THESIS FOR THE DEGREE OF LICENTIATE OF ENGINEERING IN THERMO AND FLUID DYNAMICS

Simulation and Optimization of an Axial Compressor Considering Tip Clearance Flow

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Abstract

With ever-increasing demands for high efficiency in axial compressors, it has become important to consider more geometrical features of the manufactured component in the design phase. Many possibilities open if the level of fidelity of the computational model can be increased. A higher level of detail leads to enhanced performance of components, as the need to be conservative in the design phase is reduced. Better performance is important for reducing fuel consumption and the weight of the component and, consequently, decreasing the environmental impact.

An axial compressor consists of rotating and stationary blade rows, where a distance between the rotating blades and the inner casing (shroud) is called the tip gap or tip clearance and is required to avoid contact of the blades with the shroud during engine operation. Large tip gaps in relation to the blade height can typically be found in the rear stages of transonic compressors. If the size of the tip gap is large in relation to the blade height, it can affect the flow in the rotor passage significantly. Including a tip gap in the optimization process of a compressor stage can therefore be of importance even in the early design phase to find geometries that will reach the design point at the design rotational speed.

In this thesis, different turbulence models and wall modeling approaches are used to calculate the flow in a transonic compressor stage with a large tip clearance. The benefits of including the tip clearance in the optimization process of a transonic compressor are shown and discussed. It is shown that considering the tip clearance in the optimization process is important to be able to reach the specified design point. Furthermore, from the optimization results it is shown that redistribution of the flow as a result of blockage in the tip region impacts the design variables over the entire span.

Keywords: compressor, design, CFD, tip clearance, turbulence models, validation, optimization

List of publications

This thesis is based on the work contained in the following publications:

I M. Lejon, L-E. Eriksson, N. Andersson and L. Ellbrant, 2015, Simulation of Tip-Clearance Effects in a Transonic Compressor, *Proceedings of ASME Turbo Expo 2015*, June 15–19, Montréal, Canada

II M. Lejon, N. Andersson, L. Ellbrant and H. Mårtensson, 2015, CFD Optimization of a Transonic Compressor Stage with a Large Tip Gap, *22nd ISABE Conference*, October 25–30, Phoenix, USA

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I would like to thank my family for always supporting me. Finally, I would like to thank my wonderful girlfriend Linda for being there.

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The noblest pleasure is the joy of understanding
- Leonardo da Vinci

Nomenclature

C_d	Discharge coefficient
Re	Reynolds number
U	Blade speed
U_{shroud}	Relative velocity of the shroud
V	Velocity
c	Chord
g	Tip clearance size
h	Enthalpy
p	Pitch
s	Span
β	Relative flow angle
γ	Stagger angle
ϕ	Flow coefficient, V_x/U
ψ	Stage loading coefficient, $\Delta h_0/U^2$

Subscripts

0	Total
1	Inlet
p	Pressure side
s	Suction side
x	Axial direction

Abbreviations

CFD	Computational Fluid Dynamics
LES	Large Eddy Simulation
NSGA-II	Non-dominated Sorting Genetic Algorithm-II
RANS	Reynolds Averaged Navier-Stokes
SAS	Scale-Adaptive Simulation
SST	Shear Stress Transport
URANS	Unsteady Reynolds Averaged Navier-Stokes

Contents

1 Introduction

The axial compressor is an essential part of a modern aircraft engine and is made up of a number of rotating and stationary blade rows. Work is done on the air by the compressor before it enters the combustion chamber, and a number of considerations, aerodynamic and structural, need to be taken into account in the design phase of the component.

An axial compressor consists of rotating and stationary blade rows, where a distance between the rotating blades and the inner casing (shroud) is called the tip gap or tip clearance and is required to avoid contact of the blades with the shroud during engine operation. Thermal expansion of the rotating blades and elongation of blades due to centrifugal forces are factors that need to be taken into account when setting the tip clearance for a blade row. Furthermore, g-forces e.g. during landing can momentarily alter the tip clearance in an aircraft engine.

At the tip of an aircraft wing, the pressure difference on the pressure and suction sides gives rise to a tip vortex. Similarly, the pressure difference of the pressure and suction surfaces of a compressor rotor near the tip gives rise to an air flow through the tip clearance. In an axial compressor, the proximity of the shroud and flow from nearby blade passages adds another dimension to the complexity of the flow in this region.

