



Design And Analysis Of Compact Heat Exchanger In Cfd

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ABSTRACT

Enhancing convective heat transfer while maintaining an acceptable pressure drop is a critical requirement in compact heat exchanger design. In this study, an air-side passive enhancement technique is examined using sinusoidal fins integrated with surface dimples through a comparative CFD analysis of three configurations: plain, elliptical dimpled, and circular dimpled fins. The geometries are created in SolidWorks and simulated in ANSYS Fluent under steady-state turbulent flow conditions for inlet velocities ranging from 1 to 2.5 m/s. The results indicate that the addition of dimples significantly improves heat transfer performance compared to the plain fin. Among the tested configurations, the circular dimple exhibits the highest enhancement of about 18–20%, while the elliptical dimple shows an improvement of approximately 16%. This enhancement is mainly due to increased turbulence intensity, improved fluid mixing, and disruption of the thermal boundary layer. The Nusselt number also increases for both dimpled cases, confirming the improvement in convective heat transfer. However, this benefit is accompanied by an increase in pressure drop, where the elliptical dimple results in a higher rise of about 10.6%, while the circular dimple shows a relatively lower increase of 7–8%. Overall, the circular dimpled sinusoidal fin provides a better balance between heat transfer enhancement and pressure loss, making it more suitable for compact heat exchanger applications.

Keywords: Sinusoidal fin, dimpled surface, elliptical dimple, circular dimple, heat transfer enhancement, Nusselt number, pressure drop.

1. INTRODUCTION

The dissipation of heat within a thermal system is strongly dependent on the rate at which thermal energy is transferred between a heated surface and the surrounding fluid. This process occurs through conduction, convection, and radiation, among which convection plays a dominant role in compact heat exchangers and modern thermal management systems [1]. Efficient heat transfer is essential in a wide range of applications such as electronic cooling systems, industrial heat exchangers, automotive radiators, and energy conversion devices [2].

The heat transfer rate is primarily governed by parameters such as temperature gradient [3], effective heat transfer surface area [4], and convective heat transfer coefficient [5]. Enhancement of heat transfer can be achieved by increasing the surface area and disturbing the boundary layer using extended surfaces

such as fins, ribs, and surface modifications like dimples [6]. Among these techniques, dimple-based surface enhancement is widely considered an effective passive method due to its ability to improve heat transfer with relatively low manufacturing complexity and energy requirement.

Dimpled surfaces are known to promote flow separation and reattachment, which leads to the formation of vortices and enhanced turbulence intensity near the heated surface. This flow behavior reduces boundary layer thickness, improves fluid mixing, and increases the local convective heat transfer coefficient [7]. As a result, dimpled geometries are extensively used in HVAC systems [8], power plants [9], chemical processing units [10], automotive cooling systems [11], and heat exchanger applications [12]. However, the increase in turbulence is often accompanied by a rise in frictional losses and pressure drop, which must be carefully balanced during thermal design optimization.

A number of researchers have investigated the influence of different dimple geometries on thermal–hydraulic performance. Circular dimples are widely used due to their simple geometry and strong vortex generation capability, while elliptical dimples have been reported to provide improved heat transfer performance with relatively lower pressure drop due to smoother flow transitions [13]. Studies have shown that dimple shape, depth, and arrangement significantly affect flow separation behavior and overall heat transfer characteristics [14]. For example, elliptical configurations are often associated with improved thermal efficiency, whereas circular dimples provide strong turbulence generation but may result in slightly higher flow resistance.

Although several studies have been conducted on flat surfaces and conventional channel flows, limited research has focused on the combined effect of sinusoidal fin structures with dimpled surfaces. Sinusoidal fins inherently enhance heat transfer by increasing surface area and generating secondary flow due to their wavy geometry, which promotes better fluid mixing and boundary layer disruption. However, the performance of sinusoidal fins can still be further improved by introducing surface modifications such as dimples.

In this study, a sinusoidal fin configuration is considered and modified using elliptical and circular dimples to enhance thermal performance. The geometries are developed using SolidWorks and analyzed using ANSYS Fluent under steady-state turbulent flow conditions. Air is used as the working fluid, and simulations are conducted for inlet velocities ranging from 1 m/s to 2.5 m/s to evaluate thermal and flow characteristics under different operating conditions.

Thus, the objectives of this study are:

- To design and develop sinusoidal fin configurations with elliptical and circular dimples
- To perform numerical analysis using Computational Fluid Dynamics (CFD) under turbulent flow conditions
- To evaluate thermal performance parameters such as Nusselt number and heat transfer coefficient
- To analyze flow characteristics including pressure drop, wall shear stress, and skin friction coefficient
- To compare the thermal–hydraulic performance of dimpled fins with a plain sinusoidal fin
- To identify the most efficient configuration that provides maximum heat transfer with acceptable pressure loss

2. METHODOLOGY

2.1 PHYSICAL MODEL DEVELOPMENT OF SINUSOIDAL FIN WITH DIMPLE DESIGN

The present study focuses on enhancing convective heat transfer using a sinusoidal fin integrated with dimple structures. A wavy (sinusoidal) fin is selected as the base model due to its ability to increase surface area and induce secondary flow. The geometry is developed using SOLIDWORKS. The fin has a length of 360 mm, width of 225 mm, height of 8 mm, and thickness of 4 mm. The sinusoidal profile consists of 8 waves, with an amplitude of 4 mm and wavelength of 45 mm. To improve thermal performance, dimples are introduced on the fin surface. The dimple parameters are: diameter = 11 mm,

depth = 5 mm, and pitch = 46 mm, arranged in a 4×16 pattern over the surface. Three configurations are considered: plain sinusoidal fin, sinusoidal fin with circular dimples, and sinusoidal fin with elliptical

