COMPUTATIONAL INVESTIGATION OF THE EFFECTS OF CASING TREATMENTS ON THE PERFORMANCE OF A TURBOFAN

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ABSTRACT

A computational survey focused on modifications to the casing near the rotor blade tip is carried out for the purpose of enhancing the performance and increasing the stall margin of a model turbofan stage in transonic operating conditions. The study is divided into three phases. During the first phase two types of casing treatments, inward protruding rings and circumferential grooves, were tested with relatively coarse grids. In the second phase, a grid resolution study is carried out with the results from this phase influencing the choices for the third stage. In the third phase, a comprehensive study is performed to examine the near-stall effects and stall-related behavior of the baseline case and a series of circumferential groove configurations.
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LIST OF ABBREVIATIONS

SS: Suction sides of rotor blades
PS: Pressure sides of rotor blades
SW: Smooth wall configuration
M5G: Middle five grooves configuration
SGa - SGh: Single groove configurations, each has its groove located at a to h, respectively
TABLE OF SYMBOLS

$R$ : Gas constant

$E_c$ : Eckert number

$\gamma$ : Ratio of specific heats

$Pr$ : Prandtl number. $Pr = \nu/\alpha = (\text{momentum diffusivity})/(\text{thermal diffusivity})$

where $\nu = \mu/\rho$ and $\alpha = k/\rho C_p$

$d_{tip}$ = tip gap distance

$d_{land}$ = width of land separating grooves in rotor casing

$\eta_{ad}$ = adiabatic efficiency

$p_{back}$ = back pressure specified at exit plane of computational domain

CFL = Courant-Friedrich-Lax parameter

$\bar{U}$ : The average velocity of rotor blades

$V_x$ : Axial velocity component, normalized by $\bar{U}$

$V_\theta$ : Theta velocity component (relative spanwise velocity), normalized by $\bar{U}$

$V_r$ : Radial velocity component, normalized by $\bar{U}$
CHAPTER 1
INTRODUCTION

1.1 Overview

Pumping devices of any sort, be they fans, compressors, or pumps, take mechanical shaft work as an input and transfer this to a working fluid, thereby increasing the total enthalpy of the fluid from its inlet value when entering the device to its exit value upon leaving. As the pressure ratio is always greater than 1, any device of this sort is operating in an environment of an adverse pressure gradient. At some point for a given shaft speed (i.e., for a given input of mechanical work), the pressure gradient becomes excessive, and the flow field begins to break down (Fig. 1.1). It is observed that the operation range decreases as the shaft speed increases. The point at which this happens for a given machine serves as a demarcation line beyond which the machine enters an unstable operating region. For fans and compressors in air-breathing propulsion engines, the instabilities, which limit the pressure ratio available from design, are typically manifested as varying degrees of rotating stall or surge. Stall in terms of a single blade usually means separation of the incoming flow, which often occurs on the suction side. The interactions between multiple blades, however, make the overall behavior of the stall of a compressor much more complicated.
A nondimensional parameter, $B$, is defined by Greitzer (1976) to be an indicator of the mode of compressor instability:

$$B = \frac{U}{2\omega L_c},$$

where $U$ is mean rotor velocity, $\omega$ is Helmholtz resonator frequency and $L_c$ is the effective length of equivalent duct. When $B$ is below a critical value, rotating stall occurs; otherwise the instability would transform to surge. Rotating stall can be described as a rotating annulus of severely retarded distortions of circumferential flow (Greitzer, 1981), which will cause serious vibratory stresses in the blading of compressors. For reasons not clearly understood, separation happens to a group of airfoils when rotating stall occurs, followed by a blockage which diverts the incoming flow. As a consequence, the incidence angle is increased near the pressure side and is decreased toward the suction side. The separation will propagate around the annulus but at a fraction (typically $\frac{1}{4}$ to $\frac{1}{2}$).
of the rotor speed (Emmons et al. (1955)). Surge exhibits large oscillations of mass flow rate, which leads to serious results such as inlet overpressure, and high blade and casing stress levels.

Aerodynamic stability of fans and compressors in air-breathing engines has been a concern since these devices have been in existence and will likely continue to be for some time. One of the criteria for the stability of a system is its capability to recover from a small perturbation. To this extent, rotating stall is severe since the only way to come out of it may be to significantly decrease rotational speed, resulting in a loss in pressure rise, and consequently excessive internal temperature on blades and even in the burner and turbine (Greitzer, 1976). To investigate the fluid dynamics of the pumping components (i.e., the fans and compressors themselves), especially as related to stall inception, stall control, and extension of the operating range to improve the stall/surge margin, is thus significant. A recent numerical investigation can be found in Niazi (2000).

1.2 The Effects of Tip-leakage Flow

Significant research has been carried out to investigate the onset of stall, and more often than not, the tip clearance flows were found guilty (Moore (1982), Koch (1981), Freeman (1985)). The tip clearance leakage is driven by the pressure difference between the pressure and suction sides of blades which starts as a jet with a velocity equal to the main flow but with different direction. Since the axial velocity is lower in the boundary layer than in the inviscid main flow, the rotor does more work on the boundary layer and tends to transfer part of the low energy fluid from it to the main passage in the form of tip-leakage flow. A portion of the kinetic energy of the tip-leakage flow will be dissipated
when it is mixed with the main stream, leading to an increase in entropy. The tip-leakage flow is a source of loss in general in that energy is wasted in “feeding” the tip vortex emanating from the leading edge. The interaction between the tip clearance flow and the streamwise core flow below it, due to their different directions, forces the tip clearance to roll up into a tip vortex (Fig. 1.2). The loss increases as the angle between clearance flow and mainstream flow increases (Storer & Cumpsty, 1991). Span-wise flow is also induced on the suction side of the rotor blade; this, too, represents a loss in energy that would have otherwise been transferred to the working fluid.

![Diagram of tip clearance flow](image)

**Figure 1.2**  Basic features of tip clearance flow (Vo, 2001)

The shock waves existing in transonic compressors cannot be ignored since they will interact with the tip-leakage flows, and this interaction will in turn affect the stability. Sellin *et al.* (1993) found that a large zone of blockage is created as the tip vortex passes through the leading edge shock. The tip vortex will continue to grow circumferentially and radially. Puterbaugh and Brendel (1997) proposed that the interaction between the clearance flow and the shock can be viewed as an inviscid problem as it is fundamentally the result of the change in momentum brought about by the shock-induced pressure rise.
Similar statements regarding the contribution to blockage from the expansion of vortex can be found in Adamczyk et al. (1993), and Khalid et al. (1999). Hah and Wennerstrom (1991) conducted detailed numerical and experimental analysis for a three-dimensional flow filed inside a transonic rotor, they confirmed that the strong triple interaction between passage shock, blade boundary layer and tip-leakage flow causes inefficiency in supersonic flow zone of transonic compressor.

Some researchers believe that the leading edge tip-leakage vortex will breakdown when it passes through normal shocks and then initiates stall (Hoffman and Ballman (2003), Yamada et al. (2003), etc.), while Hah and Rabe (2004) found that the tip clearance vortex does not break down for oblique shocks, but in their case it is instead the oscillation induced by the triple interaction that possibly causes the stall. Based on this, Chen et al. (2006) concluded that when the tip gap is small, stall is found to be initiated from the bursting of trailing edge separation/vortex, which is strengthened by the blade tip-leakage flow; when the tip gap is large enough, the leading edge blade tip-leakage vortex breakdown will trigger the stall instead.

The zone of blockage caused by the tip-leakage flow and expanded by the shocks can be quantified by some means. By assuming that the stall process occurs on the end-wall, Koch (1981) correlates stall to the wall boundary layers growth for compressors with moderate to high solidities. When wall stall occurs, the low-momentum region blocks the way that incoming flow passes through the passage and thus reduces the pressure rise; as a result, effective core flow area is reduced which stimulates the stall in a premature phase. The reduction of effective main flow is often referred to as end-wall blockage. Similar to the calculation of the displacement thickness associated with
boundary layers, Khalid et al. (1999) developed a new method of quantifying the blockage based on the growth of the region of low total pressure fluid associated with the tip-leakage:

$$\mathcal{A}_b = \iint \left( 1 - \frac{\rho u_m}{\rho_e U_e} \right) dA,$$

where $u_m$ is the component of velocity resolved in the mainflow direction, $\rho_e$ and $U_e$ are the velocity and density at the edge of blockage area, respectively. Usually a rapid growth of blockage is enough to initiate instability. However, as will be discussed in the next section, the removal of high-blockage flow might not postpone the stall process (Lee and Greitzer (1989), Houghton and Day (2011)).

In summary, as pressure ratio increases and mass flow rate decreases, the likelihood of a breakdown in the flow field, in the form of separation and reverse flow on the suction side of the rotor blade, would emanate in the tip-gap region. To alleviate the adverse effects of the tip clearance leakage and increase its streamwise momentum is thus important not only with-respect-to reducing energy loss, but also for stability concerns. A good discussion of tip-leakage losses and their effects may be found in Denton (1993).

### 1.3 Casing Treatments and Circumferential Grooves

The potential to improve stall margin by casing treatments depends on their ability to alter the effect of tip clearance flow. It has been shown that wall stall appears to take place where casing treatments are effective in stall margin improvement (Greitzer (1979)), where wall stall is closely related to the tip-leakage flow. As shown in Fig. 1.3, a significant part of the blade span is covered by a large wake from the suction side of the
blade when blade stall occurs; in contrast, the wall stall is produced when there is a wide region of low velocity fluid near the pressure side of the blade due to the interaction between the tip clearance vortex and the passage shock. Fig. 1.4 clearly shows the influence of different tip clearance sizes on static pressure rise capability, where SW stands for smooth wall and CT represents cases with casing treatment (axially skewed slot in this case). From the work done by previous researchers (Freeman (1985), Adamczyk et al. (1993) and Copenhaver et al. (1993)), Thompson (1997) concluded that some rotor designs can utilize the interactions of tip clearance flow to improve performance better than the others. The way that the rotor designs would affect or make use of the interaction is hence the main problem.

Figure 1.3 Illustration of wall stall and blade stall processes (Greitzer et al., 1979)
Figure 1.4  Static pressure rise characteristics at various tip clearances for the smoothed wall and treated builds; \( \phi = \text{flow coefficient} \ C_{x/\mu_M} \),
\[
\psi_{s,s} = \Delta P_s / \frac{1}{2} \rho U_m^2
\]
, nondimensional static pressure rise across rotor
(Smith & Cumpsty, 1984)

A summary of various types of end-wall treatments for the purpose of increasing stall margin was presented by Hathaway (2007). One method shown to be effective for increasing compressor stability is that of circumferential grooves cut into the casing and located radially outward from the rotor blade tip.

Despite the inherent difficulty in understanding the mechanism of grooves configurations to delay stall, there has been considerable progress in decoding it. The
difficulty comes from the lack of fundamental understanding of the stall process of the SW baseline configuration, which was attempted to be explained in the previous section. Prince (1974) identified four mechanisms of effects of circumferential grooves for the improvement of stall margin:

1. Two-dimensional boundary layer effects.
2. Tangential velocity effects.
3. End-wall boundary layer effects.
4. Segmentation of the tip-clearance vortex.

The radial velocity components of the tip clearance vortex play a significant role in its influence. By numerical means, Shabbir and Adamczyk (2005) showed that casing grooves enable axial momentum to transport radially, which augment the net axial shear stress force acting at the casing, thus increase the tolerance of near-casing flow as to balance with increased axial pressure (Fig. 1.5). Nolan (2005) hypothesized that the tip clearance vortex shifted downstream as a result of the presence of a circumferential casing groove; later, Hanley (2010) found that the tip clearance vortex location is shifted downstream as the depth of a casing groove increases, but the quantity of shifting does not directly relate to the change in stall margin. From his research Hanley (2010) also questioned the link between the stall margin improvement and the change in radial transport of streamwise momentum. Houghton (2011) found on his subsonic machine that the stall pattern is not affected by the presence of casing grooves, even when the groove alters the development and trajectory of the tip-leakage vortex.
Rabe and Hah (2002) showed, from both experimental and numerical investigations on a transonic axial compressor, that the grooves increase the stall margin by reducing the flow incidence angle on the pressure side of the leading edge (Fig. 1.6). Following the mechanisms identified by Prince (1974), they took insight into the depth of the grooves and showed that circumferential groove treatment with shallow grooves can be applied successfully; however, it is not the case for subsonic compressors according to traditional assumption. To question the proposal that segmentation of the tip-clearance vortex is the cause of the improved stall margin with circumferential grooves, Hall et al. (1996) argued that although such phenomena always exists, circumferential grooves are not always effective.
The controversy about the optimal location of grooves can be found in the works of many researchers (Shabbir and Adamczyk (2005), Vo et al. (2001), Lu et al. (2006), Müller (2007)). By numerically investigating a transonic axial compressor, Chen et al. (2006) concluded that the grooves in mid-chord are more effective when the tip gap is small, while they work well near the leading edge for large tip gap configuration. A more recent paper by Houghton and Day (2011) offers both experimental and computational investigations into the effectiveness of casing grooves for enhancing compressor stability (Fig. 1.7). Single grooves at 8% and 50% axial chords aft of the blade leading edge were found to have the maximum stall margin improvement, and entropy gains (efficiency loss) reduce when the groove is moved from the leading edge toward mid-chord. Since only the groove at 8% chord of the two optimal grooves changes the early growth and trajectory of the tip-leakage vortex, the altering is not a necessary requirement for stall margin improvement. By looking into the core flow effects, they showed that near-casing
flow total pressure, or the velocity of the near-casing flow is most important for explaining the performance of casing grooves; however, there is no direct link between stall margin and the effect grooves have on outflow blockage, blade loading, or the casing static pressure field. They also showed the effects of groove depth on the stall margin improvement. Later they (Houghton and Day (2010)) showed that less margin would be gained from multiple grooves than the sum of the gains from individual grooves.

