Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-46042

SELF-REGULATING CASING TREATMENT FOR AXIAL COMPRESSOR STABILITY ENHANCEMENT

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ABSTRACT

The operating range of an axial compressor is often restricted by a safety imposed stall margin. One possible way of regaining operating range is with the application of casing treatment. Of particular interest here is the type of casing treatment which extracts air from a high pressure location in the compressor and re-injects it through discrete loops into the rotor tip region. Existing re-circulation systems have the disadvantage of reducing compressor efficiency at design conditions because worked flow is unnecessarily re-circulated at these operating conditions. Re-circulation is really only needed near stall. This paper proposes a self-regulating casing treatment in which the re-circulated flow is minimized at compressor design conditions and maximized near stall. The self-regulating capability is achieved by taking advantage of changes which occur in the tip clearance velocity and pressure fields as the compressor is throttled toward stall.

In the proof-of-concept work reported here, flow is extracted from the high pressure region over the rotor tips and re-injected just upstream of the same blade row. Parametric studies are reported in which the flow extraction and re-injection ports are optimized for location, shape and orientation. The optimized design is shown to compare favorably with a circumferential groove tested in the same compressor. The relationship between stall inception type and casing treatment effectiveness is also investigated.

The self-regulating aspect of the new design works well: stall margin improvements from 2.2 to 6.0% are achieved for just 0.25% total air re-circulated near stall and half that near design conditions. The self-regulating capability is achieved by the selective location and orientation of the extraction hole; a simple model is discussed which predicts the optimum axial location.

INTRODUCTION

Flow instability in an aero-engine compressor leads to rotating stall or surge. Stall is avoided by restricting the compressor operating point to a safe distance from the stability limit. In some cases the application of casing treatment to the outer wall of the compressor annulus over the rotor tips may be used to regain a part of the restricted operating range. In the past, casing treatment has not been widely used in axial compressors because the efficiency penalty often outweighs the benefits of stall margin improvement.

A historical survey of casing treatment designs is given by Hathaway [1]. Traditional forms of casing treatment applied over the rotor blade tips include axially skewed slots and circumferential grooves; see, for example, Osborn et al. [2], Seitz [3], and Houghton and Day [4]. In general, axial slots provide greater stall margin improvement than circumferential grooves, however, this often comes at a higher efficiency penalty; see Prince [5] and Fujita and Takata [6].

Another common form of casing treatment involves extracting high pressure air from a downstream location in the compressor and re-injecting the air just upstream of the critical blade row. The air may be injected passively (steady injection driven by the prevailing pressure difference) or actively (pulsed injection using stabilizing feedback from measurements in the flow field).

Active control is usually tailored to a specific stalling mechanism (see, for example, Gysling and Greitzer [7] and Weigl et al. [8]) or applied to remedy a restricted operating range due to inlet distortion (see Spakovsky et al. [9, 10]). As axial compressors often exhibit a wide range of stalling mechanisms (other than spikes and modes), a general purpose passive treatment is a better option - provided, of course, that the problem of efficiency loss can be overcome.

A comprehensive numerical and experimental study on steady injection was published by Suder et al. [11]. They developed a system that injected 2% of annulus flow along the casing to increase the stability margin by 6%. This study showed that the flow mechanism responsible for stall delay was solely due to the increased mass averaged axial velocity in the tip region. They concluded that the steady tip injection increases stability by unloading the rotor tip and thereby reducing blockage.

Strazisar et al. [12] demonstrated 4% to 6% stall margin improvement by injecting 0.7% to 1.3% of annulus flow in a highly loaded transonic rig. They investigated both steady and active blowing in conditions in which the compressor stalled with both modes and spikes depending on shaft speed. They showed active control achieved the same stalling flow rate as with steady injection, but with a reduced injection mass flow rate. Their results agree well with the results of Suder et al. [11] to indicate that an increase in mass averaged axial velocity in the tip region is correlated to stall margin improvement.

A large focus in the above research was on minimizing the amount of re-circulated mass flow required to achieve an adequate gain in stall margin. This is because, by entraining and re-circulating worked flow, casing treatments inevitably lead to a reduction in efficiency. Minimizing the re-circulated flow reduces the efficiency loss. Traditionally, re-circulating designs have relied on air extracted from ports downstream of the rotor row in question, or from even further back in the compressor. In such cases the downstream pressure driving the re-circulated flow is relatively constant, and hence flow is re-circulated at all operating flow rates - including at design where loss of efficiency can least be tolerated. (Even a half percent loss of efficiency in an aero-engine compressor is unacceptable.)

The current work deals with this problem by seeking a better air extraction location, one that will selectively circulate a small amount of air at design conditions and a larger amount near stall. The approach adopted here is to self-regulate the re-circulated air by making use of the changes in the over-tip pressure field that occur as the compressor is throttled toward stall.

A similar, strategically placed, self-regulating re-circulation system is often used in centrifugal compressors; see, for example, Hunziker et al. [13]. In a radial machine, the result of flow re-circulation is to improve the operating range by re-matching flow volumes, much as is done by handling bleed in axial compressors. The similarity between the two systems is limited, however, because the amount of air re-circulated is different (5% in the radial case as opposed to 0.25% in the current axial case) and stall margin gain is achieved by flow re-matching rather than delaying stall inception.

OBJECTIVES AND SCOPE OF PRESENT STUDY

The focus of the present work is to produce a proof of concept design for a self-regulating re-circulating casing treatment. The idea is to have a localized re-circulation loop through which little air would flow at design conditions but through which more air would flow as the compressor approaches the stability limit. The objective is to provide a low weight and low complexity design for the first stage of a high pressure compressor. The proposed design is evaluated based on its ability to provide useful stall margin improvements (2 to 4%) coupled with little ($< 0.5\%$) or no loss of efficiency at design conditions. The present study includes efficiency measurements (not often reported in the literature), and a direct comparison with another form of casing treatment evaluated in the same machine.

The casing treatment proposed here has three principal components: an extraction hole, an injection nozzle and a connecting loop. A sketch of the system is shown in Fig. 1. The extraction hole (1) bleeds flow from over the rotor tips. The flow passes through the connecting loop (2) and is then directed by the injection nozzle (3) toward the leading edge tips of the same blade row. The pressure difference between the extraction hole and the injection nozzle drives the flow through the loop. Discrete re-circulation loops are used as opposed to a continuous cavity because the amount of air to be re-circulated is very small.

The proposed design is more compact than other systems that bleed air from locations further downstream in the machine (for example Strazisar et al. [12]). Extraction from within the rotor row itself will have a lower driving pressure as a result. However, both Suder et al. [11] and Strazisar et al. [12] demonstrated that reasonable stall margin improvements are possible with small amounts of injected mass flow.

The current work will be presented in five steps.

1) Experimental measurements in the tip clearance gap are presented. These results are used in a simple model to explain the self-regulating capability of the new casing treatment and predict the best extraction hole location.

