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REVISITING DIMENSIONLESS PARAMETERS QUANTIFYING FILM COOLING

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ABSTRACT

Adiabatic effectiveness of film cooling (η) has been characterized by the density (DR_h) and blowing (BR_h) ratios. In this study, dimensional analysis and computations based on Reynolds-Averaged-Navier-Stokes (RANS) were performed to identify and examine parameters needed to quantify η, where film cooling is crossflow fed instead of plenum fed. The test problem studied is film cooling of a flat plate, where the cooling air, issuing through 30-degree inclined circular holes, is fed from a cooling channel whose flow direction is perpendicular to the direction of the hot-gas flow. For this test problem, dimensional analysis shows an additional blowing ratio is needed, denoted as BR_c, to quantify η, where BR_c is the ratio of the mass flux through the film-cooling hole to the mass flux in the cooling channel upstream of the film-cooling hole. RANS results with and without conjugate heat transfer obtained by varying the mass-flow rate in the cooling channel, while keeping DR_h and BR_h constant (DR_h = 1.9 and BR_h was either 0.75 or 1.0), show reducing the mass-flow rate in the cooling channel by one half, which doubles BR_c (from 2.6 to 5.2) to slightly affect the discharge coefficient through the film-cooling holes (<5%) but up to 85% on laterally-averaged η and up to 25% on overall cooling effectiveness. RANS results also show the flow mechanisms induced by BR_c that affect η. The RANS part of this study was validated by comparing with experimental data.

Keywords: film cooling, crossflow-fed film cooling, adiabatic effectiveness, overall cooling effectiveness, RANS

INTRODUCTION

The efficiency of a gas turbine increases with the temperature of the hot gas entering the turbine component from the combustor. Thus, advanced gas turbines operate at very high temperatures – temperatures that far exceed the maximum temperature the turbine material could maintain structure integrity. This is enabled by cooling all parts of the turbine exposed to the hot gas. Film cooling is widely used to cool the first-stage vanes, blades, and endwalls, where cooling air extracted from the high-pressure compressor is ejected through film-cooling holes to form a blanket of cooler air next to the turbine material to insulate it from the hot gas [1].

The importance of film cooling has led many investigators to study it. See reviews by Han et al. [2], Bogard & Thole [3], and Bunker [4]. Most studies of film cooling have the cooling air fed from a plenum (Fig. 1(a)). However, in gas turbines, the cooling air for film cooling is invariably fed in a crossflow fashion (Fig. 1(b)). Thole et al. [5], Kohli & Thole [6], and Gritsch et al. [7] were among the first to study this problem. In their study, the direction of the cooling flow in the cooling channel is parallel to the direction of the hot-gas flow. Thole et al. [5] showed the importance of accounting for not only the crossflow at the exit of the film-cooling hole such as those due to compound-angle holes but also how cooling flow enters the film-cooling hole. The study by Thole et al [5] focused

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on how direction of crossflow affects the discharge coefficient across the film-cooling hole, the turbulent intensity in the hole, and the velocity and turbulence intensity profiles at the exit of the film-cooling hole. Gritsch et al. [8], Stratton et al. [9], McClintic et al. [10-12], Qenawy et al. [13], Sperling & Mathison [14], Wang, et al. [15], and Straub et al. [16] studied crossflow-fed film cooling, where the direction of cooling flow in the cooling channel is perpendicular to the direction of the hot-gas flow, which is more representative of film cooling in vanes and blades. All of these studies showed "perpendicularly" crossflow-fed film-cooling to introduce considerable swirl in the film-cooling jet, which significantly affected its interactions with the approaching hot-gas flow and the resulting adiabatic effectiveness.

So far, most studies of film cooling, whether plenum fed or crossflow fed, have assumed that the adiabatic effectiveness of a given film-cooling-hole configuration depends only on the density ratio (DR_h) and the blowing ratio (BR_h). When the film-cooling is crossflow fed, most investigators used the Reynolds number of the flow in the cooling channel to distinguish effects of different mass flow rates in the cooling channel. However, McClintic et al. [10-12], Qenawy et al. [13], and Straub et al. [16] showed adiabatic effectiveness to be affected by the ratio of the average velocity in the film-cooling hole to the average velocity in the cooling channel.

When performing experimental and computational studies, it is important to ensure that all relevant dimensionless parameters are accounted for so that data generated could be scaled from laboratory to engine conditions. Thus, the objective of this study is twofold. The first objective is to revisit dimensionless parameters that affect film cooling when the film-cooling flow is crossflow fed. The focus is on identifying all dimensionless parameters that affect adiabatic effectiveness where all walls are adiabatic. The second objective is to compare adiabatic effectiveness with overall cooling effectiveness and examine the discharge coefficient through the film-cooling hole with and without conjugate heat transfer when all of the dimensionless parameters are accounted for. This study is based on dimensional analysis, experimental measurements, and computational fluid dynamics (CFD). For the CFD part, the turbulent flow is modeled by steady Reynolds-averaged Navier-Stokes (RANS) with and without conjugate heat transfer.

The remainder of this paper is organized as follows. First, a dimensional analysis of the plenum-fed and crossflow-fed film-cooling problems shown in Fig. 1 is presented. Next, the experimental component of this study is described. This is followed by a description of the computational component, which includes the computational model of the experimental study, problem formulation, numerical method of solution, results of the grid-sensitivity study, and solution validation. The results from the experimental and computational study are then presented. The paper concludes with a summary of findings.