Figure 1.7  Plots of stall margin improvements (SMI) and maximum efficiency improvement (MEI) generated by the single groove as it is moved aft from the leading edge, on a subsonic compressor (Houghton & Day, 2011)

Other changes brought by the grooves on the flow field are also investigated by researchers, such as suction of the low total pressure and energizing of the tip-leakage flow (Crook et al. (1993)), etc., are also nominated as the reflections / mechanisms of stall margin improvement, and thus the indicators of stall.

It has been found by some researchers that an improvement in stall margin comes with an inevitable loss in efficiency (Fujita, 1984), but other researchers claimed to have either achieved zero sacrifice in efficiency (with circumferential groove casing treatment,
Prince (1974), or even got better efficiency at the same time (on a low-speed machine, Nolan (2005), Nezym (2004)).

Two different modes of stall, short length-scale or “spike” and long length-scale or “modal” stall, are demonstrated to be critical in deciding whether the grooves would enhance the stall margin (Fig. 1.8). The two modes of stall can be changed by the size of tip clearance or by the rotor tip incidence (Day (1993), Camp and Day (1998)). From their research, Camp and Day (1998) concluded that spike stall can be identified when the slope of stagnation-to-static pressure rise characteristic is negative. This was verified by Vo (2001). This conclusion also agrees with the previous results obtained by Smith and Cumpsty (1984) (Fig. 1.4). Vo (2001) then established a criteria for spike stall to occur, which simply states that flow shall move into the adjacent blade passage around the leading and trailing edges of the blade (Fig. 1.9), and that zero mass flow across the pitch at the trailing edge blade tip shall occur. As to modal stall inception, the Moore-Greitzer model (Moore, 1982) is widely accepted to describe the phenomenon. Practically, circumferential grooves can improve the performance of some compressors, but may not improve, or even aggravate the stall margin for others. For example, Houghton and Day (2010) showed that spike-type compressors could achieve the greatest stall margin improvement. They also identified two kinds of modal stall inception: destabilising modes and stable modes. This type of research is necessary in that it will advance the understanding of the stall mechanism; however, it is out of the scope of the current topic. Readers who are interested may start from Day (1993), Camp and Day (1998), Gong et al. (1998), Nolan (2005), Hah et al. (2006), Vo (2001), etc. The fan stage in this study has a small tip clearance, and the basic results of resulting stall patterns will
be discussed in section 5.5.

Figure 1.8 Different SMI by casing treatments because of changing the rotor tip incidence and resulting stall modes (Houghton & Day, 2010)

Figure 1.9 Simultaneous flow events in spike formation (Vo, 2001)

The purpose of this thesis, rather than to conduct comprehensive tests for many different casing treatments without any understanding of basic fluid mechanic features, is to computationally investigate the effect of casing grooves in detail and other tip region treatments for possible stability enhancement and improvement in performance of a fan
stage. This will be accomplished through the use of an in-house flow solver referred to as *Tenasi*.

The initial idea prompting this particular investigation was inspired by the use of so-called “rub strips”, which are built into the casing wall but actually contact the blade tip. The material of the rub strip is weaker than that of the blade, such that the blade will wear away the material to the point that the tip gap will be essentially zero. The tip-leakage flow, therefore, should be very nearly zero, such that tip-leakage losses are either eliminated or greatly reduced. The best efficiency with zero tip clearance was also confirmed by other researchers (Moore (1982), Freeman (1985), Nolan (2005)). Instead of trying an actual rub strip configuration, however, a test was carried out for a virtual channel in the casing by extending continuous rings inward from the casing surface at locations just upstream of the leading edge and just downstream of the trailing edge of the rotor blade. Two additional configurations were then tested; one being the leading-edge ring by itself, the other being the trailing-edge ring by itself (Lin *et al.* (2012)). It was then decided to try the grooves extending into the casing since some success with this method had been seen for core compressor machines (see the summary by Hathaway (2007)).

The fan stage to be investigated is a scale model from a commercial turbofan engine, known as the SDT2-R4. This model stage served as a baseline for aero-acoustic testing (Source Diagnostic Test) at NASA-Glenn. The nominal operating design point has a speed of 12,657 RPM (for a tip Mach number, $M_{\text{tip}}$, of approximately 1.1), with a total pressure ratio of 1.47 and a mass flow rate of 100.5 lbm/sec. Details of the aero-acoustic tests are found in Woodward and Hughes (2004) and for the performance testing of the
SDT2-R4 stage in Hughes (2002). The rotor has 22 blades and the stator has 54 vanes. For the simulations in this paper, an extra stator vane is added, so as to allow for the use of a partial wheel sector containing 2 blades and 5 vanes. A recent paper by Hyams et al. (2011) demonstrates that the addition of the extra vane has a minimal effect on the simulated fan performance, since the results compared very well with experiment. The next chapter of the thesis will provide a brief description of the flow solver and its capabilities, as well as the computing resources required for this research. The results and discussion of the various treatments will then be presented followed by the conclusions. Illuminated by the previous studies, the results will be illustrated mainly in the aspects of leading edge vortices, near-casing total pressure and velocity field. The results part will be divided into two chapters, between which there will be a chapter (Chapter 4) studying the effect of the sensitivity of grids on CFD simulation. The initial results from relatively coarse grids will be presented in Chapter 3, and the results from refined grids for groove configurations, including single grooves will be presented in Chapter 5.
CHAPTER 2
FLOW SOLVER

The in-house flow solver Tenasi has been used to investigate the effects of casing treatments. The Tenasi code is a flow solver for Computational Fluid Dynamics (CFD) research that is developed by the University of Tennessee at Chattanooga; it is capable of solving steady and unsteady three-dimensional incompressible, compressible and arbitrary-Mach flow cases. CFD allows researchers and designers to visualize and calculate detailed information inside the flow field. CFD is becoming an essential tool in turbomachinery flow analysis.

2.1 Governing Equations

The non-dimensionalized unsteady three dimensional Navier-Stokes equations without body forces in Cartesian coordinates are as follows:

\[
\frac{\partial}{\partial t} \int_V Q dV + \int_{\partial \Omega} \bar{F} \cdot \bar{n} dA = \frac{1}{\text{Re}} \int_{\Omega} \bar{G} \cdot \bar{n} dA,
\]

where \( Q = [\rho, \rho u, \rho v, \rho w, \rho e]^T \) is the volume-weighted conserved variable vector, \( \bar{n} \) is the outward unit normal and Re is the Reynolds number \( (\text{Re} = \left(\frac{\rho_{\text{ref}} U_{\text{ref}} L_{\text{ref}}}{\mu_{\text{ref}}}\right)/\mu_{\text{ref}}) \). The reference quantities for non-dimensionalization are: density \( \rho_{\text{ref}} \); velocity \( U_{\text{ref}} \);
temperature $T_{\text{ref}}$; pressure $\rho_{\text{ref}} U_{\text{ref}}^2$; length $L_{\text{ref}}$; time $L_{\text{ref}} / U_{\text{ref}}$; energy and enthalpy $h_{\text{ref}}$. In detail,

$$F = \begin{bmatrix}
\rho (u - x_i) \\
\rho u (u - x_i) + p \\
\rho v (u - x_i) \\
\rho w (u - x_i) \\
\rho h_i (u - x_i) + (\gamma - 1) M_i^2 p x_i \\
\end{bmatrix} \hat{i} + \begin{bmatrix}
\rho (v - y_i) \\
\rho u (v - y_i) + p \\
\rho v (v - y_i) \\
\rho w (v - y_i) \\
\rho h_i (v - y_i) + (\gamma - 1) M_i^2 p y_i \\
\end{bmatrix} \hat{j} + \begin{bmatrix}
\rho (w - z_i) \\
\rho u (w - z_i) + p \\
\rho v (w - z_i) \\
\rho w (w - z_i) \\
\rho h_i (w - z_i) + (\gamma - 1) M_i^2 p z_i \\
\end{bmatrix} \hat{k},
$$

$$G = \begin{bmatrix}
0 \\
\tau_{xx} \\
\tau_{yx} \\
\tau_{zx} \\
\end{bmatrix} \hat{i} + \begin{bmatrix}
0 \\
\tau_{xy} \\
\tau_{yx} \\
\tau_{zy} \\
\end{bmatrix} \hat{j} + \begin{bmatrix}
0 \\
\tau_{xz} \\
\tau_{yz} \\
\tau_{zz} \\
\end{bmatrix} \hat{k},
$$

$$\tau_{xx} = (\mu + \mu_t) \left( 2 \partial_x u - \frac{2}{3} \nabla \cdot \mathbf{u} \right),$$

$$\tau_{yy} = (\mu + \mu_t) \left( 2 \partial_y v - \frac{2}{3} \nabla \cdot \mathbf{u} \right),$$

$$\tau_{zz} = (\mu + \mu_t) \left( 2 \partial_z w - \frac{2}{3} \nabla \cdot \mathbf{u} \right),$$

$$\tau_{xy} = \tau_{yx} = (\mu + \mu_t) (\partial_y u + \partial_x v),$$

$$\tau_{xz} = \tau_{zx} = (\mu + \mu_t) (\partial_z u + \partial_x w),$$

$$\tau_{yz} = \tau_{zy} = (\mu + \mu_t) (\partial_z v + \partial_y w),$$

$$q = -\left( \frac{\mu}{\text{Pr}} + \frac{\mu_t}{\text{Pr}_t} \right) \nabla T.$$
The total enthalpy \( h = e + E_c \frac{p}{\rho} \), where \( E_c \) is an Eckert number defined by \( E_c = \frac{U_{\text{ref}}^2}{h_{\text{ref}}} \).

Pr is the Prandtl number, \( \text{Pr}_t \) is the turbulent Prandtl number. \( x, y, \) and \( z \) are the grid speed in the \( x, y, \) and \( z \) directions respectively.

2.2 Finite Volume Methods

The approach that Tenasi uses is a node-centered scheme based on control volumes surrounding each vertex. Control volume is defined as a volume fixed in space or moving with constant velocity in an inertial frame of reference. The control volume is confined by median duals which connect the centroid of each element; this defines the control volume boundary or control surface. In 2D, the median duals connect the centroids of elements and their corresponding edge midpoints (Fig. 2.1); whereas in 3D, the centroids of elements, the centroids of the surfaces and the midpoints of the edges are connected.

![Figure 2.1](image)

Figure 2.1 Illustration of control volumes, which are defined as median duals surrounding each vertex

The governing equations are discretized within control volumes; i.e., the solution variables are stored associated within nodes.
Note that the governing equations, in one-dimensional inviscid form, can be written in terms of primitive variables $q$ as

$$M \frac{\partial q}{\partial t} + AM \frac{\partial q}{\partial x} = 0,$$

where $M = \frac{\partial Q}{\partial q}$ and $A = \frac{\partial F}{\partial Q}$. With preconditioning, the equation above results in

$$M \Gamma_q^{-1} \frac{\partial q}{\partial t} + AM \frac{\partial q}{\partial x} = 0,$$

which can be written as

$$\frac{\partial q}{\partial t} + a_r \frac{\partial q}{\partial x} = 0,$$

let $\Gamma_q = \text{diag}(1, 1, \beta)$ and

$$\beta = \begin{cases} M_r^2, & M_r < 1 \\ 1, & M_r \geq 1 \end{cases},$$

which is sufficient to provide well-behaved eigenvalues.

With preconditioning, the governing equations can be written as

$$M \Gamma_q^{-1} \frac{\partial q}{\partial t} + \int_{\Omega} \mathcal{F} \cdot ndA = \frac{1}{\text{Re}} \int_{\Omega} \mathcal{G} \cdot ndA,$$

where $q = (\rho, u, v, w, p)^T$ is the volume-weighted non-conservative solution vector. The interface flux is evaluated using Roe’s flux approximation given by

$$F_{i+\frac{1}{2}} = \frac{1}{2} (F_R + F_L) - \frac{1}{2} |\tilde{A}| \Delta Q,$$

where $|\tilde{A}|$ can be considered as a dissipation matrix and is evaluated using Roe-averaged variables. This equation can be written as
\[
F_{i+1/2} = \frac{1}{2} (F_R + F_L) - \frac{1}{2} \tilde{M} |\tilde{a}| \Delta q
\]

with \( \Delta Q = \tilde{M} \Delta q \). With preconditioning, it can be written as

\[
F_{i+1/2} = \frac{1}{2} (F_R + F_L) - \frac{1}{2} \tilde{M} \tilde{I}_q^{-1} |\tilde{I}_q^{-1} \tilde{a}| \Delta q.
\]

Now the baseline flow solver of a node-centered, finite volume, implicit scheme can be formed, which is applied to general unstructured grids with non-simplicial elements. An implicit semi-discretization of the equation

\[
\frac{\partial q}{\partial t} + a \frac{\partial q}{\partial x} = 0
\]

can be written as

\[
\frac{\partial}{\partial t} (q^{n+1}) + \mathcal{R}^{n+1} = 0,
\]

where \( n \) indicates a time level and \( \mathcal{R}^{n+1} \) is the residual vector at time level \( n+1 \). The procedure of calculation, as well as for the turbulence model, includes the following basic steps: reconstruction of the solution states at the control volume faces, evaluation of the flux integrals for each control volume, and the evolution of the solution in each control volume in time. Higher-order spatial accuracy is achieved using a linear or quadratic reconstruction of the dependent variables at the control volume faces (i.e., gradient) and using these reconstructed values to evaluate fluxes. Gradients needed for the reconstruction are evaluated using an un-weighted least squares approach while the gradients for the inviscid terms are computed using a weighted least squares approach.
2.3 Time Evolution

The flow solver employs a discrete Newton relaxation approach to solve the unsteady RANS (Reynolds-averaged Navier–Stokes) equations. Newton’s method is used to drive the right-hand-side to zero. The flux Jacobians arising from this linearization can be evaluated using numerical derivatives or the complex Taylor series method. They can also be replaced by approximations which result in substantial savings in computational time. The resulting linear system is solved using a Symmetric Gauss Seidel algorithm (point relaxation). For deforming grids, the Geometric Conservation Law (GCL) has to be satisfied in order to prevent the occurrence of spurious sources in the solutions. This leads to an additional contribution to the residual.

