

This is to certify that the
thesis entitled

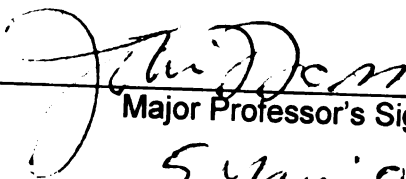
AN EXPERIMENTAL INVESTIGATION OF THE
AERODYNAMIC SHROUD WITH AN OFF-HIGHWAY
ENGINE COOLING FAN

presented by

MICHAEL D. DUSEL

has been accepted towards fulfillment
of the requirements for the

M.S. degree in Mechanical Engineering


Major Professor's Signature

5 May '05
Date

PLACE IN RETURN BOX to remove this checkout from your record.
TO AVOID FINES return on or before date due.
MAY BE RECALLED with earlier due date if requested.

DATE DUE	DATE DUE	DATE DUE

AN EX

SHR

**AN EXPERIMENTAL INVESTIGATION OF THE AERODYNAMIC
SHROUD WITH AN OFF-HIGHWAY ENGINE COOLING FAN**

By

Michael D. Dusel

A THESIS

**Submitted to
Michigan State University
in partial fulfillment of the requirements
for the degree of**

MASTER OF SCIENCE

Department of Mechanical Engineering

2005

AN

dynamic

technique

mance

tion. T

plenum

annular

Integral

reference

that the

pared to

is provid

ble-X ho

location a

shroud af

averaged a

ABSTRACT

AN EXPERIMENTAL INVESTIGATION OF THE AERODYNAMIC SHROUD WITH AN OFF-HIGHWAY ENGINE COOLING FAN

By

Michael D. Dusel

An experimental investigation of a large scale engine cooling fan with an aerodynamic shroud has been conducted to assess the effectiveness of this active flow control technique. The experimental study was motivated by the desire to increase the performance and efficiency of this axial fan which is intended for use in an off-highway application. The aerodynamic shroud utilizes a Coanda jet that is: i) delivered from a pressurized plenum, and ii) introduced along the surface of the shroud upstream of the fan inlet. This annular jet delivers high momentum fluid into the blade tip clearance region.

Experimental data were acquired to quantify the effect of the aerodynamic shroud. Integral quantities including pressure rise, flow rate, and input power were measured for a reference baseline condition and for several powered shroud conditions. The results show that the inception of fan stall is delayed to a lower flow rate (higher pressure rise) as compared to the baseline condition. Further insight into the effect of the aerodynamic shroud is provided by detailed velocity measurements in the wake of the fan. Specifically, a double-X hot-wire probe was used to resolve three components of velocity at a single axial location and various radial locations. The phase averaged data show that the aerodynamic shroud affects the “tip vortex” and the velocity field at the measurement location. Time averaged and phase averaged in-plane PIV measurements support these data.

I
through

able. I

Charles

funding

ple in th

people

Lawren

Daniel

Lawre

mately

future

would

ACKNOWLEDGEMENTS

I would like to thank my advisor, Dr. John Foss, for his guidance and support throughout my graduate work. His dedication to his work and to his students is remarkable. I would also like to thank my committee members, Dr. Ahmed Naguib and Dr. Charles Petty for their valuable input during the review process. Caterpillar provided the funding for this research and I am grateful for this as well.

I have been extremely fortunate to work with very intelligent and interesting people in the Turbulent Shears Flow Laboratory at Michigan State University. Many of these people contributed their time and effort during the course of this research including Al Lawrenz, Doug Neal, Scott Treat, Kyle Bade, Richard Prevost, Aren Hellum, Amanda Danielson, and Drew Butki. I am grateful for all their help. A special thanks to Al Lawrenz and Doug Neal who took time to teach me things as an undergraduate that ultimately lead to my interest in fluids research.

Finally, I would like to thank my family for their never ending support and my future wife Sarah McCann for her support and motivation throughout this endeavour. I would not have been able to accomplish what I did, had it not been for them.

List of F

List of T

Nomencl

1.0 Intr

1.1 M

1.2 F

1.3 C

2.0 Exp

2.1 T

2.2 T

2

2

2

2.3 E

2

2

3.0 Mea

3.1 D

3.2 I

3

3

3

3.3 p

3

TABLE OF CONTENTS

List of Figures.....	vi
List of Tables	xii
Nomenclature	xiii
1.0 Introduction	1
1.1 Motivation and Background Information	1
1.2 Previous Investigations of the Aerodynamic Shroud	3
1.3 Overview of the Experimental Study	5
2.0 Experimental Apparatus	8
2.1 Test Fan	8
2.2 The Axial Fan Research and Development (AFRD) Facility	8
2.2.1 AFRD Overview	8
2.2.2 Flow Rate Measurement System	9
2.2.2.1 Flow Meter Configuration and Measurement Technique	9
2.2.2.2 Flow Meter Calibration	11
2.2.3 Hot-wire Traversing Mechanism	14
2.3 Experimental Aerodynamic Shroud	14
2.3.1 Prototype Design	14
2.3.2 Shroud Plenum Pressurization System	16
3.0 Measurement Equipment and Technique	24
3.1 Data Acquisition System	24
3.2 Integral Measurements	25
3.2.1 Measurement Equipment	25
3.2.1.1 Pressure Transducers	25
3.2.1.2 Torque Measurement	26
3.2.1.3 Optical Encoder	27
3.2.2 Definition of Variables	27
3.2.3 Data Acquisition and Procedure	31
3.3 Particle Image Velocimetry Measurements	33
3.3.1 PIV Equipment	33

3.4

4.0 Re

4.1

4.2

5.0 Det

5.1

5

5

5

5

5.2 F

5

5

5

5

6.0 Sum

Reference

3.3.2 PIV Vector Processing	34
3.3.3 Data Acquisition and Procedure	36
3.4 Hot-wire Measurements	38
3.4.1 Hot-wire Probes, Anemometers, and Thermistor	39
3.4.2 Calibration Facility	39
3.4.3 Calibration	40
3.4.4 Data Acquisition Procedure	42
3.4.5 Double-X Hot-wire Processing	44
4.0 Results and Discussion - Integral Measurements	56
4.1 Baseline Measurements (Unpowered Shroud Condition)	56
4.2 Effect of Aerodynamic Shroud (Powered Shroud Conditions)	57
5.0 Detailed Velocity Measurements	66
5.1 Hot-wire Measurements	67
5.1.1 Hot-wire Measurement Uncertainties	68
5.1.2 Phase Averaged Velocity Data	70
5.1.3 Phase Averaged Fluctuation Intensity (RMS) Data	72
5.1.4 Phase Averaged Vorticity Data	73
5.2 Particle Image Velocimetry Measurements	75
5.2.1 PIV Measurement Uncertainties	76
5.2.2 Time Averaged PIV Data	77
5.2.3 Calculation of Flow Coefficient	81
5.2.4 Phase Averaged PIV Measurements	82
6.0 Summary and Conclusions	123
References	126

Figure

Figure 2

Figure 2

Figure 2

Figure 2

Figure 2

Figure 2

Figure 2

Figure 2

Figure 2

Figure 3

Figure 3

Figure 3

Figure 3

Figure 3

Figure 3

Figure 3

Figure 3

LIST OF FIGURES

Figure 1.1	Schematic representation of the aerodynamic shroud concept.....	7
Figure 2.1	Prototype test fan ($D = 0.711$ m).....	18
Figure 2.2	Schematic representation of the AFRD flow facility	18
Figure 2.3	Schematic representation of the flow metering device.....	19
Figure 2.4	Details of the force transducer for the flow metering device (Neal, 2002)...	19
Figure 2.5	Calibration data and fit for the flow metering devices	20
Figure 2.6	Schematic representation of AFRD hot-wire traverse.....	21
Figure 2.7	Schematic representation of the aerodynamic shroud and plenum configuration (Radiator not shown in top view).....	22
Figure 2.8	Radiator with static pressure tap downstream of the cooling fins.....	23
Figure 2.9	Schematic representation of AFRD with aerodynamic shroud assembly and pressurization system.....	23
Figure 3.1	Torque measurements acquired over a period of time with the fan removed from the shaft to measure a tare value.....	48
Figure 3.2	Schematic representation of the pressure rise across the fan blades	48
Figure 3.3	A schematic representation of the cross-correlation technique using in PIV vector processing	49
Figure 3.4	A schematic representation of the interrogation window overlap used in PIV vector processing	49
Figure 3.5	Schematic of PIV measurement regions relative to the experimental configuration.....	50
Figure 3.6	TTL voltage signals from the optical encoders for PIV phase triggering taken at a shaft speed of 20 rpm.....	51
Figure 3.7	Illustration of the double-x hot-wire probe	51
Figure 3.8	Hot-wire calibration facility (Top and side panel removed to show detail)..	52

Figure 3.9

Figure 3.10

Figure 3.11

Figure 3.12

Figure 3.13

Figure 3.14

Figure 3.15

Figure 4.1

Figure 4.2

Figure 4.3

Figure 4.4

Figure 4.5

Figure 4.6

Figure 4.7

Figure 4.8

Figure 5.1

Figure 5.2

Figure 3.9	Front-view of the calibration facility. Two axes of rotation allow the hot-wire probe to rotated in the vertical and horizontal plane	52
Figure 3.10	A representative pressure signal from a hot-wire calibration. The high standard deviation was reduced by fitting the curve with a 15th order polynomial function.....	53
Figure 3.11	A representative image showing a fluctuating tuft. This tuft was used to find the mean flow direction at a given radial location in the wake of the fan.....	53
Figure 3.12	Schematic of hot-wire probe positioning device	54
Figure 3.13	Histogram of flow angles measured by an improperly aligned hot-wire probe	54
Figure 3.14	Histogram of flow angles measured by a properly aligned hot-wire probe ..	55
Figure 3.15	Intersection of the predicted velocities from the two wires in a single x-array for use in the processing algorithm to find the measured velocity and angle.....	55
Figure 4.1	Baseline performance data for three independent tests	62
Figure 4.2	Baseline efficiency data for three independent tests	62
Figure 4.3	Total system pressure rise plotted as a function of volume flow rate in dimensionless form.....	63
Figure 4.4	Baseline characteristic curve showing the stall point and separate regions of the characteristic curve	63
Figure 4.5	Performance data for selected shroud conditions.....	64
Figure 4.6	Two performance curves, a baseline condition and a representative powered shroud condition, plotted with a low and high system resistance curve to demonstrate the advantage of delaying stall.....	64
Figure 4.7	Efficiency data for selected shroud conditions.....	65
Figure 4.8	Performance data at showing the stall point hysteresis	65
Figure 5.1	Selected shroud conditions labeled with the operating points used for the detailed velocity measurements (see Table 4.1 for numerical values).....	87
Figure 5.2	Mesh of hot-wire phase averaged data locations in r-q plane at $z = 35\text{mm}$ ($z/R_{tip}=0.098$), shows direction of fan rotation	88

Figure 5.3

Figure 5.4

Figure 5.5

Figure 5.6

Figure 5.7

Figure 5.8

Figure 5.9

Figure 5.10

Figure 5.11

Figure 5.12

Figure 5.13

Figure 5.14

Figure 5.15

Figure 5.16

Figure 5.17

Figure 5.18

Figure 5.19

Figure 5.20

Figure 5.2

Figure 5.2

Figure 5.2

Figure 5.2

Figure 5.3	Histograms illustrating the “sample by sample” difference between velocities found using the pre and post calibration constants for Condition ‘A’	89
Figure 5.4	Histograms illustrating the “sample by sample” difference between velocities found using the pre and post calibration constants for Condition ‘B’	90
Figure 5.5	Histograms illustrating the “sample by sample” difference between velocities found using the pre and post calibration constants for Condition ‘C’	91
Figure 5.6	Plots of convergence for a representative data point acquired in the hot-wire measurements	92
Figure 5.7	Phase averaged mean radial velocity for condition ‘A’	93
Figure 5.8	Phase averaged mean radial velocity for condition ‘B’	93
Figure 5.9	Phase averaged mean radial velocity for condition ‘C’	94
Figure 5.10	Phase averaged mean tangential velocity for condition ‘A’	94
Figure 5.11	Phase averaged mean tangential velocity for condition ‘B’	95
Figure 5.12	Phase averaged mean tangential velocity for condition ‘C’	95
Figure 5.13	Phase averaged mean axial velocity for condition ‘A’	96
Figure 5.14	Phase averaged mean axial velocity for condition ‘B’	96
Figure 5.15	Phase averaged mean axial velocity for condition ‘C’	97
Figure 5.16	Phase averaged velocity magnitude for condition ‘A’	97
Figure 5.17	Phase averaged velocity magnitude for condition ‘B’	98
Figure 5.18	Phase averaged velocity magnitude for condition ‘C’	98
Figure 5.19	Phase averaged RMS of radial velocity for condition ‘A’	99
Figure 5.20	Phase averaged RMS of radial velocity for condition ‘B’	99
Figure 5.21	Phase averaged RMS of radial velocity for condition ‘C’	100
Figure 5.22	Phase averaged RMS of tangential velocity for condition ‘A’	100
Figure 5.23	Phase averaged RMS of tangential velocity for condition ‘B’	101

Figure 5.2

Figure 5.2

Figure 5.2

Figure 5.2

Figure 5.2

Figure 5.2

Figure 5.2

Figure 5.2

Figure 5.3

Figure 5.3

Figure 5.3

Figure 5.3

Figure 5.3

Figure 5.3

Figure 5.3

Figure 5.39

Figure 5.24	Phase averaged RMS of tangential velocity for condition 'C'	101
Figure 5.25	Phase averaged RMS of axial velocity for condition 'A'	102
Figure 5.26	Phase averaged RMS of axial velocity for condition 'B'	102
Figure 5.27	Phase averaged RMS of axial velocity for condition 'C'	103
Figure 5.28	Phase averaged axial vorticity for condition 'A'	103
Figure 5.29	Phase averaged axial vorticity for condition 'B'	104
Figure 5.30	Phase averaged axial vorticity for condition 'C'	104
Figure 5.31	Representative contour plot of percent validated vectors for condition 'A'	105
Figure 5.32	Time averaged PIV data: velocity magnitude in the r-z plane for condition 'A'	106
Figure 5.33	Time averaged PIV data: mean radial velocity for condition 'A'. The axial location of the hot-wire survey is indicated on the plot. The two 'x' marks denote the radial range of the measurements	107
Figure 5.34	Comparison of the radial velocities measured by the hot-wire survey and the two PIV regions for condition 'A'	107
Figure 5.35	Time averaged PIV data: mean axial velocity for condition 'A'. The axial location of the hot-wire survey is indicated on the plot. The two 'x' marks denote the radial range of the measurements	108
Figure 5.36	Comparison of the axial velocities measured by the hot-wire survey and the two PIV regions for condition 'A'	108
Figure 5.37	Time averaged PIV data: velocity magnitude in the r-z plane for condition 'C'	109
Figure 5.38	Time averaged PIV data: mean radial velocity for condition 'C'. The axial location of the hot-wire survey is indicated on the plot. The two 'x' marks denote the radial range of the measurements	110
Figure 5.39	Comparison of the radial velocities measured by the hot-wire survey and the two PIV regions for condition 'C'	110

Figure 5.4

Figure 5.4

Figure 5.4

Figure 5.4

Figure 5.4

Figure 5.4

Figure 5.4

Figure 5.4

Figure 5.4

Figure 5.4

Figure 5.5

Figure 5.5

Figure 5.5

Figure 5.5

Figure 5.5

Figure 5.5

Figure 5.40	Time averaged PIV data: mean axial velocity for condition 'C'. The axial location of the hot-wire survey is indicated on the plot. The two 'x' marks denote the radial range of the measurements	111
Figure 5.41	Comparison of the axial velocities measured by the hot-wire survey and the two PIV regions for condition 'C'	111
Figure 5.42	Time averaged PIV data: velocity magnitude in the r-z plane for condition 'D'	112
Figure 5.43	Time averaged PIV data: mean radial velocity for condition 'D'	113
Figure 5.44	Time averaged PIV data: mean axial velocity for condition 'D'	113
Figure 5.45	Fan and shroud configuration showing the control surface used for the integration of the flow rate from the PIV velocity measurements	114
Figure 5.46	Phase averaged PIV data: velocity magnitude for condition 'A' at fan blade leading edge, note high velocity cell, α , and negative radial flow, β	115
Figure 5.47	Phase averaged PIV data: velocity magnitude for condition 'A' at 5 degrees past leading edge, note the high velocity cell, γ	115
Figure 5.48	Phase averaged PIV data: velocity magnitude for condition 'A' at 10 degrees past leading edge, note high velocity cell, δ	116
Figure 5.49	Phase averaged PIV data: velocity magnitude for condition 'A' at 15 degrees past leading edge	116
Figure 5.50	Phase averaged PIV data: velocity magnitude for condition 'A' at 20 degrees past leading edge, note the "path" of high velocity, ι	117
Figure 5.51	Phase averaged PIV data: velocity magnitude for condition 'A' at 25 degrees past leading edge, note the "path" of high velocity, ι	117
Figure 5.52	Phase averaged PIV data: velocity magnitude for condition 'A' at 30 degrees past leading edge, note high velocity cell, κ	118
Figure 5.53	Phase averaged PIV data: velocity magnitude for condition 'A' at 35 degrees past leading edge	118
Figure 5.54	Phase averaged PIV data: velocity magnitude for condition 'C' at the fan blade leading edge, note the high velocity cell, λ	119
Figure 5.55	Phase averaged PIV data: velocity magnitude for condition 'C' at 5 degrees past the leading edge	119

Figure 5.5

Figure 5.5

Figure 5.5

Figure 5.5

Figure 5.5

Figure 5.5

Figure 5.56 Phase averaged PIV data: velocity magnitude for condition ‘C’ at 10 degrees past the leading edge	120
Figure 5.57 Phase averaged PIV data: velocity magnitude for condition ‘C’ at 15 degrees past the leading edge, note the recirculation region, μ	120
Figure 5.58 Phase averaged PIV data: velocity magnitude for condition ‘C’ at 20 degrees past the leading edge, note the large recirculation region, v	121
Figure 5.59 Phase averaged PIV data: velocity magnitude for condition ‘C’ at 25 degrees past the leading edge	121
Figure 5.60 Phase averaged PIV data: velocity magnitude for condition ‘C’ at 30 degrees past the leading edge	122
Figure 5.61 Phase averaged PIV data: velocity magnitude for condition ‘C’ at 35 degrees past the leading edge	122

Table 5.

Table 5.

Table 5.

Table 5.

LIST OF TABLES

Table 5.1	Test Conditions for Detailed Velocity Measurements	67
Table 5.2	Spatially averaged RMS levels for each component and each test condition	73
Table 5.3	Tabulated values of the locations of the peak values for various measured variables for both shroud conditions	74
Table 5.4	Values of the dimensionless flow rate found by integration of the PIV results compared to the measured flow rate from the integral measurements	82

Rom

A

A.B.

C_d

CS

E

K

L

M_O

N

P_{atm}

$P_{receiver}$

Q

R

R

T

R_e

U

$\vec{r}$

d_H

d_S

f

NOMENCLATURE

Roman:

A	Area
A,B,n	Hot-wire calibration coefficient, Chapter 3 only
C_d	Discharge coefficient
CS	Control surface
E	Voltage output
K	Correlation coefficient
L	Characteristic Length
M_O	Moment about point O in Figure 2.3
N	Number of samples
P_{atm}	Atmospheric Pressure
$P_{receiver}$	Static pressure in upper AFRD receiver
Q	Volume flow rate
R	Universal gas constant, Chapter 2 only
R	Radius
T	Temperature
Re	Reynolds number
U	Velocity component
$\vec{v}$	Velocity vector
d_H	Hydraulic diameter
ds	Particle separation for PIV
f	Data frequency

g

k

m

n

px

φ

r

r*

t_d

u

v

w

z

z*

Greek:

ΔP

Δr

Λ

Ω

α

γ

ξ

g	Annular jet gap height
k	Calibration coefficient for flow metering device
$\dot{m}$	Mass flow rate
$\hat{n}$	Normal vector
px	pixel
ϕ	Power
r	Radial direction
r^*	Non-dimensional radial position
t_d	Time delay using PIV phase positioning
u	Velocity component in line with probe axis measured by both x-arrays
v	Velocity component normal to probe axis measured by first x-array
w	Velocity component normal to probe axis measured by second x-array
z	Axial direction
z^*	Non-dimensional axial position

Greek:

ΔP	Pressure rise
Δr	Radial increment
Λ	Non-dimensional shroud flow rate
Ω	Angular fan speed
α	Phase position
γ	Angle of the velocity vector with respect to the hot-wire probe axis
ζ	Rotation angle 1 of probe axis with respect to fan coordinates

η

θ

λ

ν

ξ

ρ

τ

ϕ

χ

ψ

ω

Subscript

atm

ci

decr

incr

n

r

s

tip

shr

sys

η	Efficiency
θ	Tangential direction
λ	Non-dimensional shroud plenum pressure
ν	Kinematic viscosity
ξ	Rotation angle 2 of probe axis with respect to fan coordinates
ρ	Air density
τ	Torque
ϕ	Flow coefficient
χ	Non-dimensional shroud power
ψ	Pressure coefficient
ω	Vorticity

Subscripts:

atm	Atmospheric
ci	Calibration inlet
decr	Decreasing
incr	Increasing
n	North
r	Radial direction
s	South
tip	Fan blade tip
shr	Shroud
sys	System, encompassing test fan and aerodynamic shroud

z

θ

Acronym

AFRD

CCD

HWA

PIV

RMS

z	Axial direction
θ	Tangential direction

Acronyms:

AFRD	Axial Fan Research and Development Facility
CCD	Charge Coupled Device
HWA	Hot-wire Anemometry
PIV	Particle Image Velocimetry
RMS	Root mean squared

1.0

1.1 Mo

in const

siderabl

system.

cooling

"ram-air

evolve a

dissipate

and oil c

of the ra

the engin

usable ou

as reduc

In

cooling fa

evant resu

study is a

sists of a p

surface of

high mome

(known as t

1.0 Introduction

1.1 Motivation and Background Information

The engine cooling fan used in a typical “off-highway” application, such as found in construction or agricultural equipment, can be quite large in diameter, consume a considerable amount of power, and contribute a measurable amount of noise to the overall system. The cooling fan in this type of application is fully responsible for the convective cooling of the radiator when the engine is operating, unlike a passenger vehicle in which a “ram-air” effect is present when the vehicle is moving. As these engines continue to evolve and the power output increases, there is a greater demand on the cooling system to dissipate the added thermal load. Moreover, the addition of air conditioning condensers and oil coolers adds to the system resistance. One obvious solution is to increase the size of the radiator and cooling fan. However, this has a negative effect on the efficiency of the engine because the larger coolant pump and fan will have a parasitic effect on the usable output power. Thus, it is apparent that any effort to optimize the cooling fan, such as reducing power input or increasing mass flow rate through the fan, is justified.

In response to this motivation, an experimental investigation of an axial flow type cooling fan intended for use in an off-highway application has been conducted and the relevant results and analysis are reported in this thesis. The defining characteristic of this study is a unique aerodynamic shroud that is used with this cooling fan. The shroud consists of a pressurized plenum that produces an annular jet of air which is injected along the surface of a curved shroud upstream of the fan inlet. This Coanda jet is designed to add high momentum flow to the blade tip region and weaken, or diminish, the reverse flow (known as the tip leakage flow) and the resulting tip leakage vortex. A schematic of the

aerodynam

local press

aerodynam

uted to a la

imental stu

1.2.

Au

amount of

representa

ance and th

studies is a

fan. Axia

manufactu

deliver. M

understan

examples

These stu

dition are

A

mance o

clearanc

the focu

of blade

aerodynamic shroud concept is presented in Figure 1.1. The reverse flow is caused by a local pressure differential between the pressure and suction sides of the fan blade. The aerodynamic shroud was originally conceived to prevent the strong reverse flow, attributed to a large tip clearance, in the blade tip region of a light truck cooling fan. An experimental study was derived from this motivation and a detailed review follows in Section 1.2.

Automotive engine cooling fans have been the focus of interest in a limited amount of literature, Baranski (1974), Mellin (1980), and Morris and Foss (2001) provide representative publications. These studies have recognized the significance of tip clearance and the effect on the performance and efficiency of the cooling fan. A wider range of studies is available when considering the cooling fan in the larger context as an axial flow fan. Axial flow fans are found in numerous applications, perhaps because of their ease of manufacture and the relatively low pressure rise and high flow rates which they can deliver. Most of the recent studies documented in the available literature are directed at understanding the source of aeroacoustical noise emitted from a rotating fan. Some recent examples of this include Fukano and Jang (2003), Sorensen (2001), and Wu et al. (1998). These studies often investigate different blading schemes or a change in the boundary condition around the fan.

A few studies, in particular, have used different shrouds to obtain enhanced performance or lower noise generation of the fan. Since the vortical flow field around the tip clearance is known to be a source of loss and production of noise, the fan shroud is often the focus. For instance, Longhouse (1978) stated that tip clearances on the order of 3-4% of blade chord can contribute up to 15 dB in overall noise levels throughout the operating

range of

mate 10

ing-ring

was dec

reduce th

tially shr

designed

reverse fl

2001). T

controllin

1.2 Previ

Th

tively larg

clearance

cooling fan

first to inv

with a dian

such as fan

measuremen

shroud incre

flow rates.

by the fan gr

tant role in de

range of the fan. Furthermore, he stated that large tip clearances can result in an approximate 10% drop in peak efficiency. In response, Longhouse introduced a contoured rotating-ring shroud that is attached to the fan blade tips. The author found that the noise level was decreased by as much as 12dB. Another study by Jang et al. (2001) attempts to reduce the production of noise by controlling the vortical flow field near the tip of a partially shrouded propeller fan using an optimized shape shroud. In particular, the newly designed shroud attempts to “hinder the development of the tip vortex by diminishing a reverse flow caused by the tip vortex between the rotor tip and the shroud” (Jang et al., 2001). Thus, the modification and optimization of the fan shroud is not a new method of controlling the flow field around a fan.

1.2 Previous Investigations of the Aerodynamic Shroud

The original development of the aerodynamic shroud was motivated by the relatively large tip clearance of cooling fans in light and medium duty trucks. The large tip clearance is necessary because of the large relative motions between the engine mounted cooling fan and the chassis mounted shroud and radiator (Morris 1997). Morris was the first to investigate the efficacy of the aerodynamic shroud on an automotive cooling fan with a diameter of 457 mm and a tip clearance of 25 mm. Integral flow measurements such as fan input power and volume flow rate were acquired in addition detailed velocity measurements in the wake of the fan. These data demonstrated that the aerodynamic shroud increased performance at higher flow rates but decreased performance at lower flow rates. Because of this result, it was concluded that system resistance, characterized by the fan grill, radiator, and air-conditioning condenser in an automobile, plays an important role in determining the efficiency of the aerodynamic shroud and fan combination. In

particu

allows

tive ob

lar jet c

an addi

immers

fan blad

fan to pr

strong ra

an axial

study (M

radial flo

shroud w

N

tural buil

dynamic s

lower syst

tural venti

low system

streamline

rate (Neal 2

downstream

particular, a large system resistance lessens the efficiency and a smaller system resistance allows the aerodynamic shroud to increase the efficiency of the fan. Furthermore, qualitative observations using a tuft in the wake of the fan showed that the high momentum annular jet changed the reverse flow in the tip clearance area to a positive axial flow providing an additional contribution to the net flow rate through the fan.

In the original investigation by Morris (1997), the cooling fan was partially immersed in the shroud. In particular, the shroud covered only the upper portion of the fan blades and had a sharp edge outlet. The partially shrouded configuration allowed the fan to produce static pressures even throughout the stall range, which is characterized by strong radial flow. However, when the aerodynamic shroud was used, it forced the flow in an axial direction explaining the decrease in performance at the lower flow rates. In a later study (Morris and Foss, 2001), the authors used a rounded outlet that accommodated this radial flow and found that the enhanced performance of the fan with the aerodynamic shroud was extended into even lower flow rates.

Neal (2002) studied the effectiveness of the aerodynamic shroud with an agricultural building ventilation fan. The observation from the Morris (1997) study that the aerodynamic shroud lead to increased performance and efficiencies with higher flow rates and lower system resistance naturally suited the use of the aerodynamic shroud with agricultural ventilation fan systems. Agricultural building ventilation fans are characterized by a low system resistance (only the protective cage and motor on the downstream side and streamlined shutters on the upstream side add to the resistance) and a high volume flow rate (Neal 2002). The propeller fans tested were used in unison with a diffuser cone on the downstream side. In these studies, the aerodynamic shroud allowed the use of a wider dif-

fuser co

to a low

ber of d

gated. T

increasin

T

mance gr

not all pr

ments. It

namic shro

aerodynam

1.3 Overview

The

configuration

much differ

design with c

on the magnit

aerodynamic

how the aerod

The exp

sented as such.

ments for one

separate aerodyn

fuser cone by allowing the flow to stay attached along the walls of the diffuser. This lead to a lower velocity at the exit of the diffuser and a more uniform velocity profile. A number of different blade designs, shroud configurations, and diffuser cones were investigated. The results show that the aerodynamic shroud increased efficiency by 15% while increasing flow rates by over 35% for the optimal propeller fan of those investigated.

The fact that different blade designs resulted in much different fan/shroud performance gives evidence that the aerodynamic shroud is sensitive to blade geometries and not all production fans can benefit from the active shroud without necessary enhancements. It also supports the suggestions of Neal (2002) and Morris (1997) that the aerodynamic shroud can benefit from a blade design specifically designed for use with the aerodynamic shroud.

1.3 Overview of the Experimental Study

The present study intends to investigate the aerodynamic shroud in a similar test configuration as the Morris (1997) study, however, the fan and boundary conditions are much different. The previous study effectively demonstrated the efficacy of the shroud design with one particular automotive cooling fan and the results were strongly dependent on the magnitude of the tip clearance. This study further investigates the feasibility of the aerodynamic shroud in a different application and extends the general knowledge of the how the aerodynamic shroud affects the performance of a fan.

The experimental program was conducted in two separate portions and will be presented as such. The first portion contains overall performance and efficiency measurements for one baseline test condition and several powered shroud conditions. The separate aerodynamic shroud conditions are characterized by a “shroud power,” defined as

the pro

variable

portion

the wak

technique

velocity

aged and

ues of th

dimension

additional

methods w

The

imental ap

prototype a

equipment

integral mea

of the detail

from all of th

the product of shroud pressure and shroud volume flow rate, and it will be shown that this variable is important in defining the contribution of the aerodynamic shroud. The second portion of the experimental program contains a detailed investigation of the flow field in the wake of the fan. Two different techniques were used to accomplish this. The first technique used a double-X hot-wire probe to simultaneously resolve three components of velocity at a singular axial location downstream of the fan. These results were phase averaged and are presented as contour plots. These data include the mean and fluctuating values of the velocity, as well as the axial vorticity. The second technique uses two-dimensional PIV measurements to present a planar view of the flow field and provide additional analysis of the radial and axial velocities. Time averaged and phase averaged methods were used and each are presented as contour plots.

The thesis structure is defined as follows. First, a detailed description of the experimental apparatus is presented in Section 2.0. This includes the test fan, test facility, and prototype aerodynamic shroud. This is followed by descriptions of the measurement equipment and techniques in Section 3.0. The experimental results and analysis of the integral measurements are presented in Section 4.0 followed by the results and discussion of the detailed velocity measurements in Section 5.0. Finally, the main conclusions drawn from all of the results are presented in Section 6.0.

