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## Abstract

Flow control using impulsive injection from the suction surface of a stator vane has been applied in a low speed axial compressor. Impulsive injection is shown to significantly reduce separation relative to steady injection for vanes that were induced to separate by an increase in vane stagger angle of 4°. Injected flow was applied to the airfoil suction surface using spanwise slots pitched in the streamwise direction. Injection was limited to the near-hub region, from 10 to 36 percent of span, to affect the dominant loss due to hub leakage flow. Actuation was provided externally using high-speed solenoid valves closely coupled to the vane tip. Variations in injected mass, frequency, and duty cycle are explored. The local corrected total pressure loss across the vane at the lower span region was reduced by over 20 percent. Additionally, low momentum fluid migrating from the hub region toward the tip was effectively suppressed resulting in an overall benefit which reduced corrected area averaged loss through the passage by 4 percent. The injection mass fraction used for impulsive actuation was typically less than 0.1 percent of the compressor through flow.

## Nomenclature

$c$	chord
$C_{pt}$	total pressure coefficient
$d$	duty cycle = $\tau/T$
$f$	frequency of harmonic oscillation, Hz
$F^+$	non-dimensional forcing frequency = $fL/\bar{U}_1$
$L$	distance from injection location to vane trailing edge = $0.65c$
$\dot{m}_1$	free stream massflow rate upstream of the vane passage
$\dot{m}_{cjet}$	controlled massflow
$\dot{m}_{jet}$	injected massflow rate
$p'_{jet}$	unsteady pressure internal to the flow control vane
$\bar{P}_{jet}$	time-averaged total pressure internal to the flow control vane
$P_1$	area-averaged total pressure upstream of the vane passage
$P_2$	area-averaged total pressure downstream of the vane passage
$p_1$	area-averaged static pressure upstream of the vane
$PR$	total pressure ratio = $P_2/P_1$
$s$	span
$T$	total time of the actuation period = $1/f$
$\bar{U}_1$	mean free stream velocity upstream of the vane passage
$U_{tip}$	rotor tip speed
$\gamma$	ratio of specific heat
$\rho$	density
$\tau$	on-time of the solenoid
$\phi$	flow coefficient = (mean inlet velocity)/(compressor tip speed)

- $\psi$  pressure rise coefficient =  $\left(\frac{\gamma}{\gamma-1}\right)\left(\frac{P_{ref}}{\rho_{ref}}\right)\left((P_2/P_1)\left(\frac{\gamma-1}{\gamma}\right)-1\right)\left(\frac{g_c}{U_{tip}^2}\right)$
- $\omega$  total pressure loss coefficient

## I. Introduction

Based on the proven success of flow control in external flows it is intuitively assumed that a reasonable expectation of success can be achieved when flow control is applied to the aerodynamic surfaces of turbomachinery. Several research groups have, in fact, produced results which demonstrate that active flow control has the potential to positively impact engine performance.

Carter et al. (ref. 1) investigated the use of an ejector pump approach to simultaneously apply suction and injection to the suction surface of a cascade blade that was separated under nominal flow conditions. They achieved a 65 percent reduction in total pressure loss and a 4.5° increase in turning when injecting 1.6 percent of the freestream flow rate. Successful application of the same approach to a subsonic turbine cascade blade experiencing suction side separation at low Reynolds number has also been reported (ref. 2). Bons et al. (refs. 3 and 4), reported on the application of skewed vortex generator jets to a low-pressure turbine cascade blade to prevent separation at low Reynolds number. They found that pulsed injection with duty cycles as low as 10 percent was just as effective as harmonic pulsing (duty cycle of 50 percent) while providing a large reduction in the time-average injected massflow. This result is similar to that found in an external flow control application (ref. 5).

More recently experiments involving turbomachinery have taken a step further. In a previous effort, Culley et al. (ref. 6), demonstrated in a low speed multi-stage compressor that biased oscillatory injection i.e., unsteady injection superimposed on a mean average flow, was superior to steady injection for stator suction surface separation control. The biased oscillatory injection produced both more loss reduction than steady injection and also required less mass flow to achieve an equivalent loss reduction. Kirtley et al. (ref. 7) developed an experiment around a full annulus of flow control stators using steady injection to successfully demonstrate an ultra-low solidity compressor stator row.

The long-term objective of our research is to develop and demonstrate flow control methods for use in high-speed compressors for separation control within stators. Successful separation control may enable improved performance in two ways; 1) by increasing the range of incidence angles over which total pressure loss is acceptable, and 2) by increasing the loading level at which an acceptable level of loss occurs. The tangible benefits may be an increase in operability and an increase in stator aerodynamic loading, which can lead to reduced engine weight and parts count through lower solidity or even stage reduction.

Ultimately the success of internal flow control lies in the implementation. To truly be effective the flow control method must realize a substantial benefit, i.e., it must largely replace the mechanical design margin and its corresponding weight penalty which is carried throughout the mission profile. Concurrently the increased system complexity must not be a manufacturing or maintenance cost driver nor should it cause untenable failure modes.

