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THE COMMISSIONING AND ENHANCEMENT OF A BLOW DOWN WIND TUNNEL

By

Michael Jeffries

B.ENG. (Mechanical Engineering)

**A thesis submitted to
The Faculty of Graduate Studies and Research
in partial fulfilment of the requirements
for the degree of**

**MASTER OF ENGINEERING
Aerospace**

**OTTAWA-CARLETON INSTITUTE FOR MECHANICAL
AND AEROSPACE ENGINEERING**

**DEPARTMENT OF MECHANICAL AND AEROSPACE ENGINEERING
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**"Commissioning and Enhancement of a
Blow Down Wind Tunnel"**

Submitted by Michael Jeffries, B.Eng.

in partial fulfilment of the requirements for
the degree of Master of Engineering

Thesis Supervisor S.A. Sjolander

Chair, Department of Mechanical and Aerospace Engineering

Carleton University

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Date

ABSTRACT

Recently, Carleton University acquired a high speed blow down wind tunnel. The tunnel was purchased from McMaster university. During its operation at McMaster, problems particularly with the pressure control system limited the wind tunnel's productivity.

This thesis describes the installation and commissioning of the wind tunnel at Carleton University. Improvements to the wind tunnel and a new pressure control system has greatly improved the operation of the tunnel. The new pressure control system provides stable and accurate pressure control throughout the operating range of the wind tunnel. In addition, increases in wind tunnel run times of over 50 percent have been obtained compared with the run times obtained at McMaster University for the equivalent operating points.

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NOMENCLATURE

A	Area
C_v	Valve Flow Coefficient
e(t)	Error Signal
G(t)	Control System Transfer Function
h	Height of Test Section Flow Passage
k_p	Proportional Constant
k_i	Integral Constant (see definition in Section 6.7)
m	Mass
M	Mach Number
n	Polytropic Index
P	Static Pressure
P_b	Back Pressure
P_o	Total Pressure
PI	Proportional-Integral
PID	Proportional-Integral-Derivative
R	Gas Constant
R_e	Reynolds Number
t	Time
T	Static Temperature
T_o	Total Temperature

T_i	Integral or Reset Time
T_D	Derivative Time
V	Volume
X	Pressure Drop Ratio
X_T	Terminal Pressure Drop Factor
y	Co-ordinates in spanwise direction
Y	Expansion Factor
γ	Coefficient of Specific Heats.
ϕ	Valve Position
μ	Viscosity
1-D	One Dimensional
2-D	Two Dimensional

Subscripts

1	Inlet
2	Outlet
i	Initial
in	Cascade Inlet
f	Final
max	Maximum
set	Set Point

T	Storage Tank
th	Cascade Throat
ts	Test Section

1.0 INTRODUCTION

During the past few years, extensive work has been done in commissioning a High Speed Blow Down Wind Tunnel at Carleton University. This chapter will discuss the motivation for acquiring the high speed facility and the scope of the work to be presented in this thesis.

1.1 Motivation For Acquiring a High Speed Wind Tunnel.

In recent years, extensive experimental studies into the performance of axial turbines have been performed at Carleton. Most of these studies have been performed using planar cascades of axial turbine blades mounted in low speed cascade wind tunnels. Although valuable data regarding axial turbine performance have been obtained from low speed testing, the need for studies into the effects of compressibility on component performance have prompted the acquisition of a high speed wind tunnel facility. This facility will be used in the future for both teaching and research in areas of transonic and supersonic aerodynamics. The first test section available for use with the wind tunnel is a planar cascade test section which will be used in the near future for research on transonic turbine flows.

1.2 Choice of Wind Tunnel Type.

To study experimentally the effects of compressibility on the performance of axial turbines and compressors, a wind tunnel capable of generating flow velocities up to and in

excess of the speed of sound is required.

In the most general terms, there are two types of wind tunnels, continuous flow and intermittent flow wind tunnels. In continuous flow wind tunnels, the flow is supplied continuously by a compressor or exhaustor. However, for test sections of reasonable size, the compressor power required to drive a continuous flow wind tunnel of this type is enormous.

In intermittent type wind tunnels such as a blow down wind tunnel, pressurized air is supplied at much lower flow rates, stored in tanks and then discharged through the test section intermittently. In this way, high test section flow rates require much less compressor power. Since the run times attainable with these types of tunnels are quite short, the facility is less productive and requires a high performance data acquisition and control system. However, in view of the much lower capital and operating costs, intermittent wind tunnels are the most feasible in a university environment.

1.3 History of the Carleton University Blow Down Wind Tunnel.

The wind tunnel discussed in this work is a blow down wind tunnel. The wind tunnel was originally designed and constructed at McMaster University under the supervision of Professor J.H.T. Wade. The construction and commissioning of the wind tunnel at McMaster is documented in detail in Stannard (1975). With the retirement of Professor Wade, use of the wind tunnel ceased and it was therefore purchased by Carleton. The first test section to

be used is a cascade test section which was originally designed and used by Orenda for the development of their engines for the AVRO Arrow project. It was then sent to the National Research Council after the termination the AVRO Arrow project. The test section has been obtained from the National Research Council by Carleton on long term loan.

1.4 Operation of a Blow Down Wind Tunnel.

Figure 1.1 shows a simplified schematic diagram of a typical blow down wind tunnel. The operation of a blow down wind tunnel involves filling a large volume or storage reservoir with high pressure air. With the storage reservoir filled, the control valve connected between the test section and the storage tanks is opened. The air stored in the reservoir is then discharged at high, controlled pressures into the test section. Since run times for these type of wind tunnels are typically of the order of 30 seconds, an essential part of the wind tunnel is the pressure control system. The control system must be able to quickly establish the desired blowing pressure in the test section and maintain it approximately constant during a run.

As a result of the high velocities in the test section and the expansion of the air in the storage reservoir, the static temperatures become very low. An air drying system is usually installed in these types of wind tunnels to lower the dew point of the air in the storage reservoir to prevent condensation in the test section during a run.

Figure 1.1: Schematic Diagram of a Typical Blow Down Wind Tunnel.

1.5 Scope of the Present Work.

This thesis describes the commissioning of the wind tunnel at Carleton and the design of an improved control system. In Chapters 3 and 4, a description of the wind tunnel and the instrumentation used during this work will be presented. During the design of the control system, an unsteady aerodynamic computer model of the wind tunnel was developed which is presented in Chapter 5. Chapter 6 discusses the design and performance of the pressure control system implemented to regulate the test section pressure. Chapter 7 discusses the results of experiments conducted to assess the overall aerodynamic performance of the wind tunnel. Finally, Chapter 8 presents the conclusions and summarizes further studies required to assess the performance of the facility. In addition, further modifications to enhance the performance of the facility are also discussed in Chapter 8.

2.0 REVIEW OF PREVIOUS RELATED WORK.

As mentioned in Chapter 1, the wind tunnel was originally designed and commissioned at McMaster University. This chapter will review the operation of the wind tunnel at McMaster and some of the experiments conducted using it.

2.1 Original Wind Tunnel Installation at McMaster University.

The original installation of the High Speed Wind Tunnel at McMaster is shown in Figure 2.1. The storage reservoir consisted of four large tanks each approximately 1 metre in diameter and 8.6 metres long. The test section used at McMaster was the cascade test section mentioned in Chapter 1. The test section was also obtained by McMaster on loan from the National Research Council and was returned when research using this tunnel ceased.

The air in the storage tanks was led to the test section through the piping also shown in Figure 2.1. The test section feed piping included two valves, a shut-off valve and a control valve. The shut-off valve provided a means of isolating the test section area from the tanks. In the event of a control valve failure, the wind tunnel run could be terminated by closing the shut-off valve. The shut-off valve was also used to initiate a wind tunnel run, as will be described in Section 2.3. The control valve was used to regulate blowing pressures in the test section under the control of the pressure control system.

Figure 2.1: High Speed Wind Tunnel Installation at McMaster University.

2.2 Previous Studies Using the High Speed Wind Tunnel.

During the operation of the wind tunnel at McMaster University, research conducted using the facility included studies of the performance of high turning-angle turbine blades. High turning-angle turbine blades are defined as blades with over 100 degrees of total turning. Some of the results of this work are presented in Stannard (1975). To study the performance of this type of turbine blade, a cascade consisting of 16 blades was mounted in the test section at the location shown in Figure 2.1. The pressure surface of one blade and the suction surface of the adjacent blade within the cascade were instrumented with static taps. Blade temperature measurements were obtained from thermocouples mounted on the surface of blades within the cascade. Measurements of the total pressures upstream of the cascade were conducted by traversing the inlet region with a Cobra type probe. The wake region downstream of the cascade was also traversed using a pitot tube. From these measurements, the losses in total pressure across the blade row and the wake distribution as a function of downstream distance were examined. The measurements were obtained at different cascade pressure ratios and inlet flow angles. Blade pressure measurements were taken using a Scanivalve system and a single pressure transducer. The output of the transducer was recorded using a pen chart recorder.

2.3 Shortcomings in the Original Wind Tunnel System.

A review of the design and operation of the wind tunnel at McMaster has identified

several shortcomings in the original wind tunnel system. This section will discuss these shortcomings.

2.3.1 Wind Tunnel Pressure Control System.

The most serious problems with the wind tunnel operation originated in the pressure control system. The control system was slow and as a result it was difficult to obtain and maintain pressure control during a run. The control system was an electro-pneumatic feedback control system based on a Foxboro Model M62U controller. The controller was an analog system containing proportional, integral and derivative components. The controller input signal was supplied by a Foxboro Model E11AH Pressure Transmitter, which measured the test section static pressure near the inlet of the test section. The controller output signal was sent to the control valve positioner. At the time of the wind tunnel operation at McMaster, high speed analog and digital control systems were available; however, these systems were expensive and beyond the means of a university research facility.

The control system was not fast enough to track and control the initial start-up transient. Therefore an elaborate start-up procedure utilizing both the shut-off valve and the control valve was developed. Before the start of the run, the control valve was manually positioned, with the shut-off valve closed, at the estimated position required to achieve the desired pressure. To initiate the run, the shut-off valve was ramped open at maximum rate. In effect, the start-up transient was controlled under feed-forward control. The control

system, which operated using feedback of the test section pressure, was activated a specified time after the opening of the shut-off valve. The time delay allowed the test section pressure to rise close to the desired pressure, reducing the pressure error signal to a value which could be handled by the control system. The start-up procedure required some degree of trial and error to obtain the correct initial control valve setting for different cascades and test section blowing pressures. This made obtaining the desired test section blowing pressure somewhat difficult and required a great deal of patience on behalf of the operator.

In addition to problems in obtaining pressure control, problems in maintaining control also occurred. It was found that the control system became unstable at high valve positions. Figure 2.2 shows a test section pressure trace where the control system becomes unstable at valve positions greater than 90 percent. This was a result of the non-linear flow characteristics of the control valve. The mass flow gain of the control valve became too large at high valve positions. This instability resulted in wind tunnel runs having to be terminated at valve positions of approximately 90 percent. As can be seen in Figure 2.2, if pressure control could be maintained at high valve positions, a considerable increase in run times could be obtained since there was still considerable pressure remaining in the tanks.

2.3.2 Shut-Off Valve.

In addition to the problems with control identified in the previous section, there were also design flaws identified with the wind tunnel system. One flaw was the selection of a 4

Figure 2.2: Test Section Pressure Trace Showing Control System Instability at High Valve Positions. Taken From Stannard (1975).

inch butterfly valve for the shut-off valve. The control valve was 4 inch ball valve which was located downstream of the shut-off valve, as shown in Figure 2.1. Due to the design of the valves, under the same conditions, a butterfly valve will have approximately half the mass flow capacity compared with the same size of ball valve. Therefore, the butterfly valve upstream of the control valve resulted in a significant pressure drop across the shut-off valve. This reduced the upstream pressure seen by the control valve and therefore reduced the possible run times for a given test section blowing pressure.