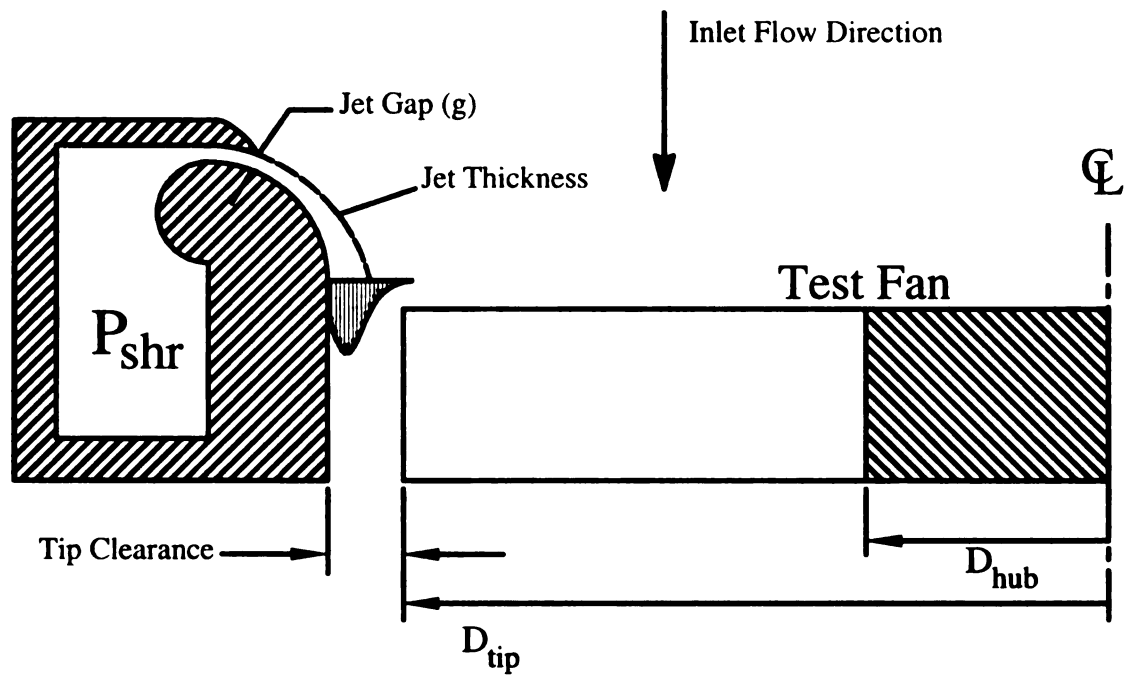


Figure 1.1 Schematic representation of the aerodynamic shroud concept

2.0 I

2.1 Test

T

imental i

hub to tip

respect to

trailing e

immersion

2.2 The A

All

The AFRD

cooling fan

Foss et al.,

and input po

try and part

sections des

2.2.1 AFRD

A sch

is comprised

lower receive

mately 2 m in

the same width

2.0 Experimental Apparatus

2.1 Test Fan

The 711 mm prototype fan pictured in Figure 2.1 was the test fan used in all experimental investigations. The nine molded plastic blades were attached to a steel hub with a hub to tip ratio of approximately 0.37. The blades had a constant pitch (chord angle with respect to plane of rotation) of 34° and a single thickness of 4.5 mm from leading edge to trailing edge. The chord length was 157 mm at the tip and the projected width (total immersion height of the fan blade) was 100 mm.

2.2 The Axial Fan Research and Development (AFRD) Facility

All test fan data contained in this thesis were acquired in the AFRD flow facility. The AFRD facility was originally designed for the testing and research of automotive cooling fans, although it has been used for agricultural ventilation fan research as well (see Foss et al., 2001). Integral flow quantities (fan or system pressure rise, volume flow rate, and input power) as well as detailed velocity measurements including hot-wire anemometry and particle image velocimetry, can be acquired in the AFRD facility. The following sections describe the AFRD facility in more detail along with the relevant calibrations.

2.2.1 AFRD Overview

A schematic representation of the test facility is shown in Figure 2.2. The AFRD is comprised of two sections: an upper receiver (C) in which the test fan is mounted, and a lower receiver (H) in which the flow rate is measured. The upper receiver is approximately 2 m in height and 3 m in width and depth. The lower receiver is 1 m in height with the same width and depth dimensions. Air is moved through the facility by both the test

fan (A)

directed

prime r

SQA-36

outlet is

the flow

fan.

T

by two b

0.68 m fr

by a Relia

tional spe

2.2.2 Flow

A

flow rate t

sists of a l

from the up

momentum

the mass flo

be provided

subsection 2

2.2.2.1 Flow

fan (A) and the prime mover (N). Air is drawn into the upper receiver from the laboratory, directed through a pair of inlet nozzles (L) into the lower receiver, and moved through the prime mover back to the laboratory. A large centrifugal blower (Chicago Blower Co. SQA-36.5) is used as the prime mover. Its inlet is located in the lower receiver (I) and its outlet is equipped with a computer controlled throttle (M). This throttle is used to adjust the flow rate through the AFRD, consequently adjusting the pressure rise across the test fan.

The vertical drive shaft is located in the center of the upper receiver and supported by two bearings (D) on a structural shaft support. The upper bearing is approximately 0.68 m from the fan exit plane to prevent obstruction of the exiting flow. The fan is driven by a Reliance Electric 15 hp motor (G) and a 2:1 belt drive pulley system so that the rotational speed of the fan is twice that of the motor.

2.2.2 Flow Rate Measurement System

A unique metering device is used to obtain an integral measurement of the mass flow rate through the AFRD facility (Morris et al., 2001). The measurement device consists of a large turning passage in the lower receiver that redirects the downward flow from the upper receiver to the horizontal direction. In redirecting the flow, a moment of momentum flux is created at the pivot point. The significance of this action in measuring the mass flow rate and a detailed explanation concerning the measurement apparatus will be provided in subsection 2.2.2.1 followed by an explanation of the calibration method in subsection 2.2.2.2.

2.2.2.1 Flow Meter Configuration and Measurement Technique

the turn

Two id

rical de

prevent

tical no

of 0.406

support

anchored

momentu

point (P)

motion.

Figure 2.

experienc

Si

moment o

ing analys

2.3 that ex

turning var

and Foss (1

Figure 2.3 illustrates the configuration of the flow metering device. The inlet to the turning vane from the upper receiver consists of a 1/4 circle of radius 0.406 m (T). Two identical inlets and turning vanes exist on either side of the AFRD provide a symmetrical delivery of air to the lower receiver. The inlets in the upper receiver are filtered to prevent drafts from influencing the flow into the turning vanes. Each inlet leads to a elliptical nozzle with a 4:1 area contraction ratio (V).

The turning vane itself is created from a curved aluminum sheet (Q) with a radius of 0.406 m and a span of 1.6 m. It is supported by a sharpened point resting on a divoted support (O) on either side of the span. These points act as a low friction pivot and are anchored directly to the ground isolating them from AFRD vibrations. The moment of momentum flux created at the pivot point (O) is balanced by the moment of the force at point (P) which supports the lower portion of the turning vane and resists rotational motion. This force is measured with a ring type strain gauge assembly as illustrated in Figure 2.4. The strain gauge assembly outputs a voltage proportional to the tensile force experienced by the ring.

Since the force ($\vec{F}_P$) can be measured, the control volume formulation of the moment of momentum flux can then used to compute the mass flow rate, $\dot{m}$. The following analysis will explain this in further detail. Using the control volume shown in Figure 2.3 that extends from nozzle outlet to the outlet of the turning vane along the span of the turning vane, the expression for the momentum of momentum flux as derived in Potter and Foss (1982) is given by:

$$\Sigma M_O = \vec{r}_{O-P} \times \vec{F}_P = \int_{cs} \rho [\vec{r} \times \vec{V}] (\vec{V} \cdot \hat{n}) dA \quad (2.1)$$

Followi

less usin

A consta

the upstre

measured

upon the s

the equatio

The coeffic

section 2.2.

m , can be re

2.2.2.2 Flow

Calib

voltage outpu

replaced the t

elliptical shape

erated the inco

Following a simplification and reduction of variables, the integral can be made dimensionless using a characteristic length (L) and velocity (U) (Morris et al., 2001):

$$\vec{r}_{O-P} \times \vec{F}_P = -\rho L^3 U^2 \int_{cs} \left[\frac{\vec{r}}{L} \times \frac{\vec{V}}{U} \right] \left(\frac{\vec{V}}{U} \cdot \hat{n} \right) \frac{dA}{L^2}. \quad (2.2)$$

A constant (c_1) will replace the dimensionless integral if it is not a function of flow rate or the upstream velocity profile. According to Morris et al. (2001), “The magnitude of the measured force can be written in terms of the mass flux $\dot{m} = c_2 \rho L^2 U$, where c_2 depends upon the selection of U and L .” With the cross-product ($\vec{r}_{O-P} \times \vec{F}_P$) written as $l \cdot F_P$ the equation simplifies to

$$F_P = \left[\frac{c_1}{\rho c_2^2 l L} \right] \dot{m}^2 = k \dot{m}^2. \quad (2.3)$$

The coefficient, k , was evaluated by the use of a calibration procedure discussed in subsection 2.2.2.2. From this equation, and given the measured force, F_P , the mass flow rate, $\dot{m}$, can be reliably measured.

2.2.2.2 Flow Meter Calibration

Calibration of the force transducers was necessary to compute a flow rate from the voltage output. A known flow rate was produced through a planar contraction which replaced the test fan at the inlet to the upper receiver. The calibrated inlet consisted of an elliptical shaped nozzle with a 6.3:1 contraction ratio. The relatively high area ratio accelerated the incoming flow ensuring a thin boundary layer at the exit and a C_d value close to

1.0. The

a previous

Th

flow rate

numbers

where U_c

the kinem

increasing

Th

from 150.

regions of

ment thick

ficient. Th

the scale m

Th

hence the

traction, th

where the s

1.0. The evaluation of $C_d = C_d(Re)$ was performed through two separate procedures in a previous study by Neal (2002) and is described in the following paragraphs.

The first procedure involved a 3/16 scale model study. By using a known mass flow rate from a sonic nozzle test stand, the C_d value was found for a range of Reynolds numbers between approximately 50,000 and 150,000 with Reynolds number is defined as:

$$Re = \frac{U_{ci} d_H}{\nu} \quad (2.4)$$

where U_{ci} is the velocity at the contraction inlet, d_H is the hydraulic diameter, and ν is the kinematic viscosity of the laboratory air. From this study, the C_d values increase with increasing Reynolds number to $C_d=0.980$ for $Re=150,000$.

The second procedure involved a hotwire study with a Reynolds number range from 150,000 to approximately 775,000. The hotwire survey traversed the near wall regions of the exit of the contraction on both the length and width sections. The displacement thickness found from this investigation was then used to estimate the discharge coefficient. The C_d value found from this procedure was also 0.980 confirming the results of the scale model study.

The maximum Reynolds number for the present study was approximately 800,000, hence the use of a C_d value of 0.980 was justified. Given the C_d value of the planar contraction, the mass flow rate through the AFRD facility was found by:

$$\dot{m} = \rho A_{ci} C_d \sqrt{\frac{2}{\rho} (P_{atm} - P_{receiver})} \quad (2.5)$$

where the subscript “ci” refers to the contraction inlet.

A
the prime
pressure t
The calibr
transfer fu

In equation
transducer
flow rates
The calibr
include th
ment devi

Sim
flow mete
to:

where the
the mass
with the v

A range of flow rates was induced through the calibrated contraction by throttling the prime mover. The voltage output from both force transducers and from the differential pressure transducer were recorded by an A/D acquisition system described in Section 3.1. The calibration data were plotted and fitted with a linear least squares fit. The resulting transfer functions for the North and South flow meters were:

$$\dot{m}_n = 2.7816 \sqrt{E_n - offset} \text{ (kg/s)} \quad (2.6)$$

$$\dot{m}_s = 2.9214 \sqrt{E_s - offset} \text{ (kg/s)} \quad (2.7)$$

In equations 2.6 and 2.7, the term ‘*offset*’ denotes the “pre-load” voltages from the force transducers. The pre-load was necessary to avoid fully unloading the strain gauges at low flow rates due to the vibrations in the system. The fitted data are shown in Figure 2.5. The calibration data were acquired in two separate trials. Correspondingly, the plots include the results from both trials to qualitatively show the repeatability of the measurement devices.

Since two independent measurements of flow rate were acquired from the two flow meters but only one integral measurement is required, they were averaged according to:

$$\dot{m}_{measured} = \frac{\dot{m}_n + \dot{m}_s}{2} \quad (2.8)$$

where the subscripts ‘*n*’ and ‘*s*’ denote the “north” and “south” flow meters. Although the mass flow rate is the actual quantity measured, the data are presented in the results with the volume flow rate, Q , as given by:

$$Q_{measured} = \frac{\dot{m}_{measured}}{\rho} \quad (2.9)$$

The difference
ically no

2.2.3 Horizontal

A
probe down
the travers
steel ring
tional mot
box. Rad
#MB4024
mounted o
ing in this
travel was
wire probe
faced with
in a custom
0.05mm/m

2.3 Experimental

2.3.1 Prototype

The
2.7. The di
the fixed ver

The differences in the measured volume flow rates between the two flow meters were typically no greater than 1.2%.

2.2.3 Hot-wire Traversing Mechanism

A stepper motor controlled traverse system allowed movement of the hotwire probe downstream of the fan in the r , θ , and z directions. A schematic representation of the traverse is shown in Figure 2.6. The entire traverse was mounted on a 1.52 m diameter steel ring that surrounded the fan allowing it to be moved in the azimuthal direction. Rotational motion of the steel ring was provided by a 6:1 pulley system and a 100:1 worm gear box. Radial motion of the hotwire probe was provided by a linear slide (Velmex Inc. #MB4024BJ-S4-20) with a maximum horizontal travel of 51 cm. Two linear slides were mounted on the steel ring (180° apart) but only one was necessary for the probe positioning in this study. A vertical linear slide (Velmex Inc. #MB4024BJ-S4-16) with 40.5 cm of travel was mounted to each horizontal linear slide to provide axial movement of the hotwire probe. All stepper motors were controlled by a Velmex #NF90-3 controller interfaced with a PC from which stepping commands were sent. The commands were written in a custom Matlab code. The manufacturer supplied accuracy of the linear slide was 0.05mm/meter.

2.3 Experimental Aerodynamic Shroud

2.3.1 Prototype Design

The experimental aerodynamic shroud assembly is shown schematically in Figure 2.7. The different parts of the complete system include the radiator, the shroud plenum, the fixed venturi shroud and the auxiliary pressurization source. A radiator was included

to mimic

condition

mance of

Th

Modine v

directly c

sealed of

upstream

static pre

tor and pr

ator and t

T

upper sec

axisymm

upper sh

designed

exiting t

T

fan diam

ter. To

on repr

shroud

dynam

to mimic an installed condition. It was important to include the pressure drop and flow conditioning associated with the radiator because of the possible influence on the performance of the cooling fan.

The radiator used in the experimental study was a production model radiator from Modine with an effective cooling area of 0.83 m by 0.83 m. The radiator was placed directly on top of the shroud plenum and the area around the sides of the radiator was sealed off to prevent air from leaking into the area downstream of the radiator and upstream of the fan inlet. The downstream side of the radiator frame was equipped with a static pressure tap to measure the pressure drop across the restrictive elements. The radiator and pressure tap location is pictured in Figure 2.8. The axial distance between the radiator and the leading edge of the fan was approximately 140 mm.

The shroud plenum was built in two sections. The lower section is fixed and the upper section moves vertically to adjust the gap height from which the jet originates. An axisymmetric, contoured carbon-fiber piece with a sharp trailing edge is attached to the upper shroud section to create the thin wall jet from the shroud plenum. The contour is designed to provide a smooth inlet to the fan and to act as a nozzle to accelerate the air exiting the plenum.

The geometric dimensions of the aerodynamic shroud were chosen based on the fan diameter, the tip clearance, and the typical dimensions of a radiator for this fan diameter. To replicate a field application, the design of the aerodynamic shroud was dependent on representative spatial constraints of an underhood installation. For this reason, the shroud plenum was made much smaller relative to the fan diameter than the previous aerodynamic shroud investigations of Morris (1997) and Neal (2002). Specifically, the shroud

plenum w

the plenu

A calibrat

TH

tip cleara

mill with

previous r

1.2. The

the shrou

Figure 2.

between

sufficient

eter of 80

delivery

2.3.2 SH

T

the plen

relation

Compre

from th

of Mon

inlet.

plenum was 0.891 m square, or 1.25 times the diameter of the fan. The upper section of the plenum rested on four 1/4 inch threaded rods which were used to adjust the gap height. A calibrated “wedge” was used to set this height to the desired value.

The inside diameter of the venturi shroud was 729 mm corresponding to a 9 mm tip clearance for the 711 mm fan. The shroud was fabricated from wood on a five-axis mill with a curved inlet and outlet. The use of the curved outlet was motivated from the previous research of Morris (2001). Those findings are discussed in more detail in section 1.2. The upper section of the venturi shroud had a thinner cross section to accommodate the shroud plenum and to allow the coverplate to be attached to the venturi shroud (see Figure 2.7). The shroud plenum was constructed on the coverplate with a 22 mm gap between the venturi shroud and the plenum sidewalls at the four quadrants. This gap was sufficient to allow a uniform axisymmetric jet to exit the plenum. Four inlets with a diameter of 89 mm each are located in the four corners of the shroud plenum to allow for the air delivery as shown in Figure 2.7.

2.3.2 Shroud Plenum Pressurization System

The plenum was pressurized by injecting air into the four inlets on the underside of the plenum. Figure 2.9 illustrates the piping system and auxiliary blower arrangement in relation to the testing facility. Air was delivered by a centrifugal blower (Spencer Turbo-Compressor, Lot NO. 37215) through 3” PVC piping. The piping entered the plenum from the upper receiver. This arrangement differed from the experimental configurations of Morris (1997) and Neal (2002) because of the presence of a radiator upstream of the fan inlet.

centrifugal

the inlet

rate was

where A_1

nozzle. m

the piping

used to m

the shroud

rate was

where

A

of P_{shroud}

The flow to the shroud was adjusted with a butterfly valve on the outlet side of the centrifugal blower. A Smith and Wang nozzle with a known discharge coefficient, C_d , on the inlet side metered the flow (see Figure 2.9). Using this method the shroud mass flow rate was found according to

$$\dot{m}_{shroud} = \rho_{atm} A_{nozzle} C_d \sqrt{(P_{atm} - P_{nozzle})} \quad (2.10)$$

where A_{nozzle} and P_{nozzle} is the area and static pressure at the throat of the Smith and Wang nozzle, respectively. Since the air flow underwent viscous heating through the blower and the piping system, a thermocouple (Omega Engineering Inc., model 670, Type ‘T’) was used to measure the air temperature in the shroud. A small hole was drilled into the top of the shroud and the thermocouple was suspended into this hole. The shroud volume flow rate was then adjusted for each data point according to the measured temperature:

$$Q_{shroud} = \frac{\dot{m}_{shroud}}{\rho_{shroud}} \quad (2.11)$$

where

$$\rho_{shroud} = \frac{P_{shroud} + P_{atm}}{RT_{shroud}} \quad (2.12)$$

A static pressure tap in the bottom on the shroud plenum allowed the measurement of P_{shroud} .



Figure 2.1 Prototype test fan ($D = 0.711\text{ m}$)

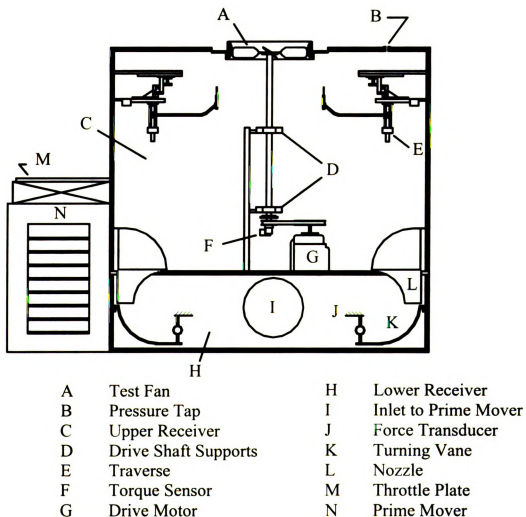


Figure 2.2 Schematic representation of the AFRD flow facility

O

R_1

F_c

Figure 2.4 DC

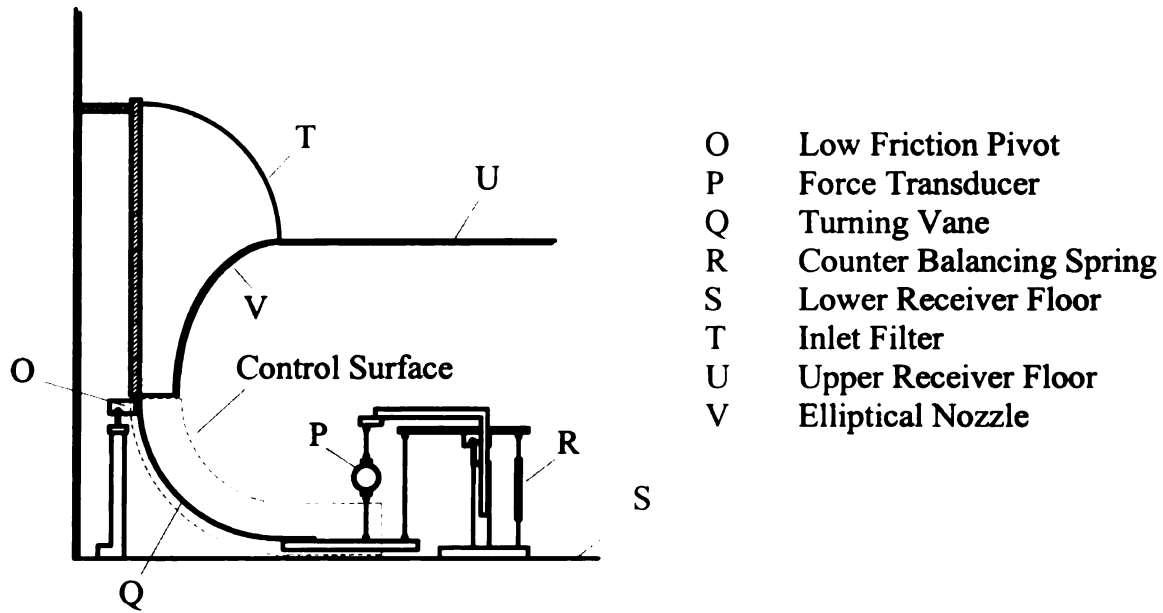


Figure 2.3 Schematic representation of the flow metering device

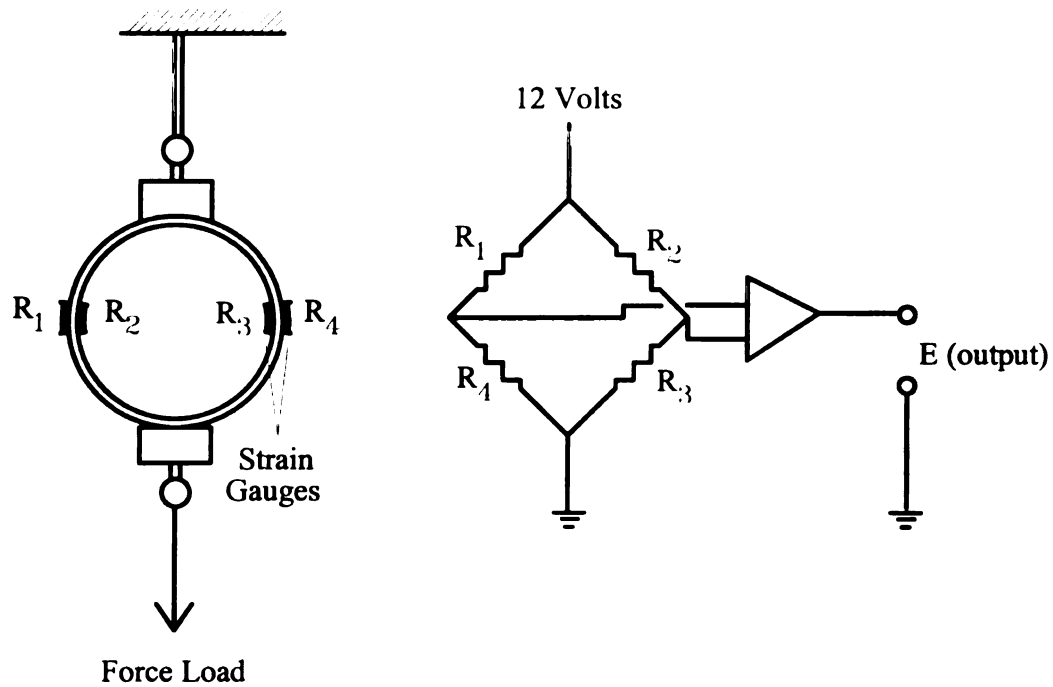


Figure 2.4 Details of the force transducer for the flow metering device (Neal, 2002)

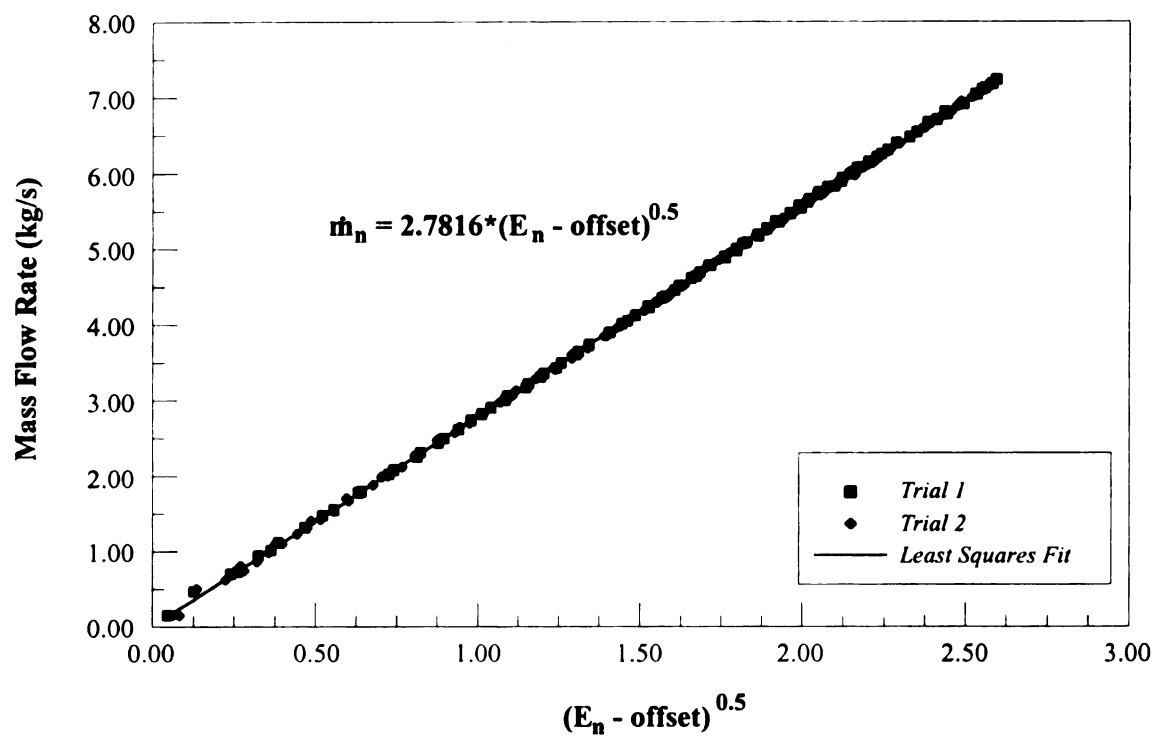
Mass Flow Rate (kg/s)

8
7
6
5
4
3
2
1
0

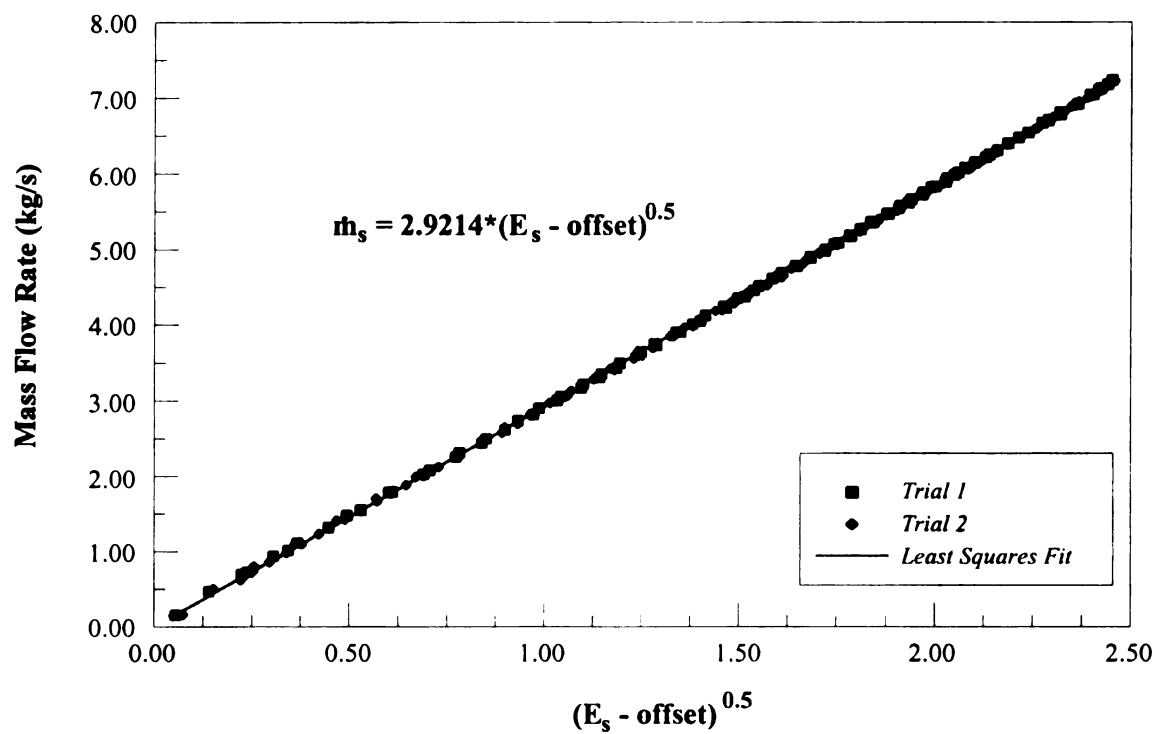
Mass Flow Rate (kg/s)

8.00
7.00
6.00
5.00
4.00
3.00
2.00
1.00
0.00
0.00

Figure



a) North Flow Meter Calibration



b) South Flow Meter Calibration

Figure 2.5 Calibration data and fit for the flow metering devices

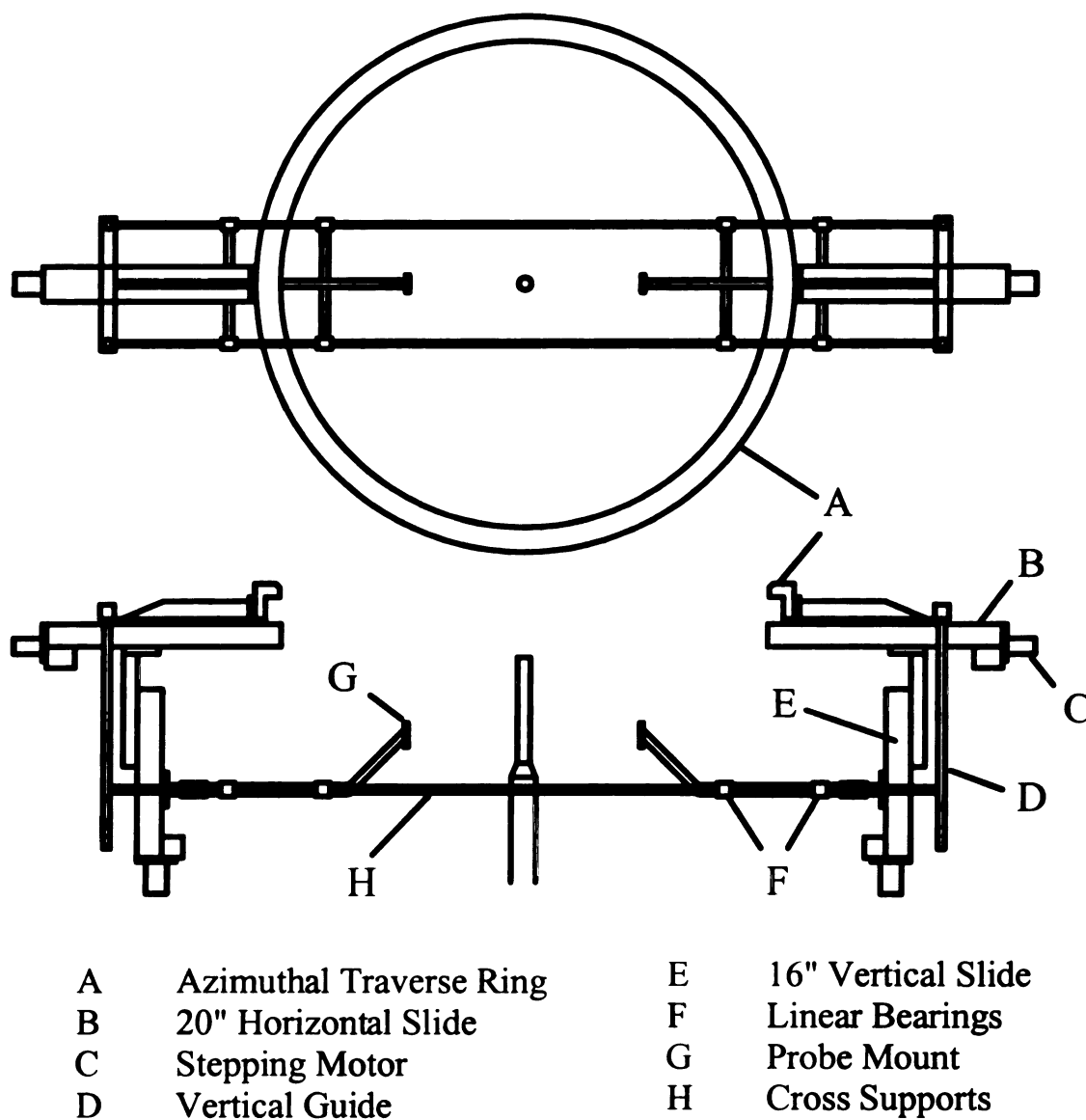


Figure 2.6 Schematic representation of AFRD hotwire traverse

U
S

Ventu

Figure 2.7

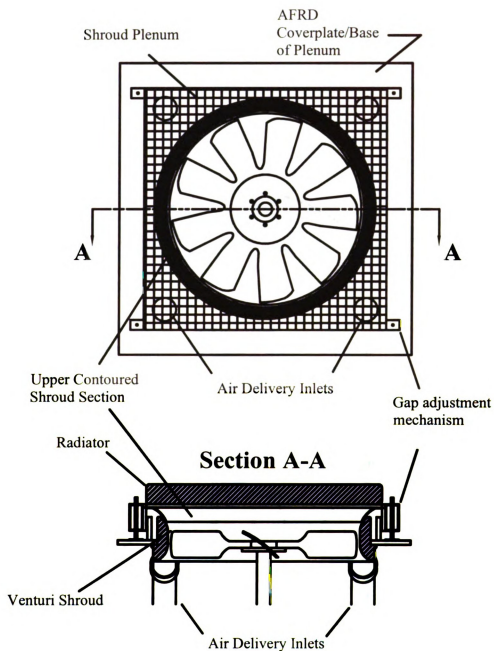
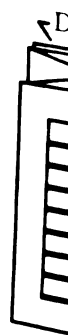


Figure 2.7 Schematic representation of the aerodynamic shroud and plenum configuration (Radiator not shown in top view)

Figure



A
B
C
D

Figure 2.9 S

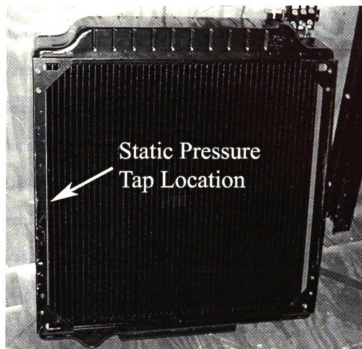
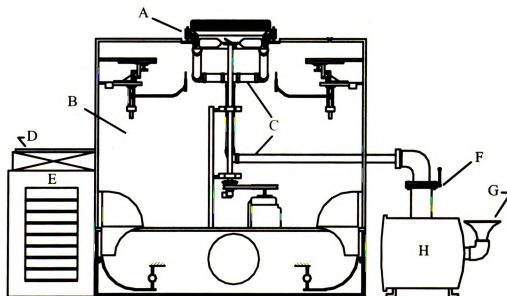


Figure 2.8 Radiator with static pressure tap downstream of the cooling fins



- | | | | |
|---|-----------------------------|---|-------------------------|
| A | Aerodynamic Shroud Assembly | E | Prime Mover |
| B | Upper Receiver | F | Spencer Blower Throttle |
| C | Shroud Piping | G | Smith and Wang Nozzle |
| D | Prime Mover Throttle | H | Spencer Shroud Blower |

Figure 2.9 Schematic representation of AFRD with aerodynamic shroud assembly and pressurization system

3.0 M

TH

obtain the

the first h

out an aer

in Section

investigate

program.

wake of the

about the P

calibration

rithms used

tion 3.4.