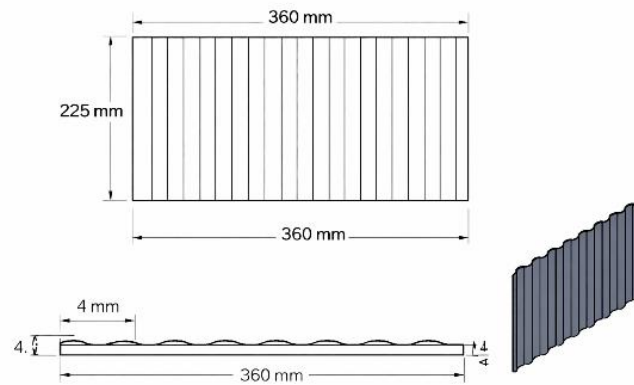


Figure.1

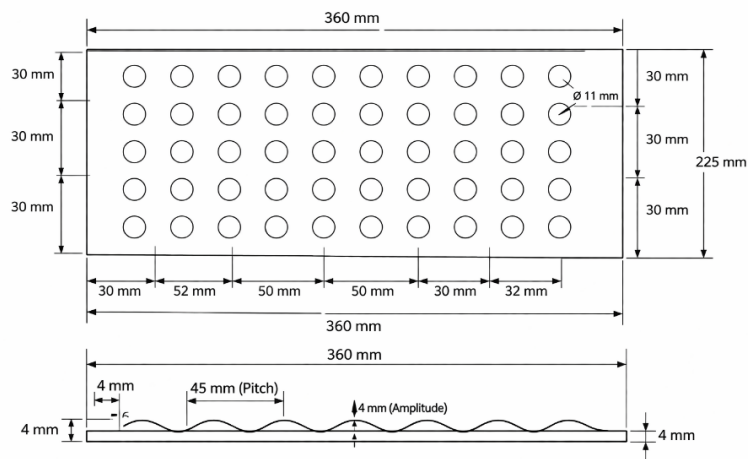


Figure.2

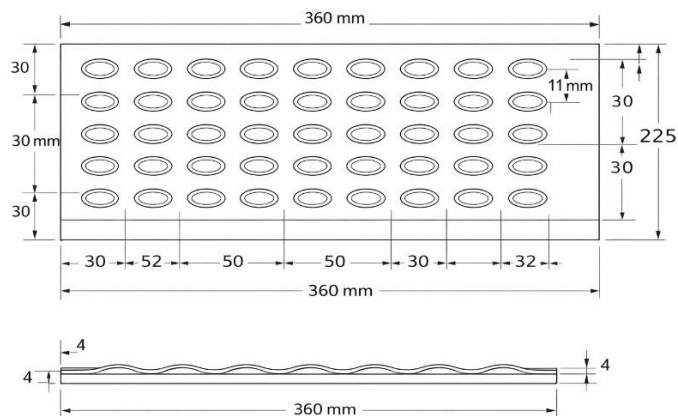


Figure.3

dimples. The 2D drawings of these configurations (Figure 1, Figure 2 and Figure 3) illustrate the dimensional details and dimple distribution pattern.

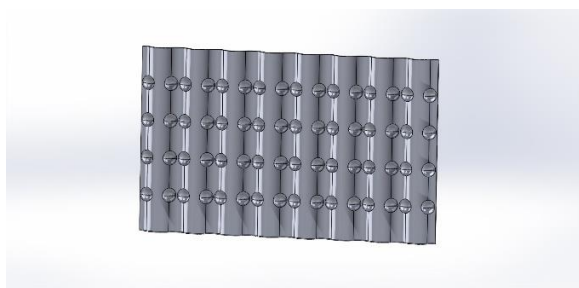
2.1.1. MATERIAL PROPERTIES AND BOUNDARY CONDITIONS

Copper is selected as the fin material due to its high thermal conductivity, which enhances heat transfer performance. The material properties are taken from standard HMT data sources: density (8960 kg/m^3), thermal conductivity ($167 \text{ W/m}\cdot\text{K}$), specific heat capacity ($385 \text{ J/kg}\cdot\text{K}$), Young's modulus (110 GPa), and Poisson's ratio (0.34). Air is used as the working fluid, and its properties are evaluated at the film temperature for better accuracy. The film temperature is calculated as the average of inlet and hot surface temperatures, resulting in 70°C . At this temperature, the air properties obtained from the HMT data book are: density (1.029 kg/m^3), dynamic viscosity ($20.59 \times 10^{-6} \text{ kg/m}\cdot\text{s}$), kinematic viscosity ($20.02 \times 10^{-6} \text{ m}^2/\text{s}$), thermal diffusivity ($28.5568 \times 10^{-6} \text{ m}^2/\text{s}$), Prandtl number (0.694), specific heat capacity ($1009 \text{ J/kg}\cdot\text{K}$), and thermal conductivity ($0.02966 \text{ W/m}\cdot\text{K}$).

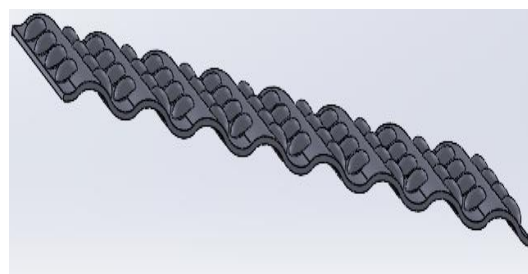
The simulations are carried out under turbulent flow conditions with inlet velocities of 1, 1.5, 2, and 2.5 m/s, corresponding to Reynolds numbers of 3503.29, 5254.94, 7006.59, and 8758.24. The inlet temperature is maintained at 27°C , the outlet temperature is set at 27°C , and the fin surface (hot body) is kept at a constant temperature of 113°C . These conditions ensure consistent comparison of thermal performance among all configurations.

3D Model Development and Numerical Setup

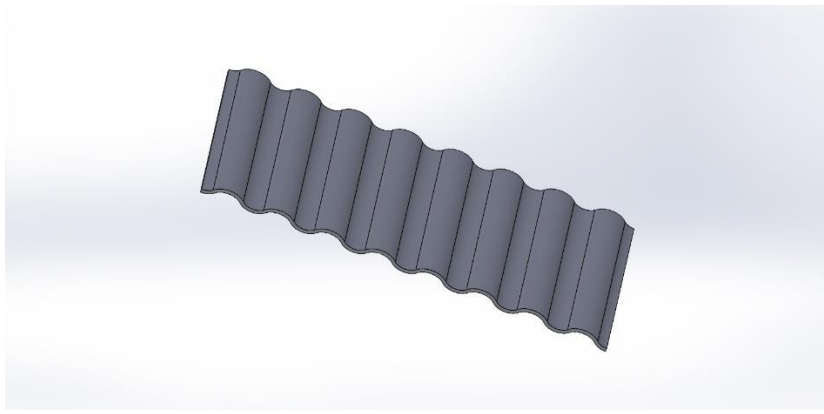
The 3D model of the sinusoidal fin is created in SOLIDWORKS and imported into ANSYS Fluent (R2 version) for numerical simulation. An outer rectangular duct is generated using the enclosure method, and the sinusoidal fin is placed inside the domain. A Boolean subtraction operation is performed to obtain the fluid region. The mesh is generated with fine elements to ensure accuracy. The solver is set to a pressure-based, steady-state condition with double precision. The realizable $k-\epsilon$ turbulence model with enhanced wall treatment is used. Boundary conditions include velocity inlet, pressure outlet, constant temperature at the hot surface, and adiabatic outer walls. The 3D models and computational setup are illustrated in Figure 3. Post processing is performed to obtain contours and parameters such as heat transfer rate, pressure drop, Nusselt number, and turbulence characteristics for performance comparison.



(a)



(b)



(c)

Figure.3.(a) Sinusoidal Fin with Circle dimple (b) Sinusoidal Fin with Ellipse dimpled (c) Sinusoidal Plain Fin.