DIMENSIONAL ANALYSIS

For the plenum-fed and the crossflow-fed film-cooling problems depicted in Fig. 1, the output sought is the adiabatic wall temperature (T_{ad}), which is dimensional, and adiabatic effectiveness (η) is its nondimensional form. The inputs that could affect T_{ad} are as follows:

1. hole geometry: circular cylinder with diameter D
2. length of hole: ℓ = H/sin α = 6D
3. inclination of hole relative to plate's surface: $\alpha = 30^o$
4. angle between flow direction in cooling channel and flow direction in hot-gas path: $\beta = 90^o$
5. geometry of hot-gas path: A_h
6. geometry of cooling channel: A_c
7. hot-gas temperature in freestream: T_h
8. hot-gas density in freestream: ρ_h
9. hot-gas freestream velocity magnitude: U_h
10. hot-gas viscosity in freestream: μ_h
11. temperature of cooling air at cooling-channel inlet: T_c
12. density at cooling-channel inlet: ρ_c
13. average velocity magnitude in cooling-channel upstream of film-cooling hole: U_c
14. viscosity of cooling air at cooling-channel inlet: μ_c
15. average temperature of cooling air in cooling hole: T_j
16. average density in cooling hole: ρ_j
17. average velocity magnitude in cooling hole: U_j
18. average viscosity in cooling hole: μ_j

Thus, T_{ad} is a function of 18 parameters: D, ℓ, α, β, A_h, A_c, T_h, ρ_h, U_h, μ_h, T_c, ρ_c, U_c, μ_c, T_j, ρ_j, U_j, and μ_j. Since four units are needed to describe the problem – namely, mass, length, time, and temperature – there should be a total of $(18+1)-4 = 15$ dimensionless parameters for the problems shown in Fig. 1. Using Buckingham Pi or some similar method yields Eqs. (1) and (2), given below, which connects η to the remaining 14 dimensionless parameters.

(1)

$$(T_h - T_{ad})/(T_h - T_c) = F\left(\frac{\rho_j}{\rho_h}, \frac{\rho_j}{\rho_c}, \frac{T_h}{T_j}, \frac{T_c}{T_j}, \frac{\rho_j U_j}{\rho_h U_h}, \frac{\rho_j U_j}{\rho_c U_c}, \frac{\ell}{D}, \frac{A_j}{A_h}, \frac{A_j}{A_c}, \frac{\alpha}{\pi}, \frac{\beta}{\pi}, Re_c, Re_h, Re_j\right)$$

$$\eta = F\left(DR_h, DR_c, TR_h, TR_c, BR_h, BR_c, \frac{\ell}{D}, \frac{A_j}{A_h}, \frac{A_j}{A_c}, \frac{\alpha}{\pi}, \frac{\beta}{\pi}, Re_c, Re_h, Re_j\right) \qquad (2)$$

Of the 14 dimensionless parameters that affect η, the following five parameters are connected to the geometry of the problem, which were not varied in this study: ℓ/D, A_j/A_h, A_j/A_c α/π, and β/π. If the walls are adiabatic and the velocity magnitudes are low (e.g., Mach number much less 0.3), then $\rho_j \approx \rho_c$ so that $DR_c \approx 1$. Also, since $\rho = p/RT$ is a good approximation of the gas and the pressure difference across the film-cooling hole is typically a small percentage of the absolute pressure in that region (i.e., $p_h \approx p_c$), $DR_h = \rho_j/\rho_h \approx \rho_c/\rho_h \approx T_h/T_c$ so that $DR_h \approx TR$. Thus, the dimensionless parameters that affect film-cooling adiabatic effectiveness can be reduced to DR_h or TR, BR_h, BR_c, Re_c, Re_h, and Re_j so that Eq. (2) becomes

$$\eta = F(TR, BR_h, BR_c, Re_c, Re_h, Re_j) \qquad (3)$$

The above equation shows six dimensionless parameters are needed to describe η for the problem shown in Fig. 1 if all geometric parameters are fixed, all walls are adiabatic, and the pressure drop across the film-cooling hole is small when compared to the absolute pressure about the hole.

Note that in most experimental studies of film cooling, $DR_c = \rho_j/\rho_c \approx 1$ and $DR_h \approx TR$. The Reynolds numbers given by Re_c, Re_h, and Re_j characterize the flow rate in the hot-gas path, cooling channel, and cooling hole; determine whether the flow is laminar or turbulent; and affect the discharge coefficient across the film-cooling hole. Of these six dimensionless parameters, most previous studies only considered the following five: $BR_h = \rho_j U_j/\rho_h U_h$, $DR = \rho_c/\rho_h \approx T_h/T_c = TR$, Re_h, Re_c, and Re_j. Thus, one parameter was missing, namely BR_c. Only McClintic et al. [10-12], Qenawy et al. [13], and Straub et al. [16] recognized the effects of $BR_c = \rho_j U_j/\rho_c U_c \approx U_j/U_c = VR_c$ while varying Re_c.

If the cooling air is plenum fed, then $\rho_j U_j A_j = \rho_c U_c A_c$ since all cooling air entering the plenum exits through the film-cooling holes. Thus, $BR_c = \rho_j U_j/\rho_c U_c = A_c/A_j$, which can be seen to depend only on geometry. This is why BR_c was not studied previously. However, for a crossflow-fed configuration, not all air entering the cooling channel exits through the film-cooling holes. With film cooling crossflow fed, BR_c depends on the geometry and on the amount of cooling air in the cooling channel. As a result, BR_c must be considered. Thus, the remainder of this study examines the effects of BR_c on film-cooling effectiveness, overall cooling effectiveness, and discharge coefficient.

Note that the dimensional analysis given above, based on Fig. 1, does not account for rotation and curved surfaces such as pressure and suction surfaces of vanes and blades, where pressure can vary appreciably throughout the flow domain. However, film-cooling from compound angle holes and crossflow-fed that are not perpendicular can readily be incorporated through α and β. The dimensional analysis presented is for flat-plate configurations typically used under laboratory conditions. If surface heat transfer needs to be accounted for, then Nusselt and Prandtl numbers need to be added. If there is conjugate heat transfer, then Biot number also needs to be added [17].

DESCRIPTION OF THE PROBLEM

This section describes the crossflow-fed film-cooling problem studied. First, the experimental configuration studied is described along with the measurement methods employed. Afterwards, the two computational models used to simulate the experimental study are described.

Experimental Component

The experimental part of this study uses the steady-state conjugate aerothermal test facility at the National Energy Technology Laboratory. This test facility has been described in detail by Ramesh et al. [18]. In this section, only the essence of the facility pertinent to this study is given. For the tests described in this paper, the hot-gas temperature was constant at T_h = 650 K and the inlet cooling air temperature was controlled to approximately T_c = 342 K to produce a hot gas-to-coolant temperature ratio of TR = 1.9. The datasets used for these averages represent multiple replications of each test condition and at least two different days of testing to measure (1) hot-gas and coolant temperatures, (2) hot-gas and coolant pressures, and (3) downstream velocity profiles.

