CFD BASED SENSITIVITY STUDY OF FLOW PARAMETERS
FOR ENGINE LIKE FILM COOLING CONDITIONS

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ABSTRACT
A standard CFD code with two-layer k-ε-model was used to calculate film cooling effectiveness of flat plate test cases. Experimental data from the literature were taken to perform extensive validation of the code for film cooling effectiveness prediction. Emphasis was put on injection of cooling gas through one row of cylindrical holes in the streamwise direction. Blowing ratio, density ratio, blowing angle, pitch, and hole length to diameter ratio were varied in a wide range. It was found that the code is well suited for the prediction of lateral averaged film cooling effectiveness for common film cooling conditions. A similarity analysis is presented for the prescribed film cooling problem to isolate the influence parameters of flow properties and geometry. A reduction of the parameters of influence was achieved using physical implications. The magnitude of the remaining parameters was compared for literature reported experimental results and gas turbine applications. It was found that experimental realized Reynolds and Eckert numbers are mostly far from turbine engine conditions. Therefore the validated CFD code was used to extrapolate the experimental configuration to engine like conditions. It was found that the examined Reynolds and Eckert numbers had no significant impact on lateral averaged film cooling effectiveness. It is hence possible to present a reduced but complete set of the governing influence parameters on the discussed film cooling problem.

NOMENCLATURE
a ejection angle
cp heat capacity at constant pressure
D ejection hole diameter
di boundary layer displacement thickness
Ec Eckert number Ec = \frac{\rho u^2}{c_p T}
Eu Euler number Eu = \frac{\rho}{\rho u^2}
h heat transfer coefficient
\eta film cooling effectiveness
\bar{\eta} lateral averaged film cooling effectiveness
I hot gas to coolant momentum flux ratio I = \frac{(\rho u^2)_{G}}{(\rho u^2)_{C}}
\kappa isentropic exponent
L ejection hole length
\lambda thermal conductivity
M blowing rate M = \frac{(\rho u)_{C}}{(\rho u)_{G}}
Ma Mach number Ma = \frac{u}{\sqrt{\kappa R T}}
\mu dynamic viscosity
\nu kinematic viscosity
Pr Prandtl number Pr = \frac{\mu c_p}{\lambda}
p pressure
q wall heat flux
R specific gas constant
Re Reynolds number Re = \frac{u D}{\nu}
\rho density
s hole pitch
T temperature
Tu turbulence level Tu = \frac{\sqrt{u'^2}}{u}
\theta dimensionless temperature \theta = \frac{T_0 - T}{T_0 - T_C}
u velocity
x downstream distance from ejection hole
z lateral distance from ejection hole centerline

Subscripts
AW adiabatic wall
C coolant
D hole diameter based
G hot gas
rec recovery temperature
W wall
0 without ejection

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INTRODUCTION

The classic way of handling the film cooling heat flux problem is to split it into a temperature and a heat transfer part to be independent of each other. The temperature problem is formulated in terms of a film cooling effectiveness

\[ \eta = \frac{T_{AW} - T_G}{T_C - T_G} \]  

(1)

If the Mach number reaches values giving a significant temperature effect in the boundary layer, effectiveness has to be expressed in terms of recovery temperatures \( T_{AW} \) and \( T_{Ceff} \). Heat flux is determined as

\[ q = h (T_{AW} - T_W) \]  

(2)

with \( h \) being the heat transfer coefficient of the film cooled surface. This heat transfer coefficient is significantly different from the heat transfer coefficient without coolant ejection:

\[ h \neq h_0 \]  

(3)

Determination of the heat transfer coefficient \( h \) of the unblown, but internal cooled surface itself is a highly complex problem [19]. One often has to accept, that research on film cooling behavior ends after having just performed the derivation of a cooling effectiveness in dependency of some geometrical and fluid dynamical parameters. Then the heat transfer coefficient is assumed to be equal to the case without coolant ejection, especially at some distance from the ejection location. Total heat flux is then calculated from these data and correlated film cooling effectiveness. Especially for locations in the vicinity of the ejection, the difference of \( h \) and \( h_0 \) usually reaches its maximum. However, at this location the designer is mostly interested in correct results of heat flux prediction, because ejection naturally takes place close to the point where cooling is of predominant importance. Considering configurations of full coverage film cooling, there are no longer locations, that can be assumed to be unaffected by coolant ejection [7]. Keeping in mind these facts, this study focuses on the effectiveness problem as a part of the heat flux prediction and a preparatory work on the general problem of heat transfer in film cooling.