## II. Experiment Description

The experiment hardware and setup is described over the next three sections. First is a brief description of the Low Speed Axial Compressor system which is extensively described in the references provided below. Changes to the basic compressor rig configuration are described in two subsequent sections. These can be thought of as internal to the compressor-the flow control vane design, and external to the compressor-the solenoid actuation system.

### A. Low Speed Compressor

The NASA-Glenn Low Speed Axial Compressor (LSAC) facility is used for this study. Air enters the facility through a filtered roof vent, is conditioned for temperature and turbulence, and then passes through a calibrated bellmouth and into the research compressor. Airflow exiting the compressor is controlled by a throttle valve, close-coupled to the collector, and discharged into either an atmospheric or altitude exhaust system. A 1500-hp variable speed motor drives the compressor rotor.

The compressor consists of an inlet guide vane and four identical stages designed for accurate low-speed simulation of the rear stages of a high-speed core compressor. A long entrance duct is used to develop thick endwall boundary layers typical of an embedded stage. The first two stages are used to setup a “repeating stage” environment. The third stage is the focus of research measurements, while the fourth stage acts as a buffer to the exit conditions. The flow path has an outer diameter of 1.219 m and a hub-tip radius ratio of 0.80. All stators have inner shrouds with a single labyrinth seal-tooth in the shrouded stator cavity. The nominal rotor tip and stator seal

clearances are 1.4 and 0.6 percent of span, respectively. Rotor tip speed is 61 m/s and nominal axial velocity,  $\bar{U}_1$ , is on the order of 25 m/s. The increased size and low speed of this facility enables intra-stage surveys of the flow field thus making possible an increased understanding of the complex flow phenomena within multi-stage axial compressors. A complete description of the LSAC facility is given by Wasserbauer (ref. 8).

The blading used for the current tests is based on the Rotor B/Stator B blading designed by General Electric for the NASA Energy Efficient Engine program. Details of the original designs are reported by Wisler (ref. 9). The stators are designed by applying modified 65-series thickness distributions to modified circular-arc meanlines. The NASA stators are slightly modified from the GE design to accommodate a difference in hub-tip radius ratio between the GE and NASA low speed compressor facilities. The NASA stator features a solidity of 1.38, an aspect ratio of 1.32, a stagger angle of 42° and a camber of 40.5°. The stator chord is 9.4 cm. Stators are sealed at both the hub and tip junctions with the flow path.

Overall performance is expressed in terms of the average pressure rise coefficient,  $\psi$ , and flow coefficient,  $\phi$ . The average pressure rise coefficient is determined from inlet and outlet static pressure measurements on the hub and casing. The flow coefficient is defined as the measured mean inlet velocity normalized by the rotor tip speed. The mean inlet velocity is determined from static pressure measurements at the exit of the inlet bellmouth using a previously determined discharge coefficient, and is corrected for humidity. Vane element performance is calculated from total pressure measurements acquired with miniature (1.64 mm) Kiel head probes and static pressure and flow angle measurements acquired with 18° wedge probes. All pressure measurements are acquired at midgap of the rotor-stator spacing (the spacing is approximately 35 percent of axial chord) and are referenced to stagnation conditions measured in the inlet plenum of the facility. The following measurement accuracies are reported by Wellborn (ref. 10):  $\Delta\psi = 1.09\%$ ,  $\Delta\phi = 0.39\%$ ,  $\Delta\omega = 2.1\%$ .

In the third stage of the compressor there is a removable window in the casing which allows simplified access to four vanes in the annulus. An experiment reconfiguration can therefore be achieved without disassembling the casing by simply removing the window and vanes as a unit. Although there are obvious limitations in changing the blading over such a short segment of the annulus this has proven to be an effective means of altering the local vane loading conditions.

## **B. Flow Control Vanes**

Internal modifications to the LSAC configuration are described below. This includes specially modified vanes for flow control, vane mounted instrumentation used for experiment control, and the configuration of the stage-3 window mentioned above.

The standard vanes used in the LSAC were fabricated from a solid construction of fiberglass with tip and hub-end trunions which serve to anchor the vane in the casing and hub ring, respectively. A threaded post of solid steel at the hub-end allows for the vane stagger angle to be adjusted and locked. The standard blading used in the LSAC has been modified to accommodate a flow path for injection air originating from outside the casing to the suction surface of the vane.

The four vanes under the stage-3 window described in this study are manufactured using a rapid prototyping process. This technique produces airfoils similar to the standard LSAC components in that the aerodynamic surfaces and mounting points are the same as the nominal vane (within the tolerance limits of the fabrication process). However, injection through the vane surface requires the fabrication of flow passages within the narrow geometry of the vane. This is accomplished cost effectively using rapid prototyping because it enables the fabrication of integral airfoils which include embedded internal flow passages. There are limitations to the process that prevent the precise replication of feature sizes smaller than 0.375 mm. Absolute accuracy of the part geometry is also not assured. However, careful location of the parts to be grown within the rapid prototyping machine minimizes these issues and enhances the surface finish and part strength. A CAD representation of the vane with the internal plenum is shown in figure 1.

