} \right)^n + \left(\frac{e}{3.75D} \right)^{1.11n} \right] \quad \text{Eq. 5.16}$$

A value of $n = 2.75$ was used in Eq. 5.16, since it correlated well with the Colebrook equation in the range of Reynolds numbers seen experimentally.

Pressure Ratio and Flow Parameter Corrections:

Near the best efficiency point, Strub et al. noted that approximately half of the change in efficiency will result in a change in the pressure ratio:

$$\frac{PR_{sp} - 1}{PR_t - 1} = 0.5 + 0.5 \frac{\eta_{sp}}{\eta_t} \quad \text{Eq. 5.17}$$

The authors assume that the pressure ratio - flow characteristic at a given point is approximately parabolic. The flow parameter is then corrected as follows:

$$\frac{\Pi_{msp}}{\Pi_{mt}} = \sqrt{\frac{PR_{sp}}{PR_t}} \quad \text{Eq. 5.18}$$

In general, the deviations of the test Reynolds number from the implied Reynolds number of the manufacturer's map were modest, and the efficiency correction was in most cases less than 1%. The magnitude of the flow parameter and pressure ratio corrections were generally less than 0.5% of the full scale for the mass flow parameter, and 1% of the full scale for the pressure ratio.

5.5 ASSESSMENT OF ERRORS IN TEST PARAMETERS

5.5.1 Uncertainties in Measured Test Quantities

In general, the uncertainty in the measured pressures, temperatures, and shaft speed can be divided into five categories:

1. **Single Sample Error:** This represents the accuracy and resolution of the sensor in taking a single test reading, and was discussed in section 5.3.
2. **Sensor Calibration Uncertainty:** As shown in section 5.1, there is some scatter in the calibration data, which also results in measurement uncertainty.
3. **Test Measurement Uncertainty.** The scatter in a single test measurement represents an additional source of uncertainty.
4. **Fluctuations of measured test quantities during a test:** Although the fluctuations in the measured quantities in a single test measurement were small, there was some fluctuation in the operating point during a test, largely due to fluctuations

in the turbine blowing pressure. This introduced additional sources of uncertainty in the calculation of test parameters.

5. **Experimental uncertainties:** Due to imperfections in the test apparatus, it can be expected that there will be some systematic errors in all measured quantities. These may include errors in pressure measurements due to small misalignments of the probes, imperfections in pressure taps, or leakage of gas from the test circuit, and errors in temperature measurements due to heat transfer through the insulated sections of the apparatus.

The sample error and sensor calibration uncertainty are reduced by taking a large number of samples at each test measurement, and by repeating each test measurement three times in succession for each data point. The scatter in the data for a single test measurement was calculated as the difference between the maximum and minimum sample readings divided by the sample average. Typical fluctuations of single test measurements are presented for each sensor in Table 5.1. The scatter in approximately 80% of the test measurements is less than the values shown in the table. Although the standard deviation calculated from the measured test samples is much smaller than the scatter in the data, a conservative estimate of the combined uncertainties of a single test measurement due to sample error, calibration uncertainty and scatter in the data, is to consider the values accurate to within the typical fluctuations of a single measurement.

ASME PTC 10 provides guidelines for the maximum allowable fluctuations in measured quantities during a test, which is calculated as the difference between the maximum and minimum test readings divided by the average of the three test readings. Test points in which the fluctuations in any of the measured quantities exceeded the guidelines in section 3.3 of PTC 10 (Table 5.2) were rejected, with the exception of test points at very low flow rates, where due of the square root relationship (Eq. 5.1) resolution on the flow velocity is poor. In these cases, flow rate measurements with up to 4% fluctuation between test measurements were accepted.

Measurement	Instrument	Sample Error	Sample Resolution	Typical Scatter of Single Measurement
Flowmeter Differential Pressure	Omega PX-139 Differential Pressure Transducer	0.1% FS	312.5 μ V (0.000694 psi)	0.1 % of Measured Differential Pressure
Inlet and Discharge Gage Pressure	Omega PX-613 Gage Pressure Transducer	0.1% FS	312.5 μ V (0.0469 psi)	0.5 % of Absolute Measured Pressure
Inlet and Discharge Temperature	Type-E Grounded Thermocouple Junction	0.1 $^{\circ}$ C	0.01 $^{\circ}$ C	0.25 % of Measured Absolute Temperature
Shaft Speed	United Sensor IRS-5W Infrared Tachometer	1 Hz	1 Hz (60 rpm)	0.5 % of Measured RPM

Table 5.1: Typical Fluctuations in Single Test Point Measurements

The total measurement uncertainty from all sources was estimated by summing the uncertainty of a single measurement and the typical fluctuation during a test. In general, the total estimated uncertainty of the measurements for a test point is lower than the maximum permissible fluctuations in the test readings, except in the case of the shaft speed, and at low flow rates, the nozzle differential pressure. The uncertainty in the measurement of the inlet and discharge pressure was therefore conservatively estimated to be equal to the maximum permissible fluctuation in test readings as given by ASME PTC 10. In other words, the inlet and discharge pressure readings are considered to be accurate to within $\pm 1\%$ of the measured absolute pressure, and the inlet and discharge temperatures are considered accurate to within $\pm 0.25\%$ of the measured absolute temperature. Since the relative uncertainty in the flow nozzle differential pressure is

increased significantly at low flow rates, the uncertainty in the measurement of the flow nozzle differential pressure was estimated to be equal to the maximum permissible fluctuation in a test reading under most circumstances, and approximately three times greater at low flow rates. The flow nozzle differential pressure is therefore considered to be accurate to within $\pm 1\%$ of the measured differential pressure in most cases, and it is considered accurate to within $\pm 3\%$ of the measured differential pressure for tests at low flow rates ($\Pi_m < 0.02$).

Measurement	Maximum Permissible Fluctuation (%)	Typical Fluctuation During Present Tests (%)
Inlet Pressure	2.0	0.2
Inlet Temperature	0.5	0.2
Discharge Pressure	2.0	0.2
Discharge Temperature	0.5	0.2
Nozzle Differential Pressure	2.0	1.5
Shaft Speed	0.5	0.2

Table 5.2: Permissible Fluctuations of Test Readings (Table 3.1, PTC 10, 1997), and Typical Fluctuations Seen During Performance Tests

5.5.2 Uncertainties in Calculated Test Parameters

The uncertainty in the test parameters is estimated by examining the effects of errors in the measured test quantities on the calculated test parameters. Classical uncertainty analysis shows that the uncertainty of a calculated result R is related to the uncertainties of the measured quantities x_i by the following relationship (Moffat, 1985):

$$\delta R = \sqrt{\sum \left(\frac{\delta R}{\delta x_i} \delta x_i \right)^2}$$

Eq. 5.19

For the parameters that are products of measured quantities raised to certain exponents, the relative uncertainties in the test parameters can be expressed as follows (Moffat):

$$R = kx_1^a x_2^b \dots x_n^m$$

$$\frac{\delta R}{R} = \sqrt{\left(a \frac{\delta x_1}{x_1} \right)^2 + \left(b \frac{\delta x_2}{x_2} \right)^2 + \dots + \left(m \frac{\delta x_n}{x_n} \right)^2}$$

This approach was used to calculate the uncertainty in the flow parameter, speed parameter, pressure ratio and efficiency, with the results summarized in Table 5.3.

For the flow parameter, since the relative error in the differential pressure increases at low flow rates, the uncertainty is given at both low and high flow rates. The error in the efficiency calculation is greatly increased at low pressure ratio, and the uncertainty was therefore calculated for representative values for each speed line. The uncertainty in the efficiency calculation is seen to be consistent with the measured scatter in the data, which is also increased at low pressure ratios. In addition, the uncertainty in the efficiency calculation is increased slightly for high values of the isentropic exponent, and the uncertainties in Table 5.3 are given for argon, which has the highest value of γ of the test gases used.

Test Parameter	Estimated Uncertainty
Flow Parameter	
$\Pi_m < 0.02$	± 0.004 (2.5% FS)
$\Pi_m > 0.02$	± 0.0008 (0.5% FS)
Speed Parameter	± 0.15 (0.7% FS)
Pressure Ratio	± 0.03 (1.3% FS)
Efficiency:	
$\Pi_N = 10.06$	± 4.7 (6.3% FS)
$\Pi_N = 15.23$	± 2.1 (2.8% FS)
$\Pi_N = 18.35$	± 1.6 (2.1% FS)
$\Pi_N = 21.03$	± 1.2 (1.6% FS)

Table 5.3: Estimated Total Error in Test Parameters

6.0 COMPUTATIONAL METHODS

In addition to measuring the actual performance of the compressor, a meanline analysis program was developed to model the compressor, and CFD computations were made for the rotor. Since only the approximate rotor geometry was known, the meanline and CFD results were not intended for direct comparison with the measurements. Instead, they were intended to investigate aspects of the performance which could not be examined experimentally, and to determine whether a meanline analysis would correctly capture the effects of γ on the performance predictions of a centrifugal compressor.

In the first place, because of the need to control the circuit leakage, it was not possible to conduct tests for all three test gases at the same Reynolds number. However, the Reynolds number could be fixed in the computations and the calculations were therefore used to provide supporting evidence that the performance differences observed in the measurements were due to the differences in γ , not the modest variations in Reynolds number.

Secondly, the losses in the test circuit are too high to allow choking of the present rotor. In addition to the pressure ratio and efficiency, the choking mass flow rate for a given speed parameter is also an important aspect of the performance. This is particularly true for the turbines in gas turbine engines: The choking mass flow rate for the turbine plays an important part in determining the steady state operating line of the compressor to which it is connected. The CFD computations were used to investigate the effect of γ on the choking mass flow rate.

Finally, since it was not known whether the losses in the diffuser and volute would be strongly affected by the change in fluid properties, the meanline analysis results were used to investigate the effect of changing the working fluid on the performance of these components. The results of the meanline analysis were also used to evaluate the ability

of a meanline analysis in predicting the performance of turbomachinery for working fluids other than air.

6.1 PREDICTION OF THE IMPELLER PERFORMANCE USING CFD

The CFD computations were performed using CFX-BladeGen+. This is a two-part code consisting of a geometric design (CAD) module and a CFD module. The CFD module is based on the CFX-5 Navier-Stokes code. Some features of the Navier-Stokes code have been disabled or supplied with default values to allow rapid specification of the computational case. For example, only one turbulence model is available, and the unstructured grid is generated semi-automatically based on the specification of a single grid refinement factor which controls the general grid density (Sjolander, 2001, CFX-BladeGen+ User's Manual, 2000).

The CFD module is intended to allow rapid aerodynamic screening of trial turbomachine geometries generated with the CAD module, rather than to give the final, detailed aerodynamic performance estimates. In the present study, the code was used to determine whether the aerodynamic analysis tools currently used in the early stages of turbomachinery design capture satisfactorily the influence of fluid properties such as γ .

For the present calculations, an approximate geometric model of the rotor was created by the author, based on the dimensions given in Fig. 4.1, and the actual computations were performed by Prof. S. Sjolander. The computational domain included a short length of channel upstream and downstream of the rotor and extended circumferentially over one main vane spacing, including one of the splitter vanes. An intermediate grid density was used, giving about 185,000 tetrahedral and prismatic cells. The inlet total temperature and pressure, and the discharge static pressure were specified as boundary conditions. Both the grid density and convergence criterion were varied to confirm that the overall performance results were adequately converged and reasonably grid independent (Sjolander, 2001).

The results of the CFD computations for the three test gases will be presented and discussed in chapter 8.

6.2 PREDICTION OF COMPRESSOR PERFORMANCE USING A MEANLINE ANALYSIS

In the early stages of turbomachinery design, a meanline analysis is often used to determine some of the main geometric features of the compressor or turbine, such as the size of the machine, and approximate blade angles. Rather than using computationally expensive CFD, a meanline analysis can be useful in evaluating rapidly a large number of proposed designs prior to performing detailed design calculations.

In a meanline analysis, the performance of the compressor is predicted by examining the flow along the mean streamline through the impeller. It is then assumed that the velocity field and thermodynamic properties at the mean streamline are representative of the bulk flow properties in the compressor. The ideal discharge flow field at the mean streamline is calculated, and empirical correlations are used to calculate a total pressure or total enthalpy loss in each of the impeller, diffuser, and volute. The losses are then used to calculate updated estimates of the discharge flow conditions, and the process is repeated until the discharge flow conditions are consistent with the known inlet mass flow. The isentropic efficiency can then be calculated by comparison of the predicted inlet and discharge temperatures to the inlet and discharge temperatures for an isentropic process at the same pressure ratio, as discussed in section 2.2.2.

A schematic representation of the compressor, and the station numbering used in the meanline analysis is shown in Fig. 6.1.

The meanline analysis program, written in the FORTRAN 90 programming language, is described below, and information about the source code for the computer program can be found in Appendix D.

Figure 6.1: Stage Geometry and Station Numbering

6.2.1 Impeller Analysis

6.2.1.1 Inlet Flow Field

The first step in a meanline analysis is to compute the inlet velocity field of the impeller (Fig. 6.2). The thermodynamic state must be specified, as well as the mass flow rate, inlet swirl angle α_i , rotor speed N , and the geometry at the mean streamline. Flow

angles in the meanline analysis are measured relative to the meridional direction, and are considered to be positive in the direction of the blade rotation. Thus, in Fig. 6.2, α_1 , β_1 , and β_1' are all considered negative.

Figure 6.2: Impeller Inlet Velocity Field

The inlet absolute velocity is computed by applying conservation of mass at the rotor inlet.

$$\dot{m} = \rho_1 A_1 C_{a1} = \rho_1 A_1 C_1 \cos \alpha_1 \quad \text{Eq. 6.1}$$

Since the inlet stagnation conditions are specified, Eq. 6.1 must be solved iteratively, using the isentropic flow relations for an ideal gas:

$$M = \frac{C_1}{\sqrt{\gamma RT}} \quad \text{Eq. 6.2}$$

$$\frac{\rho_0}{\rho} = \left(1 + \frac{\gamma-1}{2} M^2\right)^{\frac{1}{\gamma-1}} \quad \text{Eq. 6.3}$$

The inlet blade speed and relative velocity are then calculated as follows:

$$U_1 = \frac{\pi N D_1}{60} \quad \text{Eq. 6.4}$$

$$W_1 = \sqrt{C_{a1}^2 + (U_1 - C_1 \sin \alpha_1)^2} \quad \text{Eq. 6.5}$$

6.2.1.2 Discharge Flow Field and Fluid Properties

Slip Factor

The ideal discharge velocity field can be computed by assuming that the discharge relative flow W_2 exits the impeller at the blade metal angle β_2 . However, because the flow will not be perfectly guided by the impeller blades, there will be some deviation between the impeller discharge blade angle β_2' , and the flow angle β_2 . In centrifugal machines, this deviation is called slip.

Although the slip factor does not directly affect the impeller efficiency, the impeller slip is an important parameter in the performance of the compressor. As will be seen shortly, the impeller slip has the effect of reducing the discharge tangential velocity C_{u2} , which in turn decreases the work input, and thus the impeller pressure ratio.

The slip factor σ is defined as the ratio of the actual discharge radial velocity C_{u2} to the radial velocity that would exist if there was no deviation between the discharge blade angle and flow angle, and is shown schematically in Fig. 6.3.

$$\sigma = \frac{C_{u2}}{C_{u2} (\beta_2 = \beta_2')} = \frac{C_{u2}}{U_2 + C_{r2} \tan \beta_2'} \quad \text{Eq. 6.6}$$

Figure 6.3: Impeller Discharge Velocity Field Illustrating the Effect of the Slip Factor σ

The slip factor is calculated using the correlation of Wiesner (1967), which has been modified by Aungier (1995, 2000).

$$\sigma = 1 - \frac{\sqrt{\sin \beta_2'}}{Z^{0.7}} \quad \text{Eq. 6.7}$$

where Z is the number of blades in the impeller. For impellers with splitter blades, the equivalent number of blades can be estimated as follows:

$$Z = Z_{Full} + \frac{L_{Splitter}}{L_{Full}} Z_{Splitter} \quad \text{Eq. 6.8}$$

The slip factor is sometimes reduced for impellers having a high ratio of inlet to discharge mean radius ($\varepsilon = r_1/r_2$). In cases where the mean radius ratio ε is greater than the limiting value ε_{lim} , the slip factor is corrected as follows:

$$\sigma_{cor} = \sigma \left(1 - \frac{\varepsilon - \varepsilon_{lim}}{1 - \varepsilon_{lim}} \right)^{\beta_2' / 10} \quad \text{Eq. 6.9}$$

$$\varepsilon_{lim} = \frac{\sigma - \sigma^*}{1 - \sigma^*} \quad \text{Eq. 6.10}$$

$$\sigma^* = \sin(37^\circ + \beta_2) \quad \text{Eq. 6.11}$$

Discharge Velocity and Fluid Thermodynamic Properties:

The ideal discharge velocity field is computed by satisfying conservation of mass at the impeller discharge:

$$\dot{m} = \rho_2 A_2 C_{r2} \quad \text{Eq. 6.12}$$

However, the discharge density ρ_2 is not yet known, and the discharge flow field and thermodynamic properties must be calculated by iteration. With an initial estimate of the discharge density, we can estimate the discharge velocity in the radial direction C_{r2} from Eq. 6.12, and with the slip factor, we can solve for the discharge tangential and absolute velocity.

$$C_{u2} = \sigma \left(U_2 + C_{r2} \tan \beta_2' \right) \quad \text{Eq. 6.13}$$

$$C_2 = \sqrt{C_{u2}^2 + C_{r2}^2} \quad \text{Eq. 6.14}$$

Since the blade speed U_2 is known, we can then calculate the relative velocity and the flow angles in Fig. 6.3. The impeller work and discharge stagnation temperature are then calculated from the Euler equation:

$$\Delta h_0 = U_2 C_{u2} - U_1 C_{u1} \quad \text{Eq. 6.15}$$

$$T_{02} = T_{01} + \frac{\Delta h_0}{c_p} \quad \text{Eq. 6.16}$$

Finally, the impeller loss Δh_{0Loss} is computed using the method described in the following section. The impeller loss is then used to determine the discharge stagnation temperature for an isentropic compression T_{02s} , and the impeller pressure ratio.

$$T_{02s} = T_{02} - \frac{\Delta h_{0Loss}}{c_p} \quad \text{Eq. 6.17}$$

$$\frac{P_{02}}{P_{01}} = \left(\frac{T_{02s}}{T_{01}} \right)^{\frac{\gamma}{\gamma-1}} \quad \text{Eq. 6.18}$$

The discharge static pressure, temperature, and density are then calculated using the isentropic flow relations for an ideal gas. The calculated discharge density is used to compute an updated estimate of the discharge radial velocity C_{r2} in Eq. 6.12. The process of Eqs. 6.12 – 6.18 is then repeated until the computed discharge conditions satisfy continuity.

6.2.1.3 Impeller Losses

The losses in centrifugal compressor impellers are dependent on many different design parameters, and complex, three dimensional flows are present in the impeller, diffuser, and volute. However, experimental data can be used to correlate the pressure or enthalpy losses due to a particular loss mechanism in the impeller to some of the main geometric features and one-dimensional flow parameters.

Oh et al. (1997) recently reviewed the loss correlations available in the open literature, and compared the ability of different combinations of loss correlations for predicting the performance of a variety of centrifugal compressors to arrive at what they feel is an optimum set of loss correlations. With the exception of the incidence loss, the set of correlations recommended by Oh et al. is used in the meanline analysis, and is briefly described below.

Internal Losses:

(1) Incidence Loss:

An incidence loss occurs due to misalignment of the relative velocity vector W_1 and the blade angle at the impeller inlet β_1' . Since the fluid must flow along the impeller passage, there is an abrupt change in the relative velocity in the vicinity of the impeller leading edge. The incidence loss of Ferguson (1963) is used in the program:

$$\Delta h_{0inc} = \frac{(U_1 - C_{u1})^2}{2} \quad \text{Eq. 6.19}$$

(2) Blade Loading Loss:

A blade loading loss is computed to account for blade to blade pressure gradients that produce secondary flows in the impeller passage, and may lead to blade stall. The blade loading loss correlation of Coppage et al. (1956) is used:

$$\Delta h_{0blt} = 0.05(D_f U_2)^2 \quad \text{Eq. 6.20}$$

$$D_f = 1 - \frac{W_2}{W_{1t}} + \frac{0.75(U_2 C_{u2} - U_1 C_{u1})}{U_2^2 (W_{1t} / W_2) \left((Z / \pi) (1 - D_{1t} / D_2) + 2 D_{1t} / D_2 \right)} \quad \text{Eq. 6.21}$$

The subscript t is used to denote the condition at the blade tip.

(3) Skin Friction Loss:

A skin friction loss is computed to account for friction along the walls of the impeller passage. The skin friction loss correlation of Jansen (1967) is used, where the subscript h denotes the condition at the blade hub.

$$\Delta h_{0sf} = 2c_f \frac{L_b}{D_h} \bar{W}^2 \quad \text{Eq. 6.22}$$

$$\bar{W} = \frac{C_{1t} + C_2 + W_{1t} + 2W_{1h} + 3W_2}{8} \quad \text{Eq. 6.23}$$

The skin friction coefficient c_f is calculated according to the method of Schlichting (1979), from the impeller surface roughness, and the impeller passage Reynolds number. The velocity scale in the impeller passage Reynolds number is the average relative velocity in the impeller passage from Eq. 6.23, and the length scale is the average of the hydraulic diameters of the impeller passage at inlet and discharge:

$$D_h = \frac{2A_1}{P_1} + \frac{2A_2}{P_2} \quad \text{Eq. 6.24}$$

where A and P are respectively the impeller passage area and perimeter normal to the flow direction. The impeller passage Reynolds number is then calculated as follows:

$$\text{Re}_{dh} = \frac{\bar{\rho} \bar{W} D_h}{\mu} \quad \text{Eq. 6.25}$$

where $\bar{\rho}$ is the average of the inlet and discharge static density.

