Fluid Mechanics of Ported Shroud Centrifugal Compressor for Vehicular Turbocharger Applications

by

George Alexander Christou

S.M. Aeronautics and Astronautics, Massachusetts Institute of Technology, Cambridge (2010)

Diploma, Mechanical Engineering, National Technical University of Athens, Greece (2006)

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Signature redacted

Author

Department of Aeronautics and Astronautics

August 20, 2015

Certified by

Choon S. Tan

Senior Research Engineer

Thesis Supervisor

Certified by

Edward M. Greitzer

H. N. Slater Professor of Aeronautics and Astronautics

Committee Member

Certified by

Nicholas A. Cumpsty

Emeritus Professor of Mechanical Engineering, Imperial College London

Committee Member

Certified by

Borislav T. Sirakov

Manager, Aerodynamic Science Wheels and Aerodynamics

Committee Member

Accepted by

Paulo Lozano

Associate Professor of Aeronautics and Astronautics

Chairman, Department Committee on Graduate Students
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Abstract

This thesis presents an investigation of the effects of Ported Shroud (PS) self-recirculating casing treatment used in turbocharger centrifugal compressors for increasing the operable range. Computed results, assessed with experimental measurements on ported and non-ported variants of a representative turbocharger compressor, are used to determine the impact of the PS on the flow field and hence performance. It is shown that the main flow path perceives the PS flow as a combination of flow actuations that include injection and removal of mass flow, and injection of axial momentum and tangential momentum. A computational model in which the presence of the PS is replaced by imposed boundary conditions that reflect the individual flow actuations has thus been formulated and implemented. The removal of a fraction of the inducer mass flow was determined to be the dominant flow actuation in setting the performance of PS compressors. Mass flow removal reduces the flow blockage associated with the impeller tip leakage flow and increases the diffusion in the main flow path. Adding swirl to the injected flow in the direction opposite of the wheel rotation results in an increase of the stagnation pressure ratio and a decrease of the efficiency. The loss generation in the flow path has been defined to rationalize efficiency changes associated with PS operation.

Thesis Committee:
Dr. Choon S. Tan (chair)
Prof. Nicholas A. Cumpsty
Prof. Edward M. Greitzer
Dr. Borislav T. Sirakov
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## Nomenclature

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<th>Description</th>
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<tr>
<td>$\Omega$</td>
<td>angular velocity</td>
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<tr>
<td>$N$</td>
<td>angular frequency</td>
</tr>
<tr>
<td>$A$</td>
<td>area</td>
</tr>
<tr>
<td>$\delta$</td>
<td>boundary layer</td>
</tr>
<tr>
<td>$\delta^*$</td>
<td>displacement thickness</td>
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<tr>
<td>$\rho$</td>
<td>density</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>mass flow rate</td>
</tr>
<tr>
<td>$P$</td>
<td>pressure</td>
</tr>
<tr>
<td>$r$</td>
<td>radius</td>
</tr>
<tr>
<td>$U$</td>
<td>rotational velocity, $U = \Omega r$</td>
</tr>
<tr>
<td>$\tau_{ij}$</td>
<td>shear stress tensor</td>
</tr>
<tr>
<td>$s$</td>
<td>specific entropy</td>
</tr>
<tr>
<td>$h$</td>
<td>specific enthalpy</td>
</tr>
<tr>
<td>$T$</td>
<td>static temperature</td>
</tr>
<tr>
<td>$u$</td>
<td>velocity in stationary frame</td>
</tr>
<tr>
<td>$w$</td>
<td>velocity in rotating frame</td>
</tr>
<tr>
<td>$\omega$</td>
<td>vorticity</td>
</tr>
</tbody>
</table>

### Subscripts and Superscripts

- **axial**, $\theta$, $m$, $r$, str, span: axial, circumferential, main flow, radial, spanwise, streamwise direction
- $b$: blocked
- $in$: compressor inlet
- $out$: compressor outlet
- $eff$: effective
- $M$: mass flow average
- $PS$: PS
- $PS\ Slot$: PS Slot location
- $PS\ Outlet$: PS Outlet location
- $ref$: reference value
- $t$: stagnation quantity
- wheel inlet: wheel inlet
Non-Dimensional Quantities

\( \eta \)  
adiabatic efficiency

\( \beta \)  
flow angle in relative frame

\( \phi \)  
mass flow fraction, defined in text according to purpose

\( f_{\text{red}} \)  
reduced frequency

\( \pi_c \)  
stagnation pressure ratio

\( C_p \)  
static pressure coefficient, \( C_p = \frac{\Delta p}{\frac{1}{2} \rho_{\text{inlet}} U_{\text{tip}}^2} \)

\( \alpha \)  
swirl angle, \( \tan \alpha = \frac{u_{\theta}}{u_{str}} \)
Chapter 1  Introduction

In this chapter the concept of a Ported Shroud (PS) centrifugal compressor is introduced, followed by a summary of proposed flow mechanisms associated with their operation as reported in literature. Based on the limitations in the current understanding of PS compressor operation, a list of research objectives, hypotheses and research questions of this thesis are delineated. The contributions of this thesis in the research field of PS centrifugal compressors are then presented, followed by an outline of the organization of this thesis.

1.1 Ported Shroud Centrifugal Compressor

The development of modern internal combustion engines (ICE) for vehicular applications is driven by the need for improved fuel efficiency and the compliance with stringent gas emission (CO₂, NOₓ) regulations [1, 2, 3]. One way to improve fuel economy and reduce CO₂ emissions is to downsize the ICE, reducing the engine displacement. A smaller ICE is less throttled at part loads than larger engines thus contributing to higher efficiency. The drawback is that smaller ICEs provide lower peak performance. To overcome this disadvantage downsized engines are combined with pressure charging (mechanical or turbocharging) [2]. Turbocharging or supercharging an ICE increases the amount of air ingested in the engine cylinders thus increasing the engine power density while also contributing to lower combustion chamber temperatures and reduction of NOₓ [4].

In compliance with the requirements for improved fuel efficiency and lower emissions, automotive turbochargers are required to provide both high efficiency and pressure ratio over a large operating range [2]. Unfortunately, an increase in pressure ratio tends to both increase the compressor surge mass flow and decrease the mass flow at which the compressor chokes, reducing the compressor map width [5]. As a result, various techniques have been employed to increase the compressor operable range and efficiency. For centrifugal compressors used in vehicular turbochargers the use of a Ported Shroud (PS) has proven its effectiveness [6].
A PS is a self-recirculating casing treatment that consists of two slots, one of which is placed downstream of the inducer throat (PS Slot) and the other upstream of the impeller leading edge (PS Outlet). The two slots connect the PS cavity, which is contained inside the inducer shroud, with the compressor main flow channel and allow a fraction of the compressor flow to circulate through the PS cavity (Figure 1-1). The placement of the end locations of the cavity are selected in such a way so as to provide a positive pressure differential between PS Outlet and Slot for operating points (OPs) near choke ($P_{PS\text{Outlet}} > P_{PS\text{Slot}}$) as shown in Figure 1-1A. This assures that additional air flows through the PS cavity and bypasses the inducer throat. For OPs near surge, a negative pressure differential is achieved ($P_{PS\text{Outlet}} < P_{PS\text{Slot}}$). The static pressure at the PS Slot is higher than that at the PS Outlet, allowing flow to recirculate from the inducer to upstream of the impeller inlet through the PS cavity (Figure 1-1B).

Figure 1-1: PS operation near choke where a fraction of the inlet flow bypasses the inducer throat through the PS Cavity (A) and surge where a fraction of the inducer flow recirculates upstream of impeller (B) [6]
1.2 Motivation

The tools available for the analysis of the performance of PS compressors include both experiments and state of the art three-dimensional CFD. However, employing these methods on a large number of PS compressor configurations is inefficient and time consuming. The goal is to arrive at in-depth knowledge of which specific aspects of the PS operation are beneficial to the range extension so as to avoid conducting experiments and calculations for every PS compressor configuration.

1.3 Proposed Ported Shroud Flow Mechanisms Reported in Literature

To satisfy the requirements for increased pressure ratio over a wide compressor operable range, researchers have proposed a number of compressor map width enhancement devices. Passive map width control devices have been preferred in centrifugal compressor turbochargers for automotive applications due to low implementation and operation cost, durability and effectiveness [6]. A number of passive control devices providing compressor range extension have been noted in literature but the following review contains key points of pertinent research papers focusing on ported shroud casing treatment devices.

Fisher is one of the first researchers who investigated the effect of using a PS, referred to as an inducer recirculating bypass in his paper, on the compressor operable range [6]. The effect of the PS on compressor performance was experimentally investigated for a number of impellers. The use of a PS, on average, moved the compressor surge line to 70-80% of its original flow, increased the choke flow by 5-9% (Figure 1-2) and incurred a penalty in efficiency of approximately one point on most of the operating range compared to the baseline non-ported compressor [6]. Pressure measurements near the inducer tip showed that the use of a PS reduced the pressure non-uniformity compared to the baseline case. Thus, it was hypothesized that the PS decoupled the inducer from the circumferentially non-uniform flow in the rest of the stage [6].
Figure 1-2: Compressor range extension with the use of a Map Width Enhancement Device compared to a non-ported compressor as reported in [6].

Hunziker et al. numerically investigated the range extension obtained through the use of PS cavity on a single pitch centrifugal compressor with a vaned diffuser [5]. The authors noted that for OPs near surge, the flow recirculated through the PS cavity had a large circumferential velocity component in the direction of the wheel rotation as the inducer had already added work to the fluid. Hence, when the PS flow was again introduced to the main flow upstream of the impeller inlet it decreased the blade incidence angle, assisting in avoiding flow separation on the suction surface behind the main blade leading edge [5]. Even though the flow rate through the inducer increased compared to the baseline non-ported compressor of same outlet mass flow, the tip Mach number was reduced.

Decrease of the impeller incidence angle with the use of a PS has also been hypothesized by Qiu et al. [7] and Fisher [6] as a contributor to the increase of the compressor stall margin. According to their analysis, the flow recirculating through the PS cavity blocks part of the passage flow,
reducing the inlet flow area and leading to a decrease of the incidence angle of the main core flow [7].

Uchida et al. [8] and Iwakiri et al. [9] postulated that the region of reversed flow near the inducer shroud region was detrimental to setting the surge limits. Their numerical investigation showed that the introduction of a PS decreased the extent of this backflow region and it was hypothesized this responsible for decreasing the surge mass flow by approximately 25% compared to a conventional compressor [8]. Both papers investigated the effect of introducing Variable Inlet Guide Vanes (VIGVs) upstream of the impeller inlet in combination with the PS compressor [8, 9]. Ribs inside the PS cavity ensured that the circumferential velocity component of the flow through the PS cavity was removed, and setting the angle of the VIGVs near the compressor inlet controlled the flow angle of the inlet flow. The use of VIGVs in combination with the PS system resulted in the further decrease of the flow reversal near the inducer tip region. Increasing the VIGV setting angle resulted in a lower surge mass flow. Specifically, a VIGV angle of 80° decreased the surge mass flow by 59% compared to a conventional compressor [8]. On the other hand, the use of VIGVs with a non-ported compressor resulted only in a surge mass flow decrease of less than 15% [8]. It was therefore concluded that the effect of the PS in combination with the VIGVs was responsible for the surge mass flow reduction [8, 9]. However, even though the surge mass flow was reduced by increasing the angle of the VIGVs, this was accompanied by a substantial increase of the choke mass flow. Specifically, a combination of PS and VIGVs at 0° angle resulted in the decrease of the surge mass flow by approximately 25% and increase of operable range by 29%, while an angle setting of 70° resulted in the decrease of the surge mass flow by 50% and the decrease of overall operable range by 14% compared to the baseline non-ported case [8].

Yamaguchi et al. argued that although the strong swirl component of the PS flow contributes to the decrease of the incidence angle it was not clear whether this was effective for the improvement of the surge margin [10]. To test this hypothesis, they installed vanes inside the PS cavity to control the swirl angle of the recirculating flow. Three configurations of the casing treatment were tested. The baseline case did not contain any vanes allowing for the PS flow to conserve its swirl velocity component in the direction of the wheel rotation. A second configuration included radial vanes that removed the swirl component, while in the last
configuration inclined vanes were installed such that counter-swirl (opposite to the impeller rotation) would be added. Gas stand measurements showed, that introduction of counter-swirl to the PS flow reduced the surge mass flow by up to 20% and increased the pressure ratio compared to the baseline casing treatment case while incurring no additional efficiency penalty. Experimental data and CFD analysis showed that the introduction of counter-swirl to the PS flow increased the work coefficient for OPs with mass flow between peak efficiency and surge resulting in a higher static pressure at the PS slot location. This resulted in the increase of the mass flow recirculating through the PS cavity which was thought to have a larger impact on stabilizing the impeller performance than reduction of the flow incidence angle. The findings of Yamaguchi et al. [10] are also supported by the work of Sivagnanasundaram et al. [14], where vanes were inserted into the PS cavity. The vanes decreased the swirl angle of the exiting PS flow to 30° in the direction of the wheel rotation. By decreasing the circumferential velocity component in the direction of the wheel rotation, an increase of the compressor pressure ratio and surge margin were calculated, while recirculating more flow through the PS cavity.

Tamaki also investigated the effect of introducing negative swirl to the flow through the PS cavity for use in a transonic centrifugal compressor of high pressure ratio $\pi_c > 4.5$ [11]. In accordance with [10], he showed that the use of counter-swirl vanes in the PS cavity provided both a wider compressor operable range and higher pressure ratio for OPs between peak efficiency and surge compared to a conventional PS cavity. Based on CFD results and a one-dimensional analysis Tamaki showed that the introduction of counter-swirl to the PS flow increased the impeller inlet flow angle and the work coefficient of the impeller. Additionally, the work coefficient for the counter-swirl case had a larger negative slope for OPs near surge. It was hypothesized that these two factors lead to the increase of the overall compressor pressure ratio and the compressor stability.

Tamaki et al. proposed an additional reason for the range extension for the counter-swirl PS test case [11, 12]. Through CFD calculations the authors observed the presence of a shock located at the shroud region of the inducer, near the PS slot location for the non-ported compressor. The tip leakage vortex originating from the main blade would interact with the shock and be subjected to an adverse pressure gradient that would force it to expand downstream of the shock thus increasing flow blockage. In the case of the conventional PS, the low axial momentum tip
leakage flow would be removed by the PS before interacting with the shock. However, for OPs at a lower mass flow the shock would move further upstream of the PS slot for the conventional PS case. As a result, the low momentum tip leakage flow was not removed via the PS slot before interacting with the shock. The introduction of counter-swirl to the PS flow resulted in the increase of the relative Mach number near the inducer thus moving the shock downstream of the PS slot. This enabled the PS slot to remove the tip leakage flow before reaching the shock [11, 12].

A number of parametric studies have been conducted to assess the influence of key geometric parameters of the PS cavity on the compressor performance. It has been argued that the removal of the backflow region near the inducer tip is beneficial toward improving the surge margin. Since the flow through the PS cavity is pressure driven, changing the width of the PS slot would have a direct effect on the amount of recirculated mass flow. Uchida et al. [8] and Iwakiri et al. [9] showed that an increase of the PS slot width by 20% resulted in an increase of the surge margin. However, a further increase had a negative effect [8]. Numerical results by Sivagnanasundaram et al. [14] showed that increasing the PS Slot from the baseline value of 3mm to 4mm decreased the surge mass flow by 17%. Xinqian et al. [15] conducted a CFD parametric study on a number of PS cavity geometric characteristics listed in Table 1-1 and indicated in Figure 1-3. The distance between the main blade leading edge and the PS slot ($S_r$) was identified as the key parameter. Increase of $S_r$ from 75% of its baseline value to 125% resulted in the decrease of the surge mass flow by 8%. The higher value of static pressure at a location further downstream on the shroud resulted in a larger pressure difference across the PS cavity and therefore higher recirculating mass flow.
Table 1-1: PS geometry characteristics used in parametric study [15] to assess influence of PS geometry on PS compressor performance

<table>
<thead>
<tr>
<th>PS Geometry Characteristic</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$b_b$</td>
<td>width of the horizontal groove</td>
</tr>
<tr>
<td>$b_f$</td>
<td>width of front groove</td>
</tr>
<tr>
<td>$b_r$</td>
<td>width of rear groove</td>
</tr>
<tr>
<td>$h_b$</td>
<td>height of groove</td>
</tr>
<tr>
<td>$S_f$</td>
<td>distance between main blade leading edge and front groove</td>
</tr>
<tr>
<td>$S_r$</td>
<td>distance between main blade leading edge and rear groove</td>
</tr>
</tbody>
</table>

Figure 1-3: PS geometry characteristics used in parametric study [15] to assess influence of PS geometry on PS compressor performance; $S_r$ was found to be key parameter

To date, the majority of researchers investigating PS centrifugal compressor operation have not taken into account the non-axisymmetric nature of the flow inside the volute and the upstream effect on the diffuser and impeller flow field [16, 17, 18]. For OPs near surge the static pressure field in the circumferential direction has a minimum value at the pressure side of the volute tongue implying large negative incidence. For OPs near choke the pressure reaches the
maximum value at the pressure side implying positive incidence with respect to the volute tongue.

Mingyang et al. investigated how the volute tongue induced pressure non-uniformity influenced the operation of non-ported and PS compressors [18, 19]. CFD results of a non-ported compressor showed that the passages subjected to the lowest static pressure at the diffuser inlet had the largest mass flow rates [18]. For the ported compressor the individual flow passages that coincided with higher levels of static pressure experienced larger flow recirculation through the PS cavity. As a consequence, the use of a PS casing treatment reduced the level of the individual passage mass flow non-uniformity [19]. To investigate the effect of the PS device on the compressor stability, the authors of [19] used the slope of total to static pressure ratio $PR$ as a criterion [20]:

$$SP = \sum SP_i \leq 0$$

Equation 1-1

Where $SP_i$ is the stability parameter for each component:

$$SP_i = \frac{1}{PR_i} \frac{\partial PR_i}{\partial \dot{m}}$$

Equation 1-2

The results showed that the use of a PS cavity effectively increased the number of passages that experienced a negative slope, thus increasing the stability of the impeller and consequently of the entire compressor [19].

Xinqian et al. [15] experimented with a non-axisymmetric PS casing treatment that took into account the volute-tongue induced pressure non-uniformity. The non-axisymmetric PS was designed with a sinusoidal distribution in the circumferential direction of the distance between the main blade leading edge and the upstream groove of the PS slot, $S_r$ (Figure 1-3). All other geometric parameters of the PS cavity remained constant. They found that when the phase of the largest $S_r$ of the non-axisymmetric PS coincided with that of the minimum value of static pressure near the leading edge of the splitter blades, a decrease of up to 10% in surge mass flow, compared to a symmetric PS cavity, could be achieved [15]. Increasing the $S_r$ distance at the
circumferential location of minimum static pressure implies that additional mass flow is recirculated compared to an axisymmetric PS. This resulted in a more uniform mass flow distribution through the individual passages and as hypothesized a delay in the onset of surge [15].

Figure 1-4: Gas stand measurements for PS compressor with extended PS Cavity shown in red in (A) indicate decrease of the surge mass flow compared to the baseline PS compressor (B) and a penalty in efficiency of less than two points (C) [13]

A PS compressor configuration where the PS Cavity was extended upstream, shown in red in Figure 1-4 A, was assessed in [13]. Gas stand measurements show a decrease of the surge mass flow for the PS compressor with the extended cavity of up to 15% compared to the baseline PS compressor (chart B) and a penalty of up to two points.
Researchers at Honeywell Turbo Technologies tested a PS compressor configuration where the PS Outlet was blocked [13]. A cross section of the blocked cavity compressor is shown in Figure 1-5 (the impeller is not shown). This configuration did not permit flow to recirculate through the PS Cavity and as such operated as a pressure equalizing chamber. Gas stand measurements of the blocked cavity configuration (right picture of Figure 1-6) indicate that it is as effective as a conventional PS compressor (left picture of Figure 1-6) in extending the compressor operating range at OPs corresponding to $\pi_c < 2.5$. However, at OPs with $\pi_c > 2.5$ not allowing for flow recirculation results in the decrease of the operating range. In terms of efficiency, no noticeable difference between the two tested configurations was measured.

Figure 1-5: Cross section of blocked cavity compressor configuration used by HTT to assess the role of the PS Cavity in attenuating the volute induced flow non-uniformity [13]
Figure 1-6: Gas stand $\pi_c$ measurements for conventional PS (left) and Blocked Cavity (right) configuration by HTT, indicating that not allowing for flow recirculation is detrimental to the compressor performance at OPs with $\pi_c > 2.5$ [13]

1.4 Limitations to Understanding of Ported Shroud Compressor Operation

Review of the pertinent research articles indicates limitations in the understanding of PS compressor operation. There is no consensus on which flow mechanisms associated with PS compressor operation are most critical in setting the performance. An example of lack of consensus concerns the swirl angle of the injected PS flow at the PS Outlet. A number of researchers hypothesize that the pre-swirl contained in the recirculating flow is responsible for the PS compressor range extension through the decrease of the incidence angle [5, 8, 9]. On the other hand, the introduction of counter-swirl is championed by [10, 11, 21] as being more beneficial. It is not clear whether the benefit of using a PS is due to the effect of mass removal, through the removal of low momentum fluid near the inducer shroud, or because of mass
injection upstream of the wheel inlet by changing the impeller incidence angle or energizing the flow near the shroud.

During a PS compressor operation there are multiple flow actuations happening simultaneously. Mass flow is removed at the PS Slot and re-injected upstream, the PS Outlet also acts as a source of axial and angular momentum injection. Based on the literature review that was performed, no one has attempted to isolate the PS flow actuations and investigate their relative importance in setting the compressor performance. Furthermore, it is not clear whether these PS flow actuations are equally as effective for the entire compressor map or are some actuations more effective than others at different operating points and speedlines.

Finally, whilst most research papers focus on the range extension provided by a PS, assessment of the entropy generation processes associated with PS compressor operation has been mostly neglected. Quantification and assessment of the flow mechanisms responsible for the efficiency penalty normally experienced in PS centrifugal compressors would lead to more efficient PS compressors.

1.5 Hypotheses, Research Objectives and Questions

Based on the insights that have been obtained through the literature review on the subject of PS operation, the following three initial working hypotheses are formulated:

**Hypothesis 1:** It is hypothesized that mass flow removal at the PS slot removes low momentum fluid near the shroud region and decreases the extent of the backflow region near the endwall. As a consequence, the PS works in such a way as to decrease the flow blockage in the inducer and increase the compressor operable range and pressure rise capability. Furthermore, removal of the recirculation zone near the shroud endwall is conducive to improving impeller efficiency.
Hypothesis 2: The second working hypothesis is that the implementation of the PS recirculating device assists in the re-matching of individual compressor components. The recirculation of flow through the PS cavity near surge increases the mass flow through the inducer. By doing so, the inducer is essentially working at operating conditions closer to peak efficiency compared to the rest of the compressor, resulting in the postponement of inducer stall and the extension of the compressor operable range.

Hypothesis 3: It is postulated that the PS works in such a way as to enhance the attenuation of the volute induced non-uniformity for off-design OPs and consequently decouple the inducer from the remaining compressor. Enhanced attenuation of the pressure non-uniformity decreases the flow distortion at the wheel inlet and decreases the mass flow at which the compressor surges.

The goal of this thesis is to define and quantify the flow processes associated with PS compressor operation. This would allow formulation of guidelines for the design of more effective PS recirculation devices and consequently improved centrifugal compressor performance. Specifically, the following items are to be addressed:

- Identify and quantify flow mechanisms that are effective in increasing the compressor operable range and pressure ratio but decrease the associated efficiency penalty due to the implementation of a PS casing treatment. Accomplishing this would necessitate assessing the differences in flow processes between individual components (inlet, PS cavity, impeller, diffuser, volute) of compressors with and without PS.
  - How does the transport of fluid (removal and subsequent injection) along the compressor flow path influence:
    - Blockage – compressor pressure rise capability?
    - Entropy Generation – compressor efficiency?

- Identify and assess quantitatively the role of individual flow actuations associated with PS operation in setting the compressor performance. The function of the PS casing treatment consists of a number of flow actuations that are performed at the PS Slot (mass...
and momentum removal) and the PS Outlet (mass injection as well as axial and angular momentum injection).

- How do these PS flow actuations individually affect the compressor performance (pressure rise capability, efficiency, stability)?
- Which of the individual PS flow actuations have a larger impact on the compressor performance? Does the main flow perceive the presence of the PS primarily as a source/sink of mass, momentum of energy flux?
- Are the effects of the individual PS flow actuations additive?
- Can we formulate a preliminary design model to simulate the PS flow actuations on the main flow?

- Assess the mechanism through which the PS recirculating casing treatment device affects the upstream influence of the volute induced circumferential flow non-uniformity.
  - Does the PS work as a pressure equalizing device, thus effectively decoupling part of the impeller from volute induced flow non-uniformity or are the PS flow actuations dominant in setting the compressor performance?
  - Could these two mechanisms be working in unison or is one more beneficial than the other? If so, is this valid for the entire compressor map?

- Leverage the findings of this thesis in order to explain experimental measurements found in literature and Honeywell in-house data regarding PS compressor operation.
  - The data measurements reported by Yamaguchi et al. [10] demonstrating the enhanced compressor operating range achieved through the introduction of counter-swirl to the injected flow at the PS Outlet.
  - In-house Honeywell data regarding the PS compressor test configuration where the PS outlet has been blocked and the PS Cavity acts as a pressure equalizing chamber (gas stand data measurements shown in Section 1.3 [13]).
1.6 Research Contributions

The contributions of this thesis toward the understanding of PS compressor operation includes:

- It is shown that the main flow path perceives the PS flow as a combination of flow actuations that include mass flow removal at the PS Slot, mass flow injection and momentum (axial and tangential) injection at the PS Outlet.

- The effect of mass flow removal is the dominant PS flow actuation in setting the PS compressor performance. The removal of mass flow results in the diffusion of the compressor main stream and decreases the impeller flow blockage. The decrease of the blockage stems from the removal of the inducer tip leakage flow and results in the increase of the blade loading. The diffusion of the compressor main stream and reduction of the impeller flow blockage are responsible for increasing the compressor stability. The partial removal of the recirculation is shown to also be responsible for the decrease of the entropy generation inside the impeller.

- The introduction of counter-swirl (opposite to the wheel rotation) to the PS flow at the PS Outlet increases the compressor pressure ratio and static pressure rise and decreases the efficiency. The higher pressure ratio is a result of the impeller doing more work on the flow due to the higher blade inlet flow angle compared to co-swirl injection. The decrease in efficiency is due to the higher mixing loss between tip leakage and main flow. The effect of angular momentum injection on the compressor performance is shown to decrease at OPs with high flow blockage upstream of the blade LE.

- Removing and discarding the PS flow is more effective in increasing the compressor stagnation pressure ratio than recirculating it, while succumbing to no additional penalty in efficiency.

- The attenuation of the flow non-uniformity by the PS Cavity at OPs corresponding to high pressure ratios $\pi_c > 3$ is not responsible for the increased compressor operable range, but rather the effect of the PS flow actuations.
1.7 Organization of Thesis

The research compressor that serves as the test bed for this thesis is firstly introduced in Chapter 2, followed by a delineation of the research roadmap that has been used in order to address the research questions. A description of the geometry and the computational models of the compressor variants that have been employed in the context of the CFD modeling are then provided. Chapter 2 concludes with a brief summary of the techniques used in order to obtain the experimental data that has been made available by researchers at the University of Cincinnati for assessing the computed flow field.

The CFD generated flow fields of both ported and non-ported variants of the research compressor are interrogated in Chapter 3 and compared against available experimental data. Through this assessment, flow mechanisms responsible for extending the PS compressor operable range, increasing the pressure ratio and decreasing the efficiency are identified. Furthermore, the flow entering and exiting the PS Cavity is assessed with the goal of representing the PS flow through a combination of flow actuations, such as mass injection/removal, angular momentum injection etc., at the PS Outlet and Slot.

In Chapters 4, 5 and 6 the effect of PS flow actuations on the compressor performance are assessed. The computational platform for this investigation is a single passage model of the compressor which does not include the PS Cavity and where the PS flow actuations are modeled through appropriate boundary conditions. In Chapter 4, mass flow removal at the PS Slot is assessed. The effect of mass flow injection, as well as axial and angular momentum injection at the PS Outlet on the compressor performance are individually examined in Chapter 5. In Chapter 6, the effect of combinations of PS actuations at the PS Slot and Outlet on the compressor performance is assessed. A test case where mass flow is removed at the PS Slot and discarded rather than being recirculated through the PS Cavity is examined first. The combined effect of mass flow injection with co-swirl and counter-swirl at the PS Outlet with that of mass flow removal at the PS Slot is then examined. A physical interpretation of the PS compressor operation in context of the PS flow actuations is given at the end of the chapter.
The preliminary results from assessing the role of the PS cavity in attenuating the circumferential flow non-uniformity originating from the volute are presented in Chapter 7. A variant of the PS compressor is modeled in which the PS Outlet is blocked, such that flow cannot recirculate through the PS cavity, therefore acting as a pressure equalizing device. The performance of the baseline PS compressor is compared against the blocked cavity configuration with the purpose of assessing the relative importance of the PS cavity in attenuating the flow non-uniformity.

