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IMPROVED TRENCH FILM COOLING WITH SHAPED TRENCH OUTLETS

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ABSTRACT

This paper presents the influence of the shaped trailing edge of trench outlets on film cooling effectiveness and aerodynamics. A 90^o outlet wall to a trench will give a vertical slot jet into the cross flow and it was considered that improvements in the cooling effectiveness would occur if the trailing edge of the trench outlet was bevelled or filleted. CFD approach was used for these investigations which started with the predictions of the conventional sharp edged trench outlet for two experimental geometries. The computational predictions for the conventional sharp edged trench outlet were shown to have good agreement with the experimental data for two experimental geometries. The shaped trailing edge of the trench outlet was predicted to improve the film cooling effectiveness. The bevelled and filleted trench outlets were predicted to further suppress vertical jet momentum and give a Coanda effect that allowed the cooling air to attach to the downstream wall surface with a better transverse spread of the coolant film. The new trench outlet geometries would allow a reduction in film cooling mass flow rate for the same cooling effectiveness. Also, it was predicted that reducing the coolant mass flow per hole and increasing the number of holes gave, for the same total coolant mass flow, a much superior surface averaged cooling effectiveness for the same cooled surface area.

INTRODUCTION

Cooling is crucial to the operation of modern gas turbines as it operate at elevated temperatures much higher than the permissible material temperatures of the components that envelop the hot-gas-path. Moreover, trends have shown that future high performance turbines will continue to operate at increasing elevated temperatures. Thus, better film cooling scheme will be required to protect the combustors and turbine blades and to reduce the thermally induced maintenance cost of this components. Film cooling has three main components: back side convective cooling; internal passage convective cooling and film cooling. Coolant air for the film cooling is bled from the compressor and flows through the metal passages to form a film of cold air between the metal and the hot gas crossflow. For combustors, film cooling does not have any adverse effect on the cycle thermal efficiency, but for turbine blades the compressor air bleed has not been heated by the combustion and therefore there is a loss in efficiency as the expansion across the turbine blade is reduced.

This loss in cycle efficiency increases as the film cooling mass flow increases. However, the gain in efficiency, from the ability to operate at higher turbine inlet temperatures, leads to an overall cycle efficiency gain. Unfortunately, this blade cooling air makes the combustor operate richer with a higher NOx [1] and in the limit the combustor will approach stoichiometric before the overall engine equivalence ratio is stoichiometric (which is the highest thermal efficiency condition). Eventually the total mass flow of coolant used for blade film cooling will become the limiting factor in reaching the highest cycle thermal efficiency. Therefore, reducing this film coolant mass flow is beneficial for maximum cycle efficiency, and in combustors, for low NOx emissions. There are two ways of reducing the coolant mass flow rate: improving the effectiveness of film cooling through the hole outlet shape design and improving the effectiveness with which coolant mass flow is utilised, which relates to the hole array design.

Early work on film cooling concentrated on the effectiveness of the film cooling [2-14]. Inclined cylindrical hole injection was introduced to reduce the jet momentum normal to the cooled wall surface, as a means of improving the film cooling performance. The majority of these studies on film cooling utilise 30^0 or 35^0 injection angles.

Shaped film cooling hole outlet geometries were developed [4, 15, 16] to improve the surface coverage of the coolant jet surface film. Shaped hole geometries help to keep the coolant close to the hot-wall-surface and this improves the cooling effectiveness, or reduces the coolant mass flow required for the same cooling effectiveness. This is typically achieved by giving the injection hole exit a different area and geometry from the metering hole inlet that controls the coolant flow rate. Goldstein et al. [4] reported that film cooling effectiveness is independent of the blowing rate, M in the near hole region, and increases with M downstream of the holes.

The addition of a compound angle (CA) to the baseline cylindrical holes and shaped or expanded cylindrical injection holes has been experimentally investigated by Schmidt et al. [17] and Sen et al.18]. Their studies were motivated by a previous computational study by Sathyamurthy and Patankar [19] who had limited experimental data with which to validate their computational results. The addition of the CA to the baseline cylindrical inclined holes significantly improved the lateral effectiveness [17]. The combination of CA with the 15^0 forward expansion holes provided further improvement in the lateral cooling performance [17].

Recently, studies have shown that a film cooling geometry in which the coolant exits from holes embedded in a transverse trench significantly improves the film cooling effectiveness [20-24]. The interaction between the inclined film cooling hole with the downstream trench wall spreads the coolant in the transverse direction and this is the key cause of the improved transverse film cooling effectiveness. The next level in turbine cooling technology is to combine film cooling with thermal barrier coatings (TBC) [25]. The layers of TBC could be as thin as the order of the film-hole sizes. Thus, the coating may be applied on a masked hole area to form the shape of the trench. The thermal barrier coating also could be applied to give a bevel or fillet outlet to the trench and the effect of this has been investigated in the present work.

Bunker [20] investigated discrete holes embedded in a transverse trench. The investigation was motivated by the previous work conducted by Wang et al. [21], which reported a uniform velocity distribution at the outlet of the trench fed with discrete coolant holes. Waye and Bogard [22] experimentally investigated the effects of an overlaying transverse trench on the film cooling effectiveness of axial holes. They examined nine trench configurations, in addition to the baseline configuration with no trench. They also used inserts into the trench in the upstream and downstream faces of the trench of rectangular and triangular (bevel) shapes to form the lips of the trenches. The triangular hole inserts had an angle of 30^o, the same as the film cooling hole.

