EXPERIMENTAL INVESTGATION OF ENTRY LENGTH HEAT TRANSFER COFFICIENT WITHIN FAN AND CONICAL SHAPED FILM COOLING CHANNELS

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ABSTRACT

In this paper, experimental study has been carried out to quantify the distribution and magnitude of heat transfer along the entry length of fan and conical shaped film cooling holes at an inclination angle to the surface of 30°, rotation angle of 0°, and Reynolds number focus range was between 10,000 and 30,000. Experimental data were gathered from a transient liquid crystal method within a 40 times scale model of a film cooling hole to provide detailed distributions of local Nusselt number within the channel. To consider the effect of a cooling flow turning as it interacts with the main hot gas flow, a flow turning 'hood' was used. Discharge coefficient for each cooling channel configuration was provided, including the effects of the flow turning hood, due to large uncertainty, ≈15%, for low flow rates, discharge coefficients were provided up to a Reynolds number of 100,000. Circumferentially averaged Nusselt number plots were generated over the length of each film hole and compared against existing literature having similar arrangements, with general agreement in trend. Magnitudes and distribution of Nusselt number appeared to be similar for both fan and conical shaped cooling holes, with a relatively large exit hydraulic diameter between the cone and fan accounting for both a larger heat transfer surface area, offset by a lower local Reynolds number, resulting in a similar enhancement of Nusselt number relative to their identical inlet diameters. Both cooling geometries show an initial enhancement of heat transfer as the flow turns into the entry length, then decreasing along the entry length of the film cooling channel. Uncertainty for Nusselt number distributions was 8.5%, and between 1.6% - 15% for discharge coefficient.

1. INTRODUCTION

Film cooling is a technique commonly used to reduce the temperature of critical gas turbine components, such as nozzle guide vanes and compressor turbine blades relative to the high temperature mainstream gas. Cooling air is bled from the compressor, delivered through the internal aerofoil cooling channels, and on to the blades surface. Once on the aerofoil surface, the air is intended to create a cooling film between the hot combustion gasses and the aerofoil surface. Sophisticated design work is required to optimise the film effect, including the injection angle to the surface, streamwise offset angle, Reynolds number, and blowing ratio. Where a film hole exits into the gas path, the aerofoil wall locally becomes very thin, and therefore a significant heat-sink effect is observed from the internal convective heat transfer within the film hole geometry. When accounting for this heat-sink effect, the development of thermal boundary layer through the entry length of these film holes becomes important. Due to the imperfect entry arrangement into film holes, the velocity profile cannot generally be assumed as uniform, and although it is common to observe a boundary-layer trip through such an entry length, in practical applications, it has long been recognised that transition to a fully turbulent boundary layer cannot be easily predicted, Kays and Crawford [1].

Early experiments were conducted to determine heat transfer enhancement in a variety of tube entry length arrangements at 50,000 Reynolds number. Three of the variations tested are particularly relevant to a typical film hole geometry; two within an entry length to a cylindrical pipe at 0° inclination angle; one with continuous entry cross-section, and the second with a sharp entry from plenum, and a third at

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45° inclination angle, with continuous cross-section. The Nusselt number development was measured using a steam-heated tube, and observation of the condensation rate at progressive sections by Boelter et al [2]. Whilst this work exhibits valuable information relating to the Nusselt number development along the entry length, limitations in its application to typical turbomachinery cooling applications exist, chiefly; local circumferential variation was not captured, only a relatively high Reynolds number flow was tested, interpolation for angles within the 0° - 45° range is required, no variations in rotation angle are available, and there is no investigation for more modern film geometries. Another departure from a typical film cooling arrangement is that only one cylinder was fed from a plenum, and this was in the same direction of flow, and those with an effective inclination angle were instead fed directly from a supply tube with constant diameter, and therefore the flow was constrained. Although more modern film geometries, such as those with fan and cone shaped geometry variations are common in industrial applications, very little published literature is available to determine the internal heat transfer coefficients present within their entry lengths. This study has been conducted to build on that work and expand these correlations to consider both fan and conical channels within the gas turbine cooling application. The research outputs include information on the local heat transfer distributions, and the relative Nusselt enhancement compared to fully developed flow at the same hydraulic diameter of the film hole inlet.

