Design of a Jet Actuator for Active Control of Rotating Stall

by

Diego Sebastien Diaz

Ingénieur de constructions aéronautiques, diplômé de l’Ecole Nationale Supérieure d’Ingénieurs de Constructions Aéronautiques

Submitted to the Department of Aeronautics and Astronautics in partial fulfillment of the requirements for the degree of

Master of Science in Aeronautics and Astronautics

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

May 1994

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Abstract

Previous theoretical studies have shown that significant improvement can be achieved in the active control of rotating stall by using jet actuation. A jet actuator for use on the MIT Gas Turbine Lab three stage low speed research compressor was designed. Preliminary study identified the implementation requirements. Two-dimensional linear models were developed to estimate the control effectiveness of the design chosen. The final conceptual choice is a control scheme with discrete actuation: 12 valves feeding 24 injectors equally spaced around the circumference and injecting at an angle from the outer casing. A valve-injector prototype was tested in a wind tunnel, simulating 1/6th of the unwrapped compressor annulus.

The dynamic pressure ratio of the injectant to the main flow was identified as the valid parameter in the classification of the steady velocity field. The injection flow field was shown to have a highly three dimensional shape. The radially averaged circumferential spreading of the jet was satisfactory and created disturbances with 3 to 4 times larger amplitude than the precursor disturbances to rotating stall. The dynamics of the prototype were investigated. The wiggling of the jet due to jet varying penetration with varying mass flow is identified as a nonlinear behavior and its effects are quantified spatially so that meaningful transfer functions can be estimated. This transfer function between valve angle and freestream velocity perturbation shows that the injection process essentially behaves like a time delay. The injection process is shown to be quasi-steady in the range of frequency of interest (up to 100Hz).

Thesis Supervisor: Alan H. Epstein
Title: Professor
Acknowledgments

First, I would like to thank Professor Epstein for his guidance and support, and for sharing some of his enthusiasm for research with me. Thanks to Professor Greitzer for his warm welcome at the Gas Turbine Laboratory.

Thanks to Gavin Hendricks for helping in the modeling and making things look simpler than they are. Well, are they that simple? To Professor Jim Paduano, who helped me greatly to sort out the motor controller and the signal processing.

Thanks to Victor Dubrowsky and Jim Letendre, who greatly contributed to make the experimental work an enjoyable experience, and who made my “little drawings” a reality. Thanks to Robin Courchesne, Holly Rathbun, Bill Ames and Diana Park, who facilitated many aspects of my stay at the lab.

I would like to thank Roland and Chris for their proofreading and much, much more.

Thanks to the GTL french crowd (should I say the SNECMA crowd?).

During these two years, MIT gave me the opportunity to meet so many nice people, I would need an extra-appendix to name them all.

Thanks to the MIT Ballroom Dance Club to have put its destiny in my hands for a year. Thanks, to the MIT Tae-kwon-do Club, to such an intellectually challenging place like MIT you gave me the physical counterpart.

Thanks to Christophe, Carole, Marc, Sophie and Marie.

Finally, I would like to thank SNECMA for its financial support that made possible the furthering of my education at MIT. This work was supported by the US Air Force Office of Scientific Research, Major Daniel B. Fant, technical monitor. Their support is gratefully acknowledged.
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Nomenclature

\( A_c \)  
Compression area

\( A_j \)  
Total injecting area

\( C_d \)  
Nozzle discharge coefficient

\( l_a \)  
Compressor annulus height

\( \delta l_j \)  
Jet valve opening

\( L_{r,ss} \)  
Steady total pressure loss across rotors

\( L_{s,ss} \)  
Steady total pressure loss across stators

\( \delta L_{r,ss} \)  
Perturbation in total pressure loss across rotors

\( \delta L_{s,ss} \)  
Perturbation in total pressure loss across stators

\( \dot{m} \)  
Mass flow

\( n \)  
Spatial harmonic number

\( P \)  
Static pressure

\( P_t \)  
Total pressure

\( \delta P_i \)  
Inlet static pressure perturbation

\( \delta P_{ti} \)  
Inlet total pressure perturbation

\( \delta P_e \)  
Exit static pressure perturbation

\( q_{oj} \)  
Dynamic pressure ratio of the jet at the nozzle exit

\( q_{om} \)  
Dynamic pressure ratio of the main flow

\( r \)  
Compressor mean radius

\( R \)  
Feedback gain

\( s \)  
\((\alpha + i\omega)\frac{r}{U}\)

\( t \)  
Time

\( U \)  
Rotor velocity at the mean radius
$Z$ Complex gain, $Z = Re^{i\beta}$
$
\alpha$ Disturbance growth rate
$
\beta$ Feedback phase angle
$
\gamma$ Injection angle
$
\zeta$ $
\left(\frac{2}{|m|} + \mu\right)$
$
\phi$ Flow coefficient, *Axial velocity/Wheel Speed*
$
\delta \phi$ Flow coefficient perturbation
$
\lambda$ Inertia parameter for compressor rotors
$
\mu$ Total fluid inertia in the compressor
$
\vartheta$ Circumferential coordinate
$
\vartheta_a$ Feedback spatial shift
$
\rho$ Fluid density
$
\rho^*$ Fluid density through the choked valve
$
v^*$ Fluid velocity through the choked valve
$
\tau$ Nondimensional time delay
$
\tau_a$ Nondimensional actuator time constant
$
\tau_r$ Nondimensional rotor total pressure loss characteristic time
$
\tau_s$ Nondimensional stator total pressure loss characteristic time
$
\psi$ Nondimensional pressure rise
$
\psi_i$ Ideal (isentropic) total-to-static pressure rise
$
\psi_{ss}$ Steady state total-to-static pressure rise
$
\omega$ Disturbance rotational frequency
$
\omega_r$ Rotor rotational frequency

**subscripts**

$a$ actuator
$c$ compressor
$e$ exit
$i$ inlet
\( j \)  jet
\( r \)  rotors
\( s \)  stators

**superscripts**

- mean value
- \( \frac{d}{dt} \)
Chapter 1

Introduction

Active control of rotating stall based on the Moore-Greitzer model has been successfully demonstrated on two research compressors [8],[15]. Further studies [9] show that significant improvement in active control performance can be achieved if better actuator and sensor pairs are developed. These theoretical studies showed that a close coupled jet actuator would offer the best performance.

This chapter reviews past work that has been done on active control of rotating stall. The sensor-actuator selection problem is explained with a particular focus on the jet-actuator.

The first section explains the organization of the thesis. The second section gives some background on rotating stall and the stall inception process based on small amplitude disturbances. The third section overviews the experimental work done at the MIT Gas Turbine Laboratory on implementing active control of rotating stall on a single stage and a three stage research compressor. The fourth section reviews the research done by Hendricks and Gysling [9] on the evaluation of alternative sensor-actuator pairs.
1.1 Scope and organisation of the thesis

The goal of this work is to design and test a jet injector, suitable for use on the Gas Turbine Laboratory three stage compressor, which produces a flow perturbation consistent with the 2D model of Hendricks and Gysling [9]. The first chapter describes the technical background leading up to this work. The second chapter describes the modeling developed to aide the actuator design process. The third chapter explains and describes the mechanical design of the actuation scheme. The fourth chapter describes the tests done on a prototype of the injector. The fifth chapter presents the data and evaluates the performance of the design. Chapter six summarize the results and give recommendations for future research.

1.2 Background on rotating stall

Aircraft engines and compression system in general are subject to performance limiting instabilities such as surge and rotating stall. Surge is a one dimensional axisymmetric flow oscillation affecting the whole engine. Rotating stall is a multidimensional instability localized to the compressor. It consists of several regions of stalled flow rotating around the compressor annulus at a fraction of the rotor speed. These instabilities are encountered near the peak of the compressor characteristic where the performance is high, namely high pressure rise and low mass flow. To ensure a safe operation of the engine the current approach is to define a stall or surge margin and to operate the engine away from the unstable domain, thus limiting the performance a given engine can achieve.
1.3 Previous Implementations of Active Control of Rotating Stall

Paduano [15] first showed that active control of rotating stall was possible on a single stage axial compressor. This work also experimentally established the link between small pre-stall circumferential disturbances and rotating stall, because the suppression of the disturbances prevented rotating stall. The control scheme actuator consisted of a set of 12 movable inlet guide vanes equally distributed around the circumference, upstream of the rotor. The sensor array was a set of eight hot wires placed upstream of the compressor. Using this scheme Paduano was able to increase the stable operating range of the compressor by 23%.

Haynes [8] applied the technique demonstrated on the above mentioned single stage compressor to a low speed multistage compressor. He stabilized the three stage research compressor at the MIT Gas Turbine Lab using active control. That particular compressor shows strong traveling waves which grew into rotating stall cells. The scheme was similar to the one used on the single stage compressor, see figure 1-1. According to linear theory each spatial harmonic can be controlled independently. The control scheme implemented, controlled the first and second harmonic, yielding an improvement of 8% in flow range. This work demonstrated that compressor dynamics of such a low speed compressor could indeed be described by linear theory.