2) The optimization of the extraction hole is presented. The over-tip location and shape are tuned to changes in the tip clearance flow field as the compressor is throttled from design to near stall conditions. The location and shape are chosen such that re-circulation through the loop is minimized near design conditions, and maximized as stall is approached. This should give rise to a casing treatment which will maintain compressor efficiency and

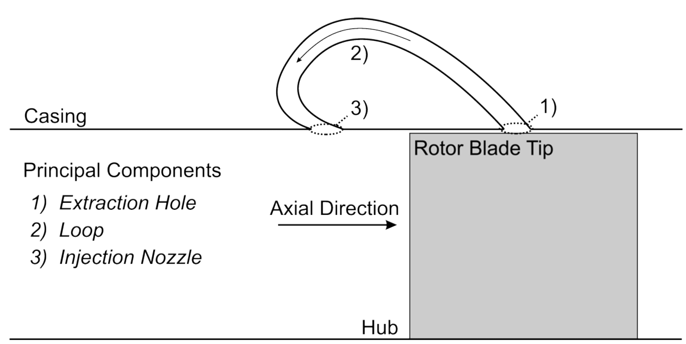

FIGURE 1. New extraction/re-injection casing treatment design.

will extend the operating range of the machine.

3) The injection nozzle is optimized and the complete casing treatment is tested in full annular form using 12 or 24 discrete re-circulation loops. Two injection nozzle geometries are considered, and the optimal angle of injection is investigated.

4) The optimized extraction/re-injection design is compared to a circumferential groove. This allows a rare comparison of two casing treatment designs in the same machine.

5) The effectiveness of the casing treatment is examined when the mechanism of stall inception in the compressor is changed.

EXPERIMENTAL RIGS

Two single stage low speed compressors were used for this work. Past experience [1] indicates that casing treatment function is not sensitive to compressibility effects; i.e., the function in a low-speed compressor is indicative of high-speed performance.

The "Deverson compressor" is a large 5 foot diameter single stage test rig modeled on an embedded stage in a high-pressure compressor. This compressor is used to obtain detailed over-tip pressure and velocity measurements and to perform the extraction hole optimization work. The smaller compressor used for full annular testing of the casing treatment designs is the so called "Mini-Deverson compressor." This machine is a scaled down model of the larger Deverson compressor.

Both compressors are nearly identical in specifications and performance; details are shown in Tab. 1. The Reynolds number of the smaller rig is lower; however, this does not affect the similarities in performance of the two compressors. The pressure and efficiency characteristics of the two machines are nearly identical, as is the stalling behavior (spikes). More details can be found in Dickens [14].

The performance of the compressor is evaluated using pressure rise and efficiency characteristics. The pressure rise coefficient, Ψ, is expressed as $(P_{s,exit} - P_{o,in})/(1/2\rho U_{mid}^2)$. The flow coefficient, ϕ, is defined $V_{x,in}/U_{mid}$. The stall margin improvement, SMI, is defined as the percent reduction in stalling flow coefficient relative to the smooth wall case. The efficiency is calculated from the shaft torque and the change in static pressure across the stage: $\eta = (\dot{m}\Delta P_s)/\rho T\omega$.

Characteristics were measured by continuously recording the inlet and casing pressures as the compressor was slowly throttled into stall. The efficiency and pressure rise characteristics for each test case were repeated several times to ensure representative results. For each parameter studied, back to back tests were performed and the smooth wall case was measured at the beginning and end of each set of measurements. In this way, it was possible to ensure results that are reliably comparative.

Stall inception measurements were recorded using six fast response pressure transducers. The transducers were evenly distributed around the annulus and mounted flush with the casing at the rotor leading edge.

TABLE 1. COMPRESSOR RIG FEATURES.

	Deverson	Mini-Deverson
IGV Blades	49	49
IGV Inlet Swirl	25°	25°
Rotor Blades	51	51
Stator Blades	49	49
Casing Diameter	1.524 m	0.45 m
Hub/Casing Ratio	0.8	0.8
Design Flow Coeff.	0.51	0.51
Stage Loading	0.46	0.46
Stage Reaction	52%	54%
Rotor Tip Clearance	1.2% Chord	1.4% Chord
Reynolds Number[1]	$3.1x10^5$	$1.6x10^5$

ROTOR OVER-TIP FLOW FIELD

This section presents a selection of results from a detailed experimental investigation of unsteady flow in the tip clearance gap (Weichert[2]). The investigation yielded high resolution pressure and velocity maps of the flow in the tip gap region that formed the basis of the present work. From the changes observed in the flow field as the compressor is throttled towards stall, the possibility arises of a new casing treatment design that will tune itself to the prevailing compressor operating conditions. The concept of the casing treatment is first described with the assistance of the over-tip results. Then, a simple model based on these experimental results is used to test the self-regulating hypothesis which forms the basis of the new casing treatment design.

Experimental Rotor Over-tip Flow Maps

The investigation was carried out on the Deverson compressor. The large size of the machine means that the tip clearance gap is also relatively large (1.4 mm), making it possible to take detailed measurements in the gap between the tips of the moving blades and the casing wall. Measurements were carried out at two flow coefficients: design and near stall conditions.

Three measuring probes were used: an unsteady static pressure transducer (flush mounted), an unsteady total pressure probe, and a miniature hot-wire. Over-tip static pressure maps are familiar measurements, however, hot-wire and total pressure measurements in the tip clearance gap have not previously been reported in this detail.

[1] Based on rotor true chord.

[2] *Tip Clearance Flows in Axial Compressors: Stall Inception and Stability Enhancement*, PhD Thesis, University of Cambridge, UK, 2011

