NUMERICAL INVESTIGATION OF THE INFLUENCE OF SPECIFIC VORTEX GENERATION ON THE MIXING PROCESS OF FILM COOLING JETS

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ABSTRACT

Vortices generated and designed in internal cooling systems of film cooled turbine blades have a significant influence on the aerodynamic mixing process of the coolant and the main flow. Of course the resulting distribution of the film cooling effectiveness is strongly influenced by the behaviour of the coolant mixing. The vortices are generated by a special design of the cooling channel and film cooling holes and by the comparable flow phenomena coming along with the problem of jets in crossflow.

The multi-block Navier-Stokes solver TRACES with special features for the simulation of complex and turbulent mixing phenomena is applied to solve the flow field and the heat fluxes and temperature distribution in the turbine blade structure simultaneously. The code is validated for the simulation of film cooling and heat transfer and was also compared to other codes taking part in the AGARD Working group 26 (Dunham, 1996 [4]) working on the test cases “NASA-Rotor-37” and DLR-Turbine guide vane “VT1B”.

As a result of extensive parameter studies (Vogel, 1996 [18]) a vortex mixing model will be established which describes the existing vortices and their effect on the film cooling mixing process in detail. It was found that vortex generation inside the internal cooling channel with an opposite direction of rotation compared with the main kidney shaped jet vortex system can reduce the amount of coolant air significantly without reducing the film cooling effectiveness. All numerical investigations presented in this paper were carried out using a model geometry. No experimental results were available for this geometry. The validation of the code concerning film cooling mixing was presented in [15], [16], [17] and [18] (Vogel, 1994-1996). The geometries applied in those validations were quite similar to those used in this paper. Nevertheless for credibility some comparisons between experimental and numerical data are presented.

LIST OF SYMBOLS

\[ M_{\text{blow}} = \frac{\rho U}{\rho_\infty U_\infty} \] local blowing ratio
\[ I_{\text{blow}} = \frac{\rho U^2}{\rho_\infty U_\infty^2} \] local momentum ratio
\[ \alpha \] ejection angle (s. Fig. 1)
\[ \beta \] inclination angle of blowing hole (s. Fig)
\[ d \] blowing hole exit diameter
\[ t \] blowing hole pitch
\[ l \] length of blowing hole
\[ \eta_{\text{ad}} = \frac{T_{\text{ad}} - T_0}{T_\infty - T_0} \] adiabatic film cooling efficiency
\[ \eta_{\text{av}} \] averaged adiabatic film cooling efficiency
\[ p_t \] Total pressure
\[ x \] distance from hole exit center in downstream direction
\[ z \] surface normal distance
\[ Q_x = \frac{\partial x}{\partial x} - \frac{\partial z}{\partial y} \] streamwise vorticity component
\[ \rho \] density
\[ U \] amount of velocity

Subscripts:
\[ ad \] adiabat
\[ av \] averaged

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INTRODUCTION

The efficiency of gas turbines is increased by increasing temperatures in the combustion chamber. Nowadays the turbine inlet temperature is much higher than the allowed temperature for modern turbine material. Therefore the turbine blade and casings have to be protected from the hot passage flow by active cooling. Film cooling has the advantage to protect the blade at the location of the ejection and further downstream. The stability of the coolant film strongly depends on the mixing with the main flow, which is influenced by several parameters; e.g. blowing rate, geometry of internal cooling system and blowing hole, upstream type of boundary layer, Reynolds number, turbulence intensity, etc.. Modern gas turbines use a considerable amount of cooling air, in some cases more than 25% of the core engine mass flow, which will decrease the efficiency and lead to highly complex cooling systems. Therefore it is important to reduce the amount of cooling air without risking turbine damage due to overheating. The reduction of the amount of cooling air is only possible if the behaviour of the coolant film and its mixing with the hot main flow is known correctly.

The interaction of the mixing process is carried out numerically using the flow solver TRACE$S$ which is validated concerning turbine flows with film cooling and heat transfer in detail (Langowsky & Vogel (1996) [11], (1997) [12], Heselhaus & Vogel (1992) [9], (1995) [10], Vogel (1993-1996) [14], [15], [16], [17], [18]).