The tip clearance can have a large effect on performance. As it increases in size, its influence ranges from negligible (for a very small tip clearance) to substantial, where it has a considerable adverse effect on efficiency and on the stable operating range of the entire turbo machine. Although the tip clearance size is usually on the order of millimeters or parts of a millimeter, it can be responsible for a large part of the total losses [1].

1.1 Problem definition

Accurately predicting the effects of tip clearance flow on performance poses a great challenge for the axial compressor engineer. At the same time it cannot be neglected if it is suspected to have a large impact.

A number of decisions must be made when doing Computational Fluid Dynamics (CFD) calculations of an axial compressor stage. The first question with respect to the tip clearance is whether the clearance should be included in the simulation model at all. If it is included, different degrees of simplification can be made to the computational domain in order to simplify the grid generation process, possibly at the expense of some important flow physics. When the computational domain has been generated, a number of important choices remain. Should a steady or time dependant solution be computed? Which turbulence model should be used? How should the boundary layers be resolved?

The multitude of aspects to consider makes it difficult to give a clear answer as to what choices to make, and the choices will depend largely on available computational resources, available time and the purpose of the calculations. The present thesis aims to give a foundation on which to base such decisions, including an analysis of how the design of an axial compressor stage can be influenced by taking the tip clearance into account.

1.2 Scope of work

The main focus of this thesis is on modeling of tip clearance flow and its impact on compressor design. Tip clearance flow and its impact on compressor performance is discussed, giving the reader an introduction to the work that has been published in the field.

2 Optimization approach

The optimization approach utilized in Paper II is based on the work done in [2] and will here only be described in brief. The design variables used for the rotor and stator blades in Paper II are the leading and trailing edge blade angles and the blade stagger angle at three spanwise positions (10%, 50% and 95%), resulting in a total of 18 design variables.

The transonic compressor stage is optimized using a Non-dominated Sorting Genetic Algorithm (NSGA-II) [3] in modeFrontier^{TM} to meet design requirements at a specified design point. An initial design set is first generated by latin hypercube sampling, which ensures good coverage of the design space. A radial basis function is then generated based on results of CFD calculations of the initial design set. Ten designs are chosen from a converged Pareto front obtained from the optimization made on the response surface. These ten designs are evaluated using CFD calculations and are subsequently used to update the response surface. This process is repeated until the response surface is considered converged. The optimization approach, illustrated in Fig. 2.0.1, is described in more detail in [4], where it was shown to be effective in finding compressor blade designs with high performance.

Two design objectives are used to rank the performance of the stages: polytropic efficiency at the design point and part speed stability. The polytropic efficiency at the design point is an appropriate objective since this is the operating condition at which the compressor stage is designed to operate most of the time. Part speed stability is used as the second objective to facilitate stable operation as the engine is throttled.

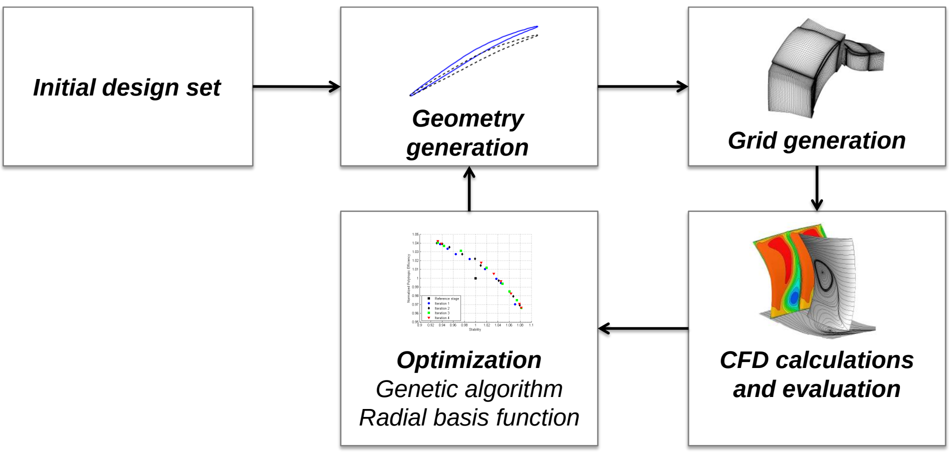


Figure 2.0.1: *Simplified overview of the optimization set-up*

3 Tip clearance flow

Tip clearance flow, illustrated in Fig. 3.0.1, is mainly driven by the pressure difference bwtween the pressure and suction sides of the blade at the tip [5]. This means that aerodynamic loading has a large impact on the strength of the tip clearance flow. It is stated in [1] that the average pressure along the chord on the suction side of the rotor blade at the tip is almost independent of the tip clearance size. Separation and re-attachment of the flow can occur in the tip clearance as discussed in [6], which has an effect on the mass flow through the clearance [7]. Part of the flow that passes through the clearance rolls up to a vortex, referred to as the tip clearance vortex. An example of a tip clearance vortex trajectory in a blade-to-blade view is illustrated in Fig. 3.0.2.

