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Performance analysis and effect of various material fin-perforations in a plate-finned heat exchanger

Bondu jyothi[1*], *Tayi* Srikanth[1], *S.V.S.S.R.* Krishna P[1], *Raju* Chittumuri[1]

[1]Mechanical Department, Ideal Institute of Technology, Kakinada, Andhra Pradesh, India.

Abstract. Finding increasingly efficient heat exchangers requires a critical task: the development of fin profiles for prolonged life of equipment. The present study uses numerical analysis for better plate-finned heat exchangers hydraulic and thermal performance. Fin patterns are used in the analysis to evaluate a four-tube in-line heat exchanger. The proposed approach involves use of simulation and numerical methods to solve the heat transfer problem on different materials and wind velocity data as boundary conditions for improved heat transfer. The Reynolds-Averaged Navier-Stokes (RANS) model will be used to correct the average turbulent air flow over finned tubes in three dimensions using the ANSYS-Fluent software. The evaluation plan compares the current findings with published Nusselt number empirical correlations for the justification. Finally, the presentation of a parametric study on the impact of fin perforation size and shape on different materials and the behaviour of fluid dynamics and heat transfer. Larger fin perforation sizes (circular, rectangle, rhombus, and ellipse shapes) are found to enhance the heat exchangers' overall PEC and thermal efficiency for sustainability of the thermal equipment. The comparison was made between non-perforated and perforated fin types. Ultimately, it was determined that the best option for plate fin heat exchangers is the circular type perforated fin. When compared to without perforated fins, the circular type perforated fin has a higher heat transfer coefficient. A circular perforated fin has high performance, and the rhombus type perforated fin has lower performance.

Keywords: Plate-fin heat exchanger, Perforated fin plates, Heat transfer on different materials

1 Introduction

1.1 Heat Exchanger

Heat exchangers play a crucial role in various industrial, commercial, and residential applications by efficiently managing the transfer of thermal energy. The particular heat exchanger type and design chosen for a given application will rely on factors such as

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[*] Corresponding Author: bondhuuvj99@gmail.com

temperature ranges, fluid characteristics, available space, and longevity of the equipment. Overall, heat exchangers are versatile devices that play a critical role in optimizing energy usage and facilitating thermal management in diverse engineering systems. In plate and fin heat exchanger design, the fin geometry has grown in importance. Liquids are frequently used in conjunction with enhanced surface geometries to cool electronic equipment so that it works for prolonged duration continuously.

A compact heat exchanger that combines the concepts of plate-finned tube heat exchangers is called a plate-fin tube heat exchanger. It typically consists of a series of flat, finned plates that are separated by tubes. The arrangement of plates and tubes creates a compact and efficient heat exchange system. Depending on the application, one or more rows of heat exchangers may be produced. Heat exchangers with tubes encircled by flat fins are known as plain plate-fin and tube heat exchangers. This design enhances the heat transfer surface area, making it an efficient choice for applications where compactness, high thermal performance, and safety are essential.

Fig. 1. Plain in-line circular tube plate heat exchanger

W.K. Rauber et al. [1] had applied, numerical and simulation for a conjugate heat transfer problem of the fin patterns such as circular, diamond-patterned rectangle perforation, and rectangle without an insert and examined an in-line, four-tube heat exchanger. Abu Madi et al. [2] the geometry of the flat and corrugated fins had an effect on the Reynolds number, the fin and tube geometries, and the number of tube rows, which evaluated the performance behaviour of finned plate and tube heat exchangers. Wang et al. [3], furthermore, the most widely used design was the plain fin because it was easy to make, simple to assemble, and it possessed features of a minimal pressure drop. Fernández-Seara et al. [4] the heat transfer coefficients during the condensation and vapour generation processes in heat exchanger tubes were measured using an experimental setup. Applying the underlying method to multiple convection heat transfer cases, they noticed that design engineers dealing with thermal issues would find it useful. Wang et al. [5] examined the louvred fin and tube heat exchangers and simple, semi-dimpled vortex generators (VGs) in terms of their airside performance and the semi-dimpled VG geometry was not as effective at transferring heat as the louvred fin geometry. Liu and colleagues et al. [6] investigated how the size, quantity, and spacing of the holes affect the air side Colburn factor using computational fluid dynamics (CFD). The air side Colburn factor of the plain fin heat exchanger was lower than that of the perforated fin heat exchanger. Kalantari et al. [7] was discovered that the heat transfer coefficient for gas-liquid finned tube heat exchangers exhibited a conjugate correlation. The correlation used three-dimensional CFD simulations for the Nusselt number using non-dimensional geometrical parameters and the Prandtl number. ltwieb et al. [8] assessed a multi-tube heat exchanger with plain fins and several geometrical alterations and its heat performance. For every geometric modification, the local fin efficiency, Fanning, and Colburn factors were found through validation experiments and CFD comparisons. The authors established two empirical correlations between the fanning and Colburn friction factors, who demonstrated a