2.2 PARAMETER DESCRIPTION

The thermo-hydraulic performance of the sinusoidal fin configurations is evaluated using key flow and thermal parameters under controlled conditions. In this study, three models are considered: a plain sinusoidal fin, a circular dimpled fin, and an elliptical dimpled fin, to examine the effect of surface modification on heat transfer and pressure drop characteristics. Air is used as the working fluid with a constant inlet temperature of 27°C, while the fin surface is maintained at a uniform temperature of 113°C. The flow inside the rectangular duct is assumed to be steady, three-dimensional, and turbulent. The inlet velocity is varied as 1, 1.5, 2, and 2.5 m/s to analyze its influence on thermal and flow behavior, corresponding to Reynolds numbers within the turbulent range. As the velocity increases, the mass flow rate also increases proportionally, which significantly affects both heat transfer rate and pressure drop across the fin. The Reynolds number is used as the primary parameter to define the flow regime, and it is calculated based on the hydraulic diameter of the duct. All cases fall within the turbulent region, ensuring a consistent and reliable comparison of thermo-hydraulic performance among the different fin configurations.

S.No	Input parameter	Parameter values			
1	Mass flow rate (kg/s) Plain fin	0.0120	0.0180	0.0245	0.0301
2	Ellipse Dimple	0.0124	0.0186	0.0248	0.0310
3	Circle Dimple	0.0128	0.0192	0.0256	0.0321
4	Velocity (m/s)	1	1.5	2	2.5
5	Inlet temperature (C)	27	27	27	27
6	Reynolds number	3503.29	5254.94	7006.59	8758.24

2.3 NUMERICAL PROCEDURE

The thermo-fluid performance of three sinusoidal fin configurations—plain, circular dimpled, and elliptical dimpled—is evaluated using Computational Fluid Dynamics (CFD) analysis. The geometries are developed in SOLIDWORKS and imported into ANSYS DesignModeler to generate the computational domain through enclosure creation and Boolean subtraction. The study considers steady, three-dimensional, incompressible turbulent airflow, and the governing continuity, momentum, and energy equations are solved using the Reynolds-Averaged Navier–Stokes (RANS) formulation. The realizable $k-\epsilon$ turbulence model with enhanced wall treatment is employed to accurately capture near-wall flow behavior and heat transfer effects. A pressure-based, double-precision solver is used with coupled pressure–velocity interaction and second-order discretization schemes to ensure numerical accuracy. Boundary conditions include a velocity inlet varying from 1 to 2.5 m/s at 27°C, a pressure outlet, and a constant temperature of 113°C applied to the fin surface, while the remaining walls are treated as adiabatic with no-slip conditions. The solution is initialized using a hybrid method and iterated until convergence is achieved, and key performance parameters such as heat transfer coefficient, Nusselt number, pressure drop, and total heat transfer rate are extracted for comparative analysis.

2.3.1.MESH MODELLING AND GRID GENERATION CHECKING

The computational domain is discretized using ANSYS Meshing to effectively capture the fluid flow and heat transfer behavior within the rectangular duct containing sinusoidal fin configurations. The study includes plain, circular dimpled, and elliptical dimpled fins, which involve complex geometrical features requiring careful mesh generation. A three-dimensional unstructured mesh with a mesh size of 5 is created in ANSYS Workbench (R2), resulting in approximately 801,147 nodes and 4,218,040 elements. Mesh refinement is applied near the fin surfaces and duct walls to accurately resolve steep velocity and temperature gradients, thereby improving boundary layer prediction and capturing turbulence effects generated by the dimples. Simulations are performed for inlet velocities of 1, 1.5, 2, and 2.5 m/s, all within the turbulent regime. The highest velocity case (2.5 m/s) is considered for detailed comparison due to its stronger turbulence effects. The adopted mesh provides a suitable balance between computational efficiency and solution accuracy, and is therefore used to evaluate key parameters such as Nusselt number, heat transfer coefficient, pressure drop, and wall shear stress.

Table.2.3.1.Mesh element size, nodes and elements of different dimples

Type of dimples	Element size	Nodes	Elements
Circle shaped	5	835,920	4395,780
	4	1095,740	5710,860
	3	1545,620	8210,950
	2	2,325,880	12,640,520
	1	4,120,950	22,450,640
Ellipse shaped	5	801,147	4218,040
	4	1,052,380	5,486,920
	3	1,486,750	7,865,430
	2	2,245,910	11,952,780
	1	3,985,620	21,864,300
Plain Sinusoidal Fin	5	720,850	3,850,420
	4	945,630	5,020,310
	3	1,325,780	7,120,540
	2	2,010,420	10,850,630
	1	3,520,860	19,420,780

2.4 DUCT SIZE

The computational domain is modeled as a rectangular duct to evaluate the thermo-hydraulic performance of plain, circular dimpled, and elliptical dimpled sinusoidal fins. The geometry is developed in ANSYS Workbench (R2) using the enclosure approach, where the fluid region is defined around the fin and obtained through a Boolean subtraction process. The duct dimensions are set as ± 360 mm in the flow direction, ± 2 mm in the transverse direction, and ± 30 mm in the vertical direction, providing sufficient space for flow development. The sinusoidal fin, placed centrally inside the duct, has a length of 360 mm, width of 225 mm, height of 8 mm, and thickness of 4 mm, with a wave profile defined by 4 mm amplitude and 45 mm wavelength, forming 8 waves.

Boundary conditions are assigned using named selections, including velocity inlet, pressure outlet, adiabatic outer walls, and a constant temperature at the fin surface. The duct size is chosen to ensure proper development of velocity and thermal boundary layers under turbulent conditions without excessive computational cost. The hydraulic diameter of the duct, calculated from its cross-sectional dimensions, is 0.0681 m and is used as the characteristic length for evaluating flow and heat transfer performance.

