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On Film Cooling of Turbine Guide Vanes

**- From Experiments and CFD-Simulations to
Correlation Development**

Hossein Nadali Najafabadi

Division of Applied Thermodynamics and Fluid Mechanics
Department of Management and Engineering
Linköping University

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Cover: Film cooling jet streamlines from the cooling hole exit and surface temperature. The curves compare computational fluid dynamics, Spalart-Allmaras (magenta) and Shear Stress Transport (green), with experiments (white circles). The equations represent the time-line of the developed correlation models, from left to right suggested by R. S. Bunker, W. F. Colban and H. N. Nadali.

*"The three great essentials to achieve anything worth while are:
Hard work, Stick-to-itiveness, and Common sense."*

Thomas A. Edison

To Elsie!
For your love and patience!

Abstract

To achieve high thermal efficiency in modern gas turbines, the turbine-inlet temperature has to be increased. In response to such requisites and to prevent thermal failure of the components exposed to hot gas streams, the use of different cooling techniques, including film cooling, is essential. Finding an optimum film cooling design has become a challenge as it is influenced by a large number of flow and geometrical parameters. This study is dedicated to some important aspects of film cooling of a turbine guide vane and consists of three parts.

The first part is associated with an experimental investigation of the suction and pressure side cooling by means of a transient IR-Thermography technique under engine representative conditions. It is shown that the overall film cooling performance of the suction side can be improved by adding showerhead cooling if fan-shaped holes are used, while cylindrical holes may not necessarily benefit from a showerhead. According to the findings, investigation of an optimum cooling design for the suction side is not only a function of hole shape, blowing ratio, state of approaching flow, etc., but is also highly dependent on the presence/absence of showerhead cooling as well as the number of cooling rows. In this regard, it is also discussed that the combined effect of the adiabatic film effectiveness (AFE) and the heat transfer coefficient (HTC) should be considered in such study. As for the pressure side cooling, it is found that either the showerhead or a single row of cylindrical cooling holes can enhance the HTC substantially, whereas a combination of the two or using fan-shaped holes indicates considerably lower HTC. An important conclusion is that adding more than one cooling row will not augment the HTC and will even decrease it under certain circumstances.

In the second part, computational fluid dynamics (CFD) investigations have shown that film cooling holes subjected to higher flow acceleration will maintain a higher level of AFE. Although this was found to be valid for both suction and pressure side, due to an overall lower acceleration for the pressure side, a lower AFE was achieved. Moreover, the CFD results indicate that fan-shaped holes with low area ratio (dictated by design constraints for medium-size gas turbines), suffer from cooling jet separation and hence reduction in AFE for blowing ratios above

unity. Verification of these conclusions by experiments suggests that CFD can be used more extensively, e.g. for parametric studies.

The last part deals with method development for deriving correlations based on experimental data to support engineers in the design stage. The proposed method and the ultimate correlation model could successfully correlate the laterally averaged AFE to the downstream distance, the blowing ratio and the local pressure coefficient representing the effect of approaching flow. The applicability of the method has been examined and the high level of predictability of the final model demonstrates its suitability to be used for design purposes in the future.

Populärvetenskaplig beskrivning

Gasturbiner används både som flygmotorer och sammankopplade med en generator för att generera elektricitet. För att moderna gasturbiner ska uppnå en hög verkningsgrad, behöver turbininloppet vara så varmt som möjligt och som en följd av detta kommer många komponenter i gasturbinen utsättas för gasflöden med höga temperaturer. För att förhindra värmerelaterade problem på dessa komponenter måste olika kylningstekniker användas, en sådan är filmkylning. Filmkylning är en teknik som bygger på att ett tunt lager av kall luft släpps ut längs med ytan genom små hål för att skydda komponenterna från de heta gasflödena. Många forskare har tidigare undersökt olika aspekter av filmkylning för att erhålla så optimal design av filmkylningen som möjligt. I den här avhandlingen undersöks viktiga aspekter av filmkylning ur tre olika perspektiv.

I första delen undersöks filmkylning av sug- och trycksidan av en turbinskovel med hjälp av en experimentell teknik, IR-termografi. Experimenttriggen finns på Siemens Industrial Turbomachinery i Finspång och representerar en turbinskovel med liknande förutsättningar som i en gasturbin. Mätresultaten har visat att för sugsidan kan flera filmkylhål på framkanten förbättra den totala filmkylningsprestandan om hål där arean ökar i flödesriktningen (här kallade divergerande hål) då används nedströms, men för cylindriska hål fås inte samma fördel av det. Arbetet har också visats att optimal filmkylning inte bara är en funktion av hålform, blåsförhållande, det inkommande flödet osv., det är även mycket beroende av om flera filmkylhål på framkanten används eller inte och dessutom på hur många rader av filmkylningshål som används. På trycksidan visar resultaten att bara flera filmkylhål på framkanten eller en rad av cylindriska hål kan höja värmeöverföringskoefficienten väsentligt, men det har också visats att en kombination av båda eller att använda divergerande hål indikerar en lägre värmeöverföringskoefficient. En mycket viktig slutsats är att mer än en rad av filmkylningshål inte förmår att öka värmeöverföringskoefficienten. Eftersom två eller flera filmkylningsrader kan höja den adiabatiska effektiviteten, kan den totala filmkylningsprestandan förväntas förbättras.

I den andra delen har datorsimuleringar (Computational Fluid Dynamics, CFD) visat att flödesacceleration ökar den adiabatiska effektiviteten. På sugsidan är den

effekten mer uttalad än på i trycksidan, därför att flödesacceleration på trycksidan är mycket låg. Det har också visats att flödesseparation uppstår för divergerande hål som har litet areaförhållande (vilket är fallet för medelstora gasturbiner) när blåsförhållandet är högre än ett. Detta indikerar att den adiabatiska effektiviteten reduceras om blåsförhållandet ökas ytterligare. Resultaten har verifierats med experiment, och indikerar att CFD kan användas i större utsträckning för till exempel parameterstudier.

Sista delen omfattar metodutveckling för härledning av korrelationsmodeller från experimentella data. Dessa ska användas av konstruktörerna i projekteringsstadiet för att erhålla en bättre kyldesign. Metoden har visat att olika korrelationsmodeller kan härledas för att korrelera den laterala genomsnittliga adiabatiska effektiviteten med nedströms avstånd, blåsförhållande och lokal tryckkoeficient. Den slutliga modellen har visat mycket god förutsägbarhet och därmed visat sig lämplig för att användas i projekteringsstadiet.

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List of Papers

This thesis is based on the following five papers, which will be referred to by their Roman numerals:

- I. **Film Cooling Performance of a Turbine Vane Suction Side: The Showerhead Effect on Film Cooling Hole Placement for Cylindrical and Fan-Shaped Holes**, Hossein Nadali Najafabadi, Matts Karlsson, Esa Utriainen, Mats Kinell, *Journal of Turbomachinery*, Vol. 137, 091005, 2015.
- II. **Film Cooling Performance of Multiple Arrays of Cylindrical and Fan-Shaped Holes**, Hossein Nadali Najafabadi, Matts Karlsson, Esa Utriainen, Mats Kinell, Lieke Wang, *Submitted for Publication*, March 2015.
- III. **Film Cooling Jet Injection Effect in Heat Transfer Coefficient Augmentation for the Pressure Side Cooling of Turbine Vane**, Hossein Nadali Najafabadi, Matts Karlsson, Esa Utriainen, Mats Kinell, *Proc. of ASME Turbo Expo 2014*, GT2014-26055, 16-20 June 2014, Düsseldorf, Germany.
- IV. **CFD Based Sensitivity Analysis of Influencing Flow Parameters for Cylindrical and Shaped Holes in a Gas Turbine Vane**, Hossein Nadali Najafabadi, Matts Karlsson, Esa Utriainen, Mats Kinell, *Proc. of ASME Turbo Expo 2012*, GT2012-69023, 11-15 June 2012, Bella Center, Copenhagen, Denmark.
- V. **Film Effectiveness Correlations for Cylindrical and Fan-Shaped Holes, Introducing Local Pressure Coefficient**, Hossein Nadali Najafabadi, Matts Karlsson, Esa Utriainen, Mats Kinell, *Proc. of ASME Turbo Expo 2012*, GT2012-69021, 11-15 June 2012, Bella Center, Copenhagen, Denmark.

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Nomenclature

All dimensions use SI units.

Upper Case Latin

AR	Area ratio for fan-shaped holes
C_p	Pressure coefficient
D	Cooling hole diameter
DR	Density ratio
L	Length characterizing the scale of the setup
L_C	Cylindrical film cooling hole length
L_F	Film cooling hole length corresponding to cylindrical part of fan-shaped holes
L_t	Material thickness
M	Local blowing ratio
P	Hole pitch
$\dot{Q}$	Net emissivity power
Pr	Prandtl number
R	Gas constant
R_{ad}^2	Adjusted coefficient of determination
Re	Reynolds number
S	Vane surface length
T	Temperature
Tu	Freestream turbulence intensity
U	Velocity $U(x,y,z)$
X	Correlation variable

Lower Case Latin

c_P	Specific heat at constant pressure
c_V	Specific heat at constant volume
h	Heat Transfer Coefficient
k	Thermal conductivity
l	Characteristic length
p	Pressure
$\dot{q}$	Heat transfer rate per unit area
s_{ij}	Strain rate tensor
$\bar{s}_{ij}$	Strain rate based on time averaged velocity
t	Time
t_F	Fan-shaped hole breakout width
u_i	Velocity components in spatial directions, $i = 1, 2, 3$
$\bar{u}_i$	Time averaged velocity
$\overline{u'_i}$	Fluctuating velocity components
$\overline{u'_i u'_j}$	Specific Reynolds stress tensor
x, y, z	Directions in a cartesian coordinate system
x_i	Directions in a cartesian coordinate system, $i = 1, 2, 3$
y^+	Non-dimensional wall distance

Greek

α	Inclination angle
α_λ	Absorbability
β	Correlation coefficient
γ	Transformation power
δ	Boundary layer thickness
δ_{ij}	Kronecker delta, i and $j = 1, 2, 3$
η	Adiabatic film effectiveness
Λ	Thermal diffusivity
μ	Dynamic viscosity
ν	Kinematic viscosity
ν_t	Turbulent viscosity
ξ	Surface normal coordinate
ζ	Film cooling scaling parameter
ρ	Density
ρ_λ	Reflectivity
ϕ	Overall film cooling effectiveness
ϕ_1	Fan-shaped hole lateral diffusion angle
ϕ_2	Fan-shaped hole forward diffusion angle
Φ	Arbitrary variable
τ_{ij}	Stress tensor
τ_λ	Transmitivity

Subscripts and Superscript

aw	Adiabatic wall temperature
c	Coolant condition
C	Cylindrical hole
$exit$	Hole exit located at vane surface
f	Film cooled vane
F	Fan-shaped hole
i	Initial condition
i, j	Correlation variable subscripts
in	Hole inlet located at the supply plenum
m	Recovery temperature
w	Wall temperature
∞	Freestream condition
$-$	Lateral/pitch wise average
$=$	Spatial (area) average
0	Uncooled condition
$1, 2, 3$	Variable counting (correlation)
$'$	Superscript, reference value for normalization

Abbreviations

AFE	Adiabatic Film Effectiveness
CFD	Computational Fluid Dynamics
HTC	Heat Transfer Coefficient
NHFR	Net Heat Flux Reduction
RANS	Reynolds-Averaged Navier-Stokes
SST	Shear Stress Transport

Chapter 1

Introduction

1.1 Background

Typical examples of using gas turbines are aircraft propulsion, land-based power generation, and industrial applications. Increasing the overall efficiency and power output of gas turbines is characterized by the rise in the compressor pressure ratio and the turbine-inlet temperature. Since turbine guide vanes and blades cannot withstand extreme turbine-inlet temperatures, cooling is essential in order to obtain a reasonable life span for the components exposed to hot gas streams.

Depending on the range of the turbine-inlet temperature, a specific type or combination of certain types of cooling may be required. Table 1 shows guidelines for the cooling type required when the turbine-inlet temperature exceeds certain ranges, Fullagar [1]. It can be seen from the table that film cooling is required in addition to convective cooling to sustain blades at required temperatures above 1450 K.

Film cooling is about providing a protective layer of air by bleeding a thin layer of coolant between the hot gases and the external surfaces. The cooling air, often taken from the compressor, is injected through discrete film holes, or rows of film holes, on the hot gas path surfaces of the turbine. This work is dedicated to investigation of some aspects of film cooling relevant to a turbine-inlet guide vane at the first stage of medium-size gas turbines.

1.2 State of the Art

A tremendous amount of research has been done over the past four decades in order to comprehend the fundamental physics of film cooling. Improvement in the state of the art has been achieved continuously through investigations of different aspects of film cooling both experimentally and computationally by a broad

Temperature range, T	Cooling methods commonly used
$T < 1200\text{K}$	No cooling required
$1200\text{K} < T < 1450\text{K}$	Internal convective system
$1450\text{K} < T < 1600\text{K}$	Convective systems augmented by rows of film cooling holes
$1600\text{K} < T < 2000\text{K}$	Combinations of convection, impingement systems and film cooling
$T > 2000\text{K}$	Some kind of transpiration cooling

Table 1: Relevant cooling systems for ranges of turbine-inlet temperatures, Fullagar [1].

spectrum of researchers. However, some aspects of film cooling have been explored to a larger extent and some others to a lesser extent, but almost all aspects have been explored. Research has focused on the effect of approach flow prior to the film hole, acceleration, mainstream turbulence properties and vorticity production, mainstream and film hole fluid dynamics and interactions, density ratio, blowing ratio, external surface curvature, cooling hole shape effect, hole spacing and orientation, hole length to diameter ratio, external surface roughness, etc.

In addition, different correlations have been developed from experimental data, depending on the number of parameters and available data, to help engineers achieve better design and performance. Since this work includes experimental and computational investigation as well as correlation development, the state of the art is reviewed with respect to each individual category in the following sub-sections. This approach offers the opportunity to better clarify and justify the research objectives and aims regarding each subject when discussed later.