(4) Tip Clearance Loss:

The correlation of Jansen (1967) is used to calculate the tip clearance loss:

$$\Delta h_{ocl} = 0.6 \frac{\varepsilon}{b_2} C_{u2} \sqrt{\frac{4\pi}{b_2 Z} \frac{r_{1t}^2 - r_{1h}^2}{(r_2 - r_{1t})(1 + \rho_2 / \rho_1)}} C_{u2} C_{a1} \quad \text{Eq. 6.26}$$

where ε is the clearance between the blade tip and the impeller casing.

(5) Mixing Loss:

The mixing loss according to Johnston and Dean (1966) is computed to account for losses due to mixing of the impeller blade wakes with the free stream flow.

$$\Delta h_{omix} = \frac{1}{1 + \tan^2 \alpha_2} \left(\frac{1 - \varepsilon_{wake} - b_3 / b_2}{1 - \varepsilon_{wake}} \right)^2 \frac{C_2^2}{2} \quad \text{Eq. 6.27}$$

where b_3 is the diffuser inlet width (see Fig. 6.4), and ε_{wake} is the wake fraction of the blade to blade space, which is estimated from the meridional velocities before and after mixing as follows (Aungier, 1995, 2000).

First, the diffusion factor D_{eq} is calculated (Lieblein, 1959):

$$D_{eq} = \frac{W_1 + W_2 + 2\pi D_2 \frac{U_2 C_{u2} - U_1 C_{u1}}{U_2 Z L_b}}{2} \quad \text{Eq. 6.28}$$

To estimate the size of the wake, we must estimate the velocity at which separation occurs W_{sep} .

$$W_{sep} = W_2 ; \quad D_{eq} < 2$$

$$W_{sep} = W_2 \frac{D_{eq}}{2} \quad D_{eq} \leq 2$$

The meridional velocities before and after mixing can then be estimated as follows (Aungier, 1995, 2000).

$$C_{mWake} = \sqrt{W_{sep}^2 - W_{u2}^2} \quad \text{Eq. 6.29}$$

$$C_{mMix} = \frac{C_{m2} A_2}{\pi D_2 b_2} \quad \text{Eq. 6.30}$$

If it is assumed that there is no significant variation in density in the tangential direction, the wake fraction can be estimated by applying conservation of mass in the mixing zone:

$$\varepsilon_{wake} = 1 - \frac{C_{mMix}}{C_{mWake}} \quad \text{Eq. 6.31}$$

Parasitic Losses:

In addition to the losses in the impeller passages, there is an additional work input that does not result in any compression of the fluid. This work input is called the parasitic loss, and includes the effects of windage and recirculating flow. Some authors also include the tip clearance loss as a parasitic loss, but since this mechanism has already been considered in the impeller losses, it is ignored here. The parasitic losses are calculated using the correlations recommended by Oh et al. (1997).

(1) Disk Friction Loss:

The disk friction loss is calculated using the method of Daily and Nece (1960).

$$\Delta h_{0df} = f_{df} \frac{(\rho_1 + \rho_2) r_2^2 U_2^3}{8\dot{m}} \quad \text{Eq. 6.32}$$

where the disk friction factor f_{df} is given by:

$$f_{df} = \frac{2.67}{\sqrt{\text{Re}_{df}}} \quad \text{Re}_{df} < 30,000$$

$$f_{df} = \frac{0.0622}{\text{Re}_{df}^{0.2}} \quad \text{Re}_{df} \geq 30,000$$

$$\text{Re}_{df} = \frac{\rho_2 U_2 r_2}{\mu}$$

(2) Recirculation Loss:

The recirculation loss represents the energy lost when high pressure fluid from the impeller discharge is leaked back to the inlet and re-compressed. The recirculation loss is calculated from Oh et al. (1997).

$$\Delta h_{orc} = 8.0 \times 10^5 \sinh(3.5\alpha_2^3) D_f^2 U_2^2 \quad \text{Eq. 6.33}$$

where the discharge absolute flow angle α_2 in Eq. 6.33 is expressed in radians.

6.2.2 Diffuser Analysis

Aungier (2000) provides a one-dimensional performance analysis for vaneless diffusers of centrifugal and mixed-flow compressors (Fig. 6.4), which is based on a classical straight pipe diffuser analysis. The method uses a finite difference formulation to calculate the flow field and thermodynamic properties at each successive downstream station based on the values at the current location.

The flow field and thermodynamic properties at the inlet to the diffuser are assumed to be equal to the conditions at the impeller exit. Since there is no work input in the diffuser, there is no change in the stagnation enthalpy from inlet to discharge. Thus, for an ideal gas, the diffuser discharge stagnation temperature is equal to the stagnation temperature at the impeller discharge.

The governing equations for one-dimensional flow in a vaneless passage are given by Aungier (2000):

(1) Continuity:

$$\dot{m} = 2\pi r \rho b C_m (1 - B)$$

Eq. 6.34

The subscript m denotes the component of velocity in the meridional direction, which is the radial direction in centrifugal compressor vaneless diffusers. B is the area blockage in the diffuser passage due to the boundary layer.

Figure 6.4: Vaneless Diffuser Geometry

(2) Conservation of Angular Momentum:

$$bC_m \frac{d(rC_u)}{dm} = -rCC_u c_f \quad \text{Eq. 6.35}$$

The variable m is the meridional distance along the diffuser. Again, for centrifugal compressor vaneless diffusers, this is in the radial direction. Equation 6.30 predicts the rate of decay of the tangential component of velocity C_u due to friction and to the increased radial distance of the fluid as it moves through the diffuser.

The area blockage factor B in Eq. 6.34, and the skin friction coefficient c_f in Eq. 6.35 are calculated using a simple boundary layer growth model, based on a $1/7^{\text{th}}$ power boundary layer velocity profile. If the boundary layer velocity profiles are assumed to be the same on both diffuser walls, integration of the velocity across the passage width yields the following expression for the area blockage factor (Aungier, 2000):

$$B = \frac{2\delta}{8b}$$

The initial boundary layer thickness δ_3 is estimated from:

$$\frac{2\delta_3}{b_3} = 1 - \left(\frac{b_3}{r_3} \right)^{0.15}$$

The boundary layer thickness at each successive downstream location is predicted from the following relation (Aungier, 2000):

$$\delta = 2.25b \left(1 - \frac{rC_u}{rC_{ue}} \right) \quad \left(\delta \leq \frac{b}{2} \right) \quad \text{Eq. 6.36}$$

where the subscript e denotes the value at the edge of the boundary layer. The maximum combined thickness of the diffuser wall boundary layers is the local diffuser width. Thus, the thickness of the diffuser wall boundary layer in Eq. 6.36 is limited to half of the local diffuser width b . The skin friction coefficient is calculated from the skin friction law for turbulent flow over a flat plate (White, 1994):

$$c_f = 0.02 \left(\frac{\rho U_e \delta}{\mu} \right)^{-1/6}$$

(3) Conservation of Linear Momentum:

$$\frac{1}{\rho} \frac{dP}{dm} = \frac{C_u^2}{r} - C_m \frac{dC_m}{dm} - \frac{CC_m c_f}{b} - \frac{dI_D}{dm} \quad \text{Eq. 6.37}$$

The first three terms on the right hand side of Eq. 6.37 are based on the stall-free straight-walled diffuser analysis of Reneau et al. (1967). The last term in Eq. 6.37 is a diffusion loss term that has been added by Aungier (2000), apparently to account for stall and flow separation in the diffuser, and is calculated as follows:

First, the divergence parameter is calculated:

$$D = -\frac{b}{C} \frac{dC}{dm} \quad \text{Eq. 6.38}$$

The diffusion losses are low for values of the diffusion parameter less than the limiting value D_m :

$$D_m = 0.4 \left(\frac{b_3}{L} \right)^{0.35} \quad \text{Eq. 6.39}$$

Aungier (2000) provides an empirical diffuser efficiency factor E , based on experimental data from various different compressor stages.

$$\begin{aligned}
 E &= 1 & D &\leq 0 \\
 E &= 1 - 0.2 \left(\frac{D}{D_m} \right)^2 & 0 < D < D_m \\
 E &= 0.8 \sqrt{\frac{D_m}{D}} & D &\geq D_m
 \end{aligned}$$

The diffusion term in Eq. 6.37 is then calculated as follows:

$$\frac{dI_D}{dm} = -2(P_0 - P)(1 - E) \frac{1}{\rho C} \frac{dC}{dm} \quad \text{Eq. 6.40}$$

The diffusion loss is increased when the passage area gradient in the meridional direction is high. This is analogous to a straight diffuser with a high divergence angle. If the diffuser passage area increases too rapidly, flow separation will occur along the diffuser wall. In classical straight diffuser analysis, the maximum, stall free, diffuser divergence angle is approximately 9° . Aungier (2000) uses this area gradient to estimate the maximum, stall free, local area for a centrifugal compressor vaneless diffuser:

$$(rb)_{\max} = (rb)_3 \left(1 + 0.16 \frac{r}{b_3} \right)$$

If the local area exceeds this value, the diffusion term is estimated from:

$$I_D = 0.65 \frac{P_0 - P}{\rho} \left(1 - \frac{(rb)_{\max}}{rb} \right)$$

The solution to the flow in the diffuser is obtained by numerical integration of the governing equations of motion. Equations 6.35 – 6.37 are cast in a finite difference formulation, which is solved explicitly by stepping in the meridional (radial) direction from the inlet to the discharge of the diffuser.

6.2.3 Volute Analysis

Aungier (2000) presents a simple, one-dimensional analysis for volutes and collectors (Fig. 6.5), which is based on the method of Weber and Koronowski (1986).

The flow field and thermodynamic properties at the inlet to the volute are assumed to be known from the conditions at the diffuser exit. As with the diffuser, there is no work done on the fluid in the volute, and the discharge stagnation temperature is equal to the stagnation temperature at the inlet. However, the flow field and remaining thermodynamic properties at the volute exit are not known, and must be calculated by iteration.

Figure 6.5: Volute Geometry and Station Numbering

Meridional Head Loss:

The radial component of the velocity at the diffuser exit becomes a swirl velocity in the volute, as shown schematically in Fig. 6.6. For the most part, this energy is not recovered, and the first loss considered corresponds to a complete loss of the radial dynamic pressure (Aungier, 2000):

$$\omega_m = \left(\frac{C_{r5}}{C_5} \right)^2 \quad \text{Eq. 6.41}$$

Figure 6.6: Volute Secondary Flow Illustrating Loss of Radial Kinetic Energy

Tangential Head Loss:

Under ideal conditions, the angular momentum rC_u is conserved in the volute. However, variations in flow area and mean radius in the circumferential direction can result in a loss of angular momentum, and some of the tangential dynamic pressure is assumed to be lost. The volute sizing parameter is defined as:

$$SP = \frac{r_5 C_{u5}}{r_6 C_6}$$

The optimum value of the sizing parameter is $SP = 1$, in which case there is no loss of tangential momentum due to variations in flow area and mean radius. For values of the sizing parameter other than $SP = 1$, a tangential velocity head loss is computed as follows (Aungier, 2000):

$$\omega_u = \frac{1}{2} \frac{r_5 C_{u5}^2}{r_6 C_5^2} \left(1 - \frac{1}{SP^2} \right) \quad SP \geq 1 \text{ Eq.} \quad 6.42 \text{ (a)}$$

$$\omega_u = \frac{r_5 C_{u5}^2}{r_6 C_5^2} \left(1 - \frac{1}{SP} \right)^2 \quad SP < 1 \text{ Eq.} \quad 6.42 \text{ (b)}$$

Skin Friction Loss:

A wall skin friction loss is computed for the volute (Aungier, 2000):

$$\omega_{sf} = 4c_f \left(\frac{C_6}{C_5} \right)^2 \frac{L}{d_H} \quad \text{Eq. 6.43}$$

where L is the average flow path length of a fluid particle in the diffuser, and d_H is the volute exit hydraulic diameter ($d_H = d_6$ for a cylindrical volute exit):

$$L = \frac{\pi}{2} (r_5 + r_6)$$

$$d_H = \sqrt{\frac{4A_6}{\pi}}$$

The solution to the flow in the volute is obtained by guessing initially the discharge density ρ_6 and computing the exit velocity C_6 by applying conservation of mass. The total pressure loss in the volute is then computed by summing the loss coefficients described above:

$$\Delta P_{0VL} = \sum \omega \times \frac{1}{2} \rho_s C_s^2$$

The discharge density ρ_6 is then computed from the known discharge stagnation temperature T_{06} , and the latest estimates of the discharge stagnation pressure P_{06} and velocity C_6 . This procedure is repeated until the mass balance between the inlet and discharge is satisfied to within a specified tolerance. Since the variations in density in the volute are small, convergence is generally rapid.

Once the flow field and thermodynamic state at the volute discharge are known, the overall stage efficiency can be calculated as follows:

$$\eta_{Stage} = \frac{\left(\frac{P_{06}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{06}}{T_{01}} - 1} \quad \text{Eq. 6.44}$$

6.3 RESULTS OF THE MEANLINE ANALYSIS USING AIR AS THE WORKING FLUID

The pressure ratio and isentropic efficiency as a function of mass flow parameter predicted using the meanline analysis are compared to the manufacturer's data in Figs. 6.7 and 6.8 respectively.

The peak efficiency predicted using the meanline analysis is approximately 3% lower than the manufacturer's quoted efficiency, and the peak efficiency is in general

predicted at a lower mass flow rate in the meanline analysis. In addition, there is also some discrepancy in the predicted pressure ratio, particularly at high flow rates.

Figure 6.7: Pressure Ratio Predicted by the Meanline Analysis – Comparison to Manufacturer's Results for Air

In the middle of the mass flow range, the discrepancies between the manufacturer's performance and the performance predicted by the meanline analysis are believed to be primarily due to inaccurate estimation of the rotor and volute geometries. In particular, the impeller backsweep angle was difficult to measure from the rotor, and it was also

difficult to estimate the tip clearance between the impeller blades and the casing. A somewhat better prediction of the actual compressor performance was achieved when a backsweep angle of 35° was used, but the original estimate of 27° was retained to maintain consistency with the CFD computations.

Figure 6.8: Isentropic Efficiency Predicted by the Meanline Analysis – Comparison to Manufacturer's Results for Air

At high values of the flow parameter, the difference between the results predicted by the meanline analysis and the manufacturer's data is believed to be due to poor prediction

of the impeller losses near the choking mass flow parameter. The meanline analysis did not predict any significant decrease in impeller efficiency between the stage peak efficiency point and the choking flow parameter, for any of the speed lines. Although this is not physically realistic, the results of the meanline analysis were found to be in good agreement with the published results of Oh et al. (1997) for the Eckhardt and KIMM impellers. The meanline analysis is therefore not recommended for use in predicting the performance of compressors operating at flow rates near choke limit.

Although the results of the meanline analysis are not directly comparable to the experimental results, as noted previously, accurate prediction of the performance of the compressor was not necessary. More importantly, the meanline analysis program was intended for use in determining whether loss correlations developed using air can be applied to other working fluids.

7.0 RESULTS AND DISCUSSION

In order to ensure proper working order of the test stand, it was operated first using air as the working fluid, and the experimental results were compared to the manufacturer's published data from Fig. 4.2. In addition, the results of the meanline analysis program were compared to the experimental results to verify the accuracy of the computed results.

Once the test stand was shown to provide good quality data, the tests were repeated using CO₂ and argon, for the same values of speed parameter as for air. The experimental results for CO₂ and argon were compared to the results for air in order to determine the effect of changing the working fluid on the aerodynamic performance of the compressor.

In addition, the results of the meanline analysis and of the CFD computations for argon and CO₂ were compared to the corresponding results for air, to verify consistency of the three methods, and to better understand the reasons for the changes in performance when using different working fluids.

7.1 COMMISSIONING THE TEST STAND

In the initial tests using air as the working fluid, the test stand was operated in open circuit arrangement (Fig. 4.3 (a)), in order to verify that it was possible to collect reliable and repeatable data, before proceeding with the purchase of the additional equipment required for closed circuit testing. Once the apparatus was proven to provide accurate and repeatable data, the tests were repeated in closed cycle arrangement, with virtually identical results.

Figures 7.1 & 7.2 show comparisons of the measured pressure ratio and isentropic efficiency for both open and closed circuit tests to the data supplied by the turbocharger supplier. Measurements for a given speed line were taken on different days to demonstrate repeatability of the test data. Care was also taken to maintain as closely as possible a constant test Reynolds number for a given speed line.

Figure 7.1: Comparison of Measured Pressure Ratio to Manufacturer's Test Data for Air

Figure 7.2: Comparison of Measured Efficiency to Manufacturer's Data for Air

The differences between the measured performance and the manufacturer's data, particularly at high pressure ratio are likely due to variations in manufacturing. The compressor volute was delivered as cast, and it can be expected that there would be some variation in performance for different machines of the same model due to manufacturing differences in the passage area and surface roughness of the volute. However, the reasonable agreement between the test data and the data supplied by the manufacturer is sufficient to show proper functioning of the test apparatus.

Additionally, in tests conducted using air, changes in relative humidity were found to have a noticeable effect on both the flow rate and efficiency measurements. This is believed to be the main source of scatter in Figs. 7.1 and 7.2. Since pure, dry argon and CO₂ gases were used in subsequent tests, the scatter in the results for these gases was much less noticeable.

7.2 RESULTS FOR CARBON DIOXIDE AND ARGON AS THE WORKING FLUID

A series of tests was conducted with CO₂ and argon, for the same values of speed parameter as the tests using air as the working fluid. Due to the differences in fluid properties, this involved running the tests at different absolute speeds. For reference, the highest speed line ($\Pi_N = 21.03$) corresponds to an absolute speed of approximately 96,000 rpm for air, and approximately 75,000 rpm and 89,000 rpm for CO₂ and argon respectively.

Although it was not possible to conduct the tests at the same Reynolds number for the three test gases used in the experiments, the flow parameter, pressure ratio, and efficiency for all test data were corrected for deviations in Reynolds number using the method described in section 5.4.3.2, for an assumed impeller surface roughness of 5 μm . The implied Reynolds numbers of the manufacturer's map (Fig. 4.2) were used as the reference Reynolds numbers for the corrections, in an effort to separate the effects of the isentropic exponent from the effects of Reynolds number. Although the ratio of the test Reynolds number to the reference Reynolds number was as high as 1.8 in some tests using CO₂, and as high as 1.3 in some tests using argon (Table 7.1), the corrections to the flow parameter and pressure ratio were in general not noticeable, and the correction to the efficiency was in most cases not greater than 1%.

Gas	Speed Parameter	Re_{b2} from Supplier's Map	Maximum Re_{b2} in Experiments	Minimum Re_{b2} in Experiments	Average Re_{b2} in Experiments	Maximum Efficiency Correction (%)
CO ₂	10.06	52660	94201	85570	91228	-1.84
	15.23	79716	141234	130473	135624	-0.72
	28.35	96077	168549	153686	158970	-0.45
	21.03	110094	197416	184969	188959	-0.38
Argon	10.06	52660	68325	63044	65055	-0.83
	15.23	79716	103761	93394	98137	-0.41
	28.35	96077	124873	114814	121246	-0.27
	21.03	110094	137677	132050	134223	-0.17
Air	10.06	52660	57386	49084	53097	0.33
	15.23	79716	84635	60297	75458	0.24
	28.35	96077	98446	62918	85548	0.27
	21.03	110094	111033	58126	82144	0.45

Table 7.1: Deviation of Test Reynolds Numbers Re_{b2} from Reference Reynolds Number Extracted from Fig. 5.2

7.2.1 Effects of γ on the Choking Mass Flow Parameter and Pressure Ratio

The measured pressure ratio as a function of mass flow parameter is presented in Fig. 7.3 for all three test gases, with the experimental results for air from Fig. 7.1 represented by a mean line to reduce clutter. It is clear from the experimental results that changing

the test gas has a significant effect on the pressure ratio, with higher pressure ratios measured for gases with higher values of the isentropic exponent.

Another interesting feature in figure 7.3 is that the speed lines for air and CO₂ cross at high values of the flow parameter, suggesting that gases with lower values of isentropic exponent may tend toward higher choking flow parameter.

Figure 7.3: Measured Pressure Ratio for Air, CO₂ and Argon as a Function of Mass Flow Parameter

The results from the meanline analysis are shown in Fig. 7.4 for all three test gases. It is clear that there is also a significant difference in performance predicted by the meanline analysis, and the changes in pressure ratio predicted by the meanline analysis are similar to the changes seen in the experiments. However, the meanline analysis does not predict that the speed lines for CO₂ and air will cross at high values of the flow parameter. This is believed to be partly due to inaccurate prediction of the losses at high values of the mass flow parameter, as was discussed in section 6.3.

Figure 7.4: Pressure Ratio for Air, CO₂, and Argon Predicted Using a Meanline Analysis

Due to high pressure losses in the circuit, it was not possible to measure experimentally the choking mass flow parameter. However, this effect could be investigated using CFD (Fig. 7.5). The pressure ratio is higher in the CFD analysis than in either the experiments or the meanline analysis since the CFD computations were conducted on the impeller only, and do not include the losses in the diffuser or volute.