To further improve the surface finish of the flow control vanes a thin coat of primer has been applied, yielding a surface finish that is comparable to that of the standard fiberglass LSAC vanes. The rapid prototype flow control vanes have been demonstrated to be entirely capable of surviving within the LSAC environment.

The injection location on the suction surface is 35 percent of chord and was previously determined to be upstream of the separation line in a wind tunnel study performed by the Illinois Institute of Technology using a custom-designed NACA airfoil whose suction surface pressure distribution closely matched that of the standard LSAC stator vane. The injection angle is pitched at  $30^\circ$  relative to the surface tangent to impart stream-wise momentum to the flow.

These vanes have been used in a previous study (ref. 6) and were designed with a span-wise slot divided into six segments separated by support webs in order to maintain structural rigidity of the vane skin. The slot width and vane skin thickness are each 0.63 mm (0.7 percent of chord). The span-wise coverage was originally designed to extend from approximately 10 to 90 percent of span, being constrained by the vane cavity design. In the current effort four of the existing six slots have been sealed in order to limit injection to the near-hub region. The two remaining open slots represent coverage from 10 to 36 percent of span.

One of the flow control vanes is also equipped with pressure instrumentation ports. These ports allow the measurement of pressure at 5 locations on the vane; the internal vane cavity pressure, and 4 static pressure measurements along the suction surface. The internal cavity pressure is used to correlate the injection forcing pressure with injection mass flow. The suction surface taps are used to monitor the state of separation along the vane surface. Each static port is at 44 percent of span, and at 5, 74, 82, and 90 percent of chord, respectively.

Of the four rapid prototype blades in the LSAC stage-3 window only the middle two are configured as flow control vanes. The outer two vanes are not plumbed for injection. Under nominal design conditions the flow over the LSAC stator vanes is not prone to strong separation prior to compressor stall. Therefore, to enable reasonable operating conditions, three of the four vanes under the window are re-staggered to approximately  $4^\circ$  from nominal to induce early flow separation. Surface pressure measurements acquired from the instrumented stator vane indicate that at this  $4^\circ$  re-stagger the vanes are not separated under open-throttle conditions but suffer significant separation at lower flow coefficients. The remaining 49 vanes in the annulus are installed at the nominal design stagger. The vane restagger is determined from changes in the circumferential position of the vane trailing edge as the vane is rotated about the trunion axis and is accurate to  $\pm 1^\circ$ . The vane configuration was never changed for the duration of the experiment. Figure 2 shows a photograph of the stage 3 window assembly with flow control vanes.

### C. Solenoid Actuation

The solenoid actuation system describes a series of components which are used to deliver a metered source of injection air to the tip-end of a pair of flow control vanes while varying the duty cycle and frequency of the modulated flow. This system is shown schematically in figure 3.

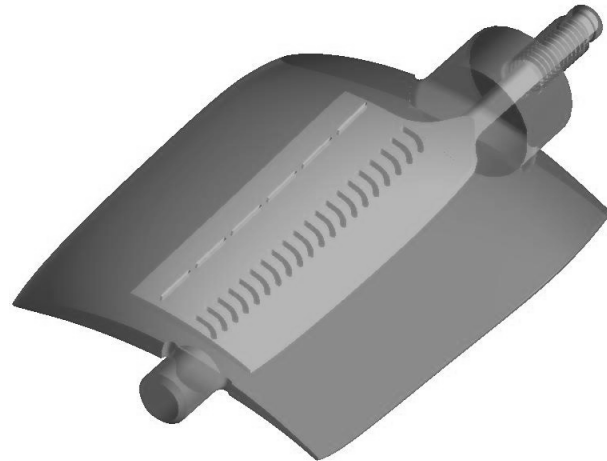


Figure 1.—Computer Aided Design drawing of the rapid prototype flow control stator vane. Injection air enters at the tip-end, travels to an internal plenum, and out the slots located on the suction surface at 35 percent chord. Out of six span-wise slots the first four from the tip end have been sealed.

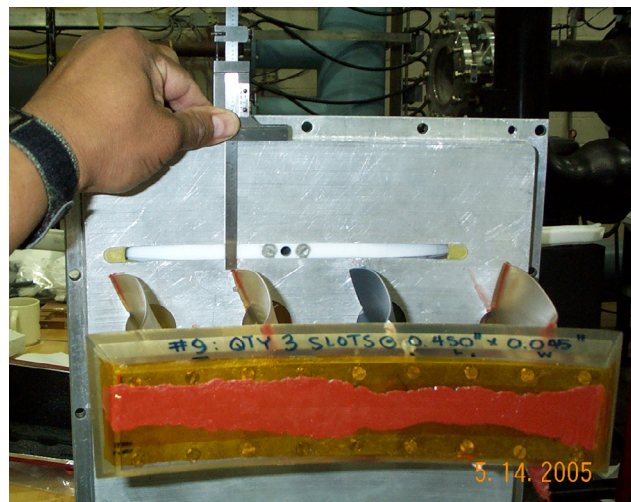


Figure 2.—Photograph of the stage-3 window with rapid prototype vanes installed. The stagger angle is set by the trailing edge location.