For very small tip clearances, the main contribution to losses associated with the tip clearance is caused by viscous losses generated at the shroud. As the clearance increases in size, losses due to mixing of the tip clearance flow become dominant [8]. It was shown experimentally in [9] that a small tip clearance can have a beneficial impact on performance compared to a sealed tip, and it was shown in [8, 9] that an optimal tip clearance size can exist. In [8], the existence of an optimal tip clearance is explained in terms of the competing loss mechanisms, viscous and mixing losses for small and large tip clearances, respectively. For some clearance, the sum of these two loss sources is at a minimum (in the range 0.1–1% of the blade height for the compressors discussed in [8]).

3.1 Impact on stability margin

An increase in tip clearance size can significantly reduce the stability margin of a compressor stage, as shown *e.g.* in [10]. The topic of stall associated with tip clearance flow is discussed in numerous papers, *e.g.* for a subsonic compressor in [11] and for a transonic compressor in [12].

The onset of stall in a transonic compressor was investigated in [12] using Large Eddy Simulations (LES) and experimental measurements. The stall process for the investigated

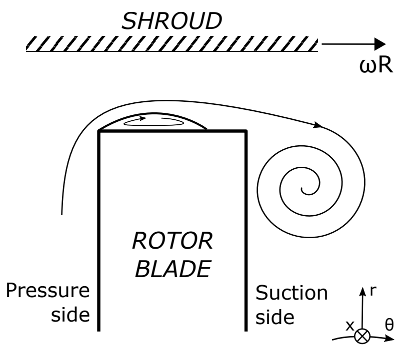


Figure 3.0.1: *Tip clearance flow illustration in a relative frame of reference*

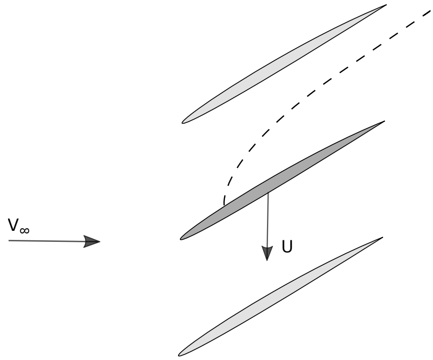


Figure 3.0.2: *Example of a tip clearance vortex trajectory shown as a dashed line*

compressor is well described and will here be summarized.

Random oscillation of the tip clearance vortex and its interaction with the passage shock is described in [12] to be the main trigger for the occurrence of a rotating stall cell. The onset of the rotating stall cell is explained by the formation of a stall cell downstream of the sonic plane in the tip region, where the region of low momentum influences the adjacent passage. The route from a near stall operating condition to the formation of a rotating stall cell is described as a step-by-step process based on LES calculation results.

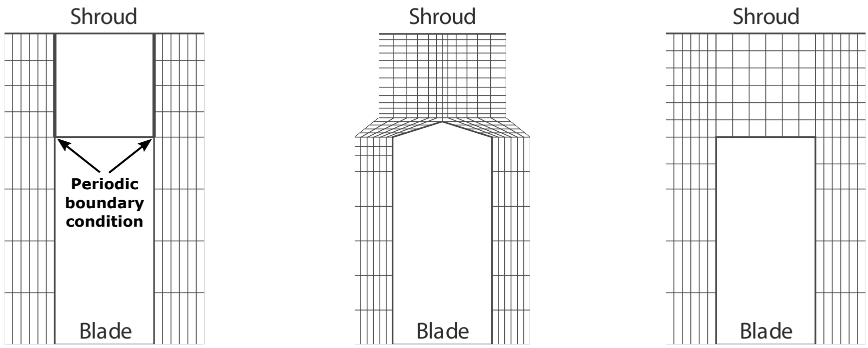
1. The passage shock fully detach.
2. The tip clearance vortex trajectory moves forward to the leading edge. Reversal of tip clearance flow occurs at the trailing edge, impinging on the pressure side of the adjacent rotor blade.
3. Tip clearance flow spill over the leading edge of the adjacent blade.