Figure 7.5: Pressure Ratio for Air, CO_2 , and Argon, Predicted Using CFD

As mentioned earlier, it was not possible to maintain the same experimental Reynolds numbers, for a given value of speed parameter, for the three different gases. For example, for the $\Pi_N = 21.03$ speed lines, the average Reynolds numbers (Re_{b2}) for air, CO_2 , and argon were respectively about 82,000, 190,000 and 135,000. However, the corresponding CFD and meanline calculations were all performed at a Reynolds number of 132,000. Thus the differences in performance seen in the meanline and CFD calculations are a result of the change in γ alone. The good agreement between the computations and the measurements, in regards to both the trends and the magnitudes of the changes in pressure ratio with working fluid, shows that the differences in behaviour seen in Fig. 7.3 are predominantly due to the differences in the isentropic exponent, rather than differences in Reynolds number.

Also shown on Fig. 7.5 is the predicted performance for helium. Helium has the same isentropic exponent as argon but all other fluid properties are significantly different. For example, the values of the gas constant R , the specific heat at constant pressure c_p and the thermal conductivity k differ by about a factor of 10. The close agreement between the results for helium and argon further supports the conclusion that the observed variations in performance with working fluid are due exclusively to the differences in the isentropic exponent.

The CFD calculations also confirm that the crossing of the speed lines seen in Fig. 7.3 is the result of differences in the choking mass flow rates. As seen from Fig. 7.5, the CFD indicates that the value of the choking mass flow parameter varies inversely with γ , with argon having the lowest and CO_2 the highest choking flow parameters respectively.

The differences in choking flow parameter seen in Figs. 7.3 – 7.5 can be predicted using arguments based on a one-dimensional, isentropic flow analysis, as discussed in the next section. In addition, the change in pressure ratio can also be predicted by a simple analysis, which is described in section 7.2.1.2.

7.2.1.1 Choking Flow Parameter

For a perfect gas, the mass flow rate for one-dimensional isentropic compressible flow through a passage with area A can be expressed as

$$\dot{m} = \rho AV = \rho_0 \left(\frac{\rho}{\rho_0} \right) AM \sqrt{\gamma RT_0} \left(\sqrt{\frac{T}{T_0}} \right)$$

From the isentropic flow relations, and the ideal gas equation of state:

$$\frac{\rho_0}{\rho} = \left(1 + \frac{\gamma-1}{2} M^2 \right)^{\frac{1}{\gamma-1}}$$

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

$$\rho_0 = \frac{P_0}{RT_0}$$

Substituting these expressions into the equation for the mass flow:

$$\dot{m} = \frac{P_0 \sqrt{\gamma}}{\sqrt{RT_0}} AM \left(1 + \frac{\gamma-1}{2} M^2 \right)^{-\frac{1}{2} \left(\frac{\gamma+1}{\gamma-1} \right)} \quad \text{Eq. 7.1}$$

If Eq. 7.1 is used in the definition of the flow parameter (Eq. 2.2), with $M = 1$, and A being the impeller throat area, we obtain the following expression for the choking flow parameter:

$$\Pi_m = \frac{\dot{m}\sqrt{T_{01}}}{P_{01}} \left(\frac{1}{D^2} \sqrt{\frac{R}{\gamma}} \right)$$

$$\Pi_m^* = \frac{A}{D^2} \left(\frac{2}{\gamma + 1} \right)^{\frac{1+\gamma}{2(\gamma-1)}} \quad \text{Eq. 7.2}$$

It is clear from Eq. 7.2 that changing the isentropic exponent will result in a change in the choking flow parameter. For two machines with the same geometry, but operating with different working fluids A and B ,

$$\frac{\Pi_{mB}^*}{\Pi_{mA}^*} = \frac{\left(\frac{2}{\gamma_B + 1} \right)^{\frac{1+\gamma_B}{2(\gamma_B-1)}}}{\left(\frac{2}{\gamma_A + 1} \right)^{\frac{1+\gamma_A}{2(\gamma_A-1)}}} \quad \text{Eq. 7.3}$$

Equation 7.3 predicts higher choking flow parameter for decreasing values of the isentropic exponent, as shown in Fig. 7.6, which is the same trend that was seen in the experimental results, and in the CFD and meanline computations. In particular, Eq. 7.3 predicts that the ratio of the choking flow parameter for CO₂ to the choking flow parameter for air ($\Pi_{mCO_2}^* / \Pi_{mAIR}^*$) will be 1.012, and that for argon and air, the ratio ($\Pi_{mARGON}^* / \Pi_{mAIR}^*$) is 0.972. By comparison, the ratio of the choking flow parameters for CO₂ and argon to the choking flow parameter for air as predicted by the meanline analysis are 1.019 and 0.971 respectively. The predicted choking flow parameter for CO₂ and argon, based on the CFD predictions for air, are shown in Fig. 7.7. For both gases, the choking flow parameter predicted using Eq. 7.3 is in very good agreement with the impeller choking flow parameter predicted using CFD, even though the flow in the impeller throat is far from being one-dimensional.

Figure 7.6: Variation of Choking Flow Parameter Π^*_m with γ , Based on One-Dimensional Isentropic Flow Analysis (Eq. 7.3)

Figure 7.7: Choking Flow Parameters for CO₂ and Argon Predicted by One-Dimensional Isentropic Flow Analysis, and by CFD

7.2.1.2 Pressure Ratio

The effect of the isentropic exponent on the pressure ratio can also be predicted using a one-dimensional analysis. If losses are neglected, matching of the speed and flow parameters for two gases *A* and *B* implies similarity of the inlet flow coefficient ($\phi_I = C_{aI}/U_I$) and work coefficient ($\psi = \Delta h_0/U_2^2$), as shown below.

Flow Coefficient

By applying continuity at the rotor inlet, the mass flow parameter (Eq. 2.2) can be expressed in the following alternative form:

$$\Pi_m = \frac{\dot{m} \sqrt{T_{01}}}{P_{01}} \left(\sqrt{\frac{R}{\gamma}} \frac{1}{D^2} \right) = \frac{\rho_1 A_1 C_{a1} \sqrt{T_{01}}}{P_{01}} \left(\sqrt{\frac{R}{\gamma}} \frac{1}{D^2} \right)$$

If we assume that the inlet density ρ_1 is approximately equal to the inlet stagnation density ρ_{01} , using the ideal gas equation of state $P/\rho = RT$, this expression can be rewritten as follows:

$$\Pi_m = \frac{C_{a1} \sqrt{RT_{01}}}{RT_{01}} \frac{A_1}{\sqrt{\gamma} D^2} = \frac{C_{a1}}{\sqrt{\gamma} RT_{01}} \frac{A_1}{D^2}$$

which is consistent with our definition of the mass flow parameter as the axial Mach number of the inlet flow to the compressor. Rearranging to solve for the inlet axial velocity C_{a1} :

$$C_{a1} = \Pi_m \frac{\sqrt{\gamma RT_{01}} D^2}{A_1} \quad \text{Eq. 7.4}$$

Similarly, the inlet blade speed U_1 , or the tip blade speed U_2 can be expressed in terms of the speed parameter (Eq. 2.1):

$$\Pi_N = \frac{N}{\sqrt{T_{01}}} \frac{D}{\sqrt{\gamma R}} \quad \text{or} \quad ND = \Pi_N \sqrt{\gamma RT_{01}}$$

$$U_2 = \omega r_2 = \left(\frac{2\pi}{60} N \right) \left(\frac{D}{2} \right) = \frac{\pi N D}{60} = \Pi_N \frac{\pi}{60} \sqrt{\gamma R T_{01}} \quad \text{Eq. 7.5}$$

$$U_1 = U_2 \frac{D_1}{D} = \Pi_N \frac{\pi}{60} \sqrt{\gamma R T_{01}} \frac{D_1}{D} \quad \text{Eq. 7.6}$$

Combining equations 7.4 and 7.6 results in the following expression for the inlet flow coefficient:

$$\phi_1 = \frac{C_{a1}}{U_1} = \frac{\Pi_m}{\Pi_N} \left(\frac{\sqrt{\gamma R T_{01}} D^2}{A_1} \right) \left/ \left(\frac{\pi}{60} \sqrt{\gamma R T_{01}} \frac{D_1}{D} \right) \right. = \frac{\Pi_m}{\Pi_N} \left(\frac{60 D^3}{\pi D_1 A_1} \right) \quad \text{Eq. 7.7}$$

Therefore, for the same geometry, if the speed parameter Π_N , and flow parameter Π_m , are matched for two gases, Eq. 7.7 shows that the inlet velocity triangles will be similar, although the magnitudes of the velocities may be different.

Work Coefficient

For a perfect gas, the impeller work input, or Euler work (Δh_0) is related to the stagnation temperature change by $\Delta h_0 = c_p(T_{02} - T_{01})$, where c_p is the specific heat at constant pressure. By substituting this expression for the specific work input and Eq. 7.5 for the blade tip speed, we obtain the following expression for the work coefficient, neglecting losses:

$$\psi = \frac{\Delta h_0}{U_2^2} = \frac{c_p (T_{02} - T_{01})}{\left(\Pi_N \frac{\pi}{60} \right)^2 \gamma R T_{01}} = \frac{c_p \left(\left(\frac{P_{02}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right)}{\left(\Pi_N \frac{\pi}{60} \right)^2 \gamma R}$$

Noting that $\gamma = c_p/c_v$, and $R = c_p - c_v$,

$$\psi = \frac{\left(\left(\frac{P_{02}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right)}{(\gamma - 1) \left(\Pi_N \frac{\pi}{60} \right)^2} \quad \text{Eq. 7.8}$$

Since the impeller slip is mainly a function of the rotor geometry, it is reasonable to assume that the slip factor for the same impeller operating using two different working fluids will be the same. In addition, if the inlet velocity triangles are similar, similarity of the slip factor implies that the discharge velocity triangles will also be similar, and thus the impeller work coefficient for the two gases will be the same. This is the same assumption that is made in ASME PTC 10 (1997) for predicting the speed at which a compressor should be operated at to obtain the same pressure ratio using a substitute gas.

The above assumption was also found to be essentially correct in a simplified analysis of the impeller, based on the Euler equation. The differences between the work coefficients predicted by the simplified analysis for CO₂ and argon, compared to the work coefficient for air, is shown in Fig. 7.8. Figure 7.8 shows that under most circumstances, the difference between the work coefficients for CO₂ and argon when compared to the values for air is less than 0.5%.

Figure 7.8: Deviation of Impeller Work Coefficient for CO₂, and Argon Predicted by Euler Analysis

Pressure Ratio

Since the impeller work coefficient ψ for two gases operating at the same mass flow parameter Π_m and speed parameter Π_N are approximately equal, Eq. 7.8 implies that the pressure ratio for two perfect gases A and B will be related by the following expression:

$$\frac{\left(\frac{P_{02A}}{P_{01A}}\right)^{\frac{\gamma_A-1}{\gamma_A}} - 1}{(\gamma_A - 1)} = \frac{\left(\frac{P_{02B}}{P_{01B}}\right)^{\frac{\gamma_B-1}{\gamma_B}} - 1}{(\gamma_B - 1)}$$

Therefore, to predict the impeller pressure ratio for a substitute gas B based on experimental results for gas A , the following equation can be used:

$$\frac{P_{02B}}{P_{01B}} = \left(1 + \left(\frac{\gamma_B - 1}{\gamma_A - 1} \right) \left(\left(\frac{P_{02A}}{P_{01A}} \right)^{\gamma_A - 1} - 1 \right) \right)^{\frac{\gamma_B}{\gamma_B - 1}} \quad \text{Eq. 7.9}$$

Application of Eq. 7.9, for a given value of the mass flow parameter, does not take into account the difference in choking flow parameter for different gases. Better results are obtained if the flow parameters are normalized to the observed values at choking. Equation 7.9 is then used to predict the pressure ratio of gas B from the pressure ratio of gas A at the same value of Π_m/Π_m^* .

$$\frac{\Pi_{mB}}{\Pi_{mB}^*} = \frac{\Pi_{mA}}{\Pi_{mA}^*} \quad \text{Eq. 7.10}$$

It should be noted that re-scaling the mass flow parameter implies that there is no longer similarity of the inlet and discharge velocity triangles. However, in practice, the difference in choking flow parameter when using substitute gases is small enough that for the purposes of this analysis, the velocity triangles can still be treated as though they are similar, even when there is a significant difference in the value of γ between the two gases.

Equations. 7.9 and 7.10 were used to project the impeller characteristic for argon and CO_2 from the computed characteristic for air, with the results shown in Fig. 7.9. Equation 7.9 is seen to predict a greater change in pressure ratio than was seen in the CFD. This discrepancy is due primarily to changes in efficiency when changing the working fluid, which is discussed further in the next section. If the efficiency is retained in the derivation of Eq. 7.9, then the correction formula takes the following form:

$$\frac{P_{02B}}{P_{01B}} = \left(1 + \frac{\eta_B}{\eta_A} \left(\frac{\gamma_B - 1}{\gamma_A - 1} \right) \left(\left(\frac{P_{02A}}{P_{01A}} \right)^{\frac{\gamma_A - 1}{\gamma_A}} - 1 \right) \right)^{\frac{\gamma_B}{\gamma_B - 1}} \quad \text{Eq. 7.11}$$

Figure 7.9: CFD Prediction of Rotor Pressure Ratio as Function of γ , and Pressure Ratio Predicted by Similarity Considerations (Eq. 7.9)

7.2.2 Effect of the Isentropic Exponent on Efficiency

The measured efficiency as a function of the mass flow parameter for CO₂ and argon are shown in Figs. 7.10 & 7.11 respectively, with the measured efficiency for air from Fig. 7.2 represented by a mean line to reduce clutter.

Figure 7.10: Measured Isentropic Efficiency for CO₂ and Air as Function of Mass Flow Parameter

Clearly, in addition to having an effect on the pressure ratio and choking mass flow parameter, there is also a significant effect of the isentropic exponent on the efficiency, with the highest efficiency being measured for CO₂, followed by air, and finally the lowest efficiency being measured for argon. To make this clearer, the measured peak values of efficiency for each value of the speed parameter are plotted in Fig. 7.12.

Figure 7.11: Measured Isentropic Efficiency for Argon and Air as Function of Mass Flow Parameter

It is also noticeable in Figs. 7.10 and 7.11 that the measured peak efficiencies for argon and CO₂ both occur at higher values of the mass flow parameter than for air. The reason for this is not yet understood, but it is believed to be due to changes in the loss characteristics of the diffuser and volute when working fluids other than air are used.

Figure 7.12: Measured Peak Efficiency for Air, CO₂, and Argon as Function of Speed Parameter

Changes in isentropic efficiency for CO₂ and argon were also seen in both the CFD and meanline computations. Figures 7.13 and 7.14 show the corresponding efficiencies as computed for $\Pi_N = 15.34$ and $\Pi_N = 21.03$ using CFD and the meanline analysis respectively.

Figure 7.13: Isentropic Efficiency for Air, CO_2 , and Argon Predicted Using CFD

For the CFD computations, the trend in efficiency with specific heat ratio is seen to be consistent with that in the measurements. As with the pressure ratio, the isentropic efficiency predicted by the CFD analysis is higher than was seen experimentally and in the meanline analysis, since the diffuser and volute losses were not computed, and the CFD computations do not include the effect of tip clearance losses.

Fig. 7.13 also includes the computed results for helium. As with the pressure ratio, the results of the efficiency computations are almost identical for helium and argon. This

again supports the conclusion that γ is the dominant fluid property that determines the variation in turbomachinery performance with changing working fluid.

Figure 7.14: Isentropic Efficiency for Air, CO₂, and Argon Predicted Using a Meanline Analysis

The trend in efficiency in the meanline computations (Fig. 7.14) is also consistent with that seen in the experiments, although the magnitude of the change in efficiency predicted by the meanline analysis is approximately half of that seen experimentally and in the CFD computations, particularly in the case of argon. In addition, the meanline

analysis does not predict that the peak stage efficiency for argon and CO₂ occurs at higher values of the flow parameter, as was seen in the experiments. Since the cause of the shift in peak efficiency toward higher mass flow for CO₂ and argon is not yet known, it is difficult to tell why this feature has not been captured by the meanline analysis.

An attempt was made to determine the cause of the apparent discrepancy in the prediction of efficiency by the meanline analysis, by comparing results from the meanline analysis for the rotor alone to the results of the CFD. Unfortunately however, reliable results could not be obtained using CFD at lower values of the flow parameter. Since the meanline analysis does not provide an accurate assessment of the impeller losses at high values of flow parameter (as was discussed in section 6.3), it was not possible to draw any useful comparisons between the two methods.

The changes in efficiency when using test gases with different values of the isentropic exponent can be predicted using a one-dimensional, isentropic analysis of the flow. For an isentropic process, the compression ratio is related to the pressure ratio as follows:

$$\frac{\rho_{02}}{\rho_{01}} = \left(\frac{P_{02}}{P_{01}} \right)^{\frac{1}{\gamma}}$$

It is clear that for two geometrically similar compressors operating at the same pressure ratio, but with different working fluids, the fluid with the higher value of the isentropic exponent will have a lower compression ratio. In order to satisfy continuity, the ratio of the discharge radial velocity C_{r2} to the inlet axial velocity C_{a1} must therefore be higher for the fluid with the higher value of γ . Since the inlet and discharge velocity triangles are similar for the two working fluids for the same values of the speed and flow parameter, the ratio of the discharge relative velocity W_2 to the inlet relative velocity W_1

will also be higher for the fluid with the higher value of specific heat ratio. Since many of the contributions to the losses vary as the square of the velocity, the higher discharge velocity should result in higher losses in the impeller, and all downstream components. This is consistent with the changes in efficiency with working fluid seen in the experiments and the computations.

As was discussed previously however, if the compressor is operated at the same speed parameter and flow parameter using a substitute working fluid, the pressure ratio will be different. In addition, it is difficult to predict directly the effect of the increase in pressure loss on the efficiency. If we make the simplifying assumption that the efficiency varies linearly with the pressure losses, and thus the average dynamic head in the impeller passage (normalized by the local stagnation pressure), then the change in efficiency when using an alternative working fluid can be approximated as follows:

$$\frac{(1-\eta)_A}{(1-\eta)_B} \cong \frac{\Delta P_{LOSS A} \bar{P}_{0A}}{\Delta P_{LOSS B} \bar{P}_{0B}} \cong \frac{\frac{1}{2} \bar{\rho}_A \bar{W}_A^2 \bar{P}_{0A}}{\frac{1}{2} \bar{\rho}_B \bar{W}_B^2 \bar{P}_{0B}}$$

The problem then lies in determining the ratio of the normalized average dynamic head of the two working fluids in the impeller passage. If we assume that the average dynamic head is the arithmetic average of the inlet and discharge values for each working fluid:

$$\frac{(1-\eta)_A}{(1-\eta)_B} = \frac{\left(\frac{\rho_{1A} W_{1A}^2}{P_{01A}} \right) + \left(\frac{\rho_{A2} W_{2A}^2}{P_{02A}} \right)}{\left(\frac{\rho_{1B} W_{1B}^2}{P_{01B}} \right) + \left(\frac{\rho_{2B} W_{2B}^2}{P_{02B}} \right)}$$

As with the pressure ratio correction, we can assume that the inlet and discharge densities for the two gases are approximately equal to their stagnation values. Noting

again that $P/\rho = RT$, and dividing each of the terms in brackets by the values at the inlet for gas B :

$$\frac{(1-\eta)_A}{(1-\eta)_B} = \frac{\left(\frac{W_{1A}}{W_{1B}}\right)^2 \frac{R_B T_{01B}}{R_A T_{01A}} + \left(\frac{W_{2A}}{W_{1B}}\right)^2 \frac{R_B T_{01B}}{R_A T_{02A}}}{1 + \left(\frac{W_{2B}}{W_{1B}}\right)^2 \frac{R_B T_{01B}}{R_B T_{02B}}} = \frac{L + M}{1 + K} \quad \text{Eq. 7.12}$$

To simplify the analysis, each term in the above equation is considered separately. Since the velocity triangles are similar for the two gases if the speed parameter and flow parameters are matched:

$$\frac{W_{XA}}{\sqrt{\gamma_A R_A T_{0XA}}} = \frac{W_{XB}}{\sqrt{\gamma_B R_B T_{0XB}}}$$

The subscript X is used to indicate that this expression is true at both the inlet and discharge conditions. The three terms in Eq. 7.12 can then be expressed as follows:

$$L = \left(\frac{\gamma_A R_A T_{01A}}{\gamma_B R_B T_{01B}} \right) \left(\frac{R_B T_{01B}}{R_A T_{01A}} \right) = \frac{\gamma_A}{\gamma_B}$$

$$M = \left(\frac{W_{2A}}{W_{2B}} \right)^2 \left(\frac{W_{2B}}{W_{1B}} \right)^2 \frac{R_B T_{01B}}{R_A T_{02A}} = \left(\frac{W_{2B}}{W_{1B}} \right)^2 \frac{\gamma_A}{\gamma_B} \frac{T_{01B}}{T_{02B}}$$

$$K = \left(\frac{W_{2B}}{W_{1B}} \right)^2 \frac{T_{01B}}{T_{02B}}$$

Applying conservation of mass across the rotor,

$$\rho_1 A_1 W_1 \cong \rho_2 A_2 W_2$$

where A_1 and A_2 are the impeller inlet and discharge areas normal to the flow direction in the blade passage respectively. Note that this expression is only approximately true since incidence and slip have been ignored, and also since the flow at the impeller inlet is not one-dimensional. For most impellers, the inlet area A_1 is slightly larger than the discharge area A_2 , but in this simplified analysis they can be considered to be approximately equal. As an example, for the T04-E impeller used in the experiments, the inlet area is approximately 15% larger than the discharge area. The continuity equation can then be approximated by:

$$\frac{W_{2B}}{W_{1B}} \cong \frac{\rho_{2B}}{\rho_{1B}} \cong \left(\frac{P_{02B}}{P_{01B}} \right)^{\frac{1}{\gamma_B}} \quad \text{Eq. 7.13}$$

Using this expression in the equations for M and K ,

$$M = \frac{\gamma_A}{\gamma_B} \left(\frac{P_{02B}}{P_{01B}} \right)^{\frac{2}{\gamma_B}} \frac{T_{01B}}{T_{02B}}$$

$$K = \left(\frac{P_{02B}}{P_{01B}} \right)^{\frac{2}{\gamma_B}} \frac{T_{01B}}{T_{02B}}$$

Replacing these terms in Eq. 7.12 and simplifying results in the following approximate expression for relating the impeller efficiency for two different working fluids:

$$\frac{(1-\eta)_A}{(1-\eta)_B} \cong \frac{\gamma_A}{\gamma_B} \quad \text{Eq. 7.14}$$

For use with Eq. 7.14, the mass flow parameter would be re-scaled in the same way as it was when performing the efficiency correction, using Eq. 7.10, in order to maintain consistency with the changes in choking flow parameter.

