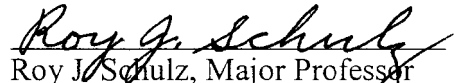
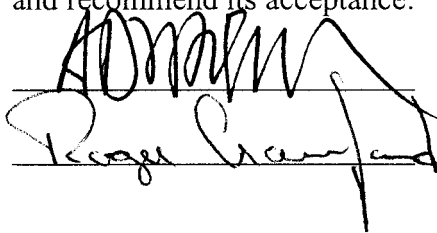


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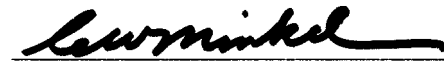
I am submitting herewith a thesis written by Sherri Lynette Smith entitled "One-Dimensional Mean Line Code Technique to Calculate Stage-by-Stage Compressor Characteristics." I have examined the final copy of the thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering.


Roy J. Schulz, Major Professor

We have read this thesis
and recommend its acceptance:


Roger Crawford

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Associate Vice Chancellor and
Dean of The Graduate School

1-D MEAN LINE CODE TECHNIQUE TO CALCULATE STAGE-BY-STAGE COMPRESSOR CHARACTERISTICS

A Thesis
Presented for the
Master of Science
Degree
The University of Tennessee, Knoxville

Sherri Lynette Smith
May 1999

DEDICATION

This thesis is dedicated to my Lord and Savior Jesus Christ who has provided me the opportunity and ability to complete it. It is also dedicated to my best friend, my husband, Scotty Smith. In addition, it is dedicated to my parents, Jimmy and Christine Holcomb, who have unselfishly given their love, guidance, and support throughout my life. It is also dedicated to my sister, Patti Roberts, who has been a continual source of encouragement. Lastly, it is dedicated to my grandmother, Ruby Holcomb, and to the loving memories of my grandmother, Dera Tallant, my grandfathers, Clifford Tallant and Woodrow Holcomb, and my great-grandmother, Vera Tallant.

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ABSTRACT

Because modern turbine engines have such complex flow fields, extensive testing is required to ensure stability. The testing of these engines and their components is very expensive. To offset testing costs, computer simulations are often used. Several computer simulations have been developed at Arnold Engineering Development Center for this purpose. These codes require stage-by-stage compressor characteristics in order to simulate flow through a compressor.

A technique to calculate compressor characteristics was developed using a mean line code (MLC). This MLC was modified to include loss and deviation correlations from open literature and to account for the conservation of angular momentum as the flow area changes between blade rows.

In this research, it was determined that the MLC could predict the compressor total pressure characteristics for the normal operating range within about 4.9-percent difference when compared to data for a single fan rotor compressor. For this same fan rotor, the MLC could predict the compressor total temperature characteristics within 1.2-percent difference.

It was also determined that for a single compressor stage, the MLC could predict the compressor total pressure characteristics for the normal operating conditions within 1.7-percent accuracy when compared to data. For this compressor stage, the MLC could predict the compressor total temperature characteristics within 1.0-percent accuracy.

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LIST OF SYMBOLS

Symbol	Definition
A	area
a	speed of sound
a/c	point of maximum camber as a fraction of chord
AVDR	axial velocity density ratio
BMFF	bladed mass flow function
c_p	specific heat at constant pressure
h	enthalpy
H^*	rothalpy
i	incidence
k	correction factor
M	Mach number
m	deviation slope factor
$\dot{m}$	mass flow
n	reference minimum-loss-incidence-angle slope factor, index
NBMFF	non-bladed mass flow function
P	pressure, interpolated term
R	gas constant for air
r	radius
T	temperature
T^*	reference temperature

U	wheel speed
V	absolute velocity
W	relative velocity, width
X	arbitrary variable

Greek

α	absolute flow angle
β	relative flow angle
β'	blade metal angle
δ	deviation
γ	specific heat ratio
ρ	density
σ	solidity
θ	camber angle
ϖ	total pressure loss coefficient
ζ	cross-sectional area correction factor

Subscript

1	Inlet
2	exit
3D	three-dimensional
10	10-percent thick NACA 65-(A ₁₀)-series blade

choke	choking condition
Data	pertaining to experimental data
hub	blade hub location
M	Mach number related
m	meridional
MLC	pertaining to the Mean Line Code
min	viscous forces related
r	relative to the blade row
ref	reference condition
s	average
shape	shape factor
stall	stalling condition
t	total, tangential direction component
th	thickness factor
tip	blade tip location
x	axial direction component
vloss	viscous loss
$\perp$	perpendicular
Superscript	
b	solidity exponent

1.0 INTRODUCTION

Modern turbine engines are very complex pieces of machinery. Furthermore, they have components that have high levels of operational complexity. One such component is the compressor. A compressor's job is to supply the turbine engine with the required airflow at the required pressure. A specific type of compressor in use today is an axial flow compressor. In an axial flow compressor, air flows in a predominately axial direction through one or more stages. A stage consists of a rotor and stator. The rotor is a rotating annular cascade of blades used to exchange kinetic energy for an increase in pressure rise through the blade row. A stator is a stationary annular cascade of blades used to diffuse and de-swirl the flow.

Discernment of the internal flow field, which can be obtained through altitude test facilities, is critical for both the design and analysis of modern turbine engines. Because testing turbine engines is expensive, a cheaper technique has been developed through the use of computer codes to simulate the compressor flow field. Arnold Engineering Development Center (AEDC) has developed several models to simulate airflow through all or part of an engine. Two such models, ATEC (Aerodynamic Turbine Engine Code) (Garrard, 1995) and DYNTTECC (DYNAMIC Turbine Engine Compressor Code) (Hale, 1992) employ the one-dimensional form of the governing equations. ATEC models an entire gas generator, and DYNTTECC models the compression system and combustor of a turbine engine. They both use a finite differencing technique to solve the equations of mass, momentum, and

energy simultaneously, using the following source terms: mass bleed, blade forces, and shaft work. For these models to work well, they must have accurate compressor stage-by-stage characteristic maps, which are normally in the form of total pressure ratio and total temperature ratio versus corrected flow, for a variety of shaft rotational speeds.

There are different ways to find these characteristics; one way is to get them from experimental data. Unfortunately, data is not always available from engine manufacturers due to proprietary restrictions. Often, available characteristics are in the form of loss and deviation correlations versus diffusion factor. Another way to find these characteristics is to use a stage-by-stage compressor code to calculate them. Codes that employ the three-dimensional form of the governing equations (3-D codes) provide detailed analyses, but are computationally demanding and require blade geometry details that are difficult to acquire. Codes that employ the two-dimensional form of the governing equations (2-D codes) are more feasible for this purpose but require a radial distribution of detailed blade geometry, which can also be difficult to obtain from engine manufacturers. Codes that employ the one-dimensional form of the governing equations (1-D codes) require the least amount of blade information and may produce adequate results. The code developed in this thesis is a 1-D mean line code, which is able to calculate stage-by-stage compressor characteristics, which can then be used in 1-D dynamic codes such as ATEC and DYNTECC.

The work included in this thesis expands a single-blade row mean line code (MLC) to simulate all of the blade rows of a compressor, including calculations to cross the space between blade rows. The MLC includes transonic loss and deviation correlations from the open literature to cross the bladed regions and presents stage-by-stage results from the MLC compared to experimental data to determine the accuracy of the predictions.

2.0 LITERATURE REVIEW

Computer simulations such as ATEC (Garrard, 1995) and DYNTECC (Hale, 1992) require stage-by-stage compressor characteristics to compute compression system performance. These characteristics can be determined from experimental data or mathematical models. The current state-of-the-art technique to obtain characteristics is typified by an example of experimental data and 1-D, 2-D, and 3-D numerical simulations. The approach of this thesis will then be summarized as a balance between the given techniques and the available data necessary to characterize the blades.

A typical example of the use of experimental data to build characteristics is given by Schreiber and Starken (1983) through the wind tunnel testing of a cascade from a transonic compressor rotor blade section. Their test included varying the Mach number from 0.8 to 1.1 and the inlet flow angle from stall to choke. The effects of changing the axial velocity density ratio, which is defined to be

$$AVDR = \frac{\rho_2 W_2 \sin(\beta_2)}{\rho_1 W_1 \sin(\beta_1)} = \frac{\rho_2 V_{x2}}{\rho_1 V_{x1}}, \quad [3.1]$$

where β_1 and β_2 are defined as the inlet and exit flow angles with respect to the front of the cascade, was investigated by using an end wall suction device. Shock losses were also examined. The losses measured in this experiment agreed well with the losses measured in the corresponding transonic compressor rotor blade element. As was expected, experimental data from cascade work or actual compressors can be used to calculate accurate stage-by-stage characteristics.

Unfortunately, unlike this example, experimental data is often not available due to proprietary restrictions.

Bloch (1991) developed a 1-D compressor stage performance model called POSTALL4 that simulated pre-stall and post-stall compressor behavior. This model assumed incompressible flow in both forward and reversed flow. Included in Bloch's thesis is a sensitivity study of small changes in design variables on steady-state performance. This model was applied to two compressors: a 3-stage, low speed compressor and a 10-stage, high-speed compressor. The model had favorable results when applied to the 3-stage compressor, and mixed results when applied to the 10-stage compressor. The unfavorable results in the 10-stage compressor were thought to be due to flow recirculation and compressibility effects, which could not be accounted for by a steady-state, incompressible, 1-D code. There was an additional limitation with Bloch's model; it assumed the flow conditions at the inlet of the stator were equal to the exit conditions of the previous rotor. This assumption does not allow angular momentum to be conserved between blade rows with a change in annular cross-sectional flow geometry.

For the forward, unstalled flow, Bloch used a deviation angle calculation based on Carter's correlation (Horlock, 1973). He used profile loss calculations from Leiblein (1959), that included trailing edge form factors from the Koch and Smith (1976) correlation. The annulus and secondary losses were estimated by equations provided by Dixon (1975). For the forward, stalled flow, Bloch uses approximations by Moses and Thomason (1986) to calculate the loss coefficient

and fully mixed flow angle. Annulus and secondary losses are not calculated for stalled flow because they are considered negligible compared to the profile loss. For the reversed flow, the flow angle is assumed to equal the metal angle, as suggested by Turner and Sparkes (1964) and Koff and Greitzer (1986). The loss was determined using a parabolic fit to five coincident curves of data presented by Carneal (1990). No losses were calculated for the last stator because it was assumed to act like an inlet guide vane, that is, with negligible pressure loss, as was suggested by Gamache (1985).

Streamline curvature methods are 2-D methods for calculating stage-by-stage compressor characteristics. These methods use equations to calculate pressure gradients in the radial, or spanwise direction (Cumpsty, 1989). They simulate a radial distribution of 2-D flow plus swirl. There are different approaches to this method. Smith (1966) obtains these equations from the equations of motion. Denton (1986) uses an approach that has a more physical focus, and can be applied to a wide range of geometries. The streamline curvature model developed by Sayari and Boics (1995) uses an averaging procedure that works normal to the central streamline instead of the more commonly used circumferential averaging procedure.

The streamline curvature model used by Klepper (1998) is basically a duct flow solver modified to include loss and deviation correlations. The Streamline Curvature Code was written by Hearsey (1994), and the correlations were a transonic set collected by Hearsey. The model was applied to a single fan blade row

and presented favorable results after calibration. Results were within 3-percent of experimental data for a 1D representation and 6-percent for a 2-D representation. Two-dimensional models require more detailed geometry information than 1-D models. These additional details are often under proprietary restrictions, and are therefore often unavailable.

Three-dimensional models would be more detailed than 1-D or 2-D models, but they require extensive blade geometry information, which is often unavailable. Also, they are computationally demanding, often taking days or weeks to obtain a solution. If a 3-D model were used to calculate stage-by-stage characteristics, there may be no need for the models that required the characteristics in the first place. Also, it would be very time consuming.

One-dimensional simulations require less extensive blade geometry information than 2-D or 3-D simulations. The model presented in this thesis is a 1-D code to predict stage-by-stage pressure characteristics. The author took an existing 1-D mean line code (MLC) and developed it from a single-blade compressor simulation into a code that can simulate all of the blade rows of a fan or core compressor, including calculations of the regions in-between blade rows to account for the conservation of angular momentum between blade rows with a change in annular geometry. This new 1-D mean line code includes the loss and deviation correlations used by Klepper (1998). Inputs into the code include: inlet total pressure and temperature, inlet absolute flow angle, inlet Mach number, inlet and exit mean radii, inlet and exit cross-sectional area, inlet and exit blade metal

angles, cross-sectional area correction factor, gas constant for air, specific heat ratio, mass flow, and angular velocity. It calculates exit total pressure and temperature, which are used to calculate the compressor characteristics, such as adiabatic or polytropic efficiencies, as well as total pressure ratio and total temperature ratio. It also calculates exit static pressure and temperature, and inlet and exit absolute, tangential, and relative velocities. The approach of this code is discussed in the following chapter.

3.0 APPROACH

To obtain stage-by-stage compressor characteristics (total pressure ratio and total temperature ratio versus corrected flow), a 1-D mean line code (MLC) was developed from a single-blade row compressor simulation into a code that can simulate all of the blade rows of a fan or core compressor. The MLC uses the compressible fluid relations to simulate modern transonic compressors. In addition to simulating the flow field across bladed regions (rotors and stators), the space between the rotor and stator will be simulated to maintain the conservation of angular momentum as the cross sectional area changes. The specification of the axial velocity density ratio (AVDR), and loss and deviation obtained from experimental data or correlations from open literature establish sufficient information for crossing the bladed regions.

3.1 Flow through a Bladed Region

As shown in Step 1 of Figure 3.1¹, common inlet conditions, such as total inlet pressure, total inlet temperature, inlet Mach number, and the inlet flow angle are known. Inlet static pressure, inlet static temperature and inlet speed of sound can be found using isentropic relationships. Inlet relative velocity, inlet relative flow angle, and inlet relative Mach number can be found by using velocity triangles (Step 2). As shown in Step 3, isentropic relationships can then be used to calculate the inlet relative total pressure and the inlet relative total temperature (these are the total conditions of the flow that the rotating blade row sees):

¹ All figures can be found in Appendix A

$$P_{t1r} = P_1 \left[1 + \frac{\gamma-1}{2} M_{1r}^2 \right]^{\frac{\gamma}{\gamma-1}} \quad [3.1]$$

$$T_{t1r} = T_1 \left[1 + \frac{\gamma-1}{2} M_{1r}^2 \right], \quad [3.2]$$

where P_{t1r} is the inlet relative total pressure, T_{t1r} is the inlet relative total temperature, P_1 is inlet static pressure, T_1 is inlet static temperature, γ is the specific heat ratio, and M_{1r} is the inlet relative mach number. The relative mass flow function for bladed regions (BMFF_r) is needed for crossing the bladed region.

3.1.1 Development of the Relative Mass Flow Function for Bladed Regions

For the bladed region, the mass flow function is defined using the area perpendicular to the flow because we know the exit flow angle as a function of the deviation angle and the exit metal angle. Mass flow is density multiplied by the dot product of area and velocity:

$$\dot{m} = \rho (\bar{A} \bullet \bar{W}) = \rho \cos(\theta) |\bar{A}| |\bar{W}|, \quad [3.3]$$

where W is the relative velocity. Velocity triangles are shown in Figure 3.2. If

$\theta = 0$, $\bar{A} = A_{\perp W_x} \hat{i} + A_{\perp W_y} \hat{j} + A_{\perp W_z} \hat{k}$ and $\bar{W} = W_x \hat{i} + W_y \hat{j} + W_z \hat{k}$, the mass flow can

then be written as

$$\dot{m} = \rho A_{\perp W} W, \quad [3.4]$$

where $A_{\perp W}$ is the area perpendicular to the relative velocity and

$$W = \sqrt{W_x^2 + W_y^2 + W_z^2}. \quad [3.5]$$

From isentropic relationships, static pressure and static temperature are given by:

$$P = P_{tr} \left[1 + \frac{\gamma - 1}{2} M_r^2 \right]^{-\left(\frac{\gamma}{\gamma - 1}\right)} \text{ and} \quad [3.6]$$

$$T = T_{tr} \left[1 + \frac{\gamma - 1}{2} M_r^2 \right]^{-1}. \quad [3.7]$$

Equations [3.6] and [3.7] are substituted into the ideal gas equation of state, which is then solved for density. This equation for density,

$$\rho = \frac{P}{RT} = \frac{P_{tr} \left[1 + \frac{\gamma - 1}{2} M_r^2 \right]^{-\left(\frac{\gamma}{\gamma - 1}\right)}}{RT_{tr} \left[1 + \frac{\gamma - 1}{2} M_r^2 \right]^{-1}}, \quad [3.8]$$

along with the product of the relative Mach number and the speed of sound,

$$W = M_r a = M_r \sqrt{\gamma RT}, \quad [3.9]$$

are substituted into equation [3.4], yielding the following equation for mass flow:

$$\dot{m} = \frac{P_{tr} \left[1 + \frac{\gamma - 1}{2} M_r^2 \right]^{-\left(\frac{\gamma}{\gamma - 1}\right)} A_{\perp w} M_r \sqrt{\gamma RT}}{RT_{tr} \left[1 + \frac{\gamma - 1}{2} M_r^2 \right]^{-1}}. \quad [3.10]$$

Solving this equation for the relative mass flow function of the bladed region

(BMFF_r) yields:

$$BMFF_r = \frac{\dot{m} \sqrt{T_{tr}}}{P_{tr} A_{\perp w}} = M_r \sqrt{\frac{\gamma}{R} \left(1 + \frac{\gamma - 1}{2} M_r^2 \right)^{\frac{-(\gamma + 1)}{2(\gamma - 1)}}}. \quad [3.11]$$

The relative mass flow function at the blade inlet (BMFF_{lr}) can then be calculated by:

$$\text{BMFF}_{1r} = M_{1r} \sqrt{\frac{\gamma}{R} \left(1 + \frac{\gamma-1}{2} M_{1r}^2 \right)^{\frac{-(\gamma+1)}{2(\gamma-1)}}}, \quad [3.12]$$

ending Step 4.

The BMFF_{1r} and three ratios, relative total temperature ratio, relative total pressure ratio, and the ratio of areas perpendicular to the flow are needed to cross the bladed region. Either experimental data or loss and deviation correlations can be used to calculate the coefficient of total pressure loss relative to the blade row, whose definition is

$$\varpi_r = \frac{P_{t1r} - P_{t2r}}{P_{t1r} - P_1} = \frac{1 - \frac{P_{t2r}}{P_{t1r}}}{1 - \frac{P_1}{P_{t1r}}}, \quad [3.13]$$

and the deviation angle, defined in Figure 3.2 as

$$\delta = \beta_2 - \beta'_2 \quad [3.14]$$

(Step 5, Figure 3.1). Once ϖ_r , δ , and the axial velocity density ratio (AVDR) are known, the three ratios can be calculated. The calculation of these three ratios, including the development of AVDR, is described in the following sections.

3.1.2 Development of the Relative Total Temperature Ratio

The first ratio needed for crossing bladed regions is the relative total temperature ratio. The calculation of the relative total temperature ratio begins with two equations: the first law of thermodynamics and the Euler turbine equation. The first law of thermodynamics is written in terms of enthalpy and absolute velocity:

$$\text{work} = \left(h_2 + \frac{V_2^2}{2} \right) - \left(h_1 + \frac{V_1^2}{2} \right), \quad [3.15]$$

where h_1 and h_2 are enthalpy at the inlet and exit, respectively, and V_1 and V_2 are absolute velocity at the inlet and exit, respectively. The Euler turbine equation gives work in terms of wheel speed, relative velocity, and absolute velocity:

$$\text{work} = \frac{1}{2} \left[(U_2^2 - U_1^2) - (W_2^2 - W_1^2) + (V_2^2 - V_1^2) \right], \quad [3.16]$$

where U_1 and U_2 are wheel speed at the inlet and exit, respectively (for derivation of equation [3.16], see Appendix E). When equations [3.15] and [3.16] are set equal to each other, the absolute velocities subtract out. This results in two equations equal to a constant rothalpy (H^*):

$$H^* = h_1 + \frac{W_1^2}{2} - \frac{U_1^2}{2} = h_2 + \frac{W_2^2}{2} - \frac{U_2^2}{2}. \quad [3.17]$$

Dividing by C_p gives a reference temperature (T^*) in terms of relative total temperature and wheel speed at the inlet and the exit:

$$T^* = T_{t1r} - \frac{U_1^2}{2C_p} \quad [3.18]$$

$$T^* = T_{t2r} - \frac{U_2^2}{2C_p}. \quad [3.19]$$

Setting equations [3.18] and [3.19] equal to each other and solving for the relative total temperature ratio yields:

$$\frac{T_{t2r}}{T_{t1r}} = 1 + \frac{U_2^2 - U_1^2}{2C_p T_{t1r}}. \quad [3.20]$$

3.1.3 Development of the Relative Total Pressure Ratio

The second ratio needed for crossing bladed regions is the relative total pressure ratio. The equation for the relative total pressure loss coefficient is given in Section 3.1.1 as

$$\varpi_r = \frac{P_{t1r} - P_{t2r}}{P_{t1r} - P_1} = \frac{1 - \frac{P_{t2r}}{P_{t1r}}}{1 - \frac{P_1}{P_{t1r}}}, \quad [3.13]$$

assuming that the streamlines come in and go out at the same radius. Solving equation [3.13] for the relative total pressure ratio yields:

$$\frac{P_{t2r}}{P_{t1r}} = 1 - \varpi_r \left(1 - \frac{P_1}{P_{t1r}} \right). \quad [3.21]$$

In compressors, however, the inlet and exit radii are not uniform due to a reduction of cross-sectional area with axial distance, associated with a rise in density. Equation [3.13] is modified to reflect the mean radius change, as shown in Leiblein (1965):

$$\varpi_r = \frac{(P_{t1r})_{id} - P_{t2r}}{P_{t1r} - P_1} = \frac{\left(\frac{P_{t2r}}{P_{t1r}} \right)_{id} - \frac{P_{t2r}}{P_{t1r}}}{1 - \frac{P_1}{P_{t1r}}}. \quad [3.22]$$

Solving for the relative total pressure ratio gives:

$$\frac{P_{t2r}}{P_{t1r}} = \left(\frac{P_{t2r}}{P_{t1r}} \right)_{id} - \varpi_r \left(1 - \frac{P_1}{P_{t1r}} \right). \quad [3.23]$$

From isentropic relationships, the ideal relative total pressure ratio is given by:

$$\left(\frac{P_{t2r}}{P_{t1r}} \right)_{id} = \left(\frac{T_{t2r}}{T_{t1r}} \right)^{\frac{\gamma}{\gamma-1}}, \quad [3.24]$$

where the relative total temperature ratio is derived in Section 3.1.2 to be

$$\frac{T_{t2r}}{T_{t1r}} = 1 + \frac{U_2^2 - U_1^2}{2C_p T_{t1r}}. \quad [3.20]$$

For derivation of equation [3.24], see Appendix E. The relative total temperature ratio can also be written as:

$$\frac{T_{t2r}}{T_{t1r}} = 1 + \frac{\omega^2 (r_2^2 - r_1^2)}{2C_p T_{t1r}}. \quad [3.25]$$

Note that if the exit radius equals the inlet radius, equation [3.23] reduces to equation [3.21].

3.1.4 Development of the Ratio of Areas Perpendicular to Relative Velocity

The third ratio needed for crossing bladed regions is the ratio of areas perpendicular to the flow. As stated in Section 3.1.1, the equation for mass flow in the direction of flow is:

$$\dot{m} = \rho A_{\perp W} W. \quad [3.4]$$

The steady continuity equation states that the mass exiting must equal the mass entering, so

$$\dot{m} = \rho_1 A_{\perp W_1} W_1 = \rho_2 A_{\perp W_2} W_2. \quad [3.26]$$

The ratio of areas perpendicular to the flow is then

$$\left(\frac{A_1}{A_2} \right)_{\perp W} = \frac{W_2 \rho_2}{W_1 \rho_1}. \quad [3.27]$$

From the velocity triangles shown in Figure 3.2, neglecting radial velocity across the annulus, this becomes

$$\left(\frac{A_1}{A_2} \right)_{\perp W} = \frac{\frac{V_{x2}}{\cos(\beta_2)} \rho_2}{\frac{V_{x1}}{\cos(\beta_1)} \rho_1}, \text{ or} \quad [3.28]$$

$$\left(\frac{A_1}{A_2} \right)_{\perp W} = \frac{\cos(\beta_1)}{\cos(\beta_2)} AVDR, \quad [3.29]$$

where β_1 and β_2 are the relative flow angles at the inlet and exit, respectively, and AVDR is the axial velocity density ratio. AVDR is found using the definition of mass flow and the continuity equation in the following way. The mass flow is equal to the density multiplied by the dot product of area and velocity:

$$\dot{m} = \rho (\vec{A} \bullet \vec{V}) = \rho \cos(\theta) |\vec{A}| |\vec{V}|. \quad [3.30]$$

If $\vec{A} = A_{\perp V_x} \hat{i} + 0\hat{j} + 0\hat{k}$ and $\vec{V} = V_x \hat{i} + V_y \hat{j} + V_z \hat{k}$, the mass flow can be rewritten as

$$\dot{m} = \rho A_{\perp V_x} V_x, \quad [3.31]$$

where $A_{\perp V_x}$ is the area perpendicular to the axial velocity. A sketch depicting the flow through an annular cascade is shown in Figure 3.3. Setting the inlet mass flow equal to the exit mass flow yields:

$$A_{\perp V_{x1}} V_{x1} \rho_1 = A_{\perp V_{x2}} V_{x2} \rho_2. \quad [3.32]$$

If both sides of equation [3.32] are divided by the inlet axial velocity and density, the result is the axial velocity density ratio (AVDR):

$$AVDR = \frac{V_{x2}\rho_2}{V_{x1}\rho_1}, \quad [3.33]$$

which is also equal to the ratio of areas perpendicular to the axial velocity,

$$AVDR = \left(\frac{A_1}{A_2} \right)_{\perp V_x}. \quad [3.34]$$

The AVDR can be determined using equation [3.33] or [3.34], depending on what is known. In using the MLC, the exit axial velocity and density are not known but the casing geometry is. The AVDR can then be determined by:

$$AVDR = \left(\frac{A_1}{A_2} \right)_{\perp V_x} = \zeta \frac{(r_{tip1}^2 - r_{hub1}^2)}{(r_{tip2}^2 - r_{hub2}^2)}, \quad [3.35]$$

where r_{tip} and r_{hub} represent the radii at the tip and hub, respectively. ζ is a cross-sectional area correction factor that accounts for

- boundary layer growth
- error associated with using a 1-D representation for a radial distribution of inlet flow angle
- and error associated with the neglecting the radial velocity.

The relative mass flow function at the exit of the blade ($BMFF_{2r}$) can then be found using the following equation (Step 6):

$$BMFF_{2r} = BMFF_{1r} \left(\frac{A_1}{A_2} \right)_{\perp W} \left(\frac{P_{t1r}}{P_{t2r}} \right) \left(\sqrt{\frac{T_{t2r}}{T_{t1r}}} \right), \quad [3.36]$$

with $\left(\frac{A_1}{A_2}\right)_{\perp w}$, $\frac{P_{t1r}}{P_{t2r}}$, and $\frac{T_{t2r}}{T_{t1r}}$ as defined in equations [3.34], [3.23], and [3.20],

respectively. Then, the exit relative mach number (M_{2r}) can be found iteratively from:

$$BMFF_{2r} = M_{2r} \sqrt{\frac{\gamma}{R}} \left(1 + \frac{\gamma-1}{2} M_{2r}^2 \right)^{\frac{-(\gamma+1)}{2(\gamma-1)}}, \quad [3.37]$$

as indicated in Step 7. Finally, using isentropic relationships and velocity triangles (Steps 8-10), the exit total pressure and exit total temperature can be calculated, completing the calculation of the simulation through the bladed row. Next, the calculation of the flow through the space between blade rows, or the non-bladed region (NBR), will be discussed.

3.2 Flow through a Non-bladed Region

Once P_{t2} , T_{t2} , M_2 , α_2 , and V_{t2} (exit tangential velocity) are found for a rotor, they are used as the subsequent inlet conditions, P_{t1} , T_{t1} , M_1 , α_1 , and V_{t1} , for the non-bladed space between the rotor and stator. An algorithm for calculating the flow through a NBR is shown in Figure 3.4. As shown in Step 1 of Figure 3.4, the mass flow function of the non-bladed region (NBMFF), developed in the following section, is constructed to maintain conservation of angular momentum between bladed regions.

3.2.1 Development of the Mass Flow Function for Non-Bladed Regions

As stated earlier, for the bladed region, the mass flow function is defined using the area perpendicular to the flow because we know the exit flow angle as a

function of the deviation angle and the exit metal angle. For the NBR, it is defined using the area perpendicular to the axial velocity, since the exit flow angle is unknown and must be calculated from the contribution of angular momentum in the analysis. In Section 3.1.3, the mass flow in the axial direction was derived as:

$$\dot{m} = \rho A_{\perp} V_x. \quad [3.31]$$

From isentropic relationships, the static pressure and temperature are given by:

$$P = P_t \left[1 + \frac{\gamma - 1}{2} M^2 \right]^{-\left(\frac{\gamma}{\gamma - 1}\right)} \quad \text{and} \quad [3.38]$$

$$T = T_t \left[1 + \frac{\gamma - 1}{2} M^2 \right]^{-1}, \quad [3.39]$$

where P_t is total pressure, T_t is total temperature, and M is mach number. These equations are substituted into the ideal gas equation and it is solved for density.

This equation for density,

$$\rho = \frac{P}{RT} = \frac{P_t \left[1 + \frac{\gamma - 1}{2} M^2 \right]^{-\left(\frac{\gamma}{\gamma - 1}\right)}}{RT_t \left[1 + \frac{\gamma - 1}{2} M^2 \right]^{-1}}, \quad [3.40]$$

along with the axial component of Mach number and the speed of sound,

$$V_x = M_x a = M_x \sqrt{\gamma RT} \quad [3.41]$$

are substituted into equation [3.31], yielding the following equation for mass flow:

$$\dot{m} = \frac{P_t \left[1 + \frac{\gamma-1}{2} M^2 \right]^{\left(\frac{\gamma}{\gamma-1} \right)} A_{\perp V_x} M_x \sqrt{\gamma R T}}{R T_t \left[1 + \frac{\gamma-1}{2} M^2 \right]^{-1}}. \quad [3.42]$$

The axial component of Mach number can be written as a function of absolute Mach number, tangential velocity, and the speed of sound:

$$M_{x2} = \sqrt{M_2^2 - M_{t2}^2} = \sqrt{M_2^2 - \left(\frac{V_{t2}}{a_2} \right)^2}. \quad [3.43]$$

The mass flow function of the non-bladed region (NBMFF) can then be written as:

$$\text{NBMFF} = \frac{\dot{m} \sqrt{T_t}}{P_t A_{\perp V_x}} = \sqrt{M^2 - \left(\frac{V_t}{\sqrt{\gamma R \frac{T_t}{1 + \frac{\gamma-1}{2} M^2}}} \right)^2} \sqrt{\frac{\gamma}{R} \left(1 + \frac{\gamma-1}{2} M^2 \right)^{\frac{-(\gamma+1)}{2(\gamma-1)}}}. \quad [3.44]$$

Therefore, the inlet NBMFF is:

$$\text{NBMFF}_1 = \sqrt{M_1^2 - \left(\frac{V_{t1}}{\sqrt{\gamma R \frac{T_{t1}}{1 + \frac{\gamma-1}{2} M_1^2}}} \right)^2} \sqrt{\frac{\gamma}{R} \left(1 + \frac{\gamma-1}{2} M_1^2 \right)^{\frac{-(\gamma+1)}{2(\gamma-1)}}}. \quad [3.45]$$

3.2.2 Development of Ratios Needed for Crossing the Non-Bladed Regions

In addition to the NBMFF_1 , four ratios are needed for crossing the NBR (Step 2). The first is the ratio of tangential velocities, which is given by the conservation of angular momentum:

$$\frac{V_{t2}}{V_{t1}} = \frac{r_1}{r_2}, \quad [3.46]$$

where r_1 and r_2 are the radii at the inlet and exit, respectively. The second is the ratio of total temperature. It is assumed to be one, with the assumption of no work or heat transfer:

$$\frac{T_{t1}}{T_{t2}} = 1. \quad [3.47]$$

The third is the total pressure ratio. Since there is no wheel velocity, equation [3.23] reduces to:

$$\frac{P_{t2}}{P_{t1}} = 1 - \varpi \left(1 - \frac{P_1}{P_{t1}} \right). \quad [3.48]$$

Often, loss correlations do not calculate a loss coefficient for the NBR. If there is no experimental data available for the space, ϖ can either be set to zero, or it can be estimated using the experimental data for the blades on either side of the NBR.

The fourth ratio needed for crossing the NBR is the ratio of areas perpendicular to axial velocity, $\left(\frac{A_1}{A_2} \right)_{\perp V_a}$. Since β_2 for the non-bladed region is unknown, the mass flow functions (NBMFF) for this region are calculated using the area perpendicular the respective axial velocity. Obtained from the continuity equation, the ratio of areas perpendicular to the axial velocities is:

$$\left(\frac{A_1}{A_2} \right)_{\perp V_a} = \varsigma \frac{\pi (r_{tip1}^2 - r_{hub1}^2)}{\pi (r_{tip2}^2 - r_{hub2}^2)}, \quad [3.34]$$

as discussed in Section 3.1.3. The mass flow function at the exit of the NBR can then be found using the following equation (Step 3):

$$\text{NBMFF}_2 = \text{NBMFF}_1 \left(\frac{A_1}{A_2} \right)_{\perp V_x} \left(\frac{P_{t1}}{P_{t2}} \right) \left(\sqrt{\frac{T_{t2}}{T_{t1}}} \right). \quad [3.49]$$

By iteration on

$$\text{NBMFF}_2 = \sqrt{M_2^2 - \left[\frac{V_{t1} \frac{V_{t2}}{V_{t1}}}{\sqrt{\gamma R \frac{T_{t1}}{1 + \frac{\gamma-1}{2} M_2^2}}} \right]^2} \sqrt{\frac{\gamma}{R}} \left(1 + \frac{\gamma-1}{2} M_2^2 \right)^{\frac{-(\gamma+1)}{2(\gamma-1)}}, \quad [3.50]$$

the exit Mach number can be calculated (Step 4, Figure 3.4). Then, using isentropic relationships, P_2 and T_2 can be found (Step 5). Once the static temperature is known, the speed of sound can be calculated and used to calculate the absolute velocity (Step 6). Finally, using velocity triangles, the exit flow angle can be calculated (Step 7), completing the calculation of flow through the non-bladed region.

4.0 EXPERIMENTAL DATA

Two machines were modeled for this thesis. Design and performance characteristics of these machines, as well as the testing of them, is presented. Finally discussed is the treatment of their experimental data required by the analysis of this thesis.

