

Effect of Helical Grooved Pin Fins in a Rectangular Channel on Heat Transfer Intensification and Friction Factor Using CFD Analysis

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ABSTRACT: The present study gives the effect of both cylindrical pin-fins and helical grooved pin-fins into the rectangular channel on heat transfer augmentation. Nusselt number and friction factor were experimentally investigated. Due to additionally induced turbulence in the flow field caused by resistance to fluid flow, the heat transfer is increased. This study focuses on different morphology of pin fin arrays with constant inter fin pitch and clearance ratio. The configurations include short pin fin elements. CFD analyses with Ansys fluent were carried out to determine the heat transfer characteristics and pressure drop. Experiment were conducted with constant heat flux and staggered configurations of pin fins as it gives more heat transfer compare to inline arrangement. The volume flow rate, pressure difference and temperature at the inlet and outlet were measured for the channel with different pin fin shapes at various Reynolds number (Re) in the range of 13500 to 42000 to obtain the friction factor. The results showed that the friction factor for all the fins decreased with the increase of Re and heat transfer increases with helical grooved pin fins as it increases turbulence.

KEYWORDS: CFD, Helically grooved pin fin, Turbulence.

I. INTRODUCTION

Fins with different geometries are used in a wide range of industrial applications such as cooling of electronic equipments, turbine blades and various heat exchanger devices for to increase performance of equipments. Pin fins provided on a heating surface increase the heat dissipation area and may cause turbulent mixing of flow. High heat flux is crucial for effective cooling of surfaces with relatively small surface area, especially in military, healthcare, electronic systems, aerospace and various industrial applications. Life of components becomes more reliable and durable with effective cooling. These pin fins located inside the flow channel promote turbulence to enhance the convective heat transfer rate, even if the pin fins are made of with low heat conductivity materials.

Fengming Wang et al. [1] studied the Investigation on flow and heat transfer characteristics in rectangular channel with drop-shaped pin fins. The result shows that the more streamlined drop-shaped pin fins were better at delaying or suppressing separation of the flow passing through them, which decreased the aerodynamic penalty compared to solid circular pin fins. The heat transfer enhancement of the drop-shaped pin fins was less than that of the circular pin fins but in terms of specific performance parameters, drop-shaped pin fins are a promising alternative configuration to circular pin fins.

M. Eren, S. Caliskan [2] studied Effect of grooved pin-fins in a rectangular channel on heat transfer augmentation and friction factor using Taguchi method. The optimum conditions were determined for pin-fin geometries. It was observed that Reynolds number affected the Nusselt number by the value of 60.15%. For performed experiment, the maximum heat transfer rate was obtained in in-line and staggered array at 15,146 Reynolds number, 3.446 b/e ratio. The most

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significant factor affecting the friction factor was the pin-fin geometry. The optimum level of pin-fin geometry which is the most important parameter for the friction factor was detected in the case of CGPFs as per experimental results. In-line array gives higher Nusselt number and thermal performance factor than the staggered array. The thermal performance factor (g) of the C-GPFs and T-GPFs was found to be the best and was about 2.27 and 2.81 respectively for performed experiment.

Hongxia Zhao [3] studied Pressure drop and friction factor of a rectangular channel with staggered mini pin fins of different shapes. The result shows that experiments were conducted on de-ionized water as working fluid, flowing across staggered mini pin fins of the same height and transverse spacing (inter fin distance) but with different pin density and different geometries of circular, elliptical, square, diamond and triangle, in a rectangular channel. The friction factor of five pin fin groups with different geometry drops as Re increases, and finally reaches a constant value. At large Re (>300), the influence on the eddy resistance by the pin fin geometry will lead to the difference in friction factors among different pin fin groups. The elliptical pin fin is the best streamlined, and hence the lowest flow resistance; the triangular pin fin is the least streamlined and hence the largest resistance as per performed experiment.

Mi-Ae Moon, Kwang-Yong Kim [4] studied Analysis and optimization of fan-shaped pin-fin in a rectangular cooling channel. The experimental results shows that a fan-shaped pin-fin was proposed and the heat transfer and friction loss in a rectangular channel with the pin-fin were evaluated and compared with those of the solid circular pin-fin using 3-D RANS analysis. Compared to the solid circular pin-fin, the fan-shaped pin-fin showed a better area-averaged Nusselt number for Reynolds numbers less than 100,000. The maximum enhancement of 22.8% was found at $Re = 80,000$. On the other hand, the pressure drop in the channel with the fan-shaped pin-fin was somewhat smaller than that with the circular pin fin for Reynolds numbers larger than 20,000.