3.1 Data A

All

using a 16 b

an 8 channe

tion of 0.3 m

nels of data

format and tr

3.0 Measurement Equipment and Technique

The details of the measurement equipment, techniques, and processing used to obtain the results are presented in this chapter. Integral measurements were acquired for the first half of the experimental program to characterize fan performance with and without an aerodynamic shroud. The quantities measured and the methods used are described in Section 3.2. The information gathered from these measurements was used to further investigate the mechanistic features of the fan flow in the second half of the experimental program. This was accomplished by acquiring detailed velocity measurements in the wake of the fan using particle image velocimetry (PIV) and hot-wire anemometry. Details about the PIV system and vector processing are given in section 3.3. Finally, the hot-wire calibration method and technique using the double-X probe as well as the processing algorithms used to simultaneously resolve three components of velocity are presented in section 3.4.

3.1 Data Acquisition System

All integral and hot-wire data acquired in this thesis were sampled and recorded using a 16 bit, 8 channel Iotech Wavebook/516 A/D acquisition module accompanied by an 8 channel expansion module. The A/D system had a range of ± 10 volts with a resolution of 0.3 mV. Measurements were acquired as “sample-and-hold” meaning that all channels of data were acquired and recorded simultaneously. The data were saved in a binary format and transferred to a PC for processing using custom Matlab programs.

3.2 In

overall

efficien

specify

are use

shroud

The eq

are criti

definitio

dure use

3.2.1 M

3.2.1.1

F

formance

the AFR

HD head

face of th

the pressu

Validyne r

used to me

sures great

dynes were

3.2 Integral Measurements

In engineering practice, it is common to use integral quantities to examine the overall performance of a fan, blower, or compressor. Namely, the pressure rise ΔP , and efficiency η , as a function of volume flow rate Q , are significant variables to consider in specifying an air moving device for a particular application. In this study, these variables are used to compare the baseline performance of the cooling fan without the aerodynamic shroud to the performance enhancement (or degradation) with the aerodynamic shroud. The equipment used to measure these quantities and produce the performance curves that are critical in assessing the effect of the aerodynamic shroud is given in Section 3.2.1. The definition of the variables presented in the results is given in Section 3.2.2. and the procedure used to acquire the integral measurements is described in Section 3.2.3.

3.2.1 Measurement Equipment

3.2.1.1 Pressure Transducers

Four separate pressure measurements were necessary to fully characterize the performance of the fan and shroud combination. The static pressure in the upper receiver of the AFRD was measured with a 10 Torr MKS Baratron pressure transducer, model 398 HD head unit with a Type 270B signal conditioner. The static pressure at the downstream face of the radiator was measured to acquire both the pressure drop across the radiator and the pressure rise across the fan (see section 3.2.2). This pressure was measured using a Validyne model DP15-20 with a range of 0-868 Pa. A Validyne DP10-22 head unit was used to measure shroud pressure when the shroud pressure was under 1245 Pa. For pressures greater than this value, a Validyne model DP15-26 was employed. The three Validynes were each used in conjunction with a model CD15 carrier demodulator. A fourth

pressu

involv

attache

pressu

tude of

Type 2

transdu

3.2.1.2

Torquem

directly t

accuracy

earthy of

ity of ± 0.1

from bendi

tribute to th

The

cision machi

lever arm to

tioner and cor

duced an outpu

A tare n

the friction from

pressure was necessary to obtain the flow rate through the shroud flow system which involved measuring the static pressure at the throat section of the Smith and Wang nozzle attached to the inlet of the auxiliary shroud blower as described in Section 2.3.2. This pressure was measured by a 10 Torr or 100 Torr MKS Baratron (depending on the magnitude of the pressure), model 398 HD head unit with a Type 270 signal conditioner and a Type 274 head selector. The reported accuracy of the Baratron and Validyne pressure transducers is $\pm 0.08\%$ and $\pm 0.25\%$ respectively.

3.2.1.2 Torque Measurement

A torque sensor from S. Himmelstein and Co. (MCRT 3120TN(2-2)NN Pulley Torquemeter) was used to measure the torque applied to the drive shaft. It was connected directly to the lower end of the shaft below the pulley and had a range of ± 200 in.lb. The accuracy of the torque readings were supplied by the manufacturer and included a nonlinearity of ± 0.2 in.lb. (0.1% of full scale), a hysteresis of ± 0.2 in.lb., and a non repeatability of ± 0.1 in.lb. It was equipped with a strain gauge bridge designed to cancel signals from bending and thrust loads. Hence, any bending load from belt tension does not contribute to the overall measurement.

The torque sensor was calibrated prior to experimental data being acquired. A precision machined lever arm was fit directly on the torque sensor. A load was applied to the lever arm to produce a known torque. This torque was then entered in the signal conditioner and controller module (S. Himmelstein and Co., 700 Series) so that the module produced an output of ± 10 volts directly proportional to the full scale torque range.

A tare measurement was taken prior to testing to account for the torque applied by the friction from the bearings. The friction is a function of grease viscosity, and in turn, a

function
before
for app
measur
ing the

approxin
measured
to testing

3.2.1.3 C

A

tional spe
infrared in
spaced ho
4093B), a
ual hole in
recorded d
rotational s

3.2.2 Defin

One c
of the pressu

function of temperature. In this manner, the bearings must reach a steady state condition before a constant tare value can be found. To find a proper tare value, the shaft was run for approximately 2 hours at the testing speed with the fan removed. A plot of this torque measurement is shown in Figure 3.1. The measured torque was then adjusted by subtracting the tare value

$$\tau_{actual} = \tau_{measured} - \tau_{tare} \quad (3.1)$$

It is evident from the plot that the tare decreases with an exponential decay to approximately 5.5 in-lb. After 45 minutes, the torque was within 0.3 in.lb. of the torque measured at 150 minutes. Consequently, the fan was operated for at least 45 minutes prior to testing with $\tau_{tare} = 5.5$ in-lb.

3.2.1.3 Optical Encoder

An optical encoder was used to accurately set, monitor, and record the fan rotational speed. The optical encoder used for the integral measurements consisted of an infrared interrupt circuit (Omron EE-SG3, slot sensing method) and a disk with 60 evenly spaced holes attached to the fan shaft. Using a Schmitt trigger (Fairchild Semiconductor 4093B), a TTL signal was produced (0 or 5 volts depending on the location of an individual hole in relation to the infrared sensor). A time series of the measured voltages was recorded during data acquisition and subsequently processed to determine an accurate rotational speed.

3.2.2 Definition of Variables

One of the most common ways to characterize the performance of a fan is in terms of the pressure rise (ΔP) as a function of flow rate (Q). The measurement of Q is

describ

into ac

tive of

air by i

is adde

quantity

vant flo

T

ΔP is d

blades an

ciently "

However

clearly de

where P_r

$P_{receiver}$

clarify this

each other

sure drop a

It is

in dimension

aerodynamic

described in section 2.2.2, however, the added flow from the shroud plenum must be taken into account in a powered shroud condition. It is important to note that a primary objective of the aerodynamic shroud is to increase the transfer of thermal energy to the ambient air by increasing the amount of convected air through the radiator. Since the shroud flow is added downstream of the radiator, this amount of flow must be subtracted from the quantity measured by the flow meters. Hence, in the powered shroud condition the relevant flow rate is

$$Q_{fan} = Q_{total} - Q_{shroud} \quad (3.2)$$

The definition of ΔP must also be clarified according to the application. Usually, ΔP is defined as the static pressure differential between the downstream side of the fan blades and the upstream approach flow. In the absence of restrictive elements and sufficiently “far upstream”, the upstream static pressure is simply the quiescent atmosphere. However, when restrictions are present (e.g., a radiator) the pressure differential must be clearly defined. In the present investigations, the pressure rise, ΔP , is defined as:

$$\Delta P_{fan} = P_{receiver} - P_{radiator} \quad (3.3)$$

where $P_{radiator}$ is the static pressure measured just downstream of the radiator and $P_{receiver}$ is the static pressure in the AFRD upper receiver. Figure 3.2 is presented to clarify this configuration. Both $P_{radiator}$ and $P_{receiver}$ were measured independently of each other and referenced to atmospheric pressure. This method was used so that the pressure drop across the radiator and the total system pressure rise could be measured as well.

It is typical in turbomachinery literature to represent the aforementioned variables in dimensionless form. Morris (1997) performed a dimensional analysis in the previous aerodynamic shroud investigations and the resulting dimensionless variables will be used

in this

the tip

effects

number

(Barans

and the

where A

shroud an

Th

cally simi

relationsh

effects are

An

of volume

ciency is u

the power i

where

in this study as well. The length and velocity scales used were the fan diameter (D) and the tip speed (U_{tip}). Because the velocity and pressure fields are insensitive to viscous effects and governed solely by inertial effects and boundary conditions, the Reynolds number is not considered a significant parameter. This is customary in many fan studies (Baranski, 1997). Hence, the pressure coefficient is defined as:

$$\psi = \frac{\Delta P_{fan}}{\rho U_{tip}^2} \quad (3.4)$$

and the flow coefficient is defined as

$$\phi = \frac{Q_{fan}/A_{flow}}{U_{tip}} \quad (3.5)$$

where $A_{flow} = \frac{\pi}{4}(D_{shr}^2 - D_{hub}^2)$ represents the effective annular flow area between shroud and the fan hub.

These non-dimensional variables are especially useful when comparing geometrically similar fans of different diameters or operating speeds. In those cases, the functional relationship, $\psi = \psi(\phi)$, will remain constant. The limiting case is when compressibility effects are present.

Another integral quantity of considerable interest is the fan efficiency as a function of volume flow rate. Many definitions of efficiency exist. For this study, a static efficiency is used where it is defined as the ratio of energy input to the fluid in unit time over the power input to the fan. In terms of the relevant variables, it is defined as

$$\eta = \frac{\Delta P_{fan} Q_{fan}}{\dot{\phi}_{fan}} \quad (3.6)$$

where

and τ

This d

from t

assess

accom

The rea

in effi

case, the

where

In equatio

air deliver

based on th

ris estimate

could be use

mine the act

was dictated

$$\dot{\phi}_{fan} = \tau \cdot \Omega \quad (3.7)$$

and τ is the shaft torque that is required to rotate the fan at Ω radians/second.

This definition of power is instructive because it is independent of the mechanical losses from the motor, belts, and bearings in the experimental configuration. Therefore, it is an assessment of the fan only.

In the active shroud condition, a modified definition of efficiency is necessary to accommodate the added power consumption associated with the aerodynamic shroud. The reader is reminded that, in using the aerodynamic shroud, it is desired to realize a gain in efficiency greater than the additional power needed to pressurize the shroud. In this case, the efficiency is defined as:

$$\eta = \frac{\Delta P_{fan} Q_{fan}}{\dot{\phi}_{fan} + \dot{\phi}_{shroud}} \quad (3.8)$$

where

$$\dot{\phi}_{shroud} = \frac{\Delta P_{shroud} \cdot Q_{shroud}}{\eta_{shroud}}. \quad (3.9)$$

In equation 3.9, the term η_{shroud} is present to account for the inefficiencies of a shroud air delivery system. An estimated value $\eta_{shroud} = 0.7$ was used for the calculations based on the previous aerodynamic shroud work of Morris (1997) and Neal (2002). Morris estimated this number from typical operating efficiencies of centrifugal blowers that could be used for a practical aerodynamic shroud application. It was not useful to determine the actual efficiency of the experimental aerodynamic shroud since the configuration was dictated by the constraints of the experimental setup and available hardware.

shrou

cally.

shrou

plenur

and

The func

The com

variable r

sionless s

two. The

This variab

the contribu

expressed as

3.2.3 Data A

The eq

ties of interest.

It is clear that the additional boundary conditions imposed by the aerodynamic shroud modify the functional dependence of the flow coefficient and efficiency. Specifically, the pressure coefficient and efficiency become a function of the flow coefficient, shroud plenum pressure, and the shroud flow rate. In non-dimensional terms the shroud plenum pressure and shroud flow rate are:

$$\lambda = \frac{\Delta P_{shroud}}{\rho U_{tip}^2} \quad (3.10)$$

and

$$\Lambda = \frac{Q_{shroud}}{U_{tip} A_{flow}}. \quad (3.11)$$

The functional relationships are then modified to $\psi = \psi(\phi, \lambda, \Lambda)$ and $\eta = \eta(\phi, \lambda, \Lambda)$. The complexity of the data analysis is greatly increased using three variables. Thus, a new variable representing a dimensionless shroud power is created by combining the dimensionless shroud power and shroud flow rate, therefore, reducing the number of variables to two. The definition adapted from Morris (1997) is:

$$\chi = \frac{\Delta P_{shroud} Q_{shroud}}{\rho U_{tip}^3 A_{flow}}. \quad (3.12)$$

This variable simplifies the data analysis and it provides a useful method of characterizing the contribution of the shroud. Specifically, using χ , the ψ and η functions can be expressed as $\psi = \psi(\phi, \chi)$ and $\eta = \eta(\phi, \chi)$.

3.2.3 Data Acquisition and Procedure

The equipment described in Section 3.2.1 was used to acquire the integral quantities of interest. All channels of data were acquired for 10 seconds at a sampling rate of 10

kHz. T

acquire

tion. To

during t

shroud p

delivere

tests, the

other. J

moved i

complet

repeated

rate con

(15 data

character

reverse

sure rise

above.

experim

gap hei

refers to

Three di

kHz. The main portion of the integral data and the results presented in this thesis were acquired at a fan speed of 950 rpm.

A characteristic curve, or performance curve, was produced for each shroud condition. To clarify, the term “shroud condition” refers to the state of the aerodynamic shroud during the test. The “shroud off” condition means that no auxiliary air is supplied to the shroud plenum. A “powered shroud” or “active shroud” condition implies that air is being delivered to the shroud creating an annular jet upstream of the fan inlet. For all integral tests, the procedure involved acquiring data from one extreme of the operating range to the other. For instance, the test would start at a very low pressure rise. The throttle was moved in small increments, acquiring data at each increment, until the AFRD throttle was completely closed to create a “full shut-off” condition (largest ψ). This procedure was repeated going from the full shut-off condition back to the low pressure rise, high flow rate condition to test for repeatability in the measurement. Approximately 30 data points (15 data points in each direction) were acquired over the course of the experiment to fully characterize the performance curve. To test the effects of hysteresis, this method was reversed for some tests, starting from full shut-off, increasing flow rate until a low pressure rise, and returning back to the starting pressure rise condition.

The procedure for a “shroud off” condition was relatively simple as explained above. Conversely, the powered shroud conditions presented an added complexity in the experimental variables. All of the powered shroud integral data consisted of a constant gap height and shroud pressure for the duration of the experiment. The gap height, g , refers to the space between the annular jet nozzle and the fixed venturi shroud surface. Three different gap heights were used including a 2 mm, 3 mm, and 4 mm gap. Further-

more, for

sures ra

3.3 Par

R

ate a vel

used to

wake of

ment use

compute

experime

3.3.1 PI

Th

computer

generator

Th

4000 sens

in front of

A

all PIV me

beam into a

a waveleng

more, four different shroud pressures were tested for each gap height. The shroud pressures ranged from nominally 250 Pa to 1500 Pa depending on the gap height.

3.3 Particle Image Velocimetry Measurements

Particle image velocimetry (PIV) is an instantaneous measurement used to evaluate a velocity field across a planar area. In this study, a two-component PIV system was used to measure the axial and radial components of velocity in the outer region of the wake of the fan. Both time averaged and phase averaged data were acquired. The equipment used in the PIV measurements is described in Section 3.3.1. The technique used to compute and validate vectors is described in Section 3.3.2 followed by details of the experimental configuration and procedure in Section 3.3.3.

3.3.1 PIV Equipment

The PIV system consisted of a CCD (charge coupled device) camera, a laser, a computer with the main processing unit and necessary software, and an atomizing aerosol generator.

The 2k x 2k pixel, 12-bit camera was a Flowmaster 2S model with a Kodak KAI 4000 sensor capable of 15 single frames per second. A 532 nm band pass filter was used in front of the lens to filter out ambient light.

A New Wave Research Minilase III, Nd:YAG dual-head 10 Hz laser was used for all PIV measurements. The laser was equipped with light-sheet optics to expand the laser beam into a planar sheet. The manufacturer specified the output of the laser to be 50 mJ at a wavelength of 532 nm with a pulse width of 5-7 ns.

A
ing Unit
data. La
ment. Th
pulse as v
used for a

Th

H with an
octyl sebac
0.2 μ m and

A s
was used fo
a reflective
was used to
trip the "log
averaged da

3.3.2 PIV V

In thi
mode. The in
is performed
tion of the pro
ical size of the

A computer equipped with dual Intel Xeon processors and a Programmable Timing Unit (PTU) card was used to control the PIV system as well as acquire and process the data. LaVision's proprietary software, DaVis, was the main interface with the PIV equipment. The software allowed the synchronization of the camera exposure with the laser pulse as well as the control of the many other laser and camera settings. DaVis was also used for all processing and post-processing of the acquired images.

The flow was seeded with a Topas Atomizer Aerosol Generator, model ATM 210/H with an adjustable particle production rate. The aerosol fluid used in the generator was octyl sebacate (DEHS) and the manufacturer reported particle sizes generated are between $0.2\mu\text{m}$ and $0.3\mu\text{m}$.

A slightly different configuration of the optical encoder described in Section 3.2.1 was used for the particle image velocimetry measurements. Instead of an interrupt circuit, a reflective type receiver/transmitter (Omron EE-SF5, phototransistor sensing method) was used to output a TTL level signal. A narrow strip of a reflective material was used to trip the "logic high" once every revolution. This provided the trigger signal for the phase averaged data that will be described in Section 3.3.3.

3.3.2 PIV Vector Processing

In this study, the CCD camera records images in the double frame/double exposure mode. The images are divided into separate interrogation regions and a cross-correlation is performed on each region to resolve a single vector (see Figure 3.3). The spatial resolution of the processed data is dictated by the number of interrogation regions and the physical size of the image area.

An
els for each
tion is the
than half t
reduced. T
longer be c

where ds is
desired inte

An a

ing to manag

Using multi

lowing. Dur

dow. In the

window and

number of ve

resolution wit

further increas

Figure 3.4. TH

ing with an ini

32x32 pixels w

Validation

cessing. Spurio

An interrogation region is composed of a number of pixels. As the number of pixels for each interrogation region decreases, the spatial resolution is increased. The limitation is the magnitude of particle shift. In particular, if the mean particle shift is greater than half the size of the interrogation window, then the accuracy of the correlation is reduced. This is because some particles move outside the interrogation region and can no longer be correlated. The optimum particle shift is given by Keane and Adrian (1990):

$$0.1px < ds < \frac{1}{4}d_{IntWin} \quad (3.13)$$

where ds is the separation of the particle images on the CCD, and d_{IntWin} represents the desired interrogation window size in pixels (designated as px.) (LaVision GmbH, 2002).

An adaptive multi-pass feature in the DaVis software was used in the PIV processing to manage the issue of particle shift and optimize the size of the interrogation window. Using multiple passes, the accuracy of the correlation is increased as explained by the following. During the first pass, a reference vector is calculated for each interrogation window. In the next pass, the interrogation window is half the size of the first interrogation window and shifted according to the magnitude and direction of the reference vector. The number of vectors calculated with each pass is increased, therefore, increasing the spatial resolution without sacrificing correlations. Moreover, a window overlap can be used to further increase the spatial resolution. An example of a 50% window overlap is shown in Figure 3.4. The PIV measurements acquired in this study use a total of four passes starting with an initial pass of 128x128 pixels and a 50% overlap. The final window size is 32x32 pixels with a 50% overlap to result in a total of 16,384 vectors for each image.

Validation schemes were used to remove spurious vectors resulting from the processing. Spurious vectors arise from a number of undesirable circumstances including

noise, poor

are available

study was

and highest

study, if the

is to prevent

a "median

vectors and

ables to set

magnitude

less than the

at least three

or the vector

remove as m

cedure was fo

3.3.3 Data A

The pr

directions near

rpm. Due to th

were used in the

tions is shown in

2 overlapped reg

the camera, region

noise, poor seeding, or high velocity gradients. A great number of validation techniques are available in the DaVis software package. The first validation technique used in this study was a “peak ratio” validation. The peak ratio is the ratio between the first and second highest correlation peaks. The criterion can be set by the user, and in the case of this study, if the peak ratio was less than a value of 1.2 the vector was tagged as spurious. This is to prevent any ambiguity of the correct shift. A second validation technique was termed a “median filter”. This involved computing a median vector from the eight surrounding vectors and comparing that value to the middle vector. Again, there are user defined variables to set the criteria for this action. In this study, the middle vector was removed if its magnitude was greater than three times the rms of the neighboring vectors or if there were less than three neighboring vectors. In addition to this, a vector was inserted if there were at least three good vectors surrounding it. The inserted vector is either the original vector or the vector from the second, third, or fourth lower correlation peaks. It is important to remove as many spurious vectors as possible when computing averages. Hence, this procedure was followed in the present investigations.

3.3.3 Data Acquisition and Procedure

The present study seeks to resolve the velocity components in the axial and radial directions near the outer portion of the fan exit plane. All PIV data were acquired at 950 rpm. Due to the limited size of the measurement field, two separate measurement planes were used in the PIV experiments. A schematic representation of the measurement locations is shown in Figure 3.5 and denoted as “region 1” and “region 2”. As shown, region 2 overlapped region 1 by approximately 12 mm. Due to the differences in placement of the camera, region 2 is a larger field of view, or smaller magnification, than that of region

1. Hence

lated even

Ti

ditions. I

processed

the veloci

The phase

phase trig

control of

It

adjust the

vidual fa

tion. Th

the instan

lowing.

encoder

were the

represen

check an

ples. N.

quency.

1. Hence, region 1 has a slightly greater spatial resolution. Specifically, a vector is calculated every 0.83 mm in region 1 compared to 0.96 mm in region 2.

Time averaged and phase averaged data were acquired for the selected shroud conditions. For the time averaged data, approximately 200 vector maps were recorded and processed in both regions of interest. Phase averaged data were also acquired to capture the velocity field at eight different phase positions between two blades of the rotating fan. The phase averaged acquisition system used the TTL output on the optical encoder as the phase trigger. A “logic high” from the TTL signal used with a delay time allowed full control of the phase plane in which data were acquired.

It was instructive to know the position of the fan at the first phase plane and then adjust the phase positions in equal increments to capture the dynamics between two individual fan blades. The leading edge of the fan blade was chosen as the first phase position. Thus, it was necessary to find the proper delay time between the trigger signal and the instant that the leading edge crossed the PIV laser plane. This is described by the following. The optical encoder on the fan shaft was used in conjunction with an optical encoder that was aligned with the leading edge of the fan blade. The two output signals were then acquired while the fan was run at various speeds from 20 rpm to 130 rpm. A representative output signal is shown in Figure 3.6. The various speeds provided an error check and a larger sample from which to average. From these data, the number of samples, N , between the rising edges of the two signals could be found. Since the data frequency, f , and the speed of the test, ω , were known, the phase lag, α , was found by:

$$\alpha = \frac{N\omega}{f} \quad (3.14)$$

where α is
speed, Ω .

where Ω is

To
was increas

leading edge

ments. Bec

100 images

3.4 Hot-wire

Hot-w

nature of the

simultaneous

downstream

averaged using

continuous sa

that were not a

uration and eq

newly fabricate

ble-x hot-wire

3.4.2 followed b

where α is in radians. With this information the proper delay time, t_d , for a given fan speed, Ω , is simply:

$$t_d = \frac{\alpha}{\Omega} \quad (3.15)$$

where Ω is in radians/second.

To acquire data at the different phase positions between the blades, the delay time was increased according to the increase in phase lag. Since there were 40° between two leading edges, data were acquired from 0° (leading edge) to 35° in five degree increments. Because of the size and number of files involved in the phase averaged data, only 100 images were recorded for each phase angle.

3.4 Hot-wire Measurements

Hot-wire anemometry data were acquired to further evaluate the instantaneous nature of the flow field in the wake of the fan. A double-X hot-wire probe was used to simultaneously measure three components of velocity at several radial locations just downstream of the fan exit plane for the selected shroud conditions. The data were phase averaged using the signal from the optical encoder. Since the hot-wire measurement is a continuous sample of a the velocity field, these results give detailed statistical information that were not available from the other measurement methods. The hot-wire sensor configuration and equipment is described in Section 3.4.1. The hot-wires were calibrated in a newly fabricated calibration unit designed specifically for the purpose of calibrating double-x hot-wire probes. A detailed description of this calibration unit is given in Section 3.4.2 followed by the calibration method in Section 3.4.3. The actual method of acquiring

the data is

tion of the

3.4.1 Hot-

The

of two x-w

a $\pm 45^\circ$ ang

seen in Fig

active sensi

and soldere

nector so th

wires were

typical noise

A the

acquisition to

thermistor ha

response of 10

3.4.2 Calibra

The ho

brated over a r

shown in Figure

a velocity at the

num inlet (blow

the data is described in Section 3.4.4. Finally, the processing algorithm and the calculation of the statistical quantities of interest is given in Section 3.4.5.

3.4.1 Hot-wire Probes, Anemometers, and Thermistor

The double-X hot-wire probe consisted of four tungsten wires in an arrangement of two x-wire pairs rotated 90° to each other. The wires on a single x-array are oriented at a $\pm 45^\circ$ angle to the probe axis and spaced 1 mm apart. An illustration of this probe can be seen in Figure 3.7. The four separate $5\text{ }\mu\text{m}$ tungsten wires were 3 mm long with a 1 mm active sensing length in the center. The two ends of the wire were copper plated to $30\text{ }\mu\text{m}$ and soldered to stainless steel broaches. These broaches were then wired to a BNC connector so the output signal could be recorded by the data acquisition device. The hot-wires were driven by four DISA type 55M10 constant-temperature anemometers with a typical noise level of 2 mV RMS (Morris, 1997).

A thermistor was placed in the vicinity of the hot-wire during calibration and data acquisition to compensate for ambient temperature fluctuations between data points. The thermistor had an accuracy of $\pm 0.2\text{ K}$, a sensitivity of 2.03 K/Kohm , and a frequency response of 10 Hz at 293 K (Bohl, 1996).

3.4.2 Calibration Facility

The hot-wire calibration facility allowed the double-X hot-wire probe to be calibrated over a range of velocities and angles. An illustration of the calibration facility is shown in Figure 3.8. A centrifugal blower was used to pressurize a small plenum creating a velocity at the outlet of the calibration unit. Multiple screens just downstream of the plenum inlet (blower outlet) were used to dampen large pressure and velocity fluctuations

and inheren

straight an

The hot-w

A brief st

ensuring th

The

axes as see

ing or re-p

#NF90-3 c

The control

stepping con

the desired a

questionable

to 10 volt ran

to the hot-w

angle. This pr

angle on both

at each angle w

age was record

angle at each ca

3.4.3 Calibration

A quasi-st

range of $\pm 36^\circ$ in

and inherent unsteadiness. The smooth converging section preceding the outlet ensured a straight and steady outlet flow. The rectangular outlet was 2.5 cm wide by 21.5 cm long. The hot-wire was aligned with the center of the outlet just downstream from the exit plane. A brief study showed that the boundary layer thickness at the exit plane was minimal ensuring that the hot-wire was in the inviscid core.

The calibration facility had the ability to rotate the hot-wire around two different axes as seen in Figure 3.9. This allowed both x-wire arrays to be calibrated without moving or re-positioning the hot-wire probe. Two stepper motors controlled by a Velmex #NF90-3 controller were used to rotate the probe in the horizontal and vertical planes. The controller was interfaced with a computer running a custom Matlab code from which stepping commands were sent. A specific amount of steps were sent to move the probe to the desired angle. However, the accuracy in determining the angle from this method was questionable because of skipped or miscounted steps. Therefore, a potentiometer with a 0 to 10 volt range was attached to the output shaft. The potentiometers were calibrated prior to the hot-wire calibration such that the output voltage was directly proportional to the angle. This process involved manually moving each traverse to a physical stop at a known angle on both the positive and negative extremes of the angle range. The voltage recorded at each angle was then used to find a linear fit. During calibration, the potentiometer voltage was recorded along with the hot-wire anemometer voltage to accurately resolve the angle at each calibration point.

3.4.3 Calibration

A quasi-steady method was used to calibrate the double-X hot-wire probe over a range of $\pm 36^\circ$ in approximately 6° increments for each x-array. The first x-array was

calibrated

in the hori

calibration

traverse b

This resul

each angle

s over a 40

with a 10

noulli equa

the unstea

Bernoulli e

several stea

confirming t

The v

mistor were a

data were use

where E is the

is the pitch angl

of the 'n' varia

algorithm resulte

of a noisy pressur

mon. Despite su

calibrated through each of the 13 angles using the horizontal traverse by rotating the probe in the horizontal plane about the vertical axis of rotation. Immediately following the first calibration, the second x-array was calibrated through the same angles using the vertical traverse by rotating the probe in the vertical plane about the horizontal axis of rotation. This resulted in a total of 26 calibration files for a full double-X probe calibration. At each angle, the flow speed was increased from a near zero velocity to approximately 30 m/s over a 40 second period. The static pressure in the calibration unit plenum was measured with a 10 Torr Baratron pressure transducer described in Section 3.2.1.1. Using the Bernoulli equation, the velocity was calculated for each sample acquired. It was assumed that the unsteady effects from varying the velocity were negligible justifying the use of the Bernoulli equation. As noted in Morris (2002), the assumption was tested by performing several steady-state calibrations over the same velocity range as a quasi-steady calibration confirming that the differences were not measurable.