1.4 Theoretical Study of Sensor-Actuator schemes for rotating stall control

Previous researchers have shown that it is possible, by using a linear model, to capture the dynamics of low speed research compressors and to improve the performance using an active control scheme. A next relevant point is, is it possible to do better? Hendricks and Gysling [9] have investigated various actuator-sensor combinations. They considered the following actuators: jets upstream of the compressor fed by
higher total pressure air than that at the compressor inlet; intakes ports which are jet fed by air at the same total pressure as the inlet air; valves downstream of the compressor; and wiggling inlet guide vanes upstream of the compressor. The conclusion of their work is that the use of jets fed by high pressure air combined with axial velocity sensing would give a significant improvement in the stable operating domain of the compressor over that achieved to date. Furthermore, jet actuation is the most robust in term of phase margin. Also this approach may be more readily adapted to an engine environment than a scheme employing a set of movable guide vanes.

As the conclusion of Hendricks and Gysling investigation [9] is one of the underlying assumptions of this thesis, the following sections review that paper. The assumptions and the model used are briefly described and the relevant results regarding the jet actuator are presented. The sensor study is not described.

### 1.4.1 Analytical model of compressor stability

The analytical model used in [9] is derived from [3] and [10]. The main assumptions are the following: two dimensional analysis is adequate for describing the flow [3]; the model is assumed to be linear; the inlet total pressure is uniform; inlet and exit ducts are long enough so that there is no reflection of the disturbance waves; and the tip speed is low enough so that the flow can be considered incompressible. The axial velocity wave is decomposed into its spatial Fourier harmonics,

$$\delta \phi_c = \sum_{n=1}^{\infty} A_n e^{s \xi} e^{in\theta}, \text{ where } s = \frac{(\alpha + i\omega)r}{U}$$

(1.1)

$\delta \phi_c$ is the flow perturbation in the compressor; $A_n$ is the spatial Fourier harmonic coefficient; $\theta$ is the circumferential coordinate; $\xi$ is a nondimensional time; $\alpha$ is the disturbance growth rate; $\omega$ is the disturbance rotational frequency; $r$ is the compressor mean radius and $U$ is the rotor velocity at the mean radius. Because the model is linear, each spatial mode can be considered independently. The complete flow perturbation is then the sum of each of the spatial modes. Thus, the $n$th mode only
is considered,

$$\delta \phi = A_x e^{x^c} e^{i n \phi}$$  \hspace{1cm} (1.2)

The compressor quasi-steady assumption assumes that the pressure rise across the compressor reacts instantaneously to variations in flow coefficient. In the model used in this study and described in [12, 14], the pressure rise across the compressor is modified by time lags introduced by the inertia of the fluid into the blade passages. Thus, the compressor does not react instantaneously to flow coefficient variations. This time lag has a stabilizing effect that results in the spatial harmonics going unstable at sequential flow coefficients (the first harmonic going unstable first). That behavior has been showed experimentally in [8, 15].

The unsteady pressure rise through the compressor can be written,

$$\frac{P_e - P_{ti}}{\rho U^2} = \psi - \lambda \frac{\partial \phi}{\partial \phi} - \frac{\mu_r}{U} \frac{\partial \phi}{\partial t}$$  \hspace{1cm} (1.3)

$$\psi = \psi_i - L_r - L_s$$  \hspace{1cm} (1.4)

where $\psi_i$ is the ideal characteristic of the compressor, $L_r$ and $L_s$ represent the losses in the rotors and stators. The complete derivation can be found in [9, 10, 13].

The variables are expressed as the sum of their steady value and the perturbation quantity,

$$\phi = \phi + \delta \phi \quad \psi_i = \psi_i + \frac{d \psi_i}{d \phi} \delta \phi$$

$$P_e = P_e + \delta P_e \quad L_s = L_s + \delta L_s$$  \hspace{1cm} (1.5)

$$P_{ti} = P_{ti} + \delta P_{ti} \quad L_r = L_r + \delta L_r$$

After linearization the pressure rise perturbation equation at stall inception is

$$\frac{\delta P_e - \delta P_{ti}}{\rho U^2} = \frac{d \psi_i}{d \psi_e} \delta \phi_e - \delta L_s - \delta L_r - \lambda \frac{\partial (\delta \phi_e)}{\partial \phi} - \mu \frac{\partial (\delta \phi_e)}{\partial t}$$  \hspace{1cm} (1.6)
\[
\psi_i = \psi_{ss} - L_{r,ss} - L_{s,ss}
\]  
\(1.7\)

The loss perturbations are related to the flow perturbation with the following equations,

\[
\tau_s \frac{\partial (\delta L_s)}{\partial t} = \frac{\partial L_{s,ss}}{\partial \phi} \delta \phi - \delta L_s
\]  
\(1.8\)

\[
\tau_r \left( \frac{\partial (\delta L_r)}{\partial t} + \frac{U}{r} \frac{\partial (\delta L_r)}{\partial \theta} \right) = \frac{\partial L_{r,ss}}{\partial \phi} \delta \phi - \delta L_r
\]  
\(1.9\)

The total pressure perturbation upstream and the static pressure perturbation downstream are related to the flow perturbation by the relations,

\[
\frac{\delta P_{ti}}{\rho U^2} = \frac{1}{|n|} \frac{\partial (\delta \phi)}{\partial \hat{t}}
\]  
\(1.10\)

\[
\frac{\delta P_e}{\rho U^2} = \frac{1}{|n|} \frac{\partial (\delta \phi)}{\partial \hat{t}}
\]  
\(1.11\)

Then by combining these equations, the dynamics of the system for a particular mode are described by a generalized eigenvalues system in \(s\),

\[
(A - sB)\delta \bar{x} = 0
\]  
\(1.12\)

with,

\[
A = \begin{bmatrix}
\left( \frac{d \phi_s}{d \phi_e} - in \lambda \right) \frac{1}{\zeta} & -\frac{1}{\zeta} & -\frac{1}{\zeta} \\
\frac{1}{\tau_s} \frac{d L_{s,ss}}{d \phi_e} & -\frac{1}{\tau_s} & 0 \\
\frac{1}{\tau_r} \frac{d L_{r,ss}}{d \phi} & 0 & -(in + \frac{1}{\tau_r})
\end{bmatrix}
\]  
\(1.13\)
\[ B = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \] (1.14)

\[ \delta \bar{x} = \begin{bmatrix} \delta \phi_c \\ \delta L_x \\ \delta L_r \end{bmatrix} \] (1.15)

and

\[ \zeta = \left( \frac{2}{|n|} + \mu \right) \] (1.16)

The real part of each of the eigenvalues corresponding to one harmonic must be negative for the system to be stable. A complete derivation of that eigenvalue problem can be found in [9].

### 1.4.2 Results of the alternate actuator-sensor study

The complete control system consists of sensors, actuators and a controller used to close the loop. The controller used for these study was a simple complex gain,

\[ \text{Actuation} = Z \times \text{Sensed variable}, \quad \text{where} \quad Z = Re^{i\vartheta_a} \] (1.17)

where \( R \) is the feedback gain and \( \vartheta_a \) is the spatial feedback shift.

The best actuator-sensor pair is the high velocity jet actuation combined with an axial velocity measurement [9]. Figure 1-3 shows the relative performances of the different actuation choices. The curve represent the points of neutrally stable operation so that the area under the curve is the stable operating domain of each actuation pair: the jet actuator and downstream velocity sensing maintain the system
stable up to a larger slope than any other actuator-sensor pair and at any slope where the system is stable this control scheme provides the largest phase margin. Another interesting feature described in [9], is that the jet actuator reduces the rotation rate of the disturbance as the gain is increased, while the other control schemes increase the rotation rate of the disturbance. That feature offers the advantage that the control performance of the jet actuator is less constrained by the bandwidth of the actuator.

1.4.3 Conclusions of alternate Actuator Study

The jet scheme seems the most promising actuator, giving maximum range extension with a large phase margin. A critical point is that this model uses simplifying assumptions that must be addressed in a practical actuator design: the fluid is injected axially, around the circumference close to the compressor (i.e., less than a compressor radius away) and is assumed to mix out completely before reaching the compressor face. From a mechanical point of view, a jet actuator is more robust in a real engine environment.
Figure 1-1: Cross-section of the mounting ring with the wiggling inlet guide vanes used by Haynes [8] to implement active control of rotating stall on a multistage compressor.
Figure 1-2: Model Schematic for the jet injection, from [9].
Figure 1-3: Comparison of alternate actuation approaches applied to the MIT three-stage research compressor. The area under each curve represents the improvement in stable operation for the corresponding scheme. Upstream jets are by far the most effective.
Chapter 2

Conceptual Design of a Jet Actuator

2.1 Introduction

This chapter presents the conceptual design of the jet actuator for use on the Gas Turbine Lab three-stage compressor [8].

The objectives of this chapter are to determine the important parameters, the requirements and the constraints, imposed by both the fluid mechanics laws and the design choices. This study produces fluid design charts of use in the detailed design. The objective is to parametrize the jet actuator design and to produce a viable preliminary design. The fluid mechanics laws and the design choices impose certain requirements and constraints on the jet actuator scheme but any design will present imperfections compare to the ideal scheme. Therefore, a set of parameters is defined to predict the impact of design imperfections on control performance.