Figure 2 shows a schematic of the experimental setup. The hot gas flow is conditioned to provide a uniform velocity and temperature profile to approach the 101 mm x 101 mm test section. The test section is designed to enable optical access on three of the four walls. The fourth wall supports the film-cooled test coupon. The cooling air fed from the coolant supply channel is perpendicular to the direction of the hot gas flow. Another viewport is located along the cooling channel wall opposite the test coupon. Infrared imaging is used to measure the surface temperatures for the hot and the cold sides of the flat plate.

The cooling channel is a 127 mm x 6.4 mm rectangular channel. The temperature of the cooling air is measured upstream and downstream of the test coupon using 1.6mm diameter Type-T thermocouples. These thermocouples are located about 25 mm from the leading- and trailing-edges of the test coupon. An electric preheater is used to control the temperature of the cooling air. The average temperature is used as a control point which should be representative of the temperature of the cooling air near the center hole. In other words, the temperature of the cooling air at the center of the test article is constant for all test cases.

The hot-gas flow is measured using a standard pressure/temperature compensated orifice meter that has been calibrated against sonic nozzles at ambient temperatures. The overall uncertainty in the experimental blowing ratio is described in the following reference (i.e., +/– 0.075). The contribution of the main gas flow error to the overall uncertainty in blowing ratio is approximately one order of magnitude smaller than the error introduced by taking the difference of two cooling-air flow readings.

The cooling-air flow rate is measured upstream of the test coupon using a MicroMotion Coriolis flow meter (accuracy +/- 0.25% of reading) and downstream of the test coupon using an Imperial V-20THD venturi meter (accuracy +/- 0.75%). A back pressure control valve is used to independently control the amount of cooling air that is diverted through the film cooling holes. The flow rate of the film cooling air is calculated as the difference between the cooling air flow measured upstream and downstream of the test coupon. For the conditions discussed in this paper, the uncertainty in the blowing ratio is +/- 0.075. The independent variable for these experiments is the cooling channel flow at the inlet to the cooling channel and the blowing ratio. This also varies the temperature of the test article and the shape of the downstream velocity profile near the wall.

Figure 3 shows the test coupon investigated. The coupon is a flat plate made of 316 stainless steel. This baseline coupon was manufactured using conventional machining processes. The surfaces exposed to the hot and cold gas streams measure 101 mm x 66 mm. The long side of the coupon is oriented parallel to the hot gas flow direction. As shown in this figure, a 2.4 mm wide groove is milled into the cold-side surface. This groove depth extends through 80-85% of the coupon thickness and reduces heat conduction from the test section walls. The cylindrical hole film cooling geometry can be summarized by the following parameters: hole diameter ($D = 3.2$ mm); hole spacing, or hole pitch (P = 3D); hole length ($\ell = 6D$), and angle of inclination (α = 30 degrees).

To measure the velocity profile in the momentum boundary layer, a 0.6-mm diameter total pressure probe with a 0.08 mm sensing hole in the tip was used (see Fig. 4). The static pressure is also measured at the wall of the test section (same axial plane as the pressure probe). The position of the total pressure probe is controlled by using a translation stage programmed to collect data at 60 probe locations over a span of roughly 25.4 mm. At each location, the stage pauses for 11 seconds before moving to the next location. The wall reference point was set when the test rig was "hot", and this reference was repeatable to within 0.075 mm.

The operating conditions are summarized in Table 1. Measurements for each test condition were taken over a 10-minute steady-state period. The temperatures, pressures, and flow rate data were sampled every second. The coolant mass flow rate was varied for each test article to achieve blowing ratio conditions of 0.75 and 1.0. These blowing ratio conditions were chosen to reduce local jet lift-off conditions.

Computational Component

Figure 5 shows a schematic of the computational model of the experimental setup described in the previous section. Figure 6 shows a schematic of a configuration that is identical to the configuration shown in Fig. 5 except instead of having five film-cooling holes, there is only one film-cooling hole. The reason for studying the configuration shown in Fig. 5 is to validate the computational study by comparing predictions with measurements from the experimental setup, which has five film-cooling holes. When there are five film-cooling holes, the BR_h and BR_c provided are for the average of five holes. With only one hole, the BR_h and BR_c provided is for that one hole. Thus, the configuration shown in Fig. 6 enables a more precise study on the effects of BR_h and BR_c.

All dimensions in Figs. 5 and 6 are given in terms of the diameter of the film-cooling hole: D = 0.125 inches (3.18 mm). As shown in those figures, the configuration consists of a flat plate that has either five or one film-cooling hole, a hot gas path, and a cooling channel with flow direction perpendicular to the direction of the hot-gas flow. For the film-cooling hole, it has a circular cross section with diameter D and a length of ℓ = 6D and is inclined at angle of α = 30° relative to the flat plate. The spacing between the centers of adjacent film-cooling holes in the spanwise direction is P = 3D.

On the computational domain, the walls of the cooling channel match those in the experimental setup. For the hot-gas path, all walls match except for the top wall, which was truncated at $z = H_h$, and the plane at $z = H_h$ was modelled as an inviscid wall. This simplification is deemed acceptable because the boundary-layer flow about the film-cooled flat plate is much smaller than $H_h = 32D$. For the hot-gas path, the distance from the inflow boundary to the film-cooling hole, L_{h1}, was adjusted to ensure that the boundary-layer velocity profile approaching the film-cooling hole matched the experimental measurements. For the cooling channel, the distance from the inflow boundary to the location of the film-cooling hole, L_{c1}, was adjusted to ensure that the flow in the cooling channel was "fully developed" before reaching the film-cooling holes. Since the flow is compressible, by "fully developed", it is meant that all boundary layers from all walls merged and there are no entrance effects.