The voltage signals from the pressure transducer, hot-wire anemometers, and thermistor were acquired during the 40 second period at a rate of 1 kHz. For each sensor, the data were used to fit the equation

$$E^2 = A(\gamma) + B(\gamma)Q^{n(\gamma)} \quad (3.16)$$

where E is the voltage from the hot-wire anemometer, Q is the measured velocity, and γ is the pitch angle of the probe. The best fit equation was found by iterating through values of the ' n ' variable until the minimum standard deviation was found. This processing algorithm resulted in 13 separate transfer functions for each of the four sensors. Because of a noisy pressure signal, typical standard deviations of approximately 0.3 m/s were common. Despite substantial flow conditioning downstream of the blower (upstream of the

calibration

tions on t

ters and

signal wa

higher ve

the mean

with the p

same tren

15th order

cal standar

deviations

tions.

Beca

before and a

the calibration

repeated. A c

to process the

from the pre-c

this thesis and

the results of th

3.4.4 Data Acq

calibration unit outlet), the high standard deviation may have been due to pressure fluctuations on the outlet side of the blower. Further investigations concluded that the anemometers and pressure transducers were not responsible for this noise. The noise from the signal was clearly fluctuating through a mean value and was much more prevalent at the higher velocities. To remedy this problem, a 15th order polynomial curve was fit through the mean of the pressure signal as a smoothing function. A representative pressure signal with the polynomial fit is shown in Figure 3.10. Since the hot-wire signals displayed the same trend as the pressure signal, each of the four hot-wire signals were also fit with a 15th order polynomial smoothing function. The combination of these actions led to typical standard deviations of approximately 0.02 m/s which is representative of the standard deviations achieved with the more time consuming discrete point, steady state calibrations.

Because of the nature of hot-wire “drift” over time, a calibration was performed before and after each experiment termed the “pre-calibration” and “post-calibration”. If the calibration did not agree well, the data were discarded and the experiment was repeated. A common method of checking for disagreement between the calibrations was to process the post-calibration files with the calibration information (A , B , and n variables) from the pre-calibration. Calibration drift did become an issue with the data presented in this thesis and the uncertainties predicted as a result of drift will be presented along with the results of the each experiment.

3.4.4 Data Acquisition Procedure

Ho

mm (r/R

trailing ed.

Pre

digital cam

each of the

the hot-wir

investigate

representative

hot-wire pro

device moun

angle to be a

the probe pos

Hot-w

kHz resulting

probe. This sa

speed of 950 m

approximately

resolved. There

acquisition. Th

explained in the f

angles found from

ing program. It w

Hot-wire measurements were acquired for 17 radial locations, in increments of 10 mm ($r/R = 0.028$), at an axial distance of 35 mm ($z/R = 0.098$) downstream from the trailing edge of the shroud for the selected shroud and operating conditions.

Preliminary tuft observations were made to estimate the mean flow direction. A digital camera configured with a long exposure time captured the fluctuating tuft image at each of the 17 locations across the fan exit plane for the shroud conditions tested during the hot-wire data acquisition. The camera was situated at two different orientations to investigate the fluctuations in the axial-radial plane and the radial-tangential plane. A representative tuft image is shown in Figure 3.11. This information was then used to align the hot-wire probe with the mean flow direction. The probe was aligned with a positioning device mounted on the hot-wire traverse described in Section 2.2.3 allowing the pitch angle to be adjusted by rotating the probe around two axes. A schematic representation of the probe positioning device is shown in Figure 3.12.

Hot-wire measurements were acquired for 60 seconds at a sampling rate of 11.4 kHz resulting in a total of 684,000 samples per radial location with the double-X hot-wire probe. This sampling rate corresponded to one sample every 0.5° of fan rotation for a fan speed of 950 rpm. The x-array hot-wire configuration can only measure angles up to approximately $\pm 36^\circ$ since any angles greater than the calibration limits cannot be reliably resolved. Therefore, it was important to check the alignment of the probe during data acquisition. The processing algorithm resolves speed and angle from each x-array as explained in the following section. To check the alignment of the probe, histograms of the angles found from each x-array were output from a modified version of the main processing program. It was desired to center the histogram around a zero angle so that the probe

was aligned
data to generate
provided a
improperly
tively. If the
of the problem
tered histogram

3.4.5 Double

The
and angle from
double-x histogram
one speed/angle

The v
From the time
(Q_1, Q_2) are for
 n) correspond
for each sensor
 $i=1,2,\dots,13$ (M
intersection of
points to the in
quite intensive
samples for each

was aligned with the mean flow direction. The modified version processed one second of data to generate these histograms. Although the data were not converged, the histograms provided a good representation of the actual angles. Representative histograms of an improperly and a properly aligned probe are shown in Figure 3.13 and Figure 3.14, respectively. If the histogram was apparently skewed or truncated, the corresponding pitch angle of the probe was corrected and the data point was repeated as necessary to produce a centered histogram.

3.4.5 Double-X Hot-wire Processing

The goal of the main processing algorithm was to solve for the instantaneous speed and angle from each x-array resulting in two separate speed/angle pairs for the complete double-x hot-wire probe. The following description details the method followed to find one speed/angle pair. The process was simply repeated for the other x-array.

The voltages from the two sensors of the x-array will be denoted as E_1 and E_2 . From the time series of recorded voltage pairs, (E_1, E_2) , the two predicted velocities (Q_1, Q_2) are found using equation (3.16) for all 13 sets of calibration coefficients (A, B , and n) corresponding to each of the 13 calibration angles. This results in 13 separate velocities for each sensor. That is, equation (3.16) is solved for $Q_1(E_1, \gamma_i)$ and $Q_2(E_2, \gamma_i)$ where $i=1,2,\dots,13$ (Morris, 2002). The measured speed and angle, (Q, γ) , from the x-array is the intersection of the two curves, and is found by interpolating between the three closest points to the intersection using a polynomial fit (see Figure 3.15). The processing can be quite intensive since this routine is performed for each sample out of the 684,000 total samples for each x-arrays of the double-X hot-wire probe.

The
ponents of $\mathbf{v}$
probe axis $\mathbf{v}$
denoted as
tion, the ve
array will b
redundant m
nate system.
tioned in a s
necessary to
 (r, θ, z) whe
with the rota
transformation

where the angle
script 'x' can be

Post-proc
obtain the desire
and *rms* of the vel
The time mean of

The speed and angle pair found from each x-array were decomposed into two components of velocity. For purpose of discussion, the velocity component in line with the probe axis will be denoted as u_1 and the velocity vector normal to the probe axis will be denoted as v where the subscript '1' denotes the first x-array. Continuing with this notation, the velocity components in the axial and normal probe directions in the second x-array will be denoted as u_2 and w . Since the velocity components u_1 and u_2 are a redundant measurement, this results in three components of velocity in the probe coordinate system. However, these velocity components are relative to the probe which is positioned in a spherical coordinate system using the device in Figure 3.12. Therefore, it was necessary to transform the velocity components to the cylindrical fan coordinate system (r, θ, z) where r is positive outward from the center of rotation, θ is positive tangential with the rotation of the fan, and z is positive downward from the fan exit plane. The transformation equations are given as follows:

$$U_r = u_x \sin \zeta \cos \xi - v \cos \zeta \cos \xi - w \sin \xi \quad (3.17)$$

$$U_\theta = u_x \sin \zeta \sin \xi - v \cos \zeta \sin \xi + w \sin \zeta \cos \xi \quad (3.18)$$

$$U_z = u_x \cos \zeta + v \sin \zeta \quad (3.19)$$

where the angles ζ and ξ are the angles from the probe positioning device, and the subscript 'x' can be from either x-array or an average of the two.

Post-processing of the initial speed and angle calculations were performed to obtain the desired velocity data and statistical quantities. These include: the time mean and rms of the velocity components, the phase averaged mean and rms, and axial vorticity. The time mean of the velocity components was calculated according to

and the m

The veloc

Phase ave

velocity fi

not be seen

nents were

and the phas

(

where T is th

angle of $0^\circ <$

The ph

The pha
calculated by

$$\bar{U}(r, \theta, z) = \frac{1}{N} \sum_{i=1}^N U_i \quad (3.20)$$

and the rms of the velocity components was given by

$$\tilde{U}(r, \theta, z) = \left(\frac{1}{N-1} \sum_{i=1}^N (U_i - \bar{U})^2 \right)^{\frac{1}{2}}. \quad (3.21)$$

The velocity data were phase averaged using the phase trigger described in Section 3.3.3. Phase averaged data provides a view of the blade-to-blade dynamics and the resulting velocity field. It can be used to discern distinguishing features of the flow field that cannot be seen from the time mean information. The phase averaged mean velocity components were computed by

$$\langle \bar{U} \rangle(r, \theta, z, \alpha) = \frac{1}{N} \sum_{i=1}^N U_i(iT + t_\alpha) \quad (3.22)$$

and the phase averaged rms calculations were computed by

$$\langle \tilde{U} \rangle(r, \theta, z, \alpha) = \left(\frac{1}{N-1} \sum_{i=1}^N [U_i(iT + t_\alpha) - \langle \bar{U} \rangle(r, \theta, z)]^2 \right)^{\frac{1}{2}} \quad (3.23)$$

where T is the time for one full revolution of the fan and t_α is the phase lag for a given angle of $0^\circ < \alpha < 360^\circ$.

The phase averaged velocity magnitude was also of interest and was calculated by

$$\langle |V| \rangle = \sqrt{\langle \bar{U}_r \rangle^2 + \langle \bar{U}_\theta \rangle^2 + \langle \bar{U}_z \rangle^2}. \quad (3.24)$$

The phase averaged axial vorticity was also found from the velocity data. It was calculated by

A second

equation 3

and a forw.

where Δh is

$$\langle \omega_z \rangle(r, \theta, z) = \frac{1}{r} \left(\frac{\partial(r \langle \bar{U}_\theta \rangle)}{\partial r} - \frac{\partial \langle \bar{U}_r \rangle}{\partial \theta} \right). \quad (3.25)$$

A second order finite difference scheme was used to find the partial derivative terms in equation 3.25. A central difference was used to calculate the interior points:

$$\left(\frac{\partial a}{\partial h} \right)_i = \frac{a_{i+1} - a_{i-1}}{2\Delta h} \quad (3.26)$$

and a forward difference was used to calculate the boundary points:

$$\left(\frac{\partial a}{\partial h} \right)_i = \frac{-3a_i + 4a_{i+1} - a_{i+2}}{2\Delta h} \quad (3.27)$$

where Δh is the distance between data points.

Torque (in-lb)

Figure

ΔP



AFRI
Recen

Figure 3
that

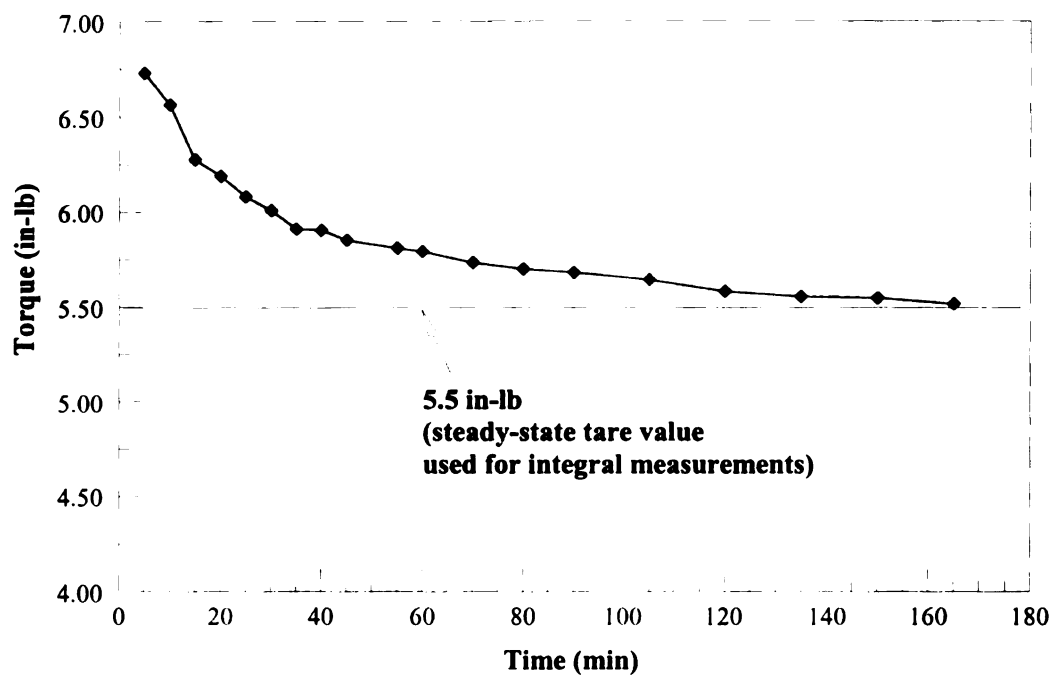


Figure 3.1 Torque measurements acquired over a period of time with the fan removed from the shaft to measure a tare value

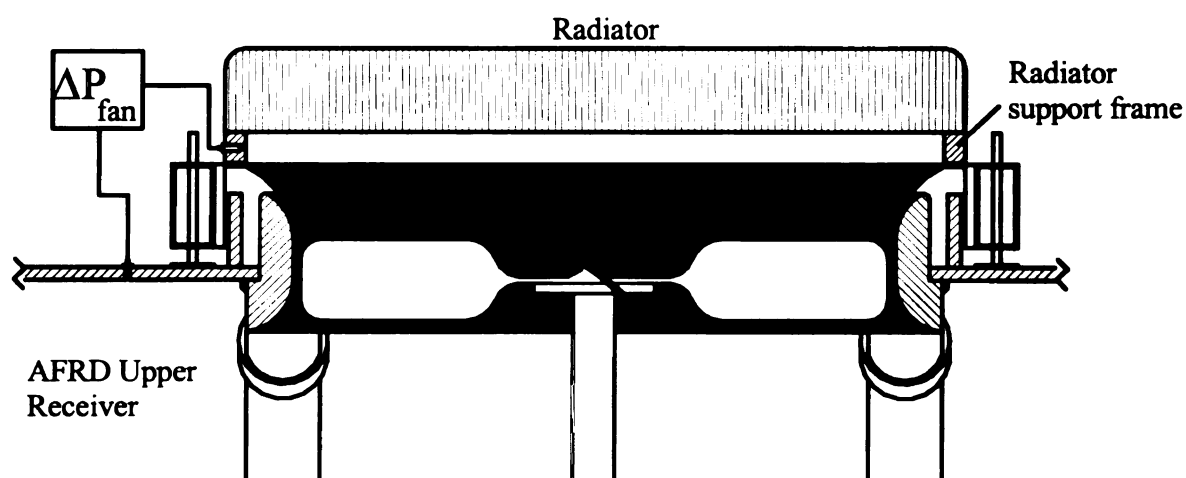


Figure 3.2 Schematic representation of the pressure rise across the fan blades. Note that the quantity, ΔP_{fan} , was actually calculated from two separate pressure measurements: $(P_{receiver} - P_{atm}) - (P_{radiator} - P_{atm})$.

intern
win

Figure 3.3 A

Figure 3.4 A

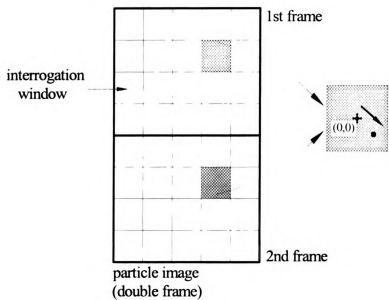


Figure 3.3 A schematic representation of the cross-correlation technique using in PIV vector processing

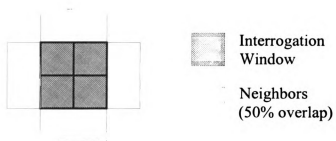


Figure 3.4 A schematic representation of the interrogation window overlap used in PIV vector processing

-0.

-0.

-0.

^{*}
Z

0.2

0.4

Figure 3

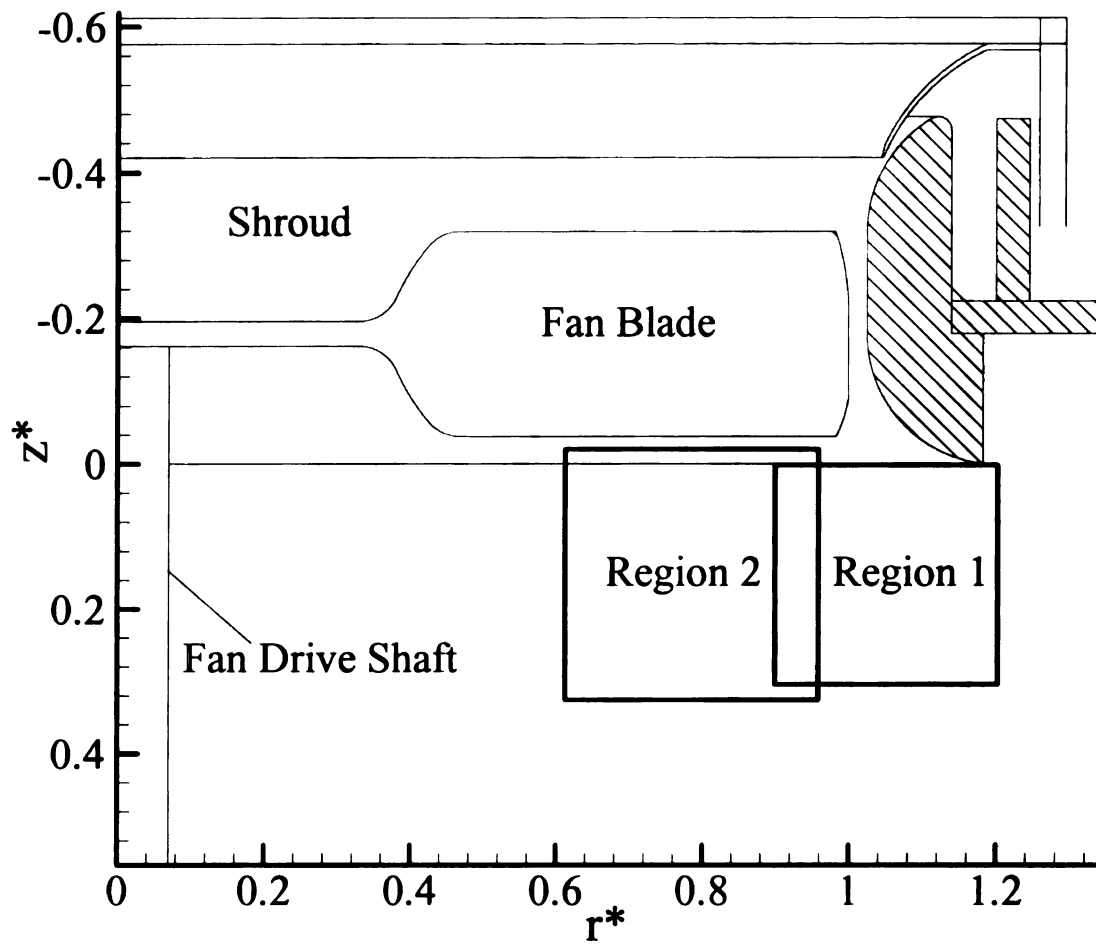


Figure 3.5 Schematic of PIV measurement regions relative to the experimental configuration

Figur

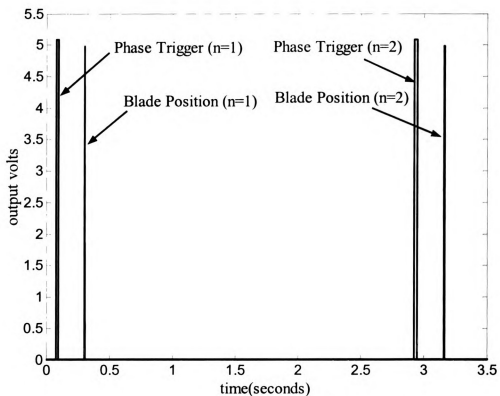


Figure 3.6 TTL voltage signals from the optical encoders for PIV phase triggering taken at a shaft speed of 20 rpm

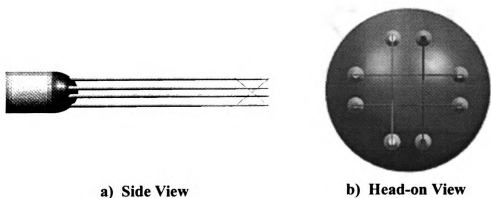


Figure 3.7 Illustration of the double-x hot-wire probe.

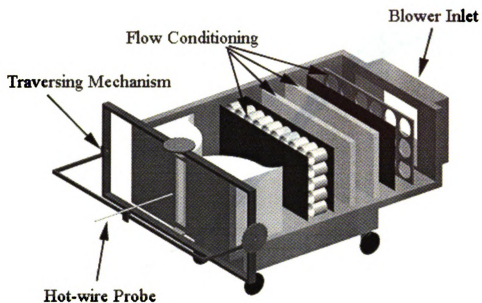


Figure 3.8 Hot-wire calibration facility (Top and side panel removed to show detail).

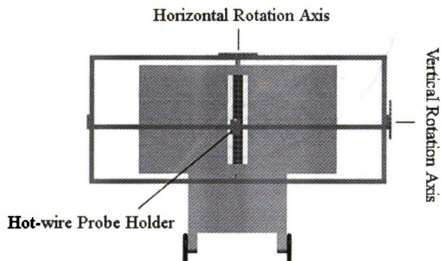


Figure 3.9 Front-view of the calibration facility. Two axes of rotation allow the hot-wire probe to rotated in the vertical and horizontal plane.

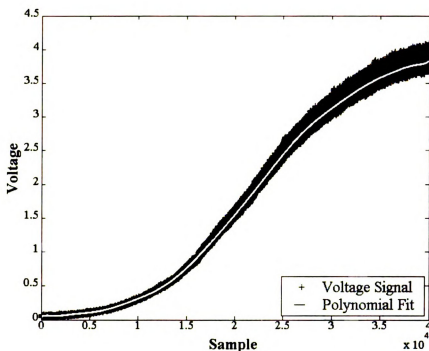


Figure 3.10 A representative pressure signal from a hot-wire calibration. The high standard deviation was reduced by fitting the curve with a 15th order polynomial function.

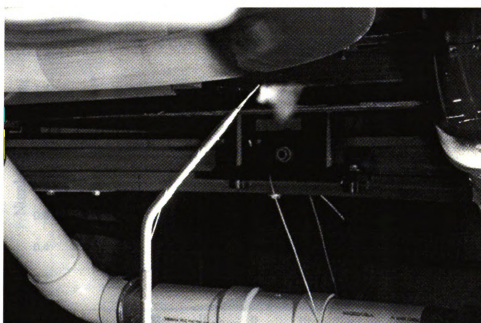


Figure 3.11 A representative image showing a fluctuating tuft. This tuft was used to find the mean flow direction at a given radial location in the wake of the fan.

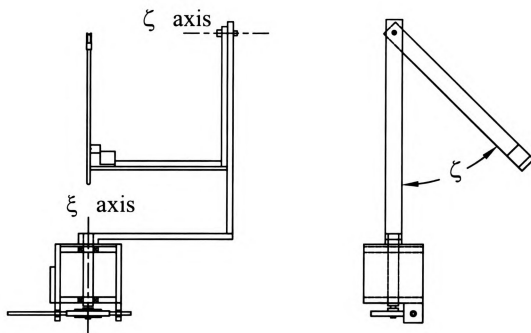


Figure 3.12 Schematic of hot-wire probe positioning device.

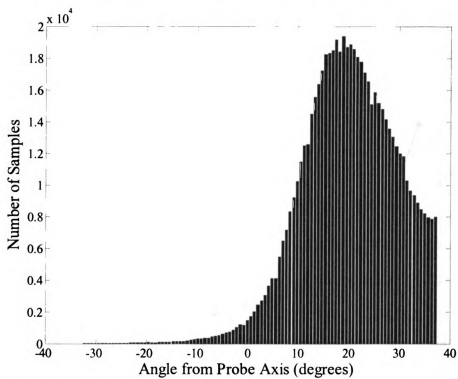


Figure 3.13 Histogram of flow angles measured by an improperly aligned hot-wire probe.

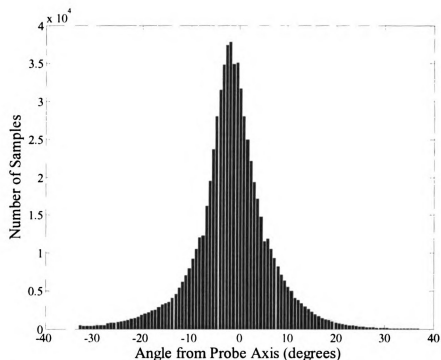


Figure 3.14 Histogram of flow angles measured by a properly aligned hot-wire probe.

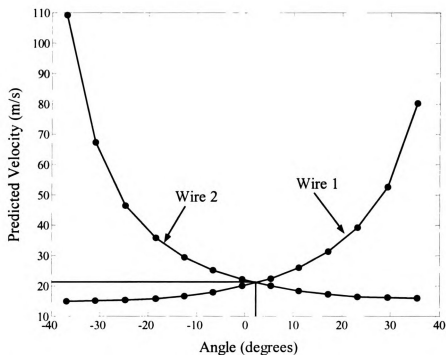


Figure 3.15 Intersection of the predicted velocities from the two wires in a single x-array for use in the processing algorithm to find the measured velocity and angle.

4.0 Results and Discussion - Integral Measurements

The results of the integral measurements are presented and discussed in this section. Integral quantities are important in assessing the overall performance of the cooling fan with and without the aerodynamic shroud. First, the baseline measurements will be presented along with a discussion of the characteristic features of the performance curve in Section 4.1. This is followed by a discussion of the change in performance due to the aerodynamic shroud in Section 4.2.

4.1 Baseline Measurements (Unpowered Shroud Condition)

Baseline performance and efficiency curves without a powered aerodynamic shroud ($\chi = 0$) were acquired to establish a benchmark from which to compare the $\chi \neq 0$ results. For all shroud conditions the integral performance data are presented as $\psi = \psi(\phi, \chi)$, which represents, in non-dimensional form, the pressure rise across the fan as a function of volume flow rate and shroud power. The corresponding efficiency data are presented as $\eta_{sys} = \eta_{sys}(\phi, \chi)$ where η was defined in equation 3.8. The functional dependence of these two integral quantities was discussed in Section 3.2.2.

Each performance test followed the procedure outlined in Section 3.2.3. Three separate baseline data sets were taken to demonstrate the repeatability of the measurements. Plots of the three separate characteristic curves are presented in Figure 4.1 and the efficiency curves are shown in Figure 4.2. Although the majority of the powered shroud data presented in this thesis used a fan speed of 950 rpm, a baseline test at 700 rpm is included in the plots to demonstrate that these data scale correctly with fan speed when presented in non-dimensional form. The data agree well between separate tests confirming the repeatability of the measurements. It is noted that these data were obtained with

the radiator upstream of the fan inlet, and that the pressure coefficient represents the non-dimensional pressure rise across the fan blades only, as defined by equation 3.3. These data do not reflect the pressure rise across the complete fan and radiator system. Consequently, the plots do not extend to $\phi = 0$. A plot of the pressure rise across the complete fan and radiator system as a function of flow rate is presented in Figure 4.3 for reference. Data were acquired at a total pressure rise less than zero to mimic a less restrictive upstream element.

Referring to the baseline performance data in Figure 4.1, a particularly notable feature is the prominent stall region of this fan. Fan stall is a condition in which air flow separates from the suction surface of the fan blade. A “stalled” condition typically develops at a high pressure rise (relative to the maximum pressure rise at a zero flow rate condition) caused by an increased system resistance. On the characteristic curve, fan stall is identified as a plateau or drop in the pressure rise across the fan blades despite a lower flow rate. It is usually followed by a second pressure increase until the point of no delivery. (No delivery is also referred to as the “full shut-off” condition.) The above description of fan stall is illustrated by the annotated baseline performance curve in Figure 4.4. As shown in the figure, the onset of stall for this fan is approximately $\phi = 0.256$. The stall point is followed by a relatively large “stalling dip” until $\phi = 0.132$, when the pressure coefficient begins to increase again. Although the fan has regained the ability to produce a static pressure rise, it operates in an extremely inefficient manner (see Figure 4.2).

4.2 Effect of Aerodynamic Shroud (Powered Shroud Conditions)

The shroud off ($\chi = 0$) condition is plotted along with selected powered shroud conditions ($\chi > 0$) in Figure 4.5. A clear effect of increasing shroud power (increasing χ)

on the performance of the fan is the movement of the stall point to a lower flow rate. Generally, as the shroud power is increased, this effect is increased, although a unique function cannot be derived. At the highest shroud power ($\chi = 22.0 \times 10^{-3}$), the stall point is moved to $\phi = 0.186$, a 27% decrease in the stall point flow coefficient as compared to the $\chi = 0$ condition. Even at the lowest shroud power investigated ($\chi = 1.13 \times 10^{-3}$), the stall point flow coefficient is decreased by 11%. For flow rates higher than the stall point, there is no measurable change between the baseline condition and the powered shroud conditions. A clear degradation of performance is seen at the low flow rates where the $\chi > 0$ characteristic curves cross the $\chi = 0$ curve. In this region, the pressure coefficients are much lower for the powered shroud conditions compared to the baseline curve. The combined effect of the aforementioned changes in the baseline performance data create the appearance that the aerodynamic shroud systematically “shifts” the defining characteristics of the baseline performance curve to lower flow rates as the χ value is increased.

By delaying the stall point to a lower flow coefficient, the effective operating range of the fan is increased. This point is further clarified by considering the fan’s performance curve along with a system load curve. A system load curve is a parabola representing the pressure drop across the restrictive elements upstream or downstream of the fan as a function of volume flow rate through those elements. When plotted on the same axes, the intersection of the two curves provides the operating point of the fan. Two system load curves are presented along with a baseline condition and the $\chi = 8.64 \times 10^{-3}$ powered shroud condition in Figure 4.6. Given the load curve denoted as “System load curve 1” in the figure, one can see that the operating point is still within the desirable operating range of the fan. If the system resistance was increased, for example by the addition

of an air conditioning condenser to a standard radiator, thereby increasing the system resistance represented by “System load curve 2”, the operating point would now lie in the stalled region of the baseline, $\chi = 0$, condition; however, the fan would not yet be stalled in the $\chi = 8.64 \times 10^{-3}$ shroud condition. Hence, the advantage using an aerodynamic shroud is clear.

The aerodynamic shroud may prove to be beneficial by extending the useful operating range of this test fan, however, this gain is at the expense of the fan’s efficiency as seen in Figure 4.7. The measured efficiency accounts for the added power from the shroud and is defined by equation 3.8. The plot shows that the efficiency is monotonically decreased as the χ value is increased. At the most energized shroud condition, $\chi = 22.0 \times 10^{-3}$, the efficiency is lowered from 49.4% to 41.1% at its peak. As expected, this is the worst case. However, even at the lowest shroud power ($\chi = 1.13 \times 10^{-3}$), the efficiency is decreased to 47.0% at its peak (cf. 49.4%).

It is difficult to compare the results of the present study to the previous study by Morris (1997). In that experimental investigation, the test fan, boundary conditions, and test conditions differed considerably from the present study. Two main differences are the geometry of the shroud outlet (a sharp edged rectangular shape in the Morris study) and the magnitude of the tip clearance. Specifically, the test fan used for the Morris study had a tip clearance of 20.7% of blade span, compared to the 4.0% tip clearance for the current test fan. Despite the differences, the present study confirmed that shroud power variable χ “can be used to fully characterize the contribution of the shroud” (Morris and Foss, 2001).

In the data presented in Figures 4.5 and 4.7, it is necessary to note that the data points shown only include the measurements acquired “in one direction.” That is, the plot only includes the data acquired while the operating point was moved from a high flow rate to a low flow rate (increasing pressure rise). It is displayed this way for clarity and to show the distinct separation of the different characteristic curves. However, the data were also acquired while moving the operating point from the low flow rate condition to the high flow rate condition (decreasing pressure rise). In doing so, hysteresis effects were discovered in the $\chi > 0$ data. The $\chi = 1.13 \times 10^{-3}$ characteristic curve is the most prominent case of this and is shown in Figure 4.8 as an example. The significant hysteresis can be seen in the stall area where the stall point occurs at a lower flow rate when loading the fan from a high flow rate to a low flow rate. The two stall points are labeled as ϕ_{incr} and ϕ_{decr} on Figure 4.8 to denote the stall point when increasing and decreasing the flow rate. This hysteresis was found to be repeatable and, although it appears in most of the $\chi > 0$ data, it is less apparent as the shroud power is increased.

A similar occurrence of this hysteresis can be found in airfoil (stall recovery) and compressor research, and has been referred to as a “hysteresis loop.” Experimental data show that the angle of attack of an airfoil that has stalled must be decreased considerably beyond the original stall angle for flow reattachment to occur (Mittal and Saxena, 2000). A hysteresis loop also occurs in compressor stall and effort has been directed at studying the cause (see D’Andrea et al, 1996).