The effects on control of the different parameters are modeled using the 2D linear approach of [9] and a proportional control law with a time delay to take into account the actuator time lag.

\[ \tilde{\tau}_a \frac{d(\text{actuation})}{dt} = \text{command} - \text{actuation} \]  

(2.1)
where $\tau_a$ is the actuator time constant (\(\frac{1}{\tau_a}\) being the actuator bandwidth). The model has been updated to take into account the three-stage characteristic and losses. Finally, a new derivation is presented for a varying total pressure injector (i.e., the exit area is fixed) and the performance of fixed and varying area approaches are compared.

This chapter is organized as follows. First, the design requirements, the constraints on the design, and the important parameters are identified. Then, each of the parameters is examined; specifically, jet penetration; injection angle; mixing imperfections; time varying jet total pressure. In each of those sections the modeling and numerical results are given. The chapter concludes with a discussion of the results.

### 2.2 Preliminary design approach

#### 2.2.1 Problem statement

The analytical two dimensional model from [9] predicts that a jet actuator will perform the best in the active control of rotating stall. That model assumes that the momentum and mass addition of a close coupled (less than a compressor radius away) jet is perfectly transmitted to the main flow (i.e., the two streams are completely mixed when they reach the compressor face) and that the injectant is uniformly distributed around the circumference.

The design requirements are derived from the model assumptions. The design should be be simple, versatile and cost effective. The concept of slightly modifying the existing three stage compressor experiment to accept the new scheme goes into the direction of lower cost. The disturbance precursor to rotating stall have been measured to be 1% or 2% of the main flow in the compressor [4]. For a first try in active control of rotating stall with jet injection, a comfortable margin of control power should be made available, therefore it was chosen to design a scheme that can inject up to 10% of the compressor mass flow.
2.2.2 Possible choices and relevant design parameters

In this section different design options are considered and the discrepancies between each design and the model assumption serve to extract the parameters relevant for a jet actuator design. For example, the fluid can be injected into the compressor at several positions—hub, tip, mid-span or uniformly along the span; uniformly around the circumference or at discrete positions. One possible design would be to use hollow inlet guide vanes with injection holes along their span, at the trailing edge or at midspan. Uniform tip injection like that of [7] could be attractive but would require large changes to the existing three stage experimental apparatus. One approach is a shower head type injector placed in the outer casing wall at discrete locations. The injection nozzles would point into the flow laterally and radially to increase the circumferential and radial spreading.

Given that the actuation is discrete, with jet injecting at angles into the flow. The massflow requirement is 10% and the choice of a subsonic injection sizes the exit nozzle area. Also, any penetration of the injector into the flow is a possible source of flow disturbance that could be detrimental to the compressor operation.

Given this general approach, several points must be investigated— the behavior of a jet into a crossflow; the variation in control effectiveness with angle injection; the spatial harmonic content of the created disturbances caused by discrete actuation around the circumference; the influence of only partial jet mixing on the control effectiveness; the difference between jet velocity versus nozzle area variation.

The existing setup has 24 discrete positions in positions in which injectors could be placed although the movable guide vanes use 12 servomotors. Therefore, discrete actuation, which would use 12 valves (one per servomotor) and 12 or 24 injector is attractive. The final design choice is of 24 injectors with 3 hole each oriented at 30 degrees from each other, fed by twelve valves. The following sections go into more detail over the conceptual design of the different parameters of concern. The choice of 24 injectors and the shape of the injectors is justified.
2.3 General Considerations Regarding Jets

2.3.1 Studies of Jet in Crossflow

In the preliminary design of section 2.2.2 the possibility of an angle injection has been considered. A jet in cross-flow has a very different behavior than does a coaxial jet. An extensive literature survey has been done by Schetz [16] on jet injection. It appears that the research on jet injection with a small angle (i.e., 30 degrees) in a confined passage is relatively meager. The cross-section of a jet in a cross-flow has a horseshoe shape, with two contra-rotating vortices as showed in figure 2-1 from [1, 16]. Because of the enhanced turbulence level, that type of jet mixes faster than the coaxial jet. For a design purpose, two empirical laws were retained from [1]. The first one derived by Ivanov,

$$\frac{x}{d} = \left(\frac{q_{omf}}{q_{oj}}\right)^{1.3} \left(\frac{y}{d}\right)^{3} + \frac{y}{d} \cot \alpha$$

(2.2)

The second one derived by Shandorov,

$$\frac{x}{d} = \left(\frac{q_{omf}}{q_{oj}}\right) \left(\frac{y}{d}\right)^{2.55} + \frac{y}{d} \left(1 + \frac{q_{omf}}{q_{oj}}\right) \cot \alpha$$

(2.3)

where $q_{omf}$ is the main flow dynamic pressure, $q_{oj}$ is the jet dynamic pressure at the exit of the nozzle and $\frac{y}{d}$, $\frac{x}{d}$ are the jet centerline coordinates nondimensionalized by the injecting nozzle diameter. In [1] the cross-section of the jet is approximated by an ellipse with an aspect ratio of 5 and a major axis dimension, $h$ that scales as,

$$h = 2.25d_o + 0.22l$$

(2.4)

where $l$ is the coordinate along the jet centerline and $d_o$ is the jet exit diameter.

Figure 2-2 shows a comparison of the predictions of equations 2.2 and 2.3. Figure 2-3 illustrates the jet cross section of a particular jet as seen from the side and from the top.
2.3.2 Dynamic jet behavior

The jet will be pulsating into the main flow, so, the effect of unsteadiness must be evaluated. Two reduced frequencies can be considered. The first is,

\[ F_d = \frac{f D}{U_{\text{mainflow}}} \] (2.5)

where \( f \) is the pulsating frequency and \( D \) is the diameter of the injection nozzle. The second one is,

\[ F_X = \frac{f X}{U_{\text{mainflow}}} \] (2.6)

where \( X \) is the downstream axial location of interest. These two reduced frequencies capture two different phenomena. The one based on the injection nozzle diameter relates the jet pulsating frequency to the convection time for the mean flow past the jet opening. In the second case, the convection time of the main flow from the jet injection point to the axial position of interest (here, the face of the first rotor) is considered. If these reduced frequencies are considerably less than one then the injection process can be considered quasi-steady. These reduced frequencies are used in chapter 5.

2.3.3 Jet penetration charts

As described in 2.3.1, jet penetration in a crossflow is a function of the jet exit velocity, of the injection hole diameter, and of the injection angle. As a direct consequence of this, the jet will move radially and laterally as the jet velocity varies. To avoid introducing new, perhaps unwanted, dynamics to the system and to meet the jet spreading requirement in a steady manner, introducing a perturbation around a steady massflow injected seems a wiser choice than a zero mean injection with non-linear behavior. For different injection angles, curves of constant mass-flow and curves of constant penetration at the axial position corresponding to the compressor face have been plotted. Figure 2-4 illustrates such a chart, used in the design process.
This approach facilitates trade offs such as that between the number of holes and the hole diameter.

The requirement for penetration of the jet is half span and a quarter of the lateral distance between two injectors when the jet are blowing half of the maximum flow and the compressor is still into the stable domain. A minimum nozzle area can be computed from the massflow requirement of 10% and the unchoked orifice condition. By inspection, it is clear that a 24 injectors configuration is able to meet that requirement with an injection angle of 30 degrees and three holes per injector (see figure 2-5). The actual shape of the design is described in chapter 3.

2.4 Effect of injection angle on control

2.4.1 Modeling

A model was constructed to evaluate the influence of jet injection angle on control effectiveness. The model used is similar to [9]. The effect of an angle was included in the momentum equation across the actuator,

\[
(P_1 - P_2)A + (\delta P_1 - \delta P_2)A + \rho(C_{z1} + \delta C_{z1})^2 A + \rho(C_{zj} + \delta C_{zj})^2 A_j \cos \gamma = \rho(C_{z2} + \delta C_{z2})^2
\]  

(2.7)

and yields the following eigenvalue problem,

\[
(A - sB)\delta \bar{x} = 0
\]  

(2.8)

with
\[
A = \begin{bmatrix}
\left(\frac{d\gamma}{d\phi} - \frac{A}{\lambda} \phi - i n \lambda \right) \frac{1}{\zeta} & -\frac{1}{\zeta} & -\frac{1}{\zeta} & \frac{1}{\zeta} (\phi \cos \gamma - \phi_2) \\
\frac{1}{r_a} \frac{dL_e}{d\phi} & -\frac{1}{r_a} & 0 & 0 \\
\frac{1}{r_c} \frac{dL_r}{d\phi} & 0 & -\left( in + \frac{1}{r_c} \right) & 0 \\
\frac{2}{r_a} & 0 & 0 & -\frac{1}{r_a}
\end{bmatrix}
\]

(2.9)

\[
B = \begin{bmatrix}
1 & 0 & 0 & -\frac{1}{|n| \zeta} \\
0 & 1 & 0 & 0 \\
0 & 0 & 1 & 0 \\
0 & 0 & 0 & 1
\end{bmatrix}
\]

(2.10)

and

\[
\delta \vec{x} = \begin{bmatrix}
\delta \phi_2 \\
\delta L_e \\
\delta L_r \\
\phi_j \delta i_j
\end{bmatrix}
\]

(2.11)

### 2.4.2 Results of injection angle on control effectiveness

As shown in figure 2-6, the increase in injection angle toward normal injection decreases the control effectiveness. This is due to the decrease in axial momentum added to the main flow. Too big an angle should be avoided. Section 2.3.3 showed that an angle of 30 degrees gives satisfactory penetration. The model predicts, that an angle of 30 degrees still gives a reasonable range of control domain compare to coaxial injection.
2.5 Effect of circumferential jet mixing on the spatial modal content

2.5.1 Modeling

Because the jets are distributed discretely around the circumference, the circumferential mixing will not be uniform and as a consequence the circumferential perturbation created will have additional spatial modal content. It is therefore of interest to determine the impact of the 12 actuators arrangement on the modal content of the disturbances generated. As an example, figure 2-7 shows the partially mixed jet array modeled by a cosine distribution of twelve 50% duty cycle square waves.

