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"A Numerical Investigation of Thermal Performance of Inline Pinfins on a Wedge Duct"

Deepthi K R [1], Dr S Shankar [2]
[1] Research Scholar, Cambridge Institute of Technology, Bengaluru,
Email ID:deepthiramesh89@gmail.com, Phone number: 8105761324
[2] Professor & Head (R&D), Mechanical Department, Cambridge Institute of Technology, Bengaluru

ABSTRACT
The current paper includes numerical analysis for flow and heat transfer co-efficient parameters of elliptical and circle pin fin on a wedge duct. Reynolds-averaged Naiver–Stokes equations are coupled with k-e turbulent model coupled and hence analysed. The disparity of flow field variables on pressure drop and amount of heat transfer was obtained for Reynolds number of 10,000 – 50,000 in steps of 10. Two different shapes of pin fins are analyzed with two different boundary conditions for Five Reynolds number namely, circular and elliptical pin fins. The results of two different inline pin fins were justified with the empirical data with change in end wall and pin–fins area-averaged Nusselt number with the variation in the Reynolds number. The numerical results of circular pin fin agree well with the experimental results. The Area-averaged Nusselt numbers at the end wall for BC [1] of air coolant increases with different Reynolds numbers for both circle and elliptical pin fins. Further the results of the pin fins are compared with two different boundary conditions for pressure drop and thermal efficiency. The rate of heat transfer is higher for elliptical pin fins which shows that the thermal efficiency is high and pressure drop is low in the wedge duct

Keywords: Elliptical, rate of heat transfer, boundary conditions[BC],pressure drop,coolant

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I. INTRODUCTION

One of the most important factor to increase the cycle thermal efficiency and turbine power output is increasing the turbine inlet temperature .The turbine inlet temperature for aircraft engines and land based turbine the turbine inlet temperature has reached 1600K and to 2000K respectively. So it is important to remove the heat for the safe mode operation since it's higher than melting point of the blade material of the turbine. Thermal stress concentrations are created at the trailing edge of the turbine blade since it is affected more. Also the trailing edge of the blades has thin profile to increase the aerodynamic efficiency adds several constraints for the cooling schemes. Hence, effective and reasonable mechanism has to be brought in to increase the cooling for the thermal protection of the trailing edge.

The enhancement of the heat transfer can be done from two ways, to increase the contact area of the fluid with high temperature walls to enhance the heat transfer or to increase the churning of all the coolants to increase the intensity of the turbulence by adopting fluidization Techniques [4] At the trailing edge, pin fins are attached which connects the upper end wall to the lower endwall increases the heat transfer but also provides the stiffness to the blade. The thickness of the blade profile is tapered towards the flow direction which shapes the trailing edge to form wedge shape. This profile also provides the aerodynamic efficiency. Figure 1.1 shows the pin fins attached at the trailing edge of the blade. The discharged coolant from the upstream cooling channels flows first into the trailing region of the wedge with cooling of pin fins and finally ejects at the slot of the trailing edge and gets mixed with the mainstream.

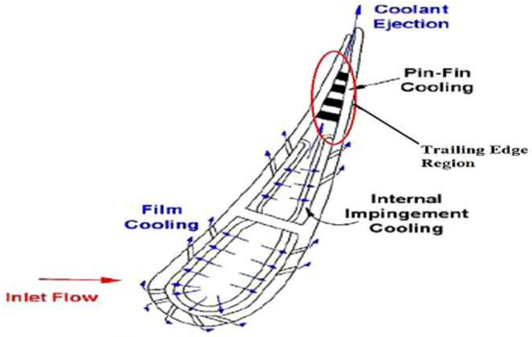


Fig 1.1: Sectional view of the blade [Courtesy: Google]

At the trailing edge the attached pin fins can increase the rate of heat transfer by triggering the wakes which further increase the mixing of the cooling fluids. It's observed that the pressure gradients and friction drag are increased and reported [1].The complicated vortex are formed at the junction of end wall pin fins and each pin fins contributes to it which consequently raise the end wall rate of heat transfer. The end wall horse shoe vortices formed at the upstream are very important since it contributes to the enhancement of the end wall heat transfer. The trade off for the disturbing the flow can be described as (1)The upstream pin fins triggers the horseshoe vortices and separated flow reattachment enhance the heat transfer (2)Thickness of boundary layer increases along with the flow until it reached the upstream of next pin fin rows. This results in trade off of increasing the end wall Nusselt number in the downstream direction. The major factor which describes the flow mechanism is pressure drop and heat transfer, There are other factors tat affects such as geometry, Reynolds Number and cooling fluids properties.