Delaying or preventing stall is also a topic of interest in compressor research. If a low speed axial fan (e.g., an automotive type cooling fan) stalls, the consequences may be lower efficiencies, a lower pressure rise, and noisy operation. The same consequences are

relevant in compressors. However, compressors may also exhibit excessive vibration, and extreme degradation of performance due to the higher pressures and rotational speeds during operation. Considering that compressors are often used in applications in which stable performance is critical, stall is unacceptable. Thus, a considerable amount of effort is directed at this subject. In specific instances, active flow control has been used as an attempt in controlling the stall. One technique was successfully used by Day et al. (1993) in which the general concept was similar to that of the aerodynamic shroud. Specifically, it involved injecting air into the tip region of the compressor blades using 12 injection valves spaced evenly around the circumference upstream of the inlet. Using this method, the author was able to delay stall and extend the useful operating range by 6.0%. Many researchers have also documented the role of tip clearance in delaying stall. Smith (1970) showed that increasing the tip clearance between the blade tips and the casing of a low-speed compressor caused the onset of stall to move to a higher flow rate.

Despite the similarities in the results, it is noted that compressor studies are only partly relevant to the present study involving an axial flow fan due to the difference in design and application. For instance, a compressor usually has a high solidity (chord divided by blade spacing) leading to greater interaction between two consecutive blades. The hub-to-tip ratio is much greater in compressors than axial flow fans as well. In terms of operation, compressors typically produce a much higher pressure rise.

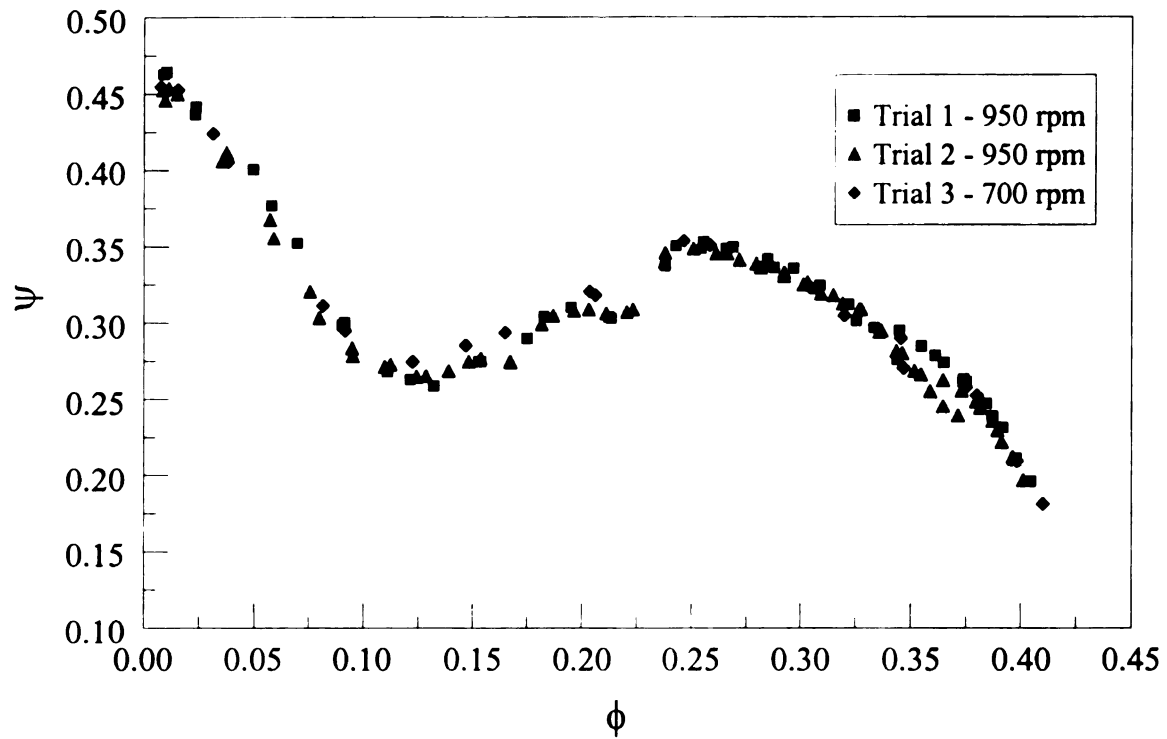


Figure 4.1 Baseline performance data for three independent tests

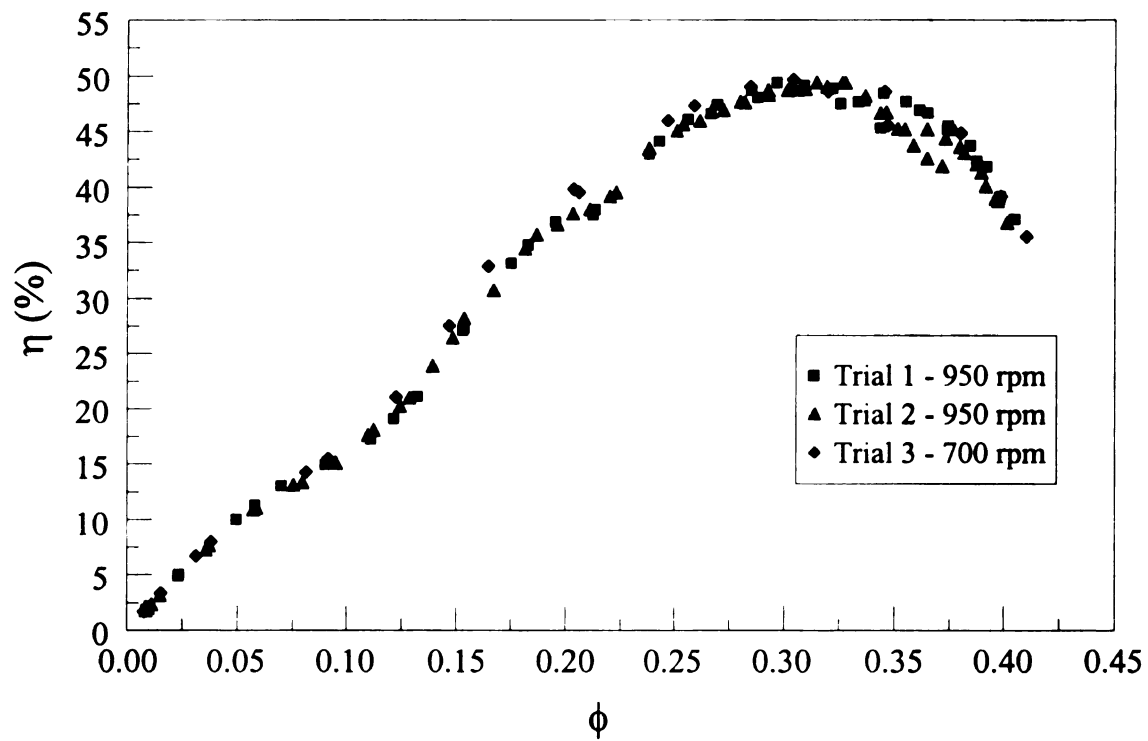


Figure 4.2 Baseline efficiency data for three independent tests

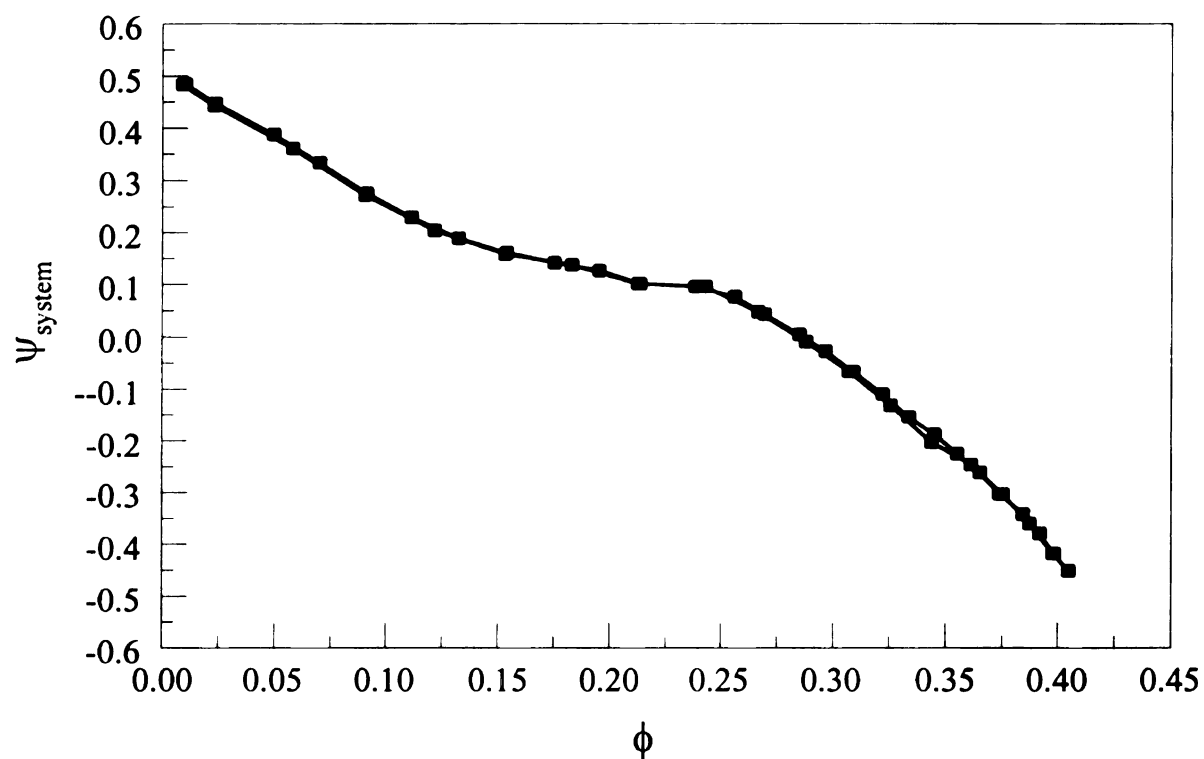


Figure 4.3 Total system pressure rise plotted as a function of volume flow rate in dimensionless form

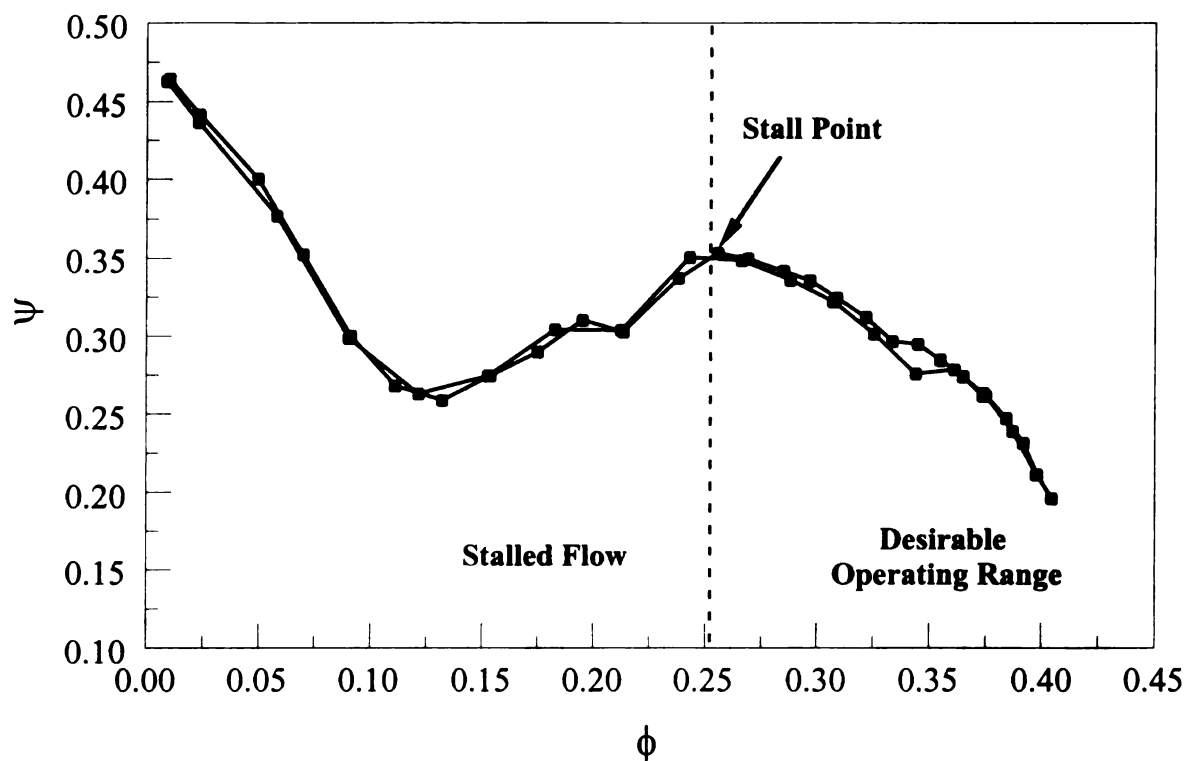


Figure 4.4 Baseline characteristic curve showing the stall point and separate regions of the characteristic curve

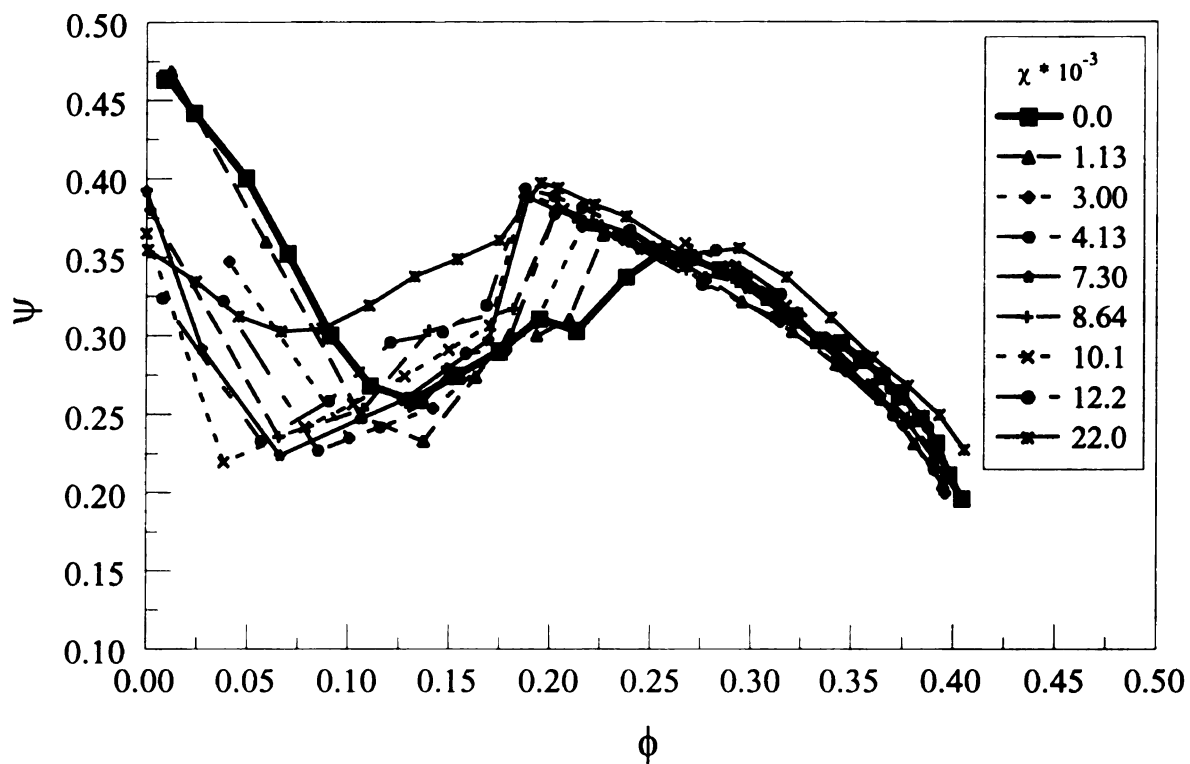


Figure 4.5 Performance data for selected shroud conditions

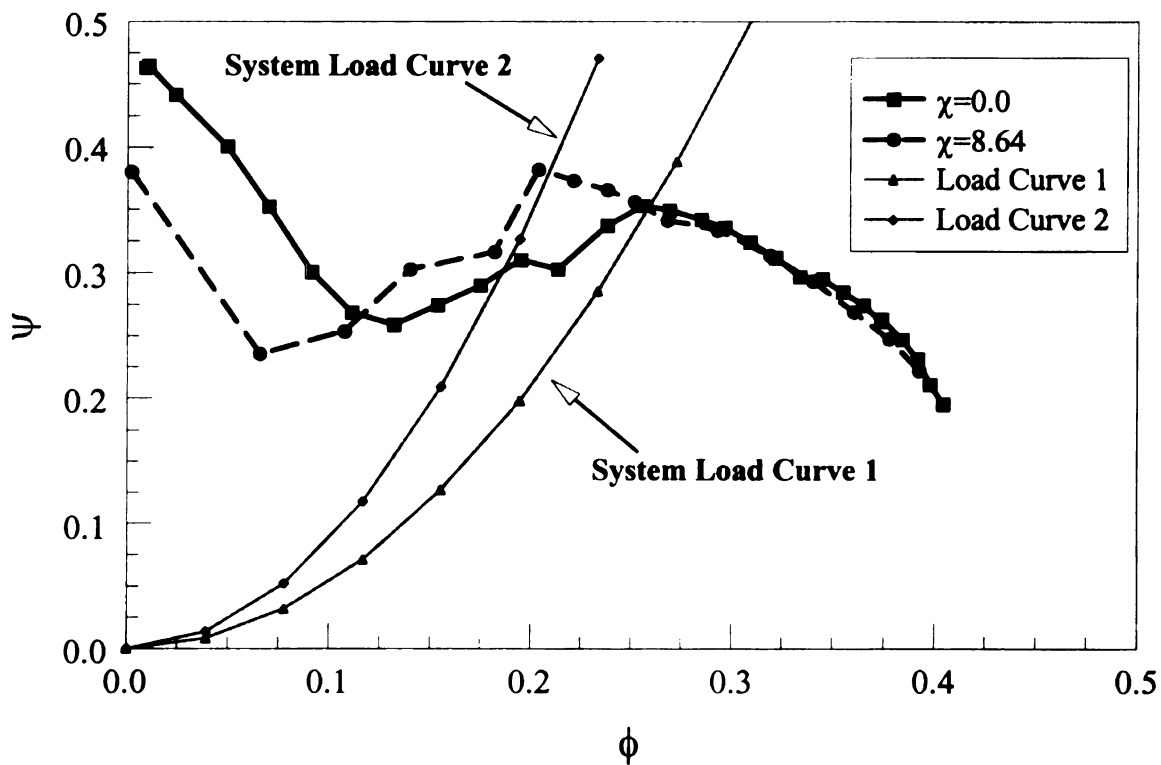


Figure 4.6 Two performance curves, a baseline condition and a representative powered shroud condition, plotted with a low and high system resistance curve to demonstrate the advantage of delaying stall.

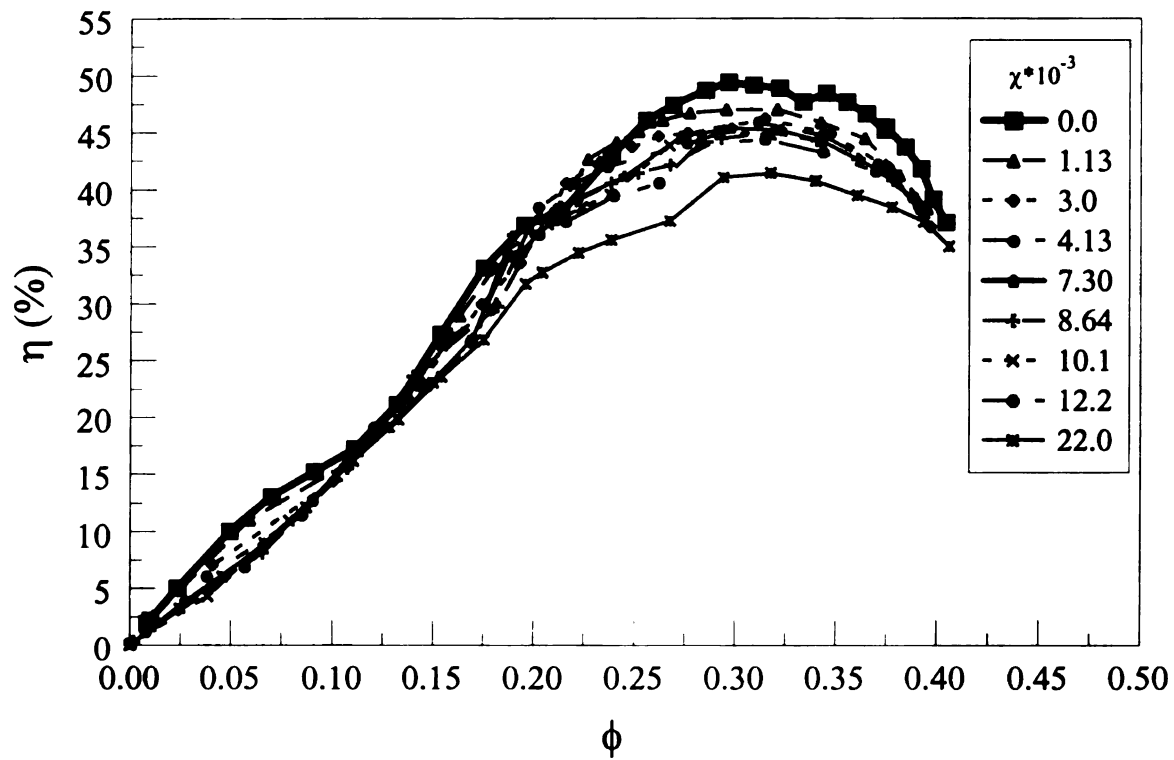


Figure 4.7 Efficiency data for selected shroud conditions

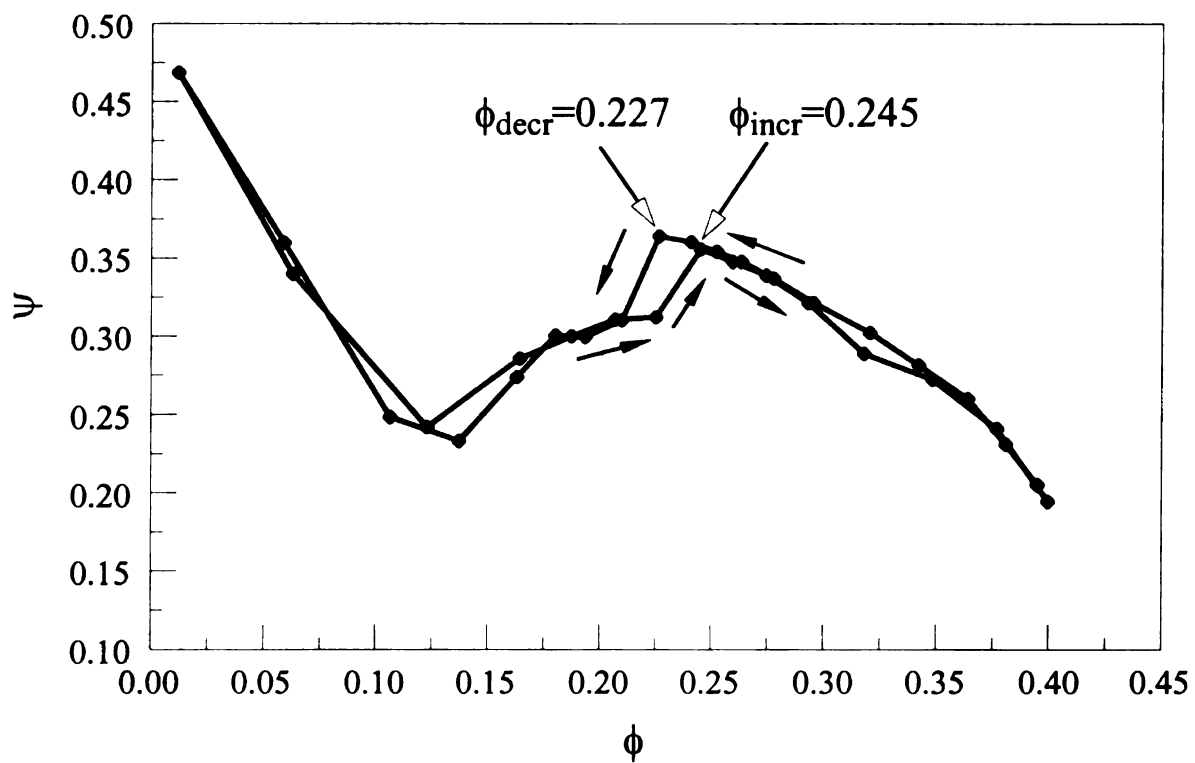


Figure 4.8 Performance data at $\chi = 1.13 \times 10^{-3}$ showing the stall point hysteresis

5.0 Detailed Velocity Measurements

Hot-wire and Particle Image Velocimetry (PIV) measurements were acquired to analyze the detailed mechanisms responsible for the integral performance results for three selected operating conditions. The procedure in which the data were acquired for each measurement method was discussed in Sections 3.3.3 and 3.4.4. The point-wise, phase averaged measurement of the three components of velocity at a fixed axial location downstream of the fan is provided by hot-wire anemometry and is presented in Section 5.1. The PIV data provide a spatial view of the velocity field using both a time averaged and phase averaged technique. Additional flow field interpretations are available from these data and the results are presented in Section 5.2.

Three different shroud conditions were selected for the detailed velocity measurements including $\chi = 0$, 8.64×10^{-3} , and 22.0×10^{-3} representing a “shroud off,” “low power,” and “high power” shroud condition. The focus conditions of the research were the operating points denoted as ‘A’ through ‘D’ as labeled on Figure 5.1 with the corresponding performance curves. Table 5.1 contains the significant details of each operating point. The stall point for the corresponding shroud condition is listed in the table for reference. The specific points were chosen to compare both a stalled (off-design) condition (B) and two powered shroud conditions (C, D) to a baseline design operating condition (A). The hot-wire study, in particular, investigated the $\chi = 0$ and the $\chi = 8.64 \times 10^{-3}$ shroud conditions including a stall condition. (Specifically, these are test conditions A, B, and C.) The PIV study investigated all three χ values but did not include a stall condition. (Specifically, test conditions A, C, and D were studied with PIV.)

Table 5.1 Test Conditions for Detailed Velocity Measurements

Test Condition	Measurement Method	χ	ϕ	ψ	Stall Point (ϕ)
A	HWA, PIV	0.00	0.27	0.35	0.256
B	HWA	0.00	0.22	0.31	0.256
C	HWA, PIV	0.0086	0.22	0.38	0.204
D	PIV	0.0220	0.21	0.39	0.195

5.1 Hot-wire Measurements

Hot-wire data were acquired at a single distance of 35 mm ($z/R_{tip} = 0.098$) downstream from the trailing edge of the shroud. The hot-wire was traversed across the exit of the fan recording data at 17 radial locations from $r^* = 1.181$ to $r^* = 0.731$ ($r^* = r/R_{tip}$) in 10 mm increments ($\Delta r/R_{tip} = 0.028$). The fan was operated at 950 rpm. A continuous time series was acquired for 60 seconds at rate of 11.4 kHz corresponding to one sample every 0.5° of fan rotation. The experimental mesh shown in Figure 5.2 was produced from the data where each grid point represents a data point. A schematic representation of the fan is included to illustrate the location and extent of the hot-wire survey. The hub of the fan is at approximately $r^* = 0.368$, however, the locations of the measurement only extend inward to $r^* = 0.731$. This is due to a large region of reverse flow. Hot-wire anemometry cannot discern the direction of flow, thus, the extent of the data was limited to the measurable range. Note that the schematic representation of the fan is drawn at an arbitrary angular position to show the general position of the fan and hub relative to the experimental mesh. The exact position of the fan blade rel-

ative to the measurement location was not known. Also note that the experimental mesh is 35 mm downstream from the shroud.

The presentation of the hot-wire investigation is divided into four sub-sections. First, a description of the uncertainties involved in the measurement technique is presented in Section 5.1.1. This is followed by the presentation of the phase averaged velocity results for the three velocity components in Section 5.1.2. The phase averaged fluctuation intensity (RMS) values for each velocity component are also presented to analyze the unsteadiness of the flow field. These data are presented in Section 5.1.3. Finally, contour plots of the axial vorticity for each shroud condition are presented in Section 5.1.4.

5.1.1 Hot-wire Measurement Uncertainties

The major cause of uncertainty in the hot-wire velocity data was due to the disagreement between the pre-and post-calibrations. Several data sets were discarded during the experimental program because of this issue. As described in Section 3.4.3, the agreement between the calibrations was checked by processing one calibration data set with the other calibration coefficients (e.g., process the pre-calibration data files with the coefficients found from the post-calibration). Once this initial check verified that the calibrations were sufficiently close, the complete data set was processed separately with both calibrations. From this information, a sample-by-sample comparison was performed to determine the true error from the calibration “drift”. After removing erroneous velocity data, a percentage difference between each pair of data points from the two separate data sets was calculated.

Figures 5.3 through 5.5 contain histograms of the percentage difference for the three velocity components and velocity magnitude for each shroud condition. Although the histograms presented are from one radial location, the results are representative of the complete data set. It was desired to have error less than $\pm 2\%$ for all velocity data. However, extreme difficulty was experienced in achieving this window of error for all velocity components since the signal from all four wires of the double-X hot-wire probe must remain consistent over the course of the experiment.

The error histograms for condition 'A' are presented in Figure 5.3. The radial component agrees very well as the histogram is centered about the zero axis. The tangential and axial components both have a mean error of 3%. However, the radial component is also much larger in magnitude than the other two components. This is reflected in the error histogram for the velocity magnitude which is centered at approximately 1%.

Conditions 'B' and 'C' both exhibit the same trends in the velocity component errors as shown by Figures 5.4 and 5.5. The histograms of the radial components are centered very close to zero and are within $\pm 2\%$. The highest error was observed in the stalled condition 'B' in which the histogram extends over $\pm 10\%$ for the axial component. However, because of the high radial velocity and low axial velocity, the mean agreement in the velocity magnitude is still only 2.2%. Determining which of the two calibrations would be more appropriate to use in processing the experimental data is difficult if not impossible. Therefore, the two data sets were averaged on sample-by-sample basis. It is these data that are presented in the subsequent section.

The phase averaged velocities were also tested for convergence of the mean and RMS quantities. Several data points were tested and the same trend among them was

observed. Representative plots of the data converging about the mean are presented in Figure 5.6 for a single data point from condition ‘A’. The plot shown includes the convergence of the mean and RMS of the three velocity components.

5.1.2 Phase Averaged Velocity Data

Phase averaged velocity data are presented as contour plots for the first 90° of fan rotation. The flow is assumed axisymmetric and more detail can be observed in this truncated view compared to the complete data set. The mean, RMS, and magnitude of velocity are scaled by the fan blade tip speed U_{tip} and are defined as $\langle \bar{U}^* \rangle$, $\langle \tilde{U}^* \rangle$, and $\langle |V^*| \rangle$, respectively.

Figures 5.7 through 5.9 show the mean radial velocity contour plots for conditions ‘A’, ‘B’, and ‘C’. It is immediately evident that conditions ‘A’ and ‘C’ are qualitatively similar. A distinct blade passing is demonstrated by an area of high radial velocity followed by a much lower radial velocity. These high velocity regions are spaced 40° apart corresponding to the spacing of the nine fan blades. The radial location of the maximum radial velocity is centered around $r^* = 1.08$ in both conditions, however, the azimuthal location in condition ‘C’ is 5° behind condition ‘A’. Condition ‘C’ also exhibits a higher radial velocity, although the velocity variation is much greater. It is assumed that the annular jet stays attached to the surface of the shroud which curves outward in the radial direction. This shroud flow may entrain the main flow towards the outer radial regions accounting for the larger radial component. In condition ‘B,’ the effects of fan stall are immediately apparent. A blade passing is not readily distinguishable since the radial flow field is no longer tightly coupled to the fan blades. The radial velocity is also much lower and nearly uniform in the outer region. From these data, it is quite apparent that the aero-

dynamic shroud changes the nature of stall and structure of the radial velocity field in the wake of the fan.

The mean tangential phase averaged velocities are presented in Figures 5.10 through 5.12. Once again, the periodic blade passing is evident in conditions 'A' and 'C', but not in the stalled condition 'B'. The "cells" of high velocity related to the blade passing are centered near the tip region ($r^* = 1$) in condition 'A'. This point is moved closer to $r^* = 1.1$ in condition 'C', another indicator that the shroud flow entrains the main flow radially outward. Unlike the radial component plots, the tangential velocity is much different between conditions 'A' and 'C'. From these data, it is clear that the aerodynamic shroud increases the tangential component of the flow field. It is also interesting to note that the areas of the highest and lowest radial and tangential velocities in condition 'C' exist at the same nominal locations; see Figures 5.9 and 5.12.