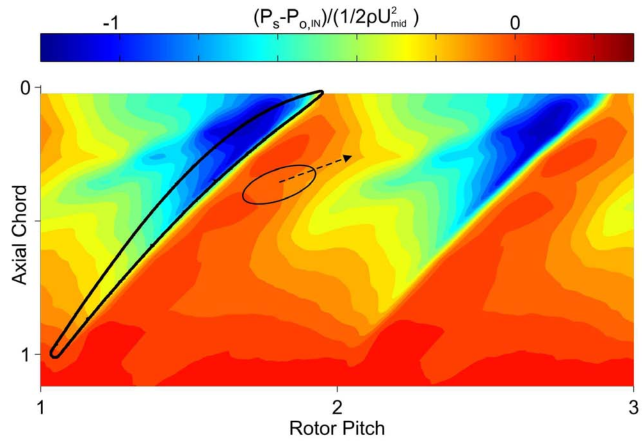

FIGURE 2. **Casing static pressure contours at design conditions.**

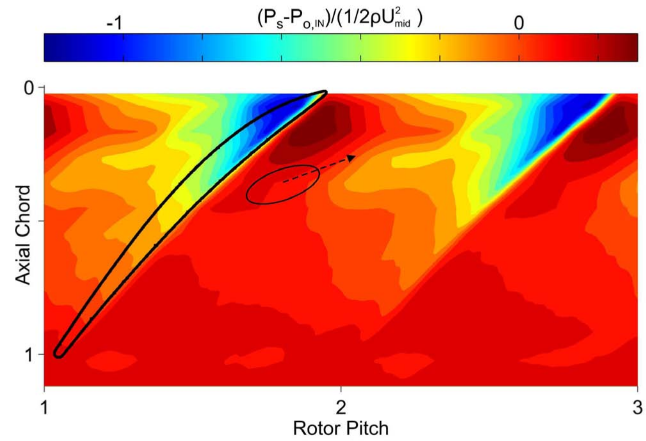

FIGURE 4. **Casing static pressure contours near stall.**

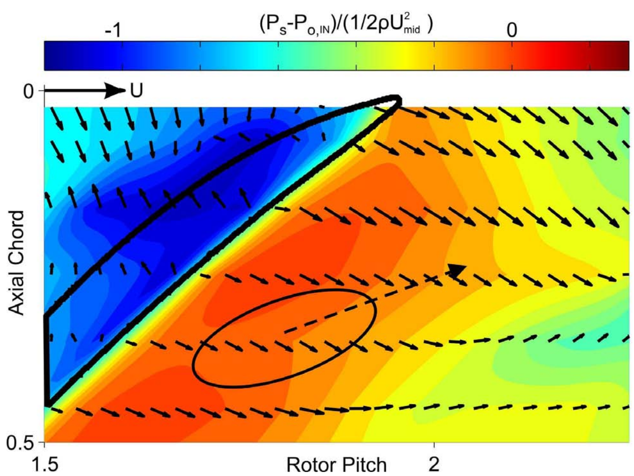

FIGURE 3. **Casing static pressure contours at design with absolute velocity vectors at 50% tip clearance height.**

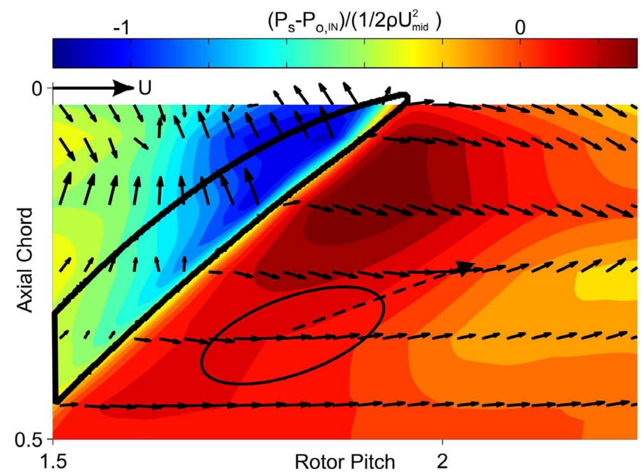

FIGURE 5. **Casing static pressure contours near stall with absolute velocity vectors at 50% tip clearance height.**

The three probes were not installed simultaneously, but one at a time. Measurements were obtained by moving the probes to each of 13 positions distributed from the leading to the trailing edge of the blade. The unused probe access holes were plugged to maintain a smooth casing. At each measuring location, the hot-wire was rotated through 10° steps through 200° in order to determine the local velocity. In addition, the hot-wire was moved through three depths in the clearance gap. The total pressure probe was rotated in 10° steps through 360°. The unsteady total pressure measurements were used to remove the ambiguity in the absolute direction of the hot-wire results and to produce contour maps of the total pressure (not shown here). The flow maps presented here are made up of ensemble averaged measurements.

A selection from the detailed over-tip measurements is presented in Figs. 2 to 5. Absolute velocity vectors are shown along with a tip speed vector (U) for reference. Figure 2 shows the casing static pressure map at compressor design conditions. The region near the tip is enlarged in Fig. 3 to show the absolute velocity vectors at 50% tip clearance height. (The oval marked in the figure is the proposed position of the extraction hole which will be discussed later.) It can be seen that the maximum pressure rise across the blade (pressure side to suction side) occurs at about 25% axial chord.

In contrast, the static pressure field at the near stall condition is shown in Fig. 4. The region near the blade tip is similarly enlarged to show the velocity vector field in Fig. 5. In Fig. 4, the maximum pressure rise across the blade has increased in magnitude and has moved toward the leading edge, to about 10% axial chord. As a result, the velocity vectors in Fig. 5 have changed size and direction compared to Fig. 3.

An appropriately placed air extraction hole would take advantage of the above changes in pressure and flow direction to produce a self-regulating extraction system. In the next section, the static pressure fields in Figs. 2 and 4 will be used to estimate the best position for the extraction hole.

Simple Extraction/Re-injection Model

Flow through the re-circulation loop will primarily be driven by the pitchwise averaged static pressure difference between the extraction hole and the injection port, $\Delta P = \overline{P}_{EX} - \overline{P}_{INJ}$, i.e., the pressure difference between positions 1 and 3 in Fig. 1. We are seeking the best extraction hole location to maximize flow re-circulation at near stall conditions (maximize ΔP_{NS}) and to minimize flow re-circulation at design conditions (minimize ΔP_D). **If there is an optimum location for an extraction hole, based on more re-circulation near stall and less at design, it will show up as a maximum of the function $\Delta P_{NS} - \Delta P_D$.**

To evaluate ΔP, the pitchwise averaged static pressure at each axial position (from the leading edge to the trailing edge in Figs. 2 and 4) is used for $\overline{P}_{EX}$. The pressure for the injection location, $\overline{P}_{INJ}$, is taken from the static pressure at 10% axial chord upstream of the rotor leading edges. The pressure difference ΔP is evaluated at near stall and design conditions to produce the function $\Delta P_{NS} - \Delta P_D$. This is shown in non-dimensional form in Fig. 6. The maximum of the function suggests that the best axial location for the extraction hole will occur at 35% axial chord.

It is clear that having the extraction hole at a part-chord position will produce less driving pressure for the re-circulating flow than if the hole were further downstream, at the exit of the blade row for example. This disadvantage is, however, outweighed by having a self-regulating system that minimizes re-circulated flow at design conditions. Figure 7 shows the difference between a re-circulation loop using an extraction hole at 100% chord and one using a hole at 35% chord. At 100% chord, the pressure difference driving the re-circulation is more or less constant from design to stall. At 35% chord there is a clear rise in the driving pressure as the flow coefficient is reduced toward stall. This rise is a natural consequence of the forward shift in blade loading which occurs as the flow rate is reduced.

Having used a simple model to find the best position for the extraction hole, it now remains to check that this position is in fact the best and to find the optimal shape and orientation of the hole.

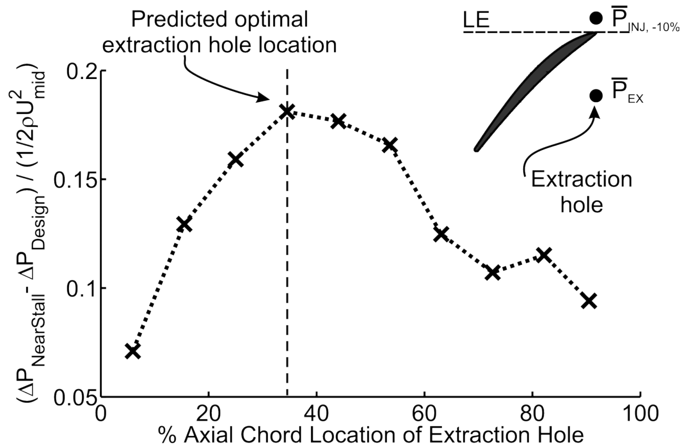

FIGURE 6. "Sweet spot" for extraction hole location predicted from re-circulation loop driving pressure ($\Delta P = \overline{P}_{EX} - \overline{P}_{INJ}$) at near stall and design conditions. Driving pressure is based on over-tip static pressure measurements (Figs. 2 and 4).