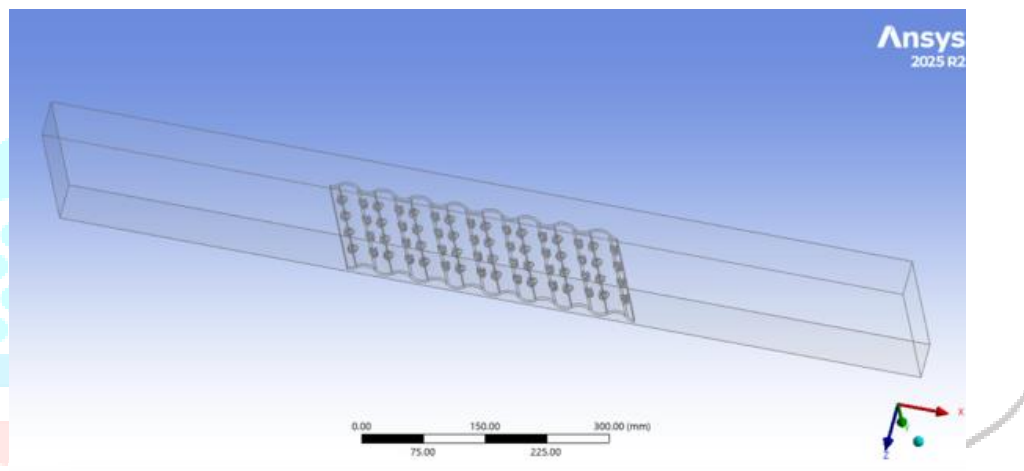


Figure 4. Duct with dimpled plate

2.5 BOUNDARY CONDITIONS AND THERMO-PHYSICAL PROPERTIES

All sinusoidal fin configurations plain, circular dimpled, and elliptical dimpled are modeled with a no-slip boundary condition. Air is used as the working fluid with an inlet temperature of 27°C , and its thermophysical properties at operating conditions are: density (1.029 kg/m^3), specific heat ($1009 \text{ J/kg}\cdot\text{K}$), dynamic viscosity ($2.059 \times 10^{-5} \text{ Ns/m}^2$), thermal conductivity ($0.02966 \text{ W/m}\cdot\text{K}$), and Prandtl number (0.694). The fin is made of copper due to its high thermal conductivity, with properties of density (8960 kg/m^3), thermal conductivity ($167 \text{ W/m}\cdot\text{K}$), and specific heat capacity ($385 \text{ J/kg}\cdot\text{K}$). A uniform inlet velocity ranging from 1 to 2.5 m/s is applied, corresponding to Reynolds numbers between 3503 and 8758, ensuring turbulent flow conditions. The fin surface is maintained at a constant temperature of 113°C , while the outer duct walls are treated as adiabatic, and a pressure outlet condition is specified for smooth flow exit. The simulations are conducted under steady-state conditions using the Reynolds-Averaged Navier–Stokes (RANS) equations with the realizable k – ϵ turbulence model and enhanced wall treatment. A tetrahedral mesh is used for discretization, and the solution is iterated until convergence, ensuring accurate and consistent evaluation of thermo-hydraulic performance.

2.6 GOVERNING EQUATIONS

The fluid flow and heat transfer within the rectangular duct are governed by the fundamental conservation equations of mass, momentum, and energy. In the present study, the flow is assumed to be steady, three-dimensional, incompressible, and turbulent. The Reynolds-Averaged Navier–Stokes (RANS) approach is employed to model the turbulent flow behavior.

The continuity equation, representing conservation of mass, is given by:

$$\frac{\partial(\rho u_i)}{\partial x_i} = 0$$

The momentum equation, which accounts for conservation of momentum in the flow field, is expressed as:

$$\rho u_j \frac{\partial u_i}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[(\mu + \mu_t) \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right]$$

The energy equation, governing heat transfer within the fluid domain, is written as:

$$\rho u_i \frac{\partial T}{\partial x_i} = \frac{\partial}{\partial x_i} \left[\left(\frac{k}{c_p} + \frac{\mu_t}{Pr_t} \right) \frac{\partial T}{\partial x_i} \right]$$

In these equations, u_i and u_j represent the velocity components, ρ is the fluid density, p is the pressure, μ is the dynamic viscosity, and μ_t is the turbulent viscosity. The temperature is denoted by T , while k , c_p , and Pr_t represent thermal conductivity, specific heat capacity, and turbulent Prandtl number, respectively.

The realizable k - ϵ turbulence model, which adds extra transport equations for turbulent kinetic energy (k) and its dissipation rate (ϵ), is used to simulate turbulence effects. For the purpose of examining heat transport in sinusoidal fins with dimples, these equations provide precise prediction of turbulence strength, energy dissipation, and increased mixing. ANSYS Fluent's finite volume method is used to solve the aforementioned governing equations numerically, offering a thorough understanding of the thermal performance and flow characteristics of the various fin designs.

2.7 POST-PROCESSING AND DATA ANALYSIS

After achieving convergence, post-processing is performed to evaluate the thermal and flow performance of the sinusoidal fin configurations. Key parameters such as mass flow rate, heat transfer rate, pressure drop, wall shear stress, skin friction coefficient, Nusselt number, and surface heat transfer coefficient are extracted for analysis. Temperature and velocity contours are used to visualize the distribution of thermal and flow fields inside the duct, highlighting the influence of circular and elliptical dimples on turbulence and fluid mixing, while streamline plots help identify vortex formation, flow separation, and reattachment regions. The results are compared for plain, circular dimpled, and elliptical dimpled fins by analyzing the variation of heat transfer rate and pressure drop with inlet velocity to assess overall thermo-hydraulic performance. Performance improvements relative to the plain fin are also calculated, and the results are presented using graphs and tables for clear comparison and interpretation.

2.8 VALIDATION OF NUMERICAL MODEL

The numerical model is validated by comparing analytical predictions with CFD results for the plain sinusoidal fin at the highest inlet velocity of 2.5 m/s. The analytical approach gives a Reynolds number of 8509, Nusselt number of 697.55, heat transfer coefficient of 12.03 W/m²K, and a heat transfer rate of about 90.2 W. In contrast, the CFD simulation predicts a Reynolds number of 8758.24, Nusselt number of 727.56, heat transfer coefficient of 17.61 W/m²K, and heat transfer rate of 370.87 W, along with a higher pressure drop of 5.90 Pa compared to 0.54 Pa from analytical estimation. The variation between analytical and numerical results is mainly due to the simplified assumptions used in empirical correlations, which do not fully represent the complex flow behavior over a sinusoidal surface. The CFD model captures additional effects such as flow separation, vortex formation, and enhanced turbulence,

leading to higher heat transfer and pressure loss. Despite this difference in magnitude, both approaches follow similar trends, confirming that the numerical model is reliable and suitable for analyzing the thermo-hydraulic performance of the fin configurations.

3. RESULT AND DISCUSSION

CFD simulations are used to examine the thermal and flow performance of sinusoidal fins with various surface shapes, such as plain fin, elliptical dimpled fin, and eye-shaped dimpled fin. The velocity distribution, temperature distribution, heat transfer properties, and pressure change are discussed in relation to the results. Additionally, the impact of input velocity on thermal performance and flow behavior is thoroughly investigated.