The results and conclusions established throughout the thesis are summarized in Chapter 8. The chapter concludes with a list of proposed topics for future research related to the operation of PS compressors.
Chapter 2  Framework of Approach

The research roadmap that has been followed to address the research questions is described in this chapter. An introduction to the research compressor that serves as the platform for this thesis is then provided. The third section of this chapter includes a description of both computational models and methods that have been employed. Finally, a brief summary of the experimental data that have been made available during this thesis is given at the end of the chapter.

2.1 Research Roadmap

The strategy adopted in this work is outlined in the following steps:

1. Investigate and assess the performance and flow field differences between ported and non-ported centrifugal compressors
   a. Develop a CFD model of both ported and non-ported variants of the centrifugal compressor to be used in this study.
   b. Assess the computed performance and flow field against available experimental measurements; gas stand data provided by HTT and PIV measurements taken at the University of Cincinnati.
   c. Identify flow processes responsible for extending the PS compressor range and decreasing the efficiency.

2. Assess how the individual flow actuations associated with the PS affect the compressor performance. An example of this would be to maintain constant mass flow recirculating through the PS cavity but alter the swirl angle of the PS flow at the PS outlet. To this end, a single passage CFD computational model that does not include the PS Cavity component of the PS compressor is constructed. Specifically, the PS cavity is removed from the computational domain of the CFD model of the full PS compressor and the
inlet/outlet regions of the PS cavity are modeled through the application of appropriate boundary conditions at the PS Outlet and Slot (Figure 2-1). This approach allows for direct and independent control of the injected/removed flow at the two ends of the PS Cavity. Based on this proposed approach, the following steps are taken:

a. Assess the compressor performance of the PS Actuation Model against that of an axisymmetric PS compressor (without the volute and PS Cavity ribs) for OPs near peak efficiency and surge.

b. Use the PS Actuation Model to assess the effect of individual PS flow actuations at the PS Slot and Outlet on compressor performance.

c. Define the PS flow actuations that are effective in setting the compressor performance.

d. Use analytical tools to describe and quantify the effect of the PS flow actuations on the compressor performance.

![Diagram](image)

**Figure 2-1:** The PS Cavity is geometrically removed from the rest of the single passage compressor in the PS Actuation Model and the PS flow is modeled through BCs at the PS Outlet and Slot

3. Assess the interaction of the PS with the flow non-uniformity that is induced by the volute tongue. Address whether the primary role of the PS is that of a pressure equalizing device or whether the PS flow actuations at the PS slot and outlet are responsible for the improved compressor performance. To address this question, the PS...
outlet of a conventional PS compressor is blocked. By doing so, the PS cavity effectively works as a non-recirculating casing treatment device.

4. Synthesize the findings from the three previous steps and suggest required attributes of PS compressors for performance improvement.

### 2.2 Research Compressor

The centrifugal compressor that has been used as a research platform for this thesis is the Honeywell Turbo Technologies (HTT) GT40 C239A used in commercial vehicle applications. The C239A is a ten bladed centrifugal compressor with back sweep of 25° at the impeller TE with respect to the radial direction, exducer tip diameter of $D_{\text{exducer,tip}}=88$ mm, exit/inlet wheel area ratio 0.475, ratio of volute throat area to radius 0.57 inches (14.48 mm) and a vaneless diffuser of area ratio 1.3 [25, 26]. The PS variant of this compressor includes a PS cavity inside the inducer shroud supported by four structural ribs positioned circumferentially, with a PS slot of 2.5 mm width. A cross sectional and frontal view of the compressor is shown in Figure 2-2.

![Cross section and frontal view of the C239A Ported Shroud Compressor, indicating key compressor components and structural ribs inside PS Cavity [25]](image-url)
Gas stand data for both ported and non-ported variants of the C239A have been provided by HTT [13]. The experimentally measured stagnation pressure ratio $\pi_c/\pi_{c,\text{ref}}$ expressed as a ratio with respect to a reference value of $\pi_{c,\text{ref}} = 1.91$, is shown in Figure 2-3. The value $\pi_{c,\text{ref}} = 1.91$ corresponds to the pressure ratio of the peak $\eta$ OP of the compressor map. Measurements of efficiency are shown in Figure 2-4. In terms of the compressor operable range, the PS decreases the surge mass flow by 27% at $0.84 \, N_{\text{ref}}$ and 12% at $1.48 \, N_{\text{ref}}$, while the choke mass flow is increased by up to 8% at $1.48 \, N_{\text{ref}}$. The $\pi_c$ of the PS compressor is increased by up to 4.5% at $1.32 \, N_{\text{ref}}$ compared to that of the non-ported variant. In terms of efficiency, the use of a PS decreases the efficiency of the compressor by up to 12 points for OPs near surge at the $0.68 \, N_{\text{ref}}$ and 2 points at $1.48 \, N_{\text{ref}}$. 

Figure 2-3: Gas stand $\pi_c$ measurements for ported and non-ported C239A variants indicating higher $\pi_c$ and wider compressor operable range for PS compressor (HTT [13])
Figure 2-4: Gas stand \( \eta \) measurements for ported and non-ported C239A variants indicating efficiency penalty associated with PS compressor operation (HTT [13]).

2.3 Computations

Computational Fluid Dynamics (CFD) has been employed to model and assess the behavior of both ported and non-ported variants of the research compressor. The commercially available general purpose fluid dynamics program ANSYS CFX5 has been used. In this section, description of the computational models, parameters and methods used is given.
2.3.1. **Computational Method**

The compressor flow fields are generated using steady state RANS with the Shear Stress Transport (SST) turbulence model. The SST turbulence model, developed by Menter, combines the k-ω model in the near wall region of the domain and the k-ε model away from the wall as a unified two-equation turbulence model [22, 23, 24]. Since steady state calculations are conducted, the interfaces between domains resolved in the stationary and the relative frame of reference are selected to be of the Frozen Rotor type. The use of the Frozen Rotor type interface is preferred over that of a mixing plane because it does not average the flow in the circumferential direction before passing information between domains resolved in different frames of reference. Therefore, the flow non-uniformity induced by the volute is able to extend upstream into the wheel domain. This would not be the case if a mixing plane interface were used. The drawback is that the relative positioning of the stationary and rotating domains is locked with each other. To assess the use of steady state calculations the reduced frequency $f_{red}$ of the compressor is estimated through the use of Equation 2-1, where $L$ and $U$ is a characteristic through-flow velocity and length respectively and $n$ the number of "lobes" of the circumferential disturbance [27].

$$f_{red} = \frac{\omega L n}{U_{\bar{r}}}$$

Equation 2-1

The circumferential non-uniformity induced by the volute tongue has a 1/rev characteristic length scale ($n=1$) and the reduced frequency of the compressor is $f_{red} = 0.4$. Based on HTT in-house experience [28], the use of steady state CFD calculations for similar values of reduced frequency are in accordance with gas stand measurements.
2.3.2. **Computational Domain of Compressor Variants**

Full wheel and single passage models of the ported and non-ported versions of the C239A have been used as platforms to answer the research questions of this thesis. In this section, a brief description of the computational domain of the compressor variants that are used is provided.

2.3.2.1. **Full Wheel Ported and Non-Ported Compressor Variants**

The computational model of the full wheel PS compressor is shown in Figure 2-5. The model contains three domains; the wheel (yellow), diffuser-volute (green) and the port-inlet (blue). The entire compressor model is discretized using a structured grid of hexahedral elements which has been generated by HTT [28]. The wheel domain is discretized using approximately 1.5 million hexahedral elements, the diffuser-volute region with one million elements and the port-inlet region with two million elements. The wheel domain is solved in the relative frame, while port-inlet and diffuser-volute domains are solved in the stationary frame. Frozen Rotor type interfaces have been used between domains resolved in different frames of reference and are depicted in red in Figure 2-5; two interfaces between wheel and port-inlet (interface A and B) domain and between the wheel and diffuser-volute domain (interface C). One of the structural ribs inside the PS cavity can be seen in the right picture in Figure 2-5. Stagnation pressure and temperature and axial direction of the flow are prescribed at the compressor inlet as boundary conditions. Mass flow is prescribed at the volute discharge as the outlet boundary condition. The specified mass flow is enforced at each timestep; hence the pressure distribution is an implicit part of the simulation [24]. The mass flow value is set at the volute outlet to simulate OPs that can be compared consistently with the available gas stand data.
Figure 2-5: Full wheel PS compressor computational domain; Frozen Rotor type interface between wheel and port-inlet domains at A and B, and between wheel and diffuser-volute domains at C

Figure 2-6: Full wheel non-ported compressor computational domain; Frozen Rotor type interface between wheel and inlet domains at A, and between wheel and diffuser-volute domains at C

The non-ported computational model (Figure 2-6) is similar to the ported case, with the difference that the PS cavity is not included and the outer diameter of the inlet domain is comprised of the upstream extension of the impeller shroud. The model contains the same wheel (yellow) and diffuser-volute (green) domain as the PS compressor case, while the inlet region
(blue) is discretized using approximately one million hexahedral cells. Two Frozen Rotor type interfaces are used; between wheel and inlet section (interface A in Figure 2-6) and between wheel and diffuser-volute domain (interface C). As with the PS compressor simulations, stagnation pressure and temperature with axial direction of the flow are prescribed as inlet boundary conditions, while the mass flow value is set at the exit.

2.3.2.2. Single Passage PS Actuation Model

A single passage model of the C239A serves as the platform for assessing the effects of the PS flow actuations on the compressor performance. As described in Section 2.1, the PS cavity is not included physically in the computational domain and the PS flow is modeled through the use of boundary conditions at the PS Outlet and Slot locations (shown in green in Figure 2-7). Depending on the location of the PS actuation that is being investigated, two versions of the single passage PS Actuation Model are used. When flow is injected at the PS Outlet location the single passage model shown in the left picture of Figure 2-7 is used. When PS flow actuations at the PS Slot are investigated, the computational model on the right of Figure 2-7 is used.

Figure 2-7: Meridional cross section of single passage PS Actuation Model variants; actuations at PS Outlet active (left), actuations at PS Slot active (right); PS Outlet and Slot locations indicated in green; Frozen Rotor type interfaces between Wheel and Inlet (A), Wheel and PS Extension (B), and Wheel and Diffuser (C)
The same wheel and diffuser computational domains are used for both variants of the PS Actuation Model and are discretized using one million and 200,000 elements respectively. The inlet domain corresponding to the version when the PS flow actuations at the PS Outlet are active consists of approximately 1.2 million elements, while the inlet used when investigating PS actuations at the slot is discretized into approximately 500,000 elements. When assessing flow actuations at the PS Slot, a radial diffuser is added as an extension of the PS Slot location. The addition of this domain is necessary when prescribing the PS Slot as an outlet of the CFD domain. Based on the full wheel PS compressor simulations, it is possible for the flow at the PS Slot location to locally have a radial velocity pointing inward, even when flow is removed through the PS Slot and injected upstream. Since outlet boundary conditions cannot be imposed at locations where the flow is not unidirectional, the radial diffuser at the PS Slot is added to increase the path so that the flow at the exit of the diffuser is unidirectional. Frozen Rotor type interfaces are used between domains resolved in stationary and rotating frames of reference and are shown in red in Figure 2-7; between wheel and inlet domain (interface A), wheel and PS extension domain (interface B) and wheel and diffuser (interface C). Stagnation pressure and temperature and axial direction of the flow are always prescribed as boundary conditions at the compressor inlet. Mass flow is prescribed as the boundary condition at the exit of the diffuser. When assessing PS flow actuations at the PS Outlet, mass flow, stagnation temperature and the desired flow angle of the injected flow are prescribed as inlet boundary conditions. When investigating mass flow removal at the PS Slot, mass flow is prescribed as an outlet boundary condition at the PS Slot.

2.4 Experiments

Experimental measurements of the flow field inside the PS compressor have been obtained by researchers at the University of Cincinnati. Guilliou [25] and Gancedo [26] used the same GT40 C239A PS centrifugal compressor as a test bed for the experimental study of the development of surge in PS centrifugal compressors. Pressure measurements have been acquired inside the
impeller and the diffuser. The use of stereoscopic and in-plane PIV measurements quantitatively maps the flow field upstream of the wheel and inside the PS cavity. Crossflow plane velocity components inside the PS cavity are measured after replacing a portion of the impeller housing with a transparent Plexiglas window thus permitting the penetration of the laser sheet inside the cavity. These PIV measurements are used for the assessment of the CFD generated flow field of the PS compressor to establish confidence in the adequacy of the computational method used.

Figure 2-8: Stereoscopic PIV set-up for acquiring data upstream of the wheel inlet [26]
Chapter 3  Flow and Performance Characteristics of Ported Shroud and Non-Ported Compressor

In this chapter the performance characteristics of ported and non-ported compressors are addressed. The CFD generated performance curves and flow field are compared against experimental measurements as a means to assess the adequacy of the computed results for addressing the research questions posed in Chapter 1. This is followed by assessing the impact of introducing a Ported Shroud on the compressor flow field. Through the interrogation of the flow field inside the PS cavity and at the PS Outlet focus is given in understanding how the main flow perceives the flow exiting the PS. Another focal point of this chapter is the quantification of the effect the PS has on the entropy generated inside the compressor and the evolution of impeller flow blockage.

3.1 Comparison between CFD and Experimental Data

CFD simulations have been conducted using the full wheel ported and non-ported variants of the C239A. The details of the computational domain, numerical methods and boundary conditions used for implementing these simulations have been described in Section 2.3. The computed compressor stagnation pressure ratio $\pi_c$ and efficiency $\eta$ are compared against the gas stand measurements and are shown in Figure 3-1 and Figure 3-2. Simulations have been undertaken at three speedlines, $0.84 \, N_{ref}$, $1.16 \, N_{ref}$ and $1.48 \, N_{ref}$, for the PS compressor and at the $0.84 \, N_{ref}$ speedline for the non-ported variant. As mentioned in Section 2.2, with respect to the gas stand measurements, the largest percentile decrease of the surge mass flow between ported and non-ported compressor variants is measured at the $0.84 \, N_{ref}$ speedline (27%), while the largest decrease in absolute terms is measured at $1.48 \, N_{ref}$. The gas stand data shows that the
efficiency penalty between ported and non-ported compressors decreases when moving to higher speedlines, from 12 points for OPs near surge at $0.68 \, N_{ref}$ to two points at $1.48 \, N_{ref}$. These three speedlines are selected because the effect of the PS on the compressor performance can be assessed over a wide range of the compressor map.

The assessment between experimentally measured and CFD computed compressor performance shows that the numerical results capture the trend of the experimental measurements and that there is good agreement between the two. The CFD over-estimates the pressure ratio by up to 3.5% at the high speedline, while the discrepancy between experimentally measured and computed $\eta$ is less than two points for OPs between peak efficiency and surge. The largest discrepancies are present for high mass flow OPs; at these OPs the steep experimental curve suggests that the compressor is close to choke and therefore the measured efficiency and pressure ratio values are sensitive to any small measurement error in mass flow rate.

Figure 3-1: Comparison between numerical and gas stand $\pi_c$ measurements for ported and non-ported variants of C239A
In addition to the gas stand data, the CFD calculations have also been assessed against detailed PIV measurements. Figure 3-3 shows a comparison of the in-plane velocity components $V_{xy}/U_{tip}$ between PIV measurements (left) and CFD (right) for the PS compressor operating at conditions near surge $\dot{m}/\dot{m}_{ref} = 0.33$ at 0.84 $N_{ref}$. The measurements are taken at a crossflow plane located at a distance of 30% of the outlet diameter upstream of the PS Outlet. The flow at the outer radius corresponds to the flow exiting the PS Outlet. The velocity vectors of the in-plane velocity indicate that the flow exits the PS Cavity in the form of four jets, circled in the left picture in Figure 3-3, due to the four structural ribs inside the PS Cavity as will be shown in subsequent sections of this chapter. The radial $u_r/U_{tip}$ and circumferential $u_\theta/U_{tip}$ velocity profiles at locations 1 and 2 (shown in the right picture of Figure 3-3) are shown in Figure 3-4 for both PIV measurements and CFD. Location 1 corresponds to a circumferential angle of one of the structural struts inside the PS cavity, while location 2 is situated at an approximate equal circumferential distance between two adjacent struts. Therefore, it is assumed that the flow field at these two locations is representative of the flow at the PS Outlet and can be used for
comparative purposes. Both CFD and PIV data indicate that the flow exiting the PS has a large circumferential component (> 0.3 $U_{tip}$) in the direction of the wheel rotation. Comparison of the CFD generated flow field and the experimental measurements upstream of the PS Outlet indicate qualitative and quantitative agreement even at conditions near surge.

The observed accord between CFD and experimental measurements (gas stand and PIV) implies that the CFD model can be used as a platform to assess the compressor performance changes associated with PS casing treatment.

![Jet exiting PS Outlet](image)

Figure 3-3: PIV (left) and CFD (right) in-plane velocity components at crossflow plane upstream of PS Outlet for PS compressor OP $\dot{m}/\dot{m}_{ref} = 0.33$ at 0.84 $N_{ref}$; the data suggests that the flow exits the PS Cavity in the form of four jets with a large circumferential component in the direction of the wheel rotation circled in red in the left chart.
Figure 3-4: Radial and circumferential velocity profiles at crossflow plane upstream of PS Outlet (refer to Figure 3-3) from PIV and CFD for PS compressor OP $\dot{m}/\dot{m}_{ref} = 0.33$ at $0.84N_{ref}$, the data indicates that the flow exiting the PS Cavity has a strong circumferential component in the direction of the wheel rotation $u_\theta/U_{tip} > 0.3$

### 3.2 Flow Inside the PS Cavity

The mass flow recirculating through the PS Cavity $\dot{m}_{PS}$, as a fraction of the compressor inlet flow, is shown in Figure 3-5. Positive values of $\dot{m}_{PS}/\dot{m}$ indicate flow being removed from the impeller passages and injected upstream of the wheel inlet, while negative values correspond to conditions where a fraction of the inlet flow bypasses the impeller LE and is injected into the main flow at the PS Slot location. The PS recirculates up to 71% of the compressor inlet flow at conditions near surge when operating at the low speedline $0.84N_{ref}$, approximately 45% at the
mid speedline $1.16 \, N_{ref}$ and only 10% at $1.48 \, N_{ref}$. The circled data points in Figure 3-5 correspond to the peak $\eta$ OP at each speedline. At conditions near peak $\eta$ less flow is recirculated through the PS Cavity. It is therefore inferred that at conditions corresponding to peak $\eta$ the PS compressor behavior is similar to that of the non-ported compressor. This is in accordance with the experimentally measured $\eta$, shown in Figure 2-4, where the difference in efficiency between ported and non-ported variants decreases at higher speedlines for peak $\eta$ OPs.

![Figure 3-5](image)

Figure 3-5: Computed fraction of PS mass flow indicating that up to 70% of the inlet mass flow is recirculated through the PS Cavity at OPs near surge; the circled data points correspond to peak $\eta$ OPs

The recirculating flow exits the PS Outlet in the form of four jets, which can be seen in Figure 3-6, where the axial velocity of the exiting flow $u_{axial}/U_{tip}$ is plotted at the PS Outlet for an OP near surge conditions $\dot{m} = 0.3\dot{m}_{ref}$ at $0.84 \, N_{ref}$. Negative values of $u_{axial}/U_{tip}$ indicates flow exiting the PS Outlet. This distinct pattern of the exiting flow is due to the presence of the four structural ribs inside the PS cavity. The high fraction of mass flow that is recirculated through the PS cavity in combination with the four jet pattern results in high values of the axial momentum $J = \int u_{axial} \, d\dot{m}$ of the flow exiting the PS. The axial momentum of the flow exiting at the PS Outlet $J_{PS\,outlet}$ is shown to be up to 9 times that of the main inlet stream $J_{inlet}$ (Figure
3-7). This implies that the main inlet flow perceives the PS flow as a source of axial momentum injection and that the effect of this PS flow actuation on the PS compressor performance should be assessed.

![Figure 3-6: Axial velocity $u_{axial}/U_{tip}$ at PS Outlet for PS compressor OP $m/\dot{m}_{ref} = 0.3$ at 0.84 $N_{ref}$; negative velocity indicates flow exiting PS Cavity in the form of four jets](image)

![Figure 3-7: Ratio of axial momentum of flow exiting PS cavity $J_{PS \text{outlet}}$ with respect to axial momentum of main inlet stream $J_{inlet}$; it is inferred that at OPs near surge the PS acts as a source of axial momentum injection with respect to the main inlet flow](image)
To assess the flow field in the PS Cavity, the flow at surface $S$, located at the mid-radius $\bar{R}_{PS}$ of the PS Cavity depicted in Figure 3-8, is examined. Figure 3-9 shows the streamlines inside surface $S$ of the PS cavity overlaid on the $Cp$ contour. The axis of rotation coincides with the $z$-axis of Figure 3-9. The flow that is removed at the PS Slot contains swirl in the direction of the wheel rotation. This flow then intercepts the structural ribs at a large incidence angle, resulting in a large separation zone and a decrease of the static pressure near the suction side of the struts (Region A in Figure 3-9). The flow exiting the PS Outlet near the pressure side of the ribs is unable to follow the corner at the TE (sudden expansion) of the struts leading to corner separation (Region B in Figure 3-9). These two regions of low static pressure entrain a portion of the flow exiting the PS cavity back inside the PS cavity thus producing the four jet formation shown in Figure 3-6.

![Figure 3-8: Surface of revolution S located at mid-radius of PS Cavity $\bar{R}_{PS}$](image-url)
One of the hypotheses of this thesis is that the PS works in such a way as to remove the low momentum fluid near the inducer shroud. To assess this hypothesis, the flow blockage along a number of crossflow planes is estimated using the blockage calculation method proposed by Khalid [29, 30].

Assuming that a main flow direction can be identified, the velocity component in the main flow direction $u_m$ and the gradient $\nabla (\rho u_m)$ at any location in the flow field can be calculated. The magnitude of the two components of $\nabla (\rho u_m)$ in the circumferential and spanwise direction are used as an appropriate scalar to identify regions of high blockage generation. Upon detecting defect regions, the flow blockage is then calculated through an equation based on the concept of
the two-dimensional boundary layer thickness [29, 30]. 0 contains a more detailed description of the procedure followed to identify defect regions and estimate the flow blockage.

Figure 3-10: PS compressor (solid red) decreases flow blockage by up to 25% compared to non-ported compressor (dashed red) at near surge OP at 0.84 \( N_{\text{ref}} \).

The blockage, as a percentage of the local geometric area of each crossflow plane, for both ported and non-ported OPs at 0.84 \( N_{\text{ref}} \) is shown in Figure 3-10. Comparison between ported (solid red) and non-ported compressor (dashed red) at the near surge OP of the non-ported compressor indicates a decrease in the passage blockage of more than 25% upstream of the PS Slot location. The decrease in flow blockage along the impeller passage is traced to the removal of the recirculation zone present near the inducer shroud region due to the inducer tip leakage flow. Figure 3-11 shows high blockage regions at a crossflow plane upstream of the PS Slot and surface streamlines on a meridional plane perpendicular to that crossflow plane. The examined OP corresponds to near surge for the non-ported compressor \( \dot{m}/\dot{m}_{\text{ref}} = 0.41 \) and \( \dot{m}/\dot{m}_{\text{ref}} = \).
0.38 for the PS compressor at 0.84 $N_{ref}$. Figure 3-11 indicates that the regions of high blockage inside the impeller correspond to locations where flow recirculation is present. The use of a PS decreases the extent of the recirculation zone near the inducer shroud compared to the non-ported compressor, resulting in the decrease of the flow blockage by up to 25% as quantified in Figure 3-10.

Figure 3-11: Regions of high blockage at a crossflow plane upstream of PS Slot correspond to locations with flow recirculation indicated by the streamlines. Removal of this recirculation results in the decrease of blockage between non-ported $\dot{m}/\dot{m}_{ref} = 0.41$ (left) and PS compressor $\dot{m}/\dot{m}_{ref} = 0.38$ (right) OPs at 0.84 $N_{ref}$.

Figure 3-12 shows the reduction in blockage between the ported OP $\dot{m}/\dot{m}_{ref} = 0.38$ and the non-ported OP $\dot{m}/\dot{m}_{ref} = 0.41$ at 0.84 $N_{ref}$. The blue curve corresponds to differences in blockage across the passage span, while the red curve corresponds to the decrease in blockage in the top 30% of the passage span. The reduction in blockage for the entire passage height is traced to the reduction in the top 30% of the span; hence it is inferred that the removal of the recirculation zone is responsible for the blockage reduction in the entire impeller.
3.4 Entropy Generation for Ported and Non-Ported Compressors

To identify and assess the entropy generation mechanisms associated with PS compressor operation, the entropy production inside the individual compressor components, delineated in Figure 3-13, for ported and non-ported compressor OPs at $0.84 N_{ref}$ is shown in Figure 3-14. The first two columns in Figure 3-14 correspond to the peak $\eta$ OP for the non-ported and PS compressor respectively. The third and fourth columns, non-ported $\dot{m}/\dot{m}_{ref} = 0.41$ and ported $\dot{m}/\dot{m}_{ref} = 0.38$, correspond to the near surge OP of the non-ported case, while the last column is the PS compressor surge OP. The entropy production shown in Figure 3-14 and throughout this thesis unless otherwise mentioned, is calculated as the net $s_{flux}$ across a control volume and normalized by the total enthalpy flux according to Equation 3-1:
\[
\Delta s = \frac{T_{t,\text{out}} \left( \int s_{\text{out}} \, d\dot{m}_{\text{out}} - \int s_{\text{in}} \, d\dot{m}_{\text{in}} \right)}{\int h_{t,\text{out}} \, d\dot{m}_{\text{out}} - \int h_{t,\text{in}} \, d\dot{m}_{\text{in}}}
\]

Equation 3-1

Figure 3-13: PS Compressor components

Figure 3-14: Compressor component entropy production for ported and non-ported compressors at 0.84 \(N_{\text{ref}}\): PS compressor (4th column) decreases the entropy generation inside the impeller by 25\% compared to non-ported compressor (3rd column) at near surge OPs
Comparison of the entropy generated between ported $\dot{m}/\dot{m}_{ref} = 0.38$ (4th column) and non-ported $\dot{m}/\dot{m}_{ref} = 0.41$ (3rd column) compressor variants (OP corresponding to near surge for the non-ported compressor) shows that the use of a recirculating device decreases the entropy generated inside the impeller by up to 25%. Figure 3-14 indicates that the entropy generated inside the PS Cavity and inlet region accounts for approximately 25% and 23% respectively of the total entropy generated inside the PS compressor.

The entropy generation rate per unit volume due to viscous dissipation, $\dot{S}_{visc}$, is plotted for an impeller crossflow plane upstream of the PS Slot in Figure 3-15. The non-dimensional $\dot{S}_{visc}$ is calculated through Equation 5-2.

\[
\dot{S}_{visc} = \frac{T_{t, outlet} \dot{S}_{visc}}{\frac{1}{2} \left( \frac{\rho_{inlet} U_{tip}^3}{c} \right) c}
\]

Equation 3-2

where $\dot{S}_{visc} = \frac{1}{T} \tau_{ij} \frac{\partial u_i}{\partial x_j}$

$c$: length of impeller blade

$U_{tip}$: Rotation velocity at wheel outlet

$\tau_{ij}$: symmetric viscous stress tensor

$T$: Static Temperature

The contour in the left of Figure 3-15 shows that regions of high entropy generation $\dot{S}_{visc}$ for the non-ported compressor are located near the shroud of the impeller and correspond to regions of flow recirculation. The removal of the flow recirculation with the use of the PS decreases the entropy generation inside the impeller passage as shown in the right of Figure 3-15. The entropy generation decrease in the top 30% of the impeller span accounts for approximately 90% of the total decrease of entropy generation inside the impeller between ported and non-ported
compressors. This implies that the removal of the flow recirculation is the mechanism through which the impeller efficiency is improved.

Figure 3-15: Regions of high local entropy generation rate $\dot{S}_{\text{visc}}$ inside impeller crossflow plane upstream of PS Slot correspond to regions of flow recirculation for non-ported $\dot{m}/\dot{m}_{ref} = 0.41$ (left); removal of the flow recirculation results in decrease of $\dot{S}_{\text{visc}}$ for PS compressor $\dot{m}/\dot{m}_{ref} = 0.38$ (right) at 0.84 $N_{ref}$.