The efficiency of a film cooling design and the aforementioned effects are formulated and often discussed as functions of adiabatic film effectiveness (AFE), heat transfer coefficient (HTC), aerodynamic losses, etc. The AFE determines how effective coolant distribution is over the surface and the goal is to reach AFE as close as possible to its maximum value that is 1.0, while having the lowest impact on the rate of heat transfer to the surface measured by HTC. Lower values of HTC indicate lower heat flux load on the surface. In the context of HTC, the ratio of HTC for a cooled case to that of an uncooled case referred to as HTC augmentation, can also be used and it should be kept less than or equal to one. The overall film cooling performance, which combines the effect of both AFE and HTC augmentation, is a quantity that may describe best whether a cooling design is favorable or detrimental. More details regarding the definition of these terms will be given in the method section.

1.2.1 Experimental Investigations

One of the earliest reviews on the film cooling research and development of the basic film geometries is provided by Goldstein [2]. Goldstein et al. [3] performed an experimental investigation on film cooling effectiveness performance of discrete hole injection into a turbulent boundary layer on a flat plate. Quantification of the film cooling performance for shaped holes has been first done by Goldstein et al. [4]. Jabbari and Goldstein [5] studied the film cooling and the heat transfer of two staggered rows of holes. The effects of hole length to diameter ratio and freestream turbulence on film cooling performance have been addressed by Burd et al. [6]. Sinha et al. [7] investigated the effect of density ratio on film cooling effectiveness. Ekkad et al. [8] and Lee et al. [9] performed experimental investigations on the influence of compound angle (the cooling hole angle relative to the freestream flow in the lateral direction) in film cooling characteristics for cylindrical holes and shaped holes, respectively.

Most of the earliest research regarding film cooling rely on flat plate investigations. Due to continuous changes in the flow around an airfoil, the available pressure gradients and the surface curvature, the implications of these studies could be limited in practice. Some researchers have therefore made an effort to investigate the effects of flow properties associated with airfoils on film cooling performance. For instance, Kruse [10] studied the wall curvature and pressure gradient effects together with the effects of hole geometry on film cooling performance. His findings suggest slightly lower AFE in the vicinity of the cooling hole when the hole is subjected to adverse pressure gradient compared to the case with favourable pressure gradients. In contrast to the study by Kruse [10], Maiteh and Jubran [11] have found that both favourable and adverse pressure gradients will decrease the film cooling performance at blowing ratio 0.6. According to a comprehensive review by Bogard and Thole [12], surface curvature along with freestream turbulence and hole shape have a strong impact on defining the performance of the film cooling.

Advances in technology and measurement techniques have provided the possibility to conduct experimental investigations on real airfoil configurations and under more realistic engine representative conditions. Such studies can examine the film cooling performance on the suction or the pressure side of turbine vanes and blades as well as leading edge cooling (referred to as showerhead cooling, which consists of few staggered rows of cooling holes, often three to five rows, located in the stagnation region with the purpose of effectively cooling the leading edge and to some extent the suction and pressure sides). For example, strong effects on suction side film cooling performance due to changes in Reynolds/Mach number have been reported by Drost and Böls [13]. Such changes indeed cause alteration in boundary layer thickness and flow acceleration and can thereby influence film performance.

In another study by Sargison et al. [14], it was found that the effects of the fan-shaped and converging slot holes are almost identical in terms of AFE and that they both perform better than cylindrical holes. Arts and Bourguignon [15] have shown that the decrease in AFE and the increase in HTC can be expected for a pair of cooling rows on the pressure side of a high-pressure nozzle guide vane if the Reynolds number is increased. Identification of the "hot spots" on the surface within the showerhead region using the overall cooling effectiveness has been addressed by Nathan et al. [16]. Their research was conducted in a simulated turbine vane with a showerhead and an additional row of cooling on both the suction and the pressure sides. The suction and pressure side cooling performance comparison indicated lower AFE and higher HTC in the latter case when the blowing ratio is above unity, as discussed by Kinell et al. [17].

Some studies have made explicit investigations of the flow and geometrical influence in only the HTC or HTC augmentation. For example, in a study by Turner et al. [18], conducted on a C3X vane in a transonic linear cascade, they showed that the HTC due to the showerhead cooling increases if the approaching Reynolds number and also cooling ejection ratios are increased. The research done by Bonanni et al. [19] showed a negligible effect on HTC due to pressure gradients, and Ammari et al. [20] found a reduction in HTC due to flow acceleration which suppresses the injection-induced turbulence.

The increase in turbulence level (from 3.6% to 11%) has proven to cause higher HTC augmentation for shaped holes compared to cylindrical holes in a study performed by Saumweber et al. [21]. The resulting HTC augmentation influenced by variation in some parameters such as blowing ratio, Reynolds number, ejection angle and hole spacing in the case of flat plate has been discussed in Baldauf et al. [22].

Bolchoz et al. [23] have shown that showerhead cooling will cause high level of HTC enhancement. Xue et al. [24] demonstrated that compared to a showerhead only cooling case, the presence of an additional row of shaped holes can lead to a lower heat flux. The influence of exit Reynolds number/Mach number on the performance of showerhead cooling with respect to both film cooling effectiveness as well as Nusselt number has been investigated by Nasir et al. [25]. In the presence of showerhead cooling, the AFE and overall cooling effectiveness of a cooling hole embedded in a trench on the pressure side is improved compared to the standard holes, as discussed by Albert and Bogard [26].

The above-mentioned studies address important aspects of film cooling either with respect to certain parameters or regarding relevant configurations employed in turbine cooling such as showerhead cooling. However, findings from these studies cannot provide an appropriate answer to the important question of how the improvement in film performance for the suction and the pressure side of a turbine guide vane should be looked into; if there are other important aspects to

be considered for achieving such improvements besides the influential parameters which may lead to higher film cooling performance of an individual row of cooling hole.

1.2.2 Computational Fluid Dynamics Studies

Since film cooling applications and concepts are still improving, there are demands for analysis, evaluation, and design optimization tools. In this context, computational fluid dynamics (CFD) has become a powerful tool for investigation of different aspects of film cooling. It offers the possibility to study wider ranges of parameters at lower cost (economically), i.e. compared to performing physical experiments, CFD calculations are often cheaper and usually faster. However, in order to ensure their reliability, it is essential to validate the results with experiments.

A systematic computational methodology may consist of four steps for CFD simulation of film-cooling that includes computational model of the physical problem, geometry and grid generation, discretization scheme, and turbulence modeling, as discussed by Walters and Leylek [27]. The CFD studies can therefore focus on investigating the effect of influencing parameters and/or the evaluation and validation of the numerical methods and turbulence modeling for film-cooling applications. As regards the latter case, i.e. validation, most computational studies utilize codes based on Reynolds-Averaged Navier-Stokes (RANS) equations. In this respect, deficiency of suitable turbulence closures has been encountered as one of the major difficulties in heat transfer predictions for gas turbines, Dunn [28].

The effect of different turbulence models in CFD simulations for prediction of film cooling has been studied for both flat-plate and vane-like geometries. According to findings by Walters and Leylek [27], based on flat-plate configuration, the standard $k - \epsilon$ model (SKE) with standard wall function (SWF) results in overprediction of lateral spread and center-line cooling effectiveness when compared to experiments. Medic and Durbin [29] also reported deficiency of the SKE model.

Ferguson et al. [30] studied different turbulence modeling approaches such as the standard $k - \epsilon$ model (SKE) with SWF, SKE with non-equilibrium wall function (NEWF), renormalization group $k - \epsilon$ with SWF (RNG), RNG with NEWF, SKE with two layer wall treatment (2LWT), Reynold stress model (RSM) and RSM with NEWF. They showed that for blowing ratio unity, a more accurate solution can be achieved if SKE with 2LWT is used in comparison with RNG and RSM models. This conclusion was valid in terms of both center-line and spanwise averaged AFE. In a study by York and Leylek [31], it was shown that the realizable $k - \epsilon$ model (RKE) could predict the flow field and heat transfer coefficients of leading edge cooling compared to SKE.

The capability of the RKE model has been further compared with the standard $k - \omega$ (SKW) and RSM turbulence models on a flat-plate model by Harrison and Bogard [32]. They reported that although laterally averaged effectiveness results from the SKW model show the best comparison with experiments, the center-line effectiveness was predicted best by RKE for two investigated blowing ratios 0.5 and 1.0. Bianchini et al. [33] studied heat transfer performance of fan-shaped film cooling holes by means of both experiments and CFD. They used the Two-Layer model both in the isotropic original formulation and with an anisotropic algebraic correction, the $k - \omega$ SST and the $\nu^2 - f$ turbulence models to evaluate the heat transfer coefficient and film cooling effectiveness over a flat-plate. Based on their findings, the SST and $\nu^2 - f$ models will give similar results, and obtained good agreement between CFD and experimental results.

Colban et al. [34] utilized the RNG $k - \epsilon$ and $\nu^2 - f$ turbulence models to evaluate the showerhead, suction and pressure side film cooling of fan-shaped holes. In that study, CFD results did not agree well with experiments, and the overall conclusion was that the RNG $k - \epsilon$ model under-predicts and the $\nu^2 - f$ model over-predicts film effectiveness.

Besides computational studies for evaluation and validation, some other researchers have used a validated computational method in some way or another to investigate the influencing parameters in film cooling performance. One extensive study in this context was made by Baldauf and Scheurlen [35]. They performed a CFD based sensitivity analysis of flow parameters for cylindrical holes under engine representing conditions on a flat-plate and, accordingly, some detailed knowledge about the effect of investigated parameters such as density ratio, blowing ratio, hole pitch to diameter ratio, etc. was provided.

Nguyen et al. [36] also made a sensitivity study of the influencing parameters utilizing a flat plate. By means of statistical analysis, they tried to identify the most influencing parameters among blowing ratio, density ratio, hole pitch and trench depth to diameter ratio for a round hole embedded in a trench. Johanson et al. [37] went some steps beyond parametric study and used Genetic Algorithm to optimize a high-pressure turbine vane pressure side cooling. They showed that by means of an efficient form of CFD an improved film cooling array could be redesigned from a baseline case. In that study, the cooling holes and corresponding plenum chambers were not included in the CFD calculations, and instead film holes are modeled as discrete sources of mass flow.

The overall conclusion from these parametric studies may have practical limitations or is associated with some level of uncertainty due to the differences in flow physics between a flat-plate and a highly curved vane configuration. The verification of such statement may be given when the results of studies that have investigated the effect of curvature and such, present in airfoil types of configuration, are reviewed, e.g. Mayle et al. [38], Ito et al. [39] and Davidson et al. [40].

The question to be addressed here is whether CFD can be used as a tool for studying some influencing parameters in film cooling performance on real vane configurations and, if so, what the possible limitations are.

1.2.3 Developed Correlations

Thermal and structural finite element (FE) analysis of hot gas exposed surfaces of turbine components is considered to be an important and crucial step in the design process. In order to perform such an FE analysis the thermal boundary condition has to be derived from either flow and/or boundary layer computations or correlations. Due to shortcomings associated with computational approach, for example the influence of the associated uncertainty with specific input parameters, correlations are more commonly used for derivation of the boundary condition. Provision of correlations with the capability to predict the distribution of the film cooling is therefore essential in the design phase.

Since development of the correlation relies on experimental data for both derivation and evaluation, they may be valid under certain circumstances. Correlations are often developed for heat transfer coefficient and adiabatic film effectiveness, though the focus in this study is the latter case. From the earliest stages of gas turbine development, different correlation models have been proposed in open literature, reflecting the variation in internal cooling geometries and film cooling parameters.

One of the earliest studies that addresses film cooling prediction was made by Goldstein and Haji-sheikh [41]. In the study by Brown and Saluja [42], two correlations for prediction of laterally averaged AFE have been suggested, one for blowing ratio less than 0.64 and one for higher blowing ratios. In that study, a few parameters such as pitch to diameter ratio, P/D , are excluded. L'Ecuyer and Soechting [43] proposed correlations for three different flow regimes based on velocity ratio that influence the distribution of AFE.

In an extensive study, Baldauf et al. [44] developed a correlation model based on flat-plate experimental data for cylindrical holes. Their model has no coefficient to be determined and is thus considered to be more general, although it is derived under the circumstance of a specific boundary layer thickness, $\delta/D = 0.1$. Four correlations commonly used in industry suggested by Bunker [45] are for predicting the AFE of cooled air blowing through slots, although they can be adapted for discrete holes. A correlation model for AFE prediction of fan-shaped holes has been developed by Colban et al. [46]. With regard to the proposed correlation models, two important questions arise, one of which is associated with limitations of the data used to derive the model since they are obtained from flat plate studies. The second question is related to the model generalization and its applicability if other data sets are used.

1.3 Aim

The primary aim was to investigate the importance of showerhead cooling in determining the film characteristics of the suction and the pressure side of a turbine guide vane, which will in turn show possible alternatives for obtaining improvements in film cooling performance in these areas. This was to be accomplished through an extensive study such that it also covers parameters that may influence an individual row of cooling independent of the showerhead cooling, e.g. the hole shape, the range of the blowing ratio and the approaching flow effects. This objective is in accordance with the question formulated in section 1.2.1.

The secondary aim, addressing the question developed in section 1.2.2, was to use CFD as an alternative tool to investigate some of the influencing parameters in adiabatic film effectiveness for a real vane configuration. This also offers the possibility to employ such investigations for further design and optimization purposes. The validity and limitations of the approach also have to be considered.

Finally, this study aimed to develop a generalized correlation method such that further extensions and developments in the correlation model can be achieved. The method needs to be verified by deriving different models based on experimental data, covering ranges of parameters, obtained from real vane configurations. Thus, the method should not have practical limitations, or if there are such restrictions they should be explored. This implies that the method can be used to derive correlation models for a variety of configurations, if necessary, such as different cooling hole shapes, the suction and the pressure side cooling and in the presence or absence of showerhead. This aim corresponds to the question established in section. 1.2.3.