Sahitiet al. [5] performed experiments regarding heat transfer and friction loss in a rectangular channel with various geometries of pin fin cross-sections; e.g., a NACA airfoil, dropform, lancet, ellipse, circle, and square. They reported that the elliptical pin fin arrays showed the maximum heat transfer rate with a similar pressure drop among the tested pin-fin geometries.

Chyu et al. [6] observed the heat transfer performance of circular, cubic, and diamond shaped pin fin arrays in a channel. Their results shown that diamond-shaped pin-fins showed the superlative heat transfer performance, but the cubic and diamond shaped pin-fins resulted in higher pressure losses than solid circular pin-fins.

Akyol and Bilen [7] conducted experiments to examine heat transfer and friction loss characteristics over a horizontal rectangular channel. Reynolds number varies between 3000 and 32,000. With the staggered array, improved heat transfer is obtained than inline arrangement due to the increase of turbulence in flow. It is also stressed that an augment in the Reynolds number causes increase in pressure drop and friction factor because of the blockage effect and it is high in staggered arrangement than in-line in a rectangular channel.

Jeng and Tzeng [8] performed an investigational study to investigate pressure drop and heat transfer characteristics of a square pin fin, and a circular fin in different arrangements i.e in-line and staggered. In-line square pin fin array has lesser pressure drop than that of in line circular pin fins. The staggered square pin fin configuration has the largest pressure drop across the channel.

Li et al. [9] studied elliptical pin fin arrays for Reynolds numbers varying from 900 to 9000. The major and minor axes of the elliptical pin fins were chosen such that the pin fin circumferences were equal to the corresponding circular pin fins. The experimental results showed that the heat transfer inside a channel embedded with elliptic pin fins was fairly higher than that with circular pin fins, whereas the resistance of the former was much lower than that of the latter arrangements.

Sahin et al. [10] investigated overall heat transfer coefficient, friction factor, inter fin pitch, clearance ratio and effects of various design parameters experimentally within a Reynolds number range of 13,500 to 42,000. For higher thermal

performance, it is found that lower clearance ratio (C/H), inter fin spacing ratio and Reynolds numbers are more efficient.

Nomenclature

A_f	wetted surface area (unit:m ²)	Re	Reynolds number
A_p	pin section area(unit:m ²)	T	thermodynamic temperature (unit: K)
B	rectangular duct width (unit: m)	u	velocity (unit: m/s)
C_p	heat capacity (unit: J/(kg.K))	V_f	fluid volume inside pin fins array region (unit:m ³)
D_h	hydraulic diameter (unit: m)		Greek letters
f	pressure loss coefficient	ν	kinematic viscosity(unit: Pa . s)
h	Convective heat transfer coefficient (unit: W/(m ² . K))	K	thermal conductivity(unit: W/(m.K))
H	rectangular duct height (unit: m)	ζ	specific performance parameter
L	length of pin fins array in x-direction (unit: m)		Subscripts
L_p	pin circumference length (unit: m)	ave	average
m	flow mass rate (unit: kg/s)	in	inlet
N	number of pin fins	max	maximum
Nu	Nusselt number	out	outlet
P	pressure (unit: Pa)	w	wall
q	density of heat flow rate (unit: W/m ²)		

