A comparison of cylindrical and row trenched cooling holes with alignment angle of 0 degree near the combustor endwall

E Kianpour, C S Nor Azwadi\textsuperscript{a} and I Golshokouh
Department of Thermo Fluid, Faculty of Mechanical Engineering, UTM
81300, Skudai, Johor, Malaysia

E-mail: \textsuperscript{a}azwadi@fkm.utm.my

Abstract. We studied the effects of cylindrical and row trenched cooling holes with alignment angle of 0° at BR=3.18 on the film cooling performance near the endwall surface of a combustor simulator. In this research, a three-dimensional presentation of gas turbine engine was simulated and analyzed with a commercial finite volume package FLUENT 6.2.26 to gain fundamental data. The current study has been performed with Reynolds-averaged Navier-Stokes turbulence model (RANS) on internal cooling passages. This combustor simulator combined the interaction of two rows of dilution jets, which were staggered in the stream wise direction and aligned in the span wise direction. The entire findings of the study declared that with using the row trenched holes near the enwall surface; film cooling effectiveness is doubled compared to the cooling performance of baseline case.

1. Introduction
Gas turbine industries try for higher engine efficiencies [1]. Bryton cycle is a key to this objective [2]. According to this cycle, the turbine inlet temperature should increase to gain more efficiency. However increasing the turbine inlet temperature creates harsh environment for downstream components. So, it is needed to design a cooling technique in this area. Film cooling is the way which is used. In this system, a thin thermal boundary layer such as buffer zone is formed and attached on the protected surface. Cylindrical and trenched cooling holes are two layouts of film cooling. With trenching the cooling holes, the injected coolant is suddenly spread before exiting the cooling holes and entering the main flow and as a result the coolant attached better on the protected surface [3-4].

According to the importance of this research, a broad literature search was conducted to collect the information. Stitzel and Thole [5] indicated that with no dilution, the exit profile was relatively uniform however, the high temperature and low total pressure mainstream flow was found. Kianpour et al.[6-7] simulated the combustor endwall cooling holes with two different layouts and exit section area. The results declared that for both configurations, the temperatures adjacent the wall and between the jets was a while cooler with less cooling holes. From Yiping et al.[8] researches, it is found that the narrow trenched holes were more effective than other trenched cases and baseline. Also they mentioned that row trenches have dominant effect on film cooling performance compared to the individual trenches. Sundaram and Thole [9] and Lawson and Thole [10] studied the effects of trenched depth and width on film cooling performance at the vaneendwall. The results showed that the maximum cooling effectiveness is obtained at the trench depth of 0.80D. According to the importance of this issue, more studies are required. There are questions should be answered: Which type of trenched cooling holes with different alignment angles has more effect on film cooling performance?
In order to measure the validity of the results, a comparison between the data gained from this study and Vakil and Thole\textsuperscript{1} project was made.

2. Methods and Materials

As seen in figure 1, the combustor was a container with a width and height of 111.8cm and 99.1cm. The length of the combustor was 156.9cm and the contraction angle was 15.8°. While the inlet cross-sectional area was 1.11m\textsuperscript{2}, the exit cross-sectional area was 0.62m\textsuperscript{2}. The contraction angle began at X=79.8cm. The combustor simulator involved four stream wise series of film-cooling panels. The starting point of these panels was at 1.6m upstream of the turbine vanes. The length of the first and second panels was 39centimeter and 41cm respectively. However the third and fourth panels were 37 and 43 centimetre in length. The thickness of the panels was 1.27cm. The thermal conductivity (k=0.037 W/mk) was such light that adiabatic surface temperature measurement was possible. There were two rows of dilution holes within the second and third panels of cooling panels. These dilution holes were located at 0.67m and 0.90m downstream of the beginning of the combustor liner panels. The diameter of the first row and second row of dilution holes was 8.5cm and 11.9cm respectively.

![Figure 1. The 3-D view of the combustor simulator.](image)

The centreline of the second row was staggered with respect to the first row of dilution holes. To verify the purpose of this study, a three-dimensional representation of a Pratt and Whitney engine was simulated. The present combustor simulator included two configurations of cooling holes. The first arrangement (baseline or case 1) was designed similar to the Vakil and Thole\textsuperscript{2}. In both cases, the film cooling holes were placed in equilateral triangles. The diameter of the film cooling holes was 0.76cm and drilled at an angle of 30° from the horizontal surface. The length of film cooling holes in the baseline case was 2.5cm. For the second case (case 2) the cooling holes placed within a row trench with alignment angle of 0 degree which is shown in figure 2. Furthermore, the trench depth and width was 0.75D and 1.0D respectively. A global coordinate system (X, Y and Z) was also selected.
The thermal distribution inside a combustor simulator was measured along the specific measurement planes. These measurement planes are shown in figure 3.

In order to get more accurate data and reasonable time consumption, about $8 \times 10^6$ tetrahedral meshes were used and this is in concurred with Stitzel and Thole [5] study. According to the specific flow ratio at the inlet of volume control, inlet mass flow boundary condition was defined. Wall boundary condition and slip less boundary condition were applied to limit the interaction zone between fluid and solid layer. Also at the end of volume control the pressure outlet boundary condition was used. In addition, both cases were completely symmetric along the $X$-$Y$ and $X$-$Z$ planes. According to this issue, symmetry boundary condition $\partial \hat{n} = 0$ was applied. In addition theses equations were used as well.