The effect of tip clearance flow is not only local, as described above, but can affect the flow far away from the tip region as shown in [9] and in Paper II. In [9], a change of the tip clearance size affected a separation at the hub, effectively increasing the stall margin of the rotor. A large separation at an end-wall, such as the rotor hub, can occur when the flow is diffused to a level at which separation occurs, similar to separation caused by a high level of diffusion in a duct. If the tip clearance increases in size, a larger region of low momentum fluid at the tip will limit the overall passage diffusion, thereby reducing the separation in regions away from the tip.

3.2 Tip clearance modeling

The tip clearance of a compressor rotor is typically modeled in one of three ways. The tip clearance can be set as a periodic boundary, it can be gridded using a "pinch-tip" approach, or it can be gridded using a multi-block approach.

By applying a periodic boundary condition over the blade tip, see Fig. 3.2.1a, the tip clearance is not resolved but still allows for a mass flow from the suction side to the pressure side. Viscous losses and mixing effects over the blade tip are neglected. Flow separation in the tip clearance, which in reality would cause blockage if the tip has a finite thickness, can be accounted for by reducing the size of the tip clearance. However, since the flow sees the blade tip with a periodic boundary as a sharp edge, there will be a contraction of the flow [7]. The relation between the size of the modeled tip clearance and the actual geometrical tip clearance must be evaluated on a case-to-case basis. The periodic boundary condition approach was used for compressors in *e.g.* [13–15] and for a two-stage turbine already in 1990 in [16].



(a) *Periodic boundary* **(b)** *Pinch-tip model* **(c)** *Multi-block approach*

Figure 3.2.1: *Tip clearance modeling approaches*

The pinch-tip approach requires a zero blade thickness at the tip. In order to achieve this, the blade tip corners can be rounded or the tip geometry can be changed to a wedge shape to extend the computational grid into the tip clearance as shown in Fig. 3.2.1b. This is done to avoid adding complexity, but often at the expense of quality, to the computational grid. The pinch-tip approach was used *e.g.* in [15, 17–19]. Similar issues as for the periodic boundary can be seen, which are discussed in [7].

The multi-block approach where the tip clearance is fully resolved with a separate grid block has become more and more common as the level of fidelity of simulations has increased. The approach is illustrated in Fig. 3.2.1c. A reason why the multi-block approach is attractive is that it requires no modification of the blade geometry. The multi-block approach was used *e.g.* in [20–22].

The study presented in [6] compared the three approaches described above for a low speed compressor. The tip clearance size was reduced for the pinch-tip and periodic boundary condition approaches since flow separation was suspected to occur along the blade tip. The pinch-tip model overpredicted the total losses, while using the periodic boundary condition in the tip clearance resulted in an underprediction. The best overall agreement with experimental data was obtained using the multi-block approach.

The multi-block approach was compared with a periodic boundary condition over the tip clearance in [23] for the transonic compressor rotor NASA Rotor 37, which has a tip clearance of 0.45% of the blade height. It was discussed that expansion occurs over the tip clearance. Consequently, the effect of separation on vena contracta was not considered when the periodic boundary condition was used. In general, results presented in the study for a periodic boundary condition were in good agreement with results obtained with the multi-block approach.

The number of grid points in the radial direction in the tip clearance region varies considerably in the literature depending on the computational power at the time, the tip clearance size, the tip clearance modeling approach and the wall modeling approach. A periodic boundary condition and 2 grid nodes were used in [13] and [14], while the multi-block approach, which compared well with experimental data, resolved a 1% tip clearance size to blade height using 31 grid nodes in [22].

3.3 Turbulence models

The predicted performance of an axial compressor stage was shown to be largely affected by the choice of the turbulence model in Paper I.

NASA Rotor 37 was investigated in [17] with steady state calculations using the k-ϵ and k-ω turbulence models, and with unsteady calculations using the k-ϵ turbulence model. The tip clearance was modeled with a pinch-tip approach (see Section 3.2), and wall functions were used for all calculations. The best agreement to one-dimensional experimental data was found for unsteady calculations using the k-ϵ turbulence model. Comparing the steady state results, it was found that the k-ϵ turbulence model underpredicted the total pressure ratio and total temperature ratio, and slightly overpredicted the adiabatic efficiency. The k-ω turbulence model showed an overprediction of the total pressure ratio, total temperature ratio and the adiabatic efficiency.

