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A review of gas turbine effusion cooling studies

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ABSTRACT

Effusion-cooling is the next logical step in gas turbine blade cooling. Research on this topic has been done since the early 1950s, but manufacturing and modelling difficulties have prevented its commercial application so far. Still, there is a multitude of scientific publications about most aspects of this technology. Here, an overview over the publications most relevant for engineering uses is provided, with a focus on its application to gas turbine blades. The topics addressed here include the basic geometric and aerodynamic parameters known from film-cooling, but also the thermal conductivity of the base material, simplified approaches for modelling effusion-cooling and finally the application to blades, which incorporates the combination of impingement- and effusion-cooling as well as influences from operation.

Contents

1. Introduction

Modern gas turbines are operated at hot gas temperatures well in excess of the softening temperature of the metallic components in the hot gas path. Therefore different cooling technologies are normally applied, the most advanced of which is film-cooling. The discrete nature of these cooling holes does not provide a closed cooling film over the blade surface and the blade is not adequately shielded from the combustion gases. The required excessive amount of coolant leads the coolant jet to overshoot and induces vortices that can considerably reduce the effectiveness. Only the far field, where the cooling film reattaches to the surface, is cooled properly. With effusion-cooling, however, the jets influence each other, and because of their lower impulse do not overshoot into the main flow but remain within the boundary layer. This

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Nomenclature

A	area [m^2]
D	hole diameter [m]
H	spanwise hole distance [m]
L	streamwise hole distance [m]
T	temperature [K]
ad	adiabatic
c	cold
d	curvature diameter [m]
h	hot
p	pitch; distance between holes [m]
u	velocity [m/s]
λ	thermal conductivity [W/mK]
ρ	density [kg/m^3]
CFD	Computational Fluid Dynamics
I	Impulse Ratio $\left[\frac{u_c^2\rho_c}{u_h^2\rho_h}\right]$
LES	Large Eddy Simulation
M	Blowing Ratio $\left[\frac{u_c\rho_c}{u_h\rho_h}\right]$
PIV	Particle Image Velocimitry
R	Density Ratio $\left[\frac{\rho_{in}}{\rho_{out}}\right]$
TBC	Thermal Barrier Coating
V	Velocity Ratio $\left[\frac{u_c}{u_h}\right]$
α	Inclination Angle of Cooling Hole [°]
η	Cooling Effectiveness $\left[\frac{(T_h-T_w)}{(T_h-T_c)}\right]$

intensifies the tendency of the vortices to reattach to the wall which increases the cooling effectiveness.

It is of paramount significance that effusion-cooling is the result of many subsequent rows of cooling holes. In other words, true effusion-cooling cannot exist after only one or two rows of cooling holes [1]. It seems 7 rows of cooling holes is the minimum for a model that encapsulates all relevant phenomena. Therefore most research on film-cooling cannot provide data that is valid for effusion-cooling. Furthermore, the flow of the coolant through the cooling holes causes convective cooling, the potential of which is utilized far better in the more densely spaced effusion-cooling holes than with film-cooling. This, as well as the cooling in the coolant plenum, can only be properly assessed using conductive materials (as far as blades are concerned) in experiments or conjugate calculations.

The manufacturing costs as well as concerns regarding the reliability of effusion cooling have so far prevented its commercial use in gas turbine blades. In combustor liners, on the other hand, effusion-cooling is much closer to commercial application and most studies therefore focus on this. Hence selected studies of effusion cooling of combustor liners are also included in this overview. Differences like the length-to-diameter ratio of the holes, the hot gas *Ma* and the thermal conductivity of the substrate material exist between blades and liners. Therefore, some results of studies on liners cannot be applied to blades, but some aspects of the cooling, like the development of the cooling film over multiple rows of cooling holes, are not affected by this. Transpiration-cooling (as defined below) s explicitly not part of this review because the manufacturing, the flow through the porous body and the distance to application all differ significantly from effusion-cooling.

The following review is structured according to topics that are the most important for the development and assessment of effusion-cooling. Since some of the larger studies occupy themselves with multiple topics, a certain amount of cross-referencing between sections is unavoidable, although it was attempted to reduce this to a minimum. Within the sections analytical publications are discussed before experimental ones and numerical work is usually discussed last. This by no means intends to be a weighing of the importance or accuracy of the publications. While the evidence in one field may suggest that numerical results do not capture the experimental data correctly (as with turbulence modelling), an exceptionally good agreement between numerics and experiments may be found in other fields. One example for this can be seen from Fig. 5, where experimental (Particle Image Velocimitry), advanced CFD (Large Eddy Simulation) and "simple" CFD (Reynolds-Averaged Navier–Stokes) using a Baldwin-Lomax turbulence model are compared. This suggests that although the detailed flow field may not be captured fully, the results suffice for more general statements on, for example, the temperature distribution.

1.1. Effusion-cooling vs. other cooling techniques

The difference between film- and effusion-cooling lies in the size and number of the cooling holes: Effusion has holes with diameters down to 0.2 mm; their large number causes a very close spacing and therewith an interaction of the coolant jets that does not exist with film-cooling. Associated with these characteristics are differences in the amount of coolant exiting each of the holes as well as its velocity and impulse. The flow through effusion-cooling holes is, due to their small size and the lesser amount of coolant, normally laminar. Typically, the coolant used for both

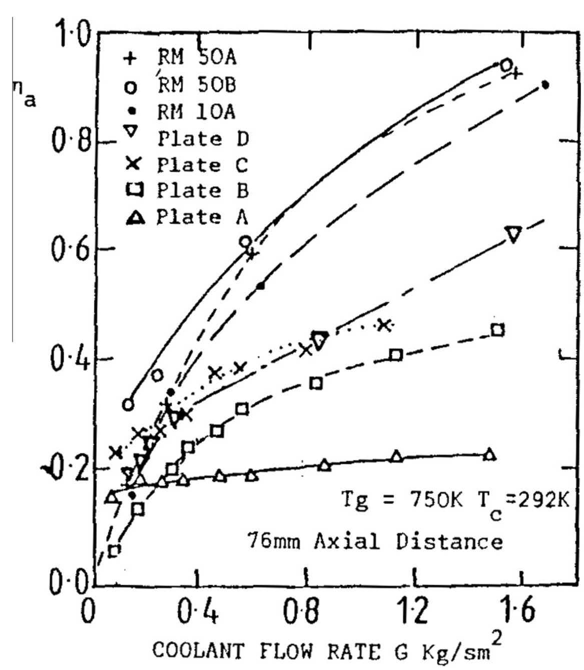

Fig. 1. Comparison of the cooling effectiveness of effusion- and transpiration-cooled plates [3].

film- and effusion-cooling is the same, i.e., high-pressure air taken from compressor bleeds.

The difference between effusion- and transpiration-cooling, on the other hand, is not always clear. In general the term "transpiration"-cooling is used when the coolant reaches a thermal equilibrium with the solid it is moving through. Additionally, a phase-change from liquid to gaseous may happen after the coolant has transpired from the solid or the transpiring fluid may be a different substance than the main flow altogether. These last differences are not always observed and they do not apply to the transpiration-cooling studies discussed here. Also, transpiration-cooling is associated with porous media like ceramic foams (porous media making attaining a thermal equilibrium much more easy), whereas effusion-cooling uses solid plates perforated with a multitude of small, discrete cooling holes. These theoretical differences are not always regarded in praxis, especially in older studies.

In [2] film- and transpiration- as well as convective cooling were investigated for flat plates using analytical equations. It was shown that transpiration-cooling has a far better η than convective cooling, but it was also greatly superior to film-cooling for $10^3 \leqslant Re \leqslant 10^9$ and $0 \leqslant M \leqslant 0.012$. A direct experimental comparison between effusion- and transpiration-cooling was made in [3]. The authors pointed out convincingly that η_{ad} is not a valid parameter for evaluating the performance of transpiration- or effusion-cooling. Furthermore, they showed that an increasing blowing ratio does not lead to coolant lift-off with transpiration-cooling, but it may lead to a detachment of the cooling film with effusion-cooling. The coolant exit temperature was considerably higher for transpiration-cooling because the convective cooling in the porous material was far better here than for effusion-cooling. With regard to overall cooling effectiveness the former was far superior to the latter, see also Fig. 1. The authors of [4] looked at a transpiration- respectively film-cooled cylinder in a hot gas flow. Transpiration-cooling in general provided the higher η and caused less disturbance to the boundary layer.

A similar investigation, this time numerical, was performed in [5] calculated that transpiration- and effusion-cooling provide a much better thermal protection of a cooled plate than simple slot- or film-cooling and showed that the effectiveness of transpiration- and effusion-cooling is nearly the same after a certain streamwise length, especially for higher blowing ratios. In another numerical study, [6], it was also pointed out that the boundary layer develops differently for transpiration- and effusion-cooling because the coolant of the former has a much higher impulse.

In light of the above, neither the results of film- nor those of transpiration-cooling investigations can be applied *a priori* to effusion-cooling.

2. Basics of effusion-cooling

Effusion-cooling still is a long way from being applied on actual turbine blades and the largest part of past and current research is devoted to understanding the basics of this cooling technique. These basics include simple geometric parameters like hole spacing, but also the investigation of the coolant-hot-gas interaction in the boundary-layer of the main flow.

2.1. Hole shape, spacing and angle

Two primary factors for increasing η are the spacing of the holes and the inclination angle of the hole against the surface. Furthermore the manufacturing of the cooling holes can also contribute to enhancement of the cooling effectiveness as this may affect the shape of the holes. Surface roughness, especially in the cooling holes, is a factor that is worthy of attention, but this has not been done systematically apart from [7], who report that rougher walls in the cooling holes enhance η (see also further down).

As far as the shape of the hole is concerned, the experimental results presented in [8] show a clear dependence of η on the hole size. The overall η increased from approximately 0.54 to 0.63 for the highest M over the three investigated hole sizes of $d = 1.18$ mm, $d = 1.42$ mm and $d = 2.16$ mm. This increase can be attributed mainly to an increase in the effectiveness of the cooling film due to a decrease in the coolant outlet velocity with increasing hole sizes. The same group of authors demonstrated in [3] that increasing the number of holes and simultaneously decreasing their diameter can lead to a substantial improvement in η because the coolant effusing from these smaller holes induces less mixing in the boundary layer.

It is known from film-cooling investigations that a diffuser-shaped cooling hole outlet with a lateral widening of e.g., 10° increases the lateral spread of the coolant and at the same time decreases the lift-off of the coolant jet. In the numerical study [9] this effect is shown to exist for effusion-cooling as well (for a detailed description of the manufacturing of shaped effusion-cooling holes see [10]). In [9] different shapes of the cooling holes were numerically investigated, using an in-house conjugate solver. As depicted in Fig. 2, one configuration had a laterally-widened diffuser in the *TBC*, the other a diffuser that is not laterally widened but simply laid-back. Both had $\alpha = 30°$, $L = 6D$ and the calculations were set for $M = 0.28$ and $M = 0.48$. For both configurations the cooling effectiveness measured in the substrate layer was considerably higher than for the cylindrical holes and the coolant spread more evenly over the surface. This resulted in a reduction of the thermal load on the *TBC* by 35% for the first configuration and 47% for the second one. In a further step, the contouring of the hole was extended to the full height of the plate. This resulted in an increase in cooling effectiveness of 5%-points for the higher M, but for the lower M the cooling effectiveness of the fully-contoured configurations was slightly lower than that of the partially-contoured ones.