The fluid in the hot-gas path and cooling channel is air. For the hot-gas path, the freestream temperature and velocity above the flat plate were set at T_h = 650 K and U_h = 107.5 m/s along the x-direction. For the cooling channel, the temperature of cooling air at its inlet was set at T_c = 342.5 K, which led to a temperature ratio of TR = 1.9. In the experimental study, two different mass-flow rates entered the cooling channel, and they were 0.03829 kg/s and 0.01835 kg/s. However, since the domain of the hot-gas path was reduced by one-half, the mass flow rates used in the computations were reduced by one-half For each mass-flow rate that entered the cooling channel, the static pressure at the outflow boundaries of the hot-gas path and the cooling channel were adjusted to produce two different blowing ratios that are defined in the traditional way, which is $BR_h = \rho_j U_j/\rho_h U_h = 0.75$ and 1.0. Since there are two mass-flow rates that enter the cooling channel for each BR_h, there are two values of $BR_c = \rho_j U_j/\rho_c U_c$. The static pressures used are summarized in Table 1. The other boundary conditions are as follows. All surfaces of solid walls are no-slip. All surfaces of solid walls are also adiabatic except for the surfaces of the coupon with the film-cooling holes. For the coupon (the region with dimensions W x L x H in Fig. 5), there are two thermal conditions: adiabatic and conjugate. The adiabatic-wall boundary condition is needed to obtain the film-cooling adiabatic effectiveness. The conjugate analysis accounts for the conduction heat transfer from the hot-gas path to the cooling channel through the coupon and is needed to obtain the overall cooling effectiveness. Results from the conjugate analysis are the ones that can be compared with experimental measurements. Note that if all walls are adiabatic, then the bulk temperature, T_b, in the cooling channel equals T_c. For the conditions of this study, even when there is conjugate heat transfer that coupled the hot-gas path and the cooling channel, $T_b \approx T_c$. Thus, TR $\approx$ 1.9 for both adiabatic and conjugate conditions.

Table 2 summarizes all parameters examined in the computational study, where subscript "ad" denotes cases with adiabatic walls, and the subscript "conj" denotes cases with conjugate heat transfer. Experiments were conducted only for the conjugate cases in Table 2 with the 5-hole configuration. CFD simulations were performed for both the 5-hole and the 1-hole configurations. The BR_h and BR_c values provided in this table are the nominal ones sought. The actual values realized in the computational and the experimental studies will be discussed in the section on validation.

PROBLEM FORMULATION, NUMERICAL METHOD OF SOLUTION, AND CODE

Since the temperature of the air in the hot-gas path is much higher than the temperature of the cooling air (TR = 1.9), the density variation in the flow is considerable. Thus, though the Mach number is low (<< 0.3), the compressible formulation is needed to address the interactions between the hot-gas flow and the cooling flow. In this study, the governing equations employed for the gas phase are the Reynolds-averaged continuity, Navier-Stokes, and total energy equations for a thermally perfect gas with temperature-dependent thermal conductivity, viscosity, and specific heats [19]. Turbulence effects were modeled by using the shear-stress transport (SST) model [20] with curvature and corner corrections [21].

For the plate with one or five film-cooling holes, the effects of thermal and mechanical stresses were not considered in the conjugate analysis. Only temperature distributions were simulated. The governing equation used for the solid phase was the thermal-energy equation closed by the Fourier law of conduction with temperature-dependent thermal conductivity [22]. In the conjugate analysis, temperature and heat flux are continuous at all gas-solid interfaces.

Solutions to the governing equations for the domains shown in Figs. 5 and 6 were obtained by using the ANSYS Fluent code [23]. The grid system used is described in the next section on Verification and Validation. For the fluid-phase governing equations, the fluxes for the inviscid terms in the continuity, Navier-Stokes, and energy equations at the cell faces were interpolated by using the second-order upwind scheme, while pressure and all diffusion terms were approximated by using second-order accurate central formulas. For the solid phase, the energy equation only has diffusion terms, and the second-order accurate central formula was used. Since only steady-state solutions were of interest, the SIMPLE algorithm was used to generate solutions to the discretized governing equations. To ensure convergence to each steady-state solution, iterations were continued until the residual for every equation reached a plateau. For all cases studied, the scaled residuals at convergence were less than 10^{-5} for the continuity and momentum equations, 10^{-7} for the energy equation (in solid phase as well in conjugate analysis), and 10^{-5} for the turbulence equations.

VERIFICATION AND VALIDATION

Verification of this computational study was conducted through a grid-sensitivity analysis. Figure 7 shows the multi-block structured grid system used to discretize the computational domain. In each cooling hole, it is made up of an H-H grid (i.e., curvilinear Cartesian) and an O-H grid (i.e., curvilinear cylindrical). In the hot-gas path, cooling channel, and coupon, H-H grids were used. For this grid system, grid points were clustered about all solid surfaces and the cooling holes to resolve the boundary layers about the walls, the flow from the cooling channel into the cooling holes, the flow structure in each cooling hole, and the interactions between the boundary layers on the hot-gas side and the flow exiting the cooling holes. The first cell away from any solid surface has a y+ value less than unity so that integration of the governing equations for the fluid phase is to the wall (i.e., wall functions were not used). Also, the grid is smooth when grid spacings change, and grid lines intersect nearly orthogonally.

For the 5-hole configuration shown in Fig. 5, three grids were examined: coarse with 29.7 million cells, baseline with 49.2 million cells, and fine with 59.4 million cells. For the 1-hole configuration shown in Fig. 6, three grids were also examined: coarse with 20.4 million cells, baseline with 33.2 million cells, and fine with 40.8 million cells. For both configurations, the grids were refined in the region about the cooling hole(s), where the flow of the cooling air interacts with the hot-gas flow.

The grid sensitivity study was conducted with the "Case 4ad" operating condition given in Table 2 because it represented the most stringent flow condition. Results obtained for the pressure coefficient (C_p), friction coefficient (C_f), and film-cooling effectiveness (η) along x/D at y = z = 0 are shown in Fig. 8(a) for the 5-hole configuration and Fig. 8(b) for the 1-hole configuration. From these figures, the solutions obtained on the baseline grid can be seen to match those from the finest grid. Thus, the baseline grids for the 5-hole and the 1-hole configurations were used to generate all solutions.