2.4 Relative Motion

The unstructured sliding interface technique was recently developed and validated at the SimCenter in the Tenasi code by Hyams et al. (2011). Arc-based mesh extrusions are constructed from each periodic interface. Virtual points are utilized to correspond to the exact location at which the client data interpolations are performed.

2.5 Turbulence Modeling

The turbulence models are implemented in a loosely coupled manner. The flow solver has the Spalart-Allmaras (1992) model, the Menter (2003) SAS model, the k-\(\varepsilon\)/k-\(\omega\) hybrid model (with and without SST), and the Wilcox (2006) Reynolds Stress model. In addition, DES modes are available for the Spalart-Allmaras, the Menter SAS and the k-\(\varepsilon\)/k-\(\omega\) hybrid models (Nichols et al., 2006). No wall functions or transition models were
used in either solver with integration being performed to the wall with grids designed to give $y+$ values on the order of unity.

2.6 Parallel Implementation

The parallel solution procedure consists of a scalable solution algorithm implemented to run efficiently on subdomains distributed across multiple processes and communicating through MPI. The algorithm has multiple nested kernels viz. time step, Newton iteration, LU/SGS iteration etc. The subdomain coupling is at the innermost level, i.e., in the solution of the linear system. A block-Jacobi type updating of the subdomain boundaries ensures efficient parallelization with a small incremental cost incurred in terms of sub-iterations required to recover the convergence rate of the sequential algorithm. Details about the parallel algorithm can be found in Hyams (2000).
CHAPTER 3

INITIAL RESULTS OF TWO TYPES OF CASING TREATMENTS

3.1 Overview

There are essentially two types of tip modifications that will be examined in this project for their effectiveness in improving performance of the fan stage. The first configuration consists of rings that protrude into the flow-field (Fig 3.1 (a)). The second configuration consists of circumferential grooves that are recessed into the casing (Fig 3.1 (b)). Three different configurations for the rings are considered: (1) ring at the leading edge of the blades (2) ring at the trailing edge of the blades and (3) rings at leading and trailing edges of the blades. The details of the groove and ring configurations are shown in Table 3.1. It should be noted that the depths were chosen such that they block the tip gap present at that location; the widths were arbitrary. Fig. 3.2 adapted from Wisler et al. (1974) gives a general layout of one of the groove configurations, as well as some guidance parameters for width, depth, etc.

Table 3.1 Details of the parameters of the groove and ring configurations

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Location</th>
<th>Depth</th>
<th>Width</th>
</tr>
</thead>
<tbody>
<tr>
<td>Groove</td>
<td>20 - 80% Chord</td>
<td>$8.75 \times d_{\text{land}}$</td>
<td>$2.5 \times d_{\text{land}}$</td>
</tr>
<tr>
<td>Ring</td>
<td>Leading edge</td>
<td>$1.93 \times d_{\text{tip}}$</td>
<td>$5.23 \times d_{\text{tip}}$</td>
</tr>
<tr>
<td>Ring</td>
<td>Trailing edge</td>
<td>$1.25 \times d_{\text{tip}}$</td>
<td>$7.7 \times d_{\text{tip}}$</td>
</tr>
</tbody>
</table>
(a) Front and back ring locations in relation to fan stage

(b) Size and location of five grooves in relation to fan stage

Figure 3.1 General layouts of the ring and groove configurations
Figure 3.2 General layout of grooves and associated parameters (from Wisler et al. (1974))

A brief summary of the run parameters and code options that were used will now be given; these will apply, unless otherwise noted, to all the configurations under consideration. The Menter (Menter et al., 2003) SAS is chosen as the turbulence model to be used. The flow field was initialized via a customized routine in the solver that adjusted the velocity vectors depending upon the location in the flow path and increased pressure from inlet to exit using linear interpolation. This method gives a reasonable approximation to the field that takes the turning of the flow and the pressure rise into account, which makes the numerical “start transient” progress more easily. The solver was used in the unsteady mode from the start, with 2\textsuperscript{nd}-order accuracy in time and 5 Newton sub-iterations per physical time step iteration. The actual time steps used were rather aggressive at the beginning, having values equivalent to as few as 45 time-step iterations per wheel revolution (i.e., 8 degrees per time step or approximately one rotor blade pitch every two time steps); this was done in order to quickly establish the overall flow field. After 2 - 3 revolutions at these larger values, the time step was then reduced to the equivalent of 180 time steps per revolution, and the solution continued for at least 10 additional revolutions (1,800 time steps). The dual time-stepping technique was
employed to increase numerical stability with the CFL number being ramped up from 1 to 4 during the first 250 time steps of the simulation.

For the higher-order spatial accuracy, quadratic reconstruction of the flux was used with the Venkatakrishnan limiter employed in the near vicinity of shocks. The variable Mach number algorithm by Briley et al. (2003) was used, although preconditioning was not required, since the reference Mach number was set to the blade tip Mach number at 100% speed which is supersonic ($M_{\text{tip}} = \text{approximately } 1.1$). In this sense, the solver acts as a primitive-variable based solver. Each simulation was run in the relative frame of reference. All solid boundaries were treated as no-slip and adiabatic. Additionally, the rotor casing used a boundary condition that spins it in the opposite direction of the rotor at the same rotational speed; the absolute velocity vectors will, therefore, go to zero on the portion of the casing just outboard of the rotor blades. The inflow boundaries were set to total temperature and pressure equal to standard-day values of 101.325 kPa and 288.15 K, respectively. A constant static pressure was applied to the exit boundary, and this value was varied to obtain the individual operating points along the speed-line curve; all results for the various configurations are at 100% speed. The start-up procedure described in the previous paragraph was applied to each of the operating points, with the exception of those near-stall. In those cases, simulations were re-started from previous solutions at stable operating points, and the exit back pressure was increased in small increments to carefully approach the stall point.

In order to illustrate how the stability limit is determined, Fig. 3.3 shows a plot of the mass flow history of the fan stage with grooved casing. Both simulations in this figure are restarted from previous stable solution at a slightly lower back pressure. Upon restarting
with a specified back pressure of 131.6 kPa, the flow rate declines slowly for the first 10 revolutions and then begins to settle out at about 83 lb\textsubscript{m}/sec; the simulation is stable at this flow rate for over 20 revolutions. By restarting from the same solution but increasing the back pressure by only 100 Pa (just under 0.1% of an atmosphere), the mass flow history follows a similar path for about 10 revolutions. It then starts to decline noticeably before dropping very rapidly at about 17.5 revolutions; the simulation terminated shortly thereafter. This plot gives a good indication that the stall limit has been reached numerically.

![Mass flow history to determine stage stability limit (with grooved casing)](image)

Figure 3.3  Mass flow history to determine stage stability limit (with grooved casing)

Fig. 3.4 below is taken from Hyams et al. (2011) and shows comparisons of total pressure ratio for a 2-rotor/5-stator section for various speed-line curves in terms of the on-design speed for the baseline case. This figure is being included to demonstrate the accuracy of the Tenasi solver for this type of problem. The results using an unstructured grid agree very well with experiment for all speeds. The two cases using a structured grid
were run only at 100% design speed level, and a noticeable loss in performance is seen. This is felt to be due primarily to a lack of grid resolution. The “2-5 sector” used with an unstructured grid allowed for an increase in grid resolution, while keeping the overall grid point count at an affordable level. The grid for this case contained approximately 6.325 M nodes, with 12.7 M tetrahedral volume elements and 7.596 M prism volumes (primarily used for the boundary layer). Tetrahedral and prismatic elements comprise the majority of the volume elements, so only the values for these elements are being reported. Further details of these simulations may be found in Hyams et al. (2011).

![Computed SDT2-R4 stage performance](image)

**Figure 3.4** Total pressure vs. corrected mass flow rate for SDT2-R4 stage (from Hyams et al. (2011))

### 3.2 Leading-Edge and Trailing-Edge Rings

#### 3.2.1 Leading-Edge and Trailing-Edge Rings Employed Together

The general dimensions of the leading-edge (or front) ring and trailing-edge (or back) ring were given in the first paragraph of the “Overview” section above. As noted in the
Introduction, this case acts as a virtual channel for the tip of the rotor to pass through, as the leading and trailing edges of the tip are “blocked off”. The grid for this configuration contains 6.199 M nodes, 11.26 M tetrahedral volumes, and 7.843 M prisms. This is very close in size to the baseline grid, which contained 6.178 M nodes, 11.14 M tetrahedral volumes, and 7.866 M prisms. Fig. 3.5 shows a cut of the grid.

![Image](image.png)

Figure 3.5  A theta cut of the grid for leading-edge and trailing-edge rings employed together

This configuration performed quite poorly (with stall occurring at a back pressure of 128 kPa as opposed to 131.1 kPa for the baseline configuration) as can be seen in the figures below (Fig. 3.6). The efficiency is also largely reduced (Fig. 3.7). In order to understand this poor performance, the leading and trailing edge rings were simulated independently of one another.
Figure 3.6  Total pressure comparisons with virtual channel protruding from casing

Figure 3.7  Efficiency comparisons with virtual channel protruding from casing
3.2.2 Leading-Edge Ring Only

The grid for this case contains 6.172 M nodes, 11.24 M tetrahedral volumes, and 7.843 M prisms, and is shown in Fig. 3.8. This grid is also very close in size to the baseline grid. This particular case did not perform very well at all, especially when compared to the baseline computational case and experiment. The “stall” condition for this case was at a back pressure setting, \( p_{\text{back}} \), of about 125 kPa (Fig. 3.8); the word “stall” here is used in the context of failure of the simulation to complete. It is not meant to imply that the aerodynamic stall mechanism is being captured, although there is probably some evidence of both physical and “numerical” stall in the solution. The \( p_{\text{back}} \) setting for the baseline case is around 131 kPa before it breaks down. So relative to the baseline, the simulation fails at a back pressure setting approximately 4.6% lower. The presence of the leading-edge ring is obviously detrimental to the fan’s performance. The physical/numerical reasons for this performance degradation cannot be stated with certainty at this point, but it is strongly suspected that the blade leading edge being just downstream of the ring is interacting adversely with the wake of the ring, perhaps in the form of an unsteady shedding mechanism in the wake. Regardless of the actual reason, it is felt that the presence of the front ring is in some manner enhancing the tip-leakage loss and, therefore, reducing the performance of the fan stage.
Figs. 3.9 and 3.10 are plots of the total pressure and efficiency, $\eta_{ad}$, for this configuration, which clearly show the performance breakdown. Total pressure ratio is adversely affected as the speed-line curve is moved down and to the left. This behavior is as if the stage were operating at a lower speed, which indicates the severity of the performance degradation. This is also reflected in the efficiency curve, although the efficiency values themselves are not terribly lower than for the baseline. The comparison of total pressure rise clearly indicates the leading-edge ring by itself should not be considered for performance enhancement.
Figure 3.9  Total pressure comparisons with leading-edge ring protruding from casing

Figure 3.10  Efficiency comparisons with leading-edge ring protruding from casing

3.2.3  Trailing-Edge Ring Only

Like the previous two configurations, the grid for this case was very close in size to that of the baseline. The node count was approximately 6.183 M, with 11.1 M tetrahedral volumes and 7.859 M prisms. So the node count and prism count are slightly higher than
the baseline and leading-edge ring cases, with the tetrahedral count slightly less. A cut of the grid is shown in Fig. 3.11.

![Image](image.png)

**Figure 3.11** A theta cut of the grid for trailing-edge ring only

Fig. 3.12 gives the performance, in terms of total pressure ratio, as compared to the experimental data and the baseline simulation. The performance curves are almost identical, with a slight drop off at the lower flow rates for the trailing-edge ring. Agreement is pretty good near the choke condition; the computations then show lower values than the experiment. At the design point flow rate, the pressure ratio is approximately 1.455 compared to 1.47 for experiment, a difference of about 1.2%. As flow decreases further from the design-point value, the percent difference between the computed values and experiment remains about the same, with the greatest difference being about 1.4% at the left-most experimental point.

Fig. 3.13 is a comparison of the adiabatic efficiencies of the computed and experimental values. In the choke region, the computed values are again practically identical and in good agreement with experiment. Both computational cases are again less than experiment in the design-point region, but the difference between the trailing-edge ring and the baseline is more noticeable. Like the values in pressure ratio, the
difference in the two computed values increases at the lower flow rates.

Figure 3.12  Total pressure comparisons with trailing-edge ring protruding from casing

Figure 3.13  Efficiency comparisons with trailing-edge ring protruding from casing

The trailing-edge ring by itself shows no real improvement in performance relative to
the baseline, but it shows no appreciable ill effects. This is very different from the leading-edge ring alone and the virtual channel. Though the reason is not fully understood at this point, it seems readily apparent that the effect on the flow field is strongly influenced by the presence, or lack thereof, of the leading-edge ring.