2.3.3 Storage Tank System.

Finally, there was a significant problem identified with the storage tank system. Figure 2.3 shows the tank configuration as installed at McMaster University. From the figure, it can be seen that the tanks are connected in series. It can also be seen that there are actually two sets of identical storage tanks. These two sets of tanks were apparently acquired at different times, with the second set being added to increase wind tunnel run times. While each set is connected by two 8 inch flanged pipe connections, the pairs are connected by a single 4 inch flanged connection. This led to a significant pressure drop between the two sets of tanks and prevented the air in the upstream set of tanks from being fully utilized during a wind tunnel run.

Figure 2.3: Storage Tank Arrangement at McMaster University.

2.4 Summary of Literature Review.

The shortcomings presented in this chapter suggest that significant improvements in wind tunnel performance could be obtained by modifying the existing wind tunnel system and designing a new pressure control system. Therefore, in addition to discussing the commissioning of the wind tunnel at Carleton, the following chapters will also discuss the modifications made to the wind tunnel system.

3.0 THE HIGH SPEED WIND TUNNEL.

This chapter discusses the installation of the High Speed Wind Tunnel at Carleton University. The modifications made to the existing wind tunnel system which improved the wind tunnel performance are also discussed.

3.1 General Description of the Wind Tunnel Installation.

The High Speed Wind Tunnel is located in the High Speed Aerodynamics Laboratory in Room 4060 of the Minto Centre. Figure 3.1 shows a schematic diagram of the components of the wind tunnel which are located in that room. The compressor which supplies the air to the storage tanks is located in the basement of the Minto Centre. Compared with Figure 2.1, the wind tunnel installation is similar to that at McMaster, however several modifications including the replacement of the shut-off valve and air dryer and the addition of the storage tank exit piping can be seen.

3.2 Modifications to the High Speed Wind Tunnel.

Some of the shortcomings of the original wind tunnel system were noted in Chapter 2. As a result, several major modifications were made to the wind tunnel during commissioning at Carleton.

Figure 3.1: High Speed Wind Tunnel Installation at Carleton University.

As discussed in Section 2.3.3, there was a problem with the original storage tank arrangement at McMaster. The original storage tank arrangement caused a significant pressure drop between the two sets of tanks. To solve this problem, a new exit piping arrangement was added to the system. Figure 3.2 shows the new tank arrangement with the storage tank exit piping in place. The exit piping allows the two sets of storage tanks to supply air roughly equally. In effect, the storage reservoir now consists of two sets of storage tanks in parallel.

The second modification to the wind tunnel system was the replacement of the shut-off valve. The original shut-off valve obtained from McMaster was a Foxboro Model M3U, 4 inch butterfly valve. There were two problems identified with this valve. The first, and probably most serious, was that even fully closed the shut-off valve leaked significantly. Therefore, the valve did not provide complete storage tank isolation and in the event of a control valve failure, the wind tunnel could not be completely shut down. Secondly there was a significant pressure loss across the fully open shut-off valve, particularly at high mass flows corresponding to high test section blowing pressures. Figure 3.3 compares the uncontrolled test section pressure traces with the shut-off valve installed and removed. These measurements were obtained by ramping the control valve fully open at maximum rate and measuring the test section static pressure at the inlet of the test section. The instrumentation used and the precise locations of the measurements are discussed in Chapter 4. From the figure, it can be seen that the presence of the shut-off valve becomes quite significant at high test section pressures. Since the shut-off valve is located upstream of the control valve, the

TOP VIEW

FRONT VIEW

Figure 3.2: Improved Storage Tank Arrangement with Storage Tank Exit Piping.

Figure 3.3: Effects of the Shut-Off Valves on Uncontrolled Test Section Pressures.

pressure loss across the shut-off valve reduces the maximum blowing pressures which can be achieved in the test section or alternatively, shortens the run time available at a given test section blowing pressure. The problem was solved by replacing the existing shut-off valve with a valve of substantially larger size. The new shut-off valve is a Neles-Jamesbury Model 815L, 6 inch butterfly valve. The uncontrolled test section pressure trace with the new valve installed is also presented in Figure 3.3. As seen in the figure, the presence of the new valve has negligible effect on the test section blowing pressures.

As mentioned in Chapter 1, air drying systems are usually installed on blow down wind tunnels to reduce the dew point of the air in order to prevent condensation in the test section during a run. Another significant modification made to the wind tunnel was the replacement of the air drying system. As will be discussed in Section 3.7, it was found that the existing Van-Air Dryer which was obtained in the purchase of the wind tunnel was not adequate for this application. Therefore, the Van-Air system has been replaced by a Comp-Air Kellogg System which provides considerably lower dew points.

3.3 The Wind Tunnel Valves.

As mentioned in the previous section, the new shut-off valve is a Neles-Jamesbury Model 815L, 6 inch butterfly valve. This valve is positioned using a Neles-Jamesbury Model QP3, pneumatic actuator. Since the shut-off valve has only two desired positions, open and closed, the air supply to the actuator is controlled by a solenoid valve. The air supply for the

shut-off valve is approximately 100 psig and is obtained from the shop air lines located in the laboratory.

The control valve is a Foxboro Model V9000, 4 inch ball valve. The control valve is actuated using a Foxboro Model P110 pneumatic actuator. The control valve actuator is controlled by a Neles-Jamesbury Model NE700, electro-pneumatic positioner. The positioner requires an input signal of between 4 and 20 mA, where 4 mA corresponds to the closed position and 20 mA to the fully open valve position. As will be discussed in Chapter 4, the positioner input signal is supplied by the control system. The output of the positioner is a pneumatic pressure between 3 and 15 psig linearly scaled based on the input signal. The output pressure from the positioner is sent to the valve actuator which positions the valve. The input air supply for the control valve positioner is also obtained from the shop air line located in the laboratory.

3.4 The Cascade Test Section.

At present, the only available test section for the High Speed Wind Tunnel is the cascade test section discussed in Chapter 1. Figure 3.4 shows a schematic diagram of the test section. The upper wall of the test section is reinforced by external ribs. The ribs supply the structural rigidity required when the test section is pressurized during a wind tunnel run. The internal components of the test section are also shown in Figure 3.4. These components include the turntable, the inlet side walls, the sliding walls and the cascade inlet side walls.

Figure 3.4: Cascade Test Section.

The turntable can be rotated to vary the angle of the cascade with respect to the inlet flow. The cascade inlet side walls are attached to the turntable by a parallelogram linkage. This linkage allows the cascade inlet side walls to adjust position with the turntable angle such that the side walls remain parallel for all turntable angles. The sliding walls move in the test section lengthwise direction to maintain the seal along the edge of the cascade inlet side walls. The sliding walls are kept in place by pneumatic pistons which keep the walls firmly sealed to the cascade inlet side walls. The test section inlet walls are rigidly fixed to the lower test section wall.

With the upper wall attached, the height of the test section flow region is approximately 2.4 inches. The width of the test section inlet is approximately 18.5 inches. The turntable is capable of rotating from a minimum angle of 0 degrees to a maximum angle of approximately 65 degrees. A turntable angle of 0 degrees corresponds to the inlet flow being perpendicular to the cascade. Since the cascade inlet side walls adjust position with the angle of the turntable, the cascade inlet area varies from a maximum width of approximately 11 inches to a minimum of approximately 4.75 inches. Figure 3.5 shows the variation in cascade inlet flow area with cascade angle.

The test section also has provisions for boundary layer bleed. The bleed slots are located approximately 15 inches upstream of the centre of the turntable on both the test section lower and upper walls. At present, the boundary layer bleed slots are blocked by

Figure 3.5: Cascade Inlet Area as a Function of Cascade Angle.

solid plates. The plates have been machined to fit into the bleed slots in order to provide a smooth flow surface inside the test section.

3.5 The Calibration Cascade.

During the work presented in this thesis, a cascade consisting of a row of cylinders was used. Figure 3.6 shows the calibration cascade, which was also obtained from McMaster University. The calibration cascade was designed to be used during wind tunnel performance and control system testing. During control system testing, rapid pressure transients and large pressure fluctuations can easily occur and damage precision machined blade rows. The calibration cascade provides a blockage to the test section outlet area equivalent to a cascade blade row without having to use an actual blade row. Since the present work focused on the control system and performance of the wind tunnel as a whole, the calibration cascade was used exclusively.

The calibration cascade consists of a row of 15 cylinders arranged as shown in Figure 3.6. Each cylinder is 0.5625 inches in diameter and 2.375 inches in exposed height. The cylinders are spaced approximately 0.825 inches apart centreline to centreline.

3.6 Compressor Installation.

The compressor is located in Room 1080 of the Minto Centre. It is a Broom Wade

Figure 3.6: Calibration Cascade.

Model VC500, two-stage, intercooled, positive displacement compressor. The compressor is driven by a 100 HP AC, 3 phase electric motor. Figure 3.7 shows the layout of the compressor equipment in Room 1080. To reduce the final delivery temperature of the air in the storage tanks, the air leaving the compressor is passed through a reverse flow aftercooler. The air is then passed through a moisture separator to remove excess moisture. The temperature of the air delivered to the storage tanks is approximately 300 K.

The final storage tank pressure is controlled by the delivery pressure control valve connected to the compressor on the air inlet side. This valve monitors the storage tank pressure and closes when the storage tank pressure reaches the preset value, thus shutting off the supply of air to the compressor. The compressor idles until the storage tank pressure drops below about 90 percent of the preset value at which point the delivery pressure control valve opens allowing air into the compressor. Although delivery pressures up to approximately 120 psig are obtainable from the compressor, for delivery pressures above approximately 100 psig the compressor drive motor exceeds its input current service rating. Therefore the final storage tank pressure has been set to approximately 100 psig.

The compressor is rated to deliver approximately 500 S.C.F.M. with a 100 HP motor installed. Therefore, the mass flow rate of the compressor is approximately 17 kilograms per minute. For the original configuration, the pumping times required to fill the tanks from empty to approximately 100 psig was about 15 minutes. However, due to the installation of

Figure 3.7: High Speed Wind Tunnel Compressor Installation.

the new dryer system, which will be discussed in the next section, the pumping times increased to approximately 20 minutes.

3.7 Wind Tunnel Air Drying System.

The purpose of the Air Drying System is to reduce the dew point of the air delivered to the storage tanks. Since the air is being accelerated to high velocity through the test section, the static temperature of the air in the test section becomes low. In addition, the air in the tanks cools as it expands, thus effectively reducing the total temperature in the test section during a run. If the static temperature in the test section falls below the dew point of the air, condensation will form. This is undesirable since condensation in the air within the test section results in a two phase flow being delivered to the blade row. In addition, condensation shocks can also occur in the test section and blade row.

The original air dryer obtained from McMaster was a Van-Air Desiccant Dryer. The dryer operated using a deliquescent drying process where the moist air was passed through a bed of desiccant. The some of moisture in the air was absorbed into the desiccant. It was almost immediately determined that this air drying system was not suitable for this application. During initial testing, large amounts of vapour were seen at the test section outlet. In addition, ice formed on the downstream side of the cylinders in the calibration cascade. Using dew point testing equipment on loan from Comp-Air Canada, the dew point of the air in the tanks was found to be approximately 5 °C. Based on information obtained

from the manufacturer, the dryer is capable of reducing the relative humidity of the air down to approximately 55 percent. Therefore, the dryer was in fact operating to its specifications or possibly beyond. However, this is clearly not sufficient drying capability for this wind tunnel and as a result the dryer has been replaced.

The new air drying system is Comp-Air Keillogg Model CDH520, Regenerative Desiccant Dryer. This system works based on the same principle as the Van-Air system, except that the desiccant is continuously regenerated or dried. From Figure 3.2, it can be seen that the air dryer consists of two tanks. While one tank is drying the air delivered from the compressor, the other tank is being regenerated. As mentioned in the previous section, the tank pumping times are increased by this dryer. This is related to the regeneration requirements. Approximately 15 percent of the air dried by one tank in the dryer is used to regenerate the other tank. Therefore, the air flow rate into the storage tanks using the Comp-Air System is approximately 15 percent less than the air flow rate using the Van-Air system.