3.1 VELOCITY DISTRIBUTION

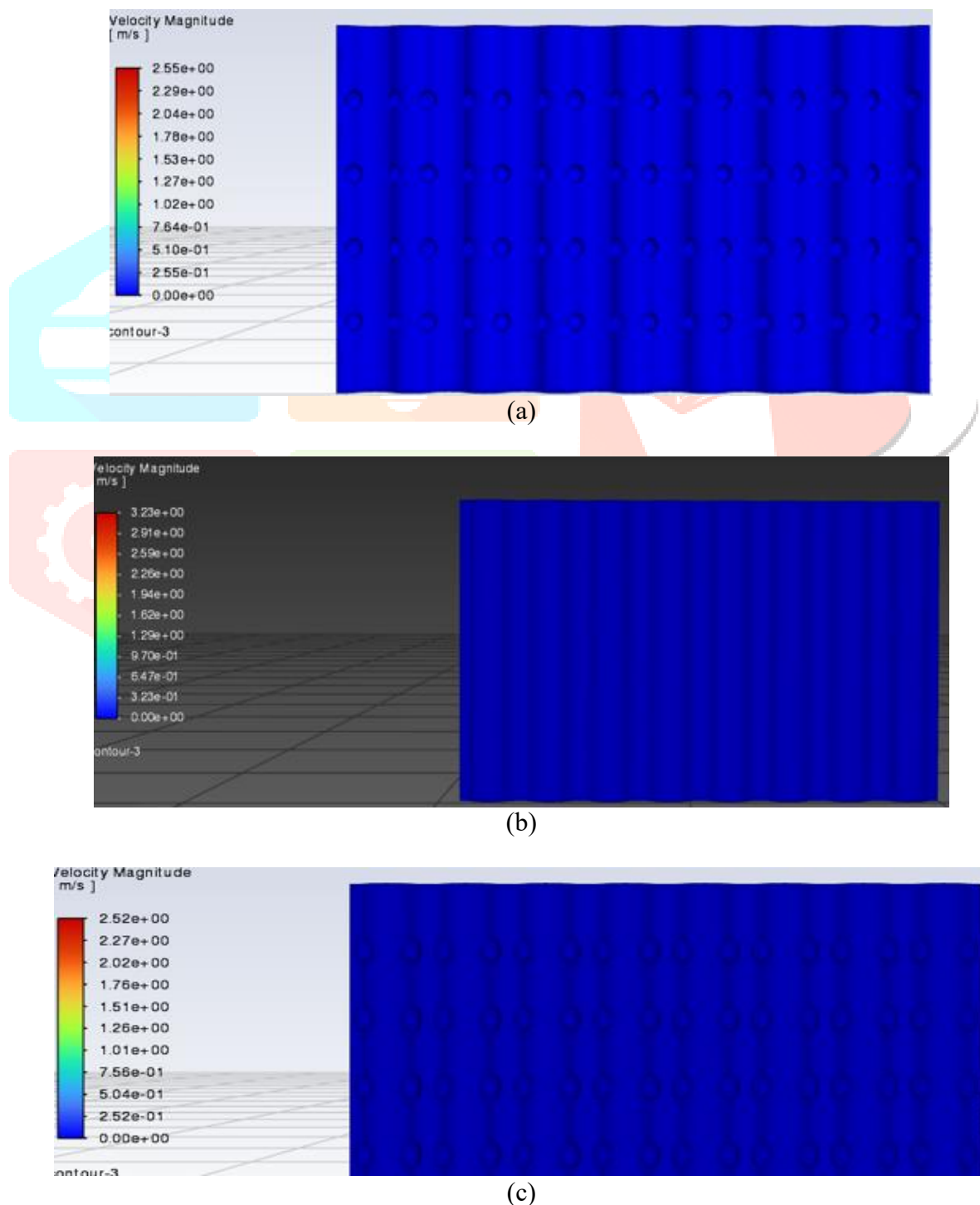
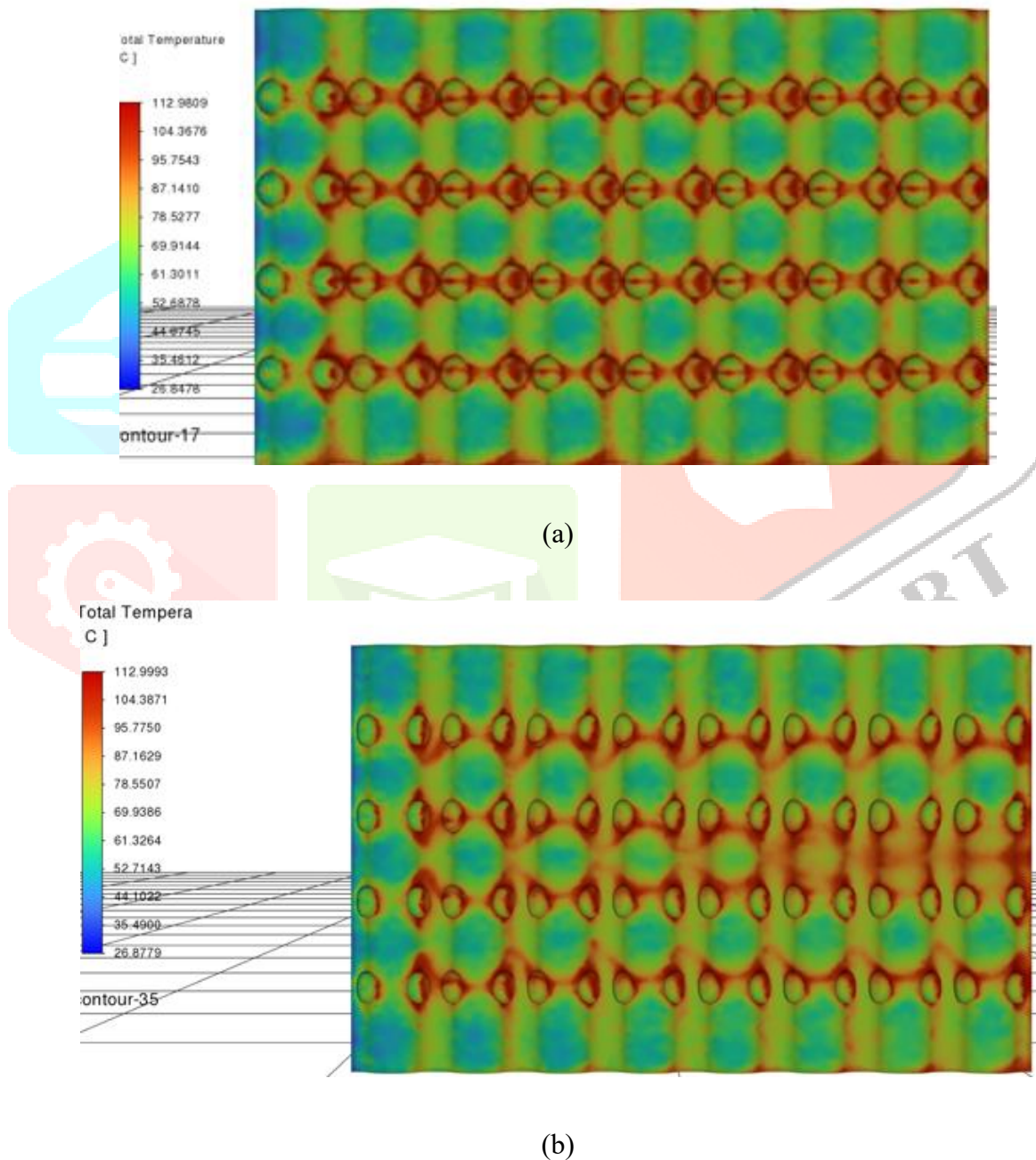
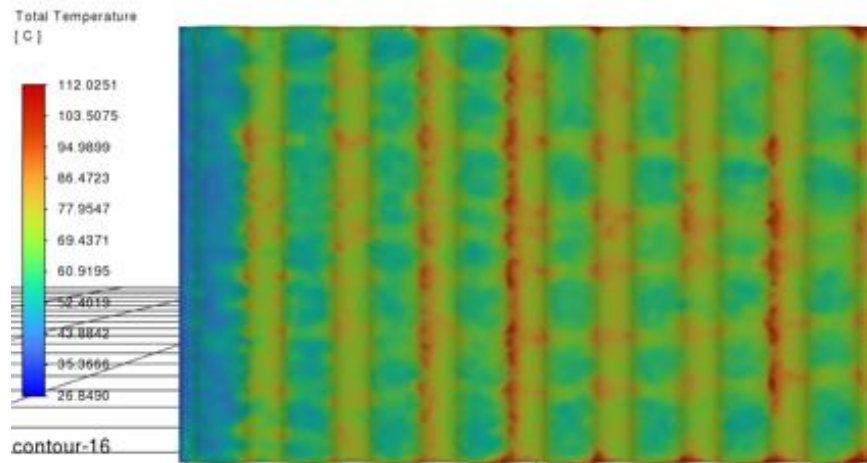