The overarching study which encompasses this work aims to investigate the internal heat transfer coefficients, and discharge coefficients found within these fan and conical shaped film cooling channels, at a range of realistic film cooling arrangement conditions for geometry, Re, rotation angle, inclination angle, and with localised distributions of Nusselt number. This paper focusses on data gathered for a rotation angle of 0°, an injection angle of 30°, with no additional cross-flow, and Reynolds number range of 10,000 to 30,000 for both fan and cone shaped cooling holes, and with the supply fed from a plenum. To experimentally determine the local heat transfer distribution, a thermochromic liquid crystal technique has been developed, and previously detailed in Wright et al [3], and is being utilised with a modular test piece design for geometry variations.

2. METHODS

A modular test section was developed to allow for testing of the desired variations and is shown in **Figure 1**, the scale model used a factor of 40 to simulate a typical single film hole cooling arrangement. The model is fed by an inlet transition duct into a main plenum, mimicking a typical aerofoil cooling feed, this plenum then feeds the film cooling hole, and also has an optional outlet to simulate crossflow, both of these outlets vent to atmosphere. Tests were conducted with and without a flow turning hood at the end of the film hole, the hood is to simulate the flow turning effect encountered by the mixing process with the mainstream gas flow. As in a typical gas turbine configuration, the fan shaped model maintains an initial diameter (D) for a length of 2D, before spreading laterally to create an outlet with double the cross-sectional area. The conical shaped geometry diameter also remains constant from the inlet for a length of 2D, before spreading both laterally and vertically to a final area of thrice its original cross-sectional area. For the purposes of this test, the inlet diameter is taken as the hydraulic diameter.

An air blower supplies a constant mass flow of air, heated through a series of heat exchangers to bring the flow to a desired temperature and flowrate. During preparation, the air is vented from the rig, until the desired flow rate and air temperature are achieved. Once achieved, fast action valves were activated to change the diverted air direction to go through the test section, and apply a near step temperature change through the film cooling hole model, which was monitored by thermocouples throughout the test section shown in Figure 1 – Assembly of Conical 1,2,3 and Fan 4,5,6 shaped cooling models, with main plenum 7, and inlet transition piece 8.

Using a thermochromic liquid crystal technique, a colour response was observed by high-speed cameras when the heated air was diverted through the model. Nusselt number distributions were then calculated from this TLC response for all internal cooling channel heated surfaces, showing the localised heat transfer. Areas of flow separation within the geometry could also be inferred from this measurement technique. Comparisons were made between the heat transfer distributions seen by the fan (**Figure 2**), and conical (**Figure 3**) shaped geometries, as well as the circumferentially averaged Nusselt number along their length, given in (**Figure 4**), which also compares the data against the existing measurements for a cylindrical hole [1].

Although heat transfer coefficient measurements were the primary aim of this testing, pressure tappings had been located within the plenum before the film hole giving a pressure drop over the section which was then used to generate a discharge coefficient (C_d) for each test.

3. RESULTS AND DISCUSSION

Both fan and conical shaped film entry lengths were tested over a length (L) of 6 times the inlet diameter (D), and an injection angle to the surface (α) of 30°. The rotation angle (Ө) relative to the upstream flow direction was modified between 0° and 90° for each geometry, and the data for 0° was compared against existing literature [2]. No crossflow was simulated in these tests, only that which occurs from the momentum and subsequent eddies of the plenum flow, and therefore the results are especially valid for a location at the end of a channel, or where no crossflow is present. Over the Reynolds number range tested the uncertainty in Nusselt number was calculated to be 8.5%.

Generally, the distribution of Nusselt number is consistent between the Reynolds number range, as shown in **Figure 2** for the fan shaped geometry, and **Figure 3** for the conical film. These figures also demonstrate that Nu distribution is highly consistent between the fan and cone shaped cooling holes. These distributions show that as the flow turns into the film hole, the highest heat transfer occurs before an L/D of 2, and on the underside of the film hole, as the momentum impinges the flow onto this surface. The upper surface of the cooling hole however sees a flow separation here, and an area of low heat transfer. This localised area of low heat transfer is also apparent in **Figure 4**, where within the initial length; at L/D<1 is due to this upper surface region being almost wholly within the separation region.

Figure 4 shows the L/D dependent, circumferentially averaged Nusselt distribution, relative to the Nusselt in a fully developed channel of the inlet diameter. Data shows that throughout the film hole length, the cone and fan had very similar heat transfer distributions. The cone however, has a larger relative hydraulic diameter as L increases, and therefore the Nusselt number is averaged from a larger observed surface area, where overall the heat transfer would in-fact be greater than captured in **Figure 4**. at the higher L/D range (L/D>4), we also see that Nu/Nu_∞ trends to, and dips just below 1, this is due to the increase in the actual hole diameter, which isn't factored into either the local Nu or Nu_∞, this was done to provide a simple way of gauging the relative Nusselt enhancement compared to a cylindrical hole using a single assumption of film hole diameter.