According to its specifications, the Comp-Air dryer system is able to achieve dew points of approximately -40°C . During its initial use, the dew point of the storage tank air has been found to be approximately -40°C . Observations suggest that the condensation problem has been solved since no vapour is seen at the test section outlet.

4.0 WIND TUNNEL DATA ACQUISITION AND INSTRUMENTATION.

This chapter discusses the data acquisition and control equipment used during the commissioning of the High Speed Wind Tunnel. The instrumentation of various wind tunnel components, including the test section and the storage tanks, are also discussed.

4.1 Data Acquisition System.

The data acquisition system used for this work was a Hewlett Packard Model 3054A, Data Acquisition and Control System. Figure 4.1 shows a schematic diagram of the system. The data acquisition system consisted of a Hewlett Packard Model HP3497A Data Acquisition Unit and a Hewlett Packard Model HP3437A System Voltmeter. The system was connected to and controlled by a 80386 based IBM Compatible Computer via a Hewlett Packard Interface Bus (HPIB). Communications between the computer and the data acquisition system are done using a specialized set of HPIB command strings issued from the computer. The software drivers used to process the command strings are part of the data acquisition system drivers purchased from Hewlett Packard.

The data acquisition and control system software was written in Turbo Pascal by the author of this thesis. The software drivers obtained from Hewlett Packard were also written in Turbo Pascal and were incorporated into the program. The main portions of the data acquisition and control system software are shown in Appendix B.

Figure 4.1: Schematic Diagram of the Data Acquisition and Control Equipment.

The Data Acquisition Unit includes a Hewlett Packard Model 44421 20 Channel Guarded Acquisition Card which is connected at the back of the HP3497A. This card contained the input channels from which the transducer voltages were measured. The voltages on each channel were sent from the Data Acquisition Unit to the System Voltmeter. The voltmeter converted the analog voltages on each channel to a digital signal. The voltmeter contains a 3.5 digit successive approximation Analog to Digital (A/D) converter. The accuracy of the voltmeter A/D conversion is approximately 0.05 percent of the full scale of the voltmeter. The voltmeter has three full scale settings, 0.1, 1 and 10 volts. The output of the transducers which are discussed later in this chapter are 1 to 5 Volts DC. Therefore, the scale of the voltmeter was set to 10 volts. The digital voltages from the voltmeter were then sent to the computer via the HP-IB connection.

The data acquisition system was capable of sampling rates of approximately 100 samples per second on a single channel. However, during the majority of this work, 5 channels were typically measured. The locations of these measurements will be discussed later in this chapter. With 5 channels, the sampling rates were reduced to approximately 20 samples per second per channel. It will be shown in Chapter 6 that this system provided sufficient sampling rates to control the test section pressure. However, this system was on loan from Professor R.J. Kind and will not be a permanent portion of the wind tunnel system. In addition, for cascade blade testing, much higher sampling rates over many channels may be required. Therefore, it is likely the data acquisition system will be replaced with a more advanced system in the near future.

4.2 Control Valve Positioning.

Figure 4.1 also shows the setup used to position the control valve from the computer. As discussed in Chapter 3, the control valve is positioned using a current to air positioner and a pneumatic actuator. The positioner requires an input signal between 4 and 20 mA. To produce the control valve input signal, or positioning signal, from the computer, a Digital to Analog (D/A) converter was manufactured and purchased from ETC Electronics. The D/A converter is contained in a DB-25 connector shell and is attached to a modified, parallel I/O expansion card inside the computer. The expansion card was also obtained from ETC Electronics. The D/A converter is an 8 bit converter and is operated from the parallel port of the computer by sending an 8 bit number to the parallel port address. The analog output of the D/A converter is between 4 and 20 mA.

4.3 Wind Tunnel Instrumentation.

During commissioning, the wind tunnel was instrumented using pressure and temperature transducers to measure important quantities. To study the performance of the wind tunnel, there are several quantities of interest including storage tank pressure and temperature and both total and static pressures within the test section at various locations. From these measurements, other quantities including Mach numbers and Reynolds Numbers within the test section could be calculated. This section will discuss the instrumentation of the wind tunnel and the locations on the wind tunnel where measurements were obtained.

4.3.1 Pressure Transducers.

Separate transducers were used for each pressure measurement location on the wind tunnel because of the fast response required due to short wind tunnel run times. All pressure transducers used for this work were Omega, Model PX 613 transducers. These transducers are strain gage type transducers. Since the pressures in different parts of the wind tunnel varied greatly, transducers of different ranges were required. The ranges of the pressure transducers used in the test section are 0 to 60 psig and 0 to 100 psig. The range of the storage tank pressure transducer is 0 to 150 psig. The output of the transducers are 1 to 5 volts DC, linearly scaled to the input pressure range of the transducer. The accuracy of these transducers is approximately 0.4 percent of the full scale of the transducer. These transducers are sealed absolute pressure units which are factory calibrated to output 1 Volt at an absolute pressure of 14.7 psi.

The transducers were calibrated using a dead weight calibration system. The calibration curves for all transducers are shown in Appendix C. The transducers have good linearity characteristics with negligible hysteresis.

4.3.2 Locations of Test Section Pressure Measurements.

Figure 4.2 shows the locations in the test section where pressure measurements were obtained. In order to provide a pressure signal from which the pressure control system would

Figure 4.2: Locations of Test Section Pressure Measurements.

operate, a static tap was located in the test section inlet region. The static tap is located approximately 49 inches upstream of the cascade and is offset from the test section centreline 4.25 inches. The static tap is installed on the upper test section wall. The Mach numbers in the test section inlet region are low, typically less than approximately Mach 0.2, therefore static pressure measured in this region is very close to the test section total pressure or blowing pressure.

To measure total and static pressure in the cascade inlet region, a pitot tube and a static tap were installed approximately 15.5 inches upstream of the cascade as shown in Figure 4.2. The pitot tube is located along the test section centreline. The static tap is offset by approximately 1.5 inches from the centreline. Both the pitot tube and the static tap are attached to the upper test section wall. The pitot tube used is shown in Figure 4.3. The probe portion of the pitot tube was constructed using two small hypodermic tubes. The inner diameter of the smaller tube is approximately 0.028 inches. This was done to allow better spatial resolution for boundary layer measurements along walls of the test section. The pitot tube was installed for purposes of the wind tunnel commissioning and calibration. It will be removed when cascade measurements are to be taken since the wake caused by the tube will distort the flow into the cascade.

4.3.3 Storage Tank Measurements.

The pressure of the air in the storage tanks was measured using the storage tank

pressure transducer discussed in Section 4.3.1. The transducer is attached to one storage tank through a 1/4" NPT opening located on the side of the tank.

Since the temperature of the air in the storage tanks will decrease throughout a run as a result of the expansion, the decreasing temperature of the air in the tanks will affect temperatures in the test section, notably the effective total temperature of the flow. Therefore the storage tanks were instrumented with a temperature probe so that the decrease of the air temperature in the tanks could be measured during a run.

To measure the temperature of the air in the storage tanks, an Omega Model NB1-ICC-18G-24 temperature probe was used. The temperature probe contains a Type J Iron-Constantan thermocouple. The thermocouple leads are connected to a temperature transmitter attached to the probe. The temperature transmitter is Omega Model TX91-J2. The output of the temperature transmitter is between 4 and 20 mA based on a range of 255 to 366 K.

The temperature probe is connected through a 1/2" NPT opening on the side of one tank. The length of the probe is approximately 24 inches and measures the temperature close to the centreline of the tank. The temperature transmitter converts the thermocouple signal to an electric current signal. The temperature transmitter was calibrated at the manufacturer. The calibration curve for the temperature transmitter was reproduced and curve fitted and is shown in Appendix C.

All Dimensions
in Inches

Figure 4.3: Cascade Inlet Pitot Tube.

5.0 UNSTEADY AERODYNAMIC MODEL OF THE HIGH SPEED WIND TUNNEL.

To aid in the design of a pressure control system for the High Speed Wind Tunnel, a computer model of the wind tunnel was developed. The modelling of the wind tunnel was done using a lumped parameter approach which is described in this chapter. At a late stage in this analysis a paper by Tanner and Dietiker (1963) which used a similar approach to modelling a blow down wind tunnel was found.

5.1 Lumped Parameter Model.

The wind tunnel shown in Figure 3.1 consists of many components including storage tanks, a shut-off valve, a control valve, a test section and a network of piping. The network of piping includes elbows, tee-connections, contractions and expansions. All these components have, to some degree, an effect on the overall performance of the wind tunnel.

To simplify the analysis several assumptions were made. As discussed in Section 3.2, the pressure loss across the fully open shut-off valve is negligible during a wind tunnel run, therefore the shut-off valve was omitted from the model. The pressure losses resulting from components, such as elbows and contractions, in the piping upstream of the control valve were estimated and found to be negligible in comparison to the loss in total pressure across the control valve. The losses resulting from friction in the piping were also estimated and found to be negligible. Therefore, it was assumed that the total pressure upstream of the

control valve was equal to the storage tank pressure at any time during a run. The total temperature of the flow through the wind tunnel was assumed to be equal to the temperature of the air in the storage tanks. That is, it was assumed there was no heat transfer into the system through the pipes or the test section. To keep the analysis reasonably simple, the losses in total pressure as a result of the elbows and expansions between the control valve and test section were accounted for by assuming that the static pressure downstream of the control valve was equal to the total pressure in the test section. Therefore, the flow between the control valve and the test section was in effect modelled as a sudden expansion.

To model the wind tunnel as a whole, a lumped parameter approach was used. This approach involves modelling each component individually then assembling the components to form the complete model. Using the assumptions previously discussed, the wind tunnel was considered to consist of three main components, the storage tanks, the control valve and the test section. To model each of the actual components of the wind tunnel, simplified modelling components were developed. For example, the storage tanks were modelled as a large volume with mass flow out only. Similarly, the test section was modelled using a volume with both mass flow into and out of the volume region. In the following sections, models for each of these components will be developed individually and then assembled to form the complete wind tunnel model.

5.2 Storage Tank Component.

Considering the generalized storage tank component shown in Figure 5.1, as mass flow leaves the tank, the pressure and temperature decrease through polytropic expansion.

Assuming the pressure and temperature are uniform throughout the tank, the relationship for polytropic expansion is given as

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} . \quad (5.1)$$

In actuality, the pressure in the tank will be relatively uniform during the expansion however the temperature will not. As a result of heat transfer there will be a temperature variation in the tank. The temperature of the air close to the walls of the tank may be significantly higher than the temperature of the air toward the centre of the tank. For simplicity both pressure and temperature are assumed uniform within the tank

The variable n is called the Polytropic Index. For air, the value of the Polytropic Index ranges between 1 and 1.4. A polytropic index of 1.0 represents isothermal expansion while 1.4 represents adiabatic expansion. The polytropic index is a function of the heat transfer into the tank during the expansion.

Figure 5.1: Storage Tank Component.