Simulations were made of a half-scale six stage axial compressor in [22] with tip clearances around 1% of the blade span. The tip gap was resolved using a multi-block approach, and a y+ value of around 1 was used for the finest mesh. The k-ϵ and the k-ω Menter Shear-Stress Transport (SST) turbulence models were compared in the study. Furthermore, RANS and URANS simulation results obtained using the k-ω Menter SST turbulence model were compared with experimental results. It was discussed that the k-ϵ turbulence model generally overpredicts the stall margin, while the k-ω Menter SST model generally underpredicts it. It was also discussed and shown that the drawback of the k-ω Menter SST turbulence model in terms of underpredicting the stall margin could be alleviated by using the re-attachment model of Menter [24]. Furthermore, it was shown that the performance at the design point was relatively insensitive to the choice of turbulence model, grid size and whether or not the simulation was done as steady state or unsteady. The choice became more important near stall, where the slope of the speedline for the unsteady simulation was closer to the experiment.

3.4 Wall modeling approach

The choice of how to resolve the boundary layer was shown in Paper I to play an important part in the performance prediction of an axial compressor stage. Two approaches were analyzed: the k-ϵ turbulence model using wall functions and the k-ϵ turbulence model using Chien's low-Reynolds formulation, where the first grid node away from the wall is placed in the viscous sub-layer to accurately resolve the boundary layer. Using wall functions was shown to predict a thicker boundary layer compared to the low-Reynolds formulation. A thicker boundary layer decreases the turning over the rotor (increase the deviation angle), which reduces the work done by the blade on the fluid. Performance parameters such as total pressure ratio and polytropic efficiency predicted using the low-Reynolds model show a closer agreement to the experimental measurements in Paper I compared to wall functions.

3.5 Tip clearance loss

Expressions for the change in efficiency caused by viscous shear at the shroud and mixing loss (using Denton's leakage mixing model [25]) are shown in [8], here reproduced here as Eq. 3.5.1 and Eq. 3.5.2.

$$\Delta\eta_{shear} \sim \mathrm{Re}^{-1}\left(\frac{g}{s}\right)^{-1}\left(\frac{c}{s}\right)^{2}\left(\frac{U_{shroud}}{V_1}\right)^{2}\frac{\phi^2}{\psi\cos^3\beta_1} \tag{3.5.1}$$

$$\Delta\eta_{mixing} \sim C_d\left(\frac{c}{p}\right)\left(\frac{g}{s}\right)\left(1-\frac{V_p}{V_s}\right)\sqrt{1-\left(\frac{V_p}{V_s}\right)^{2}}\frac{\phi^2}{\psi\cos^3\beta_1} \tag{3.5.2}$$

Where Re is the Reynolds number, g/s is the ratio of the tip clearance size to the span, c/s is the chord to span ratio, c/p is the chord to pitch ratio, U_{shroud} is the relative velocity of the shroud, V_1 is the inlet velocity, β_1 is the inlet relative flow angle, ϕ is the flow coefficient, ψ is the stage loading coefficient and C_d is a discharge coefficient. V_p/V_s is the ratio of the velocity on the pressure side to the velocity on the suction side, which is approximated in [8] as

$$\frac{V_p}{V_s} \approx \sqrt{1-2\frac{\psi}{\phi}\left(\frac{c}{p}\right)^{-1}\frac{\cos^2\beta_1}{\cos\gamma}} \tag{3.5.3}$$

where γ is the stagger angle. Good agreement between the computed level of loss with the loss calculated using Eq. 3.5.2 was found for tip clearance sizes up to 3.4% of the blade height [8]. A number of observations can be made from these expressions in terms on the dependence of the tip clearance loss on other variables. To decrease the efficiency loss due to shear stress, it would be benefitial with low values for the Reynolds number, c/s, U_{shroud}, β_1 and ϕ, and high values for g/s, ψ and V_1. To decrease the efficiency loss due to mixing it would be beneficial to have low values for g/s, γ, ψ and ϕ, and high values for c/p and β_1. An example of the effect of varying ψ, β_1, ϕ and c/p one at a time (to visualize the trend) on the mixing loss is shown in Fig. 3.5.1. By decreasing the stage loading coefficient, ψ, the mass flow through the clearance would decrease and the mixing losses would be lower.