It has been shown that the entropy generated inside the PS Cavity accounts for approximately 25% of the total compressor loss (see Figure 3-14). The entropy generation inside the PS Cavity is depicted with the solid lines in Figure 3-16. The data is plotted against the fraction of recirculating flow with respect to the compressor inlet mass flow $\dot{m}_{PS}/\dot{m}$. The dashed lines correspond to the two-dimensional profile loss for each OP. The entropy generated in the boundary layer is calculated through the use of Equation 3-3.

$$T\dot{S}_{BL} = C_d \sum_{\text{surfaces}} \rho U^3 \int_0^{x_{\text{final}}} \left( \frac{U}{U} \right)^3 \frac{d}{L} \left( \frac{x}{L} \right)$$

Equation 3-3
Where $\dot{S}_{BL}$: the total rate of boundary layer entropy generation per unit depth

$C_d$: the dissipation coefficient, taken equal to 0.002 [27]

$u_E$: the freestream velocity

$U$: reference velocity

$L$: reference length

![Graph showing entropy generated due to profile losses](image)

**Figure 3-16**: Entropy generated due to profile losses accounts for approximately 15% of the total entropy generated inside PS Cavity

Figure 3-16 indicates that the profile loss accounts for approximately 15% of the loss inside the PS cavity. The entropy generated due to profile losses corresponds to the lowest entropy generation bound for the PS Cavity. These values serve as a metric of the additional loss generated inside the PS cavity because of flow separation and mixing of the flow exiting and entering the PS cavity as shown in Figure 3-9. Therefore, the difference between the actual loss generated inside the PS Cavity and the profile loss indicates the maximum improvement in terms
of entropy that can be obtained by designing a PS Cavity that provides uniform flow at the PS Outlet and that eliminates any flow separation.

To assess the entropy generation mechanism in the inlet region, the entropy generation rate $\dot{S}_{visc}$ upstream of the PS Outlet is shown in Figure 3-17. The $\dot{S}_{visc}$ contour indicates four regions of high entropy production near the outer radius of the inlet, one of which is highlighted as Region A in Figure 3-17. These regions correspond to the four jets exiting the PS Outlet and represent high loss production due to mixing of the flow exiting and entering the PS Cavity as shown in Figure 3-9. Furthermore, due to mixing of the flow exiting the PS cavity with the flow from the upstream inlet, regions of high entropy generation are located near the PS cavity inner diameter (Regions B in Figure 3-17). The results based on the entropy generation in the inlet region imply that designing a PS Cavity that provides a unidirectional exit flow at the PS Outlet would result in decreasing the entropy generation corresponding to Region A.

![Figure 3-17: High entropy generation $\dot{S}_{gen}$ due to mixing of flow exiting/entering PS Cavity (Region A) and mixing of flow exiting PS Cavity with upstream flow (Region B) for PS compressor $\dot{m}/\dot{m}_{ref} = 0.3$ at $0.84 \, N_{ref}$](image-url)
3.5 Representation of the PS Flow through Actuations

Analysis of the flow field inside the PS compressor indicates that the PS acts as a sink of mass flow at the PS Slot and a source of mass flow, as well as axial and tangential momentum at the PS Outlet. One of the objectives of this thesis is to separate the PS flow actuations and assess their effect on the compressor performance individually. As described in Section 2.1, the proposed approach involves geometrically removing the PS Cavity from the PS compressor and modeling the PS flow through the use of boundary conditions at the PS Slot and Outlet. However, before executing this approach it is necessary to assess whether the performance of the geometrically coupled compressor is captured by representing the PS flow through boundary conditions.

The PS compressor used as a benchmark to assess the approach described includes the same inlet, wheel and diffuser as the geometry described in Section 2.3.2.1; however, the volute and the four structural ribs inside the PS cavity are removed. This is done as to remove the geometric asymmetry of the compressor. The use of a benchmark axisymmetric PS compressor is consistent with the use of a single passage compressor as a platform to investigate the effects of the PS flow actuations.

The procedure followed to assess the utility of the proposed representation of the PS flow is shown in Figure 3-18. OPs corresponding to operation near peak $\eta$ and surge for the axisymmetric PS compressor at two speedlines, 0.84 $N_{ref}$ and 1.16 $N_{ref}$, are computed. These speedlines correspond to two of the three speedlines assessed in Section 3.1. Flow quantities at the PS Outlet and Slot of the axisymmetric PS compressor are then averaged and used as boundary conditions in the geometrically uncoupled PS Actuation Model. The mass flow, mass flow averaged $\tilde{t}^M_{PS\text{Outlet}}$ and swirl angle $\tilde{a}^M_{PS\text{Outlet}} = \arctan\left(\frac{\bar{u}^M_{PS\text{Outlet}}}{\bar{w}_{axial}^M_{PS\text{Outlet}}}\right)$ of the flow are used at the PS Outlet, while $\dot{m}_{PS\text{Slot}}$ is prescribed at the PS Slot. The performance of the PS Actuation Model is then assessed against that of the benchmark axisymmetric PS compressor.
Figure 3-18: Procedure for assessing the representation of the PS flow through the application of boundary conditions

Figure 3-19 shows the computed compressor pressure ratio and efficiency for the axisymmetric PS compressor and the single passage PS Actuation Model at 1.16 $N_{ref}$. Comparison between the two shows that the compressor performance characteristics are captured through the use of the single passage PS Actuation Model. The discrepancy between the two CFD models is approximately 1.5 points in efficiency and less than 2% in terms of $\pi_c$. The impeller inlet flow angle $\beta_1$ and $T_e$ profiles for OPs near surge $\dot{m}/\dot{m}_{ref} = 0.6$, shown in Figure 3-20, indicate that both the flow entering the impeller and the temperature distribution between the two models are in close agreement.
Figure 3-19: PS Actuation Model predicts the compressor $\pi_c$ (left) and $\eta$ (right) of the axisymmetric PS compressor at 1.16 $N_{ref}$.

Figure 3-20: PS Actuation Model (red) captures the impeller LE relative flow angle (left) and $T_t/T_{t,inlet}$ (right) profiles of the axisymmetric PS compressor (blue) at $\dot{m}/\dot{m}_{ref} = 0.6$ at 1.16 $N_{ref}$.

Comparison of the compressor performance and the flow angle and temperature profiles at the blade LE indicates that the PS can be represented through a set of flow actuations at the PS Outlet and Slot. Furthermore, the agreement between geometrically coupled and uncoupled
models implies that the PS Actuation Model can be used as a platform to assess the effect of individual flow actuations on the compressor performance.

### 3.6 Summary

Comparison of the CFD generated flow field against both gas stand data and PIV measurements show agreement, thus implying adequacy in the computational method that has been employed.

The results indicate that the use of a PS reduces the impeller flow blockage by up to 25% compared to that of a non-ported compressor for OPs near surge. The decrease of flow blockage is a result of the partial removal of the recirculation zone present near the inducer shroud region which originates from the inducer tip leakage flow. The reduction of the extent of the recirculation zone results in a decrease of the entropy generated inside the impeller by up to 25% compared to a non-ported compressor.

Analysis of the flow field at the PS Outlet indicates that the flow exiting the PS Cavity is not unidirectional. The flow exits the PS Outlet in the form of four jets, while a portion of the flow exiting the PS Cavity is entrained back inside the cavity. It has also been shown that the flow removed through the PS Slot intercepts the structural struts located inside the PS cavity with a large incidence angle resulting in flow separation. As a result of the flow non-uniformity at the PS Outlet and the flow separation, the entropy generated inside the PS cavity accounts for approximately 25% of the total loss generated inside the compressor. The high velocity of the exiting flow at the PS Outlet is also responsible for the increased mixing losses with the main path flow which leads to an increase of entropy generation of up to 200% upstream of the wheel inlet compared to a non-ported case.

It has also been shown that the operation of the PS may be modeled through a set of flow actuations at the inlet and exit of the PS Cavity. Specifically, the PS acts as a sink of mass flow at the PS Slot, while at the PS Outlet, the PS flow acts as a source of mass, as well as axial and tangential momentum.
Chapter 4  Assessment of Mass Flow Removal at PS Slot

Analysis of the flow field inside the PS compressor, presented in Chapter 3, indicates that the action of the PS flow is perceived as a combination of actuations by the main flow, such as mass injection/removal as well as axial and angular momentum injection. In this chapter, the actuation of mass flow removal at the PS Slot is isolated from the flow actuations at the PS Outlet and its effect on the compressor performance is assessed. The chapter first describes the approach taken to model the effect of mass flow removal and continues with the computed compressor performance characteristics. The effect of mass flow removal on decreasing the flow blockage through the partial removal of the inducer shroud recirculation zone is quantified. By separating the static pressure rise into its contributing sources using a CV approach, the flow mechanisms which are responsible for the increase of the static pressure are assessed and quantified. The findings from this analysis indicate that decrease of the flow blockage and the diffusion of the primary stream are the two dominant factors in increasing the impeller static pressure rise. Mass flow removal is then shown to decrease the impeller entropy generation and increase the compressor stability.

4.1  Modeling - Approach

The CFD model used to assess the effect of mass flow removal on the compressor performance has been described in Section 2.3.2.2 and is shown in Figure 4-1. Mass flow removal is modeled through an outlet boundary condition at the PS Slot. The approach taken is to select a baseline inlet mass flow $\dot{m}_{in}$ at the compressor inlet and increase the fraction of mass flow removed at the PS Slot, $\varphi_{PS\ Slot} = \dot{m}_{PS\ Slot}/\dot{m}_{in}$. The $\dot{m}_{in}$ of the compressor stays constant while the $\dot{m}_{out}$ decreases with increasing values of $\varphi_{PS\ Slot}$. The baseline OPs selected for the three investigated
speedlines are listed in Table 4-1. The first OP at each speedline corresponds to the peak \( \eta \) OP for the non-ported compressor and the second OP corresponds to operation closer to surge. The ratio \( \dot{m}_{in}/(\dot{m}_{in})_{peak \eta} \) cross-references the inlet mass flow to that of the peak \( \eta \) OP at the same speedline. This ratio indicates how close each investigated OP is to the peak \( \eta \) operating conditions at that speed.

![Figure 4-1: Cross section of computational model used for assessment of mass flow removal at the PS Slot on the compressor performance with \( \dot{m}_{in} = \dot{m}_{out} + \dot{m}_{PS Slot} \)](image)

<table>
<thead>
<tr>
<th>Speedline</th>
<th>Description</th>
<th>( \dot{m}<em>{in}/\dot{m}</em>{ref} )</th>
<th>( \dot{m}<em>{in}/(\dot{m}</em>{in})_{peak \eta} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.84 ( N_{ref} )</td>
<td>Peak ( \eta )</td>
<td>( \dot{m}<em>{in}/\dot{m}</em>{ref} = 0.83 )</td>
<td>1</td>
</tr>
<tr>
<td>0.84 ( N_{ref} )</td>
<td>Near Surge</td>
<td>( \dot{m}<em>{in}/\dot{m}</em>{ref} = 0.5 )</td>
<td>0.6</td>
</tr>
<tr>
<td>1.16 ( N_{ref} )</td>
<td>Peak ( \eta )</td>
<td>( \dot{m}<em>{in}/\dot{m}</em>{ref} = 1.21 )</td>
<td>1</td>
</tr>
<tr>
<td>1.16 ( N_{ref} )</td>
<td>Near Surge</td>
<td>( \dot{m}<em>{in}/\dot{m}</em>{ref} = 0.85 )</td>
<td>0.7</td>
</tr>
<tr>
<td>1.48 ( N_{ref} )</td>
<td>Peak ( \eta )</td>
<td>( \dot{m}<em>{in}/\dot{m}</em>{ref} = 1.6 )</td>
<td>1</td>
</tr>
</tbody>
</table>

Table 4-1: Baseline OPs used for assessment of mass flow removal at PS Slot on compressor performance
The region upstream of the PS Slot (inducer) works at a higher mass flow compared to the exducer region. To assess the effect of mass flow removal consistently, the inducer and exducer regions of the impeller are treated separately when compared against non-ported cases. For instance, a case of mass removal with an inlet mass flow of $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ and a fraction of removed mass flow of $\varphi_{PS\text{Slot}} = 0.3$, will be compared against a non-ported OP with $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ at the inducer and a non-ported OP with $\dot{m}_{in}/\dot{m}_{ref} = 0.35$ at the exducer.

### 4.2 Effect of Mass Flow Removal on Compressor Performance

The effect of increasing the fraction of mass removed at the PS Slot on the compressor pressure ratio is shown for $0.84\ N_{ref}$ in Figure 4-2. The data is plotted on a $\dot{m}_{out}$ basis, since the stagnation pressure at the exit of the compressor is of importance. The blue data points correspond to single passage non-ported OPs. The red data points are OPs with mass flow removed at the PS Slot for which the baseline $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ (red circle in Figure 4-2), corresponds to peak $\eta$ operating conditions for the non-ported compressor. Up to 50% of the $\dot{m}_{in}$ is removed at increments of 10% (i.e. $\Delta \dot{m}_{PS\text{Slot}}/\dot{m}_{in}=0.1$). The effect of mass removal is also assessed using a baseline $\dot{m}_{in}$ corresponding to operating conditions of the non-ported compressor closer to surge $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ (green circle in Figure 4-2); again up to 50% of the inlet flow is removed at increments of 10%. These data points are shown in green in Figure 4-2. The same approach is taken for the other two speedlines shown in Table 4-1. The results indicate that the removal of mass flow through the PS Slot increases the compressor $\pi_c$. The flow mechanisms responsible for this increase are identified and analyzed in subsequent sections of this chapter.
Figure 4-2: Mass flow removal at PS Slot results in the increase of the compressor $\pi_c$ compared to non-ported compressor OPs of equal $\dot{m}_{out}$ at $0.84 \, N_{ref}$; red circle indicates peak $\eta$ baseline OP, green circle indicates near surge baseline OP.

The compressor efficiency is calculated through the use of entropy and stagnation enthalpy flux taking into account both PS and main flow streams (see T-s diagram in Figure 4-3). Two scenarios are considered for calculating the compressor efficiency when removing mass flow at the PS Slot. The first scenario involves reusing the removed PS flow, so that the work done on the PS stream is not wasted. This approach gives an upper bound of efficiency and is calculated through:

$$\eta = 1 - \frac{P_{loss,Main\ Stream} + P_{loss,PS\ Stream}}{P_{compressor}}$$  \hspace{1cm} \text{Equation 4-1}$$

where

$$P_{compressor} = \int h_{t,\text{out}} \dot{m}_{out} - \int h_{t,\text{PS\ Slot}} \dot{m}_{PS\ Slot} - \int h_{t,\text{in}} \dot{m}_{in}$$  \hspace{1cm} \text{Equation 4-2}$$
The second approach involves discarding the removed PS flow and all the work that the compressor has done on the flow, thus giving a lower bound of efficiency. Equation 4-1 is also used for the lower efficiency bound but with the use of the following expression for the lost work of the PS stream

\[ P_{loss,PS Stream} = \int h_{t,PS Slot} \, \dot{m}_{PS Slot} \, dt - \int h_{t,in} \, \dot{m}_{PS Slot} \]  

Equation 4-5
The effect of mass removal on the compressor efficiency is shown for 0.84 $N_{ref}$ in Figure 4-4. The blue points correspond to non-ported OPs. The red and grey points correspond to mass flow removal OPs corresponding to inlet mass flow of $\dot{m}_{in}/\dot{m}_{ref} = 0.83$. The green and yellow points correspond to mass flow removal OPs with $\dot{m}_{in}/\dot{m}_{ref} = 0.5$. The results indicate that in the case when the PS stream is fully utilized (e.g. expanded through a turbine) the effect of removing mass flow increases the compressor efficiency compared to non-ported OPs working at same $\dot{m}_{out}$ by up to 10% for mass flow removal fractions of $\varphi_{PS\ slot} = 0.5$. However, as stated previously this is the upper bound value for efficiency. In the case where the PS stream is discarded the efficiency is found to decrease by up to 12% compared to non-ported OPs of same $\dot{m}_{out}$.

The results shown so far have shown the effect of mass removal on OPs at the low speedline 0.84 $N_{ref}$. The results for the compressor pressure ratio and efficiency at the 1.16 $N_{ref}$ speedline...
are shown in Figure 4-5. The trends seen for mass flow removal at 0.84 $N_{ref}$ are also found for the OPs at the two other investigated speedlines. This implies that the mechanisms responsible for the differences in the compressor performance are the same for all investigated speedlines.

![Figure 4-5: Effect of mass flow removal at PS Slot on compressor $\pi_c$ (left) and $\eta$ (left) at 1.16 $N_{ref}$](image)

**4.3 Decrease of Flow Blockage Through Mass Flow Removal**

One of the hypotheses of this thesis stipulates that the PS works in such a way as to decrease the impeller blockage generation. The effect of removing a fraction of the inducer mass flow through the PS Slot on the impeller flow blockage is quantified. This is done with respect to an OP corresponding to a non-ported compressor working at conditions near peak $\eta$ at $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ at 0.84$N_{ref}$. Two mass flow removal cases are assessed for which the inducer mass flow is equal to $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ and the removed mass flow fraction is $\phi_{PS, SLOT} = 0.3$ and 0.5. The mass flow removal OPs are compared against the baseline non-ported compressor of $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ for inducer crossflow planes. Inside the exducer, the mass flow removal OPs are compared against non-ported compressor OPs of equal exducer mass flow respectively. The
Table 4-2: Non-ported and mass flow removal OPs used to assess influence of mass flow removal on blockage generation at near peak $\eta$ OP at $0.84N_{ref}$; color coding refers to Figure 4-6.

<table>
<thead>
<tr>
<th>Description</th>
<th>$\dot{m}<em>{in}/\dot{m}</em>{ref}$</th>
<th>$\dot{m}<em>{in}/(\dot{m}</em>{in})_{peak\eta}$</th>
<th>$\dot{m}<em>{out}/\dot{m}</em>{ref}$</th>
<th>$\varphi_{PS\ Slot}$</th>
<th>Color Code</th>
</tr>
</thead>
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<tr>
<td>Non-Ported</td>
<td>0.83</td>
<td>1</td>
<td>0.83</td>
<td>0</td>
<td>Solid Black</td>
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<tr>
<td>Non-Ported</td>
<td>0.58</td>
<td>0.7</td>
<td>0.58</td>
<td>0</td>
<td>Solid Purple</td>
</tr>
<tr>
<td>Non-Ported</td>
<td>0.41</td>
<td>0.5</td>
<td>0.41</td>
<td>0</td>
<td>Solid Blue</td>
</tr>
<tr>
<td>Mass Removal</td>
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<td>0.58</td>
<td>0.3</td>
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</tr>
<tr>
<td>Mass Removal</td>
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<td>1</td>
<td>0.41</td>
<td>0.5</td>
<td>Dashed Blue</td>
</tr>
</tbody>
</table>

The flow blockage, expressed as a fraction of the geometric area of the crossflow plane, is calculated through the method proposed by Khalid [29, 30] and shown in Figure 4-6. With respect to the non-ported compressor OPs, the results indicate that there is a buildup of blockage inside the inducer region, which then decreases inside the exducer. Even at the peak $\eta$ OP the calculated blockage upstream of the PS Slot exceeds 15% of the geometric area. This is consistent with the full wheel non-ported compressor numerical calculations shown in Section 3.3 that show a strong recirculation zone near the inducer shroud. The removal of a fraction of the inducer mass flow through the PS Slot results in the overall reduction in blockage throughout the entire impeller. The flow blockage upstream of the PS Slot for the mass flow removal cases is reduced to less than 5% of the geometric area upon removing 30% of the inducer flow. Increasing the fraction of mass flow removed to 50% has negligible effect on further decreasing the flow blockage upstream of the PS Slot, indicating that the low momentum fluid near the inducer shroud has almost entirely been removed. The removal of mass flow results in lower flow blockage downstream of the PS Slot also. The largest differences correspond to crossflow planes just downstream of the PS Slot where the calculated decrease of flow blockage is up to 25%. However, the results indicate that there is a build-up of blockage in the exducer region for
the mass removal OPs which approaches the blockage values corresponding to the non-ported compressor OPs of same $\dot{m}_{out}$.

Figure 4-6: Mass flow removal results in the decrease of the impeller blockage between mass flow removal OPs of $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ with $\varphi_{PS\, Slot} = 0.3$ and $\varphi_{PS\, Slot} = 0.5$ and non-ported compressor OPs at $0.84\, N_{ref}$

<table>
<thead>
<tr>
<th>Description</th>
<th>$\dot{m}<em>{in}/\dot{m}</em>{ref}$</th>
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<th>$\dot{m}<em>{out}/\dot{m}</em>{ref}$</th>
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</tr>
</thead>
<tbody>
<tr>
<td>Non-Ported</td>
<td>0.5</td>
<td>0.6</td>
<td>0.5</td>
<td>0</td>
<td>Solid Blue</td>
</tr>
<tr>
<td>Non-Ported</td>
<td>0.35</td>
<td>0.42</td>
<td>0.35</td>
<td>0</td>
<td>Solid Red</td>
</tr>
<tr>
<td>Non-Ported</td>
<td>0.25</td>
<td>0.3</td>
<td>0.25</td>
<td>0</td>
<td>Solid Red</td>
</tr>
<tr>
<td>Mass Removal</td>
<td>0.5</td>
<td>0.6</td>
<td>0.35</td>
<td>0.3</td>
<td>Dashed Green</td>
</tr>
<tr>
<td>Mass Removal</td>
<td>0.5</td>
<td>0.6</td>
<td>0.25</td>
<td>0.5</td>
<td>Dashed Green</td>
</tr>
</tbody>
</table>

Table 4-3: Non-ported and mass flow removal OPs used to assess influence of mass flow removal on blockage generation at near surge OP at $0.84\, N_{ref}$; color coding refers to Figure 4-7
Assessment of the effect of mass flow removal is also conducted for a case where the baseline OP is closer to surge with $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ at $0.84 \ N_{ref}$ and removing 30% and 50% of the inducer mass flow. Table 4-3 contains the description of the examined OPs used. The color coding in Table 4-3 refers to Figure 4-7 which shows the impeller blockage. Comparison of the non-ported compressor OPs indicates that increasing the inducer mass flow results in the decrease of the inducer flow blockage. Specifically, the inducer flow blockage decreases by approximately 8% between non-ported $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ (solid blue) and $\dot{m}_{in}/\dot{m}_{ref} = 0.25$ (solid green). Removing a fraction of the inducer mass flow results in a further decrease of the inducer flow blockage compared to a non-ported compressor OP of equal inducer mass flow. Removing 50% of the inducer mass flow (dashed green) further decreases the inducer flow blockage by up to 10%; however, it cannot remove all the low momentum fluid near the inducer shroud at this OP. Comparison between solid blue, dashed red and dashed green curves implies that continuously increasing the fraction of removed mass flow leads to a further decrease of the flow blockage in the inducer region.

![Figure 4-7: Mass flow removal results in the decrease of the impeller blockage between mass flow removal OPs of $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ with $\varphi_{PS\ Slot} = 0.3$ and $\varphi_{PS\ Slot} = 0.5$ and non-ported compressor OPs at 0.84 $N_{ref}$](image-url)
Figure 4-8: Differences in blockage across the entire crossflow plane are traced to the top 20% of the span inside the inducer between non-ported $\dot{m}_{\text{in}}/\dot{m}_{\text{ref}} = 0.5$ and mass flow removal $\dot{m}_{\text{in}}/\dot{m}_{\text{ref}} = 0.5$ with $\phi_{PS \text{Slot}} = 0.5$ (left) and the exducer between non-ported $\dot{m}_{\text{in}}/\dot{m}_{\text{ref}} = 0.25$ and mass flow removal $\dot{m}_{\text{in}}/\dot{m}_{\text{ref}} = 0.5$ with $\phi_{PS \text{Slot}} = 0.5$ (right) at $0.84 N_{\text{ref}}$.

To understand the mechanism through which the impeller flow blockage is reduced, the difference in blockage, $\Delta \text{Blockage}$, between non-ported compressor and mass flow removal cases is assessed. The left figure of Figure 4-8 shows the decrease in inducer blockage between the non-ported OP corresponding to the solid blue line (see Table 4-3 and Figure 4-7) and the mass flow removal OP with the dashed green line. These two compressor configurations have equal $\dot{m}_{\text{in}}$. The figure on the right shows the decrease of blockage in the exducer between non-ported OP corresponding to the solid green line and the mass flow removal case with the dashed green line. The black lines in Figure 4-8 correspond to the decrease in blockage calculated across the entire span of the crossflow plane, while the purple curves show the blockage decrease for span values above 80%. The results imply that more than 80% of the flow blockage reduction across the passage is due to the decrease in the top 20% of the span. The high flow blockage present inside the inducer is due to the recirculation zone near the shroud, which originates from the inducer tip leakage flow. This is shown in Figure 4-9, where the streamlines originating from the tip region of the blade upstream of the PS Slot are indicated in black. The removal of all or a
portion of the inducer tip leakage flow is responsible for the decrease of the inducer flow blockage. The removal of the inducer tip leakage flow before entering the exducer region, as shown in Figure 4-9, also results in the decrease of the exducer flow blockage.

![Figure 4-9: Partial removal of flow recirculation originating at inducer tip region upstream of PS Slot through mass flow removal; non-ported $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ (left) and mass flow removal $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ with $\varphi_{PS\ Slot} = 0.5$ (right) at $0.84\ N_{ref}$](image)

4.4 **Effect of Mass Flow Removal on Static Pressure Rise**

In Section 4.3 it is shown that the removal of a fraction of the inducer mass flow decreases the flow blockage throughout the entire impeller. As a consequence of the blockage reduction it is expected that the pressure rise capability of the impeller be enhanced.
Figure 4-10: Mass flow removal results in increase of impeller $C_p$ rise between mass flow removal OPs of $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ with $\varphi_{PS_{slot}} = 0.5$ and non-ported compressor OPs at $0.84 N_{ref}$; the inducer $\Delta C_p$ increase between mass flow removal and non-ported OP is more than 100%.

The impeller static pressure rise coefficient $C_p = \frac{P_{wheel_{inlet}}}{\frac{1}{2} \rho_{inlet} U_{tip}^2}$ corresponding to OPs of mass flow removal with an inlet $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ (with reference to the inducer mass flow for the OP near peak $\eta = \dot{m}_{in}/(\dot{m}_{in})_{peak} = 0.6$) and $\varphi_{PS_{slot}} = 0.5$ and two non-ported compressor OPs is shown in Figure 4-10. The dashed green curve represents the mass flow removal OP, the blue curve is the non-ported with $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ ($\dot{m}_{in}/(\dot{m}_{in})_{peak} = 0.6$), while the green curve is a non-ported OP with $\dot{m}_{in}/\dot{m}_{ref} = 0.25$ ($\dot{m}_{in}/(\dot{m}_{in})_{peak} = 0.3$). The OP corresponding to the dashed green line has the same $\dot{m}_{in}$ with the blue solid curve and the same $\dot{m}_{out}$ with the solid green curve. Figure 4-10 indicates that increasing the inducer mass flow from $\dot{m}_{in}/\dot{m}_{ref} = 0.25$ (solid green) to $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ (solid blue) has no influence on the inducer static pressure rise at these OPs near surge. This implies that the inducer is near stall. In the context of PS compressor operation, it can be inferred that at near surge OPs the fact that the inducer works at a higher mass flow than the exducer does not contribute to the increase of the impeller static.
pressure rise. Comparison between the dashed green and solid blue curves indicates that the effect of removing mass flow increases the static pressure rise at the inducer exit by more than 100%. This level of increase is also computed when removing equal mass flow fractions at higher \( \dot{m}_{\text{in}}/\dot{m}_{\text{ref}} \) OPs and at higher speedlines. With respect to the \( C_p \) rise in the exducer, comparison of the mass flow removal case (dashed green) with the non-ported compressor of same \( \dot{m}_{\text{out}} \) (solid green) yields an increase of approximately 12%. These results indicate that the increase of static pressure in the inducer is primarily responsible for the larger difference of static pressure between mass flow removal and non-ported compressor OPs at the impeller outlet.

However, the exact flow mechanisms which are responsible for the static pressure increase have not yet been identified nor has their effect been quantified. In the following sections, a method which allows for the separation of the static pressure rise into individual contributing effects is described and applied to delineate the effect of mass flow removal.