10% agreement between their predictions and the experimental data. Altwieb and Mishra et al [9] the thermal response of a plain, louvred, or semi-dimple vortex generator coupled with a fins and tube heat exchanger was investigated using numerical modeling and experimentation in a released paper. In this work, the properties of pressure drop and heat transfer were thoroughly examined. Two novel design equations for the behaviour of the heat transfer rate and pressure drop have been developed. Biswas et al. [10] used the unsteady liquid crystal thermography method to study a vortex in the form of one delta wing and one pair of delta winglets in a rectangular channel with Reynolds numbers ranging from 1360 to 2270. They concluded that local heat transfer could be increased by up to 200%. Khoshvaght-Aliabadi et al. [11] The Gnielinski equation was found to be less accurate in predicting the Nusselt number at the transitional flow than the Notter Rouse equation, which examined the impact of delta-winglets in different configurations for the plain tube. This VG (Vortex generator) at Re=8715 had a maximum PEC (performance evaluation criterion) of 1.41 compared to the plain tube. Hot spots are produced in this area by the recirculating flow that forms in the downstream zone of the vortex generators. This wonderful problem might have an amazing solution in perforated baffles. Deb et al., [12], Fin tube and plate fin heat exchanger types could benefit greatly from the use of longitudinal vortex-generators to improve heat transfer. Based on their research, winglet heat exchangers can enhance heat transfer by up to 240%. Ahmed et al. [13], researched on vortex generator applications that was recently completed. The literature indicates that there was a dearth of researched on the heat transfer characteristics of vortex generators inside plate fin heat exchangers. Gentry and Jacobi [14] examined the effectiveness of an embedded Vortex generator with delta wings in a flat plate and discovered that heat transfer was enhanced by up to 50% when compared to a basic flat plate. Chen and Shu [15] demonstrated that although the closed wall raises the delta-wing vortex generator's turbulent kinetic energy, it has no impact on the normal velocity or average axial mean. Wu and Tao [16] investigated the heat transfer properties of four plates—two with delta-winglet longitudinal vortex generators and one without—at attack angles between fifteen and sixty degrees. The average Nusselt number of plates with an attack angle of 60° is found to be slightly higher than that of plates with an attack angle of 45°, despite the possibility of a larger pressure drop. Akcayoglu [17] in a plate fin heat exchanger with triangular fin inserts, a winglet pair of vortex generators was employed to enhance heat transfer. With increasing angle of attack and Reynolds number, the average Nusselt number increases. Zeng et al. [18] examined the thermal performance of a plate fins channel through the use of four basic vortex generator types. They observed that the mixing effect between wing cascades improved heat transfer characteristics as the winglet's inclination angle increased. Khoshvaght Aliabadi et al. [19] examined the copper-water nano fluid's thermo-hydraulic performance in a variety of plate-fin channels, including wavy, louvered, plain, offset strip, pin, vortex generator, and perforated. Their results showed that pressure drop and heat transfer coefficients increased with the weight fraction of nanoparticles. Additionally, they claimed that the thermal-hydraulic performance of the vortex generator channels was sufficient. From the above literature, it has been observed that earlier researches have focused on numerical tools and simulation analysis to find out fin perforations. In this paper, the corresponding author focused to optimize the fin shapes in order to get an effective heat transfer rate with fast rate of convergence in the solution.

- To Create the high-performance 3D model for plate-finned heat exchanger for extended use and longevity of equipment by enhanced cooling provided by thermally efficient design.
- To Simulate and optimize the shape using Fluent software for fast heat transfer rate.
- To Geometry comparison with different inserts shapes.
- To find the optimum plate-finned heat exchangers' shape, size, and thermal performance.
- To obtain validation with simulation and numerical analysis.

2 Methodology

CATIA software is intended to be used for geometric model preparation. The effective heat transfer area and heat transfer coefficient can be raised by using perforated fins. The shape of the perforations controls how much the surface area changes, and the geometric parameters are displayed below

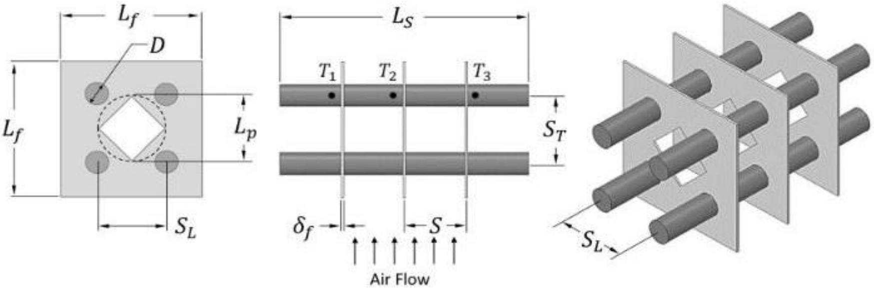


Fig. 2. Detail drawing of Perforated Finned-Tube Heat Exchanger [1]

Fin Spacing S =42.50 mm
Diameter of the tube D= 16.0 mm
Thickness of the fin δ_f =1.50 mm
Transversal pitch S_T= 48.0 mm
Number of Tubes Ns= 4
Inlet Temperature Te =296 K

Length of the fin L_f =96.00 mm
Length of the tube Ls =170.0 mm
Longitudinal pitch S_L =48.0 mm
Number of Fins N_f =3
Entrance velocity, u= 7.2, 8.6, 10.3 and 11.9 m/s

Hydraulic diameter [1] can be calculated as below,

$$D_h = \frac{4V'\psi}{A'}$$

Where, $V' = S_T S_L (S-\delta_f)$
$A' = 2(S_T S_L - \pi D^2/4) + \pi D (S-\delta_f)$
$\psi = 1-\delta_f/S - (\pi D^2/4)(S-\delta_f)/(S_T S_L S)$