II. EXPERIMENTAL SET UP

Fig.1 shows the experimental test set-up which comprises a fan blower, hydrodynamic developing section, test section, outlet and data collection system. Temperature and velocity measurements are carried out with thermocouples and anemometers respectively. The experimental set-up consisting of the main duct, test section and tested fins. The channel, constructed of wood of 20 mm thickness, had an internal cross-section of 250 mm width and 100 mm height (hydraulic diameter, $D_h = 142.86$ mm). The total length of the channel was 1000 mm. It was operated in suction mode and positioned horizontally. The test section had a cross-section of 250 mm x 100 mm square, and a length of 250 mm. The outside of the test section was coated with a layer of glass wool insulation to ensure well insulation against heat losses by conduction to the surrounding atmospheric air. The tested pin fins, made of the same material as the base plate because of the considerations of conductivity, good machinability and low cost [10]. The fins had a circular cross section of 15 mm diameter helically grooved. Fig. 2 Shows helical grooved pin fins with constant interfin pitch and clearance ratio having 50 mm height and 15 mm diameter. The Reynolds number range used in this experiment was 13,500 to 42,000, which was based on the hydraulic diameter of the channel over the test section (D_h) and the average velocity (u). The heating unit mainly consisted of an electrical heater, The dimensions of the electrical heater placed was 250 x 250 mm. The heater output had a power of 200 W at 38 V and a current of 5.2 A. The electrical-power input to the heater was controlled by a variac transformer to obtain a constant heat flux of 3200 W/m² along the test plate. The base plate with 250 mm long, 250 mm wide and 6 mm high is made of aluminium. The mean inlet velocities of the air flow entering to the test section were determined by an anemometer. The steady-state inlet and outlet temperature of the air stream was determined by RTD temperature sensors. The pressure drop across the test model was measured using differential pressure transmitters that can take measurements between 0 and 100 Pa.

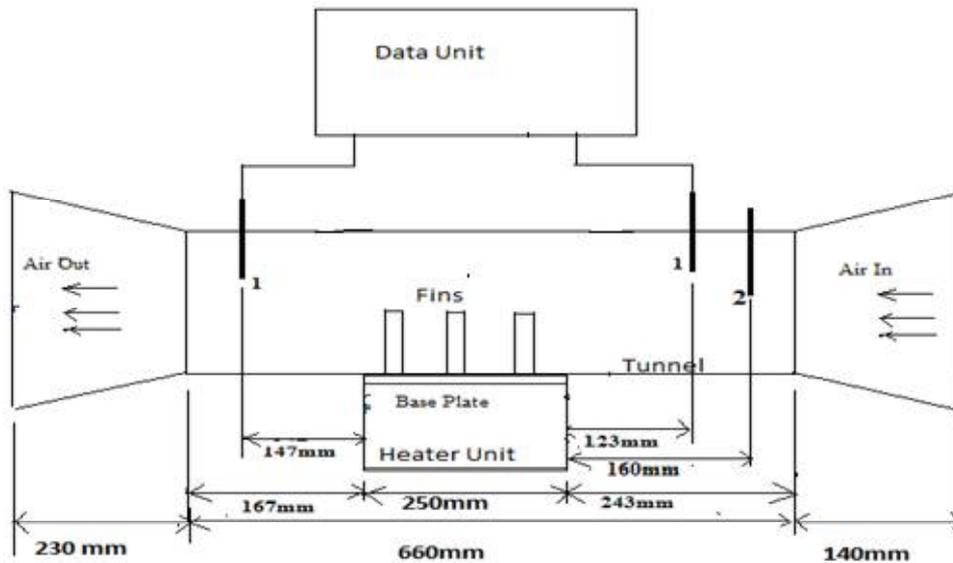


Fig. 1 Experimental set up

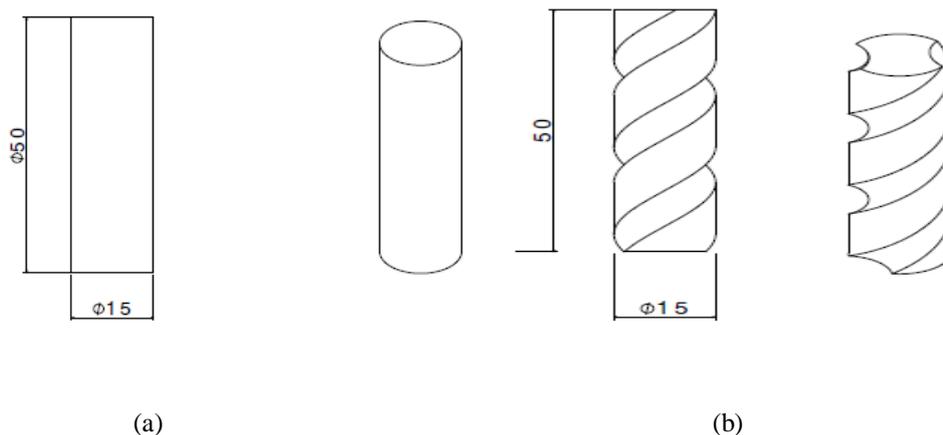


Fig. 2 Geometry of pin fin (a) Cylindrical and (b) Helical grooved

III. COMPUTATIONAL ANALYSIS

The fluid and pin fins were modelled as a 'conjugated' computational domain. The boundary conditions are summarised as: Inlet: The fluid flow temperature (T_{in}) was set to 300K and the inlet velocity (U_{in}) was determined by the corresponding Reynolds number based on hydraulic diameter; Walls: The no-slip condition was applied to all the solid walls. An adiabatic thermal condition was applied to the end walls, and a constant heat flux condition ($q = 3200 \text{ W/m}^2$) was applied to the heated test section; Pins: The pin was treated as a solid volume with a constant thermal conductivity.