EXTRACTION HOLE OPTIMIZATION

The extraction hole optimization was carried out on the large Deverson compressor using a single extraction/re-injection loop, as shown schematically in Fig. 8. It is not sufficient to rely on a model based on the pitchwise averaged static pressure measurements to choose the location of the extraction hole. The changing velocity field also plays a part and so the only real measure of merit is the flow rate through the re-circulation loop.

A composite block was fitted to the compressor casing over the rotor tips and extraction holes were cut through the block at various axial locations and angles as shown in Fig. 9. There are many holes in this block but only one is open for testing at a time; the others are blocked with the green plasticine seen in the photo.

The flow is extracted through the block at the compressor casing surface, directed around the outside of the compressor via a piece of plastic tubing, and re-injected upstream of the rotor row through a round hole 70% axial chord upstream of the blade leading edge. The location of the injection hole was influenced by constraints of the rig, however, its position is not important at this stage provided it gives a representative back pressure against which to optimize the extraction hole.

The flow rate passing through the loop is measured using a miniature Venturi meter as shown in Fig. 8. A hot-wire probe is also shown in Fig. 8. This probe was used to measure the pulsation amplitude in the re-circulation loop caused by the blade

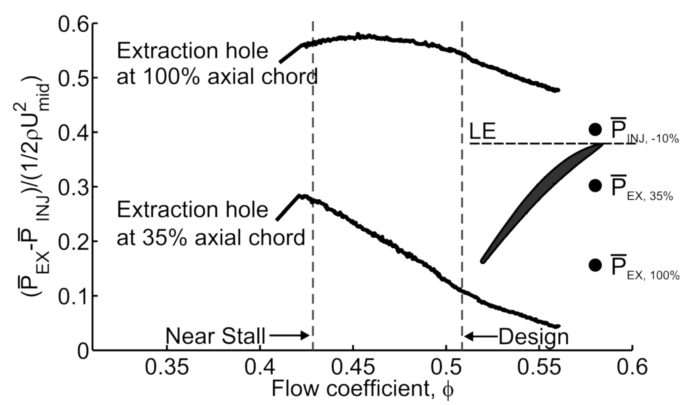

FIGURE 7. Measured characteristics of driving pressure for injection hole at 10% axial chord upstream of leading edge and for extraction holes at 100% chord and 35% chord.

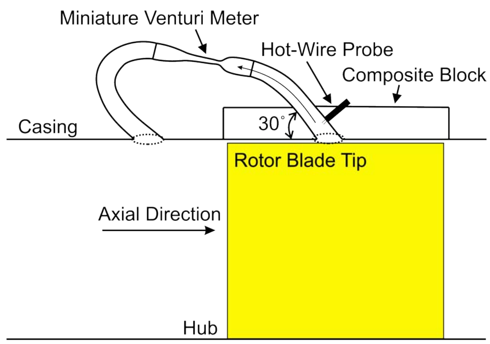

FIGURE 8. **Instrumented extraction/re-injection loop.**

passing over the extraction hole. The mass flow through the loop oscillates about ± 20% of the averaged value. Although this oscillation is significant, it does not in itself turn the injection on and off, and is a lower magnitude than that needed to be considered a pulsed jet. The results presented in the following sections quote the average mass flow through the re-circulation loop.

The extraction hole was produced by drilling through the composite block with no rounding of the break-through edges. The resulting surface profile of the hole is an ellipse. The angular orientation of the extraction hole is defined in two planes: the surface offtake angle and the tangential offtake angle. The surface offtake angle is the angle of the extraction hole relative to a plane tangent to the surface of the casing wall; a surface offtake angle of 30° is shown in Fig. 8. (This angle is used in all test cases presented here as smaller angles were found to create an edge of material too thin for reliable production.) The tangential offtake angle, on the other hand, is defined relative to a line tangential to the casing surface in the direction of blade rotation as shown in Fig. 10.

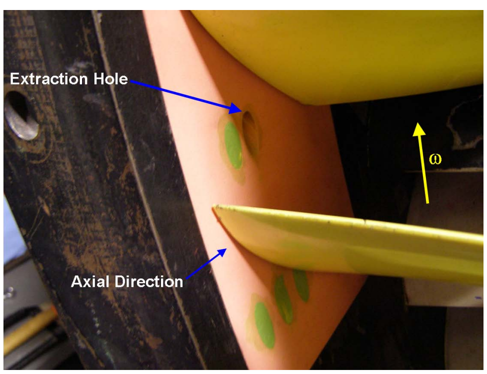

FIGURE 9. **Compressor casing with composite block and extraction hole locations viewed from upstream of the rotor.**

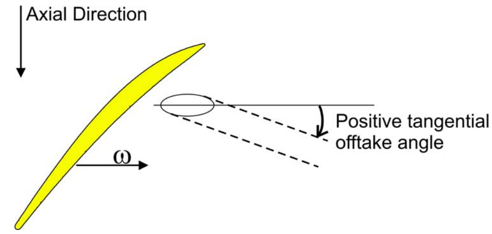

FIGURE 10. **Tangential offtake angle imposed on the casing surface viewed from the hub.**

Axial Location of Extraction Hole

To begin with, the axial location of the extraction hole was optimized using a 0° tangential angle as defined in Fig. 10. The location of the hole was varied from 10% to 90% axial chord. The results are shown in Fig. 11, reporting the percentage of the total incoming flow which is extracted and re-circulated through the loop. At all operating conditions, the figure shows a general increase in re-circulated mass flow as the hole location is moved rearward. This is expected, as the circulation is driven by the average pressure difference between the extraction and injection holes. This pressure difference increases with axial location because of the general pressure rise through the blade row.

The trend of the re-circulated mass flow versus the axial position of the extraction hole is different at compressor design and near stall conditions (Fig. 11). The blue line shows that at design conditions, the re-circulated mass flow increases continuously in a parabolic manner as the extraction hole moves downstream. At near stall operating conditions, the green line shows the re-circulated mass flow increases rapidly at first and then levels off.