Nusselt number (Nu):

$$Nu_{Dh} = \frac{h_0 D_h}{k_a}$$

Friction factor:

$$f = \frac{\Delta P}{\rho_a u_e^2} \frac{D_h}{2L}$$

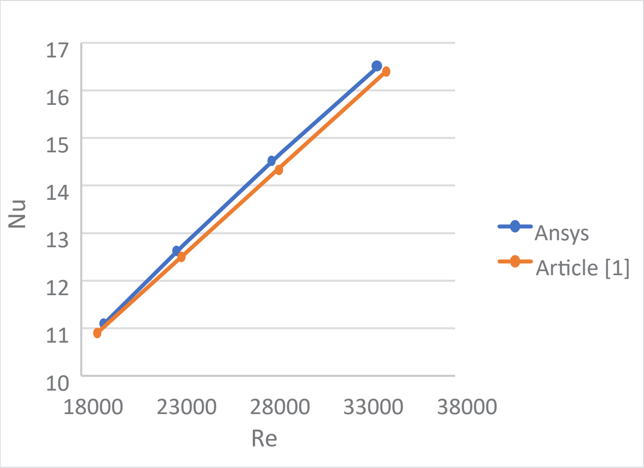
2.1 PEC (Performance Evaluation Criterion):

Performance evaluation criteria for heat exchangers are essential to assess their effectiveness in transferring heat between fluids. These standards offer a thorough evaluation of the heat exchanger's performance with regard to size, pressure drop, heat transfer efficiency, and other useful factors. The specific choice of criteria depends on the application and the priorities of the system designer or operator. This parameter is calculated.

PEC = (Nu with perforated fin/ Nu without perforated fin)/ (f with perforated fin/ f without perforated fin)

Table1. Reynolds no. vs Nusselt Number

Reynolds no.	**18853**	**23363**	**28957**	**34312**
ANSYS	111.7562	126.2603	145.6678	164.8459
[1]	109	125	144	164

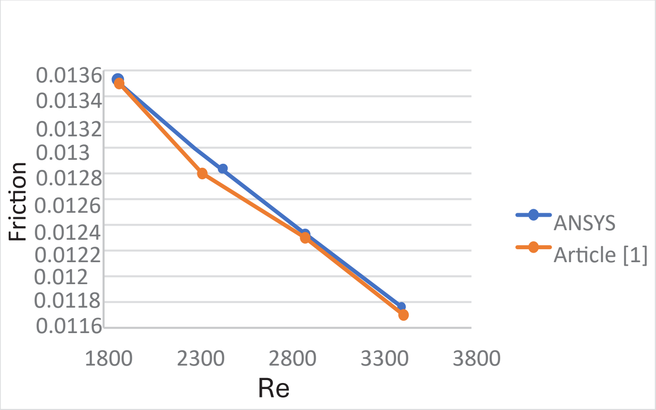


Graph1: Reynolds no. vs Nusselt Number

According to the above data, the circular type perforated fin, with a Reynolds number of 34312, has a higher Nusselt number than the fin without a perforation. Current work using the reference article [1] is validated by the Renolds number versus the Nusselt number.

Table 2: Reynolds no. vs Friction factor

Reynolds no.	**18853**	**23363**	**28957**	**34312**
ANSYS	0.01357	0.012858	0.012367	0.011799
[1]	0.0135	0.0128	0.0123	0.0117



Graph 2. Nu vs Heat transfer coefficient

When comparing the circular type perforated fin to the non-perforated fin, the above table found that the Nusselt number of the former is larger, at a value of Reynolds no. of 34312; the validation of present work with the reference article [1] is demonstrated by the relationship between Reynolds no. and Friction factor. Likewise, a circular type perforated fin has a better heat transfer coefficient than a fin without any perforation.