Contour plots of the mean axial component of velocity are presented in Figures 5.13 through 5.15 for each shroud condition. It is noteworthy that the maximum axial velocities occur well beyond the tip radius of the axial fan; note the peak values at $r^* \approx 1.2$ and the indication that the actual peak values may exist for an even larger radius position. The azimuthal location of the peak coincides with the lowest radial velocity. When the fan is stalled (condition 'B') the blade passing is evident in the axial component, in contrast to the radial and tangential components where it is not visible. Condition 'C' exhibits larger axial velocity variations than condition 'A' between two passing blades. As the fan blade passes a given azimuthal position, it appears that the magnitude of the axial velocity is maintained and "swept" radially outward from the hub. It is interesting that the axial component is smaller than the radial component in this axial flow type fan. Pre-

sumably, this is due to the operating point at which the measurements were acquired. It is well known that as the pressure rise increases and the flow rate decreases, the flow becomes more radial. Venturi shrouds with a curved outlet are often used to increase the usable operating range of the fan by accommodating this radial flow.

Contour plots of the velocity magnitude as calculated by equation 3.24 are presented in Figures 5.16 through 5.18. These plots are included to compare the contribution of each velocity component to the overall magnitude. It is apparent that the flow is predominately in the radial direction for all shroud conditions. It is also apparent that velocities in condition 'C' are consistently larger in magnitude, although the flow rate in condition 'C' is lower as reported in the integral measurements. It is important to note that the shroud flow is incorporated in the hot-wire measured velocities, whereas, the shroud flow in the integral measurements is subtracted from the total flow rate (see equation 3.2).

5.1.3 Phase Averaged Fluctuation Intensity (RMS) Data

The RMS levels of the velocity components for each shroud condition are presented in Figures 5.19 through 5.27 as contour plots. These data are an indicator of the unsteadiness of the flow. It is well known that high velocity fluctuations contribute to aerodynamic noise generation, and lower flow rates generally have higher fluctuation levels (Fukano and Jang, 2003). The RMS levels for the stalled condition (condition 'B') are slightly higher than condition 'C' and even higher than in condition 'A' according to the radial component plots. This is expected because fan stall involves a high level of unsteadiness and increased noise generation caused by the separated flow over the suction surface of the fan blade.

The tangential and axial velocity fluctuations show much more uniform areas of RMS levels. The highest levels of unsteadiness are observed near the blade tips in all the components of velocity. The axial component, in particular, has the highest RMS levels for each shroud condition. A blade passing is clearly shown in this component as lower levels of RMS are observed in the inferred region between two consecutive blades followed by the region of higher fluctuations. The higher RMS values are most likely from the wake of the pressure side of the fan blade. In addition, the tangential component velocities are generally the least unsteady of the three components as indicated by the RMS plots. Quantitatively, this is expressed by averaging the discrete RMS values over the domains for each component and each shroud condition. The values are presented in Table 5.2.

Table 5.2 Spatially averaged RMS levels for each component and each test condition

Test Condition	$\langle \tilde{U}_r \rangle$	$\langle \tilde{U}_\theta \rangle$	$\langle \tilde{U}_z \rangle$
A	0.0950	0.0688	0.1006
B	0.1264	0.0896	0.1298
C	0.1018	0.0832	0.1154

5.1.4 Phase Averaged Vorticity Data

The vortical flow field in the wake of the fan has a significant effect on the aerodynamic performance and noise characteristics. Common vortex structures observed in axial fan flows include the tip leakage vortex, the leading edge separation vortex, and the trailing edge vortex. The phase averaged axial vorticity was calculated according to equation 3.25 in order to analyze one component of vorticity present in the wake of the cooling fan

for each shroud condition. The results were normalized by 2Ω following the method of Morris (1997) and are presented as contour plots in Figures 5.28 through 5.30.

Regions of negative axial vorticity, which correspond to counter-clockwise rotation, are evident in the periodic blade passing signal. These regions, which are understood to be the signatures of the tip leakage vortex, exist before the region of high axial velocity and after the regions of high radial and tangential velocity in condition 'A'. Specifically, the highest negative axial vorticity for the 90 degree plot shown is centered at $\theta = 60^\circ$ ($r^* = 0.48$ on the vertical axis) compared to $\theta = 55^\circ$ for the radial, $\theta = 56^\circ$ for the tangential, and $\theta = 65^\circ$ for the axial component. These correlations are given in Table 5.3 for further reference. It is apparent that the negative vorticity is not directly associated with any velocity component, but is nearest to the highest velocity magnitude corresponding to the localized low pressure at the core of the vortex.

Table 5.3 Tabulated values of the locations of the peak values for various measured variables for both shroud conditions

Test Condition	Centroid $-\langle \omega_z \rangle$	Centroid $\langle \omega_z \rangle$	Peak $\langle \bar{U}_r \rangle$	Peak $\langle \bar{U}_\theta \rangle$	Peak $\langle \bar{U}_z \rangle$	Peak $\langle V \rangle$
A	60.0	52.5	55.0	56.0	65.0	55.0
C	61.0	44.0	49.0	43.0	79.0	43.0

The aerodynamic shroud clearly changes the fan blade dynamics in terms of rotational fluid in the wake of the fan. The negative vorticity appears to be weakened by the aerodynamic shroud; however, the positive vorticity following this area is much larger in magnitude. Consider the vorticity in the boundary layer (dominantly $\bar{\omega}_\theta$) on the surface of the shroud. The, assumed to be impulsive, axial velocities associated with the blade passing will give these ω_θ vorticity filaments a strong reorientation as $\{\omega_\theta(\partial \bar{U}_z / r \partial \theta)\}$

which is the equivalent of a “source term” for $\omega_z > 0$. It is suggested that this mechanism is responsible for the $\bar{\omega}_z > 0$ concentrations following the $\bar{\omega}_z < 0$ of the tip vortex. This argument is strengthened by comparing the results for conditions ‘A’ and ‘C’. (The $\bar{\omega}_\theta$ will be stronger for ‘C’ given the aerodynamic shroud Coanda jet). By the argument presented, one would expect the tip vortex to be weakened by the “gap filling” Coanda jet in agreement with the observations.

5.2 Particle Image Velocimetry Measurements

Time averaged and phase averaged PIV measurements were acquired in the $r - z$ plane at a fixed azimuthal position for the two shroud conditions used in the hot-wire measurements, $\chi = 0$ and 8.64×10^{-3} , and an additional “high power” shroud condition, $\chi = 22.0 \times 10^{-3}$. All measurements were acquired with a fan speed of 950 rpm. Operating points slightly before the stall points were chosen corresponding to the points ‘A’, ‘C’, and ‘D’ as illustrated in Figure 5.1. Therefore, two of the three test conditions (A and C) can be related to the hot-wire results. Since a single PIV measurement region covered a limited amount of radial distance, two regions were used to capture a larger area of the velocity field as shown in Figure 3.5. The original experimental program included acquiring PIV data near the fan hub to observe the hub recirculation region, however, an equipment failure with the PIV laser did not allow the acquisition of these measurements. Nonetheless, the PIV data presented is useful in analyzing the velocity field and provides a validation of the hot-wire measurements.

The PIV data include the velocity magnitude and the axial and radial velocity components, where the magnitude is defined as

$$|U_{rz}| = \sqrt{\bar{U}_r^2 + \bar{U}_z^2}. \quad (5.1)$$

It is important to note that the velocity magnitudes between the PIV and hot-wire data cannot be directly compared due to the absence of the tangential component in the two-dimensional PIV measurements. Consistent with the hot-wire survey results, the PIV data were scaled by R_{tip} and U_{tip} , where $U_{tip} = 35.37$ m/s. The relevant non-dimensional variables are defined as: $r^* = r/R_{tip}$, $z^* = z/R_{tip}$, $\bar{U}^*(r, z) = \bar{U}(r, z)/U_{tip}$, and $|U_{rz}^*| = |U_{rz}|/U_{tip}$.

The presentation of the PIV results are sub-divided into four sections. The uncertainties involved in the PIV measurements are described in Section 5.2.1. This is followed by the time averaged PIV data presented in Section 5.2.2. A flow coefficient was calculated by the integration of the PIV velocity data and compared with the integral measurements. These calculations are explained in detail and the results are presented in Section 5.2.3. Finally, phase averaged PIV data were acquired to investigate the blade to blade velocity field variation and these data are presented in Section 5.2.4.

5.2.1 PIV Measurement Uncertainties

Errors in the PIV measurements were primarily a result of the low validation of vectors in specific regions. The causal factors were glare, low quality seeding, or poor laser light. The latter was most certainly the cause of a great number of invalidated vectors. Specifically, when using a dual head laser, both heads must be working properly to obtain good correlations between the two laser pulses. In the present study, one laser was observed to be more intense and covered a wider planar area as compared to the other laser. The low validation was observed in the corners of each PIV region where the laser

light intensity was lowest, while nearly 100% of the vectors in the middle region were validated. A contour plot of the percent of validated vectors for condition 'A' is shown Figure 5.31 and displays this trend. This plot is representative of the other shroud conditions that were tested.

Additional errors were introduced by using two separate PIV regions. Although considerable care was taken to align and calibrate the image when re-positioning the camera, the slightest mis-alignment would lead to a mismatched overlap region. Also, the two regions were acquired for completely separate conditions. Since the test condition involved establishing three separate variables (flow rate, pressure rise, and shroud power), it became extremely difficult to obtain the exact flow condition as the corresponding measurement for the other region. It is difficult to quantify these errors; they are noted to explain discrepancies in the data.

5.2.2 Time Averaged PIV Data

The velocity magnitude $|U_{rz}^*|$ for condition 'A' is presented in Figure 5.32. The largest magnitude appears to be just downstream of the tip clearance region from $r^* = 1$ to $r^* = 1.06$. This high velocity region continues in a "path" travelling radially outward and does not appear to follow the contour of the shroud outlet. These data suggest that the shroud design may be improved if a larger outlet radius were utilized. The predominately radial component observed in the hot-wire measurements is evident in the PIV data as shown by the streamtraces superimposed over the velocity contours. This strong positive radial component has been observed in other axial fan research near the suction side of blades (Inoune and Kuroumaru, 1984). The authors note that this may be caused by centrifugal effects acting on the blade's boundary layer. (The Coriolis "force" will also pro-

vide positive radial components of the velocity vector). A reverse flow is observed downstream of the fan closer to the hub, however, the velocity magnitude in this area is close to zero. It appears that the strong radial flow near the outer regions entrains this stagnant fluid radially outward. Glare from the laser reflecting off of the fan blade did not allow measurement of the seeded flow directly downstream of the fan blade. However, the measurement region was within 8 mm downstream of the fan blade and it shows positive axial velocity for $r^* \geq 0.64$. These data suggest that only the outer 36% of the fan is contributing to the net volume flow rate. Tuft surveys conducted by Morris (1997) at a high pressure condition in the previous aerodynamic shroud study showed reverse flow from portions of the fan blade close to the hub but not across the complete span. Thus, at test condition 'A', the flow may be separated at the inner portions of the fan blade causing reverse flow, while the rest of the fan blade is providing axial flow.

The radial component of velocity for condition 'A' is presented in Figure 5.33. The highest time averaged radial velocities appear to be at the radial extent of the shroud downstream from the tip clearance region. These data agree well with the hot-wire measurements in the location and magnitude of the maximum radial velocity. To analyze the agreement between the two measurement techniques, a time averaged radial and axial velocity was found for each radial location of the hot-wire survey. Also, the PIV data were extracted at the downstream location of the hot-wire measurements creating a direct comparison. These two sets of velocities were then plotted together to create a graphical representation of the agreement. The radial velocities for condition 'A' are presented in Figure 5.34. It is observed that the two measurement techniques agree well at the higher r^* values of each PIV region. From this plot, it is probable that the radial velocity profile

closely follows the hot-wire measured velocities and that the left half of Region I is underpredicting the radial flow field. The inner radial region shows that the hot-wire overpredicts the velocity as compared to the PIV. Nonetheless, the disagreement between the two PIV regions (as discussed in Section 5.2.1) is quite apparent.

The axial velocity component for condition 'A' is presented in Figure 5.35 and shows that the highest axial component occurs just downstream of the blade tip. The hot-wire data suggest that this region is in the inviscid core between the fan blades. Reverse flow is evident from the lowest level contours where the highest negative axial velocity is approximately $\bar{U}_z^* = -0.06$ or about 2 m/s. The agreement between the hot-wire survey and the PIV measurements for the axial velocity field is presented in Figure 5.36. In contrast to the radial velocity field, the two regions of PIV appear to match well at the overlap region. The measured hot-wire axial velocities appear to have the same profile across most of the area, however, the magnitude is higher by as much as 14%. A correlation coefficient, $K_{\theta z} = \overline{U_z' U_\theta'} / \tilde{U}_z \tilde{U}_\theta$, was calculated for each hot-wire data location and the resulting values are in range of 0.182 to 0.558. Since these velocity fluctuations are correlated, it is suggested that the PIV data are biased because of particle migration from the light sheet for the larger U_z values.

Figure 5.37 presents the velocity magnitude for condition 'C'. Qualitatively, conditions 'A' and 'C' appear very similar signifying that the aerodynamic shroud is clearly preventing stall. A strong radial component still exists and the direction of the flow has not changed. However, the area of reverse flow has moved outward slightly to $r^* = 0.65$. The aerodynamic shroud could be creating a larger area of reverse flow, or it is possible that the lower flow rate has caused a larger portion of the fan blade to stall. Also, the mag-

nitudes are slightly lower in condition 'C' as compared to condition 'A'. This is in contrast to the hot-wire results which show that the phase averaged magnitudes are higher in condition 'C' despite the lower flow rate. The difference can be attributed to the absence of the tangential component in the PIV data since the hot-wire data showed a large tangential component for condition 'C'.

Figure 5.38 shows the axial and radial velocities for condition 'C'. Again, the contour plots are very similar between conditions 'A' and 'C'. The radial velocity is still strongest near the outer region of the shroud but this "peak" region has moved slightly more downstream. Figure 5.39 shows that the hot-wire measurements are predicting a much higher radial velocity (maximum of 16% higher) than the PIV measurements, however, the shape of the radial velocity profile appears to be very similar. The axial velocity contour plot in Figure 5.40 is nearly identical to condition 'C' except that the area of positive axial velocity is concentrated more at the blade tip and the stagnant region closer to the hub has become larger. When compared to the hot-wire measurements (Figure 5.41), the velocity profile is very similar, and, again, the axial velocities are approximately 16% higher as measured by the hot-wire. For this test condition, the calculated values for the correlation coefficient $K_{\theta z}$ are in the range of 0.067 to 0.462 suggesting that the error is due to the particle migration previously discussed.

The in-plane velocity magnitudes for condition 'D' are presented in Figure 5.42. The contour plot follows the same trend as was observed in the previous two test conditions. The velocity magnitude is considerably lower across the measurement area and the area of reverse flow has moved outward even farther to $r^* = 0.69$. Streamtraces show that the flow is more radial than conditions 'A' and 'C'. Also, the contours near the over-

lap between the two PIV measurement regions do not match as well as the other test conditions. Since this is a relatively large shroud power, it is most likely that the error was introduced when setting the experimental condition.

Figures 5.43 and 5.44 present the axial and radial velocities for condition ‘D’. Although the flow rate is very close to condition ‘C’, the velocity components show a much lower magnitude. Following the trend of condition ‘C’, the tangential component may be an even larger percentage of the total velocity, however, it is not measured in this two-dimensional view of the velocity field. The disagreement between the two regions is very evident from the component contour plots as well.

5.2.3 Calculation of Flow Coefficient

The PIV results were used to calculate a flow rate by integrating across the fan exit flow area. The integration is performed along a control surface extending from the outlet of the shroud to the fan shaft (see Figure 5.45) and revolved around the circumference. The integral is defined as

$$Q = \int_0^{R_{shr}} (\vec{V} \cdot \hat{n})(2\pi r)dr \quad (5.2)$$

where R_{shr} is the radius to the radial extent of the shroud and the dot product $\vec{V} \cdot \hat{n}$ represents the velocity vector normal to the control surface, or in this case the axial velocity component. Due to the lack of data between the fan shaft and the inner PIV measurement region, the velocity was assumed to be zero in this area. In actuality, this may not be true and the validity of this assumption can be tested by the results. Axisymmetric flow is also

assumed in this calculation since the measurements take place at a singular azimuthal position.

The flow rates integrated from the PIV results are presented in Table 5.4 along with the measured flow rate found from the integral measurements. Note that the measured flow coefficient is the total flow (measured by the flow meters) since the shroud flow cannot be subtracted from the PIV data. The results show that the integrated flow coefficient is consistently lower than the measured flow coefficients with a difference of approximately 10.0% and 8.3% for conditions 'A' and 'C' respectively. Furthermore, the 10% difference for condition 'A' (no shroud flow) indicates that mass flow can only be balanced to within 10% and the 8% difference for condition 'C' is most likely within the uncertainty of the calculation. Despite this, the small difference indicates that a large percentage of the net flow rate is contained in the PIV measurement regions. Considering condition 'D', the 25.3% difference is actually closer to 15% given the 10% difference in the baseline condition. The large disagreement in this condition further proves that the flow is not axisymmetric. This is a plausible explanation because the high pressures in the shroud plenum will be more likely to create a non-uniform annular jet as a result of the deformation of the upper Coanda jet surface.

Table 5.4 Values of the dimensionless flow rate found by integration of the PIV results compared to the measured flow rate from the integral measurements

Test Condition	ψ	ϕ_{measured}	$\phi_{\text{integrated}}$	% Difference
A	0.35	0.26	0.23	10.03
C	0.38	0.22	0.21	8.30
D	0.39	0.21	0.16	25.27

5.2.4 Phase Averaged PIV Measurements

Phase averaged data were acquired in the wake of the fan for conditions ‘A’ and ‘C’. Eight phase planes, separated by 5° , were used to cover the complete area between two fan blades starting with the laser aligned with the blade’s leading edge. Synchronization of the PIV laser pulse with the fan blade was described in Section 3.3.3. For a given phase position, 100 image pairs were acquired. The resulting contour plots are presented in this section for both shroud conditions.

The results for condition ‘A’ are presented in Figures 5.46 through 5.53. Starting with Figure 5.46 (at the leading edge), a small “cell” of high velocity is observed in the wake and is centered at approximately $r^* = 1.1$, $z^* = 0.08$. This cell is marked by “ α ” on the plot. It appears to originate at the blade tip just downstream of the fan blade. The streamline directions show that this flow is strongly radial and that it follows the contour of the shroud. A negative radial velocity is observed in the low velocity, reverse flow region near the area closer to the hub (see β on the plot). At a point farther upstream, the negative radial velocity is turned to the positive radial direction. The implication, for this phase position, is that the flow is drawn inward until it is entrained outward by the high velocity radial flow.

The progression of the high velocity cell is evident at the next phase position (5° , Figure 5.47) where it is now centered at $r^* = 1.15$, $z^* = 0.10$, see γ . It still appears to be originating from the blade tip. The negative radial velocity has decreased in the reverse flow region as the flow is almost entirely in the negative axial direction. At 10° past the leading edge (Figure 5.48), the high velocity cell (see δ) has moved even farther downstream and outward to the edge of the measurement area. The “trail” from which it appears to originate is now moved farther out past the blade tip. This trend continues in

the next three phase positions: 15° , 20° , 25° . At 15° past the leading edge (Figure 5.49), the highest velocities extend directly downstream from the tip clearance region and the flow has a larger axial component. As shown, the flow no longer follows the contour of the shroud.

Figure 5.50 presents the contour plot at 20° past the leading edge. It is instructive to note that the trailing edge of the fan blade occurs at approximately 22° past the leading edge. It is observed that the highest velocity magnitudes occur at or near this phase position. The location (downstream of the tip clearance region) of the “path” of high velocity fluid suggests that there are strong centrifugal forces acting on the suction side of the blade. This high velocity region (ι) downstream of the tip region, is continued in the next phase position (25° , Figure 5.51). At 30° past the leading edge (Figure 5.52) a new cell of high velocity fluid (κ) appears to form close to the blade tip. The direction of the flow has also reverted back to the radial direction and a higher negative axial velocity exists closer to the hub. Figure 5.53 shows that the progression of the velocity field through the last phase position agrees well with the phase position at the leading edge.

The phase averaged PIV data for condition ‘A’ agree well with the general trends of the phase averaged hot-wire data for condition ‘A’. Specifically, the PIV data show the movement of the areas of high velocity magnitude radially outward as the fan blade passes the azimuthal measurement location from leading edge to trailing edge. Referring to Figure 5.16, the hot-wire data show this general outward movement; however, the magnitudes cannot be compared directly because of the absence of the tangential component in the PIV measurements. The PIV data also verifies the inferred regions from the hot-wire data since the exact blade position was not known in those experiments. Namely, the

pressure side of the fan blade, near the leading edge (Figure 5.46), shows a strong radial velocity followed by a change towards lower radial and higher axial velocity after the trailing edge (Figure 5.51).

The phase averaged PIV results for condition 'C' are presented in Figures 5.54 through 5.61. Characteristic similarities and differences exist between the two test conditions. For instance, the "cells" of high velocity that appeared in condition 'A' near the leading edge are still apparent (see λ on Figure 5.54). However, the location is not as far downstream in condition 'C'. This agrees with the 5° lag in the location of peak velocity magnitude observed in the hot-wire data when comparing conditions 'A' and 'C' (Figures 5.16 and 5.18). A similarity observed when comparing the PIV conditions 'A' and 'C' is that the maximum velocity magnitudes appear to be nearly equal. One would expect the velocity to be lower in condition 'C' corresponding to a lower flow rate.

The phase positions of 5° and 10° past the leading edge show that the aerodynamic shroud adds a greater radial component to the velocity field. Although this was already observed in the time averaged data, the mechanism is more apparent in a phase averaged plot. Specifically, Figure 5.55 shows that the flow is forced into an almost entirely radial direction. This effect is even more prominent at 10° in Figure 5.56. Moreover, the flow near the corner of the shroud in the 10° and 15° phase positions is at a very low velocity implying that either the wall jet created by the aerodynamic shroud does not stay attached to the shroud, or that there is a secondary, rotational flow that exists at this location. This results in a decreased area from which the flow exits the fan. This may partially explain why the velocities in condition 'C' are comparable to those in condition 'A' despite the lower flow rate.

The effects from the stronger reverse flow imparted by the aerodynamic shroud becomes apparent at 15° past the leading edge (Figure 5.57) where a small recirculation region is observed at $r^* = 0.70$, $z^* = 0.08$ (see μ on plot). It appears to be created as the reverse flow interacts with the positive axial and radial flow in the same area. This recirculation is much larger at the 20° phase position (Figure 5.58) and is centered farther downstream at $r^* = 0.77$, $z^* = 0.16$ (see v on plot). It does not appear to be connected to the axial vorticity observed in the hot-wire derived vorticity data. Secondary flow features such as this one lead to greater losses due to the enhanced dissipation of kinetic energy.

The next phase positions, 30° and 35° past the leading edge (Figures 5.60 and 5.61), show that the recirculation region is no longer visible. It is interesting that the flow at these positions is more axial than the same phase positions as condition 'A'. Thus, the flow is more radial near the leading edge, and more axial in the area between the fan blades as compared to condition 'A'. This agrees well with the phase averaged hot-wire data in which all velocity components have larger spatial variations in the powered shroud condition as compared to the "shroud off" condition. This may also explain why a larger positive vorticity was observed in the axial vorticity plots. Specifically, the change in flow direction between two consecutive blades could cause this clockwise rotation to develop.

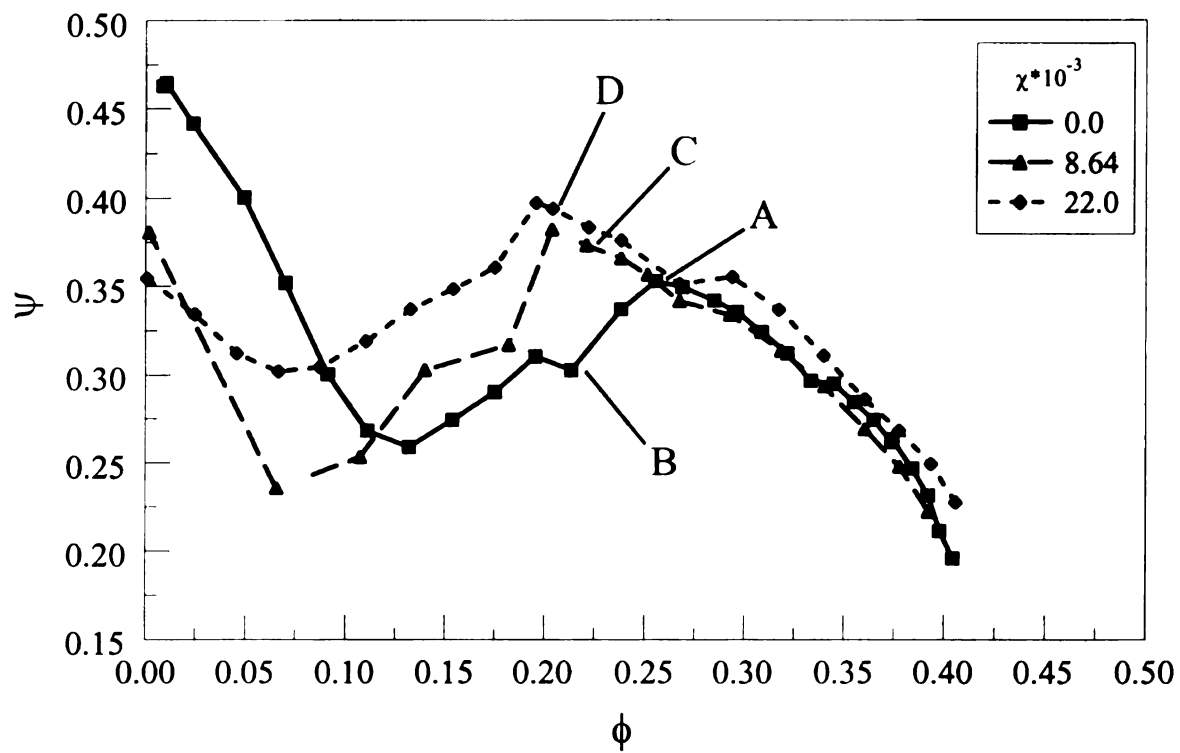


Figure 5.1 Selected shroud conditions labeled with the operating points used for the detailed velocity measurements (see Table 5.1 for numerical values)

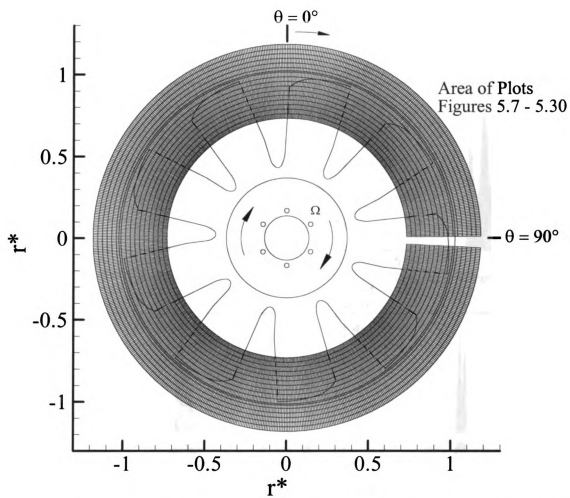


Figure 5.2 Mesh of hot-wire phase averaged data locations in r - θ plane at $z = 35\text{mm}$ ($z/R_{\text{tip}} = 0.098$), Ω shows direction of fan rotation

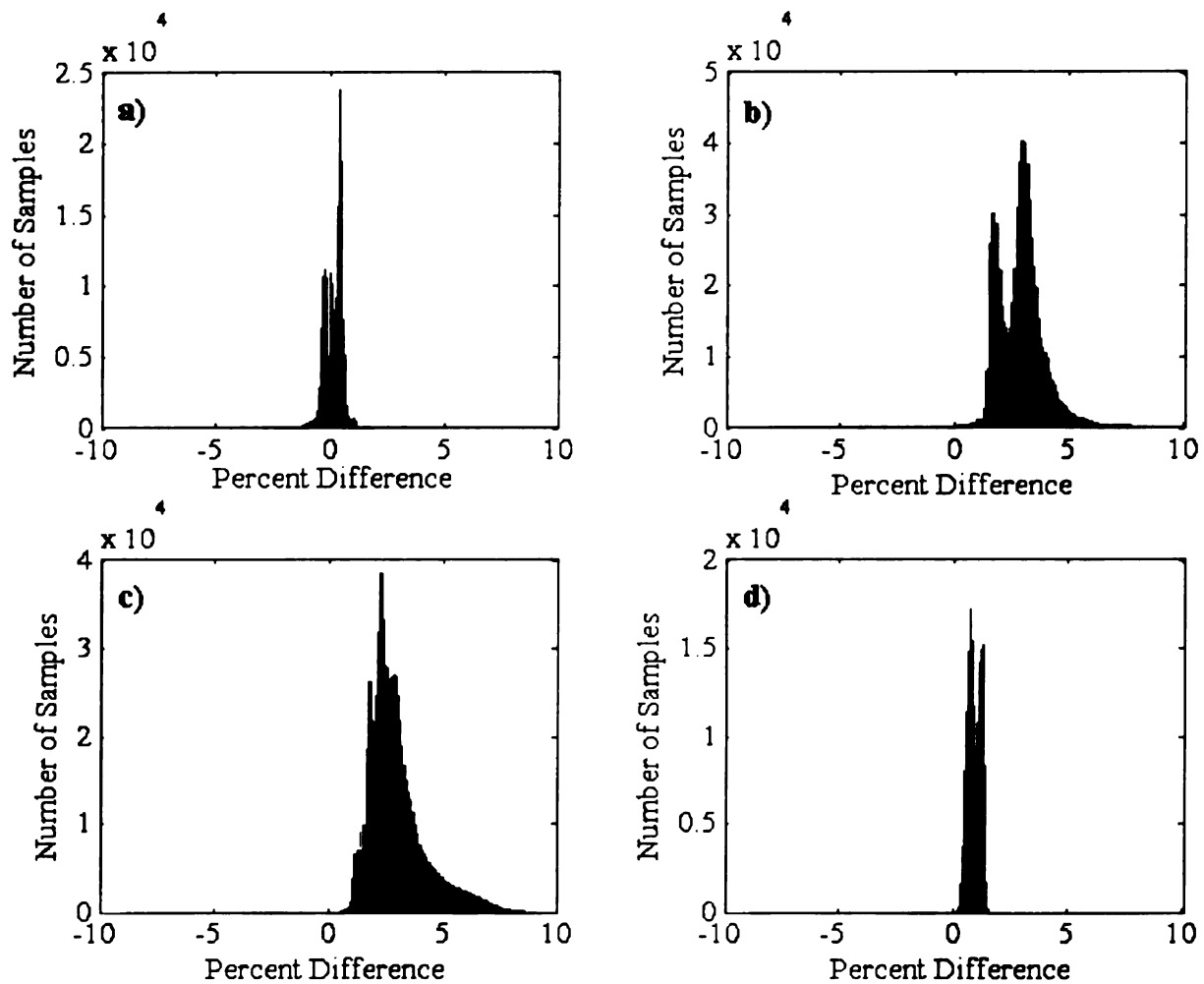


Figure 5.3 Histograms illustrating the “sample by sample” difference between velocities found using the pre and post calibration constants for Condition ‘A’: a) mean radial component, b) mean tangential component, c) mean axial component, and d) velocity magnitude