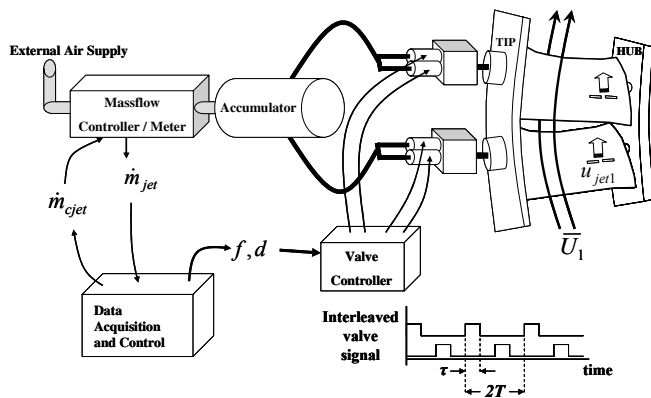


Figure 3.—A graphical representation of the solenoid injection system. Air flow is controlled by a mass flow controller prior to entering an accumulation tank which stabilizes the pressure. The solenoids are fed from the tank. Each pair of solenoids is operated in an interleaved fashion to extend operating frequency range.

Two solenoid valves are connected to each flow control vane to enable an increased frequency of operation. This is accomplished by operating each valve for the desired on-time, which must be 50 percent of the period or less, and 180° out of phase with the operation of the other valve, as shown in figure 3.

For these experiments two operating modes were used. The first used a fixed duty cycle, which is defined as the valve on-time,  $\tau$ , with respect to the total time,  $T$ . Total time is with respect to the periodic input received by the vane noting that it is fed by two interleaved solenoid valves each operating over a period of  $2T$ . Duty cycle was fixed at either 50 or 25 percent. The second operating mode enabled variable duty cycle by fixing the on-time and varying the period.

The solenoid valves used in this experiment are specially designed for high speed actuation when coupled to the electronic drive unit which powers them. This is accomplished by actually overdriving the solenoid coils for a few microseconds before reverting to the nominally rated coil voltage. Even so, this is a non-standard application. The precise response of the solenoid valve is inferred from bench measurements of the injection flow velocity near the vane slot exit and high speed pressure measurements in the vane cavity below the injection slot. An example is depicted in figure 4. It is certain that at very high frequencies of operation the solenoid does not travel the entire design stroke length. This is supported by an observed increase in the source pressure driving the valves. During the experiment maximum injection mass flow rates were limited to maintain reasonable values of source pressure in the accumulator.

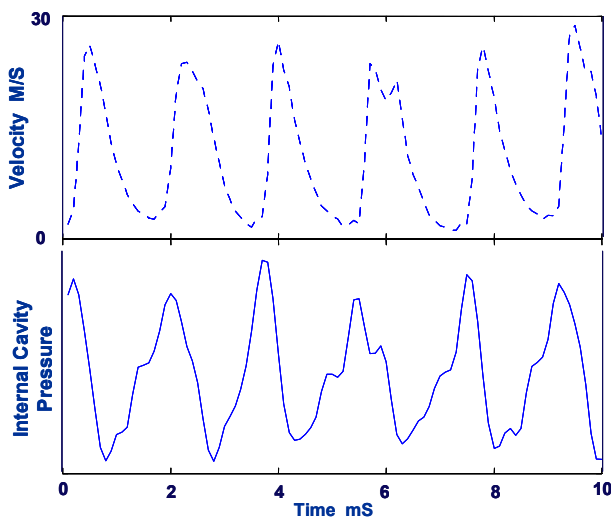


Figure 4.—This is an example of the injection velocity near the slot exit and the instantaneous pressure in the vane cavity with respect to time. The solenoids are being driven at 550 Hz, 50 percent duty cycle.

In all cases the source of injected fluid is a filtered shop air supply whose capacity far exceeds the mass flow requirement of the experiment. The 862 kPa source is regulated down to approximately 344 kPa where it supplies a mass flow controller. The mass flow controller provides a convenient method of both metering and measuring the injection mass flow. The full scale range of the device is 4.31 g/s with an accuracy of  $\pm 1$  percent of full scale.