3 Modelling of Different Plate-Finned Heat Exchanger and their effects

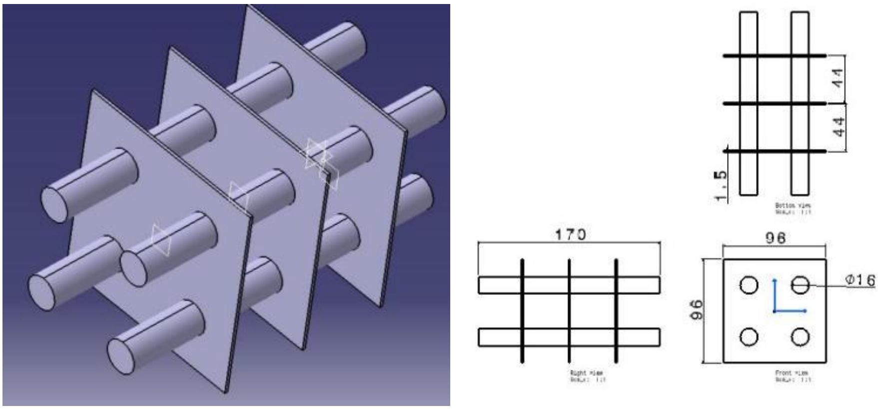


Fig. 3. 3D Model Tube and Finned Type Heat Exchanger

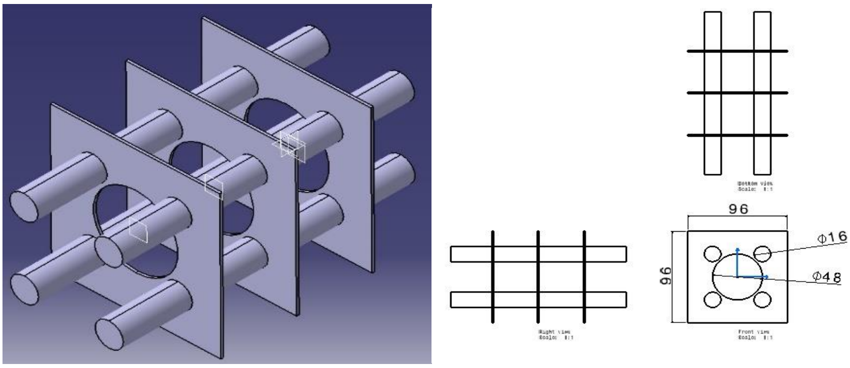


Fig. 4. 3D Model Tube and Finned Type Heat Exchanger with circular cutout

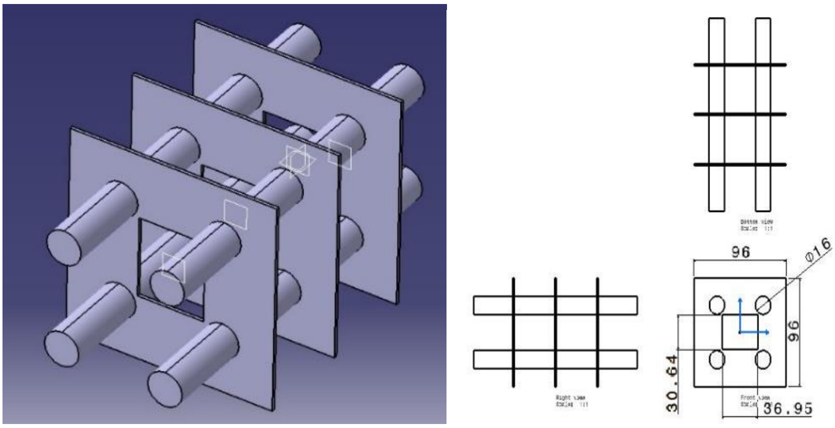


Fig. 5. 3D Model Tube and Finned Type Heat Exchanger with rectangular cutout

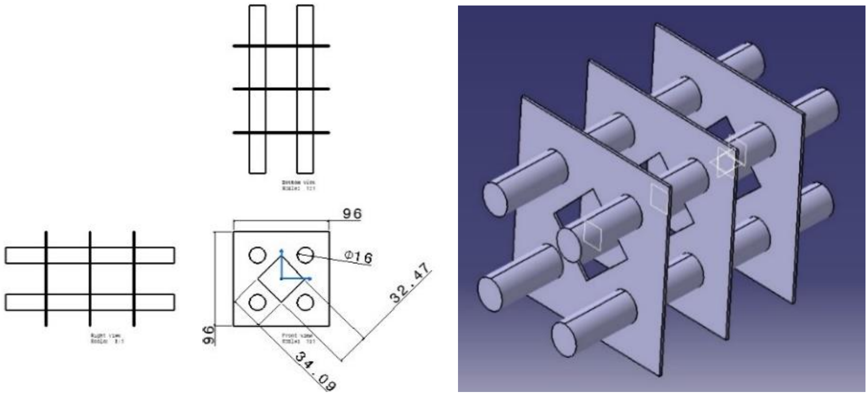


Fig. 6. 3D Model Tube and Finned Type Heat Exchanger with diamond cutout

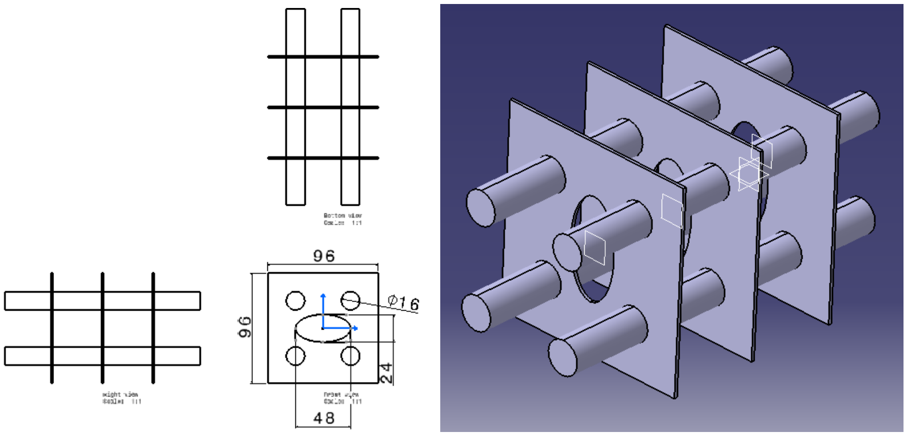


Fig 7. 3D Model Tube and Finned Type Heat Exchanger with Ellipse 1 Cutout

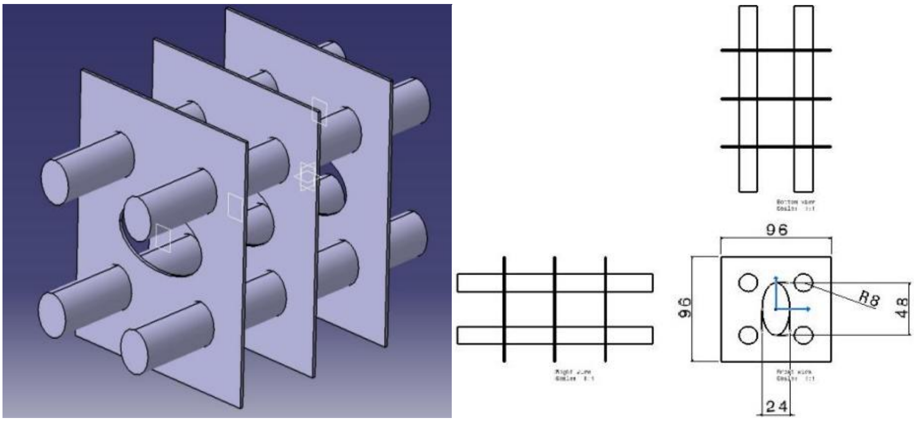
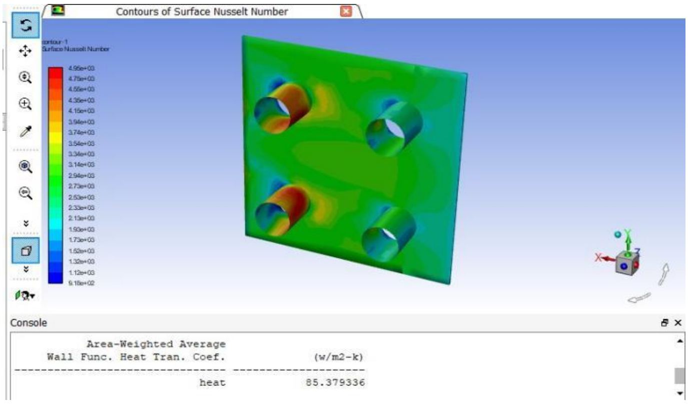