It is well known that the blowing hole geometry plays an important role for the coolant mixing with the hot turbine flow. Inclined fan shape holes are state of the art. Deformed exit areas of the holes are also considered. The mechanism of the mixing and the interaction of the generated vortex systems and their influence of the film cooling efficiency are nevertheless not clearly known. Some experiments and also numerical investigations were carried out to visualize and understand the typical flow characteristics in the case of film cooling ejection, Frick & Roshko (1989) [2], Bardina et. al. (1985) [3], Leylek (1994) [5], Lopez (1993) [6], Margason (1993) [7]. Leboeuf (1994) [13] first tried to classify and characterize the main vortices, their generation, their interaction with each other and their influence on the behaviour of the whole cooling film.

The goal of this paper is the investigation of vortex systems generated within the blade interior and their interaction with the main and known vortex systems. The major goal is to save coolant air without any reduction of the film cooling efficiency.

THE DLR-CODE TRACE$S$

TRACE$S$ is part of the whole system TRACE which furthermore consists of TRACE$U$ 2. TRACE$S$ was originally developed at DLR by D.T. Vogel (1993) e.g. [14] and since 1994 the code is applied and developed in joint-teams together with MTU (Broichhausen, 1997) [1]. TRACE$S$ is completely written in FORTRAN90, vectorized and shared memory parallelized. MPI2 allows the distributed memory parallelization of FORTRAN90 codes in the near future. The code is basically equipped with an explicit Jameson-solver on structured multi-block meshes using several two-equation turbulence models and multistage feature. Furthermore several implicit strategies combined with various preconditioning methods can be applied. TRACE$S$ is able to simultaneously simulate the aerothermic coupling of cooled turbine flow and heat conduction in the turbine blade and platform structure.

VALIDATION FOR FILM COOLING MIXING

As mentioned before TRACE$S$ was validated for film cooling. For credibility some of these results are presented in this paper. The focus of these comparisons between experimental (Wilfert, 1994 [21] and numerical data was the correct qualitative description of the phenomena and its quantitative representation. The most important mixing vortices, their sources and development should be simulated correctly. Fig. 3 (Vogel, 1996 [19]) shows a comparison of measured and predicted data for the streamwise vorticity and turbulence intensity in the mixing region of a film-cooled subsonic turbine blade. The blowing ratio was 2.0 and the row of blowing holes ($t/d = 2.5$, $l/d = 1.4$) were located at 40% axial chord length on the suction side of the blade. Fig. 3 presents two different cross-flow sections located at $x/d = 5$ and $x/d = 10$. The resulting measured and predicted streak lines on the blade surface are presented in Fig. 4.

Fig 5 shows a measured vorticity pattern at the blowing hole exit plotted in jet orthogonal direction. This figure

1TRACE$S$ is the DLR flow solver for steady turbomachinery flow problems

2TRACE$U$ simulates unsteady blade row interactions
shows the most important vortex structure which are described later on.

INVESTIGATION OF THE GEOMETRY INFLUENCE

The geometry influence of the cooling channel and blowing hole is investigated using the prescribed Navier-Stokes-solver. In all parameter cases the $k-\omega$-model of Wilcox (1991) [20] was applied with some modifications concerning the influence of the compressibility and near-wall treatment, Haidinger (1992) [8].

Table 1 presents the actual parameter variations.

Fig. 2 schematically illustrates the applied geometries described in Table 1. The ejection angle $\alpha$ in all cases is 45 deg and the pitch-to-diameter ratio $(t/d)$ is 2.5. Case 1a/1b represents an "ordinary" cylindrical hole, whereas case 2a/2b reproduces a diffusor-shaped hole (not exactly a fan-shaped hole; the ratio of exit to inlet hole diameter is 1.428) and 3a/3b shows a cylindrical hole with a bow at the inlet region turning the flow. The turning of the flow in a bow generates a kidney-shaped vortex pair due to the secondary flow which affects the overall mixing process. This will be explained in detail later on.

Looking at Table 1 the following facts can be pointed out:

- The blowing ratio is around 0.93 for $l/d = 1.5$ and $\beta = 0^\circ$ (Case 1a and 3a), the momentum ratio is around 0.8.
- The diffusor hole reduces the blowing ratio by 25% and the momentum ratio by 45%. The reason is the constant outlet area and the reduced smallest area of the blowing hole compared with case 1a/1b.
- The holes with $l/d = 3.5$ (Cases "b") reduce the blowing and momentum ratio due to stronger pressure losses inside the blowing holes. The total pressure inside the coolant channel is the same as in cases "a".
- The coolant total pressure $p_{tc}$ is reduced for the bow shaped ejection holes to gain a nearly constant blowing rate.

The density ratio of coolant to free stream is approximately 1.35. Fig. 6 shows a view of a typically 3D-mesh consisting of 457781 nodes, whereas in the case of symmetry (for non-inclined main flow) 240636 nodes are sufficient.