4.1 Sources of Experimental Data

The first machine modeled by the MLC is the NASA Rotor 1B compressor (Seyler, 1967) and the second is the NASA Stage 35 compressor (Reid, 1978). The NASA Rotor 1B compressor is a transonic, single blade row compressor fan and the NASA Stage 35 compressor is the first stage of a core compressor. Both have performance characteristics described by experimental data sets for multiple speed lines.

4.1.1 NASA Rotor 1B

In the 1960's the development of workable titanium alloys allowed the production of blades that could achieve tip speeds of 1,400 feet per second and greater. At these speeds, shock losses due to Mach number are important. NASA Rotor 1B (Seyler, 1967) is a medium aspect ratio single transonic rotor designed to be one of the rotor blades of a compressor fan. Its blades have a combination double-circular-arc and multiple-circular-arc design, which is a common design technique for modern military fans. Its double-circular-arc sections are from its hub to 60-percent of its span, and the remaining 40-percent of the blade has a multiple-

circular-arc blade profile. It was tested in 1967 and has experimental data for 100-, 90-, 70-, and 50-percent of design corrected speed.

Rotor 1B has a tip radius of 18.25 inches and a hub-tip ratio of 0.5. It is a forty-four blade rotor and has a rotor tip solidity of 1.3. Its design corrected weight flow is 215.49 lbs/sec and at design it provides a total pressure ratio of 1.6 and a diffusion factor of 0.35 at the tip for a speed of approximately 8800 rpm. It has no inlet or exit guide vanes and has a mid-span damper to insure rotor blade structural integrity.

4.1.1.1 Test Facility

NASA Rotor 1B was tested at General Electric's House Compressor Test Facility at Lynn, Massachusetts in 1967. The rotor ingested air from the atmosphere through two filter banks. The first filter was designed to filter twenty-two-percent of dust particles larger than three to five microns, and the second was designed to filter ninety to ninety-five-percent of the same size particles. After passing through the two-bank filter system, the air passed through a coarse wire inlet screen, a bell-mouth, and finally a flow straightener. There was a ratio of 2.24 reduction in area from the exit of the flow straightener to the inlet of the rotor. After exiting the rotor, the air was split into two streams. The inner stream of air went through a flow straightener and a venturi flow meter, and was then discharged to the atmosphere. The outer stream of air went through a slide cylindrical throttle valve, into a collector, and was then dispersed into pipes containing a flow

straightener and a venturi flow meter. The air was finally exhausted into the atmosphere. A schematic of the test facility is shown in Figure 4.1.

4.1.1.2 Instrumentation

The conditions at the inlet of Rotor 1B were measured by twenty-four thermocouples located on the inlet screen and six seven-element pitot-static rakes located fourteen inches downstream of the inlet flow straightener. Traverse probes were used to record blade element data at the rotor exit. These probes were immersed to five radial locations: 10-, 30-, 50-, 70-, and 90-percent of annulus height. Probes upstream and downstream of the rotor were immersed relative to where the corresponding design streamlines passed. Throughout the flow path, the casing and hub contained large numbers of static pressure taps. On the walls, the static pressures were measured at different axial locations, but at one circumferential location, except where traverse probes or rakes were located. Where the traverse probes were located, measurements were made at more than one circumferential location. The static pressure wedge probes, thermocouple rakes, and traverse probes were all calibrated for Mach number effects. Calibrations for small pitch and yaw angle variations were not necessary because the probes were sufficiently insensitive to these variations. A schematic of the instrumentation layout is shown in Figure 4.2.

4.1.2 NASA Stage 35

NASA Stage 35 (Reid, 1978) was designed in the 1970's . It was designed to be the first stage of a core compressor. It has multiple-circular-arc blade profiles.

It operates as a highly loaded stage, and has low aspect ratio blades, which is common in modern military engines. Experimental data for Stage 35 exists for 100-, 90-, and 70-percent of design corrected speed.

The rotor of Stage 35 has a tip radius of .2525 meters, and a hub-tip ratio of 0.7. It has thirty-six blades, a tip solidity of 1.3, and a maximum diffusion factor of 0.48 at a speed of 17,189 rpm. The stator has forty-six blades, a tip solidity of 1.3, a hub solidity of 1.47, and a maximum diffusion factor of 0.34. Stage 35 has a design mass flow of 20.20 kg/sec. It has an inlet rotor tip speed of 455 m/sec, and a stage design total pressure ratio of 1.82. The blades were designed with a chord - to - thickness ratio capable of insuring structural integrity to eliminate the need for a mid-span damper.

4.1.2.1 Test Facility

Stage 35 was tested in the NASA Lewis Research Center's Single-Stage Compressor Facility in Cleveland, Ohio, in 1978. An inlet in the roof of the facility allowed atmospheric air to enter. This air passed through a flow-measuring orifice and then into the plenum chamber, which was upstream of the test stage. The air passed through the test stage into a collector and then exited the facility through a vacuum exhaust system. A schematic of the test facility is shown in Figure 4.3.

4.1.2.2 Instrumentation

Measurements from a calibrated thin-plate orifice were used to determine the mass flow through the stage. The average of measurements from two chromel-constantan thermocouples was used as the orifice flow temperature. Rotational

speed was measured by an electronic speed counter in conjunction with a magnetic pickup. Calibrated transducers measured the orifice flow pressures.

Measurements were made radially across the flow upstream of the rotor were made using two combination probes, which measured total temperature, total pressure, and flow angle, and using two 18° wedge probes, which measured static pressure and flow angle. Each probe was automatically aligned with the flow direction by a null-balancing control system. Temperature was measured using chromel-constantan thermocouples. Inner- and outer-wall static pressure taps were placed at the same circumferential locations as the combination probes just mentioned. No measurements were made between rotor and stator due to their close spacing. Downstream of the stator, combination probes and wedge probes were placed both circumferentially and radially. The estimated uncertainties in the data can be seen in Table 4.1². A schematic of the instrumentation layout is shown in Figure 4.4. Station 1 is just upstream of the rotor, and Station 3 is just downstream of the stator.

4.2 Internal Consistency Check Procedure

It was determined in the present research study that the 2-D experimental data obtained from Rotor 1B and Stage 35 was internally consistent, i.e., any variable in the data set could be calculated using other variables in the data set, and the calculated value of the variable would equal the value of that variable listed in the data set. However, a question can be raised: Does a mass-weighted average

² All tables can be found in Appendix B

value (MWA) of each flow variable represent a consistent 1-D representation of the previously consistent 2-D experimental data. In other words, after the radially-distributed variables are used to obtain mass-weighted average 1-D values, can other MWA variables in the data set be used to calculate the same value for that variable? In order to check the internal consistency of the 1-D representation, an internally consistent data set was constructed and compared with the MWA values of the experimental data. For details of the mass-weighted average method used, see Appendix C. The internally consistent 1-D data set was constructed using a minimum set of mass-weighted averages of variables and other specified variables. These variables were chosen because they are natural quantities needed at the inlet of the blade and for crossing the blade. As shown in the two left columns of Figure 4.5, these variables are

- mass-weighted average: inlet total pressure, inlet total temperature, inlet flow angle, inlet mach number, loss, and deviation,
- other specified variables: ratio of specific heats, gas constant, wheel speed, mean area radii, and mean area metal angles.

AVDR is also required to calculate the flow fields. To obtain an AVDR for each mass flow, an additional exit MWA quantity, P_2 , is accepted. Using the above mentioned specified variables, the fundamental flow properties listed in Figure 4.5, and this MWA P_2 , an AVDR can be backed out for each mass flow. Once AVDR and the variables in the left columns of Figure 4.5 are known, the variables shown in the right columns of Figure 4.5 were calculated using fundamental flow

properties. Comparisons of the mass-weighted averages and these internally consistent 1-D data sets for Rotor 1B and Stage 35 will now be discussed.

4.2.1 Internal Consistency Check Applied to Rotor 1B

Comparisons of mass-weighted averaged data and the internally consistent set for total exit pressure, total exit temperature, exit velocity, and exit tangential velocity at 100-percent corrected speed are shown in Figures 4.6 - 4.10. This is shown in Figures 4.6 – 4.10, respectively. Figures 4.6 and 4.7 show that for the 100-percent speed line the exit total pressure and temperature have maximum differences of 6.35-percent and 2.30-percent, respectively. The maximum differences for the exit velocities are higher, in general, with a maximum difference in the exit absolute velocity of 6.44-percent, a maximum difference in the tangential velocity of 15.95-percent, and a maximum difference in the relative velocity of 14.67-percent (Figures 4.8, 4.9, and 4.10, respectively). Comparisons for exit static pressure and temperature for the 100-percent speed line, and exit total pressure, total temperature, static pressure, static temperature, absolute velocity, tangential velocity, and relative velocity for the 90-, 70-, and 50-percent speed lines are shown in Appendix D. These differences show that, for Rotor 1B, the mass-weighted averages of the variables are not internally consistent. From this point, any quantity listed as data for Rotor 1B will be the internally consistent 1-D representation of experimental data.

4.2.2 Internal Consistency Check Applied to Stage 35

Comparisons of mass-weighted averaged data and the internally consistent set for the 100-percent speed line for Stage 35 are shown in Figures 4.11 – 4.19. Figures 4.11 and 4.12 show that for the rotor, the exit total pressure and temperature have maximum differences of less than one-percent. The maximum difference in the exit absolute velocity, tangential velocity, and relative velocity are 1.59-percent, 0.65-percent, 2.49-percent, respectively (Figures 4.13 – 4.15). For stator 35, the exit total pressure and temperature and the exit velocity and tangential velocity all have maximum differences of less than one-percent (Figures 4.16 – 4.19). As with Rotor 1B, comparisons for exit total pressure, total temperature, static pressure, static temperature, absolute velocity, tangential velocity, and relative velocity for the 100-, 90-, and 70-percent speed lines of Stage 35 are shown in Appendix D. Although all the variables of the stator and the pressures and temperatures of the rotor are within one-percent of the consistent set of data, the velocities of the rotor are off by a more significant amount. Therefore, the mass-weighted averages of the variables for Stage 35 are considered not internally consistent. From this point, any quantity listed as data for Stage 35 will be the internally consistent 1-D representation of experimental data.

5.0 APPLICATION OF THE MLC WITH LOSS AND DEVIATION

SPECIFIED FROM DATA

The MLC was applied to each machine discussed in Section 4.1, with the results compared to the internally consistent 1-D representation of experimental data presented in the previous section. To run the MLC, certain input conditions (discussed in Chapter 3.0) were needed. The cross-sectional area correction factor (ς) is included in these requirements for crossing a bladed or non-bladed region. The calculation of ς is discussed below. Then, verification of the MLC is considered. Finally, the results using loss and deviation from data are presented.

5.1 Calculation of the Cross-sectional Area Correction Factor

As discussed in Section 3.1.3, the ratio of areas perpendicular to the flow is needed for crossing the bladed region, and as discussed in Section 3.2, the ratio of areas perpendicular to the axial velocity is needed for crossing the non-bladed region. The equations are rewritten here for convenience:

$$\left(\frac{A_1}{A_2} \right)_{\perp W} = \frac{\cos(\beta_1)}{\cos(\beta_2)} AVDR \text{ and} \quad [3.28]$$

$$\left(\frac{A_1}{A_2} \right)_{\perp V_t} = AVDR = \varsigma \frac{\pi (r_{tip1}^2 - r_{hub1}^2)}{\pi (r_{tip2}^2 - r_{hub2}^2)}. \quad [3.35]$$

Since AVDR has already been calculated in (Section 4.2), it could be used in calculating these area ratios. However, the MLC is set up to use the cross-sectional area correction factor and hub and tip radii for this calculation to give additional

boundary layer development insight. Solving equation [3.35] for the cross-sectional area correction factor yields:

$$\varsigma = \frac{AVDR}{\left(\frac{r_{tip1}^2 - r_{hub1}^2}{r_{tip2}^2 - r_{hub2}^2} \right)}, \quad [5.1]$$

where the inlet and exit hub and tip radii are known.

5.2 MLC Verification

To verify the accuracy of the MLC, a middle point on the 100-percent speed line for each blade row was run independently using its specific cross-sectional area correction factor and specified loss and deviation. For the blade rows, the specified loss coefficients are MWA's of the loss coefficients of the experimental data, and the specified deviations are the difference between the MWA exit flow angles and the mean metal angle. As shown in Tables 5.1, 5.2, and 5.3, the MLC produced results with almost zero-percent difference when compared to data. When Stage 35 was modeled collectively (rotor-space-stator) in this way, the difference was less than one-percent when compared with data (Table 5.4). Since the MLC was able to reproduce the overall performance of Rotor 1B and Stage 35 for each chosen operating point, it was assumed that the MLC was constructed correctly.

5.3 Discussion of Results of the MLC with Loss and Deviation Specified

After calculating the cross-sectional area correction factor for each point, an average cross-sectional area correction factor was calculated for each speed line of each blade row and non-bladed region. These averages were used in the MLC in

calculating the points on each respective speed line. Tables 5.5-5.8 show the values of AVDR, ζ , and mean radii used for each speed line of Rotor 1B, Rotor 35, NBR 35, and Stator 35, respectively. For each mass flow, loss and deviation was specified from data, as discussed in Section 5.2. The following is a discussion of the results using the MLC to model NASA Rotor 1B and NASA Stage 35 using a mean AVDR and loss and deviation specified from data.

5.3.1 NASA Rotor 1B

Figures 5.1 – 5.7 show the percent differences of the MLC results to data, using specified loss and deviation. The percent differences are calculated by the following equation:

$$\text{Percent Difference} = \frac{X_{\text{MLC}} - X_{\text{Data}}}{X_{\text{Data}}} * 100.0, \quad [5.2]$$

where X_{MLC} is a variable calculated by the MLC, and X_{Data} is a variable from the data set. As shown in these figures, the difference begins to grow large when the compression stage chokes. The experimental data indicates that choking occurs somewhere between the inlet and exit of the blade row. According to Wilson (1984), this is normally the case. For the 100-percent speed line, a much higher difference occurs than over the rest of the speed line. The same is true for the 90-, 70-, and 50-percent speed lines, but the differences are less pronounced at the lower speeds. The differences between the MLC results and data near choke is thought to be higher because the respective speed lines' average cross-sectional area correction factor (ζ) does not represent choke as well as it represents the rest of the

speed line. Neglecting the results near compressor choking, the maximum difference of exit total pressure and temperature are 3.1-percent and 0.9-percent, respectively. Similarly, for exit static pressure and temperature, neglecting the behavior near choke, the maximum differences are 3.0-percent and 0.9-percent, respectively. Similarly, the maximum differences in exit absolute velocity, tangential velocity, and relative velocity are 1.2-percent, 7.9-percent, and 7.2-percent, respectively.

5.3.2 NASA Stage 35

The percent differences, as calculated in equation [5.2], of the MLC results using specified loss and deviation for Stage 35 are shown in Figures 5.8 – 5.29. In this section, first the comparison of the MLC results to the data for Rotor 35 will be discussed, then the treatment of the non-bladed region (NBR Stator 35). Then, the comparison of the MLC results for Stator 35 to the data for will be discussed, and finally the comparison of the MLC to data for Stage 35 collectively (rotor-NBR-stator) will be discussed.

5.3.2.1 NASA Rotor 35

The percent differences for Rotor 35 are shown in Figures 5.8-5.14. As was the case in Rotor 1B, the percent differences at rotor choking are considerably higher than at the other mass flows. As was the case for Rotor 1B, experimental data indicates that choking occurs somewhere between the inlet and exit of the blade row. Again, the difference between MLC results and data near choke is thought to be higher because the respective speed lines' average cross-sectional

area correction factor does not represent choke as well as it represents the rest of the speed line. Neglecting choke for Rotor 35, the maximum differences between MLC results and data for exit total pressure and temperature, and exit static pressure and temperature, are 2.5-percent, 0.7-percent, 2.6-percent, and 0.8-percent, respectively. Also, neglecting the near-choke points, the maximum differences for the exit absolute velocity, tangential velocity, and relative velocity are 1.2-percent, 7.2-percent, and 5.6-percent, respectively.

5.3.2.2 Non-bladed Region of Stage 35

Section 3.2 discussed the algorithm for calculating flow through a non-bladed region. Since no loss data was available for the NBR, the total pressure loss coefficient was calculated using the exit total and static pressures from the data of Rotor 35 and the inlet total pressure from the data of Stator 35 (equation [3.48]). The effectiveness of accounting for the conservation of angular momentum between blade rows with a change in annular geometry will now be discussed.

Figures 15-17 show MLC results both with the enforcement conservation of angular momentum and without the enforcement of the conservation of angular momentum compared to data. As shown in Figures 15-17, accounting for the conservation of angular momentum across the NBR provides definite improvement in predictive capability, especially for the axial and tangential velocities. It improves the accuracy of the calculated stator inlet variables up to 5.3-percent, when compared to data.

5.3.2.3 NASA Stator 35

The percent differences between MLC results and data for Stator 35 performance variables are shown in Figures 5.18-5.23. If stage choking is included, the differences in Stator 35 are less than 3.4-percent. If choke is neglected, the differences between MLC results and data for the exit pressures and temperatures are less than 0.2-percent, and the differences for the exit velocities are less than 1.2-percent.

5.3.2.4 Stage 35 Collectively (Rotor-NBR-Stator)

After the rotor and stator of Stage 35 were modeled separately, the entire stage was modeled collectively, including calculations of the space between blade rows. The percent differences between MLC results and data for Stage 35 performance variables are shown in Figures 5.24-5.29. Again, the percent differences at choking conditions are generally worse than at the lower mass flows. Neglecting choke, the maximum difference of exit total pressure and temperature are 2.4-percent, 0.8-percent, respectively, and the maximum difference of exit static pressure and temperature are 4.2-percent, and 1.3-percent, respectively. Also, neglecting choke, the differences for the exit velocities are less than 6.0-percent.

6.0 APPLICATION OF THE MLC USING LOSS AND DEVIATION CORRELATIONS

Now that the MLC has been verified and the cross-sectional area correction factors have been calibrated using specified 1-D representations of loss and deviation, the next step is to make the MLC predictive by including loss and deviation correlations. A summary of the loss and deviation correlations used are discussed below. For a more detailed description of the correlations, see Klepper (1998). Once these correlations were included in the MLC, it was applied to NASA Rotor 1B and NASA Stage 35. A discussion of the results can be found in Section 6.3.

6.1 Discussion of Loss and Deviation Correlations

Loss and deviation correlations from open literature were included in the MLC. These correlations were collected by Hearsey (1994) to get a transonic set, using databases of machines designed in the 1950's and 1960's. These correlations were developed from both cascade and machine data from situations where NACA 65 series and double-circular-arc blade profile designs were used.

To work, these correlations must be given certain blade geometry information. The required quantities are: solidity, camber, location of maximum camber point as a fraction of chord, blade maximum thickness to chord ratio, and the inlet and exit blade metal angles. Once these quantities are known, the low-speed reference minimum-loss incidence angle (i_{ref}), and the reference minimum-

loss air inlet angle (β_{1ref}) can be calculated using equations found in the NASA SP-36 (Lieblein, 1965). i_{ref} is given by the first equation:

$$i_{ref} = (k_i)_{shape} (k_i)_{th} (i_o)_{10} - n\theta , \quad [6.1]$$

where $(k_i)_{shape}$ and $(k_i)_{th}$ are correction factors for blades that are not of the NACA 65-(A₁₀)-series, $(i_o)_{10}$ is the variation of zero-camber incidence angle for the 10-percent-thick 65-series thickness distribution and is a function of inlet air angle β_1 , n is the minimum-loss-incidence slope factor, and θ is the camber angle. β_{1ref} is given by:

$$\beta_{1ref} = \beta'_1 + i_{ref} , \quad [6.2]$$

where β'_1 is the inlet metal angle.

6.1.1 Discussion of Loss Correlations

The loss correlations are used to calculate the relative total pressure loss coefficient, ϖ_r , which is defined by equation [3.18]. Loss is calculated at a given compressor operating point by developing a “loss bucket”, which displays the total pressure loss coefficient as a function of incidence at a given Mach number. This curve usually has a U or bucket-like shape, with the middle bottom of the curve being the minimum-loss incidence point. The following equation is used to develop this curve:

$$\varpi_r = (\varpi_{min} + \varpi_M) \left[1 + \left(\frac{i - i_M}{W} \right)^2 \right] , \quad [6.3]$$

where $\varpi_{\min}$ is the loss caused by viscous forces on the blade, and ϖ_M is the loss coefficient associated with shock. Also, i is the actual incidence, i_M is the minimum loss incidence, and W is the arbitrarily defined width of the loss bucket. Figure 6.1 shows a loss bucket and how each variable effects it. An increase in the loss coefficients causes the bucket to shift up. An increase in the minimum loss incidence causes the bucket to shift to the right. A decrease in the width term causes the bucket width to decrease.

The equation for the loss coefficient associated with viscous forces on the blade is developed by Robbins et. al. (1965):

$$\varpi_{\min} = \varpi_{\text{vloss}} \left(\frac{2P\sigma}{\cos\beta_{2\text{ref}}} \right), \quad [6.4]$$

where ϖ_{vloss} is an input loss modification term and P is a term interpolated from Figure 203 (b) of Robbins, et. al (1965).

The last loss coefficient is the one associated with a strong shock in the passage. If the inlet Mach number is larger than the critical Mach number, there will be supersonic flow somewhere in the blade passage. Figure 6.2 shows a schematic of the shock loss model. If the inlet Mach number is larger than the critical one, a detached bow shock will appear somewhere in the blade passage. For simplicity, the shock will be treated as a normal shock. At point A in Figure 6.2, the Mach number is equal to the inlet Mach number. At point B, the Mach number is determined using a Prandtl-Meyer expansion from the blade tip to the point of shock impingement. Double-circular-arc (DCA) sections are assumed for

approximating the expansion angle. Blade spacing affects the point of shock impingement, so solidity is very important. The following (Miller, Lewis, and Hartman, 1961) is the equation for the shock loss coefficient:

$$\omega_M = \frac{\left(\frac{(\gamma + 1)M_s^2}{(\gamma - 1)M_s^2 + 2} \right)^{\frac{\gamma}{\gamma - 1}} \left(\frac{\gamma + 1}{2\gamma M_s^2 - (\gamma - 1)} \right)^{\frac{1}{\gamma - 1}} - 1}{\left(1 + \frac{(\gamma - 1)}{2} M_{1r}^2 \right)^{\frac{\gamma}{\gamma - 1}} - 1}, \quad [6.5]$$

where M_s is the average of the Mach numbers at point A and point B in Figure 6.2 and M_{1r} is the inlet relative Mach number.

6.1.2 Discussion of Deviation Correlations

Deviation is defined as the difference between the exit flow angle and the exit blade metal angle. A schematic is shown in Figure 6.3. This deviation can be caused by thick boundary layers or by the separation of boundary layers. There are several different reasons why this occurs. The following equation (Lieblein, 1965) represents the total deviation as the linear sum of five reasons for deviation:

$$\delta = \delta_M + \delta_i + \delta_{ref} + \delta_{v_x} + \delta_{3D}, \quad [6.6]$$

where δ_M is the deviation due to Mach number greater than one in the blade passage, δ_i is the deviation due to off-design incidence, δ_{ref} is the low-speed reference minimum loss deviation angle, δ_{v_x} is the deviation associated with axial velocity, and δ_{3D} is the deviation associated with streamline radial location.

The deviation associated with a shock in the blade passage is only used if the inlet relative Mach number is greater than one. It is developed by Jansen and Moffat (1967) as

$$\delta_M = \beta_{2M} - \beta_{2ref}, \quad [6.7]$$

where

$$\beta_{2M} = \arccos \left\{ \cos \beta_{1ref} \frac{A_1}{A_2} \frac{V_{x1}}{V_{x2}} \left[\frac{1 + \frac{\gamma-1}{2} M_{2r}^2}{1 + \frac{\gamma-1}{2} M_{1r}^2} \right]^{\frac{1}{\gamma-1}} \right\}. \quad [6.8]$$

In this equation, A_1 and A_2 are the inlet and exit areas, respectively, V_{x1} and V_{x2} are the inlet and exit velocities, respectively, and M_{1r} and M_{2r} are the inlet and exit relative Mach numbers, respectively.

δ_i is the deviation due to off-design incidence. The equations for this deviation component were developed using existing cascade data (Horlock, 1958).

There are four equations, based on the actual incidence:

$$\delta_i = \frac{1}{2} \frac{\partial \delta}{\partial i} (i_{choke} - i_{ref}), \quad i \leq i_{choke} \quad [6.9]$$

$$\delta_i = \frac{\partial \delta}{\partial i} \frac{(i - i_{ref})(i + i_{ref} - 2i_{choke})}{2(i_{ref} - i_{choke})}, \quad i_{choke} < i \leq i_{ref} \quad [6.10]$$

$$\delta_i = (i - i_{ref}) \left[1 + \frac{\left(1 - \frac{\partial \delta}{\partial i}\right)(i + i_{ref} - 2i_{stall})}{2(i_{stall} - i_{ref})} \right], \quad i_{ref} < i \leq i_{stall} \quad [6.11]$$

and

$$\delta_i = i - i_{\text{stall}} + \frac{1}{2} \left(\frac{\partial \delta}{\partial i} + 1 \right) (i_{\text{stall}} - i_{\text{ref}}), \quad i_{\text{ref}} < i \leq i_{\text{stall}} \quad [6.12]$$

where i_{stall} is the stalling incidence, i_{choke} is the choking incidence, and $\frac{\partial \delta}{\partial i}$ is obtained from Figure 177 of Robbins et. al (1965) as a function of solidity and inlet air angle.

The low-speed reference minimum loss deviation angle is given by:

$$\delta_{\text{ref}} = (k_{\delta})_{\text{shape}} (k_{\delta})_{\text{th}} (\delta_o)_{10} + \theta \frac{m + \frac{1}{2} \left(\frac{a}{c} - \frac{1}{2} \right)}{\sigma^b}, \quad [6.13]$$

where $(k_{\delta})_{\text{shape}}$ and $(k_{\delta})_{\text{th}}$ are correction factors for blades that are not of the NACA 65-(A₁₀)-series, $(\delta_o)_{10}$ is the variation of zero-camber incidence angle for the 10-percent-thick 65-series thickness distribution and is a function of inlet air angle β_1 , σ is the solidity, b is the solidity exponent, and m is the deviation slope factor. This equation for δ_{ref} , presented by Carter, (1946) is a variation of the original one presented by Lieblein (1965). The modification makes δ_{ref} a function of the point of maximum camber as a fraction of chord $\left(\frac{a}{c} \right)$.

The deviation correlation based on axial velocity ratio is presented by Horlock (1967-68). Because of end-wall boundary layer growth, the pressure increase is not as great as design, therefore the axial velocity ratio does not equal design. Horlock states that when the axial velocity ratio is greater than one, the

deviation actually decreases. The following equation is used to calculate the corresponding deviation component:

$$\delta_{v_x} = 10 \left(1 - \frac{V_{m2}}{V_{m1}} \right), \quad [6.14]$$

where V_{m1} and V_{m2} are the meridional velocities at the blade inlet and exit, respectively.

The final deviation component is based on radial location. Figure 202 (b) of Robbins et. al (1965) gives this deviation for high-speed sections or blades with DCA profiles. For blades with NACA 65-(A₁₀) series sections, the deviation for low-speed sections is equal to –0.5 degrees for heights greater than 50-percent of blade height from hub and is defined by the following equation for all other radial locations

$$\delta_{3D} = -\frac{1}{2} + \frac{75}{8} \left(\frac{1}{2} - \frac{r - r_{hub}}{r_{tip} - r_{hub}} \right)^2. \quad [6.15]$$

6.2 Calibration Technique

The correlations used in the MLC were calibrated for NASA Rotor 1B and each blade row of NASA Stage 35 separately. This was done by adding (or subtracting) loss and deviation until they matched the respective loss and deviation specified by data for a middle point on each speed line. These add-loss and add-deviation corrections were then used for each point on their respective speed lines. A comparison of the data and MLC Results with loss and deviation calculated by correlations will now be discussed.

6.3 Discussion of Results of the MLC using Loss and Deviation Correlations

Calculations were performed by using the MLC with the loss and deviation correlations discussed in Section 6.1. Tables 6.1, 6.2, and 6.3 show the add-loss and add-deviation corrections required to get the MLC with loss and deviation correlations to match the loss and deviation specified from data for a middle point on each speed line of Rotor 1B, Rotor 35, and Stator 35, respectively. Performance maps were constructed showing the pressure and temperature characteristics predicted by the MLC and the pressure and temperature characteristics from data.

6.3.1 NASA Rotor 1B

The pressure and temperature characteristics maps for Rotor 1B are shown in Figures 6.4 and 6.5, respectively. In the normal operating region, the pressure characteristics were predicted well. The lower speed lines were predicted better than the higher speed lines, with the differences between predictions and data of the lower speed lines within 2.5-percent and the differences of the higher speed lines within 4.9-percent. The pressure ratio characteristics were under-predicted for lower mass flows and over-predicted for higher mass flows of each speed line. The differences between MLC results and data are much higher for choked conditions than for normal operating conditions, with the MLC solution over-predicting the data. A choked condition is caused by poor flow quality in which there can be large negative incidence, which can be severe enough to cause flow separation on the pressure surface of the blade. This results in an increase in loss. Figures 6.6 and 6.7 show a comparison of the predicted total pressure loss coefficients and deviation

angles and the data. Figure 6.8 shows an average, or mean AVDR for each speed line compared to the specific AVDRs calculated for each mass flow on that speed line. The method for obtaining these specific AVDRs is discussed in Section 4.2. Figure 6.6 shows that the loss was often under-predicted for Rotor 1B. The percent differences of the predicted loss and the data were generally higher at choke and stall. The MLC's under-prediction of loss at choke is due to a combination of reasons: (1) the average cross-sectional area correction factor (ζ) is not representing the choked condition as well as the normal operating range, (2) the correlations are incapable of predicting the large negative incidence at choke, and finally, (3) the correlations assume the blades to have double-circular-arc blade shape. This is a problem because the blades of Rotor 1B have a multiple-circular-arc blade shape for 40-percent of their span. These reasons contribute to the large differences in predicted deviation and data, as shown in Figure 6.7. The deviation angles from data are much less inclined than the deviations calculated by the correlations. The over-sensitivity of the deviation correlations indicate they are better suited to low speeds since low rotor speeds have steeper slopes of deviation angles. Therefore, the deviation correlations do not predict high speed line deviation angles well. As shown in Figure 6.8, the mean AVDR is less than the AVDRs near stall and greater than the ones near choke for the lower speed lines, and the reverse is true for the higher speed lines. In both cases, the mean AVDR generally represents the normal operating region better than it does choked and stalled regions, which is a factor in

the fact that MLC predictions are better for predicting normal compressor operating regions than for compressors operated near choked and stalled flow regions.

The temperature characteristic map for Rotor 1B is shown in Figure 6.5. The temperature characteristics were predicted well in the normal operating region. As was the case for the pressure characteristics, the lower speed lines were predicted better than the higher speed lines, with the difference of the lower speed lines within 0.7-percent and the difference of the higher speed lines within 1.2-percent. As with the total pressure characteristics, the MLC under-predicted at lower mass flows and over-predicted at higher mass flows. Again, the differences between MLC results and data are higher for choked conditions than for the normal operating regions, with the predicted temperature being higher.

A discussion of the MLC results for the Rotor 1B exit total pressure and temperature, exit relative total pressure, exit static pressure and temperature, and exit absolute, tangential and relative velocities can be seen in Appendix F, Section F.1. The following is a summary of the results in Appendix F.1. The MLC did well in the normal operating condition, especially for total pressure and temperature, which are the quantities needed to obtain the required compressor characteristics. The greatest percent differences between predicted results and data were in the tangential velocities. This indicates that the MLC has difficulty modeling the turning of the flow. For the normal operating range, the exit relative total pressures calculated by the MLC were within 1.7-percent of data. The MLC generally predicted higher exit relative total pressure values than the values from data. This

indicates that the predicted losses calculated by the MLC were lower than the data for the normal operating condition. In general, the stalled and choked regions had higher percent differences than the normal operating regions.

6.1.2 NASA Stage 35

The pressure and temperature characteristic maps for Stage 35 are shown in Figures 6.9 and 6.10, respectively. In the normal operating region, the pressure characteristics were predicted well. The differences of the 100-percent speed line are within 1.7-percent, and the differences of the remaining speed lines are within 0.6-percent, with the MLC under-predicting at lower mass flows and, generally, over-predicting at higher mass flows. The choked regions are not predicted as well as the normal operating conditions, but the differences in this region are still within 2.73-percent. Figures 6.11-6.16 show comparisons of predicted total pressure loss coefficients and deviation angles and comparisons of the mean AVDR to AVDRs of each mass flow for the rotor and stator of Stage 35, respectively. As shown in Figures 6.11 and 6.14, the losses were predicted well for the normal operating region of Stage 35. Where the loss was missed for Stage 35, it was generally over-predicted, where for Rotor 1B it was generally under-predicted. The difference between the assumed blade shape used in the MLC and the actual blade shape of the compressor is more significant for the Stage 35 case. The blades of Rotor 1B have a multiple-circular-arc blade shape for 40-percent of their span, but the entire blade profiles for the blades of Stage 35 are multiple-circular-arc. Large differences in the predicted deviation angle and the data for Rotor 35 can be seen in Figure

6.12. As was the case for Rotor 1B, the deviation correlations are overly-sensitive. In general, differences in the stalled regions are higher than the percent differences for the normal operating region. In a stalled condition, there is an increase in loss, which occurs when there is flow separation on the suction surface of the blade. This flow separation occurs when there is poor inlet flow quality caused by a large positive incidence angle. The correlations do not accurately predict this large positive incidence. As with Rotor 1B, the mean AVDR generally represents the normal operating region better than it does choked and stalled regions, as shown in Figures 6.13 and 6.16. However, the mean AVDR seems to represent choked regions better than it does stalled regions.

The temperature characteristic map for Stage 35 is shown in Figure 6.10. The temperature characteristics are predicted well. In the normal operating region, differences are less than 1.0-percent. The differences between MLC results and data are higher for choked and stalled conditions than for the normal operating regions.

A discussion of the MLC results for the exit total pressure and temperature, exit relative total pressure, exit static pressure and temperature, and exit absolute, tangential and relative velocities for Rotor 35, Stator 35, and the combined Stage 35 can be seen in Appendix F, Section F.2. . The following is a summary of the results in Appendix F.2. In general, Stage 35 was predicted well, with a difference of predicted results and data within 4.4-percent, for the normal operating range. The quantities needed to calculate the compressor characteristics, exit total pressure

and temperature, were modeled nicely, with differences within 1.7-percent for the normal operating range. The tangential velocities were over-predicted for lower mass flows and under-predicted for higher mass flows. In general, the stalled and choked regions had higher percent differences between predicted results and data than the normal operating regions.

7.0 SUMMARY AND CONCLUSIONS

This chapter summarizes the results and conclusions of the research performed in this thesis. First, the results from the investigations of NASA Rotor 1B and NASA Stage 35 using the MLC are presented. Then, conclusions are drawn based on these results.