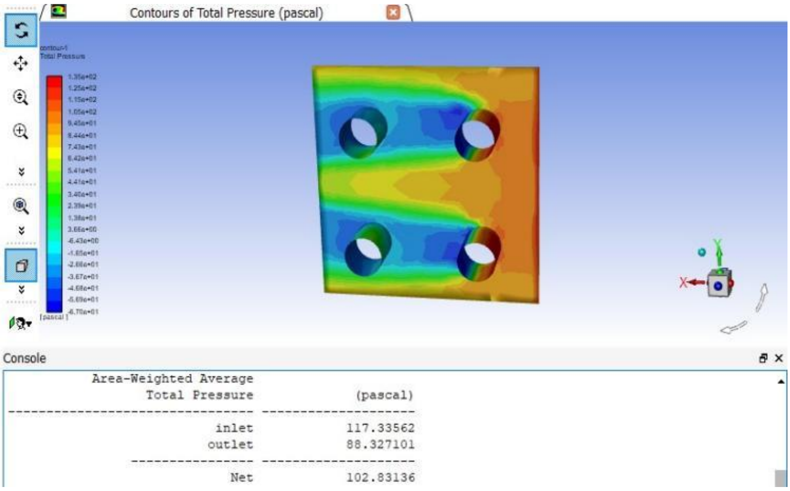
Fig 8. 3D Model Tube and Finned Type Heat Exchanger with Ellipse 1 Cutout

4 Performance Analysis



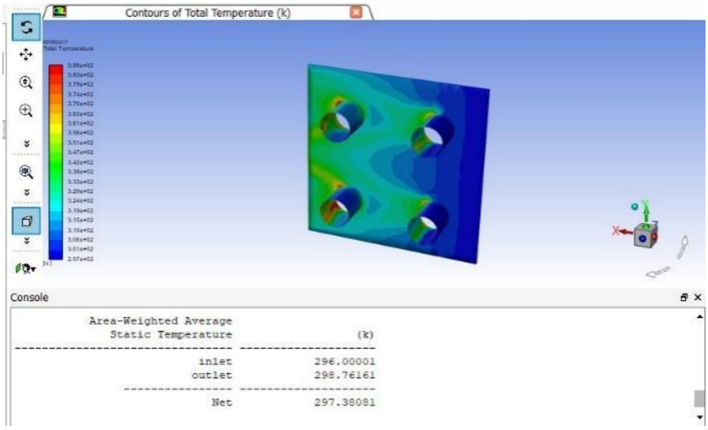
```
           Area-Weighted Average
     Wall Func. Heat Tran. Coef.                 (w/m2-k)
---------------------------------- --------------------
                              heat            85.379336
```

Fig. 9. Average of heat transfer coefficient reading for Rectangle perforated plate-fin



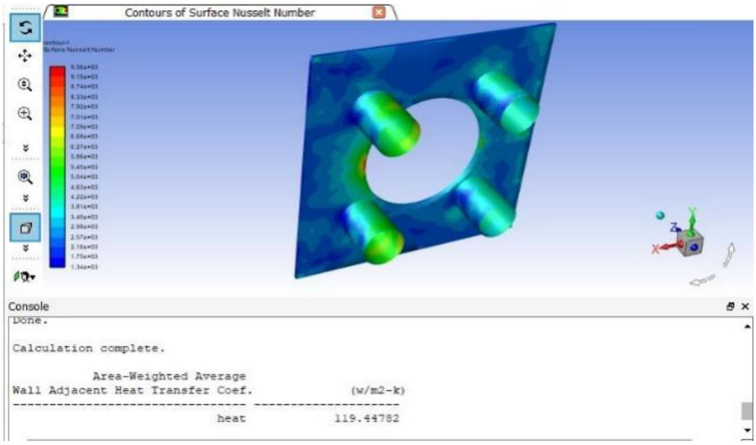
```
           Area-Weighted Average
                  Total Pressure               (pascal)
---------------------------------- --------------------
                             inlet            117.33562
                            outlet            88.327101
                  ---------------- --------------------
                               Net            102.83136
```

Fig 10. pressure reading for without perforated plate-fin



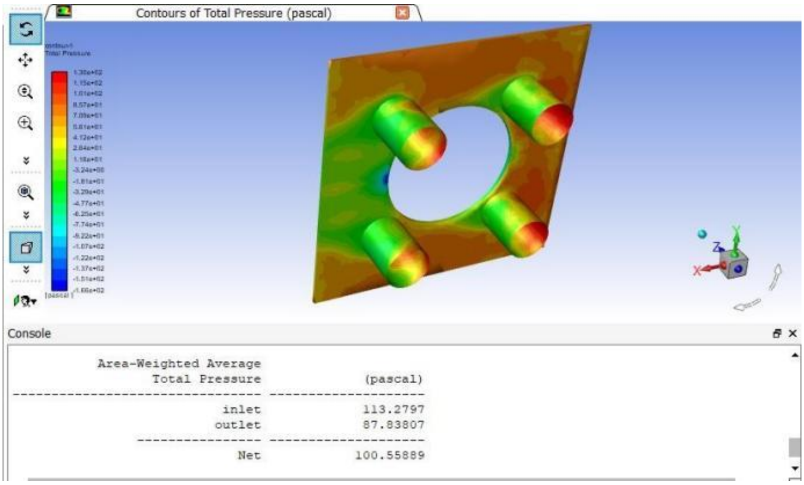
```
           Area-Weighted Average
              Static Temperature                    (k)
---------------------------------- --------------------
                             inlet            296.00001
                            outlet            298.76161
                  ---------------- --------------------
                               Net            297.38081
```