Figure 4 also shows that the Reynolds number 30,000 case had a lower Nu/Nu_∞ratio than the 10,000 case, based on a Nu_∞calculated from the Dittus-Boelter equation. This may be because the 10,000 Re case is closer to laminar flow in the upstream plenum, having a Reynolds number of 3,150, compared to a plenum Re of 9,400 for the 30,000 Re case, therefore making a boundary layer trip more significant.

When comparing the experimental data in this study to literature in **Figure 4**, it is difficult to draw a direct comparison because of multiple independent variables between the experiments, namely Re, injection angle, and the arrangement of the film hole inlet. The general trend from L/D =1 to L/D=6 can be seen to generally agree. The Nusselt number in the 0° inclination angle for a cylindrical hole is of a similar magnitude as the experiments conducted in this study. Data from the present study shows a greater decrease in Nu Nusselt number towards the higher L/D values, which is to be expected as the hydraulic diameter is increasing.

The inlet flow in the literature cases is less constrained, having no inlet plenum, and therefore reduced surface area impacted by flow separation and may explain why we see the higher enhancement. This constrained flow effect may be considered to be closer to the unheated entrance case where the flow boundary layer is already developed. The sharp entry case may also be significantly larger in Nu due to the flow already being in the same direction as the entry length, and therefore requiring no turning.

Discharge coefficients were calculated for each geometry and Reynolds number and given in **Table 1**. Comparison with existing literature shows that these are significantly higher than C_d's seen in similar configurations within the 10,000 – 30,000 Re range, Hay and Lampard [4]. Due to the large scale of test section, a relatively low flow velocity was produced through the film hole over the Re range tested, and the inaccuracy of the manometer for these very low flow rates created very large uncertainties in the $C_d$data, in the magnitude of 15% for this Re. An additional data set for C_d is therefore provided for a Re of 100,000, with a lower uncertainty of just 1.6%. Testing with and without the flow turning hood was found to have little impact on the heat transfer distribution and measured C_d.

		Discharge Coefficient, C_d	
	Reynolds Number	Free	Hood
Fan	10,000	1.02	0.96
	30,000	0.98	0.98
	100,000	0.89	0.89
Cone	10,000	0.91	0.91
	30,000	0.96	0.95
	100,000	0.85	0.87

Table 1 - Film hole discharge coefficient

Data generated from this study will be used to improve the design calculations for gas turbine cooling systems. Local distributions can be used to better understand the flow and heat transfer phenomena within these channels, and to optimise the cooling effectiveness of aerofoil geometries.