INTENTION OF THE STUDY

Despite its widely spread use, the over decades ongoing and still actual research shows, the fluid dynamics of film cooling are still not fully understood. The reason of this lack of knowledge on the real behavior of film cooling is the large number of parameters influencing this flow problem. Having a closer look on different publications concerning the single row flat plate configuration, it is found, that various parameters of geometry and fluid dynamics are pointed out. Emphasis is put on different variables expected to be the governing ones, and the list of included parameters seldom is found to be the same in two reports. The authors were not able to find a collection that seemed to be complete, nor was there any statement on an existing full set as a base for an approach to the problem. Therefore the authors want to present the derivation of such a closed set of parameters and possible further reductions using computational fluid dynamics as a tool.

To obtain reliable results from the CFD study, the used code has to be validated. The authors do not want this validation to be seen as a presentation of a new method on the film cooling problem. It is carried out to make an all purpose commercial code available as a tool described above.

Film cooling on turbine blades is in all practically used configurations a fully three dimensional problem. In addition it does have significant impact on the oncoming boundary layer, transition, and recirculation behavior. Numerical prediction can only be successful, if the complex vortex structure of the ejection situation can be resolved with the used grid. A highly detailed geometrical model of the flow domain has to be used. The coolant supply has to be modeled carefully to pull the difficult inlet condition as far from the interesting interaction zone as possible. This is the only way to assure a realistic ejection vector profile, that is affected by parameters as hole geometry, streamline curvature into hole inlet, hole length, or internal coolant flow history. These parameters result in vortex and velocity vector profiles of the ejected coolant, which have a significant impact on the mixing process and subsequently on the effectiveness of the cooling film. In addition, the main flow has to be modeled at least in form of boundary layer profiles including velocity, turbulence and, if necessary, temperature. The authors want to point out, that it is not the use of a highly specialized code to result in good predictions, but the careful modeling of the flow situation.

VALIDATION OF THE CODE

The code used in this study was Advanced Scientific Computation's TASCflow [17], version 2.4, a 3-D Navier-Stokes solver with a standard k-ε-turbulence model. It offers full multigrid capability and robust convergence behavior. A two-layer model was used on the cooled wall, with one-equation treatment for the near wall layer, adopting damping functions on the algebraic evaluated mixing length parameter \( e \) and the turbulent viscosity. For other walls a logarithmic law model was applied.

For the verification of the code's capabilities, standard film cooling configurations were taken, featuring one row of cylindrical cooling holes inclined in the streamwise direction and a flat plate surface to be cooled. Literature documented data from flat plate experiments were taken for comparison [4, 10, 13, 14, 18]. The computational setups were adjusted to match these geometrical and fluid dynamical conditions. For the rating of the results, only adiabatic wall temperature, that is film cooling effectiveness, was taken into account. The experiments were typically performed in small low speed wind tunnels with low turbulence. The range of parameters covered by the experiments is shown in Table 1.

Some of the experiments use Freon/Air mixtures or CO₂ as the coolant to produce the density ratios as high as 4.17. We will refer to the fact using foreign gas as coolant and its implications on the physics of film cooling later on. This way to manipulate the density ratio was chosen to avoid high temperature load on the test rigs, thereby being able to simulate exact adiabatic conditions. In contrast to this method, testing with air on different temperatures is accompanied by the problem of generating errors due to heat conduction. Especially in the vicinity of the ejection, high temperature gradients provoke significant heat flux, even in very low conducting materials. Considering the facts listed above, one has to recognize, that all data of "adiabatic" wall temperatures are either corrected for conduction errors by more or less detailed methods, or contain errors that originate from the not correctly