For computer modelling, the differential form of equation (5.1) is more useful. Thus equation (5.1) is first written as

$$\ln\left(\frac{T_2}{T_1}\right) = \left(\frac{n-1}{n}\right) \ln\left(\frac{P_2}{P_1}\right). \quad (5.2)$$

Equation (5.2) is equivalent to

$$\int_1^2 \frac{dT}{T} = \frac{n-1}{n} \int_1^2 \frac{dP}{P}. \quad (5.3)$$

Therefore, in differential form

$$\frac{dT}{T} = \left(\frac{n-1}{n}\right) \frac{dP}{P}. \quad (5.4)$$

Now considering the equation of state

$$P = \frac{mRT}{V}. \quad (5.5)$$

Differentiating with respect to time,

$$\frac{dP}{dt} = \frac{RT}{V} \left(\frac{dm}{dt}\right) + \frac{mR}{V} \left(\frac{dT}{dt}\right). \quad (5.6)$$

Rearranging equation (5.4) produces

$$\frac{dT}{dP} = \frac{n-1}{n} \left(\frac{T}{P}\right). \quad (5.7)$$

Since

$$\frac{dT}{dt} = \left(\frac{dT}{dP} \right) \frac{dP}{dt}, \quad (5.8)$$

combining equations (5.7) and (5.8);

$$\frac{dT}{dt} = \frac{n-1}{n} \left(\frac{T}{P} \right) \frac{dP}{dt}. \quad (5.9)$$

Now combining equations (5.6) and (5.9),

$$\frac{dP}{dt} = \frac{RT}{V} \left(\frac{dm}{dt} \right) + m \left(\frac{RT}{PV} \right) \left(\frac{n-1}{n} \right) \frac{dP}{dt} \quad (5.10)$$

and since

$$m = \frac{PV}{RT}, \quad (5.11)$$

equation (5.10) reduces to

$$\frac{dP}{dt} = \frac{RT}{V} \left(\frac{dm}{dt} \right) + \left(\frac{n-1}{n} \right) \frac{dP}{dt}. \quad (5.12)$$

Rearranging equation (5.12) produces

$$\frac{1}{n} \left(\frac{dP}{dt} \right) = \frac{RT}{V} \left(\frac{dm}{dt} \right), \quad (5.13)$$

and since by definition

$$\dot{m} = \frac{dm}{dt}, \quad (5.14)$$

combining equations (5.13) and (5.14) and rearranging yields

$$\frac{dP}{dt} = \dot{m} \left(\frac{nRT}{V} \right). \quad (5.15)$$

Combining equations (5.9) and (5.15),

$$\frac{dT}{dt} = \left(\frac{n-1}{n} \right) \left(\frac{T}{P} \right) \left(\dot{m} \frac{nRT}{V} \right) \quad (5.16)$$

which reduces to

$$\frac{dT}{dt} = \dot{m} \left(\frac{(n-1)RT^2}{PV} \right). \quad (5.17)$$

Since mass flow is leaving the tank, both equations (5.15) and (5.17) must have a minus sign. Therefore the equations governing pressure and temperature changes in the tank become

$$\frac{dP}{dt} = -\dot{m} \left(\frac{nRT}{V} \right), \quad (5.18)$$

and

$$\frac{dT}{dt} = -\dot{m} \left(\frac{(n-1)RT^2}{PV} \right). \quad (5.19)$$

5.3 Volume Component.

The volume component is defined as a volume which can store and release mass with both an inlet and outlet mass flow. Figure 5.2 presents a general volume component. For this analysis both pressures and temperatures were assumed uniform throughout the volume. Assuming the velocities within the volume are negligible, the equation of state can be written in terms of total quantities. Differentiating the equation of state with respect to time;

$$\frac{dP_0}{dt} = \frac{RT_0}{V} \left(\frac{dm}{dt} \right) + \frac{mR}{V} \left(\frac{dT_0}{dt} \right). \quad (5.20)$$

Assuming there is no heat transfer into or out of the volume, equation (5.20) reduces to

$$\frac{dP_0}{dt} = \frac{RT_0}{V} \left(\frac{dm}{dt} \right). \quad (5.21)$$

The change in mass within the volume is defined as

$$\frac{dm}{dt} = \dot{m}_1 - \dot{m}_2. \quad (5.22)$$

Therefore, the changes in total pressure within the volume becomes

$$\frac{dP_0}{dt} = (\dot{m}_1 - \dot{m}_2) \frac{RT_0}{V}. \quad (5.23)$$

The mass flow out of the volume is a function of conditions within the volume, the outlet flow area (A_2) and the back pressure (P_b) assuming the exit is not choked. If the exit becomes choked the mass flow out of the volume becomes independent of the back pressure.

Figure 5.2: General Volume Component.

Assuming isentropic acceleration inside the volume up to the exit, the non-dimensional mass flow through the exit is

$$\frac{\dot{m}_2 \sqrt{T_0}}{P_0 A_2} = M_2 \sqrt{\frac{\gamma}{R} \left(\frac{2}{2 + (\gamma - 1) M_2^2} \right)^{\frac{\gamma + 1}{\gamma - 1}}}, \quad (5.24)$$

where M_2 is the Mach number in the exit plane. The Mach number is defined as

$$M_2 = \sqrt{\frac{2}{\gamma - 1} \left(\left(\frac{P_0}{P_b} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right)}. \quad (5.25)$$

Combining equations (5.24) and (5.25) and rearranging;

$$\dot{m}_2 = P_0 A_2 \sqrt{\frac{2\gamma}{RT_0(\gamma - 1)} \left(\left(\frac{P_b}{P_0} \right)^{\frac{2}{\gamma}} - \left(\frac{P_b}{P_0} \right)^{\frac{\gamma + 1}{\gamma}} \right)}. \quad (5.26)$$

However, if

$$\frac{P_b}{P_0} = \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}}, \quad (5.27)$$

the exit of the volume becomes choked, thus $M_2 = 1.0$. Therefore equation (5.24) reduces to

$$\frac{\dot{m}_2 \sqrt{T_0}}{P_0 A_2} = \sqrt{\frac{\gamma}{R} \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma + 1}{\gamma - 1}}}. \quad (5.28)$$

Rearranging equation (5.28);

$$\dot{m}_2 = P_0 A_2 \sqrt{\frac{\gamma}{RT_0} \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma + 1}{\gamma - 1}}}. \quad (5.29)$$

Therefore, there are two equations governing the changes in pressure within the volume depending on whether the outlet is choked or unchoked. Combining equations (5.23) and (5.26), the change in total pressure within the volume becomes

$$\frac{dP_0}{dt} = \left(\dot{m}_1 - P_0 A_2 \sqrt{\frac{2\gamma}{RT_0(\gamma-1)} \left(\left(\frac{P_b}{P_0} \right)^{\frac{2}{\gamma}} - \left(\frac{P_b}{P_0} \right)^{\frac{\gamma+1}{\gamma}} \right)} \right) \left(\frac{RT_0}{V} \right) \quad (5.30)$$

for the unchoked exit case. For the choked exit, combining equations (5.23) and (5.29) produces

$$\frac{dP_0}{dt} = \left(\dot{m}_1 - P_0 A_2 \sqrt{\frac{\gamma}{RT_0} \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{\gamma-1}}} \right) \left(\frac{RT_0}{V} \right). \quad (5.31)$$

5.3 Valve Analysis.

A valve can be considered as a variable area orifice, which acts as a throttling device resulting in a total pressure loss across the valve. A valve is used to regulate the flow through a system by adjusting the orifice area of the valve. As a result of the complex geometry of valves, the mass flow capacity of valves are usually determined empirically by valve manufacturers.

To simplify the analysis, consider a valve as a contraction followed by a sudden expansion as shown in Figure 5.3. In addition, assume incompressible, frictionless flow and

Figure 5.3: Idealized Diagram of a Valve.

that the upstream area is equal to the downstream area. The loss in total pressure across the valve can be defined as

$$P_{0_1} - P_{0_2} = K_L \left(\frac{1}{2} \rho U_1^2 \right) . \quad (5.32)$$

where K_L = Total Pressure Loss Coefficient.

Equation (5.32) can be rewritten as

$$P_1 + \frac{1}{2} \rho U_1^2 - P_2 + \frac{1}{2} \rho U_2^2 = K_L \left(\frac{1}{2} \rho U_1^2 \right) . \quad (5.33)$$

Assuming the downstream pressure is taken a sufficient distance downstream of the throat such that the flow is uniform, by continuity $U_1 = U_2$ since $A_1 = A_2$, therefore equation (5.33) becomes

$$P_1 - P_2 = K_L \left(\frac{1}{2} \rho U_1^2 \right) . \quad (5.34)$$

Rearranging equation (5.34);

$$U_1 = \sqrt{\frac{2(P_1 - P_2)}{\rho K_L}} . \quad (5.35)$$

From continuity;

$$Q = U_1 A_1 = U_2 A_2 . \quad (5.36)$$

Therefore, the flow through the throat is defined as

$$Q = A_1 \sqrt{\frac{2(P_1 - P_2)}{\rho K_L}} \quad (5.37)$$

From Neles-Jamesbury (1991), the ISA Valve Flow Equation for an incompressible flow is defined as

$$Q = C_v \sqrt{\frac{P_1 - P_2}{G}} \quad (5.38)$$

where G = Specific Gravity of the working fluid.
 C_v = Valve Flow Coefficient.

Comparing equation (5.38) with equation (5.37), it is seen that they are essentially the same equation with equation (5.38) containing empirical corrections to account for the actual valve flow characteristics. The term C_v is an experimentally determined value which takes into account the varying loss across the valve with valve orifice area. The flow coefficient is used to, in effect calibrate the valve flow rate through the valve for a given static pressure drop.

From Neles-Jamesbury (1991), the compressible ISA Valve Mass Flow Equation, in kg/hr, is defined as

$$\dot{m} = 94.8(C_d P_1 Y) \sqrt{\frac{XM}{T_1}}, \quad (5.39)$$

where

Y	= Expansion Factor
X	= Pressure Drop Ratio
M	= Molecular Weight (Air=28.97)
T ₁	= Upstream Static Temperature (K).

The pressure drop ratio is defined as

$$X = \frac{P_1 - P_2}{P_1}, \quad (5.40)$$

The expansion factor is an empirical relationship which is used to account for changes in density as a result of compressibility. The expansion factor is defined as

$$Y = 1 - \frac{X}{3X_T} \quad (5.41)$$

where

X _T	= Terminal Pressure Drop Ratio.
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The terminal pressure drop ratio represents the pressure drop ratio for a given valve position for which the valve becomes choked. If the valve chokes, the mass flow through the

valve becomes independent of changes in downstream pressure. Therefore the pressure drop ratio equals the terminal pressure drop ratio, that is $X=X_T$. The expansion factor relationship reduces to a constant

$$Y = \frac{2}{3} . \quad (5.42)$$

Equation (5.39) then reduces to

$$\dot{m} = 94.8 \left(\frac{2}{3} \right) (C_d P_1) \sqrt{\frac{M X_T}{T_1}} . \quad (5.43)$$

Both the flow coefficient and the terminal pressure drop ratio are determined experimentally as a function of valve opening for different valve designs by valve the manufacturers.

5.4 The High Speed Wind Tunnel Model.

The model of the high speed wind tunnel was created by assembling the appropriate components to represent the various parts of the actual wind tunnel. As discussed in Section 5.1, the cascade test section was modelled using a volume component. The storage tanks were modelled using a storage tank component with the combined volume of the actual tanks. Thus, the wind tunnel was modelled using three components as shown in Figure 5.4

The storage tanks might have been modelled using a volume component rather than a

Figure 5.4: Diagram of the High Speed Wind Tunnel Model.

storage tank component since during most operations of the tunnel the compressor will be running, therefore mass is entering the storage tanks. The mass flow rate into the tanks is approximately 0.25 kg/s. During a run, the mass flow rates out of the tanks ranges between approximately 2 to 5 kg/s. Therefore, the mass flow entering the tanks during a run is small compared with the mass flow leaving the tanks and thus was neglected.

As also discussed in Section 5.1, it was assumed that the total temperature throughout the wind tunnel was equal to the temperature of the air in the storage tanks at any time during a run. It was also assumed that the total pressure upstream of the control valve was equal to the storage tank pressure throughout the run. In addition, the static pressure downstream of the control valve was taken to be equal to the total pressure in the test section.

The wind tunnel was modelled using the equations derived in the previous sections for each component. The mass flow through the wind tunnel is determined by the control valve and thus can be calculated using the valve equation by specifying a valve position. However, specific data, including the flow coefficient and the terminal pressure drop ratio as a function of valve position for the type of valve being modelled, must be known to estimate the valve flow characteristics.