The spacing of the holes is of primary importance for the resulting cooling effectiveness. In [11] the influence of hole spacing of both L and H on the cooling effectiveness is investigated experimentally. L is varied from 8 to 12 to 30D, H between 8 and 12D.

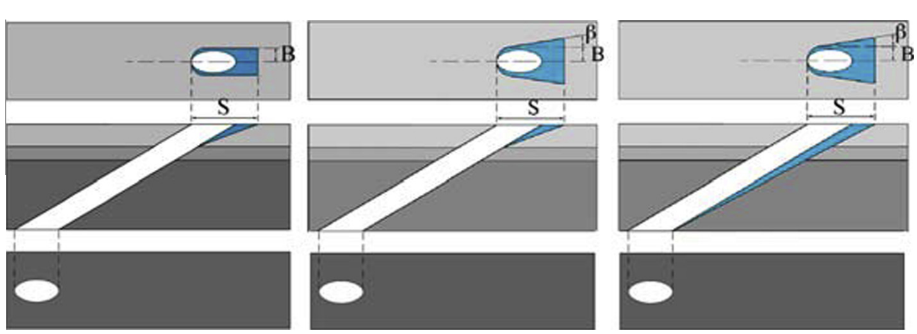

Fig. 2. Different diffuser shapes with length "S" and width "B" after [60].

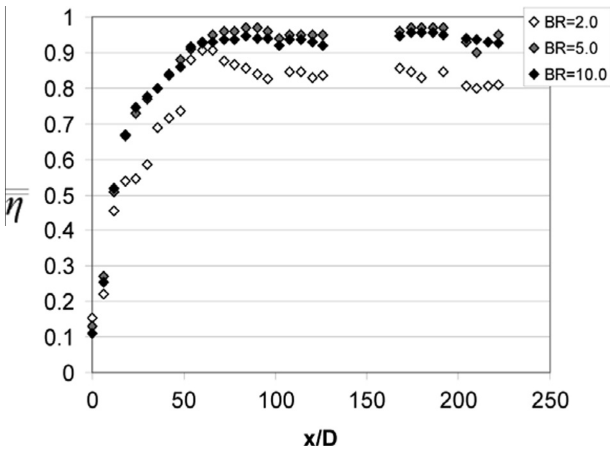

Fig. 3. Spatially-averaged adiabatic film-effectiveness for $L = 6$, $H = 3$ and a contraction ratio of 4 [12].

The authors state that the wider H prevented the coalescence of the jets and therefore decreased η, increasing L decreased η as well.

Another excellent experimental study with a field of staggered, inclined cooling holes and $H = 3,5 or 7\ D$ for $L = 6D$ and $H = 5D$ for $L = 18D$ and M of 2, 5 and 10 can be found in [12]. The surface temperatures were measured using infrared thermography. The authors report that the first configuration seems overcooled, since η did not increase anymore after the 7th row of cooling holes. It continued to increase over the full length of the investigated array when $L = 18D$, though, and remained circa 20% below that of the denser array (cf. Fig. 3). When H was increased from $3D$ to $5D$ to $7D$, the maximum cooling effectiveness dropped from nearly 1.0 to approximately 0.7 to 0.8. The authors also remark that the heat transfer coefficients in streamwise direction for the denser array of holes are higher and develop more rapidly than for the less dense array. This is probably due to the lack of interaction between the separate cooling jets for the more widely spaced array. These results were obtained for a configuration in which the main flow is contracted by a factor 4, though. Both the acceleration of the flow and the resulting increase in M were much higher here than for a channel with a constant flow area, which the authors also discuss in the same publication.

In [13] conjugate numerical simulations of three different plates with 15 rows of shaped cooling holes and $M = 0.28$ and $M = 0.48$ are described ($\eta_{max} \leqslant 0.9$). Contrary to the experimental results of Scrittore et al. [14] (discussed in more detail below) M increased with increasing row number because of a numerically prescribed constant pressure at the plenum inlet. The first configuration investigated in [13] had a constant $L = 6D$, whereas the other two configurations had a gradually or suddenly increasing axial hole distance. This change increased η, most significantly for the lower M. The last geometry proved to have an η that was 2 to 3% points higher than that of the other two. It is reported that the temperature of the coolant at the inlet of the holes decreased from the 4th to the 8th cooling hole for all three configurations and both M. This shows that, even though efforts were made to minimise the effect of convective cooling in the plenum, the flow here still contributed to the overall cooling.

The third geometrical aspect relevant for effusion-cooling holes is their angle of inclination against the hot gas surface. Because the cooling holes get longer when the angle to the surface gets shallower, the amount of convective cooling in the holes increases. This is combined with the better attachment of the coolant to the hot gas surface, which leads to an increase in the cooling effectiveness.

Conjugate calculations of an effusion-cooling configuration with inclination angles of 30°, 60° and 90° for cylindrical holes with $D = 0.5$ mm and $M = 0.5$, $M = 1.0$ and $M = 2.0$ confirm this [15]. There is little difference with regard to η between the holes with $\alpha = 30°$ and $\alpha = 60°$, although the former are said to be slightly better because of the slightly higher convective cooling within the holes. The authors note that the behaviour of the holes with $\alpha = 90°$ deviated notably from the two other configurations, especially regarding the large drop in η after the effusion-cooled region (cf. Fig. 4). This tendency decreased somewhat with increasing M because of the higher amount of coolant used here. In [16] another conjugate analysis of a similar problem is provided. Here, a field of 7 rows of staggered cooling holes each with $D = 0.2$ mm, $L = 6D$ and $H = 1.5D$ was investigated. For $\alpha = 45°$ the single jets could be distinguished clearly, which is not the case for $\alpha = 30°$. Furthermore, the cooling film showed significantly less lift-off in the latter case. the authors attributed this partially to the reduced penetration of the boundary layer, and partially to the reduced intensity of the vortices that were induced in the main flow. They also report that η can be substantially increased if L is reduced from $10D$ to $6D$.

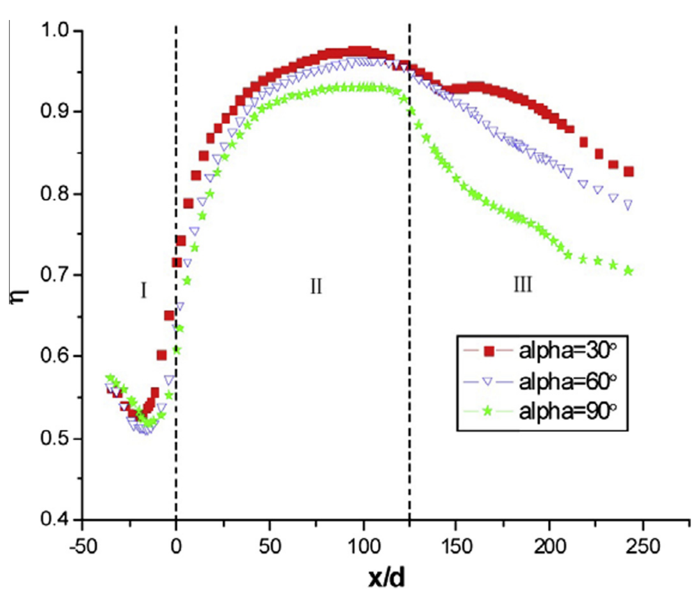

Fig. 4. Cooling effectiveness for three different cooling hole angles and $M = 0.5$ in the region leading up to the cooling hole (I), the near-field (II) and far-field (III) regions [15].

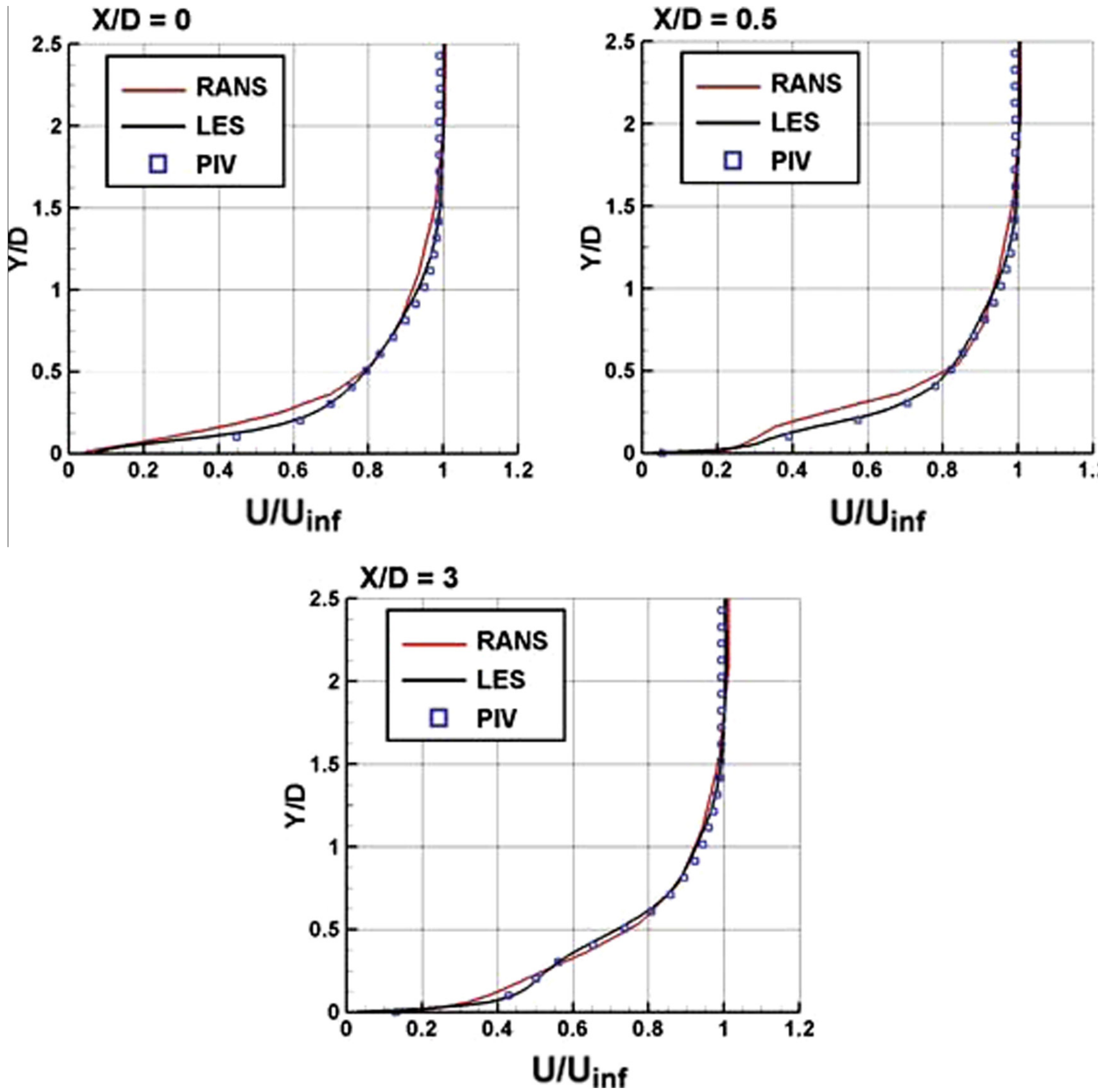

Fig. 5. Comparison of RANS, LES AND PIV in the mid-plane after a single effusion-cooling hole [60].