To validate the solutions of this computational study based on steady RANS with the SST model, comparisons were made with experimental data. Table 3 compares results obtained by conjugate analysis and experimentally measured blowing ratio (BR_h), density ratio (DR), and velocity ratio (VR_h) for all cases studied for the 5-hole configuration. From Table 3, the maximum relative differences in BR_h, DR_h, and VR_h can be seen to be less than about 4.5% for the lower blowing ratio cases and less than about 2.5% for the higher blowing ratio cases. Thus, the conjugate CFD simulated BR_h, DR_h, and VR_h of the experimental study with reasonable fidelity.

Figure 9 compares computed and measured velocity profiles downstream of the film-cooling holes at (x/D, y/D, z/D) = (2.75, 0, 0). From this figure, the boundary-layer thickness, δ, can be seen to be predicted with reasonable accuracy. Near the wall, the comparison is still reasonable but not as good. In the near-wall region, the experimental probe also has challenges. In the computational results, note that the velocity profile is non-monotonic next to the wall. The non-monotonicity is created by the swirl in the film-cooling jet induced by the crossflow-fed of the film-cooling flow, which will be explain in more detail in the results section.

Figure 10 compares computed and measured temperature profiles along the spanwise direction on the coupon being cooled by film cooling from y/D = -10.5 to y/D = 10.5 at four locations downstream of the film-cooling holes: x/D = 1.071, 5.071, 9.071, and 13.071 locations, respectively. The computed results shown in Fig. 10 account for the conjugate heat transfer from the hot-gas path to the cooling channel across the coupon containing the five film-cooling holes. From Fig. 10, the wall temperature on the coupon can be seen to be predicted with reasonable accuracy.

The reasonable match between the computed and the measured results shown in Table 3 and Figs. 9 and 10 shows steady RANS based on the SST model with curvature correction to be adequate in addressing the research questions of this study.

Since the results section focuses on the 1-hole configuration and the validation is based on 5-hole configuration, it is important to show that the 1-hole configuration reproduces the average of the 5-hole configuration. Table 4 shows the mass-flow rates entering and exiting the 1-hole and 5-hole configurations along with the experimentally measured values, and they match well. As noted, the 1-hole configuration was used because precise instead of average values of BR_h and BR_c could be used.

RESULTS AND DISCUSSION

As noted in the introduction, the objective of this study is to revisit dimensionless parameters that affect film cooling when it is crossflow fed with focus on film-cooling adiabatic effectiveness, overall cooling effectiveness, and discharge coefficient across the film-cooling flow.

Parameters Affecting Adiabatic Effectiveness

For the problem studied where geometry was not varied, dimensional analysis showed the adiabatic effectiveness (η) to be a function of six parameters: the density or temperature ratio, $DR_h \approx TR = T_h/T_c$; the blowing ratio based on conditions in film-cooling (FC) hole and the hot-gas path, $BR_h = \rho_j U_j/\rho_h U_h$; the blowing ratio based on conditions in the cooling channel and in the FC hole, $BR_c = \rho_j U_j/\rho_c U_c$; and the three Reynolds numbers based on conditions in the hot-gas path, Re_h; the cooling channel, Re_c; and the FC hole, Re_j. Of these, $BR_c = \rho_j U_j/\rho_c U_c$ is the new dimensionless parameter that needs to be added.

With computations based on steady RANS, the validity of the dimensional analysis performed could be assessed. Also, the effects of BR_c on film cooling that are crossflow fed could be examined. In this section, solutions from steady RANS are used to address the following questions:

- Are the assumptions invoked in the dimensional analysis valid for the problem studied?
- Why is swirl correlated to BR_c?
- Why is $BR_c = \rho_j U_j/\rho_c U_c$ needed in addition to Re_c?
- How does swirl affect film cooling?

The assumptions invoked in the dimensional analysis were adiabatic wall, low Mach number, and nearly the same pressure across the film-cooling hole. Thus, $\rho_j \approx \rho_c$ so that $DR_c \approx 1$ and $DR_h = \rho_j/\rho_h \approx \rho_c/\rho_h \approx T_h/T_c$ = TR. In so doing, two dimensionless parameters were eliminated – namely, DR_c and either DR_h or TR. In this study, TR is retained, but TR and DR_h could be used interchangeably.

For the conditions of this study, Table 5 shows the change in density, temperature, and pressure from the inlet to the outlet of the film-cooling hole to be less than 4.16%, 2.08%, and 2.34%, respectively, if all walls are adiabatic. Also, the change in density, temperature, and pressure from the inlet to the outlet of the cooling channel is less than 1%. Based on these results, the assumptions invoked are reasonable with maximum relative error around 4%.

On swirl induced by crossflow-fed film cooling, Fig. 11 shows the axial (V_z) and tangential (V_θ) velocities of the flow at several cross sections in the film-cooling hole as a function of BR_h and BR_c, where V_z and V_θ are defined relative to the film-cooling hole's centerline. Figure 12 shows the swirl defined as the ratio of tangential to axial velocity ($S = V_\theta/V_z$). From these figures, it can be seen that for a given BR_h, the lower the BR_c, the higher is the swirl. Since different swirl can result from the same BR_h, it indicates that another parameter, namely, BR_c is needed.

The question now is why swirl is correlated to BR_c, and why a lower BR_c produces a higher swirl? To illustrate, consider Fig. 13(a). From this figure, one can see that for a given BR_h, a lower BR_c indicates higher mass-flow rate in the cooling channel so that a smaller fraction of the flow in the cooling channel is extracted for film cooling. Thus, with lower BR_c, the cooling air extracted for film cooling is closer to the wall and approaches the film-cooling hole with a smaller angle (i.e., $\theta_1 < \theta_2$ if $Re_{c,1} > Re_{c,2}$, where higher Re_c implies a higher mass flow rate in the cooling channel. With a smaller approaching angle, the flow entering the hole has a higher velocity in the tangential direction, which produces higher swirl.

On why $BR_c = \rho_j U_j / \rho_c U_c$ is needed in addition to Re_c, Fig. 13(b) shows that for a given Re_c, one can have an infinite number of different BR_c's. Thus, specifying Re_c alone is insufficient to define film-cooling with perpendicular crossflow coolant channel configurations.