3.3 Grooved Casing: Middle Five Grooves

The dimensions of the grooves in terms of the width of the land separating the individual grooves were given in the first paragraph of the “Overview” sub-section. There are two grids run for this configuration. The first grid was slightly bigger than the ones for the previous cases, since the grooves themselves required gridding. It contained 6.924 M nodes, with 11.66 M tetrahedrons and 9.15 M prisms. The increased prism count is resulted from the spatial resolution required for the regions adjacent to the walls of the grooves. The second grid was more refined, and, therefore, the point/volume count increased noticeably; it contained 9.795 M nodes, 16.78 M tetrahedrons, and 12.97 M prisms. A cut of the grid is shown in Fig. 3.14. As described before for the baseline rotor casing, the grooves are declared to spin backward relative to the rotor, which is fixed in the relative frame of reference.
The results of middle five grooves in two different grid sizes were compared with the baseline case in a relatively coarse grid, Fig. 3.15 shows a plot of the performance in terms of total pressure ratio for solutions from both the original and the refined grids (in which the region close to the grooves were refined the most). The solutions from these two grids are very close to that of the baseline, and they are consistent with results from the previous configurations when compared to experiment, in that the curves show slightly lower pressure rise. The most significant result in this plot, however, is that stable operation is maintained to a noticeably lower value of mass flow rate. For the baseline case, the last stable operating point has a flow rate of approximately 87.5 lb_m/sec. With the grooves included, the flow rate is reduced to the previously stated value of 83 lb_m/sec. This represents a range extension of about 5.1% relative to the baseline case. Fig. 3.16 is a plot of efficiency, and it is easy to see that a small price is paid for the increase in operating range. Relative to the baseline case, the efficiency is decreased by about 2% at most from the original grooved grid; the refined grid shows some improvement in that the results from it roughly split the difference between the baseline and the original grooved
The results in this chapter have shown possible solutions to the adverse effect of tip-case.

3.4 Summary

The results in this chapter have shown possible solutions to the adverse effect of tip-
leakage flows. Two types of casing treatments were considered to mitigate this effect, but only the groove configuration was successful in achieving the desired objective. As was discussed in Chapter 1, numerous explanations for the relationship of grooves and stall margin improvement have been proposed by many researchers.

In an effort to further understand the effect of the grooves and in the hope of offering a contribution to this area, it was decided to investigate the effects of a series of single-groove configurations on the performance of the fan stage. During this process, significant sensitivity of the stall margin improvement to the way the grids were generated was observed. This leads to a grid generation/resolution study, which is the topic of the next chapter.
CHAPTER 4
EFFECT OF GRID ON THE SIMULATION RESULTS

In this chapter, the effects of grid density and distribution on the performance of the SDT2-R4 fan stage are studied. A key motivation for this study is the observation that different methods of generating meshes seem to have a significant impact on the performance of the fan stage. Two different ways of generating viscous meshes for the configurations with the grooves were considered. Five grids with varying resolutions were generated and the results of the simulations are presented here. Examination of total pressure distribution at the near tip area suggests that a reasonably dense grid is required to resolve the near-stall condition.

4.1 Overview of the Grids

CFD simulation of turbomachinery is sensitive to the grid being used. Two different approaches to the generating viscous meshes for the grooved configurations are considered. The first takes the entire geometry, including the grooves, builds an unstructured mesh and then uses VLI (Karman, 2006) to insert the viscous layers (Fig 4.1(a)). The second approach attempts to isolate the effect of the grooves and facilitate comparison to the baseline solutions by keeping the grid below the grooves (the baseline part) unchanged (Fig 4.1 (b)). This approach has the advantage that when it comes time to generate a viscous mesh for a new groove configuration, VLI will have to be
performed for just the grooves and not the entire mesh resulting in a substantially faster turnaround time. An artifact of the second approach is that there is viscous refinement at the entrance to the grooves, even though that part of the casing has been removed because of the addition of the grooves.

In order to keep the grid below the casing identical between the baseline and groove meshes, additional lines (connectors) have to be introduced into the casing surface mesh. These lines correspond to potential groove locations. Such meshes are termed to be “groove ready”. Fig. 4.2 shows the comparison between the topologies of the normal
baseline grid and the groove-ready baseline grid; the meshes on the casing are removed for clarity, and the connectors are shown in red color. In contrast with the normal baseline grid which has relatively smoother grid distribution on the area above the rotors (Fig. 4.2 (a)), the groove-ready baseline grid has less smooth but denser mesh at the same place because of the presence of the additional lines (Fig. 4.2(b)). More details can be found in Fig. 4.3 - 4.7. The area of casing above the rotors is thus segmented into several pieces so the grooves can be merged easily.

(a) Normal baseline grid                      (b) Groove-ready baseline grid

Figure 4.2 Comparison between the topologies of normal baseline grid and groove-ready baseline grid (top view, outline)

The distribution of the points on the casing turns out to be critical. The previous chapter compared a regular baseline grid with a groove grid generated using the first approach. In that case, the baseline grid was able to run at a back pressure of 131.1 kPa with a mass flow rate of 87.5 lbm/sec, while the groove configuration could run up to 131.7 kPa with a mass flow rate of 83 lbm/sec, an improvement in the stall margin of 5.1%. The current study, however, leads to different results as summarized in Table 4.1.
Note that the testing grids have the same level of points, the only difference are the groove-ready lines for baseline grid and the additional “viscous layers” in the entrances of grooves for the grooved grid.

Table 4.1 Comparisons of stall points of baseline and grooved configurations from normal and groove-ready versions of grids

<table>
<thead>
<tr>
<th>Grid</th>
<th>Stall point</th>
<th>Description of the grid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseline, normal</td>
<td>131.1 kPa</td>
<td>No segmentation lines. It has relatively smoother grid distribution on the area above the rotors</td>
</tr>
<tr>
<td>Groove, VLI after the grooves are attached</td>
<td>131.7 kPa</td>
<td>Viscous layers simply follow where the solid walls are</td>
</tr>
<tr>
<td>Baseline, groove-ready</td>
<td>131.0 kPa</td>
<td>Additional groove-ready lines on the casing on the top area of rotors</td>
</tr>
<tr>
<td>Groove, VLI before the grooves are attached</td>
<td>131.5 kPa</td>
<td>Additional “viscous layers” in the entrances of grooves</td>
</tr>
</tbody>
</table>

Five different grids with different resolutions that were tested are shown in Fig. 4.3-4.7. To illustrate the density of grid points, the grid of the rotors’ part is divided into 3 blocks and marked as A, B and C respectively, with the interfaces marked as 1 (between block A and block B) and 2 (between block B and block C). The five different grids were for two different configurations: one with five grooves and one with eight grooves. The five grooves configuration allowed for grooves to be placed over the mid-section of the blade tip, while the eight groove configuration provided for additional locations at the leading and trailing edges. Only one grid was built for the five-groove configuration (Case 1; Fig 4.3). The remaining four grids were built for the eight-groove configuration with varying degrees of refinement in the three regions. A coarse grid comparable to the five-groove case was built and is labeled Case 2 (Fig 4.4). The grid for Case 3 had
additional refinement in block C; however, it had a poor transition in terms of point spacing between the grooves and the rest of the casing (Fig 4.5). This was fixed in the mesh for Case 4 (Fig 4.6). Case 5 was a grid based on the one from Case 4 with additional refinement in blocks A and B (Fig 4.7). Detailed comparisons of the test grids are shown in Table 4.2.

Table 4.2 Comparisons of five test grids of groove-ready baseline case

<table>
<thead>
<tr>
<th>Grid</th>
<th># Nodes</th>
<th># Grooves</th>
<th>Connector spacing Δs</th>
<th># Nodes on the casing</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>Case 1 (Fig. 4.3)</td>
<td>6.17M</td>
<td>5</td>
<td>0.1</td>
<td>0.1</td>
</tr>
<tr>
<td>Case 2 (Fig. 4.4)</td>
<td>6.24M</td>
<td>8</td>
<td>0.1</td>
<td>0.1</td>
</tr>
<tr>
<td>Case 3 (Fig. 4.5)</td>
<td>9.36M</td>
<td>8</td>
<td>0.05</td>
<td>0.04</td>
</tr>
<tr>
<td>Case 4 (Fig. 4.6)</td>
<td>9.51M</td>
<td>8</td>
<td>0.05</td>
<td>0.04</td>
</tr>
<tr>
<td>Case 5 (Fig. 4.7)</td>
<td>11.65M</td>
<td>8</td>
<td>0.04</td>
<td>0.04</td>
</tr>
</tbody>
</table>

Figure 4.3 Middle five groove-ready baseline grid with relatively coarse resolution. Top view (left) and side view (right)
Figure 4.4  Full coverage eight groove-ready baseline grid with relatively coarse resolution. Top view (left) and side view (right)

Figure 4.5  Full coverage eight groove-ready baseline grid with relatively refined resolution in rotor blade area. Top view (left) and side view (right)

Figure 4.6  Full coverage eight groove-ready baseline grid with relatively refined resolution in rotor blade area and smoother casing domains. Top view (left) and side view (right)
4.2 The Results obtained with the Grids

Strong sensitivity to grid resolution is observed as the simulations approach the stall point. The results for the various cases, both in terms of the back pressure as well as the achieved mass flow rate, are summarized in Table 4.3. Note that the difference between the backpressures can be up to 0.55 kPa (or 1.7375 lb\textsubscript{m}/sec in terms of mass flow rate), which is significant.

<table>
<thead>
<tr>
<th>Case</th>
<th>Mass flow rates lb\textsubscript{m}/sec at near-stall conditions of different grids</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>131.0 kPa</td>
</tr>
<tr>
<td>Case 1</td>
<td>86.9200</td>
</tr>
<tr>
<td>Case 2</td>
<td>88.4184</td>
</tr>
<tr>
<td>Case 3</td>
<td>Not tested</td>
</tr>
<tr>
<td>Case 4</td>
<td>Not tested</td>
</tr>
<tr>
<td>Case 5</td>
<td>Not tested</td>
</tr>
</tbody>
</table>
The contours of relative total pressure ($P_{t,\text{rel}}$) for the coarse grids (Cases 1, and 2) and the refined grids (Cases 4, and 5) at their near-stall conditions are shown in Figs. 4.8 - 4.9. $P_{t,\text{rel}}$ is one of the most important indicators of stall phenomena and is selected to exhibit the effect of the grids. The differences in the distribution of $P_{t,\text{rel}}$ are quite evident, most noticeably in the low energy area illustrated in blue color in the middle of two blades, which covers approximately 20% to up to 80% axial chord of the suction surface of the rotor blades.

In contrast with the refined grids, the blue areas in Cases 1 and 2 (Fig. 4.8) seem unable to be captured (or resolved) very well. On their own operation limitations, Case 1 (Fig. 4.8 (left)) exhibits a large blue area which takes most of the space between suction and pressure surfaces in a blade passage; the blue area seems to be smeared in Case 2 (Fig. 4.8 (right)). Although they are both operating in the near-stall condition, their flow fields largely differ from each other.

Figure 4.8 $P_{t,\text{rel}}$ contours near blade tip from relatively coarse grids. Case 1 at 131.0 kPa back pressure (left) and Case 2 at 131.2 kPa back pressure (right), both at their near-stall points
The diffusion of the blue area is believed to be the effect of numerical viscosity, which should be diminished as the grid is refined. Although each has different stall point, cases 3-5 resolve the flow fields near the casing to a more reasonable level. Pictures in Fig. 4.9 show the total pressure contour at Case 4’s near-stall point as well as the contours of Case 5’s three different near-stall conditions. At the same back pressure of 131.4 kPa, the length of the blue area of Case 4 (Fig. 4.9 (a)) is roughly 20 percent longer than Case 5 (Fig. 4.9 (b)); at the approximately same mass flow rate of Case 4 (at 131.4 kPa back pressure), Case 5 (at 131.45 kPa back pressure) has relatively shorter and less intense low energy area near casing (Fig. 4.9 (c)). Case 5 at its near-stall point (Fig. 4.9 (d)) has a similar pattern as that of Case 4 but with slightly lower total pressure in the center of the blue area.
In addition to the refinement of the grids, the importance of smoothness on the casing mesh above the rotor is also emphasized by the contrast of Case 3 and 4. The presence of the groove-ready segmentation lines requires more work on the manual smoothing of the grid, especially on the casing above the rotor blades. The difficulty comes from the match of the spacing, $\Delta s$, of these lines when they come across the top of the rotor blades: the spacing has to be matched. The number of points of each segmented connector and the distribution of points need to be carefully tuned in order to obtain a smoother grid. As a result, the grid points in this area become denser than before. Current results indicate that, as the mesh in that area becomes smoother, it can run at higher back pressure and lower mass flow rate.

Grid density and distribution on the level of Case 5 are believed to be reliable in resolving the near-stall phenomena in the current type of simulation. However, the inherent issues of two-passage simulation limit the accuracy, and it is often not affordable to run a full-wheel simulation with a grid of reasonably high resolution.

4.3 Summary

This chapter studies the effects of grid density and distribution on the performance of the SDT2-R4 fan stage for the purpose of building a reliable groove-ready grid. The groove-ready grid is the type of baseline grid that is suitable for the merge of grooves. As a result, the baseline part of the corresponding grooved grid is identical to the groove-ready baseline.