It is worth noting that Eq. 7.14 will always predict lower efficiency for higher values of the isentropic exponent. In the experiments, the efficiency for argon was seen to increase at the lowest speed. This is believed to be an experimental problem. The experimental uncertainty for all measured data is increased at low speed, since the pressure and temperature differences from inlet to discharge are the lowest for this speed line. In addition, in spite of the insulation, heat transfer out of the test circuit from the hot duct between the impeller and the location of the discharge total temperature measurement will result in over-prediction of the efficiency for argon of up to 2% for the $\Pi_N = 10.06$ speed line. For the remaining speed lines however, the effect of heat loss through the insulation is negligible. Estimates of the effect of the heat losses in the discharge pipe section on the efficiency for each speed line are given in Appendix F.

The results from Eq. 7.14 are compared to the CFD results in Fig. 7.15. Equation 7.14 was found to predict a greater change in efficiency than was seen in the CFD analysis of the rotor. This over-prediction is partly due to inaccuracies in the simplifying assumptions in the analysis. In particular, there are a number of loss mechanisms present in the stage that do not depend on the impeller outlet conditions. For example, the inlet shock losses and losses due to incidence at the impeller leading edge are dependent only on the flow conditions at the stage inlet. Therefore, Eq. 7.14 was modified in a semi-empirical way to arrive at a final correction formula for the effects of fluid properties on efficiency. The following revised correction formula is proposed:

$$\frac{(1-\eta)_A}{(1-\eta)_B} = \left(\frac{\gamma_A}{\gamma_B} \right)^n \quad \text{Eq. 7.15}$$

Figure 7.15: CFD Prediction of Rotor Efficiency as Function of γ , and Efficiency Predicted by Similarity Considerations (Eq. 7.14)

Note that Eq. 7.15 has the same form as many Reynolds number corrections for efficiency. The value of the exponent n must be determined by comparison to experimental results. However, the value of n should be less than 1, since not all of the loss mechanisms are necessarily affected by the change in fluid properties. A value of $n = 0.8$ is proposed for the efficiency correction, as will be discussed in the following section.

In the meanline analysis, the correlations for all of the loss mechanisms (section 6.2.1.3) are calculated based on inlet and discharge conditions, with the exception of the incidence loss. The correlations should therefore include the effects of the changes in velocity and compression ratio for different working fluids, and there should not be any need for modifications to the meanline analysis program in order to capture the effects of the change in isentropic exponent on the performance of the impeller, provided the loss correlations themselves are valid for gases other than air. As shown in Figure 7.14, the meanline analysis appears to underestimate the change in efficiency due to a change in the isentropic exponent, and some adjustments to the loss correlations presented in section 6.2.1.3 may be necessary. However, the meanline analysis does at least capture the major effects of the isentropic exponent on the performance of the compressor, and it appears as though a meanline analysis that was developed for use with air can be used to predict the performance of compressors operating using working fluids other than air with reasonable accuracy, provided the loss correlations used are based on inlet and discharge conditions.

7.3 PREDICTION OF COMPRESSOR PERFORMANCE FOR ALTERNATIVE WORKING FLUIDS

7.3.1 Prediction of Impeller Performance

With a better understanding of the effects of the isentropic exponent on the choking flow parameter, pressure ratio, and efficiency, it is now possible to provide a more refined method for predicting the impeller performance for a substitute working fluid. The following set of equations is proposed to predict the change in flow parameter, efficiency, and pressure ratio for a gas *B*, based on test data for gas *A*.

$$\frac{\Pi_{mB}}{\Pi_{mA}} = \frac{\left(\frac{2}{\gamma_B + 1} \right)^{\frac{1+\gamma_B}{2(\gamma_B-1)}}}{\left(\frac{2}{\gamma_A + 1} \right)^{\frac{1+\gamma_A}{2(\gamma_A-1)}}} \quad \text{Eq. 7.16}$$

$$\frac{\eta_B}{\eta_A} = \frac{1}{\eta_A} + \left(\frac{\gamma_B}{\gamma_A} \right)^{0.8} \left(1 - \frac{1}{\eta_A} \right) \quad \text{Eq. 7.17}$$

$$\frac{P_{02B}}{P_{01B}} = \left(1 + \frac{\eta_B}{\eta_A} \left(\frac{\gamma_B - 1}{\gamma_A - 1} \right) \left(\left(\frac{P_{02A}}{P_{01A}} \right)^{\frac{\gamma_A-1}{\gamma_A}} - 1 \right) \right)^{\frac{\gamma_B}{\gamma_B-1}} \quad \text{Eq. 7.18}$$

A value of $n = 0.8$ is proposed for the efficiency correction for the compressor used in this study, since when using this value, good correlation is obtained with the CFD data.

The pressure ratio and efficiency for argon and CO₂, predicted using Eqs. 7.16–7.18, for a value of $n = 0.8$, are compared to the performance predicted using CFD in Figs. 7.16 & 7.17. The above correction procedure appears to give reasonable predictions of the rotor performance for both argon and CO₂, when compared to the results from CFD.

It should be noted that for multi-stage axial compressors, since the inlet and discharge velocity triangles are similar, matching of the first stage flow and work coefficients should result in these parameters being matched in subsequent stages. However, for multi-stage centrifugal compressors, the flow in diffusers and return channels will be significantly different when using substitute working fluids, and it does not seem reasonable to expect that the inlet flow coefficient and thus the work coefficient for subsequent stages will be matched. Thus, corrections to the pressure ratio and efficiency due to changing fluid properties should be applied to each stage separately for centrifugal compressors if possible.

Figure 7.16: CFD Prediction of Rotor Efficiency for Argon and CO_2 and Predicted Impeller Efficiency Using Proposed Efficiency Correction (Eqs. 7.16 & 7.17), and CFD Results for Air

Figure 7.17: CFD Prediction of Rotor Pressure Ratio for Argon and CO_2 , and Predicted Impeller Efficiency Using Proposed Pressure Ratio Correction (Eqs. 7.16 & 7.18), and CFD Results for Air

Fig. 7.18 also shows the results of applying Eq. 7.17 to the measured peak stage efficiencies. Although the scatter in the experimental data makes the assessment less clear, Eq. 7.17 appears to provide reasonable corrections to these data as well.

Figure 7.18: Predicted Stage Peak Efficiencies Using Proposed Efficiency Correction (Eq. 7.16), and Experimental Results for Air

It is clear from Figs. 7.16 & 7.17 that the correction methods presented here more closely match the predicted CFD performance for CO₂ than they do for argon, particularly at lower rotational speed. This is not altogether unexpected, since the difference in isentropic exponent between air and argon (19%) is greater than the difference between air and CO₂ (8%). Due to the simplifying assumptions made in the analysis, it is expected that the prediction will be more accurate when the two gases have similar values of the isentropic exponent. In addition, because of the assumption that the relative velocity ratio (W_2/W_1) for the impeller can be predicted from the pressure ratio alone (Eq. 7.13), it can be expected that the efficiency correction will not be as accurate when the compressor is operated far from the design incidence. However, in spite of the

simplifications made in the analysis, the correction method outlined in Eqs. 7.16-7.18 still provides reasonably good predictions of the changes in performance due to a change in working fluid.

7.3.2 Prediction of Stage Performance

An attempt was made to predict the performance map of the compressor used in the experiments using the measured results for air and Eqs. 7.15–7.17, with the results shown in Figs. 7.19-7.21. In the middle of the mass flow range, Eqs. 7.15–7.17 appear to provide a good approximation of the actual performance. However, Eq. 7.16 tends to underestimate the change in stage efficiency at high flow rates, resulting in lower predicted pressure ratios at high mass flow parameters than were seen experimentally.

Figure 7.19: Stage Efficiency Predicted Using Eqs. 7.16 & 7.17 and Experimental Data for Air - Comparison with Measured Values for Argon

Figure 7.20: Stage Efficiency Predicted Using Eqs. 7.16 & 7.17 and Experimental Data for Air – Comparison to Measured Values for CO₂

Figure 7.21: Stage Pressure Ratio Predicted Using Eqs. 7.16 & 7.18 and Experimental Data for Air – Comparison to Measured Values For Argon and CO₂

As mentioned earlier, the measured peak efficiency for a given speed line occurred at a higher flow parameter for both CO₂ and argon than for air, and this phenomenon is not predicted by Eq. 7.16. If the actual ratio of efficiency is used instead in Eq. 7.17, a much better prediction of the pressure ratio is obtained (Fig. 7.22). The ability of Eq. 7.17 to predict the pressure ratio using an alternative working fluid, given accurate values of the efficiency for the two gases, provides evidence that the pressure ratio correction presented here is based on sound physical arguments. However, it is clear that further investigation into the effects of changing the working fluid on compressor efficiency is required in order to provide better estimates of the stage performance for alternative working fluids.

Figure 7.22: Revised Stage Pressure Ratio Predicted Using Eqs. 7.16 & 7.18, Experimental Data for Air, and Measured Efficiency for Argon and CO₂ – Comparison to Measured Values for Argon and CO₂

7.4 IMPLICATIONS FOR TURBINE AERODYNAMICS

If the assumptions presented in the previous discussion regarding the causes of the difference in performance of the compressor are correct, then it should be possible to predict at least qualitatively the change in performance of turbines when substitute gases are used. For instance, it is expected that the choking mass flow parameter for the turbine would increase for decreasing values of the isentropic exponent, for the same reasons this is true for the compressor. An increase in the isentropic exponent should also result in a decrease in expansion (and therefore in the discharge velocity) in the turbine, for the same pressure ratio. This should in turn lead to an increase in turbine efficiency for increased values of the isentropic exponent, as opposed to the decrease in compressor efficiency that was seen. Finally, by equating the work coefficient and speed parameter for two identical turbines operating with different working fluids, it is predicted that the fluid with the lower value of the isentropic exponent should require a lower pressure ratio to drive the turbine at a given value of the speed parameter. Unlike the compressor characteristic, where the speed lines for different working fluids crossed at high mass flow parameter, the above observations imply that the turbine load lines for different working fluids operating under similar loading conditions will not cross at high values of the mass flow parameter, as shown schematically in Fig. 7.23.

Figure 7.23: Schematic Representation of Predicted Turbine Characteristics for Different Working Fluids

8.0 CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

The effect of the isentropic exponent γ on the aerodynamic performance of turbomachinery has been investigated both experimentally and computationally. The results of the investigation have confirmed that γ is a significant criterion of similarity in the performance of turbomachinery. For compressors in particular, the pressure ratio, the choking mass flow rate and the isentropic efficiency were all found to be significantly affected by changes in γ . As expected, the changes in aerodynamic performance when substitute working fluids are used were found to be dependent only on the isentropic exponent.

Based on simple arguments, methods have been developed for predicting the changes in performance of compressors when substitute gases are used. The initial correction formula for efficiency appeared to over-estimate the changes with γ . A simple modification, involving a single empirical factor, improved the results noticeably. The value of the empirical factor was determined from the present limited data and it would be desirable to confirm it using a larger body of data. The correction formulas for the pressure ratio and choking mass flow rate produced very good results without apparent need for further empirical adjustments. When measured values of the efficiency were used to predict the stage pressure ratio for an alternative working fluid, the accuracy of the pressure ratio prediction was noticeably improved. This illustrates how accurate prediction of the efficiency is essential to the accurate prediction of pressure ratio for a substitute gas. Although the efficiency correction presented provides reasonable predictions of the changes in impeller efficiency, further work is required to examine the effects of fluid properties on the performance of the diffuser and volute. It is believed that changes in performance in these components are responsible for the increased flow parameter for peak efficiency when using both argon and CO_2 , and that further work in

this area will allow designers to obtain more accurate predictions of the change in stage efficiency for alternative working fluids.

In semi-closed O_2/CO_2 gas turbine engines, the turbines will also see values of γ which differ substantially from those experienced by the turbines in conventional open-cycle engines, or in the typical rotating rigs used for prototype turbine testing and development. In addition, for gas turbine engines operating using air as the working fluid, the isentropic exponent of the exhaust gases is approximately 1.33. This represents a significant change in the isentropic exponent, and manufacturers of gas turbine engines who make use of data collected from indirectly heated cold flow turbine test rigs should be made aware of the differences in performance that can be expected due to the differences in the fluid properties. Thus, there is a need to conduct a similar investigation to the present one for turbines.

REFERENCES

- Aungier, R.H. (1995), "Mean Streamline Aerodynamic Performance Analysis of Centrifugal Compressors", *Journal of Turbomachinery*, **117**, pp. 360-366.
- Aungier R.H. (2000), Centrifugal Compressors: A Strategy for Aerodynamic Design and Analysis, ASME Press, New York, 320 p.
- Beitler, S.R., (ed.), (1959), Fluid Meters: Their Theory and Application, 5th ed., Report of ASME Research Committee on Fluid Meters, ASME Press, New York, 203 p.
- Buckingham, E. (1914), "On physically similar systems; illustrations of the use of dimensional equations", *Phys. Rev.*, **4**, pp. 345-376. (As referenced by White, 1994)
- Carter, A.D., Moss, C.E., Green, G.R., Annear, G.G. (1960), "The Effects of Reynolds Number on the Performance of a Single Stage Compressor", Aeronautical Research Council Reports and Memorandum, Memorandum 3184, May 1957, U.K.
- CFX BladeGen+ v. 4.0.018 User's Manual, (2000), AEA Technology. (Supplied with Software)
- Colebrook, C.F. (1939), "Turbulent Flow in Pipes, With Particular Reference to the Transition Region Between the Smooth and Rough Pipe Laws", *J. Inst. Civ. Eng.*, **11**, pp. 133-156. (As referenced by Haaland, 1983)
- Coppage, J.E., Dallenbach, F., Eichenberger, H.P., Hlavaka, G.E., Knoernschild, E.M., and Van Le, N. (1956), "Study of Supersonic Radial Compressors for Refrigeration and Pressurization Systems", WADC Tech. Rep. 55-257.

- Daily, J.W., Nece, R.E. (1960), "Roughness Effects on Frictional Resistance of Enclosed Rotating Disks", *J. of Basic Engineering*, **82**, pp. 553-562.
- De Ruyck, J. (1992), "Efficient CO₂ Capture Through a Combined Steam and CO₂ Gas Turbine Cycle", *Proc. 1st Int. Conf. on CO₂ Removal*, Pergamon Press, Toronto.
- Ferguson, T.B. (1963), The Centrifugal Compressor Stage, Butterworths, Toronto, 151 p.
- Flowmeter Computation Handbook, ASME Research Committee on Fluid Meters, ASME Press, New York, 169 p.
- Jeffries, M. (1994), The Commissioning and Enhancement of a Blow Down Wind Tunnel, M.Sc. Thesis, Carleton University, Ottawa, Canada.
- Jeffries, M. (2001), Initial Investigations of Transonic Turbine Aerodynamics Using the Carleton University High-Speed Wind Tunnel, Ph.D. Thesis, Carleton University, Ottawa, Canada.
- Haaland, S.E. (1983), "Simple and Explicit Formulas for the Friction Factor in Turbulent Pipe Flow", *J. of Fluids Eng.*, **105**, pp. 89-90.
- Harman, R.T.C. (1981), Gas Turbine Engineering: Applications, Cycles, and Characteristics, MacMillan Press, New York.
- Holman, J.P. (1990), Heat Transfer, 7th ed., McGraw-Hill, Inc., Toronto, 714 p.
- Hunter, I.H. (1994), Design of Turbomachinery for Closed and Semi-Closed Gas Turbine Cycles, M.Sc. Thesis, Cranfield University, Cranfield, UK.
- Irvine, T.F., Liley, P.E. (1994), Steam and Gas Tables with Computer Equations, Academic Press, Orlando, 185 p.

- Jackson, J.B., Neto, A.C., Whellens, M.W., and Audus, H. (2000), "Gas Turbine Performance Using Carbon Dioxide as Working Fluid in Closed Cycle Operation", *Proceedings of ASME Turbo Expo 2000*, 2000-GT-153, Munich, Germany.
- Jansen, W. (1970), "A Method for Calculating the Flow in a Centrifugal Impeller when Entropy Gradients are Present", *I. Mech. E. Internal Aerodynamics (Turbomachinery)*, **92**, pp. 132-140.
- Johnston, J.P., Dean, R.C. (1966), "Losses in Vaneless Diffusers of Centrifugal Compressors and Pumps", *ASME J. Engineering for Power*, **88**, pp. 49-62.
- Kline, S.J. (1965), Similitude and Approximation Theory, McGraw-Hill Inc., New York, 229 p.
- Lee, J.F. (1954), Theory and Design of Steam and Gas Turbine Cycles, McGraw-Hill Book Company, Inc., Toronto.
- Lieblein, S. (1959), "Loss and Stall Analysis of Compressor Cascades", *J. of Basic Engineering*, **81**,
- MacDonald, P. (2001), Private Communication.
- Mathieu, P., Dechamps, P., and Distelmans, M. (1994), "Concepts and Applications of CO₂ Gas Turbines", *Proceedings of Power-Gen Europe 1994*, Cologne Germany.
- Mathieu, P., and De Ruyck, J. (1993), "The CO₂ Gas Turbine Option for Recovery of CO₂ from CC and IGCC Plants", *ASME Cogen-Turbo Conference Proceedings*, Bournemouth U.K.

- Moffat, R.J., (1985), "Using Uncertainty Analysis in the Planning of an Experiment", *J. Fluids Engineering*, **107**, pp. 173-182.
- Navaratnam, M., (1994), The Investigation of an Aero Derivative Gas Turbine Using Alternative Working Fluids in Closed/Semi Closed Cycles, M.Sc. Thesis, Cranfield University, Cranfield, UK.
- Oh, H.W., Yoon, E.S., Chung, M.K. (1997), "An Optimum Set of Loss Models for Performance Prediction of Centrifugal Compressors", *Proc. I. Mech. Eng.*, **211**, Part A, pp. 331-338.
- Performance Test Code on Compressors and Exhausters, ASME PTC 10-1997.
- Reneau, L.R., Johnston, J.P., Kline, S.J. (1967), "Performance and Design of Straight, Two-Dimensional Diffusers", *J. of Basic Engineering*, **89**, pp. 141-150.
- Saravanamuttoo, H.I.H., Cohen, H., Rogers, G.F.C., 2001, Gas Turbine Theory, 5th ed., Prentice-Hall, Toronto, 414 p.
- Schlichting, H. (1979), Boundary Layer Theory, 7th ed., McGraw-Hill Inc., Montreal, 817 p.
- Shepherd, D.G. (1963), Principles of Turbomachinery, The MacMillan Company, Toronto, 463 p.
- Sjolander, S.A. (2001), Private Communication.
- Stein, H. (1996), "Electric Power Generation Market Forecast, 1996-2005", *Turbomachinery Handbook*, **37** (6), pp. 16-20.

- Strub, R.A., Bonciani, L., Borer, C.J., Casey, M.V., Cole, S.L., Cook, B.B., Kotzur, J., Simon, H., and Strite, M.A. (1987), "Influence of the Reynolds Number on the Performance of Centrifugal Compressors", *ASME Journal of Turbomachinery*, 109, pp. 541-544.
- Ulizar, I., Pilidis, P. (1999), "Handling of a Semiclosed Cycle Gas Turbine With a Carbon Dioxide-Argon Working Fluid", *Proceedings of the International Gas Turbine & Aeroengine Congress and Exhibition*, 99-GT-374, Indianapolis, Indiana.
- Urner, G. (1997), "Pressure Loss of Orifice Plates According to ISO 5167-1", *Flow Measurement and Instrumentation*, 8, pp 39-41.
- Weber, C.R., Koronowski, M.E. (1986), "Meanline Performance Prediction of Volute in Centrifugal Compressors", *ASME Paper No. 86-GT-216*. (As referenced by Aungier, 1995)
- White, F.M. (1994), Fluid Mechanics, 3rd ed., McGraw-Hill Inc., Toronto, 736 p.
- Wiesner, F.J. (1967), "A Review of Slip Factors for Centrifugal Impellers", *ASME J. of Engineering for Power*, 89, pp558-572.
- Zhou, M., and Gauthier, J.E.D. (1998), "A Literature Survey on the Use of O₂/CO₂ Mixtures as the Primary Working Fluid in Combustion Engines", Mechanical Engineering Report #980503, Royal Military College of Canada, Kingston, Canada.