Some studies combine the effects of hole spacing and hole angle to determine their relative importance. In the (primarily) experimental studies [17,18] it is reported that the cooling effectiveness depends more on L (0.5 vs. 0.6 mm) than on α (24° vs. 30°). It should be noted, though, that the two angles are very similar and a larger difference in η should be expected when the angle increases. The same tendency is also reported in the experiments [19,20] where only a limited influence of a variation of α from 15° to 20° to 30° on η was observed in adiabatic investigations. But, since all investigated injection angles were extremely shallow, no great difference in cooling film attachment should be expected. The authors note that a reduction of the hole spacing increases the cooling effectiveness. In this case this phenomenon should be at least partially attributed to the experimental setup, in which M decreased with an increasing number of holes.

Other studies are dedicated to more advanced cooling hole patterns that have a hole angle relative to the flow direction. A numerical study in which the flow field and adiabatic cooling effectiveness distribution of two different hole patterns were compared with an experimental one was presented in [21]. Both patterns consisted of multiple rows of staggered cylindrical cooling holes with $\alpha = 30°$ either in or against the streamwise direction. The authors point out that in the former case mixing with the hot gas was reduced, but the jets tended to lift-off at higher M. In the latter case the coolant did not only cool the area directly behind the cooling hole but also the surrounding area. In [22] a flat plate with compound angle injection as well as injection angles of 30° in and against the main flow direction were investigated both numerically and experimentally. The blowing ratios varied between 1.0 and 4.0 with $R \approx 1$ and $D = 1.5$ mm. The plate used for the experiments had 14 rows of cooling holes with alternating 9 or 10 cooling holes per row. The authors note that for the high M investigated here the effusion-cooling did not behave fundamentally different from film-cooling and M did not influence η_{ad}.

Summarizing, these "geometrical" aspects of effusion-cooling are understood rather well. The optima for the different parameters have, roughly, been found and applying them is now a question of manufacturability and costs.

2.2. The influence of temperature level and different ratios for coolant/hot gas interaction

In general neither experimental nor numerical studies take account of real turbine inlet temperatures. These have firstly an effect due to the resulting radiation, but also because of the resulting density ratios R between coolant and hot gas. Among the few

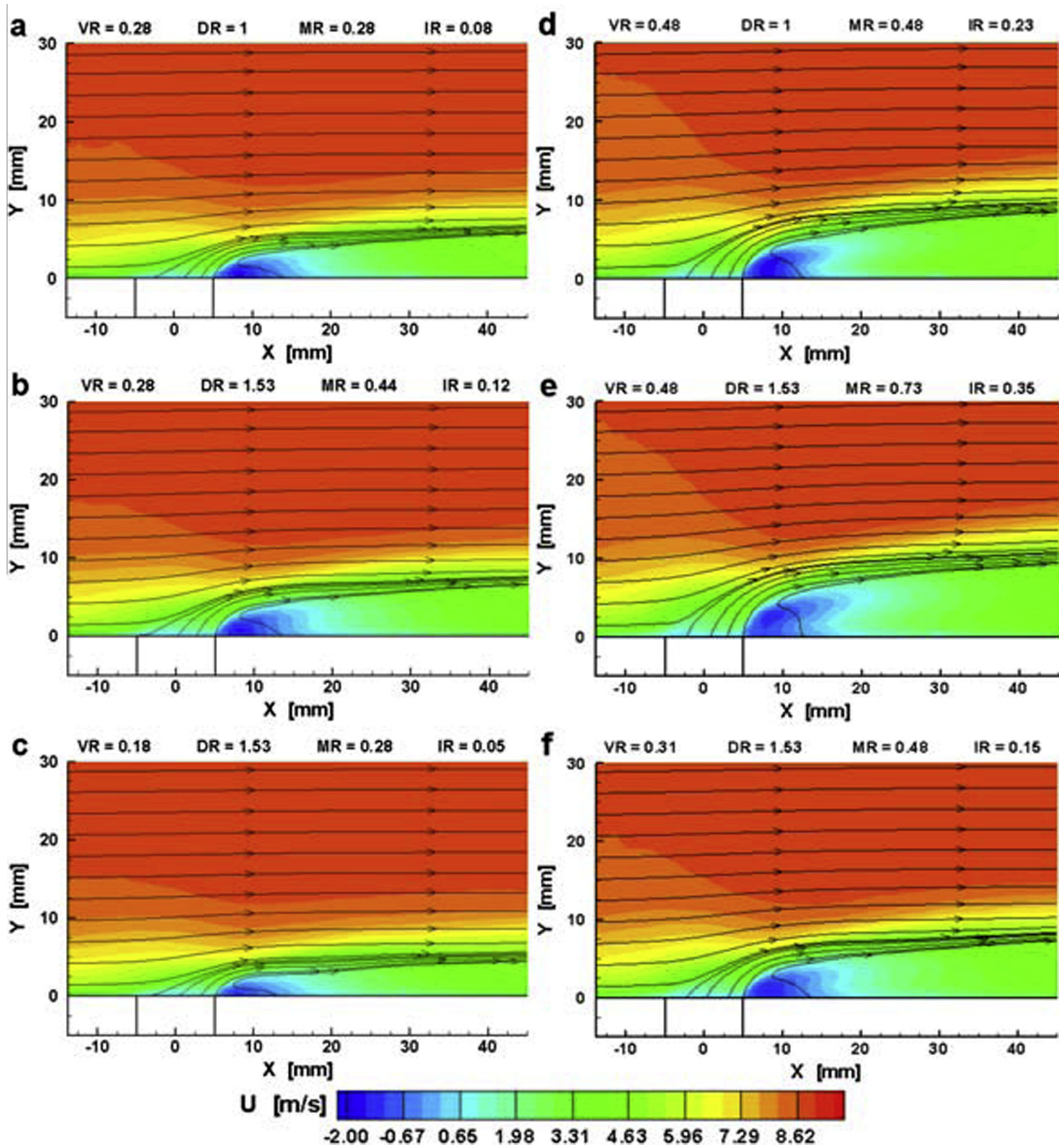

Fig. 6. Flow field at the cooling hole outlet for $\alpha = 90^\circ$ and different V and M [27].

investigations that did try to quantify the differences between low- and high-temperature tests, the experiments in [7,8] can be considered to be the best. The authors showed the large effect of radiation by looking at the same effusion-cooled plate with hot gas/coolant temperatures of 750 K/293 K and 2100 K/770 K, respectively. This yielded an R of approximately 2.5 respectively 3.0. The resulting η dropped by up to approximately 9%-points for the higher hot gas temperature. The authors partially attribute this to the different coolant exit velocities and partially to radiation. In an early paper [23] the authors measured the surface temperatures on one and the same blade for 250 °C, 500 °C, 770 °C and 1000 °C. They note that radiation effects only become noticeable above 500 °C, but from this temperature onwards approximately 90% of the measured drop in cooling effectiveness of approximately 8%-points is due to radiation effects.

There are several dimensionless numbers that aim at quantifying the physically relevant differences that dominate the interaction between the coolant and the hot gas. The most important among these are the density ratio R, the velocity ratio V and the blowing ratio M (all of which are defined in the Nomenclature). Most of the work on this topic has been in the experimental field.

For normal compressible fluids the temperature ratio is proportional to R. This may influence η as on the one hand denser coolant flows show less lift-off and on the other hand cool the surface better. In [24], conducted with cooling holes with $\alpha = 90^\circ$ it is stated that when R increases (the coolant becomes colder relative to the main flow) the convective cooling remained dominant but the overall relation between the film-cooling effectiveness and the overall effusion-cooling effectiveness changed. The same group also reported on the importance of considering the convective cooling in the holes for the temperature increase of the coolant and hence the change in the coolant density at the hole exit as well as its influence on the film cooling effectiveness on the plate surface [25].

In [19,20] it was shown that the effect of the temperature ratio on the cooling effectiveness significantly decreases with increasing

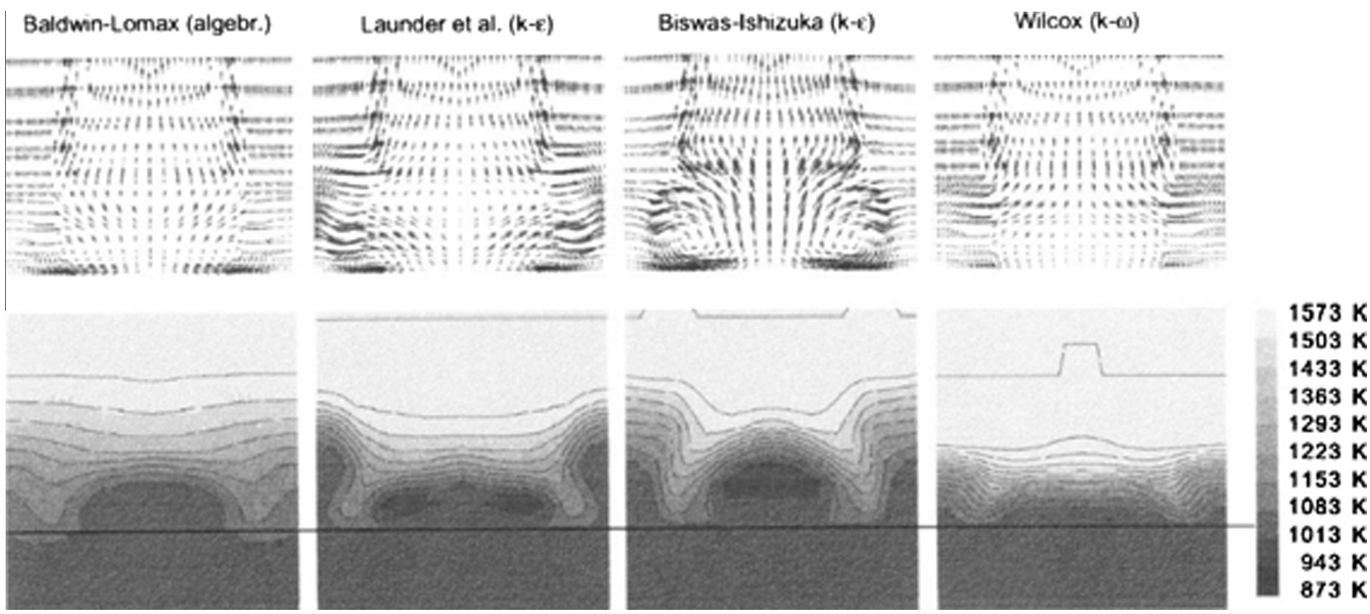

Fig. 7. Flow and temperature field at effusion outlet for different turbulence models [36].