All nine trenched configurations examined by Waye and Bogard [22] had a total trench depth of 0.5D. It was shown that the configuration with a rectangular insert situated both up-stream and down-stream of the trench (narrow trench) provided the best laterally averaged film cooling effectiveness. They concluded that the key component of the trenched configuration to improve the film cooling effectiveness was a rectangular lip immediately downstream of the trailing edge of the coolant hole. For the triangular insert there was no vertical wall to the trench at the hole outlet and this resulted in little benefit in film cooling effectiveness compared with the baseline with no trench. Dorrington et al. [28] experimentally investigated the effect of geometrical design parameters on the trenched and crated outlet. Their results were mainly presented in terms of spatially average effectiveness. Similar to the findings of Waye and Bogard [22], they showed that the narrow trench had a better cooling performance. Also, they showed that a trench of depth 0.75D was the optimum trench depth for blowing ratios ranging between 0.5 to 1.5 and that slanting the trailing edge of the trench resulted in degradation of the cooling performance.

Lu et al. [23] further investigated the effect of the trench depth on film cooling performance for two trench widths. A CFD simulation was also used to show the jet-mainstream interaction and the distributions of both the film cooling effectiveness and heat transfer coefficient. They examined trench depths of 0.5D, 0.75D and 1.0D at 2D and 3D trench widths. They showed that the optimum trench depth was 0.75D for both widths. Also, Lu et al. [23] results showed for the narrow 0.75D trench depth that there was a significant trench wall vertical velocity that produced a lifted jet at the trench exit compared to the wide 0.75D trench depth configuration. This lifted jet is a detrimental feature. The present work aimed to show how this configuration could be improved on by using either fillet or bevel outlets to the trench.

Jia et al. [24] experimentally investigated the effect of coolant jet compound angle on the cooling effectiveness of the trenched configuration. They examined six configurations: baseline 30^0 inclined hole configurations with and without the 45^0 and 90^0 CA, and trenched configurations with and without the 45^0 and 90^0 CA. Their study showed that the 45^0 CA holes embedded in a transverse trench gave the highest overall film cooling effectiveness.

Oguntade et al. [26] and Harrison et al. [27] used computational fluid dynamics to determine if the dramatic improvement in the film cooling effectiveness from the use of trenched holes configuration is predictable. The success of their studies showed that CFD can be used as a tool for studying and identifying promising film cooling configurations. The present work uses computational approach to explore further improvements in the trench film cooling using changes in the geometry of the downstream face of the trench. The computational approach was validated [26] for inclined holes against the classic film cooling data of Sinha et al. [13]. The present work for trench film cooling has been validated for a straight edged trench outlet by comparing with the results of Waye and Bogard

[22] for a 0.5D straight walled trench and with the results of Lu et al. [23] for a 0.75D straight walled trench

This work used CFD to guide improvements of the film cooling performance of the trench outlet geometrical designs. The effect of bevelled or filleted trailing edge of the trench outlet on film cooling performance was investigated. The addition of the bevel or fillet was in the last 0.25D of the 0.75D overall trench depth. The aim of the bevelled and the filleted trench outlet was to further suppress the vertical velocity component of the exiting jet from the trench so as to improve the film cooling effectiveness. The experimental data of the authors [22, 23] for the convectional sharp edge trench outlet form the base cases for validation of these CFD procedures and reasonable good agreement was achieved the base cases. Bevelled or filleted trailing edge of the trench outlet is found to improve cooling performance.

EXPERIMENTAL DATA USED FOR THE CFD VALIDATION

The experiments of Waye and Bogard [22] and Lu et al. [23] were the data base for validation of the CFD procedures established for conventional inclined holes [26] applied to the conventional sharp edge trenched configuration. Waye and Bogard [22] used a trench depth of 0.5D and Lu et al. [23] found that 0.75D trench depth was the optimum configuration for the best cooling effectiveness and this was modelled in the present work. There was a difference in hole size between the two studies and the two P/D ratios were 2.775 [22] and 3.0 [23]. There were also significant differences in the inclined hole length to diameter ratio with the trench in place, L/D, with 5.7 used by Waye and Bogard [22] and 2.25 by Lu et al. [23]. This difference will result in different hole outlet velocity distributions, which will influence the interaction of the jet with the crossflow.
An aerofoil hollow blade was used by Waye and Bogard [22] scaled up nine times with the coolant flow entering the plenum parallel to the hole feed, but with a limited plenum volume. Lu et al. [23] used a conventional flat plate film cooling configuration with a large plenum volume and an inlet parallel to the crossflow. In the experiments of Lu et al. [23], the x = 0 location was defined as a point closer to the upstream edge of the coolant hole. Thus for the trench of depth 0.75D trench, the x = 0 location is 16.498mm axial distance downstream of the trench leading edge. These significant geometrical differences in hole and plenum chamber designs resulted in separate CFD validation, in spite of the small differences in hole geometries. The experimental results showed a very significant difference in the axial variation of transverse average cooling effectiveness for these two investigations. These differences are likely to be to a significant effect of the geometrical differences between the two test facilities.

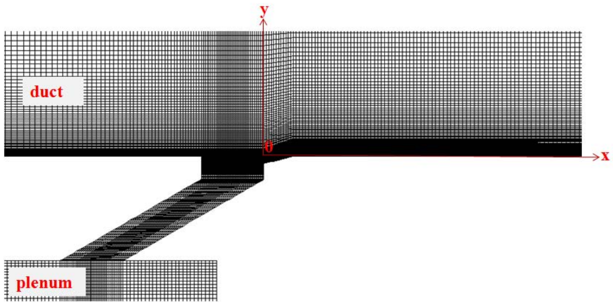
The present work explores the effect of bevelled and filleted downstream trench outlet on the cooling performance. The experimental data of Waye and Bogard [22] and Lu et al. [23] form the base for the CFD validation of the baseline cases. The region of interest in the experiments [22] was the suction surface of the vane which was relatively flat. Thus, the computational studies have been conducted on a film cooled flat plate and it is a simplified replica of the experimental configuration.