Fig 11. Temperature distribution on without perforated plate-fin



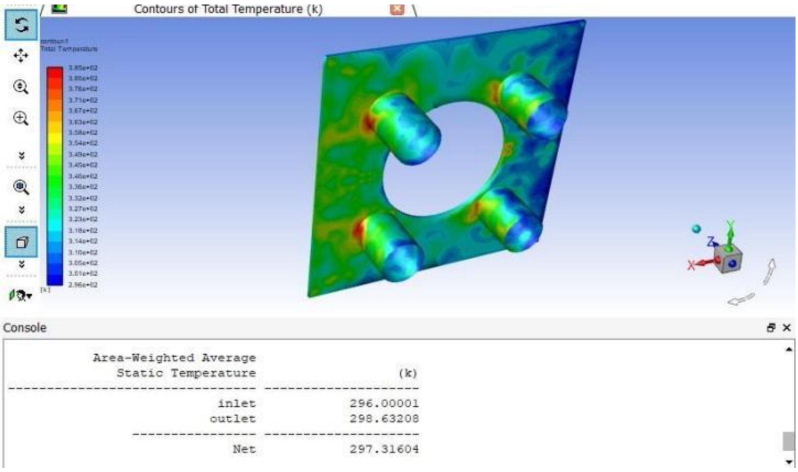
```
Done.

Calculation complete.

            Area-Weighted Average
Wall Adjacent Heat Transfer Coef.                (w/m2-k)
--------------------------------  --------------------
                            heat             119.44782
```

Fig. 12. average of heat transfer coefficient reading for Circular perforated plate-fin



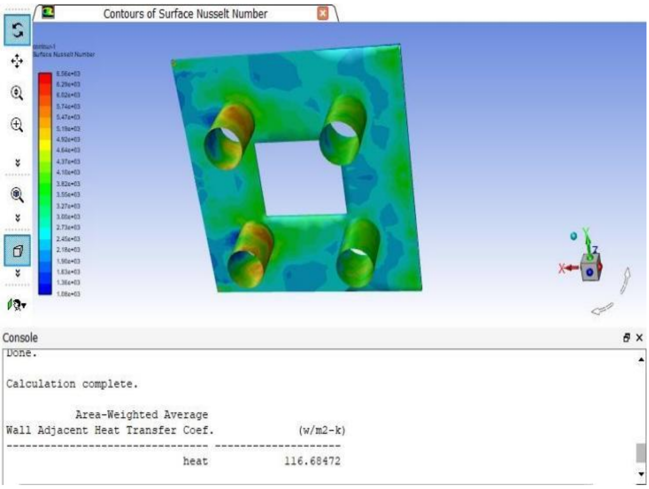
```
            Area-Weighted Average
                   Total Pressure              (pascal)
--------------------------------  --------------------
                            inlet             113.2797
                           outlet             87.83807
                ----------------  --------------------
                              Net            100.55889
```

Fig. 13. pressure reading for Circular perforated plate-fin



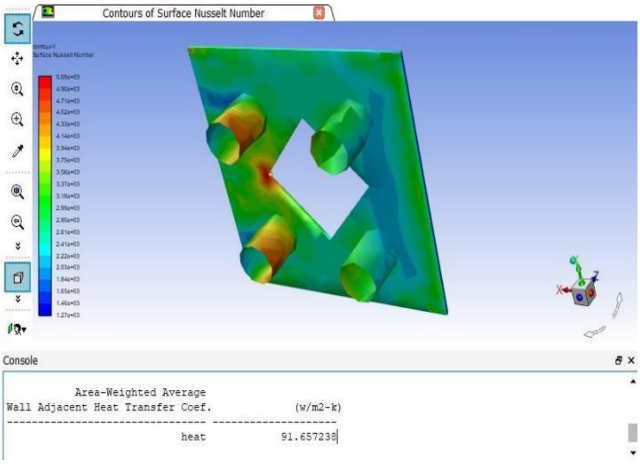
```
            Area-Weighted Average
               Static Temperature                   (k)
--------------------------------  --------------------
                            inlet            296.00001
                           outlet            298.63208
                ----------------  --------------------
                              Net            297.31604
```

Fig. 14. Temp distribution of Circular perforated fins



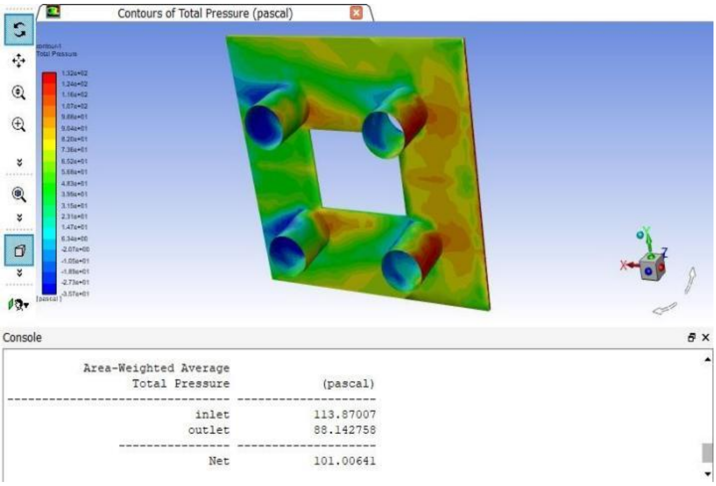
```
Done.

Calculation complete.

             Area-Weighted Average
Wall Adjacent Heat Transfer Coef.                (w/m2-k)
-------------------------------- --------------------
                            heat            116.68472
```

Fig. 15. average of heat transfer coefficient reading for Rectangle perforated plate-fin.