4. FIGURES

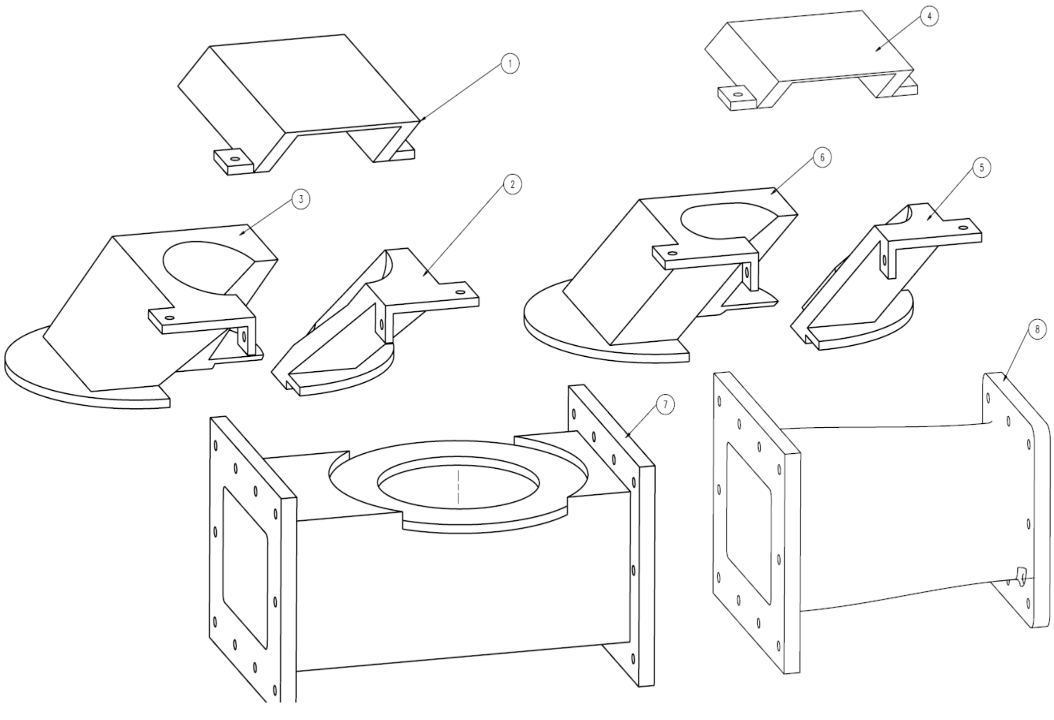

Figure 1 – Assembly of Conical 1,2,3 and Fan 4,5,6 shaped cooling models, with main plenum 7, and inlet transition piece 8.

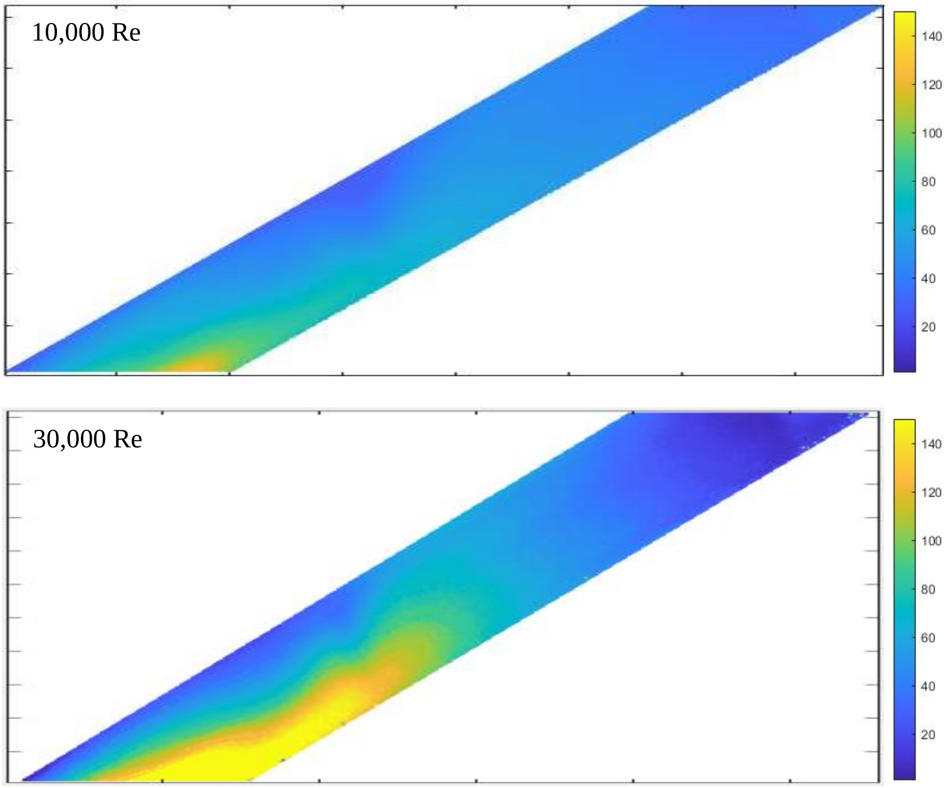

Figure 2 - Internal Nusselt number distribution - Fan shaped cooling channel - Upper - 10,000 Re, Lower - 30,000 Re

