1. Introduction

In order to improve the engine power and thermal efficiency, modern gas turbines employ various cooling techniques for turbine blades, such as internal cooling, impinging jet cooling, and film cooling. Film cooling (Goldstein, 1971) is a most popular external cooling technique, which uses discrete holes located on external surface of a turbine blade to eject coolant air to generate a coolant layer between the blade surface and hot gas.

Due to simple configuration and manufacturing convenience, cylindrical holes have been used widely for film cooling. Burd et al. (1998) obtained flow measurements with different lengths of a cylindrical film-cooling hole. They suggested that length-to-diameter ratio ($L/D$) has strong effect on film-cooling performance. Lutum and Johnson (1999) investigated the influence of $L/D$ on film-cooling effectiveness experimentally. The results revealed that the film-cooling effectiveness of a cylindrical hole generally decreased with decreasing $L/D$ for a relatively short hole configuration ($L/D < 5.0$), but changed insignificantly with the hole length in a range of $5 < L/D < 18$. Yuen and Martinez-Botas (2003) examined the effect of streamwise angle of film-cooling holes on film-cooling effectiveness with three streamwise angles ($30^\circ$, $60^\circ$, and $90^\circ$) experimentally. Ligrani et al. (1994) investigated the effect of a compound angle on film-cooling performance for two staggered rows. They reported that the compound angle enhances uniformity of film-cooling effectiveness by increasing the lateral momentum. Lee and Kim (2009) optimized a cylindrical hole to enhance film-cooling effectiveness with two design variables, $L/D$ and ejection angle of the hole, using various surrogate models. The optimized cylindrical hole showed 3.6% improvement in the spatially-averaged film-cooling effectiveness compared to a reference design.

However, because the film-cooling performance of simple cylindrical holes is insufficient, especially, at high level of turbine inlet temperature, various shapes of hole exit have been developed to improve the film cooling performance. Saumweber and Schulz (2008) studied the cooling performance of a fan-shaped film-cooling hole by varying the inclination angle, expansion angle of diffuser, and length of cylindrical part. Gritsch et al. (2005) also investigated the
fan-shaped holes with various geometric parameters, such as $L/D$, hole inlet-to-outlet area ratio, hole coverage ratio, hole pitch ratio and hole compound angle. Among them, the hole pitch ratio had the largest effect on the film-cooling effectiveness. Lee and Kim (2010) performed an optimization of a fan-shaped hole with three design variables. The optimum design of the fan-shaped hole showed a film-cooling effectiveness enhanced by 28% compared to a reference fan-shaped hole.

Sargison et al. (2002) proposed a “console” hole (converging-slot hole), which showed similar film-cooling performance and reduced aerodynamic loss in comparison with a fan-shaped hole. Zhang and Hassan (2006) introduced a novel film-cooling configuration called a “louver” hole. They compared the film-cooling performance of the louver hole with that of cylindrical, console, and fan-shaped holes. As the results, the louver hole showed similar performance to the console hole and much better performance than the cylindrical and fan-shaped holes. Lu et al. (2007) compared the film-cooling performances among various film-cooling hole configurations, such as cylindrical, trenched, cratered, slot, and crescent holes. Liu et al. (2010) presented new film-cooling hole geometries, i.e., dumbbell-shaped and bean-shaped holes. Their research indicated that the thermal performances of the proposed two holes were better than those of crescent, louver, cylindrical, and fan-shaped holes with the same boundary conditions. A novel shaped film-cooling hole called “leaf-shaped” hole was developed by Lee and Kim (2012). Film-cooling performance of the leaf-shaped hole was evaluated in comparison with that of a fan-shaped hole. The leaf-shaped hole yielded similar effectiveness to that of a fan-shaped hole at a blowing ratio of $M = 0.5$, however, the effectiveness was significantly enhanced as the blowing ratio increased.

The above mentioned shaped film-cooling holes, have high film-cooling performances, but, are difficult to be manufactured. For this reason, combined-hole systems using double-jet (Kusterer et al. 2006) and anti-vortex (Heidmann, 2008) holes have been developed. Kusterer et al. (2006) proposed a double-jet film-cooling system, which combines two cylindrical holes with different compound angles. The double-jet film-cooling system showed an improved film-cooling effectiveness due to anti-vortex pair induced by the double jets. Heidmann (2008) introduced an anti-vortex hole, which comprises of one primary cylindrical hole and two cylindrical holes branching out from a primary hole. The anti-vortex hole generated an anti-kidney vortex pair located on both side of a kidney vortex.