The actual control valve used in the wind tunnel is a Foxboro ball valve. However, limited information from Foxboro was available regarding this valve. Therefore the flow characteristics of the Foxboro valve were assumed to be similar to the flow characteristics

obtained from Neles-Jamesbury for a similar type of valve. It is worth noting that the control valve was sold as a Foxboro valve, however cast into the side of the valve body is the name Jamesbury. Therefore, it was assumed that the data regarding similar valves available from Neles-Jamesbury were a good approximation to the Foxboro characteristics. The curves for the flow coefficient and terminal pressure drop as a function of valve position for the control valve are shown in Appendix A. The control valve position versus time was measured using a position transducer by stroking the valve from closed to fully open and vice versa. It was found that the valve position versus time response was approximately linear for the valve moving from closed to fully open. The position versus time response for the valve moving from fully open to closed was found to be roughly exponential. Since during most wind tunnel operations the valve will be opening, the valve position versus time was assumed linear based on the time required to stroke the valve from closed to fully open.

By continuity the mass flow through the valve is equal to the mass flow leaving the storage tanks and entering the test section. Therefore, the storage tank pressure and temperature drop and the changes in test section pressure can be calculated. The Mach number and thus static pressure and temperature upstream of the control valve can be determined from the mass flow. In its present configuration, the wind tunnel exhausts to atmosphere. Therefore, the back pressure on the test section outlet is atmospheric throughout the run.

A computer program was written using the equations derived in this chapter to simulate the wind tunnel performance. Appendix A contains the specific data for the wind tunnel components which were used to model the wind tunnel. The wind tunnel model was used to design a pressure control system which is discussed in the next chapter.

5.5 Experimental Verification of the Wind Tunnel Model.

Using experimental data obtained from uncontrolled wind tunnel tests, the accuracy of the wind tunnel model was assessed. The data from wind tunnel tests was obtained by ramping the control valve fully open at maximum rate and recording the test section pressures with time.

As previously discussed, in its present configuration the test section outlet area is the effective flow area of the cascade installed in the test section. The effective flow area of the calibration cascade was estimated by measuring the upstream Mach number with the cascade choked. The results are presented in Chapter 7. The effective flow area of the calibration cascade at an angle of 0 degrees to the inlet flow was found to be approximately 12.9 in².

Figure 5.5 compares the uncontrolled wind tunnel response predicted using the model with the actual wind tunnel response for a test section outlet area of 12.9 in². From the figure it is seen that the model over predicts the test section pressures by a large amount. This is mainly a result of inaccuracies in the modelling of the control valve.

Figure 5.5: Comparison of Predicted Uncontrolled Wind Tunnel Response with Actual Wind Tunnel Response Using a Control Valve $C_v=550$.

The flow coefficients published by manufacturers are determined experimentally under an ideal set of conditions with constant pressure drop across the valve using incompressible laminar flow. However, as discussed in Neles-Jamesbury (1991), when installed in a system, the flow characteristics of the valve become a function of flow conditions within the system. Therefore, the installed valve characteristics may be significantly different than the characteristics quoted by the manufacturer.

To estimate the fully open flow coefficient of the control valve in the wind tunnel, the valve was ramped fully open. During the majority of the wind tunnel run the control valve is choked, thus rearranging the equation for a choked valve produces

$$C_v = \frac{3}{2} \left(\frac{\dot{m}}{94.8 P_1} \right) \sqrt{\frac{T_1}{X_1 M}} \quad (5.44)$$

The mass flow was estimated by measuring the total and static pressures in the cascade inlet region in the test section. It was assumed that the total pressure upstream of the control valve was equal to the storage tank pressure. It was also assumed that the total temperature throughout the wind tunnel was equal to the storage tank temperature at any time during the run. From the mass flow, the Mach number upstream of the valve was calculated. Using the Mach number upstream of the valve, the static pressure and temperature upstream of the control valve were calculated by assuming 2-D flow. The changes in tank temperature were estimated by measuring the changes in tank pressure and assuming adiabatic expansion. The value of the fully open terminal pressure drop ratio quoted by the manufacturer was used

for this analysis. It was assumed to be reasonably accurate since examining other models of ball valves produced by Neles-Jamesbury, the terminal pressure drop ratios are the same for all types of ball valves. Therefore, it appears that the terminal pressure drop is a function only of the valve position and design.

Figure 5.6 presents the measured fully open flow coefficients as a function of pressure drop across the valve. At lower values of the pressure drop coefficient (X), the data appears to be grouped in a series of lines. The reason for this is not entirely clear. The data shown in Figure 5.6 was taken during a single run therefore further measurements would be required to investigate the reason for the grouping. However, the trend in the data, as shown by the line in Figure 5.6, is reasonably clear. The measured values appear to be substantially less than the values quoted by the manufacturer at high pressure drops across the valve. As the pressure drop decreases, the flow coefficient approaches the manufacturer's value. This is a result of the flow approaching incompressible conditions for which the flow coefficients are determined by the manufacturer.

Figure 5.7 shows the predicted uncontrolled wind tunnel response using a value of 350 for the fully open flow coefficient for the control valve. From the figure, it is seen that much better agreement between experimental and predicted wind tunnel response has been obtained. The effect of the tank polytropic index was also examined and presented in Figure 5.7. As expected, the actual value of the polytropic index lies between 1 and 1.4. The figure suggests a tank polytropic index of approximately 1.2 is a good estimate.

Figure 5.6: Measured Valves of the Control Valve Fully Open Flow Coefficient.

Figure 5.7: Predicted Wind Tunnel Uncontrolled Response Using an Estimated Value of 350 for the Control Valve Flow Coefficient.

In the following chapter, the wind tunnel model will be used to design a pressure control system for the wind tunnel and to estimate the order of magnitude of the control system gains. The predictions of the controlled wind tunnel performance will then be compared to experimental data.

6.0 PRESSURE CONTROL SYSTEM

This chapter discusses the feedback control system designed to control the test section blowing pressure of the High Speed Wind Tunnel.

6.1 Purpose of the Pressure Control System.

A typical wind tunnel run can be divided into three parts, the start-up transient, the quasi-steady portion and the wind tunnel shut down. The quasi-steady portion of the run refers to the part of the run where the desired blowing pressure has been achieved and the mass flow into the test section is equal to the mass flow out of the test section.

The pressure control system therefore has three specific functions. The controller must first bring the test section pressure up to the specified blowing pressure at the start of the run. Once the specified blowing pressure has been established, the test section achieves a quasi-steady state. The control system must then maintain the blowing pressure approximately constant, as close to the desired blowing pressure as possible, for as long as there is sufficient air in the storage tanks. Once the pressure in the storage tanks becomes too low that pressure control cannot be maintained, the control system must shut down the tunnel in order to conserve the remaining air in the tanks. This minimizes the pumping time required to recharge the storage tanks between runs.

6.2 General Approach to Control.

For reasons which will be discussed in the following sections, feedback control was the best choice for the control system. Both the start-up transient and the quasi-steady portion of the wind tunnel run are controlled using feedback control.

6.2.1 FeedBack Control of the Start Up Transient.

There are many possible operating conditions, corresponding to different blowing pressures and test section outlet areas, over which the wind tunnel can operate. Feedback control was determined to be the most suitable control system to regulate the test section pressure during the start-up transient. In the present wind tunnel configuration described in Chapter 3, the test section outlet area is the effective throat area of the cascade blade row installed in the test section.

The start-up transient could have been controlled using a feed-forward type control system where for a specified blowing pressure, an initial control valve position is specified. The valve would then be opened to this position as quickly as possible. For a given desired blowing pressure, the valve position at the end of the start-up transient is related to the test section outlet area. The initial valve position would therefore have to be specified based on the desired blowing pressure and the test section outlet area. Since the test section outlet area, or more precisely the effective throat area of the cascade, is not clearly defined, the

initial valve position required to achieve a desired blowing pressure would have to be established through extensive calibration measurements. In addition, further calibration measurements of the initial valve position would have to be conducted for each new cascade.

It should also be noted that the storage tank pressure and hence the upstream pressure on the control valve, is not exactly the same for each run because of the crude compressor control system. In addition, to save time a run might be initiated before the tank pressure reaches its full value. Therefore, setting the control valve position to achieve a particular test section pressure would also need to be determined as a function of the initial storage tank pressure.

With feedback control, the control valve position is determined by the actual test section pressure in relation to the specified blowing pressure. While the test section pressure is below the set point the control system will continue to open the control valve. Calibration measurements are needed to establish suitable gains or range of gains for the feedback loop. However, such measurements are considerably less than those needed to define initial valve positions for all possible combinations of wind tunnel conditions.

The main advantage of open-loop or feed-forward control is that it would use the maximum rate at which the control valve can be opened. Thus it will, in principle, result in the shortest start-up transient. However, it was found that feedback control only lengthens the start-up transient by a relatively small and acceptable amount. The performance of the

feedback control system is discussed later in this chapter.

6.2.2 Feedback Control of the Quasi-Steady Portion of the Run.

Once the specified blowing pressure has been achieved at the end of the start-up transient, the pressure control system must maintain the test section pressure approximately constant. The obvious choice of control system is to use feedback control since, once again, the control system operates on the actual test section pressure. Therefore, disturbances and deviations of the test section pressure away from the specified pressure are directly sensed and corrected for by the control system.

6.2.3 Wind Tunnel Shut Down.

Once the valve position reaches approximately fully open, the control system should close the control valve as quickly as possible to conserve the remaining air in the storage tanks. Thus, during the shut-down transient the control valve is effectively under open-loop control.

6.3 The Wind Tunnel Block Diagram.

Figure 6.1 shows the block diagram representing the wind tunnel system under feedback control.

Figure 6.1: High Speed Wind Tunnel Block Diagram.

From the block diagram

$$\phi(t) = G(s)e(t) , \quad (6.1)$$

where $\phi(t)$ is the non-dimensional valve position. The error signal, $e(t)$, is the difference between the specified test section pressure and the actual test section pressure. Thus

$$e(t) = P_{set} - P_{ts} , \quad (6.2)$$

or in non-dimensional terms, the error signal is written

$$e(t) = \frac{P_{set} - P_{ts}}{P_{set}} . \quad (6.3)$$

The term $G(s)$ in equation (6.1) is the control system transfer function. This term dictates how the control system responds to the error signal. This term may contain proportional, integral and derivative terms, or even more elaborate control schemes to act on the error signal. The control system transfer function also contains gains which can be adjusted to vary the overall system performance and achieve specific performance characteristics. The specific contents of the control system transfer function and its gains will be discussed in later sections.

6.4 General Remarks Concerning Control System Gains.

There are several considerations which must be taken into account when selecting the control system gains. Since the air in the storage tanks is limited, to obtain maximum run times, the gains of the control system must be set such that the specified test section pressure

is obtained in the shortest possible time. This minimizes the amount of air used during start up. Therefore the gains of the controller must also be set such that there is minimal overshoot of the specified blowing pressure. This suggests that the gains of the control system should be set such that the wind tunnel system be slightly over damped, but not highly since the start-up transient time is increased by increased damping. The investigation of the specific values of the control system gains are discussed in Section 6.7.

6.5 Design of the Pressure Control System.

Using the wind tunnel model derived in Chapter 5, a suitable pressure control system was designed to regulate the test section blowing pressure. This section will present the predicted performance of the pressure control system using the wind tunnel model. The following section will present experimental verification of the performance of the pressure control system.

As already mentioned, there are many possible operating points for the wind tunnel. The discussion will focus on a single operating point. The magnitude of the control system gains were selected based on the considerations discussed in Section 6.4.

6.5.1 Proportional (P) Control.

The simplest possible control system would have only proportional control. The

control function is of the form

$$\phi(t) = k_p e(t) , \quad (6.4)$$

where k_p is the proportional constant.

With proportional control, the control valve position is proportional to the error signal. Therefore, the error signal will always be non-zero during the wind tunnel run. Thus there will always be a difference between the specified and the achieved blowing pressure. As a result, the specified blowing pressure must be set considerably higher than the actual desired blowing pressure.