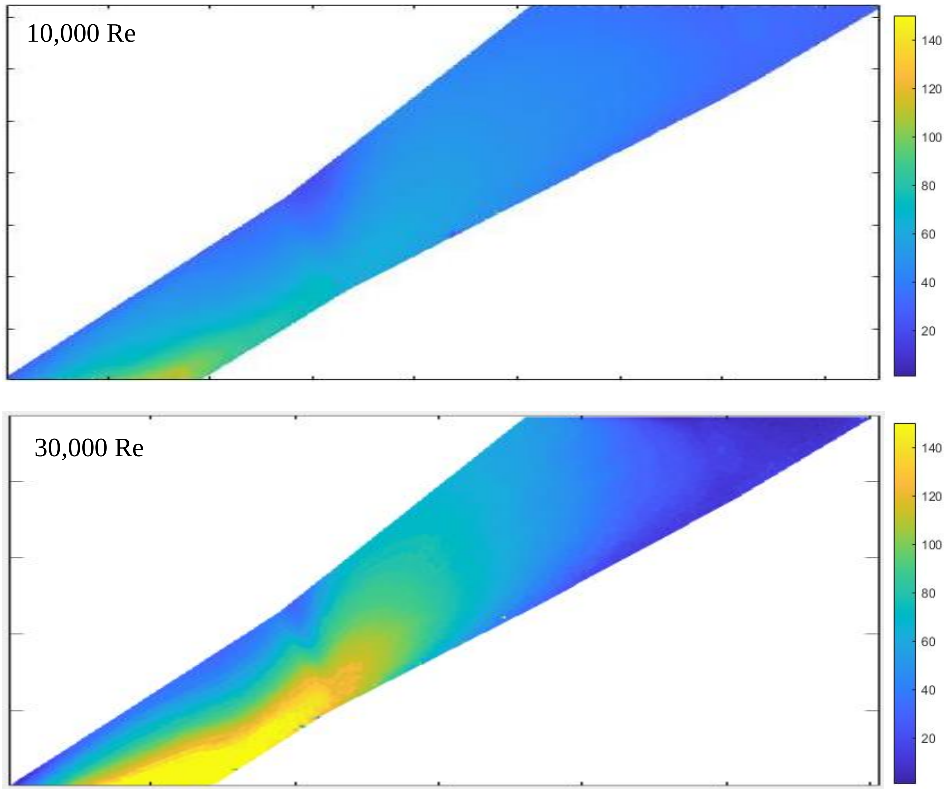

Figure 3 - Internal Nusselt number distribution - Conical shaped cooling channel - Upper - 10,000 Re, Lower - 30,000 Re

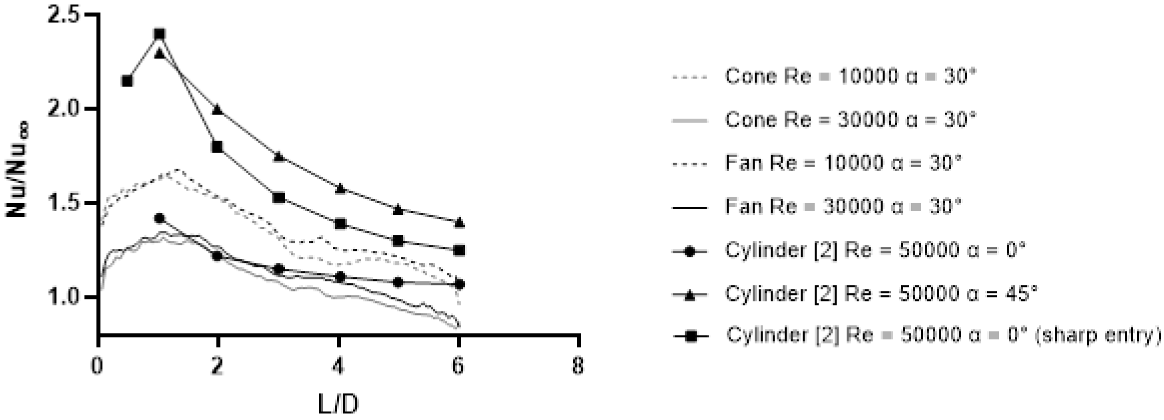

Figure 4 - Circumferentially averaged Nusselt number

5. CONCLUSIONS

Heat transfer measurements were conducted for fan and cone shaped film cooling holes, at a range of Reynolds number from 10,000 – 30,000. The Nusselt number enhancement as a function of flow length was given and compared against existing literature. The heat transfer performance of fan and cone shaped cooling holes were seen to be similar in magnitude and distribution. The heat transfer performance over the Reynolds number range was also seen to be broadly similar in magnitude, but greater in magnitude for higher Reynolds number. Nusselt enhancement was not as high as expected from previous literature, but geometry related effects including the entrance plenum may play a greater role than expected, especially where there is a transition towards turbulence into the entry length. Nusselt number distribution contours show that the higher heat transfer coefficient is concentrated on the lower surface of the film hole where the flow impinges. The upstream momentum carries the flow onto this surface, restarting the thermal boundary layer. Nusselt number enhancement at a Reynolds number of 10,000 was observed to be higher than the enhancement at 30,000. This is likely due to the increased upstream momentum causing a reduction in heat transfer on the upper surface, and the effect of a larger transition towards turbulent flow as the fluid enters the restricted diameter of the film hole.

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