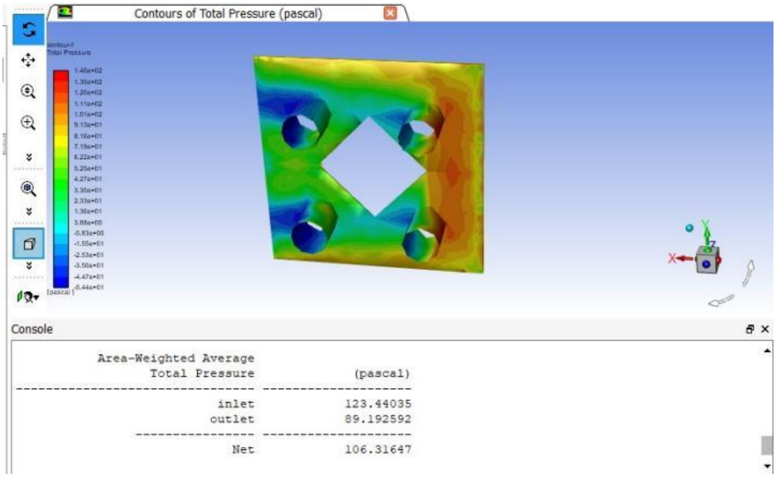
```
             Area-Weighted Average
Wall Adjacent Heat Transfer Coef.                (w/m2-k)
-------------------------------- --------------------
                            heat            91.657238
```

Fig. 16. pressure reading for Rectangular perforated plate-fin



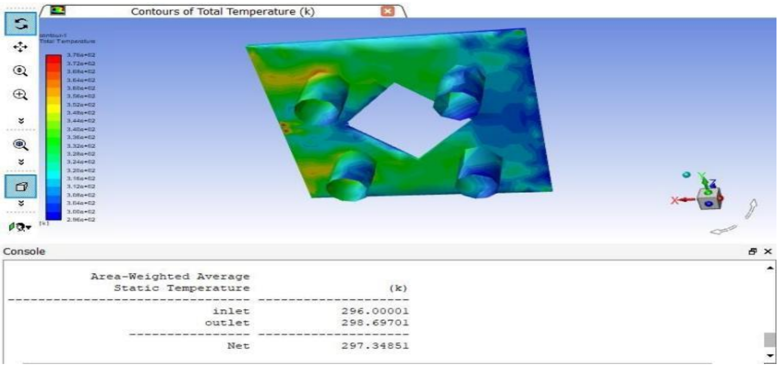
```
             Area-Weighted Average
                Total Pressure                  (pascal)
-------------------------------- --------------------
                          inlet             113.87007
                         outlet             88.142758
                ---------------- --------------------
                            Net             101.00641
```

Fig. 17. Temp distribution of Rectangular perforated fins



```
            Area-Weighted Average
                   Total Pressure                (pascal)
-------------------------------- --------------------
                            inlet            123.44035
                           outlet            89.192592
                ---------------- --------------------
                              Net            106.31647
```

Fig. 18. average of heat transfer coefficient reading for Rhombus perforated plate-fin.



```
            Area-Weighted Average
               Static Temperature                     (k)
-------------------------------- --------------------
                            inlet            296.00001
                           outlet            298.69701
                ---------------- --------------------
                              Net            297.34851
```

Fig. 19. Temp distribution of rhombus perforated fins

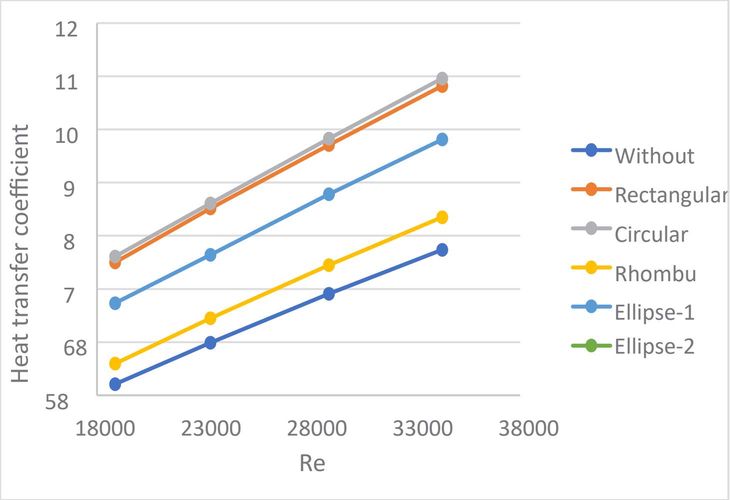
5 Analysis

Table 3. Performance analysis

variation of parameter in different fin configuration	Without Perforated Fins	With Perforated Fins				
		Square	Circular	Rhombus	Ellipse-1	Ellipse-2
Net temperature, T(k)	296	296	296	296	296	296
Net pressure, P(Pa)	117.33	113.97	113.27	123.44	119.33	112.43
Avg. heat transfer coefficient rate, h(w/m²-k)	85.379	116.68	119.44	91.65	106.1	114.5