7.1 Summary of Results

Several points were learned from this research. The question of the internal consistency of the 1-D MWA representation of the consistent 2-D data set was explored. The MLC was applied to Rotor 1B and Stage 35 with loss and deviation specified from data. Finally, the MLC was applied to Rotor 1B and Stage 35 with loss and deviation correlations included.

After an internal consistency check was performed on the 1-D MWA representation of the consistent 2-D data of Rotor 1B and Stage 35, it was determined that, in general, 1-D representations of 2-D flows may not be internally consistent. A consistent set of data was obtained using selected mass-weighted-averaged variables and the fundamental flow properties. Results obtained by the MLC were compared to this set.

Results from the MLC accounting for the conservation of angular momentum between blade rows with a change in annular geometry were compared with MLC results that did not account for this. It was determined that accounting for the conservation of angular momentum between blade rows with a change in annular geometry improves the accuracy of the inlet variables to the stator.

The MLC with loss and deviation specified from data produced favorable results at normal operating conditions for both Rotor 1B and Stage 35. With the calibration of the MLC using a middle point on each speed line, the MLC with loss and deviation correlations also produced favorable results at normal operating conditions for both Rotor 1B and Stage 35.

The MLC experienced limitations at stall, especially at low speeds. A stalled condition occurs when there is poor flow quality in which there is a large positive incidence, which causes a flow separation on the suction surface of the blade. This causes an increase in loss and deviation. The correlations do not accurately predict the performance of cascade profiles at this condition. The correlations are limited in that they are based on blades having double-circular-arc blade shapes and the machines modeled in the present thesis have either combination double-circular-arc / multiple-circular-arc blade profiles or completely multiple-circular-arc blade profiles. Another limitation is that for Rotor 1B, the correlations do not calculate accurate shock losses. The correlations are applied at the mean flow radius with no knowledge of the strong shocks at the blade tips or severe flow turning near the hub. Therefore, the correlations are estimating losses, which do not adequately represent the entire annular cross section of the compressor for a given blade. Also the deviation correlations were written for low speed machines, so the slopes of the deviation curves were steeper than the data.

The MLC also experienced limitations at and near choked conditions. A choked condition is caused by poor flow quality in which there is a large negative incidence, which causes a flow separation on the pressure surface of the blade, causing an increase in loss. One reason for the MLC's limitation at choke is that the average cross-sectional area correction factor (ζ) cannot represent the boundary layer thickening that occurs near the choked condition as well as over the rest of the speed line. Another is the limitation of the loss correlations at choke, due to the previously stated differences in the blade profile assumed by the correlations and the actual blade profile of the machines modeled.

7.2 Conclusions

Several conclusions were made as a result of this study. These conclusions are listed below.

- Mass weighted averages of the multi-dimensional data in may not be internally consistent, therefore the generation of a consistent set of 1-D data for the machines was required.
- For Rotor 1B, when loss and deviation are specified from data, the MLC calculates exit total pressure within 2.1-percent difference with respect to data and exit total temperature within 0.6-percent difference with respect to data for the normal operating range.
- For Stage 35, when loss and deviation are specified from data, the MLC calculates exit total pressure within 1.4-percent difference with respect to data

and exit total temperature within 0.6-percent difference with respect to data for the normal operating range.

- For Stage 35, accounting for the conservation of angular momentum between blade rows with a change in annular geometry improves the accuracy of the inlet variables to the stator up to 5.3-percent
- For Rotor 1B, when loss and deviation are calculated using calibrated correlations, the MLC calculates total pressure characteristics within 4.9-percent difference with respect to data and total temperature characteristics within 1.2-percent difference with respect to data for the normal operating range.
- For Stage 35, when loss and deviation are calculated using calibrated correlations, the MLC calculates total pressure characteristics within 1.7-percent difference with respect to data and exit total temperature within 0.6-percent difference with respect to data for the normal operating range.
- There are clear limitations in the MLC approach to compressor performance modeling when a compressor operated near stalled and choked conditions.
- A mean line analysis is not a good method to use for high-aspect ratio compressor fans.
- The curves of the predicted deviation angles have a steeper slope than the data.
- With the calculation of cross-sectional area correction factors and the calibration of the MLC using total pressure loss coefficients and deviation

angles from data, the MLC can predict flow field quantities accurately for the normal operating regions.

8.0 RECOMMENDATIONS

Although the MLC was successful in calculating compressor stage flow parameters in the normal operating range, modifications could be employed to improve its performance at choked and stalled conditions. Recommendations for future work will now be discussed.

1. Use the MLC to model modern military Fan or core compressor and compare the results to experimental data. It should be possible to see how trends develop for similar machines.
2. In this study, the loss and deviation correlations were calibrated at a middle point for each speed line. An improvement upon this is to calibrate three points per speed line: one at stall, one at the normal operating range, and one at choke.
3. Perform research to obtain or develop correlations for other blade profiles, such as multiple-circular-arc blade profiles. A switch in the input file could indicate whether to use double-circular-arc or multiple-circular-arc blade profiles.
4. A recommendation similar to the one just stated is to perform research to obtain or develop correlations that are sensitive to combination blade profile shapes.
5. Perform research to obtain or develop correlations specifically designed to model choked conditions and stalled conditions, respectively. A switch in the input file could indicate whether the point being modeled

is closer to stall, choke, or the normal operating range. Possible models to be employed are Bloch's (1996) shock-loss model, which is investigated in Cahill (1997), and Dick Hersey's (1994) stall model.

6. Perform research to obtain or develop correlations to include losses at the hub and tip. Work would be required to determine how to include hub and tip losses into the 1-D stage loss value.
7. Perform research to obtain or develop high speed deviation correlations
8. Perform research to obtain or develop AVDR correlations

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APPENDICES

APPENDIX A

FIGURES

Known: $P_{t1}, T_{t1}, M_1, \alpha_1, \gamma, R, U_1, U_2, \dot{m}, A_1, A_2$, Blade Shape

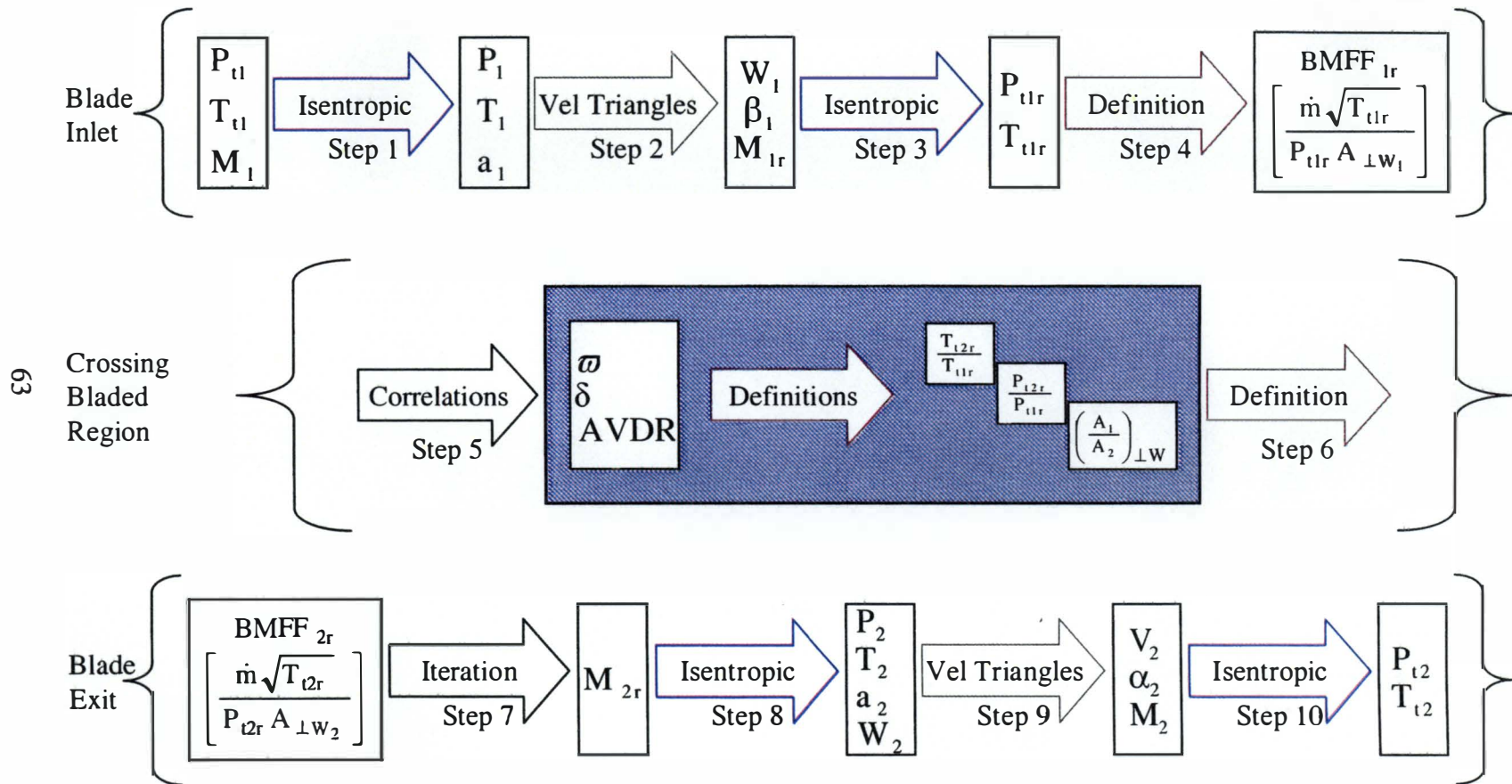


Figure 3.1. Algorithm for Calculating Flow Across a Bladed Region

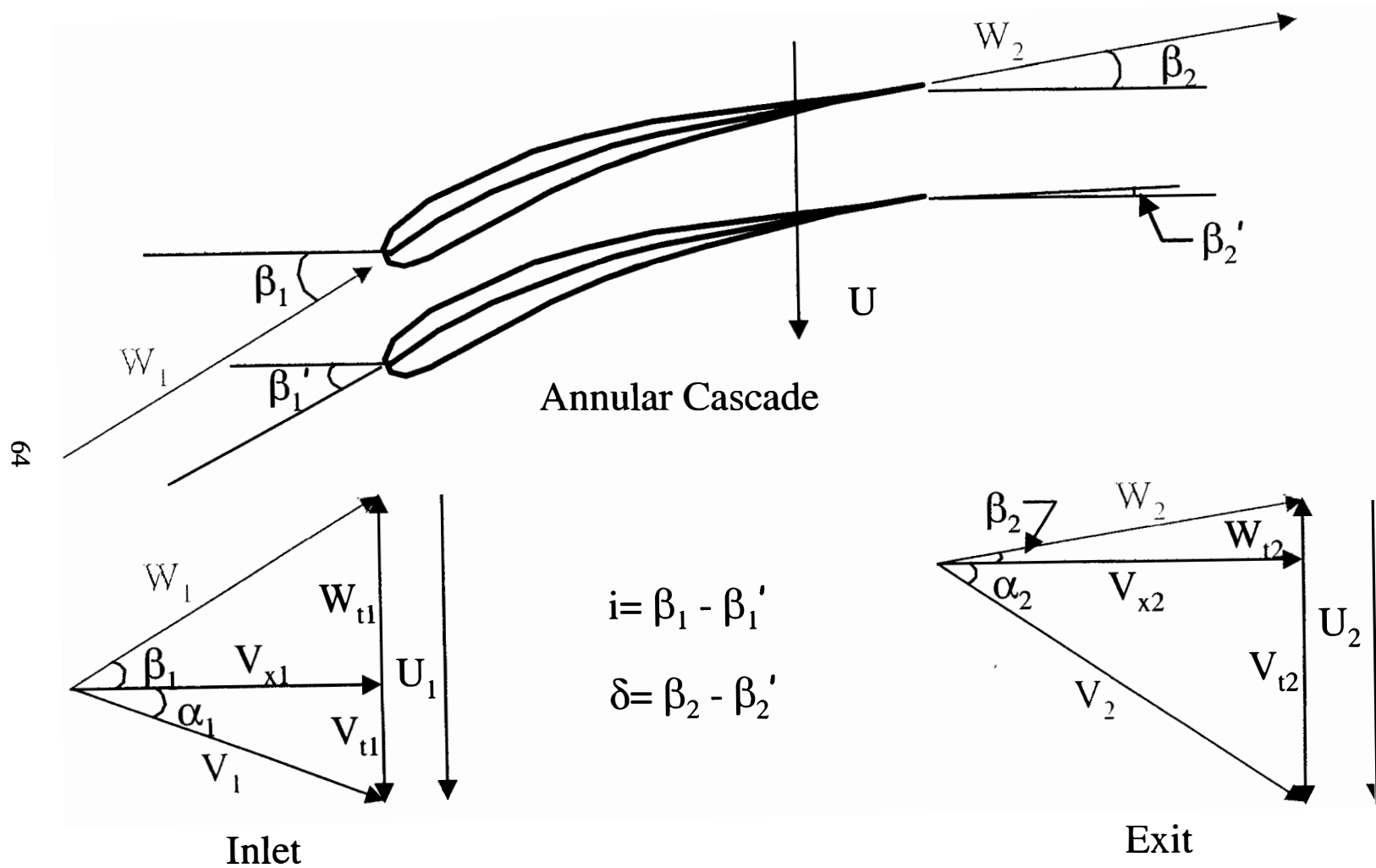


Figure 3.2. Velocity Triangle Nomenclature Used in the MLC

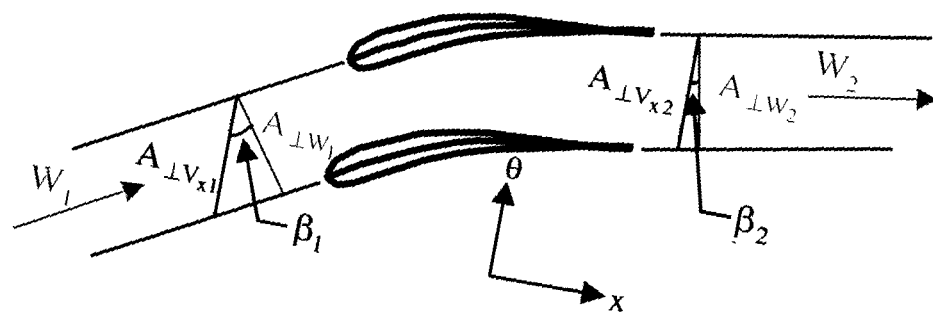


Figure 3.3. Schematic of a Mean Area Cascade

Known: $P_{t1}, T_{t1}, M_1, \alpha_1, \gamma, R, V_{t1}, \dot{m}, A_1, A_2$

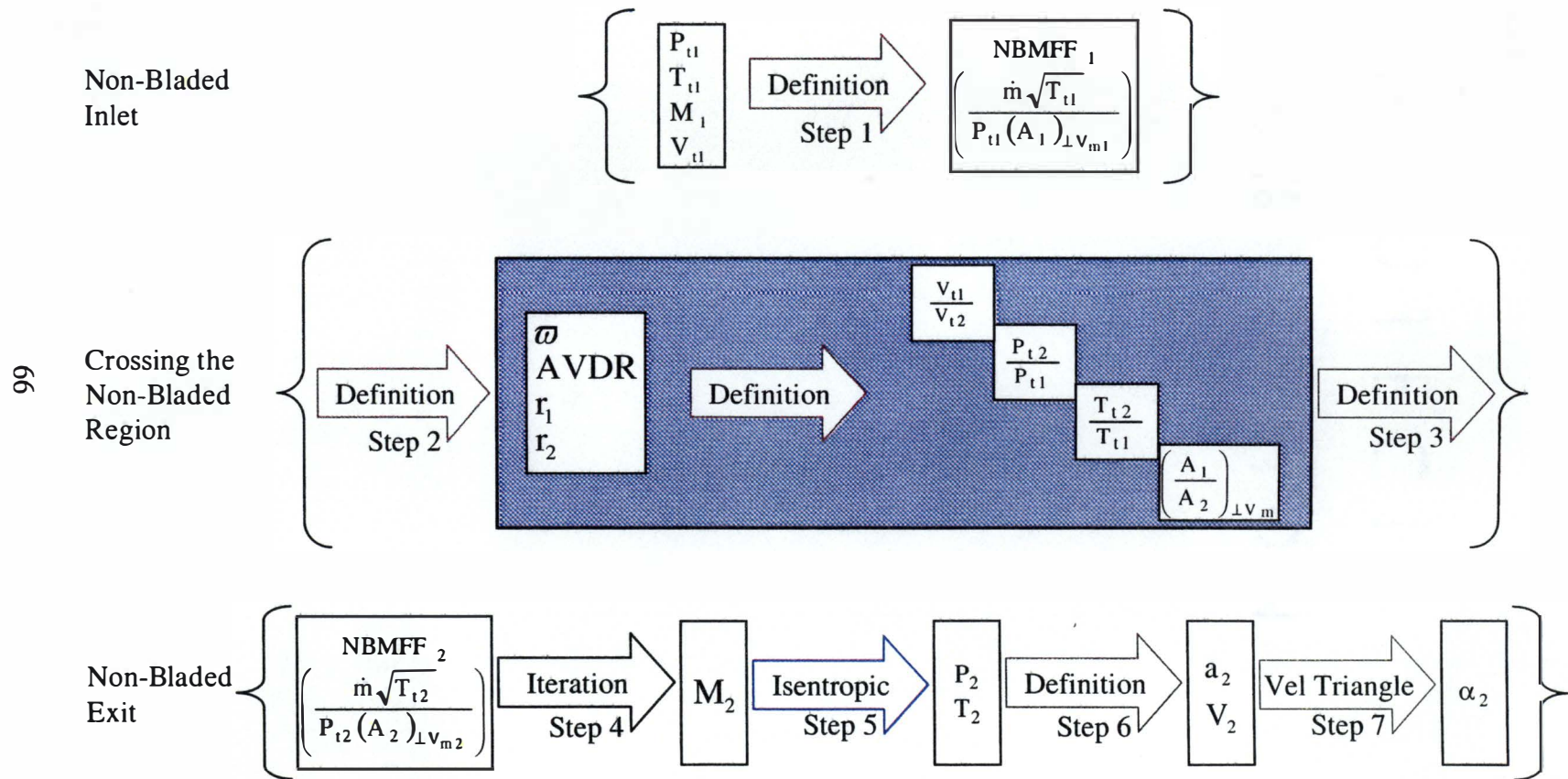


Figure 3.4. Algorithm for Calculating Flow Across a Non-bladed Region

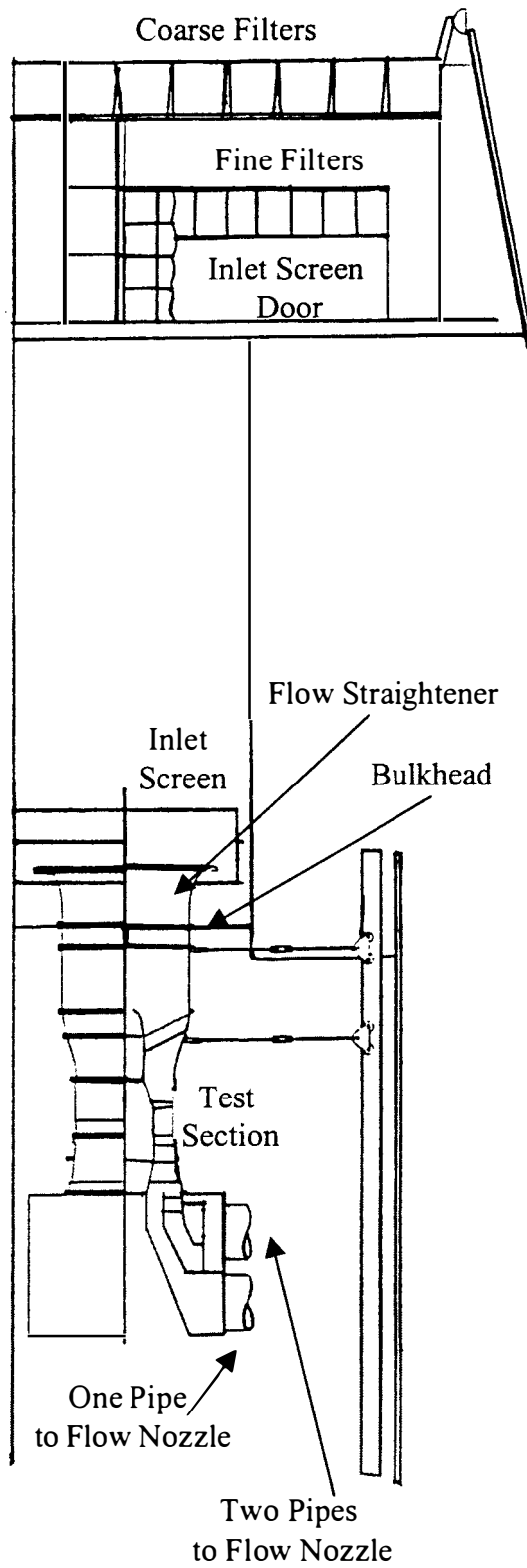


Figure 4.1. Schematic of NASA Rotor 1B Test Setup. (Seyler and Goselow, 1967)

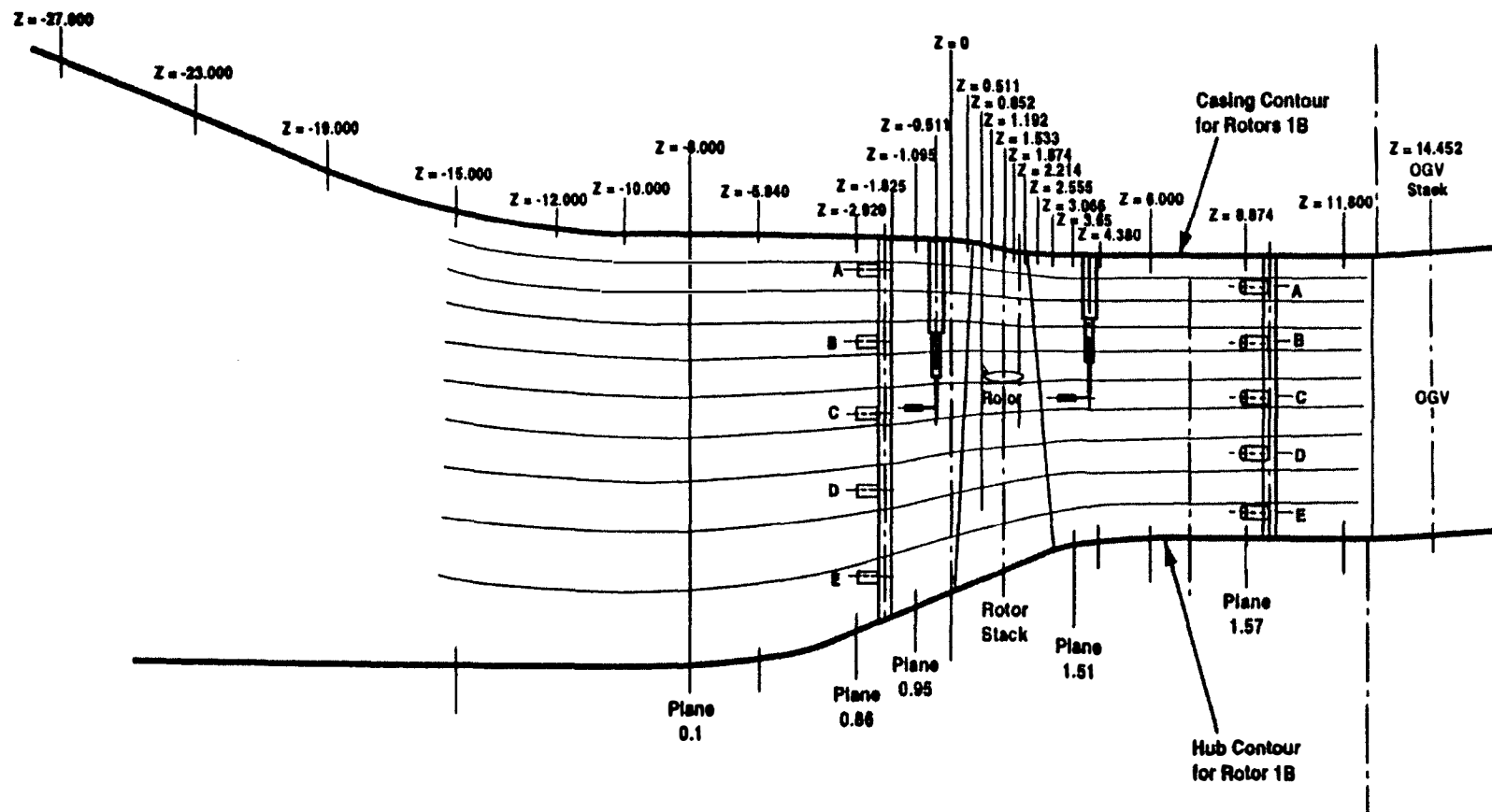


Figure 4.2. Schematic of NASA Rotor 1B Instrumentation Setup. (Seyler and Goselow, 1967)

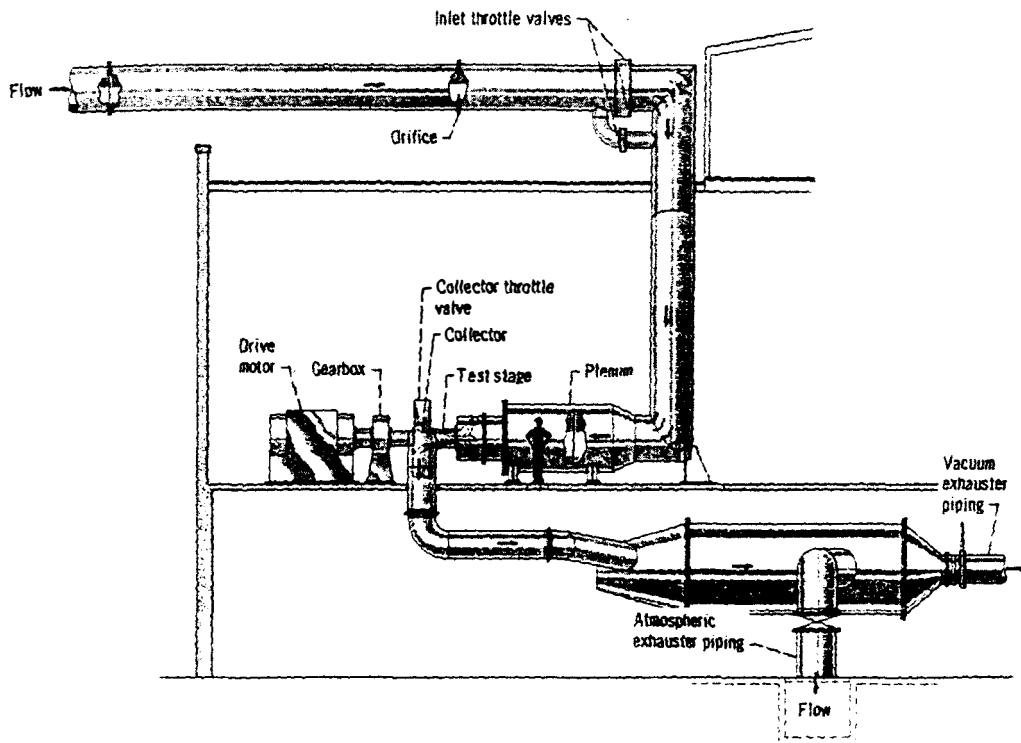


Figure 4.3. Schematic of NASA Stage 35 Test Facility.
(Reid and Moore, 1978)

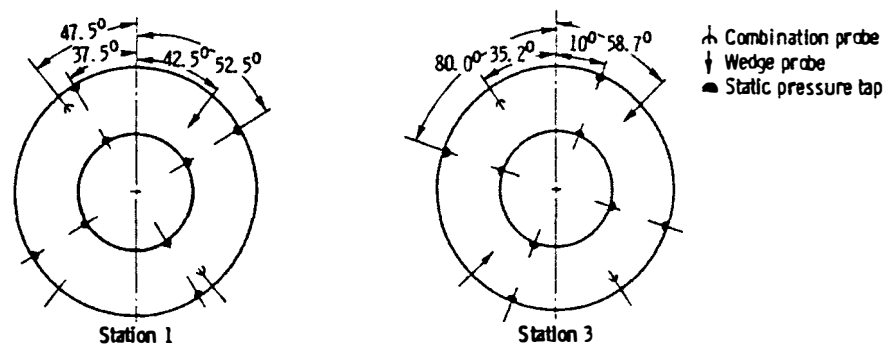


Figure 4.4. Schematic of NASA Stage 35 Instrumentation Layout.
(Reid and Moore, 1978)

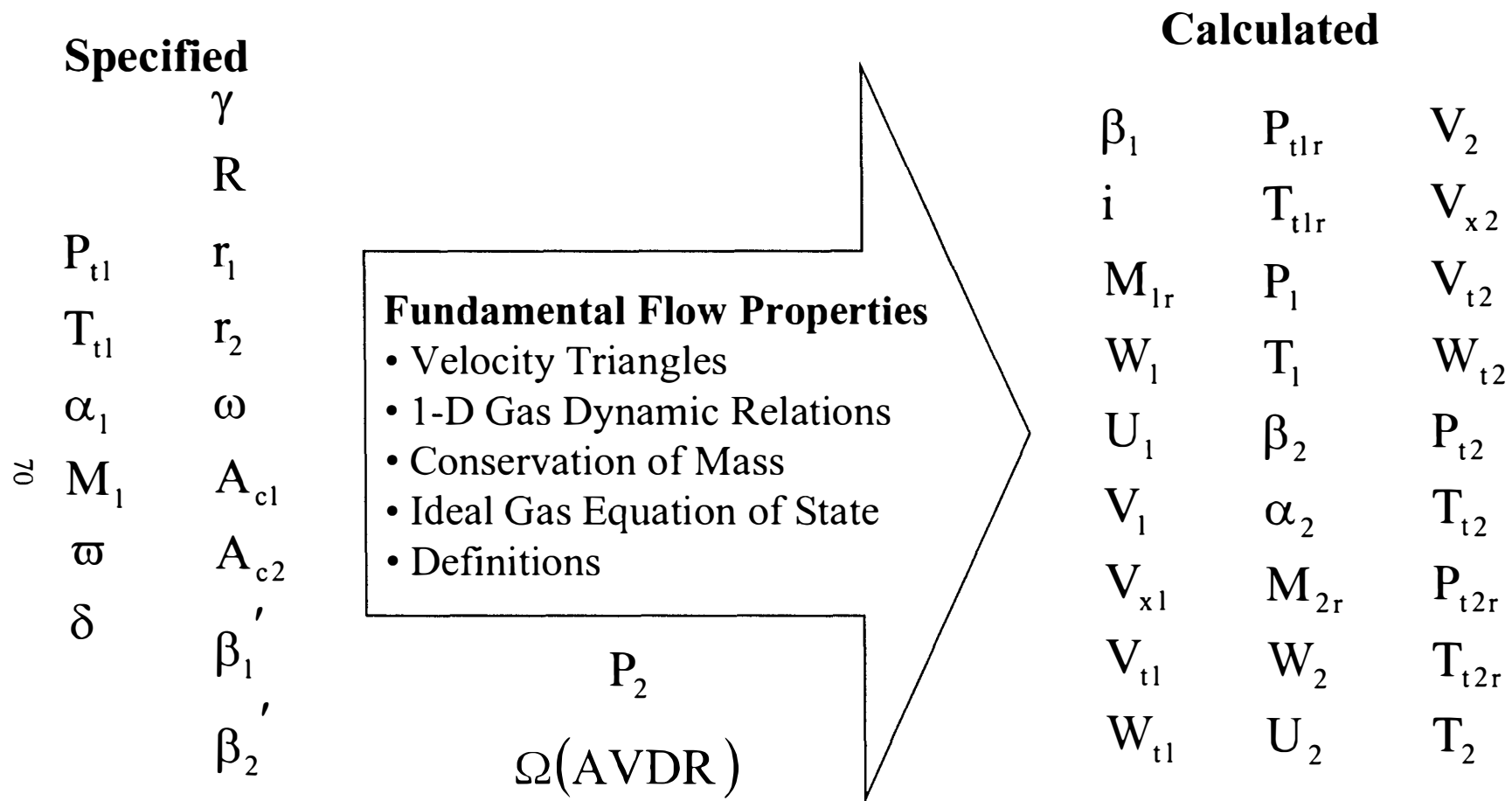


Figure 4.5. Schematic for Internal Consistency Check

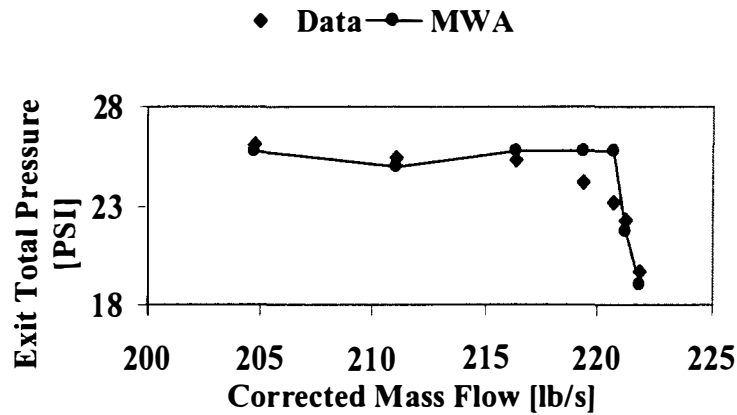


Figure 4.6. Comparison of MWA Data and Internally Consistent Data for Exit Total Pressure at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 1B

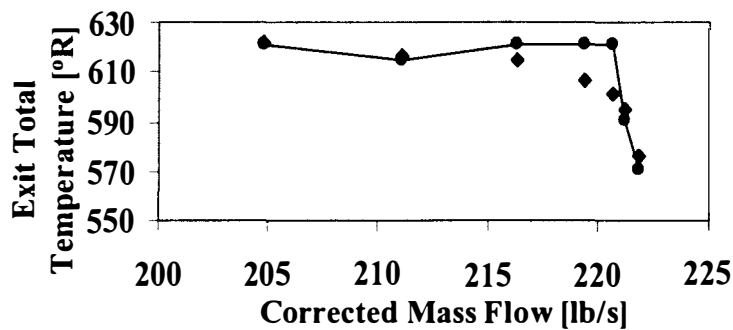


Figure 4.7. Comparison of MWA Data and Internally Consistent Data for Exit Total Temperature at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 1B

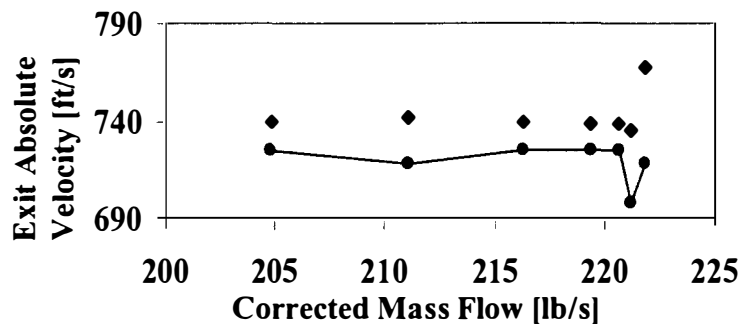


Figure 4.8. Comparison of MWA Data and Internally Consistent Data for Exit Absolute Velocity at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 1B