The most important aspect of Fig. 11 is the vertical distance between the blue and green lines, as this represents the change in the re-circulated mass flow as the compressor is throttled from design to near stall conditions. To achieve an effective self-regulating system, the objective is to find the maximum ver-

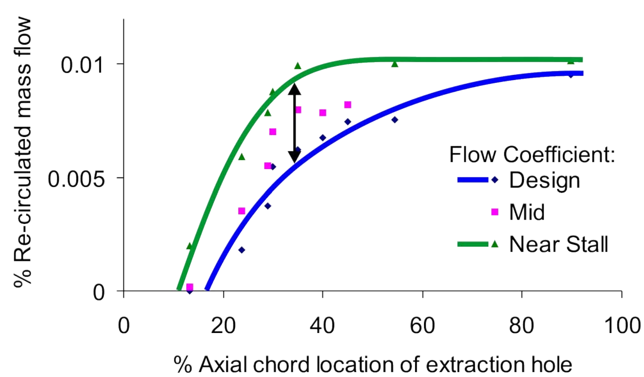

FIGURE 11. **Extraction hole optimization: axial chord location.**

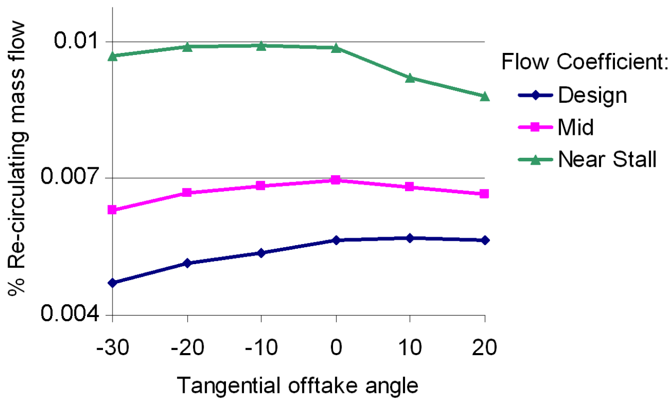

FIGURE 12. **Extraction hole orientation.**

tical distance between the two lines. Figure 11 shows the greatest vertical distance is in the region between 30% and 40% axial chord. For an extraction hole located in this region, the re-circulated mass flow near design conditions is about half of that re-circulated near stall. This location thus achieves the desired self-regulating aspect of the new casing treatment design, and agrees with the optimum location found by the simple model in Fig. 6.

All further angle and shape optimization is completed with the extraction hole set in the "sweet spot" region centered at 35% axial chord.

Tangential Offtake Angle

The tangential offtake angle, as defined in Fig. 10, was varied from $-30°$ to $+20°$; the results are shown in Fig. 12. The lines are nearly parallel, indicating low sensitivity of the re-circulated mass flow to tangential angle. The re-circulated mass flow near stall (green line) is maximized when the tangential offtake angle is in the range $-20°$ to $0°$. For this range, the re-circulated mass flow near design conditions (blue line) is lowest at a tangential offtake angle of $-20°$. An offtake angle of $-20°$ is thus chosen as the optimal angle for the self-regulating extraction hole. At an axial location of 35% chord, and an offtake angle of $-20°$, the extraction/re-injection loop re-circulates about twice the mass flow near stall as it does near design.

The optimum axial location and offtake angle determined here are represented by the oval and trajectory sketched in Figs. 2 to 5. The optimization experiments have produced an extraction hole which is misaligned with the flow vectors at design conditions (where minimum flow extraction is required), and aligned with the prevailing flow direction near stall (where maximum extraction is required). This orientation of the hole thus aids flow extraction near stall because the favorable flow direction reduces entry losses.

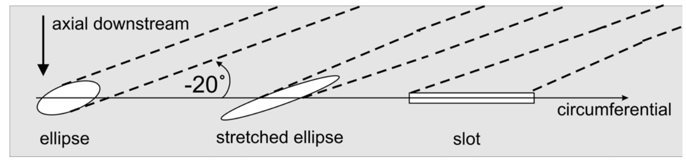

FIGURE 13. **Extraction hole geometries viewed on casing surface.**

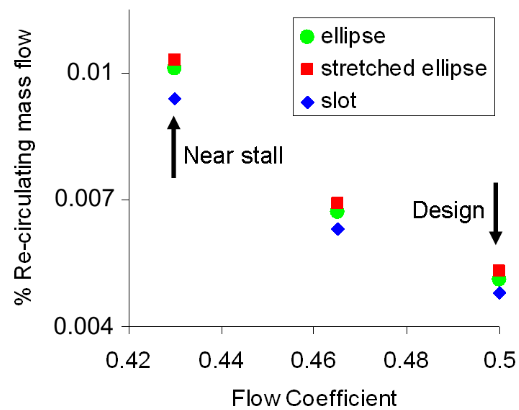

FIGURE 14. **Extraction hole geometry optimization.**

Extraction Hole Shape

With the optimal axial chord location and tangential offtake angle defined, the geometrical shape of the extraction hole can be investigated. The extraction hole shapes chosen for investigation are shown in Fig. 13, their cut profiles being viewed as on the casing surface. The casing surface area of each hole is the same. The first shape is a simple hole drilled through the surface at an angle, hence the elliptical profile on the surface. The second shape, the stretched ellipse, shows a wider opening in the direction of the offtake trajectory; this shape was meant to further minimize the entrainment of flow near design conditions. The third shape, a slot, widens the opening of the extraction hole along a tangential line; this was done in the hope of extracting as much axially reversed flow as possible at conditions near stall. The latter two were produced using a 3D rapid prototyping printer.

The results for the extraction hole shape optimization are shown in Fig. 14. It can be seen that changes to the extraction hole shape have little effect on the re-circulated mass flow. Further, the effect of the individual shapes was to either increase or decrease the re-circulated mass flow at all flow coefficients, not just near stall as is desired. In addition to the geometries shown in Fig. 13, the sensitivity of the extraction hole to profiling of the lip edges was also investigated. In none of the cases tested did the profiling produce any measurable advantage.

As none of the additional hole shapes enhanced the self-regulating feature of the extraction system, the drilled hole (elliptical surface profile) was maintained for ease of manufacture.

Extraction Hole Optimization Summary

It has been shown that it is possible to find an optimum flow extraction configuration which is self-regulating in its ability to re-circulate a minimum amount of flow at compressor design conditions and a maximum amount of flow near stall. An optimized loop will re-circulate about 0.01% of the annulus flow near stall, and half that near compressor design conditions. This self-regulating feature is essential for an effective form of casing treatment which aims to maintain compressor efficiency at design conditions and extend operating range near stall.

INJECTION NOZZLE DESIGN OPTIMIZATION

The injection nozzle is the second component of the new extraction/ re-injection design that needs to be optimized. The objective of this section is to develop an injection nozzle that will provide the greatest possible stall margin improvement from the air provided by the extraction hole discussed in the previous section. In addition, the nozzle design should minimize impact on the efficiency of the compressor. The optimization of the nozzle will include the manner of air injection (into the main stream or along the casing), the direction of injection (axial or swirling) and the position of the nozzle relative to the rotor leading edge.

A single injection nozzle will not have a measurable effect on compressor performance and therefore multiple injection nozzles (12 or 24) will be tested in full annular form. For practical reasons, the full annular testing was performed in the smaller Mini-Deverson compressor.

The Mini-Deverson compressor is a vertical axis machine built up of multiple aluminum rings. These rings are stacked one on top of the other and can be recognized by the horizontal lines in Fig. 15. In order to test the new casing treatment, the aluminum casing ring that sits over the rotor tips has been replaced by a ring of rapid prototyped plastic segments. These segments can easily be replaced and rearranged to create numerous experimental variations.