Many researchers have investigated factors influencing the heat transfer and pressure drop characteristics of cooling duct with pin–fins. Parametric studies, outlining pin–fins tip clearance, cross-sectional shapes, H/D, and SX. La ET AL. [10] found that, unlike the cross-flow heat changer (H/D > 8, were the end wall heat transfer is negligible) and the plate-fin-and-tube compact heat exchangers (H/D < 0.25, in which the tube heat transfer is relatively small), the heat transfer from pin–fins and end walls contributes to the overall heat transfer effect for short pin–fins (0.5 < H/D < 4).

Sparrow and Ramsey [12], Babus Haq et al. [13], Tahat et al. [14], Hwang and Lui [17] and Jeng et al. [16], studied the heat transfer coefficient and pressure drop characteristics of cooling duct with in-line and staggered arrays. The studies verified that a staggered pin–fin array gives higher end wall heat transfer enhancement and more pressure drop penalty compared to in-line array. The current study investigates the thermal performance of inline pin fins in a wedge duct for two conditions of coolants with 2 different shapes, one is circular pin fins and other is elliptical pin fins

II. NUMERICAL PROCEDURE

2.1 Geometric Modelling

The baseline configuration is the wedge duct model consisting of five rows twenty five circular pin fins [1] with inlet, outlet, wall, upper wall, and lower wall and proposed model with five rows of twenty five elliptical pin fins. The proposed configuration of the elliptical pin fins wedge model used is shown in fig 2.1.The pin–fin height to diameter ratio H/D ranges from about 1.3 to 3.6.The structured grid was generated in ANSYS ICEM-CFD 13.0 for elliptical pin fins are shown in below figure 2.1. Structured O-grid is maintained around the pin–fins and also refinement of the grid at the end walls area maintained. To satisfy the requirement of the ANSYS CFX13.0 Solver the resolution at the near-wall condition y+ is maintained less than 1.