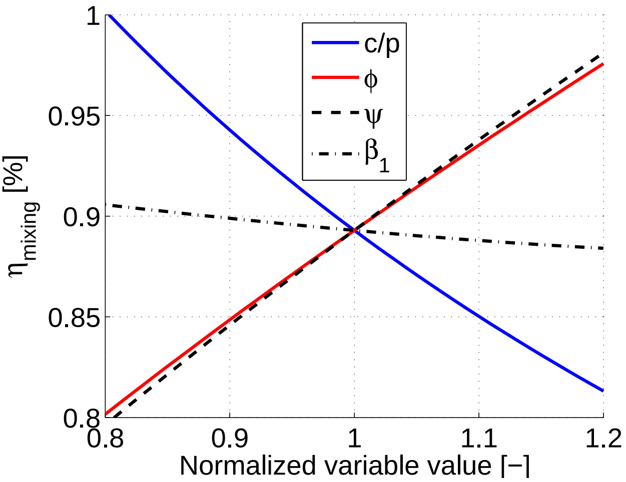


Figure 3.5.1: *Effect of indidvidual variables on mixing loss*

4 Concluding remarks

The following subsections present conclusions and lessons learned by the author, in part from the literature study, but mainly from the work presented in Paper I and Paper II.

4.1 Meshing

- The tip clearance should be resolved with a separate grid block to accurately represent the geometry, to facilitate good grid quality and to avoid making assumptions about flow features in the clearance. As discussed in Section 3.2, using a pinch-tip approach or a periodic boundary condition at the tip can require artificially changing the size of the tip clearance to match experimental data in terms of mass flow through the clearance.

- If wall functions are used, it can be a good rule-of-thumb to use evenly spaced grid points in the radial direction in the tip clearance region. Placing the first node away from the shroud and the blade tip at a distance suitable for wall functions determines the radial grid spacing in the clearance.

4.2 Turbulence model and wall treatment

- The k-ϵ turbulence model with wall functions was shown to give overall good agreement with experimental data in Paper I. The overall underprediction of the stage performance compared to experimental results indicates that this approach may give a conservative performance estimate. The relatively low computational time needed, compared to Chien's low-Reynolds model evaluated in the study, makes it suitable to use in an optimization set-up. Ranking of designs in terms of efficiency was evaluated in Paper II using wall functions and the low-Reynolds model, where it was shown that the same ranking was predicted by both wall modeling approaches.

- For detailed studies at an operating point away from stall, *e.g.* at the design point, it can be suitable to use the k-ϵ turbulence model with a low-Reynolds model to resolve the boundary layers (Chien's low-Reynolds formulation of the k-ϵ turbulence model was compared with wall functions and experimental data in Paper I).

- Considering the ability to predict performance near stall in [22], it can be suitable to evaluate the capability of the k-ω Menter SST turbulence model with the re-attachment model of Menter [24] to predict near stall performance for other axial compressors and other tip clearance sizes.

- The author of the present thesis has found an improved stall margin prediction when making unsteady calculations, e.g. the SAS-SST simulation in Paper I, which was closest to the experimental stall line. Unsteady calculations were recommended in [22] to accurately predict the slope of the speedline near stall.

4.3 Design

- If including the tip clearance has a large effect on the results from CFD calculations, then it can be appropriate to include the tip clearance in the optimization process. As discussed in Paper II, the tip clearance flow gives rise to blockage in the tip region causing a redistribution of the flow in the blade passage. Redistribution of flow in the blade passage was in turn shown to affect the design variables over the entire blade span.

5 Future work

Ongoing work includes evaluation of the impact of manufacturing imperfections and deviations on a blade design. Since certain tolerances are used in manufacturing, the blades are never perfectly smooth and geometrical variations can be present that could adversely affect the performance. Furthermore, wear and tear during engine operation will increase the level of surface roughness over time. Investigating blade designs that are robust to manufacturing and in-service variations would be an interesting topic of high relevance for industry applications. Furthermore, work is currently being done on the stability criteria used to rank compressor stages in the optimization process.

The tip clearance is an important aspect to consider in the design of an axial compressor stage, and it would be interesting to further investigate means of reducing its influence on performance. A shrouded rotor blade could have a beneficial impact, and the topic would make an interesting research project. An interesting topic would also be to assess the capability of different turbulence models to predict the onset of stall for a transonic compressor stage configuration with a large tip gap, comparing predictions with detailed measurement data.

6 Summary of Papers

6.1 Paper I

M. Lejon, L-E. Eriksson, N. Andersson and L. Ellbrant, 2015, Simulation of Tip-Clearance Effects in a Transonic Compressor, *Proceedings of ASME Turbo Expo 2015*, June 15–19, Montréal, Canada

6.1.1 Division of work

My contribution, besides being the lead author, was grid generation, a mesh convergence study, CFD simulations, post-processing and interpreting the results. Co-authors supervised the work and provided feedback on the results. Co-authors also provided tools for post processing the results.