V (achieved by reducing the main stream velocity). Since the temperature differences between the main flow and the coolant matter less at the high η achieved for the higher V, this seems logical. In [26] similar conclusions were reached for flows with $0.043 \leqslant Ma \leqslant 0.075$ and $42{,}000 \leqslant Re \leqslant 93{,}000$. The temperature ratio was changed by increasing the hot gas temperature. In this case the main reason for the observed effect seems to be the variation of the number, spacing and diameter of the holes. The amount of coolant entering the plenum was only measured (not regulated) and the mass flow rates for the different configurations consequently changed from 0.17–0.89 kg/m^{2}s, which corresponds roughly to a $M = 0.2 - 2.0$ for densely spaced holes with $D = 2.24$ mm and $M = 0.8 - 12.4$ for $D = 1.0$ mm. On this basis the authors concluded there is no simple correlation between temperature ratio and η. It should be noted that, in accordance with the point made in the Introduction, one of the further main conclusions was that the overall cooling effectiveness was significantly higher than the adiabatic effectiveness.

An extensive systematic investigation on which one of the different dimensionless ratios can be best used to evaluate effusion-cooling configurations has been performed recently. The experimental studies by [27] and Large Eddy Simulation (LES)-studies by [28] used the same geometries and boundary conditions; their results match extremely well (cf. Fig. 5). The effusion-cooling holes were very small and densely packed ($D = 0.2$ mm, $H = 3D$, $L = 6D$) and $\alpha = 90°$ or $\alpha = 30°$ in streamwise direction. The holes were either cylindrical or shaped. The Particle Image Velocimetry (PIV)-measurements were performed using a CO_2-injection with $R = 1.53$ to simulate the difference in density between coolant and hot gas [27]. The authors of both the PIV- and LES-studies focussed on $V = 0.1, 0.28$ and 0.48 as well as $M = 0.28$ and 0.48. The effect of the difference in density on the cooling effectiveness could be seen most clearly with M, although a weak effect could be discerned in the evaluation of V as well. The authors show that for holes with $\alpha = 90°$, R had little effect when V was evaluated, with only small differences in the recirculation area behind the cooling hole. When the evaluation was based on M, though, R did become important. As can be seen in Fig. 6, the higher density coolant penetrated further into the hot gas boundary layer. When the holes were inclined, the same tendencies could be observed, but the penetration of the coolant in the main flow was far less pronounced and the recirculation zone behind the cooling hole practically disappeared. The penetration into the main flow almost vanished when the inclined holes had a fan-shaped diffuser at the outlet. R can thus be said to be of little importance here.

In summary, getting the correct temperature level (radiation) and the temperature difference between coolant and hot gas (blowing ratio, density ratio) seem to have a considerable influence on the obtained results. Since radiation is not an issue unique to effusion-cooling, "standard" correlations may suffice, but this remains to be proven.

2.3. Turbulence

A large portion of basic research on turbulence in effusion-cooling configurations focuses on the determination of the turbulence in a boundary layer with foreign gas "transpiration". Very often these are highly specialist publications that are not always relevant for the direct technical application of effusion-cooling to blades. Therefore only some representative studies shall be discussed.

Combined experimental and numerical studies were presented in [14,29]. In the first publication, a large plate with 20 rows of cooling holes is investigated. The development of the cooling film over these holes is looked at for several blowing (momentum) ratios. These are described in terms of turbulence levels, though, and these scaled exactly with M, but had little influence on η_{ad}. In [29] a rather large array of 10 rows with 9 effusion-cooling holes each was investigated ($P/D = 7.14$). Two flows, one with a turbulence level of 0.5%, the other with a turbulence level of 18% were compared. It is said that the high level of turbulence significantly decreased η_{ad}. This view is in opposition to that of [30], but in the latter study $R \approx 1$ whereas the density ratio in [29] was with $R = 1.7$ much closer to real engine (combustor liner) conditions.

The two conjugate numerical investigations [31,32] also focussed on the effects of turbulence. The studies were validated using experimental data from other research groups. The results from the former publication were validated using experiments on an effusion-cooled combustion chamber liner [33]. With $\alpha = 17°$ the investigated holes had an angle of inclination very similar to film-cooling holes, but this made turbulence modelling very difficult. Consequently, the main deviations between experiments

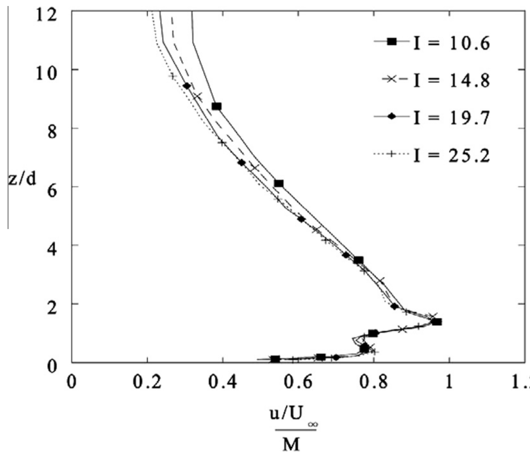

Fig. 8. S treamwise velocity profiles measured one row downstream of rows 1, 5 and 10 for $I = 10.6$ and 25.5 using M to normalize the profiles [14].

and numerics lay with the turbulent kinetic energy above the holes. The authors attribute this to insufficient turbulence production generated by the model at the hole inlet which is due to the low Re inherent to effusion-cooling. The temperature field deviated considerably from the experiments, but the heat transfer coefficients matched well. The numerical results from [32] were validated with the experiments from [34,35] for a single and a double row of cooling holes. Two different turbulence models were used, an anisotropic model and a standard k-ϵ-model, for $M = 0.5$ and $M = 1.0$. The models deviated in their predictions of the local temperature field, but the laterally averaged temperatures were almost the same and in good agreement with the experiments. Since one or two rows of holes are not enough to simulate effusion-cooling, the validity of these results may be limited. A comparison of both the Biswas–Ishizuka and Launder k-ϵ-models, the Wilcox k-ω-model and the Baldwin–Lomax algebraic turbulence model was performed by [36] for 7 rows of inclined, shaped holes. Their conjugate calculations showed no significant deviations in either the flow field or the temperature distribution around the holes between the models, as can be seen in Fig. 7.

It should be clear from the above that the turbulence level has a distinct influence on the cooling effectiveness, as is also the case for film-cooling. The currently available RANS-models cannot predict the detailed turbulence distribution on an effusion-cooled plate when compared with experimental results. The numerical results as such are consistent, though, and the deviations with experimental data seem to have little influence on the resulting predicted temperature distribution.

2.4. Boundary layer development

Most work on boundary layer development has been done in the experimental field, but recently some interesting LES-based work has been published as well. The largest part of these publications is concerned with the penetration of the coolant jets in the boundary layer.

The cooling film development for a flat plate with cylindrical cooling holes with $L = 4.4D$ and $H = 4.9D$ was looked at in a very good experimental investigation by Scrittore et al. [14]. The investigated M were with $3.2 \leqslant M \leqslant 5.0$ high, but for combustor liners not extraordinarily so. The authors show that the jet penetration in the boundary layer changed significantly with an increasing number of rows of cooling holes. Peaks in the velocity profile occurred nearer to the wall when the cooling hole was located further downstream, although the boundary layer was thicker here, as can be seen in Fig. 8. Due to a decrease in static pressure along the flow path M decreased by nearly 20%, but the variation in penetration depth does not change noticeably with M. For the last rows of cooling holes the authors report an η_{ad} of up to approx. 80%.

The authors of [30] performed their experiments on a plate with cylindrical cooling holes with $\alpha = 90°$ for $0.21 \leqslant M \leqslant 0.77$ at $R \approx 1$. The penetration depth of the coolant gradually decreased from the 1^{st} to the 15th cooling hole until it reached a constant value. It was only near the penultimate and last rows that η_{ad} reaches a constant value. A drop in η is noted after the first few rows of cooling holes. This is by the authors attributed to the high initial penetration of the jets in the main flow and the resulting vortices.

Elaborate experimental investigations into the boundary layer development of effusion-cooled plates have been also performed by Kruse [37] with excellent results. The experiments were conducted on quasi-adiabatic flat and curved plates with $\lambda = 0.209$W/mK and cylindrical cooling holes with $D = 0.1$ mm and $\alpha = 90°$. The boundary layer profile was measured using Pitot tubes for pressure and hot wire probes for velocity profiles. Regardless of the hole spacing, which varied between $p/D = 3$ and $p/D = 12$, even the smallest amount of coolant caused a significant increase in the velocity in the boundary layer (cf. Fig. 9). The V given by the author varies between 0.002 and 0.0085, which would roughly correspond to $M = 0.05$ to $M = 0.12$. This V was used to

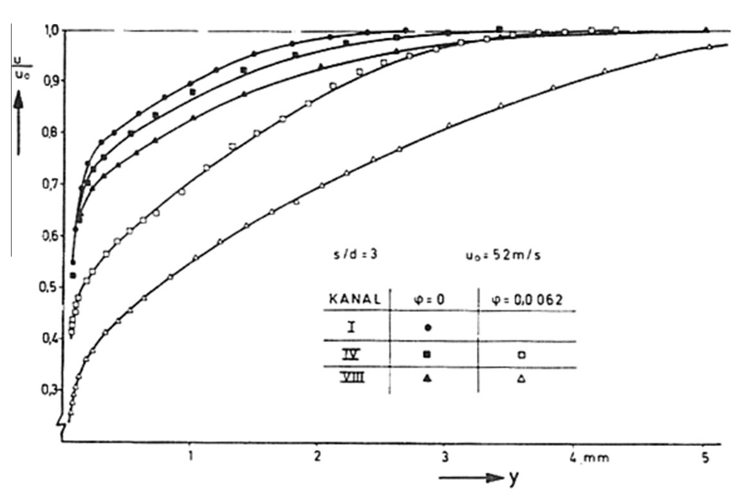

Fig. 9. Flow field in the boundary layer with and without coolant injection (''KANAL VIII''=8th measurement position; ϕ = blowing ratio) [37].

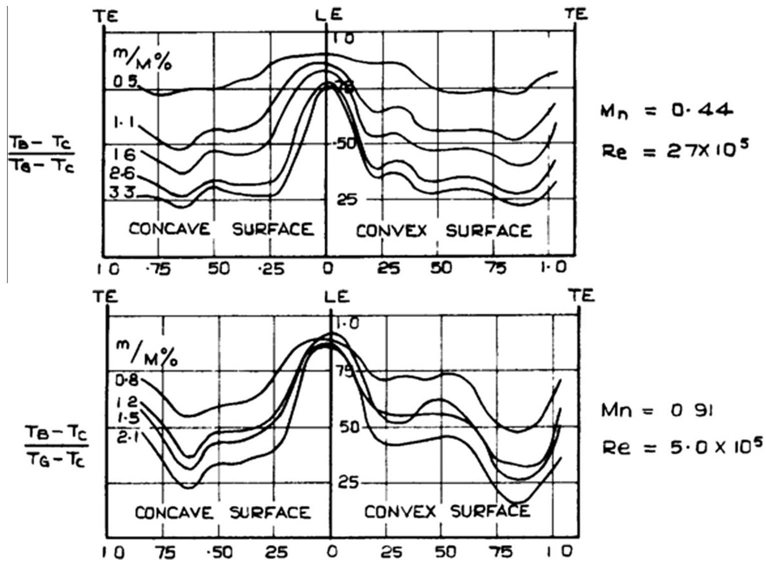

Fig. 10. Cooling effectiveness distribution along an effusion-cooled blade for two mach-numbers [23].

calculate an equivalent coolant velocity perpendicular to the surface. The maximum of η is approx. 0.78 in the 8th row for $M = 0.12$ and $p/D = 5$, for a p/D of 12 this decreases to approx. 0.42.