Chapter 2

Method

This study is devoted to three approaches for investigating different aspects of film cooling, which in turn demands that the methods associated with each approach be explored. This chapter will therefore address important features of the experimental, CFD and correlation approaches.

2.1 Experimental Approach

Surface heat-transfer measurements obtained from experimental techniques are important for a successful cooling design. They are thus the essence of experimental correlations that are used in the design stage and should cover ranges of parameters influencing a particular design. Moreover, such measurements are extensively used for validation and verification of the computational studies, and make a major contribution in the development and application of the computational techniques.

To obtain measurements on heat-transfer surfaces, there are different experimental techniques which can provide either film cooling effectiveness (AFE) or heat transfer coefficient (HTC) and in some cases both quantities. Examples of such techniques are heat flux gages, mass-transfer analogy, liquid crystal Thermography, and IR-Thermography to name just a few. While each technique has its own pros and cons, neither finding the best surface heat-transfer technique nor the design of the test facility have been the scope of this work. In fact, a well established test facility to be used along with IR-Thermography for measuring surface heat-transfer was provided prior to the project start. In the following sections, therefore, first an introduction to heat transfer theory for film cooling is given and then short summaries of important aspects of the experimental methodology used and the test facility characteristics will be reviewed.

2.1.1 Heat Transfer Theory for Film Cooling

There are three heat transfer mechanisms by which the heat transport can take place, known as conduction, radiation and convection, where the latest case is the focus of this study. Convective heat transfer, which describes the transport of heat from fluid to solid and vice versa, consists of two mechanisms. The first mechanism deals with the fluid flow motion near the wall where the random molecular motion or diffusion is dominant in defining the fluid flow properties. In the second mechanism, which happens off the wall, the fluid flow characteristics are affected to a large extent by the macroscopic motion of the fluid. The rate of heat transfer per unit area, $\dot{q}$ is thus governed through the combination of these mechanisms and is formulated by Newton's law of cooling, defined in Eq. 1.

$$\dot{q} = h (T_w - T_\infty) \quad (1)$$

In this equation the proportionality of the surface heat flux to the temperature difference between the wall and the fluid is related to the heat transfer coefficient, h . In general, T_∞ in Eq. 1 is the main-flow static temperature. However, in case of high speed flows, flows with Mach number above 0.3 with the compressibility effects, the relevant main-flow temperature is required to account for the effect of friction heating. This leads to the so-called recovery temperature, given as

$$T_m = T_\infty + Pr^\phi \frac{U_\infty^2}{2c_p} \quad (2)$$

According to Schlichting [47], the exponent ϕ has the value 1/2 for a laminar and 1/3 for a turbulent boundary layer. The recovery temperature is therefore used instead of T_∞ in Eq. 1. Another important issue to consider is how to explain the heat transfer rate if a fluid with different temperature, for example coolant in film cooling, than the freestream is ejected onto the surface. By injecting coolant into the freestream a third temperature, which can range between the coolant temperature and freestream temperature due to mixing, is introduced. The rate of heat flux is related to the difference between the wall temperature and this temperature that is known as the film temperature (T_f) or adiabatic wall temperature (T_{aw}). Figure 1 demonstrates the principle of film cooling injection. The surface heat flux definition is then modified to

$$\dot{q}_f = h_f (T_w - T_f) \quad (3)$$

Equation 1 for an uncooled case, denoted by subscript 0, becomes

$$\dot{q}_0 = h_0 (T_w - T_\infty) \quad (4)$$

Equation 3 states that a decrease in the temperature difference governing the heat

Figure 1: Film cooling principle.

transfer due to coolant injection will lead to a reduction in the surface temperature load. According to this equation, the adiabatic wall temperature is equal to the measured wall temperature if the rate of surface heat flux is zero, $\dot{q} = 0$. This implies that by applying film cooling on an adiabatic surface ($\dot{q} = 0$), the unknown T_f would be known. The dimensionless form of the adiabatic wall temperature, which is referred to as adiabatic effectiveness (or adiabatic film effectiveness, AFE, as used in this study), η , is defined as

$$\eta = \frac{T_\infty - T_f}{T_\infty - T_c} \quad (5)$$

where T_c is the coolant temperature, T_∞ is the freestream temperature, and the effectiveness ranges between 0 and 1.0. It becomes 0 when the film temperature is equal to the hot main flow temperature and will become 1.0 when it is equal to the cooling air temperature. Accordingly, the film cooling injection's target is to reach the maximum level of AFE. Although this equation is an estimate of the favorable effect of film cooling, it is not sufficient to conclude if an effective cooling design is achieved. There are thus other important aspects to be considered to obtain complete knowledge of the pros and cons of a film cooling design such as heat transfer augmentation, aerodynamic losses, etc.

It is known that flow disturbances caused by film injection can enhance the heat transfer coefficient and lead to an increase in the heat load to the surface, which is an unfavorable effect. The efficiency of a film cooling design is therefore evaluated in terms of AFE along with the so-called heat transfer coefficient augmentation, the ratio of the HTC with film cooling to that without film cooling, and is defined as h_f/h_0 . The objective of a film cooling design, in this regard, will then be to keep the HTC augmentation, h_f/h_0 , less or equal to unity. If the HTC itself, i.e. h_0 or h_f , is the subject of discussion, then the lower value of HTC would be of interest since it indicates lower heat load on the surface.

Further discussion on the changes in HTC is related to the flow characteristics near the wall, where viscous effects are dominant. This region, which is called the

boundary layer, is one of the mechanisms contributing to convective heat transfer, as discussed earlier. Indeed, the increase or decrease in HTC is to a large extent dependent on the state of the boundary layer.

A turbulent boundary layer will enhance the HTC as it increases the mixing of the fluid. The flow around an airfoil, as in turbomachinery applications, is subject to continuous change and often experiences a laminar to turbulent boundary layer with different thicknesses. The consequence of such changes will be that the HTC will be subjected to changes along the airfoil.

2.1.2 Test Facility

Figure 2 shows a schematic of the test facility built for film cooling investigations at Siemens Industrial Turbomachinery, Finspång.

Figure 2: The experimental set-up and corresponding components. The test section indicates the cascade geometry, which restricts the flow path by one vane and two vane-shaped side-walls. The red and blue arrow-lines denote the hot and cold gas path, respectively.

The mainstream airflow, shown by red arrow-lines in Fig. 2, is provided by a blowing machine which supplies 3 kg/s of air at a pressure of about 1.5 bar. The airflow passes a shut-off valve and a diffuser and then through a settling chamber with a honeycomb which makes the flow uniform and straight. By passing a transition duct the airflow reaches a turbulence grid in order to maintain a desirable level of turbulence. After the turbulence grid, a bypass valve is indicated in order to bypass the mainstream air prior to the transient test. The test section, which is followed by a diffuser, is located after the bypass valve (labeled P in figure 2).

The mainstream air is heated by the blower through compression and friction losses. As mentioned earlier, the air is bypassed through a duct prior to the test. This facilitates the heating of the piping between the test section and the blower. After a steady state air temperature of about (331 K) is achieved, a pneumatic actuator valve closes the bypass and the heated air enters the test section. To ensure steady state flow within 1 s from the test start-up time, which is due to limitation in the total run-time, the bypass duct is connected to the test section exhaust to obtain the same pressure drop.

Figure 3: Semi-transparent view of the test section with test object and window frames. The vane simulates a 2D profile.

Cooling air is supplied to the test section through a secondary unit which is connected to different chambers by means of tubes of about 100 mm diameter. The secondary unit consists of a Coriolis mass-flow meter, presented as pressure reducing valve in Fig. 2, followed by a regulator for controlling the fraction of the cooling air. The cooling air path is demonstrated by blue arrow-lines in Fig. 2. Since the main inlet has constant air-flow, the cooling air mass-flow rate is adjusted such that desirable blowing ratio, defined as in Eq. 6, is obtained.

$$M = \frac{\rho_c U_c}{\rho_\infty U_\infty} \quad (6)$$

with the coolant and the freestream densities denoted by ρ_c and ρ_∞ . Also, U_c and U_∞ represent the coolant and the freestream velocities, respectively. Since both mainstream and coolant fluids are air, the density ratio will be around unity, ($DR \approx 1$). The coolant is at room temperature which is about 294 K.

Figure 3 shows a semi-transparent view of the test section, which is manufactured from 5 mm thick sheet metal. As illustrated in this figure, there are 9

different window frames in total that give the accessibility to the camera to record surface temperature of the vane on the suction and pressure sides as well as show-erhead region.

The model experiment is performed at a larger-scale through the dimensional analysis, which according to Eckert [48], for adiabatic film effectiveness, is described by Eq. 7.

$$\eta = \frac{T_\infty - T_f}{T_\infty - T_c} = f\left(\frac{x}{L}, \frac{y}{L}, Re_\infty, M, \frac{T_\infty}{T_c}\right) \quad (7)$$

In this relation the length characterizing the scale of the setup is denoted by L . To maintain engine representing conditions dynamic similarity with respect to Reynolds number and length scales are obtained by adjusting the scale of the model and the properties of the flow. The ejection ratio, M , can be varied to include ratios similar to engine.

Figure 4: Non-dimensional pressure distribution C_p . The maximum throat velocity is indicated by the vertical dashed line.

In addition, the pressure coefficient has been considered in similarity analysis in order to account for the pressure variation along the vane contour, which may influence the boundary layer development and consequently heat transfer. The pressure coefficient is defined as

$$C_p = \frac{p_s(s) - p_{s\infty}}{p_{d\infty}} \quad (8)$$

with $p_s(s)$ representing the static pressure along the vane surface, $p_{s\infty}$ and $p_{d\infty}$ denoting the inlet static and dynamic pressures, respectively. The pressure coefficient distribution of the test facility is compared to engine test results in Fig. 4. The good agreement between engine test results and experiments can be clearly seen from the figure. Note that due to the existence of a film cooling hole for engine

test results, the pressure distribution over the suction side close to the maximum throat velocity is slightly different from the test facility data, which are for an uncooled vane.

Figure 5: The prototype vane with showerhead cooling, denoted as SH1-SH5, positions and numbering of the film cooling rows. The cavities supplying cooled air are marked [C1-C4]. The definition of the S coordinate starts with $S = 0$ on the suction side's trailing edge and becomes 1.0 at the trailing edge on the pressure side.

Figure 5 shows the prototype vane and cavities corresponding to the cooling holes at various locations. Since the numbering of the rows differs from what appears in Papers I-V, Tab. 2 represents the numbering match between the figure here and the corresponding case in the appended paper. Definition of the S coordinate starts from the trailing edge on the suction side and ends at the pressure side's trailing edge as shown in Fig. 5. The double row cooling hole cases are interlaced. That is the first row is in stagger alignment with respect to the second row, see Fig. 6.

Figure 6: The alignments for showerhead cooling and double row cooling holes on the suction side.

It is worth mentioning that the leading edge region is often cooled by means of several, typically three up to five, rows of cooling holes, which is referred to as

showerhead cooling. In this study the showerhead cooling consist of five rows, denoted as SH1-SH5 in Fig. 5, with staggered alignment between different rows shown in Fig. 6. Accordingly, the presence of showerhead implies that all these five rows are injecting coolant. The absence of showerhead means that neither coolant is injected nor the physical cooling holes exist.

Current Numbering	Numbering in appended papers				
	I	II	III	IV	V
Row #1	#3	#4	-	#1	#1
Row #2	#2	#3	-	-	-
Row #3	#1	#2	-	#2	#2
Row #4	-	#1	-	-	-
Row #5	-	-	-	#3	#3
Row #6	-	-	-	-	-
Row #7	-	-	-	#4	#4
Row #8	-	-	-	-	#5
Row #9	-	-	-	#5	#6
Row #10	-	-	#1	-	#7
Row #11	-	-	#2	#6	#8
Row #12	-	-	#3	-	-

Table 2: Cooling hole numbering match in Fig. 5 to that given in appended Papers I-V.

Definition of the parameters for fan-shaped and cylindrical holes is shown in Fig. 7 and the corresponding value of the film cooling hole parameters for the suction and pressure side holes are presented in Tab. 4. In addition, in Tab. 3 the characteristics of showerhead cooling holes and the blowing ratio corresponding to each row are reported. Note that the blowing ratio for the showerhead cooling is kept constant throughout the work and it is refereed to as nominal blowing ratio.

Figure 7: Detailed cooling hole geometry for fan-shaped holes to the top-left (side view) and bottom-left (top view) and for cylindrical holes to the right.

2.1. EXPERIMENTAL APPROACH

Row	SH1	SH2	SH3	SH4	SH5
M	3.17	5.04	6.00	3.21	1.68
Axial angle [°]	54.7	68.9	81.4	64.4	54.3
Radial angle [°]	90.0	63.6	60.4	62.6	90.0

Table 3: Showerhead film-cooling hole parameters and blowing ratios

Configuration	Fan-shaped				Cylindrical	
	AR	t_F/P	L_F/D	$\alpha(^{\circ})$	L_C/D	$\alpha(^{\circ})$
Row #1	1.8	0.24	4.3	40	5.5	40
Row #2	1.8	0.24	4.5	37	5.8	37
Row #3	1.3	0.15	5.1	26.6	7.05	26.6
Row #4	1.33	0.15	4.9	31.7	6.39	31.7
Row #5	1.43	0.18	4.05	41.4	5.1	41.4
Row #6	1.43	0.11	4.03	41.3	5.1	41.3
Row #7	1.72	0.19	5.3	28.1	7.05	28.1
Row #8	1.75	0.24	4.7	35.9	6.13	35.9
Row #9	1.8	0.23	4.6	37.3	6.0	37.39
Row #10	1.75	0.26	4.72	36.4	6.1	36.5
Row #11	1.75	0.26	4.72	36.1	8.2	34.5
Row #12	1.83	0.23	4.3	40.7	5.45	37.7

Table 4: Cooling hole parameters for the suction and the pressure side cooling rows. The cylindrical holes maintain $P/D = 4.3$ regardless of the row number.