As mentioned above, various hole designs have been studied to improve film-cooling performance, however, there was no systematic study of bended film-cooling holes (Ghorab and Hassan, 2010). In the present study, a bended cylindrical film-cooling hole was investigated and optimized to maximize its film-cooling performance based on the authors’ previous parametric study (Kim and Kim, 2017). The film-cooling performance and flow structure of the bended hole were analyzed using three-dimensional (3-D) Reynolds-averaged Navier-Stokes (RANS) equations. Optimization of the bended film-cooling hole was performed to enhance film-cooling effectiveness using surrogate model-based optimization techniques. Three design variables related to the shape of the bended film-cooling hole were chosen by referring to the results of a parametric study.

2. Numerical analysis

The governing differential equations including the continuity, 3-D RANS, and energy equations were solved by using a commercial code ANSYS-CFX 15.0® (2014) to analyze the flow and heat transfer through and downstream of the film-cooling hole. In this study, the shear stress transport (SST) turbulence model (Menter, 1994; Bardina et al., 1997) was used in numerical analysis. The SST model was established by combining the k-ω and k-ε models with a blending function. In the near-wall region, k-ω model is used, and, in the bulk-flow region, k-ε model is activated in this model. Results of numerical analysis using SST model for flat-plate film cooling showed good agreements with experimental data for cylindrical, fan-shaped, and laidback fan-shaped holes (Lee and Kim, 2009; Lee and Kim, 2010; Lee and Kim, 2012).

Figure 1 shows the geometry of the bended cylindrical film-cooling hole considered in the present study. The bended hole is constructed by connecting two cylindrical holes with different injection angles ($\theta_1$, $\theta_2$). Diameter ($D$) of the bended cylindrical hole is 12.7 mm, which is same as that used by Lu et al. (2007). The diameters of the two cylindrical parts are identical. Computational domain consists of a main channel of hot gas, a coolant plenum with closed end, and a film-cooling hole. The distance from the inlet of the main channel to the center of the hole exit is 381 mm. The total lengths of the main channel and the plenum are 889 mm and 508 mm, respectively. The height of the film-cooling hole ($H$) in z direction is 25.4mm. The heights of the main channel and the plenum are 101.6 mm and 76.2 mm, respectively.
Fig. 1 Geometry of bended film-cooling hole and computational domain (Kim and Kim, 2017)

The boundary conditions (Fig. 1(c)) were applied to match those of the experiment performed by Lu et al. (2007) as closely as possible. The working fluid was assumed to be ideal gas (air). Uniform velocity in y-direction was prescribed at the mainstream inlet, and the Reynolds number, based on the diameter of the film-cooling hole, was 11,000. The inlet temperature of the mainstream was kept constant at 321K. At the inlet of the coolant plenum, mass flow rate boundary condition was applied to fix the blowing ratio at \( M = 0.5 \) with the temperature of 296 K. The blowing ratio is defined as the mass flux ratio of the coolant to the mainstream. At the outlet of the main channel, atmospheric pressure was specified. An no-slip and adiabatic conditions were applied to all the walls. Since an array of film-cooling holes was considered, periodic conditions were used at the side boundaries of the main channel. The distance between adjacent holes was 3D. The density ratio of the coolant to the hot gas in the main stream was 1.083, and the turbulence intensity at the main channel inlet was 3%.

An example of the computational grid is shown in Fig. 2. ANSYS ICEM 15.0® was used to construct a hybrid grid in the computational domain. The grid in the main channel and the plenum including the film-cooling hole consists of tetrahedral and hexahedral meshes, respectively. The meshes are clustered near the cooling surface in the main channel and the upper surface of the coolant plenum. Additionally, sufficiently dense meshes are concentrated inside the film-cooling hole and around the exit of the hole to resolve the interaction of the ejected cooling jet with mainstream of the hot gas. Prism layers are stacked at the wall to settle the high-velocity gradient near the wall and the first prism layer adjacent to the wall is placed at \( y^+ < 1 \) to implement the low-Re SST model.