The mass flow meter is followed by a large pressure vessel, approximately 5500 cm<sup>3</sup>, which serves two purposes; as an accumulator to provide a stable operating pressure for the solenoid valves, and to isolate the meter from resonance conditions which could affect its accuracy. The accumulator is also instrumented with a 344 kPa pressure transducer to monitor the source pressure feeding the solenoid valves. Independent supply tubes, 6.35 mm ID, originate at the accumulator and Tee off to feed two high speed solenoid valves attached to the tip end of the two flow control vanes.

Two solenoid valves are connected to each flow control vane to enable an increased frequency of operation. This is accomplished by operating each valve for the desired on-time, which must be 50 percent of the period or less, and 180° out of phase with the operation of the other valve, as shown in figure 3. For these experiments two operating modes were used. The first used a fixed duty cycle, which is defined as the valve on-time,  $\tau$ , with respect to the total time,  $T$ . Total time is with respect to the periodic input received by the vane noting that it is fed by two interleaved solenoid valves each operating over a period of  $2T$ . Duty cycle was fixed at either 50 or 25 percent. The second operating mode enabled variable duty cycle by fixing the on-time and varying the period. The solenoid valves used in this experiment are specially designed for high speed actuation when coupled to the electronic drive unit which powers them. This is accomplished by actually overdriving the solenoid coils for a few microseconds before reverting to the nominally rated coil voltage. Even so, this is a non-standard application. The precise response of the solenoid valve is inferred from bench measurements of the injection flow velocity near the vane slot exit and high speed pressure measurements in the vane cavity below the injection slot. An example is depicted in figure 4. It is certain that at very high frequencies of operation the solenoid does not travel the entire design stroke length. This is supported by an observed increase in the source pressure driving the valves. During the experiment maximum injection mass flow rates were limited to maintain reasonable values of source pressure in the accumulator.

Impulsive injection is thus defined as a special case of unsteady injection where the constant component of the injected flow is at or near zero. This constant component of the injected flow, typically due to leakage in the actuator, represents a continuous source of low momentum fluid added to the separation region.

### III. Experiment Results

All results reported herein are obtained at design speed and with a flow coefficient of 0.35, which is sufficiently far away from stall yet produces well separated flows across the restaggered flow control vanes. Examining figure 5 we can observe the change in the downstream wake which occurs when a vane is restaggered under these conditions. Circumferential surveys of the vane passage are performed at 30 percent span. The nominal vane referred to in the figure is of solid fiberglass construction and is located in the same stage but on the opposing side of the compressor from the restaggered flow control vane. By analyzing the changes to the baseline wake pattern during injection we can gain a qualitative understanding of the effect of impulsive actuation.

In figure 6 a series of wake profiles are displayed which demonstrate the effect of steady injection over

a range of low injection mass fractions. Consistent with our previous work (ref. 6) the low injection rates only serve to aggravate the separation by introducing additional low momentum fluid to an area of separation. In this case the injection was only increased to a point where the wake nearly matched the non-blowing profile. If the injection rate was sufficiently increased we would expect the wake to begin decreasing in width and depth until overblowing was apparent.

Contrast the previous wake profiles with figure 7 which depicts impulsive injection at a frequency of 500 Hz and 50 percent duty cycle. In this instance a substantial benefit in the wake profile is observed. Interestingly, the largest mass fraction used is approximately half that of the lowest steady blowing case presented in the previous figure. Even more curious is the fact that the total pressure deficit in the wake was never increased by impulsive injection as it was by steady injection. This last condition holds true for every impulsive injection case with duty cycle 50 percent or less which was attempted over the course of the experiment.

A lower injection mass fraction was expected in this study for two reasons; 1) the spanwise coverage of injection on the vane surface was decreased by a factor of 67 percent from previous experiments (ref. 6), and 2) the solenoid valves exhibit little to no leakage when in the off condition since they are designed as shutoff valves (although this may not hold true at very high frequency for reasons previously described). The curves in figure 7 are at 50 percent duty cycle. By reducing the duty cycle it is expected that mass fraction can be further reduced.

A reduction of the wake total pressure deficit at low injection massflows was an unexpected result. From several previous works (refs. 6 and 11 to 13) it

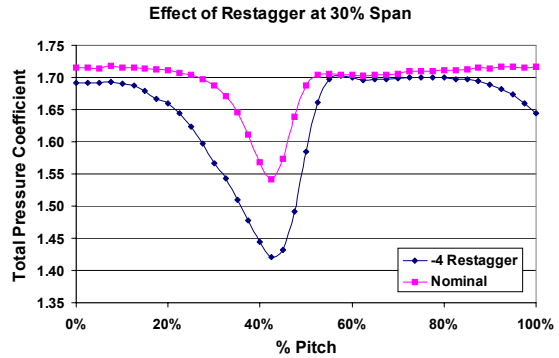


Figure 5.—A change in the vane stagger angle is used to induce separation locally while allowing the compressor to operate away from stall. The staggered vane wake is wider and deeper indicating a separated condition. The narrower wake is from a vane at nominal design stagger.