Computational approach

Three-dimensional numerical simulation was performed using AnsysFluent-CFD software. First-order upwind was selected for the discreteness of the governing equations, and the standard k–e turbulence model was applied. The result are plotted with the help of line diagrams.

IV. PARAMETERS AND DATA TREATMENT

The hydraulic diameter is defined as the ratio of the open duct volume available for flow to the total wetted surface area inside the pin fin array region.

$$D_h = \frac{4V_f}{A_f} \quad (1)$$

where V_f is the entire fluid volume inside the pin fins array area. A_f is the wetted surface area.

$$V_f = BHL - N_p A_p H \quad (2)$$

$$A_f = 2(B+H)L + N_p (L_p H - 2A_p) \quad (3)$$

The Reynolds number is defined based on the hydraulic diameter considering pin fin as.

$$Re = \frac{U_{max} D_h}{\nu} \quad (4)$$

The local and average convective heat transfer coefficients are defined as

$$h = \frac{q}{(T_w - T_f)} \quad (5)$$

$$h_{av} = \frac{q}{(T_{w,av} - T_{f,av})} \quad (6)$$

The Exit temperature can be calculated as below using an energy balance across the ends of the test section.

$$T_{f,out} = T_{f,in} + \frac{(2 q_{Aheater})}{m C_p} \quad (7)$$

The local and average Nusselt numbers are distinct based on the hydraulic diameter with different pin fin geometries.

$$Nu = \frac{h D_h}{K} \quad (8)$$

$$Nu_{avg} = \frac{h_{avg} D_h}{K} \quad (9)$$

The total pressure loss coefficient or friction coefficient is defined as follows

$$f = \frac{P_{in} - P_{out}}{(1/2) \rho \mu^2 L_{in}} \quad (10)$$

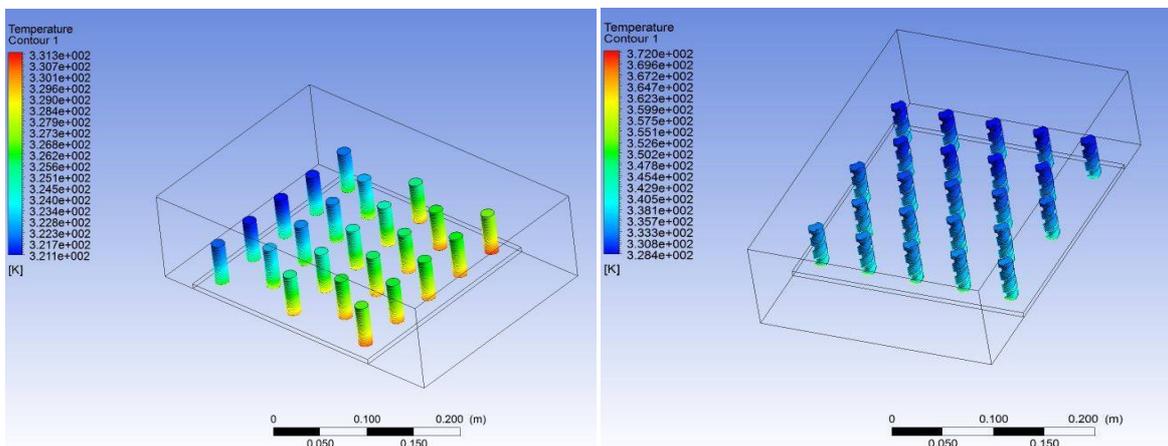
A specific performance parameter (ξ) that expect both heat transfer improvement and pressure loss is introduced as

$$\xi = \frac{f}{Nu_{avg}} \quad (11)$$

V. RESULTS AND ANALYSIS

Heat transfer characteristics

The local heat transfer was significantly enhanced in the area near the pin fin circumferences for all the cases. In the rear wakes of pin fins, the local convective heat transfer capacities also remained at greater levels. These features may be recognized to the actions of the horseshoe and wake vortices as per fluid flow characteristics. In addition, the windward blockage by helical grooved pin fins array was maximal for all cases; the flow was forced to follow a more twisted path around the pins, which increased the pressure drop. Figure 3 and Figure 4 shows the computational results of temperature distribution and local Nusselt number distributions across the pin fins at the heated area of base plate with different types of pin fin arrays at 42000 Reynolds number.