### 4.4.1. Separation of Impeller Static Pressure Rise

Let us consider the control volume shown in Figure 4-11 which consists of the flow passage between blade pressure side and suction side along an incremental meridional distance of \( dx \). The radial direction coincides with the \( z \) axis. The geometric and effective flow area at the CV inlet are \( A \) and \( A_{\text{eff}} \) respectively. An effective velocity in the meridional direction expressed through Equation 4-6 is defined which takes into account that the flow does not occupy the geometric passage area \( A \) but rather the effective area \( A_{\text{eff}} \)

\[
\text{Equation 4-6} \quad u_{\text{eff}} = \frac{\dot{m}}{\rho A_{\text{eff}}}
\]

Flow enters on the left side of the CV with \( \dot{m}, u_{\text{eff}}, \rho, p \), while the flow and geometric quantities associated with the CV outlet are \( \dot{m} + d\dot{m}, u_{\text{eff}} + du_{\text{eff}}, \rho + d\rho, p + dp \) and \( A + dA, A_{\text{eff}} + dA_{\text{eff}} \). A portion of the incoming flow \( d\dot{m} \) is removed through an opening located on the shroud of the CV. Finally, the effect of the blade loading is taken into account by assigning \( P_{ps} \) and \( P_{ss} \)
on the sides of the CV. The $P_{PS}$ and $P_{SS}$ values are taken to be constant along the meridional distance $dx$ of the CV.

Conservation of mass is applied on the CV taking into account that the flow does not occupy the geometric passage area $A$ but rather the effective flow area $A_{eff} = A - A_b$:

$$
\rho u_{eff} A_{eff} = d\dot{m} + (\rho + d\rho)(u_{eff} + du_{eff})(A_{eff} + dA_{eff})
$$  \hspace{1cm} \text{Equation 4-7}

Conservation of momentum along the meridional direction, assuming inviscid flow and that the pressure acts on the geometric area $A$ is written as:
\[-u_{\text{eff}} \dot{m} + (u_{\text{eff}} + du_{\text{eff}})(\dot{m} + d\dot{m}) + u_{\text{inj}} d\dot{m}\]
\[= pA - (p + dp)(A + dA) + (p_{PS} - p_{SS}) A_{\text{blade}} \sin \beta \]

Equation 4-8

where \(u_{\text{inj}}\) is the velocity component of the flow removed in the meridional direction.

Combining Equation 4-7 and Equation 4-8, keeping only the first order terms and including the effect of the centrifugal force in increasing the pressure along the meridional direction, where \(\gamma\) is the angle between the radial and meridional direction gives:

\[
dp = \]
\[
(-2 \frac{u_{\text{eff}}}{A} - \frac{u_{\text{inj}}}{A}) d\dot{m} + \frac{u_{\text{eff}}^{2} A_{\text{eff}}}{A} d\rho + \frac{\rho u_{\text{eff}}^{2}}{A} dA_{\text{eff}} + \rho \Omega^{2} r \cos \gamma dr \]
\[
+ \frac{(p_{PS} - p_{SS}) A_{\text{blade}} \sin \beta}{A} \]

Equation 4-9

The last term in Equation 4-9 is the blade loading since it is the pressure difference between pressure and suction side of the blade \(\Delta p_{\text{blade}} = (p_{PS} - p_{SS})\). Finally, non-dimensionalizing using \(q_{\text{ref}} = \frac{1}{2} \rho_{\text{inlet}} U_{\text{tip}}^{2}\) the following expression for the impeller \(Cp\) is obtained:

\[
Cp = \frac{\partial Cp}{\partial \dot{m}} d\dot{m} + \frac{\partial Cp}{\partial \rho} d\rho + \frac{\partial Cp}{\partial A_{\text{eff}}} dA_{\text{eff}} + \frac{\partial Cp}{\partial r} dr + \frac{\Delta p_{\text{blade}}}{A_{\text{ref}} A_{\text{blade}} \sin \beta} A_{\text{blade}} \sin \beta \]

Equation 4-10

where

\[
\frac{\partial Cp}{\partial \dot{m}} = - \frac{u_{\text{eff}}}{A_{\text{ref}}} \left(2 + \frac{u_{\text{inj}}}{u_{\text{eff}}} \right) \]

Equation 4-11

\[
\frac{\partial Cp}{\partial \rho} = \frac{u_{\text{eff}}^{2} A_{\text{eff}}}{A_{\text{ref}}} \]

Equation 4-12

\[
\frac{\partial Cp}{\partial A_{\text{eff}}} = \frac{\rho u_{\text{eff}}^{2}}{A_{\text{ref}}} \]

Equation 4-13
Equation 4-10 can be used to assess how the effect of mass flow removal at the PS slot, changes in density, $A_{eff}$ (and therefore blockage), blade loading and radius influence the $Cp$ of the impeller.

The first term in Equation 4-10 $\frac{\partial Cp}{\partial m} d\dot{m}$, quantifies the effect of removing a fraction of the inlet flow through the PS Slot on the static pressure. The additional static pressure rise from this mechanism is due to the change in flow area of the main core stream (that exits the domain outlet) from upstream to downstream of the PS Slot location, thereby experiencing a larger level of diffusion. This mechanism is primarily one-dimensional in nature and can be rationalized through the help of Figure 4-12. Assuming incompressible and inviscid flow, the static pressure increase between Stations 1 and 2 can be calculated by applying Bernoulli equation for a streamtube of the main flow $\dot{m}_a$ from Station 1 to 2:

$$\frac{p_2 - p_1}{1/2 \rho u_1^2} = 1 - \left(\frac{A_{1a}}{A_2}\right)^2$$

Equation 4-15
The third term in Equation 4-10 $\frac{\partial cp}{\partial A_{eff}} dA_{eff}$ quantifies how the static pressure is affected by changes in the $A_{eff} = A - A_b$ through the compressor flow path, while $\frac{\Delta p_{blade}}{\Delta q_{ref}} A_{blade} \sin \beta$ represents the influence of the blade loading.

The use of this sensitivity analysis based approach for estimating the $Cp$ through the sum of its individual components is assessed against the CFD generated values in the context of two examples in Appendix B. The first example is for a case where mass flow is removed from a nonuniform flow through a straight duct, while the second assessment is against the CFD generated $Cp$ values for the case of mass flow removal at the PS Slot. Based on the results shown in Appendix B, it can be inferred that the use of this approach provides an adequate estimation of the static pressure rise trends and values as reflected in the CFD solutions. Thus, using this approach to analyze the contributions of individual flow mechanisms to the static pressure rise through the impeller is justifiable.

### 4.4.2. Contribution of Flow Mechanisms to Impeller Static Pressure Rise

The separation of the impeller $Cp$ rise into its individual components is examined in this section. In accordance with the analysis presented in Section 4.4.1, the contributions of the following parameters are quantified:

- Removal of a fraction of the inducer mass flow through the PS Slot. This can be thought of as the diffusion of the main core stream due the change of its area between locations upstream and downstream of the PS Slot
- Blade loading
- Compressibility
- Effective flow area $A_{eff} = A - A_b$, which takes into account changes of both geometric area $A$ and blockage $A_b$
- Changes in radius through the centrifugal force
The performance of inducer and exducer regions is conducted separately and compared against non-ported compressor OPs of the same $\dot{m}_{in}$ and $\dot{m}_{out}$ respectively. Figure 4-13 shows the separation of the inducer $Cp$ when removing mass flow with respect to a non-ported compressor OP near peak $\eta$. The non-ported compressor corresponds to $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ ($\dot{m}_{in}/(\dot{m}_{in})_{peak \eta}=1$) while 30% and 50% of the inlet flow is removed for the two mass flow removal cases. The results show that the two most important mechanisms in setting the inducer $Cp$ are the diffusion of the main core stream and that of the blade loading.

![Graph showing the breakdown of inducer $Cp$ rise indicating blade loading dominant mechanism for non-ported compressor OP $\dot{m}_{in}/\dot{m}_{ref} = 0.83$; blade loading and diffusion of primary stream dominant mechanisms for mass flow removal OPs $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ with $\varphi_{PS\text{Slot}} = 0.3$ and $\varphi_{PS\text{Slot}} = 0.5$ at $0.84 N_{ref}$](image)

The exducer $Cp$ values are shown in Figure 4-14. The left chart corresponds to a non-ported compressor OP of $\dot{m}_{in}/\dot{m}_{ref} = 0.58$ ($\dot{m}_{in}/(\dot{m}_{in})_{peak \eta}=0.7$) and a mass flow removal case of $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ with 30% mass removal. The right chart shows the separation of the exducer $Cp$ for the non-ported compressor OP $\dot{m}_{in}/\dot{m}_{ref} = 0.41$ ($\dot{m}_{in}/(\dot{m}_{in})_{peak \eta}=0.5$) and a mass flow removal OP of $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ with 50% mass removal. The centrifugal force is the
dominant mechanism in increasing the exducer pressure rise, while blade loading and changes of the effective area are of secondary importance. It has been shown that changes in $A_{eff}$ result in the decrease of the static pressure inside both inducer and exducer regions. This is to be expected, since both $A$ and $A_{eff}$ continuously decrease throughout the entire impeller passage for this centrifugal compressor design.

![Figure 4-14: Breakdown of exducer $C_P$ rise indicating blade loading and centrifugal force dominant mechanisms; non-ported compressor and mass flow removal $\dot{m}_{in}/\dot{m}_{ref} = 0.58$ with $\varphi_{PS,slot} = 0.3$ (left) and non-ported $\dot{m}_{in}/\dot{m}_{ref} = 0.41$ and mass flow removal $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ with $\varphi_{PS,slot} = 0.5$ (right) at 0.84 $N_{ref}$](image)

The contribution of each flow mechanism to the impeller $C_P$ increase has been shown in Figure 4-13 (inducer) and Figure 4-14 (exducer). However, to assess the effect of mass flow removal on the compressor pressure rise capability, the contribution of these flow mechanisms to the difference of $C_P$ between non-ported and mass flow removal OPs, $\Delta C_P = C_{Pmass flow removal} - C_{Pnon-ported}$, should be quantified. As discussed in Section 4.4, the difference of static pressure increase inside the inducer, between mass flow removal OPs and non-ported compressor OPs, accounts for more than 70% of the static pressure difference at the impeller outlet. Therefore, the region of primary interest for this analysis is the inducer. Figure 4-15 is a breakdown of the inducer $\Delta C_P$ between mass flow removal OPs and the baseline non-ported compressor at $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ ($\dot{m}_{in}/(\dot{m}_{in})_{peak \eta}=1$). The chart on the left in Figure 4-15 compares the non-
ported compressor against the mass flow removal OP with 30% of the flow removed and the chart on the right with 50% removed.

The results show that approximately 70% of the inducer ΔCp between the non-ported compressor and that of ϕ_{PSSlot} = 0.3 mass flow removal is due to the diffusion of the main stream flow. When comparing the non-ported case against the mass flow removal of ϕ_{PSSlot} = 0.5, the contribution of the main flow diffusion increases to 77%. The difference in blade loading between mass removal and non-ported baseline case accounts for approximately 17% of the inducer ΔCp, while the increase of the effective flow area accounts for approximately 6%.

Mass flow removal \( \hat{m}_{in}/\hat{m}_{ref} = 0.83 \) with \( \varphi_{PSSlot} = 0.3 \) vs. non-ported compressor OP \( \hat{m}_{in}/\hat{m}_{ref} = 0.83 \)

Mass flow removal \( \hat{m}_{in}/\hat{m}_{ref} = 0.83 \) with \( \varphi_{PSSlot} = 0.5 \) vs. non-ported compressor OP \( \hat{m}_{in}/\hat{m}_{ref} = 0.83 \)

Figure 4-15: Breakdown of inducer ΔCp rise between mass flow removal OPS and non-ported compressor OP at peak η \( \hat{m}_{in}/\hat{m}_{ref} = 0.83 \) indicates diffusion of primary stream dominant mechanism for ΔCp

The baseline non-ported compressor OP that is shown in Figure 4-15 corresponds to the peak η OP. The results of a similar investigation with a baseline non-ported compressor OP near surge \( \hat{m}_{in}/\hat{m}_{ref} = 0.5 \) \( \hat{m}_{in}/(\hat{m}_{in})_{peak η=0.6} \) are shown in Figure 4-16. The breakdown of the inducer ΔCp between the non-ported baseline compressor and that of a mass flow removal case with \( \hat{m}_{in}/\hat{m}_{ref} = 0.5 \) and \( \varphi_{PSSlot} = 0.3 \) is shown in the chart on the left of Figure 4-16 and with that of \( \varphi_{PSSlot} = 0.5 \) in the right chart of Figure 4-16. When removing 50% of the inducer
flow, the diffusion of the core flow accounts for approximately 40% of the inducer static pressure rise, the increase of the blade loading accounts for 50% and the increase of the $A_{eff}$ for the remaining 10%.

Mass flow removal $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ with $\varphi_{PS \ slot} = 0.3$

Mass flow removal $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ with $\varphi_{PS \ slot} = 0.5$

Figure 4-16: Breakdown of inducer $\Delta C_p$ rise between mass flow removal OPs and non-ported compressor OP at near surge $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ indicates blade loading and diffusion of primary stream dominant mechanisms for $\Delta C_p$

The results regarding the breakdown of the inducer $\Delta C_p$ into its components show that the blade loading increases through the removal of mass flow at the PS Slot. Specifically, Figure 4-15 contains the dissection of the inducer $\Delta C_p$ when removing mass flow with respect to the peak $\eta$ non-ported compressor. The increase in $C_p$ due to blade loading is calculated to be $\Delta C_{P\ blade} = 0.032$ when removing 50% of the inducer flow. In the case when 50% of the inducer flow is removed with respect to a non-ported OP closer to surge ($\dot{m}_{in}/(\dot{m}_{in})_{peak \ \eta=0.6}$), the increase due to blade loading is $\Delta C_{P\ blade} = 0.08$ (see Figure 4-16). Thus, it can be inferred that mass flow removal is more effective in increasing the blade loading at OPs closer to surge. This is traced to the larger increase of the blade relative inlet flow angle between mass flow removal and non-ported compressor when operating closer to surge. The relative inlet flow angle $\beta_1$ near the blade LE is shown in Figure 4-17 for mass flow removal OPs with respect to baseline non-ported compressor at peak $\eta$ (left chart) and a non-ported compressor $\dot{m}_{in}/(\dot{m}_{in})_{peak \ \eta=0.6}$ (right
The increase of blade loading for the mass flow removal OPs is not only due to the increase of the blade relative inlet flow angle but also due to the decrease of flow blockage near the inducer shroud which results in a larger turning of the flow. Figure 4-18 shows the angle by which the flow is turned with respect to the relative flow angle at the blade LE for non-ported compressor.
and mass flow removal OPs with $\dot{m}/\dot{m}_{ref} = 0.83$. Regions of higher flow turning correspond to regions where the flow blockage has been decreased as shown in Figure 4-19.

![Figure 4-18](image)

**Figure 4-18:** Mass flow removal increases the inducer flow turning $\Delta \beta$ at span > 80%; $\Delta \beta$ at crossflow plane upstream of PS Slot with respect to inlet flow angle at blade LE for non-ported compressor OP at $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ (left) and mass removal OP at $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ with $\psi_{PS\text{Slot}} = 0.5$ at 0.84 $N_{ref}$

![Figure 4-19](image)

**Figure 4-19:** Regions of high flow turning correspond to regions where mass flow removal decreases flow blockage; flow blockage at crossflow plane upstream of PS Slot for non-ported compressor OP at $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ (left) and mass removal OP at $\dot{m}_{in}/\dot{m}_{ref} = 0.83$ with $\psi_{PS\text{Slot}} = 0.5$ at 0.84 $N_{ref}$ and span > 80%
In conclusion, the results in this section show that through the increase of the inducer effective flow area, the actuation of mass flow removal is directly responsible for 10% of the calculated increase in the inducer $C_p$ rise between mass flow removal OPs and non-ported compressors. By increasing the flow turning, which results in the increase of blade loading, the decrease of the flow blockage is shown to account for up to 50% of the static pressure rise for OPs near surge. Therefore, the decrease of the flow blockage accounts for up to 60% of the calculated inducer $C_p$ increase for OPs near surge. It has been shown that the contribution of the blade loading in the increase of the inducer $\Delta C_p$ between ported compressor OP and mass flow removal OPs is larger at OPs near surge. This is due to the larger available potential through the decrease of blockage. As shown in Section 4.3, with respect to a baseline non-ported OP at peak $\eta$, increasing the fraction of removed mass flow from 30% to 50% has negligible effect on further reducing the flow blockage (see Figure 4-6). Therefore, the potential available for increasing the blade loading is small. For OPs near surge however, due to the higher values of flow blockage for the baseline non-ported case (> 30%), increasing the fraction of removed flow results in a continuous decrease of the inducer blockage (see Figure 4-7).

### 4.5 Effect of Mass Flow Removal on Entropy Generation

The effect of mass flow removal has so far been assessed in light of the associated decrease in flow blockage and increase in static pressure rise. However, as shown in Chapter 3 the use of a ported shroud results in the decrease of the entropy generated inside the impeller. Therefore, the effect of removing mass flow in terms of the entropy generation is assessed in this section.
Figure 4-20: Mass flow removal decreases entropy generation both upstream of wheel inlet (left) and inside impeller (right) due to partial removal of inducer shroud flow recirculation; entropy flux for mass flow removal OP at $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ with $\varphi_{PS\,Slot} = 0.5$ (dashed green), non-ported compressor OPs $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ (solid blue) and $\dot{m}_{in}/\dot{m}_{ref} = 0.25$ (solid green) at $0.84 \, N_{ref}$.

Figure 4-20 shows the entropy generated upstream of the wheel inlet (left chart) and inside the impeller (right chart) based on the CFD calculations using the single passage PS Flow Actuation model. The solid blue and green curves represent non-ported compressor OPs with $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ ($\dot{m}_{in}/(\dot{m}_{in})_{peak\,\eta}=0.6$ with respect to the peak $\eta$ OP at this speedline) and $\dot{m}_{in}/\dot{m}_{ref} = 0.25$ ($\dot{m}_{in}/(\dot{m}_{in})_{peak\,\eta}=0.3$) respectively. The dashed green curve corresponds to a mass flow removal case of $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ with $\varphi_{PS\,Slot} = 0.5$. Hence, the dashed green curve has the same $\dot{m}_{in}$ with the solid blue curve and the same $\dot{m}_{out}$ with the solid green curve. Comparison between non-ported compressor $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ (solid blue) and $\dot{m}_{in}/\dot{m}_{ref} = 0.25$ (solid green) OPs shows that increasing the inducer mass flow results in the decrease of the entropy generated upstream of the PS Slot. Specifically, the decrease of the entropy generated upstream of the wheel inlet and inside the inducer corresponds to approximately 7 and 3.5 points of efficiency respectively. These results indicate that in the case of a recirculating PS compressor, the fact that the inducer works at a higher mass flow results in the increase of the compressor efficiency compared to a non-ported compressor OP of equal outlet mass flow. The entropy generated upstream of the wheel inlet for the solid blue curve is 2.5 times higher than that of the mass flow removal OP (dashed green curve), which corresponds to approximately 1.5 points.
lower overall efficiency. Upstream of the PS Slot, the results indicate a decrease of the entropy generated by up to 20%; which corresponds to an increase of the overall efficiency of approximately one point. The decrease in loss generation is due to the partial removal of the recirculation zone associated with the inducer tip leakage flow. The reduction of loss generation is traced to regions near the inducer shroud for span values greater than 70%. This is shown in Figure 4-21, where $\dot{S}_{\text{visc}}$ of a crossflow plane upstream of the PS Slot for the non-ported compressor OP (left figure) and mass flow removal OP (right figure) is plotted.

![Figure 4-21](image)

Figure 4-21: Local entropy generation rate $\dot{S}_{\text{visc}}$ on flow plane upstream of PS Slot indicates largest decrease due to mass flow removal calculated for span > 70%; non-ported compressor OP at $\dot{m}_{\text{in}}/\dot{m}_{\text{ref}} = 0.5$ (left) and mass flow removal case at $\dot{m}_{\text{in}}/\dot{m}_{\text{ref}} = 0.5$ with $\varphi_{PS\text{ slot}} = 0.5$ (right) at 0.84 $N_{\text{ref}}$. 

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Figure 4-22: Local entropy generation rate $\dot{s}_{visc}$ on flow plane downstream of PS Slot indicates largest decrease due to mass flow removal calculated for span > 70%; non-ported compressor OP at $\dot{m}_{in}/\dot{m}_{ref} = 0.25$ (left) and mass flow removal case at $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ with $\phi_{PS Slot} = 0.5$ (right) at 0.84 $N_{ref}$.

With regards to the exducer region, Figure 4-20 indicates that the entropy generated downstream of the PS Slot for the mass flow removal case is approximately 25% less than the for the non-ported compressor OP at $\dot{m}_{in}/\dot{m}_{ref} = 0.25$. Figure 4-22 shows $\dot{s}_{visc}$ of a crossflow plane downstream of the PS Slot for the non-ported compressor (left figure) and the mass flow removal case (right figure). The decrease in the exducer entropy generation is similarly traced to regions of the flow passage corresponding to span values greater than 70%. The partial removal of the inducer tip leakage flow before it moves downstream into the exducer region and mixes out with the main core flow is the cause for this reduction. It has been ruled out that the higher entropy generated for the non-ported compressor OP is due to higher mixing losses between the exducer tip leakage and main flow. This is inferred from the fraction of tip leakage mass flow with respect to the inlet mass flow $\dot{m}_{tip leakage}/\dot{m}_{inlet}$, plotted in Figure 4-23, which indicates that the tip leakage flow in the exducer for the mass flow removal OP is higher than that of the non-ported OP. The higher tip leakage flow is a direct result of the higher blade loading in the compressor exducer, as shown in Figure 4-24.
Figure 4-23: Exducer tip leakage mass flow fraction is higher for the mass flow removal case of $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ with $\phi_{PS\text{Slot}} = 0.5$ (dashed green) compared to that of non-ported OP $\dot{m}_{in}/\dot{m}_{ref} = 0.25$ (solid green) at 0.84 $N_{ref}$.

Figure 4-24: Increase of impeller blade loading due to mass flow removal at PS Slot; mass flow removal OP of $\dot{m}_{in}/\dot{m}_{ref} = 0.5$ with $\phi_{PS\text{Slot}} = 0.5$ and non-ported OPs at 0.84 $N_{ref}$
4.6 Effect of Mass Flow Removal on Compressor Stability

It has been shown that the use of a ported shroud increases the compressor operating range and thus the stability compared to that of a non-ported compressor. The aim of this section is to assess whether the actuation of mass flow removal at the PS Slot is effective in enhancing the compressor stability. The total to static pressure rise characteristic \( \Delta p_{t-s} = p_{out} - p_{t,in} \) of a compressor is considered to be closely linked to the compressor stability [27, 31]. When the slope of \( \Delta p_{t-s} \) becomes positive, \( \frac{\partial \Delta p_{t-s}}{\partial \dot{m}} > 0 \), the compressor system can amplify disturbances leading to compressor surge. The compressor total to static pressure rise \( \Delta p_{t-s} \) is non-dimensionalized using \( q_{ref} = \frac{1}{2} \rho_{inlet} U^2_{tip} \), hence \( \Delta \dot{p}_{t-s} = \Delta p_{t-s} / q_{ref} \). The slope of the compressor total to static pressure rise, \( \frac{\partial \Delta \dot{p}_{t-s}}{\partial (\dot{m}_{out}/\dot{m}_{ref})} \), is used to assess the effect of mass flow removal on the compressor stability.

![Graph showing the effect of mass flow removal on compressor stability](image)

**Figure 4-25:** Mass flow removal (red and green) results in more negative values of the slope of \( \Delta p_{t-s} \) compared to non-ported compressor OPs (blue) at 0.84 \( N_{ref} \), thus increasing the compressor stability.
Figure 4-25 shows $\frac{\partial \Delta \rho}{\partial (m_{out}/m_{ref})}$ for non-ported compressor OPs (blue) and mass flow removal OPs with a baseline inlet flow of $m_{in}/m_{ref} = 0.83$ (red) and $m_{in}/m_{ref} = 0.5$ (green). The slope of the total to static pressure rise for OPs with mass flow removal is negative and has a lower value compared to non-ported OPs of same $m_{out}$ thus increasing the compressor stability. It is therefore inferred that the static pressure increase between mass flow removal and non-ported compressor OPs due to the diffusion of the primary stream and the decrease of flow blockage leads to more negative values of $\frac{\partial \Delta \rho_{c-s}}{\partial (m_{out}/m_{ref})}$ and therefore increases the compressor stability.

In addition to the results based on the slope of the total to static pressure rise, the extent of the recirculation zone near the inducer shroud is also used as a qualitative metric of the stability of the compressor. This is in accordance with the metric used to detect surge during gas stand measurements, where the static pressure is measured at the compressor inlet. Once the gauge pressure measures a positive value, because work has been done to the fluid in the recirculation zone, the compressor is assumed to have started to surge. The effect of mass removal has been shown to decrease the extent of the recirculation zone as shown in Figure 4-9.

### 4.7 Summary

The effect of mass flow removal at the PS Slot on the compressor performance has been assessed in this chapter. Increasing the fraction of mass flow removed results in the increase of the compressor pressure ratio. The removal of mass flow results in the decrease of the impeller flow blockage and increases the diffusion of the compressor main stream.

It is shown that removing 50% of the inducer mass flow results in the increase of the inducer static pressure rise between mass flow removal and non-ported compressor OPs of equal inducer mass flow by up to 100%. The diffusion of the main flow stream, a primarily one-dimensional effect, accounts for 80% of the inducer static pressure increase for OPs where the inducer
corresponds to peak $\eta$ conditions. At OPs near surge, the diffusion mechanism accounts for approximately 40% of the calculated increase.

The decrease of flow blockage is due to the partial removal and reduction of the recirculation zone near the inducer shroud through the removal of the inducer tip leakage flow. For OPs near surge, removal of 50% of the inducer flow decreases the inducer blockage by up to 10%. The increase of the effective flow area directly accounts for approximately 10% of the higher inducer pressure rise capability between mass flow removal and non-ported OPs. It has been shown that the removal of flow blockage leads to the increase of the blade relative inlet flow angle and higher turning of the flow at the top 20% of the span. Both of these effects increase the blade loading between mass removal and non-ported OPs of equal inducer mass flow which accounts for an additional 17%-50% of the static pressure increase inside the inducer. Therefore, the decrease of the flow blockage inside the inducer accumulatively accounts for up to 60% of the inducer static pressure increase for the mass flow removal OPs.

In terms of efficiency, by removing the tip leakage flow originating from the inducer, a decrease of entropy generation corresponding up to 2.5 points of efficiency is shown upstream of the PS Slot location; a similar decrease of entropy generation inside the exducer region is also found. The efficiency penalty associated with mass flow removal at OPs near surge, assuming that the PS flow is discarded, may be as high as 12% compared to non-ported OPs of equal outlet mass flow. Through the use of the slope of the compressor total to static pressure rise, it is found that mass flow removal increases the stability of the compressor. It is therefore inferred that the static pressure increase due to the diffusion of the primary stream and the decrease of flow blockage enhances the compressor stability.
Chapter 5  Physical Interpretation of PS Actuations at PS Outlet

The results of Chapter 3 have shown that the PS is perceived as a source of mass as well as axial and tangential momentum at the PS Outlet. The effect of the PS flow actuations at the PS Outlet on the compressor performance is assessed in this chapter. The effect of mass flow injection is firstly assessed, followed by axial and angular momentum injection.

5.1  Mass Flow Injection at PS Outlet

In this section the actuation of mass flow injection at the PS Outlet is modeled and its effect on the compressor performance assessed. The results indicate that the difference in the compressor stagnation pressure ratio and efficiency between OPs with mass flow injection and non-ported OPs of equal outlet mass flow is negligible.

5.1.1.  Modeling – Approach

The single passage PS Actuation Model used as a platform for the investigation of mass flow injection at the PS Outlet is shown in Figure 5-1. The approach followed is to select a baseline inlet mass flow $\dot{m}_{in}$ at the compressor inlet and increase the fraction of mass flow injected at the PS Outlet $\varphi_{PS\ Out} = \dot{m}_{PS\ Outlet}/\dot{m}_{in}$. The $\dot{m}_{in}$ of the compressor remains constant while the $\dot{m}_{out}$ increases with increasing values of $\varphi_{PS\ Out}$. The flow at the PS Outlet is injected upstream along the axial direction (no circumferential velocity component) at the same $T_t$ as the flow at the upstream main flow inlet. Setting the $T_t$ at the PS Outlet equal to that of the main inlet is considered justified since the focus of this investigation is to assess only the actuation of mass...
flow. No mass flow is removed at the PS Slot. The OPs selected as the baseline for two investigated speedlines are listed in Table 5-1.

