International Communications in Heat and Mass Transfer 96 (2018) 12–19

Contents lists available at ScienceDirect

International Communications in Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ichmt

Investigation of boundary layer thickness and turbulence intensity on film cooling with a fan-shaped hole by direct numerical simulation

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A R T I C L E I N F O

Keywords:
Film cooling
Direct numerical simulation
Boundary layer thickness
Turbulence intensity

A B S T R A C T

Effects of the mainstream boundary layer thickness and the turbulence intensity on film cooling under low Reynolds number conditions are studied in this work by the direct numerical simulation (DNS). In other to solve low-speed compressible flow problems, several methods of Roe scheme, preconditioning, dual time stepping, and LUSGS are adopted to solve governing equations. Results reveal that a horseshoe vortex appears with a thicker mainstream boundary layer, and thus the lateral coverage of the coolant fluid has increased significantly. In addition, the existence of turbulence intensity eliminates the blow-off phenomenon, which happens in a thin mainstream boundary layer condition and enhances the film cooling effectiveness.

1. Introduction

In order to improve the efficiency of a gas turbine engine, increasing the temperature of working fluids is one of the solutions in a modern gas turbine engine. However, increasing the temperature of the working fluid leads to an extreme working environment with high temperature and pressure, which causes the turbine blades to be broken easily. Therefore, film cooling technology has usually been used on turbine blades to protect the blades from the extreme working fluids.

Several parameters, such as blowing ratio, density ratio, geometry of film cooling holes, mainstream turbulence intensity and Reynolds number, etc., are always used to study and analyze the film cooling effectiveness. In the previous study (Fu et al. [1]), the effects of mainstream Reynolds number and turbulence intensity on a fan shaped cooling hole had been studied. Results showed that the coolant jet with a low Reynolds number was not scattered laterally by the divergent delivery channel. Therefore, the film cooling effectiveness gained was much lower than that of a high mainstream Reynolds number, because the coolant jet has the lateral development inside the divergent delivery channel. Furthermore, when the turbulence intensity was increased, the coolant jet with a low mainstream Reynolds number tended to separate from the wall easily, which caused a worse film cooling effectiveness to be obtained. However, when the mainstream Reynolds number was increased, the variation of the turbulence intensity did not have a significant influence on the film cooling behavior. Nevertheless, besides the mainstream Reynolds number and the turbulent intensity, the mainstream boundary layer thickness is also recognized as an important factor for the behavior of the film cooling. Regretfully, effects of the boundary layer thickness on the film cooling effectiveness was not investigated in the previous study due to the limitation of the content.

Zhong et al. [2] studied the effects of approaching boundary layer on an inclined jet in cross flow. In this study, a laminar approaching boundary layer and a turbulent approaching boundary layer had been applied on the mainstream. A horseshoe vortex appeared near the cooling hole with a laminar approaching boundary layer, and enhanced the merging of the jet and the mainstream. Therefore, the lateral distribution of the jet near the cooling hole increased with a laminar approaching layer. However, with a turbulent approaching boundary layer, the horseshoe vortex was not evident. Thus, the film cooling effectiveness was then reduced near the cooling holes. Liess [3] investigated effects of the boundary layer displacement thickness of the mainstream on film cooling with round cooling holes. Results showed that the film cooling effectiveness decreased considerably for a thicker boundary layer. Anderson et al. [4] also studied the effects of boundary layer displacement thickness with shaped cooling holes. In this study, in general, the thickest boundary layer produced the highest film cooling effectiveness. The above results contrasted with results from the study [3] for round holes, and Anderson et al. [4] concluded that this might be due to the separation of coolant jets was less of an issue for shaped holes.

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https://doi.org/10.1016/j.icheatmasstransfer.2018.05.017

Nomenclature

D	Width of the shaped film cooling hole (m)
e	Internal energy ($J \cdot kg^{-1}$)
I^+	Normalized momentum flux $I^+ = \frac{(\rho U)_{Local}}{(\rho U)_{Max}}$
k	Thermal conductivity ($W \cdot m^{-1} \cdot K^{-1}$)
L	Distance between the leading edge of the wall and the cooling hole (m)
L_D	Length of the delivery channel (m)
M	Blowing ratio $M = \frac{\rho_C U_C}{\rho_M U_M}$
Ma_m	Mainstream Mach number
N_x, N_y, N_z	Number of grids in X, Y, and Z directions
P	Pressure (Pa)
Pr	Prandtl number
R	Gas constant ($J \cdot kg^{-1} \cdot K^{-1}$)
Re_M	Mainstream Reynolds number $Re_M = \frac{\rho_M U_M D}{\mu_M}$
Re_C	Coolant stream Reynolds number $Re_c = \frac{\rho_c U_c D}{\mu_c}$
T	Temperature of fluid (K)
T_C	Inlet temperature of coolant (K)
T_M	Inlet temperature of mainstream (K)
Tu	Turbulence intensity
u, v, w	Velocities in X, Y, and Z directions ($m \cdot s^{-1}$)
u_τ	Friction velocity ($m \cdot s^{-1}$) $u_\tau = \sqrt{\frac{\tau_w}{\rho}}$
U	Velocity of the fluid ($m \cdot s^{-1}$)
U_C	Velocity of coolant stream at the inlet of delivery channel ($m \cdot s^{-1}$)
U_M	Inlet velocity of mainstream ($m \cdot s^{-1}$)
x, y, z	Cartesian coordinates
x^+, y^+, z^+	Non-dimensional wall distances in X, Y, and Z direction $\frac{\rho u_\tau x}{\mu}, \frac{\rho u_\tau y}{\mu}, \frac{\rho u_\tau z}{\mu}$
X, Y, Z	Dimensionless Cartesian coordinates