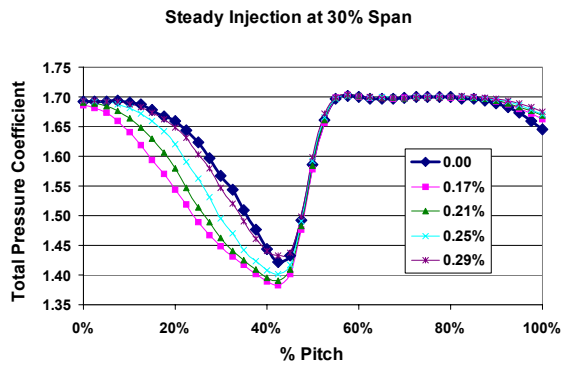


Figure 6.—Steady injection, shown at various mass fractions, tends to increase the size of the wake at low blowing values because it adds additional low momentum fluid. The highest blowing rate attempted here almost returns the wake to the baseline condition.

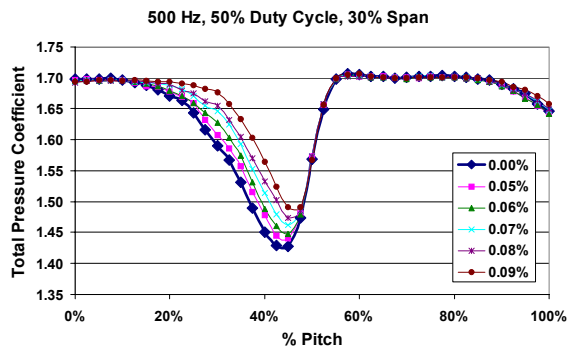


Figure 7.—Impulsive injection provides an immediate benefit to the wake at very low injection mass fraction. Blowing rates were not reduced sufficiently to observe a worsening of the separation as inferred from the wake.

is known that unsteady injection is a more effective method of mixing which is the primary mechanism which reattaches the separated flow. Impulsive injection is meant to convey that there is no residual injection flow during the actuator off-time in contrast to most unsteady implementations which have a constant mean residual flow. At the resulting ultra-low mass fractions it appears that little, if any, momentum imparted from injection is used to directly energize the stalled fluid. Rather, it is believed that the impulsively injected fluid more effectively mixes with the freestream and it is the energy in the freestream that clears the separation condition. With the existing experiment setup we were unable to sufficiently reduce injection mass flow to ever observe a detrimental change in the wake.

While wake response plots provide useful qualitative information loss production is the primary metric used for evaluating compressor performance. The conventional definition of loss coefficient for a vane passage with no mass addition is,

$$\omega = \frac{P_1 - P_2}{P_1 - p_1}$$

Here,  $P_1$  and  $P_2$  are the area averaged total pressure upstream and downstream of the vane and  $p_1$  is the area averaged static pressure upstream of the vane. This must be corrected to account for the injected flow, which can have a different total pressure than the fluid in the vane passage. With injection present, there are two loss mechanisms within the vane passage: 1) the viscous dissipation due to the vane surface boundary layers; and 2) the mixing loss generated between the injected jet and the freestream flow.

As discussed by Brocher (ref. 14), an energy balance across the vane row yields the following total pressure loss coefficient which represents the dissipation generated within the vane passage per unit exit massflow,

$$\omega = \left( \frac{(\dot{m}_1 P_1 + \dot{m}_{jet} \bar{P}_{jet}) - P_2}{\dot{m}_1 + \dot{m}_{jet}} - P_1 - p_1 \right) \left( 1 + \frac{\dot{m}_{jet}}{\dot{m}_1} \right)$$

where  $\bar{P}_{jet}$  is the mean vane cavity pressure,  $\dot{m}_{jet}$  is the injected massflow rate, and  $\dot{m}_1$  is the massflow rate of the freestream entering the vane passage. All loss figures presented in this paper are cast in terms of this corrected loss coefficient.

Figure 8 shows quantitatively the local reduction in loss at 30 percent span for a range of actuation conditions. Various frequencies and injection mass fractions were evaluated, all at 50 percent duty cycle. The trend line follows the results for the circumferential wake surveys depicted in figure 7 from which loss reduction was previously inferred. The overall trend is a reduction in loss for an increasing injection mass fraction.

Closely examining the data there do appear to be frequency affects with 400 Hz, 50 percent duty cycle being the most effective frequency of actuation. In fact most actuation frequencies performed better than the 500 Hz originally chosen. The control parameter used in the experiment was injection mass flow; therefore there are a few nicely grouped points over a range of frequencies which can be used to investigate the effect of frequency.

Instead of plotting the entire wake to investigate frequency dependence, in figure 9 we chose to look at a single fixed point in the wake at 40 percent pitch. Plotting the total pressure coefficient,  $C_{pt}$ , for the point in the wake we get a reliable indication of the change in size of the wake and therefore an inference of loss as injection mass is held constant. In an ideal situation, since the duty cycle is constant at 50 percent, the source pressure upstream of the solenoids would remain constant as well. This is not the case, however, as it continues to rise with an increase in frequency due to the finite response time of the valve. Even with a continued increase in pressure there is a maximum wake response near the frequencies of 400 and 700 Hz.