The investigation shows that the near-stall simulations of the fan stage are influenced by the grid to a large extent. Five groove-ready grids with varying resolutions
were generated and the results of the simulations presented here. Grids were refined and smoothed in the areas of the casing above the rotor blades and in front of them, respectively. It was found that there is a large discrepancy between the stall points obtained from different grids. The highest back pressure that the baseline can run is 131.55 kPa with the mass flow rate of 85.18 lb_m/sec; whereas the lowest back pressure is 131.0 kPa with the mass flow rate of 86.92 lb_m/sec. This indicates that the difference between the backpressures can be up to 0.55 kPa (or 1.7375 lb_m/sec in terms of mass flow rate). According to the cases that were run, the finer the grid resolution is, the higher the stall point will be. Coarse grids can only achieve low backpressures in that they are not able to resolve the low energy areas inside of a blade passage. Besides, the coarser grid has the lower mass flow rate when it is run at the same back pressure. In addition, the smoothness of the grid is also important. The effect of grid resolution is thus not to be underestimated especially for the evaluation of near-stall conditions.
CHAPTER 5

RESULTS OF GROOVE CONFIGURATIONS WITH REFINED GRIDS

5.1 Overview

The success of stall margin improvement by the middle five grooves configuration (M5G) necessitates the investigation for the effect of single grooves. In Chapter 1, several concepts/theories regarding tip-leakage flow as it pertains to stall and losses were presented. In this chapter, several of those parameters such as incidence angles, blade loadings, radial transport of axial velocity etc. will be considered, and the effects of the grooves on them will be discussed. It will be seen whether some of them can serve as reasonably good indicators of the benefits brought by the grooves. The loss in general from tip-leakage flows will also be discussed.

Eight locations were considered for the single groove configurations. Out of these, five were identical to the M5G configuration. Additionally, one groove location at the trailing edge and two groove locations closer to the leading edge were added. The extra groove at the leading edge was added so that its effect on the strong tip-leakage flow could be studied.

For the reasons discussed in Chapter 4, the grid which is 8 grooves-ready and is most refined (Case 5), is adopted for the all the test cases considered here. Given the sensitivity of the stall point to the choice of grid, it was decided to rerun the smooth wall baseline case as well as the M5G configuration.
Fig. 5.1 shows the overall look of the grooved configuration where eight grooves are implemented, the relative location of eight grooves covers 96.36% axial chord of a rotor blade. A close side view of the eight grooves is shown in Fig. 5.2, and it can be seen that each groove is marked as $a - h$ ($SG_a - SG_h$) alphabetically following the streamwise direction. The width of each groove and the width of the lands between grooves in the rotor casing ($d_{land}$) are kept the same as in Chapter 3, which implies that the original middle five grooves now consist of grooves $SG_c$, $SG_d$, $SG_e$, $SG_f$ and $SG_g$. $SG_h$ and $SG_h$ are extended to the upstream and downstream with the same groove width, depth and $d_{land}$; they, together with the middle five grooves, cover the middle 92.72% axial chord of a given rotor blade. For the reason that the leading edge flow usually has more influence on the overall flow phenomena, an additional groove at the leading edge location is to be investigated as well, which is marked with the letter $a$. Note that $SG_a$ extends forward beyond the leading edge of rotor blades by 9.08% of the axial chord length of a given rotor blade.

Figure 5.1  Size and location of eight grooves in relation to fan stage
The summary of the run parameters and code options that were used are given in Chapter 3, except for the time step. To capture the shock motion in a transonic rotor, a time step of 0.00025 sec was necessary according to Copenhaver et al. (1993). A different run parameter adapted in the present study is a small time step of 0.000013 sec, which is equivalent to 360 time steps per revolution. This small time step is used for both the baseline and the grooves cases after the establishment of the overall flow field. As described before for the baseline rotor casing, the grooves are declared to spin backward relative to the rotor, which is fixed in the relative frame of reference.

5.2 Fundamental Observations from Baseline Configuration

To understand the reason that grooves bring stall margin improvement, an overview of the flow field of the baseline SW configuration is presented in this section.

Observation 1: At near-stall condition, the tip-leakage flow is strongest at 0-40% chord location and becomes weaker at 60-100% chord. The strong part of tip-leakage...
flow covers a large part of the rotor passage, while the weak part forms a so-called secondary leakage.

The strength of the tip-leakage flow is largely influenced by the magnitude of the pressure difference between the pressure and suction sides of the rotor blades. Fig. 5.3 shows the top views of the changes of static pressure as the back pressure is raised from peak efficiency to near-stall operation on the casing and at 98% span. The jump in pressure is because of the presence of shocks.

![Figure 5.3](image)

**Figure 5.3** Pressure contours. Casing and 98% span. SW at peak efficiency and near-stall conditions

From the near-stall pressure contours, the area where the interaction of shock and the boundary layers on the end-wall and SS is visual near 40% chord of SS. The interaction is in addition to with the tip-leakage flow, as will be seen in observation 2.
In Fig. 5.4 the theta velocities (pitch-wise velocity components) are exhibited in the theta cuts. The theta velocity components are evident in the locations 1 – 3 but not very strong in the locations 4 – 6, which are corresponding to chord locations of approximately 0 – 40% and 60 – 100%, respectively. The strength of the tip-leakage flow in terms of theta velocity components agrees with the distribution of pressures in the near tip region.

Fig. 5.5 shows the streamlines across the tip gap region, which are also extended to their sources. The difference between the coverage of the tip-leakage flow can be visualized straightforwardly in these two pictures. At the peak efficiency operation, the tip-leakage flow travels along with the main flow and does not have strong negative axial velocity components. Whereas at the near-stall point, it covers a significant portion of the passage in near tip region as the leading edge tip-leakage flow is strengthened by the increased leading edge pressure gradient on SS-PS. Mainly formed by the cross flow in the tip of leading edge, the tip-leakage vortex travels with the rotor blades closer to PS.
than the peak efficiency condition, but it is still near the SS. It prevents the incoming flow and the tip-leakage flow (mainly from 3 – 50% chords) from passing by, which diverts the incoming flow towards the PS and the tip-leakage flow towards the radially outward direction, forcing it to pass over it. The incoming flow is also diverted towards the PS in the near tip region and will be accelerated because of the blockage, as can be seen from the velocity field at 98% span (Fig. 5.6). Some of the leading edge tip-leakage flows from 3% - 50% chords then stays in the rotor passages and interacts with the incoming flow, but the other crosses the main coverage area of tip-leakage flow and travels to adjacent passages via the chord near trailing edge of PS. This is referred to as secondary leakage.
Figure 5.5  Streamlines of the tip-leakage flows. SW at peak efficiency and near-stall conditions
**Observation 2:** As the operation is moving from peak efficiency to near-stall condition, the leading edge shock moves toward the upstream and starts to strongly interact with the tip-leakage vortex on the near casing regions, followed by a large low axial velocity area in the rotor passages.

Fig. 5.7 shows the relative Mach number contours of SW in its peak efficiency and near-stall conditions at 98% span. At the peak efficiency condition, there exists an oblique shock which is attached to the leading edge (Fig. 5.7 (a)). Since both the shock and the leading edge vortex are relatively weak, the interaction between them is not strong either, followed by a small low velocity area close to the trailing edge of SS. In contrast, at the near-stall condition, the shock is transformed from oblique shock to normal shock, and is detached from the leading edge of blades. The shock then interacts with the strong leading edge vortex and tip-leakage flow as well as the boundary layer of SS of the adjacent blade, which smears the shock in that area to a significant degree (Fig.
5.7 (b)). The interaction is so complicated that it will need further investigation. A large low-velocity area is visible not very far from the interaction point, which is mainly the product of tip-leakage flow. By examining the near-stall streamline picture again in Fig. 5.5, the low-velocity area is formed by tip fluid from about 35-50% chord that climbs over the leading edge tip-leakage vortex, as was discussed in observation 1.

![Mach contours. 98% span. SW in its peak efficiency and near-stall conditions](image)

**Figure 5.7** Mach$_{rel}$ contours. 98% span. SW in its peak efficiency and near-stall conditions

Fig. 5.8 shows the radial velocity contours of SW in its peak efficiency and near-stall conditions at 98% and 90% spans. Because of the complex interaction between shock, boundary layer and vortex, the flow gains a significant radially inward velocity right after the interaction. The inward velocity area happens at both peak efficiency and near-stall conditions but is much more evident in the latter case. Since the tip-leakage flow is not strong and the flow velocity is lower at the 90% span, the influence of the triple interaction on the main passage is less obvious.
Fig. 5.8 $V_r$ contours. 98% and 90% spans. SW in its peak efficiency and near-stall conditions

Fig. 5.9 shows the axial velocity contours for the same cases at 98% and 90% spans. It can be seen that the low axial velocity area exits right after the core of radially inward velocity area, which indicates that this area is actually a flow blockage. The triple interaction is not obvious at 90% span, but the interaction between the shock and the boundary layer on the SS is still visible
Observation 3: Separation happens on the SS.

Separation shows up often because of the adverse pressure gradient. At the stall condition, the leading edge tip-leakage vortex prevents the incoming flow from passing through, leading to lower pressure in the near tip region which aggravates the separation. The incoming flow is diverted by the vortex, and a portion of it then moves radially upward to the tip separation region and forms a strong spanwise vortex attached to the SS of rotor blades (Fig. 5.10).

From the aspect of axial velocity, the role of tip-leakage flow is more easily identified. As is shown in Fig. 5.11, most of the tip-leakage flow has a negative axial velocity component when it is ejected from the tip gap region, most noticeably right after the shock / tip-leakage vortex interaction attached to the suction sides of rotor blades, at the near-stall condition. The flow with negative axial velocity also has significant theta and

Figure 5.9  $V_x$ contours. 98% span. SW in its peak efficiency and near-stall conditions
radial velocity components, which bring it to the middle of the rotor passages and form
the low axial velocity area. Usually the negative axial velocity will be canceled by its
mixture with the incoming flow; however, because of the blockage by the leading edge
tip-leakage vortex, the separation occurs in a premature period.

![Streamwise direction](image)

Figure 5.10  Streamline of reversed flow of SW at near-stall condition

![Vx contours. Theta cuts. SW in its near-stall point](image)

Figure 5.11  $V_x$ contours. Theta cuts. SW in its near-stall point

**Observation 4**: The blockage at the rotor passage exit plane is significantly increased.
One of the basic mechanisms that happen at near-stall is a high level of blockage. Fig. 5.12 shows relative total pressure contours at the rotor passage exit plane of SW at peak efficiency and near-stall conditions, where the lower the value is, the higher the total pressure loss would be. It is found in this case, the relative total pressure near the end-wall of the passage is greatly decreased thus a large blockage is generated.

![Relative Total Pressure Contours](image)

(a) Peak efficiency condition  
(b) Near stall condition  

Figure 5.12 $P_{t,rel}$ shown in local color. Rotor passage exit plane. SW at peak efficiency and near-stall conditions

**Observation 5:** The tip blade loading near the leading edge is significantly increased.

The increase of tip blade loading near the leading edge is the result of forward shifting of passage shock following the decrease of mass flow rate (Fig. 5.13).
To sum up, the fundamental investigations on the baseline indicate that, as the operation is moving from peak efficiency to near-stall, the strongest tip-leakage flow at the first half chord near the leading edge (observation 1) starts to interact with the leading edge shock, followed by a low velocity area formed by the tip fluid in near mid-chord which climbs over the leading edge tip-leakage vortex (observation 2). The leading edge shock also interacts with the boundary layer of the SS, results in separation started near mid-chord, which is aggravated by the tip-leakage vortex (observation 3). These behaviors caused by the tip-leakage flow would produce adverse effects on the main flow and serve as a mechanism to induce the stall. The blockage at the rotor passage exit plane is significantly increased (observation 4), also is the tip blade loading (observation 5).

5.3 Classic Middle Five Grooves Configuration

The dimensions of the grooves in terms of the width of the land separating the individual grooves were given in the first paragraph of the “Overview” sub-section of
Chapter 3. The 8 grooves-ready baseline grid adapted after Chapter 4 contains 11.65 M nodes, with 33.30 M tetrahedrons and 11.14 M prisms. The middle five grooves with viscous layers are attached directly to the baseline grid without changing it, which contains 12.84 M nodes, with 40.16 M tetrahedrons and 11.14 M prisms.

The middle five grooves configuration (M5G) is compared with the baseline case (SW) in terms of their performance. Fig. 5.14 shows a plot of the performance in terms of total pressure ratio. The solutions from these two grids are very close to each other, and they are consistent with results from experiment, in that the curves show slightly lower pressure rise. The stable operation for the groove configuration is maintained to a lower value of mass flow rate, but the difference is less noticeable than it was reported in Chapter 3 because of the “enhancement of stall margin” brought by the refinement of the baseline grid. For the baseline case, the last stable operating point has a flow rate of approximately 85.18 lbm/sec. With the grooves included, the flow rate is reduced to the previously stated value of 83.23 lbm/sec. This represents a range extension of about 2.3% relative to the baseline case. Fig. 5.15 is a plot of efficiency, and there is again a small price to be paid for the increase in operating range. Relative to the baseline case, the efficiency is decreased by about 0.43% at most from the grooved grid.
The next effort is to try to understand why the grooves make the increase in operating range possible.
5.3.1 Mechanisms of Grooves by Pumping and Drawing Fluid Into and Out of the Grooves

Despite the arguments raised in the relationship between stall margin improvement and the effect of grooves on the leading edge vortex, it is felt that the relationship can serve as at least a critical indicator. As was showed in section 5.2, the tip-leakage flow starts with a jet and results in tip vortex which then interacts with passage shock, followed by low velocity fluid and growing blockage, which are the signals of the initiation of stall. The impact of grooves on the tip-leakage flow will thus lead to the change of the intensity of tip vortex, and it is still possible that the initiation of stall would be therefore postponed.