Most investigations focus on flat plates, but the experiments discussed in [38] focussed at flows with an adverse pressure gradient. The authors state that for flat plates the penetration increased with increasing row number. This tendency increased for the adverse pressure gradient flow. The turbulent intensity in the near-wall boundary layer was furthermore reduced, and although that in the upper boundary layer increased, the vortex intensity decreased there.

A combined experimental and numerical investigation of a flat plate with inclined, staggered cylindrical cooling holes was conducted in [33]. The experimental set-up consisted of seven rows with alternating eight or six cooling holes. The author reports some significant deviations between numerics and experiments in the velocity distribution in the boundary layer, but an overall very good match in the adiabatic wall temperature is stated. The experiments showed that for an M of up to 0.5 the cooling film developed directly from the first holes onwards, whereas it only developed further downstream when M was increased. Due to cooling film lift-off the first two rows in particular were cooled very poorly. This is in line with the results of [30].

In [39,6] a single cylindrical inclined cooling hole with $D = 5$ mm, $H = 6.74D$ and $L = 5.84D$ from an infinite array is investigated using LES. Overlapping boundary conditions around a unit cell were used (cf. also [40]). Thereby any number of upstream cooling holes could be simulated. The calculations were validated using experimental data from the LARA-project [41]. The density ratio equalled unity and the main flow Reynolds-number was with Re = 17,750 very low. The authors show that the flow pattern of the coolant entering the main flow depends on the number of upstream cooling holes. Although the deviations between the LES-calculations and the experiments were rather large, the qualitative agreement was good.

The development of the cooling film and the flow in the effusion-jet are understood rather well. Modern CFD calculations can give qualitatively sound results, although the quantitative agreement needs to be improved and some specialist problems, like flows with pressure gradients, have hardly been addressed yet.

2.5. Mach number

In the early investigations by Andrews et al. [23] it was experimentally shown that (for $M \approx 1$) an increase of Re and Ma on an effusion-cooled blade does not noticeably change the cooling effectiveness for blade exit Mach-numbers between 0.4 and 0.9, see also Fig. 10. This view is supported by the experimental data of [19,20], which state that the main flow Re did not have a significant effect on η. In these experiments an increase in Re implied an increase in Ma. The authors looked at an application for combustion liners, so very low Ma of approximately 0.035–0.115 were used and the provided V indicate high values of M. The results of [23] were also confirmed by the experiments in [42] for $0.15 \leqslant Ma \leqslant 0.4$ and $0.5 \leqslant M \leqslant 1.0$. When M rose above this value, though, the effect of Ma also increased, leading to lower values of the cooling effectiveness.

The conjugate numerical investigation [43] looks at plates with lower values of Ma and M ($M = 0.28/0.48$ for both $Ma = 0.1$ and $Ma = 0.25$). Keeping M constant for the higher Ma implies a significant increase in the amount of coolant introduced into the hot gas path. Nevertheless the authors report that the increase in the hot gas Mach number decreased the average cooling effectiveness with up to 8%-points.

The available data indicate that the influence of Ma on η is extremely limited, with the possible exception of very low M.

3. Thermal conductivity

The overall cooling effectiveness of blades depends sensitively on the convective cooling in both the cooling holes and the coolant plenum. Associated with this is the thermal conductivity of the base material of the blade. Traditionally, this has been the field of experiments because a numerical solution would require either a coupled CFD/FEM solver or a full conjugate calculation. Still, some interesting numerical work has been done. This is presented at the end of this section.

In the detailed and excellent study [44] all experiments were conducted on effusion-cooled walls with staggered cylindrical holes with $D = 12.0$ mm. The wall was either of brass or an acrylic resin and $0.28 \leqslant M \leqslant 0.75$ for $p/D = 5$, respectively $1.0 \leqslant M \leqslant 2.6$ for $p/D = 10$, both with $R = 0.5$. The results showed a considerably higher η for the brass wall with a measurable downstream

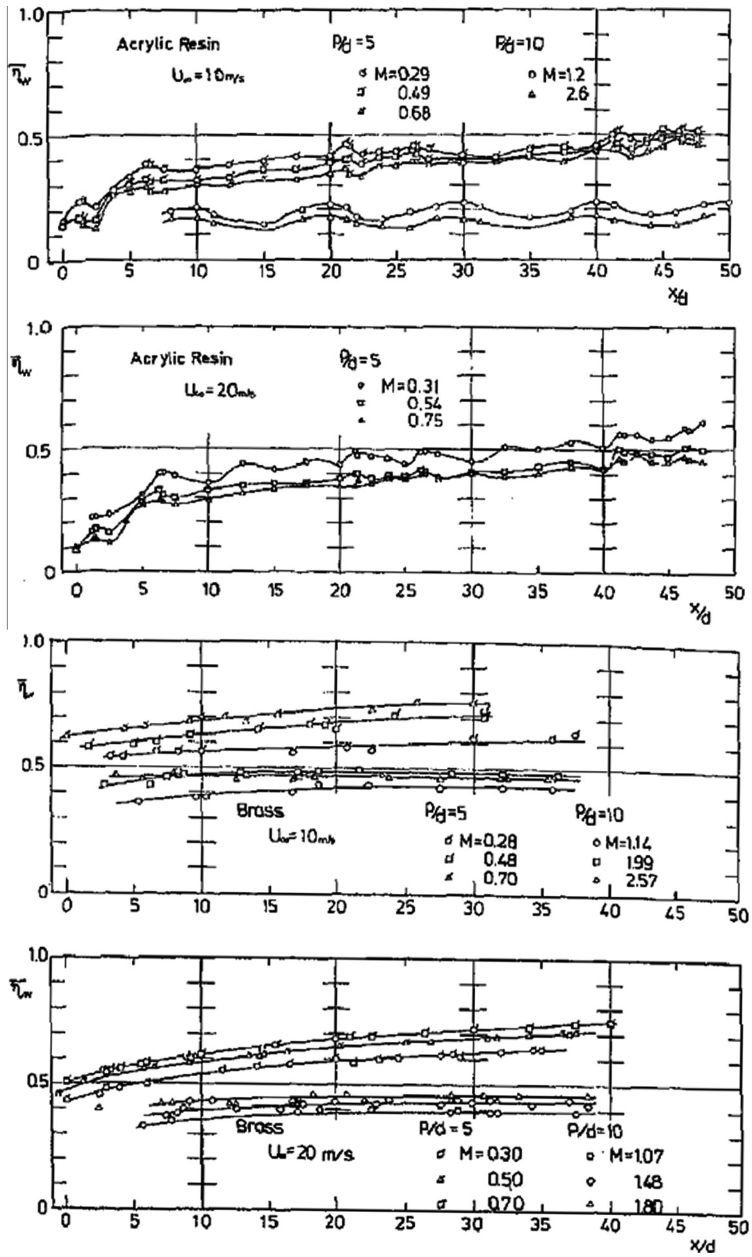

Fig. 11. Spanwise averaged cooling effectiveness for acrylic and brass plates top to bottom: Acrylic Resin U = 10 m/s – Brass U = 10 m/s – Acrylic Resin U = 20 m/s – Brass U = 20 m/s [44].

influence of the holes on the cooling effectiveness. No such influence could be detected for the acrylic resin wall. The spanwise averaged η for the brass wall steadily increased from 0.4 for the first holes to 0.5 for the last holes for the widest hole spacing. For the denser hole spacing η varied between 0.52 and 0.78, but for the acrylic resin wall it did not exceed 0.55 at any time and was on average well below 0.5. This is visualised in Fig. 11. In the companion paper [45] the heat transfer on the hot gas and inner wall surfaces of a thermally conductive wall was looked at by means of the analogy with mass transfer. The authors state the internal heat transfer in the holes and on the plenum wall dominated the overall η via the thermal conductivity of the plates. The experimental investigations discussed in [7] focus on much smaller holes (D = 1.4 mm), with M = 5 and $L \approx 10$. The measurements with conductive plates showed $0.4 \leqslant \eta \leqslant 0.6$, but $\eta_{ad} \leqslant 0.1$ for a non-conductive case, see Fig. 12. The significant contribution of the convective cooling in the plenum and λ to the overall cooling effectiveness was also noted by these authors.

Some of the publications discussed previously also looked at the influence of λ on the cooling, but in general these are of less importance. In [17,18] six flat plates with different cooling hole configurations were investigated. The temperature in the experiments was 600 °C, the pressure 1.07 bar. The authors provide a correlation for transferring these results to realistic turbine hot gas parameters, including a constant for radiation. The authors furthermore provide a correlation for calculating the relation between the heat transfer coefficients on both sides of the plate and within the cooling hole based on simple empirical and analytical equations. They point out that the heat transfer on the inner wall is largely independent of that within the holes and on the surface. In addition to this finite element analyses based on an estimated λ were conducted that enabled the authors to show that the stress pattern

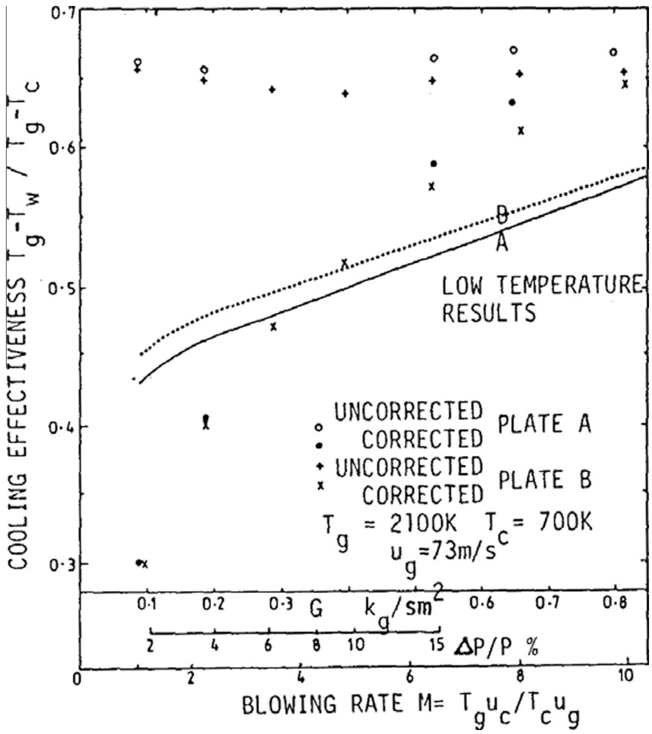

Fig. 12. Cooling effectiveness of effusion cooled plates at different hot gas temperatures both corrected and uncorrected for radiation influences for plates with drilled (A) and corroded (B) cooling holes [7].

mainly followed that of the temperature. Depending on the streamwise position along the plate η varied between 0.55 and 0.70 for $M = 0.5$, but for $M > 1.0$ cooling film lift-off is reported. The decreased cooling of the first two rows of the test rig used by [33] was especially bad when an adiabatic wall was used. When a conductive wall was chosen, the cooling of the first few cooling holes increased with increasing M due to the increased convective cooling in the holes and the heat conduction from the downstream area. It should be noted that the authors of [19,20] also looked at conductive (steel) and non-conductive (Teflon) materials claim that the different λ do not significantly change the cooling effectiveness. It seems likely that this behaviour, which differs notably from all other experimental investigations discussed so far, should be explained by peculiarities of the experiment.