COMPUTATIONAL PROCEDURES

The computational procedures and the justification for their use are those set out by the authors [26] and validated against the inclined hole film cooling experimental data of Sinha et al. [13] and Pederson et al. [6]. The geometries and meshing were generated using GAMBIT and the steady state simulations were performed using the FLUENT code. A hexahedral meshing system was used for the coolant supply plenum, duct and injection hole. In the previous work [26], the CFD results are shown to be sensitive to the applied turbulence model and the realizable k-ε does a better job at predicting the laterally average and local effectiveness distribution [26] than other RANS models and the Spalart Allmaras model. The enhanced wall treatment option is used to resolve the viscosity affected inner region of the turbulent boundary layers. Consequently, the meshes were clustered around the critical regions of the test wall surface where the flow parameters have a large gradient. All the computations used adaptive meshing which continued to increase the number of cells and nodes until there was no change in the output.
The computational domain consisted of the coolant supply plenum, duct and injection hole. Figure 1 illustrates the computational grid before adaption for the 15^0 bevelled trench outlet geometry. The dimension of the 3D computational domain in length, height, and width were 51D, 13.35D and 1.38D, respectively. The grid for all the simulations were initially about 0.7million cells and this increased to approximately 2million cells (on the average) after the final adaption of the grid. The final value of $y+ \approx 2$ was obtained for all simulations. Moreover, it is important to have a value of $y+ \leq 4$ so that the first near-wall node is placed in the laminar sub-layer. Also, a value of $y+ \leq 4$ indicates that the flow gradients are well resolved in the viscous sub-layer [29, 30, 31].

A grid independence check could be conducted in two ways. One is by using different grid sizes to identify the minimum grid size required for a minimal change in the results obtained with an increase in the number grids – this is more suitable for non-wall bounded turbulent flow. The other way is through grid optimization and grid adaption using the pre-processor and the solver respectively and the flow results obtained will be independent of grid if there is a negligible change between flow results obtained from two successive adaptions of an optimized grid. The later approach is more suitable for wall bounded turbulent flow where the region of interest is close to the wall and this was used in this study.

The exact operating conditions in the experiments were specified at the inlets of the computational domain with the assumption of a uniform velocity profile at the duct and plenum inlets. In the computations performed to modelled the experiments of Waye and Bogard [22] and Lu et al. [23], the duct inlet were respectively at 10D and 19D upstream of the hole centreline outlet (without trench). The upstream mean velocity of the hot gas used for the simulations is the same as the reported velocity measured at the suction side of the vane [27]. The duct flow upstream boundary conditions remain fixed through the computation and matched that reported in the experimental test conditions, as shown in Table 1. The state of the test and duct wall surfaces was not stated in the experiments and was assumed to be perfectly smooth in this study. The plenum coolant temperature was set to 231K and the velocity was varied to simulate different blowing ratios range M = 0.35 to 1.4. A SIMPLE scheme solution method with Green-Gauss Cell Based and Second-Order-Upwind discretization scheme was used. The convergence criteria for all residuals were set to be 10^{-5}, except for the energy residual which was set to be 10^{-6}. The net mass flow rate imbalance for all simulations was less than 0.002%. The simulations were performed using a computer capacity of 3GHz dual core processor, 8GB of RAM and 64 bit operating system. Convergence of the initial grid for all simulations was achieved after approximately 3700 iterations and convergence of subsequent grid adaption was achieved after approximately a further 200 iterations.



(a) The centreline plane view

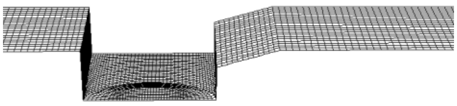

(b) Close-up view of a shaped trench outlet

Fig. 1 View of the computational grid for the 15^0 bevelled trench configuration

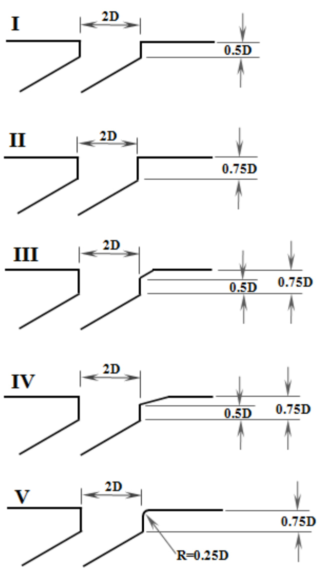


Fig.2 Trenched outlet configurations

HOLE CONFIGURATIONS INVESTIGATED

The geometrical and flow conditions used in the experiments [22, 23] and the computations are summarized in Table 1 and Fig. 2. Five cases of the trenched holes outlet configurations were investigated using CFD with a trench width of 2D. Cases I and II have a conventional sharp edge trenched outlet with a trench depth of 0.5D and 0.75D, respectively. Cases III, IV and V have a total trench depth of 0.75D and a vertical face depth of 0.5D prior to the start of the bevel or fillet outlet. Cases III and IV (bevelled trench configurations) have a bevel slope angle of 30^0 and 15^0 respectively. Case V has a fillet edge at the trailing edge of the trench outlet with fillet radius, R, of 0.25D.