![Diagram](image)

**Figure 5-1:** Cross section of computational model used as platform to assess effect of mass flow injection at PS Outlet on compressor performance with \( \dot{m}_{out} = \dot{m}_{in} + \dot{m}_{PS\, Outlet} \)

<table>
<thead>
<tr>
<th>Speedline</th>
<th>Baseline 1 ( \dot{m}_{in} )</th>
<th>Baseline 2 ( \dot{m}_{in} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.84 ( N_{ref} )</td>
<td>( \dot{m}<em>{in}/\dot{m}</em>{ref} = 0.41 )</td>
<td>( \dot{m}<em>{in}/\dot{m}</em>{ref} = 0.25 )</td>
</tr>
<tr>
<td>1.16 ( N_{ref} )</td>
<td>( \dot{m}<em>{in}/\dot{m}</em>{ref} = 1.21 )</td>
<td>-</td>
</tr>
</tbody>
</table>

**Table 5-1:** Baseline OPs used for assessment of mass flow flow injection at PS Outlet on compressor performance

### 5.1.2. Effect of Mass Flow Injection on Compressor Performance

The effect of increasing the fraction of the injected flow at the PS Outlet on the compressor pressure ratio and efficiency is shown for the 0.84 \( N_{ref} \) speedline in Figure 5-2 and Figure 5-3 respectively. Pressure ratio and efficiency are plotted on a \( \dot{m}_{out} \) basis. The blue data points
correspond to single passage non-ported compressor OPs. The red data points correspond to OPs where mass flow is injected at the PS Outlet with respect to a baseline case of \( \dot{m}_{in}/\dot{m}_{ref} = 0.41 \) (indicated with the red circle in Figure 5-2). The fraction of flow injected is \( \varphi_{PS\,Out} = 0.2, 0.4 \) and 0.6. The green data points correspond to cases of mass flow injection where the inlet mass flow is set to \( \dot{m}_{in}/\dot{m}_{ref} = 0.25 \) (indicated with the green circle in Figure 5-2) with \( \varphi_{PS\,Out} = 0.2, 0.45 \) and 0.75. The compressor \( \pi_c \) and \( \eta \) for the 1.16 \( N_{ref} \) speedline are shown in Figure 5-4.

![Figure 5-2: Negligible effect of mass flow injection at PS Outlet on compressor \( \pi_c \) compared to non-ported compressor OPs of same \( \dot{m}_{out} \) at 0.84 \( N_{ref} \); red circle indicates baseline OP of \( \dot{m}_{in}/\dot{m}_{ref} = 0.41 \), green circle indicates baseline OP of \( \dot{m}_{in}/\dot{m}_{ref} = 0.25 \)](image)

The difference in \( \pi_c \) between OPs where mass flow is injected normal to the PS Outlet and that of non-ported compressor OPs of the same \( \dot{m}_{out} \) is found to be less than 0.55%. It is found that mass flow injection results in a penalty in efficiency of up to one point. In the context of PS compressor operation, the actuation of mass flow injection results in the inducer working at a higher mass flow than the rest of the compressor and therefore further away from surge.
However, the data suggests no additional benefit from mass flow injection, such as energizing of the tip flow, compared to the performance of the inducer of a non-ported OP at same $n_{inducer}$.

![Diagram](image)

Figure 5-3: Mass flow injection at PS Outlet results in penalty of up to one point in $\eta$ compared to non-ported compressor OPs of same $m_{out}$ at $0.84 \cdot N_{ref}$; red circle indicates baseline OP of $m_{in}/m_{ref} = 0.41$, green circle indicates baseline OP of $m_{in}/m_{ref} = 0.25$

![Diagram](image)

Figure 5-4: Mass flow injection at PS Outlet has negligible effect on $\pi_c$ (left) and results in penalty of less than one point in $\eta$ (right) compared to non-ported compressor OPs of same $m_{out}$ at $1.16 \cdot N_{ref}$
5.2 Axial Momentum Injection at PS Outlet

The effect of axial momentum injection on the compressor performance is assessed in this section. Increasing the axial momentum of the flow injected at the PS Outlet is found to deteriorate the compressor performance.

5.2.1 Modeling - Approach

The same variant of the PS Actuation Model used for the assessment of mass flow injection at the PS Outlet is also used here. For each computed OP, the mass flow at the exit of the compressor $\dot{m}_{out}$ and the mass flow fraction injected at the PS Outlet with respect to the outlet mass flow $\varphi_{PS\,out} = \dot{m}_{PS\,outlet}/\dot{m}_{out}$ are held fixed. The flow at the PS Outlet is injected upstream along the axial direction and at the same $T_e$ as the flow at the upstream main flow inlet. The level of axial momentum is controlled by varying the area of the PS Outlet. Maintaining the mass flow constant and decreasing the area of the PS Outlet, results in the increase of the axial momentum $J_{PS,\,axial} = \int \rho u_{axial} dm_{PS\,Outlet} = \frac{\dot{m}_{PS\,Outlet}^2}{\rho^2 A_{PS\,Outlet}}$. No mass flow is removed at the PS Slot location.

The effect of axial momentum at peak $\eta$ and near surge is assessed for three speeds described in Table 5-2. The mass flow fraction injected for all cases is set to $\varphi_{PS\,out} = 0.4$. The $\dot{m}_{out}/(\dot{m}_{out})_{peak\,\eta}$ ratio is a cross-reference of the outlet mass flow of the assessed OP with respect to that of the peak $\eta$ OP at the same speedline.
5.2.2. **Effect of Axial Momentum Injection at PS Outlet**

The effect of increasing the axial momentum of the injected flow at the PS Outlet on the compressor pressure ratio and efficiency is assessed for the peak $\eta$ and near surge OPs at $1.16 \, N_{ref}$ and $1.48 \, N_{ref}$ described in Table 5-3. The color coding in Table 5-3 refers to all the figures in this section.

<table>
<thead>
<tr>
<th>Speedline</th>
<th>Description</th>
<th>$\dot{m}<em>{out}/\dot{m}</em>{ref}$</th>
<th>$\dot{m}<em>{out}/(\dot{m}</em>{out})_{peak\eta}$</th>
<th>$\varphi_{PS, Out}$</th>
<th>Color Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.84 $N_{ref}$</td>
<td>Peak $\eta$</td>
<td>$\dot{m}<em>{out}/\dot{m}</em>{ref} = 0.83$</td>
<td>1</td>
<td>0.4</td>
<td>Blue</td>
</tr>
<tr>
<td>0.84 $N_{ref}$</td>
<td>Near Surge</td>
<td>$\dot{m}<em>{out}/\dot{m}</em>{ref} = 0.41$</td>
<td>0.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.16 $N_{ref}$</td>
<td>Peak $\eta$</td>
<td>$\dot{m}<em>{out}/\dot{m}</em>{ref} = 1.21$</td>
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<td></td>
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</tr>
<tr>
<td>1.16 $N_{ref}$</td>
<td>Near Surge</td>
<td>$\dot{m}<em>{out}/\dot{m}</em>{ref} = 0.63$</td>
<td>0.52</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.48 $N_{ref}$</td>
<td>Peak $\eta$</td>
<td>$\dot{m}<em>{out}/\dot{m}</em>{ref} = 1.6$</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>1.48 $N_{ref}$</td>
<td>Near Surge</td>
<td>$\dot{m}<em>{out}/\dot{m}</em>{ref} = 1.39$</td>
<td>0.85</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 5-2: Investigated OPs during axial momentum injection at PS Outlet assessment

Table 5-3: Baseline OPs at peak $\eta$ and near surge used to assess influence of axial momentum injection on compressor performance at $1.16 \, N_{ref}$ and $1.48 \, N_{ref}$
The compressor $\pi_c$ and $\eta$ for 1.16 $N_{ref}$ and 1.48 $N_{ref}$ are shown in Figure 5-5 and Figure 5-6 respectively and are plotted as a function of $J_{PS,axial}/J_{PS,axial,ref}$ where $J_{PS,axial,ref}$ is the axial momentum of the injected stream corresponding to a reference/baseline PS Outlet area. Increasing the momentum of the jet at the PS Outlet is found to have a detrimental effect on both compressor $\pi_c$ and $\eta$ regardless of the OP or speedline. The largest effects are observed at the peak $\eta$ OP $m_{out}/m_{ref} = 1.6$ at the high speedline 1.48 $N_{ref}$ where an efficiency penalty of up to 9 points and a decrease in $\pi_c$ of up to 10% is computed. To assess the effect of the axial momentum injection on the impeller, the impeller $\pi_c$ at 1.48 $N_{ref}$ is shown in Figure 5-7. Injection of axial momentum of more than four times that of the reference/baseline value is found to decrease the impeller $\pi_c$ by less than 1% (compared to 10% for the entire compressor). Therefore, it is inferred that the amount of axial momentum injected upstream of the wheel inlet has negligible effect on the impeller itself and that the performance deterioration of the compressor is due to flow mechanisms upstream of the wheel inlet.

Figure 5-5: Higher values of injected axial momentum at PS Outlet result in decrease of compressor $\pi_c$ (left) and $\eta$ (right) at 1.16 $N_{ref}$
Figure 5-6: Higher values of injected axial momentum at PS Outlet result in decrease of compressor $\pi_c$ (left) and $\eta$ (right) at 1.48 $N_{ref}$.

Figure 5-7: Negligible effect of axial momentum injection at PS Outlet on impeller $\pi_c$ at 1.16 $N_{ref}$.
5.2.3. Loss due to Mixing of Main and PS Flow Stream

To trace the origin of the detrimental effect of higher levels of axial momentum injection at the PS Outlet, the entropy generated upstream of the wheel inlet and inside the impeller is shown in Figure 5-8 for $\frac{m_{out}}{m_{ref}} = 1.21$ at $1.16 \, N_{ref}$. All the additional loss for OPs with larger values of axial momentum injection at the PS Outlet is generated upstream of the wheel inlet. The entropy generation inside the impeller is unaffected by the level of the injected axial momentum at the PS Outlet as shown by the right chart of Figure 5-8. The entropy generation in the region upstream of the wheel inlet is due to the mixing losses between the injected PS stream at the PS Outlet and the incoming main flow from the upstream inlet.

The entropy generation due to the mixing of two streams can be determined by using the following equation proposed by Young and Wilcock [27]. The main stream flow direction is considered to be in the $x$ direction, the ratio of the injected to mainstream flow is $d\dot{m}/\dot{m}$, the static temperature of the main stream flow is $T$ and that of the injected stream is $T_{inj}$. It is also assumed that the injected flow enters the control volume at the static pressure of the main stream.
\[
\frac{ds_{\text{irrev}}}{C_p} = \frac{d\dot{m}}{\dot{m}} \left\{ \left[ \frac{(u_x - u_{x,\text{inj}})^2 + u_y^2}{2C_pT} \right] + \int_T^{T_{\text{inj}}} \left[ \frac{1}{T} - \frac{1}{\bar{T}} \right] d\bar{T} \right\}
\]

Equation 5-1

The first square bracket in Equation 5-1 represents the entropy change due to the dissipation of the kinetic energy of the two streams in both mainstream, \(x\) direction, and normal to the mainstream direction as both velocities mix to a uniform state. The second bracket represents the entropy change due to thermal mixing of the two streams [25]. For the cases that have been assessed, the dominant effect in setting the entropy generation due to mixing is the velocity difference in the main stream direction between \(u_{\text{axial}}\), of the main inlet flow and that of the flow injected at the PS Outlet. The entropy generation due to differences in static temperature is between one and two orders of magnitude smaller than the kinetic energy dissipation term and there is no component of the injected flow normal to the main stream.

The entropy generated in the region upstream of the wheel inlet using the CV analysis is compared to that of the CFD for the peak \(\eta_{\text{OP}} = \dot{m}_{\text{out}}/\dot{m}_{\text{ref}} = 1.21\) at \(1.16 N_{\text{ref}}\) in Figure 5-9. The CFD generated \(\Delta s\) is lower than that based on the CV analysis, which indicates that the flow has not mixed out completely to a uniform state. The actual entropy generation upstream of the wheel inlet scales with the square of the velocity difference of the two streams in the axial direction \(\Delta u_{\text{axial}}^2\). These results imply that to minimize the loss generation upstream of the wheel inlet, the PS Outlet should be designed in such a way as to minimize the velocity difference between PS and main flow stream.
5.3 Angular Momentum Injection at PS Outlet

The effect of the angular momentum injection at the PS Outlet, and more specifically the direction of the injected PS flow, on the compressor performance is the focus of this section. Injection of the flow at the PS Outlet in a direction opposite of the wheel rotation increases the compressor stagnation pressure ratio and the impeller static pressure rise while succumbing to a penalty in efficiency compared to OPs where flow is injected with zero or co-swirl. The effect of angular momentum injection on the compressor performance is shown to decrease at OPs with high flow blockage upstream of the blade LE.
5.3.1. **Modeling – Approach**

The same variant of the PS Actuation Model used for the assessment of mass flow and axial momentum injection at the PS Outlet, described in Section 2.3.2.2, is also used here. For each investigated OP, the compressor exit mass flow $\dot{m}_{out}$ and mass flow fraction of the injected flow at the PS Outlet are fixed $\phi_{PS\,Outlet} = \dot{m}_{PS\,Outlet}/\dot{m}_{out}$. The stagnation temperature of the injected flow is equal to that of the main flow inlet. The swirl angle of the injected stream $\alpha = \arctan(u_\theta/u_{axial})$ is varied from $-50^\circ \leq \alpha \leq 50^\circ$. Positive values of $\alpha$ correspond to flow injected in the direction of the wheel rotation (co-swirl) and negative values to the opposite direction of the wheel rotation (counter-swirl). The effect of angular momentum at peak $\eta$ and a near surge OP is assessed for three speedlines described in Table 5-4. The mass flow fraction injected for all cases is set to $\phi_{PS\,out}=0.4$.

<table>
<thead>
<tr>
<th>Speedline</th>
<th>Description</th>
<th>$\dot{m}<em>{out}/\dot{m}</em>{ref}$</th>
<th>$\dot{m}<em>{out}/(\dot{m}</em>{out})_{peak\eta}$</th>
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<tr>
<td>0.84 $N_{ref}$</td>
<td>Peak $\eta$</td>
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<td>0.84 $N_{ref}$</td>
<td>Near Surge</td>
<td>$\dot{m}<em>{out}/\dot{m}</em>{ref} = 0.41$</td>
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</tr>
<tr>
<td>1.16 $N_{ref}$</td>
<td>Peak $\eta$</td>
<td>$\dot{m}<em>{out}/\dot{m}</em>{ref} = 1.21$</td>
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<tr>
<td>1.16 $N_{ref}$</td>
<td>Near Surge</td>
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<td>0.52</td>
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<tr>
<td>1.48 $N_{ref}$</td>
<td>Peak $\eta$</td>
<td>$\dot{m}<em>{out}/\dot{m}</em>{ref} = 1.6$</td>
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<td>1.48 $N_{ref}$</td>
<td>Near Surge</td>
<td>$\dot{m}<em>{out}/\dot{m}</em>{ref} = 1.39$</td>
<td>0.85</td>
</tr>
</tbody>
</table>

Table 5-4: Investigated OPs during angular momentum at PS Outlet assessment

5.3.2. **Effect of Angular Momentum Injection at PS Outlet**

The effect of the swirl angle of the injected flow at the PS Outlet on the compressor pressure ratio and efficiency is assessed for the OPs between peak $\eta$ and near surge at 1.16 $N_{ref}$ and
1.48 \( N_{ref} \) described in Table 5-5. The \( \dot{m}_{out}/(\dot{m}_{out})_{peak \eta} \) ratio is a cross-reference of the outlet mass flow of the assessed OP against that of the peak \( \eta \) OP at that speedline. The color coding in Table 5-5 refers to Figure 5-10 and Figure 5-11.

<table>
<thead>
<tr>
<th>Speedline</th>
<th>( \dot{m}<em>{out}/\dot{m}</em>{ref} )</th>
<th>( \dot{m}<em>{out}/(\dot{m}</em>{out})_{peak \eta} )</th>
<th>( \varphi_{PSO ut} )</th>
<th>Color Code</th>
</tr>
</thead>
<tbody>
<tr>
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<td>1.21</td>
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<td>0.4</td>
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<td>0.85</td>
<td>0.4</td>
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</table>

Table 5-5: Baseline OPs used to assess influence of angular momentum injection on compressor performance with \( \varphi_{PSO ut}=0.4 \) at 1.16 \( N_{ref} \) and 1.48 \( N_{ref} \).

The compressor pressure ratio and efficiency at 1.16 \( N_{ref} \) and 1.48 \( N_{ref} \) are plotted as a function of the swirl angle \( \alpha \) of the injected flow in Figure 5-10 and in Figure 5-11 respectively. Table 5-6 shows the \( \Delta \pi_c = \frac{\pi_{c, \alpha=-50^\circ} - \pi_{c, \alpha=50^\circ}}{\pi_{c, \alpha=50^\circ}} \) and \( \Delta \eta = \eta_{\alpha=-50^\circ} - \eta_{\alpha=50^\circ} \) between counter-swirl \( \alpha = -50^\circ \) and co-swirl \( \alpha = 50^\circ \) angular momentum injection OPs at 1.16 \( N_{ref} \) and 1.48 \( N_{ref} \).
Table 5-6: Differences in $\pi_c$ and $\eta$ between counter-swirl $\alpha = -50^\circ$ and co-swirl $\alpha = 50^\circ$ angular momentum injection OPs with $\varphi_{PS_{out}} = 0.4$ at 1.16 $N_{ref}$ and 1.48 $N_{ref}$

<table>
<thead>
<tr>
<th>Speedline</th>
<th>$\dot{m}<em>{out}/\dot{m}</em>{ref}$</th>
<th>$\dot{m}<em>{out}/(\dot{m}</em>{out})_{peak\eta}$</th>
<th>$\Delta \pi_c$ [%]</th>
<th>$\Delta \eta$</th>
<th>Color Code</th>
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<tbody>
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<td>0.84</td>
<td>6</td>
<td>-4.8</td>
<td>Blue</td>
</tr>
<tr>
<td>1.16 $N_{ref}$</td>
<td>0.83</td>
<td>0.69</td>
<td>1.5</td>
<td>-4.2</td>
<td>Yellow</td>
</tr>
<tr>
<td>1.16 $N_{ref}$</td>
<td>0.63</td>
<td>0.52</td>
<td>0.3</td>
<td>-2.6</td>
<td>Red</td>
</tr>
<tr>
<td>1.48 $N_{ref}$</td>
<td>1.6</td>
<td>1</td>
<td>20</td>
<td>-6.5</td>
<td>Black</td>
</tr>
<tr>
<td>1.48 $N_{ref}$</td>
<td>1.39</td>
<td>0.85</td>
<td>5.5</td>
<td>-5</td>
<td>Purple</td>
</tr>
</tbody>
</table>

Figure 5-10: Counter-swirl angular momentum injection at the PS Outlet results in increase of compressor $\pi_c$ (left) and decreases of $\eta$ (right) at 1.16 $N_{ref}$.
Introduction of counter-swirl to the injected PS stream tends to increase $\pi_c$ and decrease $\eta$. This effect is amplified at OPs working at higher $\dot{m}_{out}$. For peak $\eta$ OPs, a change of $\alpha$ corresponding to 100° results in the increase of $\pi_c$ by approximately 12% and decrease of the efficiency by five points at 1.16 $N_{ref}$. For the peak $\eta$ OP $\dot{m}_{out}/\dot{m}_{ref} = 1.6$ at 1.48 $N_{ref}$, changing the injected flow swirl angle from $\alpha = 50^\circ$ to $\alpha = -50^\circ$ increases the $\pi_c$ by 20% and decreases the efficiency by 6.5 points. The same change in swirl angle results in less than 0.5% increase of $\pi_c$ and a 2.5 point drop in efficiency at the near surge OP $\dot{m}_{out}/\dot{m}_{ref} = 0.63$ ($\dot{m}_{out}/(\dot{m}_{out})_{peak \eta} = 0.52$) at 1.16 $N_{ref}$. For OPs between peak $\eta$ and near surge, the differences in compressor performance between counter and co-swirl angular momentum injection are found to be between those calculated for the peak $\eta$ and near surge OP as shown in Table 5-6.
Figure 5-12: Angular momentum injection at the PS Outlet is more effective in changing the impeller relative flow angle at OP near peak $\eta$ at $\dot{m}_{out}/\dot{m}_{ref} = 1.21$ (A) and between peak $\eta$ and surge OP at $\dot{m}_{out}/\dot{m}_{ref} = 0.83$ (B); the effect is negligible for OPs near surge at $\dot{m}_{out}/\dot{m}_{ref} = 0.63$ (C) with $\phi_{PSout} = 0.4$ at $1.16 N_{ref}$

The compressor performance at the near surge OP is less sensitive to changes of the injected flow swirl angle (for an equal fraction of injected mass flow at the PS Outlet $\phi_{PSout}$) compared to the peak $\eta$ OP. The different behavior can be understood when plotting the relative flow angle at the impeller blade LE. The relative inlet flow angle profile for the peak $\eta$ OP $\dot{m}_{out}/\dot{m}_{ref} = \ldots$
1.21 with \( \varphi_{PSout} = 0.4 \) is shown in chart (A) of Figure 5-12, for the \( \dot{m}_{out}/\dot{m}_{ref} = 0.83 \) (\( \dot{m}_{out}/(\dot{m}_{out})_{peak} \eta = 0.69 \)) OP between peak \( \eta \) and surge in chart (B), while that of the near surge \( \dot{m}_{out}/\dot{m}_{ref} = 0.63 \) (\( \dot{m}_{out}/(\dot{m}_{out})_{peak} \eta = 0.52 \)) is shown in chart (C). The difference of the impeller relative flow angle between counter-swirl (\( \alpha = -50^\circ \)) and co-swirl (\( \alpha = 50^\circ \)) is found to be up to 12\(^\circ\) for the peak \( \eta \) OP and is located near the shroud (span>80%). For the OP between peak \( \eta \) and surge the difference in flow angle is approximately 9\(^\circ\). In the case of the near surge OP \( \dot{m}_{out}/\dot{m}_{ref} = 0.63 \), the largest difference between counter and co-swirl is found to be at mid-span and less than 5\(^\circ\), implying that the incidence angle is almost the same for both co-swirl and counter-swirl.

Figure 5-13: \( u_{axial}/U_{tip} \) (left) and \( u_\theta/U_{tip} \) in absolute frame (right) indicating upstream extent of inducer recirculation zone for angular momentum injection at PS Outlet for \( \dot{m}_{out}/\dot{m}_{ref} = 0.63 \) with \( \varphi_{PSout} = 0.4 \) and \( \alpha = -50^\circ \) at 1.16 \( N_{ref} \).

To address why the relative flow angle profile for the near surge OPs is unaffected by changes of 100\(^\circ\) of the swirl angle of the injected PS stream, the flow field upstream of the impeller LE is assessed. The axial velocity on a meridional plane upstream of the blade LE for the OP near surge \( \dot{m}_{out}/\dot{m}_{ref} = 0.63 \) (\( \dot{m}_{out}/(\dot{m}_{out})_{peak} \eta = 0.52 \)) is shown in the left chart of Figure 5-13 and the circumferential velocity in the absolute frame \( u_\theta \) is shown in the right chart of Figure 5-13. For OPs near surge there is a recirculation zone near the inducer shroud region with swirl
component in the direction of the wheel rotation, since work has already been added to the flow. The upstream flow of fluid originating from the inducer and its subsequent re-injection inside the impeller is responsible for setting the impeller inlet flow angle for span locations greater than 70% at near surge OPs.

To assess how the recirculation zone near the inducer shroud affects the impeller relative flow angle the CV shown in Figure 5-14 is used. Station i corresponds to a location upstream of the blade LE, where the PS and main stream flow are assumed to have mixed out entirely. Station e corresponds to a location between Station i and the blade LE, where the recirculation zone blocks a portion of the geometric area, which results in the flow passing through a smaller effective area $A_{e,\text{eff}}$.

![Recirculation Zone](image)

**Figure 5-14: Control volume upstream of impeller inlet**

Mixing out of the PS flow, which contains angular momentum, with the main stream flow results in a flow at Station i corresponding to solid body rotation of angular velocity $\Omega_i$ and uniform $u_{axial,i}$. The angular velocity $\Omega_i$ is given by:
\[ \Omega_i = 2\varphi u_{\text{axial},PSt} \tan \alpha_{PSt} \frac{r_{PS}}{r_{i}^2} \]

Equation 5-2

Where \( \varphi = \frac{\dot{m}_{PS}}{\dot{m}_{PS} + \dot{m}_{\text{inlet}}} \)

The solid body rotation of the flow at Station \( i \) corresponds to uniform streamwise vorticity of \( \omega_i = 2\Omega_i \). The swirl angle at Station \( i \) is given by:

\[ \tan \alpha_i = \frac{u_{\theta,i}}{u_{\text{axial},i}} = \frac{\Omega_i r_i}{u_{\text{axial},i}} = 2\varphi^2 \frac{A_i}{A_{PS}} \tan \alpha_{PSt} \frac{r_{PS}}{r_{i}^2} r \]

Equation 5-3

Equation 5-4 shows the swirl angle at Station \( i \) is a function of the domain geometry, fraction of injected mass flow and swirl angle at the PS Outlet. Therefore \( \alpha_i \) is the same for all OPs provided that the mass flow fraction \( \varphi_{PSout} = \frac{\dot{m}_{PSout}}{\dot{m}_{out}} \) and swirl angle \( \alpha \) of the injected stream are equal.

The flow from Station \( i \) to Station \( e \) can be thought of a flow through a nozzle. Assuming incompressible and inviscid flow, the ratio of the streamwise vorticity \( \omega \), at Station \( i \) to Station \( e \) is:

\[ \frac{\omega_e}{\omega_i} = \frac{u_{\text{axial},e}}{u_{\text{axial},i}} \]

Equation 5-4

Based on the analysis shown in [25], the ratio of swirl angle at Station \( i \) to Station \( e \) is related by:

\[ \frac{\alpha_e}{\alpha_i} \sim \frac{A_e}{A_i} \]

Equation 5-5

It is inferred through Equation 5-5, that for OPs near surge the decrease of the effective flow area upstream of the blade LE, due to the recirculation zone, leads to the decrease of the swirl angle of the injected flow in the absolute frame. Reducing the swirl angle in the absolute frame results in a subsequent reduction of the absolute and relative frame flow angle difference between co-
swirl and counter-swirl angular momentum injection OPs. The difference in the inlet flow angle between counter and co-swirl angular momentum injection cases decreases when moving from the peak η OP to OPs near surge along the speedline. This is because the extent of the inducer recirculation zone, in both upstream and spanwise direction, continuously increases when operating closer to surge conditions as indicated in Figure 5-15. Figure 5-15 shows the absolute circumferential velocity $u_\theta/U_{tip}$ upstream of the blade LE for the peak η OP $\dot{m}_{out}/\dot{m}_{ref} = 1.21$ in figure (A), an OP between peak η and surge $\dot{m}_{out}/\dot{m}_{ref} = 0.83$ ($\dot{m}_{out}/\dot{m}_{out,peak} = 0.69$) in figure (B) and the near surge OP $\dot{m}_{out}/\dot{m}_{ref} = 0.63$ ($\dot{m}_{out}/\dot{m}_{out,peak} = 0.52$) in figure (C). Positive values of $u_\theta$ (pre-swirl) indicate fluid to which the compressor has already imparted work and is therefore used as an indicator to define the extent of the induce recirculation zone.

It is found that the introduction of $\alpha = 50^\circ$ co-swirl at the PS Outlet at the near peak η OP at 1.16 $N_{ref}$ results in the decrease of the compressor pressure ratio by approximately 6% compared to an OP with $\alpha = 0^\circ$ (refer to Figure 5-10). However, it is well documented in literature that the use of Inlet Guide Vanes (IGVs) imparting co-swirl to the flow upstream of the wheel inlet (bulk swirl) reduces the pressure ratio by more than 20% even when set at angles below $\alpha = 40^\circ$ [13, 35]. It is found that the compressor performance is more responsive (up to three times) to equal $\Delta\alpha$ specified at the inlet than at the PS Outlet. This is because bulk inlet swirl spans the entire blade span. On the other hand, setting the swirl angle of the flow exiting the PS Cavity affects the swirl angle of the flow upstream of the blade over a fraction of the blade span (refer to Figure 5-12). Additional details regarding the differences between bulk inlet swirl and angular momentum injection at the PS Outlet can be found in Appendix C.
Figure 5-15: \(u_\theta/U_{tip}\) in absolute frame indicating that the extent of the inducer recirculation zone increases when moving closer to surge conditions; peak \(\eta\) OP \(\dot{m}_{out}/\dot{m}_{ref} = 1.21\) (A), \(\dot{m}_{out}/\dot{m}_{ref} = 0.83\) (B) and OP near surge \(\dot{m}_{out}/\dot{m}_{ref} = 0.63\) (C) with \(\varphi_{PSout} = 0.4\) and \(\alpha = -50^\circ\) at \(1.16N_{ref}\)

5.3.3. **Efficiency Penalty Associated with Counter-Swirl Angular Momentum Injection**

It has been determined that the introduction of counter-swirl results in the decrease of the compressor efficiency (see Figure 5-10 and Figure 5-11). The computed decrease is higher for OPs working at higher mass flows and for higher fractions of injected mass flow at the PS Outlet.
In this section the flow mechanisms responsible for the efficiency penalty associated with counter-swirl angular momentum injection are assessed.