Figure.3.1.Velocity distribution (a) Sinusoidal Fin with Circle dimple (b) Sinusoidal Fin with Ellipse dimpled (c) Sinusoidal Plain Fin

The velocity distribution for the plain and dimpled sinusoidal fin configurations is shown in Figure 3. The plain fin exhibits a typical turbulent flow profile, where the velocity is highest at the core region and gradually decreases towards the wall due to boundary layer development. In contrast, the dimpled fins show noticeable disturbances in the velocity field, with flow acceleration occurring at the upstream edge of the dimples and deceleration within the cavity regions. The circular dimple generates stronger velocity gradients and more intense vortex structures near the surface, while the elliptical dimple produces comparatively smoother velocity variation. These flow disturbances enhance turbulence and improve heat transfer, but also increase pressure drop. Among the configurations, the circular dimple fin exhibits the highest velocity disturbance and better thermal performance compared to the plain and elliptical dimpled fins.

3.2. TEMPERATURE DISTRIBUTION



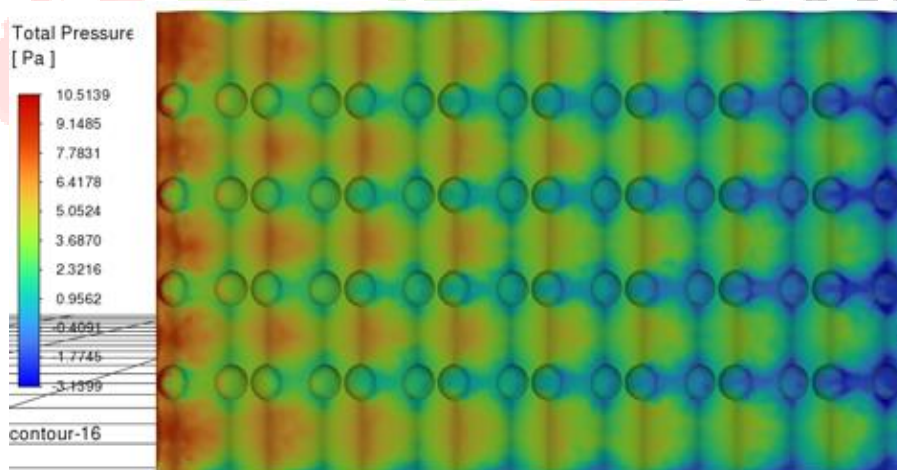


(c)

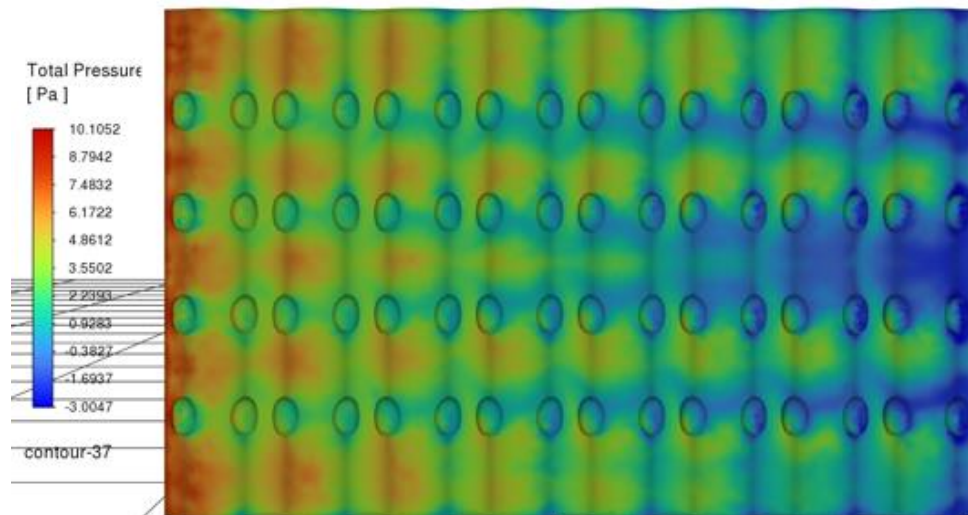
Figure.3.2. Temperature distribution (a) Sinusoidal Fin with Circle dimple (b) Sinusoidal Fin with Ellipse dimpled (c) Sinusoidal Plain Fin

The temperature distribution along the sinusoidal fin surface for different configurations is illustrated in Figure 6. In the plain sinusoidal fin, a relatively thicker thermal boundary layer develops, resulting in gradual temperature variation and lower heat transfer performance. In contrast, the dimpled fins significantly alter the temperature field due to enhanced fluid mixing and vortex formation, which reduce the boundary layer thickness. The circular dimple generates stronger turbulence and higher local temperature gradients near the surface, while the elliptical dimple provides smoother temperature variation with moderate enhancement. Overall, both dimpled configurations improve heat dissipation compared to the plain fin, with the circular dimpled fin showing the highest thermal performance and the plain sinusoidal fin the lowest.