APPENDIX A-1: EXPERIMENTAL DATA (AIR)

Table A-1.1: Test Data (Air)

Test Name	Date	Shaft Speed (rpm)	Flow Nozzle ΔP	Nozzle Throat Press. (psi)	Inlet Press. (psi)	Disch. Press. (psi)	Inlet Temp. (°C)	Disch. Temp. (°R)
Ztest01	Apr 20 01	68768	0.4642	13.921	14.149	22.137	529.89	624.36
Ztest02	Apr 20 01	68888	0.2717	14.245	14.389	22.631	529.84	631.68
Ztest03	Apr 20 01	68684	0.1714	14.420	14.498	22.827	530.00	642.27
Ztest04	Apr 20 01	68996	0.8240	13.326	13.702	20.686	530.26	615.51
Ztest05	Apr 20 01	68392	1.1243	12.846	13.349	19.052	529.64	608.57
Ztest06	Apr 20 01	68584	1.4448	12.368	13.011	17.552	529.56	604.94
Ztest 07	Apr 23 01	45636	0.3812	13.976	14.158	16.744	532.14	567.52
Ztest08	Apr 23 01	45464	0.4753	13.803	14.025	16.249	532.20	566.28
Ztest 09	Apr 23 01	45608	0.2077	14.205	14.310	17.479	532.60	573.23
Ztest10	Apr 23 01	83180	0.8773	13.124	13.522	25.187	532.34	667.68
Ztest 11	Apr 23 01	83424	0.4985	13.690	13.936	26.145	532.91	678.58
Ztest12	Apr 23 01	83128	0.7242	13.345	13.677	25.554	533.18	673.24
Ztest13	Apr 23 01	83380	0.5757	13.571	13.840	25.904	533.04	677.65
Ztest14	Apr 23 01	82876	0.8143	13.178	13.540	25.190	532.35	669.48
Ztest15	Apr 23 01	83052	0.9878	12.889	13.354	24.604	531.40	664.59
Ztest16	Apr 23 01	83112	0.9889	12.888	13.343	24.600	531.43	664.46
Ztest17	Apr 23 01	83852	0.7107	8.975	9.238	17.238	531.33	665.94
Ztest18	Apr 23 01	83164	0.7012	8.737	8.977	16.515	531.89	663.35
Ztest19	Apr 23 01	83616	0.9057	9.026	9.361	17.036	532.16	660.72
Ztest20	Apr 23 01	83228	0.9174	9.127	9.475	17.086	532.85	659.57
Ztest21	Apr 23 01	82584	1.830	11.667	12.409	20.492	533.02	648.24
Ztest22	Apr 23 01	82364	1.607	11.955	12.635	21.480	532.66	650.09
Ztest23	Apr 23 01	82768	1.466	12.140	12.775	22.261	532.73	653.52
Ztest24	Apr 24 01	45860	0.1617	13.496	13.569	16.694	532.17	574.72
Ztest25	Apr 24 01	95672	0.8230	10.074	10.394	23.106	531.84	714.44
Ztest26	Apr 24 01	95952	0.9547	8.7950	9.140	20.198	532.14	709.89
Ztest27	Apr 24 01	95416	1.0421	7.8665	8.219	17.791	532.20	703.30
Ztest28	Apr 24 01	95380	0.9709	7.3432	7.664	16.562	532.51	701.45
Ztest29	Apr 24 01	95156	0.9945	7.1957	7.515	16.099	532.70	699.63
Ztest30	Apr 24 01	94688	1.0312	7.0925	7.410	15.712	531.97	696.53
Ztest31	Apr 24 01	94880	1.0640	6.9139	7.240	15.254	534.04	698.03
Ztest32	Apr 24 01	95440	1.3346	6.9952	7.416	15.224	534.03	694.90
Ztest33	Apr 24 01	94940	1.2993	7.0140	7.418	15.227	533.11	693.02
Ztest34	Apr 24 01	95876	1.4428	7.0233	7.463	15.310	532.28	693.44
ccair01	May 31 01	44756	0.0983	13.904	14.035	17.227	520.11	565.97
ccair02	May 31 01	45368	0.0576	13.952	14.049	17.295	519.97	572.05
ccair03	May 31 01	44780	0.1409	13.878	13.929	17.160	519.02	562.13

ccair04	May 31 01	44780	0.1707	13.837	13.940	17.112	518.59	560.59
ccair05	May 31 01	45108	0.1859	13.923	14.050	17.235	518.44	560.42
ccair06	May 31 01	67436	0.5262	12.91	13.242	20.783	518.63	608.34
cc2air01	Jun 11 01	68988	0.8002	10.382	10.712	15.809	527.52	609.49
cc2air02	Jun 11 01	83328	1.2691	9.299	9.744	16.982	530.07	650.60
cc2air03	Jun 11 01	95804	1.8613	8.564	9.186	18.698	533.55	693.40
cc2air04	Jun 11 01	45220	0.3532	12.974	13.131	15.579	526.48	565.06
cc2air06	Jun 11 01	45092	0.2878	13.561	13.693	16.418	526.10	565.30
cc2air07	Jun 11 01	82628	1.1030	10.976	11.412	20.198	531.70	655.26
cc2air08	Jun 11 01	68784	0.6967	11.554	11.827	17.864	528.24	613.38
cc2air09	Jun 11 01	93996	1.5047	10.061	10.620	21.690	535.26	696.41
cc2air10	Jun 11 01	44628	0.2098	13.153	13.231	15.993	524.56	565.38
cc2air11	Jun 11 01	68044	0.6026	12.032	12.264	18.720	525.31	612.55
cc2air12	Jun 11 01	82800	0.977	11.459	11.836	21.502	527.42	655.78
cc2air13	Jun 11 01	94924	1.3644	10.960	11.460	24.291	530.59	700.14
cc2air14	Jun 11 01	82216	0.9125	11.737	12.087	21.788	528.15	656.05
cc2air15	Jun 11 01	93786	1.2762	11.225	11.719	24.494	530.04	695.42
cc2air17	Jun 14 01	82112	1.0106	12.795	12.901	23.201	532.76	660.04
cc2air18	Jun 14 01	82664	0.7216	13.072	13.136	24.293	531.77	668.57
cc2air19	Jun 14 01	93832	1.0001	13.062	13.293	28.093	535.16	711.22
cc2air20	Jun 14 01	68360	0.4405	13.156	12.997	20.177	529.11	625.01
cc2air21	Jun 14 01	44268	0.1693	14.106	14.056	17.068	526.22	567.70
cc2air22	Jun 14 01	68768	0.3138	13.378	13.271	20.830	527.39	626.47
cc2air23	Jun 14 01	93592	0.6948	13.908	14.148	30.131	532.31	714.55
cc2air24	Jun 14 01	45776	0.1126	13.828	13.727	16.969	527.65	577.66
cc2air25	Jun 14 01	82852	0.4689	13.378	13.320	24.889	529.67	673.34
cc2air25b	Jun 25 01	45752	0.4111	14.674	14.902	17.678	529.68	566.13
cc2air26	Jun 25 01	68596	1.0700	14.222	14.702	21.407	533.23	614.42
cc2air27	Jun 25 01	82940	1.6707	13.243	13.668	23.352	537.66	657.83
cc2air28	Jun 25 01	94200	1.6476	13.520	13.999	28.731	540.67	703.80
cc2air29	Jun 25 01	68816	0.7343	14.013	14.061	20.865	533.50	619.48
cc2air30	Jun 25 01	83008	1.2154	13.821	14.163	25.474	538.21	665.96
cc2air32	Jun 25 01	68724	0.7329	13.918	13.953	21.199	535.58	624.16
cc2air33	Jun 25 01	46108	0.2335	14.119	14.054	17.088	532.81	574.47
cc2air34	Jun 25 01	83184	1.2253	13.905	14.229	25.699	538.50	667.04
cc2air35	Jun 25 01	83352	1.0948	14.741	13.970	25.180	540.08	669.15
cc2air36	Jun 25 01	69068	0.6597	14.733	13.882	21.144	535.63	624.56
cc2air37	Jun 25 01	45960	0.2734	14.724	14.149	17.135	532.08	573.06
cc2air38	Jun 25 01	94664	1.5586	14.724	13.347	27.818	539.90	705.59
cc2air39	Jun 25 01	46540	0.1737	14.727	13.922	17.153	532.01	577.52
cc2air40	Jun 25 01	69892	0.4713	14.722	13.386	21.036	533.30	629.92
cc2air41	Jun 25 01	83528	0.7946	14.722	13.970	25.908	536.93	674.92
cc2air42	Jun 25 01	95816	1.0060	14.710	13.048	28.172	539.12	720.53
cc2air43	Jun 25 01	70024	0.2618	14.717	13.185	20.926	533.88	641.37
cc2air44	Jun 25 01	83820	0.3792	14.733	12.927	24.260	535.01	685.37

Table A-1.2: Reduced Test Data (Air)

Test Name	Date	Corrected Map Data:			
		Π_N	Π_m	PR	η (%)
Ztest01	Apr 20 01	15.23	0.05459	1.559	75.6
Ztest02	Apr 20 01	15.23	0.04189	1.567	71.2
Ztest03	Apr 20 01	15.23	0.03319	1.574	64.9
Ztest04	Apr 20 01	15.23	0.07261	1.498	76.3
Ztest05	Apr 20 01	15.23	0.08382	1.421	70.0
Ztest06	Apr 20 01	15.23	0.09469	1.337	60.5
Ztest 07	Apr 23 01	10.06	0.04972	1.179	72.4
Ztest08	Apr 23 01	10.06	0.05528	1.155	65.4
Ztest 09	Apr 23 01	10.06	0.03677	1.219	76.3
Ztest10	Apr 23 01	18.35	0.07495	1.849	75.2
Ztest 11	Apr 23 01	18.35	0.05698	1.862	71.2
Ztest12	Apr 23 01	18.35	0.06821	1.859	73.3
Ztest13	Apr 23 01	18.35	0.06113	1.858	71.3
Ztest14	Apr 23 01	18.35	0.07219	1.853	74.1
Ztest15	Apr 23 01	18.35	0.07943	1.828	74.8
Ztest16	Apr 23 01	18.35	0.07954	1.828	74.9
Ztest17	Apr 23 01	18.35	0.08427	1.835	75.7
Ztest18	Apr 23 01	18.35	0.08145	1.824	75.7
Ztest19	Apr 23 01	18.35	0.08992	1.793	75.7
Ztest20	Apr 23 01	18.35	0.08951	1.785	75.6
Ztest21	Apr 23 01	18.35	0.10564	1.639	69.1
Ztest22	Apr 23 01	18.35	0.09929	1.694	72.3
Ztest23	Apr 23 01	18.35	0.09557	1.731	74.0
Ztest24	Apr 24 01	10.06	0.03358	1.225	75.6
Ztest25	Apr 24 01	21.03	0.08240	2.193	73.4
Ztest26	Apr 24 01	21.03	0.09337	2.170	74.6
Ztest27	Apr 24 01	21.03	0.10053	2.135	75.2
Ztest28	Apr 24 01	21.03	0.10048	2.133	76.0
Ztest29	Apr 24 01	21.03	0.10205	2.119	75.9
Ztest30	Apr 24 01	21.03	0.10385	2.106	75.7
Ztest31	Apr 24 01	21.03	0.10649	2.091	75.5
Ztest32	Apr 24 01	21.03	0.11550	2.021	73.7
Ztest33	Apr 24 01	21.03	0.11383	2.030	74.1
Ztest34	Apr 24 01	21.03	0.11979	2.010	73.4
ccair01	May 31 01	10.06	0.0254	1.229	67.9
ccair02	May 31 01	10.06	0.0197	1.227	60.7
ccair03	May 31 01	10.06	0.0307	1.232	73.3
ccair04	May 31 01	10.06	0.0337	1.227	73.8
ccair05	May 31 01	10.06	0.0352	1.223	73.5
ccair06	May 31 01	15.23	0.0590	1.573	78.4

cc2air01	Jun 11 01	15.23	0.08003	1.461	74.1
cc2air02	Jun 11 01	18.35	0.10176	1.717	73.8
cc2air03	Jun 11 01	21.03	0.12116	1.997	73.1
cc2air04	Jun 11 01	10.06	0.04939	1.184	67.1
cc2air06	Jun 11 01	10.06	0.04374	1.198	70.4
cc2air07	Jun 11 01	18.35	0.08880	1.762	74.7
cc2air08	Jun 11 01	15.23	0.07162	1.500	76.2
cc2air09	Jun 11 01	21.03	0.10323	2.052	73.4
cc2air10	Jun 11 01	10.06	0.03782	1.212	70.7
cc2air11	Jun 11 01	15.23	0.06543	1.526	76.1
cc2air12	Jun 11 01	18.35	0.08340	1.801	75.0
cc2air13	Jun 11 01	21.03	0.09758	2.102	73.4
cc2air14	Jun 11 01	18.35	0.07972	1.800	74.4
cc2air15	Jun 11 01	21.03	0.09307	2.099	73.6
cc2air17	Jun 14 01	18.35	0.08493	1.806	75.1
cc2air18	Jun 14 01	18.35	0.07240	1.848	73.5
cc2air19	Jun 14 01	21.03	0.08171	2.138	71.2
cc2air20	Jun 14 01	15.23	0.05815	1.553	73.0
cc2air21	Jun 14 01	10.06	0.03327	1.222	71.7
cc2air22	Jun 14 01	15.23	0.04876	1.563	72.4
cc2air23	Jun 14 01	21.03	0.06524	2.158	69.4
cc2air24	Jun 14 01	10.06	0.02877	1.230	65.4
cc2air25	Jun 14 01	18.35	0.05871	1.862	71.2
cc2air25b	Jun 25 01	10.06	0.04989	1.180	71.2
cc2air26	Jun 25 01	15.23	0.07736	1.451	72.7
cc2air27	Jun 25 01	18.35	0.09988	1.701	72.2
cc2air28	Jun 25 01	21.03	0.09591	2.071	74.0
cc2air29	Jun 25 01	15.23	0.06860	1.479	72.8
cc2air30	Jun 25 01	18.35	0.08478	1.795	75.4
cc2air32	Jun 25 01	15.23	0.06891	1.518	75.5
cc2air33	Jun 25 01	10.06	0.04053	1.209	72.6
cc2air34	Jun 25 01	18.35	0.08485	1.799	75.6
cc2air35	Jun 25 01	18.35	0.08231	1.796	75.3
cc2air36	Jun 25 01	15.23	0.06630	1.517	75.7
cc2air37	Jun 25 01	10.06	0.04331	1.205	72.0
cc2air38	Jun 25 01	21.03	0.09889	2.092	74.5
cc2air39	Jun 25 01	10.06	0.03574	1.220	71.1
cc2air40	Jun 25 01	15.23	0.05959	1.552	75.1
cc2air41	Jun 25 01	18.35	0.07184	1.843	73.9
cc2air42	Jun 25 01	21.03	0.08521	2.144	71.8
cc2air43	Jun 25 01	15.23	0.04572	1.567	69.4
cc2air44	Jun 25 01	18.35	0.05532	1.860	69.3

APPENDIX A-2: EXPERIMENTAL DATA (CARBON DIOXIDE)

Table A-2.1: Test Data (Carbon Dioxide)

Test Name	Date	Shaft Speed (rpm)	Flow Nozzle ΔP (psid)	Nozzle Throat Press. (psi)	Inlet Press. (psi)	Disch. Press. (psia)	Inlet Temp. (°C)	Disch. Temp. (°R)
Co2001	Jul 28 01	54147	1.040	15.93	15.40	22.20	538.97	600.11
Co2002	Jul 28 01	54153	1.083	15.79	15.28	22.00	538.36	598.35
Co2003	Jul 28 01	54150	1.065	14.63	15.22	21.80	538.12	597.66
Co2004	Jul 28 01	53880	0.698	14.27	14.69	21.82	537.04	600.02
Co2005	Jul 28 01	53507	0.504	14.17	14.47	21.69	536.31	601.52
Co2006	Jul 28 01	54430	0.277	14.27	14.39	22.04	535.48	608.65
CO2007	Jul 29 01	53907	1.018	14.45	15.08	21.40	537.61	595.80
Co2008	Jul 29 01	53953	0.726	14.32	14.75	21.85	537.08	599.36
Co2009	Jul 29 01	54020	0.734	14.29	14.73	21.85	537.31	599.82
Co2010	Jul 29 01	54113	0.503	14.34	14.64	22.26	536.75	604.23
Co2011	Jul 29 01	54230	0.505	14.33	14.63	22.27	536.69	604.39
Co2012	Jul 29 01	53980	0.312	14.51	14.70	22.51	535.45	607.07
Co2013	Jul 29 01	53987	0.314	14.50	14.69	22.52	535.53	607.23
Co2014	Jul 29 01	54127	0.172	14.43	14.53	22.57	535.27	616.69
Co2015	Jul 29 01	54093	0.172	14.44	14.54	22.57	535.30	616.57
Co2016	Jul 29 01	54340	0.109	14.45	14.51	22.40	534.78	623.29
Co2017	Jul 29 01	54387	0.111	14.48	14.53	22.43	534.74	623.79
Co2018	Jul 29 01	53933	0.064	14.59	14.61	22.06	533.88	630.98
Co2019	Jul 29 01	54000	0.063	14.54	14.57	22.04	533.86	630.94
Co2020	Jul 29 01	65213	1.770	13.92	15.05	24.73	542.07	627.26
Co2021	Jul 29 01	65233	1.714	13.88	14.97	24.46	542.16	626.95
Co2022	Jul 29 01	64860	1.281	13.94	14.76	25.18	541.02	628.60
Co2023	Jul 29 01	64920	1.284	13.87	14.69	25.10	541.45	629.36
Co2024	Jul 29 01	64860	0.938	13.81	14.40	25.22	540.34	632.54
Co2025	Jul 29 01	64907	0.929	13.78	14.37	25.19	540.01	632.43
Co2026	Jul 29 01	65140	0.605	14.06	14.44	25.79	539.47	639.34
Co2027	Jul 29 01	64640	0.611	14.18	14.56	25.93	539.18	637.79
Co2028	Jul 29 01	64513	0.307	14.14	14.32	25.79	538.49	646.77
Co2029	Jul 29 01	64627	0.317	14.08	14.28	25.72	538.31	646.74
Co2030	Jul 29 01	64593	0.181	14.13	14.24	25.59	537.11	657.71
Co2031	Jul 29 01	64267	0.180	14.14	14.24	25.52	537.36	657.68
Co2032	Jul 29 01	64847	0.102	14.18	14.26	25.20	535.15	670.05
Co2033	Jul 29 01	64953	0.102	14.16	14.23	25.17	535.71	672.32
Co2034	Aug 10 01	35420	0.339	14.50	14.66	17.09	540.22	565.91
Co2035	Aug 10 01	35567	0.341	14.48	14.65	17.08	540.10	565.94
Co2036	Aug 10 01	34953	0.328	14.43	14.64	16.99	540.03	565.07
Co2037	Aug 10 01	35113	0.220	15.28	15.41	18.27	539.39	566.58

Co2038	Aug 10 01	35433	0.219	15.20	15.33	18.30	539.53	567.67
Co2039	Aug 10 01	35607	0.222	15.18	15.31	18.31	539.62	568.08
Co2040	Aug 10 01	35500	0.145	15.24	15.32	18.49	539.57	570.28
Co2041	Aug 10 01	35600	0.145	15.19	15.27	18.44	539.54	570.36
Co2042	Aug 10 01	35567	0.145	15.18	15.26	18.42	539.57	570.48
Co2043	Aug 10 01	35647	0.109	15.22	15.28	18.53	539.40	571.74
Co2044	Aug 10 01	35520	0.108	15.28	15.34	18.58	539.38	571.56
Co2045	Aug 10 01	35760	0.109	15.25	15.31	18.58	539.41	571.93
Co2046	Aug 10 01	35100	0.060	15.15	15.19	18.36	539.02	574.45
Co2047	Aug 10 01	35087	0.061	15.15	15.18	18.35	539.00	574.39
Co2048	Aug 11 01	35120	0.065	15.84	15.86	19.23	537.22	571.20
Co2049	Aug 11 01	35340	0.065	15.68	15.71	19.06	537.65	572.73
CO2050	Aug 11 01	35673	0.036	15.58	15.60	18.92	537.80	577.95
CO2051	Aug 11 01	35760	0.036	15.55	15.57	18.91	537.88	578.78
CO2052	Aug 11 01	35633	0.036	15.55	15.56	18.87	537.87	578.46
CO2053	Aug 11 01	36067	0.024	15.53	15.54	18.83	537.76	582.34
CO2054	Aug 11 01	36227	0.024	15.51	15.52	18.84	537.77	583.79
CO2055	Aug 11 01	75013	2.785	13.48	15.24	28.79	552.38	664.05
CO2056	Aug 11 01	75753	2.799	13.53	15.22	28.97	552.79	666.56
CO2057	Aug 11 01	74807	1.827	14.49	15.66	31.95	551.99	670.26
CO2058	Aug 11 01	74647	1.860	14.46	15.64	31.87	551.92	669.90
CO2059	Aug 11 01	74600	1.242	14.47	15.25	31.87	550.79	676.14
CO2060	Aug 11 01	74987	1.274	14.18	14.97	31.83	550.26	676.83
CO2061	Aug 13 01	74760	1.330	14.59	15.42	32.42	548.32	672.72
CO2062	Aug 13 01	74860	1.303	14.51	15.32	32.33	547.75	673.00
CO2063	Aug 13 01	75727	0.520	14.90	15.22	32.77	546.23	689.43
CO2064	Aug 13 01	75853	0.539	14.80	15.13	32.67	545.91	689.87
CO2065	Aug 13 01	75853	0.539	14.80	15.13	32.67	545.91	689.87
CO2066	Aug 13 01	75020	1.119	14.44	15.14	32.15	548.40	677.35
CO2067	Aug 13 01	75013	0.958	14.58	15.18	32.33	547.32	678.69
CO2068	Aug 13 01	75113	0.448	14.75	15.03	32.06	545.25	688.65
CO2069	Aug 13 01	74800	0.270	14.76	14.92	31.43	544.51	701.78
CO2070	Aug 13 01	74900	0.283	14.74	14.90	31.40	544.28	702.52
CO2071	Aug 13 01	75413	0.126	14.72	14.79	29.97	542.61	729.14
CO2072	Aug 13 01	75567	0.128	14.71	14.77	29.93	542.14	729.54
CO2073	Aug 14 01	74980	2.124	14.53	15.83	32.25	547.21	662.83
CO2074	Aug 14 01	75447	2.101	14.31	15.51	31.99	549.75	667.65
CO2075	Aug 14 01	75147	0.716	14.59	14.62	32.38	546.05	681.36
CO2076	Aug 14 01	75193	0.712	14.63	15.03	32.42	546.06	681.56
CO2077	Aug 14 01	75220	0.710	14.74	15.09	32.45	546.73	682.87
CO2078	Aug 14 01	64853	0.452	14.77	15.06	27.28	543.49	647.44
CO2079	Aug 14 01	64987	0.461	14.80	15.09	27.38	543.30	647.33
CO2080	Aug 14 01	54540	0.292	15.07	15.26	23.55	540.80	615.75
CO2081	Aug 14 01	54267	0.289	15.16	15.32	23.57	540.48	614.31
CO2082	Aug 28 01	35633	0.021	15.50	15.51	18.00	533.15	577.55
CO2083	Aug 28 01	35200	0.158	15.48	15.59	18.74	533.99	564.35