Convergence of numerical solution was assumed when the root-mean-square residual values of all flow parameters fell below \( 1.0 \times 10^{-6} \), and imbalances of mass and energy in the entire computational domain were less than 0.001%. An Intel i7 3.41 GHz PC was used for the computation. The calculation for a single analysis was terminated within 3000 iterations, which took approximately 36 to 48 hr of computational time depending on the geometry.

3. Design variables and objective function

In order to optimize the bended film-cooling hole, three geometric parameters, the injection angle of the lower cylindrical part (\( \theta_1 \)), the injection angle of upper cylindrical part (\( \theta_2 \)), and height of the bending point were selected as design variables for optimization. The ranges and reference values of these design variables are listed in Table 1. For the reference design, \( \theta_2 \) is 30 degree, which is the most commonly used value of the film-cooling hole angle. Reference value of \( \theta_1 \) was selected as 60 degree, which is the middle value between 30 degree (reference \( \theta_2 \)) and 90 degree (vertical hole). And, \( h/D \) value of 1.0 was chosen as the reference value. The design ranges were determined from a preliminary parametric study.

Spatially-averaged film-cooling effectiveness (\( \eta_s \)) was selected as an objective function, which is defined as follows;
Fig. 2 An example of computational grid system 
(Kim and Kim, 2017).

\[
\eta_\alpha \left( \frac{x}{D}, \frac{z}{D} \right) = \frac{\int_0^{2\alpha} \int_{-1.5}^{1.5} \eta \left( \frac{x}{D}, \frac{z}{D} \right) d\left( \frac{x}{D} \right) d\left( \frac{z}{D} \right)}{20 \times 3} 
\]  

(1)

where

\[
\eta \left( \frac{x}{D}, \frac{z}{D} \right) = \frac{T_{aw} \left( \frac{x}{D}, \frac{z}{D} \right) - T_h}{T_c - T_h} 
\]  

(2)

Here, \( T_{aw} \) is adiabatic wall temperature, and \( T_h \) and \( T_c \) are mainstream temperature in the main channel and coolant temperature, respectively. The spatially-averaged film-cooling effectiveness was averaged over an area of \( 3D \) in width and \( 20D \) in streamwise length on the cooling surface.

### 4. Optimization methods

Optimization procedure used in the present work is shown in Fig. 3. An objective function and design variables are selected, and a parametric study is performed to determine a design space. Some design points are required for filling the design space to build a surrogate model to approximate the objective function. Latin hypercube sampling (Sacks et al., 1989; Afzal et al., 2017), which is a kind of design-of-experiment technique, generates design points. Generated points ensure that the all portions of the design space are represented. Then, the objective function values at these design points are obtained by numerical analysis. Based on these values, the surrogate model is constructed, and an optimal point is sought by sequential quadratic programming (SQP), which is a kind of gradient-based search algorithm.

In the present work, surrogate modeling was employed to approximate the objective function in order to reduce the computing time required for evaluating the objective function values in optimization. In this work, Kriging (KRG) model (Martin and Simpson, 2005) was used for the surrogate modeling.

The KRG model (Martin and Simpson, 2005) uses an interpolating meta-modeling technique, which consists of two parts as follows:

\[
F(x) = f(x) + z(x) 
\]  

(3)

where \( x \) is a vector of design variables and \( f(x) \) is a known function approximating the trend of the design space called the “global” trend model, and \( z(x) \) generates a localized deviation to interpolate the sampled data points through quantification of the correlation of points with a Gaussian correlation having a zero mean and nonzero covariance.

SQP (using the function \( fmincon \) in MATLAB R2015a (2015)) was employed to find an optimum point on the constructed surrogate model. SQP is a robust algorithm for nonlinear continuous optimization based on a generalization of Newton’s method to multiple dimensions. In order to avoid local minimum or maximum, a multiple initial guess method was used at multiple locations in the design space. The best ten points among the design points were used as the initial guesses.