Figure 2-8 shows the modal content of the jet pattern modeled in figure 2-7 and figure 2-9 presents the ratio of the first mode over the twelfth mode for a cosine distribution of a varying duty cycle square wave. Therefore, good mixing circumferentially is important in order to have enough control power in the mode that has to be controlled.

2.6 Influence of Jet mixing on control effectiveness

2.6.1 Modeling

To estimate an order of magnitude for the impact of jet mixing on the control effectiveness, it will be assumed that, if the jet is not fully mixed when it reaches a stage, then there is no effect on that particular stage. To simplify the algebra the losses have been neglected and the compressor split into its three stages with the actuator placed successively at the different inter-stage locations, $\xi$ models the number of stages controlled by the jet actuator, where $\xi = 0$ corresponds to the three stages being controlled and $\xi = 2/3$ corresponds to 2 stages left uncontrolled. The eigenvalue system is then,
\[(A - sB)\delta \tilde{z} = 0 \quad (2.12)\]

with

\[A = \begin{bmatrix}
\left(\frac{d\phi}{d\xi} - in\lambda\right) \frac{1}{\xi} & (\xi(-\frac{d\phi}{d\xi} + in\lambda) + \phi) - (\xi + 1)\phi_2) \frac{1}{\xi} \\
\frac{Z}{\gamma} & -\frac{1}{\gamma}
\end{bmatrix} \quad (2.13)\]

\[B = \begin{bmatrix} 1 & -\frac{1}{\xi} \left(\frac{1}{|\mu|} + \xi\mu\right) \\
0 & 1
\end{bmatrix} \quad (2.14)\]

and

\[\delta \tilde{z} = \begin{bmatrix} \delta \phi_2 \\
\phi_2 \frac{\delta \gamma}{\gamma}
\end{bmatrix} \quad (2.15)\]

### 2.6.2 Results of influence of jet mixing on control effectiveness

Figure 2-10 shows that the control power decreases by one half for each stage left uncontrolled but the phase margin remains constant. The large phase margins mean that the measurement of the perturbations in the compressor do not need to be very accurate.

### 2.7 Modification of the control analysis for time varying injection velocity

In the preliminary design, it appears that a scheme with a valve upstream of the injection nozzle is simpler to implement than a scheme with a valve varying the size
of the injection nozzle. Thus, the control effectiveness of such a scheme has to be modeled. Because, the fixed area and fixed velocity scheme both use high momentum fluid, it is expected that they will perform similarly.

2.7.1 Modeling

The assumptions used in [9] still hold. Now a jet with time varying exit velocity will be included into the model. Mass conservation is,

$$\rho(C_{x1} + \delta C_{x1}) + \rho(C_{xj} + \delta C_{xj})\frac{A_j}{A} = \rho(C_{x2} + \delta C_{x2})$$  \hspace{1cm} (2.16)

Compressibility is neglected. The equation is split into a steady part and unsteady part:

$$\begin{cases} C_{x1} + C_{xj}\frac{A_j}{A} = C_{x2} \\ \delta C_{x1} + \delta C_{xj}\frac{A_j}{A} = \delta C_{x2} \end{cases}$$  \hspace{1cm} (2.17)

The momentum equation is,

$$(P_1 - P_2)A + (\delta P_1 - \delta P_2)A + \rho(C_{x1} + \delta C_{x1})^2A + \rho(C_{xj} + \delta C_{xj})^2A_j\cos\gamma = \rho(C_{x2} + \delta C_{x2})^2A$$  \hspace{1cm} (2.18)

By neglecting the second order terms equation 2.18 becomes,

$$\begin{cases} (P_1 - P_2) + \rho C_{x1}^2 + \rho C_{xj}^2\frac{A_j}{A} = \rho C_{x2}^2 \\ (\delta P_1 - \delta P_2) + 2\rho C_{x1}\delta C_{x1} + 2\rho C_{xj}\delta C_{xj}\frac{A_j}{A}\cos\gamma = 2\rho C_{x2}\delta C_{x2} \end{cases}$$  \hspace{1cm} (2.19)

The velocities are nondimensionalized by the rotor tip velocity $U$ and the pressure by $\rho U^2$. The parameter that will be sensed is the velocity downstream of the actuator. Thus, the nondimensionalized velocity and velocity perturbation upstream are eliminated from the momentum perturbation equation, using the mass conservation
equation. After simplification, we obtain,

\[
\frac{\delta P_1 - \delta P_2}{\rho U^2} + 2 \left( -\frac{A_j}{A} (\phi_2 \delta \phi_j + \phi_j \delta \phi_2) + \phi_j \delta \phi_j \frac{A_j}{A} (\cos \gamma + \frac{A_j}{A}) \right) = 0
\]  

(2.20)

expressing the momentum perturbation equation with total pressure instead of the static pressure: \( \delta P = \delta P + \phi \delta \phi \) and neglecting the second order terms, equation 2.20 becomes,

\[
\frac{\delta P_{t1} - \delta P_{t2}}{\rho U^2} - \frac{A_j}{A} (\phi_2 \delta \phi_j + \phi_j \delta \phi_2) + \phi_j \delta \phi_j \frac{A_j}{A} (2 \cos \gamma + \frac{A_j}{A}) = 0
\]

(2.21)

using a derivation similar to [9], the dynamics of the system are modeled by the following eigenvalue system,

\[
(A - sB)\delta = 0
\]  

(2.22)

with

\[
A = \begin{bmatrix}
\left( \frac{d\phi_2}{d\phi_2} - \frac{A_j}{A} \phi_j \frac{1}{\zeta} \right) & -\frac{1}{\zeta} & -\frac{1}{\zeta} & A_{(1,3)} \\
\frac{1}{\tau_\omega} \frac{dL_{\omega,\omega}}{d\phi} & -\frac{1}{\tau_\omega} & 0 & 0 \\
\frac{1}{\tau_\omega} \frac{dL_{\phi,\phi}}{d\phi} & 0 & -1 \left( \frac{1}{\tau_\omega} + \frac{1}{\tau_\omega} \right) & 0 \\
\frac{\tau_\omega}{\tau_\omega} & 0 & 0 & -\frac{1}{\tau_\omega}
\end{bmatrix}
\]

(2.23)

\[
A_{(1,3)} = \frac{1}{\zeta A_c} (\phi_j (2 \cos \gamma + \frac{A_j}{A_c}) - \phi_2) \left( \frac{\rho^* v^*}{\rho_j U} \right)
\]

(2.24)
\[ B = \begin{bmatrix}
1 & 0 & 0 & -\frac{A_x}{\lambda_j} \frac{1}{\lambda_j \zeta} (\frac{\rho_j v_j^*}{\rho_j U}) \\
0 & 1 & 0 & 0 \\
0 & 0 & 1 & 0 \\
0 & 0 & 0 & 1
\end{bmatrix} \tag{2.25} \]

and

\[ \delta \vec{x} = \begin{bmatrix}
\delta \phi_2 \\
\delta L_u \\
\delta L_r \\
\frac{\delta A_x}{\lambda_j}
\end{bmatrix} \tag{2.26} \]

### 2.7.2 Conclusions on the control effectiveness of varying velocity injection

Figure 2-11 shows a comparison of the varying velocity injection and the varying area injection. Figure 2-12 shows the control domain for the scheme with varying gain for the control law and figure 2-13 shows the decrease of the perturbation rotation rate as the control gain increases. Figure 2-14 shows the control performance decrease of the varying jet velocity scheme as the injection angle is increased, thus, the scheme is not sensitive to actuator bandwidth variations. Figure 2-15 illustrates the low impact on control effectiveness of increased actuator time constant. The injection with fixed exit area and the one with constant exit velocity have the same characteristics of high control performance.
2.8 Discussion of the results and of the design choice