Table 1 Geometrical and flow parameters used to simulate the experimental data

Reference	Waye [22]	Lu [23]
D	4.11 mm	12.7mm
P/D	2.775	3
Trench width, w	2D	2D
Trench depth, d	0.5D	0.75D
Injection angle	30^0	30^0
Test wall thickness	13.77mm	23.81mm
L/D (with trench)	5.7	2.25
N	8	6
DR	1.3	1.08
T_g	300K	321K
T_c	230.8K	296K
u_g	30.82m/s	13.8m/s
Re	8078.5	9908.4
T_u (duct inlet)	1%	2%
T_u (plenum inlet)	2%	2%

RESULTS AND DISCUSSIONS

Validation of the CFD Techniques

The predictions of Case I were validated using the experimental data of Waye and Bogard [22] and for Case II by using the experimental data of Lu et al. [23]. Comparison of predictions and experiments for Case 1 are presented in Fig. 3. The predicted laterally averaged adiabatic film cooling effectiveness for all blowing ratios in Fig. 3(a-c) demonstrated reasonable good agreements with the experimental data. However, at low blowing ratios the predictions show better agreement with the experiment than at high blowing ratios. It has been shown previously [26, 27] that film cooling is better predicted when the coolant jet is attached to the wall surface than for the lifted jets that occur at higher blowing ratios and consequently, a higher coherent structure of the jet in cross flow, characterized by high unsteady and anisotropic behaviour of the flow field.
The predictions of Case II using the experimental results of Lu et al [23] are shown in Fig. 4a. Reasonably good agreement between the predictions and experiments are shown in Fig. 4a. In the latter validation case, the predictions at low blowing ratio were not as good as the

agreement with the experiments obtained for Case I. However, there was a difference in trend of the lateral averaged cooling effectiveness between these two experimental data used for validation, as shown in Fig. 4b.

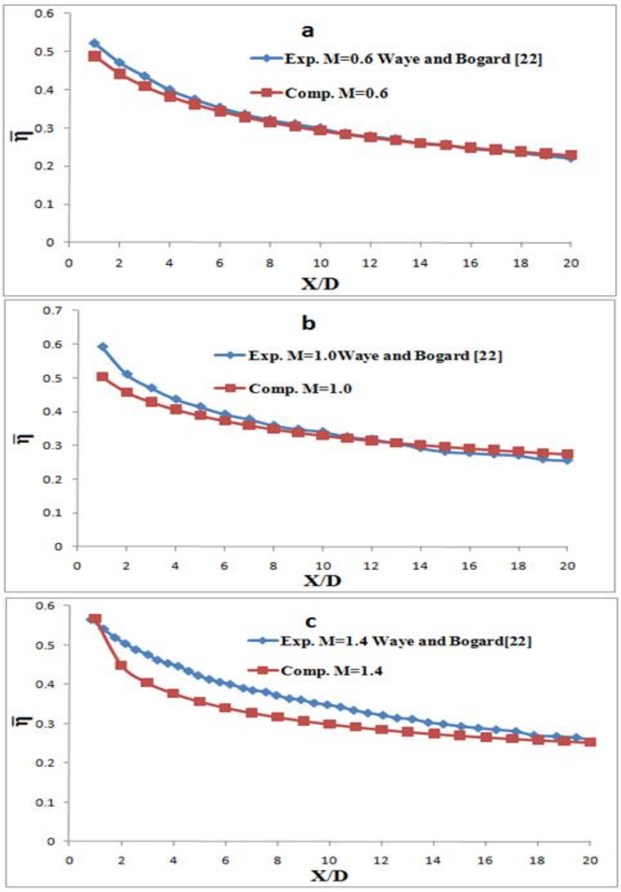


Fig.3 Comparison of the computed and measured data of the lateral averaged film cooling effectiveness for blowing ratio of: (a) M=0.6; (b) 1.0; (c) 1.4

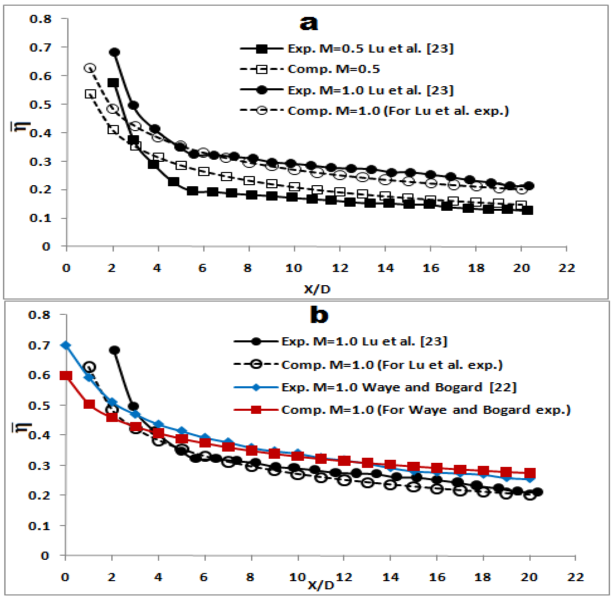


Fig.4 Comparison of lateral averaged film cooling effectiveness between: (a) Computed and measured results for Case II; (b) Experimental data

Lu et al. [23] for the same M = 1 measured higher cooling effectiveness in the near hole region but for an X/D = > 3 the cooling effectiveness was lower relative to Waye and Bogard [22] data. In addition to the differences in the trench depth used by Lu et al. [23] and Waye and Bogard [22], the difference in trend of the cooling performance between these two cases shown in Fig. 4b, could be partly due to the differences in the way the coolant air was fed into plenum or the differences in the L/D. There was also a density ratio difference between the two results.

In the computations, the exact coolant plenum chamber shapes were not reproduced, as insufficient detail was given. Thus, a rectangular coolant-air inlet that spanned the width of the plenum was used to reduce the difficulties associated with meshing the domain. Also, the plenum coolant air inlet for the prediction and the experiment were on the same side of the plenum. The predictions do show a steeper fall in cooling effectiveness for the geometry of Lu et al. [23], but not as steep as in the experiments. The difference in trend of the lateral averaged cooling effectiveness between the two experiments was well predicted as shown in Fig. 4b.