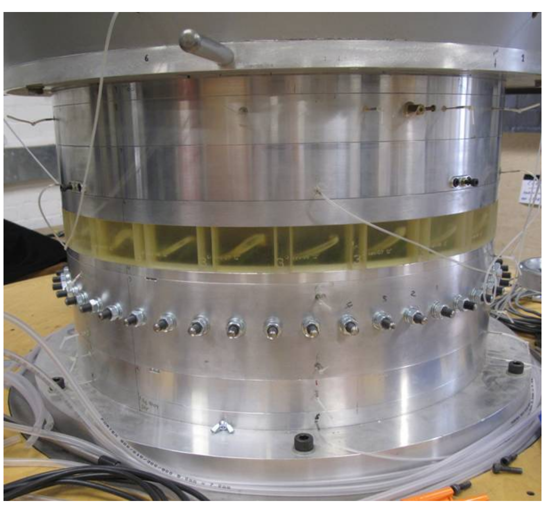

FIGURE 15. **Photo of Mini-Deverson compressor with casing treatment (transparent ring) installed.**

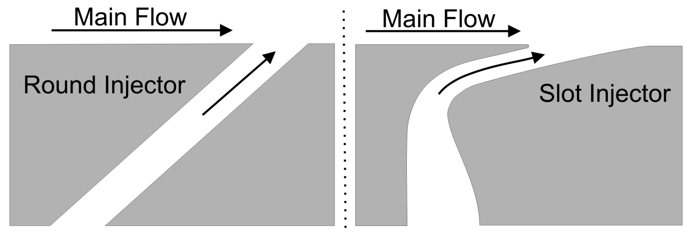

FIGURE 16. **Injection Nozzle Designs.**

Injection Nozzle Geometries

Two injection nozzles shapes were considered, as shown in Fig. 16. Both injection nozzles have the same exit area.

The round injector design is simply a hole drilled at an angle of 30° to the casing surface. The injected flow is meant to interact with the main flow like a vortex generator jet producing a general increase in axial momentum in the tip region. The slot injector design, on the other hand, is intended to inject the flow along the casing surface to energize the casing boundary layer in the vicinity of the rotor tips.

Injection Nozzle Optimization with External Air Supply

The optimization of the injection nozzle is carried out in isolation from the extraction/re-injection loop by supplying the injector with flow from an external air supply. This approach makes it possible to regulate the amount of injected air and simplifies the testing arrangement.

Special injection nozzle segments were produced for the compressor; see Fig. 17. The leftmost piece shows a segment as viewed from outside the compressor. Note that this segment is made up of two pieces: a removable plug containing the injection nozzle and a larger piece making up the body of the segment. The

FIGURE 17. **Pieces for injection nozzle testing driven by an external air supply.**

second segment from the left shows the same piece but viewed from inside the compressor. A slot design injection nozzle can be seen. To the right are two views of the removable plug. The direction of injection can be changed by rotating the plug using the protruding wings. A protractor printed into the outside of the piece (seen far left) is used to set the angle of injection. The far left piece in Fig. 17 shows a round hole in the back of the plug. Tubing is inserted into this hole and connected to a central manifold to which external air is supplied. Twelve such injector segments are evenly distributed around the annulus. The injection nozzle exit is located 50% axial chord upstream of the rotor leading edge. When using the external air supply, the most convenient way of expressing flow rate is to use a velocity ratio such as V_i/V_x, where V_i is the injected flow velocity, and V_x is the average axial velocity through the stage.

Injection Nozzle Design (Penetrating vs. Wall Jet)

The round and slot injector designs shown in Fig. 16 were compared in order to determine whether it is more beneficial for the injected flow to mix with the main stream or to follow the casing wall with a minimum of mixing.

The test results were clearly in favor of the slotted design in which air is injected close to the casing. This conclusion is in line with others reported in the literature, e.g., Strazisar et al. [12] and Weigl et al. [8].

Angle of Injection. Injection angle studies are reported in the literature but consensus is not achieved. Suder et al. [11] and Strazisar et al. [12] showed axial injection is best (with axial inflow). However, other studies with axial inflow show different results. D'Andrea et al. [15] indicated that an injection angle of 30° in the direction of rotation was optimal in a low speed compressor. Another injection study in a low speed compressor by Deppe et al. [16] indicated optimum injection was achieved at 75° against the direction of rotation. Finally, Weigl et al. [8] reported that 15° against the direction of rotation was best (also with axial inflow) in a transonic rig. Because of the apparent "compressor specific" nature of the best injection angle, optimization is a necessary step in the current work.

The angle of injection was varied from $-90°$ to $+90°$ using the rotating plugs shown in Fig. 17. Axial injection is defined as 0°, and injection in the direction of blade rotation is defined as positive. The slot type injectors are used for this work.

Figure 18 shows the results for $V_i/V_x = 2$. The black characteristic line indicates the smooth wall configuration. Each colored characteristic line indicates that all 12 injectors have been rotated to direct the flow to the angle specified in the legend. The results in Fig. 18 indicate that axially injected flow improves the stall margin the most. Results for other injected velocity ratios lead to the same conclusion.

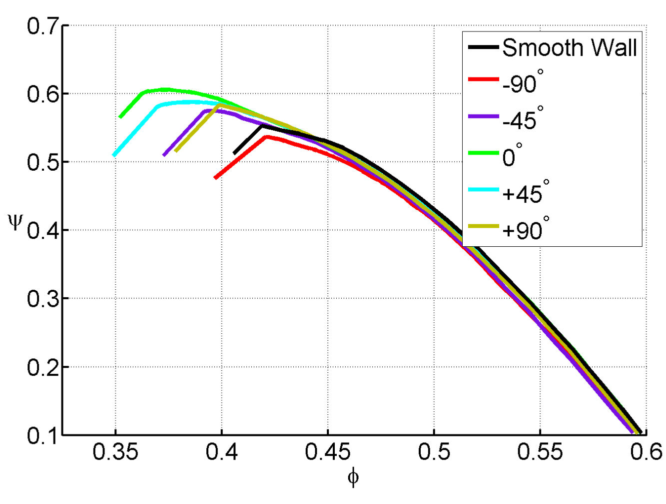

FIGURE 18. **Pressure rise characteristic for various injection angles. Twelve slot injectors with $V_i/V_x = 2$.**

Injection Nozzle Integrated with Re-circulation Loop

The optimization process continues with the slot design injection nozzle as an integrated part of the complete extraction/re-injection system. The axial position of the nozzle is tested at two locations: 10% and 50% axial chord upstream of the rotor leading edge.

A complete extraction/re-injection loop segment is shown in Fig. 19. The extraction hole is at the lower right hand corner of the segment. The hole extends into a loop which leads to the slot injection nozzle, seen on the left side of the segment. The injection nozzle is deliberately offset circumferentially from the extraction hole. This is done to minimize the possibility of overheating due to repeated re-circulation of the same flow.

All tests were conducted in full annular form with 12 or 24 loop segments evenly distributed around the annulus. The optimized extraction hole design from the previous section is used and the air is injected axially downstream.