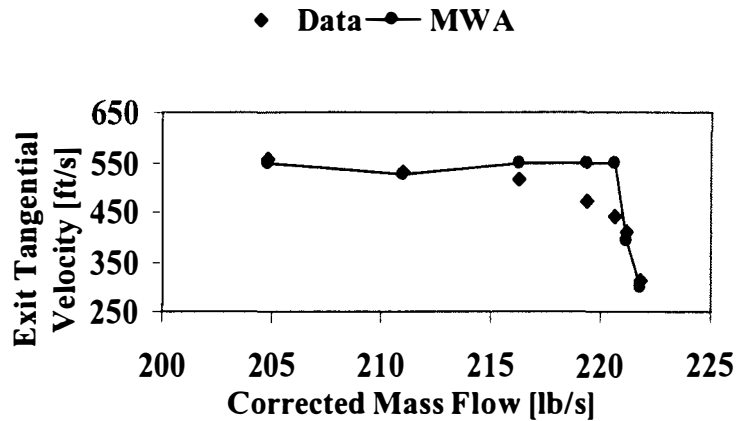


Figure 4.9. Comparison of MWA Data and Internally Consistent Data for Exit Tangential Velocity at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 1B

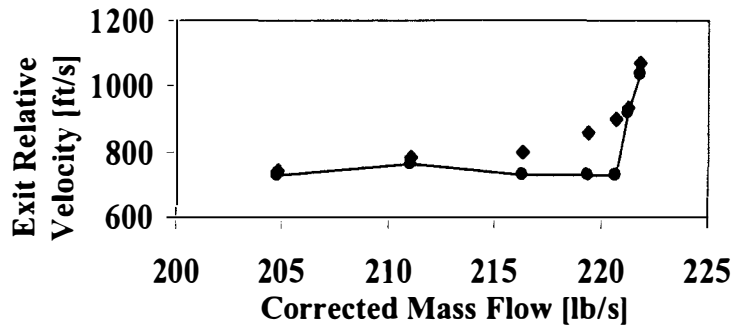


Figure 4.10. Comparison of MWA Data and Internally Consistent Data for Exit Relative Velocity at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 1B

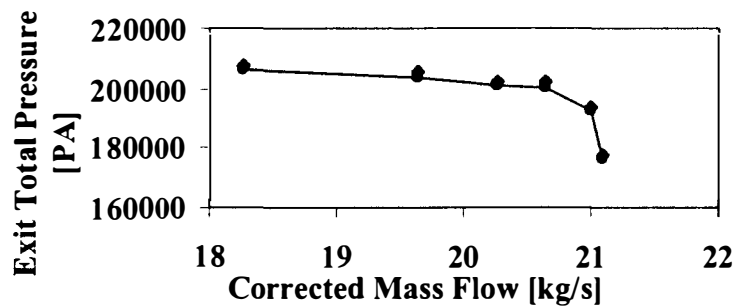


Figure 4.11. Comparison of MWA Data and Internally Consistent Data for Exit Total Pressure at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 35

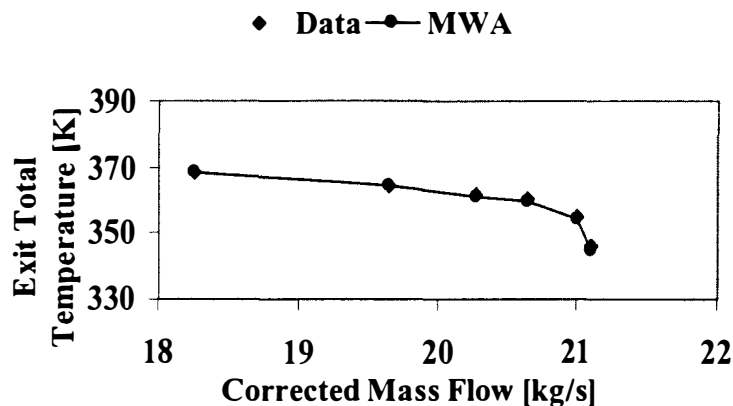


Figure 4.12. Comparison of MWA Data and Internally Consistent Data for Exit Total Temperature at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 35

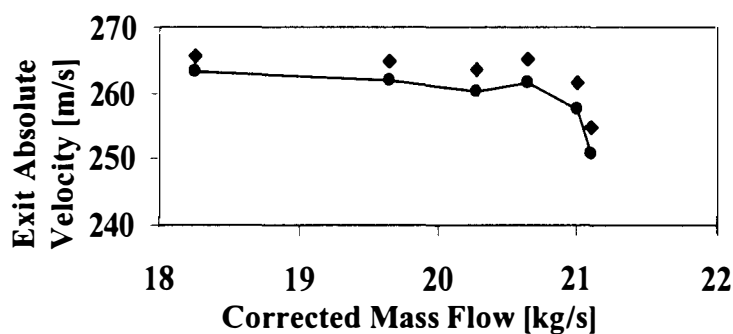


Figure 4.13. Comparison of MWA Data and Internally Consistent Data for Exit Absolute Velocity at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 35

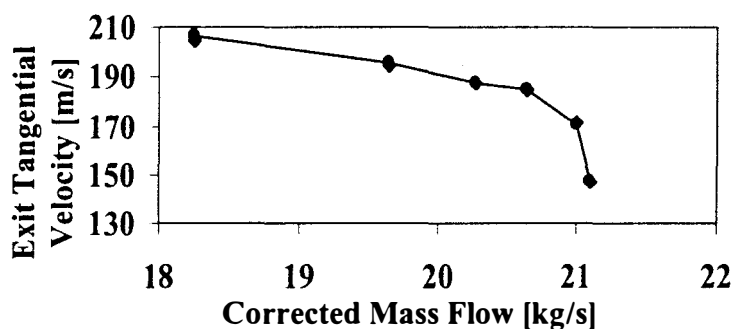


Figure 4.14. Comparison of MWA Data and Internally Consistent Data for Exit Tangential Velocity at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 35

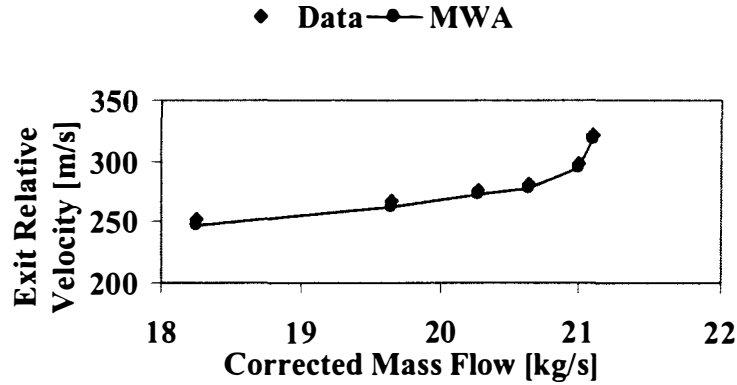


Figure 4.15. Comparison of MWA Data and Internally Consistent Data for Exit Relative Velocity at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 35

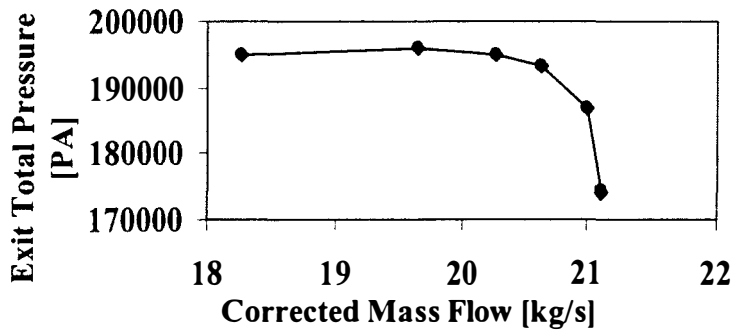


Figure 4.16. Comparison of MWA Data and Internally Consistent Data for Exit Total Pressure at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Stator 35

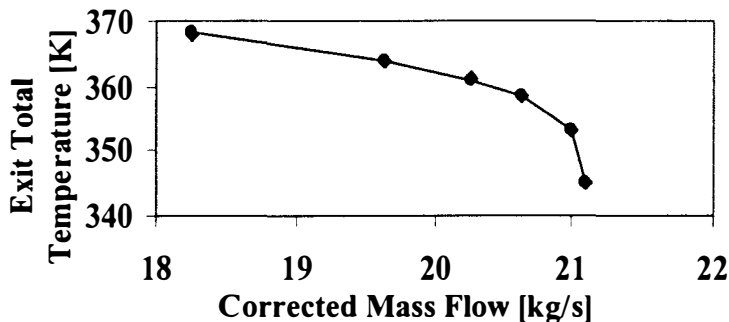


Figure 4.17. Comparison of MWA Data and Internally Consistent Data for Exit Total Temperature at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Stator 35

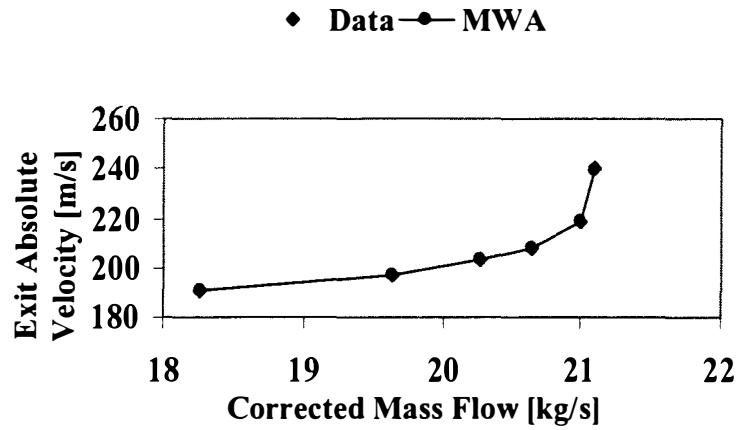


Figure 4.18. Comparison of MWA Data and Internally Consistent Data for Exit Absolute Velocity at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Stator 35

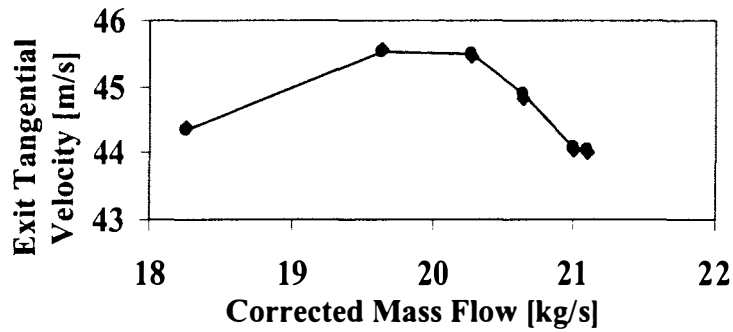


Figure 4.19. Comparison of MWA Data and Internally Consistent Data for Exit Tangential Velocity at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Stator 35

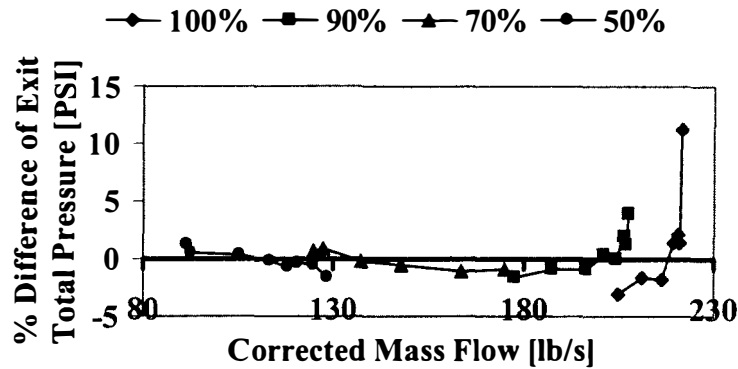


Figure 5.1. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Total Pressure at Different Corrected Mass Flows and Different Speeds for Rotor 1B

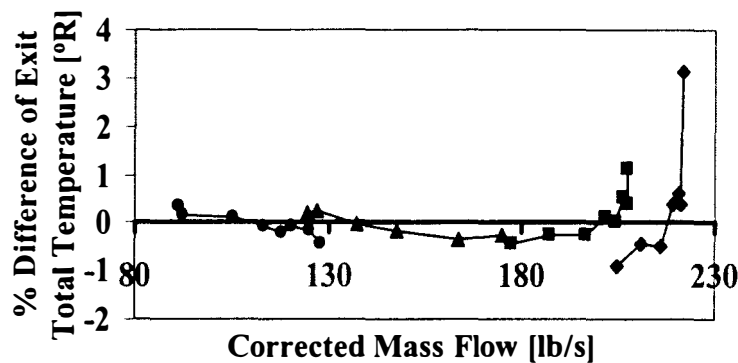


Figure 5.2. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Total Temperature at Different Corrected Mass Flows and Different Speeds for Rotor 1B

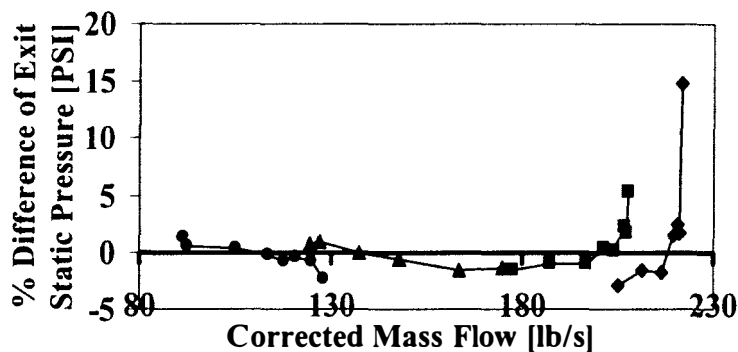


Figure 5.3. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Static Pressure at Different Corrected Mass Flows and Different Speeds for Rotor 1B

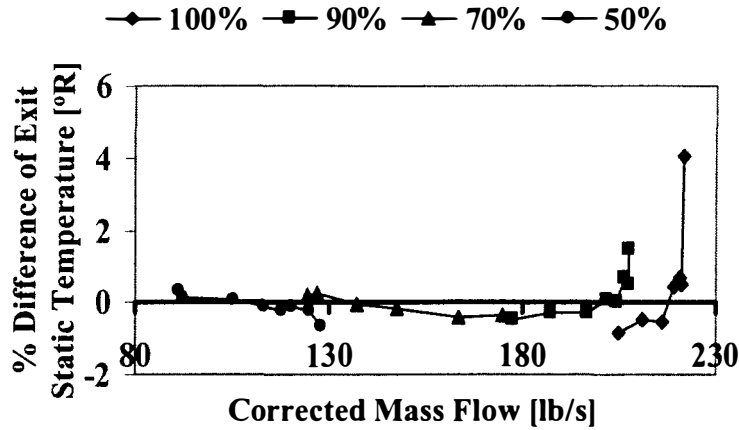


Figure 5.4. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Static Temperature at Different Corrected Mass Flows and Different Speeds for Rotor 1B

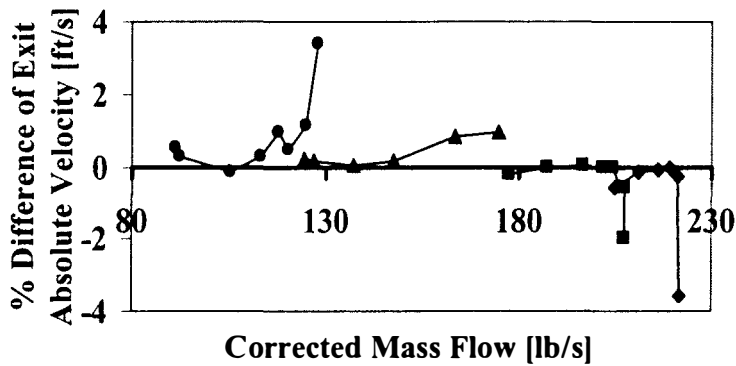


Figure 5.5. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Absolute Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 1B

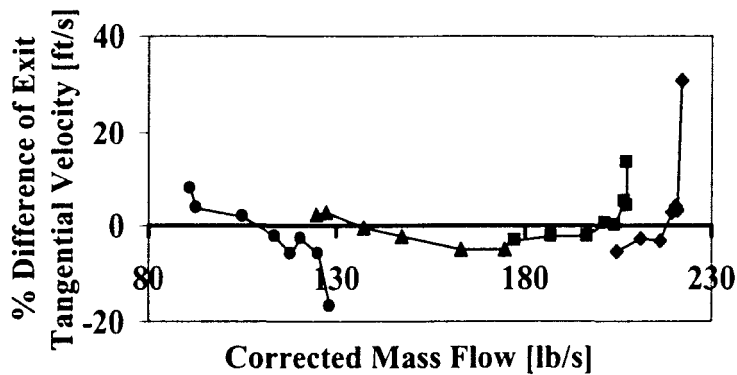


Figure 5.6. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Tangential Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 1B

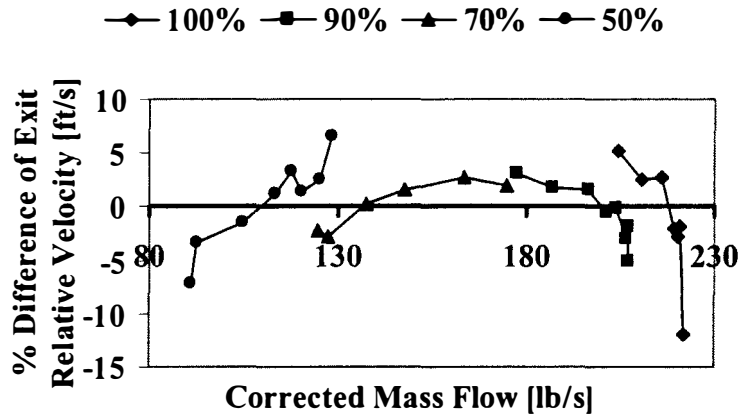


Figure 5.7. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Relative Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 1B

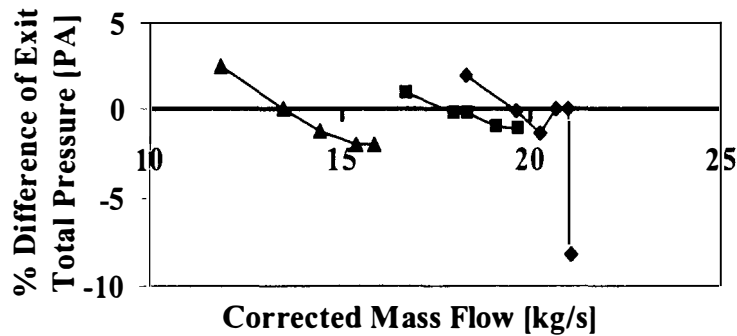


Figure 5.8. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Total Pressure at Different Corrected Mass Flows and Different Speeds for Rotor 35

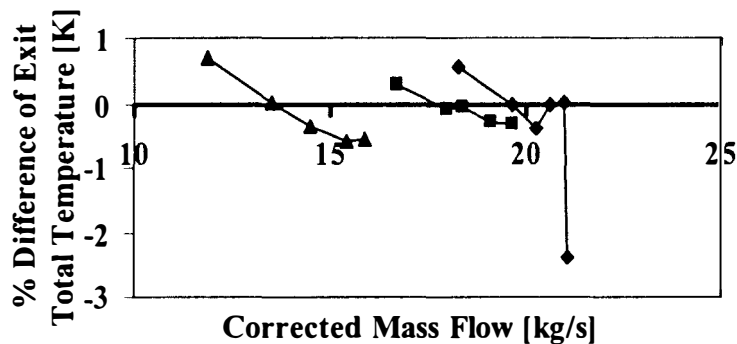


Figure 5.9. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Total Temperature at Different Corrected Mass Flows and Different Speeds for Rotor 35

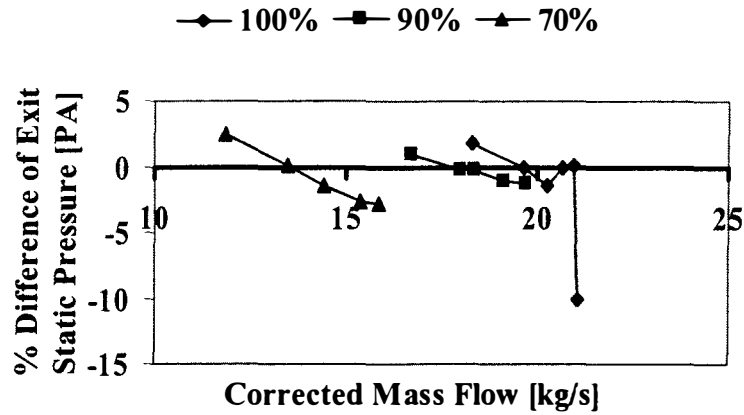


Figure 5.10. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Static Pressure at Different Corrected Mass Flows and Different Speeds for Rotor 35

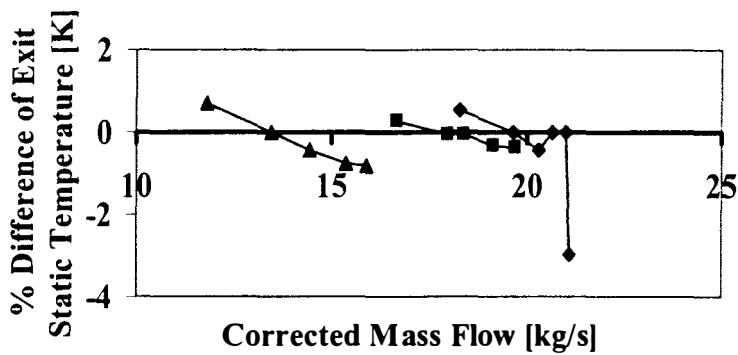


Figure 5.11. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Static Temperature at Different Corrected Mass Flows and Different Speeds for Rotor 35

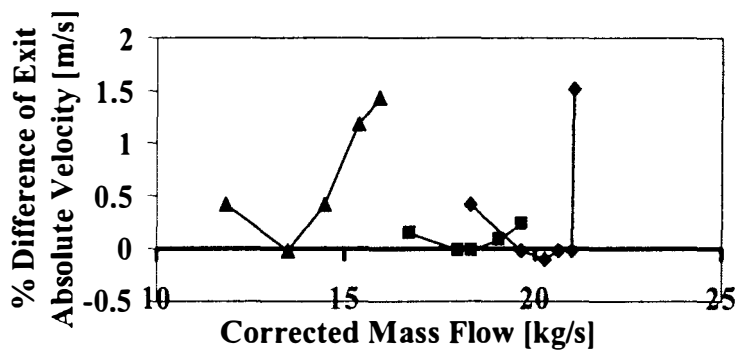


Figure 5.12. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Absolute Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 35

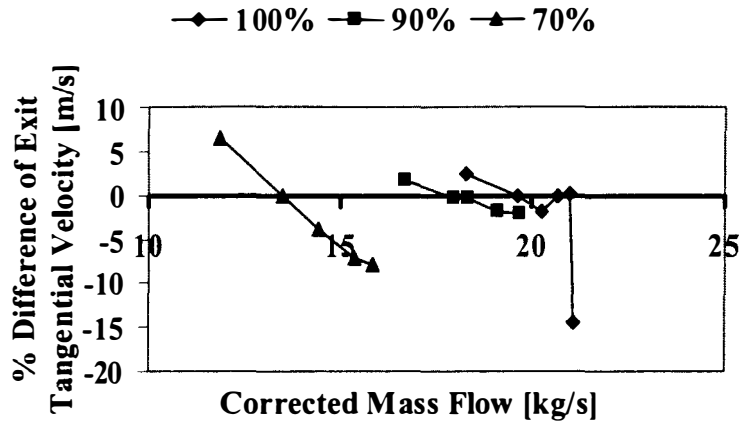


Figure 5.13. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Tangential Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 35

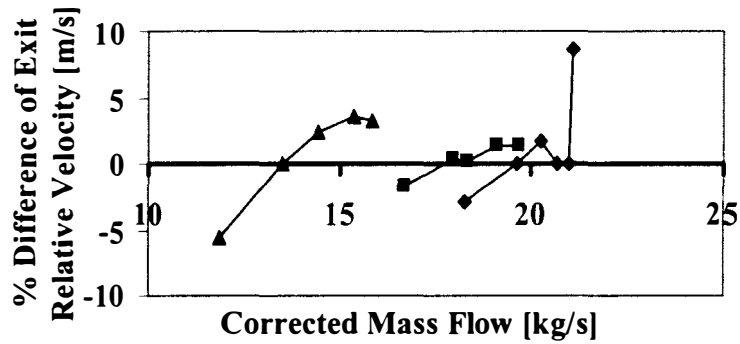


Figure 5.14. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Relative Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 35

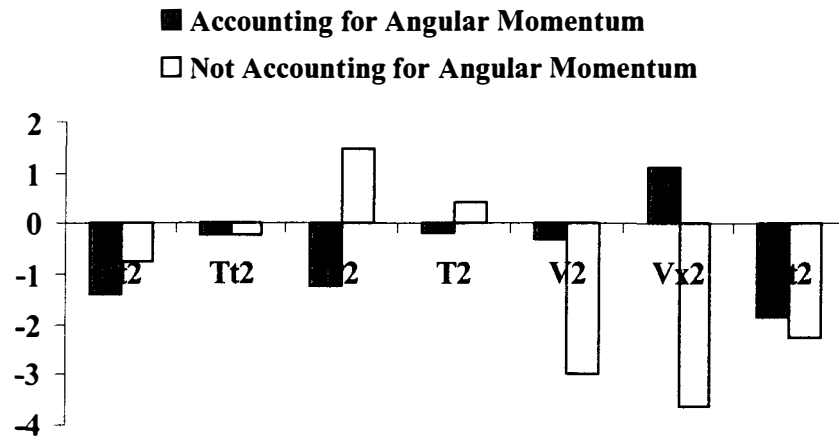


Figure 5.15. Comparison with Data of MLC Results Accounting for the Conservation of Angular Momentum through the NBR and Not Accounting for it for the 100% Speed Line of Stage 35

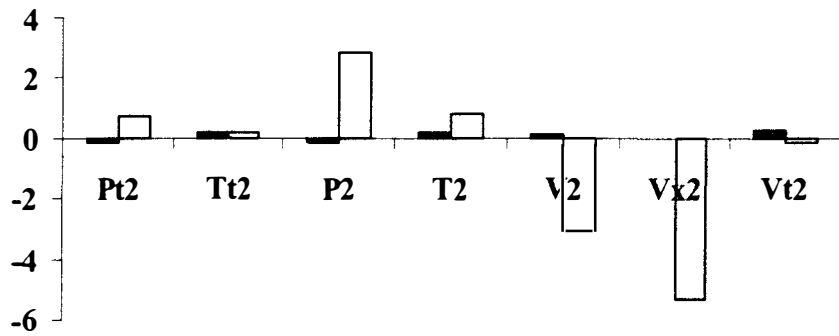


Figure 5.16. Comparison with Data of MLC Results Accounting for the Conservation of Angular Momentum through the NBR and Not Accounting for it for the 90% Speed Line of Stage 35

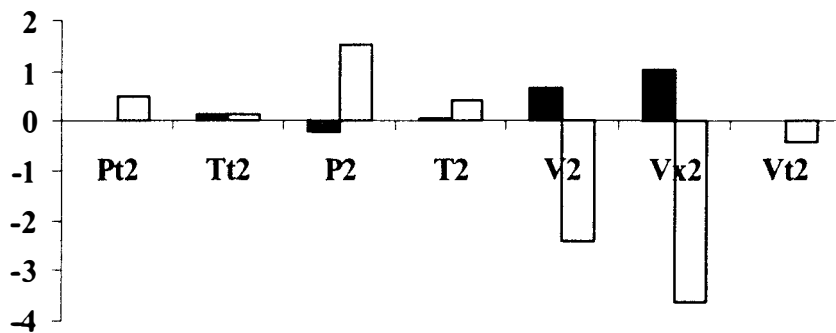


Figure 5.17. Comparison with Data of MLC Results Accounting for the Conservation of Angular Momentum through the NBR and Not Accounting for it for the 70% Speed Line of Stage 35

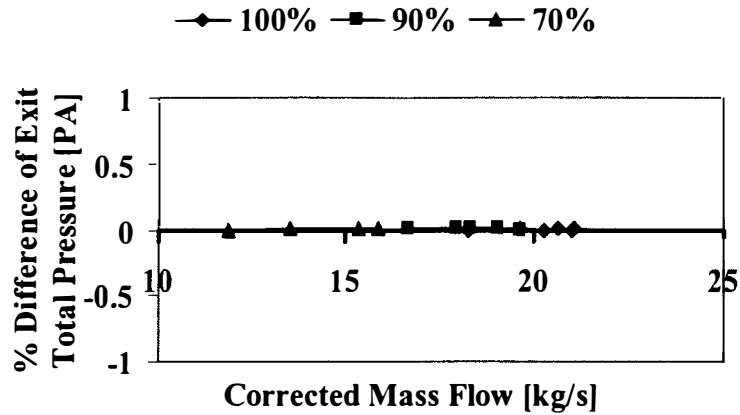


Figure 5.18. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Total Pressure at Different Corrected Mass Flows and Different Speeds for Stator 35

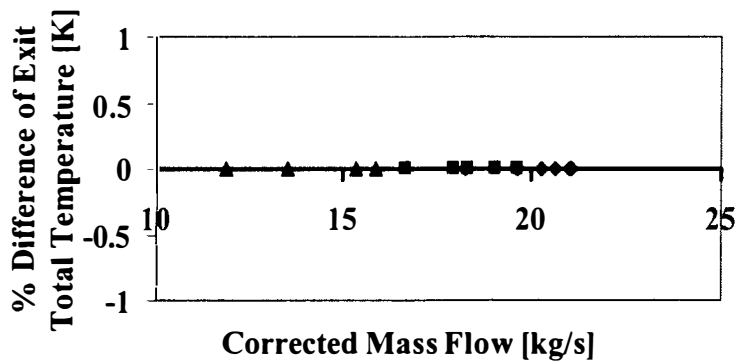


Figure 5.19. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Total Temperature at Different Corrected Mass Flows and Different Speeds for Stator 35

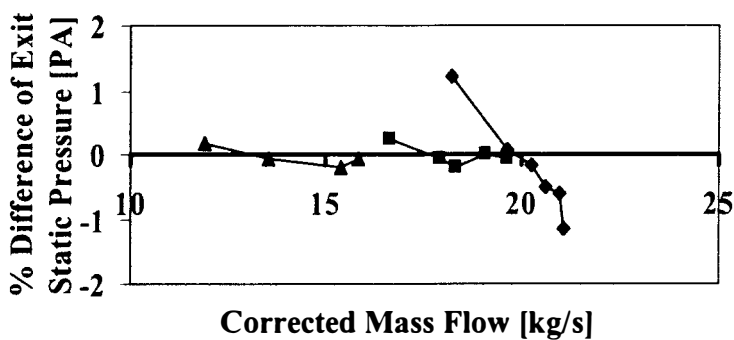


Figure 5.20. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Static Pressure at Different Corrected Mass Flows and Different Speeds for Stator 35

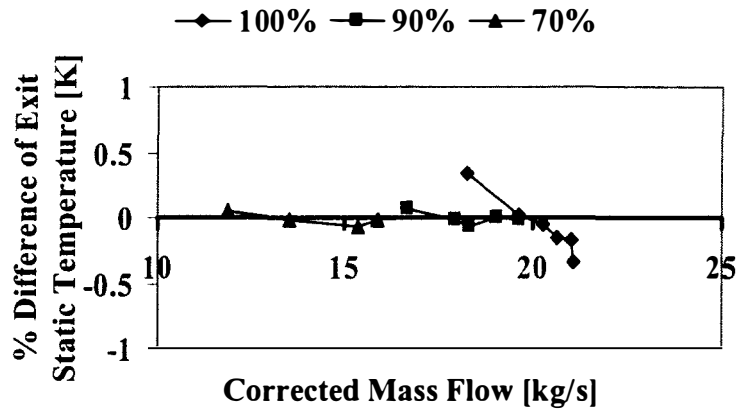


Figure 5.21. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Static Temperature at Different Corrected Mass Flows and Different Speeds for Stator 35

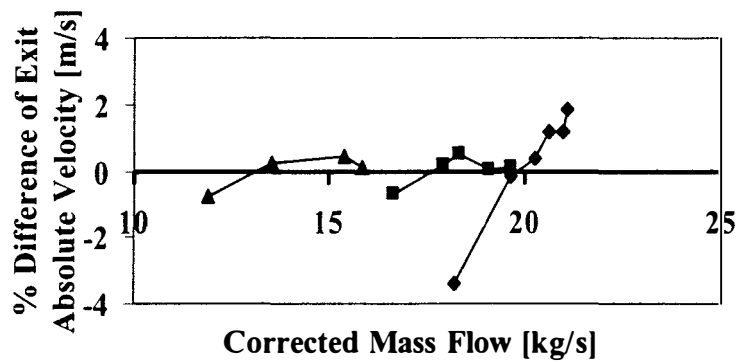


Figure 5.22. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Absolute Velocity at Different Corrected Mass Flows and Different Speeds for Stator 35

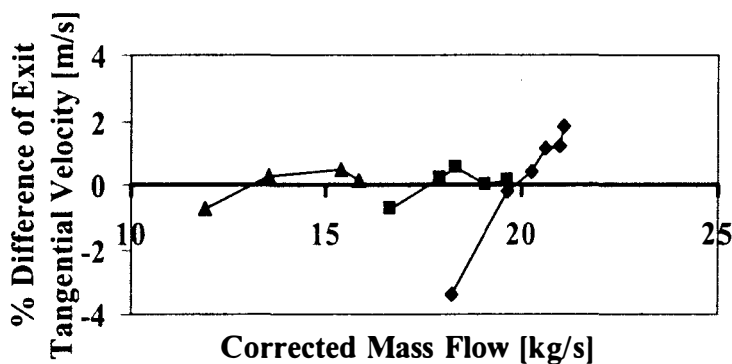


Figure 5.23. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Tangential Velocity at Different Corrected Mass Flows and Different Speeds for Stator 35

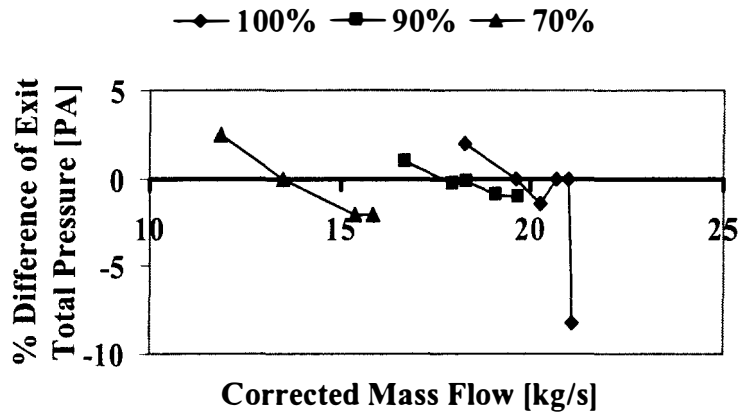


Figure 5.24. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Total Pressure at Different Corrected Mass Flows and Different Speeds for Stage 35

