# Thermal Analysis of Wedge Duct with in-Line Pin-Fin Configuration

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***Abstract*— This paper presents a Conjugate Heat Transfer (CHT) analysis of wedge duct of the trailing edge of the turbine blade for flow and heat transfer co- efficient characteristics for 2 different cases. The Reynolds-averaged Navier–Stokes equations, coupled with a k-ε turbulent model ,are considered and hence solved. Reynolds numbers (Re) of 10,000, 20,000, 30,000, 40,000 and 50,000 are considered to determine the effect of flow parameter on the pressure drop and heat transfer. The variation in the end wall and pin–fins area-averaged Nusselt numbers with the variation in the Reynolds numbers is obtained and compared with the experimental data. Results of the comparison shows that the case of air coolant with a 26 K temperature difference agrees satisfactorily with the experiments and shows a better heat transfer coefficient than that of a 50 K temperature difference. The end wall area-averaged Nusselt numbers for case (1) coolant increase with the Reynolds numbers. Conclusively, compared with two case of coolants, case( 2)produces a lower friction coefficient and a higher thermal performance factor, which significantly improve the heat transfer enhancement of the disturbed pin–fins arrayed in the wedge duct.**

***Keywords – In-line pin fin cooling, Reynolds Number, Nusselt Number, Friction Coefficient.***

## I. INTRODUCTION

Turbine blade is a single component which makes up the turbine section of a gas turbine. The blades are responsible for extracting energy from the high temperature, high pressure gas produced by the combustor. The turbine blades are often the limiting component of gas turbines. To survive in this difficult environment, turbine blades often use exotic materials like super alloys and many different methods of cooling, such as internal air channels, boundary layer cooling, and thermal barrier coatings. To protect blades from these high dynamic stresses, friction dampers are used.

In gas turbine cooling of components must be achieved by air cooling. Now a day, trailing edge cooling is a complicated cooling method. In trailing edge of turbine blade normally used cooling is pin-fin cooling so, in-line pin-fins are installed in trailing edge of the turbine blade. Pin–fins at the trailing edge, connecting the upper end wall to the lower end wall, can not only effectively intensify internal heat transfer, but can also improve the strength of the blade fig-1. In order to improve aerodynamic efficiency, the blade profile thickness is gradually reduced in the flow direction, which shapes the trailing edge region into the form of a wedge. The coolants discharged from the upstream cooling channels.

The pin–fins at the trailing edge of the blades can improve heat transfer performance by cooling fluids.

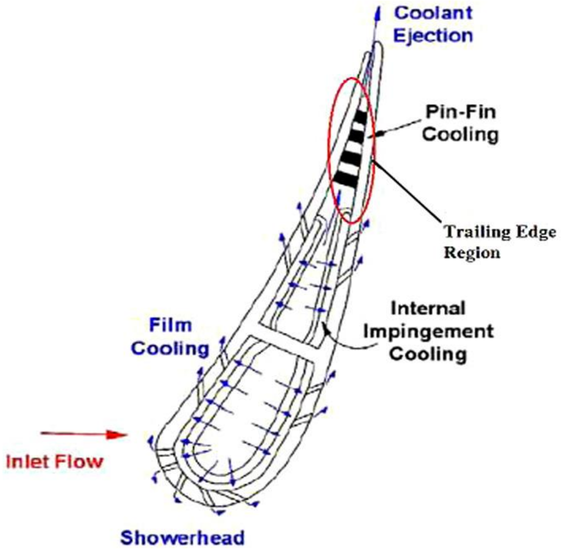



Fig1**:**Cross-sectional view of a cooled turbine blade. [1]

### A. *NUMERICAL METHOD*

#### 1) *Geometric model*

In this study, the design configuration is adopted from the [1] which consists of a wedge duct with pin–fins in inline configuration with an extended entrance section and exit section. A two dimensional sketch of the wedge model is shown in the Fig 2. Twenty-five pin–fins with a diameter of 12 mm span is placed with the distance between the up end walls and the down end walls.

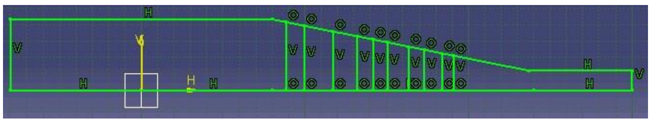


Fig 2**:** wedge model.