Fig. 7 illustrates the resulting area-averaged adiabatic film cooling effectiveness $\eta_{av}$ downstream of the ejection holes as a function of $x/d$ (the non-dimensional downstream distance from the hole exit center) for all parameter variations of table 1. The result of the reduction of the momentum ratio for the diffusor-shaped hole configuration (case 2a/2b) is clearly visible in the near hole ejection region. Even far downstream $(x/d > 10)$ the efficiency of this hole geometry is better than for the cylindrical hole configuration (case 1a/1b). As expected cases "b" $(d/l = 3.5)$ show slightly better efficiency due to the reduced momentum and blowing rate for cases 1 and 2. Case 4a/4b presents the best results which is due to the inclined ejection of the coolant relative to the main flow.

A very interesting result is shown for the cases 3a/3b. The efficiency shows very good results in comparison to cases 1 and 2. An optimum is found for $x/d = 5$ for case "a" $(l/d = 1.5)$. Case "b" $(l/d = 3.5)$ nevertheless presents efficiencies which are below those for case "a" which is opposite to cases 1 and 2. The reason for those good efficiencies for cases 3 and 4 will be explained in the following sections:

Mixing process for case 3a/3b:

For case 1a Fig. 8 presents the distribution of the adiabatic film cooling efficiency and the streamwise component of the vorticity $(\Omega_z)$ which indicates the main mixing vortices, the well known kidney-shaped vortex pair ($\Omega_2$-Structure, [18]). The presented quasi-orthogonal cross flow section is located 3.8 hole diameters downstream of the ejection hole exit center. In comparison Fig. 9 shows the same cross flow section for case 3a, whereas Fig. 10 illustrates the downstream development of the main vortices of the mixing process for case 3a. Fig. 9 shows a better lateral entrainment of coolant at the wall surface.
and coolant jets which do not entrain as much as in case 1a into the main hot gas flow normal to the wall surface.

The reason for the better and equalized distribution of the coolant film for the bow-shaped (case 3a/3b) blowing hole is visible when looking at the evolution and deveoptiment of the streamwise vorticity component: Fig. 10 shows the streamwise vorticity component at three different locations immediately downstream of the ejection hole \((x/d = 0.25, 0.5, 1.0)\). In addition to the \(\Omega_2\)-Structure a so-called \(\Omega_5\)-Structure is ejected out of the hole and is embedded inside of the main \(\Omega_2\)-Structure. The \(\Omega_5\)-Structure is generated by the bending of the flow in the bow in front of the ejection hole and rotates in opposite direction compared with the \(\Omega_2\)-Structure. As clearly demonstrated in Fig. 10 the weaker \(\Omega_4\)-Structure is transported by the \(\Omega_2\)-Structure from the jet centerline towards the outer jet border and dissipates there. The interesting fact is that this convection process also transports coolant from the cold jet center towards the outer region of the cooled surface (between the neighboring jets), which leads to a much better lateral entrainment of coolant. Furthermore the vortex transportation process leads to a rather flat coolant film and reduced coolant entrainment in surface-orthogonal direction as mentioned before.

The advantage of this process is also valid for coolant holes with larger \(l/d\) ratio as shown in Fig. 11 and 12. The \(\Omega_5\)-Structure is weaker at the hole exit but strong enough to affect the \(\Omega_2\)-Structure again.

The interaction of the \(\Omega_5\)-Structure (generated inside the blowing hole) and the \(\Omega_2\)-Structure (generated outside the blowing hole) has a very positive influence on the coolant film mixing as pointed out and presented in Fig. 7. This more academical investigation should demonstrate the capability and impact of specific vortex generation and interaction on the mixing process and its influence on the adiabatic film cooling efficiency. These investigations show that it should be possible to save coolant especially for higher blowing rates.

Mixing process for case 4a/4b:

Fig. 13 shows the coolant flow behaviour on the cooled surface indicated by the distribution of the film cooling efficiency for case 4a. Fig. 14 shows in comparison the same plot for case 1a. As demonstrated in Fig. 7 case 4a/4b shows a much better film cooling efficiency as case 1a/1b which is due to the enforced entrainment of coolant in lateral direction to the coolant jets. Fig. 15 presents the mixing process again in a cross flow section located at \(x/d = 3.8\) for case 4a. Compared with Fig. 8 the mixing and convection of coolant is clearly demonstrated and the advantages of this ejection configuration are visible.