The conceptual design has been driven by the design preference for a showerhead type injector. Figure 2-16 shows the planned arrangement of the control jet actuator. For the given design requirements, the parametric studies gave reasonable numbers for the characteristics (angle, diameter of the holes, number of the holes) of the injectors and confidence in the effectiveness of the scheme to control rotating stall.
- Correlations for centerline:
  \[
  \left( \frac{x}{d} \right) = \left( \frac{y}{d} \right)^{\alpha} \left( \frac{U_d^2}{\rho} \frac{U_{in}^2}{\rho} \right)^{\beta} + K \cot \alpha
  \]

- Estimate of jet penetration

Figure 2-1: Jet mixing in cross-flow depends on jet and free stream velocities and on jet injection angle, from [1].
Figure 2-2: Comparison of Ivanov and Shandorov (from [1]) experimental correlations for jet penetration, for two different dynamic pressure ratios $\frac{q_{ij}}{q_{om,f}}$ between the main flow and the jet.
Figure 2-3: Theoretical computation of the jet penetration and spreading [1].
Figure 2-4: The correlation for penetration has been applied to the geometry of the three stage compressor at one particular injection angle.
Figure 2-5: Theoretical computation of the jet penetration and spreading [1].
Figure 2-6: Calculated effect of jet injection angle on control. Increasing the jet injection angle toward normal injection decreases the control effectiveness, since the axial momentum of the jet is decreased. The area under each curve represents the stable operation domain for each case.
Figure 2-7: This cosine distribution of twelve 50% duty cycle square waves is used to represent a partially mixed jet array.
Figure 2-8: The modal content of the jet pattern with 50% duty cycle square wave.
Figure 2-9: The modal content of the jet pattern with varying width square wave.
Figure 2-10: Influence of jet mixing on controlled compressor performance. The numbered curves refer to the stage by which the injectant is assumed to become effective in influencing the flow. The area under each curve represents the stable operation domain for each case.
Figure 2-11: The fixed area and the fixed velocity have similar behavior.
Figure 2-12: Varying gain for control with constant jet area and varying jet velocity.
Figure 2-13: The rotation rate of the perturbation decreases as the feedback gain increases.
Figure 2-14: Comparison of control performance of the time varying jet velocity injection with increasing injection angle. The area under each curve is the stable operation domain for the corresponding case.
Figure 2-15: The stable operation domain, estimated by the area under each curve, does not vary significantly with increasing actuator time constant.
Figure 2-16: Three-stage compressor showing planned arrangement of control jet actuators.
Chapter 3

Mechanical Design of the Actuator

3.1 Introduction

This chapter describes the detailed design of the jet actuator. The objective is to adapt the mounting ring of the MIT Gas Turbine Lab three stage compressor to 12 valves and 24 injectors in a simple and cost effective manner. The injectors should be easily orientable and removable.

This chapter explains and justifies the mechanical design choices. The general scheme is described first and then the injector and the valve details are presented.

3.2 The complete scheme

Figure 2-16 and 3-1 show the complete actuation scheme. The injector fits into the mounting ring at the same location as the current control vanes. The mounting ring has to be machined only slightly to accept the new scheme. The servomotor mounts on the valve body which mounts to the mounting ring and holds the injectors in position. The injectors are easily removable and interchangeable.
3.3 Flow requirements

The design requirement is 10% of the compressor massflow near stall, which gives 0.02 kg/s per valve. The system should be able to control disturbances harmonics up to 60 Hz. Thus, the bandwidth of the actuator should be at least 120 Hz. The valve is chosen to be choked so that the relationship between valve opening and massflow is linear.

3.4 The valve

Use of the existing vane servomotors as the valve actuator of a custom made valve was chosen as a cost effective solution. The valve body is multi-purpose, in that, it holds the servomotors and the injectors to the mounting ring. Figure 3-2 shows the detail of the existing motor and shaft. The scheme uses the slot in the middle of the shaft as the valve regulator. The massflow requirements coupled with a small valve area require that the valve be choked. A screen was inserted into the exit chamber to break up the exit jet so that the valve feeds the two injectors equally. To avoid that much air leaking through the shaft bearing, the hole in which the valve shaft rotates has been machined with a very small clearance such the shaft still able to turn freely but the leakage to the upper part of the valve is minimized. There is also some clearance between the bottom of the shaft and the plate which seals the valve. Because of these clearances, leakage is allowed through the valve when it is in closed position. Figure 3-3 shows the body of the valve and figure 3-4 show the detail of the valve working principle.
3.5 Injector Design

A showerhead type design was chosen, with the following characteristics: 3 injection holes at 30 degrees separation in the circumferential direction with a mechanical diameter of $\frac{3}{16}$ inches. Design constraints include no choking within the injector and that the injection holes should be long enough to create a jet (i.e., twice the exit diameter). Also, the main flow should be disturbed as little as possible, thus the injector should protrude as little as possible into the flow path. Figure 3-5 shows the detail of the machining of the injector.

3.6 Conclusions

The theoretical design of the actuation scheme has been implemented in a simple and cost effective way.
Figure 3-1: Complete scheme: 12 valve bodies + 12 injectors on the mounting ring
Figure 3-2: Pacific-Scientific servomotors schematic.
Figure 3-3: Technical drawing of the valve. The dark arrows show the flow path.
Figure 3-4: Detailed schematic of the valve.
Figure 3-5: Technical drawing of the injector. The dark arrows show the flow path.
Chapter 4

Description of Experiment

4.1 Introduction

This chapter describes the testing of a prototype injector described in chapter 3. First, the prototype is described. Then, the wind tunnel test section is described. The measurement and data acquisition apparatus is presented. The last section overviews the type of test runs. The data are described generically and will be reduced and analyzed in chapter 5.

4.2 The prototype

4.2.1 Description

A test set has been built which modeled 1/6th of the compressor annulus. The annulus was flattened to a rectangle however. This annulus section consists of a valve and four injectors; two fed by a valve, with one on either side to provide realistic end conditions should the flows of two adjacent injector interact (this proved not to be the case).
4.2.2 Bench tests of valve and injector

Before the tests of jet behavior, performance of the valve and injector were independently measured.

The valve

The steady state mass flow versus pressure drop characteristics of the valve with no injectors was determined, using a pressure gage and a flow meter. As the massflow through the valve was increased, the pressure drops on both sides were recorded. Figure 4-1 shows the output uniformity versus shaft position. If the screen are removed, the jet produced by the servomotor shaft, unbalances massflow on each side of the valve. A valve characteristic was measured for a 100 psi supply pressure and is shown in figure 4-2. The leakage through the valve was 0.0082 kg/s, at a supply pressure of 100 psig. There is a trade off between the amount of mass flow per degree of rotation of the shaft and the steady state leakage.

The injectors

A discharge coefficient of 0.65 was assumed for the design of the injector. The discharge coefficient of two injectors was measured by using a flowmeter and a pressure gage. Figure 4-3 shows the mass flow through the injectors versus the square root of the pressure drop across them. The discharge coefficient is given by:

\[ \dot{m} = \sqrt{\rho \text{Area}_{geometric} C_d \sqrt{2\Delta P}} \], from [11] \hspace{1cm} (4.1)

The slope of the curves give respectively 0.613 and 0.637.
4.3 The test section

Figure 4-4 presents the wind tunnel test section. One end of the test section fits into an existing one foot by one foot low speed wind tunnel. This section includes a contraction that reduces the height of the passage to the height of the compressor and models a flattened 2D compressor annulus. A support has been designed to hold the four injectors and the valve on the test section. Downstream, an opening has been machined to allow the probe to travel in a plane perpendicular to the flow direction. The mass flow through the valve and the two side injectors is regulated by individual valves and monitored with three flowmeters and pressure gages.

4.4 Instrumentation

The wind tunnel speed was measured using a pitot tube and static pressure taps. The pitot tube was placed at the entry of the test section before the contraction and the pressure taps at the beginning of the passage. Two pressure taps were placed across the width of the passage and were connected with Teflon tubing and a T-junction in order to average the measurements. The pitot tube measured the total pressure only. The pitot tube and the static taps were connected to a MKS-Baratron pressure meter with a 10 psi range. The mean velocities were obtained from the difference between the total and static pressure using the incompressible Bernouilli equations. A 2D traversing system was designed using two Unislide traversers and a Velmex NF90 stepping motor controller. The arrangement consisted of a long traverser placed horizontally and holding a shorter traverser placed vertically. The hotwire anemometer used is a TSI-1210-T1.5 (4μ Tungsten wire) with a 90 degrees angle adapter, TSI-1012. The hotwire was calibrated in-situ by using the MKS-Baratron and varying the wind tunnel speed.
4.5 Data acquisition and motor control apparatus

Data were recorded along two path: through an A/D board and through the motor controller board. The A/D board is a Data Translation Model DT2801-12 bit A/D converter, it was used to record the analog output coming from the range multiplier used with the MKS-Baratron and the analog output from the amplifier and filter used with the hotwire. The hotwire signal was filtered with a 500 Hz cut-off frequency low-pass filter. The motor controller board is a Galil DMC400-10. An optical encoder was connected to that board using a connection card Galil ICB930 and was used to record the shaft position. The valve uses the same low-inertia servomotor used in the three stage compressor. The motors are Pacific Scientific 4VM62-220-1 permanent-magnet servomotors. They have an inertia of $3.8 \times 10^{-6} \text{ kg} \times \text{m}^2$. Haynes [8] described in his thesis the working set-up for these motors in the three stage compressor experiment. The computer used is a Hewlett Packard Model HP-Vectra-RS/20 Microcomputer 20 MHz 386. The data were acquired at 1000 Hz sampling rate per channel. The A/D data acquisition was done using a DMA process with a pacing loop in which the command was send to the motor; the hotwire voltage was read; and the motor position was read from the optical encoder through the motor controller board.