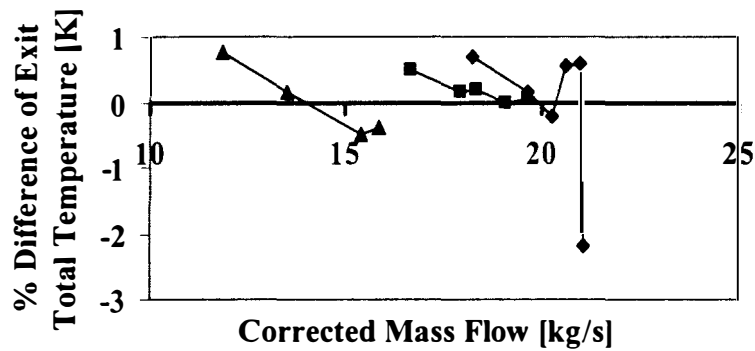


Figure 5.25. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Total Temperature at Different Corrected Mass Flows and Different Speeds for Stage 35

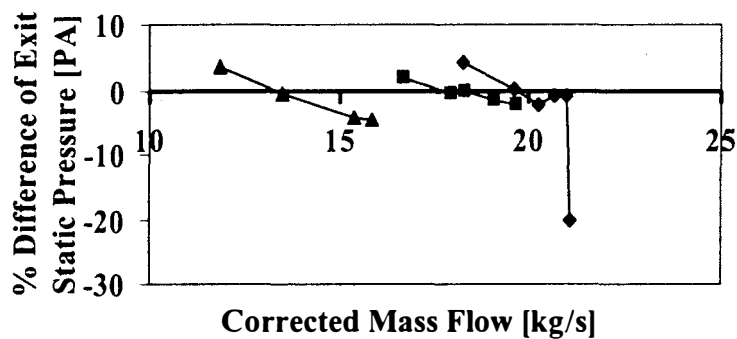


Figure 5.26. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Static Pressure at Different Corrected Mass Flows and Different Speeds for Stage 35

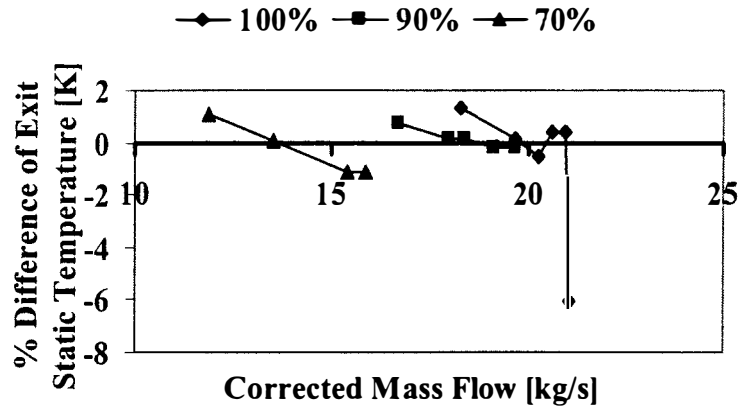


Figure 5.27. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Static Temperature at Different Corrected Mass Flows and Different Speeds for Stage 35

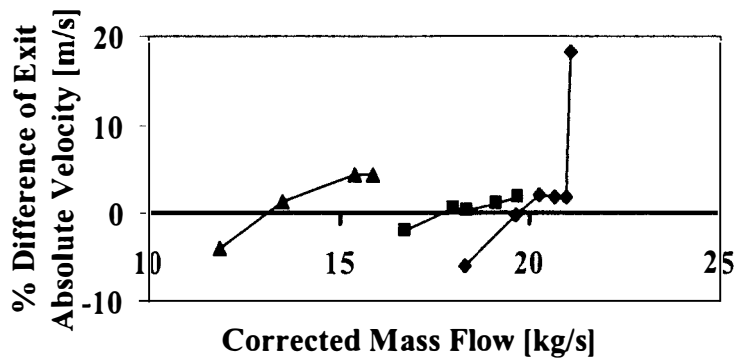


Figure 5.28. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Absolute Velocity at Different Corrected Mass Flows and Different Speeds for Stage 35

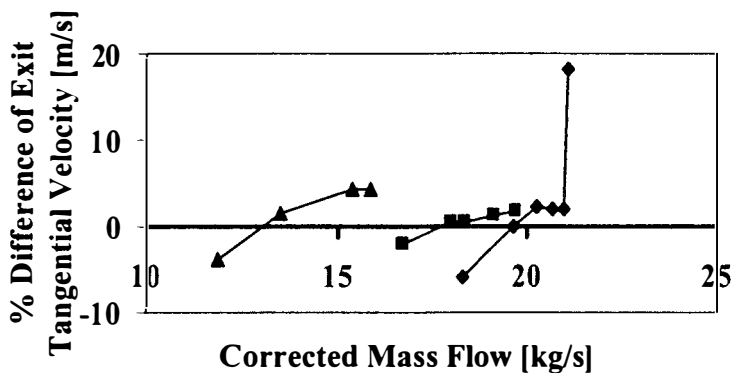


Figure 5.29. Percent Difference of MLC Results with Loss and Deviation Specified for Exit Tangential Velocity at Different Corrected Mass Flows and Different Speeds for Stage 35

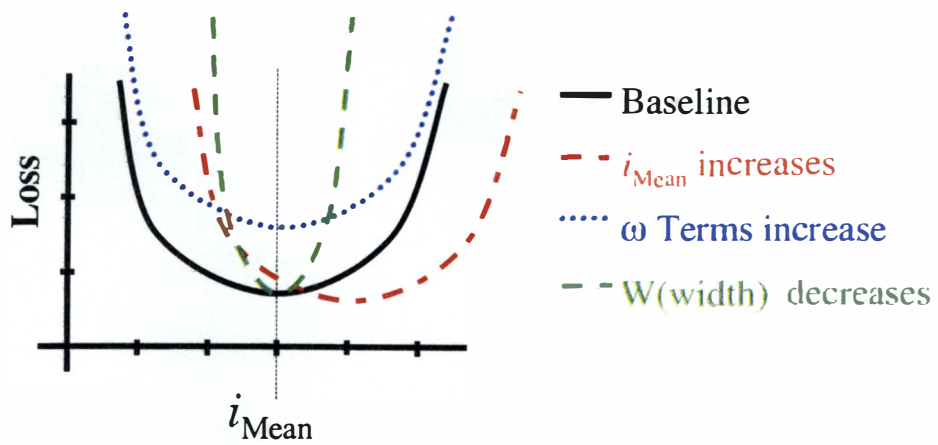


Figure 6.1. Representative Loss Bucket.
(Klepper, 1998)

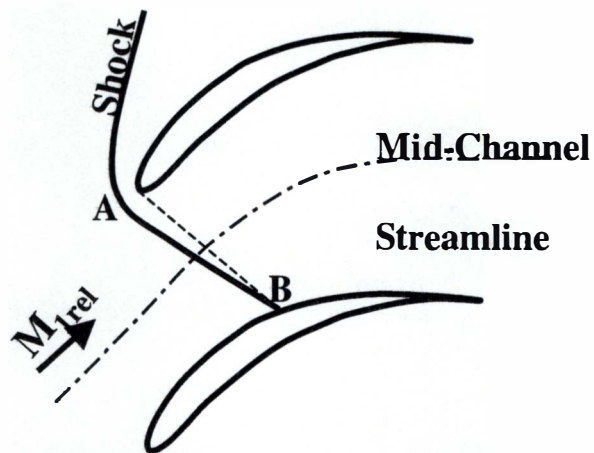


Figure 6.2. Schematic of Lewis, Hartman, Miller Shock Model.
(Klepper, 1998)

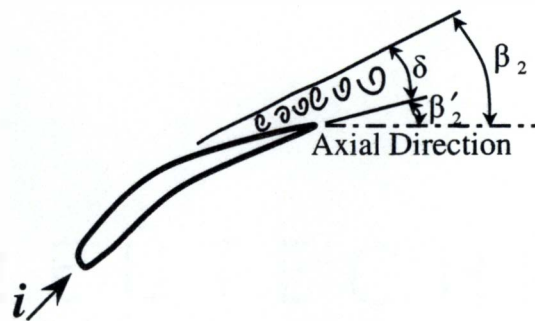


Figure 6.3. Schematic of Deviation.
(Klepper, 1998)

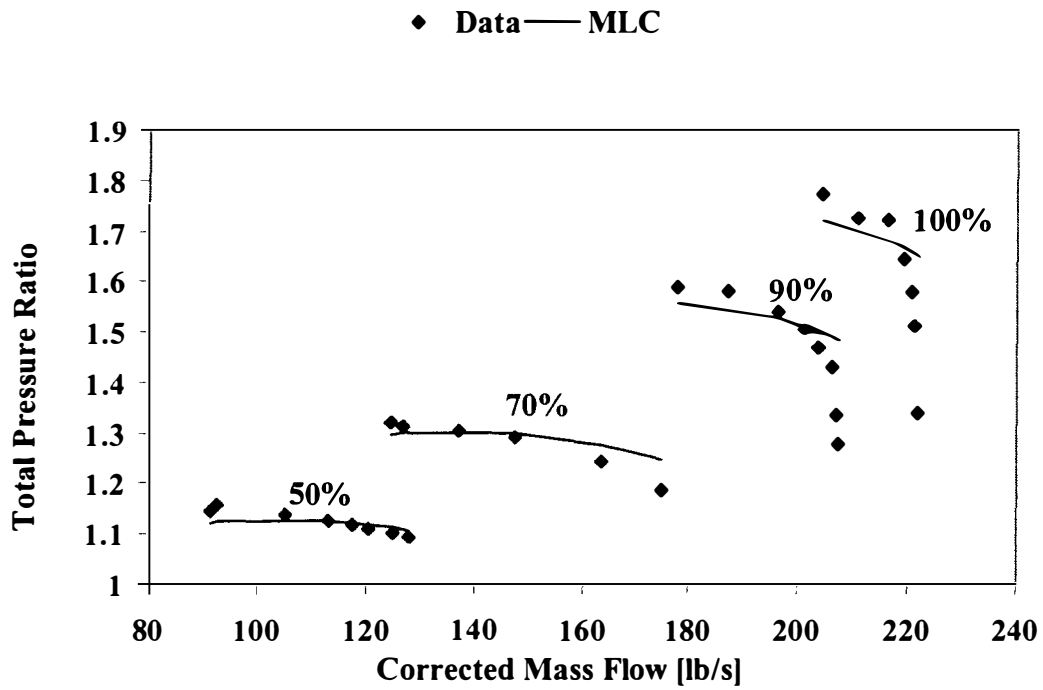


Figure 6.4. Pressure Characteristic Map for Rotor 1B with Predicted Results and Data

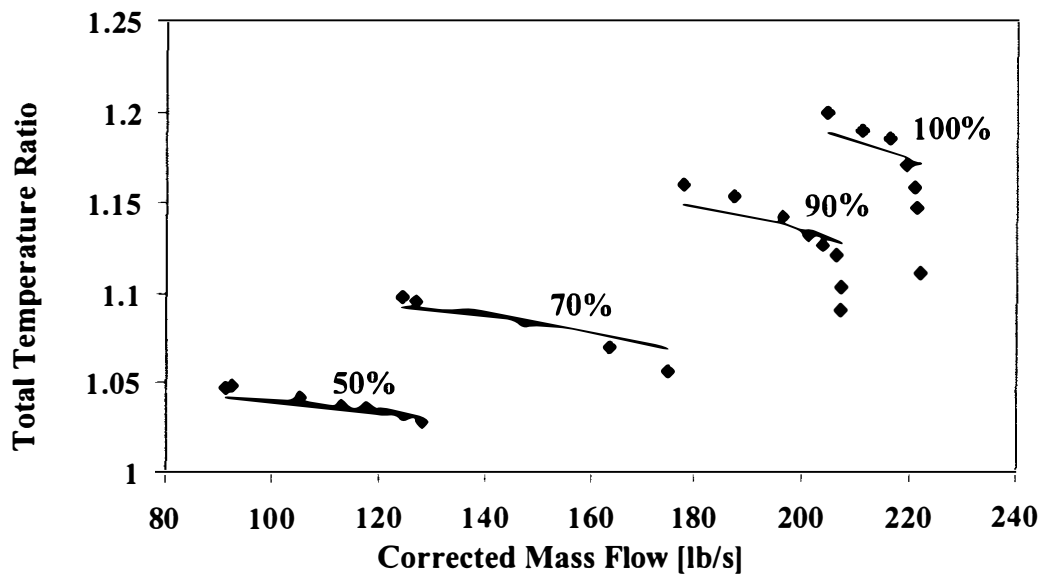


Figure 6.5. Temperature Characteristic Map for Rotor 1B with Predicted Results and Data

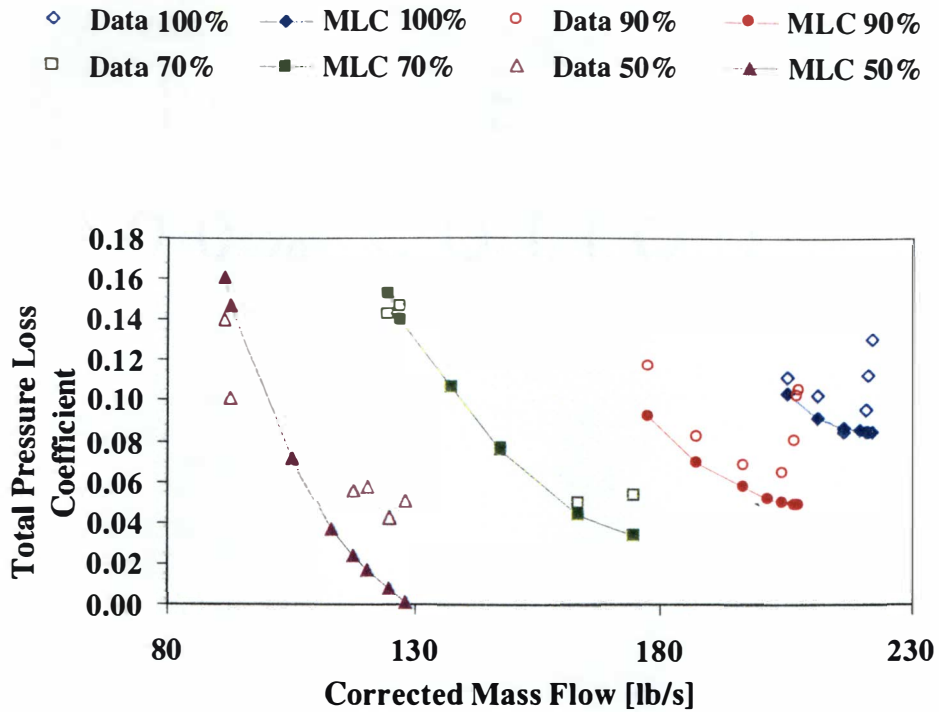


Figure 6.6. Predicted Loss Compared to Data for Different Speeds of Rotor 1B

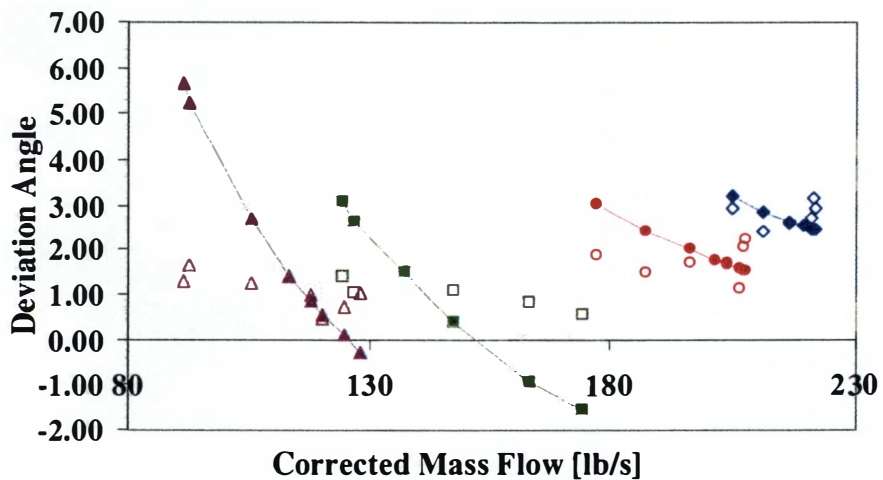


Figure 6.7. Predicted Deviation Compared to Data for Different Speeds of Rotor 1B

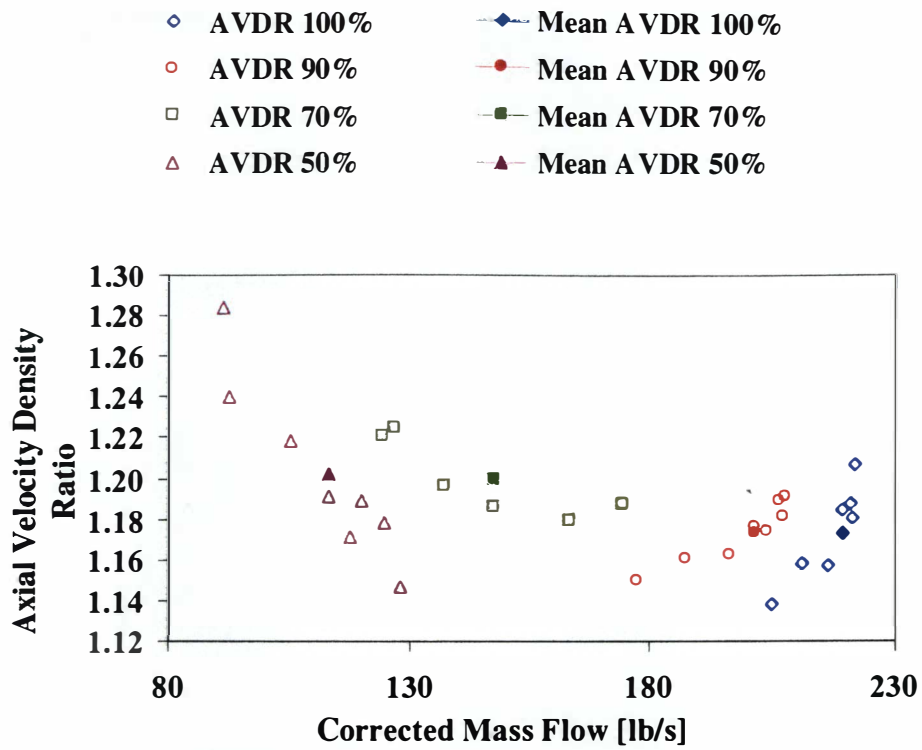


Figure 6.8. Mean AVDR Compared to Specific AVDRs for Different Speed Lines of Rotor 1B

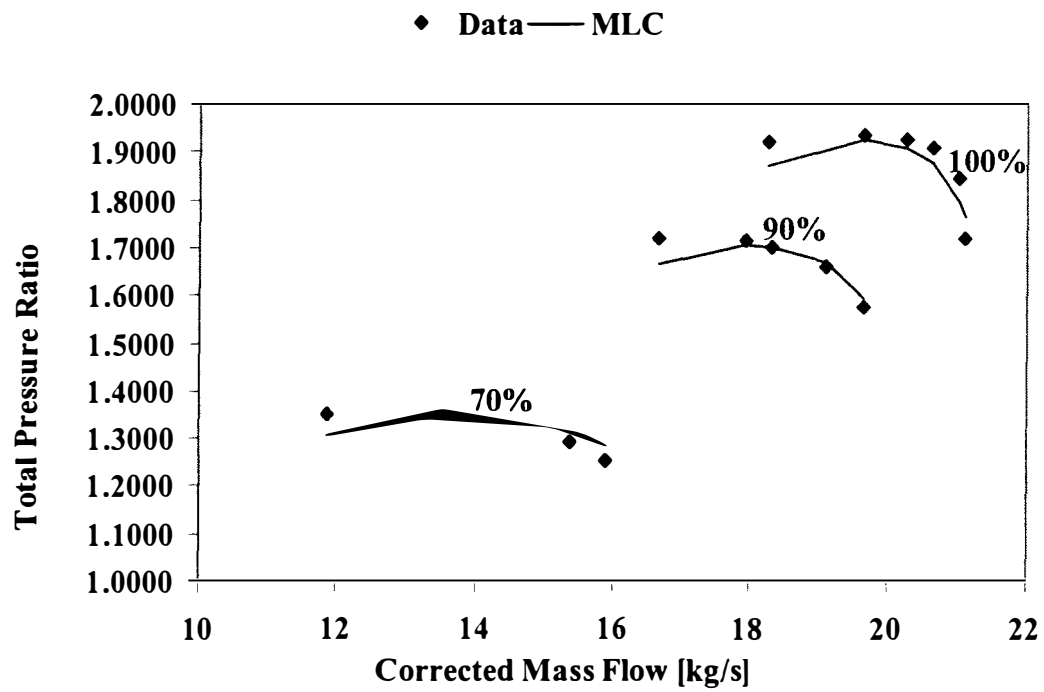


Figure 6.9. Pressure Characteristic Map for Stage 35 with Predicted Results and Data

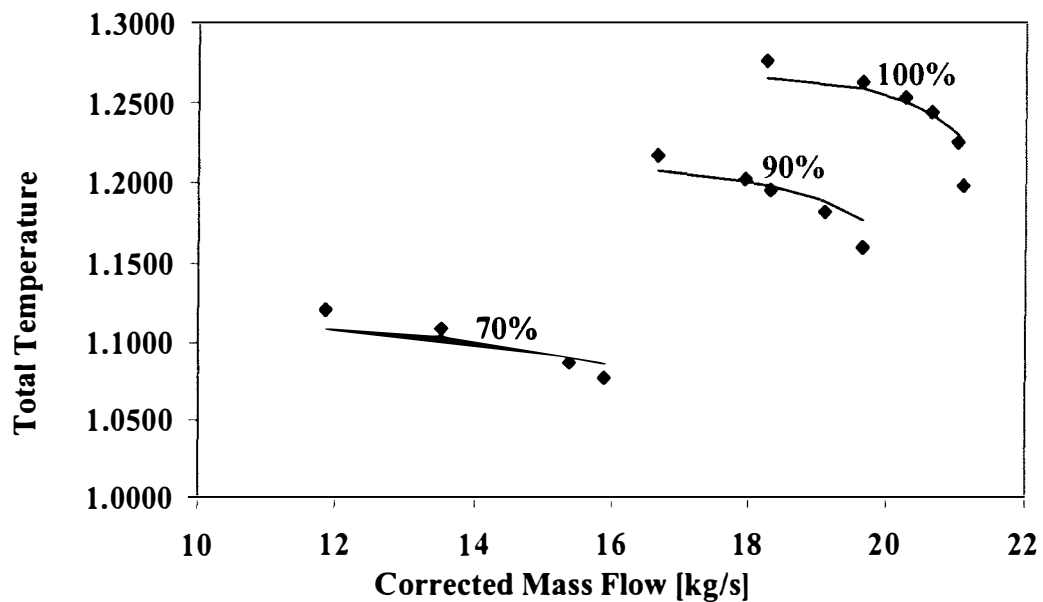


Figure 6.10. Temperature Characteristic Map for Stage 35 with Predicted Results and Data

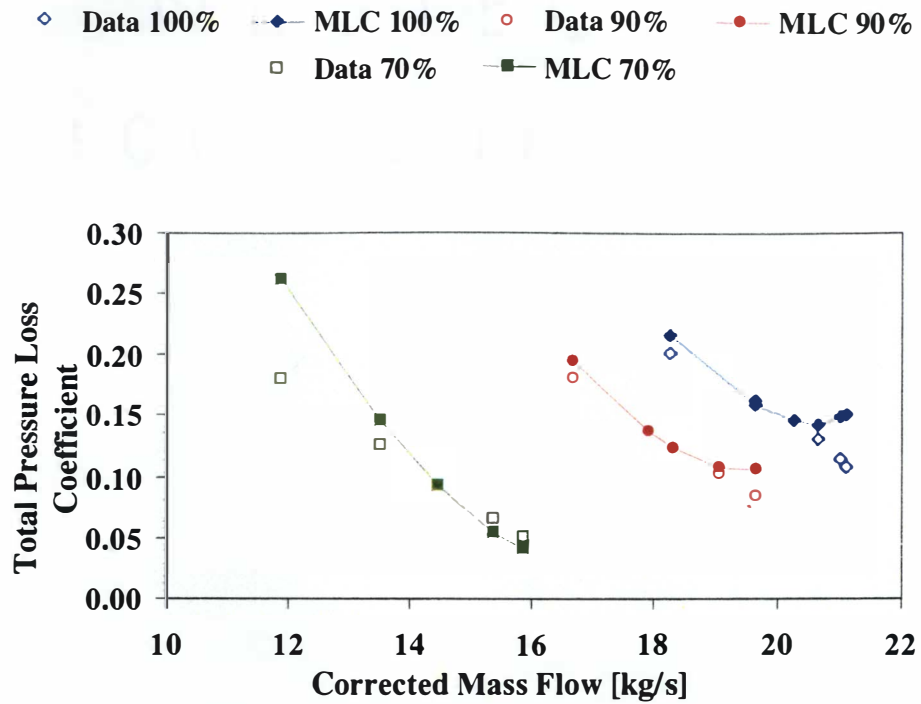


Figure 6.11. Predicted Loss Compared to Data for Different Speeds of Rotor 35

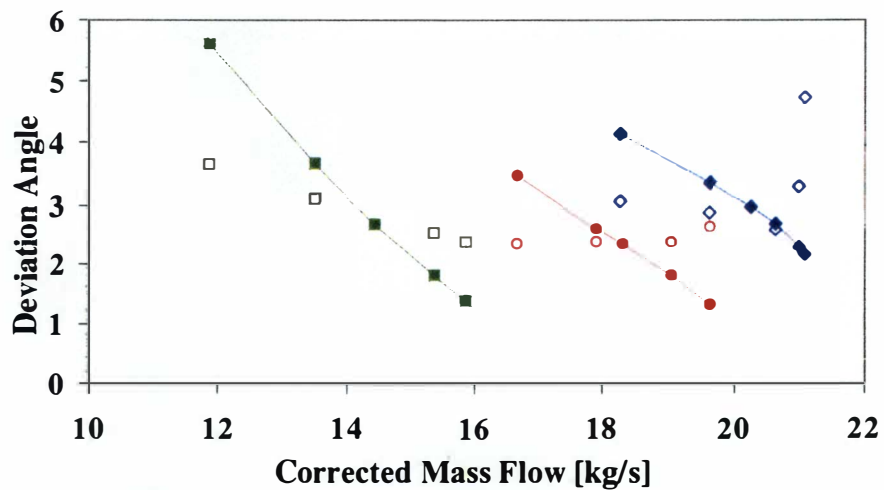


Figure 6.12. Predicted Deviation Compared to Data for Different Speeds of Rotor 35

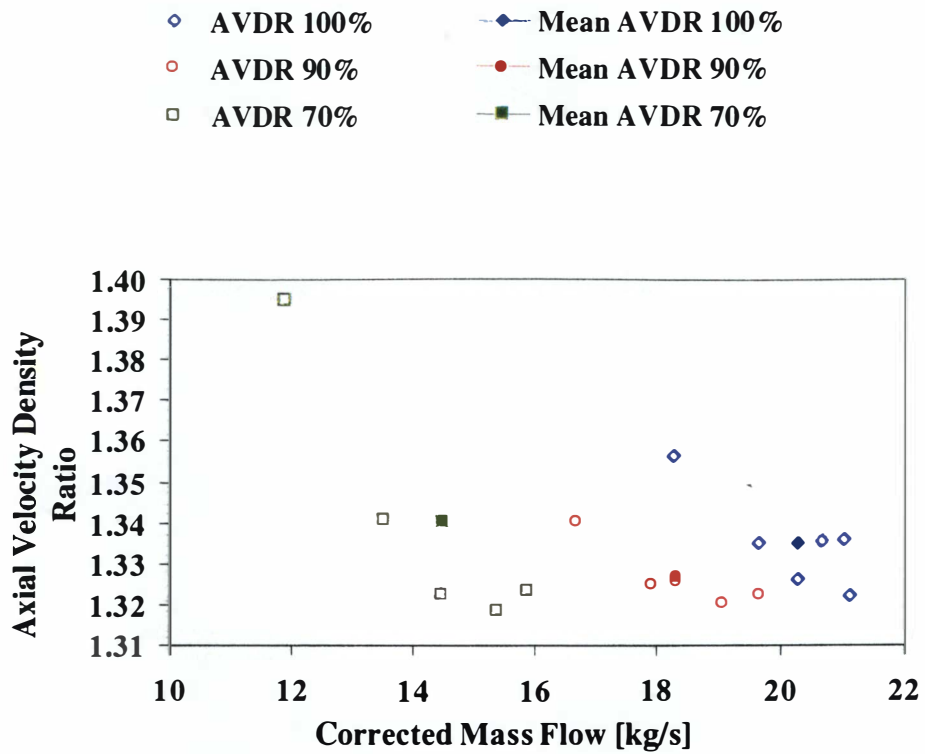


Figure 6.13. Mean AVDR Compared to Specific AVDRs for Different Speed Lines of Rotor 35

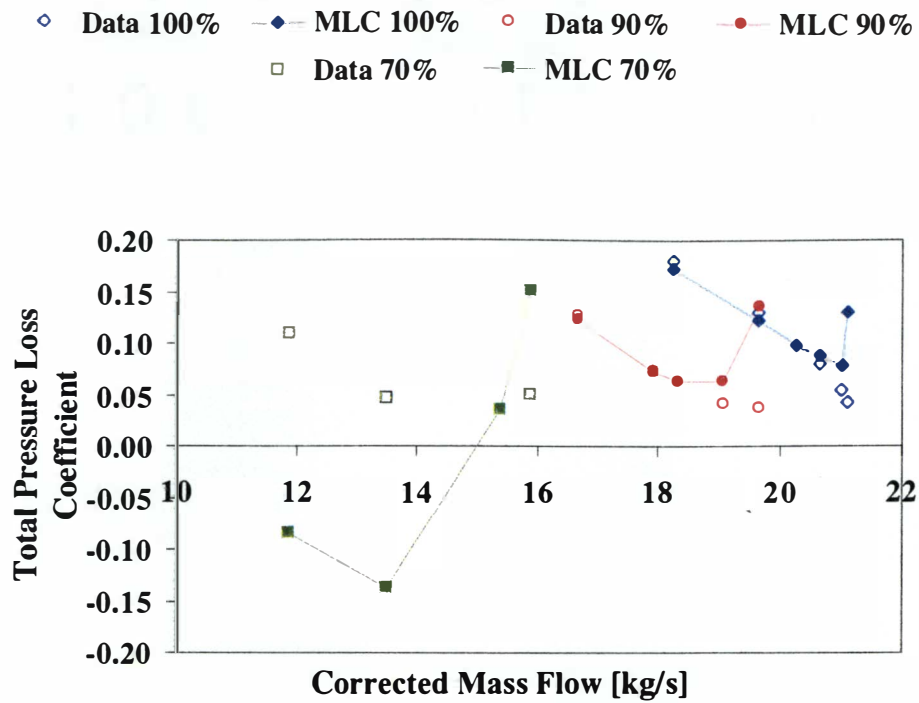


Figure 6.14. Predicted Loss Compared to Data for Different Speeds of Stator 35

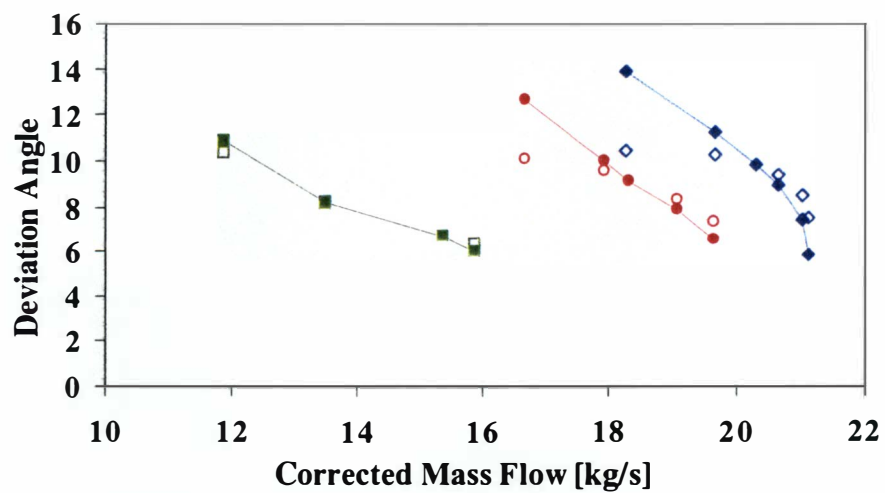


Figure 6.15. Predicted Deviation Compared to Data for Different Speeds of Stator 35

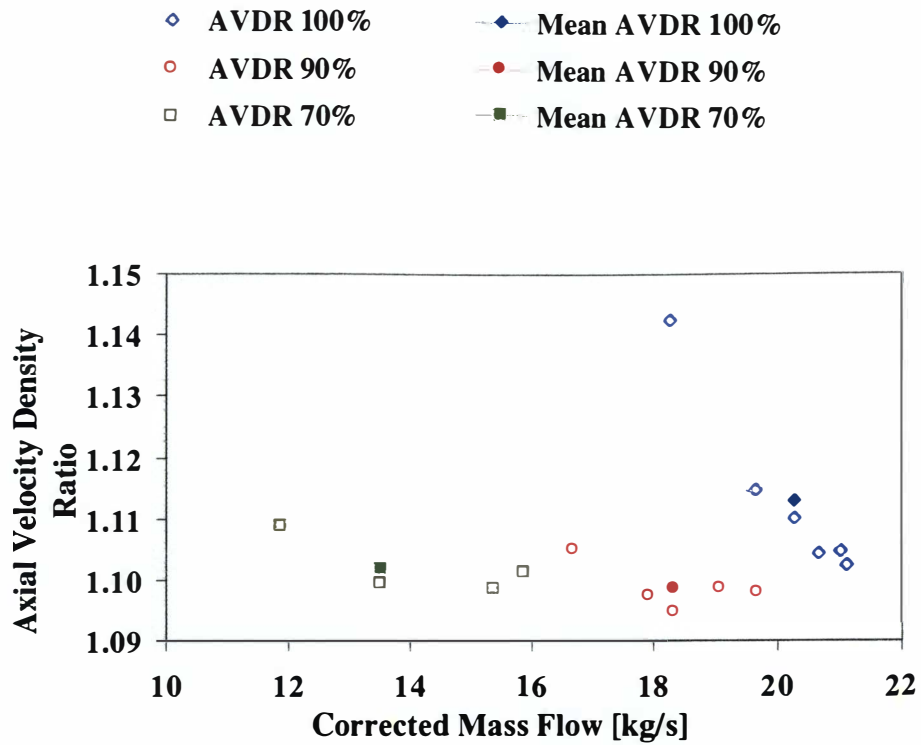


Figure 6.16. Mean AVDR Compared to Specific AVDRs for Different Speed Lines of Stator 35

APPENDIX B

TABLES

Table 4.1. Estimated Uncertainties in Stage 35 Experimental Data

Variable	Estimated Uncertainties
Mass Flow	+/- 0.3 (kg/s)
Rotative Speed	+/- 30 (rpm)
Flow Angle	+/- 1.0 (deg)
Temperature	+/- 0.6 (K)
Rotor-Inlet Total Pressure	+/- 0.01 (N/cm ²)
Rotor-Inlet Static Pressure	+/- 0.03 (N/cm ²)
Stator-Outlet Total Pressure	+/- 0.17 (N/cm ²)
Stator-Outlet Static Pressure	+/- 0.10 (N/cm ²)

Table 5.1. Percent Difference of MLC with Loss and Deviation Specified for a Middle Point on the 100% Speed Line of Rotor 1B with the Specific ς Calculated for this Point

Variable	Data	MLC	% Error
Pt2	24.20	24.20	0.00
Tt2	606.82	606.82	0.00
P2	18.43	18.42	0.00
T2	561.36	561.36	0.00
V2	738.87	738.87	0.00
Vt2	474.33	474.33	0.00
W2	856.00	856.00	0.00

Table 5.2. Percent Difference of MLC with Loss and Deviation Specified for a Middle Point on the 100% Speed Line of Rotor 35 with the Specific ς Calculated for this Point

Variable	Data	MLC	% Error
Pt2	202469.90	202283.20	-0.09
Tt2	361.71	361.63	-0.02
P2	142437.50	142316.39	-0.09
T2	327.13	327.06	-0.02
V2	263.57	263.51	-0.02
Vt2	187.81	187.62	-0.10
W2	275.96	276.14	0.06

Table 5.3. Percent Difference of MLC with Loss and Deviation Specified for a Middle Point on the 100% Speed Line of Stator 35 with the Specific ζ Calculated for this Point

Variable	Data	MLC	% Error
Pt2	194989.42	194992.22	0.00
Tt2	361.09	361.09	0.00
P2	158779.35	158783.05	0.00
T2	340.50	340.50	0.00
V2	203.35	203.35	0.00
Vt2	45.47	45.47	0.00
W2	203.35	203.35	0.00

Table 5.4. Percent Difference of MLC with Loss and Deviation Specified for a Middle Point on the 100% Speed Line of Stage 35 with the Specific ζ Calculated for this Point

Variable	Data	MLC	% Error
Pt2	194989.42	196092.47	-0.57
Tt2	361.09	361.63	-0.15
P2	158779.35	160234.17	-0.92
T2	340.50	341.35	-0.25
V2	203.35	201.82	0.75
Vt2	45.47	45.13	0.75
W2	203.35	201.82	0.75

Table 5.5. AVDR, ζ , and Inlet and Exit Mean Radii for Rotor 1B

Speed Lines				
Variable	100%	90%	70%	50%
AVDR	1.1730	1.1734	1.1996	1.2024
ζ	1.0325	1.0329	1.0559	1.0584
R_1	14.3716 in.	14.3716 in.	14.3716 in.	14.3716 in.
R_2	14.5381 in.	14.5381 in.	14.5381 in.	14.5381 in.