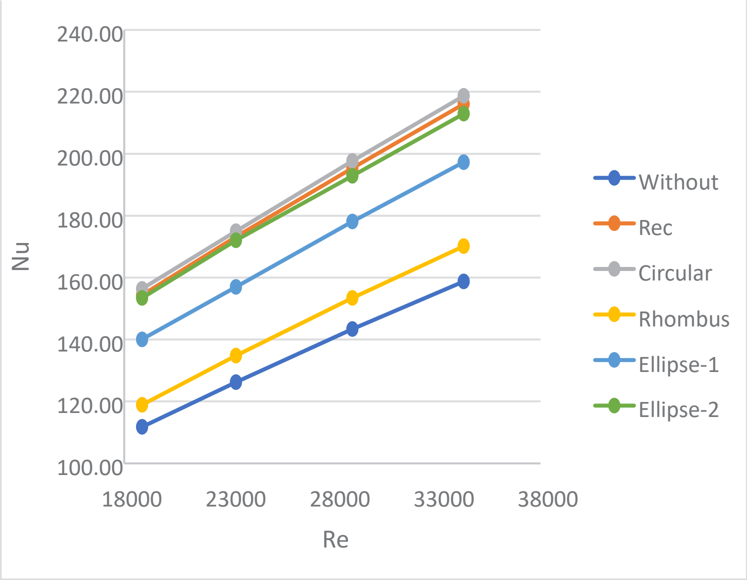
The above table concluded that the circular type perforated fin improves the heat transfer coefficient by 29% of the total transport coefficient value when compared to a fin without perforations. Plate finned heat exchangers with a circular form often have a greater heat transfer coefficient because the fluid or airflow between the fins is more effectively facilitated by the circular shape. Heat transmission between the fluid and the fins is improved by this design, which also decreases flow separation and enhances turbulence. Furthermore, the circular form reduces pressure drop, increasing the efficiency of heat transfer overall.

6 Results and Discussion



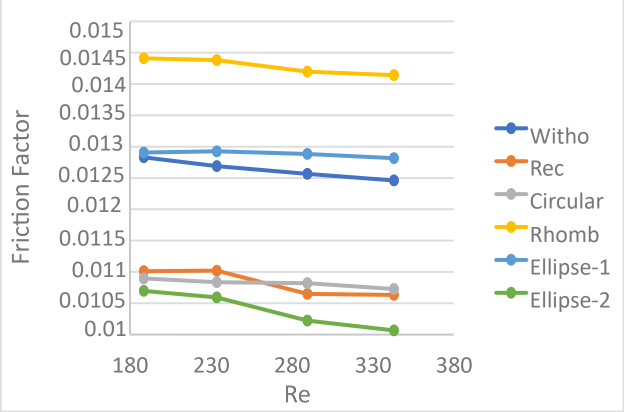
Graph 3. Reynolds no. vs Heat transfer coefficient

The above graph concluded that the circular type perforated fin has a higher heat transfer coefficient value than fins without perforations and other fin shapes.



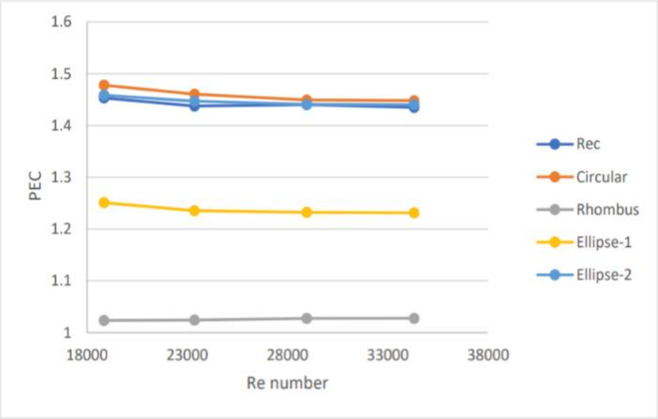
Graph 4. Reynolds no. vs Nusselt Number

The above graph concluded that the circular type perforated fin has a higher Nusselt number when compared to without perforated fin and other fin perforations.



Graph 5. Reynolds no. vs Friction factor

The above graph concluded that the ellipse-2 perforated fin has a low friction factor value and the rhombus type perforated fin has a high friction factor value.



Graph 6. Reynolds no. vs PEC

Reynolds no. vs. PEC graph shows that, when compared to a fin without a perforation, the circular type perforated fin has a higher PEC value and the rhombus type perforated fin has a lower PEC value.

7 Conclusions

The analysis was performed through numerical analysis with various types of perforated fin models.

- Initially, non-perforated fin was performed and compared with previous work. The friction factor and Nusselt number are in good agreement with earlier research.
- When compared to without perforated fins, the circular type perforated fin has a higher heat transfer coefficient, increasing 29% of the total transport coefficient value.
- The Nusselt number is also higher and increased by 29% with the circular type of perforated fin.

- Increasing the Reynolds number increases the heat transfer coefficient while decreasing the friction factor thus improving life of the equipment.
- A circular perforated fin has high performance, and the rhombus type perforated fin has lower performance.
- Finally, it was concluded that the circular type perforated fin is the good choice for plate fin heat exchangers for thermal performance and sustenance of the thermal equipment.

8 Future Scope

Further analysis can be carried out regarding plain staggered flat tubes with wavy staggered or plain staggered circle tubes. Practical can be conducted for the justification of simulated values and prolonged life of thermal equipment can be measured.

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