(a) (b)
Fig. 3 temperature distribution across the pin fins (a)cylindrical (b) helically grooved

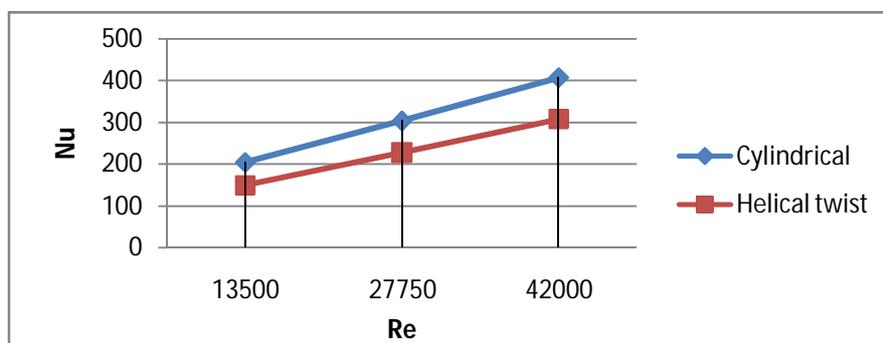


Fig. 4 local Nusselt number Vs Reynolds Number

Friction characteristics

The heat transfer behaviourunaccompanied does not provide anabsolutevaluation of heat exchanger performance but the increase in pressure drop, which is a determine of the energy required by the system, must be weighed against the improvement in heat transfer for each pin fin configuration of staggered arrangement. This optimization process is significant for improving energy efficiency and due to which it compares the contrast between heat exchanger gains

and losses. The specific friction loss was adopted to evaluate the optimization of the pin fin shapes compared to solid cylindrical and it increases because of turbulence. The specific friction loss of helical grooved pin fins was more than that of cylindrical solid pin fins, which indicates that the helical grooved pin fins is more advantageous than the others.

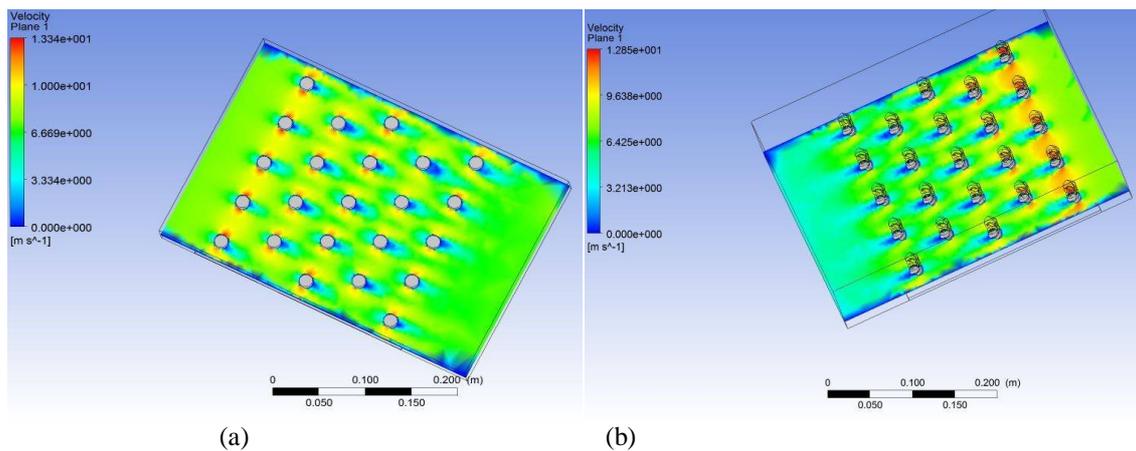


Fig. 5 local vorticity distribution diagram of pin fin (a) cylindrical (b) helically grooved