Fig. 5.16 shows the radial velocity at near rotor blade tip zones (98% span). The SW is run in its near-stall point (Fig. 5.16 (a)) and the M5G is run at the same mass flow rate that the SW case has (Fig. 5.16 (b)). The radial velocity is presented in different colors with contour lines in the figure, where the blue areas indicate a high radially inward velocity region pointing into the hub. The highest radially inward velocity area is most evidently located at the 40% chord on the SS for both SW and M5G cases. From the comparison, the trade between the main flow and the grooves is clear in the aspect of radially transporting flow. Due to the pressure difference between the PS and the SS, the flow tends to move radially upward into the groove in the PS and reverse back in the SS. The vortices in the entrances of grooves exist even after the blade passes by. The high radially inward velocities of the main flow seem to be transported into the grooves and are thus less influential on the incoming flow. Since the large pressure gradients from PS to SS exist at where large radial and theta velocity components are, the fluid exchange in
the corresponding grooves is more intensive than the others. It can also be seen that the interaction between the main flow and the grooves is very strong in the first groove near the leading edge, and becomes weaker as the grooves get closer to the trailing edge of the rotor blade (Fig. 5.17). This observation agrees with the statement of Rabe (2002) that any mechanism that redirects the flow more axially at the blade leading edge would improve stall margin. The integration of the normalized radial velocity at this span, however, indicates that the average radial velocity is raised, but not by a significant amount. The values of the average $V_r$ are -0.0277614 (SW case) and -0.0297722 (M5G case) at 98% span, -0.0120166 (SW case) and -0.0147798 (M5G case) at 99.9% span.

![Figure 5.16: $V_r$ contours. 98% span. SW in its near-stall point and M5G at the same mass flow rate](image)

As was discussed in observation 2 in section 5.2, the radially inward velocity indirectly represents the area of blockage. It is noticeable from the comparison of the radial velocity contours of SW (Fig. 5.16 (a)) and M5G (Fig. 5.16 (b)) cases that, at the same mass flow rate, the presence of grooves does not only weaken the strength of radially inward velocity, but also shrinks the area, near the leading edge. The “vanished”
radial velocity travels in the opposite direction of blade motion and is moved to the trailing edge; it eventually strengthens the radially inward velocity in the area attached to the leading edge of the PS of blades for a small amount. This behavior can be termed as the segmentation of the leading edge tip-leakage vortex, which then redirects the incoming flow and leads to lower incidence angle for some rotor blades. As a reference, the average incidence angles at 2% of the blade pitch away from the leading edge of the tip of rotor blades, are 72.2284 (SW) and 70.7068 (M5G). The discussion in the aspect of incidence angle will be discussed again in the next section (section 5.4).

Figs. 5.17 – 5.21 will be used to help visualize the flow field in the tip-gap/groove region for the near-stall operating point. Figs. 5.18 and 5.19 are at a theta cut near the blade leading edge. At this point in space and time, the blade is approaching the first groove. Figs. 5.20 and 5.21 are also at a constant theta plane near the mid-chord of the
blade at the same point in space and time. Since the grooves are continuous around the circumference, the mid-chord location is between the second and third groove channels. In Fig. 5.18, the relative velocity vectors near the leading edge show the tip-leakage vortex as it begins to form without the presence of the grooves. The vortex can be seen to be relatively intense. In Fig. 5.19, the leading edge vortex is still visible, but its intensity appears to be diminished somewhat. It can also be seen in this figure, that the blade is beginning to pump fluid into the first groove. It is believed the pumping action of the blade “bleeds” off some of the fluid that would otherwise feed the formation of the vortex; this appears to weaken the strength of the vortex, which probably helps to suppress the tendency for instabilities to grow, thus extending the operating range.

Similarly in Fig. 5.20, the mid-chord location still shows the presence of the vortex. Though the vortex has diffused somewhat by this point, its influence is still noticeable. In Fig. 5.21, the blade is between two grooves. As such, it is beginning to pump fluid into the groove channel just ahead of it, while drawing fluid out of the groove channels that it has recently passed. This, too, appears to weaken the effect of the baseline vortex, which, again, probably contributes to the extension in range (or increase in stall margin).
Figure 5.18  Velocity vectors near leading edge of blade without grooves

Figure 5.19  Velocity vectors near leading edge of blade with grooves
Figure 5.20  Velocity vectors near mid-chord of blade without grooves

Figure 5.21  Velocity vectors near mid-chord of blade with grooves
On the other hand, this pumping of fluid into and out of the grooves does increase the loss of the machine, as reflected in the efficiency plot. In addition to the viscous losses, the radial fluid motion in the grooves as well as the circumferential motion does not contribute to the rotor’s primary purpose of pumping fluid in the axial direction (adding further to the loss). However, the degree of loss shown in the efficiency plot is quite small when considering the increase in stable operating range. It is; therefore, felt that further investigation into the flow physics of the grooves and the resulting effect on the tip-leakage flow of the rotor is needed.

However, as it was discussed in Chapter 1, it is interesting to know that there are certain types of compressors where grooves would not lead to operation range extension even though the segmentation of the leading vortex exists, but it could be another investigation that might reveal more phenomena.

**5.3.2 Changes of theta and axial velocities, and blockage**

Fig. 5.22 shows the radial planes with contours of theta velocity from near-stall SW and the corresponding M5G. From the pictures the reduction of $V_t$ is evident especially in the center of the high $V_t$ area of SW case. Because of the interaction between the main flow and the grooves, the reductions of both the strength and coverage of the high $V_t$ areas are probably the production of the radial transport of the $V_t$ and the loss because of the viscosity inside the grooves. As a reference, the averages of normalized $V_t$ are -0.700491 (SW case) and -0.799459 (M5G case) at 98% span, -0.716885 (SW case) and -0.849692 (M5G case) at 99.9% span. The high $V_t$ area relative to the rotor blades is mainly from the tip-leakage flow, which travels from the PS to the SS at 98% span. The
reduction of the area directly causes less blockage at the leading edge of the PS, leading to even lower $V_t$, since the flow now has more potential to be energized by the rotor blades. The reduction of blockage is visible in Fig. 5.23, which shows the relative total pressure contours at 99% span.

Axial velocity as a key monitor of the blockage will be helpful to visualize the changes brought by the grooves. The pumping and drawing of fluid into and out of the grooves reduces the intensity of the leading edge vortex as well as the theta velocity of a large portion of the tip-leakage flow, which eventually reduces the strength of its interaction with the main incoming flow.

By the overview on the 98% span for axial velocity contours of the SW near-stall condition and the corresponding M5G case (Fig. 5.24), one can correlate them with the previous radial and theta velocity comparisons. The center of the lower axial velocity area is smeared along with the stretched radial and theta transports of fluids. The side effect of the reduction of blockage is that the subsonic incoming flow after the normal shock will experience less acceleration at the leading edge on both the SS and PS, but to a small degree. The relative Mach number contours (Fig. 5.25) show the same pattern that the axial velocity contours have. It is more obvious from the relative Mach number contours that the “stilled” area in the center of the low large low velocity area receives some kinetic energy thanks to the existence of grooves. As a reference, the averages of the normalized axial velocity are 0.446111 (SW case) and 0.443037 (M5G case) at 98% span, 0.336672 (SW case) and 0.325175 (M5G case) at 99.9% span. Thus the average of axial velocity is reduced, but only by a small magnitude.
Figure 5.22  \( V_t \) contours. 98% span. SW in its near-stall point and M5G at the same mass flow rate.

Figure 5.23  \( P_{t,rel} \) contours. 99.9% span. SW in its near-stall point and M5G at the same mass flow rate.

Figure 5.24  \( V_x \) contours. 98% span. SW in its near-stall point and M5G at the same mass flow rate.
Together with radial velocity, the integrations of the radial transport of axial momentum term at the tip gap region (and from the entrance to the exit of the rotor passages) can be calculated. Fig. 5.26 illustrates the control volume that is used to calculate the magnitude of the transportations. They are -0.993413 (SW) and -0.058948 (M5G) in the first half chord, -0.18813 (SW) and 0.0438689 (M5G) in the second half chord. The decrease of radial transport of negative axial velocity will reduce the loss of streamwise velocity in the tip region and will lead to higher tolerance to net axial pressure force, as was discussed in the introduction chapter. This physical behavior will also be brought up again in section 5.4.

Figure 5.25  Mach_{rel} contours. 98% span. SW in its near-stall point and M5G at the same mass flow rate
As was shown in the previous section, the end-wall blockage grows when the machine is being operated near stall, and as it was noticed that the low relative total pressure at the exit plane is somehow shifted towards the PS due to the presence of grooves (Fig. 5.27, green area near end-wall), so is the blockage. The blockage at the exit axial plane seems also to be reduced by the grooves, but probably not to a significant amount. Fig. 5.28 shows the particle traces of the tip-leakage flows of the two cases which also clearly exhibit the reduction of the blockage which covers a large portion of the passage near the end-wall.
5.3.3 In terms of tip blade loadings and reversed flows

Figs. 5.29 and 5.30 are close views for the pressure and axial velocity contours, respectively. The pressure contours are shown in localized color to distinguish the changes, so they are not consistent with the colors used in previous pressure contours. It can be seen that the pressure at and after the first groove decreases near the SS, and that
the core of low static pressure area upstream of grooves is moved further upstream. The high pressure gradient at the tip gap region, which is believed to be the cause of tip-leakage flow with high theta and negative axial velocity components, will now be used together with grooves to relieve the adverse effects brought by the leakage flow. The pressure difference will not only push the flow from the PS to the SS, but also from main baseline part into the grooves.

![Figure 5.29](image)

(a) SW in its near-stall point  
(b) M5G at the same mass flow rate

Figure 5.29  Pressure contours shown in local colors. 98% span. SW in its near-stall point and M5G at the same mass flow rate

With the assistance of Cp plots at 98% and 99% spans (Fig. 5.30), it can be seen that, following each single groove, there is a rise of static pressure on the PS near the leading
edge, followed by a drop of static pressure on the PS. At the location of the each groove, an increase on the PS and a decrease on the SS can be noticed, in terms of their magnitudes. The static pressure rises on both the PS and the SS right after each groove, most noticeably after the first groove near the leading edge. A high gradient of static pressure on the rotor tip is thus moved downstream by the presence of grooves, which indirectly shifts the point of reversed flow by 3.8% chord to the same direction (Fig. 5.31). Besides, the center of the second lowest static pressure area is moved upstream by the presence of the first groove, which leads to higher blade loading at that chord. The magnitude of the tip blade loading is, however, not significantly reduced by the presence of grooves, if at all.

(a) 98% span
(b) 99% span

Figure 5.30  Cp of the rotor blades at 98% and 99% spans. SW in its near-stall point and M5G at the same mass flow rate

Beside the pressure difference, the shifting of the separation point can also be explained by the increased tolerance at the tip gap boundary region (refer back to section 5.3.2).
Fig. 5.31  $V_x$ contours. 98% span. SW in its near-stall point and M5G at the same mass flow rate

Fig. 5.32 shows the streamlines across the reversed flows. As is just discussed, the location is moved to downstream. Besides, in comparison with Fig. 5.10, the strength of the vortex also seems to be weakened somehow, and incoming flow is less interacted with the vortex. Different from the SW case, the flow then joins the main stream but is dissipated by the radial and circumferential transport by grooves, which diverts its direction towards PS.
5.3.4 Loss

Fig. 5.33 shows the entropy distributions which serve as the indicator of loss in the machine. Again the patterns of the entropy distributions are similar to the axial velocity contours, which roughly indicate the trace of the motion of tip-leakage flow and the separation on the mid-chord of SS.

Not surprisingly, most of the high entropy is associated with the tip-leakage flow, which indirectly means that the loss (low entropy) is not evident in the main flow (Fig. 5.28). The highest entropy occurs at the separation areas as is indicated in Fig. 5.33 as well. It is felt that the grooves actually mitigate the growth of entropy in the main flow, but the additional loss inside the grooves complements the reduced part; therefore the efficiency improvement is not noticed at this mass flow rate, if there is any. The entropy contours are also presented in the axial cuts (Fig. 5.34). From the pictures only, slight changes on the main flow can be identified in the M5G cases, and the entropy created inside the grooves cannot be ignored, which can be visualized by Fig. 5.35.
Figure 5.33  Entropy contours. 98% span. SW in its near-stall point and M5G at the same mass flow rate

Figure 5.34  Entropy contours. Axial cuts. SW in its near-stall point (row #1 and #3) and M5G at the same mass flow rate (row #2 and #4)
5.3.5 Stalling processes of SW and M5G cases

The stalling process of the SW case can be visualized with Figs. 5.36 and 5.37. Fig. 5.36 shows the relative velocity vectors colored by $\text{Mach}_{rel}$, and Fig. 5.36 shows the streamlines of the tip-leakage flow, for both developing and developed periods. The velocity vectors show that the backflow starts in the middle of the passage where the blockage is (Fig. 5.36 (a)) and then spreads toward the upstream and downstream directions (Fig. 5.36 (b)). The fluid adjacent to the PS and SS appear to have positive axial velocity even at the developed stall condition. The tip-leakage streamlines again show that the backflow initiates from the middle of the passage near the end-wall (Fig. 5.37 (a)) and begins to spill around the leading and trailing edges below the blade tip (Fig. 5.37 (b)). Besides the backflow, it also appears that the tip-leakage vortex starts to break down during the stall process (Fig. 5.37 (b)).
In comparison, the stalling process of the M5G case has slightly different pattern. The flow in the middle of the passage near the end-wall appears to carry much less negative axial velocity component; however, the fluid adjacent to the SS near the mid-chord seems to spill forward during the process of stall (Fig. 5.38).