6.1.2 Summary and discussion

This paper compared results obtained using different wall modeling approaches and turbulence models with experimental results of a transonic compressor with a large tip gap. It was shown that the k-ϵ turbulence model using wall functions underpredicted the total pressure ratio of the stage, which was discussed to be in part due to predicting a thicker boundary layer. Using Chien's low-Reynolds model to resolve the boundary layer showed good agreement with experimental data away from stall. The stall margin was shown to be underpredicted by all models, most noticeably for the k-ϵ turbulence model using Chien's low-Reynolds model. The SAS-SST model was shown to predict performance well but required significantly more computational resources.

The SAS-SST model was thus shown to be a good candidate for detailed studies, while using the k-ϵ turbulence model with wall functions could work well for optimization purposes where a large set of designs must be evaluated. Using Chien's low-Reynolds model formulation instead of wall functions could be an appropriate choice to use to evaluate performance at a design point, as it compares well with experimental data away from stall.

6.2 Paper II

M. Lejon, N. Andersson, L. Ellbrant and H. Mårtensson, 2015, CFD Optimization of a Transonic Compressor Stage with a Large Tip Gap, *22nd ISABE Conference*, October 25–30, Phoenix, USA

6.2.1 Division of work

My contribution, besides being the lead author, was grid generation, a mesh convergence study, CFD simulations, optimization, post processing and interpreting the results. Co-authors supervised the work and provided feedback on the results. Co-authors also provided the geometry for the reference stage used in the study, and assisted in setting up the optimization process.

6.2.2 Summary and discussion

Results of two optimizations of a transonic compressor stage are presented in this paper. The first approach does not consider any tip clearance during optimization, and the impact of tip clearance flow on performance is assessed after the optimization. The second approach considers a tip clearance in the optimization process. The results are compared in order to discuss the importance of including a large tip clearance in the design process. The optimization was made using the k-ϵ turbulence model and wall functions.

It was concluded in the paper that including the tip clearance has an influence on design variables at all radial positions. Redistribution of the flow in the passage has a clear influence on the incidence angle. It was shown that the rotor stagger angle was reduced at lower spanwise positions to allow for a higher mass flow away from the tip region, which is needed to compensate for blockage caused by the tip clearance flow. The designs obtained from the optimizations must all be able to reach the specified design point. Designs obtained from the optimization which did not consider any tip clearance flow fail in this respect when evaluated with a tip clearance, as the speed line is shifted due to flow blockage and additional losses.

To ensure that wall functions were capable of predicting the performance trend in terms of efficiency, three designs with tip clearances were chosen from a Pareto front and evaluated at the design point using Chien's low-Reynolds model to resolve the boundary layers. The ranking of the designs in terms of polytropic efficiency was compared to that obtained using wall functions and was shown to be the same, although the efficiency was overall lower.

Bibliography

[1] Storer, J., and Cumpsty, N., 1994. "An Approximate Analysis and Prediction Method for Tip Clearance Loss in Axial Compressors". *Journal of Turbomachinery,* ***116***(4), October, pp. 648–656.

[2] Ellbrant, L., 2014. "Multi-objective CFD-based Design Method for Axial Compressors". PhD thesis, Chalmers University of Technology.

[3] Deb, K., Pratap, A., Agarwal, S., and Meyarivan, T., 2000. A Fast and Elitist Multi-Objective Genetic Algorithm: NSGA-II. Tech. rep., Indian Institute of Technology Kanpur. KanGAL Report No. 200001.

[4] Ellbrant, L., Eriksson, L.-E., and Martensson, H., 2013. "Balancing Efficiency and Stability in the Design of Transonic Compressor Stages". In Proceedings of ASME Turbo Expo 2013, no. GT2013-94838.

[5] Storer, J., and Cumpsty, N., 1991. "Tip Leakage Flow in Axial Compressors". *Journal of Turbomachinery,* ***113***(2), April, pp. 252–259.

[6] Glanville, J. P., 2001. "Investigation into Core Compressor Tip Leakage Modelling Techniques using a 3D Viscous Solver". In Proceedings of ASME Turbo Expo 2001.

[7] Denton, J. D., 2010. "Some Limitations of Turbomachinery CFD". In Proceedings of ASME Turbo Expo 2013: Power for Land, Sea and Air, no. GT2010-22540.