Comparison of the differences between the 0.5D and 0.75D conventional sharp trench cooling effectiveness results in Fig. 5(b) with that of fig.4 (b) shows that Fig. 5(b) has higher difference. This differences could partly be due to the difference in the DR, L/D, the location x=0, and the way the plenum coolant air was fed. However, Lu et al. [23] found no improvement in the adiabatic cooling effectiveness with an increase in the trench depth from 0.5D to 0.7D. Meanwhile, the present computational findings and those of Dorrington et al. [28] experimental findings show a significant improvement in cooling performance when the trench depth is increased from 0.5D to 0.75D.

Influence of the Shaped Trailing Edge of the Trench Outlet on the Film Cooling Effectiveness

The predicted results shown in Fig. 5 - 10 were all carried out at a density ratio of 1.3 over a range of blowing ratios from 0.35 to 1.4 for the base geometry of Waye and Bagard [22]. The benefit of the shaped trailing edge of the trench outlet on the film cooling effectiveness is presented in Figs. 5 to 8. Figure 5 presents the lateral averaged adiabatic film cooling effectiveness results for the conventional and shaped trench outlet configurations. It was shown in Fig. 5 that the conventional sharp trench with depth of 0.75D (Case II) had a superior laterally averaged adiabatic film cooling effectiveness to that of 0.5D (Case I). This improvement in the cooling performance with the increases in the narrow trench depth from 0.5D to 0.75D for the blowing ratios investigated is consistent with the experimental findings of Dorrington et al. [28]. However, this finding is inconsistent with the experimental findings of Lu et al. [23]. At M = 0.5, Lu et al. [23] found that the 0.75D narrow trench had an inferior cooling effectiveness to that of the 0.5D narrow trench and the baseline case. Also, they found the same cooling performance for the 0.75D and 0.5D narrow trench at M = 1.0 and 1.5.

The addition of 30^0 and 15^0 bevels or fillet in the last 0.25D of the 0.75D overall trench depth was predicted to give a further improvement in the laterally averaged adiabatic film cooling effectiveness for M of 0.6 and 1.0, as shown in Fig. 5. The significance and percentage of this further improvement will be discussed later with the aid of the local lateral distribution shown in Fig. 10. As for M of 1.4, shown in Fig. 5c, both bevelled trench cases and the 0.75D conventional trench had equal cooling effectiveness up to about X/D=3. Beyond X/D=3, the 0.75D conventional trench had a superior lateral averaged film cooling effectiveness. The fillet outlet had a superior cooling performance up to about an X/D=6 for all blowing ratios. For X/D above 6, the 0.75D conventional trench and all shaped trench cases had equal laterally averaged film cooling effectiveness at all blowing ratios, with the exception of the reverse trend in the bevel effect at M=1.4. However, the experimental data of Dorrington et al. [28] demonstrated a degradation of cooling performance with the chamfered (45^0 bevel at the trench downstream wall) trench configuration compared to the 0.75D sharp trench. The key feature of the trench film cooling is the obstruction of the coolant jet from the holes by the downstream vertical trench wall [22]. It is possible that the downstream vertical trench wall of depth 0.375D for the chamfered trench configuration is insufficient to obstruct the coolant air when ejected from the holes. The degradation of the cooling performance of their chamfered trench configuration could probably be due to the jet hitting the chamfered wall and worsen the situation of the jet.

The results shown in Fig. 5 indicates that the improvement in the shaped trench outlet film cooling effectiveness is in the near hole region and is more pronounced for low blowing ratios with typical jet attachment for cylindrical film cooling holes without a trench. The momentum of the vertical slot jet exiting the trench is attenuated by shaped the trailing edge of the trench. Also, the lateral spread of the slot jet at the exit of the trench is intensified due to the Coanda effect as the jet flows along the shaped trailing edge of the trench. Fig. 6 shows that the increases in cooling effectiveness is much more significant at M = 0.35 - 0.6 compared to at M = 0.6 - 1.4. At high M, the trench will only suppress the jet separation and not the total eradiation of jet separation. The laterally average cooling effectiveness is plotted in Fig. 6 as a function of the blowing ratio, M, for a range of X/D from the hole exit. This shows that increasing blowing ratio M, which increases the coolant mass flow rate G, results in an improved cooling effectiveness until the point at which the jet lifts off from the surface, which is M >0.6. Further increases in M and the mass flow rate do not improve the cooling effectiveness, apart from at large X/D >5.

These results show that if increased cooling effectiveness is required for the optimum hole outlet configuration V, then increasing the coolant mass flow and M will not be effective. A more effective approach will be to increase the number of holes and for a fixed value of G and D, the value of M will be low for each hole. This is effusion cooling where only the near hole region is important as the X/D is rarely above 10.

The benefit of keeping the blowing rate per hole low, and increasing the number of holes, is shown in Fig. 7. This shows the comparison of the cooling effectiveness for a surface area X/D = 10 by Z/D = 2.8, using the same coolant mass flow rate for one film hole and for 4 holes with a blowing rate per hole of ¼ that of the single hole.