CO2084	Aug 28 01	35327	0.112	15.40	15.49	18.73	533.50	565.13
CO2085	Aug 28 01	35300	0.207	15.33	15.48	18.51	533.58	562.09
CO2086	Aug 28 01	54173	0.425	14.88	15.29	23.05	537.87	603.42
CO2087	Aug 28 01	54367	1.028	14.72	15.44	22.09	538.73	598.46
CO2088	Aug 28 01	54193	0.598	14.78	15.23	22.80	536.69	600.48
CO2089	Aug 28 01	53980	0.643	14.79	15.24	22.79	537.36	601.02
CO2090	Aug 28 01	54160	0.241	15.02	15.18	23.40	536.36	610.45
CO2091	Aug 28 01	54140	0.094	14.96	15.00	23.10	535.84	624.62
CO2092	Aug 28 01	64647	0.144	14.67	14.77	26.49	537.95	664.63
CO2093	Aug 28 01	64433	0.740	14.39	14.80	26.53	540.98	636.68
CO2094	Aug 28 01	64460	0.768	14.44	14.73	26.53	540.74	636.32
CO2095	Aug 28 01	35630	0.421	15.13	15.36	17.99	533.46	559.46
CO2096	Aug 28 01	64927	1.803	14.32	15.44	25.20	540.66	623.59

Table A-2.2: Reduced Test Data (Carbon Dioxide)

Test Name	Date	Corrected Map Data:			
		Π_N	Π_m	PR	η (%)
Co2001	Jul 28 01	15.23	0.1676	1.435	76.1
Co2002	Jul 28 01	15.23	0.1710	1.431	77.0
Co2003	Jul 28 01	15.23	0.1699	1.424	76.4
Co2004	Jul 28 01	15.23	0.1422	1.482	80.2
Co2005	Jul 28 01	15.23	0.1230	1.505	79.6
Co2006	Jul 28 01	15.23	0.0925	1.520	74.9
Co2007	Jul 29 01	15.23	0.1667	1.414	76.0
Co2008	Jul 29 01	15.23	0.1444	1.477	80.5
Co2009	Jul 29 01	15.23	0.1452	1.478	80.6
Co2010	Jul 29 01	15.23	0.1221	1.515	79.7
Co2011	Jul 29 01	15.23	0.1224	1.514	79.7
Co2012	Jul 29 01	15.23	0.0970	1.529	76.5
Co2013	Jul 29 01	15.23	0.0972	1.530	76.6
Co2014	Jul 29 01	15.23	0.0729	1.549	69.7
Co2015	Jul 29 01	15.23	0.0728	1.548	69.6
Co2016	Jul 29 01	15.23	0.0583	1.535	63.0
Co2017	Jul 29 01	15.23	0.0587	1.535	62.6
Co2018	Jul 29 01	15.23	0.0445	1.509	54.0
Co2019	Jul 29 01	15.23	0.0443	1.510	54.3
Co2020	Jul 29 01	18.35	0.2120	1.634	75.5
Co2021	Jul 29 01	18.35	0.2096	1.625	75.0
Co2022	Jul 29 01	18.35	0.1862	1.706	79.7
Co2023	Jul 29 01	18.35	0.1867	1.708	79.7
Co2024	Jul 29 01	18.35	0.1642	1.754	79.8
Co2025	Jul 29 01	18.35	0.1637	1.753	79.7
Co2026	Jul 29 01	18.35	0.1341	1.784	76.6
Co2027	Jul 29 01	18.35	0.1341	1.791	77.2
Co2028	Jul 29 01	18.35	0.0974	1.816	71.7
Co2029	Jul 29 01	18.35	0.0990	1.813	71.6
Co2030	Jul 29 01	18.35	0.0756	1.810	64.0
Co2031	Jul 29 01	18.35	0.0753	1.812	63.8
Co2032	Jul 29 01	18.35	0.0568	1.772	55.2
Co2033	Jul 29 01	18.35	0.0569	1.771	54.6
Co2034	Aug 10 01	10.06	0.1012	1.165	71.5
Co2035	Aug 10 01	10.06	0.0997	1.164	71.2
Co2036	Aug 10 01	10.06	0.0997	1.164	71.2
Co2037	Aug 10 01	10.06	0.0801	1.190	75.8
Co2038	Aug 10 01	10.06	0.0802	1.195	76.4
Co2039	Aug 10 01	10.06	0.0808	1.195	76.3

Co2040	Aug 10 01	10.06	0.0654	1.208	74.5
Co2041	Aug 10 01	10.06	0.0656	1.207	74.5
Co2042	Aug 10 01	10.06	0.0656	1.207	74.0
Co2043	Aug 10 01	10.06	0.0569	1.212	72.6
Co2044	Aug 10 01	10.06	0.0565	1.212	72.3
Co2045	Aug 10 01	10.06	0.0569	1.212	72.7
Co2046	Aug 10 01	10.06	0.0427	1.215	64.9
Co2047	Aug 10 01	10.06	0.0427	1.215	65.0
Co2048	Aug 11 01	10.06	0.0433	1.218	68.6
Co2049	Aug 11 01	10.06	0.0435	1.217	66.8
CO2050	Aug 11 01	10.06	0.0327	1.212	57.8
CO2051	Aug 11 01	10.06	0.0327	1.213	57.1
CO2052	Aug 11 01	10.06	0.0326	1.213	57.2
CO2053	Aug 11 01	10.06	0.0268	1.207	51.4
CO2054	Aug 11 01	10.06	0.0267	1.207	50.2
CO2055	Aug 11 01	21.03	0.2532	1.877	74.2
CO2056	Aug 11 01	21.03	0.2540	1.874	73.9
CO2057	Aug 11 01	21.03	0.2118	2.040	79.8
CO2058	Aug 11 01	21.03	0.2134	2.041	79.9
CO2059	Aug 11 01	21.03	0.1817	2.098	78.2
CO2060	Aug 11 01	21.03	0.1853	2.121	79.3
CO2061	Aug 13 01	21.03	0.1862	2.100	79.1
CO2062	Aug 13 01	21.03	0.1853	2.104	78.9
CO2063	Aug 13 01	21.03	0.1222	2.125	71.2
CO2064	Aug 13 01	21.03	0.1246	2.127	71.1
CO2065	Aug 13 01	21.03	0.1246	2.127	71.1
CO2066	Aug 13 01	21.03	0.1742	2.116	77.5
CO2067	Aug 13 01	21.03	0.1624	2.121	76.3
CO2068	Aug 13 01	21.03	0.1146	2.124	70.1
CO2069	Aug 13 01	21.03	0.0900	2.107	62.7
CO2070	Aug 13 01	21.03	0.0922	2.104	62.3
CO2071	Aug 13 01	21.03	0.0622	2.007	49.6
CO2072	Aug 13 01	21.03	0.0628	2.003	49.3
CO2073	Aug 14 01	21.03	0.2243	2.020	80.6
CO2074	Aug 14 01	21.03	0.2263	2.038	81.0
CO2075	Aug 14 01	21.03	0.1467	2.201	78.3
CO2076	Aug 14 01	21.03	0.1427	2.143	75.3
CO2077	Aug 14 01	21.03	0.1416	2.137	74.7
CO2078	Aug 14 01	18.35	0.1149	1.819	74.1
CO2079	Aug 14 01	18.35	0.1158	1.818	74.2
CO2080	Aug 14 01	15.23	0.0924	1.531	73.1
CO2081	Aug 14 01	15.23	0.0919	1.531	73.6
CO2082	Aug 28 01	10.06	0.0248	1.210	51.1
CO2083	Aug 28 01	10.06	0.0677	1.204	72.9
CO2084	Aug 28 01	10.06	0.0573	1.210	72.2

CO2085	Aug 28 01	10.06	0.0776	1.196	75.1
CO2086	Aug 28 01	15.23	0.1109	1.499	78.4
CO2087	Aug 28 01	15.23	0.1662	1.416	73.8
CO2088	Aug 28 01	15.23	0.1305	1.486	78.8
CO2089	Aug 28 01	15.23	0.1348	1.488	78.8
CO2090	Aug 28 01	15.23	0.0844	1.533	73.2
CO2091	Aug 28 01	15.23	0.0535	1.532	60.7
CO2092	Aug 28 01	18.35	0.0665	1.801	59.1
CO2093	Aug 28 01	18.35	0.1457	1.802	78.3
CO2094	Aug 28 01	18.35	0.1481	1.809	79.0
CO2095	Aug 28 01	10.06	0.1098	1.167	71.9
CO2096	Aug 28 01	18.35	0.2118	1.622	74.1

APPENDIX A-3: EXPERIMENTAL DATA (ARGON)

Table A-3.1: Test Data (Argon)

Test Name	Date	Shaft Speed (rpm)	Flow Nozzle Press. (psid)	Nozzle Throat Press. (psia)	Inlet Press. (psia)	Disch. Press. (psia)	Inlet Temp. (°R)	Disch. Temp. (°R)
Ar 018	Jul 25 01	65047	0.316	15.15	15.34	25.32	538.19	704.89
Ar 019	Jul 25 01	64227	0.222	14.99	15.11	24.75	537.05	712.27
Ar 020	Jul 25 01	64493	0.223	14.94	15.06	24.76	537.53	713.55
Ar 021	Jul 25 01	64613	0.226	14.98	15.11	24.88	537.68	715.70
Ar 022	Jul 25 01	64040	0.181	14.89	15.00	24.46	536.87	718.15
Ar 023	Jul 25 01	64020	0.180	14.92	15.02	24.47	537.48	719.60
Ar 024	Jul 25 01	64113	0.180	14.90	15.01	24.47	537.49	720.15
Ar 025	Jul 25 01	64613	0.121	14.86	14.93	24.22	537.08	734.76
Ar 026	Jul 25 01	64633	0.121	14.83	14.90	24.18	536.94	734.87
Ar 027	Jul 25 01	64547	0.119	14.83	14.89	24.14	536.92	734.84
Ar 028	Jul 25 01	64567	0.076	14.87	14.90	23.69	536.03	748.84
Ar 029	Jul 25 01	64710	0.076	14.84	14.88	23.70	535.87	747.94
Ar 030	Jul 25 01	64520	0.078	14.84	14.88	23.64	535.98	748.43
Ar 031	Jul 26 01	42100	0.4658	14.987	15.27	18.374	534.72	588.88
Ar 032	Jul 26 01	42673	0.4708	14.901	15.191	18.372	535.22	592.07
Ar 033	Jul 26 01	42553	0.4646	14.841	15.129	18.284	535.44	592.65
Ar 034	Jul 26 01	42667	0.4637	14.802	15.088	18.245	535.54	593.23
Ar 035	Jul 26 01	42587	0.4628	14.779	15.071	18.214	535.48	593.07
Ar 036	Jul 26 01	42800	0.4682	14.751	15.050	18.220	535.47	593.56
Ar 037	Jul 26 01	42400	0.3835	14.730	14.982	18.312	535.14	594.21
Ar 038	Jul 26 01	42333	0.3880	14.975	15.214	18.591	534.71	593.23
Ar 039	Jul 26 01	42027	0.3818	15.017	15.254	18.583	534.91	592.76
Ar 040	Jul 26 01	42260	0.3431	14.997	15.164	18.630	534.84	594.52
Ar 041	Jul 26 01	42433	0.3089	14.876	15.062	18.646	534.69	596.19
Ar 043	Jul 26 01	42440	0.2674	14.908	15.070	18.806	534.61	598.46
Ar 044	Jul 26 01	42640	0.2085	14.933	15.057	18.986	534.36	601.62
Ar 045	Jul 26 01	42520	0.2066	14.927	15.048	18.938	534.40	601.54
Ar 046	Jul 26 01	42653	0.1357	14.932	15.019	19.014	534.15	605.34
Ar 047	Jul 26 01	42567	0.1365	14.967	15.036	19.066	534.04	605.44
Ar 048	Jul 26 01	42600	0.1363	14.910	15.002	19.010	533.96	605.20
Ar 049	Jul 26 01	42160	0.0845	14.818	14.873	18.818	532.50	606.57
Ar 050	Jul 26 01	42507	0.0850	14.787	14.842	18.849	532.84	608.86
Ar 051	Jul 26 01	42333	0.0848	14.785	14.839	18.817	533.01	609.21
Ar 052	Jul 26 01	42100	0.0537	14.790	14.818	18.700	532.94	615.70
Ar 053	Jul 26 01	41833	0.0526	14.806	14.834	18.668	532.94	615.30
Ar 054	Jul 26 01	42227	0.0538	14.811	14.841	18.747	533.02	617.63
Ar 055	Jul 26 01	42027	0.0240	14.840	14.854	18.526	532.67	624.95

Ar 056	Jul 26 01	42380	0.0244	14.856	14.869	18.568	532.63	628.79
Ar 057	Jul 26 01	42500	0.0240	14.864	14.880	18.616	532.60	629.11
Ar 058	Jul 26 01	64047	1.285	14.40	15.16	22.87	538.62	667.27
Ar 059	Jul 26 01	64107	1.267	14.49	15.22	22.92	540.66	668.69
Ar 060	Jul 26 01	64320	1.277	14.54	15.23	22.95	541.19	669.29
Ar 063	Jul 26 01	65360	0.670	14.50	14.94	24.47	538.31	691.22
Ar 064	Jul 26 01	65013	0.671	14.57	15.01	24.53	538.57	691.10
Ar 065	Jul 26 01	64513	0.409	14.08	14.34	23.46	537.85	695.58
Ar 066	Jul 26 01	64773	0.411	14.07	14.34	23.50	537.54	695.12
Ar 068	Jul 28 01	77613	2.1015	14.334	15.665	27.930	544.86	738.61
Ar 069	Jul 28 01	78173	1.7062	14.900	15.961	29.849	545.57	752.35
Ar 070	Jul 28 01	78013	1.7126	14.892	15.962	29.847	545.38	752.23
Ar 071	Jul 28 01	78100	1.7130	14.879	15.951	29.815	545.44	753.20
Ar 072	Jul 28 01	78140	1.2956	15.147	15.986	30.880	545.48	764.70
Ar 073	Jul 28 01	78300	1.3010	14.995	15.824	30.675	544.39	765.07
Ar 074	Jul 28 01	78227	1.2968	14.992	15.821	30.617	545.13	765.97
Ar 075	Jul 28 01	78233	1.1284	15.112	15.829	31.007	543.57	768.33
Ar 076	Jul 28 01	77760	0.8273	15.074	15.599	30.635	541.76	774.37
Ar 077	Jul 28 01	77727	0.8168	15.112	15.634	30.706	541.18	773.78
Ar 078	Jul 28 01	77767	0.5343	15.246	15.591	30.674	539.87	782.05
Ar 079	Jul 28 01	77753	0.5345	15.259	15.602	30.698	539.28	781.83
Ar 080	Jul 28 01	78100	0.3436	15.218	15.446	30.483	537.81	802.61
Ar 081	Jul 28 01	78307	0.3461	15.186	15.412	30.511	537.70	803.70
Ar 082	Jul 28 01	78280	0.2139	15.203	15.344	29.874	536.99	822.94
Ar 083	Jul 28 01	78040	0.2115	15.209	15.344	29.753	537.21	824.34
Ar 084	Aug 15 01	64100	1.1796	14.708	15.488	23.617	542.49	671.39
Ar 085	Aug 15 01	88640	2.8152	13.192	14.933	30.891	548.91	796.61
Ar 086	Aug 15 01	89513	2.8499	13.111	14.887	31.023	548.62	801.52
Ar 087	Aug 15 01	89433	2.8932	13.434	15.248	31.513	549.70	798.35
Ar 088	Aug 15 01	89133	2.2630	13.418	14.884	31.976	550.02	808.98
Ar 089	Aug 15 01	88827	2.2945	13.515	14.958	32.086	549.19	808.66
Ar 090	Aug 15 01	89353	1.7967	13.748	14.900	33.105	550.55	826.88
Ar 091	Aug 15 01	89333	1.8256	13.735	14.896	33.146	548.52	824.70
Ar 092	Aug 15 01	89687	1.3640	14.011	14.896	33.908	547.56	840.78
Ar 093	Aug 15 01	89520	1.3624	14.060	14.925	33.907	547.55	839.22
Ar 094	Aug 15 01	89507	0.9691	14.952	15.453	35.544	546.82	852.24
Ar 095	Aug 15 01	89445	0.9402	14.784	15.411	35.446	546.76	853.16
Ar 096	Aug 15 01	90033	0.9594	14.700	15.329	35.399	546.96	855.89
Ar 097	Aug 15 01	90340	0.5532	14.668	15.010	34.867	544.60	873.99
Ar 098	Aug 15 01	90253	0.5537	14.653	14.998	34.904	543.47	872.46
Ar 099	Aug 15 01	90070	0.2826	14.648	14.825	33.644	541.53	904.01
Ar 101	Aug 19 01	42827	0.1490	14.99	15.07	19.22	535.17	608.00
Ar 102	Aug 19 01	43107	0.1472	15.00	15.08	19.27	535.35	609.40
Ar 103	Aug 19 01	64180	0.3515	14.56	14.78	24.30	537.15	698.10
Ar 104	Aug 19 01	64347	0.3500	14.54	14.75	24.29	538.79	703.27
Ar 105	Aug 19 01	77507	0.5511	14.39	14.74	29.04	540.26	775.66

Ar 106	Aug 19 01	77707	0.5452	14.37	14.72	29.02	540.88	777.96
Ar 107	Aug 19 01	78333	1.3999	14.14	15.03	28.46	546.36	751.45
Ar 108	Aug 19 01	78220	1.3939	14.17	15.05	28.52	545.55	750.43
Ar 109	Aug 19 01	64387	0.8179	14.24	14.76	23.45	541.04	684.49
Ar 110	Aug 19 01	64413	0.8205	14.24	14.77	23.48	540.57	684.24
Ar 111	Aug 19 01	42653	0.3084	14.86	15.04	18.66	537.19	603.69
Ar 112	Aug 19 01	42673	0.3097	14.89	15.03	18.67	536.10	601.02
Ar 113	Aug 19 01	42647	0.3445	15.64	15.83	19.67	535.83	600.09
Ar 114	Aug 19 01	42713	0.3281	15.58	15.76	19.61	535.67	599.90
Ar 115	Aug 19 01	64380	0.8822	14.98	15.55	24.75	541.25	685.62
Ar 116	Aug 19 01	64433	0.8753	14.92	15.49	24.68	540.70	685.46
Ar 117	Aug 19 01	77860	1.4931	15.05	15.79	29.97	544.60	753.11
Ar 118	Aug 19 01	77987	1.4791	14.85	15.78	29.96	545.12	754.45
Ar 119	Aug 19 01	64087	1.0220	14.89	15.54	24.23	541.29	681.02
Ar 120	Aug 19 01	63860	1.0233	14.99	15.64	24.34	540.55	678.86
Ar 121	Aug 19 01	78327	1.7735	14.46	15.61	28.98	543.83	747.69
Ar 122	Aug 19 01	78273	1.7750	14.80	15.71	29.12	545.25	750.00
Ar 123	Aug 19 01	78080	1.1460	14.91	15.65	30.59	543.59	764.33
Ar 124	Aug 19 01	78260	1.1423	14.89	15.61	30.55	544.14	766.35
Ar 125	Aug 19 01	64107	0.6759	15.24	15.67	25.33	542.15	695.66
Ar 126	Aug 19 01	64520	0.6773	15.02	15.48	25.14	540.48	694.40
Ar 127	Aug 27 01	64293	1.1777	14.599	15.396	23.498	539.45	673.48
Ar 128	Aug 27 01	64260	1.3475	14.701	15.496	23.589	540.21	674.96
Ar 129	Aug 27 01	64400	1.1428	14.633	15.418	23.520	540.35	675.91
Ar 130	Aug 27 01	64140	0.9413	14.627	15.287	23.945	539.74	679.57
Ar 131	Aug 27 01	64067	0.9527	14.634	15.294	23.919	539.72	679.29
Ar 132	Aug 27 01	64193	0.4203	14.818	15.186	24.827	538.49	695.13
Ar 133	Aug 27 01	64027	0.7173	14.691	15.185	24.372	539.16	686.88
Ar 134	Aug 27 01	64100	0.6580	14.684	15.154	24.483	539.23	689.68
Ar 135	Aug 27 01	64147	0.6324	14.701	15.119	24.597	538.50	691.61
Ar 136	Aug 27 01	64800	0.2797	14.930	15.114	25.121	537.23	710.42
Ar 137	Aug 27 01	64807	0.2758	14.928	15.113	25.126	536.96	710.08
Ar 138	Aug 27 01	64373	0.0855	15.038	15.102	24.341	535.47	745.86
Ar 139	Aug 27 01	77613	0.1207	14.934	15.013	28.414	535.34	834.72
Ar 144	Aug 28 01	42547	0.4208	14.990	15.295	18.608	534.99	593.77
Ar 145	Aug 28 01	42873	0.3672	15.031	15.290	18.898	535.06	597.48
Ar 146	Aug 28 01	42413	0.2229	15.100	15.261	19.247	534.68	602.73
Ar 147	Aug 28 01	42773	0.0890	15.150	15.223	19.472	534.21	611.34
Ar 148	Aug 28 01	42680	0.0432	15.205	15.226	19.296	533.75	626.53