On how swirl affects film cooling, Figs. 11 and 12 show the region, where the axial velocity (V_z) is high at the exit of the film-cooling hole, depends on (1) the mean axial speed in the film-cooling hole, U_j and (2) the strength of the swirl in the hole – both of which affect the residence time of the cooling air in the hole. This location strongly affects how the hot-gas flow interacts with the film-cooling jet and hence the adiabatic effectiveness. With swirl, the velocity profile next to the wall can be non-monotonic as shown in Fig. 9.

Figure 14 shows the adiabatic effectiveness, $\eta = (T_h - T_{ad})/(T_h - T_c)$, and the effects of swirl on adiabatic effectiveness. For the conditions of the present study, when BR_c is low ($BR_c = 2.6$), the swirl induced was so high that the film-cooling jet lifted off the surface shortly after exiting the film-cooling hole for both $BR_h = 0.75$ and 1.0. This is because the swirl oriented the film-cooling jet to entrain hot-gas underneath it. When BR_c is high ($BR_c = 5.2$), the swirl induced was lower and produced a lateral spreading of the film-cooling jet on the surface that resembled those from a compound-angle hole, especially at $BR_c = 5.2$. Thus, swirl is a parameter that could be exploited in the design of film-cooling holes to enhance film cooling. For example, threads could be built into film-cooling holes to create the desired swirl and flow conditions at the exit of film-cooling-hole to produce the maximum lateral spreading of the film-cooling jet without lift off and thereby increase adiabatic effectiveness.

Figure 15 shows the laterally-averaged adiabatic effectiveness ($\bar{\eta}$), averaged over P = 3D, the spacing between film-cooling holes in the 5-hole configuration. From this figure, when $BR_h = 0.75$ and $BR_c = 5.2$, $\bar{\eta}$ can be seen to be quite high, approximately 0.4, from x/D = 0 to 15. Thus, swirl, if utilized effectively, could significantly increase $\bar{\eta}$.

Parameters Affecting Overall Cooling Effectiveness and Discharge Coefficient

Since adiabatic effectiveness only reveals how well film cooling insulates the surface from the hot-gas flow, one also needs to examine the overall cooling effectiveness that accounts for the heat transfer between the hot-gas path and the cooling channel including the film-cooling holes as well as the discharge coefficient across the film-cooling holes.

Figure 16 shows the overall cooling effectiveness, $\phi = (T_h - T_w)/(T_h - T_c)$, and Fig. 17 shows the laterally-averaged overall cooling effectiveness ($\bar{\phi}$), averaged over P = 3D. These results were obtained by steady RANS with conjugate analysis. Although $BR_c = 5.2$ provided better adiabatic film effectiveness (Figs. 13 and 14), $BR_c = 2.6$ provided better overall cooling effectiveness. This because the heat-transfer rate across the coupon depends on the heat transfer coefficient (HTC) on the hot-gas side and in the cooling channel. When BR_c was decreased from 5.2 to 2.6, the Reynolds number in the cooling channel doubled, which greatly increased the HTC in the cooling channel. For the conditions of the problem studied, it is the HTC in the cooling channel that is limiting the heat transfer rate across the coupon. This is because the HTC in cooling channel is much less than the HTC on hot-gas side. Thus, increasing the HTC in the cooling channel greatly increased the heat-transfer rate across the coupon.

Table 6 shows the discharge coefficient (C_D) with adiabatic wall and with conjugate heat transfer. From this table, the variation in C_D due to BR_c can be seen to be less than 5%. This table also shows how the velocity ratio (VR) and momentum flux ratio (I) affect the discharge coefficient for adiabatic wall and conjugate heat transfer. The relative differences are within 5%.

CONCLUSION

Film-cooling holes in gas-turbine vanes and blades are often fed in a crossflow fashion. Dimensional analysis and computations based on steady RANS were performed to revisit the dimensionless parameters needed to characterize film cooling adiabatic effectiveness and overall cooling effectiveness of a flat plate where the film-cooling flow is crossflow fed. Key findings are as follows:

- With the geometry fixed, dimensional analysis showed the adiabatic effectiveness, η, to be a function of five parameters: density ratio or temperature ratio ($DR_h = \rho_j/\rho_h \approx \rho_c/\rho_h \approx T_h/T_c$), blowing ratio based on conditions in film-cooling hole and hot-gas path ($BR_h = \rho_j U_j/\rho_h U_h$), blowing ratio based on conditions in film-cooling hole and the flow in the cooling channel ($BR_h = \rho_j U_j/\rho_c U_c$), and the two Reynolds numbers based on conditions in the hot-gas path (Re_h) and cooling channel (Re_c).
- When film-cooling is crossflow fed, specifying Re_c is inadequate because for a given Re_c, one could have different BR_c.
- Film-cooling holes that are crossflow fed create swirl in the film-cooling hole that correlate with BR_c or $VR_c = U_j/U_c$ since $\rho_j/\rho_h \approx 1$. The lower the BR_c for a given BR_h, the higher the swirl because the flow entering the film-cooling hole approaches the hole inlet with a smaller angle and has a higher velocity in the tangential direction.

- Swirl if utilized and controlled appropriately could significantly increase film-cooling effectiveness. This is because swirl can increase the spreading of the film-cooling flow.
- Though the swirl induced by crossflow-fed film cooling can significantly improve adiabatic effectiveness, the overall cooling effectiveness may not improve because the overall heat transfer depends on the heat-transfer coefficient on the hot-gas side and in the cooling channel.
- For the conditions of this study, the discharge coefficient was relatively unaffected by BR_c (< 5%).