Table 5.6. AVDR, ζ , and Inlet and Exit Mean Radii for Rotor 35

Speed Lines			
Variable	100%	90%	70%
AVDR	1.3350	1.3268	1.3402
ζ	1.0411	1.0348	1.0451
R_1	0.2184 m	0.2184 m	0.2184 m
R_2	0.2180 m	0.2180 m	0.2180 m

Table 5.7. AVDR, ζ , and Inlet and Exit Mean Radii for NBR 35

Speed Lines			
Variable	100%	90%	70%
AVDR	1.0273	1.0312	1.0340
ζ	0.9595	0.9632	0.9658
R_1	0.2180 m	0.2180 m	0.2180 m
R_2	0.2171 m	0.2171 m	0.2171 m

Table 5.8. AVDR, ζ , and Inlet and Exit Mean Radii for Stator 35

Speed Lines			
Variable	100%	90%	70%
AVDR	1.1130	1.0987	1.1019
ζ	0.9957	0.9829	0.9858
R_1	0.2171 m	0.2171 m	0.2171 m
R_2	0.2175 m	0.2175 m	0.2175 m

Table 6.1. Add_loss and Add_dev for Rotor 1B

Speed Lines				
Variable	100%	90%	70%	50%
Add_loss	-0.0298	-0.0324	-0.0191	-0.0534
Add_Dev	-2.4513	-2.0745	-2.9113	-0.8756

Table 6.2. Add_loss and Add_dev for Rotor 35

Speed Lines			
Variable	100%	90%	70%
Add_loss	-0.1309	-0.0813	-0.0729
Add_Dev	-4.0285	-3.5628	-0.8371

Table 6.3. Add_loss and Add_dev for Stator 35

Speed Lines			
Variable	100%	90%	70%
Add_loss	-0.0098	-0.0403	-0.2577
Add_Dev	2.568	3.3841	3.5145

APPENDIX C

DISCUSSION OF MWA METHOD

APPENDIX C – DESCRIPTION OF MASS-WEIGHTED AVERAGE METHOD

A 1-D representation of the flow field was obtained by a mass-weighted average of the radial distribution of experimental data. For this to be done, the radial distribution of mass flow was required. A radial distribution of flow quantities was measured using probes located at 10-, 30-, 50-, 70-, and 90-percent immersion points in the flow field just upstream and downstream of the blade. Immersion lines are shown in Figure C.1. The density and axial velocity are known from this experimental data. The total mass flow can be calculated by the equation:

$$\dot{m}_T = \rho AV = \rho(\vec{A} \cdot \vec{V}), \quad [C.1]$$

where $\dot{m}_T$ = total mass flow. If $\vec{A} = A_{\perp V_x} \hat{i} + 0\hat{j} + 0\hat{k}$ and $\vec{V} = V_x \hat{i} + V_y \hat{j} + V_z \hat{k}$, the mass flow can be written as

$$\dot{m}_T = \rho A_{\perp V_x} V_x \quad \text{or} \quad [C.2]$$

$$\dot{m}_T = \sum_{i=1}^n \dot{m}_i = \rho_1 V_{x1} A_{\perp V_{x1}} + \rho_2 V_{x2} A_{\perp V_{x2}} + \dots + \rho_n V_{xn} A_{\perp V_{xn}}. \quad [C.3]$$

The mass-weighted average of a discretely measured variable is defined by:

$$X = \frac{\sum_{i=1}^n \dot{m}_i X_i}{\sum_{i=1}^n \dot{m}_i}, \quad [C.4]$$

where $\dot{m}_i$ is the mass flow in the streamtube in which X_i was measured.

The radial distribution of mass flow is not known, in general, and therefore, must be estimated from available experimental data. In Figure C.2, the areas

represented by the dot-dashed lines are from mid-point to mid-point of the area between the immersion lines. A distribution of proportionality factors, f , is established as:

$$f_1 = \frac{A_1}{A_T}, f_2 = \frac{A_2}{A_T}, \dots, f_n = \frac{A_n}{A_T}, \quad [C.5]$$

where A_T is

$$A_T = \sum_{i=1}^n A_i. \quad [C.6]$$

It is assumed that the properties at the blade leading or trailing edge, respectively, are the same as at the location of its respective area (A_1, A_2 , etc.) shown in Figure C.2. In doing this, it is assumed that the proportionalities of [C.5] are the same as for the areas at the leading or trailing edge, respectively, or

$$f_1 = \frac{A_{\perp V_{x1}}}{a_T}, f_2 = \frac{A_{\perp V_{x2}}}{a_T}, \dots, f_n = \frac{A_{\perp V_{xn}}}{a_T}, \quad [C.7]$$

where a_T is assumed to be the cross-sectional area scaled to pass the specified mass flow, and is calculated by:

$$a_T = \frac{\dot{m}_T}{\sum_{i=1}^n \rho_i V_{xi} f_i}. \quad [C.8]$$

The mass flow at each radial location is then defined as:

$$\dot{m}_1 = \rho_1 V_{x1} f_1 a_T, \dot{m}_2 = \rho_2 V_{x2} f_2 a_T, \dots, \dot{m}_n = \rho_n V_{xn} f_n a_T. \quad [C.9]$$

Equation [C.4] was then used to calculate the mass weighted averages of the compressor flow fields.

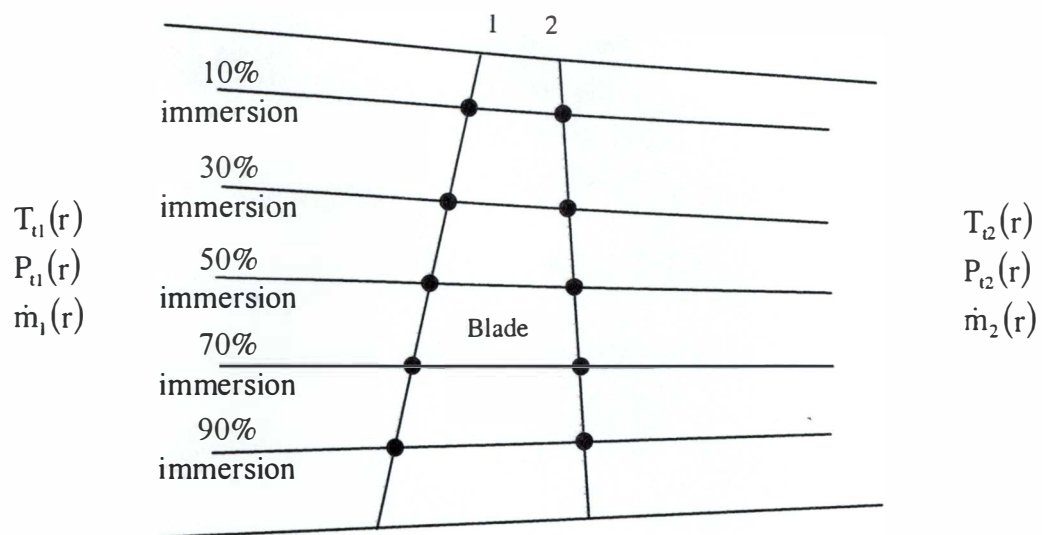


Figure C.1. Location of Instrumentation Immersion Lines

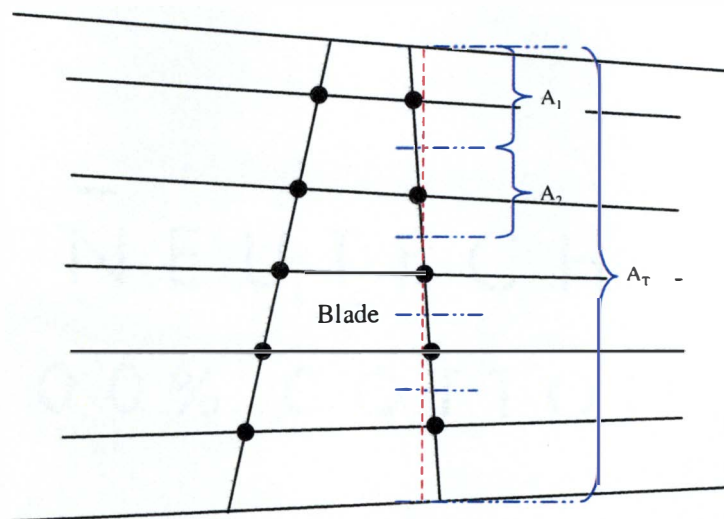


Figure C.2. Location of Instrumentation Immersion Lines and Areas

APPENDIX D

COMPARISON OF MWA DATA TO CONSISTENT SET OF DATA

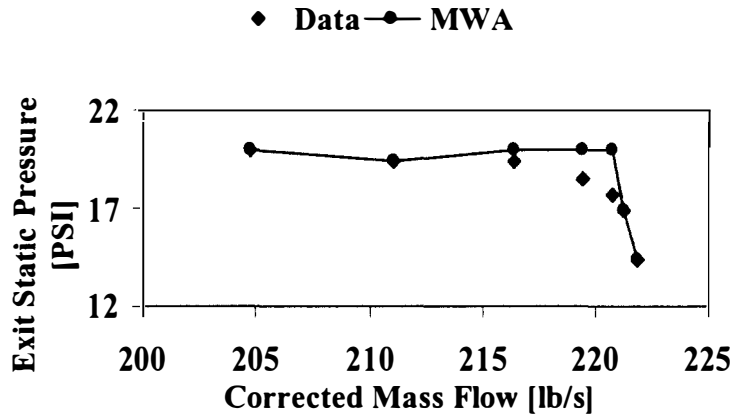


Figure D.1. Comparison of MWA Data and Internally Consistent Data for Exit Static Pressure at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 1B

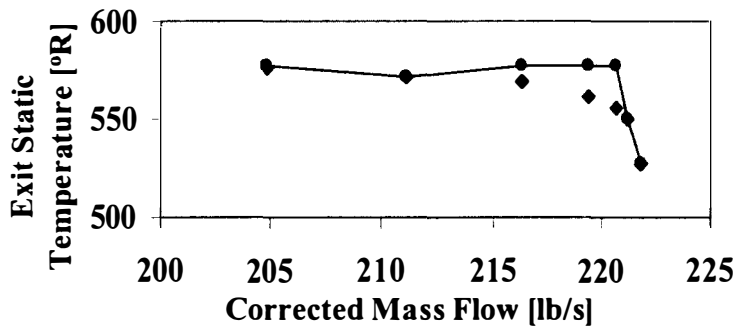


Figure D.2. Comparison of MWA Data and Internally Consistent Data for Exit Static Temperature at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 1B

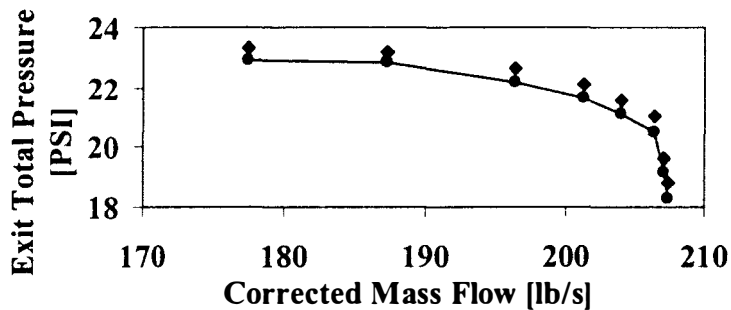


Figure D.3. Comparison of MWA Data and Internally Consistent Data for Exit Total Pressure at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 1B

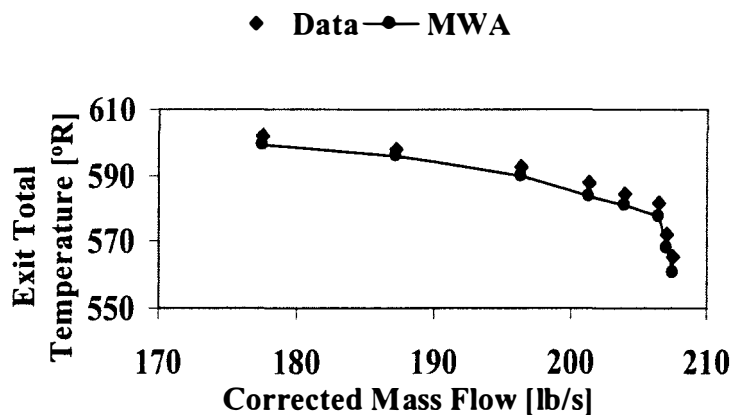


Figure D.4. Comparison of MWA Data and Internally Consistent Data for Exit Total Temperature at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 1B

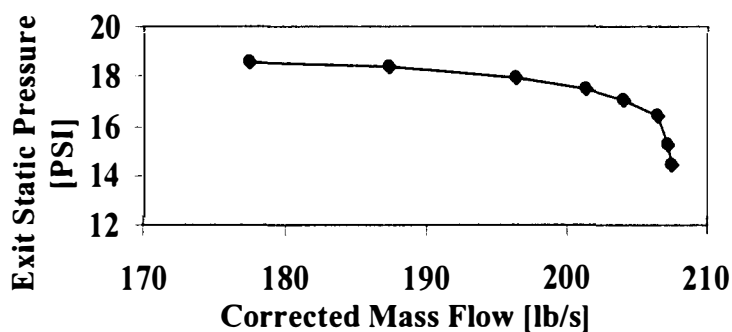


Figure D.5. Comparison of MWA Data and Internally Consistent Data for Exit Static Pressure at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 1B

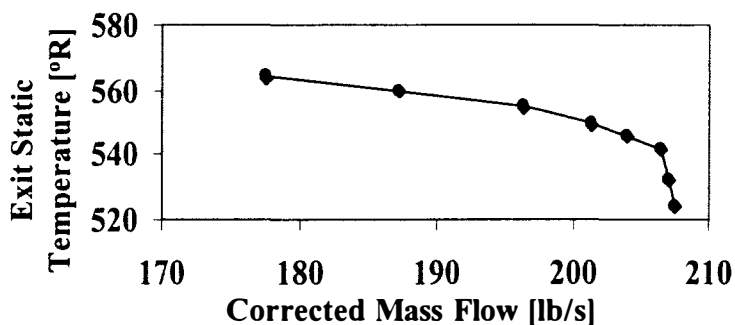


Figure D.6. Comparison of MWA Data and Internally Consistent Data for Exit Static Temperature at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 1B

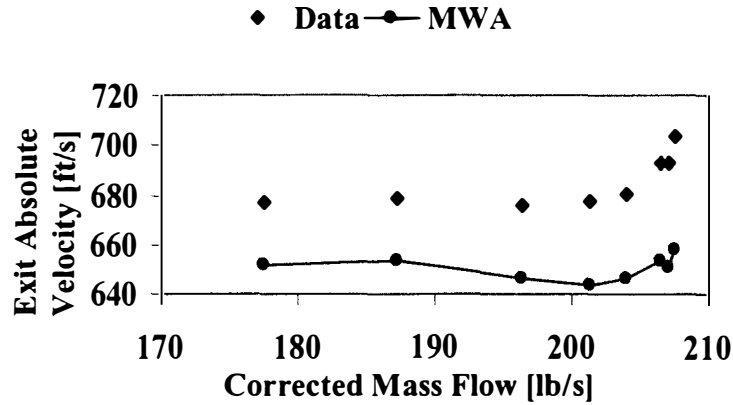


Figure D.7. Comparison of MWA Data and Internally Consistent Data for Exit Absolute Velocity at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 1B

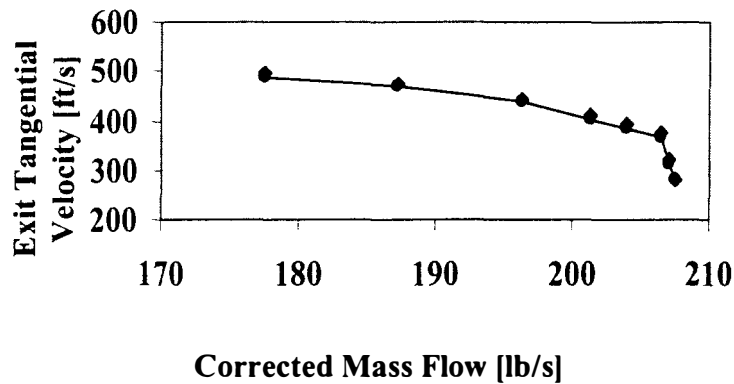


Figure D.8. Comparison of MWA Data and Internally Consistent Data for Exit Tangential Velocity at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 1B

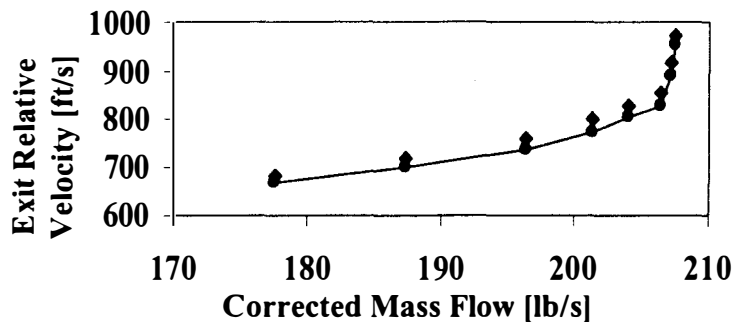


Figure D.9. Comparison of MWA Data and Internally Consistent Data for Exit Relative Velocity at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 1B

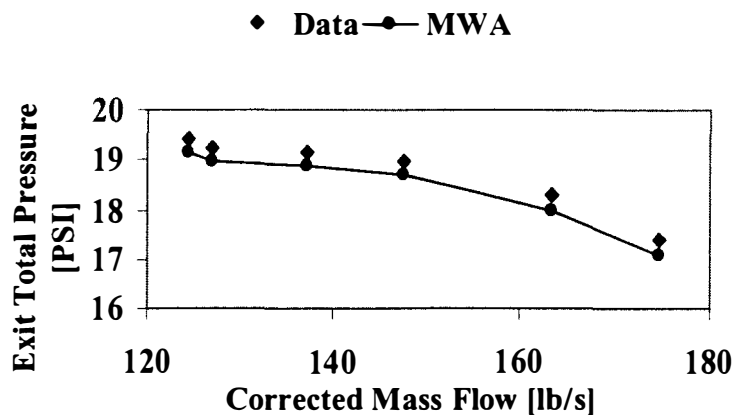


Figure D.10. Comparison of MWA Data and Internally Consistent Data for Exit Total Pressure at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 1B

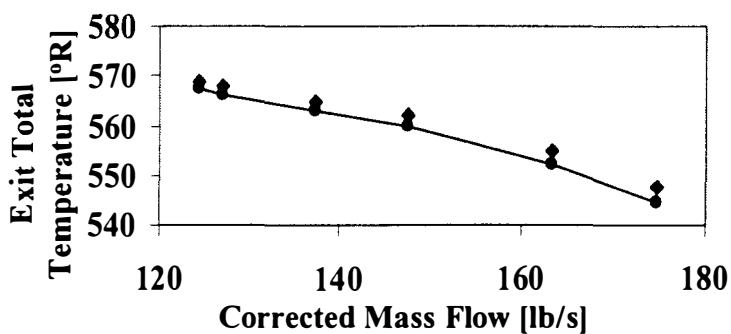


Figure D.11. Comparison of MWA Data and Internally Consistent Data for Exit Total Temperature at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 1B

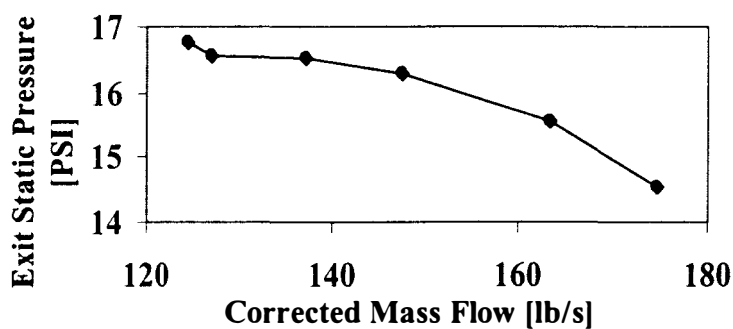


Figure D.12. Comparison of MWA Data and Internally Consistent Data for Exit Static Pressure at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 1B

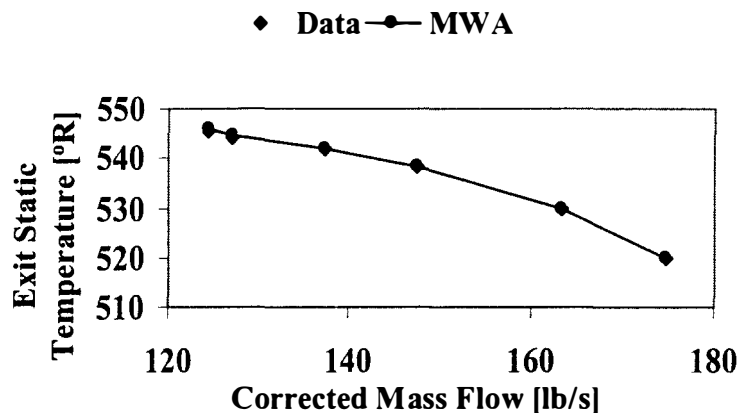


Figure D.13. Comparison of MWA Data and Internally Consistent Data for Exit Static Temperature at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 1B

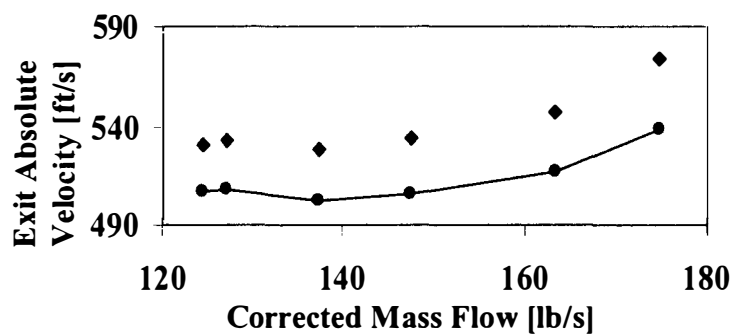


Figure D.14. Comparison of MWA Data and Internally Consistent Data for Exit Absolute Velocity at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 1B

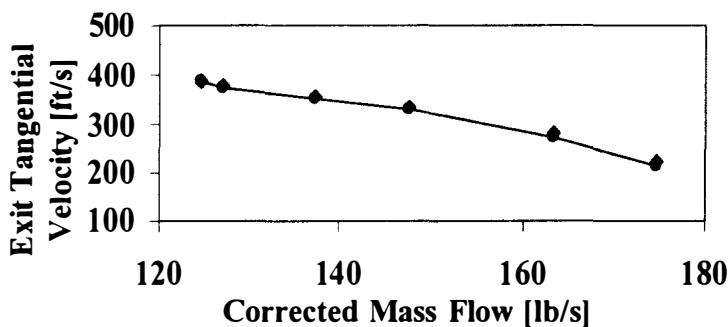


Figure D.15. Comparison of MWA Data and Internally Consistent Data for Exit Tangential Velocity at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 1B

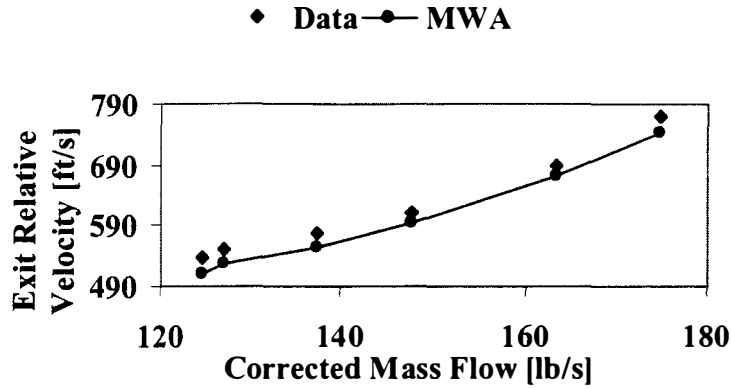


Figure D.16. Comparison of MWA Data and Internally Consistent Data for Exit Relative Velocity at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 1B

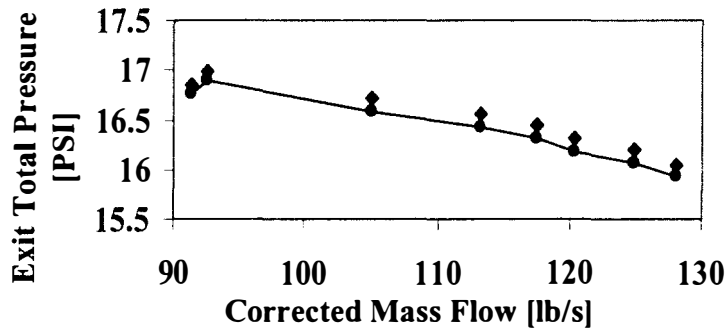


Figure D.17. Comparison of MWA Data and Internally Consistent Data for Exit Total Pressure at Different Corrected Mass Flows of the 50% Design Corrected Speed Line for Rotor 1B

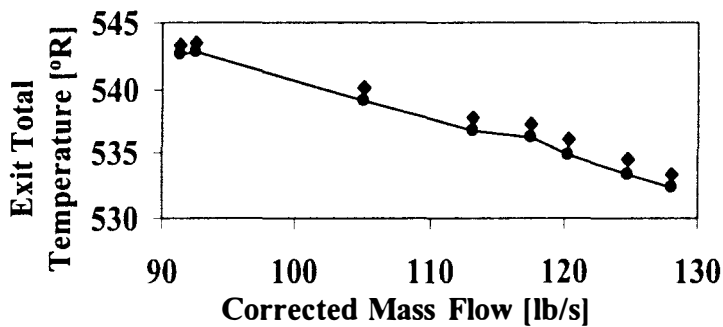


Figure D.18. Comparison of MWA Data and Internally Consistent Data for Exit Total Temperature at Different Corrected Mass Flows of the 50% Design Corrected Speed Line for Rotor 1B

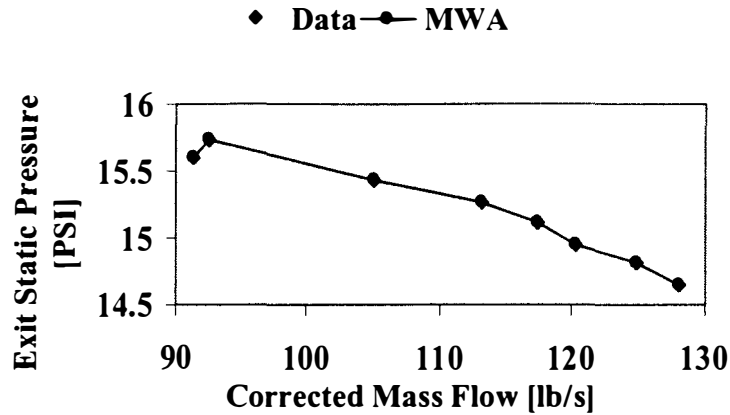


Figure D.19. Comparison of MWA Data and Internally Consistent Data for Exit Static Pressure at Different Corrected Mass Flows of the 50% Design Corrected Speed Line for Rotor 1B

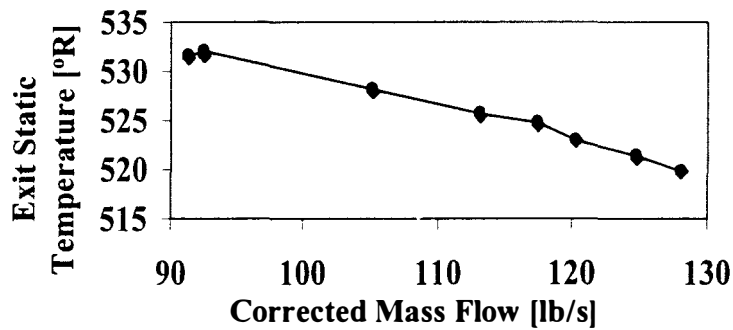


Figure D.20. Comparison of MWA Data and Internally Consistent Data for Exit Static Temperature at Different Corrected Mass Flows of the 50% Design Corrected Speed Line for Rotor 1B

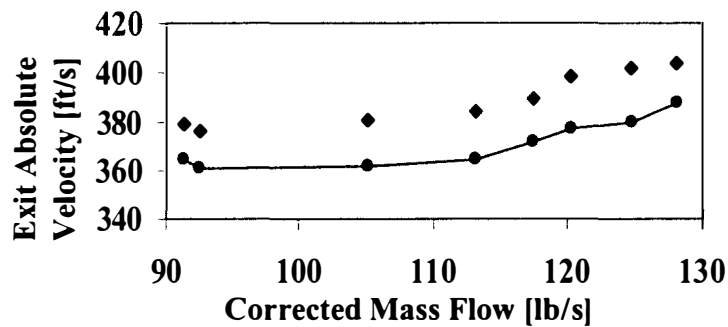


Figure D.21. Comparison of MWA Data and Internally Consistent Data for Exit Absolute Velocity at Different Corrected Mass Flows of the 50% Design Corrected Speed Line for Rotor 1B

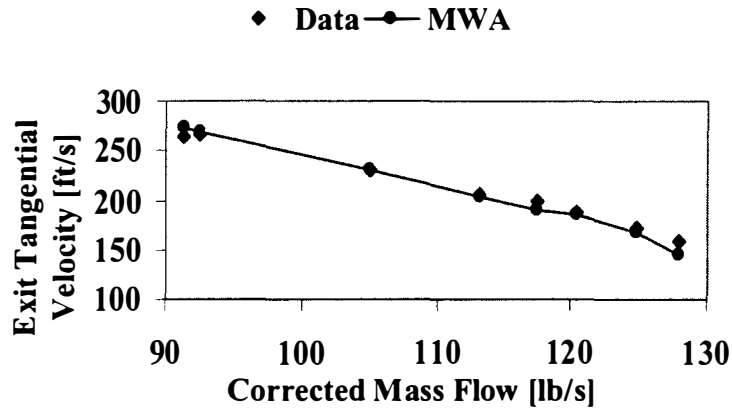


Figure D.22. Comparison of MWA Data and Internally Consistent Data for Exit Tangential Velocity at Different Corrected Mass Flows of the 50% Design Corrected Speed Line for Rotor 1B

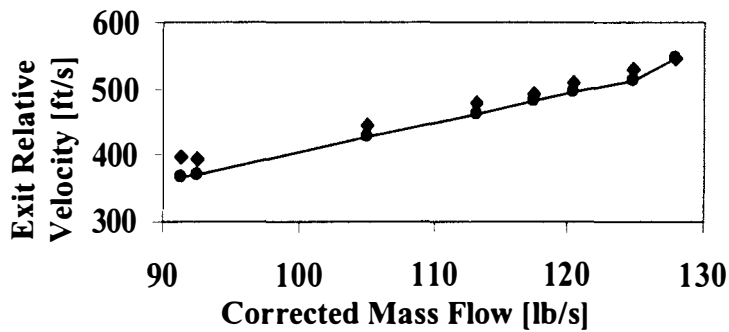


Figure D.23. Comparison of MWA Data and Internally Consistent Data for Exit Relative Velocity at Different Corrected Mass Flows of the 50% Design Corrected Speed Line for Rotor 1B