4.6 Data acquired and data processing

4.6.1 Steady state data

The flow was mapped with a steady injected flow and a constant wind tunnel speed. The velocity was measured on a grid, 8 inches wide, 1 inch high in 0.2 inch increments. The hotwire was traversed in a plane perpendicular to the main flow direction, using the traversing mechanism. At each position, data were taken for five seconds and then averaged. The velocity measured is nondimensionalized by the main stream velocity. The data are presented in three dimensional plots where the horizontal axis corresponds to the position on the grid and the vertical axis correspond to the nondimensionalized velocity.
4.6.2 Dynamic data

Dynamic data were taken in two ways. First, because the jet in a crossflow has a penetration that depends on its injection to freestream velocity ratio, the oscillations of the jet will have some effect on the measured dynamics. To better understand any possible new dynamics and nonlinearities introduced by this behavior, a discrete sweep in frequency from 10Hz to 100Hz in 10Hz increments was sent to the valve. The commanded sine-wave corresponds to the maximum travel of the valve. Then, for each frequency the measured signal was ensemble averaged over at least 20 cycles per frequency. Second, the open loop transfer function was measured by exciting the valve with white noise (generated with Matlab and digitally low-pass filtered at 200 Hz). There were three relevant parameters: the command, the shaft position, the velocity measured by the hotwire. The dynamic plots display the different transfer functions between those three signals.
Figure 4-1: Steady state valve performance.
Figure 4-2: Valve characteristic at a 100 psi supply pressure and exhausting to the atmosphere with the two injectors attached. The control scheme is designed to operate between 0.01 kg/s and 0.025 kg/s.
Figure 4-3: The design injector discharge coefficient assumed was 0.65.
Figure 4-4: Wind tunnel set-up.
Chapter 5

Data

5.1 Introduction

The measured steady state and dynamic jet behavior is analyzed in this chapter. Its suitability for control purposes in accordance with a two dimensional compressor stability model is discussed.

This chapter presents the data that were taken in the wind tunnel with the valve and injector prototypes. The data are reduced and processed to provide a better understanding of that particular design and of the feasibility of a two dimensional jet actuator.

First, the steady state data are described and the spatial features of the jet given-penetration and radial and circumferential spreading are determined. The capability of creating steady spatial harmonics is then investigated. Second, the dynamic data are presented. The linearity of the process is considered, then appropriate transfer functions are computed. The quasi-steadiness of the jet movement is confirmed.

5.2 Presentation of data

The steady state and the dynamic data were measured at the axial position corresponding to the location of the first rotor face. The distance between the jet location and the compressor first rotor face is 2.5 inches or 14 jet diameters in the three stage
MIT research compressor and the passage height is 1.44 inches.

5.2.1 Ratio of dynamic pressure as a non-dimensionalized parameter

The nondimensionalized flowfield shape of a jet in a crossflow is characterized by the ratio of the jet dynamic pressure at the nozzle exit to the main flow dynamic pressure. Steady state flow field measurements have been run at different test conditions, different injected mass flow and different wind tunnel velocities. Characterizing a test by that dynamic pressure ratio, provides a convenient way to use the nondimensionalized velocity fields, determined at particular conditions, for other conditions of mass flow injected and main flow velocity, given that the dynamic pressure ratio stays constant. To compute the dynamic pressure of the jet at the injection holes, the massflow through the valve and effective area of the injection holes was used. The flow inside the valve body was assumed compressible. To justify that this dynamic pressure ratio fully characterizes a nondimensionalized velocity profile, two steady test were run at different testing condition. The first one at 0.022 kg/s injection and 40 m/s wind tunnel speed, the second one at 0.0163 kg/s injection and 30 m/s wind tunnel speed. They both have a dynamic pressure ratio of 40. Figure 5-1 shows horizontal slices of the flowfield going from the tip (#1) to the hub (#6). The vertical axis corresponds to the velocity nondimensionalized by the main flow velocity, the horizontal axis represents an arc of circumference unwrapped. The nondimensionalized velocity field of these two test have been plotted on top of each other; the crosses represent the first test, the circles the second one. They are similar enough to consider that dynamic pressure ratio is the valid parameter to classify the static velocity fields.

5.2.2 Description of the steady state data

Figure 5-2 to 5-7 show the velocity field measured at the downstream axial distance corresponding to the compressor face. The plots are ordered with increasing dynamic pressure ratio between the jet and the main flow. The data are presented in two differ-
ent types of plots- contour plot of the nondimensionalized velocity field interpolated from the discrete grid measurements and three dimensional plots of discrete grid of the flow field. The contour plot are plotted to give a better understanding of the three dimensional flowfield features such as the number of jet observable, however, they do not show any quantitative information on the magnitude of the velocity field. On the three dimensional plots the two horizontal axes correspond to the cross-stream plane (circumferential and radial directions; 0 on the radial direction axis corresponds to the tip). On the vertical axis, the nondimensional velocity has been plotted. At that axial position the flow can be considered incompressible, so that it is equivalent to plot the ratio of dynamic pressure or the ratio of velocities. One can notice a velocity defect on the sides of the plot (points (0,0) (0,8)), this is due to the wake created by the nonblowing side injectors which protrude into the flow. In figure 5-2, dynamic ratio 17, the jets do not penetrate and spread as much as the data in the following plots. At the dynamic pressure ratio increases, the jets grow in size laterally and radially, the center line penetration into the flow increases as expected. The three jet patterns also cover a bigger arc of the circumference. Another feature is that the jet coming out the injector center nozzle is more visible at higher dynamic pressure ratios, this is particularly visible on the contour plots. One should be aware that the finite definition of the grid contributes to the apparent unsymmetrical shape of the flow from one injector: if the maximum velocity in a jet does not fall right on one grid point, the jet will appear unsymmetric and “shorter” than it really is.

5.2.3 Jet penetration

From the steady flow field data, the jet center position can be determined. Figure 5-8 shows the theoretical jet position computed as described in section 2.3.1 and the experimentally determined jet position from the steady state plots. The penetration presented are measured from the center of each injector exit nozzle, the fact that the injector has a finite size and that it protrudes into the flow has been taken into account and the measured penetrations have been corrected accordingly. The penetrations plotted are for one injector- the radial penetration for the jet corresponding to the
injector center nozzle; and for the injector lateral nozzles the jet lateral and radial penetrations. There is a factor of 0.6 between the measured penetration and the predicted one. This may be due to the fact that the prediction does not take into account that the injection is made into a confined passage and that the correlations were based on angles between 45 degrees and 90 degrees.

5.2.4 Circumferential spreading

It is of interest to determine the circumferential influence of the injection system. Because the control model with which this actuator will be used is two dimensional, the velocity profiles have been averaged in the radial direction. Figures 5-9 and 5-10 present the radially averaged velocity profiles corresponding to the three dimensional data. In this setup, the injectors are placed 3 inches from each other, so that there is an injector at 2.5 inches and at 5.5 inches on the circumferential axis in figures 5-9 and 5-10. The circumferential arc of one injection unit (two injectors fed by one valve) should control, cover 6 inches. For the highest dynamic pressure ratio (see the third plot of figure 5-10) the influence of the injection is seen over 6 inches, for the lowest (see the first plot in figure 5-9) the influence of the jets is seen only over three inches.

5.2.5 Steady spatial harmonics

Each valve feeds two injectors and the injectors are equally spaced 3 inches apart around the circumference. In section 5.2.4 it was shown that the valve influence is confined to six inch long arc. One valve is responsible for 6 inches in the circumferential direction. The velocity profiles, from figures 5-9 and 5-10 can be used to infer the actual range of influence for a valve (i.e., from 1 inch to 7 inches in the plots). A complete circumferential arrangement of the 12 valves can be synthesized by simply cutting and pasting these velocity profiles together. The spatial spectral content of these steady injection pattern was so computed. Figures 5-11, 5-13, 5-15 and 5-17 show different injection patterns from 12 pairs of injectors and their corresponding spectral content. The magnitude of the PSD coefficient of each mode is
non-dimensionalized by the magnitude of the PSD coefficient of the 0th mode. The amplitude of the perturbation corresponding to a particular mode can be determined by taking the square root of the nondimensional coefficient for that particular mode. In figure 5-13, the perturbation corresponding to the first mode is 5% of the mean flow. In figure 5-11 a step function is synthesized with half of the jets closed and the other half full open, this pattern produces a strong first mode and all the higher modes have PSD magnitude less than 10% than the PSD magnitude of the first mode. In figure 5-13 a first mode disturbance is reproduced, and as expected the spectral content shows a strong first mode but also harmonics of the 24th mode. In figure 5-15 the spatial mode content of one jet at full blowing is computed, the modes up to the 8th mode are significantly represented and the perturbation corresponding to the first mode represents 2.2% of the mean flow and the one corresponding to the 8th mode, 1% of the main flow. In figure 5-17 a second mode is synthesized, the second mode perturbation is 5% of the main flow.