### 5. Results and discussion

A grid-independency test and validation of numerical solution in comparison with experimental data (Saumweber and Schulz, 2008; Lu, 2007; Saumweber et al., 2003) were performed for laterally averaged film-cooling effectiveness in an authors’ previous study (Kim and Kim, 2017), as shown in Figs. 4 and 5, respectively. Four different numbers of nodes in a range of 1.3 - 5.9 million, were used for the grid test. From the results shown in Fig. 4, it can be seen that the results of 4.1 million grid nodes show no significant difference with those of 5.9 million grid nodes. Therefore, the optimum number of grid nodes was selected as 4.1 million. The numerical results for typical cylindrical holes are
compared with two different experimental data obtained by Lu et al. (2009) and Saumweber et al. (2003, 2008) in Fig. 5, which represents the distributions of the laterally averaged film-cooling effectiveness at $M = 0.5$. The numerical results agree well with the experimental data obtained by Saumweber et al. (2003, 2008). The results also show good agreement with the experimental data obtained by Lu et al. (2009) in the region $x/D > 8$, but some deviations are found near the hole. Difficulty in achieving perfect adiabatic condition in the experiments and limitation of the turbulence closure model in predicting transition region might be reasons for the deviations.

A parametric study was performed with the three design variables in the previous study (Kim and Kim, 2017) to investigate the geometric effects of the bended film cooling hole on cooling performance and also to determine the design range of each variable. Figure 6 presents the results of the parametric study, i.e., effects of the three design variables on the spatially-averaged film-cooling effectiveness defined by Eq. (1). When each variable was varied, the other variables were kept constants as the reference values shown in Table 1. As shown in Fig. 6, the spatially averaged film-cooling effectiveness increases with increasing $\theta_1$ until it reaches a maximum at $\theta_1 = 75^\circ$. The spatially averaged film-cooling effectiveness increases with $\theta_2$, but the increasing rate decreases with $\theta_2$. In case of $h/D$, the spatially averaged film-cooling effectiveness has a maximum at $h/D = 0.75$, and decreases thereafter with nearly constant decreasing rate. Maximum variations of the effectiveness for the three design variables in the tested ranges are almost same (around 0.1). This means that the sensitivities of the spatially averaged film-cooling effectiveness are similar for the three variables.

The bended cylindrical film-cooling hole was optimized by using KRG model in order to improve the spatially averaged film-cooling performance at a blowing ratio of 0.5. Twenty-seven design points generated by the LHS and the objective function values calculated using 3-D RANS equations at the design points, are shown in Table 2. KRG model was constructed based on these objective function values. Results of the optimization are listed in Table 3. The values of the design variables and the objective function for optimum design are compared with those for the cylindrical hole and the reference design in this table. The cylindrical hole indicates a straight hole with circular cross section. The optimum objective function value predicted by KRG model shows good agreement with the value calculated by RANS analysis.
Table 2: Values of design variables and objective function at LHS designs.

| Design | \(\theta_1\) (°) | \(\theta_2\) (°) | \(h/D\) | Objective function |
|--------|-----------------|-----------------|--------|-------------------|
| 01     | 52.158          | 21.324          | 0.733  | 0.2253            |
| 02     | 59.046          | 57.548          | 0.630  | 0.2071            |
| 03     | 37.266          | 48.896          | 0.779  | 0.1664            |
| 04     | 40.026          | 36.624          | 1.165  | 0.2068            |
| 05     | 86.490          | 27.568          | 1.217  | 0.2376            |
| 06     | 89.472          | 33.476          | 1.461  | 0.1963            |
| 07     | 66.756          | 27.568          | 1.217  | 0.2069            |
| 08     | 66.756          | 27.568          | 1.217  | 0.2069            |
| 09     | 48.786          | 39.704          | 1.422  | 0.2129            |
| 10     | 55.506          | 53.780          | 1.470  | 0.2317            |
| 11     | 85.038          | 51.064          | 1.173  | 0.2893            |
| 12     | 31.134          | 47.044          | 0.547  | 0.2116            |
| 13     | 33.060          | 11.122          | 0.828  | 0.2094            |
| 14     | 41.826          | 51.880          | 1.031  | 0.1530            |
| 15     | 73.566          | 40.880          | 1.259  | 0.2581            |
| 16     | 78.072          | 29.468          | 0.834  | 0.2116            |
| 17     | 70.026          | 59.084          | 1.291  | 0.2213            |
| 18     | 63.252          | 55.764          | 1.322  | 0.2138            |
| 19     | 63.588          | 42.512          | 0.705  | 0.3210            |
| 20     | 52.416          | 45.812          | 0.595  | 0.2777            |
| 21     | 79.344          | 44.916          | 0.914  | 0.3296            |
| 22     | 47.556          | 24.088          | 1.015  | 0.2202            |
| 23     | 76.086          | 26.096          | 1.088  | 0.2217            |
| 24     | 40.026          | 36.624          | 1.234  | 0.2186            |
| 25     | 57.378          | 25.200          | 0.523  | 0.2097            |
| 26     | 69.696          | 55.308          | 0.877  | 0.3152            |
| 27     | 81.732          | 21.512          | 1.359  | 0.2065            |