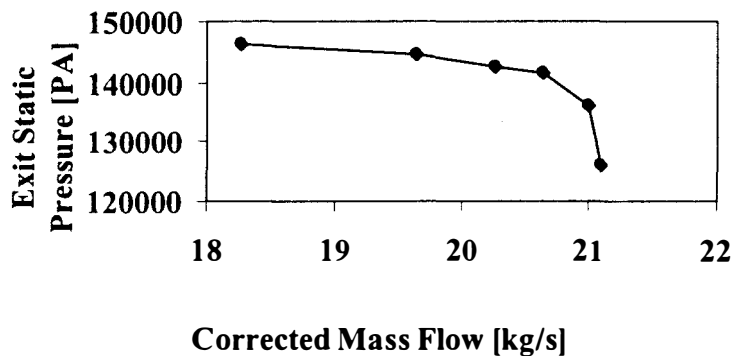


Figure D.24. Comparison of MWA Data and Internally Consistent Data for Exit Static Pressure at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 35

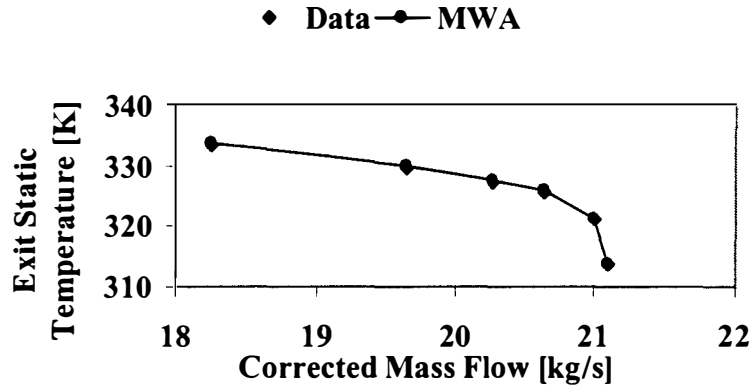


Figure D.25. Comparison of MWA Data and Internally Consistent Data for Exit Static Temperature at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Rotor 35

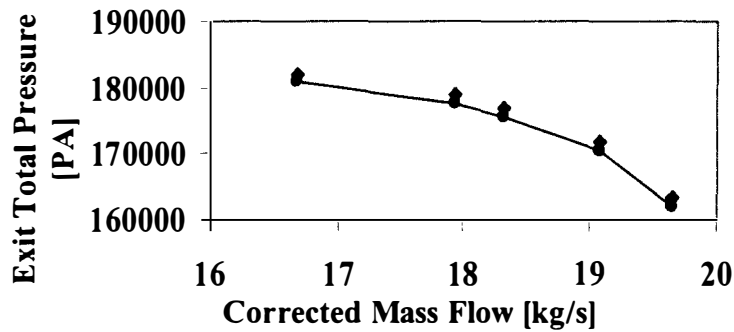


Figure D.26. Comparison of MWA Data and Internally Consistent Data for Exit Total Pressure at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 35

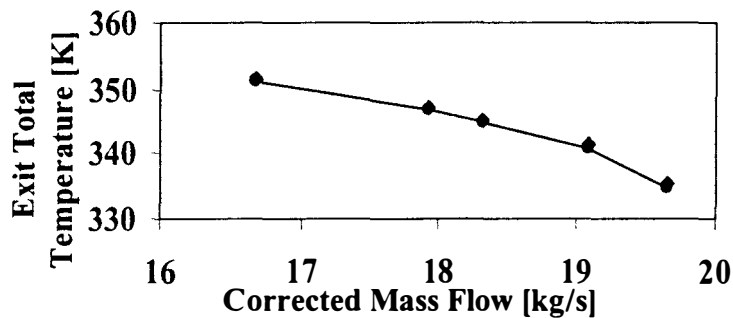


Figure D.27. Comparison of MWA Data and Internally Consistent Data for Exit Total Temperature at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 35

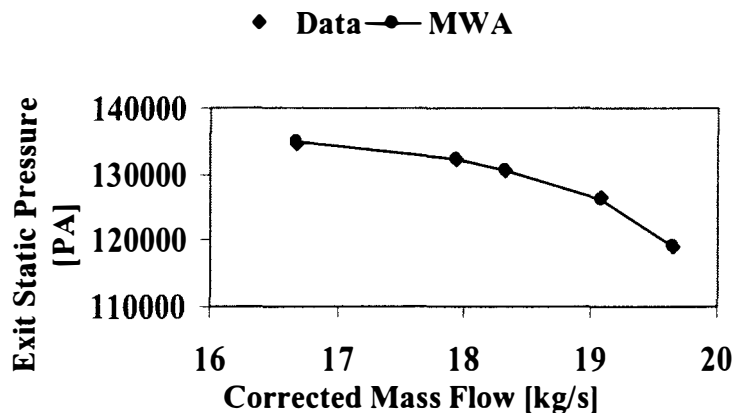


Figure D.28. Comparison of MWA Data and Internally Consistent Data for Exit Static Pressure at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 35

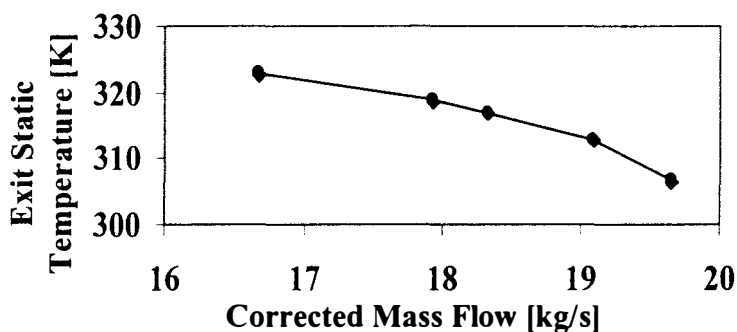


Figure D.29. Comparison of MWA Data and Internally Consistent Data for Exit Static Temperature at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 35

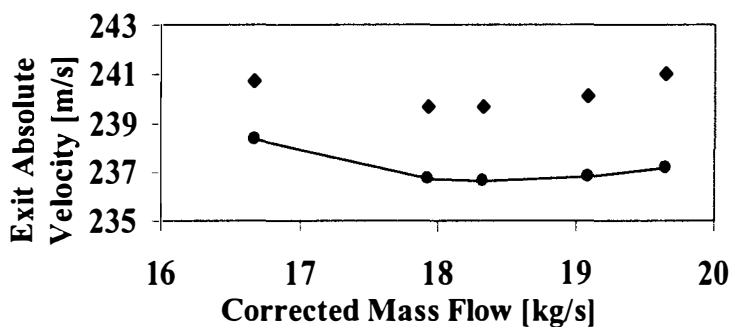


Figure D.30. Comparison of MWA Data and Internally Consistent Data for Exit Absolute Velocity at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 35

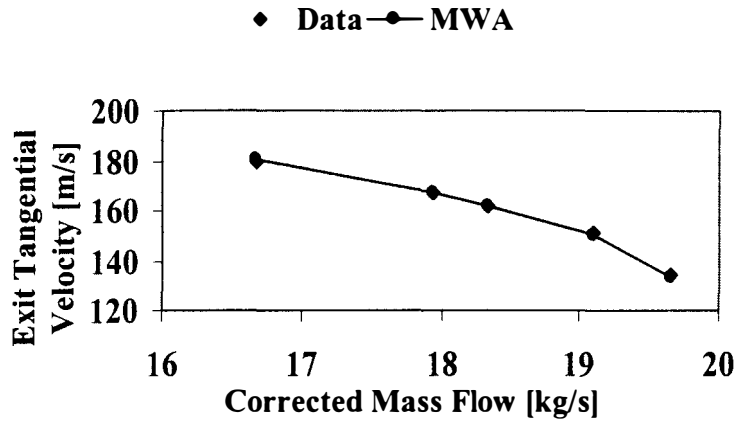


Figure D.31. Comparison of MWA Data and Internally Consistent Data for Exit Tangential Velocity at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 35

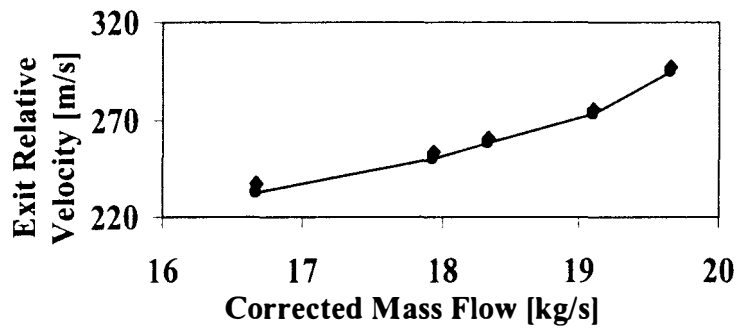


Figure D.32. Comparison of MWA Data and Internally Consistent Data for Exit Relative Velocity at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Rotor 35

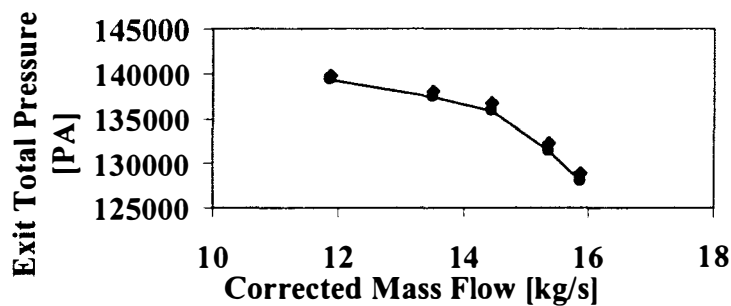


Figure D.33. Comparison of MWA Data and Internally Consistent Data for Exit Total Pressure at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 35

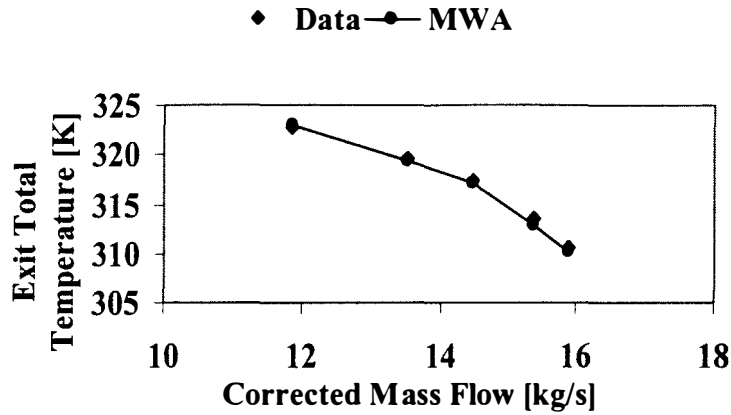


Figure D.34. Comparison of MWA Data and Internally Consistent Data for Exit Total Temperature at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 35

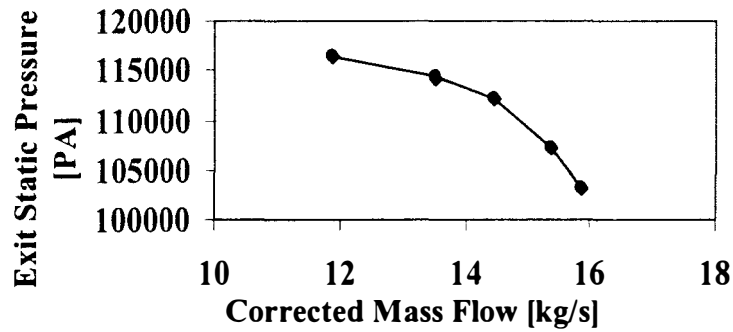


Figure D.35. Comparison of MWA Data and Internally Consistent Data for Exit Static Pressure at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 35

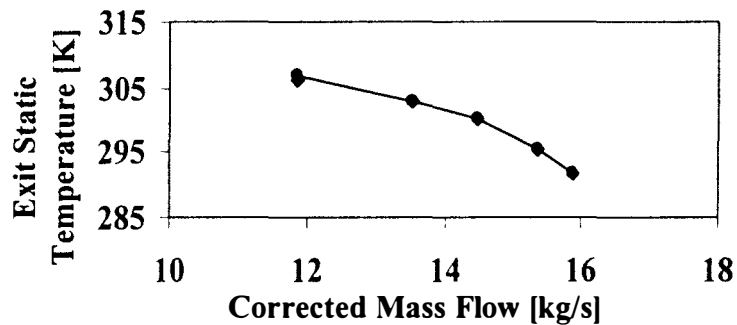


Figure D.36. Comparison of MWA Data and Internally Consistent Data for Exit Static Temperature at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 35

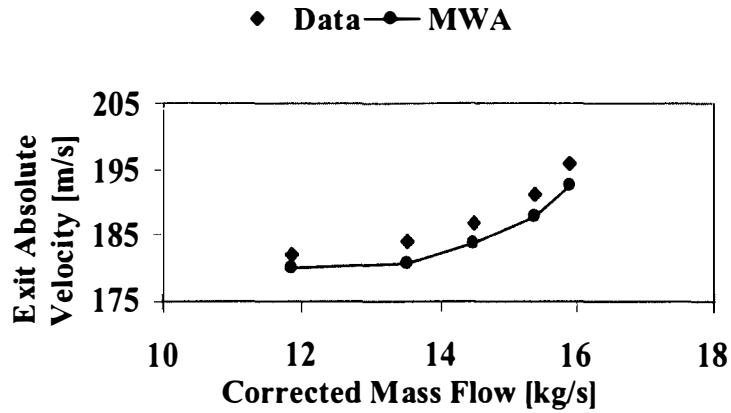


Figure D.37. Comparison of MWA Data and Internally Consistent Data for Exit Absolute Velocity at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 35

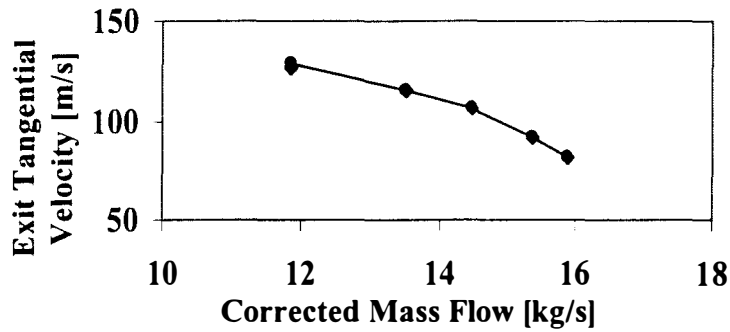


Figure D.38. Comparison of MWA Data and Internally Consistent Data for Exit Tangential Velocity at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 35

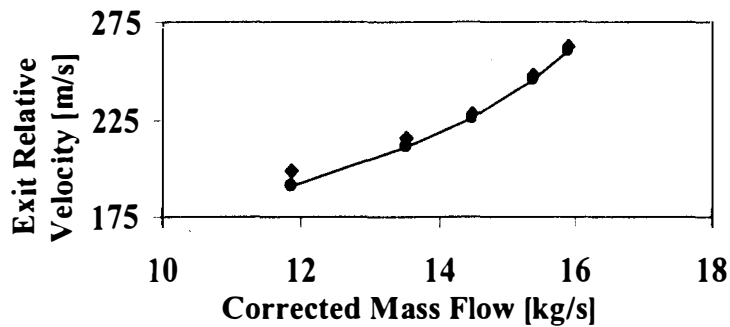


Figure D.39. Comparison of MWA Data and Internally Consistent Data for Exit Relative Velocity at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Rotor 35

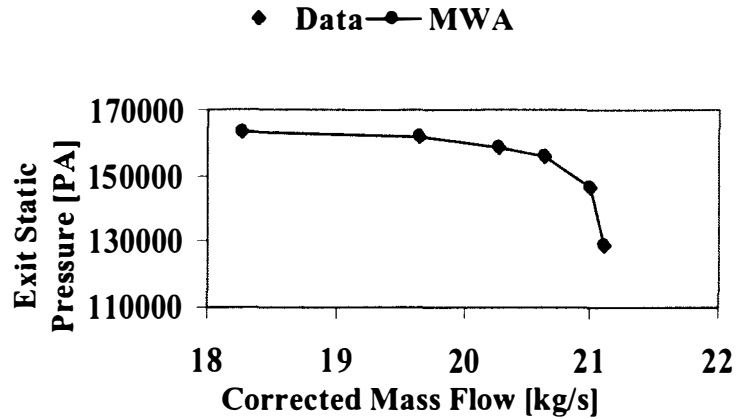


Figure D.40. Comparison of MWA Data and Internally Consistent Data for Exit Static Pressure at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Stator 35

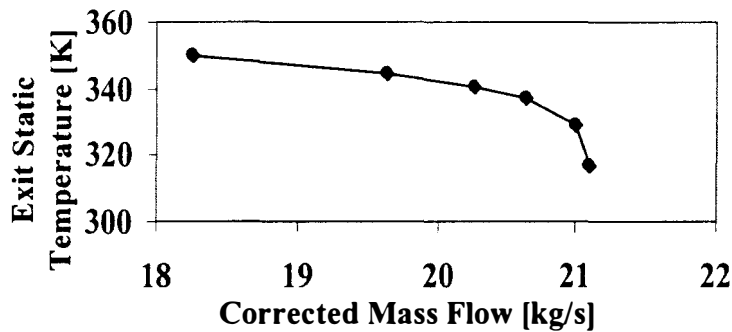


Figure D.41. Comparison of MWA Data and Internally Consistent Data for Exit Static Temperature at Different Corrected Mass Flows of the 100% Design Corrected Speed Line for Stator 35

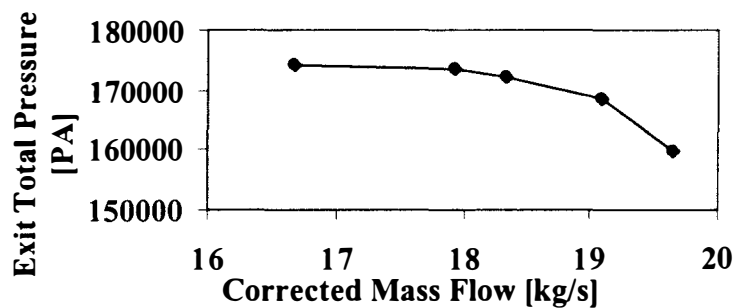


Figure D.42. Comparison of MWA Data and Internally Consistent Data for Exit Total Pressure at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Stator 35

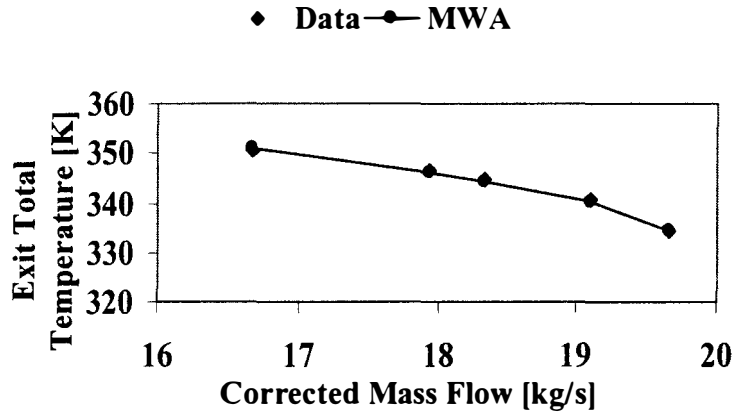


Figure D.43. Comparison of MWA Data and Internally Consistent Data for Exit Total Temperature at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Stator 35

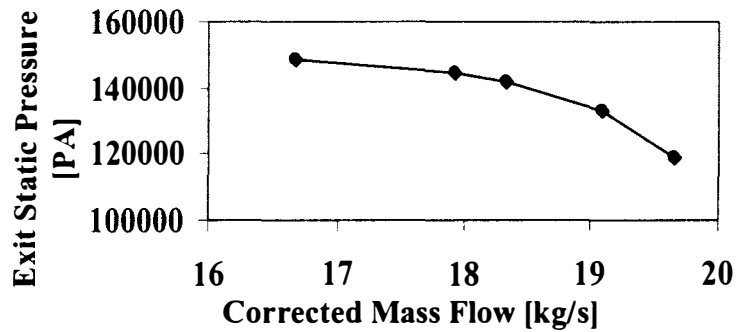


Figure D.44. Comparison of MWA Data and Internally Consistent Data for Exit Static Pressure at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Stator 35

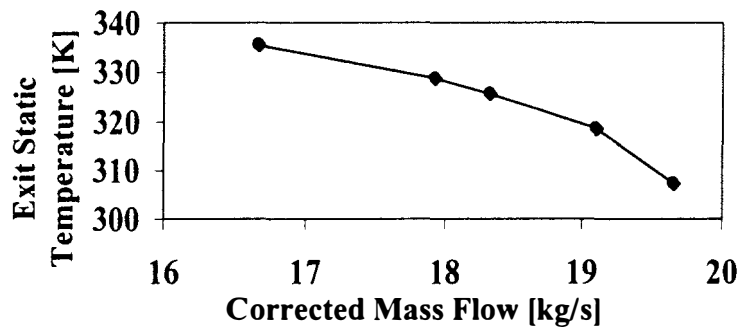


Figure D.45. Comparison of MWA Data and Internally Consistent Data for Exit Static Temperature at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Stator 35

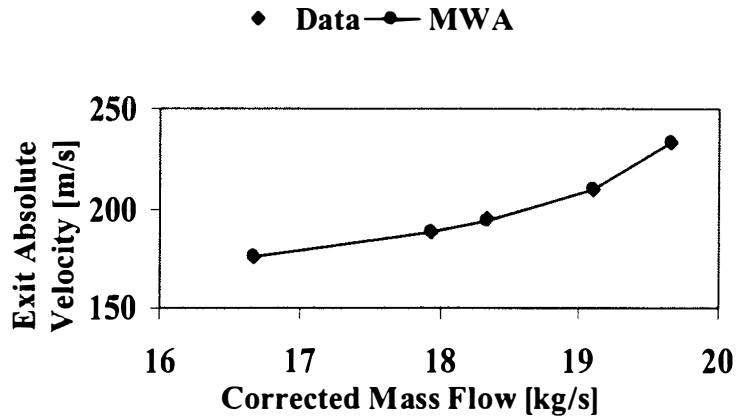


Figure D.46. Comparison of MWA Data and Internally Consistent Data for Exit Absolute Velocity at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Stator 35

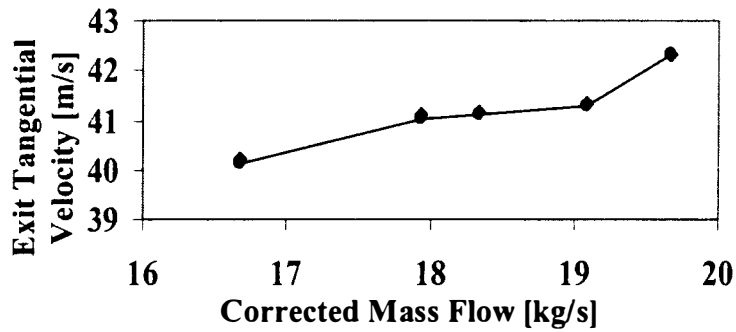


Figure D.47. Comparison of MWA Data and Internally Consistent Data for Exit Tangential Velocity at Different Corrected Mass Flows of the 90% Design Corrected Speed Line for Stator 35

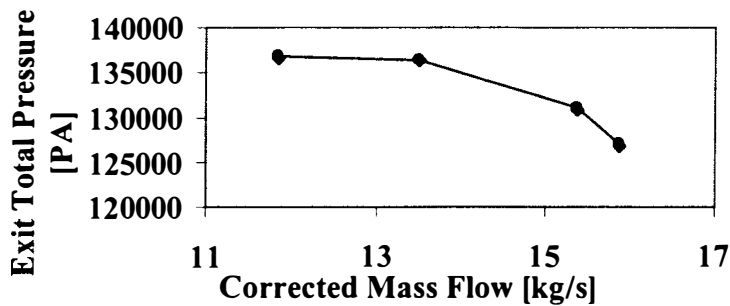


Figure D.48. Comparison of MWA Data and Internally Consistent Data for Exit Total Pressure at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Stator 35

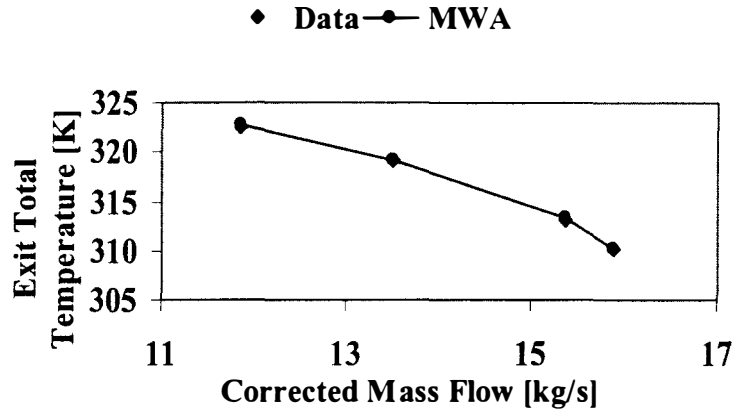


Figure D.49. Comparison of MWA Data and Internally Consistent Data for Exit Total Temperature at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Stator 35

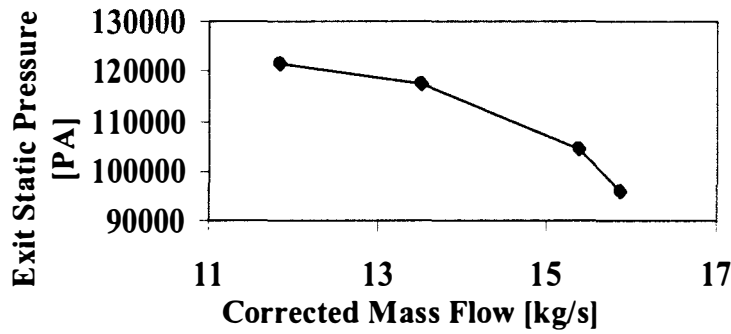


Figure D.50. Comparison of MWA Data and Internally Consistent Data for Exit Static Pressure at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Stator 35

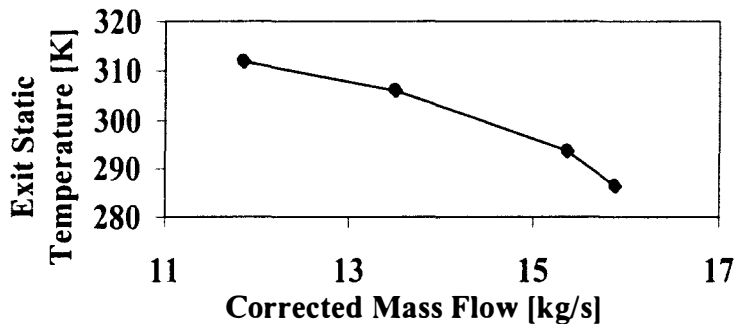


Figure D.51. Comparison of MWA Data and Internally Consistent Data for Exit Static Temperature at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Stator 35

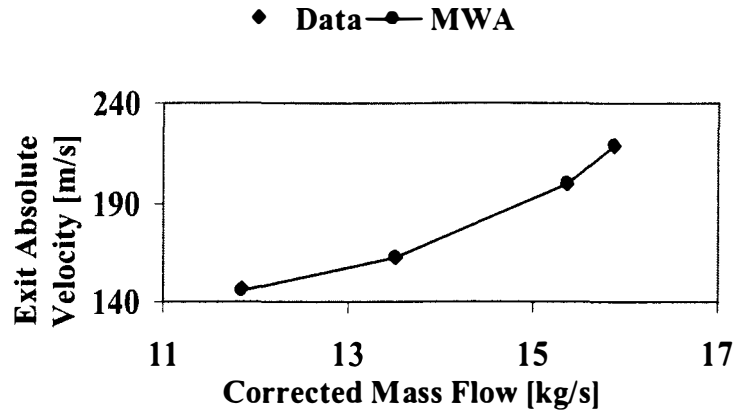


Figure D.52. Comparison of MWA Data and Internally Consistent Data for Exit Absolute Velocity at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Stator 35

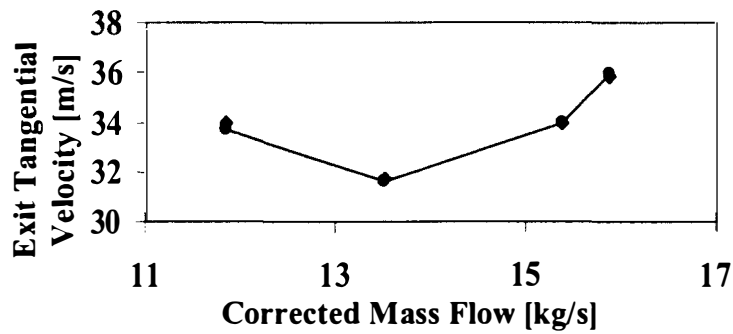


Figure D.53. Comparison of MWA Data and Internally Consistent Data for Exit Tangential Velocity at Different Corrected Mass Flows of the 70% Design Corrected Speed Line for Stator 35

APPENDIX E

DERIVATION OF EQUATIONS

APPENDIX E – DERIVATION OF EQUATIONS

This appendix discusses the derivation of equations. First, equation [3.16] will be derived, then equation [3.24].

E.1 Derivation of equation [3.16]

If a compressor or pump has steady flow,

$$\tau = \dot{m}(r_2 V_{t2} - r_1 V_{t1}) \quad [\text{E.1}]$$

(Mattingly, 1996), where τ is the applied torque. Assuming that all torques exerted on the fluid are associated with work done on the fluid, the power input is equal to:

$$\dot{W} = \omega \tau = \omega \dot{m}(r_2 V_{t2} - r_1 V_{t1}) = \dot{m}(U_2 V_{t2} - U_1 V_{t1}), \quad [\text{E.2}]$$

which is the Euler pump equation. Using the velocity triangles in Figure 3.2,

$$W^2 = W_t^2 + W_x^2, \quad [\text{E.3}]$$

$$V^2 = V_t^2 + V_x^2, \text{ and} \quad [\text{E.4}]$$

$$W_x^2 = V_x^2. \quad [\text{E.5}]$$

Solving equation [E.3] for W_x^2 and equation [E.4] for V_x^2 yields:

$$W_x^2 = W^2 - W_t^2 \text{ and} \quad [\text{E.6}]$$

$$V_x^2 = V^2 - V_t^2. \quad [\text{E.7}]$$

Equating equations [E.6] and [E.7] gives:

$$W^2 - W_t^2 = V^2 - V_t^2. \quad [\text{E.8}]$$

From the velocity triangles in Figure 3.2,

$$U = W_t + V_t \text{ or} \quad [\text{E.9}]$$

$$W_t = U - V_t. \quad [E.10]$$

Substituting equation [E.10] into [E.8] yields:

$$W^2 - (U - V_t)^2 = V^2 - V_t^2 \text{ or } W^2 - U^2 + 2UV_t - V_t^2 = V^2 - V_t^2. \quad [E.11]$$

Solving for UV_t yields:

$$UV_t = \frac{(V^2 - W^2 + U^2)}{2}. \quad [E.12]$$

Substituting equation [E.12] into equation [E.2] gives:

$$\dot{W} = \dot{m} \left(\frac{(V_2^2 - W_2^2 + U_2^2)}{2} - \frac{(V_1^2 - W_1^2 + U_1^2)}{2} \right). \quad [E.13]$$

Dividing both sides by mass flow gives:

$$\text{work} = \frac{1}{2} \left[(U_2^2 - U_1^2) - (W_2^2 - W_1^2) + (V_2^2 - V_1^2) \right]. \quad [3.16]$$

E.2 Derivation of equation [3.24]

A T-S diagram is shown in Figure E.1. In this figure, $(P_{t2r})_{id}$ represents the exit relative total pressure for an ideal, isentropic flow, and P_{t2r} represents the actual

exit relative total pressure. The ratio $\left(\frac{P_{t2r}}{P_{t1r}} \right)_{id}$ represents the ratio of the ideal exit

relative total pressure to the inlet relative total pressure. For isentropic conditions,

the ratio of relative total pressures and relative total temperatures is given by

$$\left(\frac{P_{t2r}}{P_{t1r}} \right)_{id} = \left(\frac{T_{t2r}}{T_{t1r}} \right)^{\frac{\gamma}{\gamma-1}}. \quad [3.24]$$

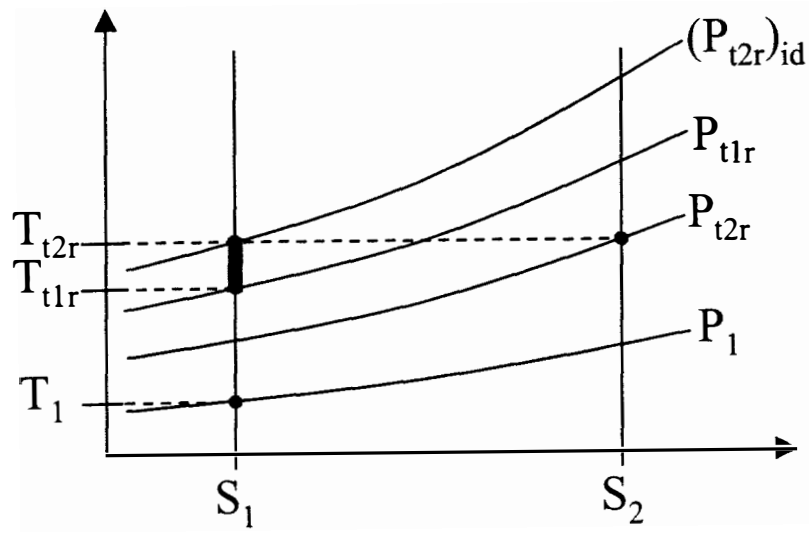


Figure E.1. T-S Diagram for Equation [3.24]

APPENDIX F

DISCUSSION OF INDIVIDUAL BLADE ROW RESULTS OF MLC WITH LOSS AND DEVIATION Correlations

APPENDIX F – DISCUSSION OF INDIVIDUAL BLADE ROW RESULTS OF THE MLC WITH LOSS AND DEVIATION CORRELATIONS

The correlations discussed in Section 6.1, with the add-loss and add-deviation corrections for each speed line, were used in modeling NASA Rotor 1B and NASA Stage 35. Performance characteristics produced by the MLC were discussed in Section 6.3. Now, the results of other flow variables for each blade row and machine for the normal operating range will be discussed. The stalled and choked conditions for both Rotor 1B and Stage 35 are addressed in Section F.3.

F.1 NASA Rotor 1B

A comparison of the data and the MLC results using the loss and deviation correlations for Rotor 1B is shown in Figures F.1 – F.8. Table 6.1 shows the add-loss and add-deviation corrections used for each speed line of Rotor 1B. Excluding the choked and stalled conditions for Rotor 1B, the highest differences in exit total pressure and exit total temperature are 4.89-percent and 1.19-percent, respectively. Similarly, the exit static pressure and temperature have maximum differences of 5.11-percent and 1.25-percent, respectively. For exit absolute velocity, tangential velocity, and relative velocity, the maximum percent differences are 2.89-percent, 8.67-percent, and 5.31-percent.

In general, there was a higher percent difference in the velocities than the other variables. The greatest percent difference was in the tangential velocity. This indicates that the MLC has difficulty modeling the turning of the flow. The static pressure and temperatures are closer to the data. The total pressure and temperature,

which are the quantities needed to obtain the required compressor characteristics, are within 4.89-percent difference.