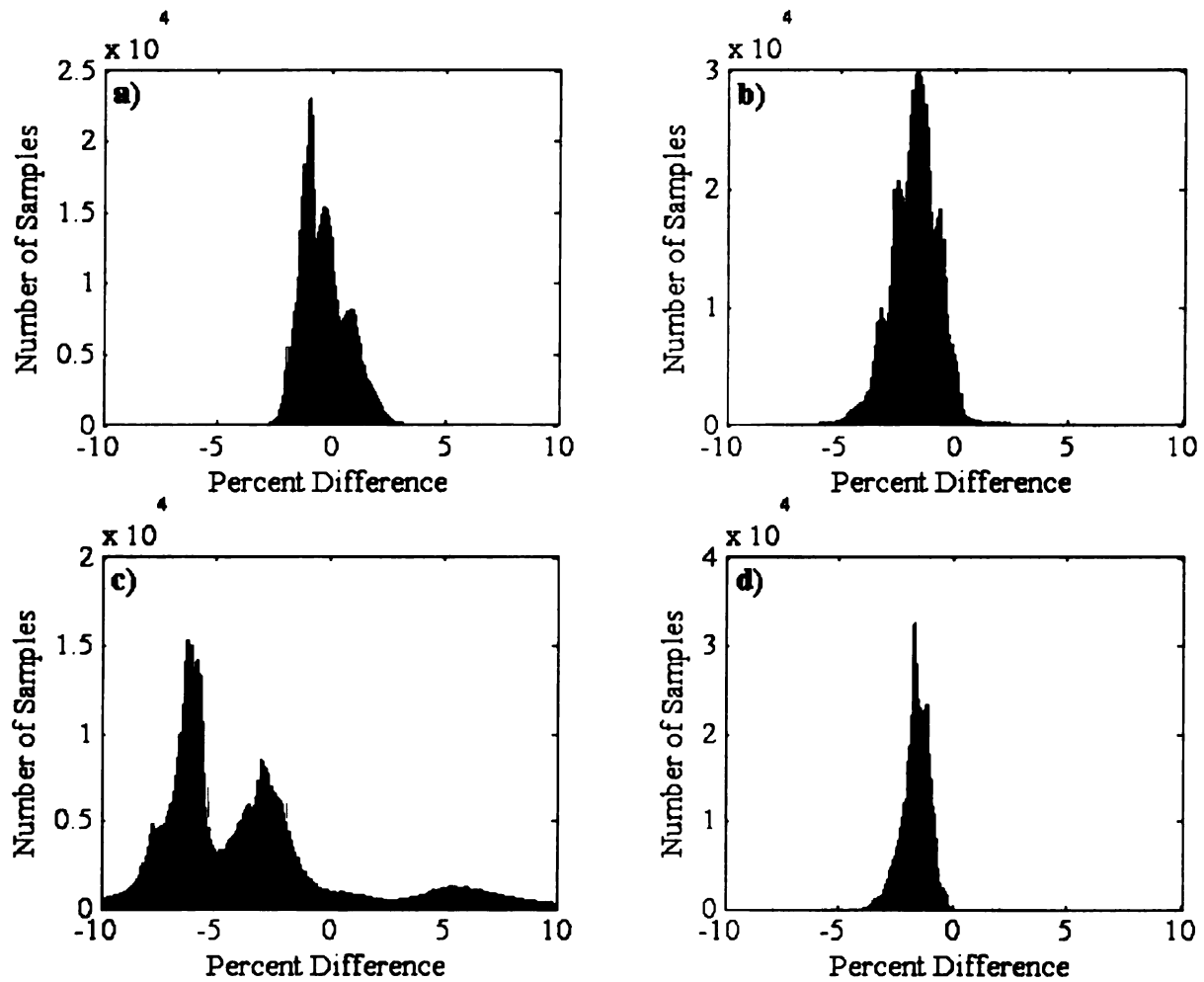


Figure 5.4 Histograms illustrating the “sample by sample” difference between velocities found using the pre and post calibration constants for Condition ‘B’: a) mean radial component, b) mean tangential component, c) mean axial component, and d) velocity magnitude

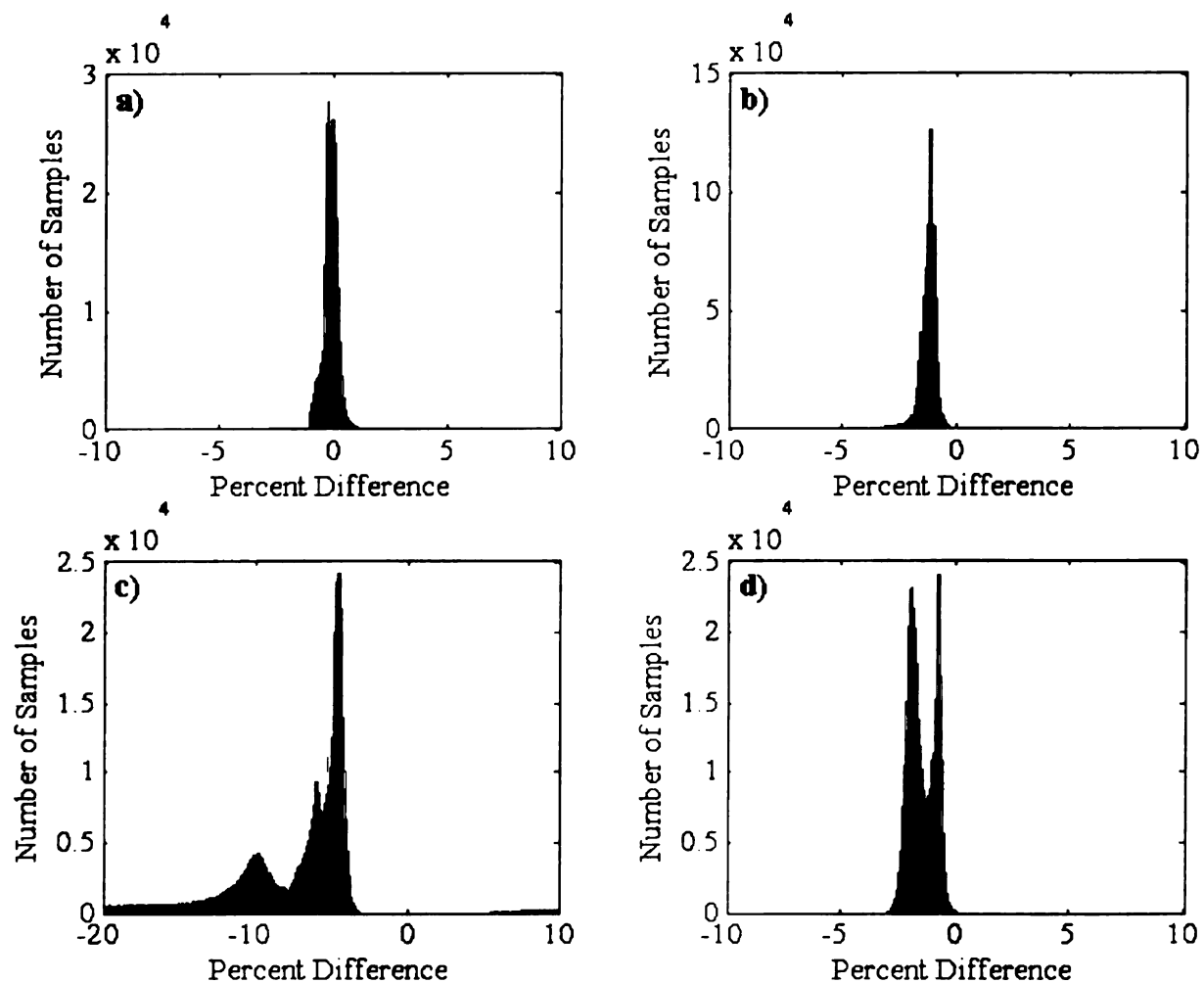


Figure 5.5 Histograms illustrating the “sample by sample” difference between velocities found using the pre and post calibration constants for Condition ‘C’: a) mean radial component, b) mean tangential component, c) mean axial component, and d) velocity magnitude

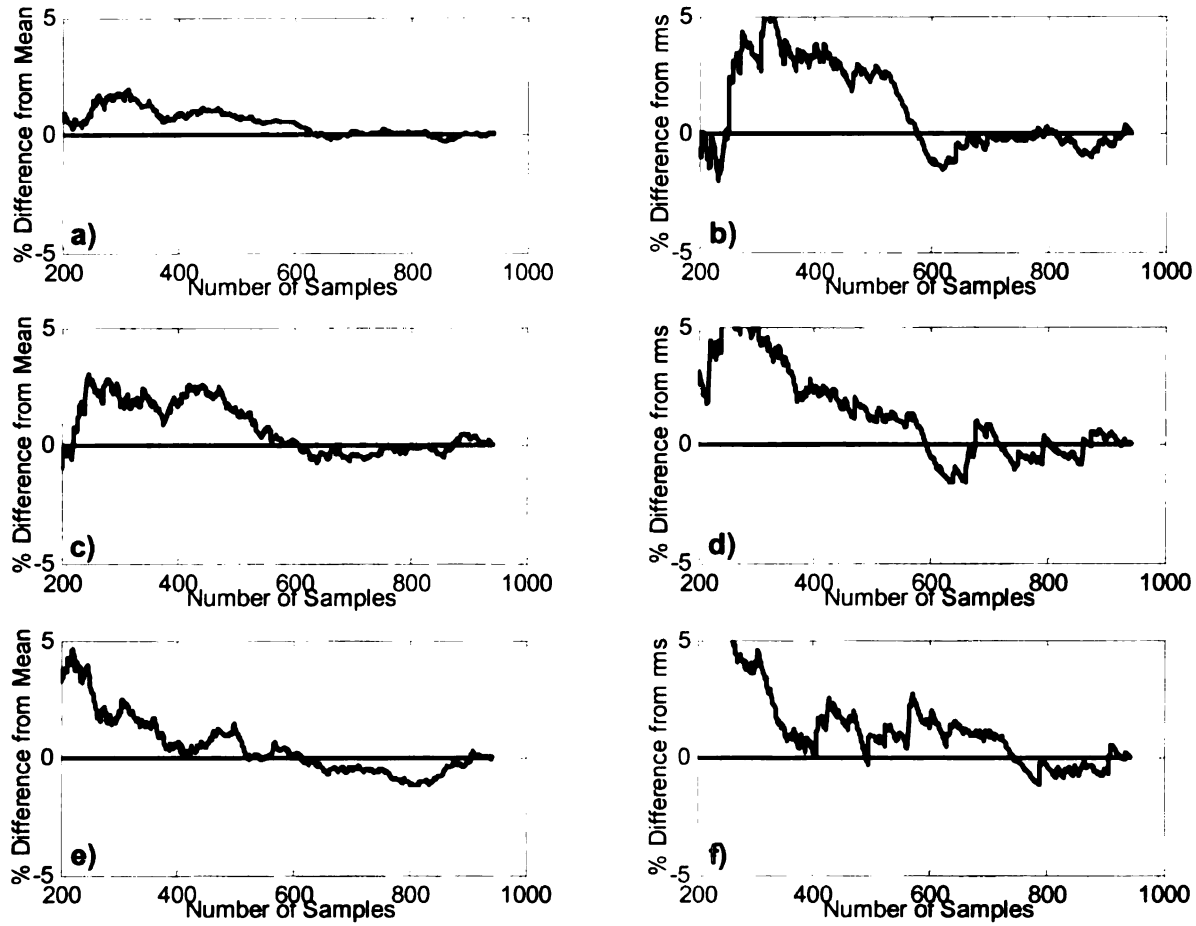


Figure 5.6 Plots of convergence for a representative data point acquired in the hot-wire measurements: a) mean radial component b) RMS of radial component c) mean tangential component d) RMS of tangential component e) mean axial component f) RMS of axial component

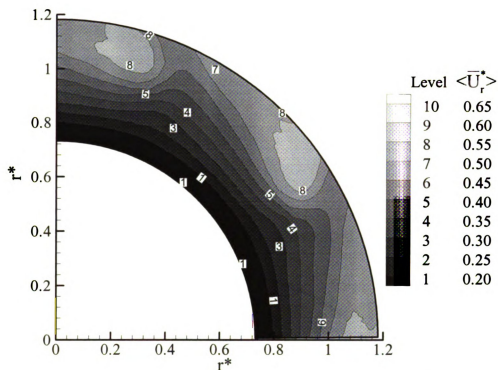


Figure 5.7 Phase averaged mean radial velocity for condition 'A'
($\chi=0.0, \phi=0.27$)

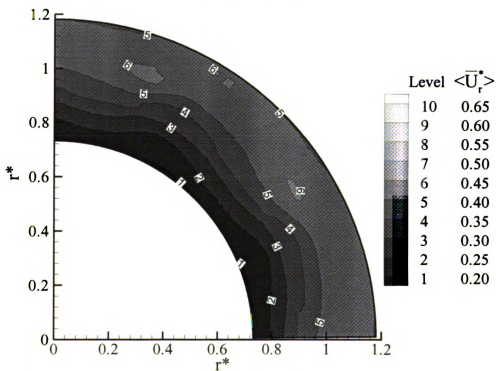


Figure 5.8 Phase averaged mean radial velocity for condition 'B'
($\chi=0.0, \phi=0.22$)

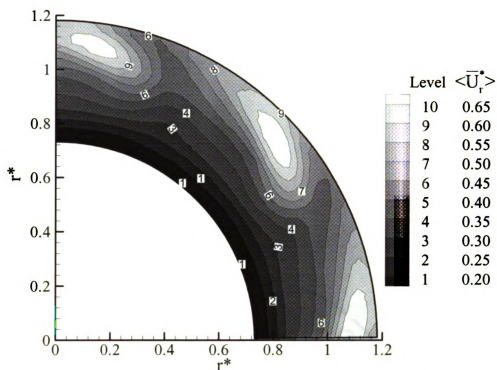


Figure 5.9 Phase averaged mean radial velocity for condition 'C'
($\chi=0.008, \phi=0.22$)

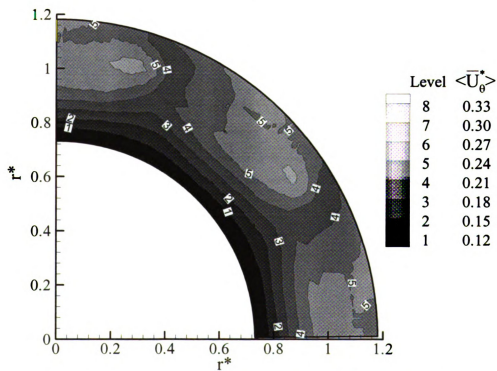


Figure 5.10 Phase averaged mean tangential velocity for condition 'A'
($\chi=0.0, \phi=0.27$)

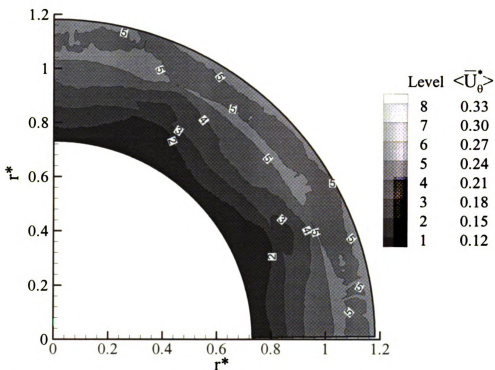


Figure 5.11 Phase averaged mean tangential velocity for condition 'B'
($\chi=0.0, \phi=0.22$)

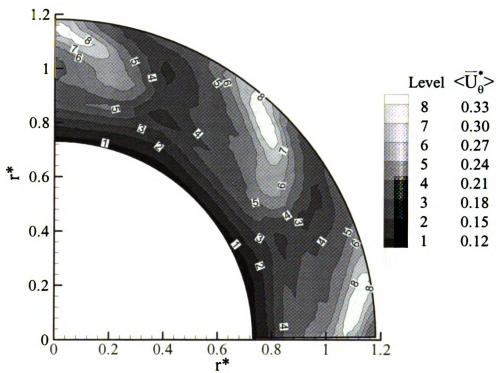


Figure 5.12 Phase averaged mean tangential velocity for condition 'C'
($\chi=0.008, \phi=0.22$)

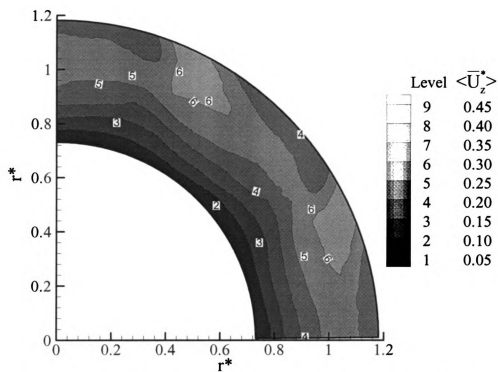


Figure 5.13 Phase averaged mean axial velocity for condition 'A'
($\chi=0.0, \phi=0.27$)

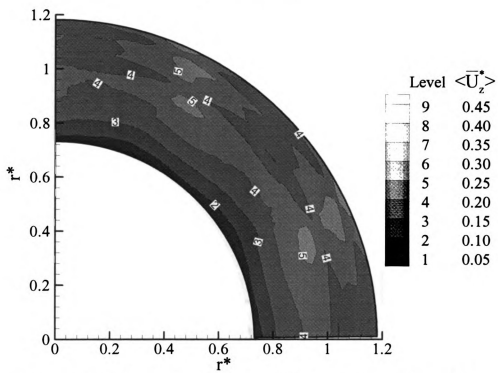


Figure 5.14 Phase averaged mean axial velocity for condition 'B'
($\chi=0.0, \phi=0.22$)

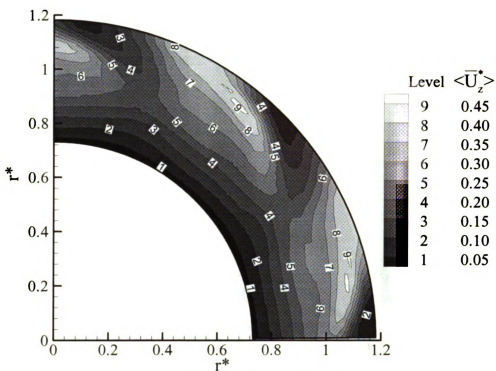


Figure 5.15 Phase averaged mean axial velocity for condition 'C'
($\chi=0.008, \phi=0.22$)

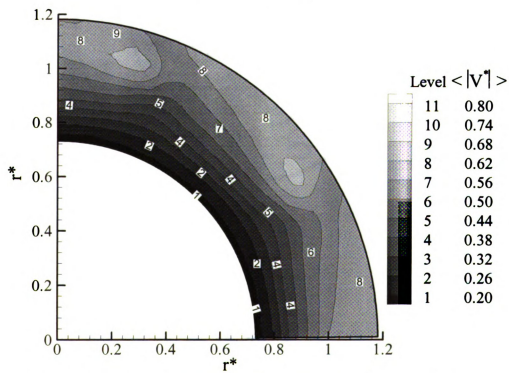


Figure 5.16 Phase averaged velocity magnitude for condition 'A'
($\chi=0.0, \phi=0.27$)

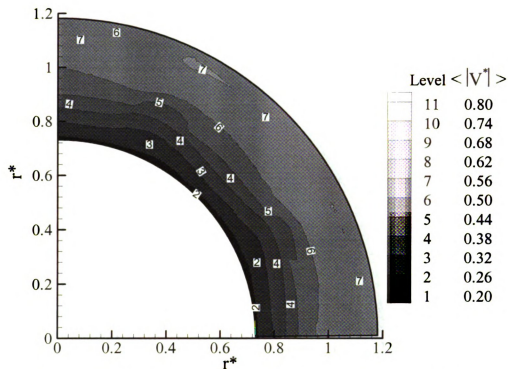


Figure 5.17 Phase averaged velocity magnitude for condition 'B'
($\chi=0.0, \phi=0.22$)

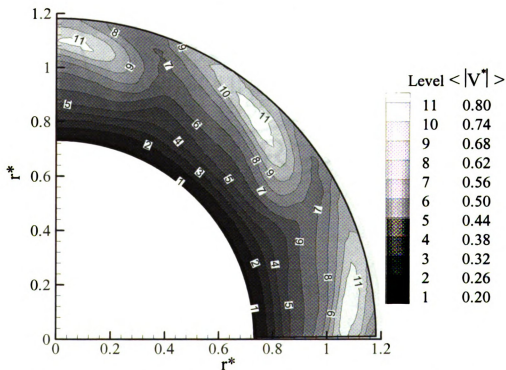


Figure 5.18 Phase averaged velocity magnitude for condition 'C'
($\chi=0.008, \phi=0.22$)

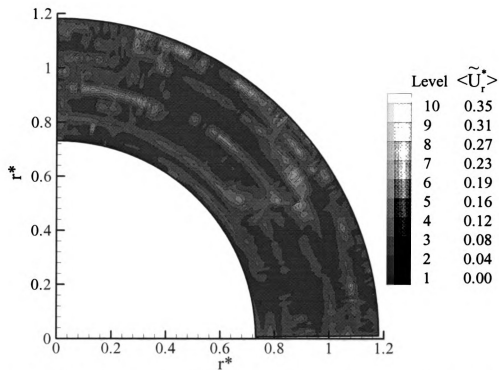


Figure 5.19 Phase averaged RMS of radial velocity for condition 'A'
($\chi=0.0$, $\phi=0.27$)

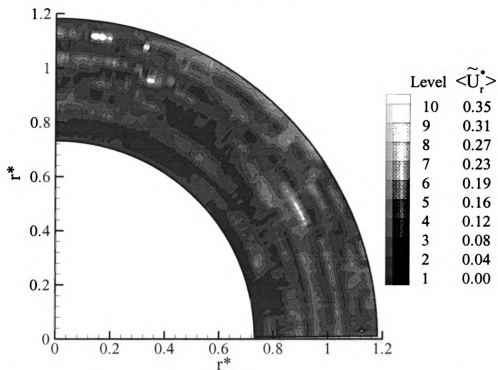


Figure 5.20 Phase averaged RMS of radial velocity for condition 'B'
($\chi=0.0$, $\phi=0.22$)

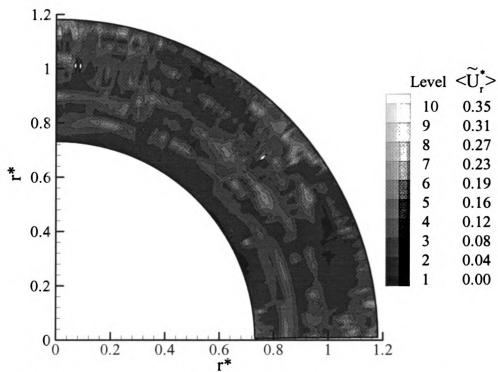


Figure 5.21 Phase averaged RMS of radial velocity for condition 'C' ($\chi=0.008, \phi=0.22$)

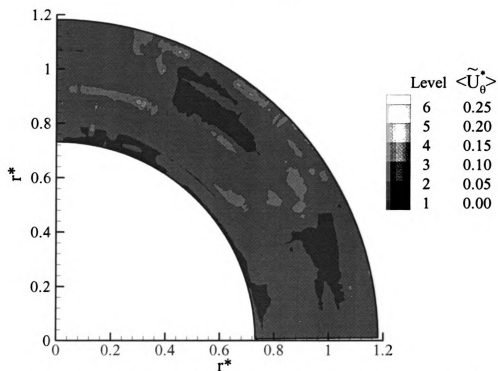


Figure 5.22 Phase averaged RMS of tangential velocity for condition 'A' ($\chi=0.0, \phi=0.27$)

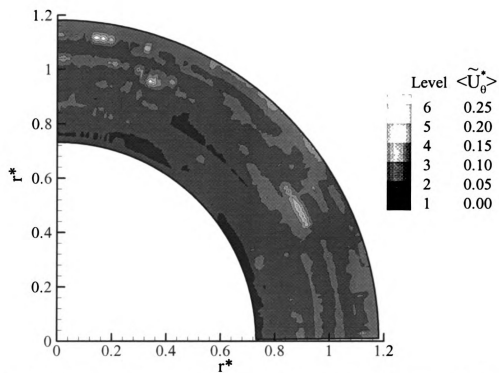


Figure 5.23 Phase averaged RMS of tangential velocity for condition 'B'
($\chi=0.0, \phi=0.22$)

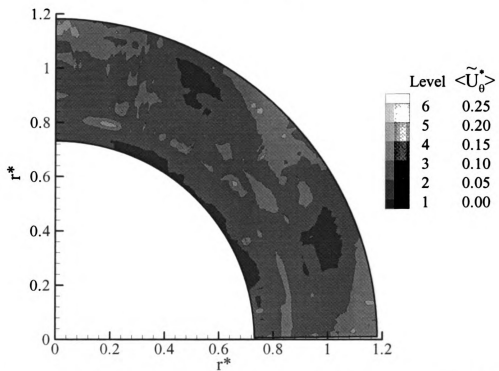


Figure 5.24 Phase averaged RMS of tangential velocity for condition 'C'
($\chi=0.008, \phi=0.22$)

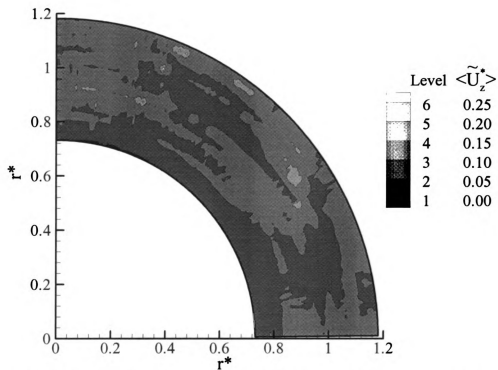


Figure 5.25 Phase averaged RMS of axial velocity for condition 'A'
($\chi=0.0, \phi=0.27$)

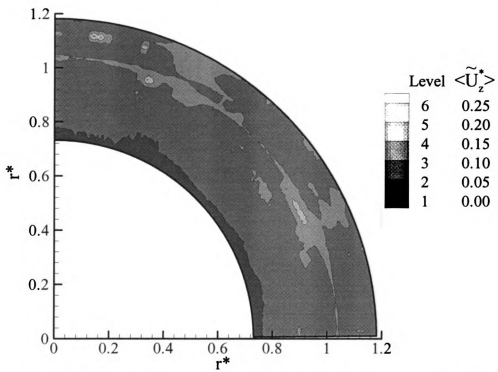


Figure 5.26 Phase averaged RMS of axial velocity for condition 'B'
($\chi=0.0, \phi=0.22$)

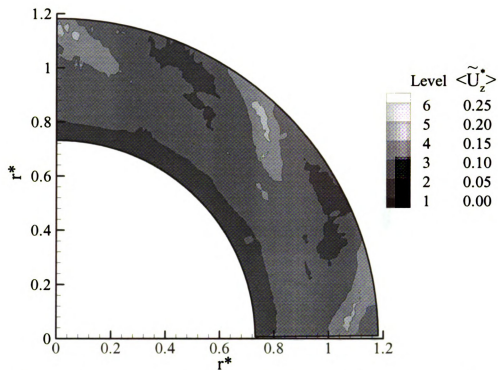


Figure 5.27 Phase averaged RMS of axial velocity for condition 'C'
($\chi=0.008, \phi=0.22$)

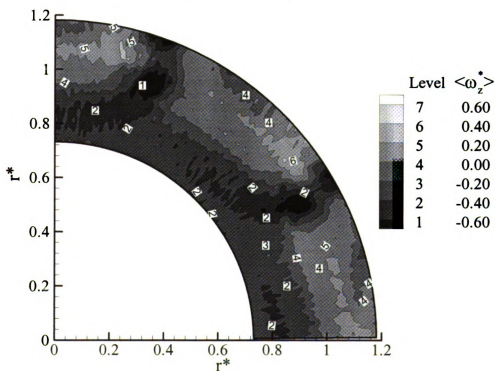


Figure 5.28 Phase averaged axial vorticity for condition 'A'
($\chi=0.0, \phi=0.27$)

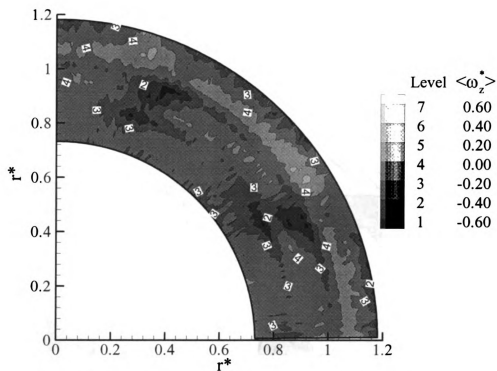


Figure 5.29 Phase averaged axial vorticity for condition 'B'
($\chi=0.0, \phi=0.22$)

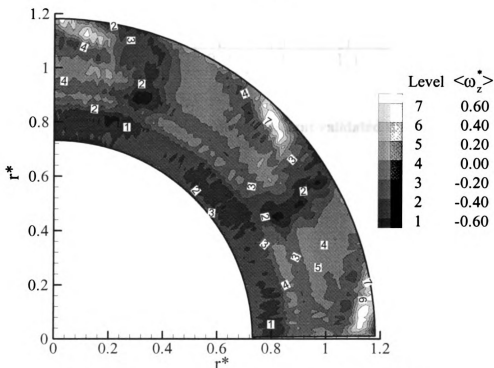


Figure 5.30 Phase averaged axial vorticity for condition 'C'
($\chi=0.008, \phi=0.22$)

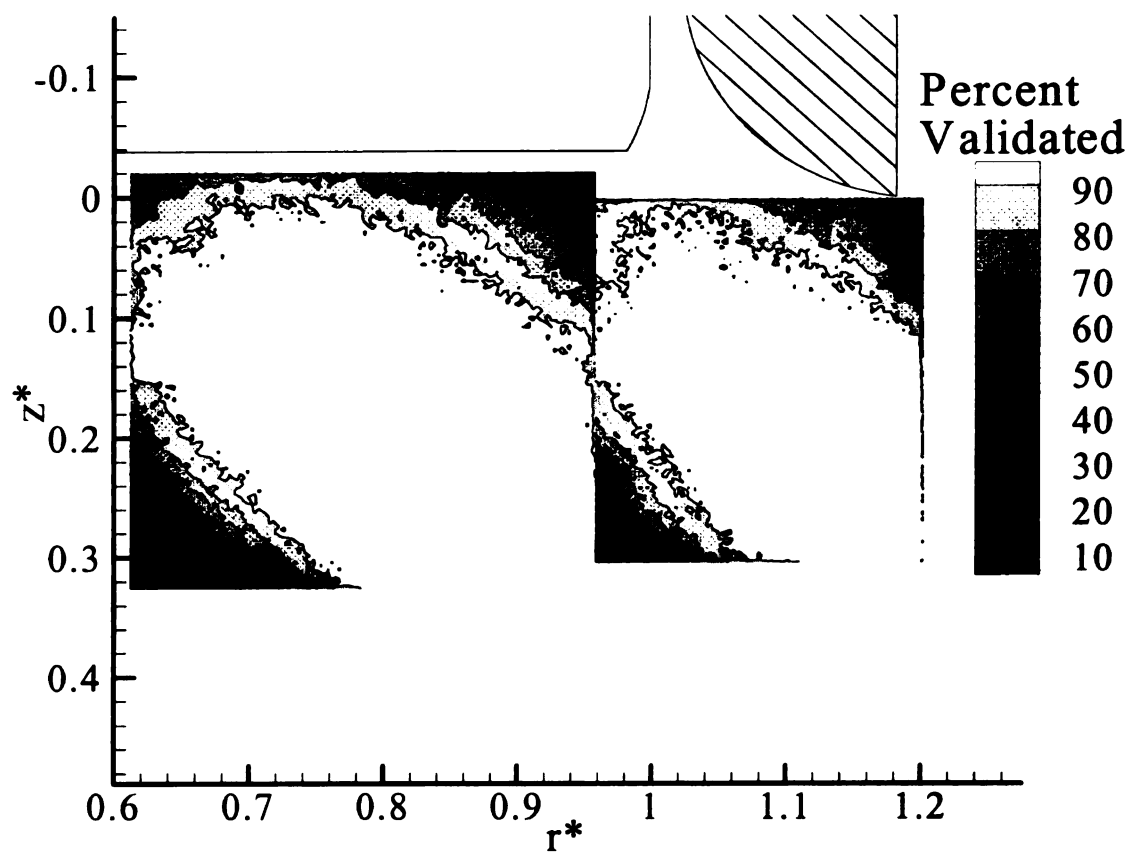


Figure 5.31 Representative contour plot of percent validated vectors for condition 'A'

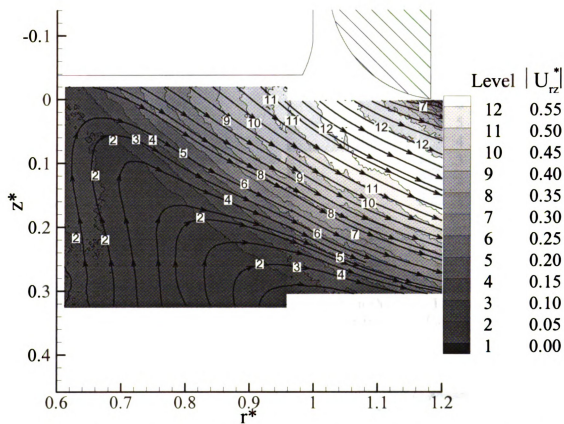


Figure 5.32 Time averaged PIV data: velocity magnitude in the r - z plane for condition 'A' ($\chi=0.0$, $\phi=0.27$)

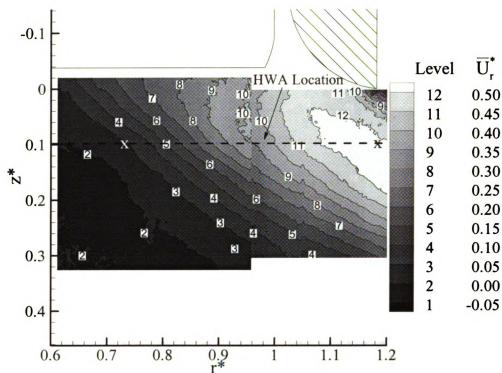


Figure 5.33 Time averaged PIV data: mean radial velocity for condition ‘A’. The axial location of the hot-wire survey is indicated on the plot. The two ‘x’ marks denote the radial range of the measurements.