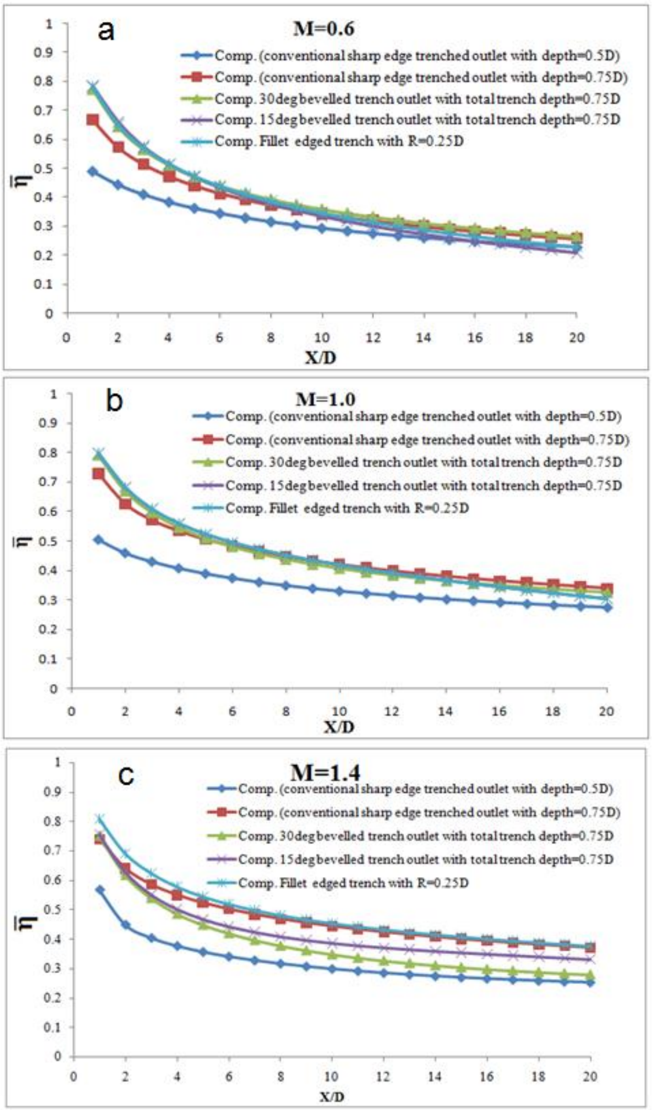


Fig. 5 Influence of the shaped trench outlet on film cooling effectiveness for (a) M=0.6, (b) M=1.0, and (c) M=1.4

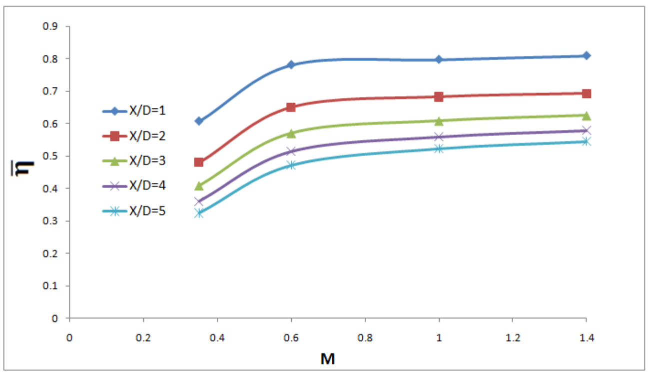


Fig. 6 Laterally averaged adiabatic film cooling effectiveness as a function of m at a range of x/d from the hole exit for Case V

These results essentially compare the cooling effectiveness for one hole at M=1.4 in Fig. 5c for configuration V, with four holes at M=0.35, but only cooling up to the X/D=2.5 distance from the hole. Figure 7 assumes that there would be no influence of the upstream film on the subsequent films and this is not correct, but the interaction is beneficial to the film cooling so that the prediction in Fig. 7 is a worst case.

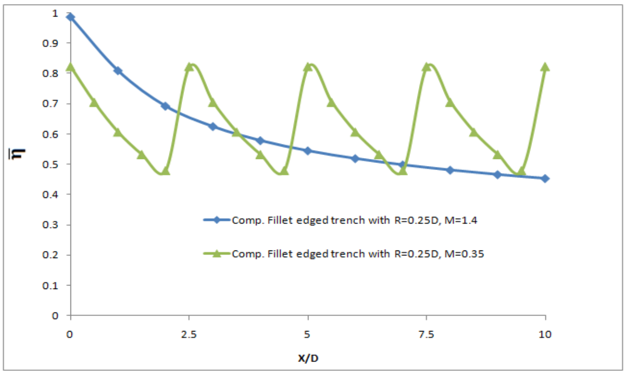


Fig. 7 Comparison of the cooling of a surface area X/D = 10 by Z/D = 2.8 using the same coolant mass flow rate for one film hole and for 4 holes with a blowing rate per hole of ¼ that of the single hole

Figure 7 shows that the four rows of holes give a far superior surface cooling for the same coolant mass flow rate. This is the key advantage of effusion cooling. The use of four rows of holes with the same hole diameter increases the total hole area and this reduces the coolant pressure loss. However, there will be upstream backside cooling of the wall in reality and this upstream flow can be designed to carry the main flow metering pressure loss. One way of doing this is to have upstream impingement cooling jets, resulting in impingement/effusion cooling. However, the comparison in Fig. 7 will be irreproachable, provided the pressure loss for the four row scheme is controlled and is the same as that of the single row of holes. The authors [26] have previously shown, for a simple inclined film cooling geometry with no outlet trench, that increasing the hole diameter and reducing its pressure loss is a very effective way of improving the cooling effectiveness at the same coolant mass flow, but lower blowing rate. This has the greatest benefit for blowing ratios at the higher pressure loss that result in lifted jets, but at the lower pressure loss the lower M gives an attached surface cooling film.