The tip-leakage flow also shows differences. During the stall developing process, the
trailing edge tip-leakage flow does not exit the grooves on the current passage but stays in the grooves until the next passage. The adjacent passage near the trailing edge seems to be covered by this secondary leakage near the end-wall (Fig. 5.39 (a)), which then evolves into backflow at the SS of the blade (Fig. 5.39 (b)). Besides, the intensity of the stalled area that covers most of the passage seems to be decreased by the presence of the grooves (Figs. 5.37 (b) and 5.39 (b)). Nevertheless, the stalling process of the M5G case shares two common behavior with the SW case, one is that the tip-leakage flow spills around the leading edge below blade tip, the other is the breakdown of tip-leakage vortex (Fig. 5.39 (b)).

![Velocity vectors colored with Mach_{rel}. 99% span. M5G’s stalling process](image)

(a) Developing  (b) Developed

Figure 5.38
5.3.6 Summary

In summary, the stall margin improvement brought by M5G was presented and its mechanisms were discussed. The mechanisms can be concluded with six observations:

1. Radial interaction between the flow near tip region and the grooves (pumping and drawing fluid into and out of the grooves) appears to weaken the leading edge vortex; it actually diffuses the vortex towards the trailing edge of the PS.

2. The incidence angle appears to be reduced because of the presence of grooves. The reason might be related to the reduction of leading edge vortex.

3. The relative pitch-wise velocity of flow at the near tip region appears to be reduced by its interaction with grooves, which is the result of the observation 1 above. The low-momentum flow then has less opportunity to block the main flow,
leading to increased average $V_x$ and $V_t$ near the end-wall of the passage. The change can be viewed in the aspect of relative total pressure at the span near end-wall.

4. The blockage at the exit axial plane of rotor passages is shifted towards the PS.

5. Radial transport of axial momentum is reduced, most noticeably in the half-chord region towards the leading edge.

6. The blade loading at the tip is changed, but the change does not clearly correlate with the improvement. However, the change of pressure at the tip seems to shift the separation point towards the trailing edge by a small amount.

The six observations listed can further be categorized in four consequential phenomena which could be illustrative for stall margin improvement: shifting of blockage, reduction of incidence angle, change of tip blade loading and improvement of radial transport of axial momentum. They will be discussed in the cases of single grooves.

Qualitative analysis of the stalling processes of SW and M5G cases shows that the SW case has the spike-like pattern according to Fig. 1.9 (Vo 2008), whereas the M5G does not. The difference is mainly the trailing edge back flow in the SW case. However, the stall pattern happens in SW case is after the peak of the total-to-static pressure plot, which indicates it actually carries modal inception. Besides, in contrast with the SW case where the reversed flow starts from the middle of the passage, the strong backflow of M5G case seems to initiate from the trailing edge. The similarities are that the stall mechanisms are accompanied with vortex breakdown and tip-leakage flow spillage below the tip from the leading edge. The stall inception will be discussed more in section 5.5.
5.4 Single Grooves at Eight Locations

As was presented in the 5.1 overview section, single grooves at eight locations (SG₁ – SG₈) are tested, and their SMI are shown in Fig. 5.40 together with M5G case. The red spot in the picture for SG₁ indicates its last operation point that the solver could run, but its mass flow rate oscillates periodically within a range; whereas the yellow spots indicate the last relatively stable operation points of groove cases. The unstable operation of SG₁ will be discussed more in section 5.4.1. The decrease of stall margin because of SG₁ is evident from the picture, and SG₂, SG₄ and SG₆ also appear to trigger the stall in an even earlier period. In contrast, SG₃, SG₅, SG₇ and SG₈ actually bring some improvements, where the magnitude of SG₃ case is noticeable. It is interesting to see that the sum of the gains of the grooves SG₃ - SG₈ individually is less than the SMI from multi-grooves (M5G); this is different from the results obtained by Houghton (Houghton 2011). However, the difference might be caused by the tests on different machines. The following discussion will be divided into two parts based on the current observation: the first part is for the single groove configurations that improve stall margin, while the other one serves as an attempt to explain the reason why the rest of the cases do not have their expected results. If not otherwise mentioned, group A will be investigated at SW’s near-stall mass flow rate (SG₃, SG₅, SG₇, SG₈), and group B will be presented with individual stall mass flow rate with SW’s corresponding case (SG₂, SG₆, SG₄, SG₈).
Figure 5.40  Stall margin improvement by groove configurations

The attempt to distinguish the changes brought by single grooves will follow the four phenomena speculated in section 5.3.6 one by one with more quantitative discussions.

5.4.1 Influence on blockage

Due to the small SMI by single grooves, perhaps the most distinct changes brought by grooves to the tip region flow field found in the current investigation can be illustrated by the relative total pressure contours. As was seen in section 5.3.2, the low total pressure area is located mainly in the first half chord of the passage near the end-wall, so that area will be concentrated on. Fig. 5.41 shows $P_{rel}$ contours of the first half chord at 99.9% span from group A. The low energy fluid where the grooves are present is successfully segmented by the M5G configuration, and part of it is transferred towards the PS as shown in yellow color. In contrast, $SG_e$ by itself affects the main flow underneath it but not too much on the flow downstream of it. $SG_e$ also segments the low $P_{rel}$ in the mid-
chord within a small area. \( \text{SG}_f \) and \( \text{SG}_g \) do not seem to influence the overall blockage near the end-wall.

![Figure 5.41](image1.png)

Figure 5.41 \( P_{t,rel} \) contours. 99.9% span. SW case and cases in group A

![Figure 5.42](image2.png)

Figure 5.42 \( P_{t,rel} \) contours. 99.9% span. Cases in group B and corresponding SW cases. Brackets indicate that the operations are at the near-stall mass flow rate of the case inside.

In comparison, Fig. 5.42 shows the cases in group B where the single grooves do not bring stall margin improvement and the corresponding SW cases that are run at their near-stall mass flow rates. In comparison with SW (\( \text{SG}_{b,h} \), brackets indicate that the
operations are at the near-stall mass flow rate of the case inside), SG$_b$ does alter the low $P_{t,rel}$ area but in a different direction: it shifts the part of fluid towards the circumferential direction only but not downstream. This behavior shrinks the effective tip-leakage vortex area, but it might strengthen the forward spillage and increase the opportunity to trigger stall prematurely. Similar behaviors can be found in SG$_a$ and SG$_d$ cases as well. In comparison with SW (SG$_a$), the low energy area is expanded towards the PS in the SG$_a$ case, followed by even larger low energy coverage near the end-wall. This might explain why SG$_a$ significantly decreases the stall margin. The circumferential transport of low energy fluid in SG$_d$ case results in larger separation near the SS, thus it would not be surprising to see if the stall initiates from the trailing edge in this case. SG$_b$ is not effective for the same reason that SG$_f$ and SG$_g$ are.

The observations from the $P_{t,rel}$ near the end-wall might suggest that the grooves near the leading edge are effective in altering the tip-leakage vortex. For grooves that would enhance the stall margin, the coverage from the tip-leakage flow should be transferred towards the downstream direction but not the circumferential-only or even the opposite.

Blockage at the exit plane of rotor passage can also be illustrative. The contours are divided into two groups just like before. As was discussed, the M5G shifts the blockage towards the SS in comparison with the SW case (Fig. 5.43). SG$_c$ shows the same pattern but with a smaller magnitude. The shifting cannot be clearly seen in the SG$_c$, SG$_f$ and SG$_g$ cases. In contrast, the $P_{t,rel}$ contours of cases in group B at the rotor passage exit planes exhibit the PS-direction shifting of the blockage (Fig. 5.44) except the SG$_h$ case where the change is not visible. This observation might suggest that the blockage at the exit plane should move towards the PS for the grooves to be effective. The shifting
towards the opposite direction might result in adverse effects.

Figure 5.43  $P_{t,rel}$ contours shown in local colors. Rotor passage exit plane with the SS on the left side and the PS on the right side. SW case and cases in group A

Figure 5.44  $P_{t,rel}$ contours shown in local colors. Rotor passage exit plane with the SS on the left side and the PS on the right side. Cases in group B and corresponding SW cases. Brackets indicate that the operations are at the near-stall mass flow rate of the case inside.

The change of the energy of the fluid near the leading edge in $SG_a$ and $SG_b$ cases is discussed in the beginning of this section. This part of fluid is not favorable as will be seen in the following context. As a result of the re-energized fluid near the leading edge, when the $SG_a$ is run in the unstable near-stall condition, the shock starts to oscillate and its position is different from blade to blade (Fig. 5.45, corresponding to the red spot in Fig. 5.40), which fluctuates the mass flow rate periodically (Fig. 5.46).
Figure 5.45  Mach$_{rel}$ contours representing the oscillations of shocks and blockages. 98% span. SG$_a$ at its unstable near-stall operation

Figure 5.46  Mass flow rate history of SG$_a$ case in unstable near-stall condition

When the back pressure is slightly raised from the near-stall condition, the overall flow field starts to break down from the leading edge, and the process can be visualized with the traces of particles released from the tip gap (Fig. 5.47). The reversed flow emanates from the tip gap and starts to interact with the groove, shock and incoming flow. As the flow travels through the groove to the adjacent blade, it begins to spill.
around it below the tip gap. The spillage prevents the incoming flow from passing through, which eventually leads to developed passage stall. The intensity of the leading edge spillage is much stronger than both SW and M5G cases (refer to Figs. 5.37 and 5.39). Fig. 5.48 shows the intensified leading edge spillage in terms of relative velocity vectors. The stall process will be discussed again in section 5.5.

Figure 5.47  Traces of particles from the tip gap. SG_a’s stalling process
5.4.2 Reduction of incidence angle

It is well known that the incidence angle is critical to the performance of rotor blades, and that high incidence angle is not favorable since it will cause separation and high blade loading. It was found in section 5.3.1 that the incidence angle to the blade tip is reduced by the M5G, which inspires the investigation on the changes of incidence angle in different cases. The full data that will be used is listed in Tables 5.1 and 5.2, where the negative differences mean that the incidence angle is reduced. Again the average incidence angles are evaluated at 2% of the blade pitch away from the leading edge of the tip of rotor blades.

It is shown in the following data that, M5G, SGb and SGc reduce the incidence angle to the rotor blade tip. SGa and SGD appear to increase the incidence angle for a larger amount. The single grooves near the leading edge do not seem to be effective on the incidence angle (SGe, SGf, SGg and SGh), where SGg is the only case that decreases the angle slightly. It is also shown that the incidence angles increase when the groove cases are operating near-stall, but the critical incidence angles vary from case to case. Fig. 5.49
shows the normalized reduction of incidence angle at the same mass flow rate for each case with normalized SMI, and that the reduction of incidence angle does not always coincide with the increase of stall margin.

Table 5.1 Incidence angles of single groove cases in group A, M5G case and the related SW case

<table>
<thead>
<tr>
<th>Case</th>
<th>Incidence angle when SW is near-stall</th>
<th>Incidence angle at SW near-stall</th>
<th>Difference</th>
<th>Percentage</th>
<th>Incidence angle at individual near-stall</th>
</tr>
</thead>
<tbody>
<tr>
<td>M5G</td>
<td>72.2284</td>
<td>70.7068</td>
<td>-1.5216</td>
<td>-2.11</td>
<td>72.6382</td>
</tr>
<tr>
<td>SGc</td>
<td>72.2284</td>
<td>70.8656</td>
<td>-1.3628</td>
<td>-1.89</td>
<td>71.6565</td>
</tr>
<tr>
<td>SGd</td>
<td>72.2284</td>
<td>72.2613</td>
<td>-0.0160</td>
<td>0.02</td>
<td>72.5183</td>
</tr>
<tr>
<td>SGf</td>
<td>72.2284</td>
<td>72.2143</td>
<td>-0.0141</td>
<td>0.02</td>
<td>72.5406</td>
</tr>
</tbody>
</table>

Table 5.2 Incidence angles of single groove cases in group B and related SW cases

<table>
<thead>
<tr>
<th>Case</th>
<th>Incidence angle of corresponding SW</th>
<th>Incidence angle at individual near-stall</th>
<th>Difference</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>SGa</td>
<td>67.5379</td>
<td>72.3684</td>
<td>4.8305</td>
<td>7.15</td>
</tr>
<tr>
<td>SGb</td>
<td>72.1139</td>
<td>70.709</td>
<td>-1.4049</td>
<td>-1.95</td>
</tr>
<tr>
<td>SGd</td>
<td>68.3827</td>
<td>72.0937</td>
<td>3.7110</td>
<td>5.43</td>
</tr>
<tr>
<td>SGh</td>
<td>72.1139</td>
<td>72.2133</td>
<td>0.0994</td>
<td>0.14</td>
</tr>
</tbody>
</table>
These facts might suggest that the reduction of incidence angle is not a direct result of, and is thus not a requirement for, stall margin improvement by circumferential grooves.

### 5.4.3 Change of tip blade loading

The trade of fluid with the grooves may result in changes to the tip blade loading as well. Plots in Fig. 5.50 suggest that the single grooves near the trailing edge ($SG_e - SG_h$) are not as effective on the blade loading as the ones near the leading edge ($SG_a - SG_d$). The effects may be described as

1. $SG_a$ reduces the blade loading in the leading edge before the shock; however, the blade loading is increased after the shock.

2. $SG_b$ reduces the pressure on PS near the leading edge as well as at the groove location. The pressure on the SS after the shock is reduced and increased again near the trailing edge.
3. $SG_c$ increases and then decreases the pressure on the PS near the leading edge. The pressure is decreased on both PS and SS at the groove location and is then both increased immediately downstream of the groove.

4. $SG_d$ reduces the blade loading upstream and at the groove location but increases the pressure on the SS downstream of the groove.