Greek symbols

α	Inclination angle (∘)
β	Diffuser angle (∘)
θ	Non-dimensional temperature $\theta = \frac{T_M - T_{local}}{T_M - T_C}$
θ^*	Boundary layer momentum thickness (m)
δ^*	Boundary layer displacement thickness (m)
γ	Specific heat ratios
η	Film cooling effectiveness $\eta = \frac{T_M - T_{surface}}{T_M - T_C}$
μ	Viscosity ($kg \cdot m^{-1} \cdot s^{-1}$)
μ_C	Viscosity of the coolant stream at inlet condition ($kg \cdot m^{-1} \cdot s^{-1}$)
μ_M	Viscosity of the mainstream at inlet condition ($kg \cdot m^{-1} \cdot s^{-1}$)
ρ	Density ($kg \cdot m^{-3}$)
ρ_C	Density of the coolant stream at inlet condition ($kg \cdot m^{-3}$)
ρ_M	Density of the mainstream at inlet condition ($kg \cdot m^{-3}$)
τ_w	Wall shear stress (Pa)

As mentioned above, the boundary layer thickness really is an important factor for the behavior of film cooling as well. However, to the authors' best knowledge, investigations into the effects of the boundary layer thickness on film cooling with shaped film holes are relatively few in the existing literature. Also, the most effective method to vary the boundary layer thickness is to change the distance, L, from the leading edge of the wall to the cooling hole. Thus, the aim of this study is to investigate effects of variations of L and the turbulence intensity, Tu, on the film cooling effectiveness for a fan-shaped cooling hole. The direct numerical simulations (DNS) conducted in this study follow the configuration of the previous study [1]. In addition, in order to economize calculation time further, two parallel computing methods, open multi-processing (open MP) and message passing interface (MPI), are implemented in the current calculation. Results show that the increment of the distance L thickens the thickness of the mainstream boundary layer, and it induces the formation of the horseshoe vortex around the cooling hole, which increases the lateral coverage of the coolant fluid. As a result, the thicker mainstream boundary layer enhances the lateral averaged film cooling effectiveness. However, with the existence of the horseshoe vortex, the influence of turbulence intensity is not remarkable on the film cooling effectiveness.

2. Physical model and governing equations

The physical model composed of a semi-infinite wall, a delivery channel, and a coolant channel is shown in Fig. 1. A uniform flow with the velocity, U_M, and the temperature, T_M, flows over the semi-infinite wall, of which the wall is adiabatic and the lengths in X and Z directions are $50D$ and infinite, respectively. The delivery channel is connected to the wall at a distance L away from the leading edge of the wall. Coolant fluid flows into the delivery channel with the averaged velocity, U_C, and the temperature, T_C. The delivery channel is connected to the coolant channel at a distance $10D$. In order to investigate effects of the mainstream boundary layer on the film cooling effectiveness exclusively and economize on computing time, a computational domain is designed on the wall, and it forms the main channel shown in Fig. 1. The width and height of the main channel are $14D$ and $5D$, respectively. Details of the size of the channels are shown in Fig. 1. The blowing ratio of all the simulations conducted in the present study is 1.5. The mainstream Reynolds number, Re_M, is 3600, and the coolant stream Reynolds number, Re_C, is 7860. Temperature of the mainstream fluid, T_M, is 350 K, and temperature of the coolant fluid, T_C, is 200 K. More details of the film cooling parameters is indicated in Table 1.

Sizes of the computation domain on the wall are $50D \times 5D \times 14D$ ($X \times Y \times Z$). The inlet boundary condition of the main channel is set as constant velocity inlet with two conditions of 0% and 5% turbulence intensity, and the outlet boundary condition is non-reflecting boundary condition derived by Fu et al. [5].

For the purpose of investigating the effects of varying boundary layer thickness on the film cooling performance, three different lengths ($L = 1D$, $L = 9D$ & $L = 18D$) are used in this study. As shown in Fig. 2, the dark gray zone on wall is the test surface where the time averaged film cooling effectiveness is calculated, and the distance between the leading edge of the wall and the cooling hole, L, is varied from $1D$ to $18D$. As a result, variations of L naturally causes the boundary layer thicknesses to be varied. The boundary layer thickness is indicated by the momentum thickness, θ^*, and the displacement thickness, δ^*. Formulas used to calculate the boundary layer thickness for compressible flow are indicated as follows.

$$\theta^* = \int_0^{L_f} \frac{\rho(y) U(y)}{\rho_{L_f} U_{L_f}} \left(1 - \frac{U(y)}{U_{L_f}}\right) dy \tag{1}$$

$$\delta^* = \int_0^{L_f} \left(1 - \frac{\rho(y) U(y)}{\rho_{L_f} U_{L_f}}\right) dy \tag{2}$$

where L_f is the thickness of the mainstream boundary layer, and it is varied from $L_f = 0.08D$ ($L = 1D$) to $L_f = 0.23D$ ($L = 9D$) and $L_f = 0.31D$ ($L = 18D$). The distributions of L and the boundary layer thickness of the simulations conducted in this study are shown in Table 2.

Several assumptions are made in the present simulations and indicated as follows.

(1) Properties of fluids follow the equation of state of an ideal gas.