In general, the exit relative total pressures calculated by the MLC were higher than that of the data for the normal operating range. This indicates that the losses calculated by the MLC were lower than that of the data. The tangential velocity was under-predicted for the lower mass flows, meaning there was more flow turning than that for which the MLC accounted. It was over-predicted for the higher mass flows, meaning the MLC accounted for more flow turning than was actually present.

F.2 NASA Stage 35

A comparison of the data and the MLC results using the loss and deviation correlations for Stage 35 is shown in Figures F.9 – F.28. Tables 6.2 and 6.3 show the add-loss and add-deviation corrections used for each speed line of Rotor 35 and Stator 35, respectively. First the results of the rotor and stator of Stage 35 will be discussed separately, then the collective modeling (Rotor-NBR-Stator) of Stage 35 will be discussed. The NBR is calculated as was discussed in section 5.3.2.2, and will, therefore, not be re-discussed here.

Figures F.9 - F.16 show the comparisons for the rotor of Stage 35. Excluding the choked and stalled conditions for Rotor 35, the highest difference in exit total pressure and temperature and exit static pressure and temperature are 3.21-percent, 0.68-percent, 3.26-percent, and 0.70-percent, respectively. For exit

absolute velocity, tangential velocity, and relative velocity, the maximum percent differences are of 1.42-percent, 3.83-percent, and 2.82-percent, respectively.

As was the case of Rotor 1B, the greatest percent differences for Rotor 35 were in the tangential velocities. In general, the tangential velocity was under-predicted for the lower mass flows and over-predicted for the higher mass flows, as it was for Rotor 1B. However, the magnitude of the differences in the tangential velocities calculated by the MLC and data was much less than that of Rotor 1B. This is probably because there is more flow turning in a rotor fan than there is in a core compressor rotor. Therefore, the MLC has more difficulty modeling the flow turning of Rotor 1B. Unlike Rotor 1B, the exit relative total pressures calculated by the MLC were generally lower than data. This indicates that the MLC over-predicted the losses. Since the majority of the loss calculated was in the profile loss, it is assumed that this was a result of the difference between the blade profile that the loss correlations assumed and the actual blade profile of the machine.

A comparison of the data and the MLC results using the loss and deviation correlations for the stator of Stage 35 is shown in Figures F.17 – F.22. Excluding the choked and stalled conditions for Stator 35, the highest difference in exit total pressure is 0.60-percent, and the differences in exit total temperature are all around zero-percent. The highest differences for exit static pressure and temperature are 2.29-percent and 0.49-percent, respectively. For exit absolute velocity and tangential velocity, the maximum differences are 3.46-percent and 7.87-percent.

As in the results of the rotor, the greatest percent differences were in the tangential velocities. Unlike the rotor results, the tangential velocities were generally over-predicted, meaning that the MLC accounted for more flow turning than was actually present. The required quantities of total pressure and temperature were modeled very well (within 1-percent) for the normal operating range.

After the rotor and stator of Stage 35 were modeled separately, the stage was modeled collectively, including calculations of the space between blade rows. Comparison of data at the exit of the stator and the results calculated by the MLC using the loss and deviation correlations from Section 6.1 with add-loss and add-deviation corrections is shown in Figures F.23 – F.28. Excluding the choked and stalled conditions for Stage 35, the highest differences in exit total pressure and temperature are 1.96-percent and 0.35-percent, respectively. The highest differences for exit static pressure and temperature are 3.40-percent and 0.60-percent, respectively. For exit absolute velocity and tangential velocity, the maximum differences are 3.71-percent and 5.00-percent.

For the normal operating range, Stage 35 as a whole had a percent difference within 5.00-percent. The quantities needed to calculate the compressor characteristics, exit total pressure and temperature, were modeled nicely, with differences within 1.96-percent. The tangential velocities were over-predicted for lower mass flows and under-predicted for higher mass flows.

F.3 Discussion of Stalled and Choked Conditions

For Rotor 1B, the differences are generally higher at stalled conditions than at normal operating conditions. As mentioned in Section 6.3.2, a stalled condition occurs when there is poor flow quality caused by a large positive incidence, which causes a flow separation on the suction surface of the blade. This causes an increase in loss. The correlations do not accurately predict the profile at this condition. The correlations are limited in that they assume the blades to have double-circular-arc blade shape. This is a problem because the blades of Rotor 1B have a multiple-circular-arc blade shape for 40-percent of their span. This is why the tangential velocity was too low compared to data at stall. Another limitation is that the correlations do not calculate high enough shock losses for Rotor 1B. At the blade tip there are losses due to strong shocks, and at the hub there are losses due to severe turning of the flow. The correlations are applied at the mean flow radius with no knowledge of the strong shocks at the blade tips or severe flow turning near the hub. This means the correlations are estimating losses which are noticeably too low to adequately represent the entire annular cross section of the compressor for a given blade.

In addition, the differences between MLC results and data are much higher for choked conditions than for normal operating conditions. Section 6.3.1, a choked condition is caused by poor flow quality in which there is a large negative incidence, which causes a flow separation on the pressure surface of the blade. This also causes an increase in loss. The MLC's limitation at choke is due to a

combination of reasons. For one, the average cross-sectional area correction factor (ζ) is not representing the choked condition as well as the rest of the speed line. Another is the limitation of the loss and deviation correlations at choke. This is due to the previously stated differences in the blade profile assumed by the correlations and the actual blade profile of the machines modeled.

For Stage 35, both the losses at stall and at choke were generally over-predicted. As in Rotor 1B, one reason for the differences in loss is that the average cross-sectional area correction factor (ζ) is not representing the choked condition as well as the rest of the speed line. Another is, as was previously stated, the loss correlations assume the blades to have double-circular-arc blade shape. The blades of Stage 35 have a completely multiple-circular-arc blade profile. Since the greatest loss component was the profile loss, it is determined that the loss was over-predicted due to the differences in the blade profile assumed and the actual blade profile.

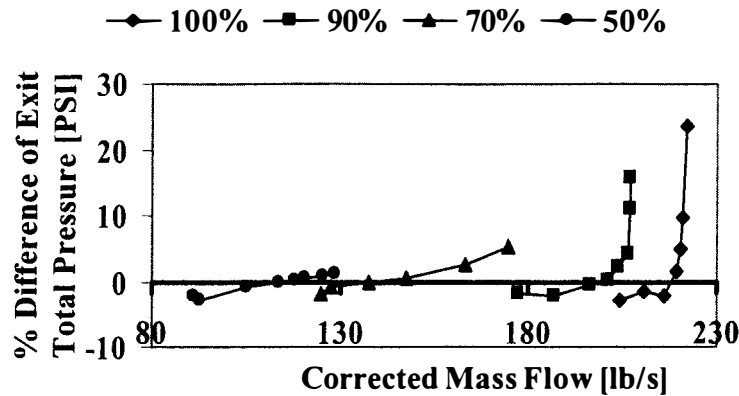


Figure F.1. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Total Pressure at Different Corrected Mass Flows and Different Speeds for Rotor 1B

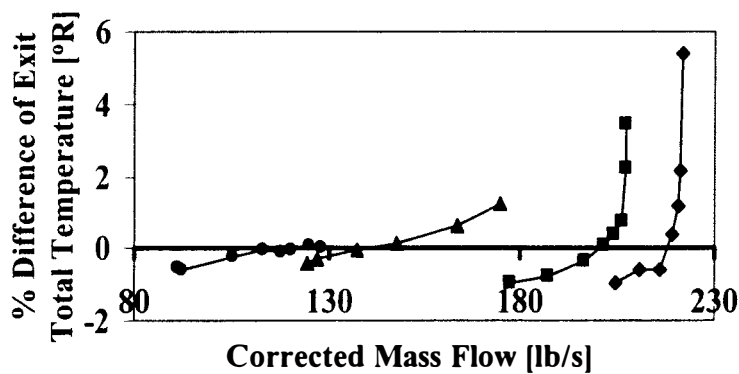


Figure F.2. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Total Temperature at Different Corrected Mass Flows and Different Speeds for Rotor 1B

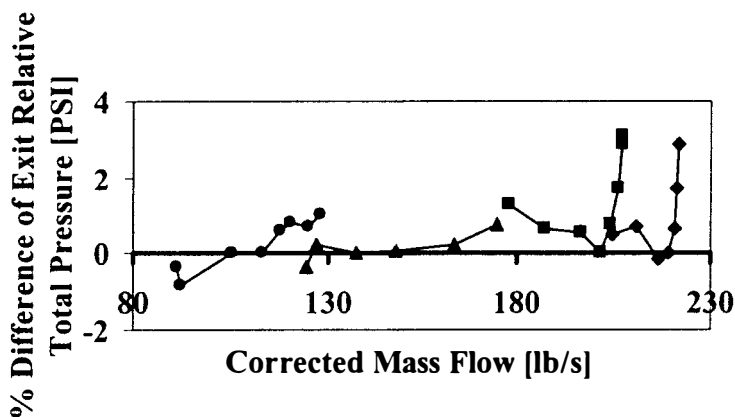


Figure F.3. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Relative Total Pressure at Different Corrected Mass Flows and Different Speeds for Rotor 1B

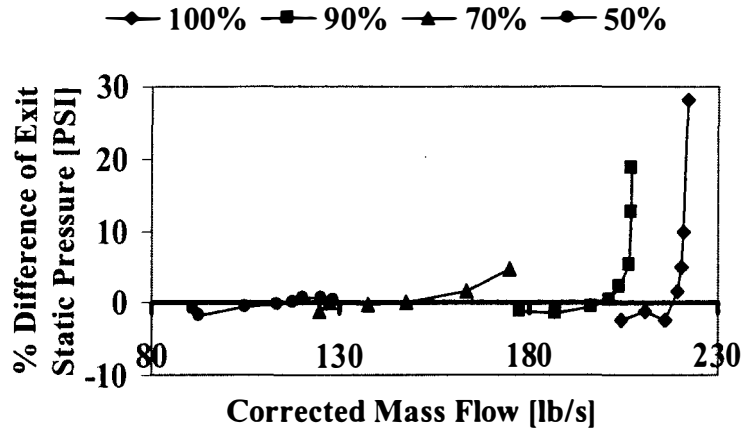


Figure F.4. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Static Pressure at Different Corrected Mass Flows and Different Speeds for Rotor 1B

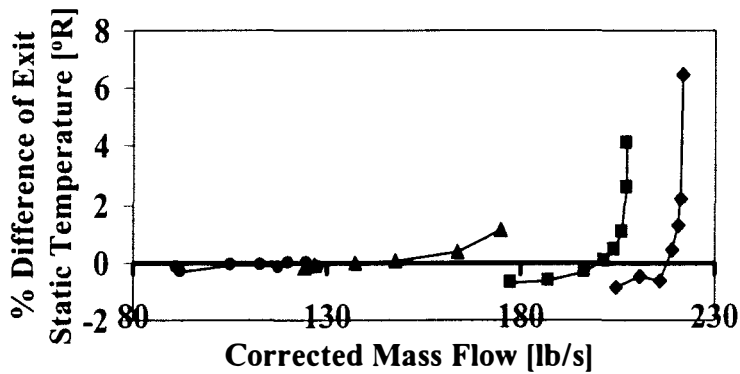


Figure F.5. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Static Temperature at Different Corrected Mass Flows and Different Speeds for Rotor 1B

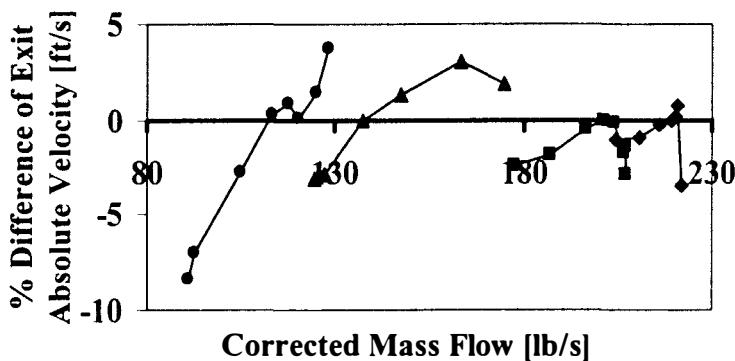


Figure F.6. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Absolute Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 1B

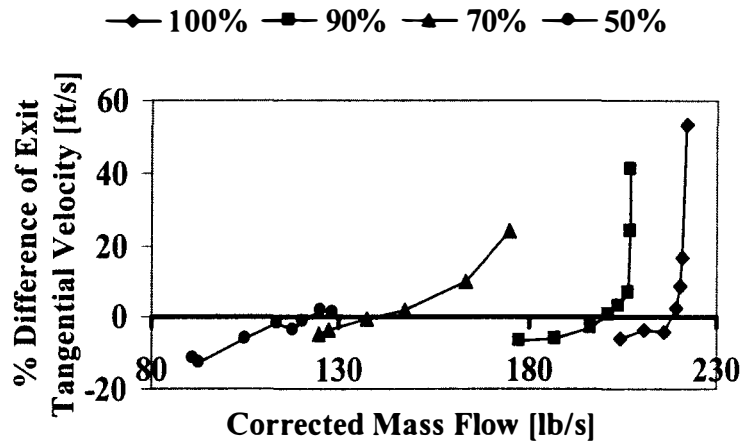


Figure F.7. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Tangential Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 1B

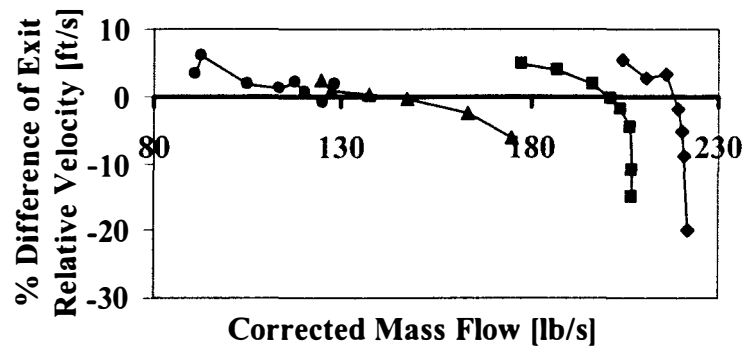


Figure F.8. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Relative Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 1B

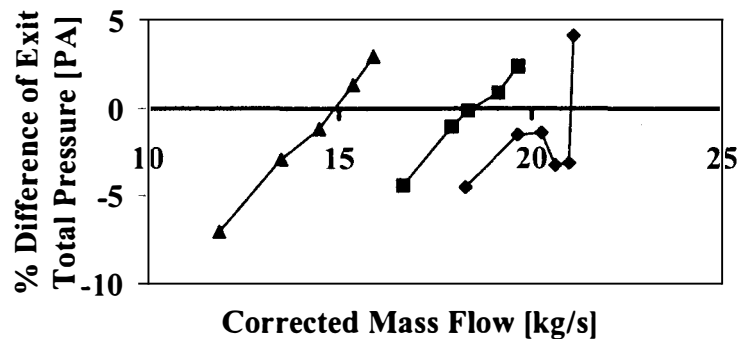


Figure F.9. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Total Pressure at Different Corrected Mass Flows and Different Speeds for Rotor 35

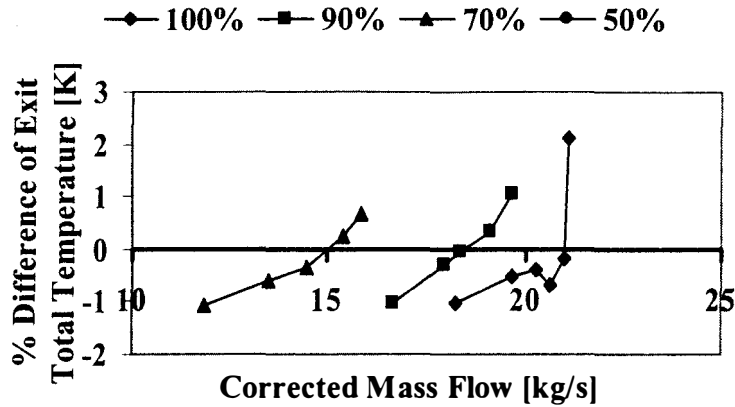


Figure F.10. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Total Temperature at Different Corrected Mass Flows and Different Speeds for Rotor 35

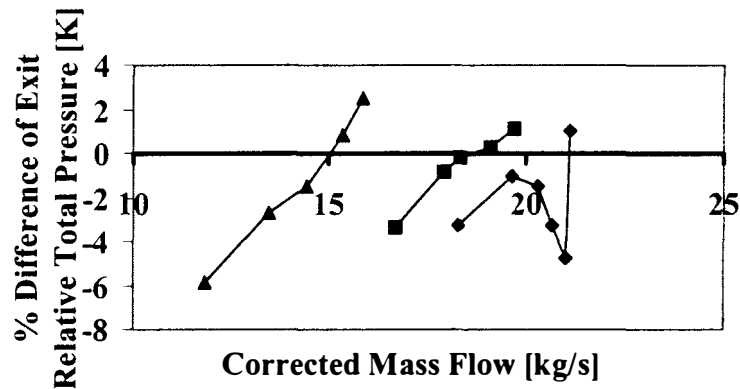


Figure F.11. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Relative Total Pressure at Different Corrected Mass Flows and Different Speeds for Rotor 35

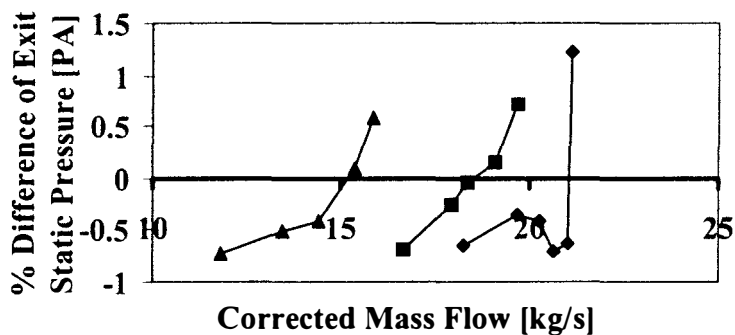


Figure F.12. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Static Pressure at Different Corrected Mass Flows and Different Speeds for Rotor 35

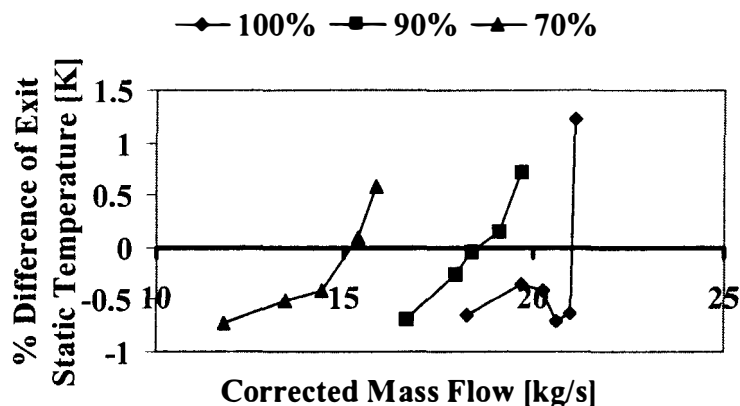


Figure F.13. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Static Temperature at Different Corrected Mass Flows and Different Speeds for Rotor 35

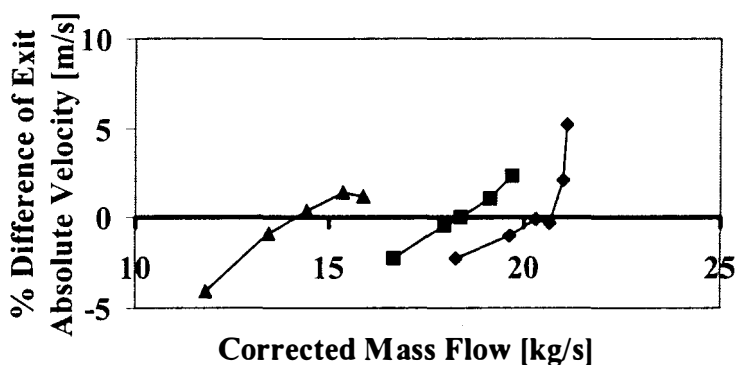


Figure F.14. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Absolute Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 35

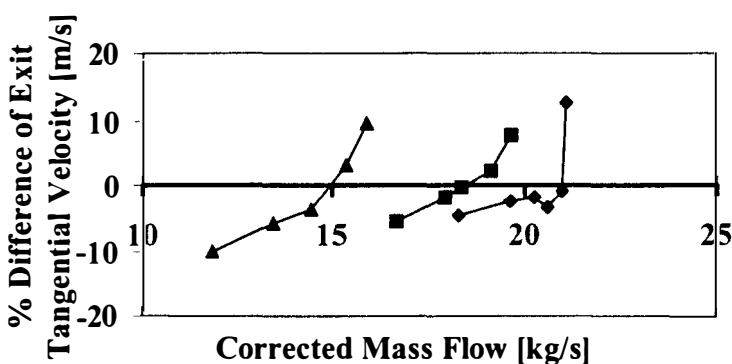


Figure F.15. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Tangential Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 35

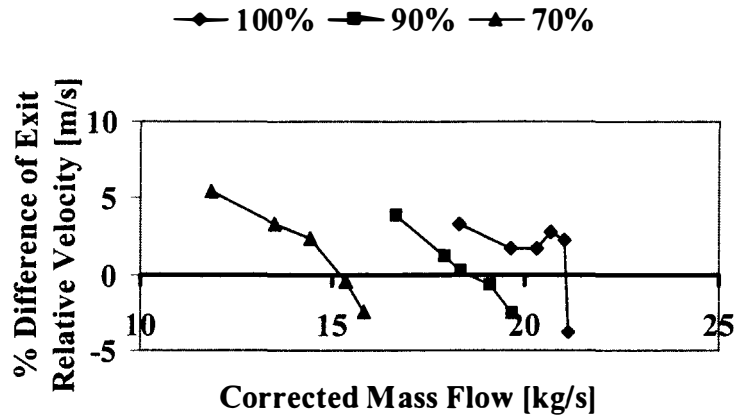


Figure F.16. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Relative Velocity at Different Corrected Mass Flows and Different Speeds for Rotor 35

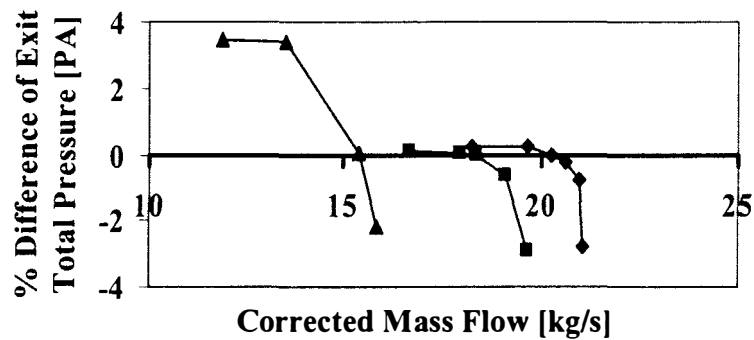


Figure F.17. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Total Pressure at Different Corrected Mass Flows and Different Speeds for Stator 35

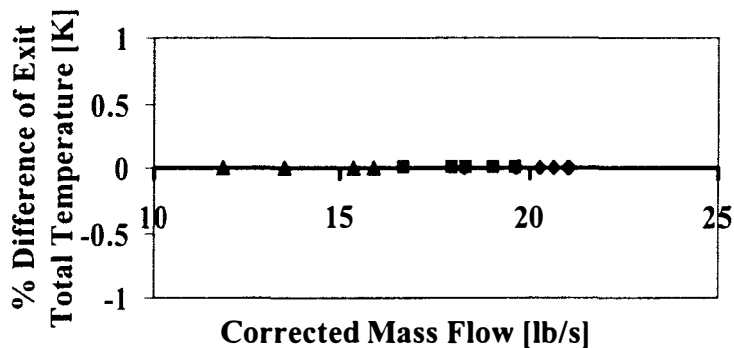


Figure F.18. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Total Temperature at Different Corrected Mass Flows and Different Speeds for Stator 35

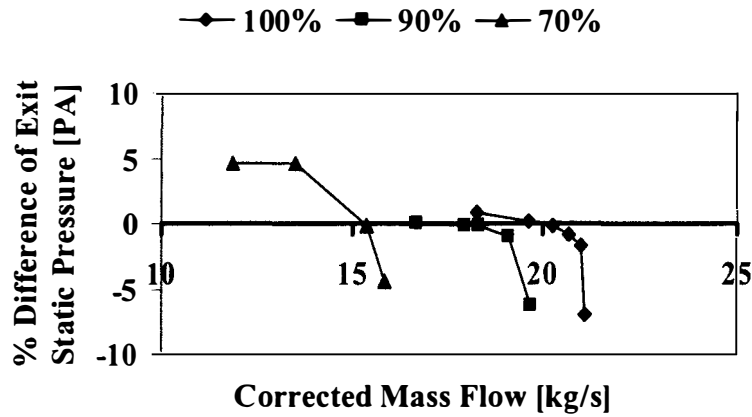


Figure F.19. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Static Pressure at Different Corrected Mass Flows and Different Speeds for Stator 35

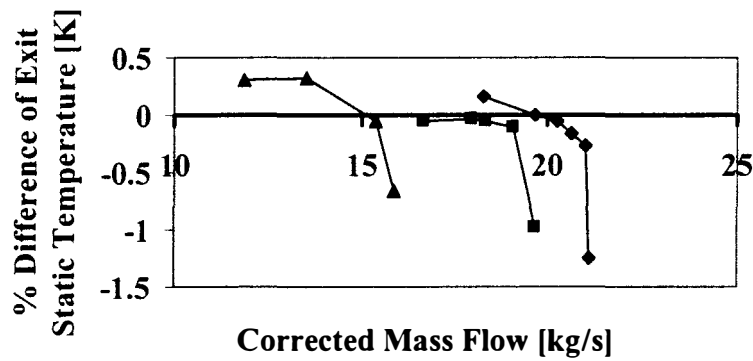


Figure F.20. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Static Temperature at Different Corrected Mass Flows and Different Speeds for Stator 35

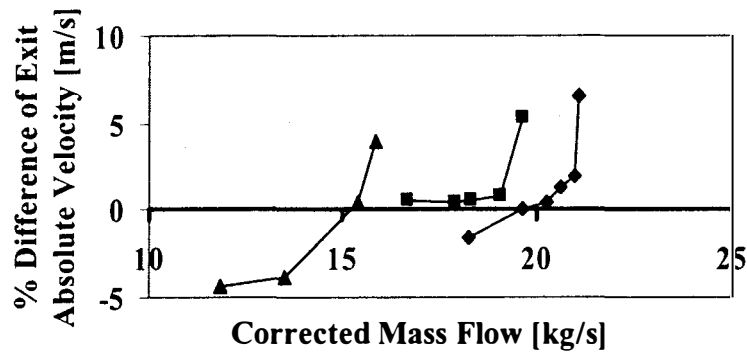


Figure F.21. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Absolute Velocity at Different Corrected Mass Flows and Different Speeds for Stator 35

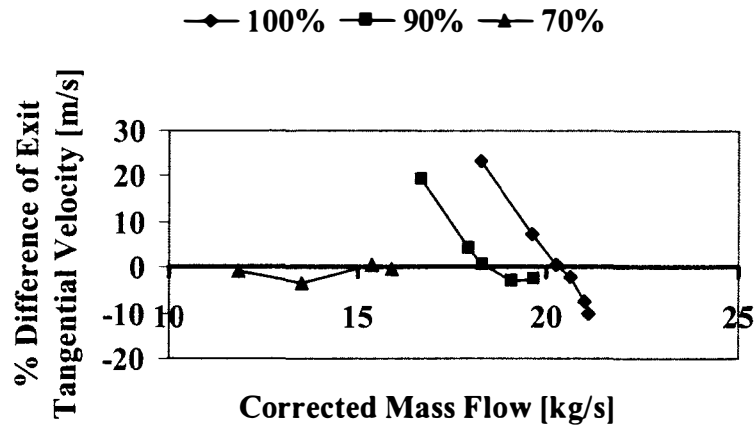


Figure F.22. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Tangential Velocity at Different Corrected Mass Flows and Different Speeds for Stator 35

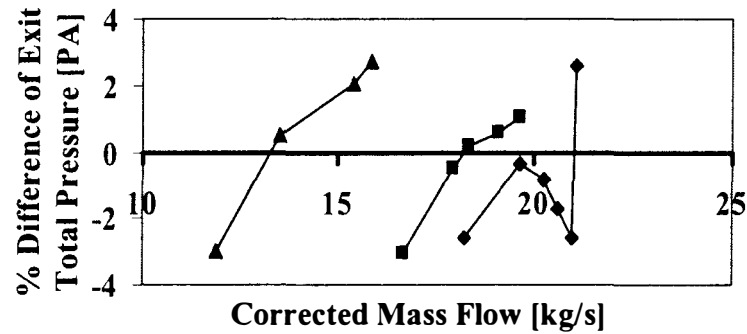


Figure F.23. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Total Pressure at Different Corrected Mass Flows and Different Speeds for Stage 35

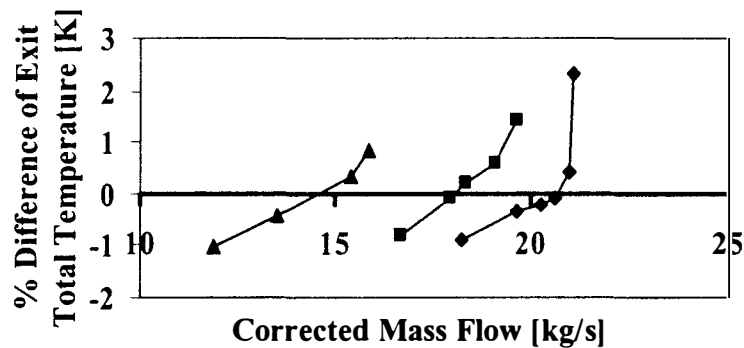


Figure F.24. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Total Temperature at Different Corrected Mass Flows and Different Speeds for Stage 35

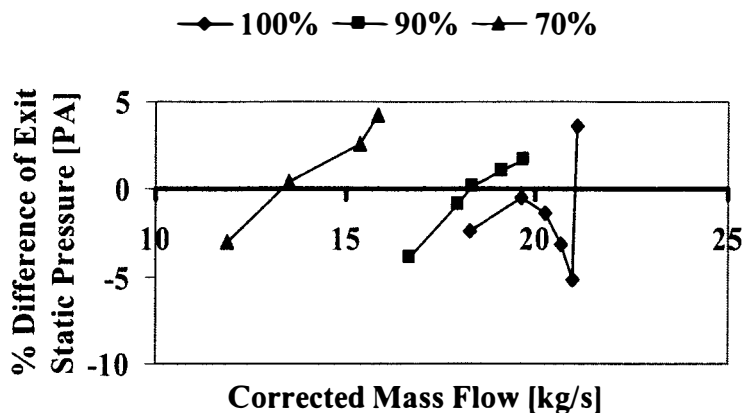


Figure F.25. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Static Pressure at Different Corrected Mass Flows and Different Speeds for Stage 35

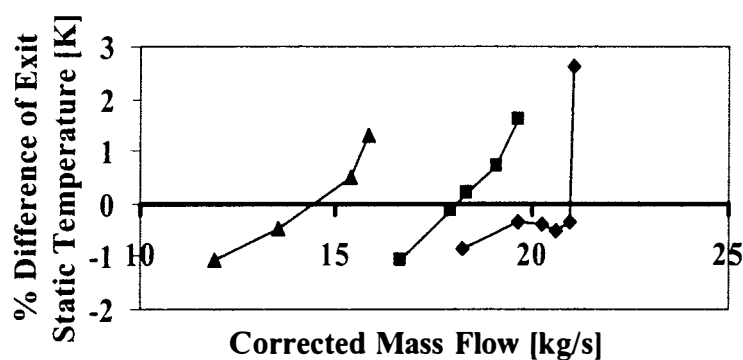


Figure F.26. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Static Temperature at Different Corrected Mass Flows and Different Speeds for Stage 35

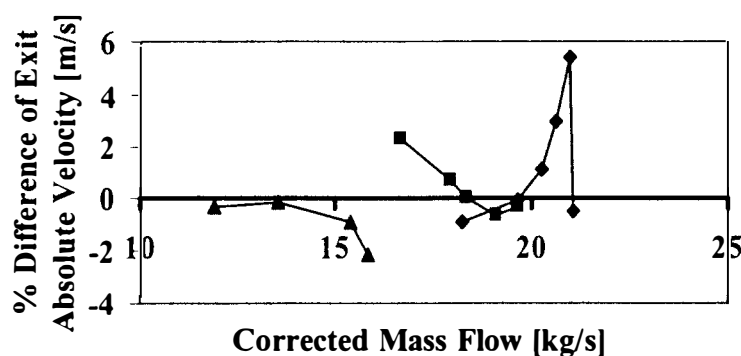


Figure F.27. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Absolute Velocity at Different Corrected Mass Flows and Different Speeds for Stage 35

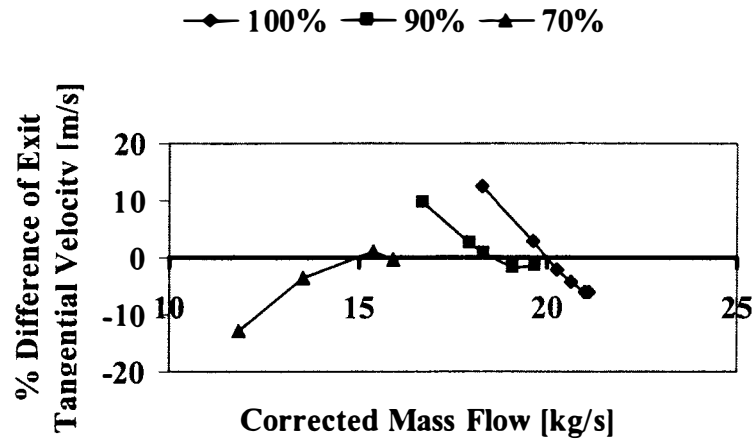


Figure F.28. Percent Difference of MLC Results with Loss and Deviation from Correlations for Exit Tangential Velocity at Different Corrected Mass Flows and Different Speeds for Stage 35

VITA

Sherri Lynette Holcomb Smith was born in Amory, MS in 1972 to Jimmy and Christine Holcomb. She grew up with her older sister, Patti, in the Salem Community, just outside of Fulton, MS. She attended Fairview Jr. High School and graduated from Itawamba Agricultural High School. After graduating high school, she attended Itawamba Community College, receiving an Associate of Arts degree in Engineering in 1993. In the Fall of 1993, she enrolled in the Mechanical Engineering Department at Mississippi State University. While at MSU, she participated in the co-op program for three semesters. On May 25, 1996, she married Scotty Lynn Smith. She graduated MSU in 1997 with a Bachelor of Science degree in Mechanical Engineering. In the Fall of 1997, she enrolled in the University of Tennessee Space Institute. On May 12, 1999, Sherri received her Master of Science degree in Mechanical Engineering.