In [46] a conjugate numerical investigation of shaped, inclined effusion-cooling holes is performed. The authors compared an adiabatic with a diabatic and a conjugate calculation. This enabled them to assess the impact of λ on the overall cooling effectiveness. The coolant exit temperature was considerably lower for the adiabatic calculation than it was for the conjugate one, but that the overall η on the plate surface was higher for the latter. The calculation with the fixed wall temperatures had a coolant exit temperature comparable to that of the conjugate calculation, but η on the plate surface was lower. The normal procedure for the calculation of heat transfer coefficients therefore seems questionable in the case of effusion cooling. The influence of different high-conductivity base materials was looked at in a conjugate calculation by [47], who investigated 15 rows of inclined staggered cooling holes. The reference material was CMSX-4 with $\lambda = 22.535$ W/mK, which was compared with NiAl-FG75 with $\lambda = 52.2$ W/mK. It is noted by the authors that the increase in λ had an adverse effect on η in the first and last rows because more heat was transported from the non-cooled areas towards the measurement area. In the middle of the calculational area η was nearly the same for the two materials.

All in all it is safe to say that λ cannot be neglected for the proper assessment of effusion-cooled blades. The effects of convective cooling within the cooling holes and the plenum and the thermal conductivity itself are too large. For combustor liners, with their low-conductivity materials, adiabatic boundary conditions seem justified, though.

4. Simplified approaches

One of the main challenges with the simulation of effusion-cooling is the sheer amount of cooling holes that need to be modelled. Therefore numerous attempts have been made to extrapolate results from much smaller measurements and calculations for the prediction of the cooling effectiveness of effusion-cooling.

The adiabatic cooling effectiveness of a single hole and multiple rows of holes was measured with an infrared camera and compared with an additive superposition model in [48]. The model only gave good results for laterally fairly widely spaced holes and a region of approximately 3D downstream of the hole. In [49] a flat plate made from aluminium with short cooling holes was investigated experimentally. Both an in-line and staggered arrangement with $p/D = 4.8$ were looked at. The authors assessed the influence of the convective cooling in the holes and the thermal conductivity of the wall on the overall cooling effectiveness by transient tests in

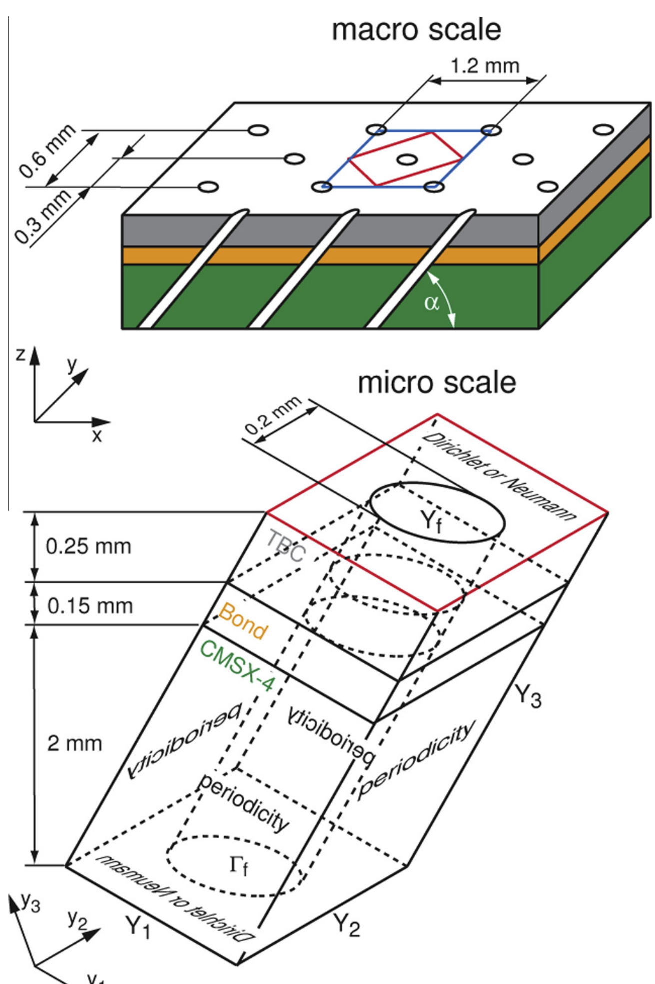

Fig. 13. Periodic reference cell as used by Laschet [57] and similarly by Mendez and Nicoud [6] after [60].

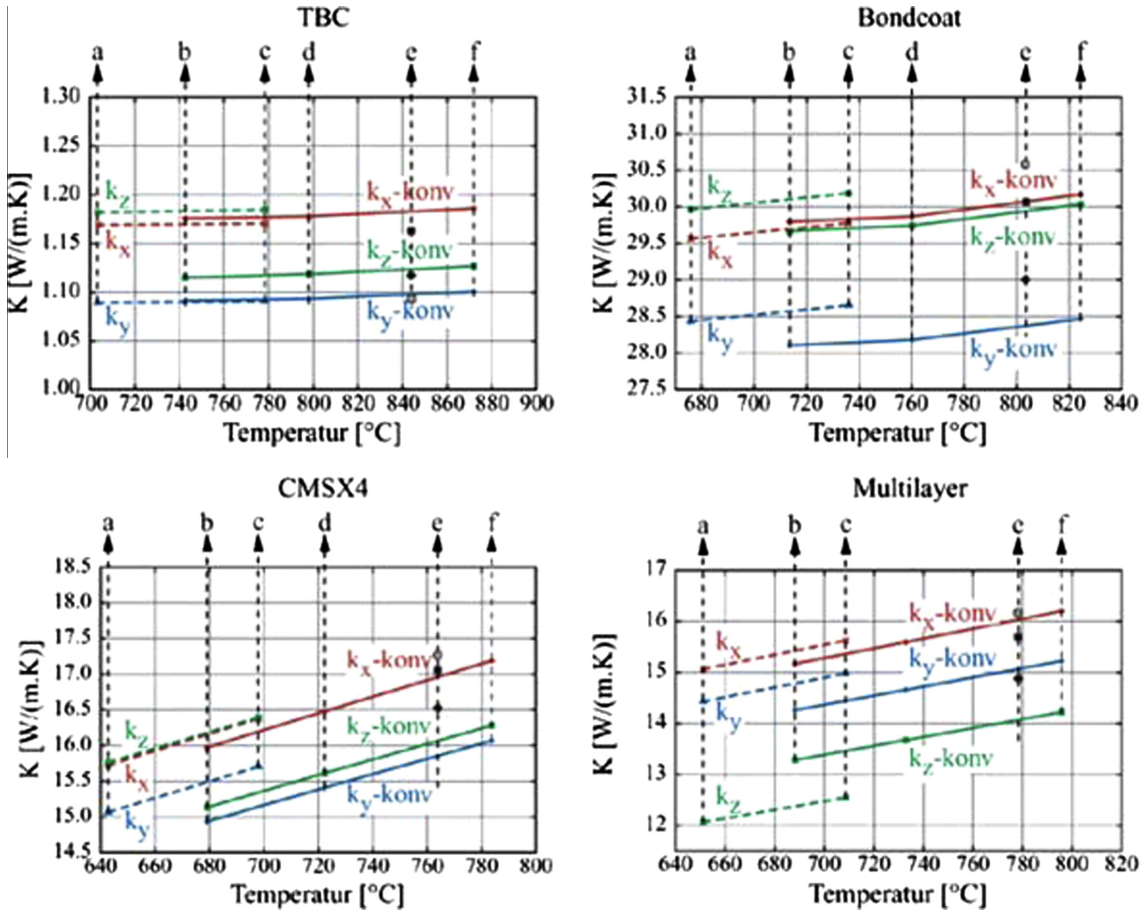

Fig. 14. Equivalent thermal conductivity of an effusion-cooled three-layer system of curved plates (A) concave – 0.48 M; (B) convex – 0.48 M; (C) flat – 0.48 M; (D) concave 0.28 M; (E) convex 0.28 M; (F) flat 0.28 M [57].

which the cooling of the aluminium was measured while cold air flowed over its surfaces. These effectivenesses were superposed both in streamwise and lateral direction. It is shown that the predictions can be fairly good for staggered arrangements when the interaction of the coolant with downstream holes was neglected. For an in-line arrangement the superposition predicted an η that was too high. Only when the effect of a cooling hole on the effectiveness was limited to the region 10*D* downstream of the hole, reasonable results were obtained. Superposition was also used in [50], where the leading edge region of a blade with 11 rows of cooling holes was looked at. Heat transfer coefficients were calculated on the basis of an experimentally determined adiabatic wall temperature. The inclusion of heat sinks at the position of the cooling holes proposed by [51] did not lead to satisfactory results. Recently [52] combined experiments with a coupled calculation of fluid and solid, also using superposition. Two experiments were performed, one with an adiabatic and one with a diabatic wall. A fairly good parabolic correlation between coolant mass flow rate and η could be established for the diabatic, but not for the adiabatic wall. The correlation derived with this approach and the superposition theory showed a good agreement with the experiments.

A different approach was chosen in [53–55]. The authors compared experimental data from a flat copper plate with three different hole angles: $\alpha = 90°$, and $\alpha = 30°$ with and without a compound angle of 45° all with $p/D = 5$. M was varied between 0.0 (i.e., uncooled) and 1.3, the temperature (density) ratio between 0.0 (adiabatic cooling effectiveness) and 1.0. In the latter case, the wall temperature attained the coolant temperature. The heat flux was lowest when $M \approx 0.5$. These results were the foundation of a numerical model in which an injection model was implemented and the turbulent mixing length adjusted accordingly. Apart from the transition region this model could calculate the downstream region of the first few rows of cooling holes fairly well. If M became too high, however, the program did not yield reliable results.

A purely numerical approach is discussed in [40,56,57]. Relatively small arrays of effusion-cooling holes are simulated using a conjugate-code. On this basis the equivalent material properties of the “porous” material are calculated. The authors based their calculations of the equivalent material properties on a periodic reference cell, an approach that was used later on by e.g. [6], see Fig. 13. In [57], this approach is applied to different curved plates. The evaluation of the results showed considerable differences between the plates (cf. Fig. 14). The equivalent properties can be back-implemented in a CFD-code, but this has not been done yet and the effectiveness of this technique thus remains unproven.