The design was done by the Catia v5. The Fig 3 shows the unstructured meshes generated in ANSYS ICEM-CFD for the numerically computated domain. An O-grid mesh is adopted around the pin–fins. To obtain the near-wall resolution, it has to be ensured that y+ is less than 1, fig 4 shows the y+ value for lower end wall of case (1), to meet the requirement of ANSYS CFX 15.0 solver.

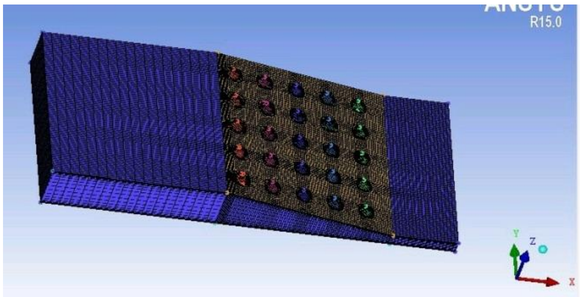



Fig 3: Computational mesh.

*2) Boundary conditions*

5% of turbulent intensity is allocated at the duct inlet. The static inlet temperature for case (1) and case (2) is 65C and 100C respectively. The static pressure is 1pa given at the outlet. A constant temperature i.e wall temperature (Tw) has been given to the pin–fin surfaces (row1 to row5). the upper and lower end walls in the wedge duct. no-slip boundary conditions and adiabatic conditions are applied to other walls of the wedge duct.

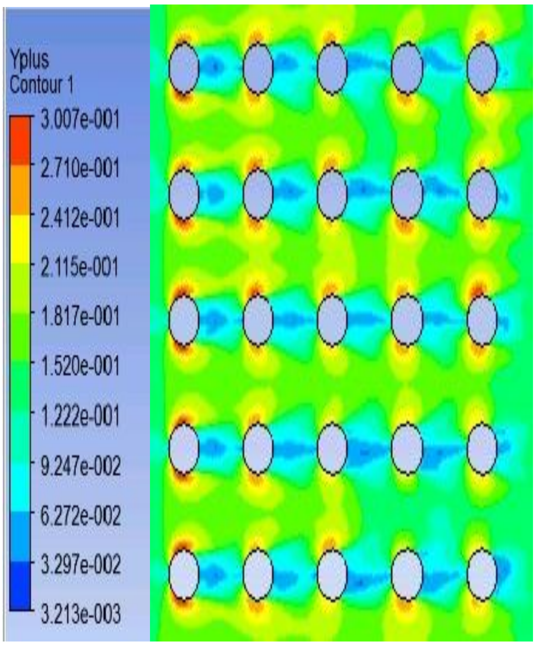



Fig 4: Y+ value for the case (1) lower end wall,

*3) Formula used*

Two Reynolds numbers are discussed herein [1]. One is based on the mean velocity (U) and the equivalent hydraulic diameter ($D_h$) at the entrance, namely ‘‘duct Reynolds number’’, defined as:

Re= (ρU $D_h$)/μ [Duct Reynolds number]

The other one is based on the average velocity (Umax) of the minimum cross-section in each pin–fin row and pin diameter (D), namely [1] ‘‘pin Reynolds number” , expressed as:

$Re_d$= (ρ$U_{max}$ D)/μ [Pin Reynolds number]

Table 1: 2 case of coolants

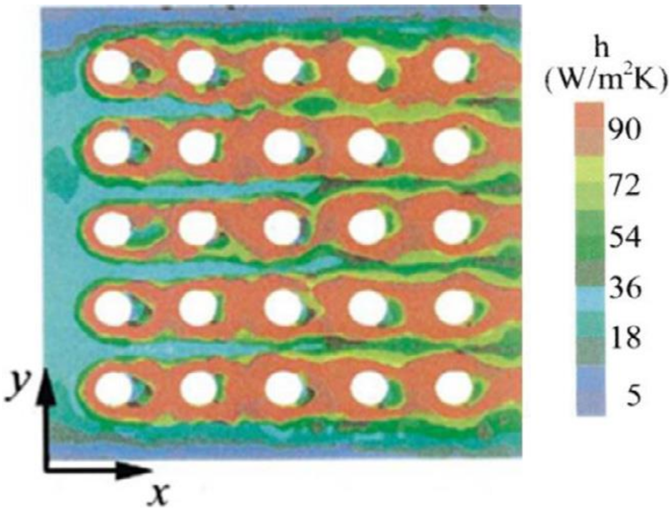
| cases | Fluid | T1 © | Tw © |
|---|---|---|---|
| case 1 | Air | 65 | 39 |
| case 2 | Air | 100 | 150 |