Axial Location of Injection Nozzle. Figure 20 shows that with the injection nozzles at 10% and 50% axial chord up-

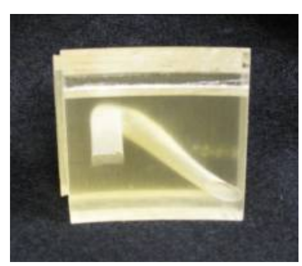

FIGURE 19. **Inner detail of casing treatment loop segment.**

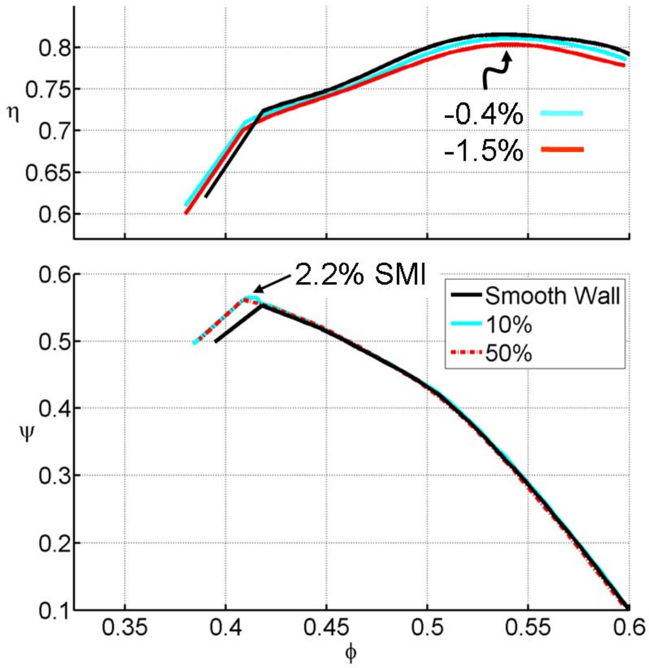

FIGURE 20. **Efficiency and pressure rise characteristics for 12 extraction/re-injection casing treatment loops. The injectors are placed at 10% and 50% axial chord length upstream of the rotor leading edges.**

stream of the rotor face, the same improvement in stall margin (2.2%) is achieved. The efficiency characteristics, on the other hand, are very different. The loops injecting at 50% axial chord upstream of the rotor leading edges incur an efficiency penalty of 1.5%. The loops injecting at 10% axial chord upstream of the rotor leading edge fair better with a drop in maximum efficiency of only 0.4%.

From the results optimizing the shape of the injection nozzle and work from Strazisar et al. [12] and Weigl et al. [8], we know that injection close to the casing wall is most effective. The axial location influences the amount of spreading that occurs in the injected jet before it reaches the rotor blade row. The results here show that although the spreading has not affected the stall margin improvement, it has affected the efficiency of the compressor. It is possible that the reason for the lower efficiency in the case of the injectors located further upstream is due to the injected flow having more time to mix out.

When the number of loops is doubled from 12 to 24, the stall margin improvement increases to 3%. This is a diappointingly low increase for double the number of extraction/re-injection loops. The reasons for this are investigated in the next section.

THE INFLUENCE OF STALLING MECHANISM

The aim of this section is to investigate the influence of the compressor stall inception mechanism on the effectiveness of the extraction/re-injection casing treatment. A study published by Houghton and Day [17] found that the maximum benefit of casing treatment is only achieved if the compressor is inclined to stall by means of a spike stalling mechanism (where inception occurs near the casing). They concluded that casing treatment becomes less effective the more modal the stall inception mechanism becomes (where spanwise extent is greater).

The Mini-Deverson compressor with a smooth wall stalls by means of a spike mechanism. However, fast response pressure measurements indicate that the stalling mechanism is modal with casing treatment installed. The study by Houghton and Day [17], using a different compressor, suggests that if the rotor stagger is changed, the stall inception mechanism might also change. If the stall inception mechanism changes, an improvement in the stall margin might be expected.

With the rotor stagger in the Mini-Deverson compressor increased by 2°, the stalling mechanism of the compressor, with the casing treatment installed, did in fact change from modes to spikes. The blue line in Fig. 21 shows that the 12 loops now provide 2.6% SMI and no change in stage efficiency. The red line shows the SMI has more than doubled to 6.1% with 24 loops. The drop in maximum efficiency in this case is 0.8%. These results are summarized in the first two rows of Tab. 2 to compare the performance of the casing treatment before the

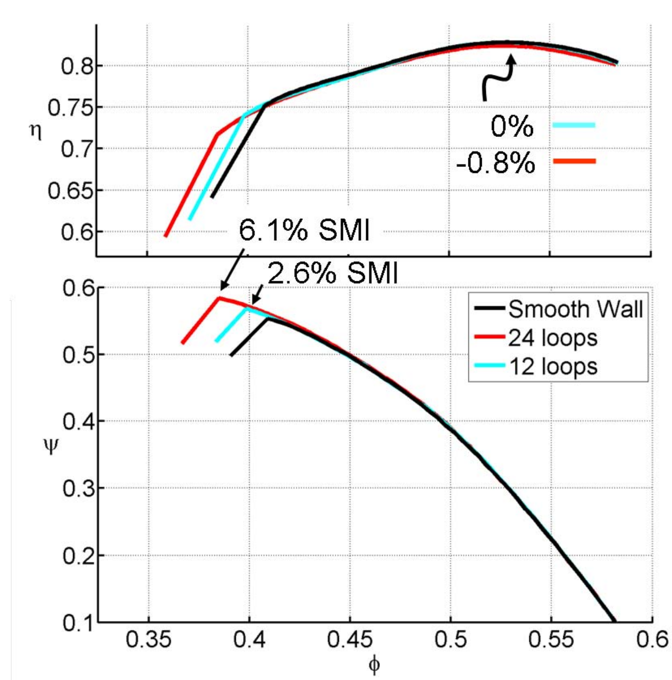

FIGURE 21. **Efficiency and pressure rise characteristics for extraction/re-injection loops with rotor blades stagger increased 2°.**

TABLE 2. CASING TREATMENT PERFORMANCE SUMMARY: SMI AND CHANGE IN MAXIMUM EFFICIENCY.

Stalling Mechanism	Modal		Spike	
(w/treatment installed)	SMI	$\Delta\eta$	SMI	$\Delta\eta$
Extraction/Re-injection				
12 loops	2.2%	-0.4%	2.6%	0%
24 loops	3%	-0.6%	6.1%	-0.8%
Circumferential Groove	0%	0%	2.3%	+0.2%

re-stagger (modal stalling mechanism) to the performance after the re-stagger (spike stalling mechanism). The circumferential groove results listed in the table will be discussed in the next section.

The table shows that for each casing treatment, a greater stall margin improvement is possible when the stalling mechanism is with spikes. Modal stall inception curtails the effectiveness of casing treatment; as suggested by Houghton and Day [17].

SINGLE CIRCUMFERENTIAL GROOVE

The new extraction/re-injection design was compared to a circumferential groove tested in the same compressor. The design of the single groove was based on the parametric study reported by Houghton and Day [4]. The leading edge of the groove was at 50% chord, with a width of 14% axial chord and a depth of 28% axial chord.

The groove design was tested in the Mini-Deverson compressor both before the re-stagger (modal stall inception) and after the re-stagger (spike stall inception). The groove provided no SMI or efficiency penalty before the re-stagger and 2.3% SMI and a slightly favorable change to efficiency when spikes were present after the re-staggering.