2.1.3 Measurement Technique

Film cooling performance evaluation requires accurate surface temperature measurements which can be accomplished using either discrete point techniques such as thermocouples or methods with sufficient spatial resolution such as IR- Thermography. The latter case is a well-established technique with many applications and is frequently used in the field of thermo-fluid dynamics. Features of IR-Thermography have been addressed in books and papers of excellent quality and completeness. The basic principles of IR-Thermography will be reviewed here only to the extent that is used in the present study.

This non-intrusive measurement technique is defined based on the electromagnetic radiation, due to the molecular agitation within the matter, that is emitted by a body as its temperature exceeds absolute zero. Thermal radiation emitted from bodies can cover a continuous range of wavelength depending on the material and the conditions. An important concept in this regard, and which the thermal radiation description relies on, is the so called black-body. The definition of the black-body with respect to thermodynamics equilibrium is a body with the ability to absorb all radiation regardless of the wavelength and a perfect emitter of all the

energy, in this case especially considering the IR spectrum.

The emitted radiation from a black-body as a function of wavelength is described using the following equation known as Planck's law:

$$M(\lambda, T) = \frac{C_1}{\lambda^5 (\exp(\frac{C_2}{\lambda T}) - 1)} \quad (9)$$

where $C_1 = 2\pi^5 h_p^6 c^2$, $C_2 = \frac{h_p c}{k_b}$ and k_b is the Boltzmann's constant equal to 1.380710^{-23} J/K. In real world applications the object radiation is not only dependent on its temperature but it is also affected by its surroundings and path. To tackle this problem, the energy balance of a body exposed to radiant energy should be considered. This implies that irradiation to an object by incident thermal radiation, leads to absorption, reflectance or/and transmittance from the body. The equivalent expression for this statement is;

$$\alpha_\lambda + \tau_\lambda + \rho_\lambda = 1 \quad (10)$$

with monochromatic coefficient α_λ as the absorptivity, τ_λ as the transmittivity and ρ_λ as the reflectivity. In a perfect black-body, the reflectance and transmittance properties in equation 10 are zero. However, real bodies are not perfect black-bodies and they emit less than black-bodies at the same temperature. A new property referred to as the emissivity is therefore introduced. Accordingly, the net emissivity power for an object is given by Stefan-Boltzmann's law as,

$$\dot{Q} = \epsilon A \sigma (T^4 - T_\infty^4) \quad (11)$$

where the object with surface area of A has temperature T in a surrounding temperature of T_∞ . The above relation thus serves as a basis for evaluation of the heat exchange by thermal radiation. The monochromatic absorptivity and emissivity follow an equality, according to Kirchhoff law, which implies

$$\alpha_\lambda = \epsilon_\lambda \quad (12)$$

This equation is the main reason for preparing the surfaces that are to be scanned by IR radiometers with a uniform emissivity coating such that they behave like a black-body. This section was a short description to some basic principles of IR-Thermography. Establishing a complete framework, considering all involved aspects such as emissivity, transmittance and reflectivity due to test object, window and surroundings and the camera calibration, is more complex. Presenting the full framework is not within the scope of this work. The following concluding remark therefore encloses this section. A post-processing framework has been accomplished in Matlab (Mathworks, Natick, MA, USA) to take into account

the background radiation, window reflection, absorption, transmittance and vane emissivity and also the camera calibration. The output of IR camera which is digital level (a 14 – bit(0 – 16383) number) that describes the total radiation input detected by each pixel during the exposure time has therefore been post-processed using Matlab.

2.1.4 Data Reduction

To deduct the heat transfer coefficient (HTC) and adiabatic film effectiveness (η) from the time-resolved surface temperature measurements, the method employed by Drost and Bolcs [13] and Reiss [49] has been used. In this method, the test object at a uniform initial temperature is subjected to sudden step changes in the main flow temperature and time-resolved surface temperature data are recorded, in this case by means of IR camera.

In order to determine the convective heat flux using an analytical model of transient conduction, which is to be used here, certain conditions must be met. The first condition is that the heat is transferred only perpendicularly to the surface. The second condition requires that the boundary condition on one side of the surface does not influence the boundary condition on the other side of the surface. In other words the model can be regarded as semi-infinite. The validity of the semi-infinite model sets a test duration limit which according to Schultz and Jones [50] can be estimated by

$$t_{max} = \frac{L_t^2}{16\Lambda} \quad (13)$$

with L_t as the material thickness here and Λ as the thermal diffusivity of the material defined as

$$\Lambda = \frac{k}{c_P \rho} \quad (14)$$

where k is the material thermal conductivity [$\text{Wm}^{-1}.\text{K}^{-1}$], ρ is the density [kg.m^{-3}] and c_P is the specific heat capacity [$\text{J.kg}^{-1}.\text{K}^{-1}$]. The above constraint limits the running time of the test, in this study, to 40 seconds with the material used, however the evaluation of the results are performed on the first 22.5 s. Under such circumstances the heat equation is written as

$$\nabla^2 (kT) = c_P \rho \frac{\partial T}{\partial t} \quad (15)$$

To simplify this equation further, two other conditions must hold, one being the constant thermal properties of the material regardless of the temperature. The

other is that the heat conduction must be one-dimensional, i.e. heat transfer only perpendicular to the surface. These two conditions lead to

$$k \frac{\partial^2 T}{\partial \xi^2} = c_P \rho \frac{\partial T}{\partial t} \quad (16)$$

where ξ is a surface-normal coordinate with $\xi = 0$ at the surface. One of two boundary conditions to be used for solving Eq. 16 stems from the continuity of the heat transfer at the surface (no possibility of heat accumulation) and the other from the semi-infinite model assumption. These conditions are given in Eqs. 17a and 17b:

$$\dot{q} = k \frac{\partial T}{\partial \xi} \Big|_{\xi=0} = h (T_f - T(0)) \quad (17a)$$

$$T(\xi \rightarrow \infty) \rightarrow T_i \quad (17b)$$

and the final boundary condition states that

$$T(t = 0) \rightarrow T_i \quad (18)$$

To solve the problem specified by equations (16-18), they should be made homogenous by subtracting the initial temperature (T_i). The problem can then be solved using the Laplace transform approach. Since, here, only the surface temperature ($T_w = T(\xi = 0)$) is of interest, we obtain

$$\frac{T_w(t) - T_i}{T_f - T_i} = \left[1 - \operatorname{erfcx}\left(\frac{h\sqrt{\Lambda t}}{k}\right) \right] \quad (19)$$

By rearranging Eq. 5 in Sec. 2.1.1, we can find T_f as a function of η ,

$$T_f = T_\infty - \eta(T_\infty - T_c) \quad (20)$$

and replacing T_f from Eq. 20 in 19 will conclude to

$$T_w(t) = T_i + (T_\infty - \eta(T_\infty - T_c) - T_i) \left[1 - \operatorname{erfcx}\left(\frac{h\sqrt{\Lambda t}}{k}\right) \right] \quad (21)$$

with erfcx known as the conjugated error function, $\operatorname{erfcx}(x) = \exp(x^2)\operatorname{erfc}(x)$. Note that the relevant main-flow temperature will be the recovery temperature as discussed in Sec. 2.1.1. In Eq. 21 all the variables are known except η and h . One extra equation is therefore needed to be able to solve the problem. This extra equation comes from the IR sequence which contains multiple pairs of wall temperature at corresponding times. We can thus solve Eq. 21 at pairs of time

points. This implies an overdetermined system of equations. By means of least-square fitting of the measured data, the values of η and h that corresponds to the smallest error will be the result.

To obtain IR images with measured wall temperature in Eq. 21, a Cedip Titanium 560M (SC7600M) MWIR camera fitted with a 50 mm /2.0 lens has been used. This camera has an extended sensitivity range of $1.5 - 5.1\mu\text{m}$, the lens however limited the actual detected spectrum to $2.5 - 5.1\mu\text{m}$. Full camera resolution which is 1280×1024 pixels at a frame rate of 5 Hz is maintained to acquire the images.

The uncertainty analysis for this measurement set-up follows the method used by Moffat [51] and has been discussed in detail in Gustavsson et al. [52]. The following sources of uncertainty have been reported to be taken into account when doing the uncertainty quantification.

- Uncertainty in freestream and cooling temperature measurements using thermocouples
- Uncertainty in background and window temperature
- Uncertainty in material properties of the test object
- Uncertainty in observation angle which might influence the reflectivity and transmittance
- Uncertainty in IR camera signal
- Uncertainty in window properties

The overall uncertainty was estimated to be 5% in h and 0.06 in η . Note that no further quantification of the uncertainty has been made as it was not within the scope and time frame of this work and it is assumed that the same values hold.

Both η and h can be presented in forms of contour plots, see for example Fig. 8, and as a function of lateral direction and stream-wise direction. The lateral direction is normalized by the cooling hole diameter, Z/D , and the zero coordinate indicates the center of the cooling hole in the middle in Fig. 8. Similarly the downstream distance is normalized by the cooling hole diameter, S/D , and $S/D = 0$ refers to the cooling hole center. Note that through this work the data are normalized with respect to reference values of η' and h' .

These quantities, η and h , are also presented in terms of laterally averaged, averaging across the lateral dimension Z . They are denoted as $\bar{\eta}$ (Eq. 22a) and $\bar{h}$ (Eq. 22b), respectively. The averaging in this work is performed in the central region of the 2D profile which includes 4 cooling holes out of 9. The results are then presented as a function of S/D with the same definition as in contour plots.

Figure 8: Contour plots for fan-shaped holes cooling row #3. The lateral and streamwise distribution of normalized AFE, top, and normalized HTC, bottom.

If it is necessary to present the spatially averaged (area averaged) AFE denoted as $\bar{\bar{\eta}}$, then Eq. 22c is used.

$$\bar{\eta}(S) = \frac{\int \eta(S, Z) dZ}{\int dZ} \quad (22a)$$

$$\bar{h}(S) = \frac{\int h(S, Z) dZ}{\int dZ} \quad (22b)$$

$$\bar{\bar{\eta}} = \frac{\iint \eta(S, Z) dZ}{\iint dS dZ} \quad (22c)$$

After averaging in the lateral direction, the data were smoothed using high order polynomial fit with shape preservation to be presented in Papers (I, II, III). Figure 9 shows a comparison between the row data and smoothed data where only interpolation of the data has been used. The discontinuity in the data is associated with limited accessibility to the test object through different windows when observed by the IR-camera.

In addition to h and η , another important parameter often used to evaluate the overall film cooling performance is net heat flux reduction (NHFR). Since this parameter considers the effect of both HTC augmentation and AFE due to coolant injection, as defined in Eq. 23, it can provide a better impression of whether one specific cooling design is beneficial or unfavorable.

$$NHFR = 1 - \frac{\dot{q}_f}{\dot{q}_0} = 1 - \frac{h_f}{h_0} \left(1 - \frac{\eta}{\phi} \right) \quad (23)$$

Figure 9: Comparison of raw data and smoothed data for cooling row #3, fan-shaped without showerhead cooling, to the left laterally averaged AFE, to the right laterally averaged HTC.

In this equation the overall film cooling effectiveness (ϕ) is measured as

$$\phi = \frac{T_w - T_m}{T_c - T_m} \quad (24)$$

and its value under engine-representative conditions for gas turbines is typically between 0.5 and 0.7 according to Mehendale and Han [53]. Although the value of (ϕ) in real engines is dependent on the surface location, in this study a constant value of 0.6 has been used. The same value was used by Lu et al. [54] and Drost and Bölcs [13] in low- and high-speed testing facilities, respectively. A positive value of NHFR indicates that the film cooling effect is beneficial and a negative value of this parameter indicates an unfavorable effect due to film cooling. In this study, always the laterally averaged NHFR results will be presented.

2.2 Computational Fluid Dynamics Approach

The governing equations of fluid dynamics representing mathematical statements of the conservation laws of physics defines what is called CFD. These physical laws are the law of mass conservation, Newton's second law and the first law of thermodynamics. The first physical law derives the continuity equation as given in Eq. 25:

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (25)$$

Note that Einstein summation convention is used through this text for derivation of the Navier-Stokes equations. The mathematical model for Newton's second law leads to derivation of the so-called Navier-Stokes equations. The general form of Navier-Stokes equations in non-dimensional form for a 3D time independent problem and including the compressibility effect is written as

$$u_j \frac{\partial(\rho u_i)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} \quad (26)$$

with

$$\tau_{ij} = 2\mu s_{ij} \quad (27)$$

where

$$s_{ij} = \frac{1}{2} \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \quad (28)$$

The conservation law of energy gives

$$\frac{\partial(\rho c_V T u_i)}{\partial x_j} = -p \frac{\partial u_i}{\partial x_j} + k \frac{\partial T}{\partial x_j} + u_i \frac{\partial \tau_{ij}}{\partial x_j} \quad (29)$$

where p is a function of ρ and T and as an example for a perfect gas we have $p = \rho RT$. The variables in the equations are denoted as density ρ , velocity components u_i , pressure p , viscosity μ , coordinate directions x_i with $i = 1, 2, 3$ in the Cartesian coordinate system, temperature T , gas constant R , thermal conductivity k , and specific heat at constant volume c_V . Note that the energy equation is required here as the heat transfer is of significance. For a three-dimensional problem, such as in this study, Eq. 26 will represent three equations corresponding to each spatial coordinate.

Since there is no known general analytical solution to Eqs. 25, 26 and 29, they have to be solved numerically. To achieve a numerical solution for a given fluid flow problem utilizing these equations, a computational domain bounded in space first must be defined. This domain and its associated boundaries define the flow region to be simulated, where inadequate representation of either can conclude to inaccurate or even unphysical solution of the problem.

In the second step and to be able to solve the problem in hand using numerical schemes, the computational domain has to be subdivided into a number of smaller sub-domains known as mesh, cells or elements. Mesh generation constitutes one of the most important steps in performing a fluid flow analysis and solving the flow physics within the domain geometry.