<table>
<thead>
<tr>
<th>M</th>
<th>0.21 — 2.0</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \rho c p_0 )</td>
<td>0.74 — 4.17</td>
</tr>
<tr>
<td>( x/D )</td>
<td>0 — 90</td>
</tr>
<tr>
<td>( s/D )</td>
<td>1.5 — 5</td>
</tr>
<tr>
<td>( L/D )</td>
<td>1.5 — 90°</td>
</tr>
<tr>
<td>( \delta/D )</td>
<td>0.04 — 0.171</td>
</tr>
</tbody>
</table>

\( Re \) = \( 3480 — 51150 \)
\( Ma \) = \( 0.07 — 0.9 \)
\( T_0 \) = \( 273 K — 773 K \)
\( T_{Ceff} \) = \( 0.76 — 1.34 \)
\( Tu \) = \( 0.002 — 0.02 \)

Table 1: Parameter range of validation
A typical grid is presented in Figure 1. It consists of three blocks representing the hot gas channel, the ejection hole and a cubic plenum chamber. In cases of non-cubical plenums, length and height of this block were adjusted to give the best representation of the experiment's plenum. Inlet of the cooling air was set to the side giving the direction of the experiment's coolant supply. The single blocks were implemented as H-grids adapted to the special geometric requirements.

The model always extended from the hole centerline to the midline between the actual and the adjacent hole. Symmetry conditions were applied to the corresponding sides of the domain to get the effect of an infinite row of holes. The lower side of the hot gas channel block was the examined adiabatic wall, the upper side of this block was also set to symmetry conditions, expecting that this gives a good representation of the channel flow of the experiments. The height of the domain was 15 diameters, the total length 75 diameters with 10 diameters from the inlet to the exit hole location. Typically the inlet velocity and turbulence profiles and a constant temperature were set. As various results from calculations with Mach numbers as high as about 0.3 and above showed, the constant inlet temperature could not be held for these cases. Specially refined models which included the development of a boundary layer analogous to the experiment's boundary were used to obtain temperature profiles showing the proper recovery effects to verify and correct these results.

Typical dimensions of the grid are 112 nodes in streamwise, 44 nodes in vertical and 16 nodes in lateral direction. The hole was modeled by 15 to 7 nodes in the cross section and up to 20 nodes in length, the plenum consisted of about 26 to 10 to 12 nodes. This resulted in a total of about 83000 nodes per run taking about 13 hours of CPU time on a HP 735 workstation.

The results of the calculations concerning the flow patterns related to the ejection situation is totally consistent with publications of 3-D calculations of other authors. There are the typical observations of vortex behavior of the ejected jet in streamwise shifted cross sections, vortex development in the hole, jetting effects and turbulence production in the cooling air flow ([11, 12, 15, 16, 21]) (Figures 2 and 3). Also the temperature and velocity profiles in the centerline show the classic behavior of the discrete hole ejection problem ([3, 5, 21, 22]), as do the temperature contours in streamwise shifted cross sections (Figure 6) ([5, 11, 12]). Figure 7 shows temperature contours of typical low and high blowing rates in the centerline and on the adiabatic wall. All temperatures are presented as dimensionless 0 values.

Documented in Figures 6 and 7 and also referring to [2, 3, 12] there is a special deficiency of the standard k-ε-model. This model assumes isotropic turbulence structure. This jet exiting problem is well presented physics. On the other hand, the computational model is free of restrictions from such conduction effects or maximum temperature load. A wall assigned to be adiabatic will exactly represent this physical attribute. In the computation all cases were calculated using air as the only fluid and the given density ratios were adjusted by choosing the corresponding cooling air temperature. Hence, it is very unlikely to predict exactly the same film cooling effectiveness given in the experimental data.
known to be a case of non-isotropic turbulence, i.e. the turbulent velocities $z'$ in the lateral direction are greater than the turbulent velocities $y'$ in the vertical direction by a factor of about two. Resulting from this physically wrong modeled turbulence, the turbulent exchange in the lateral direction is underestimated. Therefore jet spreading appears too weak, pushing coalescence of the single coolant jets to a farther downstream position.