The entropy generation mechanisms for OPs near peak $\eta$, where the extent of the inducer recirculation zone is small compared to OPs near surge, are firstly assessed. The efficiency penalty at the peak $\eta$ OP $\dot{m}_{out}/\dot{m}_{ref} = 1.21$ with $\varphi_{PSout} = 0.4$ at $1.16 N_{ref}$ between counter-swirl injection of $\alpha = -50^\circ$ and co-swirl injection of $\alpha = 50^\circ$ is found to be five points as shown in Figure 5-10. The impeller entropy generation is shown in Figure 5-16. The solid lines correspond to the entropy generated across the entire passage span, while the dashed lines correspond to the entropy generated in the top 20% of the span. The data shows that the counter-swirl injection case increases the impeller entropy generation by approximately 50%, which corresponds to a decrease of five points in efficiency. More than 60% of the higher loss between counter-swirl and co-swirl inside the impeller is accounted for by the higher entropy generated in the shroud region.

Figure 5-16: Approximately 60% of the additional entropy generation between counter-swirl and co-swirl angular momentum injection at PS Outlet is accounted for through the higher entropy generation in the impeller shroud region for peak $\eta$ OP $\dot{m}_{out}/\dot{m}_{ref} = 1.21$ with $\varphi_{PSout} = 0.4$ at $1.16 N_{ref}$
It is hypothesized that the higher entropy generated in the impeller shroud region for the counter-swirl OP is due to the higher mixing loss between tip leakage and main flow. The tip leakage mass flow fraction with respect to the outlet mass flow, $\dot{m}_{tip}/\dot{m}_{out}$, shown in Figure 5-17, indicates that the tip leakage mass flow for the counter-swirl OP is up to 25% higher than that of the co-swirl case. The higher tip leakage mass flow for the counter-swirl OP is a result of the higher blade loading for the counter-swirl OP as shown in Figure 5-17, where $\Delta C_p = \frac{P_{PS} - P_{SS}}{0.5 \rho \omega_{wheel} U_{tip}^2}$ and is caused by the higher blade inlet flow angle shown in Figure 5-12.

Figure 5-17: Counter-swirl angular momentum injection at PS Outlet results in the increase of the tip leakage mass flow for peak $\eta$ OP $\dot{m}_{out}/\dot{m}_{ref} = 1.21$ with $\varphi_{PS\text{out}} = 0.4$ at $1.16 N_{ref}$.
To assess whether the higher entropy generated near the impeller shroud is because of higher mixing losses of the tip leakage flow with the main flow, a mixed out analysis is carried out. The mixed out approach takes into account that the two streams initially have different stagnation temperature and pressure and assumes that the mixing takes place instantaneously as in a constant area duct. Since the tip leakage flow rate is small compared to the main passage flow (<3%) the following expression for the entropy generated due to mixing of the two streams, proposed in [32], is used:

$$\frac{\Delta s}{c_p} = \frac{m_{inj}}{m_{main}} \left\{ \left(1 + \frac{\gamma - 1}{2} M^2_{main}\right) \frac{T_{t, inj} - T_{t, main}}{T_{t, main}} + (\gamma - 1) M^2_{main} \left(1 - \frac{V_{inj}}{V_{main}} \cos \gamma \right) \right\}$$

Equation 5-6

In the equation above, quantities with the subscript “main” denote properties of the main passage flow, while the subscript “inj” refers to quantities of the injected flow. The relative angle between the injected and main flow is $\gamma$ as shown in Figure 5-19.
Figure 5-19: Mixing of injected flow with main passage flow at different velocity and temperature

Figure 5-20 compares the entropy computed in the top 20% of the impeller using CFD with that calculated using the mixed out analysis for the CV. The entropy generated through the mixed out analysis is consistent with the CFD generated values, indicating that the additional entropy generated in the top 20% span for the counter-swirl OP is due to the higher mixing losses. The CFD calculated entropy generation for the counter-swirl case is close to the entropy calculated through the mixed out analysis. It is hypothesized that this is due to additional loss generated near the LE of the impeller blade due to the higher incidence angle of the flow. It is found that up to 10% of the higher entropy generated inside the impeller for the counter-swirl OP can be traced to the higher entropy generated near the vicinity of the blade LE across the entire blade span.
Figure 5-20: Entropy generation due to mixing of the tip leakage flow and main passage flow for CFD and mixed out CV analysis for $\dot{m}_{out}/\dot{m}_{ref} = 1.21$ with $\varphi_{psout} = 0.4$ at $1.16 \, N_{ref}$

The results presented so far have shown the effect of angular momentum injection on the entropy generation at peak $\eta$ OPs. The entropy generation mechanisms for OPs near surge will now be assessed. The efficiency penalty at the near surge OP $\dot{m}_{out}/\dot{m}_{ref} = 0.63$ ($\dot{m}_{out}/(\dot{m}_{out})_{peak \eta=0.52}$) with $\varphi_{psout} = 0.4$ at $1.16 \, N_{ref}$ between counter-swirl injection of $\alpha = -50^\circ$ and co-swirl injection of $\alpha = 50^\circ$ is approximately 2.5 points as shown in Figure 5-10 of Section 5.3.2. The entropy generated upstream of the wheel inlet is shown in the left chart of Figure 5-21 and that inside the impeller is shown in the right chart of Figure 5-21. The solid lines correspond to the entropy generated across the entire passage span, while the dashed lines correspond to the entropy generated at span values greater than 80%. The increase of the entropy generated between counter-swirl and co-swirl OPs is approximately equally split between additional loss generation upstream of the wheel inlet and inside the impeller. The differences in entropy generated upstream of the wheel inlet and inside the impeller correspond to a penalty of approximately one and 1.5 point in efficiency respectively for the counter-swirl compared to the co-swirl OP.
The additional loss upstream of the wheel inlet is due to the larger mixing loss between the injected flow at the PS Outlet and the recirculation zone extending from the inducer shroud region. As shown in Figure 5-13, the backflow region extends upstream of the wheel inlet and has swirl in the same direction as the wheel rotation. Even though the same mass flow is injected between co-swirl and counter-swirl cases, the swirl direction for the counter-swirl is in the opposite direction of the wheel rotation. Subsequently, there is a higher velocity difference between injected flow and backflow in the circumferential direction, which is responsible for the higher entropy generation in accordance with Equation 5-1.

The swirl angle of the injected flow at the PS Outlet is found to have no effect in the entropy generated in the impeller tip region, as shown in the right chart of Figure 5-21. This is to be expected since, as shown in Figure 5-12, the relative inlet flow angle for OPs near surge is similar regardless of the direction of the injected flow which results in similar levels of blade loading and tip leakage flow as shown in Figure 5-22. This is due to the recirculation zone present near the inducer shroud, which has been shown to reduce the difference in blade inlet flow angle between co-swirl and counter-swirl OPs.

![Figure 5-21: Efficiency penalty for counter-swirl angular momentum injection at PS Outlet equally split between additional loss generation upstream of the wheel inlet (left) and inside the impeller (right) for OP near surge](image)

\[
\dot{m}_{out}/\dot{m}_{ref} = 0.63 \ \varphi_{PS\text{Out}} = 0.4 \ a = 0^\circ
\]

\[
\dot{m}_{out}/\dot{m}_{ref} = 0.63 \ \varphi_{PS\text{Out}} = 0.4 \ a = 50^\circ
\]

\[
\dot{m}_{out}/\dot{m}_{ref} = 0.63 \ \varphi_{PS\text{Out}} = 0.4 \ a = -50^\circ
\]
Figure 5.22: Negligible effect of direction of injected flow at PS Outlet on tip leakage mass flow for OP near surge
\[ \dot{m}_{out}/\dot{m}_{ref} = 0.63 \] with \( \varphi_{PSOut} = 0.4 \) at \( 1.16 N_{ref} \)

5.3.4. Stagnation Pressure Ratio Increase due to Counter-Swirl Angular Momentum Injection

In Section 5.3.2 it is shown that the injection of counter-swirl at the PS Outlet results in the increase of the compressor pressure ratio for peak \( \eta \) OPs. For OPs near surge, where there is a strong recirculation zone near the inducer shroud, the effect of angular momentum injection at the PS Outlet on the compressor pressure ratio has been shown to be negligible.

For the peak \( \eta \) OPs it is found that more than 75% of the stagnation pressure increase between compressor outlet and inlet \( P_{t, outlet} - P_{t, inlet} \) can be accounted for due to the impeller static pressure rise increase \( P_{wheel, outlet} - P_{wheel, inlet} \). The static pressure coefficient \( C_p \) for the case where counter \((\alpha = \) \(-50^\circ\)) and co-swirl \((\alpha = \) \(50^\circ\)) is injected at the PS Outlet at the peak \( \eta \) OP \( \dot{m}_{out}/\dot{m}_{ref} = 1.21 \) at \( 1.16 N_{ref} \) with \( \varphi_{PSout} = 0.4 \) is shown in Figure 5-23. The counter-swirl case is shown to increase the impeller static pressure rise by 33% compared to the co-swirl case. The increase of the static pressure rise of the impeller is traced to that of the inducer, indicating
that the increase is a result of the higher inlet flow angle for the counter-swirl angular momentum injection case.

![Graph showing the static pressure coefficient Cp for the case where counter (a = -50°) and co-swirl (a = 50°) is injected at the PS Outlet at the near surge OP. The direction of the injected flow has no effect on the impeller static pressure rise at the near surge OP. This is to be expected since, as shown in Figure 5-12, the flow angle at this OP is similar for counter and co-swirl angular momentum injection.]

The static pressure coefficient $C_p$ for the case where counter ($a = -50°$) and co-swirl ($a = 50°$) is injected at the PS Outlet at the near surge OP $m_{out}/m_{ref} = 0.63 (m_{out}/(m_{out})_{peak \eta = 0.52})$ at $1.16 N_{ref}$ with $\varphi_{PSout} = 0.4$ is shown in Figure 5-24. The direction of the injected flow at the PS Outlet is shown to have no effect on the impeller static pressure rise at the near surge OP. This is to be expected since, as shown in Figure 5-12, the flow angle at this OP is similar for counter and co-swirl angular momentum injection.

In conclusion, these results imply that the effectiveness of counter-swirl injection in increasing the static pressure rise of the impeller is dependent on the inducer recirculation zone. For equal fractions of injected flow at the PS Outlet the effectiveness of counter-swirl in increasing the impeller static pressure rise decreases as the compressor approaches surge.
5.3.5. Effect of Angular Momentum Injection on Compressor Stability

To assess the effect of angular momentum injection at the PS Outlet on the compressor stability, the slope of the total to static compressor rise $\frac{\partial \Delta \rho_{t-s}}{\partial (\dot{m}_{out}/\dot{m}_{ref})}$, defined in Section 4.6, is used. The slope of the total to static compressor rise is plotted against $\dot{m}_{out}$ in Figure 5-25 for OPs at $1.16 \ N_{ref}$ corresponding to a fraction of injected mass flow at the PS Outlet of $\varphi_{PSout} = 0.4$ with $\alpha = 0^\circ, 50^\circ$ and $-50^\circ$. The results indicate that imparting counter-swirl to the PS flow decreases the stability of the compressor, since the values of $\frac{\partial \Delta \rho_{t-s}}{\partial (\dot{m}_{out}/\dot{m}_{ref})}$ even though negative are higher than the values for OPs with co-swirl. The largest difference in the slope $\frac{\partial \Delta \rho_{t-s}}{\partial (\dot{m}_{out}/\dot{m}_{ref})}$ between counter and co-swirl angular momentum injection is found at the peak $\eta$ OP. This trend is consistent with the difference in the blade inlet flow angle between counter and co-swirl cases, which is found to decrease when moving from peak $\eta$ to near surge OPs along a speedline due to
the larger extent of the inducer recirculation zone. At the peak $\eta$ OP the blade inlet flow angle for the co-swirl case is found to be approximately $12^\circ$ lower than that of the counter-swirl injection while at the near surge OP this drops to $5^\circ$. It is inferred through the results that counter-swirl angular momentum injection is destabilizing for the compressor performance due to the higher blade incidence angle.

The increased stability for the co-swirl case compared to injection at $a = 0^\circ$ is in accordance with the results published in [34]. Y. Mingyang et al. introduce a circumferential rib on the inner wall of the inlet duct with the aim of imposing co-swirl only to the flow near the impeller shroud. Their results indicate that the increase of compressor stability is due to the decrease of the flow incidence angle near the inducer tip.

![Graph](attachment:image.png)

Figure 5-25: Counter-swirl angular momentum injection at PS Outlet results in less negative values of the slope of $\Delta p_{c-s}$ and decreases the compressor stability compared to co-swirl injection with $\varphi_{PSout} = 0.4$ at $1.16 \ N_{ref}$
5.4 Summary

In this chapter the effects of individual PS flow actuations at the PS Outlet (while no mass flow is removed at the PS Slot) are assessed in terms of their effect on the compressor performance.

Mass Flow Injection

The difference in the compressor stagnation pressure ratio and efficiency between OPs with mass flow injection and non-ported OPs of equal outlet mass flow is found to be negligible. In the context of PS compressor operation, where flow recirculates through the PS Cavity, mass flow injection at the PS Outlet results in the inducer working at an OP further away from surge. However, there is no additional benefit from mass flow injection in the inducer, such as energizing the inducer tip flow, compared to the performance of a non-ported OP at same $m_{\text{inducer}}$.

Axial Momentum Injection

The effect of increasing the axial momentum of the flow injected at the PS Outlet is only detrimental to the compressor performance. It is shown that this is due to the higher losses associated with the larger velocity difference between main inlet and injected streams.

Angular Momentum Injection

Injection of the flow at the PS Outlet in a direction opposite of the wheel rotation increases the compressor stagnation pressure ratio and the impeller static pressure rise while succumbing to a penalty in efficiency compared to OPs where flow is injected with zero or co-swirl. The effect of angular momentum injection on the compressor performance is shown to decrease at OPs with high flow blockage upstream of the blade LE. When moving from the peak $\eta$ OP to OPs near surge the extent of the inducer recirculation zone gradually increases thus decreasing the effective flow area. The decrease of the effective flow area upstream of the blade LE results in
the decrease of the swirl angle of the injected flow in the absolute frame; the flow becomes more aligned with the axial direction. As a result, in the case of counter-swirl angular momentum injection, the blade inlet flow angle decreases compared to flow conditions where there is no recirculation zone, therefore reducing the effectiveness of counter-swirl. The higher entropy generation associated with counter-swirl injection is due to the higher mixing loss between tip leakage and main flow and the larger blade incidence angle. Finally, it is shown that the injection of counter-swirl decreases the stability of the compressor.
Chapter 6  Combination of PS Flow Actuations at PS Outlet and Slot

The effect of the PS flow actuations at the PS Outlet and Slot on the compressor performance have so far been assessed individually. In this chapter the combination of PS flow actuations active at both PS Outlet and Slot locations is assessed. A case where the PS stream at the PS Slot is discarded rather than recirculated through the PS Cavity is examined first. A second case where the combined effect of co-swirl and counter-swirl flow injection at the PS Outlet with mass flow removal at the PS Slot is then assessed. The Chapter concludes with an overall summary of the effect of all PS flow actuations on the compressor performance and a physical interpretation of the PS compressor operation.

6.1 Discarding the Flow at the PS Slot versus Recirculating

The PS Flow Actuation model is used in this section to assess how discarding the flow removed at the PS Slot compares against recirculating it through the PS Cavity. To address this question, the volute and structural ribs inside the computational domain of the full wheel PS compressor (shown in Figure 2-5) are removed. This is done to obtain an axisymmetric PS flow field, where the only circumferential non-uniformity is due to the impeller blades, and to consistently compare the results against those from the single passage PS Actuation model. A number of OPs for the axisymmetric PS compressor are computed after which the mass flow recirculating through the PS Cavity $\dot{m}_{PS}/\dot{m}_{ind}$, shown in Figure 6-1, is used as a boundary condition on the PS Slot of the PS Actuation model. The $\dot{m}_{out}$ between the two models is specified to be the same, while the compressor inlet mass flow for the PS Actuation model is equal to the inducer mass flow $\dot{m}_{ind}$ of the axisymmetric PS compressor (see Figure 6-2).
Figure 6-1: Mass flow fraction of recirculating flow $\dot{m}_{PS}/\dot{m}_{ind}$ for axisymmetric PS compressor at $1.16 N_{ref}$ used as boundary condition at PS Slot for PS Flow Actuation model.

Figure 6-2: Meridional cross section of axisymmetric PS model (left) and PS Actuation Model (right)
The computed compressor pressure ratio and efficiency at 1.16 $N_{ref}$ for both axisymmetric PS compressor and mass flow removal OPs are shown in Figure 6-3. The $\eta$ values for the mass removal OPs are calculated taking into account that the PS flow removed is discarded and not reused (see analysis in Section 4.2). Comparison of the compressor performance between the axisymmetric PS and mass flow removal OPs on an $\dot{m}_{out}$ basis shows that discarding and not recirculating the PS stream increases the $\pi_c$ by up to 10% while succumbing to no additional penalty (Figure 6-3). The increase of the wheel $\pi_c$ between mass flow removal OPs and recirculating PS is approximately 7% (Figure 6-4). The remaining 3% is accounted for by the lower $\Delta P_t$ upstream of the wheel inlet for the mass removal case since there is no flow exiting from the PS Outlet to mix with the main inlet flow.

Figure 6-3: Discarding the PS flow at the PS Slot results in the increase of the compressor $\pi_c$ (left) while succumbing to no additional $\eta$ penalty (right) compared to the axisymmetric PS at 1.16 $N_{ref}$
Figure 6-4: Discarding the PS flow at the PS Slot results in the increase of the wheel $\pi_e$ compared to the axisymmetric PS at 1.16 $N_{ref}$

Figure 6-5: The pre-swirl contained in the recirculating flow for the axisymmetric PS compressor results in a lower impeller relative inlet flow angle compared to the mass flow removal OP at $m_{out}/m_{ref} = 0.59$ at 1.16 $N_{ref}$
The flow exiting the PS Outlet for the axisymmetric PS OPs contains pre-swirl due to the work already added by the inducer. At the near surge OP $\dot{m}_{out}/\dot{m}_{ref} = 0.59$ the swirl angle of the flow at the PS Outlet is approximately $65^\circ$. As a result, the impeller relative inlet flow angle, shown in Figure 6-5, is approximately $10^\circ$ less for the axisymmetric PS compressor compared to that of the mass flow removal OP. The higher relative inlet flow angle for the mass flow removal OPs leads to a higher blade loading, as shown in Figure 6-6, which results in higher compressor $\pi_c$, in accordance with the analysis shown in Section 4.4.

![Figure 6-6: The higher impeller relative flow angle for the mass flow removal OPs results in higher blade loading compared to the PS axisymmetric compressor OP at $\dot{m}_{out}/\dot{m}_{ref} = 0.59$ at 1.16 $N_{ref}$](image)

The entropy generated inside the impeller at near surge OP $\dot{m}_{out}/\dot{m}_{ref} = 0.59$ is shown in Figure 6-7. The solid lines correspond to the entropy generated across the entire passage span, while the dashed lines to the entropy generated in the top 20% of the span. The entropy generation inside the impeller for the mass flow removal OP is approximately 30% higher than that of the axisymmetric ported OP. The higher entropy generation for the entire flow passage is traced to the top 20% of the span. The larger generation of entropy near the impeller shroud is
due to the higher mixing loss between tip leakage and main flow stemming from the larger blade loading, in accordance with the findings shown in Section 5.3.3.

![Impeller entropy generation for axisymmetric ported and mass removal OP](image)

Figure 6-7: Impeller entropy generation for axisymmetric ported and mass removal OP $\dot{m}_{out}/\dot{m}_{ref} = 0.59$ at $1.16 \times N_{ref}$

In addition to the higher entropy generated inside the impeller, the mass flow removal OPs incur a penalty in efficiency due to the discarding of the useful work of the PS stream. For the near surge OP $\dot{m}_{out}/\dot{m}_{ref} = 0.59$ this accounts for approximately 13 points in efficiency. For the axisymmetric PS compressor, the entropy generated inside the PS cavity and upstream of the wheel inlet accounts for approximately 17 points of efficiency. The net result is a similar efficiency between recirculating and mass flow removal OPs as shown in Figure 6-3.

In conclusion, discarding rather than recirculating the PS flow combines the effect of both mass flow removal and removing the pre-swirl from the injected flow at the PS Outlet. The combined result of these actuations is the increase of the compressor pressure ratio while the difference in the efficiency is shown to be negligible.
6.2 Mass Flow Removal and Angular Momentum Injection

Assessment of the PS flow actuations on the compressor performance has yielded that removal of mass flow at the PS Slot and counter-swirl angular momentum injection at the PS Outlet enhance the performance of the PS compressor. The combined effect of these two actuations is the focus of this section.

6.2.1. Modeling – Approach

A total of five OPs are used to assess the combined effect of mass removal at the PS Slot and angular momentum injection at the PS Outlet. Table 6-1 shows the operating conditions for each assessed OP. The color code listed in Table 6-1, refers to all the figures in Section 6.2 unless otherwise stated.

<table>
<thead>
<tr>
<th>Name</th>
<th>$m_{out}/m_{ref}$</th>
<th>$m_{in}/m_{ref}$</th>
<th>$m_{ind}/m_{ref}$</th>
<th>$m_{PS Outfit}/m_{out}$</th>
<th>$m_{PS Slot}/m_{out}$</th>
<th>$\alpha$</th>
<th>Color Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\hat{m} = 0.63$</td>
<td>0.63</td>
<td>0.63</td>
<td>0.89</td>
<td>0.4</td>
<td>0.4</td>
<td>65°</td>
<td>Blue</td>
</tr>
<tr>
<td>$\hat{a} = 65°$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\hat{m} = 0.63$</td>
<td>0.63</td>
<td>0.63</td>
<td>0.89</td>
<td>0.4</td>
<td>0.4</td>
<td>-65°</td>
<td>Red</td>
</tr>
<tr>
<td>$\hat{a} = -65°$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MassRemoval</td>
<td>0.63</td>
<td>0.89</td>
<td>0.89</td>
<td>-</td>
<td>0.4</td>
<td>-</td>
<td>Green</td>
</tr>
<tr>
<td>NonPorted</td>
<td>0.63</td>
<td>0.63</td>
<td>0.63</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>Grey</td>
</tr>
<tr>
<td>$\hat{m} = 0.89$</td>
<td>0.89</td>
<td>0.89</td>
<td>0.89</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>Yellow</td>
</tr>
</tbody>
</table>

Table 6-1: Table of computed OPs for assessing effect of combination of mass removal and angular momentum injection
The operating conditions for the computed OPs are as follows:

i) \( \dot{m} = 0.63\ a = 65^\circ \) and \( \dot{m} = 0.63\ a = -65^\circ \)

The computational model used is shown in Figure 6-8. Mass flow is removed at the PS Slot and mass flow is injected at the PS Outlet. The mass flow injected at the PS Outlet \( \dot{m}_{PS\ Outlet} \) and removed at the PS Slot \( \dot{m}_{PS\ slot} \) are set to be equal. The swirl angle of the injected flow at the PS Outlet for the \( \dot{m} = 0.63\ a = 65^\circ \) OP is set to \( a = 65^\circ \). These operating conditions correspond to the values calculated at the near surge OP for the axisymmetric PS compressor at 1.16 \( N_{ref} \).

The swirl angle for the second \( \dot{m} = 0.63\ a = -65^\circ \) OP is set to \( a = -65^\circ \).

ii) \( \dot{m} = 0.63 \) Mass Removal

The computational domain used for this OP is described in Chapter 2.3.2.2. Only mass flow is removed at the PS Slot. The mass flow at the upstream compressor inlet is equal to the mass flow through the inducer. The \( \dot{m}_{ind} \) and \( \dot{m}_{PS\ slot} \) (and therefore \( \dot{m}_{out} \)) is equal to that for the two first OPs \( \dot{m} = 0.63\ a = 65^\circ \) and \( \dot{m} = 0.63\ a = -65^\circ \). The purpose of this OP, where only mass flow is removed at the PS Slot, is to serve as a reference case, where the effect of mass flow
removal is clearly distinguishable from the effect of the combination of mass removal and angular momentum injection.

iii) $\hat{m} = 0.63$ NonPorted and $\hat{m} = 0.89$ NonPorted

The $\hat{m} = 0.89$ NonPorted OP corresponds to the $\hat{m}_{\text{ind}}$ and the $\hat{m} = 0.63$ NonPorted OP to the $\hat{m}_{\text{out}}$ of the three previously described OPs.

6.2.2. Combined Effect of Mass Removal and Angular Momentum Injection

The compressor and impeller pressure ratio for all five OPs are shown in Figure 6-9. The OP with mass flow removal and counter-swirl angular momentum injection (red) is found to have the highest compressor and wheel pressure ratio, followed by the OP with only mass flow removal (green). The difference in compressor and wheel $\pi_c$ between these two OPs is less than 1% and 3% respectively. The higher increase found for the wheel $\pi_c$ compared to the compressor $\pi_c$, between the OPs corresponding to the red and green points, is due to the $P_t$ drop upstream of the wheel inlet for OPs with both counter-swirl mass flow injection and removal. The mixing loss between injected and inlet stream decreases the $P_t$ leading to a lower $\pi_c$ from compressor inlet to outlet. This additional loss, which decreases the overall compressor efficiency by 1.5 point, is absent in the case of only mass removal. With respect to the non-ported OP corresponding to the same outlet mass flow (grey), the wheel $\pi_c$ for the mass flow removal OP (green) is approximately 14% higher and for the OP combined mass flow removal and counter-swirl injection (red) 15% higher. It is therefore inferred that the removal of mass flow is the dominant actuation in setting the PS compressor $\pi_c$. Imparting counter-swirl to the PS flow at the PS Outlet results in the additional increase of $\pi_c$ with respect to the mass flow removal case.
Figure 6-9: Mass flow removal combined with counter-swirl angular momentum injection is found to have the highest compressor $\pi_c$ (left) and impeller $\pi_c$ (right).

Figure 6-10: Mass flow removal at the PS Slot (green) increases the impeller $C_p$ by 30% compared to non-ported $\bar{m} = 0.63$ OP (grey); mass flow removal and counter-swirl angular momentum injection (red) results in a further increase of 5%.
The impeller static pressure rise $Cp$ for the five OPs is shown in Figure 6-10. The difference in static pressure increase between OPs at the inducer exit is approximately equal to that at the impeller exit, implying that the static pressure increase of the impeller can be traced to the increase in the inducer. The mass flow removal OP (green) increases the impeller $Cp$ by approximately 30% compared to the non-ported OP of same $\dot{m}_{out}$ (grey), while the OP with mass flow removal and counter-swirl injection (red) by 35%. At the inducer exit, the static pressure rise for the mass flow removal OP (green) is more than twice that of the $\dot{m} = 0.89$ NonPorted OP (yellow) and the $Cp$ for the OP with mass flow removal and counter-swirl (red) is approximately 10% higher than that of the mass flow removal OP (green).

![Figure 6-11: Diffusion of primary stream (blue bar) and increase in blade loading (black bar) account for more than 80% of inducer $\Delta Cp$ between $\dot{m} = 0.63$ MassRemoval and $\dot{m} = 0.89$ NonPorted OP (left); increase of blade loading is the dominant mechanism for inducer $\Delta Cp$ between $\dot{m} = 0.63 \alpha = -65^0$ and $\dot{m} = 0.63$ MassRemoval (right)](image)

The breakdown of the inducer $\Delta Cp$ between the $\dot{m} = 0.63$ MassRemoval and $\dot{m} = 0.89$ NonPorted OP is shown in chart A in Figure 6-11. The deceleration of the main stream accounts for 40% of the higher static pressure increase between the mass flow removal and non-ported OP of equal $\dot{m}_{ind}$, the increase of the effective flow area accounts for 10% and the higher blade
loading for approximately 44%. The blade loading increase is not due to difference in the blade inlet flow angle since this is less than 4° but to the decrease of the flow blockage shown in Figure 6-12. The reduction of inducer flow blockage between the \( \tilde{m} = 0.89 \) NonPorted (low blockage) and \( \tilde{m} = 0.63 \) NonPorted (high blockage) OPs is up to 10%. The reduction between the non-ported OPs can be considered as a consequence of the inducer region of the PS compressor working at operating conditions further away from surge. The mass flow removal OP (green) and the OPs combining PS flow actuations at the PS Outlet and Slot (blue and red) are all shown to decrease the inducer flow blockage by an approximately 10% compared to the \( \tilde{m} = 0.89 \) NonPorted OP. The reduction between OPs with mass flow removal and the non-ported OP of equal \( m_{ind} \) can be considered as the result of the mass flow removal actuation. The decrease of blockage for the impeller passage is traced to the decrease computed at span values greater than 30%, in accordance with the findings in Section 4.3. The decrease of the flow blockage therefore accounts for approximately 55% of the static pressure increase, both directly due to the increase of \( A_{eff} \) and indirectly through the increase of the blade loading.