Then algebraic approximations of the partial differential equations (Eq. 25, 26 and 29) are solved in each cell. The resulting solution provides details of the flow field variables including velocity, temperature, pressure, etc. There are a number of commercial codes which provide a complete CFD analysis package, including defining the domain, meshing, solving and post-processing. A thorough description of all the steps involved is not within the scope of the present study and complementary information regarding different aspects of the computations are addressed in Subsections 2.2.2 and 2.2.3 .

2.2.1 RANS Models for Turbulent Flows

The state of motion for laminar flow regimes can be described by the continuity, momentum and energy equations (Eqs. 25, 26 and 29). However, to be able to

explain the turbulence associated with the existence of random fluctuations in the fluid, more information is required. The change in the flow regime from laminar to turbulent depends on the ratio of inertia force to viscous force, indicated by Reynolds number given as

$$Re = \frac{\rho U l}{\mu} \quad (30)$$

where ρ is the fluid density, U the freestream velocity, l the characteristic length and μ the dynamic viscosity. In a turbulent flow that occurs at high Reynolds number, amplification of the disturbances in flow will occur due to the sufficiently large inertia forces. Such disturbances will lead to random fluctuations in the fluid with properties such as being unsteady, three-dimensional and including ranges of turbulence scales. Depending on the level of resolving or modeling the turbulence scales, different level of solution accuracy may be achieved at different computational cost.

While resolving all eddies associated with different turbulent scales through direct numerical simulation (DNS) leads to the most accurate solution, high spatiotemporal resolution required for such simulations make them only available for lower ranges of Re numbers. In contrast to DNS, Reynolds-Averaged Navier-Stokes (RANS) turbulence models can offer a reasonable and/or often acceptable level of solution accuracy at much lower computational cost. The solutions obtained from RANS, however, contain less physical information and may suffer in accuracy compared to DNS solutions mainly due to modeling of the entire turbulence spectrum.

In-between of the two abovementioned approaches, there are models such as large eddy simulations (LES), detached eddy simulations (DES), and scale adaptive simulations (SAS) which tend to resolve the larger eddies and modeling of the smaller eddies associated with the turbulent motion. Although these models are not as expensive as DNS, they still demand high spatiotemporal resolution, especially for moderate to high Re , to provide sufficiently accurate solutions.

This work is confined to using RANS models for the following reasons. As stated in the experimental method section, (Sec. 2.1), dynamic similarity is obtained for Re number in order to achieve engine representative conditions, dynamic similarity is obtained for different parameters, including Re number. This implies that the problem to be solved involves local Re numbers in the order of several millions. Accordingly, the use of either DNS or scale-resolved models such as LES and DES may not be feasible within the scope of this study. Furthermore, even if it would have been possible to perform such computationally expensive simulations in one or two cases, using such models for parametric studies may not be convenient at the moment.

Time averaging of the turbulent motion of fluid flow by means of RANS equations will lead to complete modeling of all the turbulent motion. To establish

this time averaging, an instantaneous variable ϕ is first decomposed into a time-averaged component $\bar{\phi}$ and a fluctuating component ϕ' such that

$$\Phi(t) = \bar{\Phi} + \Phi'(t) \quad (31)$$

Applying this in Eq. 25, 26 and 29 and subsequently time averaging based on the following equation

$$\bar{\Phi} = \frac{1}{T} \int_{-T/2}^{T/2} \Phi(t) dt \quad (32)$$

yields to derivation of the so-called RANS equations, defined in Eqs. 33 and 34. Note that for the compressibility effects these equations can be interpreted as Favre-averaged Navier-Stokes equations, [55].

$$\frac{\partial (\rho \bar{u}_i)}{\partial x_i} = 0 \quad (33)$$

$$\frac{\partial (\rho \bar{u}_i \bar{u}_j)}{\partial x_j} = -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j} \left(2\mu \bar{s}_{ij} - \rho \overline{u'_i u'_j} \right) \quad (34)$$

with

$$\bar{s}_{ij} = \frac{1}{2} \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \quad (35)$$

The solution to the RANS equations will provide the time-averaged properties. However, to be able to solve this problem, the system of equations should be closed. Due to introducing of the averaged products of unknown fluctuating components, $(-\rho \overline{u'_i u'_j})$, referred to as Reynolds stresses, the number of unknowns is thus greater than the number of equations. Finding an appropriate way to accurately determine the Reynolds stresses to close the RANS equations, known as the "closure problem", is the challenge in turbulence modeling. Although a few methods have been proposed in recent decades, there is still no unique solution which suits for all turbulent problems. Of the different methods, some have been used more frequently due to their superior performance for certain engineering problems. Most of the frequently used techniques have been developed based on the Bosussinesq hypothesis. The inherent assumption within this hypothesis is the linear relation between the Reynolds stresses and the velocity gradients of the mean flow, written as

$$-\rho \overline{u'_i u'_j} = 2\mu_t \bar{s}_{ij} - \frac{2}{3} \rho k \delta_{ij} \quad (36)$$

with μ_t referred to as the eddy (or turbulent) viscosity, k defined as

$$k = \frac{1}{2} \overline{u'_i u'_i} \quad (37)$$

and δ_{ij} representing the Kronecker delta with the property of being equal to unity if $i = j$ and zero otherwise. There are models of varying complexity for solving the eddy viscosity, three of which have been considered for benchmarking in this study. These are the one-equation Spalart-Allmaras model [56] that solves a modeled transport equation for the kinematic eddy (turbulent) viscosity, the two equation model realizable $k - \epsilon$ which has two model transport equations for the turbulent kinetic energy k and its dissipation rate ϵ as presented by Shih et al. [57] and the shear stress transport (SST) model developed by Menter [58] that effectively blends the $k - \omega$ formulation for the near wall region with the $k - \epsilon$ model in the far field.

2.2.2 Computational Domain and Mesh

The physical experiments have a domain similar to the one presented in Fig. 10. The test object of this model simulates a 2D profile vane as shown in the figure. This full model vane is about 10 cm long in the span direction, which allows up to 9 cooling holes for one row with the desired characteristics. Preliminary CFD calculations for validation of the uncooled model, including non-dimensional pressure distribution and HTC validation, have been performed on this model, see for example [59]. However, in order to reduce the computational cost by an order of magnitude only a slice of the model including one cooling hole has been used for film cooling investigations. Figure 10 shows an example of the sliced computational domain with one cooling hole on the suction side and one on the pressure side that are used for CFD based sensitivity analysis in this study.

The structured multi-block Hexa mesh for the domain of interest was constructed using ICEM/CFD (ANSYS, Inc., Canonsburg, PA, USA). Increasing the orthogonality of the mesh is achieved by means of an O-grid. It is established among CFD communities that sufficient spatial mesh resolution is a necessary (but not sufficient) condition to obtain valid CFD results. This implies that CFD solutions should be mesh-independent. A mesh sensitivity study was carried out for the narrow model, cylindrical hole #1 at a blowing ratio of $M = 0.6$ and the laterally averaged AFE results proved to be mesh-independent for approximately 20 M cells. The mesh is constructed such that the details of the jet will be captured in the inside and vicinity of the holes by means of high mesh density. To resolve the boundary layer, the established mesh should also satisfy the constraint of a maximum y^+ of about unity on the vane. Figure 11 illustrates a schematic view of the generated computational grid close to the cooling hole.

Figure 10: The computational domains with the full domain considering uncooled vane, to the left, and a sliced model (narrow) consisting of one cooling hole on the suction side and one on the pressure side, to the right. The zoom box shows a close-up of the plenum chamber and fan-shaped cooling hole on the pressure side.

Figure 11: Schematic view of the computational grid close to the hole exit of the fan-shaped hole.

2.2.3 Numerical Settings and Boundary Conditions

The finite-volume based solver Ansys Fluent 13.0 was used to conduct the CFD simulations. The pressure-velocity coupling in the momentum equations is made using the COUPLED scheme. Spatial discretization of the momentum, energy, pressure, density and turbulent equations is handled by means of a second-order scheme [60].

The selected fluid was air as an ideal gas to include the compressibility effects. While the main inflow has a constant mass-flow rate, the plenum chambers sustain a mass-flow corresponding to a specific blowing ratio, similar to the experiment, defined in Eq. 6. Turbulence intensity and length scales for the main flow were chosen as 17% and 0.012 m, respectively. The turbulence level for the coolant is set to 3% and a corresponding hydraulic diameter, depending on the plenum chamber characteristics, is used.

All the simulations were carried out on the Linux Cluster Neolith at National Supercomputer Centre (NSC, Linköping University, Sweden).

2.3 Correlation Approach

Empirical correlations are commonly used to predict laterally averaged adiabatic film effectiveness. They are mathematical formulations of different levels of complexity derived from experimental test results and may cover ranges of flow as well as geometrical parameters. When selecting a correlation model, two important topics should be considered, one related to the limitations of the model and the other concerning the predictability of the model.

The first topic addresses the practical limitations that a correlation model may have. These include the number of parameters used, the interval to which the variation in one specific parameter is limited, uncertainty and error associated with experimental results, the level of complexity associated with the correlation model, and also the conditions under which the experiments are conducted. Since correlations are employed in the engines' design stage, the last factor in fact represents the validity of the derived correlation model when it is compared to real engine conditions. This may or may not be reflected on the conducted experiments.

The second topic, model predictability, which is not explicitly considered by others, to the author's knowledge, is often of greater importance. This is due to the fact that design engineers demand models that can predict the status of one specific design as accurately as possible when a design parameter is changed. In other words, information regarding the correlation model's predictability will help the design engineers to determine the level of accuracy that a model can offer.

Nevertheless, the objective of this study was to develop a method of deriving correlation models which can be varying levels of complexity and that has the potential for further extension if necessary. Accordingly, the aforementioned topics should be considered and possible proposed models need to be investigated in detail. The subsequent sections aim to address the essence of the developed method.

2.3.1 Influencing Parameters

Film cooling performance is influenced by a variety of parameters, including both geometrical aspects and flow properties. Numerous studies have addressed the importance and impact of the relevant parameters giving the highest contribution. Baldauf et al. [35] have investigated a broad set of parameters on a flat plate for cylindrical holes and according to them in general the laterally averaged AFE can be formulated as a function of

$$\bar{\eta} = f \left(M, DR, Tu, \frac{S}{D}, \alpha, \frac{P}{D}, \frac{\delta}{D}, \frac{L}{D} \right) \quad (38)$$

It should be noted that some of the variables are adapted to the definitions used in this study and are therefore different to the original postulated functionality given by Baldauf et al. [35]. In "A review of shaped hole turbine film-cooling technology", Bunker [61] reviewed some of the relevant parameters that affect AFE of the shaped holes.

The most common types of cooling holes are cylindrical and shaped holes, where the first category is simpler and has been used since the early stages of film cooling development. Shaped holes have proven to be more efficient and since the manufacturing barriers to producing these type of holes have been overcome, they are being used instead of cylindrical holes more frequently as of late. Fan-shaped holes with a lateral expansion and fan-shaped laid-back holes with a longitudinal expansion are two well-known examples of shaped holes.

Some of the influencing parameters have the same definition for both cylindrical and shaped holes such as downstream distance and blowing ratio. While some others may have different definitions or do not exist for one type compared to the other type. For example, the ratio of hole breakout to the hole spacing, t_F/P , is defined only for shaped holes. Even though a parameter might be defined the same for both cooling hole types, they may not affect the performance of film cooling in a similar manner. Nevertheless, among various important parameters here only three will be reviewed as they are considered when developing the method to derive correlation.

Effect of Downstream Distance (S/D): Both AFE and HTC are often plotted as functions of downstream distance, non-dimensional with respect to the hole diameter. In general, AFE has its maximum value at the hole exit and decays to lower levels further downstream. The location of this maximum moves further away from the hole exit for cylindrical holes as the blowing ratio increases and the jet starts to separate from the surface, often from blowing ratio 0.5. Fan-shaped holes with a large enough cross-sectional area ratio, see for instance Baldauf et al. [35], can sustain a maximum level of AFE at the hole exit up to blowing ratio as high as 4.0.

Effect of Blowing Ratio (M): Blowing ratio is defined similarly for both cooling hole shapes, as given in Eq. 6. Note that the effective blowing ratio, $M_{eff} = M/AR$, at the hole exit is not employed here since it was decided to have a constant mass-flow for both cooling hole shapes. In general, the level of AFE increases as M increases but will start to decrease as soon as the jet starts to lift-off from the surface and is thus penetrated into the hot gas stream.

It has been reported by Colban et al.[46] that fan-shaped holes can sustain a rise in AFE up to blowing ratio 4.0, although the increase is not proportional to the increase in M . It is worth mentioning that such a conclusion is made based on the specific film cooling configuration being investigated, i.e. $AR = 3.9$, $P/D = 6.5$ and $t/P = 0.48$.

For cylindrical holes, on the other hand, increasing M up to only about 0.5 can cause an increase in AFE and a further rise in M will conclude to lower AFE due to jet lift-off, which allows the hot gas to slip between the jet and the surface. This will also cause migration of the location of peak AFE towards downstream of the hole, see [44]. This discussion is clearly valid for low density ratios and based on flat-plate studies.

Effect of Hole Location: Among various parameters, film cooling effectiveness has proven to be affected by a number of approaching flow parameters, e.g. Re , δ/D and etc., as reported by Baldauf et al. [35]. This implies that if there are circumstances such that either of such parameters is subjected to continuous change along the surface on which the film cooling hole is located, then the performance of the cooling will vary since different jet interactions can occur depending on the placement of the cooling hole. Occurrence of such circumstances, i.e. continuous change in flow, is in no doubt as regards flow around an airfoil. It can therefore be concluded that film cooling performance of a cooling hole positioned at one specific location on the suction or pressure side of a guide vane may differ from the cooling hole with the same characteristics but at a different location. In addition to the approaching flow condition, curvature and pressure gradients (after the cooling hole) can also affect the AFE, see for example [62] and [11].