[8] Sakulkaew, S., Tan, C., Donahoo, E., Cornelius, C., and Montgomery, M., 2013. "Compressor Efficiency Variation With Rotor Tip Gap From Vanishing to Large Clearance". *Journal of Turbomachinery,* ***135***(3), May, pp. 031030-1–031030-10.

[9] McDougall, N. M., 1990. "A comparison between the design point and near-stall performance of an axial compressor". *Journal of Turbomachinery,* ***112***(1), January, pp. 109–115.

[10] Copenhaver, W., Mayhew, E. R., Hah, C., and Wadia, A., 1996. "The Effect of Tip Clearance on a Swept Transonic Compressor Rotor". *Journal of Turbomachinery,* ***118***, April, pp. 230–239.

[11] Vo, H. D., Tan, C. S., and Greitzer, E. M., 2008. "Criteria for Spike Initiated Rotating Stall". *Journal of Turbomachinery,* ***130***(1), pp. 011023-1–011023-9.

[12] Hah, C., Bergner, J., and Schiffer, H.-P., 2006. "Short Length-Scale Rotating Stall Inception in a Transonic Axial Compressor - Criteria and Mechanisms". In Proceedings of ASME Turbo Expo 2006, no. GT2006-90045.

[13] Adamczyk, J., Celestina, M. L., and Greitzer, E., 1993. "The Role of Tip Clearance in High-Speed Fan Stall". *Journal of Turbomachinery,* ***115***(1), January, pp. 28–38.

[14] Suder, K. L., and Celestina, M., 1996. "Experimental and Computational Investigation of the Tip Clearance Flow in a Transonic Axial Compressor Rotor". *Journal of turbomachinery,* ***118***, pp. 218–229.

[15] Van Zante, D. E., Strazisar, A. J., Wood, J. R., Hathaway, M. D., and Okiishi, T. H., 2000. "Recommendations for Achieving Accurate Numerical Simulation of Tip Clearance Flows in Transonic Compressor Rotors". *Journal of Turbomachinery,* ***122***(4), October, pp. 733–742.

[16] Kirtley, K. R., Beach, T. A., and Adamczyk, J., 1990. "Numerical Analysis of Secondary Flow in a Two-Stage Turbine". In 26th AIAA/ASME/SAE/ASEE Joint Propulsion Conference, no. AIAA Paper 90-2356.

[17] Merz, L., Dailey, L., and Orwkis, P., 2004. "A CFD Study of the Flow Through a Transonic Compressor Rotor with Large Tip Clearance". In 40th AIAA/ASME/SAE/ASEE Joint Propulsion Conference and Exhibit, no. AIAA Paper 2004-3934.

[18] Williams, R., Gregory-Smith, D., and He, L., 2006. "A Study of Large Tip Clearance Flows in an Axial Compressor Blade Row". In Proceedings of ASME Turbo Expo 2006: Power for Land, Sea and Air, no. GT2006-90463.

[19] Williams, R., Gregory-Smith, D., He, L., and Ingram, G., 2010. "Experiments and Computations on Large Tip Clearance Effects in a Linear Cascade". *Journal of Turbomachinery,* ***132***(2), January, pp. 021018-1–021018-10.

[20] Du, J., Lin, F., Chen, J., Nie, C., and Biela, C., 2013. "Flow Structures in the Tip Region for a Transonic Compressor Rotor". *Journal of Turbomachinery,* ***135***(3), May, pp. 031012-1–031012-11.

[21] Cameron, J. D., Bennington, M. A., Ross, M. H., Morris, S. C., Du, J., Lin, F., and Chen, J., 2013. "The Influence of Tip Clearance Momentum Flux on Stall Inception in a High-Speed Axial Compressor". *Journal of Turbomachinery,* ***135***(6), September, pp. 051005-1–051005-10.

[22] Cornelius, C., Biesinger, T., Galpin, P., and Braune, A., 2014. "Experimental and Computational Analysis of a Multistage Axial Compressor Including Stall Prediction by Steady and Transient CFD Methods". *Journal of Turbomachinery,* ***136***, June, pp. 061013-1–061013-12.

[23] Chima, R., 1998. "Calculation of Tip Clearance Effects in a Transonic Compressor Rotor". *Journal of Turbomachinery,* ***120***(1), January, pp. 131–140.

[24] Menter, F., 1994. "Two-Equation Eddy-Viscosity Turbulence Models for Engineering Applications". *AIAA Journal,* ***32***(8), pp. 1598–1605.

[25] Denton, J., 1993. "Loss Mechanisms in Turbomachines". *Journal of Turbomachinery,* ***115***(4), October, pp. 621–656.