Because the dynamic pressure ratio depends on the velocity of the main flow at the entry to the compression system and because the massflow injected is limited for any particular setup, the control power achievable could be limited by the fact that at critical flow coefficient the maximum reachable dynamic ratio will be too low to create a sufficiently large perturbation. Figure 5-19 shows the curve of the maximum achievable dynamic pressure ratio with the compressor axial velocity in m/s. The vertical dotted line shows the compressor stability limit without control. In the previous spatial harmonics, which were created with dynamic pressure ratios ranging from 17 to 74, a 8% perturbation in the first mode was feasible. This curve shows that even in the stable domain of the compressor, the injection scheme is able to create significant disturbances since the precursor disturbances to rotating stall have been measured to be on the order of 1% or 2% of the mean flow.

For control design purposes, it is easier to model the jet by a square wave. Figures 5-12, 5-14, 5-16 and 5-18 present the square wave approximations of the 2D jet patterns obtained by taking the mean over 6 inches circumferentially of the radially averaged profiles. There is a good agreement, between the pattern synthesized with
only a radial average and the pattern synthesized with a radial and circumferential average, in the corresponding spectral contents for the lower modes (< 12th).

5.3 Dynamic data

5.3.1 Dynamic behavior

The reduced frequencies defined in section 2.3.2 can be computed. The lowest speed in the compressor will be 30 m/s, the exit diameter of each injection nozzle is 3/16 inch, so for 100 Hz the reduced frequency \( F_d \) is 0.0156. Thus, near the injection hole the injection process appears quasi-steady to the main flow. The compressor face is located at 2.5 inches from the injectors, so for 100 Hz, the reduced frequency \( F_X \) is 0.085. Thus the jets and their mixing process will also look highly quasi-steady to the compressor.

Another aspect of the dynamic behavior of concern is due to the fact that, as the mass flow is varied, the penetration of the jet varies. The velocity profiles of the jets are not only varying in magnitude and spreading but are also oscillating: radially for the center jet and both radially and laterally for the two lateral jets in each injector. How will the linearity of the signal sent to the compressor face be altered by that behavior? To answer that question the following approach has been taken. A sequence of repetitive sine waves in a discrete frequency sweep have been sent to the valve and the resulting velocity variation has been measured at different radial and circumferential locations. To suppress the residual noise from the hotwire signal, the signal was ensemble averaged over at least 20 periods for each frequency. The measurements were made on a vertical line (radial direction) in front of the injector center nozzle and on a horizontal line intersecting the center line of the jets at mean injection flow (figure 5-20 shows the measurement locations). The resulting ensemble averaged signals are plotted in figures 5-21 to 5-25 for the vertical line and in figures 5-26 to 5-31 for the horizontal line. The radial location (position #3), that best reproduces the commanded sine wave, is replotted on top of each plot as a reference.
The horizontal axis correspond to time in ms, the vertical axis is in m/s. In plot 5-32 the signal at position #3 fits the command well, while in figure 5-21 the sine wave has altered. In figure 5-22, the measured sine wave has been cut-off, because as the massflow increases the jet penetration increases, so the jet center, after passing in front of the probe and allowing a signal faithful to the command to be measured, tends to go away from it and as a consequence the measured signal is degraded. The same explanation holds when the penetration of the jet decreases as the mass flow is decreased. If linear spectrum analysis is applied to these signals, it is expected that the better the sine wave shape is conserved the better the coherence will be. That means that at some radial positions the signal may be too degraded by nonlinear behavior to give meaningful results in a transfer function estimation. In figures 5-33, 5-34, 5-35, 5-36 and 5-37 the data are plotted in the following way: the top plot shows the spectrum of the input signal, the second shows the gain of the corresponding transfer function, the third shows the phase of the transfer function, and the last one shows the coherence between the two signals. Figure 5-33 illustrates the poor coherence caused by the strong nonlinear jet behavior at position 1, while figure 5-34 shows that, at position 3, the effects of nonlinear jet behavior are negligible.

To obtain a better definition in frequency for the transfer function, white noise lowpass filtered at 200 Hz was used. The point in space chosen to determine the transfer function coincided with the one where the sine wave was best reproduced. Figure 5-35 presents the transfer function between the digital command send by the computer to the measured position of the servomotor shaft which command the opening of the valve. Figure 5-36 illustrates the transfer function from the command to the velocity measured by the hotwire, figure 5-37 shows the transfer function from the measured valve position to the velocity measured by the hotwire. The command to position plots are presented to give an idea of the motor controller performance. The controller provided with the motor does not allow a fine optimisation of the motor performance since the shaft has relatively small inertia and there is nothing attached to increase it. The transfer function of interest is the one from the position of the valve to velocity measurement of the hotwire (figure 5-37). Up to 170 Hz the
coherence between the two signals is high and the gain is flat.

Figure 5-38 shows the comparison between the measured transfer function and a pure time delay. The phase compare well and the corresponding time delay gives, using the nondimensionalisation of chapter 2, an actuator time constant of 0.6. A better fit between a theoretical model and the measured transfer function can be obtained by taking into account the effect of the low pass filter an the measured signal and adding a pole at 500 Hz to the time delay (see figure 5-39).

5.4 Jet dynamic position

The reduced frequencies $F_d$ and $F_X$, defined in section 2.3.2 and computed in section 5.3.1, identified the jet mixing behavior to be highly quasi-steady for frequencies up to 100Hz. Thus, the dynamic shape of the flowfield and the wiggling of the jet should be predictable using the steady data. To illustrate the fact that the jet position is oscillating in a quasi-steady way as the massflow varies, the jet positions (i.e., the position of the center of the jet in the plane corresponding to the entry of the compressor), steady and dynamic, were extracted from the steady tests and from the sine wave sweeps. The assumption is that, at one point in time for the six vertical ensemble averaged sine wave sweep signals, the center of the jet is at the position corresponding to the signal which has the highest velocity. To perform the steady approximation of the dynamic jet centerline oscillations, the following approach was taken: the steady penetration of the jet centerline was plotted against the dynamic pressure ratio of the corresponding test and a second order polynomial fit was performed to have the jet centerline radial penetration as a function of the dynamic pressure ratio between the injectant and the main flow (see figure 5-40), this correlation is,

$$\frac{y}{l_c} = \frac{1}{1000} \left( -0.0621 \left( \frac{q_{oj}}{q_{om,f}} \right)^2 + 11.8176 \left( \frac{q_{oj}}{q_{om,f}} \right) - 7.06 \right) \quad (5.1)$$
where \( \frac{h}{r_c} \) is the radial penetration into the flow nondimensionalized by the height of the compressor passage, the injection angle is 30 degrees, and the axial distance is the distance between the injector location in the compressor and the first face of the compressor; the main flow field velocity was fixed to be the wind tunnel velocity used for the dynamic tests (30 m/s), so that the valve characteristic could be represented in a plot of dynamic pressure ratio against valve opening. A polynomial fit was also performed on these data points (see figure 5-41). Then, for any valve opening one can compute the dynamic pressure ratio of injectant over main flow and using that dynamic pressure ratio the corresponding radial location for the jet centerline. The phase lag identified in the previous section is not taken into account. The measured valve position was used with these correlations to predict the jet centerline position when the valve oscillates. Figure 5-42 shows three curves- the jet centerline position estimated from the measurement and represented by discrete points at the radial grid positions; the predicted jet position; and the nondimensionalized valve angle included as reference. The vertical axis is only significant for the radial position curves and represents the height of the passage where the measurements were taken, 0 corresponds to the tip and 1 to the hub. The horizontal axis represents time in ms. The accuracy of the estimated jet position from the dynamic jet is limited by the radial distance between the dynamic sine sweep measurements. Still, there is no big discrepancies between the measured and the estimated jet centerline position to contradict the quasi-steady assumption for the jet behavior.