To account for different effects dictated by the curvature of a guide vane when developing a correlation, there are two alternative solutions. Either different correlations have to be derived for cooling holes in various positions or one correlation can be derived for the entire surface of interest by adding a parameter which takes into account the effect of hole position regarding flow changes, curvature effect, etc. However, in an attempt to develop the method the first approach has been used, see Nadali et al. [63]. However, implementation of the later approach offers the possibility to consider the cooling hole position effect for possible improvements in the film cooling performance.

Although, different parameters could be considered for the hole position effect such as boundary layer thickness, flow acceleration and/or non-dimensional

pressure coefficient, the last case was selected for the purpose of this study. For further discussion on the choice of variable, see Paper V. The local C_p values at the exit of a cooling row were therefore used to represent the effect of approaching flow at a particular point in the film-cooling effectiveness. Note that curvature and pressure gradient effects that could possibly influence the film cooling performance after the hole exit are included in the data in this case due to the fact that the experiments were conducted on a vane geometry.

To conclude, for the final correlation model derived in this work and presented in Sec. 3.4, the variables X_1 , X_2 and X_3 are used and are defined as

$$X_1 = \ln \zeta \quad (39a)$$

$$X_2 = M \quad (39b)$$

$$X_3 = C_P \quad (39c)$$

In Eq. 39a the following definitions hold:

$$\zeta_{Cyl} = \frac{4}{\pi} \frac{S}{D} \frac{P}{D} \quad (40a)$$

$$\zeta_{Fan} = \frac{4}{\pi} \frac{S/D}{AR} \frac{P/D}{AR} \quad (40b)$$

with AR denoting area ratio for fan-shaped holes and is defined as the ratio of the cross-sectional area at the hole exit, A_{exit} , to the cross-sectional area at the hole entry, A_{in} .

2.3.2 Statistical Considerations

Since derivation of a correlation model of any type demands the use of at least a regression analysis, for determination of the coefficients, an appropriate statistical procedure should be followed. A further requirement for a developed correlation model is to know its predictability, which may in fact identify the limitations associated with the model. This would be possible to obtain only if correct statistical schemes are employed. It can therefore be concluded that special attention should be paid to important aspects of the statistical procedure when developing a correlation model.

The first aspect to take into account is obviously the uncertainty and error associated with the data set, which has been described in detail in data reduction, Sec. 2.1.4.

Furthermore, collected data must have a normal distribution which is a function of mean and standard deviation. This is indeed a necessity for using regression analysis and further statistical analysis. For reasons such as data complexity, a data set may not have a normal distribution by its nature, so a transformation of

some kind should be applied to the data to achieve this. In this work, the Box-Cox method was used to obtain normally distributed data for further analysis, see Box and Cox [64] and [65].

Another important aspect to be considered is the regression analysis. In order to explore the interaction effects between predictors during the model building process stage, a tentative regression model or several appropriate regression models should be built. One must then decide whether one model is preferred over another or not. For this reason, an appropriate procedure for investigation of different models must first be found, and thereafter the best model selected. The number of possible models, 2^{N-1} , which can be suggested increases rapidly as the number of predictors, N , increase and consequently the process will become time-consuming. Finding a suitable procedure that can make an automatic computer search to find the best model will therefore be imperative. In this respect, the stepwise regression method was used here, see Paper V and [66]. Afterwards, the appropriateness of the model should be measured quantitatively. This was done using the adjusted coefficient of determination, R_{adj}^2 , which is useful when models with different numbers of predictors are investigated.

A further essential aspect deals with the two-way interaction between variables, for instance $X_i.X_j$ or even one variable powered by two $X_i.X_i$. If the correlation model contains the two-way interaction, special attention should be paid to the possibility of multicollinearity, i.e. there might be in fact no two-way interaction and including such terms will lead to predictability failures. To prevent multicollinearity, it is a requirement to subtract the mean value of a corresponding predictor from the data. This means that the variables are multiplied by each other or raised to the power of two according to

$$X_i^2 = [X_i - \text{mean}(X_i)]^2 \quad (41a)$$

$$X_i.X_j = [X_i - \text{mean}(X_i)].[X_j - \text{mean}(X_j)] \quad (41b)$$

In addition, measuring the variance inflation factor, VIF , can be used as a tool to determine the presence or absence of multicollinearity and was followed in this work. In general, it is recommended to have a VIF value below 5.0, [66], and monitoring this value for all investigated correlation models has guaranteed the absence of multicollinearity while indicating the existence of two-way interaction between variables. The commercial software Minitab [66] to build the regression models and also perform the statistical analysis was used here.

2.3.3 Method Development

The proposed correlation method follows a number of simple steps. The first step is data collection for ranges of important parameters with desirable intervals of

variation corresponding to each parameter, e.g. $0.3 \leq M \leq 2$. In this regard, it is essential to note that the interval sets the boundaries which the model predictability lies on, i.e. outside these intervals a high level of uncertainty may be associated with the predictions.

A convenient increment in the variations of the values in the specified interval will also help the model to capture different physical scenarios associated with each parameter. This is in fact a requirement if a parameter is believed to have non-linear behavior with respect to the response, in this case laterally averaged or spatially averaged adiabatic film effectiveness. For further clarification an example of this issue is given below.

Different studies, including Paper IV, have shown that the variation in blowing ratio as a function of spatially averaged AFE is nonlinear. This non-linearity reflects the flow separation from the surface and jet lift-off, which in turn decreases the laterally averaged AFE when regarded as a function of downstream distance and accordingly the surface integral of quantity is decreased. It is thus shown that the spatially averaged AFE may increase up to a certain blowing ratio, depending on the hole shape, etc., and then it starts to decrease. Including data which represents both lift-off and non-lift-off scenarios will therefore give the model the capability to predict such physical phenomena.

From the second step, the statistical analysis phase is started and using an appropriate commercial or in-house program is recommended. In this step, collected data are prepared in terms of the response, $\bar{\eta}$, as a function of the selected parameters, in this case S/D , M and C_p and the two-way, if required three-way, interaction between the parameters. To prevent multicollinearity, which is discussed in Sec. 2.3.2, the mean value of the parameters must be subtracted from the main variables. In the next step, the distribution of the data is monitored to confirm the normal distribution and if necessary a transformation of some kind needs to be made to convert the data. This transformation will give a coefficient which will be part of the correlation model.

Afterwards, a stepwise regression analysis is performed on the data to define the possible two-way interaction between variables, if there is any multicollinearity, the error, and model predictability. In the final step, the coefficients of the correlation model are obtained from regression analysis with the inclusion of the variables highlighted in the previous step and exclusion of the variables with no effect.

Chapter 3

Results and Discussion

In order to accomplish the objectives of this work, experimental and numerical investigations were conducted on the suction and pressure sides of a simulated turbine nozzle guide vane where a number of film cooling configurations were examined. The availability of an extensive experimental database provided the opportunity to study ranges of influencing parameters, including hole shape, blowing ratio, showerhead effect, heat transfer coefficient augmentation due to cooling injection, performance of multiple cooling rows, etc. Some of the major findings from the experimental work are presented in Papers I, II, and III. A CFD-based sensitivity analysis, Paper IV, was also incorporated into exploration of some of the influencing parameters. A method has been developed for deriving correlation models by means of experimental data and as an example a correlation model is suggested in Paper V.

The objective of this chapter is to summarize the results of the papers, supported by some supplementary materials and comments formulated towards the aims of the work.

3.1 Suction Side Cooling

The suction side of a turbine guide vane is often subjected to high-to-moderate convex surface curvature and experiences flow acceleration followed by adverse pressure gradients after maximum throat velocity, see for example Fig. 4 in Sec. 2.1.2. In Papers I and II, the film cooling performance of cooling holes at two different positions on the suction side is investigated. One position is close to the maximum throat velocity, rows #1 in Fig. 5, and the other is located at rather high convex surface curvature, rows #3 in the same figure.

Since the laterally averaged data are presented in the papers, examples of the lateral and stream-wise variation in AFE and HTC results from experiments are

presented here as complement. The data are normalized by the reference values of η' and h' and as regard of HTC, the normalization differs from the HTC augmentation that is h_f/h_0 . The overall objective, when looking into different film cooling configurations, is to achieve higher level of AFE and the lower level of HTC.

Figure 12: Normalized AFE distribution for cooling row #1 in the absence of showerhead cooling (top) and in its presence (bottom). In each case, $M = 0.6$, $M = 0.9$ and $M = 1.2$ are from top to bottom, respectively, and fan-shaped hole results are presented to the left and cylindrical holes to the right. Note that $Z/D = 0$ is the center of the middle cooling hole and $S/D = 0$ indicates the cooling hole center for this row.

For cooling row #1, close to maximum throat velocity, Fig. 12 shows the reduction in AFE when the blowing ratio is increased for fan-shaped holes in the absence of showerhead cooling. This figure shows the decrease in effective film jet in the lateral direction quite clearly. While $M = 1.2$ has the most noticeable decrease in AFE in the stream-wise direction, the other two blowing ratios give almost similar results after $S/D = 35$, also illustrated by means of laterally averaged data presented in Paper I.

It is known that fan-shaped holes with a large area ratio, AR , can maintain an increase in AFE up to blowing ratios as high as 4.0, which is due to lateral diffusion of the film jet, a decrease in jet momentum at the hole exit indicating no jet separation and good coverage, see Baldauf et al. [44]. The fan-shaped holes used in this study have a low area ratio ($AR < 2$) as indicated in Tab. 4), which is dictated by the design constraints for medium-size gas turbines. This in turn leads to moderate jet separation occurring on row #1 when the blowing ratio is increased from 0.6 to 1.2, which enhances the mixing and reduces the AFE, see fan-shaped hole without showerhead cooling in Fig. 12. For the same cooling hole, #1, but in the presence of showerhead cooling, this phenomena is not obvious, although the highest blowing ratio seems to maintain a lower level of AFE. Cylindrical holes on this row also exhibit reduced AFE when M is increased; however, the difference is that in this case strong flow separation and a narrower effective film jet can be seen, Fig. 12. Similar to fan-shaped holes, the presence of showerhead will not have any influence on flow separation at blowing ratios 0.9 and 1.2 despite of increased level of effectiveness.

For film cooling row #3, fan-shaped holes, an increase in M will improve the AFE when showerhead cooling is not present, Fig. 13. However, the improvement is not pronounced if the two highest M are compared. This figure shows that presence of showerhead cooling will increase the AFE to a higher level. It is also indicated in the figure that the two highest blowing ratios have identical performance in terms of AFE, also seen from laterally averaged AFE data in Paper I. Moderate to severe jet lift-off can be clearly seen in this figure for the cylindrical holes in both the presence and the absence of a showerhead.

A cross-comparison between Figs. 13 and 12 for fan-shaped holes in the absence of showerhead cooling gives the impression that the bimodal effectiveness pattern related to the separation bobble downstream of the diffuser can be more clearly seen for row #3 than row #1. This is indicated by an oval shape just downstream of the cooling hole. However, Saumweber and Schulz [67] reported this phenomena for fan-shaped holes with $L_F/D \approx 2$ (L_F is the length of the cylindrical part of fan-shaped holes). The same pattern can be seen here with $4.3 \leq L_F/D \leq 5.1$ for row #3 but not for row #1. In addition, from the same comparison the coolant concentration is in the center-line area for the latter cooling row, while row #3 has a wider jet footprint.

It is also worth mentioning that the lateral variation in the AFE for all the considered cases clearly demonstrates the jets in isolation. This concludes to better mixing between the hot gas flow and coolant and consequently reduced AFE. The surface coverage is also weak under this circumstance.

The superposition effect due to the showerhead and rows of cooling holes on the suction side has been addressed in Paper I. It has been demonstrated that the additive method suggested by Sellers [68] will under-predict the AFE of the com-

Figure 13: Normalized AFE distribution for cooling row #3 in the absence of showerhead cooling (top) and in its presence (bottom). In each case, $M = 0.6$, $M = 0.9$ and $M = 1.2$ from top to bottom, respectively, and fan-shaped hole results are presented to the left and cylindrical holes to the right. Note that $Z/D = 0$ is the center of the middle cooling hole and $S/D = 0$ indicates the cooling hole center for this row.

bin effect of a single row and showerhead cooling in isolation when compared to a single row of cooling holes in the presence of showerhead cooling. However, it can be recommended that the methods be further investigated to predict the superposition effect of multiple film cooling rows such as the ones discussed in Andreini et al.[69], Zhu et al. [70] and Benjamin and Thomas [71]. This can be beneficial for design purposes to predict heavily cooled turbine vanes which use multiple arrays of cooling holes.

Illustrated in Fig. 14 are the resulting HTC for row #1. The figure shows the increase in HTC when the blowing ratio is increased for both fan-shaped and cylindrical holes in the absence of showerhead cooling. For the lowest blowing ratio, $M = 0.6$, the jet is most likely attached to the surface for both hole types. An increase in M leads to flow separation. The consequence of such a flow alteration

will be enhancement in the mixing caused by induced turbulence and disturbed boundary layer, which concludes to an increase in the level of HTC.

Cylindrical holes exhibit lower HTC compared to fan-shaped holes as indicated in Fig. 14, which is consistent with the findings of Dittmar et al. [72]. Since the latter cooling hole type has a wider jet footprint and the area of disturbed boundary layer is increased, a higher level of HTC is experienced by fan-shaped holes and this is more pronounced for the highest blowing ratio.

Figure 14: Normalized HTC distribution for cooling row #1 in the absence of showerhead cooling (top) and in its presence (bottom). In each case $M = 0.6$, $M = 0.9$ and $M = 1.2$ from top to bottom, respectively, and fan-shaped hole results are presented to the left and cylindrical holes to the right. Note that $Z/D = 0$ is the center of the middle cooling hole and $S/D = 0$ indicates the cooling hole center for this row.