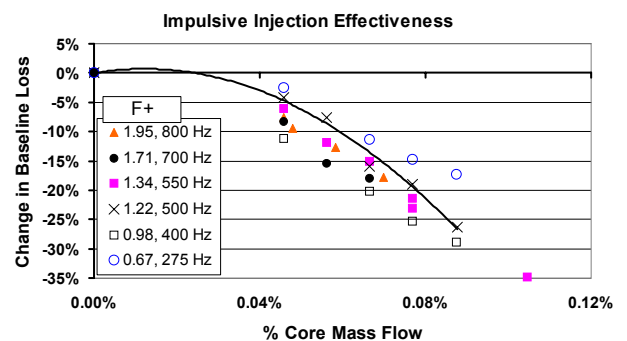


Figure 8.—The loss reduction from the baseline non-blowing condition for various injection mass fractions at various frequencies of actuation. All curves depicted are at 50 percent duty cycle and 30 percent span. The trend line is plotted for the family of wake profiles shown in figure 7.

This same response is observed in figure 10 which plots the static pressure rise over the flow control vane suction surface for the same set of data points in figure 9. One function of the stator annulus is to provide diffusion which results in a static pressure rise across the stator. A separated flow presents blockage in the diffusing path thereby limiting the static pressure rise. The state of separation, and therefore loss, can be inferred in this manner.

One concern in performing these experiments is the frequency response characteristics of the actuation system. By closely coupling the actuators to the flow control vanes many of the resonance issues can be mitigated. Potential sources of resonance could be the Helmholtz frequency of the vane cavity, estimated at roughly 2000 Hz, or pipe resonance whose quarter wave length may excite a roughly 500 Hz response based on the stand-off distance of the solenoids from the injection slot.

The aerodynamic response of the airfoil and its flow field is another consideration. The non-dimensional forcing frequency is defined as,

$$F^+ = fL/\bar{U}_1$$

where the characteristic length is defined as the distance from the point of injection to the trailing edge of the vane. For this configuration an  $F^+ = 1$  occurs at approximately 410 Hz which is below any expected actuation system resonance.

Interestingly, the peak response occurs near the  $F^+ = 1$ . The data set is sparse, however, and it is difficult to draw precise conclusions.

Additional test points were run to investigate changes in duty cycle, although this ability was limited due to the finite valve response. The actuation frequency of 275 Hz ( $F^+ = 0.67$ ) was chosen because the valves could feasibly respond without an inordinate increase in source pressure upstream of the valves.

Very interesting results occurred by reducing the duty cycle from 50 to 25 percent at 275 Hz, shown in figure 11. Previously, this actuation frequency produced the least improvement in reducing loss. At 25 percent duty cycle and the same repetition rate the loss reduction effectiveness becomes greater than the best performance previously achieved at 400 Hz ( $F^+ = 0.98$ ), 50 percent duty cycle.

A more surprising result occurred, however, when fixed pulse width injection was used. Fixing the on-time,  $\tau$ , of the solenoid at 1.5 ms and varying the actuation frequency, we could effectively reduce the duty cycle continuously over a broad range. In this instance we maintained a constant source pressure upstream of the solenoid valves. This seemed to be

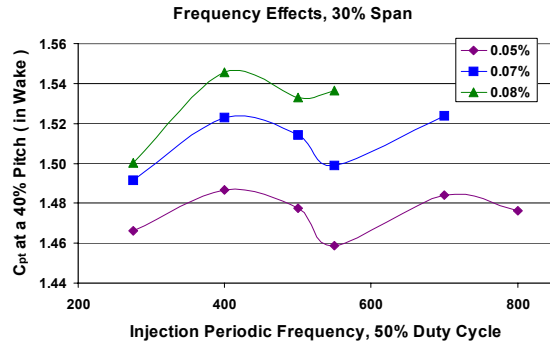


Figure 9.—Shown is the change in the wake response to impulsive injection at various frequencies and a fixed 50 percent duty cycle. The peak response for a fixed mass fraction appears near 400 Hz and again around 700 Hz.

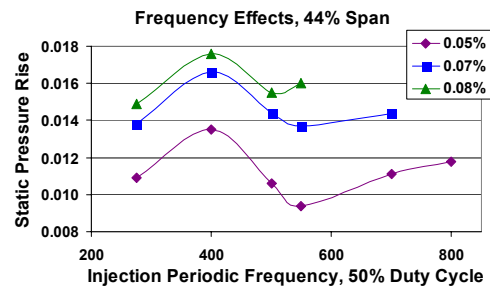


Figure 10.—The same change in wake response with frequency at a fixed 50 percent duty cycle and mass flow rate can be observed with static pressure rise across the flow control vane surface at 44 percent span.