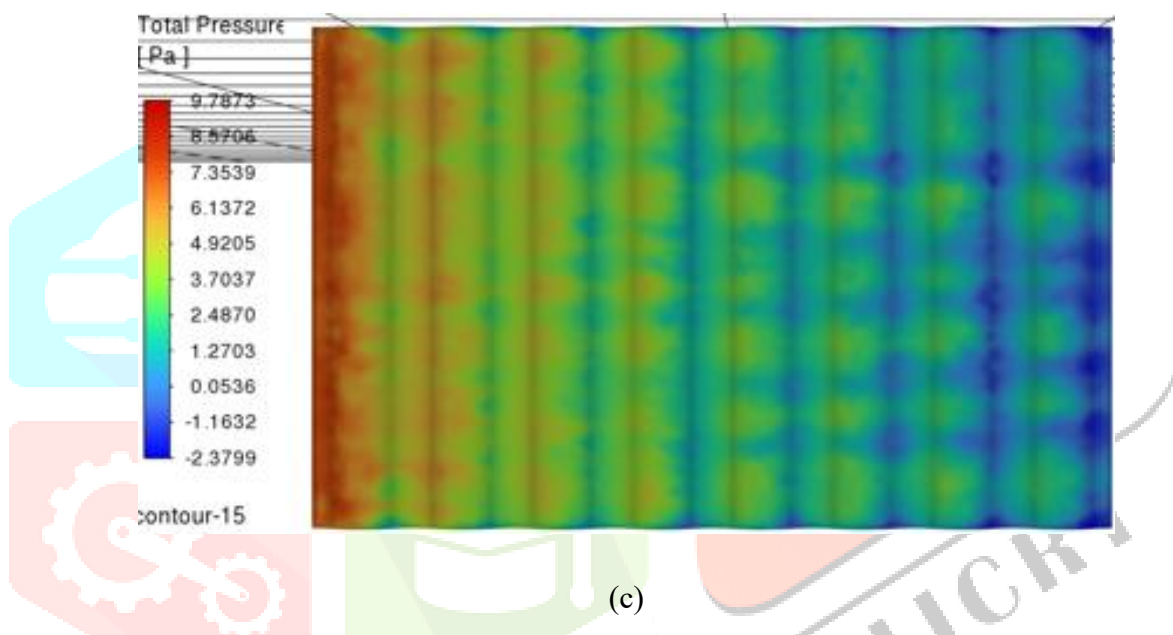
3.3. PRESSURE DISTRIBUTION



(a)



(b)



(c)

Figure.3.3. Heat transfer coefficient distribution (a) Sinusoidal Fin with Circle dimple (b) Sinusoidal Fin with Ellipse dimpled (c) Sinusoidal Plain Fin

A gradual pressure reduction is observed from inlet to outlet due to frictional losses along the flow path. In the plain sinusoidal fin, the pressure variation is smooth with minimal local disturbances. In contrast, the dimpled configurations exhibit noticeable local pressure fluctuations. As the flow passes over each dimple, a slight pressure increase occurs at the leading edge, followed by a drop within the cavity due to flow separation and recirculation, and partial recovery downstream during reattachment. These repeated variations indicate increased flow resistance in the dimpled fins. However, they also enhance turbulence and fluid mixing, which improves heat transfer. Overall, the dimpled sinusoidal fins show higher pressure drop than the plain fin, highlighting the trade-off between thermal performance and hydraulic loss.

3.4 COMPARISON OF NUSSELT NUMBER HEAT TRANSFER CO EFFICIENT

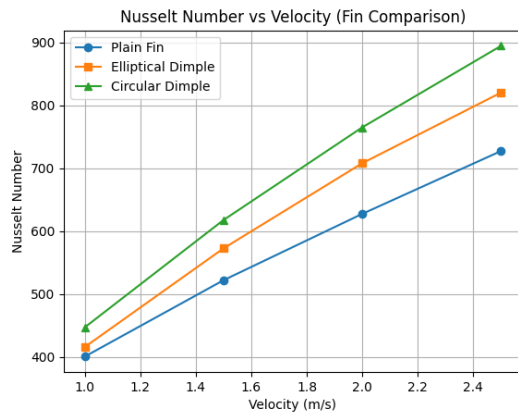


Figure 3.4. (a) Nusselt number variations

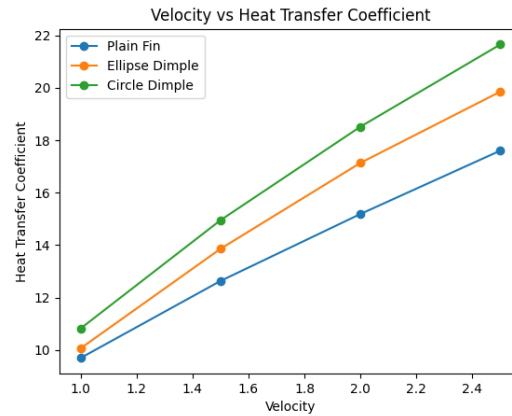


Figure 3.4. (b) Heat transfer co efficient variations

The variation of Nusselt number with inlet velocity for different sinusoidal fin configurations shows a clear increasing trend. As the velocity increases from 1 m/s to 2.5 m/s, the Reynolds number rises, leading to higher turbulence intensity and improved convective heat transfer for all cases.

Among the configurations, the circular dimpled fin exhibits the highest Nusselt number across all velocities, followed by the elliptical dimpled fin, while the plain fin shows the lowest values. At the maximum velocity of 2.5 m/s, the Nusselt number reaches approximately 894.87 for the circular dimple, 820.53 for the elliptical dimple, and 727.56 for the plain fin. This corresponds to an enhancement of about 23% for circular dimples and 13% for elliptical dimples compared to the plain fin. Additionally, the circular dimple provides nearly 9% higher performance than the elliptical dimple due to stronger vortex formation and more effective boundary layer disruption.

A similar trend is observed in the heat transfer coefficient. As velocity increases, the convective heat transfer coefficient increases due to thinning of the thermal boundary layer and enhanced fluid mixing. At 2.5 m/s, the circular dimple achieves the highest value of approximately 21.65 W/m²K, followed by the elliptical dimple (19.85 W/m²K) and the plain fin (17.61 W/m²K). The enhancement is around 23% for circular dimples and 13% for elliptical dimples compared to the plain configuration.

Overall, the circular dimpled sinusoidal fin demonstrates superior thermal performance due to intensified turbulence, repeated flow separation, and efficient reattachment mechanisms. The elliptical dimple also improves heat transfer, but to a lesser extent, while the plain fin shows the lowest efficiency due to limited flow disturbance.