Transverse Spread of the Film Cooling

Figure 8 presents the film cooling effectiveness contours for all the studied cases for a range of blowing ratios from 0.6 to 1.4. The shaped trench configuration with a sufficient downstream vertical face depth, suggest that the coolant air ejected from the holes is characterized by: obstruction of the jet, suppression of the vertical momentum of the jet and transverse spreading of the jet inside the trench before it emerges on the wall surface. The above characteristics should manifest in the spatial film cooling effectiveness of the wall surface. Also, the spatial film cooling effectiveness contour shown in Fig. 8 for the area within the mid-span and X/D = 0 -10 is a visualization of the surface area adiabatic cooling effectiveness. Effectiveness contour for Cases I and II as shown in Fig. 8 indicates an increase in the surface area distribution of the film cooling effectiveness with increases in blowing ratios from 0.6 to 1.0 was significant at both the jet exit and downstream regions compared to that for M=1.0 to 1.4.

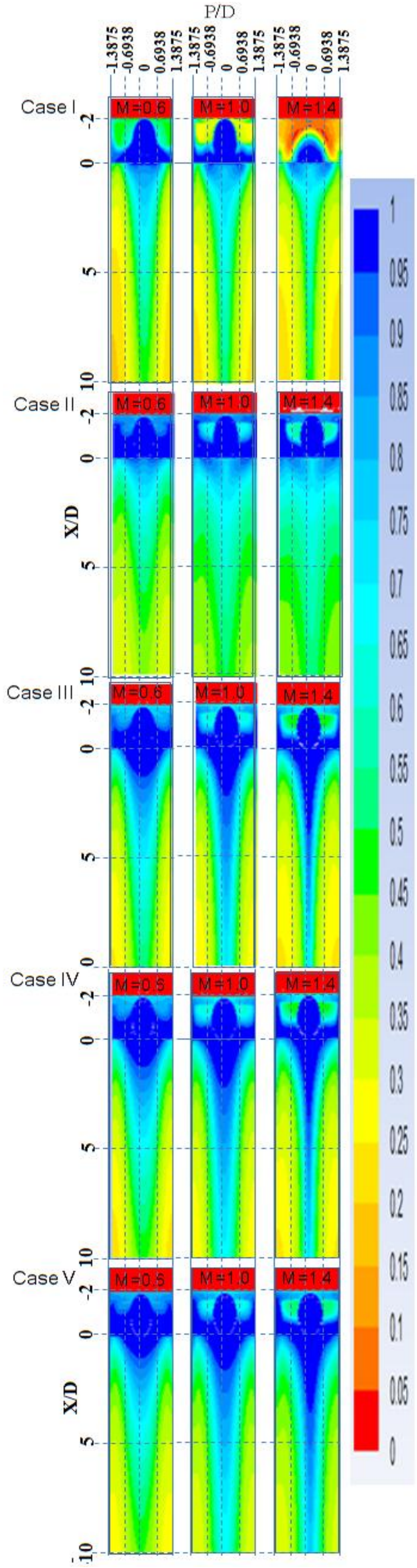


Fig. 8 Spatial film cooling effectiveness contours of the test surface

The comparison of the contours between Cases I and II at M = 1.0 and 1.4 in Fig. 8 suggest that the optimum trench depth is larger than 0.5D. The contours for Case V indicates an increase in the surface area distribution of the adiabatic film cooling effectiveness with increases in blowing ratios from a range of M=0.6 to 1.0 was only significant in the downstream region. Similarly, the contours for Cases III and IV, indicates that the surface area distribution of the cooling effectiveness increase slightly with an increases in blowing ratios from 0.6 to 1.0. In the same Cases III and IV, the contours indicate that the surface area distribution of the adiabatic effectiveness steeply decreases with an increase in M from 1.0 to 1.4. Ideal gas turbine cooling is the one in which the surface of the hot turbine component is evenly cooled. However, the contours of spatial cooling effectiveness between an X/D of 0 to 5 shown in figure 8 suggests that the cooling performance of case V is closer to that of ideal cooling than the other configurations investigated.

Figure 9 presents the development of the coolant thermal profile normal to the wall, at blowing ratios of 0.6, 1.0 and 1.4. However, for all the Cases shown in figure 9, the dimensionless temperature contours for the lateral plane at x/D=3 indicate that the coolant jet penetration increases with an increase in the blowing ratio. Case II had the best lateral coolant jet spreading with an increase in the blowing ratio. This indicates that the optimum vertical face depth prior to the start of the bevel and the fillet outlet could be 0.75D. Case V had the overall lowest jet penetration for the range of blowing ratios investigated. These qualitative results show that for the base Case I with a high vertical jet momentum at the hole centreline, filleting the trailing edge of the trench outlet suppresses the vertical jet momentum, due to the Coanda effect of the jet and consequently produces a better attached cooling jet.