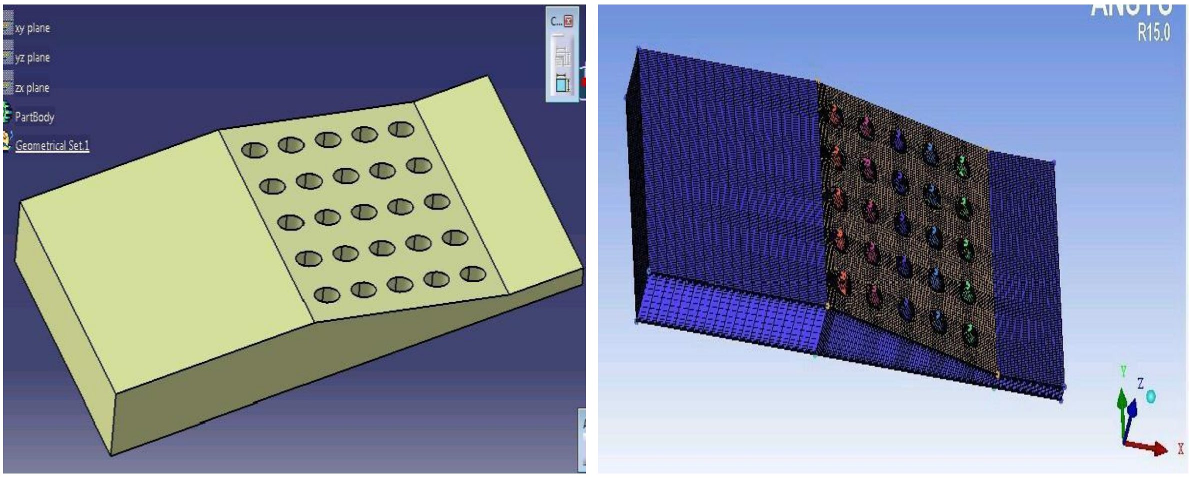


Fig 2.1:Elliptical Pinfin model and O-grid Meshed elliptical Model

2.3 Boundary Conditions

At the inlet the velocity vector and static temperature are specified and at the outlet static pressure is given .The pin fins surface, upper and lower end wall a constant temperature (Tw) is specified. No slip boundary condition on the wall and upper and lower end wall and heat transfer is adiabatic. Air as coolant with different static temperature and wall temperature are specified. BC (1) is same as empirical study conditions [1].Same boundary conditions are imposed on baseline model and proposed model.

Table 1.1: Two cases of coolants-Boundary conditions for elliptical and circular pin fins[Ref:1]

Conditions	Fluid	T1 ©	Tw ©
BC 1	**Air**	65	39
BC 2	**Air**	100	150

3.3 Formula Used

For the sake of understanding, two Reynolds numbers are considered. First one is at the entrance based on the mean velocity (U) and the equivalent hydraulic diameter (Dh) ,named ''duct Reynolds number'', defined as:

Re= (ρU Dh)/μ [Duct Reynolds number] ------ (1)

The second one is based on the average velocity (Umax) of the minimum cross-section in each pin–fin row and pin diameter (D), named ''pin Reynolds number", expressed as:

Red= (ρUmax D)/μ [Pin Reynolds number] ------- (2)

The non-dimensional pressure drop throughout the computed domain is represented by

Friction coefficient as: f= (2ΔP)/ρN(Vmax)2 ------------ (3)

Nu∞=0.023Re0.8Pr0.4 [Dittus- Boelter correlation] -------- (4)

f∞=0.0791Re0.25 [Blasius-equation] -------- (5)

η= (Nu/Nu∞)/(f/f∞)0.333 [Thermal performance factor] -------- (6)

Where, Pr is Prandtl number. The values of pr for air is 0.7

The fitting formulas of Nu and Re for air is Nu= 0.06Re0.78294 ---------- (7)

III. RESULTS AND DISCUSSION

The study is carried out for wedge shaped of circular (baseline) and elliptical (proposed) pin fin configuration. To determine the flow effect on the pressure drop and the convective heat transfer for circular pin fin wedge duct [1] and the elliptical pin fins, five low Reynolds number are considered [Ref:1].

A grid independent study was done to check the numerical accuracy of the results obtained has no influence on grid size and a grid of size 5,69,502 elements was choosen for the ellpitical pinfins.

The baseline model of the wedge duct with two cases of air coolant on twenty five circular pin fins was carried out [Ref:1]. The results show that highest thermal efficiency for air coolant BC [1] when compared to the BC [2]. BC [1] condition is same as empirical results. The results shows that the best thermal efficiency is for empirical condition and co efficient of friction is high for increased static and wall temperature condition which reduces the thermal efficiency .Same approach is used for analysis of elliptical pin fins design.

3.1. Impact of the coolants on Heat transfer

The analysis used two equations turbulence model k-epsilon with Reynolds-averaged Navier–Stokes to capture the fully developed flow and rate of heat transfer characteristics. Both the boundary conditions are assessed on the proposed configuration and the results are compared to evaluate the best design model which provides higher thermal efficiency and lower pressure drop with inline pin fin configuration. These conditions are assessed for different Reynolds number and on the bottom end wall the heat transfer coefficient are shown in below Fig 3.1 to 3.5.

For the circular pin fns, the empirical boundary conditions shows that the heat transfer coefficient has better value than the increased temperature values for static and wall temperature. For empirical l simulated conditions of Re=50,000, ROW 5 has the highest heat transfer value=390 (W/m^2k) followed by increased temperature values of 378 (W/m^2k) .The HTC_w value gradually increases for the Reynolds number and also based on the analytical formula Nusselt number of 179 is calculated and highest is at Reynolds number=50000 at row 5.The BC [2] pin Reynolds number for Re=50,000, the ROW 5 has the highest value than the other four Reynolds number i.e. Re_d=2.2E+10^4.