3.5.COMPARISON OF PRESSURE DROP AND FRICTION FACTOR ANALYSIS

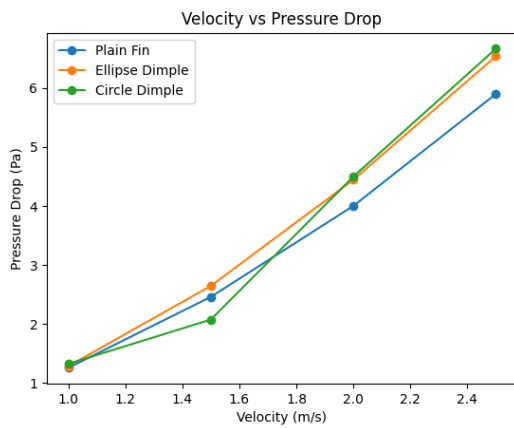


Figure.3.5.(a). Pressure drop

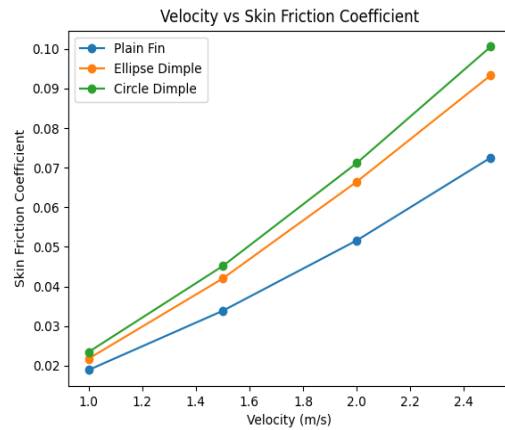


Figure .3.5. (b). Co efficient of friction

The graph shows the variation of pressure drop with velocity for plain sinusoidal, elliptical dimpled, and circular dimpled fin configurations, where the pressure drop increases with velocity from 1 m/s to 2.5 m/s due to higher flow resistance and inertial effects. At 2.5 m/s, the circular dimple fin exhibits the highest pressure drop (6.66 Pa), followed by the elliptical dimple fin (6.53 Pa), while the plain fin shows the lowest value (5.90 Pa), with increases of about 13% and 11% respectively compared to the plain fin. A similar trend is observed for the skin friction coefficient, where the plain fin has the lowest values, and the dimpled fins show higher values due to increased wall shear stress. At the maximum velocity, the circular dimple fin shows approximately 38% higher skin friction coefficient than the plain fin, indicating stronger fluid–surface interaction. Overall, dimpled fins enhance turbulence and heat transfer but also result in higher pressure drop and friction losses compared to the plain sinusoidal fin.

3.6 COMPARISON OF SHEAR STRESS AND OVER ALL HEAT TRANSFER RATE

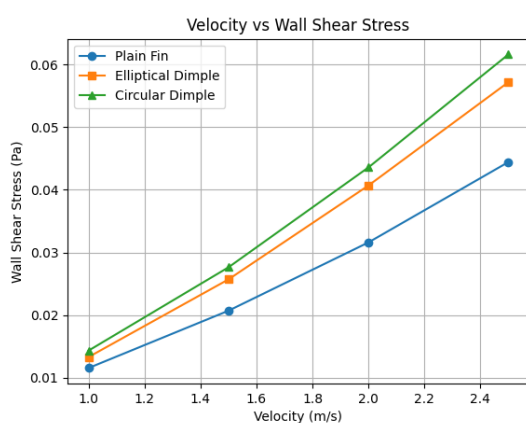


Figure.3.6. (a).Shear stress

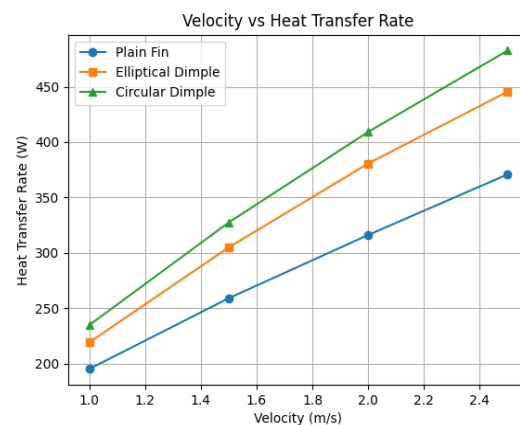


Figure.3.6. (b) . Heat transfer rate

The performance of different sinusoidal fin configurations at the maximum inlet velocity of 2.5 m/s shows a clear improvement in heat transfer along with an increase in flow resistance. For all cases, the presence of dimple structures enhances turbulence and mixing, leading to higher thermal performance compared to the plain sinusoidal fin.

At 2.5 m/s, the plain sinusoidal fin shows a wall shear stress of 0.0444 Pa with a heat transfer rate of 370.87 W, representing the lowest thermal performance among the tested configurations. The elliptical dimple fin increases the wall shear stress to 0.0571 Pa, along with an improved heat transfer rate of 445.56 W. This corresponds to approximately a 20% increase in heat transfer rate compared to the plain fin, due to enhanced flow disturbance and better fluid mixing.

The circular dimple fin exhibits the highest performance, with wall shear stress reaching 0.0616 Pa and heat transfer rate rising to 482.67 W. Compared to the plain fin, this represents an improvement of about 30% in heat transfer rate, driven by stronger vortex formation and more effective boundary layer disruption.

Overall, the results indicate that increasing turbulence intensity through dimpled surfaces significantly improves heat transfer performance. However, this enhancement is accompanied by higher wall shear stress, indicating increased flow resistance. Among all configurations, the circular dimple fin provides the best thermo-hydraulic performance, followed by the elliptical dimple fin, while the plain sinusoidal fin shows the lowest effectiveness.

3.7 OVERALL HEAT TRANSFER RATE

The overall heat transfer rate is analyzed at the maximum inlet velocity of 2.5 m/s for plain sinusoidal, elliptical dimpled, and circular dimpled fin configurations. At this higher velocity, all cases show improved thermal performance due to increased turbulence and enhanced fluid mixing. The plain sinusoidal fin provides a heat transfer rate of 370.87 W, while the elliptical dimple fin increases it to 445.57 W, representing an improvement of about 20% compared to the plain fin. The circular dimple fin achieves the highest value of 482.68 W, which is approximately 30% higher than the plain fin and about 8% greater than the elliptical configuration. Overall, the circular dimple fin demonstrates the best heat transfer performance, followed by the elliptical dimple fin, whereas the plain sinusoidal fin shows the lowest performance due to limited flow disturbance.

CONCLUSION

Over a Reynolds number range of 3503.29–8758.24, the thermal and hydraulic performance of sinusoidal fin heat sinks with elliptical and circular dimples was numerically investigated and compared with a plain sinusoidal fin. The following conclusions are drawn from the present study:

- The circular dimple fin shows the highest heat transfer performance, with an improvement of about 30% over the plain fin and around 8% over the elliptical dimple fin, due to stronger vortex generation and enhanced mixing.
- The elliptical dimple fin also enhances heat transfer, achieving approximately 20% higher performance than the plain fin, confirming improved thermal boundary layer disruption compared to the smooth surface.
- The Nusselt number and overall heat transfer rate increase with Reynolds number for all configurations, indicating improved convective heat transfer at higher velocities.
- The increase in thermal performance is mainly due to intensified turbulence, stronger secondary flows, and better fluid–surface interaction created by the dimple structures.
- However, pressure drop also increases with dimpled surfaces. Compared to the plain fin, the pressure drop rises by about 11% for the elliptical dimple and 13% for the circular dimple, due to higher flow resistance.
- The skin friction coefficient shows a similar trend, with the circular dimple fin exhibiting approximately 38% higher values than the plain fin, indicating increased wall shear stress.

- Overall, the circular dimple sinusoidal fin provides the best thermo-hydraulic performance, achieving the highest heat transfer rate while maintaining an acceptable pressure penalty compared to the other configurations.
- The study confirms that dimple-modified sinusoidal fins are effective for enhancing heat transfer in compact heat exchanger applications, where improved thermal performance is prioritized with manageable pressure losses.

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