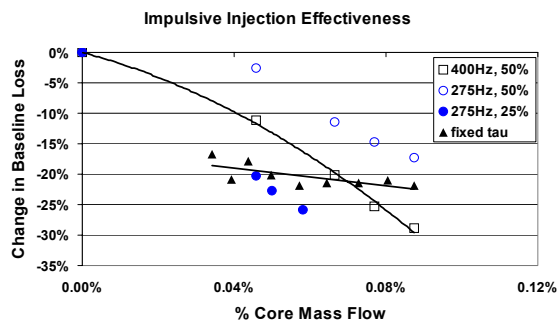


Figure 11.—Loss reduction for impulsive injection performance at reduced duty cycle is shown. Reducing the duty cycle to 25 percent at 275 Hz produces substantial gain in loss reduction. Also shown is a fixed on-time pulse width of 1.5 ms at repetition rates from 350 to 125 Hz. Similar loss reduction performance is achieved at very low repetition rates.

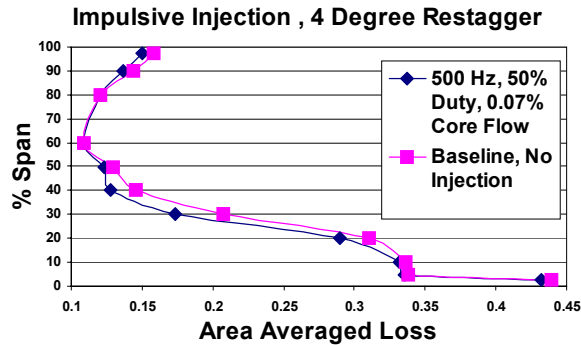


Figure 12.—Impulsive injection reduces the amount of loss in the near hub region. The calculated area-averaged loss was reduced, in this case, by 4 percent.

the area of injection which extends from 10 to 36 percent of span, corrected for injected mass, was reduced by 4 percent.

#### IV. Conclusion

The effect of impulsive injection actuation on the stator vane suction surface has been demonstrated in a low speed multistage axial compressor. Using high speed solenoid valves at 50 percent duty cycle a very small amount of air was injected from the vane suction surface. The characteristic of the injected flow was a relatively sharp pulse with no constant component or residual “leakage” flow during the valve off-time thus defining impulsive injection.

The injection was shown to reduce the width and depth of the downstream wake as measured by a time averaged total pressure probe. The wake response in relation to the baseline, non-blowing condition was shown to be a reliable indication of injection performance. Static pressure rise on the vane suction surface was also shown to reliably correlate with loss performance.

At 30 percent of span a comparison of impulsive injection at 500 Hz ( $F^+ = 1.22$ ), 50 percent duty cycle was made with steady injection. It was found that impulsive injection produce 25 percent local corrected loss reduction at 0.09 percent of the core mass flow. Steady injection, at a mass fraction of 0.29 percent, barely returned the wake to the baseline, non-blowing wake shape. Since substantially more mass is introduced in the latter case a corrected loss calculation would reveal a net increase in loss for steady blowing.

The solenoid actuation system was closely coupled to the flow control vanes to minimize resonance issues. The response of the flow field appeared to peak at 400 Hz ( $F^+ = 0.98$ ) which is closely aligned with the non-dimensional forcing frequency,  $F^+ = 1$ .

The duty cycle of impulsive injection was varied over a wide range (50 to 19%) using a fixed on-time and increasing the period between pulses. Pressure upstream of the solenoid valves was kept constant to maintain a fixed injection pulse velocity. The loss performance remained relatively flat even though the reduced frequency,  $F^+$ , was reduced to as low as 0.31.

Impulsive injection appears to approach the zero-mass injection performance of the synthetic jet. We propose that instead of directly providing the energy to reattach separated flow it leverages the existing energy in the freestream by effectively perturbing the main flow to enhance mixing and thereby increasing the momentum in the boundary layer.

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<b>13. ABSTRACT (Maximum 200 words)</b>  Flow control using impulsive injection from the suction surface of a stator vane has been applied in a low speed axial compressor. Impulsive injection is shown to significantly reduce separation relative to steady injection for vanes that were induced to separate by an increase in vane stagger angle of 4 degrees. Injected flow was applied to the airfoil suction surface using spanwise slots pitched in the streamwise direction. Injection was limited to the near-hub region, from 10 to 36 percent of span, to affect the dominant loss due to hub leakage flow. Actuation was provided externally using high-speed solenoid valves closely coupled to the vane tip. Variations in injected mass, frequency, and duty cycle are explored. The local corrected total pressure loss across the vane at the lower span region was reduced by over 20 percent. Additionally, low momentum fluid migrating from the hub region toward the tip was effectively suppressed resulting in an overall benefit which reduced corrected area averaged loss through the passage by 4 percent. The injection mass fraction used for impulsive actuation was typically less than 0.1 percent of the compressor through flow.				
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