5. $SG_e$, $SG_f$ and $SG_g$ do not affect the blade tip loading too much. The pressure on the PS is slightly reduced where the grooves are.

6. $SG_h$ slightly decreases the pressure on the SS downstream of the shock.

However, the straightforward relationship between the stall margin improvement and the changes on the blade loading cannot be identified by looking at these effects.
Figure 5.50  Blade loading near the tip, 99% span
5.4.4 Improvement of radial transport of axial momentum

Fig. 5.51 shows the data of normalized improvements of radial transport of axial momentum with normalized SMI as the reference. Radial transport of axial momentum is believed to be a source to balance the adverse pressure thus, potentially increasing the stall margin. It is, however, found not to be strongly related to the trend of SMI. As is shown in the figure, the single grooves near the leading edge all increase the radial transport of axial momentum, even when the configuration actually reduces the stall margin. It is thus believed that the improvement on the radial transport of axial momentum by itself cannot serve as the main mechanism of why the stall margin enhancement can be achieved. Fig. 5.52 shows the normalized resulting velocity changes at the 98% and 99.9% spans. The plots are divides into groups A and B.

![Figure 5.51](image)

**Figure 5.51**  Normalized improvements of radial transport of axial momentum with normalized SMI. Values and SMI are both normalized by the M5G case.
Figure 5.52 Normalized changes of velocity components in three directions of groove cases with normalized SMI. Values and SMI are both normalized by the M5G case.

In Fig. 5.52, it is shown from the plots of $V_x$ that it tends to be increased at 99.9% span as a beneficial sign with SGb as an exception. The plots of $V_t$ in group A seem to be
consistent with the trend of SMI, but it again fails to predict the trend in group B, where \( V_t \) near the end-wall also seems to be energized by \( SG_b \), \( SG_d \) and \( SG_h \). A similar trend is observed in the plots of \( V_r \), where it is indicated that the interaction between the main flow and the grooves becomes weaker when the grooves are moved aft.

### 5.4.5 Summary

It is found from the current simulation that the single groove located at the c location (20% chord) is the most effective one. Single grooves located upstream of \( SG_c \) or immediately downstream of it could precipitate the stall even before that of the baseline case. The single groove located in the mid-chord does not improve the stall margin the most in the current configuration.

Qualitative and quantitative analysis on single groove cases leads to several conclusions about the mechanisms of grooves. The first statement is that the blockage shall be shifted towards the downstream and pressure side for the grooves to be effective. The second is that the reduction of incidence angle is not always a result of stall margin improvement. The third is that the improvement on the radial transport of axial momentum is not found to be strongly related to the improved stall margin either. The last but not the least, the spanwise velocity shall be energized thanks to the presence of grooves. The result that the blade tip loading is not reduced is found to be consistent with the previous one for M5G case.

These results stress the importance of groove location and give some insight for future investigation, such as groove depth. Computational analysis has the potential to give a comprehensive view of the flow field and is thus recommended for design issues,
especially like the design of groove treatments for improving stall margin with a small efficiency penalty.

5.5 Aspects of Stall Inception

As was discussed in the introduction chapter, stall inception is found to be critical in deciding whether or how much the casing treatment would enhance the stall margin on a compressor/fan stage. On one hand, the behavior of casing treatments such as circumferential grooves seems straightforward in the sense that it provides a path for flow near the end-wall to pass from the PS and be recirculated from the SS, thus presenting the potential to adjust the adverse situation nearby. On the other hand, however, casing treatments would inevitably change the size of tip gap in some places, leading to possible switching between different stall inceptions since the tip gap size is one of the most significant parameters in deciding its stall mode (refer back to Fig. 1.4, for example).

The baseline configuration is not deemed to have a stable modal inception during its stall process. The stagnation-to-static pressure rise $\Delta P_{t-s}$ (or ratio $\frac{P_{t}}{P_{s}}$ in this context) is proposed by Camp and Day (1998) to be the indicator of stall modes. In other words, if the stagnation-to-static rise characteristic has a negative slope at the calculated stall limit, then spikes will be initiated before modal activity has its opportunity to develop; otherwise modal oscillations shall be considered to be the main phenomenon. Fig. 5.53 presents the $\frac{P_{t}}{P_{s}}$ plots from the SW, M5G and SGa cases where the data of static pressure is collected from the middle axial plane of the whole fan stage. It is clearly shown from the plots that both the SW and M5G plots roll over the peak of the characteristic and
reach the positive slope before stall; therefore both of them shall be considered to have modal type of stall, although the fan stage has a small tip gap (0.8% tc). However, the stall happens rapidly if the mass flow rate is decreased more, within about 12 revolutions (Fig. 5.54), which suggests that the SW might have “destabilising mode” stall inception. The relatively small SMI obtained by M5G thus seems to agree with the results shown in Fig. 1.8; that is, much smaller SMI could be reached by the grooves when the SW presents modal pattern of stall rather than spike type.

Figure 5.53  Stagnation-to-static pressure ratio $\frac{P_{st}}{P_s}$ plots of SW, M5G and SGₐ cases
Further investigations of single groove cases suggest that the possible “destabilising mode” stall inception is unstable when casing treatments are added to the baseline configuration. The stall pattern might even be changed from modal to spike inception by casing grooves. The pressure rises $P/P_r$ of single groove cases (except SG$_a$, which has been shown in Fig. 5.53) at their last two stable operations are plotted in Fig. 5.55, and they are summarized in Table 5.3. The data suggests that the mode pattern stall that SW has is not changed by M5G, SG$_c$, SG$_f$ and SG$_h$, but is switched to spike pattern for the others. Although the alteration of stall pattern seems not related to stall margin enhancement in this configuration (at least by the current simulation), it could cause the machine to stall earlier, such as SG$_a$ (Fig. 5.53).

The adjustment of partial tip gap size, the recirculation of fluid through the grooves, and resulting changes such as altered incidence angles and blockage, etc., would be contributors to the transformation of stall pattern. It might be that one mechanism is dominating over the other that finally leads to increase or decrease in stall margin. The mechanisms are thus felt to be more complicated than solely the aspects discussed in
section 5.3 and 5.4; they could rather be related to the possibly modified stall pattern and the resulting favorable or adverse effects.

Table 5.3 The slope of stagnation-to-static pressure ratio at the near-stall point of each case

<table>
<thead>
<tr>
<th>Case</th>
<th>$\frac{P_t}{P_s}$ at the last point</th>
<th>Case</th>
<th>$\frac{P_t}{P_s}$ at the last point</th>
</tr>
</thead>
<tbody>
<tr>
<td>SW</td>
<td>Positive</td>
<td>M5G</td>
<td>Positive</td>
</tr>
<tr>
<td>$SG_a$</td>
<td>Negative</td>
<td>$SG_c$</td>
<td>Negative</td>
</tr>
<tr>
<td>$SG_b$</td>
<td>Positive</td>
<td>$SG_f$</td>
<td>Positive</td>
</tr>
<tr>
<td>$SG_c$</td>
<td>Positive</td>
<td>$SG_g$</td>
<td>Negative</td>
</tr>
<tr>
<td>$SG_d$</td>
<td>Negative</td>
<td>$SG_h$</td>
<td>Positive</td>
</tr>
</tbody>
</table>

Figure 5.55 Stagnation-to-static pressure ratio $\frac{P_t}{P_s}$ plots of single groove cases except $SG_a$

It is also valuable to discuss in the aspect of stall process. As was discussed in section 5.3.5, both the spillage below tip from leading edge and backflow below tip from trailing edge of tip clearance flows happen when the baseline configuration stalls (Fig. 5.37). Although the two phenomena coincide with the spike pattern criteria, they were observed after the stagnation-to-static pressure rise characteristic reaches zero slope, which thus
does not suggest the baseline configuration to have spike inception (Vo, 2001). The trailing edge backflow is not observed in either M5G (Fig. 5.39) or SGa (Fig. 5.47) cases, but in different senses: M5G weakens the trailing edge reversed flow by reducing the blockage near the end-wall, whereas SGa seems to shift the backflow more towards the leading edge and thus strengthen the leading edge spillage instead. It is also noticed that, even though SGa is deemed to have spike inception, its stall process is different from the criterion established by Vo (2001) in that the backflow does not travel below the blade tip at the trailing edge. The reason might be explained by the absence of spike inception capturing model in the current simulations, or it might suggest further validation of the second criterion of trailing edge backflow. In fact, Vo himself suggests this criterion to be examined experimentally on multi-stage compressors that can be made to exhibit both spike and modal stall inceptions (Vo, 2001), since it has not been rigorously verified on common rotor geometries exhibiting spike disturbances. Regardless of the actual type of stall, future work on the examination of the stall process of this fan stage is thus recommended to serve as a theoretical preparation to control the adverse effects that might trigger the stall.

The importance of stall inception also raises the necessity of the verification for current two passage simulation. The current simulation might have run further than the machine actually could, since a disturbance of sufficient amplitude introduced a higher mass flow could trigger spike formation (Vo, 2001). Stall inceptions shall be considered to be asymmetric physical phenomena. As was discussed, modal stall inception is essentially a circumferential perturbation which would be superimposed to the main flow field, while spike stall means a finite stall cell is the first to appear. Either one will
destroy the symmetry of the flow field. A full-wheel or multiple passages simulation with specialized models designed for stall inceptions (for example, Gong et al. (1998)), or even an experimental test is recommended to examine the current result before further casing treatment-based optimization be considered on this fan stage.
CHAPTER 6
SUMMARY AND FUTURE WORK

In this thesis, a computational technique, the SimCenter’s in-house flow solver Tenasi, has been used to investigate the effects of casing treatments. The flow solver employs a discrete Newton relaxation approach to solve the unsteady RANS equations. Two types of passive flow control have been investigated to determine their impact on the stability of a fan stage. The first method used two continuous rings extending a small distance inward from the casing to form a channel just upstream and/or downstream of the leading and trailing edges of the rotor blade. This type of configuration does not work well on the extension of operating point. The other method is the so-called circumferential groove casing treatment. This method has been shown to work by improving performance for core compressors, so it was reasonable to expect similar improvements for the fan stage in this study. An extension in operating range of about 5% was seen in the initial simulation results with a small efficiency penalty.

The motivation of testing the effects of single grooves led to the investigation of the effect of computational grids on the simulation results. The way to insert viscous layers before adding grooves, which keeps the baseline part identical, is believed to be appropriate. After an iteration of mesh refinement experiments, a fairly fine grid was chosen to minimize the unfavorable effect from the grid.

The simulations of the baseline smooth wall and the five circumferential groove cases
were re-conducted as well as eight single groove configurations, with fine grids. Based on the new simulation results, a range extension of about 2.3% relative to the baseline case was found to be achieved by the five groove configuration, with 0.43% efficiency decrease at most.

Observations from several different vantage points were used to explain the effect of casing grooves on baseline near-stall operations. It was observed that as the operating point gets closer to stall, the leading edge tip-leakage flow is strengthened by the increasing pressure gradient. The interaction between detached normal shocks, tip-leakage vortex, and blade boundary layers results in an unfavorable low-momentum area in the passage near the end-wall. This can also be observed at the exit plane of the rotor passages. By adding the five grooves above the middle of the blades, it was found that the leading edge vortex is weakened because of the drawing and pumping of fluid, which therefore leads to smaller low-momentum area.

Further numerical experiments of single grooves lead to several conclusions based on qualitative and quantitative analyses. The first is that the blockage should be shifted towards the downstream and pressure side for the groove to be effective. The second is that although the middle five grooves configuration reduces the incidence angle, the reduction does not always indicate stall margin improvement. The third is that the improved radial transport of axial momentum is not found to be strongly related to the stall margin improvement. Besides, the effect of the grooves on blade tip loading is investigated, but it may not be used to explain the mechanism of casing grooves on the stall margin improvement. In addition, the groove at 20% chord near the leading edge is found to be most effective. Single grooves located upstream of SGc or immediately
downstream of it could precipitate the stall in an even earlier stage than having no groove at all.

The stall processes are examined for different cases. It is found that when the baseline configuration stalls, its tip-leakage flow forms leading edge forward spillage and trailing edge backflow below the blade tip. The stall tends to be leading-edge-spillage dominated when the grooves are present. The similarities are that the stall mechanisms are accompanied with vortex breakdown and tip-leakage flow spillage below the tip from leading edge. It is also found that the stall pattern is also changed by the presence of some grooves in this configuration. The baseline smooth wall case is considered to have “destabilising mode” stall inception, which is found to be transformed into spike or mode stall that is more stable with grooves.

There are several recommendations of future work:

1. Verification of the current two passage simulation. Stall inceptions shall be considered to be asymmetric physical phenomena. A full-wheel or multiple passages simulation with specialized models designed for stall inceptions, or even experimental testing is thus recommended to examine the current result before further casing treatments-based optimization be considered on this fan stage.

2. Regardless of the actual type of stall, future work on the examination of the stall process of this fan stage is recommended to serve as a theoretical preparation to control the adverse effects that might trigger the stall.

3. The grid used can be further refined along the tangential lines of the rotor blades in order to resolve the wakes.
REFERENCES


VITA

Weiyang Lin was born in Rui’an, Zhejiang, China, to the parents of Zhenxiong Lin and Yuqian Huang. He is the fifth of five children, three older sisters and an older brother. He attended Shiyan Elementary and continued to Wansong Middle school and Rui’an High School in Rui’an, Zhejiang, China. After graduation, he attended South China Agricultural University. Weiyang completed the Bachelors of Science degree in July 2009 in Information and Computing Sciences under the Supervision of Mingliang Fang. Weiyang was accepted by the University of Tennessee at Chattanooga in the Computational Engineering Program with a graduate assistantship.