These results are included in Tab. 2. When comparing the performance of the groove with the extraction/re-injection casing treatment with 12 loops, it is seen that for spike type stall inception the performance is nearly identical. Considering the complexity of the extraction/re-injection system, the circumferential groove may be more advantageous when only the spike stalling mechanism is expected. However, when stall occurs via modes, the extraction/re-injection system provides some SMI (2.2%) wheres the groove provides none. In this case, the added complexity of the extraction/re-injection system may be acceptable. Changing the number of loops in the extraction/re-injection system may also be used to adjust the SMI if more is needed.

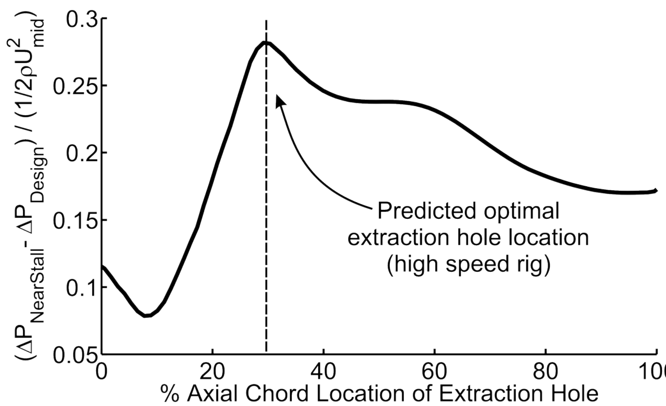

FIGURE 22. **"Sweet spot" for extraction hole location in a high speed compressor predicted from re-circulation loop driving pressure ($\Delta P = \overline{P}_{EX} - \overline{P}_{INJ}$) at near stall and design conditions, as explained for Fig. 6. Driving pressure is based on CFD casing pressure calculations.**

EXTRACTION/RE-INJECTION CASING TREATMENT PERFORMANCE AND RECOMMENDATIONS

The extraction/re-injection design fulfilled the initial design objective to provide a modest stall margin improvement (2 to 4%) with little ($< 0.5\%$) or no loss of efficiency at design conditions. As seen in Tab. 2, with 12 loops, the initial aim has been met. When 24 loops are installed, a higher SMI is achieved but with an unacceptable efficiency penalty. With further optimization, we expect the efficiency loss can be reduced.

The extraction/re-injection casing treatment design presented here was shown to provide at least a 2% SMI with 12 loops (re-circulating less than 0.13% of the inlet mass flow near stall and half that at design) and up to 6% SMI with 24 loops (re-circulating 0.25% of the inlet mass flow near stall and half that at design). This amount of re-circulated mass flow compares favorably to previous work. Strazisar [12] report 4-6% SMI with 0.7 to 1.3% of annulus flow re-circulated, and Suder [11] report 6% SMI with 2% of annulus flow injected.

The results in Tab. 2 indicate that the extraction/re-injection system proposed in this paper may be an improvement on existing re-circulating casing treatments. However, a true comparison cannot be made with other re-circulating systems reported in the literature because efficiency measurements are rarely reported. None are included in the comparable studies [7–9, 11, 12, 15, 16].

As a final step, CFD simulations[3] of the flow field in the tip gap region of a high speed machine were examined. The high speed compressor showed similar changes in the pressure and velocity fields accompanying changes in compressor flow rate as were discussed in the introduction. A similar analysis of the casing pressure field was conducted with the same model as was used to find the optimum extraction hole location in the Dever-

[3]Unpublished data made available through the European project NEWAC.

son compressor (Fig. 6). The model applied to the high speed machine (Fig. 22) produced a well defined peak in the pressure difference function at 30% chord. As the presence of a "sweet spot" is the main driver of the self-regulating action, it is concluded that the new casing treatment design might also be effective in a high speed machine.

In general, all axial flow compressors demonstrate a forward shift of blade loading as the flow rate is reduced toward stall. This forward shift is sufficient to suggest that all axial compressors will exhibit some self adaptability in terms of pressure driven re-circulation. It is thus possible that the new casing treatment may be usefully applied in other compressors.

CONCLUSIONS

1) Proof of concept experiments have been carried out on a new form of over-tip re-circulating casing treatment for axial compressors. The system met design objectives and has the ability to self-regulate the amount of flow being re-circulated. The configuration is such that a minimum amount of air is re-circulated at compressor design conditions (thus minimizing any loss of efficiency) and a maximum amount of air is re-circulated near the stability limit (thus maximizing stall margin).

2) In a departure from previous re-circulating systems in axial compressors, the new casing treatment makes use of bleed extracted over the rotor tips, rather than downstream of the row or stage. The self-regulating capability of the system makes use of changes in the over-tip pressure and velocity fields which accompany changes in compressor flow rate. New high resolution pressure and velocity measurements in the tip clearance gap are used to explain why the new treatment possesses self-regulating capabilities.

3) Experimental measurements have demonstrated that a reasonable level of stall margin improvement, and a reasonable level of self-regulation, in terms of the amount of air re-circulated, can be achieved by locating the extraction hole at about 35% chord downstream of the rotor leading edges. The best chordwise location of the air extraction hole will differ from compressor to compressor.

4) It has been shown that the angular orientation of the extraction hole (relative to the tangential direction of blade travel) is important in achieving the self-regulating feature of the new design. This is explained in terms of the changes in the tip clearance velocity field that accompany changes in compressor flow rate.

5) It was shown that the new extraction/re-injection design is preferable to a single circumferential groove when both spike and modal stall inception can be expected, or when more stall margin improvement is needed than a groove can provide.

6) The type of stall inception mechanism (spikes or modes) in a given compressor has recently been shown [17] to be important in determining the effectiveness of casing treatment. The current work, using a different compressor, confirms this observation. The current work shows that over-tip casing treatment is less effective if the compressor has a disposition to stall in modal form.

7) Stall margin improvements between 2.2 and 6.0% were achieved depending on the number of re-circulating loops and on the compressor stalling mechanism. Efficiency penalties at the compressor design condition varied between 0 and 0.8% for less than 0.25% of the inlet mass flow re-circulated near stall and half that at design. In general, modest stall margin improvements (2%) were possible without any loss of compressor efficiency at design conditions.

8) A simple model using casing static pressure measurements has been found to be useful in predicting the best location for the air extraction hole required for the new self-regulating system. The model maximizes the difference in re-circulation driving pressure between design and near stall operating conditions. Application of the simple model to CFD pressure maps for an aero-engine compressor suggests that a similar self-regulating casing treatment design might also be effective in high speed machines.

NOMENCLATURE

$\dot{m}$	Mass flow rate	η	Stage efficiency
T	Rotor torque	ρ	Inlet air density
U_{mid}	Mid-height blade speed	ϕ	Flow coefficient
V_i	Injected velocity	Ψ	Pressure rise coefficient
$V_{x,in}$	Mass-ave axial velocity	ω	Rotational speed
P	Pressure; $P_s = static$, $P_o = total$		

ACKNOWLEDGMENT

The authors appreciate the financial support of the European Union Framework 6 Project NEWAC: NEW Aero engine Core concepts (FP6-030876). Additional support from Rolls-Royce is also acknowledged.

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