*4) Turbulence models*

A commercial software ANSYS CFX15.0 has been utilized to investigate the effect of two cases of coolants on the flow and heat transfer characteristics of pin–fins in the wedge duct.
The two-equation turbulence models based on the Reynolds-averaged Navier–Stokes equation are the most commonly used in simulations. In order to find a turbulence model suitable for a good prediction of the flow and heat transfer characteristics, k-ε model is carried out. Case 1, with the conditions similar to that of the experiment [1], has been numerically investigated and the numerical results have been compared with the experimental data.

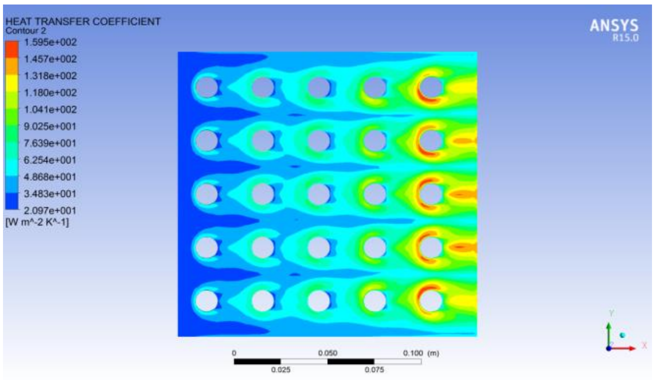
The heat transfer coefficient distributions on the bottom end wall in the wedge duct. Fig 6, the results of standard k-ε gives better results.

The advanced scheme and turbulence are solved with high order accuracy. The root mean square (RMS) residual of continuity and momentum equations and that of energy and turbulent kinetic energy is less than 1× $10^{-4}$, and then the computerized domain is considered as convergent.




(a)

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 7 Issue 04, April-2018




(b)

Fig 5: Distribution of heat transfer coefficient on the end wall, Re= 20,000, (a) experimental result, (b) standard k -ε model

*5) Convergence Study*

To make the good use of computer resources, a reasonable number of grids should be selected to guarantee the accuracy of the numerical analysis. The results demonstrate that the grid number has little influence on $Nu_d$ when the number is greater than 1, 58,922 elements. The grid number of 5,69,502 elements is finally adopted in this study.

*B. RESULTS AND DISCUSSION*

*1) Temperature difference*

The temperature difference is defined as the absolute value of the difference between the inlet temperature and the wall temperature, represented as below:

$\Delta T= T_1-T_w$

According to this definition, ΔT of Case 1 and Case 2 are 26 K and 50 K respectively, and ΔT of Case 1 is the same as that of the experimental condition.

*2) Pressure drop and Thermal Performance factor (TPF)*

The dimensionless friction coefficient varying with Reynolds number from fig 8 and fig 9 .The density of case (1) air is greater than that of case (2) when the temperature is at 100 C, and the air velocity is the lowest. The pressure drop for case (1) is lower when compared to case (2) and the thermal performance factor is high in case (2) condition when compared to case (1)

Table 2: Table for case(1)

| TPF(efficiency) | Re.no |
|---|---|
| 0.788732 | 10000 |
| 0.739791 | 20000 |
| 0.7097 | 30000 |
| 0.691555 | 40000 |
| 0.67742 | 50000 |

Table 3: Table for case (2)

| TPF(efficiency) | Re.no |
|---|---|
| 0.794151 | 10000 |
| 0.744189 | 20000 |
| 0.713556 | 30000 |
| 0.69496 | 40000 |
| 0.680502 | 50000 |

Table 4: HTC for case (1)

| case 1 | HTC | $Nu_d$ | TPF(efficiency) |
|---|---|---|---|
| row 1 | 35.5912 | 16.36377 | 0.788732 |
| row 2 | 40.0303 | 18.40474 | 0.739791 |
| row 3 | 45.7387 | 21.02929 | 0.7097 |
| row 4 | 55.8282 | 25.66814 | 0.691555 |
| row 5 | 73.3488 | 33.72359 | 0.67742 |