Chart B in Figure 6-11 shows the breakdown of the inducer \( \Delta Cp \) between the \( \tilde{m} = 0.63 \) \( \alpha = -65^0 \) and \( \tilde{m} = 0.63 \) MassRemoval OPs. The increase of blade loading due to the higher inlet flow angle is responsible for more than 75% of the higher inducer \( \Delta Cp \) for the \( \tilde{m} = 0.63 \) \( \alpha = -65^0 \) OP.
Figure 6-12: Reduction of inducer flow blockage dominated by mass flow removal actuation at PS Slot; negligible effect of angular momentum injection

Mass flow removal at the PS Slot is primarily responsible for the higher impeller static pressure rise compared to the non-ported compressor OP. The injection of counter-swirl at the PS Outlet results in the further increase of the inducer static pressure rise by approximately 10% compared to the mass flow removal OP. If the PS Cavity were included in the computational model, the higher static pressure rise at the inducer due to counter-swirl would result in a higher pressure difference between the PS Slot and Outlet. The higher pressure difference would lead to an estimated increase of the recirculated mass flow ratio from $\dot{m}_{PS\,Outlet}/\dot{m}_{out} = 0.4$ to approximately $\dot{m}_{PS\,Outlet}/\dot{m}_{out} = 0.43$ and the inducer mass flow would increase from $\dot{m}_{Inducer}/\dot{m}_{out} = 0.89$ to $\dot{m}_{Inducer}/\dot{m}_{out} = 0.92$. Consequently the inducer would be working at operating conditions further away from surge and the fraction of the recirculating flow through the PS Cavity would increase. Both of these effects would result in increasing the compressor operable range and are due to the higher inducer static pressure rise associated with counter-swirl angular momentum injection superimposed on the actuation of mass flow removal.
The compressor efficiency, shown in Figure 6-13, is computed taking into account that the flow removed at the PS Slot is reused. The efficiency for the mass flow removal OP (green) is approximately equal to that of the \( \hat{m} = 0.63 \alpha = 65^\circ \) OP (blue), while the efficiency of the counter-swirl injection \( \hat{m} = 0.63 \alpha = -65^\circ \) OP (red) is approximately 3.5 points lower than that of the co-swirl injection.

Figure 6-14 shows the entropy generation inside the impeller for the five OPs. The effect of only mass flow removal (green) is responsible for the decrease of the impeller entropy generation by approximately 25% compared to the \( \hat{m} = 0.63 \) NonPorted OP of same outlet mass flow. The decrease in entropy generation is due to the reduction of the backflow region near the inducer shroud and corresponds to approximately a four point increase in efficiency. The OP including mass flow removal and co-swirl angular momentum injection (blue) results in a further decrease of the impeller generated entropy corresponding to an additional 1.5 point in efficiency compared to the mass flow removal OP (green). The entropy generation for the counter-swirl injection OP (red) results in a penalty of 1.5 points in efficiency compared to the mass flow removal OP. The entropy generation decrease computed in the impeller through the actuation of...
mass flow removal and co-swirl injection is consistent with the decrease in entropy found inside the impeller between full wheel ported and non-ported OPs shown in Figure 3-14. These results indicate that the decrease in entropy generated inside the impeller is due to both the effect of mass flow removal at the PS Slot and the pre-swirl of the PS stream.

![Graph showing impeller entropy generation comparison]

Figure 6-14: The decrease of impeller entropy generation between Mass Removal and non-ported $\tilde{m} = 0.63$ OP corresponds to an increase of 4 points in efficiency; counter-swirl angular momentum injection reduces the efficiency by up to 1.5 points compared to the Mass Removal OP

The higher efficiency penalty in the case of counter-swirl is due to the larger entropy generation inside the impeller and more specifically the top 20% of the span. The entropy generated in the top 20% of the impeller span is shown in Figure 6-15 for OPs with PS flow actuations active at the PS Slot and Outlet. The increase of the entropy generated in the tip region for the $\tilde{m} = 0.63 a = -65^\circ$ OP compared to mass flow removal OP is in accordance with the findings in Section 5.3.3 where the effect of angular momentum injection on the compressor performance is assessed. The higher entropy generation for the counter-swirl OP is traced to the larger tip leakage flow and resulting higher mixing loss with the main flow.
6.2.3. Assessment of Efficiency Penalty due to Counter-Swirl Angular Momentum Injection

It has been shown in Sections 6.2.2 and 5.3.3 that the introduction of counter-swirl angular momentum at the PS Outlet decreases the efficiency of the impeller and therefore the compressor. However, the data published in [10] indicate that the introduction of vanes inside the PS cavity to add counter-swirl to the flow exiting the PS cavity has negligible effect on the overall compressor efficiency.

To assess the consistency of the results presented in this thesis with those published in [10], a single passage of the C239A PS compressor, including the PS Cavity, with guide vanes near the PS Outlet is modeled. The purpose of the guide vanes is to impart swirl to the flow exiting the PS cavity. To model both co-swirl and counter-swirl injection at specific angles, two variants of the model are used. The vanes are oriented so as to impart swirl angles of $a = 65^\circ$ in the direction of
the wheel rotation and $\alpha = -65^\circ$ opposite the wheel rotation. The PS Cavity vanes are shown in Figure 6-16 for the counter-swirl injection case. The noise suppressor shown in Figure 6-16 is added in accordance with the compressor geometry of [10].

Figure 6-16: Side view and frontal view of PS cavity guide vanes and noise suppressor with counter-swirl orientation

The computed compressor pressure ratio and efficiency with the PS guide vanes set at $\alpha = \pm 65^\circ$ at $1.16 N_{ref}$ is shown in Figure 6-17, while the mass flow fraction of the recirculating flow defined as $\dot{m}_{PS}/\dot{m}_{in}$ is shown in Figure 6-18.
Figure 6-17: Introducing counter-swirl through vanes inside the PS Cavity increases the compressor $\pi_c$ (left) and decreases the $\eta$ (right) by 1% compared to co-swirl OPs at 1.16 $N_{ref}$.

Figure 6-18: Introducing counter-swirl through vanes inside the PS Cavity increases the mass flow fraction recirculating through the PS cavity compared to co-swirl OPs at 1.16 $N_{ref}$.

For the OP near surge $\dot{m}/\dot{m}_{ref} = 0.63$, there is a higher $\pi_c$ by 1.5% for $\alpha = -65^\circ$ compared to the $\alpha = 65^\circ$ OP. This increase is comparable to the 2% increase computed in Section 6.2.2, where mass flow removal and angular momentum injection are modeled (with the same values of the injected swirl angle).
The counter-swirl OP succumbs to a penalty of one point in efficiency compared to the co-swirl OP. The decrease in efficiency between counter and co-swirl case is considered to be consistent with the data published in [10], where no efficiency penalty is measured between the recirculating PS compressor and the compressor with counter-swirl PS vanes. To assess the entropy generation mechanisms between counter and co-swirl injection OPs the entropy generated inside the individual compressor components is shown in Figure 6-19. The higher entropy generated inside the impeller with counter-swirl accounts for 3.5 points of efficiency. Same values of impeller $\Delta \eta$ are shown when mass flow removal and angular momentum injection are modeled based on the PS Actuation model (where the PS Cavity is not included) and shown in Figure 6-13 and Figure 6-14. This implies that the differences in entropy generation between counter and co-swirl angular momentum injection inside the impeller are captured with the PS Actuation model. The higher entropy generated inside the impeller for the counter-swirl OP is counter-balanced by the lower entropy generation rate inside the PS cavity, which accounts for three points of total efficiency. Even though a larger mass flow fraction is
recirculated for the counter-swirl case, the swirl angle of the flow removed at the PS Slot is smaller by approximately 30°. The lower swirl component of the velocity leads to a decrease of the profile loss generated inside the PS cavity and is found to account for one point of efficiency, using Equation 3-3. The remaining two points difference in efficiency are traced to the lower entropy generated at the PS guide vanes due to the lower incidence angle of the recirculating PS flow.

In conclusion, the use of the single passage PS compressor, where the PS cavity is included and contains guide vanes, has demonstrated that the introduction of counter-swirl to the recirculating PS flow increases the entropy generation rate inside the impeller. The increase of the entropy generated is in good agreement with the efficiency penalty computed when using the PS Actuation model. This efficiency penalty is counter-balanced by the lower entropy generated inside the PS cavity itself.

6.3 Summary

Discarding the PS flow that is removed at the PS Slot rather than recirculating it through the PS Cavity is found to be more beneficial in terms of increasing the compressor stagnation pressure ratio. This is because the recirculating PS flow contains pre-swirl which decreases the blade inlet flow angle and therefore the pressure ratio. In terms of efficiency, almost no difference is found between the mass flow removal case and the recirculating PS compressor, even when taking into account that the work done to the PS flow is discarded. The additional entropy generated inside the impeller for the mass flow removal case is counter-balanced by not having to account for the loss associated with recirculating flow through the PS Cavity and mixing with the inlet flow.

The effect of combining mass flow removal at the PS Slot and angular momentum injection at the PS Outlet on the compressor performance at an OP near surge is assessed in Section 6.2.2. The results indicate that mass flow removal is the dominant actuation in setting the PS compressor performance by reducing the extent of the flow recirculation near the inducer shroud.
and through the diffusion of the primary stream. The effect of mass flow removal at an OP near surge is shown to increase the inducer static pressure rise by more than 100% compared to the non-ported OP at the same inducer mass flow. The diffusion of the main flow stream and the decrease of the flow blockage are found to account for approximately 40% and 55% of the inducer static pressure increase respectively. The introduction of \( \alpha = -65^\circ \) counter-swirl to the PS flow at the PS Outlet increases the inducer static pressure by approximately 10% compared to the OP with only mass flow removal. The higher static pressure at the inducer due to counter-swirl would result in a higher pressure difference between the PS Slot and Outlet thereby increasing the recirculated mass flow of the PS compressor (if the PS Cavity were included in the computational model). Consequently the inducer would be working at operating conditions further away from surge and the fraction of the recirculating flow through the PS Cavity would increase. Both of these effects would result in increasing the compressor operable range and are due to the higher inducer static pressure rise associated with counter-swirl angular momentum injection superimposed on the actuation of mass flow removal.

6.4 Overall Summary of PS Flow Actuations- Interpretation of PS Compressor Operation

The findings of Chapters 4, 5 and 6, where the effects of individual and combination of PS flow actuations are assessed, are summarized and synthesized to provide an explanation of how a PS compressor works.

Overall Summary

Mass flow removal at the PS Slot acts in such a way as to decrease the flow blockage by partially removing the inducer recirculation zone and increase the diffusion of the primary stream. The findings in Chapter 4 and 6 show that at OPs near surge, mass flow removal results in the increase of the inducer static pressure rise by more than 100% compared to the non-ported OP
with equal inducer mass flow. The partial removal of the inducer recirculation zone is found to decrease the entropy generated inside the entire impeller and increase the efficiency by up to four points.

Mass flow injection at the PS Outlet results in the inducer working at a higher mass flow and therefore further away from surge. No additional benefit from the mass flow injection, such as energizing the inducer tip flow, compared to the performance of the inducer of a non-ported OP at same \( \dot{m}_{inducer} \) is found. Because the inducer is working at an OP closer to peak \( \eta \), the extent of the recirculation zone near the inducer shroud decreases which decreases the impeller flow blockage and entropy generation.

Increasing the velocity of the PS flow exiting the PS Cavity (while keeping the mass flow fixed) is shown to have a detrimental effect on the compressor performance due to increased mixing loss with the inlet stream.

The introduction of counter-swirl to the PS flow at the PS Outlet is shown to increase the compressor pressure ratio and decrease the efficiency. The higher pressure ratio is a result of the impeller doing more work on the flow and is traced to the higher blade inlet flow angle compared to OPs where the PS flow contains pre-swirl. The decrease in efficiency is shown to result from the higher mixing loss between tip leakage and main flow. The effectiveness of angular momentum injection at the PS Outlet is shown to be dependent on the extent of the inducer recirculation zone which continuously increases when moving from the peak \( \eta \) OP to near surge. The decrease of the effective flow area upstream of the blade LE results in the flow (in the absolute frame) becoming more aligned with the axial direction. As a result, in the case of counter-swirl angular momentum injection, the blade inlet flow angle decreases compared to flow conditions where there is no recirculation zone, therefore reducing the effectiveness of counter-swirl.
Interpretation of PS Compressor Operation

The findings of the effect of the PS flow actuations on the PS compressor are synthesized to explain the operation of a PS compressor. The effect of the PS near the peak \( \eta \) OP is assessed initially. A fraction of the inducer mass flow is assumed to be recirculated through the PS Cavity. The inducer mass flow is higher than that of the remaining compressor and working closer to the peak \( \eta \) OP. Therefore, the extent of the recirculation zone near the inducer shroud decreases compared to a non-ported compressor at equal outlet mass flow. As a result, the impeller flow blockage and entropy generation are also decreased. The described decrease in flow blockage and entropy generation is a consequence of the inducer working at operating conditions closer to peak \( \eta \). The effect of removing a fraction of the inducer mass flow is shown to further decrease both flow blockage and entropy generation compared to non-ported OPs of same inducer mass flow. Removal of mass flow at the PS Slot is shown to increase the impeller static pressure rise and the compressor stagnation pressure. When injecting the flow at the PS Outlet with counter-swirl, the compressor pressure ratio will further increase compared to a case where mass flow is injected at \( \alpha = 0^\circ \). Counter-swirl mass flow injection is shown to increase the inducer static pressure rise which results in the larger pressure difference across the PS Cavity thus forcing more flow to recirculate.

When moving along the speedline to an OP closer to surge, both the outlet and the inducer mass flow of the PS compressor decrease, which results in the gradual increase of the extent of the inducer recirculation zone. The mass flow fraction of the PS flow is also shown to increase, implying that the effects associated with mass flow removal are amplified. In the case where the flow at the PS Outlet is injected with counter-swirl, due to the larger fraction of recirculating flow compared to a case with zero or co-swirl, the development of the recirculation zone is delayed. As a result, the beneficial effects of counter-swirl in increasing the pressure ratio and inducer static pressure rise, and therefore the mass flow through the PS, are extended to OPs at lower outlet mass flow. It can therefore be inferred, that counter-swirl mass injection at the PS Outlet decreases the surge mass flow of the PS compressor compared to a PS compressor where the flow at the PS Outlet is injected with zero or co-swirl. In terms of the compressor efficiency, counter-swirl increases the entropy generation inside the impeller. This efficiency penalty,
however, is found to be counter-balanced by the lower entropy generated inside the PS cavity itself due to the lower swirl angle of the PS flow removed at the PS Slot.

The interpretation of the PS compressor operation described above is consistent and has been used to explain the findings of [10], where vanes at the PS Outlet add counter-swirl to the exiting PS flow. Gas stand measurements showed, that introduction of counter-swirl to the PS flow reduced the surge mass flow by up to 20% and increased the pressure ratio compared to the baseline (co-swirl) casing treatment. No additional efficiency penalty was measured between counter-swirl and co-swirl case. The additional range extension measured for the counter-swirl OPs is due to the effect of increasing the inducer static pressure rise because of the counter-swirl, which results in a larger fraction of recirculating PS flow thus delaying the onset of surge. The higher pressure ratio is shown to be a result of the counter-swirl increasing the blade inlet flow angle. Because of the larger fraction of PS flow for the counter-swirl case, the extent of the recirculation zone is smaller compared to co-swirl injection at OPs with the same outlet mass flow. Therefore, the effect of counter-swirl on increasing the pressure ratio is extended to lower mass flow OPs. In terms of the efficiency, it has been shown that the larger entropy generated inside the impeller is counter-balanced by the lower entropy generation inside the PS Cavity.

The findings of this thesis are used to interpret the range extension measured for PS compressor with an extended PS Cavity (shown in Section 1.3). Due to the extended length of the PS Cavity, the additional loss due to skin friction results in the decrease of the swirl angle (which is in the direction of the wheel rotation) of the PS flow. The lower co-swirl angle of the flow at the impeller LE results in a higher blade inlet flow angle compared to the baseline (short PS Cavity) PS compressor. Through the arguments presented above, the decrease of the surge mass flow for the extended cavity PS compressor is rationalized.
Chapter 7  Attenuation of Volute Induced Flow Non-Uniformity

One of the proposed mechanisms through which the PS extends the compressor operable range is by decoupling a portion of the impeller from the upstream influence of the volute. To test this hypothesis the PS Outlet is blocked so as to not allow flow to recirculate through the PS Cavity, thus effectively acting as a pressure equalizing chamber. The performance of this blocked configuration is assessed against that of a recirculating PS compressor.

7.1  Enhanced Attenuation of Volute Induced Flow Non-Uniformity with PS

It has been documented in literature that non-axisymmetric elements of a centrifugal compressor, namely the volute, create non-uniform flow fields that extend upstream to the impeller inlet [37, 38, 39, 40]. The impeller is thus subject to circumferentially varying inlet and outlet conditions which triggers flow instability in individual impeller passages at a higher compressor mass flow than if working under uniform conditions.

The static pressure field inside the diffuser and volute is shown in Figure 7-1 for OPs near choke $\dot{m} = 1.1 \dot{m}_{ref}$ (left) and surge $\dot{m} = 0.3 \dot{m}_{ref}$ (right) for the PS compressor at 0.84 $N_{ref}$. Figure 7-2 shows the static pressure for OPs near choke $\dot{m} = 1.89 \dot{m}_{ref}$ (left) and surge $\dot{m} = 1.23 \dot{m}_{ref}$ (right) at 1.48 $N_{ref}$. The flow exiting the diffuser impinges the volute tongue at positive incidence near choke OPs at both speedlines, while at OPs near surge the incidence angle is negative.
Figure 7-1: Diffuser and volute pressure field for PS compressor OP indicates positive incidence angle at volute tongue near choke $\dot{m} = 1.1 \dot{m}_{ref}$ (left) and negative incidence angle near surge $\dot{m} = 0.3 \dot{m}_{ref}$ (right) at $0.84 N_{ref}$.

Figure 7-2: Diffuser and volute pressure field for PS compressor OP indicates positive incidence angle at volute tongue near choke $\dot{m} = 1.89 \dot{m}_{ref}$ (left) and negative incidence angle near surge $\dot{m} = 1.23 \dot{m}_{ref}$ (right) at $1.48 N_{ref}$. 
In this discussion the variations of the flow quantities about the average value at the crossflow plane of interest are most important. Therefore, unless otherwise stated, the variation of static and stagnation pressure about the mean value at a crossflow plane is non-dimensionalized as follows:

$$\bar{C}_{p_i} = \frac{p - \bar{p}_i}{0.5\rho_{inlet}U_{tip}^2}$$  

Equation 7-1

The subscript $i$ denotes the crossflow plane of interest; $\bar{p}_i$: the average value of pressure at crossflow plane $i$

The mass flow of each passage, at a crossflow plane $i$, $\dot{m}_{j,i}$ (where the subscript $j$ denotes passage number) is normalized by the total mass flow through the crossflow plane of interest $\dot{m}_i = \sum_j \dot{m}_{j,i}$ divided by the number of passages $n$. This value corresponds to how significant the variation in each passage is compared to a case were the flow is uniform.

$$\tilde{\dot{m}}_{j,i} = \frac{n \dot{m}_j}{\dot{m}_i}$$  

Equation 7-2

Figure 7-3: Circumferential coordinate with respect to volute tongue
The circumferential coordinate with respect to the volute tongue used in this section is shown in Figure 7-3; \( \theta = 0^\circ \) corresponds to the volute tongue location and \( \theta \) increases in the direction of the wheel rotation. The volute induced pressure non-uniformity as a function of circumferential coordinate at the diffuser outlet for \( \dot{m} = 0.38 \dot{m}_{ref} \) and non-ported \( \dot{m} = 0.41 \dot{m}_{ref} \) OPs near surge is shown in Figure 7-4. The variations in static pressure for both ported and non-ported variants of the compressor imply that the same static pressure non-uniformity is applied at the impeller exit. However, the PS compressor decreases the flow non-uniformity at the wheel inlet compared to the non-ported compressor as shown through the stagnation pressure non-uniformity in Figure 7-5. The stagnation pressure non-uniformity at the wheel inlet stems from the work added to the flow recirculating through the PS Cavity for the PS compressor. In the case of non-ported compressor OPs, the stagnation pressure non-uniformity is due to the inducer shroud backflow. The PS compressor reduces the circumferential variations of \( P_t \) by 50% compared to the non-ported compressor. The non-uniformity of the non-ported case exhibits a 1/rev periodicity because the flow non-uniformity is induced by the compressor scroll. The ported compressor exhibits an additional 4/rev periodicity from the presence of the four structural ribs inside the PS cavity. These results show that the PS compressor enhances the
attenuation of the flow non-uniformity; however it is not clear how important the role of the PS cavity as a pressure equalizing device is in setting the PS compressor performance.

Figure 7-5: Stagnation pressure variation in circumferential direction at wheel inlet for ported \( \dot{m} = 0.38 \dot{m}_{ref} \) (blue) indicates more effective attenuation of flow non-uniformity compared to that of non-ported \( \dot{m} = 0.41 \dot{m}_{ref} \) OP (purple) at \( 0.84 N_{ref} \)

7.2 Role of PS Cavity as Pressure Equalizing Device

To assess the importance of the PS cavity in attenuating the flow non-uniformity the role of the PS as a pressure equalizing device is isolated from that of its associated flow actuations.

7.2.1. Approach

The approach taken to address the importance of the PS Cavity as a pressure equalizing device consists of assessing the flow behavior of three PS compressor configurations, shown in Figure
7-6. The first configuration is that of a conventional PS compressor which allows for flow recirculation and includes the four structural struts inside the PS cavity. The second configuration, No Ribs Ported Shroud, also allows for flow recirculation but does not contain struts inside the cavity. By removing the struts the flow field in the impeller will not be subject to the additional 4/rev non-uniformity seen in Figure 7-5, thus providing a clearer picture of how the recirculation device affects the volute induced flow non-uniformity. Finally, in the Blocked Cavity configuration the PS Outlet is blocked and flow recirculation is not allowed, thus effectively transforming the PS cavity into a pressure equalizing chamber.

![Figure 7-6: Schematic of compressor configurations used to assess the importance of PS Cavity as a pressure equalizing device](image-url)
The computed performance for the three compressor variants shown in Figure 7-6 and for that of the non-ported compressor at the 0.84 N_{ref}, 1.16 N_{ref} and 1.48 N_{ref} speedlines is shown in Figure 7-7. For the Blocked Cavity configuration only OPs corresponding between peak $\eta$ and surge are computed since the behavior near surge is of most interest. The results indicate that the Blocked Cavity configuration is as effective as the conventional PS compressor with respect to extending the range at the low 0.84 N_{ref} and mid 1.16 N_{ref} speedlines. At the high speedline 1.48 N_{ref} the surge mass flow is approximately equal to that of the non-ported compressor (shown in Figure 2-3 of Section 2.2) and is 15% larger than that of the conventional PS compressor. The results are qualitatively in agreement with the gas stand measurements provided by HTT, shown in Figure 1-6 of Section 1.3, where the PS Outlet of a PS compressor (not the C239A compressor used as a platform in this thesis) is blocked as shown in Figure 1-5. The calculations show that the Blocked Cavity case exhibits a higher efficiency of up to 9 points for OPs near surge. The increase in efficiency results from there being no flow recirculation inside the PS Cavity. As a consequence, the loss inside the PS Cavity corresponds to approximately one point of efficiency compared to approximately ten points for PS compressor OPs near surge as shown in Figure 3-14. Furthermore, there is no loss due to the mixing of the flow exiting the PS Cavity with that of the main inlet.
7.2.2. **Flow Non-Uniformity Inside Impeller**

To evaluate the role of the PS Cavity as a pressure equalizing device, the flow non-uniformity inside the impeller passages is assessed. The passages inside the impeller are numbered according to Figure 7-8, where $\theta = 0^\circ$ corresponds to the circumferential location of the volute tongue.

![Volute Tongue](image)

**Figure 7-8:** Frontal view of impeller indicating the number corresponding to each passage inside the impeller

The flow non-uniformity for the three test configurations and that of the non-ported compressor for the OP near surge at $\dot{m}/\dot{m}_{ref} = 0.41$ at $0.84 N_{ref}$ is shown in Figure 7-9. The static pressure non-uniformity $\bar{C}_p$ (Equation 7-1) inside each impeller passage is shown in the left column of Figure 7-9 and the mass flow non-uniformity $\hat{m}$ (Equation 7-2) is shown in the right column. The top row corresponds to the blade TE, the middle row to a cross flow plane just downstream of the PS Slot location and the bottom row to the blade LE. The relative position of each passage at the blade TE with respect to the volute tongue is shown in the charts in the top row.
The static pressure distribution at the blade TE indicates that all compressor variants are subject to the same non-uniformity in the circumferential direction. The static pressure non-uniformity is attenuated by more than 55% between blade TE and the PS Slot location and becomes negligible at the blade LE for all cases except the conventional PS. The static pressure distribution of the conventional PS compressor demonstrates a 4/rev periodicity that is due, as already mentioned, to the four structural ribs inside the PS Cavity. Between the PS Slot and the blade LE there is a redistribution of the flow inside the passages for the conventional PS and NoRibs Ported Shroud configuration which leads to a more uniform flow field inside the inducer. This redistribution results in the decrease of the mass flow non-uniformity by approximately 40% and 60% for the conventional PS and NoRibs Ported Shroud compressor respectively. The inducer mass flow distribution of the NoRibs Ported Shroud configuration is shown to be the most uniform with a maximum value of less than 10% of that corresponding to uniform flow. On the other hand, the distribution of mass flow inside the Blocked Cavity and Non-Ported compressor passages is found to remain constant throughout the entire length of the impeller.

The pressure and mass flow distribution inside the impeller passages at the nearest converged OP to surge, $\dot{m}/\dot{m}_{ref} = 1.39$ at $1.48 N_{ref}$, for the Blocked cavity is shown in Figure 7-10. The results indicate that as opposed to the results of the low speed line, the pressure non-uniformity that the Blocked Cavity compressor is subject to at the impeller exit is approximately 30% higher than that of the two recirculating cases. Even though the amplitude of the static pressure non-uniformity decreases by 80% when reaching the blade LE it is still more than twice that of the recirculating compressors. The amplitude of the mass flow non-uniformity at the impeller TE of the Blocked Cavity compressor is approximately 5% higher than that of the recirculating compressors. Similarly with the results at the low speedline, the mass flow non-uniformity remains constant throughout the entire impeller for the Blocked Cavity configuration which results in a mass flow non-uniformity of amplitude equal to 10% of the impeller mass flow. In the case of the recirculating compressors, the flow near the impeller LE is almost uniform.

The results indicate that the compressor variants that allow for flow recirculation are more effective in attenuating the mass flow non-uniformity compared to the non-ported and Blocked Cavity compressors. This is due to flow redistribution inside the inducer which results in a more uniform flow field. The Blocked Cavity compressor is subject to higher flow non-uniformity at
the impeller exit at the OP near surge at \(1.48 \, N_{ref}\) compared to the recirculating PS compressor variants. Because no flow redistribution is allowed, the amplitude of the mass flow non-uniformity at the blade LE for the Blocked Cavity configuration is approximately 10% of that corresponding to uniform flow. At these operating conditions the flow field for the recirculating compressor variants is uniform.
Figure 7-9: Pressure (left column) and mass flow (right) column non-uniformity at Blade TE (top row), downstream of PS Slot location (middle row) and Blade LE (bottom row) at $\dot{m}/\dot{m}_{ref} = 0.41$ at $0.84 N_{ref}$
Figure 7-10: Pressure (left column) and mass flow (right column) non-uniformity at Blade TE (top row), downstream of PS Slot location (middle row) and Blade LE (bottom row) at $\dot{m}/\dot{m}_{ref} = 1.39$ at $1.48 N_{ref}$
7.3 Summary

In this chapter the preliminary results of the effect of the PS Cavity in attenuating the volute induced non-uniformity are assessed. The PS Outlet of the PS Cavity is blocked and therefore does not allow for flow recirculation. This results in the cavity acting solely as a pressure equalizing chamber.