**VORTEX MIXING MODEL**

Due to the results obtained by the above parametric investigation and literature study a schematic vortex mixing model which classifies the most important vortices can be installed: Fig. 16 illustrates this model and names the dominant vortices.

1. The \(\Omega_1\)-Structure represents ring-vortices at the free shear layer between jet and main flow.

2. The \(\Omega_2\)-Structure was already explained and shows the main kidney-vortex in the jet, whereas the

3. \(\Omega_3\)-Structure indicates the horse-shoe vortex generated upstream of the ejected coolant jet quite similar to the horse-shoe vortex in front of blade leading edges.

4. The \(\Omega_4\)-Structure illustrates van-Karman-like unsteady vortices downstream of the ejected jet.

5. The \(\Omega_5\)-Structure is also generated inside the coolant channel just in the inlet region of the ejection hole by the suc-in process. \(\Omega_5\) has the same direction of rotation as \(\Omega_2\) and combines with this vortex pair outside of the ejection hole.

6. \(\Omega_6\) was already explained and represents an artificially generated vortex pair which counter-rotates against \(\Omega_2\).

Leboeuf, 1994 \([13]\) and Vogel, 1996 \([18]\) explain the source of those vortices in more detail.

For inclined main flow relative to the ejected flow only one part of all vortex pairs is dominating. A very interesting effect was found when looking at Fig. 17 which shows the influence of inclined main flow on the vortex structure inside an ejection hole. Fig. 17a qualitatively shows the numerical results and Fig. 17b demonstrates the experimental picture obtained by the laser-light-sheet method. Already inside the ejection hole the deformed kidney-shaped vortex pair invokes a deformed vortex pair with one strong and one weak part.

**CONCLUSIONS**

The numerical parametric investigation of the influence of the hole and coolant channel influence on the adiabatic film cooling efficiency shows a very interesting effect: It is possible to generate a vortex pair inside the coolant channel (\(\Omega_6\)-Structure) which counter-rotates against the main kidney-shaped vortex structure of the coolant jets. This
vortex structure is transported from the jet centerline towards the lateral jet border. The transportation process supplies coolant out of the jet center onto the cooled surface and therefore generates a flat coolant film with forced lateral coolant entrainment even for non-inclined ejection.

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References

\( \alpha \): Angle between blowing vector and its projection on the wall surface
\( \beta \): Angle between projection of blowing vector on the wall surface and main flow direction

Figure 1: Definition of blowing angles \( \alpha \) and \( \beta \)

Figure 2: Schematic description of geometries of Table 1
Vorticity, $x/d = 5$

Turbulence intensity, $x/d = 5$

Vorticity, $x/d = 10$

Turbulence intensity, $x/d = 10$

Figure 3: Predicted and measured turbulence intensity and vorticity, ejection at 40%

Measured, $x/l = 72%$

Predicted

Figure 4: Measured and predicted streak lines

Figure 5: Measured vorticity pattern near the exit of the blowing hole
Figure 6: Surface mesh of the reference film cooling configuration (case 1 a/b)

Figure 7: Area-averaged distribution of the adiabatic film cooling efficiency for all cases defined in Table 1
a) Film Cooling efficiency at $x/d = 3.8$, Case 1a

b) Streamwise vorticity distribution at $x/d = 3.8$, Case 1a

Figure 8: Film Cooling efficiency and Streamwise vorticity distribution at $x/d = 3.8$, Case 1a

a) Film Cooling efficiency at $x/d = 3.8$, Case 3a

b) Streamwise vorticity distribution at $x/d = 3.8$, Case 3a

Figure 9: Film Cooling efficiency and Streamwise vorticity distribution at $x/d = 3.8$, Case 3a
Figure 10: Development of the streamwise vorticity in downstream direction, Case 3a

a) Film Cooling efficiency at $x/d = 3.8$, Case 3b

b) Streamwise vorticity distribution at $x/d = 3.8$, Case 3b

Figure 11: Film Cooling efficiency and Streamwise vorticity distribution at $x/d = 3.8$, Case 3b

Figure 12: Development of the streamwise vorticity in downstream direction, Case 3b
Figure 13: Development of the film cooling efficiency on the cooled surface, Case 4a

Figure 14: Development of the film cooling efficiency on the cooled surface, Case 1a/1b
Figure 15: Film Cooling efficiency and Streamwise vorticity distribution at $x/d = 3.8$, Case 4a
Figure 16: Schematic vortex model

Figure 17: Vortex development inside the blowing hole (Case 1a), Experiments by C. Langowsky (1997)