Table 3: Results of optimization \((M=0.5)\)

| Design | Design variable | Objective function | Error (%) |
|--------|-----------------|--------------------|-----------|
| Cylindrical hole | \(\theta_1\) (°) | \(\theta_2\) (°) | \(h/D\) | Surrogate Prediction | RANS Calculation | |
| Cylinder | 30 | 30 | - | - | 0.1999 |
| Reference | 60 | 30 | 1.0 | - | 0.2544 |
| Optimum | 82.3 | 51.3 | 0.845 | 0.3447 | 0.3559 | 3.14 |

with a relative error of 3.14%. The optimization achieved a remarkable improvement in the objective function (i.e., the spatially-averaged film-cooling effectiveness) by 39.9% and 78.0% compared to those of the reference design and the cylindrical hole, respectively.

Figure 7 shows the distributions of laterally-averaged film-cooling effectiveness for the straight cylindrical hole, reference design, and optimum design of the bended hole. As shown in this figure, the reference design and the cylindrical hole show similar film-cooling effectiveness just downstream of the hole, but the reference bended-hole design shows much better film-cooling effectiveness in far-downstream region than the cylindrical hole. The optimum design shows great improvement in film-cooling effectiveness just downstream of the hole compared to the other two designs, but the film-cooling effectiveness decreases rapidly from the maximum around \(x/D = 7\) and becomes less than that of the reference design beyond \(x/D = 23\).

Distributions of local film-cooling effectiveness on the film-cooling surface are shown in Fig. 8. The optimum design shows much wider region of high film-cooling effectiveness than cylindrical hole and reference design. On the other hand, the cylindrical hole and reference design show similar widths of the region just downstream of the hole, but
Fig. 8 Distributions of local film-cooling effectiveness on film-cooling surface ($M=0.5$).

Fig. 9 Velocity contours at the hole exits of the cylindrical, reference and optimized designs.
the reference bended-hole design shows increased width of the region in further downstream compared to the cylindrical hole. This is consistent with the trends of the laterally averaged film-cooling effectiveness shown in Fig. 7. The great increase in the laterally averaged film-cooling effectiveness of the optimum design just downstream of the hole is caused by the rapid widening of the high film-cooling effectiveness region on the film-cooling surface.

Figure 10 represents the velocity contours on x-y plane at z/D = 0 for the cylindrical hole, reference and optimum designs. In the cylindrical hole, a skewed vena contracta is created, which is called the jetting effect (Saumweber and Schulz, 2008). The coolant entering the inlet of the hole flows into a narrow jetting region with high velocity near the upper wall of the hole, while a recirculation region is generated near the bottom wall of the hole. Due to the significantly skewed profile of the velocity caused by the jetting effect, there occurs locally high momentum at the upper part of the hole exit, which causes greater penetration of coolant into the mainstream reducing the film-cooling effectiveness. In the reference shape, although the jetting effect is still found, the velocity in the high momentum region decreases and the recirculation region diminishes. The exit profile of the jet is less skewed for reference shape than the cylindrical hole as shown in Fig. 9. In the case of the optimum design, it can be seen that the jetting region moves to the trailing edge of the film-cooling hole and the recirculating region largely reduces.

Figure 12 shows the temperature distributions and streamlines on y-z planes at x/D = 2.5, 5 and 10. The reference design shows the lowest height of the coolant flow. The optimum design shows the highest and the widest low temperature
region among the three tested holes. A pair of kidney vortex is found at the center in each case. It is interesting to see a pair of counter-rotating vortices outside of the kidney vortices from $x/D = 2.5$ in case of the optimum design. These counter-rotating vortices push the coolant to the surface and promote lateral spreading as shown in Fig. 12(c). And, thus, the cross section of the coolant stream becomes a ‘hat’ shape as the flow proceeds downstream. Additional vortices are also shown in the case of the reference design downstream of $x/D = 5$ (Fig. 12(b)). However, in this case, these vortices are not strong enough to spread the coolant stream.