It is found that the Blocked Cavity compressor is as effective as the conventional recirculating PS compressor in extending the operable range at OPs where \( \pi_c/\pi_{c,\text{ref}} < 1.7 \) \((\pi_c < 3)\). Assessment of the compressor flow at OPs where \( \pi_c/\pi_{c,\text{ref}} < 1.7 \) indicates that the attenuation of the pressure non-uniformity by the Blocked Cavity is similar to that of the recirculating compressor variants. However, no flow redistribution between passages throughout the impeller is found since mass flow is not allowed to be re-injected upstream of the wheel inlet. In the case of the PS compressor, the redistribution inside the inducer decreases the mass flow non-uniformity by up to 60% compared to that at the impeller TE.

At OPs where \( \pi_c/\pi_{c,\text{ref}} > 1.7 \) the surge mass flow for the Blocked Cavity compressor is almost identical to that of the non-ported compressor. At these operating conditions the Blocked Cavity configuration is subject to higher static pressure non-uniformity than the PS compressor. As a result, the amplitude of the mass flow non-uniformity for the Blocked Cavity configuration is approximately 10%, compared to an almost uniform flow for the PS compressor.

Through the preliminary results of this chapter, it is inferred that the role of the PS Cavity as a pressure equalizing device is not the dominant factor in setting the performance of the PS compressor at high pressure ratio OPs \((\pi_c > 3.5)\) but rather the effect of the PS flow actuations.
Chapter 8 Summary and Conclusions

This chapter provides a summary of the research approach taken to address the proposed research questions. The key findings of the thesis are then summarized followed by suggestions for future work in the context of PS compressor operation.

8.1 Summary

A computational model of both ported and non-ported variants of the research compressor used in this thesis is developed and assessed against gas stand and PIV measurements resulting in accord between the two. The analysis of the flow field inside the PS compressor and PS Cavity reveals that the main flow stream perceives the PS flow as a combination of PS flow actuations.

Assessment of the flow inside the PS reveals that up to 70% of the inlet mass flow is recirculated through the PS cavity, thus the PS is perceived as a source of mass flow by the main flow. The PS flow exits the PS Outlet in the form of four jets due to the four structural ribs inside the PS Cavity. The high velocity of the exiting flow, as a result of both the large fraction of recirculating flow and the four jet formation, indicates that the PS acts as a source of axial momentum with respect to the main inlet stream. The PS is also perceived as a source of angular momentum; the injected flow at the PS Outlet has pre-swirl as the impeller has imparted angular momentum to the PS stream. Comparison of the flow field inside the impeller between ported and non-ported compressors shows that the removal of mass flow through the PS Slot decreases the extent of the recirculation zone near the inducer shroud. This results in the reduction of the flow blockage and entropy generated inside the impeller. Thus, the PS is also perceived as a sink of mass flow at the location of the PS Slot.

To assess the effect of the PS flow actuations on the compressor performance a single passage CFD model of the compressor, which does not include the PS cavity, is used as the
computational platform. The PS flow actuations are modeled through appropriate boundary conditions at the PS Outlet and Slot locations. This approach allows for a flow actuation to be isolated and its effect on the performance individually assessed. The effect of four actuations on the compressor performance are assessed in this thesis; mass flow removal at the PS Slot, as well as mass flow, axial momentum and angular momentum injection at the PS Outlet.

A computational model of a PS compressor where the exit of the PS Cavity is blocked is developed so that flow is not allowed to recirculate between the impeller and inlet. The purpose of this configuration is to assess the role of the PS Cavity as a pressure equalizing device. The performance of the blocked cavity compressor variant is assessed against that of a PS compressor where flow is allowed to recirculate through the PS Cavity.

8.2 Conclusions

The key findings of this thesis are as follows:

- Mass flow removal is the dominant PS flow actuation in setting the PS compressor performance. Increasing the fraction of mass flow removed results in the increase of the compressor pressure ratio. The removal of mass flow results in the deceleration of the compressor main stream and reduction of the impeller flow blockage. The decrease in blockage is due to the reduction of the recirculation zone extent near the inducer shroud through the removal of the inducer tip leakage flow and is shown to increase the blade loading. The partial removal of the recirculation zone decreases the entropy generated inside the impeller compared to a non-ported compressor OP of equal outlet mass flow.

- The introduction of counter-swirl (opposite to the wheel rotation) to the PS flow at the PS Outlet increases the compressor pressure ratio and static pressure rise and decreases the efficiency. The higher pressure ratio is a result of the impeller doing more work on the flow
due to the higher inlet flow angle compared to co-swirl injection. The decrease in efficiency is due to the higher mixing loss between tip leakage and main flow. The effect of angular momentum injection on the compressor performance is shown to decrease at OPs with high flow blockage upstream of the blade LE.

- Removing and discarding the PS flow is more effective in increasing the compressor stagnation pressure ratio than recirculating it, while succumbing to no additional penalty in efficiency.

- No additional benefit from mass flow injection is found in the inducer when compared to a non-ported compressor of same mass flow.

- It is hypothesized that the attenuation of the flow non-uniformity by the PS Cavity at OPs corresponding to high pressure ratios $\pi_c > 3$ is not responsible for the increased compressor operable range, but rather the effect of the PS flow actuations.

**Rationale of PS Compressor Operation**

The operation of a PS compressor is rationalized through the combination of the PS flow actuations through the following arguments. Flow recirculation through the PS Cavity implies that the inducer works at a higher mass flow and closer to peak $\eta$ compared to the remaining compressor. This results in the decrease of the inducer recirculation zone and subsequent decrease of flow blockage and entropy generation. The effect of removing a fraction of the inducer mass flow is shown to further decrease both flow blockage and entropy generation compared to non-ported OPs of same inducer mass flow. Injection of the flow at the PS Outlet with counter-swirl further increases the compressor pressure ratio and inducer static pressure rise compared to co-swirl mass flow injection. The higher inducer static pressure forces additional flow through the PS Cavity, thus amplifying the effect of mass flow removal in increasing the stagnation pressure ratio. The combined effect of mass flow removal and counter-swirl injection results in the inducer working at an OP further away from surge compared to co-swirl mass flow
injection for equal compressor outlet mass flow. Because a larger fraction of mass flow is recirculated through the PS Cavity and the inducer is working at a higher mass flow, the flow blockage upstream of the blade LE is lower compared to co-swirl injection OPs. Thus the effectiveness of counter-swirl on increasing the inducer static pressure rise and therefore PS mass flow is extended to OPs closer to surge compared to co-swirl injection. It is therefore inferred that the synergistic effect of counter-swirl injection and mass flow removal is responsible for the larger range extension and stagnation pressure ratio compared to a PS compressor with pre-swirl in the PS flow.

8.3 Recommendations for Future Work

The results of this thesis show that the effect of the PS as a pressure equalizing device at OPs corresponding to high pressure ratios $\pi_c > 3$ is not sufficient in providing the range extension of the PS compressor. Thus there is a need for future research to address the interaction between volute tongue induced flow asymmetry and the Ported Shroud.

The benefit of mass flow removal at the ported shroud suggests the formulation and development of a shroud casing treatment that has a dominant mass flow removal characteristic. It is suggested that an effort be taken to explore such a casing treatment device.
Chapter 9  References


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Appendix A Blockage Estimation

The blockage estimation process used in this thesis follows the method developed by Khalid in [29] for the use in axial compressors based on the concept of two-dimensional boundary layer displacement thickness. The displacement thickness of a two-dimensional boundary layer \( \delta^* \) (shown in Figure 9-1) can be thought of as the distance necessary for the solid boundary to be displaced in the normal to the boundary direction so that the entire mass flow has a uniform velocity. This concept was extended by Khalid and employed in more complex flow fields such of those inside axial compressors.

![Figure 9-1: Two-dimensional boundary layer](image)

\[
\delta^* = \int_0^\delta \left(1 - \frac{u(y)}{V}\right) dy
\]

For a two-dimensional profile the main flow direction is trivial to identify. However, even in the case of flow inside a compressor the main flow direction can be determined. Figure 9-2 shows the relative flow angle \( \beta \) along the midpitch location for a crossflow plane located downstream of the impeller LE for a non-ported single passage OP for \( \dot{m}_{out}/\dot{m}_{ref} = 0.83 \) at 0.84 \( N_{ref} \). Even though the flow angle of the main velocity varies in the spanwise direction, a main flow direction
can be defined and the velocity component in the main flow direction \( u_m \), at any location can be calculated. The dashed line in Figure 9-2 shows the extrapolated main flow direction inside the defect regions. Velocity components other than the main flow component are regarded as crossflow.

![Figure 9-2: Relative flow angle for inducer crossflow plane for non-ported single passage \( \dot{m}_{out}/\dot{m}_{ref} = 0.83 \) at \( 0.84 \, N_{ref} \)](image)

![Figure 9-3: High blockage regions (red) and edge locations (green) for inducer crossflow plane](image)
Regions identified as high blockage regions are colored in red in Figure 9-3. The edge locations, whose velocity is considered to be equivalent to that of the free stream velocity in a two-dimensional boundary layer, are colored in green. The displacement thickness associated with each defect region is calculated using the core flow velocity $V_e$ of the edge location nearest to each defect region. The total area that corresponds to blockage is calculated through:

$$A_b = \int \left(1 - \frac{\rho u m}{\rho e V_e}\right) dA$$

Equation 9-2

Up to this point, it has been presumed that the extent of each defect area is known. However, a criterion identifying the extent of each defect region must be established. Khalid proposed a criterion for identifying defect regions based on gradients of the main flow velocity component $\nabla(\rho u m)$, since defect regions demonstrated higher main flow velocity gradients [29]. More specifically, the magnitude of the two components of $\nabla(\rho u m)$ which lie in the plane for which the blockage is to be calculated, are used as an appropriate scalar to detect regions of high blockage. In this thesis, since blockage is calculated across planes in the streamwise direction, the magnitude of the components of $\nabla(\rho u m)$ in the spanwise and circumferential direction is used:

$$|\nabla(\rho u m)|_{\text{span, } \theta} = \sqrt{(\nabla_{\text{span}}(\rho u m))^2 + (\nabla_{\theta}(\rho u m))^2}$$

Equation 9-3

The magnitude of the main flow gradient in the directions normal to the crossflow plane $|\nabla(\rho u m)|_{\text{span, } \theta}$, normalized by $\rho_{\text{wheel inlet}u_{\text{axial, wheel inlet}}/c}$, where $c$ is the blade chord, is used as the criterion for identifying regions of high blockage. The core flow region roughly corresponds to regions where $|\nabla(\rho u m)|_{\text{span, } \theta}/(\rho_{\text{wheel inlet}u_{\text{axial, wheel inlet}}/c) < 15$. The value of this criterion is set so that identified defect regions are in accordance with the defect regions identified through the calculation of the main flow angle at the midpitch location of each crossflow plane. The left picture in Figure 9-4 shows a contour plot of $|\nabla(\rho u m)|_{\text{span, } \theta}/(\rho_{\text{wheel inlet}u_{\text{axial, wheel inlet}}/c)$, where regions of high blockage are shown in
The right picture shows the local entropy generation $\dot{S}_{\text{visc}}$ for the same crossflow plane. A direct comparison between the two contours indicates that regions of high blockage match with regions of high $\dot{S}_{\text{visc}}$, which provides additional support in favor of the selection of the blockage cutoff value.

![Figure 9-4: Contours of $|\nabla (\rho u_m)|_{\text{span,}P}/(\rho_{\text{wheel inlet}} u_{\text{axial, wheel inlet}}/e)$ (left) indicate regions of high blockage correspond to regions of high $\dot{S}_{\text{visc}}$ (right) for inducer crossflow plane for $\dot{m}_{\text{out}}/\dot{m}_{\text{ref}} = 0.83$ at $0.84 N_{\text{ref}}$]

Khalid developed this blockage estimation method based on the geometry of axial compressors. In the case of this thesis, the compressor used is centrifugal. In centrifugal compressors the Coriolis force has an impact on the flow field inside the exducer. This mechanism can be demonstrated by assuming a two-dimensional channel of constant width $W$ rotating around the $z$ axis with angular velocity $\Omega$ as shown in Figure 9-5 [25]. The channel draws inviscid fluid from a reservoir in which the flow is irrotational in the stationary frame. Assuming that the length/width ratio of the channel is large enough such that variations along the channel length can be neglected compared to those across the channel, it can be approximated that $\frac{\partial}{\partial x} = 0$ in the relative frame $xyz$. From the use of Kelvin’s Theorem it can be shown that the velocity in the
$x$ direction in the relative frame $w_x$ can be described by the following (the channel width spans from $y = -W/2$ to $y = +W/2$):

$$w_x = 2\Omega y + \bar{w}_x$$  \hspace{1cm} \text{Equation 9-4}

The flow field inside the relative passage can be considered to be composed of a uniform background flow $\bar{w}_x$ and a uniform shear of $2\Omega$ [25].

![Figure 9-5: Two-dimensional inviscid flow in rotating channel where coordinate system $xyz$ in relative frame](image)

This flow mechanism is considered to be a manifestation of the relative eddy [25] and should not be considered responsible for any changes in the flow field blockage. Therefore, its contribution must be taken into account when estimating flow blockage in exducer crossflow plains especially when comparing exducer blockage between cases at different speedlines. In the case of crossflow planes inside the exducer, the shear component due to the Coriolis force $2\Omega r\theta\sin\varphi$ is subtracted from the meridional velocity. With respect to the previous equation, $r\theta$ is the passage width in the circumferential direction and $\varphi$ is the angle between the meridional direction of the crossflow plane and the axis of rotation. For inducer crossflow planes the shear component due to the Coriolis force does not impact the flow field since the radial velocity of the flow is negligible.
Appendix B Separation of Static Pressure Rise

A sensitivity analysis method for dissecting the impeller static pressure rise is presented in Chapter 4.3. The added benefit of such an approach is that the underlying flow mechanisms can be identified and their relative importance evaluated. In this Appendix chapter, the $C_p$ values calculated through the sensitivity analysis method are compared against the CFD generated values. Since the purpose of this investigation is to assess the validity of calculating the $C_p$ through the sum of its individual components, the purpose of this investigation is to compare the total $C_p$ calculated through the sensitivity analysis and compare it to the CFD generated values.

This assessment is done initially for a simple case of removing a fraction of the mass flowing through a straight duct. Finally, a comparison between CFD and analytical based values of $C_p$ will be shown for cases of removing mass flow at the PS Slot location of the single passage PS Actuation Model.

**B.1 Mass Removal of Flow Through a Duct**

In the analysis presented in Chapter 4.3, it is assumed that while the pressure acts on the geometric area $A$ of the crossflow plane, the mass flow corresponds to the effective area $A_{eff}$. This approach is considered to be acceptable since the method of calculating flow blockage is based on the concept of a boundary layer displacement thickness. Nevertheless, the approach of dissecting static pressure is applied to a less complex flow through a straight duct in order to assess its validity.
With respect to Figure 9-6, the duct inlet flow consists of two streams of different stagnation pressure. The low stagnation pressure stream in this analysis is assumed to model the low momentum fluid present near the shroud of the compressor, while the high stagnation stream plays the role of the main core flow. The total flow blockage is calculated through the displacement thickness Equation 9-1, where the free stream velocity is taken to be that of the high stagnation pressure stream. The mass flow fraction of the low momentum stream with respect to the total mass flow is \( a = \frac{\dot{m}_{1A}}{\dot{m}_1} \), while the mass flow removed through an opening in the duct surface is given by \( \delta = \frac{d\dot{m}}{\dot{m}_{1A}} \). For values of \( \delta < 1 \), only a portion of the defect flow is removed through the opening, while for \( \delta \geq 1 \), the entire mass flow defect and part of the core flow is also removed. The area ratio of the two streams is given by \( \beta = \frac{A_{1A}}{A_1} \).

The expression for the dissection of the static pressure rise into its components is a simplified form of Equation 4-9:

\[
\frac{p_2 - p_1}{1/2 \rho u_{1B}^2} = \frac{1}{1/2 \rho u_{1B}^2} \left[ \left(-2 - \frac{\dot{m}}{\rho A_{eff} A} - \frac{u_{inj}}{A} \right) d\dot{m} + \frac{\dot{m}^2}{A A_{eff}} \frac{d\rho}{\rho^2} + \frac{\dot{m}^2}{\rho A_{eff}} dA_{eff} \right]
\]  

Equation 9-5
The duct geometry shown in is modeled and simulated with CFD using inviscid flow for two values of the removed mass flow fraction $\delta = 0.5$ and $\delta = 1.5$. Comparison of the static pressure coefficient $C_p = \frac{P_2 - P_0}{\rho u_0^2}$ calculated through the CFD and the sensitivity analysis approach is shown in Figure 9-7. The results show that the sensitivity analysis approach overestimates the $C_p$ rise by approximately 20% and 10% for $\delta = 0.5$ and $\delta = 1.5$ respectively. However, the trends observed in the CFD are captured and considering that this is a linearized sensitivity analysis approach it is assumed that this model is able to capture the mechanisms responsible for the static pressure rise. Figure 9-8 shows the breakdown of the duct $C_p$ into its components for $\delta = 1.5$. Approximately 83% of the total $C_p$ rise is due to the effect of mass removal and subsequent increase of the diffusion of the main core stream, while 15% is due to decrease of the flow blockage.
Figure 9-8: Breakdown of $C_p$ rise across duct with mass removal into components indicates diffusion of primary flow dominant flow mechanism for increasing static pressure

B.2 Mass Removal at PS Slot

After assessing the results of the static pressure rise sensitivity analysis versus the CFD generated results for the simple case of mass removal of flow through a duct, the next step is to compare the results for a more complex flow case. Accordingly, the $C_p$ rise inside the impeller for the case of removing mass flow through the PS Slot is assessed using Equation 4-10. The impeller passage is split into inducer and exducer regions. The CFD and sensitivity analysis based generated $C_p$ values are shown for cases working at both low and high speedline where the inducer is working at conditions near peak $\eta$. Specifically, at the low speedline $0.84 N_{ref}$ the inducer mass flow corresponds to $\dot{m}_{ind}/\dot{m}_{ref} = 0.83$ while 30% and 50% of the $\dot{m}_{ind}$ is removed at the PS Slot Figure 9-9. At the high speedline $1.48 N_{ref}$ the inducer mass flow corresponds to $\dot{m}_{ind}/\dot{m}_{ref} = 1.6$ while 20% and 30% of the inducer flow is removed Figure 9-10. The results indicate that the sensitivity analysis is able to accurately capture the $C_p$ rise in both inducer and exducer regions for both speedlines.
Figure 9-9: CFD vs. analytically calculated $C_p$ values for $\dot{m}_{\text{ind}}/\dot{m}_{\text{ref}} = 0.83$ at $0.84N_{\text{ref}}$ for inducer (left) and exducer (right).

Figure 9-10: CFD vs. analytically calculated $C_p$ values for $\dot{m}_{\text{ind}}/\dot{m}_{\text{ref}} = 1.6$ at $1.48N_{\text{ref}}$ for inducer (left) and exducer (right).
Appendix C Assessment of Angular Momentum Injection at PS Outlet and Bulk Swirl at the Compressor Inlet

The use of Inlet Guide Vanes (IGVs) to impart pre-swirl to the inlet flow and decrease the blade relative inlet flow angle has been documented in literature [35, 36]. Adding pre-swirl to the inlet flow results in the decrease of the compressor pressure ratio, as well as the surge and choke mass flow. Figure 9-11 shows gas stand measurements of \( \pi_c \) for a non-ported centrifugal compressor (not the C239A) in black and for the same compressor with IGVs at the inlet set to impart pre-swirl of \( \alpha = 35^\circ \) in red.

![Figure 9-11: Gas stand measurements of \( \pi_c \) showing decrease of surge mass flow but overall decrease of compressor operable range for non-ported compressor with IGVs set to \( \alpha = 35^\circ \) (red) compared to non-ported without IGVs [13]](image-url)
It is shown in Section 5.3.2 that the introduction of $\alpha = 50^\circ$ co-swirl at the PS Outlet at the near peak $\eta$ OP at $1.16 \, N_{ref}$ results in the decrease of the compressor pressure ratio by approximately 6% compared to an OP where an equal fraction of mass flow is injected at $\alpha = 0^\circ$. Even though the gas stand measurements shown in Figure 9-11 do not correspond to the compressor used in this study, it is postulated that the decrease in pressure ratio may be used as a metric to assess the effect of introducing co-swirl to the inlet flow. The values of pressure ratio measured at 0.70 $\bar{N}_{ref}$ for the compressor in Figure 9-11 correspond approximately to the values computed at 1.16 $N_{ref}$ for the C239A compressor. With reference to Figure 9-11 the pressure ratio decrease associated with the introduction of $\alpha = 35^\circ$ between OPs A (no IGVs) and B is approximately 20%. The decrease measured through the gas stand data for $\alpha = 35^\circ$ is more than three times higher than that computed when adding $\alpha = 50^\circ$ to the PS flow at the PS Outlet.

To address the difference in behavior, a single passage non-ported compressor CFD model (shown in Figure 9-12) is used where the effect of the IGVs in adding swirl to the inlet flow is modeled by specifying a constant swirl angle at the inlet. The CFD model does not contain IGVs and no flow is injected at the PS Outlet. The swirl angle of the inlet flow is set to $\alpha = \pm 50^\circ$ and three OPs, described in Table 9-1, at the $1.16 \, N_{ref}$ speedline are assessed.

![Figure 9-12: Sketch of single passage non-ported compressor model indicating where the inlet flow angle is specified](image-url)
Table 9-1: Baseline OPs used to assess influence of swirl angle of the inlet flow on compressor performance with at 1.16 $N_{ref}$

<table>
<thead>
<tr>
<th>Speedline</th>
<th>$\dot{m}<em>{out}/\dot{m}</em>{ref}$</th>
<th>$\dot{m}<em>{out}/(\dot{m}</em>{out})_{\text{peak} \eta}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.16 $N_{ref}$</td>
<td>1.02</td>
<td>0.84</td>
</tr>
<tr>
<td>1.16 $N_{ref}$</td>
<td>0.83</td>
<td>0.69</td>
</tr>
<tr>
<td>1.16 $N_{ref}$</td>
<td>0.63</td>
<td>0.52</td>
</tr>
</tbody>
</table>

Figure 9-13: Adding co-swirl to the inlet flow (dashed red) results in the larger decrease of $\pi_c$ (left) and increase of $\eta$ (right) compared to $a = 0^\circ$ (dashed blue) than adding co-swirl to the PS flow (solid red) exiting the PS Outlet at 1.16 $N_{ref}$

Figure 9-13 shows the compressor pressure ratio and efficiency for OPs described in Table 9-1 where the flow angle at the inlet is set to $a = 0^\circ$ (dashed blue curve), $a = 50^\circ$ (dashed red) and $a = -50^\circ$ (dashed green). The solid lines correspond to OPs where a mass flow fraction of $\varphi_{PS\text{Outlet}} = \frac{\dot{m}_{PS\text{Outlet}}}{\dot{m}_{out}} = 0.4$ is injected at the PS Outlet at $a = 0, 50^\circ$ and $-50^\circ$. The introduction of $a = 50^\circ$ to the inlet flow (dashed red) results in the decrease of the $\pi_c$ by approximately 12% compared to the baseline $a = 0^\circ$ OP (dashed blue) at $\dot{m}_{out}/\dot{m}_{ref} = 1.02$. For the case with co-
swirl injection at the PS Outlet, the decrease in $\pi_c$ between mass flow injection at $\alpha = 50^\circ$ (red) and $\alpha = 0^\circ$ (blue) is approximately 4%. The OPs corresponding to pre-swirl at the inlet have a higher $\eta$ compared to the baseline non-ported OPs by up to five points. However, it is expected that including the IGVs in the computational model would lead to lower $\eta$ and $\pi_c$ due to the additional loss associated with their operation.

Figure 9-14: Introduction of counter-swirl to the inlet flow (dashed green) results in higher $T_{t,\text{outlet}}/T_{t,\text{inlet}}$ across the impeller compared to counter-swirl mass injection at the PS Outlet with $\varphi_{PS\text{Outlet}} = 0.4$ (solid green) at $1.16 \, N_{\text{ref}}$; the opposite is shown when imparting co-swirl.

The effect of adding counter-swirl $\alpha = -50^\circ$ to the inlet flow (dashed green) results in the decrease of $\pi_c$ by 3% compared to $\alpha = 0^\circ$ at the $\dot{m}_{\text{out}}/\dot{m}_{\text{ref}} = 1.02$ OP, while an increase of approximately 1.5% is shown for the $\alpha = -50^\circ$ at the PS Outlet OP. The difference in trend is due to the larger efficiency penalty associated with introducing counter-swirl to the inlet flow as compared to imparting counter-swirl on the flow at the PS Outlet. This is confirmed when assessing the $T_{t,\text{outlet}}/T_{t,\text{inlet}}$ for OPs with counter-swirl at the inlet and PS Outlet, shown in
Figure 9-14, indicating that more work is added to the flow for OPs with counter-swirl at the inlet. For OPs with co-swirl, less work is added to the flow when swirl is imparted on the inlet compared to the PS flow. It is therefore inferred that the compressor work input is more responsive to equal $\Delta \alpha$ at the inlet than at the PS Outlet with $\varphi_{PS_{out}} = \frac{\dot{m}_{PS_{out}}}{\dot{m}_{out}} = 0.4$.

The larger differences in $T_{t, outlet}/T_{t, inlet}$ between OPs with $\alpha = 50^\circ$ and $\alpha = -50^\circ$ at the inlet compared to $\alpha = 50^\circ$ and $\alpha = -50^\circ$ at the PS Outlet flow is a result of the larger difference in the relative inlet flow angle at the impeller LE, $\beta_1$. The $\beta_1$ profiles for OPs corresponding to angular momentum injection at the PS Outlet at $\dot{m}_{out}/\dot{m}_{ref} = 1.02 \,(\dot{m}_{out}/(\dot{m}_{out})_{peak \eta = 0.84})$ at 1.16 $N_{ref}$ are shown in the left chart in Figure 9-15 and those for OPs with an inlet specified swirl in the right chart. The $\beta_1$ profiles of counter and co-swirl OPs at the PS Outlet are almost identical for the bottom half of the passage and the maximum $\Delta \beta_1 = 9^\circ$ is found at 70% of the span. For OPs with $\alpha = 50^\circ$ and $\alpha = -50^\circ$ at the inlet $\Delta \beta_1$ is found to be approximately 16$^\circ$ between span values of 45% and 70%. The $\beta_1$ profiles for the near surge $\dot{m}_{out}/\dot{m}_{ref} = 0.63 \,(\dot{m}_{out}/(\dot{m}_{out})_{peak \eta = 0.52})$ OP with angular momentum injection at the PS Outlet is shown in the left chart in Figure 9-16, while those where swirl is specified at the inlet are shown in the right. The maximum $\Delta \beta_1$ between $\alpha = 50^\circ$ and $\alpha = -50^\circ$ at the PS Outlet is less than 5$^\circ$ at the midspan, while $\Delta \beta_1 > 10^\circ$ between span values of 25% and 50% when swirl is specified at the inlet. The recirculation zone at the inducer shroud is responsible for the backflow at span>80% for the OPs at the near surge OP.
Figure 9-15: Differences in the impeller relative inlet angle $\beta_1$ between OPs with $\alpha = 50^\circ$ and $\alpha = -50^\circ$ are higher when the swirl angle is specified at the compressor inlet (right) compared to swirling the flow at the PS Outlet for $m_{out}/m_{ref} = 1.02$ at $1.16 \, N_{ref}$

![Diagram showing differences in impeller relative inlet angle $\beta_1$ between OPs with $\alpha = 50^\circ$ and $\alpha = -50^\circ$.]

Figure 9-16: Differences in the impeller relative inlet angle $\beta_1$ between OPs with $\alpha = 50^\circ$ and $\alpha = -50^\circ$ are higher when the swirl angle is specified at the compressor inlet (right) compared to swirling the flow at the PS Outlet for near surge $m_{out}/m_{ref} = 0.63$ OP at $1.16 \, N_{ref}$

![Diagram showing differences in impeller relative inlet angle $\beta_1$ between OPs with $\alpha = 50^\circ$ and $\alpha = -50^\circ$.]

In conclusion, the compressor performance is more responsive to equal $\Delta \alpha$ specified at the inlet than at the PS Outlet. This is because differences in blade inlet flow angle between OPs where counter and co-swirl is specified at the inlet are larger across a larger fraction of the passage span compared to OPs where the swirl angle of the flow exiting the PS flow is set.