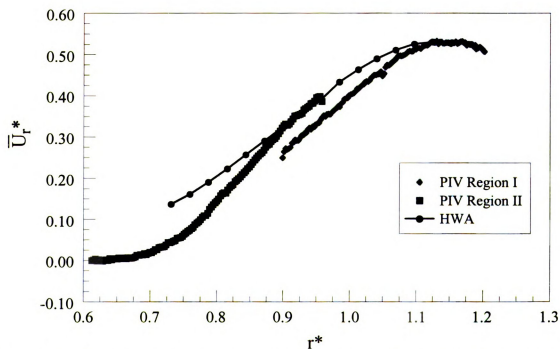


Figure 5.34 Comparison of the radial velocities measured by the hot-wire survey and the two PIV regions for condition ‘A’

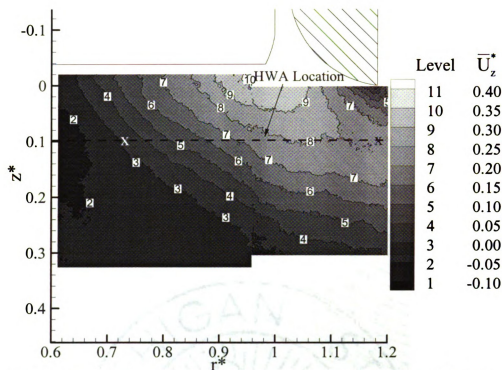


Figure 5.35 Time averaged PIV data: mean axial velocity for condition 'A'. The axial location of the hot-wire survey is indicated on the plot. The two 'x' marks denote the radial range of the measurements.

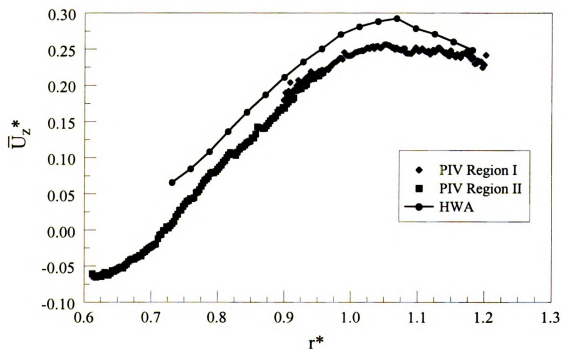


Figure 5.36 Comparison of the axial velocities measured by the hot-wire survey and the two PIV regions for condition 'A'

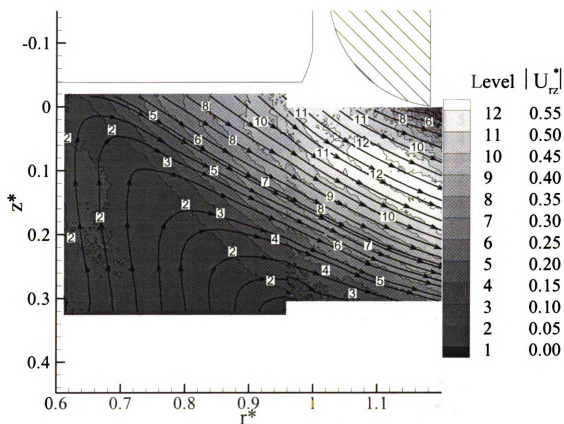


Figure 5.37 Time averaged PIV data: velocity magnitude in the r - z plane for condition 'C' ($\chi=0.008$, $\phi=0.22$)

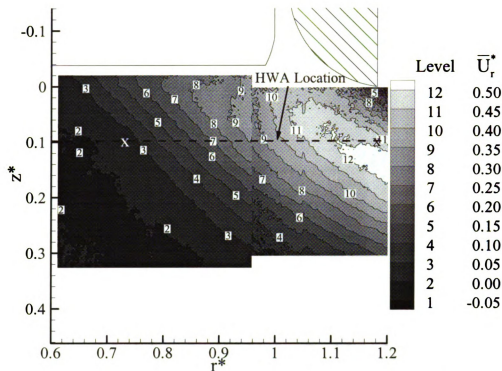


Figure 5.38 Time averaged PIV data: mean radial velocity for condition ‘C’. The axial location of the hot-wire survey is indicated on the plot. The two ‘x’ marks denote the radial range of the measurements.

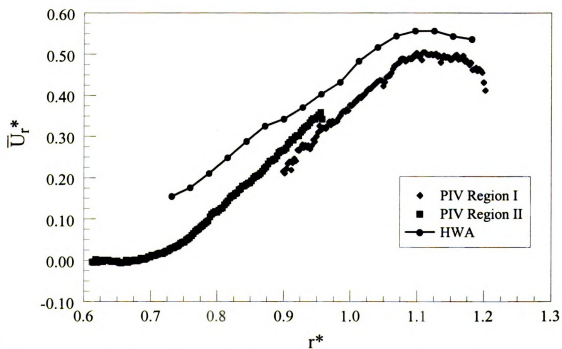


Figure 5.39 Comparison of the radial velocities measured by the hot-wire survey and the two PIV regions for condition ‘C’

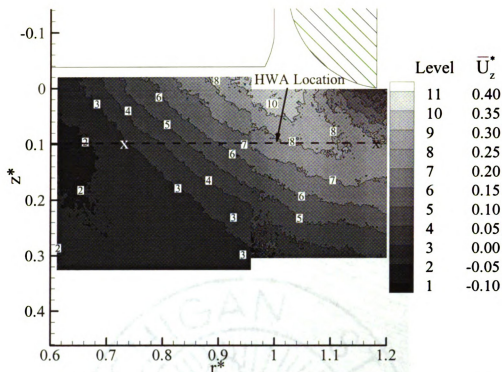


Figure 5.40 Time averaged PIV data: mean axial velocity for condition 'C'. The axial location of the hot-wire survey is indicated on the plot. The two 'x' marks denote the radial range of the measurements.

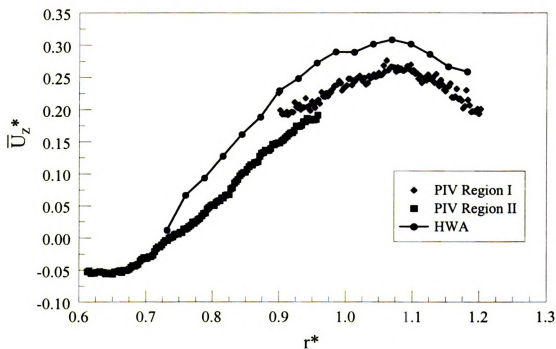


Figure 5.41 Comparison of the axial velocities measured by the hot-wire survey and the two PIV regions for condition 'C'

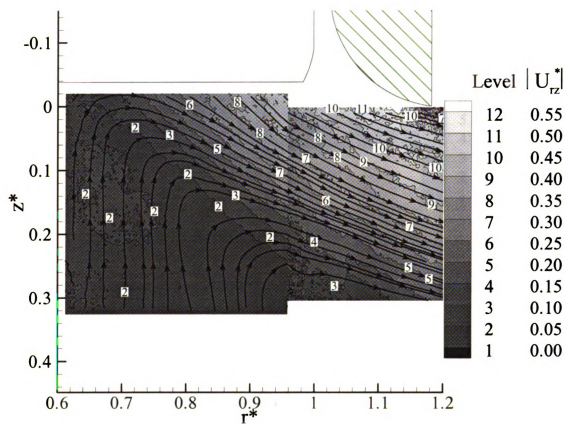


Figure 5.42 Time averaged PIV data: velocity magnitude in the r - z plane for condition 'D' ($\chi=0.022$, $\phi=0.21$)

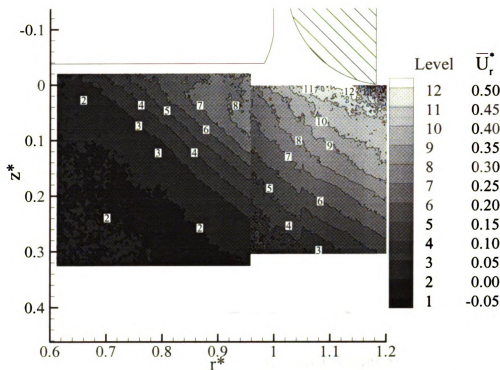


Figure 5.43 Time averaged PIV data: mean radial velocity for condition 'D' ($\chi=0.022$, $\phi=0.21$)

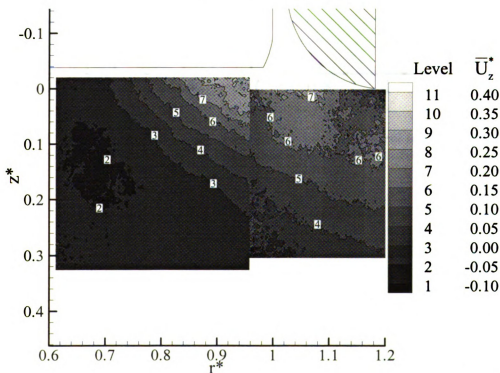


Figure 5.44 Time averaged PIV data: mean axial velocity for condition 'D' ($\chi=0.022$, $\phi=0.21$)

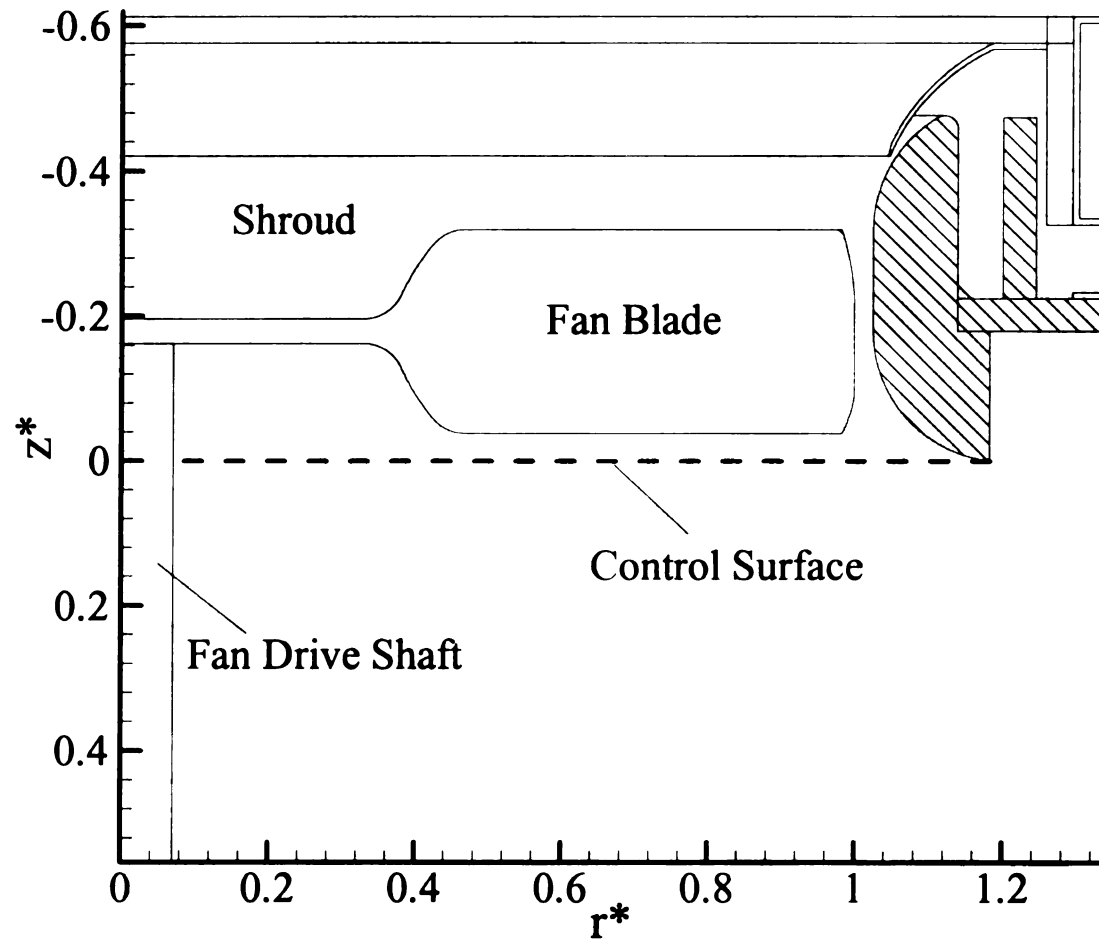


Figure 5.45 Fan and shroud configuration showing the control surface used for the integration of the flow rate from the PIV velocity measurements

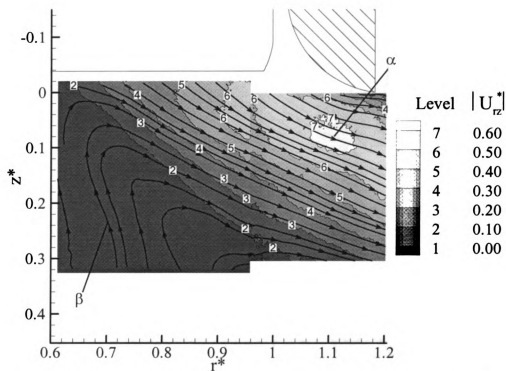


Figure 5.46 Phase averaged PIV data: velocity magnitude for condition 'A' at fan blade leading edge, note high velocity cell, α , and negative radial flow, β

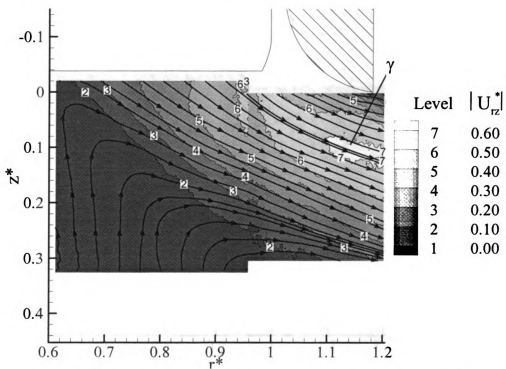


Figure 5.47 Phase averaged PIV data: velocity magnitude for condition 'A' at 5 degrees past leading edge, note the high velocity cell, γ

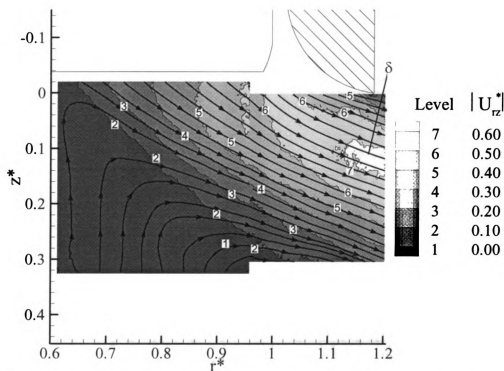


Figure 5.48 Phase averaged PIV data: velocity magnitude for condition 'A' at 10 degrees past leading edge, note high velocity cell, δ

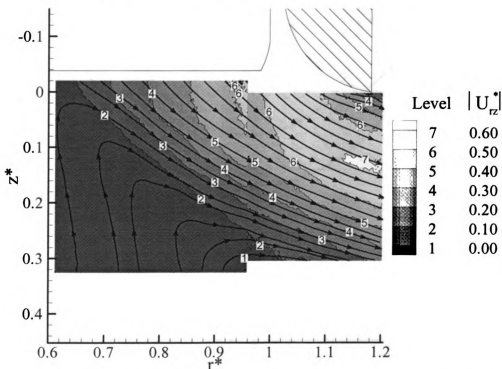


Figure 5.49 Phase averaged PIV data: velocity magnitude for condition 'A' at 15 degrees past leading edge

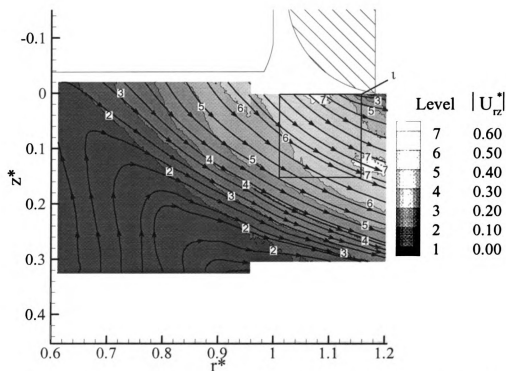


Figure 5.50 Phase averaged PIV data: velocity magnitude for condition 'A' at 20 degrees past leading edge, note the "path" of high velocity, u

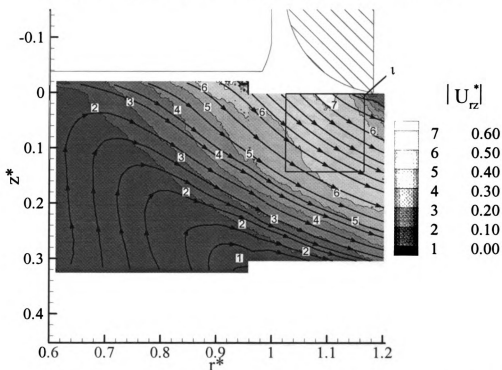


Figure 5.51 Phase averaged PIV data: velocity magnitude for condition 'A' at 25 degrees past leading edge, note the "path" of high velocity, u

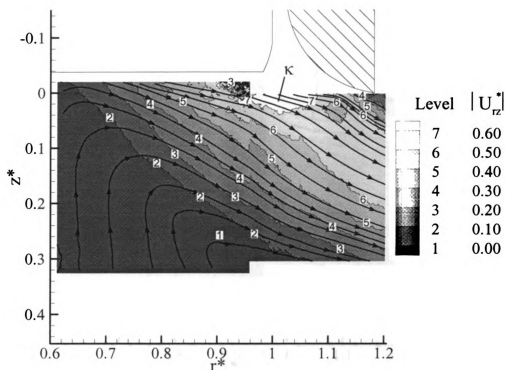


Figure 5.52 Phase averaged PIV data: velocity magnitude for condition 'A' at 30 degrees past leading edge, note high velocity cell, κ

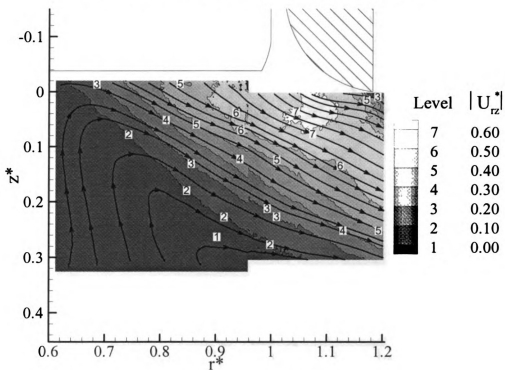


Figure 5.53 Phase averaged PIV data: velocity magnitude for condition 'A' at 35 degrees past leading edge

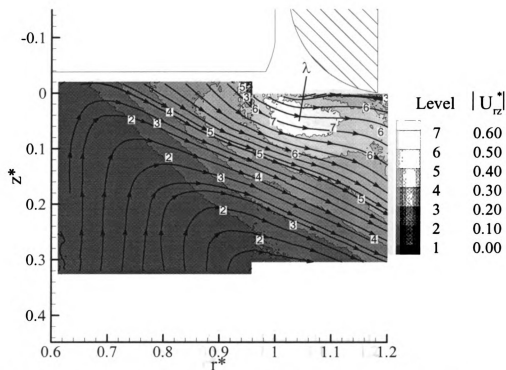


Figure 5.54 Phase averaged PIV data: velocity magnitude for condition ‘C’ at the fan blade leading edge, note the high velocity cell, λ

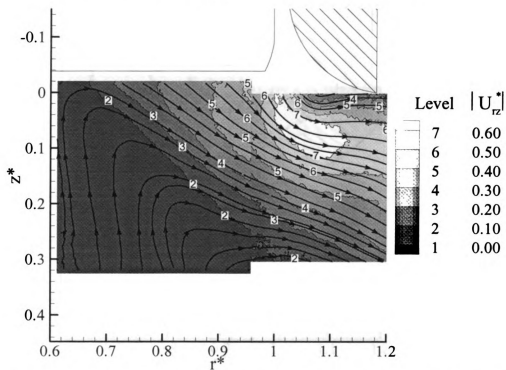


Figure 5.55 Phase averaged PIV data: velocity magnitude for condition ‘C’ at 5 degrees past the leading edge

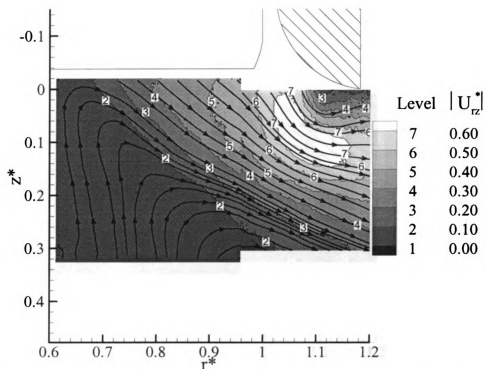


Figure 5.56 Phase averaged PIV data: velocity magnitude for condition 'C' at 10 degrees past the leading edge

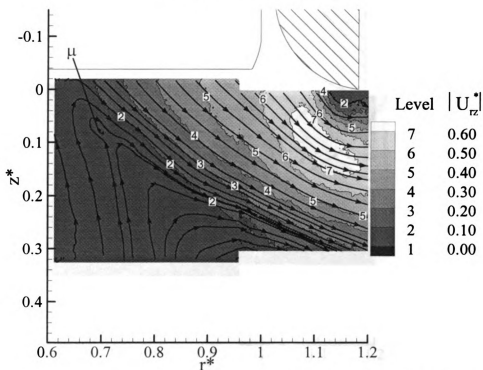


Figure 5.57 Phase averaged PIV data: velocity magnitude for condition 'C' at 15 degrees past the leading edge, note the recirculation region, μ

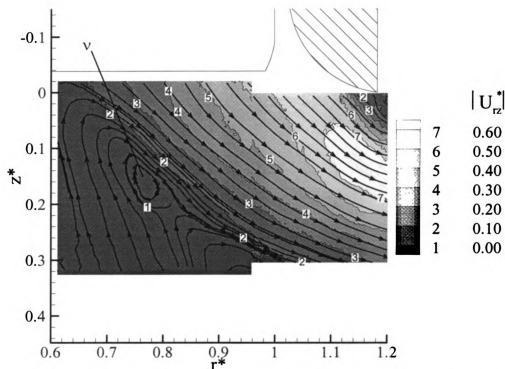


Figure 5.58 Phase averaged PIV data: velocity magnitude for condition 'C' at 20 degrees past the leading edge, note the large recirculation region, v

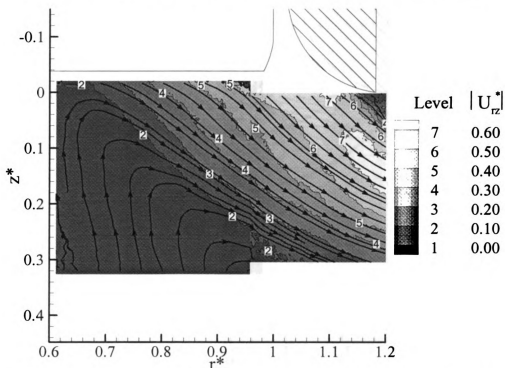


Figure 5.59 Phase averaged PIV data: velocity magnitude for condition 'C' at 25 degrees past the leading edge

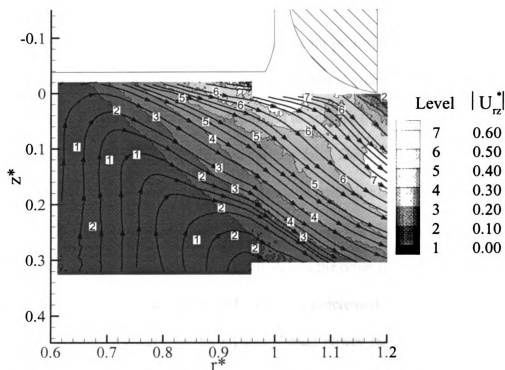


Figure 5.60 Phase averaged PIV data: velocity magnitude for condition 'C' at 30 degrees past the leading edge

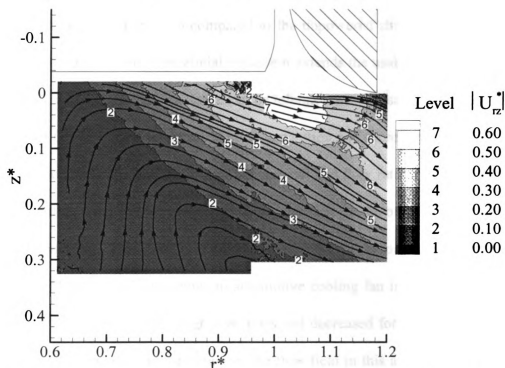


Figure 5.61 Phase averaged PIV data: velocity magnitude for condition 'C' at 35 degrees past the leading edge

6.0 Summary and Conclusions

An experimental investigation of a large scale prototype cooling fan utilizing a unique aerodynamic shroud has been conducted. Integral quantities were measured first to establish a baseline performance condition and to analyze the effect of the aerodynamic shroud on the overall performance of the cooling fan. Detailed velocity measurements were then acquired in the wake of the fan for selected operating conditions. The following conclusions are supported by the results of these experimental investigations.

1. The aerodynamic shroud effectively delays the onset of stall. The integral measurements clearly show that as the shroud power χ is increased, the stall point flow coefficient is decreased. In fact, for the highest shroud power tested, the stall point flow coefficient is decreased by 27%. A hot-wire survey supports these data by showing that the characteristics of the flow field in the wake of the fan for the powered shroud condition are qualitatively similar when compared to the unpowered shroud condition before stall. The delay of stall can be beneficial because it extends the usable operating range of the fan. Thus, a higher system resistance can be tolerated without sacrificing the undesirable effects of fan stall, such as increased noise generation and decreased efficiency.

2. The aerodynamic shroud does not increase the flow rate or efficiency of this large scale cooling fan for flow coefficients larger than that for the onset of the unpowered shroud stall condition. This result is in contrast to the previous aerodynamic shroud study conducted by Morris (1997) involving an automotive cooling fan in which the flow rate and efficiency were increased for high flow rates and decreased for low flow rates. The magnitude of tip clearance and the nature of the flow field in this area is the most likely cause for this new result. Specifically, the tip clearance area accounted for 31% of the

total effective flow area in the Morris (1997) study as compared to 5.6% in the present study. Moreover, the baseline efficiency of the fan in the present study is much higher (approximately 50% compared to 25%).

3. The dimensionless shroud power variable, χ , can fully characterize the contribution of the aerodynamic shroud. That is, only the combination of shroud pressure and shroud flow rate is necessary to determine how the aerodynamic shroud will affect the performance of the fan. This means that a small annular jet gap height (small volume flow rate) can be used with a large shroud pressure; or a large jet gap height (large volume flow rate) can be used with a smaller shroud pressure to achieve the same result. It should be noted that this can only be stated for the limitations of the present study (a 1 mm to 5 mm jet gap height). Thus, the combination of the shroud pressure and shroud flow rate allows a reduction of variables so that the pressure coefficient is only dependent upon the flow coefficient and the dimensionless shroud power. This result confirms the findings of Morris (1997) in that aerodynamic shroud study.

4. The aerodynamic shroud has a strong effect on the vortical flow field in the wake of the fan. The hot-wire axial vorticity data suggest that the areas of negative vorticity, that are assumed to be signatures of the tip leakage vortex, are weakened by the high momentum annular jet of the aerodynamic shroud. In fact, the peak negative vorticity is reduced by as much as 15% in the representative powered shroud condition. This is significant because the tip leakage vortex plays an important role in the total losses of the system. In particular, this secondary flow leads to increased viscous dissipation and can interact with the main (through) flow in the blade tip area causing additional losses from turbulent mixing.

5. The positive contribution of the aerodynamic shroud is strongly dependent on the design of the fan and the nature of the specific loss mechanisms present in the fan flow. Losses in a fan flow can occur from a number of sources including blade profile loss, shroud and passage loss, tip clearance loss and secondary flow loss (Neal, 2002). Lakshminarayana (1996) attributes 20-35% of total losses to tip clearance effects. However, Storer and Cumpsty (1994) suggest that a more realistic proportion attributed to the tip clearance flow itself is 10%. Nonetheless, tip clearance loss varies widely according to application. While the aerodynamic shroud may, in fact, decrease losses attributed to tip leakage flow, it may also be increasing losses from other sources such as secondary flows. The experimental results provide examples of this including the phase averaged PIV data, which show that, for the powered shroud condition, a region of recirculating flow downstream of the fan was observed that was not seen in the baseline shroud condition. The larger reverse flow region in the powered shroud condition is an example of the aerodynamic shroud negatively affecting other aspects of the flow field.

REFERENCES

- Baranski, B., (1997), "Designing the Engine Cooling Fan," SAE #740691.
- Bohl, D., (1996), "An Experimental Study of the Near Field Region of a Free Jet With Passive Mixing Tabs," M.S. Thesis, Department of Mechanical Engineering, Michigan State University, East Lansing, MI.
- D'Andrea, R., Behnken, R., Murray, R., (1996), "Rotating Stall Control of an Axial Flow Compressor using Pulsed Air Injection," *ASME J. of Turbomachinery*, 119, 742-752.
- Day, I.J., (1993), "Active Suppression of Rotating Stall and Surge in Axial Compressors," *ASME J. of Turbomachinery*, 115, 40-47.
- Foss, J.F., Morris, S.C., Neal, D.R., (2001), "Axial Fan Research for Automotive and Building Ventilation Applications", Tech Brief in *ASME J. of Fluids Eng.*, Spring 2001.
- Fukano, T., and Jang, C.M. (2003), "Tip Clearance Noise of Axial Flow Fans Operating at Design and Off-design Condition," *J. of Sound and Vib.*, Corrected proof, page numbers not assigned yet.
- Inoue, M., Kumoumaru, M., (1984), "Three Dimensional Structure and Decay of Vortices Behind an Axial Flow Rotating Blade Row," *Journal of Engineering for Power*, 106.
- Jang, C., Furukawa, M., and Inoue, M, (2001), "Noise Reduction by Controlling Tip Vortex in a Propeller Fan," *JSME Int. Journal*, 44(4), 748-755.
- Keane, R.D., Adrian, R.J., (1990), "Optimization of particle image velocimeters. Part I: Double Pulsed Systems," *Meas. Sci. Tech.*, 1, 1202-1215.
- LaVision GmbH., (2002), *FlowMaster Manual*, Gottingen, Germany, p. 37.
- Longhouse, R.E., (1978), "Control of Tip-Vortex Noise of Axial Flow Fans by Rotating Shrouds," *J. of Sound and Vibration*, 58(2), 201-214.
- Mellin, 1980, "Noise and Performance of Automotive Cooling Fan," SAE paper No. 740691.
- Mittal, S., Saxena, P., (2000), "Prediction and Hysteresis Associated with the Static Stall of an Airfoil," *AIAA Journal*, 38, (5), 933-935.
- Morris, S.C., (1997), "Experimental Investigation of an Aerodynamic Shroud for Cooling Fan Applications," M.S. thesis, Department of Mechanical Engineering, Michigan State University, East Lansing.

- Morris, S.C., Foss, J.F., (2001), "An Aerodynamic Shroud for Automotive Cooling Fans," *ASME J. Fluids Eng.*, 123, 287-292.
- Morris, S.C., Neal, D.R., Foss, J.F., Cloud, G.L., (2001), "A Moment-of-Momentum Flux Mass Air Flow Measurement Device," *Meas. Sci. Tech.*, 12, 9-13.
- Morris, S.C., Good, J.J., Foss, J.F., (1998), "Velocity Measurements in the Wake of an Automotive Cooling Fan," *Exp. Thermal and Fluid Sci.*, 17, 100-106.
- Morris, S.C., (2002), "The Velocity and Vorticity Fields of a Single Stream Shear Layer," Ph.D. Dissertation, Department of Mechanical Engineering, Michigan State University, East Lansing, MI.
- Neal, D.R., (2002), M.S. thesis, Department of Mechanical Engineering, Michigan State University, East Lansing.
- Potter, M.C., Foss, J.F., (1982), *Fluid Mechanics*, Great Lakes Press, Okemos.
- Smith, L.H., (1970), "Casing Boundary Layers in Mult-stage Compressors," *Proceedings of the Symposium of Flow Research on Blading*, Elsevier Publishing Company.
- Sorensen, D.N., (2001), "Minimizing the Trailing Edge Noise From Rotor-Only Axial Fans Using Design Optimization," *J. of Sound and Vibration*, 247(2), 305-323.
- Wu, S.F., Su, S., and Shah, H., (1998), "Noise Radiation From Engine Cooling Fans," *J. of Sound and Vibration*, 216(1), 107-132.

MICHIGAN STATE UNIVERSITY LIBRARIES



3 1293 02736 0951