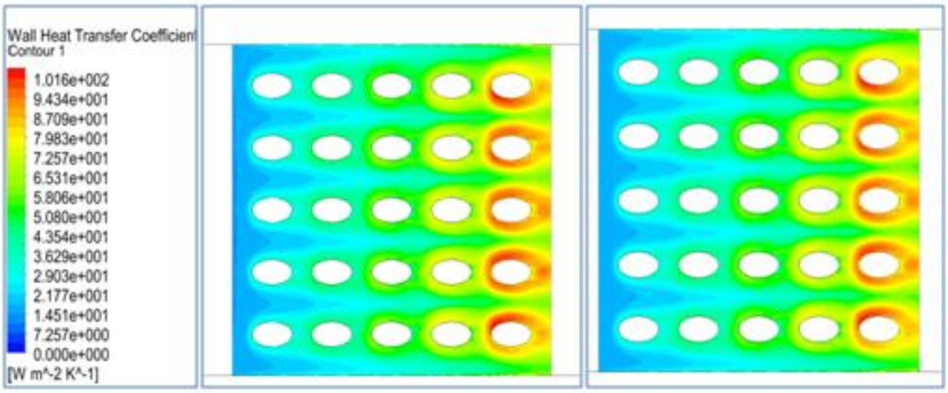


Fig 3.1: BC [1] & [2] HTC_W for Re = 10,000 at lower end wall

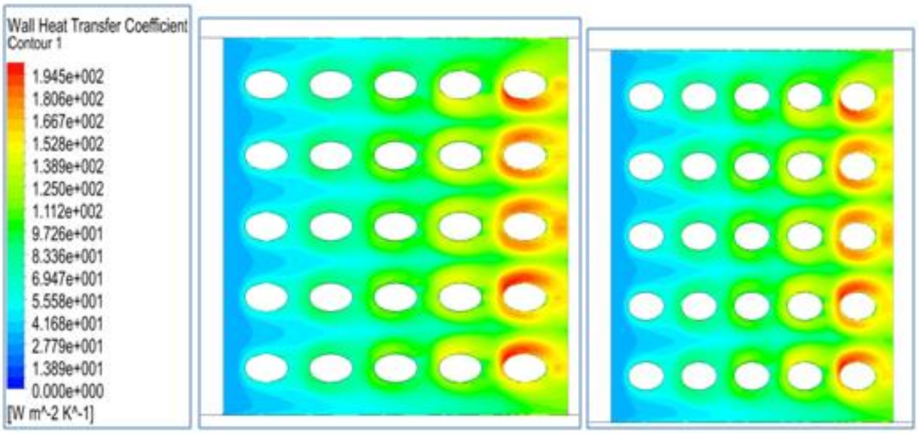


Fig 3.2: BC [1] & [2] for Re = 20,000 HTC_W at lower end wall

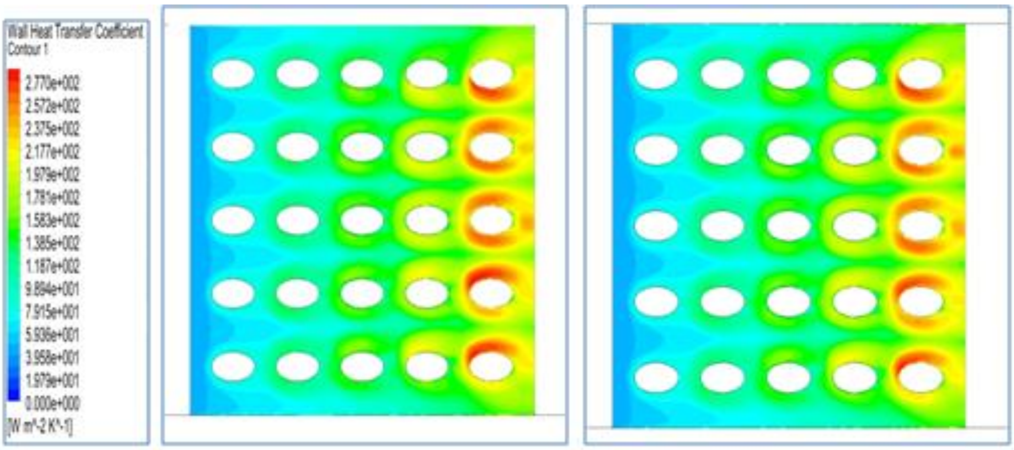


Fig 3.3: BC [1] & [2] HTC_W for Re = 30,000 at lower end wall

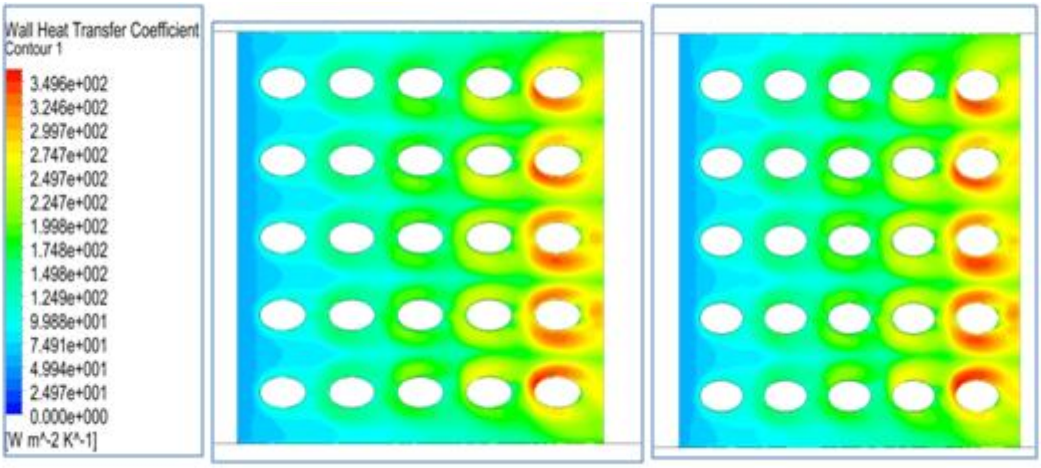


Fig 3.4: BC [1] & [2] HTC_W for Re = 40,000 at lower end wall

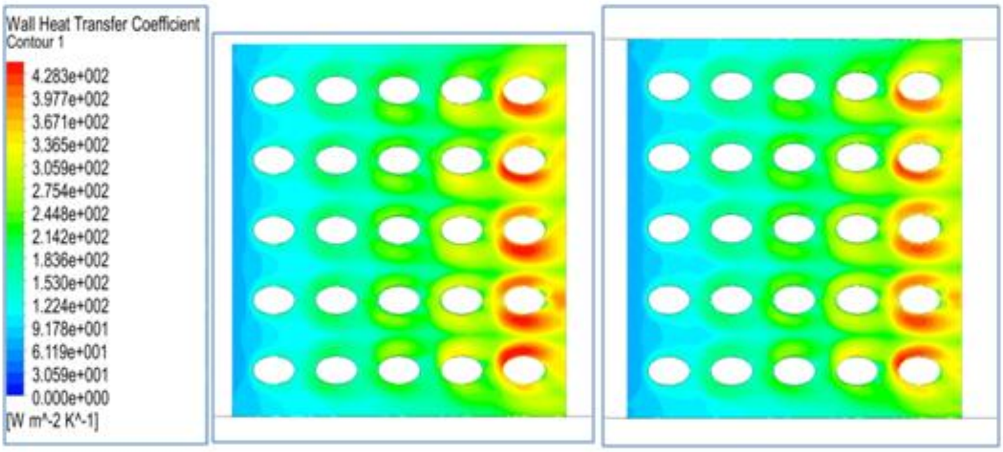


Fig 3.5: BC [1] & [2] HTC_W for Re = 50,000 at lower end wall

3.2. Pressure drop and Thermal performance

The two configurations baseline and proposed are compared to verify variation of dimensionless friction coefficient and efficiency of thermal with Reynolds number shown in Table 1.2 and 1.3 for the BC (1) and BC (2).The air velocity is the lowest and the density of case (1) air is greater than that of case (2) when the temperature is at 100 ^oC.