5.5 Conclusions

The dynamic pressure ratio is identified to be the parameter to characterize the injection into the flow. The flowfield appears to be highly three dimensional with the penetration of the jet centerline being around 60% of the predicted one. The circumferential spreading of the radially averaged flow field out of the injector covers a significant part of the 1/12th of circumference each valve is responsible for. The steady plots were averaged radially to evaluate the average size of the perturbation,
the complete scheme is able to produce. It appears that the prototype injection scheme is able to produce disturbances 3 to 4 times higher than the precursor disturbances to rotating stall. The wiggling of the jet, due to the jet varying penetration as the massflow through the valve varies, was identified as a nonlinear phenomena. The use of sine wave sweeps at various radial and lateral positions, helped to identify the locations were the nonlinear behavior is negligible and linear spectrum theory can be applied. At those particular locations, transfer functions were estimated and satisfyingly compared with a pure time delay. Up to 100 Hz the jet behavior can be considered highly quasi-steady, this was showed by computing characteristic reduced frequencies and comforted by a comparison of a quasi-steady prediction of particular jet centerline dynamic position with the dynamic position of the jet centerline estimated from the dynamic data.
Figure 5-1: Plot of two nondimensionalized velocity fields with a same dynamic ratio of 40 but different testing conditions: the circle represent a test run at 40 m/s and 0.022 kg/s injected massflow and the crosses represent a test run at 30 m/s and 0.0163 kg/s injected massflow.
Figure 5-2: Three dimensional plot of nondimensionalized velocity field. The distances are in inches.
Figure 5-3: Three dimensional plot of nondimensionalized velocity field. The distances are in inches.
0.01345 kg/s injection with 29.4 m/s wind, dyn. press. ratio of 31

Figure 5-4: Three dimensional plot of nondimensionalized velocity field. The distances are in inches.
Figure 5-5: Three dimensional plot of nondimensionalized velocity field. The distances are in inches.
Figure 5-6: Three dimensional plot of nondimensionalized velocity field. The distances are in inches.
Figure 5-7: Three dimensional plot of nondimensionalized velocity field. The distances are in inches.
Figure 5-8: Theoretical and experimental steady jet position.
Figure 5-9: Radially averaged nondimensionalized velocity field, for steady jets at increasing dynamic pressure ratios.
Figure 5-10: Radially averaged nondimensionalized velocity field, for steady jets at increasing dynamic pressure ratios.
Figure 5-11: Spatial spectral content of the radially averaged jet pattern. The top picture shows the nondimensionalized flowfield in the compressor. The bottom picture shows the magnitude of the spatial PSD coefficients for each harmonic number divided by the magnitude of the spatial PSD coefficient for the 0th harmonic.
Figure 5-12: Spatial spectral content of the square wave approximation of the jet injection. The top picture shows the averaged nondimensionalized flowfield in the compressor. The bottom picture shows the magnitude of the spatial PSD coefficients for each harmonic number divided by the magnitude of the spatial PSD coefficient for the 0th harmonic.
Figure 5-13: Spatial spectral content of the radially averaged jet pattern. The top picture shows the nondimensionalized flowfield in the compressor. The bottom picture shows the magnitude of the spatial PSD coefficients for each harmonic number divided by the magnitude of the spatial PSD coefficient for the 0th harmonic.
Figure 5-14: Spatial spectral content of the square wave approximation of the jet injection. The top picture shows the averaged nondimensionalized flowfield in the compressor. The bottom picture shows the magnitude of the spatial PSD coefficients for each harmonic number divided by the magnitude of the spatial PSD coefficient for the 0th harmonic.
Figure 5-15: Spatial spectral content of the radially averaged jet pattern. The top picture shows the nondimensionalized flowfield in the compressor. The bottom picture shows the magnitude of the spatial PSD coefficients for each harmonic number divided by the magnitude of the spatial PSD coefficient for the 0th harmonic.
Figure 5-16: Spatial spectral content of the square wave approximation of the jet injection. The top picture shows the averaged nondimensionalized flowfield in the compressor. The bottom picture shows the magnitude of the spatial PSD coefficients for each harmonic number divided by the magnitude of the spatial PSD coefficient for the 0th harmonic.
Figure 5-17: Spatial spectral content of the radially averaged jet pattern. The top picture shows the nondimensionalized flowfield in the compressor. The bottom picture shows the magnitude of the spatial PSD coefficients for each harmonic number divided by the magnitude of the spatial PSD coefficient for the 0th harmonic.
Figure 5-18: Spatial spectral content of the square wave approximation of the jet injection. The top picture shows the averaged nondimensionalized flowfield in the compressor. The bottom picture shows the magnitude of the spatial PSD coefficients for each harmonic number divided by the magnitude of the spatial PSD coefficient for the 0th harmonic.
Figure 5-19: Maximum dynamic pressure ratio achievable with the prototype configuration as a function of main flow velocity.
Figure 5-20: Positions on the measurement grid for the sine waves experiments.
Figure 5-21: Signal at position 1 compared with signal at position 3 (dash-dotted curve).
Figure 5-22: Signal at position 2 compared with signal at position 3 (dash-dotted curve).
Figure 5-23: Signal at position 4 compared with signal at position 3 (dash-dotted curve).
Figure 5-24: Signal at position 5 compared with signal at position 3 (dash-dotted curve).
Figure 5-25: Signal at position 6 compared with signal at position 3 (dash-dotted curve).
Figure 5-26: Signal at position a compared with signal at position 3 (dash-dotted curve).
Figure 5-27: Signal at position b compared with signal at position 3 (dash-dotted curve).
Figure 5-28: Signal at position c compared with signal at position 3 (dash-dotted curve).
Figure 5-29: Signal at position d compared with signal at position 3 (dash-dotted curve).
Figure 5-30: Signal at position e compared with signal at position 3 (dash-dotted curve).
Figure 5-31: Signal at position $f$ compared with signal at position 3 (dash-dotted curve).
Figure 5-32: Best radial ensemble averaged sine wave measurement (position 3, dash-dotted curve) and ensemble averaged valve position (continuous curve).
Figure 5-33: Transfer function between the measured shaft position and the velocity measured at position 1 where the nonlinear behavior of the jet has a strong effect.
Figure 5-34: Transfer function between the measured shaft position and the velocity measured at position 3 where the nonlinear behavior of the jet is negligible.
Figure 5-35: Transfer function of the servomotor command to the measured shaft position.
Figure 5-36: Transfer function of the servomotor command to the measured flow velocity at position 3.
Figure 5-37: Transfer function of the measured shaft position to the measured flow velocity at position 3.
Figure 5-38: Comparison of data to a model of a pure time delay.
Figure 5-39: Comparison of data to a model of a time delay and a zero.
Figure 5-40: The measured radial position of the jet coming out the injector center nozzle is plotted as function of the dynamic pressure ratio between the injection and the main flow in the wind tunnel.
Figure 5-41: Valve characteristic. The ratio of dynamic pressure between the jet injection and the main flow is plotted as a function of the valve opening for a wind tunnel speed of 30 m/s.
Figure 5-42: Comparison of estimated and measured jet centerline radial position when a discrete sine wave sweep is commanded to the valve.
Chapter 6

Summary, Conclusions, Recommendations

6.1 Summary and Conclusions

- Previous studies showed that significant improvement can be achieved in active control of rotating stall by using jet actuation. A preliminary design of a jet actuator was done to identify the implementation requirement of such a scheme. Theoretical models were developed to estimate the impact of the imperfection of a particular design on the control effectiveness.

- According to the previous modeling a conceptual design of 12 actuators and 24 injectors with 3 holes each was chosen. That injection scheme was mechanically designed to fit onto the MIT GTL three stage compressor and a prototype was built.

- A wind tunnel section was designed to reproduce 1/6th of the circumference of the unwrapped MIT three stage compressor. A probe traversing system and the required measurement apparatus was installed to investigate the flowfield.

- The valve prototype was found to meet the massflow requirement and provide equal injection to two injectors. The actual injector discharge coefficient was
found to be close to the one assumed in the design process, and non-choked injection was verified.

- The ratio of the dynamic pressure at the nozzle exit to the main flow dynamic pressure was identified as the valid parameter to classify the static velocity field. The flowfield was measured for increasing jet dynamic pressure ratio to main flow dynamic pressure ratio. The flowfield has a highly three dimensional shape. The jet centerline penetration measured was 60% of the predicted one.

- The circumferential spreading of a radially averaged flow field was satisfactory. A complete injection scheme was synthesized using the data and it was showed that the injection system is able to produce disturbances significantly bigger than the disturbances precursing rotating stall.

- Highly three dimensional dynamic behavior was identified and observed by using sine waves to evaluate the impact of that nonlinear behavior at different spatial location in the crossflow plane corresponding to the face of the compressor. Transfer function were identified at the spatial positions where the nonlinear behavior has the least effect. The transfer function of the valve position to that measured velocity can be compared with a pure time delay as assumed in the modeling.

- Up to 100 Hz, the jet behavior was showed to be quasi-steady and a prediction of the jet dynamic behavior using the steady state properties of the injection was performed.

### 6.2 Recommendations

- A set of 12 valves and 24 injectors has been manufactured to be tested on the three stage compressor. The injector nozzle still have not been machined.

- Before going into the compressor a new prototype of the injector should be tested to validate the parametrical approach: a three hole injector could be
tested at 45 degrees angle, and the same type with zero angle to act only at the tip could be tested. A five hole injector could be tested, to get more mixing but have less jet penetration because of the smaller size of the injection nozzles.

- The effect of the 3D shape of the jets can have diverse effects on the compressor, in the 2D model approach they will work perfectly. Work is currently pursued at the Gas Turbine Laboratory on the effects of radial and circumferential distortions on the compressor stability.

- The estimation of a 2D transfer function for use in the 2D control model, appears to be more challenging than for the movable guide vanes. Research should be done to model in a simple way the transfer functions of a three dimensional jet actuator.
Bibliography