Continuity equation:

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_i)}{\partial x_i} = -\frac{\partial \rho}{\partial x_i} + \rho \frac{\partial \tau_{ij}}{\partial x_i} + \rho g_i + F_i$$

Momentum equation:

$$\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial (\rho u_i u_j)}{\partial x_j} = -\frac{\partial \rho}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_i} + \rho g_i + F_i$$

Energy equation:
\[
\frac{\partial}{\partial t} (\rho E) + \frac{\partial}{\partial x_i} (\rho u_i E + P) = \frac{\partial}{\partial x_i} \left( K_{eff} \frac{\partial T}{\partial x_i} - \sum_j h_j f_j + u_j (\tau_{ij})_{erf} \right) + S_h \tag{3}
\]

and RNG K-\epsilon equation:

\[
\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_i} (\rho k u_i) = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_i} \right] + P_k - \rho \epsilon \tag{4}
\]

\[
\frac{\partial}{\partial t} (\rho \epsilon) + \frac{\partial}{\partial x_i} (\rho \epsilon u_i) = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_t}{\sigma_\epsilon} \right) \frac{\partial \epsilon}{\partial x_i} \right] + C_1 \epsilon \frac{\rho}{k} P_k - C_2 \epsilon \rho \frac{\epsilon^2}{k} \tag{5}
\]

To understand the thermal field results, the quantities should be defined. Film cooling effectiveness is defined as below:

\[
\eta = \frac{T-T_{\infty}}{T_{C}-T_{\infty}} \tag{6}
\]

3. Findings and Discussion

The comparison has been made for baseline case among the numerical and experimental results by Stitzel and Thole[5] and Vakil and Thole [2], and the current study. Figure 4 presents the comparison of film cooling effectiveness for plane 1p and 2p at Y/W=0.4. The deviation was calculated as follows:

\[
\%Diff = \sum_{i=1}^{n} \frac{x_i-x_i,benchmark}{x_i,benchmark} \times 100 \tag{7}
\]

According to this formula, the deviation was equal to 8.34% and 9.76% compared to findings for plane 1p and equal to 11.96% and 13.36% compared to numerical and experimental data for plane 2p.

![Figure 4. Film cooling effectiveness comparison for plane 1p and 2p along Y/W=0.4.](image)

The distribution of film cooling effectiveness of plane 1p is shown in figure 5a (case 1) and 5b (case 2). Note that, at the right side of figure 5a and 5b (50cm<Y<54cm) thermal field contours show that film cooling is being entrained by upward motion of dilution jet. The v and w velocity vectors of plane 1p are shown as well. From the middle of the temperature distribution contour, a significant movement of vortexes toward the left and right sides is found. This is the effect of dilution injection on the thermal behaviour of flow.
Figure 5. The vectors of v and w with film cooling effectiveness contours for plane 1p (a) case 1 (b) case 2.

Figure 6a (baseline) and 6b (case 2) show the film cooling distribution for plane 2p. It is declared that the rotating flow is seen on the left side of figure and it is entrained along the span wise direction. However, this rotating area is weaker for the trenched case. Also, at the right side of figure 6a the hot gases covered more extended area in comparison with trenched case. Lastly, these figures show the v and w velocity vectors superimposed on the thermal field contours of this measurement plane. The sweeping of the coolant toward the second row of dilution jet is visible.

Figure 6. The vectors of v and w with film cooling effectiveness contours for plane 2p (a) baseline (b) case 2.

The variation of film cooling effectiveness for different measurement planes at Y=30cm and along Z axis is shown in figure 7.

Figure 7. The variation of film cooling effectiveness for different measurement planes.
It is declared that for the three first measurement planes, case 2 performed better. Also, the film cooling effectiveness difference between the baseline and trenched hole is growing step by step from plane op to plane 2p. While, at the end of first cooling panel, film cooling effectiveness is increasing 40% by trenching cooling holes, at the end of second cooling panel, this ratio is equal to 66%. Also, it is seen that film cooling effectiveness increased intensively (about 92%) at further distance compared to baseline at the end of third cooling panel, although, at the first of this panel, this ratio increased only 21%.

4. Conclusion and Recommendation

In this study a three-dimensional representation of a Pratt and Whitney engine was analyzed. To sum up, the usage of trenched cooling holes significantly to development of the film cooling layer. Also, the central part of the plane 2p showed the intense penetration of the coolant and a thick film cooling layer creation in the trenched case. However, the temperature adjacent the wall and between the jets was cooler with trenching the cooling holes because by trenching the cooling holes, coolant spread better in this area. Initially, the results declared that at the end of three first measurement cooling panels, trenching cooling holes has intense effect on film cooling. Based on the results, there are several recommendations to consider. In future research within this area, different configurations of trenched cooling holes and baseline case should be considered for different cooling panels. The effects of different blowing ratios on the film cooling performance should notice as well.

Acknowledgements

The authors would like to thanks the Universiti Teknologi Malaysia and Ministry of Higher Education, Malaysia for sponsoring this research. This research is supported by research grant Vote 4F114 and 4L074.

References

[1] Ibrahim T K and Rahman M M 2013 J Mech Eng Sci 4 383
[2] Vakil S S and Thole K A 2005 J. Eng. Gas Turb. Power 127 257
[3] Smajevic I 2010 Inter J Automot Mech Eng 1 1.
[4] Wandel A P, Noor M M and Yusaf T F 2012 Inter J Automot Mech Eng 6 731
[5] Sitzel S and Thole K A 2004 J. Turbomach. 126 122.
[6] Kianpour E, Azwadi C S N and Agha S M B M 1992 Jurnal Teknologi 58 5
[7] Kianpour E, Azwadi C S N and Agha S M B M Int. J. Heat Mass Tran. 61 389
[8] Yiping L, Dhungel A, Ekkad S V and Bunker R S 2009 J. Turbomach. 131 1
[9] Sundaram N and Thole K A 2008 J. Turbomach. 130 1
[10] Lawson S A and Thole K A 2012 J. Turbomach. 134 1