Figure 14 also shows that fan-shaped holes in the presence of showerhead cooling will lead to a lower level of HTC, while cylindrical holes conclude to lower HTC in the absence of showerhead cooling. One possible explanation for this contradictory performance may be given with respect to difference in the performance of these cooling hole shapes when exposed to different states of bound-

ary layer. Cooling row #1 is located at the beginning of the formation of a new boundary layer when there is no showerhead cooling and in the presence of showerhead cooling it experiences a settled boundary layer (see Fig. 11 in Paper I). Note that injection through a showerhead will make a thicker boundary layer while at the same time it will enhance the HTC. This figure thus suggests that fan-shaped holes may reduce the kinetic energy within a thick and well-developed turbulent boundary layer and hence reduce the HTC, see Bunker [61], while cylindrical holes will elevate the turbulence level further and enhance the HTC under the same circumstance.

Figure 15: Normalized HTC distribution for cooling row #3 in the absence of showerhead cooling (top) and in its presence (bottom). In each case $M = 0.6$, $M = 0.9$ and $M = 1.2$ from top to bottom, respectively, and fan-shaped hole results are presented to the left and cylindrical holes to the right. Note that $Z/D = 0$ is the center of the middle cooling hole and $S/D = 0$ indicates the cooling hole center for this row.

Figure 15 indicates that general findings concerning row #1, such as a higher level of HTC for fan-shaped holes compared to cylindrical holes and also elevated HTC when M is increased, can be extended to row #3. The performance of this

row, however, is different from row #1 when cases in the presence and absence of showerhead cooling are compared.

This row experiences either the beginning of a transitional boundary layer from laminar to turbulent, in the absence of showerhead cooling, or the end of boundary layer thinning and the start of development of a thick turbulent boundary layer in the presence of showerhead cooling (see Fig. 11 in Paper I). Injecting coolant through this row in the presence of showerhead cooling will disturb the state of a fresh boundary layer development with high kinetic energy content due to the showerhead and consequently it will enhance the HTC to a greater extent, which is consistent with the explanation given by Bunker [61]. In this case, both cooling hole types react in the same way to the state of approaching flow.

As concluded in Paper I, on the one hand the effect of showerhead cooling on film cooling performance on rows #1 and #3 may be different when either AFE or HTC augmentation is considered. On the other hand, from the overall film cooling performance, i.e. net heat flux reduction (NHFR), point of view, this is the cooling hole shape which is influenced by the presence or absence of the showerhead cooling regardless of the hole position, although in this case the cooling hole close to the maximum throat velocity was considered to be more beneficial in general compared to the cooling row at high convex surface curvature. It should be noted that for cross-comparison between fan-shaped holes on two different rows, it has been assumed that the small geometrical differences between the two rows will not have any impact on the film cooling performance. According to Gritsch et al. [73], most hole geometry parameters such as L_F/D and AR will have negligible effect on fan-shaped hole film cooling performance.

Paper II addresses the performance of double row cooling holes, double row #1 and double row #3 in Fig. 5, and compares that with the single row cases in the presence of showerhead cooling for fan-shaped and cylindrical holes. It is shown that the double row cylindrical and fan-shaped holes close to maximum throat velocity, i.e. rows #1, lead to almost identical NHFR results for low blowing ratios. However, their overall performance starts to deviate if the blowing ratio is increased, especially for the downstream distances below $40D$. Furthermore, it has been found that at this position there are circumstance that a double row cylindrical hole without showerhead cooling may conclude to higher NHFR results compared to a single row fan-shaped hole in the presence of showerhead cooling.

In addition, as indicated in Fig. 16 a double row fan-shaped hole in the absence of showerhead cooling gives better overall film cooling performance compared to the single row case with showerhead cooling. This holds for virtually all blowing ratios to some extent. For cylindrical holes, the same conclusion holds more or less as seen in Fig. 16, although for $M = 1.2$ the double row case has a high drop in NHFR in the vicinity of $S/D < 50$, which is due to severe jet lift-off at

the hole exit. This causes very low AFE and relatively high HTC augmentation downstream of the hole and in turn reduces the level of NHFR in this region.

Comparing fan-shaped and cylindrical holes also shows that there are circumstances where the performance of a single or double row of one cooling hole shape could be similar to a different case with the other cooling hole shape.

Figure 16: Comparison of the NHFR results for single row cooling, #1, in the presence of showerhead cooling and double row cooling, #1, without showerhead cooling. Fan-shaped hole results are presented to the left and cylindrical holes to the right.

For the double or single row cooling hole at high convex surface curvature, i.e. rows #3, one of the findings in Paper II was that a double row cylindrical hole in the absence of showerhead cooling may result in the same level of adiabatic film effectiveness compared to a single row cylindrical hole in the presence of showerhead cooling, although the earlier case concludes to better NHFR.

Figure 17: Comparison of the NHFR results for single row cooling, row #3, in the presence of showerhead cooling and double row cooling, #3, without showerhead cooling. Fan-shaped hole results are presented to the left and cylindrical holes to the right.

Figure 17 shows that the general remarks taken from Fig. 16 can be extended for double row #3 without and single row #3 with showerhead cooling for both fan-shaped and cylindrical holes. One exception observed in Fig. 17 is that the single row fan-shaped case at $M = 1.2$ has slightly higher NHFR compared to the double row of the same cooling hole shape case at $M = 0.6$. Moreover, in the case of cylindrical holes this figure reveals that a double row case at low

to moderate blowing ratios can be regarded as the best configuration, since it can achieve positive NHFR all the way from the cooling hole to the trailing edge, i.e. $S/D \approx 160$. In addition, comparison between cylindrical and fan-shaped holes gives the impression that the overall performance of the latter cooling hole shape is better, especially if the first $40D$ is also of great importance, since in this case positive NHFR is obtained when this type of holes is used.

3.2 Pressure Side Cooling

Pressure side cooling has been investigated by looking into film cooling performance of rows of holes located at various positions. Due to high heat load close to the trailing edge and its importance from the point of view of both HTC and AFE, the focus was laid to cooling configurations close to this region. Therefore, the results of last cooling hole on the pressure side, row #12 in the absence of showerhead cooling, its combination with row #11 in the presence of showerhead (double rows) and also the combination of three rows #10, #11 and #12 with showerhead cooling (triple rows) are presented both here and in Paper III. Note that the cooling holes for row #11 are staggered with respect to the other two rows. Since laterally averaged HTC augmentation results are presented in the paper, the lateral and streamwise variation in both AFE and HTC results, from experiments, are illustrated and discussed here.

Figure 18: Normalized AFE distribution for cooling row #12 in the absence of showerhead cooling. Fan-shaped hole results are presented to the left and cylindrical holes to the right. $M = 0.6$, $M = 0.9$ and $M = 1.2$ corresponding to each case are from top to bottom, respectively.

As shown in Fig. 18, for single row #12 in the absence of showerhead cooling the AFE is slightly decreased in the vicinity of the hole exit in both the lateral direction and also to some extent in the main-flow direction when the blowing

ratio is increased for fan-shaped holes. For $S/D > 25$, this figure shows a small increase in overall AFE when the blowing ratio is raised. The figure also shows the separation bubble at the exit of the hole for the lowest blowing ratio. The same conclusion could be drawn for cylindrical holes, the difference being that in this case a clear reduction can be seen in AFE for the region $S/D < 6$ when the blowing ratio is increased, which is due to severe jet lift-off. This consistent increase for larger S/D may be caused by concave curvature which helps the lifted jet to be reattached to the surface further downstream of the hole.

Figure 19: Normalized AFE distribution for double row #11 and triple row #10 in the presence of showerhead cooling, top and bottom, respectively. Fan-shaped hole results are presented to the left and cylindrical holes to the right for $M = 0.6$, $M = 0.9$ and $M = 1.2$ from top to bottom. The two $S/D = 0$ correspond to the cooling hole center first for row #11 and then row #12 from left to right.

Comparing the double and triple rows in Fig. 19 with the single row case in Fig. 18 shows the effect of showerhead cooling together with row #11 in the first step and then row #10 in the second step when fan-shaped holes are considered. Note that in this figure, $12 < S/D < 18$ to the left side of the triple row corresponds to the downstream of the first cooling hole which is not included. This

interval also compacts the triple row contour and this effect must be considered in discussions.

Figure 19 shows that the significance of the improvement that may be achieved for cylindrical holes corresponds to the triple row case. The double row cylindrical holes in the presence of showerhead cooling may thus perform only slightly better, hardly visible in the figure, than the single row in the absence of showerhead cooling, considering $S/D > 25$ after row #12.

Moreover, for fan-shaped double row and triple row cooling holes when M is increased from 0.6 to 0.9, the improvement in AFE is clear but further increases in M do not seem to be influential, see Fig. 19. For cylindrical holes, although qualitative comparison between different blowing ratios seems not to be straightforward and require laterally averaged data. The increase in AFE in the lateral direction after row #11, Fig. 19, due to the existence of an upstream cooling row, i.e. row #10, is clearly seen for both cooling hole types when double row and triple row cases are compared.

In fact, an upstream cooling row staggered with a further downstream cooling row leads to better coverage due to expansion of the film jet footprint after the second row even when the distance between two rows is greater than $12D$.

Figure 20: Normalized HTC distribution for cooling row #12 in the absence of showerhead cooling. Fan-shaped hole results are presented to the left and cylindrical holes to the right. $M = 0.6$, $M = 0.9$ and $M = 1.2$ corresponding to each case are from top to bottom, respectively.

Figure 20 shows the high level of heat transfer coefficient due to injection of cylindrical holes compared to fan-shaped holes for row #12 in the absence of showerhead cooling. This is sustained for virtually all investigated blowing ratios and specifically for a downstream distance greater than $25D$.

It can also be seen from the figure that in contrast to suction side cooling, here the fan-shaped holes lead to lower HTC even in the vicinity of the cooling hole

Figure 21: Normalized HTC distribution for double row #11 and triple row #10 in the presence of showerhead cooling, top and bottom, respectively. Fan-shaped hole results are presented to the left and cylindrical holes to the right for $M = 0.6$, $M = 0.9$ and $M = 1.2$ from top to bottom. The two $S/D = 0$ correspond to the cooling hole center first for row #11 and then row #12 from left to right.

despite a wider lateral jet footprint compared to cylindrical holes. In general, a higher blowing ratio will conclude to a higher level of HTC.

It is discussed in Paper III that close to the trailing edge, the position of this row of cooling holes, the flow is subjected to acceleration and boundary layer thinning. Now, coolant injection through fan-shaped holes appears to nicely make the boundary layer thicker and also decrease the kinetic energy within the boundary layer while cylindrical holes will trigger the boundary layer and due to acceleration, the HTC is elevated to a much higher level compared to fan-shaped holes. Although it is shown in Paper III that cylindrical holes for this row in the absence of showerhead cooling causes a high level of HTC augmentation (the level of HTC in the cooled case to that of the uncooled case), from Fig. 20 it can be seen that fan-shaped holes without showerhead cooling for this row, #12, may not indicate the same conclusion.

Comparing the case considered above and the double and triple row in the presence of showerhead cooling in Fig. 21 reveals a reduction in HTC for the double and triple row cases for both cooling hole types. For each hole type there seems to be negligible differences between the double and triple row cases when there is an upstream showerhead cooling. This suggest that within a thick enough turbulent boundary layer sought to be due to showerhead cooling, further coolant injection through rows of cooling holes will not enhance the HTC. Under this condition, the kinetic energy of the boundary layer can even be reduced, if coolant is injected at low momentum, which consequently decreases the HTC. This may be a solution for reducing HTC in accelerating and boundary layer thinning flows.

In addition, comparison of fan-shaped and cylindrical holes for the double and triple rows in Fig. 21 indicates slightly lower HT for the latter cooling hole type for the lowest blowing ratio. For higher blowing ratios the performance of both hole types is to a large extent similar. These findings are consistent with the HTC augmentation results presented in Paper III.

3.3 The Effect of Approaching Flow

CFD calculations were conducted on a number of configurations to investigate the effect of approaching flow and curvature on film cooling performance of cylindrical and fan-shaped holes for ranges of blowing ratio. For this purpose a turbulence model investigation and validation study was first carried out. It has been shown in Paper IV that while the Realizable $k - \epsilon$ and $k - \omega$ SST turbulence models suffer from over-prediction of the laterally averaged adiabatic film effectiveness, the one-equation turbulence model, Spalart-Allmaras, could provide the best prediction of this quantity as compared to experiments and for the investigated case, i.e. fan-shaped hole row #3 blowing ratio 0.49.

The lateral and stream-wise distributions of the AFE results for one representative cooling hole for this row from the CFD calculations and the experiments indicated in Fig. 22, show an overall agreement between the two methods in terms of increase in jet stretching when M is increased. Note that only the selected turbulence model (S-A) results are presented. This figure also shows the over-prediction of this quantity in the flow direction and its underestimation in the lateral spread for all investigated blowing ratios. The turbulence model predicts too weak jet spreading and may suggest an isotropic property for the turbulence structure of the exiting jet, which is known to be non-isotropic. This isotropic behavior implies the same turbulent velocities in the lateral Z and vertical Y directions and thereby causes underestimation of the turbulence exchange in the lateral direction, see also Baldauf and Scheurlen [35].

Figure 22 also shows underestimation of the jet entrainment and a much slower

Figure 22: Normalized AFE distribution for fan-shaped hole, cooling row #3, in the absence of showerhead cooling from experiments to the left and from CFD calculations using Spalart-Allmaras turbulence model to the right. In each case, $M = 0.6$, $M = 0.9$ and $M = 1.2$ are presented from top to bottom, respectively. $S/D = 0$ indicates the cooling hole center of this row.

decay in jet momentum for the CFD results compared to the experiments. Such behaviors may be responsible for high level of over-prediction of AFE when flow separation occurs (see Fig. 7 in Paper IV). Finally, it is discussed in the paper that the turbulence model successfully simulates the vortical structures and vortex dynamics due to coolant injection such as Downstream Spiral Separation Node (DSSN) and Counter Rotating Vortex Pair (CRVP).

Despite the weaknesses of the turbulence model, the laterally averaged data were in good agreement with experimental data as long as no severe lift-off was present. Since the main interest of film cooling design is laterally averaged AFE, the performance of the turbulence model was considered to be satisfactory for the purpose of this study while taking its limitations into account.