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

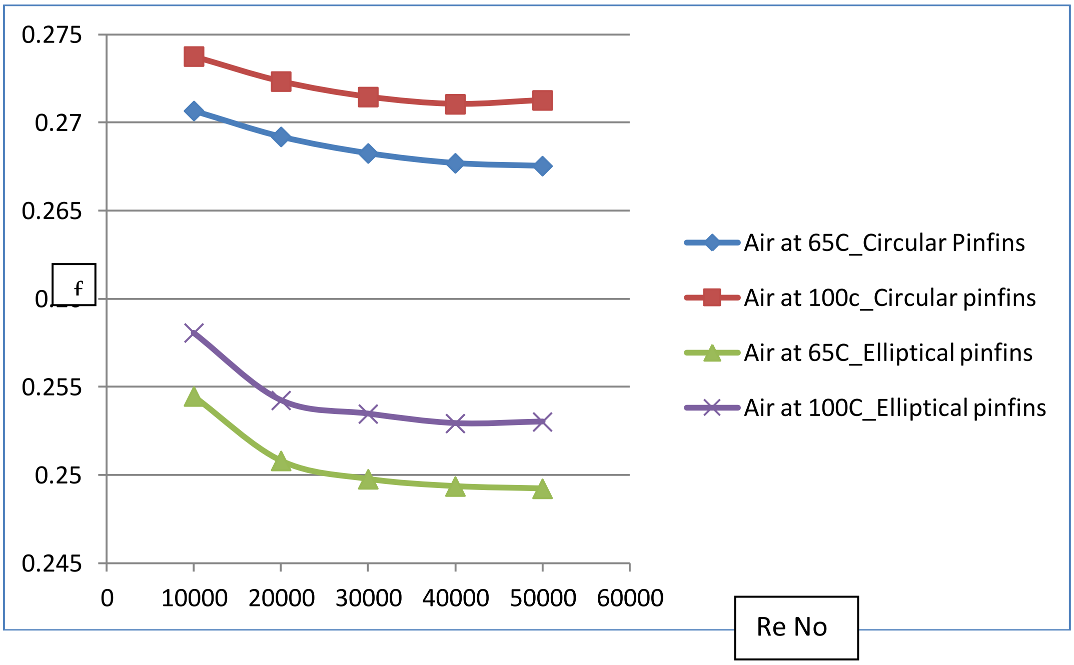
$$\Delta T = T_1 - T_w$$

By definition, ΔT of Case 1=26 K and Case 2 =50 K, case (2) of elliptical pin fin design gives higher thermal efficiency than the circular pin fin design [1]

For circular pinfin configuration, the pressure drop is highest at BC (2) of 1742(Kg/Sec) for the Reynolds number of 50000 followed by 1101(kg/sec at 40000 Re. The coefficient of friction is calculated based on the above given formula which results in highest of 0.27376 at BC (2) for Re=10000, followed by least at Re=50000 for BC (1) of 0.26754. Comparing the results with proposed configuration of elliptical pinfins the highest pressure drop of 1659(kg/sec) for BC (2) at Re=50000 followed by 1047(kg/sec) at Re=40000.The coefficient of friction is highest 0.25807 for Re=10000 at BC (2).The comparison of the results leads to confirm proposed design has better coefficient of friction.

Table 1.2: BC (1) and BC (2) Coefficient of friction for pin fin design

	Circular pinfin Design		**Elliptical pinfin design**	
Re	**Air at 65C**	**Air at 100C**	**Air at 65C**	**Air at 100C**
10000	0.27067	0.27376	0.25447	0.25807
20000	0.2692	0.27234	0.25081	0.25424
30000	0.26826	0.27146	0.24977	0.25347
40000	0.2677	0.27105	0.24936	0.25293
50000	0.26754	0.27127	0.24923	0.25302