Table A-3.2: Reduced Test Data (Argon)

Test Name	Date	Corrected Map Data:			
		Π_N	Π_m	PR	η (%)
Ar017	Jul 25 01	15.23	0.0465	1.629	71.7
Ar018	Jul 25 01	15.23	0.0477	1.633	71.0
Ar019	Jul 25 01	15.23	0.0400	1.637	66.4
Ar020	Jul 25 01	15.23	0.0403	1.638	66.7
Ar021	Jul 25 01	15.23	0.0405	1.639	66.2
Ar022	Jul 25 01	15.23	0.0361	1.633	63.5
Ar023	Jul 25 01	15.23	0.0360	1.633	63.1
Ar024	Jul 25 01	15.23	0.0361	1.633	63.1
Ar025	Jul 25 01	15.23	0.0300	1.615	57.6
Ar026	Jul 25 01	15.23	0.0300	1.614	57.5
Ar027	Jul 25 01	15.23	0.0297	1.614	57.3
Ar028	Jul 25 01	15.23	0.0239	1.583	50.8
Ar029	Jul 25 01	15.23	0.0239	1.582	51.2
Ar030	Jul 26 01	15.23	0.0240	1.582	50.8
Ar031	Jul 26 01	10.06	0.0567	1.202	74.1
Ar032	Jul 26 01	10.06	0.0579	1.203	72.7
Ar033	Jul 26 01	10.06	0.0574	1.204	72.0
Ar034	Jul 26 01	10.06	0.0576	1.203	71.6
Ar035	Jul 26 01	10.06	0.0575	1.203	71.5
Ar036	Jul 26 01	10.06	0.0581	1.203	71.6
Ar037	Jul 26 01	10.06	0.0524	1.219	74.3
Ar038	Jul 26 01	10.06	0.0523	1.219	74.9
Ar039	Jul 26 01	10.06	0.0514	1.219	74.5
Ar040	Jul 26 01	10.06	0.0491	1.227	75.6
Ar041	Jul 26 01	10.06	0.0471	1.235	76.3
Ar043	Jul 26 01	10.06	0.0439	1.245	76.5
Ar044	Jul 26 01	10.06	0.0390	1.256	76.3
Ar045	Jul 26 01	10.06	0.0388	1.255	75.8
Ar046	Jul 26 01	10.06	0.0316	1.261	73.5
Ar047	Jul 26 01	10.06	0.0316	1.264	73.8
Ar049	Jul 26 01	10.06	0.0249	1.266	70.3
Ar050	Jul 26 01	10.06	0.0252	1.267	69.6
Ar051	Jul 26 01	10.06	0.0250	1.267	69.0
Ar052	Jul 26 01	10.06	0.0198	1.264	62.1
Ar053	Jul 26 01	10.06	0.0195	1.264	61.6
Ar054	Jul 26 01	10.06	0.0199	1.264	61.0
Ar055	Jul 26 01	10.06	0.0132	1.250	52.5
Ar056	Jul 26 01	10.06	0.0134	1.248	50.6
Ar057	Jul 26 01	10.06	0.0133	1.248	50.9

Ar058	Jul 26 01	15.23	0.0916	1.502	73.1
Ar059	Jul 26 01	15.23	0.0908	1.501	73.5
Ar060	Jul 26 01	15.23	0.0913	1.499	73.5
Ar063	Jul 26 01	15.23	0.0697	1.612	75.8
Ar064	Jul 26 01	15.23	0.0692	1.616	75.8
Ar065	Jul 26 01	15.23	0.0555	1.628	73.5
Ar066	Jul 26 01	15.23	0.0559	1.625	73.9
Ar068	Jul 28 01	18.35	0.1120	1.773	71.5
Ar069	Jul 28 01	18.35	0.1021	1.852	73.7
Ar070	Jul 28 01	18.35	0.1021	1.855	73.6
Ar071	Jul 28 01	18.35	0.1023	1.852	73.3
Ar072	Jul 28 01	18.35	0.0902	1.917	73.9
Ar073	Jul 28 01	18.35	0.0911	1.918	73.7
Ar074	Jul 28 01	18.35	0.0908	1.918	73.5
Ar075	Jul 28 01	18.35	0.0853	1.940	73.7
Ar076	Jul 28 01	18.35	0.0741	1.956	71.4
Ar077	Jul 28 01	18.35	0.0736	1.956	71.4
Ar078	Jul 28 01	18.35	0.0603	1.959	68.7
Ar079	Jul 28 01	18.35	0.0604	1.959	68.6
Ar080	Jul 28 01	18.35	0.0493	1.955	63.0
Ar081	Jul 28 01	18.35	0.0496	1.956	63.0
Ar082	Jul 28 01	18.35	0.0393	1.925	56.9
Ar083	Jul 28 01	18.35	0.0390	1.923	56.3
AR085	Aug 15 01	21.03	0.1293	2.068	73.1
AR086	Aug 15 01	21.03	0.1312	2.062	72.4
AR087	Aug 15 01	21.03	0.1305	2.049	72.8
AR088	Aug 15 01	21.03	0.1186	2.142	74.6
AR089	Aug 15 01	21.03	0.1186	2.145	74.1
AR090	Aug 15 01	21.03	0.1075	2.215	73.8
AR091	Aug 15 01	21.03	0.1084	2.214	73.7
AR092	Aug 15 01	21.03	0.0958	2.258	71.9
AR093	Aug 15 01	21.03	0.0955	2.258	72.0
AR094	Aug 15 01	21.03	0.0799	2.290	70.1
AR095	Aug 15 01	21.03	0.0793	2.292	69.9
AR096	Aug 15 01	21.03	0.0807	2.284	69.7
AR097	Aug 15 01	21.03	0.0632	2.288	65.8
AR098	Aug 15 01	21.03	0.0633	2.292	65.9
AR099	Aug 15 01	21.03	0.0459	2.240	57.6
AR101	Aug 19 01	10.06	0.0332	1.269	74.4
AR102	Aug 19 01	10.06	0.0332	1.268	73.8
AR103	Aug 19 01	15.23	0.0506	1.643	72.8
AR104	Aug 19 01	15.23	0.0506	1.644	71.8
AR105	Aug 19 01	18.35	0.0629	1.968	70.9
AR106	Aug 19 01	18.35	0.0627	1.966	70.6
AR107	Aug 19 01	18.35	0.0966	1.875	76.3

AR108	Aug 19 01	18.35	0.0962	1.878	76.4
AR109	Aug 19 01	15.23	0.0758	1.583	75.7
AR110	Aug 19 01	15.23	0.0759	1.584	75.6
AR111	Aug 19 01	10.06	0.0473	1.237	71.7
AR112	Aug 19 01	10.06	0.0474	1.236	73.5
AR113	Aug 19 01	10.06	0.0487	1.237	74.3
AR114	Aug 19 01	10.06	0.0477	1.238	74.9
AR115	Aug 19 01	15.23	0.0765	1.587	75.5
AR116	Aug 19 01	15.23	0.0764	1.587	75.4
AR117	Aug 19 01	18.35	0.0960	1.887	75.1
AR118	Aug 19 01	18.35	0.0964	1.885	74.9
AR119	Aug 19 01	15.23	0.0815	1.558	74.0
AR120	Aug 19 01	15.23	0.0811	1.559	74.4
AR121	Aug 19 01	18.35	0.1056	1.832	73.5
AR122	Aug 19 01	18.35	0.1043	1.832	73.1
AR123	Aug 19 01	18.35	0.0864	1.939	74.7
AR124	Aug 19 01	18.35	0.0866	1.938	74.5
AR125	Aug 19 01	15.23	0.0668	1.619	73.8
AR126	Aug 19 01	15.23	0.0678	1.617	74.2
AR127	Aug 27 01	15.23	0.0878	1.518	72.7
AR128	Aug 27 01	15.23	0.0927	1.515	71.7
AR129	Aug 27 01	15.23	0.0866	1.517	71.9
AR130	Aug 27 01	15.23	0.0792	1.563	74.7
AR131	Aug 27 01	15.23	0.0796	1.562	74.5
AR132	Aug 27 01	15.23	0.0544	1.634	74.0
AR133	Aug 27 01	15.23	0.0699	1.605	75.1
AR134	Aug 27 01	0.00	0.0673	1.615	74.9
AR135	Aug 27 01	15.23	0.0662	1.625	74.7
AR136	Aug 27 01	15.23	0.0451	1.649	69.4
AR137	Aug 27 01	15.23	0.0448	1.649	69.4
AR138	Aug 27 01	15.23	0.0249	1.608	53.1
AR139	Aug 27 01	18.35	0.0297	1.885	51.6
AR144	Aug 28 01	10.06	0.0545	1.212	72.7
AR145	Aug 28 01	10.06	0.0514	1.228	74.5
AR146	Aug 28 01	10.06	0.0399	1.259	75.5
AR147	Aug 28 01	10.06	0.0255	1.273	71.0
AR148	Aug 28 01	10.06	0.0177	1.263	56.3

APPENDIX A-4: GAS ANALYSIS TEST REPORTS

Normalized Percent Report

1

Sample Name: CU 2 010726

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10792.D

Injection Date : 7/26/2001 1:12:37 PM

Acq Operator : pmm

Acq. Method : ARGON.M

Analysis Method : C:\HPCHEM\1\METHODS\ARGON.M

RT (min)	Compound Name	Amount %
0.000	hydrogen	0.00
0.000	C7 Plus	0.00
2.561	carbon dioxide	0.76
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
7.705	argon-oxygen	92.84
0.000	1,3 butadiene	0.00
8.226	nitrogen	6.40
0.000	3-methyl-1-butene	0.00
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00

Instrument 1

Thu, 26. Jul. 2001

01:29:17 pm

Normalized Percent Report

1

Sample Name: CU 4 010726

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10793.D

Injection Date : 7/26/2001 1:30:26 PM

Acq Operator : pmm

Acq. Method : ARGON.M

Analysis Method : C:\HPCHEM\1\METHODS\ARGON.M

RT (min)	Compound Name	Amount
0.000	hydrogen	0.00
0.000	C7 Plus	0.00
2.569	carbon dioxide	0.02
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
7.699	argon-oxygen	99.95
0.000	1,3 butadiene	0.00
0.000	3-methyl-1-butene	0.00
8.241	nitrogen	0.03
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00

Instrument 1

Thu, 26. Jul. 2001

01:41:37 pm

Normalized Percent Report

1

Sample Name: CU 4 010726b

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10794.D

Injection Date : 7/26/2001 1:43:56 PM

Acq Operator : pmm

Acq. Method : ARGON.M

Analysis Method : C:\HPCHEM\1\METHODS\ARGON.M

=====		
RT	Compound Name	Amount
(min)		%

0.000	hydrogen	0.00
0.000	C7 Plus	0.00
2.569	carbon dioxide	0.02
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
7.713	argon-oxygen	99.95
0.000	1,3 butadiene	0.00
0.000	3-methyl-1-butene	0.00
8.250	nitrogen	0.04
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00

Instrument 1 Thu, 26. Jul. 2001 02:03:05 pm

Normalized Percent Report

1

Sample Name: CU 9 010726

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10795.D

Injection Date : 7/26/2001 2:04:22 PM

Acq Operator : pmm

Acq. Method : ARGON.M

Analysis Method : C:\HPCHEM\1\METHODS\ARGON.M

=====		
RT	Compound Name	Amount
(min)		%

0.000	hydrogen	0.00
0.000	C7 Plus	0.00
2.571	carbon dioxide	0.01
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
7.714	argon-oxygen	99.77
0.000	1,3 butadiene	0.00
0.000	3-methyl-1-butene	0.00
8.254	nitrogen	0.21
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00

Instrument 1 Thu, 26. Jul. 2001 02:37:28 pm

Normalized Percent Report

1

Sample Name: CU 6 010730

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10801.D

Injection Date : 7/30/2001 11:23:47 AM

Acq Operator : pmm

Acq. Method : CARLETON.M

Analysis Method : C:\HPCHEM\1\METHODS\ARGON.M

=====		
RT	Compound Name	Amount
(min)		%

0.000	hydrogen	0.00
0.000	C7 Plus	0.00
2.557	carbon dioxide	0.01
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
7.713	argon-oxygen	99.84
0.000	1,3 butadiene	0.00
0.000	3-methyl-1-butene	0.00
8.247	nitrogen	0.15
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00
=====		

Instrument 1 Mon, 30. Jul. 2001 11:40:34 am

Normalized Percent Report

1

Sample Name: CU 7 010730

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10803.D

Injection Date : 7/30/2001 11:44:52 AM

Acq Operator : pmm

Acq. Method : ARGON.M

Analysis Method : C:\HPCHEM\1\METHODS\CARLETON.M

=====			
RT	Compound Name	Amount	
(min)			

0.000	helium	0.00	
2.226	carbon dioxide	94.80	
0.000	C7 Plus	0.00	
0.000	methane	0.00	
0.000	ethane	0.00	
0.000	ethylene	0.00	
0.000	propane	0.00	
0.000	propylene	0.00	
0.000	acetylene	0.00	
0.000	isobutane	0.00	
0.000	n-butane	0.00	
0.000	hydrogen sulfide	0.00	
0.000	t-2 butene	0.00	
0.000	1-butene	0.00	
0.000	isobutylene	0.00	
0.000	c-2-butene	0.00	
0.000	isopentane	0.00	
0.000	n-pentane	0.00	
0.000	1,3 butadiene	0.00	
7.845	argon-oxygen	1.17	
0.000	3-methyl-1-butene	0.00	
8.286	nitrogen	4.03	
0.000	t-2-pentene	0.00	
0.000	2-methyl-2-butene	0.00	
0.000	1-pentene	0.00	
0.000	2-methyl-1-butene	0.00	
0.000	c-2-pentene	0.00	
0.000	carbon monoxide	0.00	
0.000	hexane PLUS	0.00	
Total		100.00	
=====			

Instrument 1 Mon, 30. Jul. 2001 11:57:17 am

Normalized Percent Report

1

Sample Name: CU 7 010730b

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10804.D

Injection Date : 7/30/2001 11:58:38 AM

Acq Operator : pmm

Acq. Method : CARLETON.M

Analysis Method : C:\HPCHEM\1\METHODS\CARLETON.M

=====		
RT	Compound Name	Amount
(min)		%

0.000	helium	0.00
2.195	carbon dioxide	94.93
0.000	C7 Plus	0.00
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
0.000	1,3 butadiene	0.00
7.842	argon-oxygen	1.14
0.000	3-methyl-1-butene	0.00
8.282	nitrogen	3.93
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00
=====		

Instrument 1 Mon, 30. Jul. 2001 00:21:07 pm

Normalized Percent Report

1

Sample Name: CU 8 010730

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10805.D

Injection Date : 7/30/2001 12:22:19 PM

Acq Operator : pmm

Acq. Method : ARGON.M

Analysis Method : C:\HPCHEM\1\METHODS\CARLETON.M

=====		
RT	Compound Name	Amount
(min)		%

0.000	helium	0.00
2.186	carbon dioxide	99.54
0.000	C7 Plus	0.00
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
0.000	1,3 butadiene	0.00
7.849	argon-oxygen	0.12
0.000	3-methyl-1-butene	0.00
8.300	nitrogen	0.35
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00
=====		

Instrument 1

Mon, 30. Jul. 2001

01:07:05 pm

Normalized Percent Report

1

Sample Name: CU 2 010817

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10853.D

Injection Date : 8/17/2001 11:42:53 AM

Acq Operator : pmm

Acq. Method : CARLETON.M

Analysis Method : C:\HPCHEM\1\METHODS\CARLETON.M

RT (min)	Compound Name	Amount %
0.000	helium	0.00
2.200	carbon dioxide	98.52
0.000	C7 Plus	0.00
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
0.000	1,3 butadiene	0.00
7.849	argon-oxygen	0.60
0.000	3-methyl-1-butene	0.00
8.297	nitrogen	0.88
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00

Instrument 1 Fri, 17. Aug. 2001 11:57:10 am

Normalized Percent Report

1

Sample Name: CU 2 010817b

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10854.D

Injection Date : 8/17/2001 11:58:44 AM

Acq Operator : pmm

Acq. Method : CARLETON.M

Analysis Method : C:\HPCHEM\1\METHODS\CARLETON.M

=====			
RT	Compound Name	Amount	
(min)		%	

0.000	helium	0.00	
2.227	carbon dioxide	98.52	
0.000	C7 Plus	0.00	
0.000	methane	0.00	
0.000	ethane	0.00	
0.000	ethylene	0.00	
0.000	propane	0.00	
0.000	propylene	0.00	
0.000	acetylene	0.00	
0.000	isobutane	0.00	
0.000	n-butane	0.00	
0.000	hydrogen sulfide	0.00	
0.000	t-2 butene	0.00	
0.000	1-butene	0.00	
0.000	isobutylene	0.00	
0.000	c-2-butene	0.00	
0.000	isopentane	0.00	
0.000	n-pentane	0.00	
0.000	1,3 butadiene	0.00	
7.851	argon-oxygen	0.59	
0.000	3-methyl-1-butene	0.00	
8.299	nitrogen	0.88	
0.000	t-2-pentene	0.00	
0.000	2-methyl-2-butene	0.00	
0.000	1-pentene	0.00	
0.000	2-methyl-1-butene	0.00	
0.000	c-2-pentene	0.00	
0.000	carbon monoxide	0.00	
0.000	hexane PLUS	0.00	
Total		100.00	
=====			

Instrument 1

Fri, 17. Aug. 2001

00:11:42 pm

Normalized Percent Report

1

Sample Name: CU 9 010817

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10855.D

Injection Date : 8/17/2001 12:12:53 PM

Acq Operator : pmm

Acq. Method : CARLETON.M

Analysis Method : C:\HPCHEM\1\METHODS\CARLETON.M

RT (min)	Compound Name	Amount %
0.000	helium	0.00
2.199	carbon dioxide	97.88
0.000	C7 Plus	0.00
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
0.000	1,3 butadiene	0.00
7.847	argon-oxygen	0.50
0.000	3-methyl-1-butene	0.00
8.293	nitrogen	1.62
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00

Instrument 1

Fri, 17. Aug. 2001

01:06:22 pm

Normalized Percent Report

1

Sample Name: CU #6 0830

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10916.D

Injection Date : 9/4/2001 11:32:44 AM

Acq Operator : hc

Acq. Method : ARGON.M

Analysis Method : C:\HPCHEM\1\METHODS\ARGON.M

RT (min)	Compound Name	Amount %
0.000	hydrogen	0.00
0.000	C7 Plus	0.00
0.000	carbon dioxide	0.00
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
7.709	argon-oxygen	99.67
0.000	1,3 butadiene	0.00
0.000	3-methyl-1-butene	0.00
8.240	nitrogen	0.33
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00

Instrument 1

Tue, 4. Sep. 2001

11:44:16 am

Normalized Percent Report

1

Sample Name: CU #6 0830B

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10915.D

Injection Date : 9/4/2001 11:15:33 AM

Acq Operator : hc

Acq. Method : ARGON.M

Analysis Method : C:\HPCHEM\1\METHODS\ARGON.M

RT (min)	Compound Name	Amount %
0.000	hydrogen	0.00
0.000	C7 Plus	0.00
2.561	carbon dioxide	0.01
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
7.696	argon-oxygen	99.64
0.000	1,3 butadiene	0.00
0.000	3-methyl-1-butene	0.00
8.235	nitrogen	0.35
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00

Instrument 1

Tue, 4. Sep. 2001

11:30:17 am

Normalized Percent Report

1

Sample Name: CU #8 0830

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10913.D

Injection Date : 9/4/2001 10:48:00 AM

Acq Operator : hc

Acq. Method : CARLETON.M

Analysis Method : C:\HPCHEM\1\METHODS\CARLETON.M

RT (min)	Compound Name	Amount %
0.000	helium	0.00
2.227	carbon dioxide	97.73
0.000	C7 Plus	0.00
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
0.000	1,3 butadiene	0.00
7.849	argon-oxygen	0.93
0.000	3-methyl-1-butene	0.00
8.294	nitrogen	1.34
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00

Instrument 1

Tue, 4. Sep. 2001

11:02:05 am

Normalized Percent Report

1

Sample Name: CU #8 0830B

Data file : C:\HPCHEM\1\DATA\CARLETON\SIG10914.D

Injection Date : 9/4/2001 10:59:56 AM

Acq Operator : hc

Acq. Method : CARLETON.M

Analysis Method : C:\HPCHEM\1\METHODS\CARLETON.M

=====		
RT	Compound Name	Amount
(min)		%

0.000	helium	0.00
2.197	carbon dioxide	97.75
0.000	C7 Plus	0.00
0.000	methane	0.00
0.000	ethane	0.00
0.000	ethylene	0.00
0.000	propane	0.00
0.000	propylene	0.00
0.000	acetylene	0.00
0.000	isobutane	0.00
0.000	n-butane	0.00
0.000	hydrogen sulfide	0.00
0.000	t-2 butene	0.00
0.000	1-butene	0.00
0.000	isobutylene	0.00
0.000	c-2-butene	0.00
0.000	isopentane	0.00
0.000	n-pentane	0.00
0.000	1,3 butadiene	0.00
7.847	argon-oxygen	0.93
0.000	3-methyl-1-butene	0.00
8.291	nitrogen	1.32
0.000	t-2-pentene	0.00
0.000	2-methyl-2-butene	0.00
0.000	1-pentene	0.00
0.000	2-methyl-1-butene	0.00
0.000	c-2-pentene	0.00
0.000	carbon monoxide	0.00
0.000	hexane PLUS	0.00
Total		100.00
=====		

Instrument 1

Tue, 4. Sep. 2001

11:13:17 am

APPENDIX B: EXPERIMENTAL PROCEDURE

The following is a list of instructions for use of the High Speed Wind Tunnel for use with the Rotating Test Stand. The instructions for operation of the High Speed Wind Tunnel are only briefly described, and most of the instructions presented here are specific to the Rotating Test Stand. For further information on operation of the High Speed Wind Tunnel, the reader is referred to the work of Jeffries (1994 & 2001).