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NOMENCLATURE

A_c	cross-sectional area of cooling channel
A_h	cross-sectional area of hot-gas channel
A_j	cross-sectional area of film-cooling hole
BR_c	blowing ratio (new to account for cooling channel): $BR_c = \rho_j U_j / \rho_c U_c$
BR_h	blowing ratio (traditional): $BR_h = \rho_j U_j / \rho_h U_h$
C_D	discharge coefficient of film-cooling hole: $C_D = U_j / U_{j,ideal}$
C_f	skin friction coefficient: $C_f = \tau_w / (0.5\ \rho_h U_h^2)$
C_p	pressure coefficient: $C_p = (p-p_b)/(0.5 \rho_h U_h^2)$
D	diameter of film-cooling hole
DR_c	density ratio (new to account for cooling channel): $DR_c = \rho_j/\rho_c$
DR_h	density ratio (traditional): $DR_h = \rho_j/\rho_h$
I_c	momentum flux (new to account for cooling channel): $I_c = (BR_c)(VR_c)$
I_h	momentum flux ratio (traditional): $I_h = (BR_h)(VR_h)$
ℓ	length of film-cooling hole
L_i	length scale used in $Re_i = \rho_i U_i L_i / \mu_i$,
p	static pressure
$p_{b,c}$, $p_{b,c}$	static pressure at cooling-channel and hot-gas path outlet boundaries
$p_{c,in}$	static pressure at cooling channel inlet
P	spacing between film-cooling holes
Pr	Prandtl number: Pr = Pr(T)
Re_i	Reynolds number: $Re_i = \rho_i U_i L_i / \mu_i$, where i = h for flow in hot-gas path, i = j for flow in cooling hole, and i = c for flow in cooling channel
S	swirl in film-cooling hole: $S = V_\theta / V_z$
T	temperature
T_{ad}	adiabatic wall temperature
T_{ad}	bulk temperature in cooling channel; $T_b = T_c$ if all walls are adiabatic
T_c	temperature of cooling flow at cooling-channel inlet

T_h	temperature of hot gas in freestream
T_j	bulk temperature in film-cooling hole; $T_j = T_c$ if all walls are adiabatic
T_w	wall temperature
TR	temperature ratio (traditional): $TR = T_h/T_c$
TR_c	temperature ratio (new to account for cooling channel): $TR_c = T_c/T_j$
TR_h	temperature ratio (that accounts for T in cooling hole): $TR_h = T_h/T_j$
U_c	average velocity magnitude of cooling flow entering cooling channel
U_e	velocity magnitude at edge of boundary layer on hot-gas side
U_h	velocity magnitude of hot gas in freestream
U_j	average velocity magnitude in film-cooling hole: $U_j = \dot{m}_j/\rho_j A_j$
$U_{j,ideal}$	ideal average velocity magnitude in film-cooling hole: $U_j = \sqrt{2(p_{c,in} - p_{b,h})/\rho_j}$
U_τ	friction velocity: $U_\tau = (\tau_w/\rho_w)^{0.5}$
VR_c	velocity ratio (new to account for cooling channel): $VR_c = U_j/U_c$
VR_h	velocity ratio (traditional): $VR_h = U_j/U_h$
$\|V\|$	velocity magnitude
V_z, V_θ	axial and tangential velocity relative to the film-cooling hole
r-θ-z	coordinate system with the z-axis coinciding with the film-cooling hole's centerline
x-y-z	coordinate system with origin at film-cooling hole exit
y	normal distance from wall
y^+	normalized distance from wall: $y^+ = \rho U_\tau y/\mu$

Greek

α	film-cooling hole inclination angle (Fig. 1)
β	angle between flow direction in cooling channel and flow direction in hot-gas path (Fig. 1)
δ	boundary-layer thickness of hot-gas flow over plate
η	film-cooling adiabatic effectiveness: $\eta = (T_h - T_{ad})/(T_h - T_c)$
ϕ	Overall-cooling effectiveness: $\phi = (T_h - T_w)/(T_h - T_c)$
ρ	density
ρ_c	density of cooling flow
ρ_h	density of hot gas flow
ρ_j	average density of flow in film-cooling hole: $\rho_j = (\rho_{j,in} + \rho_{j,out})/2$
$\rho_{j,in}$, $\rho_{j,out}$	density of cooling jet at film-cooling hole's inlet and outlet
μ_i	dynamic viscosity, where i = h denote hot-gas and i = c denotes cooling air
τ_w	wall shear stress

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Figure Captions List

Fig. 1	(a) Plenum fed. (b) Crossflow fed.
Fig. 2	Experimental configuration studied.
Fig. 3	Coupon with one row of five inclined film-cooling holes.
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Fig. 13 Schematic showing how air in cooling channel approach the film-cooling hole as a function of BR_c and Re_c for a given BR_h

Fig. 14 Adiabatic effectiveness (η).

Fig. 15 Laterally-averaged adiabatic effectiveness ($\overline{\eta}$).

Fig. 16 Overall cooling effectiveness (ϕ).

Fig. 17 Laterally-averaged cooling effectiveness ($\overline{\phi}$).

Table Caption List

Table 1. Flow Conditions in Hot-Gas Path and Cooling Channel

<table>
<tr><th colspan="5">Operating Conditions</th></tr>
<tr><td rowspan="6">hot-gas</td><td rowspan="2">inflow</td><td>mass flow rate
(volume flow rate)</td><td colspan="2">0.3175 kg/s
(33,068 SCFH)</td></tr>
<tr><td>temperature</td><td colspan="2">650 K</td></tr>
<tr><td rowspan="4">outflow</td><td rowspan="4">Pressure</td><td colspan="2">BR=0.75</td></tr>
<tr><td>105,900 Pa</td><td>106,429 Pa</td></tr>
<tr><td colspan="2">BR=1.0</td></tr>
<tr><td>105,450 Pa</td><td>106,312 Pa</td></tr>
<tr><td rowspan="3">coolant</td><td rowspan="2">inflow</td><td>mass flow rate</td><td>0.03829 kg/s</td><td>0.01835 kg/s</td></tr>
<tr><td>temperature</td><td>343.35 K</td><td>342.18 K</td></tr>
<tr><td>outflow</td><td>pressure</td><td>111,769 Pa</td><td>110,909 Pa</td></tr>
</table>

Table 2. Summary of Cases Studied

Case*		BR_h	BR_c
1ad	adiabatic wall	0.75	2.6
2ad			5.2
3ad		1.0	2.6
4ad			5.2
1conj	conjugate	0.75	2.6
2conj			5.2
3conj		1.0	2.6
4conj			5.2