Jet entrainment also is underestimated, forcing the jets of higher momentum ratio farther from the surface. This is due to the fact that the mean jet momentum is not reduced as fast as in the experiment. Figures 8 and 9 show these effects on computed effectiveness data compared to experimental data. In Figure 8 centerline, midline and lateral average effectiveness are plotted against downstream distance. Figure 9 shows plots of effectiveness versus the lateral coordinate. As can be seen, coalescence of the midline and centerline effectiveness is shifted far downstream and the low midline effectiveness is accompanied by too high centerline effectiveness. The result is the match of the lateral average effectiveness except some discrepancies for small downstream distances. By the average effectiveness being the main interest of film cooling design, this result is very satisfying. Analysis of all these test cases extending to the parameter range of Table 1 showed the following result of the validation:

The code performed generally well for all those cases, where the exiting jet showed no significant lift off from the surface. For typical density ratios of the experiments near unity, this includes blowing rates up to about 1.0 for the range of all other parameters. For low density ratios of film heating experiments (e.g. $\rho_c/\rho_0=0.79$), too low effectiveness was given for lower blowing rates, for high density ratios (e.g. $\rho_c/\rho_0=2.0$) good results are obtained up to $M=1.4$ (Figure 10).
Figure 6: Temperature contours in cross sections (dimensionless $\theta$)
$M=0.34$ & 1.43, $\rho_C/\rho_G=0.79$, $\alpha=35^\circ$, $s/D=3$, [13]

Figure 7: Temperature contours (dimensionless $\theta$)
$M=0.34$ & 1.43, $\rho_C/\rho_G=0.79$, $\alpha=35^\circ$, $s/D=3$, [13]

Figure 8: Measured and calculated effectiveness over downstream distance. $M=0.57$, $\rho_C/\rho_G=1.31$, $\alpha=31^\circ$, $s/D=2$, [18]

Figure 9: Measured and calculated effectiveness over lateral coordinate. $M=0.5$, $\rho_C/\rho_G=2$, $\alpha=35^\circ$, $s/D=3$, $x/D=10.3$, [14]
At large ejection angles, where more turbulence is produced by the jet than at shallow angles, validity is given for blowing rates up to \( M = 2 \) (Figure 11). With low pitch to diameter ratios \( s/D \), the problem of too low calculated effectiveness for small downstream distances due to incorrect prediction of jet mixing reduces in cases of low blowing rates. For high blowing rates the low entrainment rates support formation of a stable film over the surface, resulting in a nearly constant effectiveness on a high level (Figure 12). The latter observation is not present to this extent in the experimental results.

It is concluded, that the use of the presented code leads to reliable results of lateral averaged film cooling effectiveness in the preferred range of blowing rates up to 1.0. The reason for this restriction is the use of the standard k-\( \epsilon \)-model not being able to calculate the inclined jet in crossflow problem. Beside this known problem, the representation of the cooling gas ejection flow field is of very high quality and offers the possibility of detailed numerical studies. Therefore the examined code is suitable as a tool to carry out sensitivity studies in the range of parameters specified above.

**PARAMETERS OF INFLUENCE ON THE FILM COOLING PROBLEM**

The following parameters are collected for the problem of single row flat plate film cooling with cylindrical streamwise inclined holes. It is presumed, that there is no specific crossflow relative to a sharp edged hole inlet on the side of the cooling supply. The state of the hot gas stream is given by its velocity, density and temperature, further the properties of viscosity, heat capacity (at constant pressure) and thermal conductivity:

\[
U_G, \rho_G, T_G, \mu_G, c_p G, \lambda_G
\]  

The same properties are related to the coolant flow at hole exit:

\[
U_C, \rho_C, T_C, \mu_C, c_p C, \lambda_C
\]  

Further the turbulence of the hot gas stream and the pressure within the flow domain have to be taken into consideration. The displacement thickness of the hot gas boundary layer at the ejection location being present without injection was chosen as a representative for the boundary layer influence. This gives the additional parameters of the hot gas stream:

\[
\delta_u, \rho, D
\]  

Finally, there are the geometries of the ejection, namely hole diameter, ejection angle, downstream length from ejection location, lateral distance from the hole centerline, hole pitch and hole length:

\[
D, \alpha, x, z, s, L
\]  

Together with resultant effectiveness \( \eta \), there comes a system of 22 entities, forming some kind of functional relationship of independent variables:

\[
\eta = f(U_G, \rho_G, T_G, \mu_G, c_p G, \lambda_G, U_C, \rho_C, T_C, \mu_C, c_p C, \lambda_C, U_C, \rho_C, T_C, \delta_u, \rho, D, x, z, s, L)
\]  