Table 5: HTC for case (2)

| case 2 | HTC | $Nu_d$ | TPF(efficiency) |
|---|---|---|---|
| row 1 | 34.7485 | 15.97632 | 0.794151 |
| row 2 | 38.9668 | 17.9157 | 0.744189 |
| row 3 | 44.4779 | 20.44961 | 0.713556 |
| row 4 | 54.373 | 24.99908 | 0.69496 |
| row 5 | 71.765 | 32.9954 | 0.680502 |

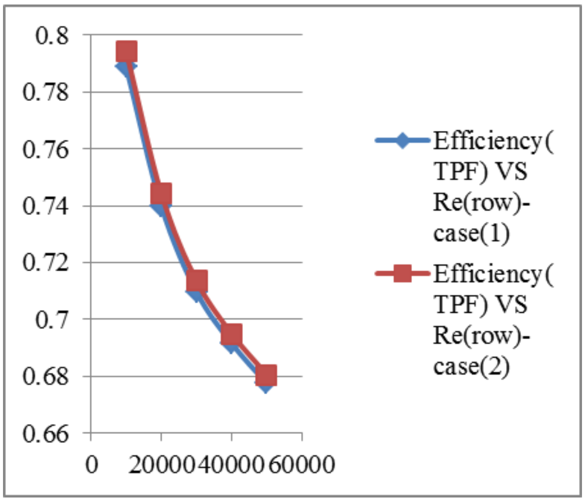



Fig 6: Reynolds number dependence of the thermal performance factor for case (1) experimental and case (2).

For the thermal performance factor for case (1) is lower when compared to the case (2) thermal performance factor , from the above figure we absorbed that case (1) maximum efficiency is 78% but the case(2) maximum efficiency is 79% respectively.

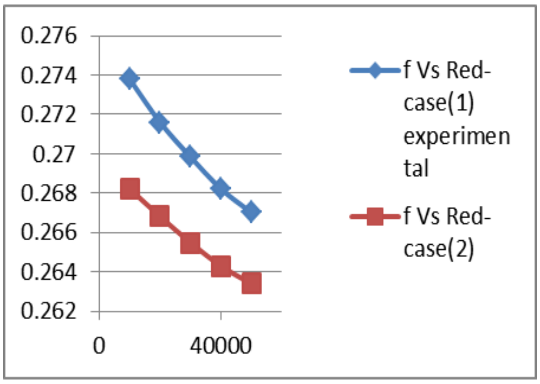



Fig 7: Reynolds number dependence of the friction coefficient for case (1) experimental and case (2).

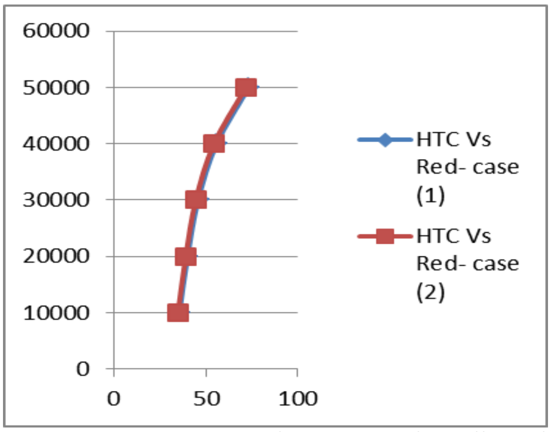



Fig 8: Reynolds number dependence of the Heat Transfer coefficient for case (1) experimental and case (2).

The above fig 7 and fig 8 shows the friction coefficient Vs $Re_d$ for 2 cases and HTC Vs $Re_d$ for 2 cases. The friction coefficient is high in case (1) when compared to case (2). The difference is 0.01.

*C. CONCLUSION*

Our analysis of the two cases of duct Reynolds numbers has come to the following conclusions:

1. For the air coolant, the predicted values of the case of $\Delta T$ = 26 K agree satisfactorily with the experimental results. Both the Nu on the bottom end wall of the wedge duct and the $Nu_d$ on the pin–fins for the case of $\Delta T$ = 26 K are higher than that for the case of $\Delta T$ = 50 K.
2. For the case (2), the Re=10,000 gives higher efficiency than the others.
3. For the Re=10,000, the Heat transfer coefficient of case (1) is higher than the case (2).
4. The friction coefficient of case (1) is higher than the case (2)
5. Compared to case (1) and case (2), case (2) gives more thermal performance factor than the case (1) condition.

So, the case (2) is considered as best thermal performance factor than the case (1) condition.

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