***TR = 1.9**

Table 3. Predicted and Measured BR_h, DR_h, and VR_h

Cases		Blowing ratio (BR_h)	Density ratio (DR_h)	Velocity ratio (VR_h)
1conj	**Exp**	0.749	1.884	0.398
	CFD	0.715	1.879	0.381
	Diff.	**4.54%**	**0.27%**	**4.27%**
2conj	**Exp**	0.743	1.885	0.394
	CFD	0.719	1.886	0.381
	Diff.	**3.23%**	**0.05%**	**3.30%**
3conj	**Exp**	1.003	1.889	0.531
	CFD	0.994	1.915	0.519
	Diff.	**0.90%**	**1.38%**	**2.26%**
4conj	**Exp**	0.994	1.882	0.528
	CFD	0.986	1.909	0.517
	Diff.	**0.80%**	**1.43%**	**2.08%**

Table 4. Mass Flow Rates: 1-Hole vs 5-Hole Configurations

Mass flow rate (10^{-2} kg/s)	BR=0.75 (Conjugate)			BR_h=1.0 (Conjugate)		
	5-hole (Exp)	5-hole (CFD)	1-hole (CFD)	5-hole (Exp)	5-hole (CFD)	1-hole (CFD)
coolant inlet	3.838	3.839	3.839	3.838	3.829	3.829
coolant outlet	-	3.665	3.803	-	3.781	3.587
hole in	0.03591	0.03484	0.03536	0.04987	0.04840	0.04839
hole out	0.03591	0.03484	0.03536	0.04987	0.04840	0.04839
hotgas inlet	31.93	31.75	31.75	32.00	31.75	31.75
hotgas outlet	-	31.92	31.79	-	31.80	31.99

Table 5. Mass Flow Rate, Pressure, Density, and Temperature

Adiabatic: BR_h=0.75, TR=1.9

		Mass flow rate (10^{-2} kg/s)		Pressure (10^5 Pa)		Density (kg/m³)		Temperature (K)	
BR_c		2.6	5.2	2.6	5.2	2.6	5.2	2.6	5.2
coolant	In	3.839	1.919	1.093	1.094	1.112	1.111	342.4	343.1
	Out	3.801	1.881	1.083	1.091	1.102	1.108	342.4	343.1
hole	In	0.03792	0.03775	1.083	1.088	1.103	1.106	342.1	342.7
	Out	0.03792	0.03775	1.058	1.064	1.058	1.079	349.3	343.7
hot gas	In	31.75	31.75	1.062	1.065	0.5693	0.5706	650.2	650.2
	Out	31.79	31.79	1.062	1.064	0.5692	0.5706	649.7	649.7

Adiabatic: BR_h=1.0, TR=1.9

		Mass flow rate (10^{-2} kg/s)		Pressure (10^5 Pa)		Density (kg/m³)		Temperature (K)	
BR_c		2.6	5.2	2.6	5.2	2.6	5.2	2.6	5.2
coolant	In	3.829	1.919	1.107	1.112	1.126	1.129	342.5	343.1
	Out	3.778	1.868	1.098	1.109	1.113	1.126	343.5	343.1
hole	In	0.05104	0.05099	1.093	1.101	1.114	1.120	341.7	342.3
	Out	0.05104	0.05099	1.051	1.063	1.049	1.086	349.5	341.0
hot gas	In	31.75	31.75	1.055	1.064	0.5656	0.5701	650.1	650.2
	Out	31.80	31.80	1.055	1.063	0.5657	0.5703	649.5	649.5

Conjugate: BR_h=0.75, TR=1.9

		Mass flow rate (10^{-2} kg/s)		Pressure (10^5 Pa)		Density (kg/m³)		Temperature (K)	
BR_c		2.6	5.2	2.6	5.2	2.6	5.2	2.6	5.2
coolant	In	3.839	1.919	1.093	1.095	1.112	1.111	342.4	343.1
	Out	3.803	1.884	1.083	1.091	1.085	1.083	348.0	351.3
hole	In	0.03536	0.03554	1.083	1.088	1.012	1.011	375.6	378.7
	Out	0.03536	0.03554	1.058	1.064	0.9233	0.9289	400.1	400.7
hot gas	In	31.75	31.75	1.062	1.065	0.5692	0.5706	650.2	650.2
	Out	31.79	31.79	1.062	1.064	0.5701	0.5712	648.8	649.0

Conjugate: BR_h=1.0, TR=1.9

		Mass flow rate (10^{-2} kg/s)		Pressure (10^5 Pa)		Density (kg/m³)		Temperature (K)	
BR_c		2.6	5.2	2.6	5.2	2.6	5.2	2.6	5.2
coolant	In	3.829	1.919	1.108	1.112	1.127	1.129	342.5	343.1
	Out	3.587	1.871	1.098	1.109	1.099	1.100	348.0	351.3
hole	In	0.04839	0.04843	1.093	1.101	1.037	1.038	369.8	373.3
	Out	0.04839	0.04843	1.051	1.063	0.9412	0.9615	389.9	387.4
Hot gas	In	31.75	31.75	1.055	1.064	0.5655	0.5701	650.1	650.2
	Out	31.99	31.80	1.055	1.063	0.5665	0.5709	648.8	648.9

Table 6. Discharge Coefficient for Adiabatic and Conjugate

BR_h=0.75, TR=1.9

	Input		Output				
	BR_h	BR_c	VR_h	VR_c	I_h	I_c	C_D
conjugate	0.75	2.6	0.44	2.99	0.33	7.77	0.563
		5.2		5.96		30.98	0.591
adiabatic	0.75	2.6	0.40	2.68	0.3	7.0	0.583
		5.2		5.29		27.5	0.589

BR_h=1.0, TR=1.9

	Input		Output				
	BR_h	BR_c	VR_h	VR_c	I_h	I_c	C_D
conjugate	1.0	2.6	0.57	2.96	0.57	7.70	0.592
		5.2		5.87		30.55	0.612
adiabatic	1.0	2.6	0.52	2.71	0.52	7.04	0.604
		5.2		5.32		27.67	0.620