Figure 6.2 shows the predicted test section pressure control using proportional control with a proportional gain of 2.3. From the figure it can be seen that there is a large error signal since the achieved test section pressure only reaches approximately 62 percent of the set-point blowing pressure. In theory, increasing the proportional gain will reduce the error signal, however for gains higher than approximately 2.3, the test section pressure tended to oscillate. The specified blowing pressure was set to achieve an actual test section pressure of approximately 2 bars absolute. Figure 6.2 shows that the desired test section pressure is reached only at the end of the start up transient. The pressure then gradually falls away from the desired blowing pressure as a result of the decreasing storage tank pressure. The decreasing storage tank pressure acts as a continuous disturbance to the wind tunnel system.

Figure 6.2: Predicted Test Section Pressure Control Using Proportional Control.

There are two shortcomings in the performance of the proportional control system. The specified blowing pressure must be set higher than the desired blowing pressure. This makes it difficult to achieve accurate test section pressure settings. Secondly, proportional control is not very good at handling the disturbance caused by decreasing storage tank pressure. Therefore, a control system which drives the error signal to zero and has better disturbance rejection characteristics is required.

6.5.2 Proportional-Integral (PI) Control.

Incorporating integral control into the control system transfer function produces

$$\phi(t) = k_p \left(e(t) + \frac{1}{T_i} \int_0^t e(t) dt \right), \quad (6.5)$$

where T_i is the Integral Time or Reset Time.

The integral term will attempt to drive the error signal to zero. This results since as long as the error signal is non-zero, the value of the integral will be changing with time, thus changing the valve position. The integral portion of the control system operates on past values of the error signal. At the start of the run, the dominant term in the control system is the proportional term since the error signal is large. This causes the valve to open and the test section pressure to rise. As the test section pressure increases toward the specified pressure, the error signal decreases toward zero. As the error signal decreases the integral term becomes more dominant, since the large error signals from the start of the run eventually

cause the value of the integral to become large. As the error signal tends to zero, the integral tends to a constant, non-zero value. Since the error signal is driven toward zero, the test section pressure will approach the specified blowing pressure.

As mentioned, the decreasing storage tank pressure acts as a continuous disturbance to the system. The integral portion of the control system also offers better disturbance rejection since as long as there is a non-zero error signal, the integral will attempt to drive the error to zero.

Figure 6.3 shows large improvements in pressure control as a result of using PI control. The achieved blowing pressure is very close to the set point pressure and the deviation of the blowing pressure away from the set point is greatly reduced. There is still a slight error signal in the test section pressure as a result of the decreasing storage tank pressure. However, the test section pressure is within approximately 1 percent of the set-point blowing pressure which is considered acceptable.

6.5.3 Proportional-Integral-Derivative (PID) Control.

For completeness, derivative control was incorporated into the control system. The control system function then becomes

$$\phi(t) = k_p \left(e(t) + \frac{1}{T_i} \int_0^t e(t) dt + T_d \left(\frac{de(t)}{dt} \right) \right), \quad (6.6)$$

Figure 6.3: Predicted Test Section Pressure Control Using Proportional-Integral Control.

where T_d is the Derivative Time.

By incorporating derivative control into the control system function, the control system attempts to drive the time derivative of the error signal to zero. The addition of derivative control provided negligible improvements in wind tunnel control over PI control. The reason is that over most of the run, the derivative of the error signal is small. As discussed in the next section PID control was tested experimentally. It was found that the derivative term caused the control system to become unstable.

6.6 Experimental Verification of the Wind Tunnel Pressure Control System.

The results of the previous section suggest that a PI or PID control system should provide acceptable test section pressure control. This section presents the results of experimental testing of both PI and PID control systems.

Figure 6.4 shows the test section pressure measured during a typical run using PI control. From the figure it can be seen that PI control provides reasonably good pressure control. The actual test section blowing pressure remains within approximately 2 percent of the specified pressure over most of the run. The start-up transient is approximately 4 seconds which also is acceptable.

During experimental testing of a PID control system it was found that the derivative

Figure 6.4: Controlled Test Section Pressure Using Proportional-Integral Control.

term caused the wind tunnel to become unstable. This was the result of the fluctuations in the instantaneous values of the test section pressure also shown in Figure 6.4. The unsteadiness in the instantaneous values of the test section pressure and thus the error signal caused the instantaneous values of the derivative to vary widely in both sign and magnitude. This caused large changes in valve position and as a result, the wind tunnel system became unstable. Therefore the derivative portion was not implemented into the final wind tunnel control system.

6.7 Experimental Investigation of the Control System Gains.

The pressure control system developed in Section 6.5 and tested in Section 6.6 demonstrated that acceptable pressure control could be achieved using Proportional-Integral (PI) control. Since there is a large number of possible wind tunnel operating points, the performance of the control system over range of operating points was examined.

To examine the performance, the control system was used to regulate the test section pressure over the range of pressures attainable for a given test section outlet area. Using the calibration cascade described in Chapter 3, Figure 6.5 shows the test section pressure traces for various test section blowing pressures with the cascade set at 65 degrees. The gains of the control system which provided acceptable pressure control were found experimentally.

Figure 6.5: Test Section Pressure Control at Various Blowing Pressures.

2 of/de 2

PM-1 3½"x4" PHOTOGRAPHIC MICROCOPY TARGET
NBS 1010a ANSI/ISO #2 EQUIVALENT

PRECISIONSM RESOLUTION TARGETS

The following values of the control system gains are suggested,

$$k_p \approx 1.0 \quad (6.7)$$

and defining

$$k_i = \frac{k_p}{T_i} \quad (6.8)$$

it was found that suitable values of k_i varied from 0.6 at low pressures to approximately 0.4 at high pressures (see Figure 6.5). For the test section blowing pressure of approximately 2.4 bars shown in Figure 6.5, there is some overshoot during the start-up transient. This curve is included to demonstrate what happens when the integral gain is set too high. Therefore, the integral gain should be slightly reduced below 0.4.

The test case presented in this section corresponds to only one test section outlet area. It was found by experiment that the gains presented in this section worked reasonably well for the same blowing pressures at different test section outlet areas.

At blowing pressures below approximately 1.2 bars it was found that the test section pressure tended to cycle about the set point. This was a result of the mass flow in the test section becoming too low. The incremental changes in the control valve position were too large compared to the changes required to accurately regulate the test section pressure. That is, the smallest possible change in valve position caused changes in mass flow into the test section which were too large. It also was found that as the pressure upstream of the control valve decreased, the pressure control at low blowing pressures improved. This was a result of

the decreased mass flow gain of the control valve since for a given change in valve position, the change in mass flow is smaller at lower upstream pressures.

This problem could be solved by using a smaller size control valve. The wind tunnel could be modified by adding a second control valve which would be in parallel to the present control valve. For high blowing pressure runs, the present valve would be used for pressure control and for low pressure runs, below approximately 1.2 bars absolute, the smaller valve would be used.

At high blowing pressures, the test section pressure was somewhat lower than the set point pressure. This is the result of the larger disturbance caused by the more rapidly decreasing storage tank pressure. The pressure control system was unable to completely drive the error to zero. However, the difference between the specified blowing pressure and the controlled blowing pressure was still acceptably small and could be compensated for by slightly increasing the specified pressure.

The start up transient can be seen in Figure 6.5 to be approximately 4 seconds for all blowing pressures. The minimum time taken to move the control valve from closed to fully open is approximately 3.5 seconds under open-loop control. Therefore the increase in the start-up time as a result of using feedback control during the start-up transient is acceptably small.

The gains suggested in this section are intended as starting points. When new cascades are developed, a few test runs should be performed to fine tune the gains of the control system. The gains presented in this sections should provide reasonably good initial estimates.

6.8 Discussion of the Wind Tunnel Model.

Figure 6.6 compares the controlled wind tunnel performance predicted using the wind tunnel model to the actual performance obtained from experiment. From the figure, it is seen that the predicted wind tunnel performance shows reasonably good agreement with the actual wind tunnel performance. However, it appears that the model somewhat under predicts the wind tunnel run times. This is a result of the inaccuracies in modelling the flow characteristics of the control valve. In the model, the flow coefficient of the control valve was assumed to be constant for a given valve position. A value of 350 was used for the control valve fully open flow coefficient since it provided good estimates of the uncontrolled wind tunnel response. However, as shown in Figure 5.6, the valve flow coefficient somewhat varies with the pressure ratio across the valve. The inaccuracies in the wind tunnel model could be reduced by further investigating the actual installed flow coefficients of the control valve as a function of both valve position and pressure drop ratio. In any case, the model does predict reasonable wind tunnel run times.

Figure 6.6: Comparison of the Predicted Test Section Pressure Control with the Actual Pressure Control.

The main reason of modelling the wind tunnel was to obtain a first guess at a possible control system and the order of magnitude of the control system gains. The control gains used in the model were similar to those found experimentally. In addition, the model also provided some insight into the sampling rates required to obtain reasonable pressure control. Using the model, it was found that approximately 100 samples per second were required to obtain pressure control. This was confirmed by experiment, however, in actuality as few as 20 samples per second were found to provide acceptable pressure control.

In conclusion, the wind tunnel model served the purpose for which it was developed. Future wind tunnel modifications and configurations, such as the addition of a settling chamber and an aeronautical test section which have been proposed, could be first modelled to assess the overall performance of the wind tunnel and estimate the values of the control system gains.

7.0 WIND TUNNEL PERFORMANCE.

The purpose of this chapter is to present the aerodynamic performance of the wind tunnel.

7.1 General Remarks Regarding the Wind Tunnel Performance Results.

Some of the results which will be presented in this chapter are specific to the cascade used during the wind tunnel testing. As discussed in Chapter 3, the calibration cascade, consisting of a row of circular cylinders, was used for the work done in this thesis. Quantities such as wind tunnel run times and cascade inlet Mach numbers are functions of the effective cascade throat area. Therefore, future cascades which will be tested using this wind tunnel will produce, to some degree, different wind tunnel run times and cascade inlet Mach numbers. For example, cascades of much larger effective throat areas than the calibration cascade will significantly reduce run times for a given test section pressure since larger mass flows are required to maintain a given test section pressure. Thus, the wind tunnel run times presented in the next section represent only typical values of run times for a given test section pressure. The run times will be compared to the those obtained by Stannard (1975), using the same cascade, to demonstrate the improvements made to the wind tunnel system.

The run times obtainable for a given cascade throat area and test section pressure are a function of the initial storage tank pressure. As discussed in Chapter 3, the compressor has

been set to deliver a final storage tank pressure of approximately 100 psig or approximately 8 bars absolute. Therefore, this chapter will examine the operation of the wind tunnel over a range of test section pressures and effective cascade throat areas for an initial storage tank pressure of approximately 8 bars. The effective throat area of the calibration cascade was varied by rotating the test section turntable thus varying the angle of the cascade with respect to the inlet flow. Figure 7.1 shows the effect of variations in cascade effective throat area on the uncontrolled test section pressure. The largest effective cascade throat area corresponds to a cascade angle of 0 degrees, that is, when the inlet flow direction is perpendicular to the cascade. The minimum cascade throat area occurs at the maximum cascade angle, which is approximately 65 degrees.

7.2 Wind Tunnel Run Times.

The improvements to the wind tunnel system and the new control system have greatly increased run times as compared to the original system. In addition, the range of obtainable test section pressures has also been increased. Figure 7.2, taken from Stannard (1975), shows wind tunnel run times for different test section pressures for the original wind tunnel. The throat areas shown in Figure 7.2 are of some uncertainty. It is not discussed in Stannard (1975) how these throat areas were calculated, however, it is stated that the calibration cascade was used during the testing. Therefore, it was hoped that by investigating the performance of the wind tunnel over the full range of cascade angles using the calibration

Figure 7.1: Uncontrolled Test Section Pressure for Different Cascade Throat Areas.