To achieve a reduction of this formulation, a similarity analysis is performed following the Buckingham Theorem. After some transformations, this process leads to dimensionless similarity numbers, forming an analogue functional relationship:

\[
\eta = f \left( Re_D, Ec_G, Pr_G, Eu_G, Tu, \frac{\mu_C}{\mu_G}, \frac{c_p C}{c_p G}, \frac{\lambda_C}{\lambda_G}, \frac{T_C}{T_G}, \frac{x}{D}, \frac{z}{D}, \frac{\alpha}{D}, \frac{s}{D}, \frac{\delta_u}{D}, \frac{L}{D} \right)
\]
Figure 13: Prandtl numbers, viscosity and thermal capacity of gases [8, 9]

We find common parameters of film cooling as blowing rate, downstream distance to hole diameter ratio, lateral distance to hole diameter ratio, blowing angle, pitch, displacement thickness, and hole length to diameter ratios. Beside these, also the similarity numbers of Reynolds, Eckert, Prandtl, and Euler, all based on hot gas properties, come up. At least hot gas turbulence level and ratios of viscosity, heat capacity, thermal conductivity, temperature, and momentum of hot gas to coolant flows have to be considered. It should be mentioned, that the not so common Eckert number can be seen as a representation of the Mach number. The functionality is formed as

$$Ec = \frac{u^2}{c_p T} = Ma^2 (\kappa - 1)$$

with the isentropic exponent $\kappa$. By examining this set of parameters, it can be recognized, that the fluid dynamic properties of viscosity, heat capacity and thermal conductivity of gases can be assumed to be independent of the pressure within an interesting range of about 1 to 40 bars. Pressure influence is represented in its effect on density and therefore the Euler number can be removed from the list of independent variables. As the influence of the ratio of thermal conductivity can be replaced by the ratio of Prandtl numbers, it can be seen: Prandtl number varies in a very small band over the hole range of possible temperatures, e.g. for air within 0.7 and 0.73 (Figure 13). Accepting an error of 2% of this value allows to replace Prandtl numbers by a constant value, thereby excluding thermal conductivity as a parameter.

There still can be made further simplifications. Assuming coolant and hot gas being the same gas, some statements concerning ratios of temperature, heat capacity and viscosity are possible. The above assumption is not offended by the hot gas being the combustion exhaust and the coolant being air. For fluid dynamical discussion, the differences in properties of air and combustion gases are neglectable (Figure 13). In the present range of pressures and temperatures most gases can be seen as ideal gases with the equation of state given as

$$\frac{P}{\rho} = RT$$

For constant pressure we may write

$$RT = \frac{R}{\rho} = \text{const}.$$ 

This results in

$$\frac{T_c}{T_G} = \frac{P_o}{P_c}$$

eliminating $T_c/T_G$ as independent variable. Considering all these assumptions, ratios of viscosity and heat capacity can be expressed as

$$\frac{\mu_C}{\mu_G} = f\left(\frac{P_c}{P_o}, T_G\right)$$

$\frac{c_{pC}}{c_{pG}}$ as independent variable. Considering all these assumptions, ratios of viscosity and heat capacity can be expressed as

$$\frac{\mu_C}{\mu_G} = f\left(\frac{P_c}{P_o}, T_G\right)$$

Figure 14: Viscosity and heat capacity ratios over hot gas temperature and density ratio for air [8, 9]
Figure 14 shows a plot of these ratios against density ratio and hot gas temperature. The edge of the presented surfaces at low temperature has no physical reason, it marks the lower end of the used property data set. These plots show, that viscosity ratio is virtually independent from hot gas temperature. For higher temperatures, this is valid for heat capacity ratios as well, for lower temperatures, we have to accept some dependency. Examining the total range of heat capacity ratio values reveals, that this band is very small. Presuming independence from hot gas temperature, we only have to accept a maximum error of about 3% in the given range of density ratios. So we postulate heat capacity being independent from hot gas temperature and find the examined ratios reduced to expressions of the density ratio and not being independent variables. The closed set of all parameters is now

$$\eta = f\left(Re_D, Ec_O, Tu, l, M, \frac{x}{D}, \alpha, \frac{s}{D}, \frac{d_1}{D}, \frac{L}{D}\right)$$ (15)