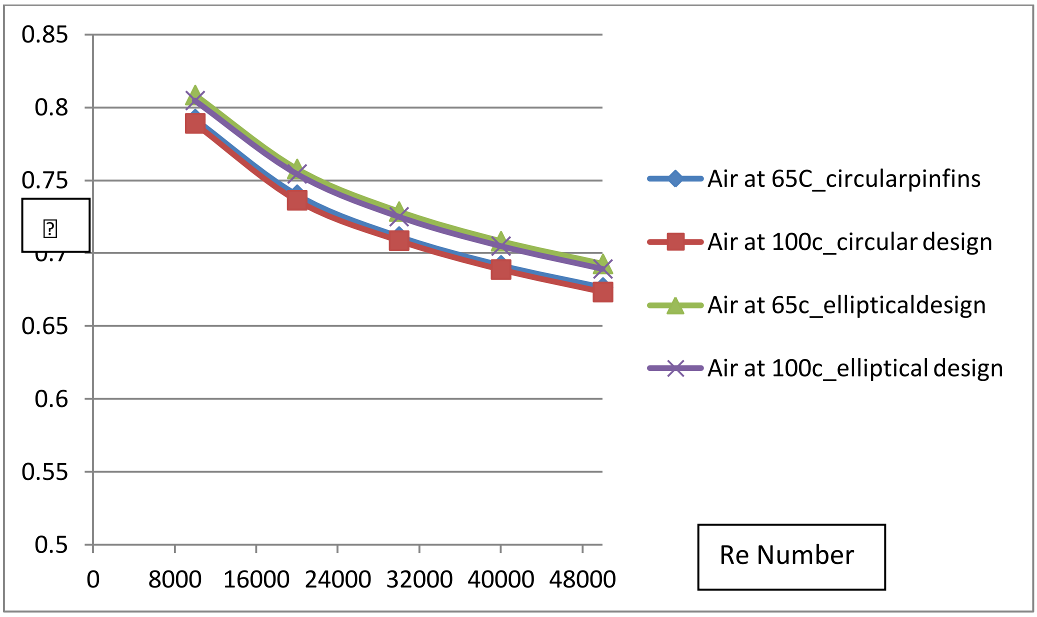


Graph 1.1: Comparison of coefficient of friction for BC [1] and [2] for pinfin

From the above graph the coefficient of friction is least at BC [1] for elliptical design of 0.25447 compared to BC [2].

Table 1.3: BC (1) and BC (2) Thermal performance factor for pin fin design

	Circular Pinfin Design		**Elliptical pinfin design**	
Re	Air at 65C	Air at 100C	Air at 65C	Air at 100C
10000	0.791996	0.789005	0.808458	0.804681
20000	0.740099	0.736272	0.757763	0.75434
30000	0.711407	0.7086	0.728545	0.724983
40000	0.691636	0.688775	0.708192	0.704845
50000	0.676449	0.673334	0.692624	0.689148



Graph 1.2: Comparison of coefficient of friction for BC [1] and [2] for pinfin

The above table 1.3 populates the data of circular pinfin and elliptical pinfin design condition for the thermal performance factor. The thermal efficiency is highest at Re=10000 for case (1) of 80.8% and 80.4% for BC (2) for elliptical pinfin design. The circular pinfin design shows lowest efficiency at BC (2) being the least for Re=10000.

IV. CONCLUSION

The study evaluated the two different conditions of coolants for in line pin fins with the effect of the heat transfer and pressure drop parameters. The results of the inline pins for circular and elliptical design shapes for two boundary conditions of the coolnats has come to following closure

- The higher friction coefficient for BC (2) is noticed when compared to all the other cases for baseline configuration. The lowest f=0.267 is achieved for Reynolds number of 50,000 for BC(1) in comparison to other four Reynolds number conjointly.
- The Thermal performance factor (η) for BC[1] is obtained than the other cases for baseline configuration. The highest Thermal performance factor i.e. η=79% is with Re= 10,000 BC[1]
- The friction coefficient for BC [2] is high when compared with rest of the cases for proposed configuration. The lowest f=0.240 when compared to other four Reynolds number respectively for Re=50,000 for BC [1]
- The thermal performance factor (η) for BC[1] is higher than BC[2].The highest Thermal performance factor is 80% for Re= 10,000 of BC[1].
- Both the Nusselt numbers on the bottom end wall and on the pin fins are higher for BC [1] compared to BC[2] for proposed configuration. This results in higher thermal efficiency and lower friction coefficient which leads to intensifying the rate of heat transfer .The shape of the pin fins has considerable effect on the heat transfer enhancement.
- This concludes that the wedge shaped with elliptical pin fins for the trailing edge is much more preferred.

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