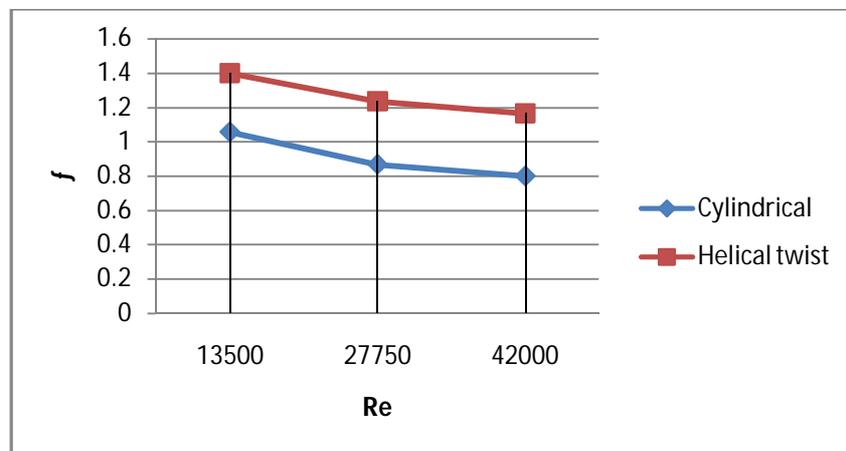


Fig. 6 Friction factor Vs Reynolds Number

Specific performance parameter Characteristics

The specific performance parameter that considers both heat transfer enhancement and pressure loss which proves that specific performance parameter characteristic of helically grooved fins is more than solid cylindrical pin fins as shown in Fig 7. Specific performance parameter Characteristics decreases with increase in Reynolds number.

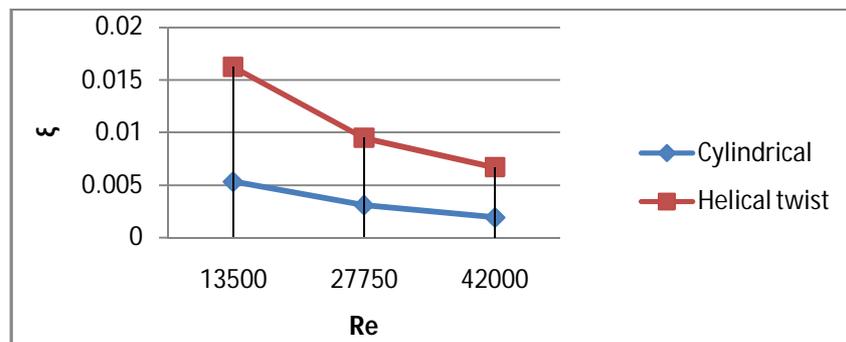


Fig. 7 Specific performance parameter Vs Reynolds Number

VI. CONCLUSIONS

In combined experimental and computational work proves that, the effects of geometrical fin parameters on heat transfer characteristics are investigated for turbulent air flow in rectangular cross section duct. A correlation for Nusselt number is developed and presented for steady state, turbulent, helical grooved fin array. The fluid flow phenomenon is further examined with CFD analysis in Ansys Fluent software to examine the effects of geometry of fin. The conclusions are

Summarized as:

1. Increase of the Reynolds number produces more turbulence and Nusselt number increases because of the turbulence effect.
2. For the specific performance, the specific friction loss in helical twisted pin fins was higher than that of circular pin fins, which indicates that the helical grooved pin is more advantageous than the others. Thus, helical grooved pin fin is a promising alternative configuration to cylindrical pin fins.
3. The heat transfer enhancement of helical grooved pin fin is stronger than that of circular pin fins. The reduction in average Nusselt number between the helical grooved and circular pins was much considerable.
4. For the specific performance, the specific friction loss in helical grooved pin fins was higher than that of circular pin fins, which indicates that the helical grooved pin is more advantageous than the others.
5. Friction factor considerably increases with helical grooved pin geometries.
6. Utilization of helical grooved fins increases heat dissipation rate, simultaneously reduction in fins weight, low weight means saving material of fin and which decreases the expenditure on the fin material and related equipment. Hence it may be utilized for cooling of electronic devices, air cooled I C engines and many other industrial applications of heat exchangers.
7. Computational results are in good conformity with the experimental findings. Computational fluid dynamics analysis is helpful for predicting heat transfer characteristics of the turbulent flow of helical grooved pin fins.

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