Figure 7.2: Original Wind Tunnel Run Times. Taken from Stannard (1975).

cascade, the range of cascade throat areas used by Stannard would be covered. Figure 7.3 presents the run times as a function of test section pressures for the present wind tunnel system. The pressures in Figure 7.3 are presented in psig to allow direct comparison with the results obtained by Stannard (1975). The highest controlled test section pressure shown in Figure 7.2 is approximately 20 psig. As shown in Figure 7.3, controlled test section pressures up to approximately 26 psig have been obtained. Figure 7.2 also shows that the maximum run time for a test section pressure of approximately 15 psig was approximately 20 seconds. By comparison, test section pressures of approximately 15 psig have been maintained for over 30 seconds with the new configuration. The increased wind tunnel run times and test section pressures are a result of the modifications made to the system during the commissioning of the wind tunnel at Carleton.

The most significant improvement to the wind tunnel system was the control system. As discussed in Chapter 2, the original wind tunnel had to be shut down at high valve positions because of control system instability. The present control system allows test section pressure control to be maintained throughout the full valve travel, thus the air in the storage tanks is fully utilized for a given test section blowing pressure.

The replacement of the shut-off valve, discussed in Chapter 3, has also contributed to increased run times. Figure 7.4 compares the run times attainable for a test section blowing pressure of approximately 2 bars absolute with the original shut-off valve and the new shut-off valve. From the figure it is evident that a gain in run time of approximately 3 seconds

Figure 7.3: Wind Tunnel Run Times as a Function of Test Section Blowing Pressures.

Figure 7.4: Improvements in Run Time with New Shut-Off Valve.

has been obtained.

7.3 Cascade Inlet Flow.

The Mach numbers at the cascade inlet were measured using the pitot tube and static tap located as discussed in Chapter 4. Figure 7.5 shows the Mach numbers during a typical wind tunnel run as measured in the centre of the cascade inlet. The figure shows some fluctuations in the instantaneous Mach numbers; however, the average Mach number is seen to be essentially constant throughout most of the run.

As a matter of interest, Figure 7.6 presents the measured cascade inlet Mach numbers as a function of the test section blowing pressure for the calibration cascade. The Mach numbers shown in Figure 7.6 are a function of the effective throat area of the cascade. Therefore, future cascades with different effective throat areas will have different inlet Mach numbers for the same cascade angles.

Examining Figure 7.6, with the cascade set normal to the inlet flow, that is a cascade angle of 0 degrees, the cascade appears to choke at somewhere between 1.8 and 2 bars absolute. From this point, the Mach number upstream of the cascade remains essentially constant. A test section blowing pressure of approximately 1.9 bars absolute corresponds to the critical pressure ratio across the cascade where choking is expected. However, with the cascade set at 65 degrees to the inlet flow, the Mach number appears to slightly increase with

Figure 7.5: Measured Cascade Inlet Mach Numbers during a Typical Run.

Figure 7.6: Measured Cascade Inlet Mach Numbers as a Function of Test Section Blowing Pressure.

test section pressure even with the cascade choked. This may be a result of the complexity of the flow through the inclined row of cylinders. The effective throat area of the calibration cascade may be slightly changing with test section pressure.

With the cascade choked, the upstream Mach number is a function only of the ratio of the throat area to the upstream area. Thus using the upstream Mach numbers shown in Figure 7.6 with the cascade choked, the effective throat area of the calibration cascade was estimated assuming 1-D flow. The equation relating the upstream Mach numbers to the area ratio for a choked throat is defined as

$$\frac{A_1}{A_{th}} = \frac{1}{M_1} \sqrt{\left(\frac{2 + (\gamma - 1)M_1^2}{\gamma + 1} \right)^{\frac{\gamma + 1}{\gamma - 1}}} \quad (7.1)$$

Where M_1 = Upstream Mach Number,
 A_1 = Upstream Area,
 A_{th} = Choked Throat Area.

Therefore, using Figure 3.5, the effective throat area of the calibration cascade at 0 degrees is approximately 12.9 in². Similarly at 65 degrees, the effective throat area of the calibration cascade is approximately 8.9 in². Compared with the throat areas quoted in Figure 7.2, the range of throat areas used during this work was similar to that used in Stannard (1975).

The Mach numbers were also measured at different vertical distances below the upper

test section wall on the test section centreline. The purpose of these measurements was to examine the velocity distribution across the cascade inlet in the span-wise direction. Figure 7.7 shows the results as a function of distance from the upper wall. The flow appears relatively uniform around midspan. However, the boundary layer heights along the upper and lower walls appear to be fairly large at approximately 20 to 30 percent of the height of the flow passage. The height of the boundary layer along the upper test section wall also appears to be substantially thicker than the lower wall boundary layer. Further studies of the flow in the pitch-wise direction across the cascade inlet are required to fully establish the degree of inlet flow uniformity. As described next, at a late stage in the measurements some indirect evidence was obtained indicating that there may be some non-uniformity in the flow away from the centreline.

The new dryer system, discussed in Chapter 3, includes an after-filtering system to remove any residue of desiccant from the air leaving the dryer. However, the original dryer had no after-filtering system. Therefore, residue from the desiccant was also present in the storage tank air. Some of this residue and perhaps some dirt from other sources was found to have fouled the walls of the test section. The residue formed black lines along the test section walls, indicating the direction of the wall shear stress vectors. These lines were noticed when the test section upper wall was removed. It was observed that there appeared to be some flow disturbances caused by the ends of the sliding walls. Figure 7.8 presents a sketch of the flow pattern inferred from the flow visualization. The wakes caused by the sliding walls appear to extend well towards the centreline. These flow disturbances may also

Figure 7.7: Measured Mach Number Distribution Across the Cascade Inlet in the Spanwise Direction.

Figure 7.8: Sketch of Test Section Flow Visualization.

have contributed to the fairly large boundary layers shown in Figure 7.7.

Therefore, further flow visualization tests and measurements of the flow at the cascade inlet are required. Modifications to the sliding walls may be required to eliminate the separations.

7.4 Storage Tank Temperature Drop.

The expansion of the air in the storage tanks during a run causes the temperature of the air in the tanks to decrease. As discussed in Chapter 4, the storage tanks were instrumented with a temperature probe to measure the storage tank temperature during a run. Figure 7.9 presents the tank temperature drop as a function of the pressure drop in the tanks. Both the temperatures and pressures shown in Figure 7.9 have been normalized using the initial tank conditions. The figure compares the changes in tank temperatures as measured by the probe, over runs at different blowing pressures. Examining the figure, for a given pressure drop, runs of high blowing pressures and thus high mass flows appear to have less of a temperature drop than do low mass flow runs. This is opposite of what would be expected. In high mass flow runs, the tank discharge occurs much faster than in low mass flow runs. Thus there is less time for heat transfer into the tanks. Therefore, one would expect for a given pressure drop, the temperature drop in the tanks would be greater at high mass flows than at low mass flows.

Figure 7.9: Measured Storage Tank Temperature Drop as a Function of Tank Pressure Drop.

The reasons for the inaccurate temperature measurements in the tanks during a run may be temperature variations within the tanks. The heat transfer through the tank walls affects the air near the tank walls. There will be some conduction heat transfer towards the centreline, however the most dominant mechanism for transferring heat within the tanks will be mixing. It may be that more mixing occurs during high blowing pressure runs since the tanks are emptied more rapidly. The conclusion is that the temperature measured in the tanks cannot be used to provide a reliable indication of the total temperature of the flow in the test section.

To estimate possible changes in storage tank temperatures during a run, an analysis was done using polytropic relations. The polytropic relationship in terms of initial and final conditions is given as

$$\frac{T_f}{T_i} = \left(\frac{P_f}{P_i} \right)^{\frac{n-1}{n}}, \quad (7.2)$$

where n is the Polytropic Index.

During a series of controlled runs at different blowing pressures, the tank pressures were tracked throughout the runs. The pressure in the tanks during a run should be fairly uniform since significant pressure gradients within the tanks cannot be sustained. In addition, storage tank air temperature measured before the start of a run should also be reasonably accurate. By measuring the initial tank pressure (P_i) and the final tank pressure (P_f) at the end of a run and using polytropic relations, the changes in temperature within the tanks were

estimated. For this analysis, a Polytronic Index of 1.4 was assumed. This corresponds to the adiabatic case, where there is no heat transfer into the tanks during a run. This represents the worst case, since in actuality there will be some degree of heat transfer into the tanks during a run and the effective polytronic index of the tanks will lie somewhere between 1.0 and 1.4.

Figure 7.10 presents the measured storage tank pressure ratios and the corresponding adiabatic temperature ratios for runs at different test section blowing pressures. From the figure, the tank temperature drops range from approximately 10 percent of the initial tank temperature at high test section pressures to approximately 40 percent at low test section pressures. As mentioned, this analysis represents the worst case where there is no heat transfer into the tanks during a run. Therefore, one would expect that the actual temperature drops for the given test section blowing pressures will be somewhat less than those shown in Figure 7.10.

7.5 Effects of Storage Tank Temperature Drop on Test Section Reynolds Numbers.

The reduction in storage tank temperatures during a run causes temperatures in the test section also to decrease. The effect of the decreasing test section temperatures is that the Reynolds number measured in the test section will increase during a run.

The most appropriate way to determine the test section temperature is by locating a total temperature probe within the test section. However, at the time of this work such a

Figure 7.10: Adiabatic Storage Tank Temperature Ratio as a Function of Tank Pressure Ratio.

probe was not available. Therefore, the results of Section 7.4 are used to investigate the possible changes in test section Reynolds numbers during a run.

During a pressure controlled run, both blowing pressure and Mach number within the test section will be essentially constant throughout most of the run. Therefore, the test section Reynolds number will vary only with the test section temperature. Consider the Reynolds number equation in compressible terms based on some arbitrary length scale, c ;

$$Re = \frac{\left(\frac{P}{RT}\right)(M\sqrt{\gamma RT})c}{\mu}, \quad (7.3)$$

where

P	= Static Pressure
T	= Static Temperature
M	= Mach Number
μ	= Viscosity.

Normalizing equation (7.3) using an initial Reynolds number (Re_i), based on the initial temperature (T_i) in the test section and assuming the test section pressure and Mach number remain constant, we get

$$\frac{Re}{Re_i} = \sqrt{\frac{T_i}{T} \left(\frac{\mu_i}{\mu} \right)}. \quad (7.4)$$

The variations in viscosity with temperature can be estimated using the Sutherland Law thus

$$\frac{\mu_i}{\mu} = \left(\frac{T_i}{T} \right)^{\frac{3}{2}} \left(\frac{T+110}{T_i+110} \right), \quad (7.5)$$

where temperatures are in K.

As discussed in Section 7.5, the temperature drops in the storage tanks ranged from 10 percent at high blowing pressures to approximately 40 percent at low blowing pressures. Assuming there is no heat transfer into the system between the tanks and the test section, the total temperature of the flow in the test section is equal to the effective total temperature of the air in the storage tanks. Assuming the Mach number remains constant, the ratio of static temperature changes is equal the ratio of the total temperature changes. Therefore from Figure 7.11, possible changes in Reynolds numbers during a run in the worst case, could range from approximately 12 percent at high blowing pressures to approximately 100 percent at low blowing pressures.

The assumption of adiabatic expansion represents the worst case, thus the actual variations in Reynolds number during a run will be somewhat less. However, it is clear that the change in Reynolds number during a run will be large enough that it cannot be ignored. Therefore, the test section total temperature will have to be measured and recorded continuously during normal wind tunnel operations.

Figure 7.11: Changes in Reynolds Number as a Function of Changes in Temperature.

8.0 CONCLUSIONS AND RECOMMENDATIONS.

8.1 Conclusions.

The wind tunnel is now operational. The improved control system provides stable control throughout the ranges of attainable test section pressures for a given cascade throat area. The modifications made to the existing wind tunnel and the improved control system have both increased the ranges of test section pressures which can be achieved and the wind tunnel run times for a given test section pressure compared to the original wind tunnel system. Wind tunnel run times have been increased by over 50 percent compared to the run times of the original system. In addition, the replacement of the air drying system has eliminated the condensation problem in the test section.

The method of modelling the wind tunnel presented in Chapter 5 seems to provide a reasonable way of estimating the wind tunnel performance. Therefore future wind tunnel configurations could be first modelled to assess the performance of the new system before design and construction commences.