PRE-TEST CHECKLIST

Before using the Rotating Test Stand, make sure that the following hardware is installed and functioning properly.

1. If it is not already installed, connect the Scani-Valve head used with the Rotating Test Rig to the Scani-Valve. The Scani-Valve head is located in the metal cabinet in the HSWT, along with the pressure transducers and other accessories specific to the Rotating Test Stand. Connect the Tygon tubes from the Scani-Valve head to the corresponding tubes from the Test Stand (the tubes are lettered A-G). Connect the PX-613 pressure transducer to the Scani-Valve. Normally, the Scani-Valve is moved close to the Rotating Test Rig, to minimize the lengths of Tygon tubing. Make sure that the Scani-Valve is turned on, and that the parallel port cable from LPT1 of the data acquisition computer is connected.
2. Connect the PX-613 pressure transducer to the 12 VDC power supply located near the test section of the HSWT. Connect the signal wires to the appropriate channel in the signal box, also located near the test section of the HSWT. The PX-613 transducer uses the same channel as the Kulite pressure transducer used in the HSWT, which must be disconnected.

3. Connect the PX-139 pressure transducer to the 5 VDC power supply located near the test section of the HSWT. Connect the signal wires to the channel in the control box marked "PX-139".
4. Unplug the power cord to the Test Section control valve actuator, and make sure that the power cord for the ejector-diffuser system control valve is connected.

STARTING THE HSWT COMPRESSOR

1. Make sure that the power supply to the HSWT control valves (switch on wall in HSWT control room) is in the OFF position. Also make sure that the dampers in the HSWT test room are open (switch on wall in HSWT test room, near mercury manometer).
2. Open the shop air valve for the control valves, located behind the compressed air tank nearest the air drier in the HSWT test room. SLOWLY open the feedback valve for the HSWT compressor. This valve is located on the ½ inch pipe connected to the compressed air tank nearest the air drier. Turn the air drier on, and make sure that there is no water leaking from the air drier. If there is water leaking from the air drier, contact a service representative.

The compressor is located in the gallery in the basement of the Minto Centre. Before turning the compressor on, make sure of the following:

1. Open the valve to the compressor aftercooler, located on a ½ inch copper pipe near the aftercooler, just enough that water begins to flow. Fully open the water supply valve, located on a vertical section of ½ inch pipe near the blue cylinder above the drive belt of the compressor.
2. Make sure that the drive belt of the compressor is clear of obstructions.

3. Turn on the power supply to the compressor. Press the “FAULT RESET” button on the power supply panel. Turn the power switch to the “AUTO” position to start the compressor.
4. Once the compressor is running, make sure that the oil pressure gage is in the middle of the range on the dial ($\sim 2.7 \text{ kg/cm}^2$)
5. Once the compressor has been running for a few minutes, check that there is water vapour visible in the drain near the compressor power supply panel. If no water vapour is visible, check to make sure that there is some flow of air through the Tygon hose in the drain. If there is no detectable flow of air, and the compressor is not in recirculating mode, open the aftercooler drain valve SLOWLY until air and/or water vapour flows into the drain (the aftercooler drain valve is the globe valve located on the wall opposite the instrument panel of the compressor). If the compressor is in recirculating mode, run a test in the HSWT to empty some of the air from the tanks, and check the drain again after the test, while the tanks are being filled. If there is still no air or water vapour, open the drain valve as described above.

NOTE: The aftercooler drain valve can become clogged after long periods of use, and should be opened approximately half way occasionally from time to time to clear the valve. If the aftercooler drain valve is not functioning properly, the compressor can be seriously damaged.

USING THE ROTATING TEST STAND

Once the compressor is running and the compressed air tanks are filling, return to the HSWT control room, and turn on the power supply to the control valves. The Rotating Test Stand is now ready for use. Before starting a test, make sure that the inlet and discharge throttle valves are set correctly. If the valves are not opened sufficiently, the

compressor will surge, and the shaft may over-speed, causing destruction of the turbocharger, and possibly damaging the instrumentation. If in doubt as to whether the valves are opened sufficiently, they should be opened fully for the first test, and throttled gradually in subsequent tests.

Make sure that the oil system is functioning properly by briefly turning on the power supply to the oil pump. The oil pressure should be set to between 5-8 psi by adjusting the relief valve at the pump discharge. Turn the cold water tap connected to the heat exchanger fully on.

Evacuating and Filling the Rotating Test Stand

If the test stand is to be used with an alternative working fluid, the following procedure should be used to evacuate and fill the circuit with the desired test gas:

1. Turn on the gas leak detector in the control room, and make sure that it is set to detect the gas currently being used. If, at any time, a significant concentration of gas in the control room is detected, close the filling bottle valves (if open), and leave the laboratory until it is safe to return.
2. Connect the shop air to the vacuum pump, and open the vacuum pump valve. Connect the argon or CO₂ tank to the valve on the inlet side of the circuit, upstream of the flow nozzle. Close the pressure regulator on the argon or CO₂ bottle by turning the regulator knob counter-clockwise until the spring tension is no longer felt.
3. Open the shop air valve to begin evacuating the test stand. It is recommended that the data acquisition program be run during this process to monitor the circuit pressure (see following section). The vacuum pump is normally capable of evacuating the circuit to approximately 3 psia (-12 psig).

4. Once the circuit pressure is stabilized, open the filling bottle valve fully. Turn the pressure regulator knob clockwise until the pressure reads approximately 1 psig. Close the vacuum pump valve and turn the shop air supply to the pump off.
5. Once the circuit pressure has returned to the ambient pressure, open the small valve below the vacuum pump to bleed the remaining air from the circuit. This usually takes approximately 3-5 minutes. During this time, monitor closely the concentration of gas in the laboratory using the gas leak detector.
6. Close the bleed valve. The filling bottle valve should be left open during tests to ensure that a high concentration of the desired test gas is maintained in the circuit. However, the valves can become very cold if the test stand is operated for extended periods of time. If frost begins to form on the valve or regulator, discontinue testing for a few minutes and close the filling bottle valve until the frost melts.
7. Gas sample bags can also be attached to the small valve next to the vacuum pump during a test if gas chromatography is to be performed.

Data Acquisition and Control

The following sections describe the use of the control and data acquisition programs. Before starting the data acquisition and control programs, make sure that the Hewlett-Packard control and data acquisition mainframe and extenders, the system voltmeter, and the control and data acquisition desktop computers are all powered on before continuing.

Data Acquisition Program:

The data acquisition program file is located in the c:\rrig\data directory of the data acquisition desktop computer. If the data acquisition program has not been used since the computer has been turned on, the data acquisition mainframe must first be initialized by entering the following command at the prompt:

```
comptest 0 .0005
```

This will initialize the data acquisition instrumentation, and set the sample rate to 0.5 milliseconds. If another sample rate is desired, change the switch “.0005” to the desired sample rate, in seconds. If the initialization sequence does not report any errors, the data acquisition can be begun by entering the following command at the prompt:

```
comptest 1 <PSamples> <TSamples> <output file>
```

where <PSamples> is the number of pressure samples to be taken for each channel, <TSamples> is the number of temperature samples to be taken for each channel, and <output file> is the name of the output file (no extension). The format of the output file is described in ReadMe.txt, located in the c:\rrig\tests directory.

Once the zero readings of the transducers are completed, there will be an output of the current shaft speed, discharge pressure, discharge temperature, and approximate flow rate on the screen. The information on the screen is updated approximately once per second.

To take a set of data, press the “m” key on the keyboard. During measurements, the information described above is not displayed to the screen. To quit the program, press the “q” key on the keyboard. It is usually necessary to hold the “q” key down for a few seconds before the program exits.

Control Program

The wind tunnel control program is located in the c:\rrig\hswtcont directory of the control desktop computer. The program is executed by entering "hswtcont" at the prompt.

If the control program has not been used since the computer has been turned on, the data acquisition mainframe must first be initialized by selecting "Initialize data acquisition" from the "Data Acquisition" pull-down menu. The program is then run by selecting "Start Wind Tunnel" from the "Run" pull-down menu. A dialog box appears from which the wind tunnel settings can be changed. Make sure that the "Ejector-Diffuser" check box is checked. Set the desired blowing pressure by modifying the value in the dialog box. The PID control gains can also be modified in this dialog box, but the default values should be accepted unless the user is encountering difficulties with the blowing pressure control. Exit the dialog box to begin the test. As soon as the control valve begins to open, and air is supplied to the turbine, switch on the oil pump to prevent failure of the bearings.

During the test, the blowing pressure can be adjusted to match as closely as possible the desired speed. The blowing pressure is increased or decreased by 0.01 bar by pressing "u" and "j" respectively, and the blowing pressure is increased or decreased by 0.1 bar by pressing "i" and "k" respectively.

To end the test, press the spacebar. This will gradually decrease the blowing pressure to approximately 1 bar. Once the compressor has slowed down sufficiently, press the spacebar again to stop the program. Once the compressor has stopped, turn off the power to the oil pump.

In an emergency, the program can be stopped by pressing the spacebar twice in rapid succession, but this is not recommended under normal circumstances, as it causes undue wear of the shaft bearings.

Maintenance

Although regular maintenance of the Rotating Test Rig is not normally required, users should check the following regularly:

1. Make sure there is sufficient oil in the reservoir, and that the oil is clean.
2. When the Rotating Test Stand is used in closed circuit operation for extended periods of time, oil can build up in the test circuit. The test circuit should be disassembled approximately after every 200 hours of use, and the pipes and heat exchanger should be thoroughly flushed with detergent, and rinsed and allowed to dry fully. This task requires approximately one full day to complete.

APPENDIX C: DETAILED DRAWINGS OF TEST EQUIPMENT

Figure C-1: Frame Sub-Assembly (Top View)

Figure C-2: Assembly Drawing of Test Setup (Open Cycle Arrangement)

ITEM	QTY.	DESCRIPTION
1	36"	3 IN. NOM. ID CLASS 150 STEEL PIPE, CLASS 150 FLANGE WELDED BOTH ENDS
2	6"	3 IN. NOM. ID CLASS 150 STEEL PIPE, CLASS 150 FLANGE WELDED BOTH ENDS
3	10"	2 IN. NOM. ID CLASS 150 STEEL PIPE, CLASS 150 FLANGE WELDED BOTH ENDS
4	20"	2 IN. NOM. ID CLASS 150 STEEL PIPE, CLASS 150 FLANGE WELDED BOTH ENDS
5	32"	2 IN. NOM. ID CLASS 150 STEEL PIPE, CLASS 150 FLANGE WELDED BOTH ENDS, WITH EXPANSION JOINT
6	1	CAPRI BRONZE BALL VALVE, 150 PSI STEAM-BASIC RATING, 9302 SCREWED END WITH WELDABLE 2" IRON/STEEL NIPPLE AT ONE END
7	2	CAPRI BRONZE BALL VALVE, 150 PSI STEAM-BASIC RATING, 9302 SCREWED END WITH FLANGE FITTING AT BOTH ENDS
8	1	90 DEG. ELBOW, LONG RADIUS, 3 IN. NOM. ID, CLASS 150 FLANGE WELDED ONE END
9	1	3" X 2" CONCENTRIC STANDARD WEIGHT REDUCER, CLASS 150 FLANGE WELDED AT 2" END, 3" END WELDED TO ITEM 8
10	3	STEEL GASKETS FOR 3" FLANGED CONNECTIONS
11	7	STEEL GASKETS FOR 2" FLANGED CONNECTIONS
12	1	1/8" PIPE PLUG (NOT SHOWN)
13	2	ADAPTER: 3/8" NPT EXT. TO 1/2" HOSE BARB
14	1	ADAPTER: 1/4" NPT EXT. TO 1/2" HOSE BARB
15	1	ADAPTER: 1/2" NPT EXT. TO 1/2" HOSE BARB
16	3	RISER CLAMPS ON 18" LONG 1" DIA. PIPE PEDESTAL FOR 3" PIPE SECTIONS (NOT SHOWN)
17	7	RISER CLAMPS ON 18" LONG 1" DIA. PIPE PEDESTAL FOR 2" PIPE SECTIONS (NOT SHOWN)
18	2	4' FLEXIBLE STEAM HOSE WITH 3" 300 CLASS 3" DIA. FLANGE ONE END, 150 CLASS 2" DIA. FLANGE ONE END (NOT SHOWN)
19	1	HEAT EXCHANGER, SECESPOL JAD-X-6.50
20	1	FRAME SUB-ASSEMBLY (FIGURE C-1)
21	1	ALBANY CEP-90 GEAR PUMP
22	1	COLE-PARMER VENTURI VACUUM PUMP P/N 78165-30
23	VAR.	PIPE AND HOSE FITTINGS FOR LUBRICATION SYSTEM AND FILLING
24	1	1/2" NYLON HOSE, APPROX. 20 FT. TOTAL LENGTH

Table C-1: List of Main Materials for Test Stand

Figure C-3: Inlet Adapter Flange

Figure C-4: Bellmouth (Used in Open Cycle Tests)

Figure C-5: Compressor Test Stand – Additional Views

Figure C-6: Detail View of Test Apparatus Showing Lubrication System

Figure C-7: Detail View of Discharge Pipe Traversing Mechanism

**Figure C-8: Detail View of Test Stand Showing Turbocharger
Assembly**

APPENDIX D-1: SOURCE CODES OF COMPUTER PROGRAMS

The source codes for the data acquisition and control programs and the meanline analysis, as well as an executable version of the meanline analysis program is included on a CD-ROM in the examination copies of this thesis. If this document does not include the CD-ROM, copies of the software can be ordered from Prof. S. Sjolander at the following address:

Rm. 3135 MacKenzie Bldg.

Carleton University

Ottawa, ON

CANADA

K1S 5B6

ssjoland@mae.carleton.ca

APPENDIX D-2: CFX-BLADEGEN+ COMPUTER FILES

The impeller geometry and CFD output files are included on a CD-ROM in the examination copies of this thesis. If this document does not include the CD-ROM, copies of the files can be ordered from Prof. S. Sjolander at the following address:

Rm. 3135 MacKenzie Bldg.

Carleton University

Ottawa, ON

CANADA

K1S 5B6

ssjoland@mae.carleton.ca

APPENDIX E: CURVE FIT EQUATIONS OF GAS PROPERTIES

The following are the curve fit equations used in determining the gas properties of CO₂ and moist air, and are taken from Irvine (1994):

AIR

$$c_p(T) = \sum_{n=0}^4 a_n T^n \quad \text{kJ/kgK}$$

$$a(0) = 1.03409 \quad a(1) = -284.887 \text{ E-6} \quad a(2) = 781.6818 \text{ E-9}$$

$$a(3) = -497.0786 \text{ E-12} \quad a(4) = 107.7024 \text{ E-15}$$

$$\gamma = \frac{c_p}{c_p - R}, \quad R = 0.287040 \text{ kJ/kgK}$$

WATER VAPOUR

$$c_p = 4.184(5.68 \times 10^{-8} T^2 - 2.34 \times 10^{-5} T + 0.4414) \quad \text{kJ/kgK}$$

$$\gamma = \frac{c_p}{c_p - R}, \quad R = 0.46189 \text{ kJ/kgK}$$

CARBON DIOXIDE

$$c_p(T) = \sum_{n=0}^6 a_n T^n \quad \text{kJ/kgK}$$

$$a(0) = 0.45386452$$

$$a(1) = 1.5334795 \text{ E-3}$$

$$a(2) = -4.195556 \text{ E-7}$$

$$a(3) = -1.871946 \text{ E-9}$$

$$a(4) = 2.862388 \text{ E-12}$$

$$a(5) = -1.6962 \text{ E-15}$$

$$a(6) = 3.717285 \text{ E-19}$$

APPENDIX F: EFFECT OF HEAT LOSS ON EFFICIENCY

The following is a brief summary of the estimated heat losses through the insulated compressor discharge pipe. The heat loss is estimated from Fourier's Law for one-dimensional conductive heat transfer in the radial direction through a cylinder (Holman, 1990):

$$q = \frac{2\pi L(T_w - T_\infty)k_{ins}}{\ln\left(\frac{r_2}{r_1}\right)} = \dot{m}c_p\Delta T$$

where:

q = Heat loss through the insulation (in W)

L = Length of insulated pipe ($L = 0.508$ m)

T_w = Inside temperature of insulation

T_∞ = Outside temperature of insulation (assumed to be 20°C)

k_{ins} = Thermal conductivity of insulation (0.042 W/m°C, according to supplier)

r_1 = Radius of inside of insulation (31.8 mm)

r_2 = Radius of outside of insulation (108.0 mm)

$\dot{m}$ = Mass flow rate of fluid in pipe (kg/s)

c_p = Specific heat at constant pressure of fluid in pipe

ΔT = Temperature change of fluid from compressor discharge to temperature measurement location (°C)

Although there is a change in temperature between the compressor discharge and the thermocouple sensor (T_w is not constant), the heat losses are calculated assuming a constant fluid temperature, since the change in fluid temperature along the pipe is in general negligible when compared to the difference between the fluid temperature and the ambient temperature. In addition, the thermal resistance of the pipe wall is also neglected, since it is much lower than the thermal resistance of the insulation. The outside temperature of the insulation is assumed to be 20°C, which is fairly consistent with observations during the tests, and the inside temperature of the insulation is assumed to be the same as the fluid temperature. These assumptions will tend to overestimate the heat loss through the insulation, and thus overestimate the changes in efficiency due to heat loss. This analysis however has been included to demonstrate that the heat loss through the insulation is in most cases negligible, and a high degree of accuracy is not required.

The above equation was used to estimate the change in temperature from the compressor discharge to the temperature measurement location (ΔT). The effect of the temperature change on the efficiency can then be estimated by examining its effect on the efficiency calculation (Eq. 2.7):

$$\Delta \eta = \frac{\Delta T}{T_{02} - T_{01}}$$

where T_{01} and T_{02} are the inlet and discharge stagnation temperatures of the compressor respectively.

The above analysis was performed at the highest and lowest mass flow rates for each gas at each speed line. In general, the highest discharge temperature was recorded at the lowest mass flow rate, and the lowest discharge temperature was recorded at the highest mass flow rate, for a given speed parameter, so the analysis summarized below

demonstrates the maximum and minimum effects of heat loss on the efficiency measurements.

Gas	Speed Parameter	Mass Flow Rate (kg/s)		Pipe Temperature (°C)		ΔT (°C)		Change in Efficiency (%)	
		Min.	Max.	Max.	Min.	Max.	Min.	Max.	Min.
Air									
	10.06	0.05	0.20	44	50	0.12	0.04	0.50	0.13
	15.23	0.10	0.18	68	83	0.16	0.07	0.25	0.14
	18.35	0.12	0.22	107	91	0.18	0.08	0.21	0.11
	21.03	0.18	0.22	126	119	0.15	0.11	0.14	0.11
CO ₂									
	10.06	0.04	0.19	48	39	0.21	0.03	0.75	0.16
	15.23	0.08	0.29	75	58	0.21	0.04	0.38	0.10
	18.35	0.09	0.36	100	75	0.27	0.05	0.33	0.08
	21.03	0.15	0.42	131	105	0.21	0.06	0.23	0.07
Argon									
	10.06	0.03	0.14	70	72	0.81	0.18	1.62	0.35
	15.23	0.08	0.21	90	98	0.41	0.18	0.61	0.23
	18.35	0.12	0.28	150	140	0.53	0.21	0.40	0.17
	21.03	0.29	0.34	225	185	1.11	0.24	0.54	0.14

The analysis shows that the effect of the heat loss through the insulation is most pronounced for argon. This is mainly because the discharge temperatures for this gas were the highest, but also partly because argon has the lowest specific heat (c_p) of the three gases.