Yet another approach was chosen in the LES-based studies [6,39] touched upon previously. The authors used their LES-simulations to formulate sink-terms in the plenum and source terms on the hot gas surface. This only required coarse grids for the main flow, but the model required input about e.g., the pressure drop in the “porous” perforated plate. The correlation between the experiments and the most refined version of the model had a maximum error of approximately 20%.

Although the need for simplified models has decreased with rising computational power, such models would still serve their purpose for a preliminary design. Unfortunately, none of the models presented so far yields results of sufficient accuracy.

5. Effusion-cooled full blades

The previously discussed investigations all focus on important aspects of effusion-cooling, but they do not enable a safe application of effusion-cooling to turbine blades. The publications dedicated to the peculiarities inherent to turbine blades are discussed in this section.

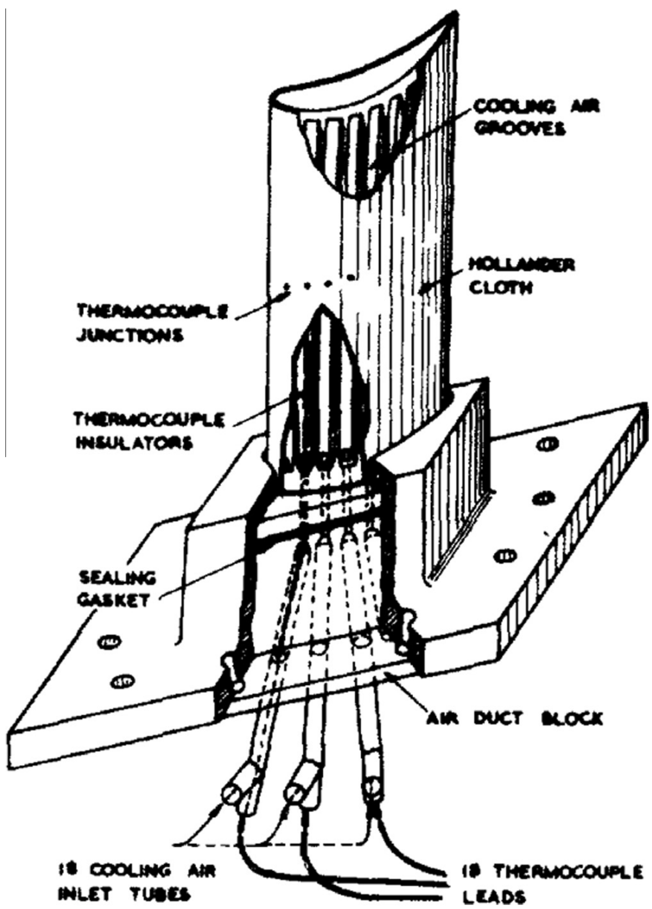

Fig. 15. Experimental set-up used in [23].

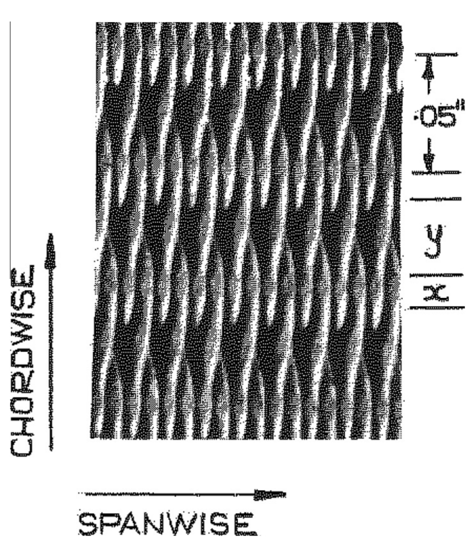

Fig. 16. Wire-mesh used in [23].

5.1. Early work on cooled blades

There are some early investigations on the topic of effusion-cooling and although they concentrated on "pure" porous materials rather than smallest holes, they already address issues that are still of primary concern today.

In [23] (the results of which have already been partially discussed above) the experimental investigation of an effusion-cooled turbine vane is discussed. The vane had a solid core made from steel, to which an outer contour made from a woven wire-mesh was brazed (see Figs. 15 and 16). The coolant could effuse from the pores between the steel wires from different coolant chambers. The experiments were conducted for $0.4 \leqslant Ma \leqslant 0.95$ and $250\,°C \leqslant T_h \leqslant 1000\,°C$. Furthermore, in one series of experiments the effusion-cooling mass flow rate was kept constant over the blade, whereas in a second series of experiments the surface temperature was kept constant. The authors noted a distinct influence of the effusion-cooling on the flow around the blade ad the resulting aerodynamic losses.

In [58], also an experimental study, effusion-cooling was achieved via porous materials. These materials and their lack of strength, alongside the experimental and analytical investigation of η, are the focus of the publication. One pertinent problem was the quantification of the cooling effectiveness that resulted from the inner convective cooling in relation to the cooling film on the surface. For an accurate prediction of the effectiveness of the effusion-cooling, an analytical solution based on algebraic equations was developed to calculate the amount of convective cooling in the porous material. The comparison of the analytical solution of the relation between convective and film-cooling with experimental data yielded excellent results and predicted $0.5 \leqslant \eta \leqslant 0.8$ for flat plates and $0.4 \leqslant \eta \leqslant 0.92$ for porous cylinders. Apparently the thermal conductivity was of less importance for the high-conductivity base-materials used, although it is said that the convective cooling contributed 60–90% of the cooling effectiveness. The possible clogging up of the cooling holes was an issue as well. Experiments showed that depositions constituted no problem, though, since the whole surface was shielded from the hot gas by the effusion-cooling film. The permeability of the effusion-cooled cylinder did decrease by almost 10%-points after approx. 150 operating hours. Since the testing temperature was well above the maximum operating temperature of the material, this was attributed mainly to oxidation.

5.2. Manufacturing and operation

A research group at RWTH Aachen University looked at some influences of manufacturing and operation on an effusion-cooled blade using an in-house conjugate flow solver. Consequently, they took the cooling effectiveness in the substrate layer for their evaluation of the different influences. The basic geometry was a plate with 7 rows of staggered, inclined cooling holes. The width was 2 times half a cooling hole, which was expanded to infinity with overlapping boundary conditions at the sides.

As was noted above, one of the most important dangers is clogging of the cooling holes. Although Grootenhuis [58] did not see any deterioration due to clogging, a poor quality of the cooling air may still lead to the closure of multiple cooling holes. In a first study, [59], the authors closed the middle of the 7 rows of cooling holes and found a substantial but not catastrophic deterioration of the cooling. When the middle hole out of an area with 15 holes was closed, this had virtually no impact on η, [60]. It therefore seems that clogging really is of little relevance for large arrays of holes, but a critical limit.

In order to investigate the influence of main flow incidence on the cooling effectiveness, the basic geometry was enlarged to 5

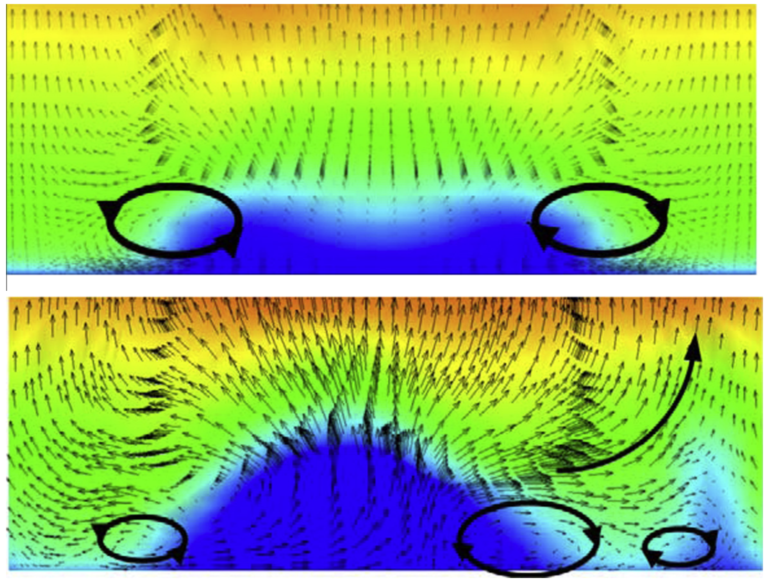

Fig. 17. Comparison of flow field after a symmetrical and asymmetrical hole for $M = 0.28$ [60].

holes in lateral direction [61]. The blowing ratios were between approximately $M = 0.26$ and $M = 0.44$. A substantial influence from the main flow incidence on the flow in the boundary layer, which could lower η by approximately 5–7%-points was recorded. This implies that (as far as contoured holes are concerned) incidence should be a design parameter.

It was already noted that in [37] curved plates were investigated alongside flat ones. The measurements showed that the boundary layer thickness increased dramatically for concave surfaces but less so for convex ones. This is confirmed in one of the very few numerical publications on effusion-cooling on a curved wall, [62]. The authors reported that η on the convex plates is lower than that of flat plates for the investigated geometry with $d = 180D$ and $D = 0.2$ mm. In other publications, [57,60], the authors consider curved geometries with $d = 240D$ for concave and $d = 350D$ for convex plates ($D = 0.2$ mm) and different streamwise and lateral distances of the holes. A reduction of the curvature lead to changes in the interaction of the cooling jets and the opposed tendencies of coolant lift-off and pressure gradients on the surface increased η considerably.

The other studies of this group are mainly relevant for contoured and or laser-drilled cooling holes. In one study they looked at the breaking-away of the TBC [63]. The authors state that this enabled a more even and earlier spread of the coolant effusing from the holes. In another study, [64], scans from real drilled holes were taken and these were incorporated in a simplified manner in the numerical grid. The results showed that the asymmetry introduced into the hole geometry had a direct influence on the cooling film on the hot gas surface (cf. Fig. 17). One dominant vortex developed and for $M = 0.28$ this increased η compared to the ideal geometry. For $M = 0.48$ it was lower than that of the ideal geometry due to vortex lift-off of the dominant vortex, which also lead to an increased transport of high-temperature fluid to the surface. Based on [65] the group stated that oxidation in the cooling holes had little (but a slightly positive) effect on the overall cooling effectiveness. It is difficult to generalize this statement due to some simplifications in the modelling and the fact that the oxidation was only present in the substrate–layer.

It should be said that some of these findings are very difficult to extend to more generic effusion-cooling configurations. Other results, like those on clogging or curvature, offer a basis for more detailed investigations.