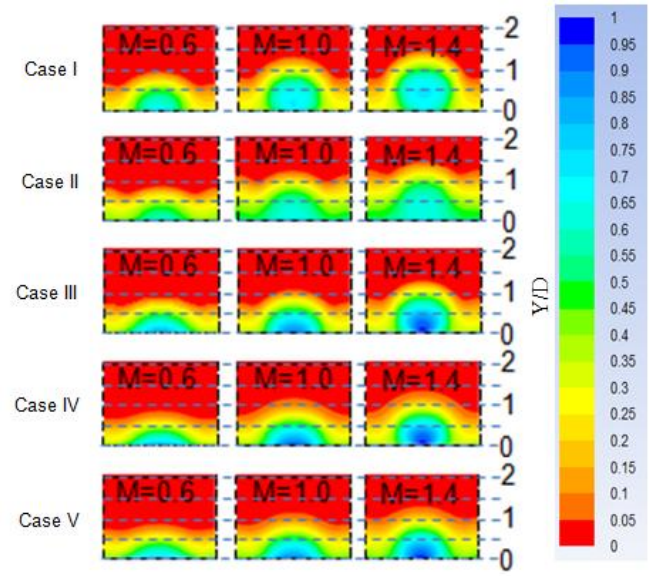


Fig. 9 Dimensionless spatial temperature profile at the plane of X/D = 3

Previously the authors have shown [26] that comparison of the CFD results with experimental data on a lateral averaged basis can give misleading results. This is because the average may be correctly predicted but the local lateral distribution may be incorrect because CFD over-predicts at the holes centreline region and under-predicts at the holes mid-span region. Moreover, the prominent asset of trench and shaped trench is the lateral spread and in our opinion this asset is best characterized by the local lateral effectiveness distribution. In view of this, the local lateral distribution of the adiabatic film cooling effectiveness is presented in Fig. 10. Effusion film cooling, which is the long term goal of these studies, typically has same values of X/D and P/D. Thus the local lateral effectiveness distributions presented in Figure 10 is at X/D = 3, close to that of the P/D. The shaped trench outlet configurations were shown to have a better local lateral effectiveness distribution compared to the conventional sharp edge trench, except for the mid-span where Case II had a slightly better local lateral effectiveness distribution.

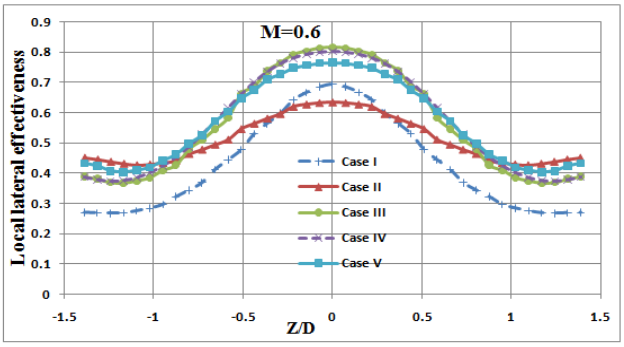


Fig. 10 Local lateral distributions of adiabatic film cooling effectiveness at X/D = 3

The cooling performance of the shaped trench outlets, especially for the fillet edged trench outlet configuration, has the potential to provide improved film cooling. Case I and bevelled trench outlet cases (Cases III and IV) had similar trends of the local lateral effectiveness distributions. While the similarity in trend was due to the initial 0.5D vertical face depth of shaped trench outlets been the same as the trench depth of Case I, the improvement was due to the Coanda effect of the jet at the last 0.25D of the trench outlet. The comparison between the shaped trench outlets and the 0.5D conventional sharp edge trench outlet are shown in Fig. 10. This shows that the bevels improved the cooling effectiveness by approximately 45% at the mid-span region and 17% at the hole-centreline. The fillet improved the cooling effectiveness by about 60% at the mid-span region and 10% at the hole-centreline. Also, comparison between all the cases with a total trench depth value of 0.75D (cases II, III, IV, and V) in fig. 10 indicates that the bevels and the fillet improved the cooling performance by approximately 28% and 20% respectively at the hole-centreline. However, the same comparison shows that the bevels and the fillet reduced the cooling performance by approximately 14% and 4 % respectively at the mid-span. This result suggests that the optimum vertical face depth of shaped trench could be above 0.5D.

CONCLUSIONS

The effects of the shaped trailing edge on trench outlet film cooling performance have been investigated using CFD. It was revealed in these computational investigations that

shaping the trailing edge of the trenched film cooling configuration resulted in further improvement of the film cooling performance of the trenched film cooling geometry. The improved film cooling performance of the shaped trench outlet was due to the Coanda effect obtained from shaping the trailing edge of the trench outlet. The fillet edged trench outlet configuration had the overall superior film cooling performance and this was due to the absence of contour at the trailing edge of the trench which further strengthens the Coanda effect. For cases I, III, IV and V, all of which had a vertical face depth of 0.5D at the trailing edge of the trench outlet, the bevels and fillet improved the cooling performance by approximately 45% and 60% respectively at the mid-span. Also, for all the 0.75D trenches, the bevels and the fillet improved the cooling performance by approximately 28% and 20% respectively at the hole-centreline.
In the near hole region, the cooling effectiveness of the filleted trench outlet was almost even and this will be a potential asset for effusion film cooling. Thus, shaped trench outlet film cooling configurations have the prospect to improve turbine film cooling effectiveness or to reduce the mass flow of coolant for a particular desired cooling effectiveness.

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NOMENCLATURE

A = Area of the coolant hole
CA = Compund angle
D = Film cooling air hole diameter, m
DR = Coolant to crossflow density ratio, ρ_j/ρ_g
VR = Coolant jet to crossflow velocity ratio, u_j/u_g
G = Coolant mass flow rate per surface area kg/sm^2 at 1 bar
M = Blowing ratio = $\rho_j u_j$ / $\rho_g u_g$
N = Number of holes in a single row
T = Temperature, K
u = Mean velocity, m/s
ρ = Density, kg/m^3
*X*Z = Surface area cooled by an individual film cooling hole in an effusion hole array, m
x, *y*, *z*= Cartesian coordinate system
Y/D = Dimensionless vertical distance
P/D = Hole pitch to diameter ratio
X/D = Dimensionless downstream distance
Z/D = Dimensionless lateral distance
L/D = Injection hole length to hole diameter ratio
α = Coolant injection angle
y+ = Normalized distance from the wall = $\rho u_\tau y / \mu$
η = Adiabatic film cooling effectiveness = $(T_g - T_{aw})/(T_g - T_c)$
$\overline{\eta}$ = Laterally averaged film cooling effectiveness
R = Fillet radius
Re = Reynolds number based on duct upstream velocity and film hole diameter.
T_u = Turbulence intensity

SUBSCRIPTS

c = coolant conditions upstream of the hole
g = hot gas crossflow
j = film coolant jet
aw = adiabatic wall

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