Streamlines on x-y plane upstream of the hole exit are shown in Fig. 13. In the case of the optimized design, unlike the reference design, a small recirculating flow appears just upstream of the film-cooling hole, which is called horseshoe vortex (Fric and Roshoko, 1994). This horseshoe vortex is also shown on both sides and downstream of the hole as shown in Fig. 8(c). Figure 14 shows 3-D streamlines and streamlines on the planes perpendicular to the main stream in the case of the optimized design. The trajectory of the center of the horseshoe vortex is seen in this figure. This trajectory passes
(a) Cylindrical hole

(b) Reference design

(c) Optimized design

$x/D = 2.5$

$x/D = 5$

$x/D = 10$

Fig. 12 Temperature distributions and streamlines on $y$-$z$ planes at $x/D=2.5$, 5 and 10.

(a) Reference design

(b) Optimized design

Fig. 13 Streamlines on $x$-$y$ plane upstream of the hole exit.

Fig. 14 Development of vortices downstream of the optimized bended film-cooling hole.
through the center of the additional vortices located outside of the kidney vortices. Thus, the horseshoe vortex contributes to the generation of the additional counter-rotating vortices which spread the coolant stream as shown in Fig. 12(c).

Figure 15 shows the temperature distributions in the y-direction at \( z/D = 0 \) and \( x/D = 3, 5, \) and 10. As shown in Fig. 15(a), the minimum temperature occurred in the region of \( 0.4 < y/D < 0.6 \) for all cases. However, the temperature is significantly lower in the optimum design than those in the cylindrical hole and reference design. At \( x/D = 5 \), the temperature of the optimum design is lower than those of the other designs throughout the whole \( y \) range. In Fig. 15(c) at \( x/D = 10 \), the temperature of the optimum design becomes higher than that of the cylindrical design in most of the \( y \) range, but is lowest at the wall (\( y/D = 0 \)).

6. Conclusion

A bended film-cooling hole was optimized to improve film-cooling effectiveness at a blowing ratio of 0.5 using surrogate modeling and 3-D RANS analysis with SST model. The numerical model used in this work was validated for cylindrical holes using two different experimental data sets. The injection angles of the lower and upper cylindrical parts, and the height of the bending point were selected as design variables, and their effects on the film-cooling effectiveness were evaluated in a preliminary parametric study. The results showed that the effectiveness is equally sensitive to all the design variables. A single-objective optimization of the bended film-cooling hole was performed to improve the spatially-averaged film-cooling effectiveness by using surrogate model, KRG, which was constructed based on the objective function values calculated at 27 design points selected by LHS in the design space. The KRG surrogate prediction for the objective function shows 3.14% relative error at the optimum point compared to the RANS calculated value. The optimum design improved significantly the spatially-averaged film-cooling effectiveness by 39.9% and 78.0% compared to the reference design and cylindrical hole, respectively. The computational flow field for the optimum bended hole showed that a large lateral velocity component is produced at the hole exit by the bending of the hole and a pair of counter-rotating vortices appear outside of the kidney vortices, which contribute to the lateral spreading of the coolant stream downstream of the hole resulting in enhancement of the film-cooling effectiveness.
NOMENCLATURE

\( D \) diameter of the film-cooling hole

\( DR \) density ratio: \( \rho_c/\rho_h \)

\( H \) height of the film-cooling hole

\( h \) height of the bending point between upper and lower cylindrical part

\( M \) blowing ratio: \( (\rho_c U_c)/(\rho_h U_h) \)

\( L \) length of the film-cooling hole

\( T_{aw} \) adiabatic wall temperature

\( T_h \) hot gas temperature in the main channel

\( T_c \) coolant jet temperature

\( U_c \) velocity of coolant at the hole exit

\( U_\infty \) velocity at the main stream inlet

\( y^+ \) \( y \) in law-of-the-wall coordinate

Subscripts

\( aw \) adiabatic wall

\( \infty \) main stream

\( c \) coolant

\( h \) hot gas

\( 1 \) laterally-averaged

\( s \) spatially-averaged

\( 1 \) lower cylindrical part

\( 2 \) upper cylindrical part

Greek symbols

\( \rho \) density, kg m\(^{-3}\)

\( \theta \) injection angle

\( \eta \) film-cooling effectiveness: \( (T_{aw}-T_\infty)/(T_c-T_\infty) \)

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