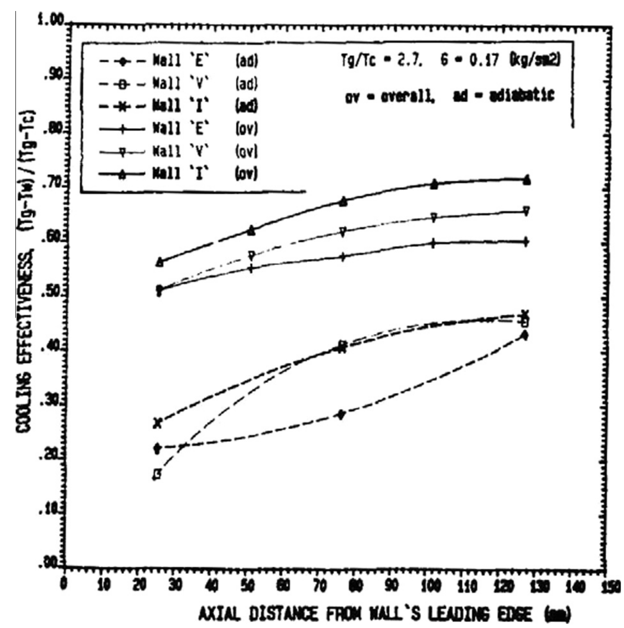

Fig. 18. Comparison of adiabatic and overall cooling effectiveness for an impingement- and effusion-cooled plate [69].

5.3. Impingement cooling

The advantage of combining film- or effusion- with impingement-cooling, i.e. the film-cooled plate is at the same time the target plate for the impingement jets, which considerably increases the overall cooling effectiveness. The commercially available Lamilloy is based on this concept [66]. One of the earliest investigations in this field is [67], which focussed on discharge coefficients for the impingement- and pressure loss coefficients for the effusion-cooling of a vane. The authors showed that both can be correlated with *Re* and *Ma*, but the latter was preferred because it gave a common point in all tests at the choked flow condition. Research groups at Leeds University, e.g., [68,69] put the investigations on a broader (theoretical) basis. More recently a group at Yonsei University dedicated a long series of combined numerical and experimental publications to a broad variety of problems related with this topic, cf. for example [70–73]. Both groups showed that the general characteristics of the cooling do not differ too much from "normal" impingement cooling and stated that the small size of the cooling holes made some difference.

The experimental investigation described in [68] focussed on the optimum number of impingement-cooling holes for a given number of effusion-cooling holes. Furthermore, the heat transfer coefficients due to both the effusion-cooling and the recirculating flow in the gap between impingement- and target plate were calculated. The hole diameter was changed for a constant number of cooling holes and therefore an increase in diameter equalled a lower coolant exit velocity and hence a better cooling effectiveness. An optimum was found when the number of holes for both cooling techniques was the same and the impingement-cooling holes were exactly in the middle of four effusion-cooling holes (i.e. there was exactly half-a-pitch distance in both streamwise and lateral direction between the impingement and effusion-cooling holes). In another excellent experimental investigation, [69], the same group varied the number and size of the effusion-cooling holes. For $M \leqslant 0.5$ the number of holes had no noticeable influence on η, but a significant dependence existed when M rose above this value due to the lift-off of the effusion-cooling jets. There was a noted decrease in the difference between the overall and the adiabatic cooling effectiveness with increasing M. This could be explained with the importance of the convective cooling in the space between the impingement- and effusion-cooling plates. In this particular case the application of impingement-cooling to an effusion-cooled surface increased η with nearly 10%-points to approximately 0.7, as can be seen in Fig. 18.

In [70] the findings of [68] were confirmed. The experimental and numerical set-up of [70] consisted of three configurations of the cooling holes, one staggered, one shifted (one plate offset half a pitch to the other) and one in-line arrangement of the cooling holes. The distance between the plates was either 1, 2 or 3D for D = 25.4 mm. The quantitative differences between the experimental and numerical results were sometimes substantial, but the qualitative agreement was quite good. Both the staggered and shifted arrangements substantially increased the heat transfer compared to the in-line arrangement, especially around the effusion-cooling holes and where the coolant jet impinged on the wall. In [71] the effect of the crossflow in the gap between impingement and target plate was explored in a combined experimental numerical study. The authors note that when the ratio between the crossflow mass flow and the total cooling mass flow became larger than 0.5, the effectiveness of the impingement was seriously reduced. Based on measurements of the Sherwood-number, they state that the combination of the two cooling techniques yielded an effectiveness that was 20–30% higher than a simple impingement-cooling and 3.9–9 times higher than convective cooling. Correlations for the calculation of the Sherwood number in dependence of the massflow ratio were derived. The same group suggested introducing fins into the space between the impingement- and effusion-cooling plates to increase the heat transfer [72]. Furthermore, they looked at the effects of rotation on the flow in the gap between the two plates [73].

The combination of impingement- and effusion-cooling has been investigated more thoroughly than most other aspects of effusion-cooling and therefore seems to be understood fairly well.

5.4. Full blades

Numerical investigations of entire blades with film cooling configurations have been presented from the late 90's onwards. Effusion-cooled blades have many more holes and calculations of full blades are therefore still virtually impossible. Some experimental studies or studies that focus on a part or slice of a full blade do exist, though.

The large influence of effusion-cooling on the flow around the blade noted by previous studies is confirmed in [74] (a summary of earlier experimental work by the same group). They showed that if a cold-air turbine is fitted with effusion-cooled blades, the aerodynamic efficiency of the blade can drop by up to 10%-points; a wire-mesh vane is only slightly worse than a full-coverage film-cooled one.

In the, also experimental, studies [75,76] blades that consisted of a solid inner core and an outer contour that were both made from X5-CrNi-188 are discussed. The outer contour had numerous cooling holes with D = 0.2 mm and $\alpha = 30°$. This angle was increased to 60° and then 90° near the leading edge. The pressure of the coolant was adapted to the outer pressure distribution along the blade. The cooling passages were shaped to ensure a constant M over the whole blade height. Both the temperatures in the core and on the blade surface were measured. The results of [75] show that η increased with increasing blowing ratio until $\eta \approx 0.7$ was reached with a cooling mass flow of 3% of the main mass flow. After this, the cooling effectiveness remained more or less constant. When Ma increased but the temperatures and M remained constant, η increased as well. This (partially) contradicts the work on flat plated discussed before. The coolant injection in the boundary layer had an effect only in the region of the strongest curvature on the suction side. In line with the findings of [74], the authors of [76] state that the profile loss coefficient was nearly 9% for a coolant mass flow of 3.75% of the main flow and increased almost linearly with increasing coolant mass flow. The results from these experiments were extrapolated to temperatures and pressures relevant in the late 1970's.

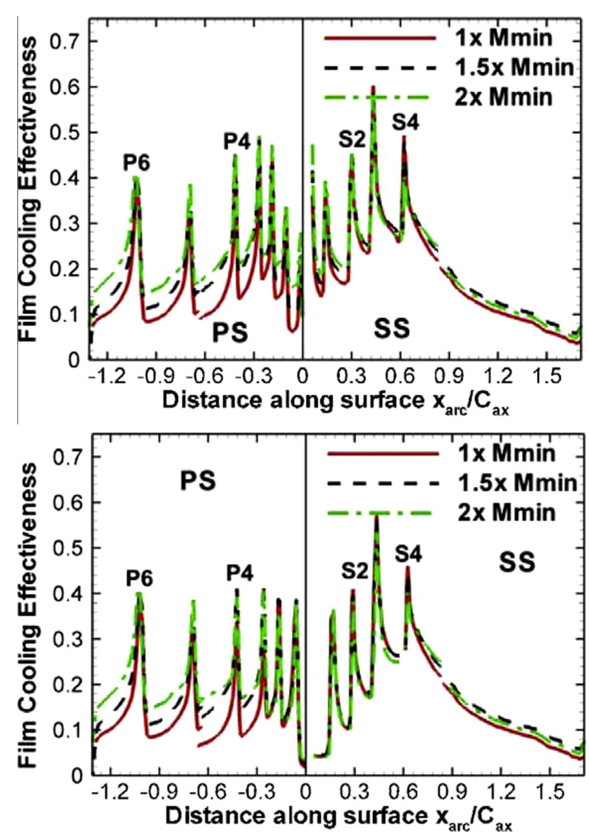

Fig. 19. Adiabatic film cooling effectiveness on a blade top: with showerhead cooling, bottom: without showerhead cooling [77].

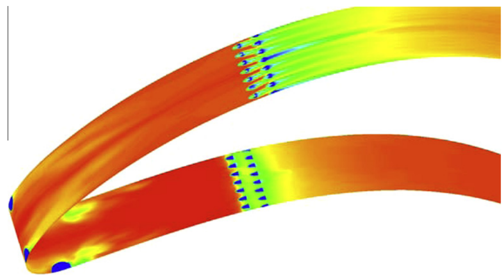

Fig. 20. Film- and effusion-cooled blade showing the different cooled areas on the pressure and suction side [78].

An experimental investigation of a effusion-cooled blade with six rows of cooling holes on the pressure side and four rows of cooling holes on the suction side and approximately 24 cooling holes for each row was presented in [77]. In order to have a basis for comparison, both a blade with and a blade without showerhead cooling in the leading edge were investigated. As with film-cooling, the coolant remained attached to the suction side and individual traces of coolant could be observed here. On the pressure side the coolant traces were very short and smeared out, as is usual with film-cooling. Furthermore the coolant was deflected towards the mid-span, especially on the suction side, which resulted in a decrease in η. Introducing showerhead cooling lowered the temperatures around the leading edge, but did not influence the rest of the blade. The authors report that an increase of $Ma = 0.36$ to $Ma = 0.45$ did not affect η as long as M was kept constant (cf. Fig. 19).

A numerical study on a combined film-/effusion-cooled blade was provided in [78]. The numerical grid included the coolant plena and consists of 4 half cooling holes in the leading edge and 14 effusion-cooling holes divided over two rows on both the pressure and the suction side as depicted in Fig. 20. The authors focussed on the interaction of film- and effusion-cooling. The counter-rotating vortices from the film-cooling negatively affected the development of the effusion-cooling. Because of its (previosuly noted) effect on the boundary layer thickness the effusion-cooling influenced the film-cooling as well.

All in all the effects of effusion-cooling on full blades have neither been investigated very thoroughly nor are they understood very well. Simply combining the body of basics research on effusion-cooling of more basic geometries does not suffice. The bulk of the future work should therefore be conducted in this field.

6. Conclusions

Effusion cooling has been investigated for nearly 60 years now, but has not been commercially applied to the cooling of gas turbine blades yet. The scientific basis for such an application has been laid, though. The fundamental development of the cooling film, the influences of Mach-number, turbulence and thermal conductivity of the base material, density and velocity ratios, i.e., all aspects one would expect to be relevant on the basis of the vast body of work on film-cooling, have been investigated thoroughly. Contradictory reports on some of these aspects exist, but it is likely that these can be explained with the different boundary conditions in the tests and simulation that are necessary due to the complexity of the flows and the amount of (interacting) cooling holes. This complexity is also the reason why simplified modelling approaches have not met with great success so far.

Still, a considerably amount of work has already been done on very specialist subjects like the combination of impingement- and effusion-cooling or the influence of operational influences like clogging on effusion-cooling. Studies of full blades, which previously were only possible under laboratory conditions on test rigs with all the associated drawbacks in terms of boundary conditions, can now be performed using CFD. Even though CFD shows weaknesses in the prediction of detailed flow phenomena, it has been proven that the overall cooling can be predicted rather well. Furthermore, LES offers a considerable potential. Therefore an application of this technology to real machines seems now more likely than ever.

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