**COMPARISON OF EXPERIMENTAL AND TURBINE CONDITIONS**

Knowing this complete set of parameters as similarity numbers, it can be verified, if the similarity conditions are respected in the experimental approach to the film cooling problem. Comparing data for geometry, blowing rate, and momentum ratio it was found, that the numerous experiments cover the range of possible realizations in turbo machinery. Even turbulence influence was examined to a wide extent [1]. For Re_D and Ec_O numbers the situation is different. Figure 15 shows a chart of Reynolds and Eckert numbers of reviewed reports and some hypothetical working points of turbo machinery applications within an area, where real turbine conditions can be found. In addition to these markers, there are calculated points that will be discussed later.

It is obvious from this chart, that most experiments are far outside the limits of this area. In publications with Reynolds and Eckert numbers within this area, comparisons of runs with different Mach numbers have also different Reynolds numbers and are usually considered to bring nearly the same results for lateral averaged effectiveness [13, 18]. It was not possible to recognize a definite statement on the separate effect of a change in Reynolds or Mach or Eckert numbers. So the question is: What does this violation of similarity laws mean to the results of these experiments?

To answer this question, numerical setups of experiments, that were used for the prescribed validation of the CFD code, were taken and extrapolated to engine conditions. For a case of Kruse setup (M=0.5) first separately Reynolds and Eckert number were raised by manipulating temperature and velocity levels. Finally both Reynolds and Eckert numbers were adjusted to the high value (markers a, b, c in Figures 15 and 16). A supplement run was performed pushing Eckert and thereby Mach number to maximum realizable values with the constant cross section channel model (marker d in Figures 15 and 16). Several further calculations were performed using the Liess setup (M=0.5 and 1.5, markers e, f in Figures 15 and 16). Figure 16 shows the resultant distributions of lateral averaged effectiveness of these runs. The range of examined Reynolds and Eckert numbers in this numerical experiment is presented in Table 2.

The diagrams show, that there is no significant influence of Eckert and Reynolds numbers on $\eta$. Slight differences in the setup conditions are unavoidable even in the CFD calculation, because the adjustment at ejection point is a sensitive and iterative process of manipulating the in- and outlet conditions. This time expensive operation was not brought to full convergence with the experimental conditions. Ac-
cepting this unavoidable deviation, we exclude Reynolds and Eckert numbers from the collection of influence parameters on $\eta$. We get now the final set of parameters for the examined simplified film cooling problem:

$$\eta = f\left(\frac{Tu, l, M}{D}, \frac{s}{D}, \frac{L}{D}\right)$$  \hspace{1cm} (16)

These parameters seem to be familiar to someone who has followed literature reports from film cooling research. Yet having in the past only to accept that these are in fact parameters of influence, we are now able to state: this is the closed set of parameters of this problem, all other possible sources of influence had been excluded by obvious physical reasons and experimental and numerical research.

CONCLUSIONS

Full 3-D Navier-Stokes calculation of the plate flat single row of cylindrical streamwise inclined holes film cooling problem was performed using a commercial all purpose CFD code. With respect to the restrictions of the code, e.g. use of standard $k-e$-turbulence modeling, best results were obtained. The available code was validated for prediction of $\eta$ in the interesting range of moderate momentum ratios and blowing rates. Hence we conclude, it is not a question of a highly specialized code to get a reasonable approach to the film cooling problem, rather it is important to model the examined flow situation carefully.

Further all possible parameters affecting the temperature problem of the discussed film cooling situation were recognized. Examination of physical facts of the flow situation allowed a significant reduction of the parameter set. By using the validated code as a tool, insensitivity of the problem to Reynolds and Eckert numbers was shown, in agreement with experimental results. From a total of 21 parameters the number of independent variables on $\eta$ was condensed to 8, giving the complete base of the problem. It has to be pointed out, that the above assumptions for the reduction of the parameter set are valid for standard turbine conditions or experiments using the same gas as coolant and hot gas stream. For experiments on foreign gas ejection or applications on steam injection a more general set of parameters has to be chosen.

REFERENCES


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