8.2 Recommendations for Further Work.

Although substantial wind tunnel calibration testing has been performed, there are further studies which must be conducted on the wind tunnel to investigate its operational

performance. Measurements of the turbulence intensity must be conducted in the test section to see if they are within acceptable limits for cascade blade testing. A complete investigation of the cascade inlet flow must be performed by traversing the cascade inlet region in both the span-wise and the pitch-wise directions. Flow visualization experiments should be conducted to investigate the flow distortions caused by the Test Section Sliding Walls. Further studies into the magnitude of the actual changes in test section Reynolds numbers during a run must be performed by locating a total temperature probe within the test section.

8.3 Further Modifications to the Wind Tunnel System.

As discussed in Chapter 6, the present control valve is too large to control very low test section blowing pressures. A second and substantially smaller control valve could be installed in parallel with the present control valve to provide good pressure control at test section blowing pressures below approximately 1.2 bars absolute.

REFERENCES

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- Stannard, J.,H., 1975, "*The Aerodynamic Design and Testing of High Turning Angle Turbine Blades*", Doctoral thesis, Engineering, McMaster University.
- Tanner, J.,A., Dietiker,W.,1963 "*The Design Study of a Pressure Control System for a 5 ft. by 5 ft. Blowdown Wind Tunnel*", Automatic and Remote Control, Second Congress of IFAC. Basle Switzerland, 1963, Butterworth & Co (Canada) Ltd, 1964.

Appendix A

High Speed Wind Tunnel Simulation Data.

This appendix contains all data which was used in the wind tunnel simulation program. The program was written in Turbo Pascal Version 7 and was simply an implementation of the equations found in Chapter 5. The program used discrete time steps to solve the equations with time. The time step used was 0.001 seconds. The program is found on the computer in the High Speed Wind Tunnel laboratory.

Storage Tanks.

Initial Pressure	= 8 bars (absolute)
Initial Temperature	= 300 K
Approximate Volume	= 26 m³
Polytropic Index	= 1 to 1.4

Test Section.

Approximate Volume	= 0.029 m³
Cascade Throat Area	= 12.9 in²
Upstream Flow Area	= 12.57 in²

Control Valve.

Cv_{full} (manufacturer)	= 550
Cv_{full} (experimental)	= 350
Full Stroking Time	= 3.5 seconds.

Figure A.1 shows the valve characteristics as a function of valve position.

Figure A.1: Foxboro Control Valve Characteristics.

The curves in Figure A.1 were curve fit for computational purposes. The curve fits were in the form

$$\frac{C_v}{C_{v_{\text{max}}}} = \sum a_i \phi^i \quad (\text{A.1})$$

and

$$X_T = \sum b_i \phi^i. \quad (\text{A.2})$$

Table A.1: Control Valve Curve Fit Coefficients.

i	a_i	b_i
0	0.0	0.819072
1	-0.317126	0.214104
2	4.90186	-2.61186
3	-23.6348	12.3267
4	54.8373	-30.855
5	-57.4866	30.3073
6	22.6976	-10.049

Appendix B

Data Acquisition and Control System Software Routines.

This appendix contains the main procedures used in the Data Acquisition and Control System Software. The program was written in Turbo Pascal Version 6. The two units VALVE and HPIB represent the core of the program. The procedures found in HPIB are used to send commands between the computer and the Data Acquisition System from which the transducer voltages are measured. The procedure MOVEVALVE is used to issue the positioning command to the control valve positioner.

The rest of the program contains user interface procedures and can be found on the computer in the High Speed Wind Tunnel Laboratory.

The code contained in PROCEDURE MoveValve is used to send the valve positioning commands from the computer to the valve positioner. This is done by sending an 8 bit number to the parallel port address, in this case the address is 632. The input to the procedure is a valve position between 0 and 1. The procedure then converts the number to an 8 bit number between 0 and 255. The 8 bit number is then sent to the parallel port address. Attached to the parallel port is a D/A converter which outputs a current signal between 4 and 20 mA which is sent to the control valve positioner.

UNIT Valve;

INTERFACE

PROCEDURE MoveValve(VAR ValvePos:REAL);

IMPLEMENTATION

PROCEDURE MoveValve(VAR ValvePos:REAL);

VAR ValvePosition:INTEGER;

BEGIN

ValvePosition:=Round(ValvePos*255);

IF (ValvePosition < 0) THEN BEGIN

ValvePosition:=0;

END; { IF }

IF (ValvePosition > 255) THEN BEGIN

ValvePosition:=255;

END; { IF }

Port[632]:=ValvePosition;

END; { PROCEDURE MoveValve }

END. { UNIT Valve }

The unit HPIB contains all the commands and procedures used to send information between the Data Acquisition System and the computer.

UNIT HPIB;

INTERFACE

TYPE Measurements=ARRAY[0..10] OF REAL;

PROCEDURE SetupHpSystem(VAR Hp3497, Hp3437, FirstChannel, LastChannel: INTEGER);

PROCEDURE CloseChannel(VAR Hp3497, Channel: INTEGER);

PROCEDURE StepChannel(VAR Hp3497: INTEGER);

PROCEDURE ReadChannel(VAR Hp3437: INTEGER; VAR Voltage: REAL);

PROCEDURE ScanChannels(VAR Hp3497, Hp3437, FirstChannel, LastChannel: INTEGER;
VAR Reading: Measurements);

IMPLEMENTATION

{SIFDEF VER60}

USES P60HPIB; This unit was contained in the system drivers obtained from Hewlett
Packard

{SENDIF}

{SI TIODECL.EX} Also from Hewlett Packard.

TYPE DelayTime=STRING[9];

PROCEDURE ErrorHandler(VAR Error: INTEGER; Routine: STRING);

BEGIN

IF (Error <> NoErr) THEN BEGIN

WRITELN;

WRITELN ('Error in call to ', routine);

WRITELN (error:6, Errstr (error));

WRITELN;

WRITE('Press <RETURN> to continue program: ');


```

    READLN;
  END; { IF }
END;{ PROCEDURE ErrorHandler }

```

```

PROCEDURE HpSetupData(VAR ISC, Hp3497, Hp3437, FirstChannel, LastChannel: INTEGER;
    VAR AE, T, F, R: CHAR; VAR Delay: DelayTime);
VAR DataFile: TEXT;
BEGIN
  ASSIGN(DataFile, 'HPIB.INT');
  RESET(DataFile);
  READLN(DataFile, ISC);
  READLN(DataFile, Hp3497);
  READLN(DataFile, Hp3437);
  READLN(DataFile, FirstChannel);
  READLN(DataFile, LastChannel);
  READLN(DataFile, AE);
  READLN(DataFile, T);
  READLN(DataFile, F);
  READLN(DataFile, R);
  READLN(DataFile, Delay);
  CLOSE(DataFile);
END; { PROCEDURE HpSetupData }

```

```

PROCEDURE SetupHPIB(VAR ISC: INTEGER);
VAR Error: INTEGER;
BEGIN
  Error := IORESET(ISC);
  ErrorHandler(Error, 'IORESET');
  Error := IOTIMEOUT(ISC, 100);
  ErrorHandler(Error, 'IOTIMEOUT');
  Error := IODMA(ISC, 0, 3);
  ErrorHandler(Error, 'IODMA');
END; { PROCEDURE SetupHPIB }

```

```

PROCEDURE SetupHp3497(VAR Hp3497, FirstChannel, LastChannel: INTEGER;
    VAR AE: CHAR);
VAR Error, Leng: INTEGER;
    Command, AF, AL: STRING;
BEGIN
  Error := IOCLEAR(Hp3497);
  ErrorHandler(Error, 'IOCLEAR');
  STR(FirstChannel, AF);
  STR(LastChannel, AL);

```

```

    Command:=('AF'+AF)+('AL'+AL)+('AE'+AE));
    Leng:=Length(Command);
    Error:=IOOUTPUTS(Hp3497,Command,Leng);
    ErrorHandler(Error,'IOOUTPUTS');
END; { PROCEDURE SetupHp3497 }

```

```

PROCEDURE SetupHp3437(VAR Hp3437:INTEGER; VAR T,F,R:CHAR; VAR
Delay:DelayTime);
    VAR Error,Leng:INTEGER;
    Command:STRING;
BEGIN
    Error:=IOCLEAR(Hp3437);
    ErrorHandler(Error,'IOCLEAR');
    Command:=('T'+T)+('F'+F)+('R'+R)+('D'+Delay)+'S');
    Leng:=Length(Command);
    Error:=IOOUTPUTS(Hp3437,Command,Leng);
    ErrorHandler(Error,'IOOUTPUTS');
END;{ PROCEDURE SetupHp3437 }

```

```

PROCEDURE SetupHpSystem(VAR Hp3497,Hp3437,FirstChannel,LastChannel:INTEGER);
    VAR ISC:INTEGER;
    AE,T,F,R:CHAR;
    Delay:DelayTime;
BEGIN
    HpSetupData(ISC,Hp3497,Hp3437,FirstChannel,LastChannel,AE,T,F,R,Delay);
    SetupHPiB(ISC);
    SetUpHp3497(Hp3497,FirstChannel,LastChannel,AE);
    SetUpHp3437(Hp3437,T,F,R,Delay);
END;

```

```

PROCEDURE CloseChannel(VAR Hp3497,Channel:INTEGER);
    VAR Command:STRING;
    Leng,Error:INTEGER;
    ChannelNo:STRING;
BEGIN
    STR(Channel,ChannelNo);
    Command:='AC'+ChannelNo;
    Leng:=Length(Command);
    Error:=IOOUTPUTS(Hp3497,Command,Leng);
    ErrorHandler(Error,'IOOUTPUTS');
END; { PROCEDURE StepChannel }

```

```

PROCEDURE StepChannel(VAR Hp3497:INTEGER);

```

```

VAR Command:STRING;
    Leng>Error:INTEGER;
BEGIN
    Command:='AS';
    Leng:=Length(Command);
    Error:=IOOUTPUTS(Hp3497,Command,Leng);
    ErrorHandler(Error,'IOOUTPUTS');
END; { PROCEDURE StepChannel }

PROCEDURE ReadChannel(VAR Hp3437:INTEGER; VAR Voltage:REAL);
VAR Error:INTEGER;
BEGIN
    Error:=IOENTER(Hp3437,Voltage);
    ErrorHandler(Error,'IOENTER');
END; { PROCEDURE ReadChannel }

PROCEDURE ConvertVoltage(VAR Channel:INTEGER; VAR Voltage,Reading:REAL);
BEGIN
    CASE ( Channel ) OF
        0: Reading:=((1.74455*Voltage)-0.737784);
        1: Reading:=((1.06282*Voltage)-0.066209);
        2: Reading:=((1.04439*Voltage)-0.0448145);
        3: Reading:=((2.62545*Voltage)-1.63234);
        4: Reading:=((30.7539*Voltage)+227.445);
    END; { CASE }
END; { PROCEDURE ConvertVoltage }

PROCEDURE ScanChannels(VAR Hp3497, Hp3437, FirstChannel, LastChannel:INTEGER;
    VAR Reading:Measurements);
VAR Channel:INTEGER;
    Voltage:REAL;
BEGIN
    FOR Channel:=FirstChannel TO LastChannel DO BEGIN
        StepChannel(Hp3497);
        ReadChannel(Hp3437,Voltage);
        ConvertVoltage(Channel,Voltage,Reading[Channel]);
    END; { FOR }
END; { PROCEDURE ScanChannels }

END. { UNIT HPIB }

```

Appendix C

Transducer Calibration Curves.

Figure C.1: Pressure Transducer, Serial No: 143044 Calibration Curve.

Figure C.2: Pressure Transducer, Serial No: 263716 Calibration Curve.

Figure C.3: Pressure Transducer, Serial No: 143393 Calibration Curve.

Figure C.4: Pressure Transducer, Serial No: 143391 Calibration Curve

Figure C.5: Pressure Transducer, Serial No: 263714 Calibration Curve.

Figure C.6: Tank Temperature Transducer Calibration Curve.

END

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FIN