From spatially averaged AFE data, it was concluded in Paper IV, that the performance of row #1 is better compared to row #3 for $M < 1.0$ for fan-shaped holes and $M < 0.6$ for cylindrical holes. This was believed to be due to higher acceleration for the row with improved performance, which is located in the lower convex curvature region.

Nevertheless, from Fig. 23, where laterally averaged AFE results from experiments are presented, it can be clearly visualized that for fan-shaped holes with $M = 0.6$, AFE is higher for row #1 compared to row #3. For $M = 1.2$ the figure shows much higher AFE for the latter row case. The findings of the paper using CFD calculations can be therefore confirmed with conclusions drawn for experimental data.

It is worth mentioning that Winka et al. [74] have reported that, consistent with the results in [62] and [75] the AFE for the row of holes located in the strong

convex curvature region is higher than the low curvature position for cylindrical holes and when the momentum flux ratio is less than unity. The obtained results here confines with these studies when cylindrical holes with blowing ratios higher than 0.6 are considered.

Figure 23: Experimental comparison of the normalized laterally averaged AFE between rows #1 and #3 for the interval $0 < S/D < 120$.

For cylindrical holes Fig. 23 shows that for the first $60D$ row #3 has better performance than row #1 for $M = 0.6$ and thereafter they are both confined to the same level of $\bar{\eta}$, whereas CFD simulations have suggested almost identical spatially averaged results for the same case. This disagreement is correlated to the fact that moderate flow separation is experienced by both cooling rows for this range of M , shown by means of both experiments and CFD. As earlier mentioned when jet lift-off appears the validity of CFD results is questionable and special attention should be paid to such circumstances when drawing conclusions. This holds even if the predictions by CFD seem to match the experiments. For instance, Fig. 23 shows slightly higher AFE for row #1 compared to the other cooling row when $M = 1.2$, which is consistent with the CFD results (denoted in Fig. 10 in Paper IV). However, this may not imply appropriate physics and the outcome of the CFD should be viewed with caution as the validation is limited to non-lift-off film cooling injection.

3.4 Correlation Development

Different correlation models have been proposed by a number of researchers, e.g. Bunker [45], Baldauf et al. [44] and Colban et al. [46]. This work has taken the initiatives from earlier correlations and in a progressive procedure modifications and improvements, in the frame of the proposed method, are introduced in order to obtain a generalized correlation model. Within all of these progression stages, the proposed method presented in Sec. 2.3.3 has been followed. This not only

showed the applicability of the method but also provided the opportunity to find the strengths and/or weaknesses of the new models in every stage.

First, progression is achieved by means of testing the already existing correlation models, if applicable, and then small modifications are introduced into the model to improve the model's predictability. It was found that by adding one extra term in one of the original correlations proposed by Bunker [45], model predictability can be greatly improved. This term changed the model from linear to nonlinear form, which could reflect the variability of the parameter more accurately, see Nadali et al. [76].

Further improvement in model predictability in a second progression stage is established by extending the conclusions from the earlier stage and then making any necessary changes to the model's variables. It was found that compared to the original form of the correlations developed in open literature, if each influencing variable is treated independently the correlation model's predictability will improve considerably. In such a case, the interaction between the variables should be accounted for by introducing a third parameter, which is the combination of two interacting variables. This in turn will add to the model's complexity, see Nadali et al. [63]. During these two stages, comparison between derived correlation models and the existing models clearly showed the improvement in model predictability.

Extension of the previous stage with the objectives of adding the effect of hole position and further evaluating the method was accomplished in the third progression stage. This gave the possibility to examine the applicability of the developed method even further and for practical use. For this purpose, two already existing variables used in the second stage, related to downstream distance and blowing ratio, along with the local pressure coefficient as a new variable were put into a general correlation formula given as

$$\bar{\eta} = \left[\beta_0 + \sum_{i=1}^3 X_i \left(\sum_{j=i}^3 \beta_{ij} X_j + \beta_i \right) \right]^{1/\gamma} \quad (42)$$

where the definitions of the variables can be found in Sec. 2.3.3. Expansion of the summation in Eq. 42 illustrates the two-way interaction between different variables. Note that this does not imply that there should be a two-way interaction, but only gives an indication of its possibility for consideration. In the statistical phase, it will be examined whether there are any interactions between the variables or not.

The results of predictions using the formulated correlation model are presented in Paper V. The paper clearly shows the high predictability of the derived model in two different ways. First, the quantification of the model's predictability is demonstrated based on the measured adjusted coefficient of determination, R_{adj}^2

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and a given table shows that the corresponding value for this quantity can be as high as 95% in the case of suction side cooling. This value is expected to be rather high for such a complex model and thus signifies the reliability of the model for design purposes.

In addition, plots of laterally averaged AFE predicted from the correlation model are compared to the experimental data for different examined cases for different blowing ratios. The results give the impression that qualitatively speaking the model also has good predictability and can accurately indicate both lift-off and non-lift-off scenarios. Since the model has been tested for number of configurations, it can be concluded that the established method has satisfactory reliability.

Chapter 4

Concluding Remarks

A summary of the findings from the presented results are presented in this chapter. The relevant outlooks based on the outcome of the work are also suggested.

4.1 Conclusions

The presence of showerhead cooling has major contribution in determining the overall film cooling performance, i.e. net heat flux reduction, of fan-shaped holes regardless of the hole position. The position of the cooling holes on the suction side has a large impact in both adiabatic film effectiveness and heat transfer coefficient, and it was found that the from overall film cooling performance point of view the cooling holes close to the maximum throat velocity is more beneficial since positive NHFR is obtained. However, the cooling row at high convex surface curvature can also maintain positive NHFR directly downstream of the cooling hole exit and further if fan-shaped holes with moderate blowing ratio is used.

Furthermore, it can be concluded that optimum film performance may not be achieved if only the performance of an individual row of cooling holes including the local position is considered. According to the findings, improving the suction side cooling performance is therefore correlated with the presence/absence of showerhead cooling as well as the number of rows to be used. Furthermore, it is shown that for investigation of the optimum cooling design the combined effect of AFE and HTC augmentation, i.e. net heat flux reduction, should be taken into account. An important note to be mentioned here is that the presence or absence of showerhead cooling may make a big difference in terms of manufacturing costs, aerodynamic losses and other design constraints. Therefore, its replacement with rows of cooling holes further downstream of the leading edge may be beneficial. Alternatively, one may consider improving the performance of the showerhead

cooling.

Moreover, the findings suggest that to maintain a low level of HTC augmentation close to the trailing edge on the pressure side, either the showerhead cooling along with another row of cooling holes or a row of fan-shaped holes should be used. It has been shown that an additional row or rows of cooling holes may not necessarily increase the level of HTC augmentation. Under certain circumstances, e.g. using multiple rows of fan-shaped holes with low blowing ratio, the HTC will therefore remain relatively low whereas AFE will increase. This will result in improved overall film cooling performance considering the effect of both AFE and HTC.

In addition, it was shown that the trend predictions obtained from CFD results can be reliable if caution is taken in model validation. As an example, in this study the effect of blowing ratio, cooling hole shape and flow acceleration on film cooling performance indicated by CFD results have been validated later by means of experiments. This suggests that despite the fact that deficiencies may be encountered in RANS simulations such as the isotropic turbulence structure assumption, which is not the case with film cooling jets, parametric studies can nevertheless be made, perhaps with a certain degree of limitation.

Finally, the correlation method that has been developed has shown a great potential to be used for deriving correlation models with different levels of complexity. The ultimate derived correlation model has not only proven the applicability and reliability of the method, but it has also shown high level of predictability when it has been examined for different configurations such as cylindrical holes, fan-shaped holes and in the presence or absence of showerhead cooling.

4.2 Outlook

This study has addressed important aspects of film cooling for a turbine guide vane under engine representative conditions. Although some other important aspects such as aerodynamic losses due to cooling, design constraints, streamline curvature etc. have not been in the scope of the work, based on the investigations conducted here, a few of the most relevant outlooks are presented.

The effect of showerhead cooling on the performance of suction and the pressure side cooling has been investigated quite extensively in this study, although the performance of the showerhead cooling in itself with respect to variations in blowing ratio, cooling hole shape, cooling hole angle, etc. has not been considered. Even though it is shown here that there are circumstances where improvements in the cooling performance may be achieved in the absence of showerhead cooling, an efficient showerhead cooling may also indicate that there is no need for any further suction or pressure side cooling or perhaps fewer cooling holes

may be required.

Further and indeed extensive use of CFD for parametric studies and/or trend predictions can be beneficial, since in general CFD simulations are less expensive. In this context, however, the importance of the turbulence model's validation and its limitations should not be overlooked as it has a high impact on the outcome. Such parametric studies may also serve as inputs for deriving correlations if necessary.

Finally, the applicability and practical limitations, if there should be any, of the correlation method that has been developed here and the final derived model require further extension and the inclusion of other important parameters.

Chapter 5

Review of Appended Papers

Paper I

Film Cooling Performance of a Turbine Vane Suction Side: The Showerhead Effect on Film Cooling Hole Placement for Cylindrical and Fan-Shaped Holes, Hossein Nadali Najafabadi, Matts Karlsson, Esa Utriainen, Mats Kinell, *Journal of Turbomachinery*, Vol. 137, 091005, 2015.

This paper investigates the importance of showerhead cooling for the suction side cooling when it is combined with a cooling row at high convex surface curvature or with a cooling row close to maximum throat velocity. It has been found that the performance of the cooling rows at these two positions are influenced to a different degree in the presence of showerhead cooling. The degree of impact has been shown to differ also with respect to what parameter is of interest, i.e. adiabatic film effectiveness, heat transfer coefficient or net heat flux reduction.

Author Contributions: Nadali did the experiments, the data post-processing, and most of the writing. Karlsson, Utriainen and Kinell have guided through the work and provided comments and discussions. Kinell has also contributed in writing the experimental apparatus section.

Paper II

Film Cooling Performance of Multiple Arrays of Cylindrical and Fan-Shaped Holes, Hossein Nadali Najafabadi, Matts Karlsson, Esa Utriainen, Mats Kinell, Lieke Wang, *Submitted for Publication*, March 2015.

The paper compares the film cooling performance of double row and to some extent single row cooling of the fan-shaped and the cylindrical holes in the presence and absence of showerhead cooling. The aim was to study the possibility of using a certain configuration which may have similar performance compared to

other configuration with different properties. The paper suggests that finding an optimum film cooling performance should be looked upon based on the number of cooling rows to be involved, the presence or absence of showerhead cooling, the variable to be looked, i.e. laterally averaged adiabatic film effectiveness, heat transfer coefficient or NHFR. These are in addition to other influencing parameters such as the blowing ratio, the hole shape, the state of approaching flow and etc.

Author Contributions: The experiments, post-processing of the data and most of the writing are done by Nadali. Karlsson, Kinell, Utriainen and Wang have contributed by providing comments and discussions through the work. Kinell have assisted also in writing the experimental apparatus.

Paper III

Film Cooling Jet Injection Effect in Heat Transfer Coefficient Augmentation for the Pressure Side Cooling of Turbine Vane, Hossein Nadali Najafabadi, Matts Karlsson, Esa Utriainen, Mats Kinell, *Proc. of ASME Turbo Expo 2014*, GT2014-26055, 16-20 June 2014, Düsseldorf, Germany.

Heat transfer coefficient augmentation due to injection of single and multiple row of cooling holes have been investigated for the pressure side. The impact of showerhead cooling on this context has also been studied. According to findings of the paper, showerhead only cooling or a single row cooling on the pressure side causes substantial HTC augmentation, while the combination of the two, i.e. showerhead plus an additional cooling row further downstream, can lead to a much lower level of HTC augmentation. The negligible differences between cylindrical and fan-shaped holes on this regard has also been shown.

Author Contributions: Utriainen initiated the work. The experiments, post-processing of the data and most of the writing was done by Nadali. Kinell has assisted in writing the experimental apparatus section. Karlsson and Utriainen have contributed in the discussion and provided comments on the writing.

Paper IV

CFD Based Sensitivity Analysis of Influencing Flow Parameters for Cylindrical and Shaped Holes in a Gas Turbine Vane, Hossein Nadali Najafabadi, Matts Karlsson, Esa Utriainen, Mats Kinell, *Proc. of ASME Turbo Expo 2012*, GT2012-69023, 11-15 June 2012, Bella Center, Copenhagen, Denmark.

CFD simulations are performed on number of configurations to research on the effect of hole shape, blowing ratio and hole position, i.e. approaching flow

and curvature, in film cooling performance. First the CFD code validation and its limitations are addressed. Then the investigations showed that fan-shaped holes may have better film cooling performance if subjected to higher acceleration. The paper also indicates that due to low acceleration rate on the pressure side the film cooling performance is not influenced much by its position.

Author Contributions: Experimental part is performed by Kinell. Nadali has done the CFD simulations, the post-processing and most of the writing. Karlsson and Utriainen have assisted and guided through the work.

Paper V

Film Effectiveness Correlations for Cylindrical and Fan-Shaped Holes, Introducing Local Pressure Coefficient, Hossein Nadali Najafabadi, Matts Karlsson, Esa Utriainen, Mats Kinell, *Proc. of ASME Turbo Expo 2012*, GT2012-69021, 11-15 June 2012, Bella Center, Copenhagen, Denmark.

This paper presents the possibility of the developed method for deriving correlation which can take into account the effect of approaching flow by introducing the local pressure coefficient as a new parameter. Developed model correlates the experimental data, either for cylindrical or for fan-shaped holes, by means of three main variables as downstream distance, blowing ratio and local C_p and the two way interaction between these variables. The correlation model predictability has been proven to be very good both quantitatively which is measured by means of adjusted coefficient of determination (R_{adj}^2) and qualitatively when the predictions are compared with experimental data.

Author Contributions: Kinell did all the experiments. Correlation development, statistical analysis and most of the writing is done by Nadali. Karlsson and Utriainen have provided suggestions and discussions through the work.

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