Frontiers in Heat and Mass Transfer (FHMT), 19, 8 (2022)
DOI: 10.5098/hmt.19.8

Global Digital Central
ISSN: 2151-8629

Frontiers in Heat and Mass Transfer

Available at www.ThermalFluidsCentral.org

THE EFFECTS OF NUSSELT, REYNOLDS NUMBER, AND PRESSURE DROP ON THE THERMAL PERFORMANCE OF PIERCED PIN FINS

Wadhah Hussein Al doori*

Engineering of Petroleum Process College, University of Tikrit, Iraq

ABSTRACT

Perforated fin forced convection heat transfer is the primary focus of this investigation. The purpose of this research is to see if perforated pin fins can help with heat transmission in the devices. Each pin's perforation diameter and number of holes are rigorously examined. The Nusselt numbers for perforated pins are 47 percent higher than those for solid pins, according to the study, and this number raises as the number of holes increases. The pressure drop is reduced by 19% when perforated pins are used instead of solid pins. Heat transmission in a round-holed pin fin was studied using forced convection in tests. Perforations in the shape of circles were among the options available. For the sake of this investigation, a number of perforations were made. For example, all of the fins and readings have one to four perforated holes. Perforated fins can promote heat transfer because of the improved Nusselt number, increased convective heat transfer coefficient, and decreased pressure gradient provided by this model is 40% smaller and up to 85% more effective at transferring heat. As a result, eddy currents are lessened.
Keywords: *Forced convection*, *Thermal performance, Pierced pin fins*

1. INTRODUCTION

System failure can come from dangerous overheating problems caused by failing to quickly disperse heat generated during operation of various engineering systems (T. K. Ibrahim et al. 2019). As a result, in order to maintain the system's ideal temperature, heat generated within the system must be expelled into the environment (Walunj and Palande 2014). Its great heat transfer efficiency makes pin fins ideal for cooling electronic components, heat exchangers, and gas turbine rotor blades. This is according to (Bahadure and Gosavi 2014). The expanded surface was utilized to boost heat transmission in a variety of applications. Using metal fins in a variety of methods, engineers were able to enhance the total surface area of heat transfer without sacrificing too much of the main surface area of the device. Fins increase a device's total weight and mass, as well as its production cost. So fin geometry optimization has taken center stage in recent years (Al Taha 2018).

Heat exchanging devices rely heavily on extended surfaces, such as fins. Fins are used to increase the surface area of a surface, hence increasing the rate at which heat is transferred to the surrounding fluid (Al Doori, 2019; Ridouane & Campo, 2008). There are a variety of fin shapes, from simple shapes like squares and cylindrical to more complex shapes like annular and tapering. Microprocessors and ICs, which generate a lot of heat, are well-suited to be used in heat sinks with fins, which have been discovered over the previous three decades to effectively dissipate the heat they generate (Al-Doori 2011). The ever-increasing need for cooling systems necessitates the development of new fin designs. The diameter of the perforated pin fin array was adjusted numerically by (Huang, Liu, and Ay 2015). The analysis focused on how to minimize temperature differences between the base plate and the surrounding environment. For example, when compared to solid pin-fin heat sinks, the base plate temperature of the designed ideal heat sinks dropped by 6.3–7.3%.

Plate fins with variously shaped geometric holes have been the subject of several investigations(Ibrahim et al., 2020; AlEssa et al., 2009). Forced convection heat transfer could be improved by using square holes instead of plate fins. Researchers found that rectangular and triangular holes transport heat at a greater rate per unit mass than ordinary plate fins in additional research. According to one theory (Prasad and Gupta 1998), cutting a portion of an irregularly-shaped fin could boost heat transfer per weight. Various radius cuts were shown to be more effective, and the fin efficiency was also proved to be superior. There has been a lot of research into perforated fin heat exchangers(Wadhah Hussein Aldoori & Ahmed, 2020; Sahin & Demir, 2008). A square-fin perforation was used to increase the Nusselt numbers. Perforations improve heat transfer efficiency by 1.4–2.6 percent (depending on the fin clearance ratio).Heat can move more quickly when a part of a pin fin is removed (Ganesh Murali and Katte 2008).

Additionally, rectangular fin arrays with varying numbers and sizes of perforations were subjected to thermal and hydraulic analyses by (Ehteshum et al. 2015). Their experiment was carried out on a 1088 mm^2 basis. From zero to two perforations, the diameter of the perforations ranged from 0 to 3 millimeters. For all fins, the Reynolds number (Re) increased along with heat transfer and pressure drop. In the studies, it was discovered that efficiency and effectiveness improved while thermal resistance and pressure drop were reduced with increasing numbers and sizes of perforations. (Karabacak and Yakar 2011) Heat exchangers with perforated fins were tested for hole placements, each round fin on a heating tube includes six millimeter-diameter holes. They investigated how heat transfer is impacted by pressure decreases and turbulence. In this experiment, six different angles can be used to determine the best angular placement. Their attention is drawn to the differences between perforated and imperforated finned heaters. When the critical value is exceeded, perforated finned sites have Nusselt values that are 13% higher than the imperforate state. The Re and Nu values above and below the critical value have a strong relationship. (W. H. A. Al Doori 2019), he found that it is possible to analyze the pressure drop and heat transfer characteristics of heat exchangers with annular-finned channels numerically. With more fins, heat transfer improved and was enhanced due to greater surface contact area. The largest heat transfer enhancement represented by Nusselt number was ascribed to the finned channel with twelve fins, followed by eight and four fins. Dense finned channels with

* Corresponding author. Email: wadhah.h.abdulrzzaq@tu.edu.iq

Frontiers in Heat and Mass Transfer (FHMT), 19, 8 (2022)
DOI: 10.5098/hmt.19.8

Global Digital Central
ISSN: 2151-8629

twelve longitudinal fins outperform finned channels with eight longitudinal fins and four longitudinal fins. Increasing the number of fins improved both Nusselt and Euler numbers. The current finding and known correlations agreed well with a maximum variance of 10% for the Nusselt number and 12% for the Euler number in all cases.

The overall performance of fins with suitably constructed lateral perforations was examined by (Ganorkar and Kriplani 2012). In the rectangular channel, many types of perforated fins are used. Other Reynolds numbers show the effect of perforated fins in a rectangular channel. Investigations were conducted to determine both Nusselt and heat transfer coefficient values. As for the diameter of the perforated holes, they were measured between 6 and 10 mm in the range of the Reynolds number. The ratio of ($Nu_{Perforated}/Nu_{Solid}$) fins increases as the Re number increases. The greater the number of holes, the greater the ratio ($Nu_{Perforated}$ /Nu_{Solid}). Increasing the diameter of the holes leads to an increase in this ratio. There is no discernible improvement in the ($Nu_{Perforated}/Nu_{Solid}$) as the number of holes increases.

The heat transfer and pressure drop properties of pin fins staggered in a rectangular duct were investigated (Baruah, Dewan, and Mahanta 2011). Pin fins with three elliptically perforated holes were also considered. The full performance, as measured by heat transfer per unit pressure drop at the heat exchanger, is depicted, with heat transfer and pressure drop parameters stated in computational terms. In terms of heat transmission and pressure drop, perforated elliptical pin fins surpass solid pin fins. Perforations in the solid elliptical fin are believed to minimize pressure drop by 12% on average while increasing heat transmission by 23%. "More holes in the elliptical pin fin could further increase this performance."

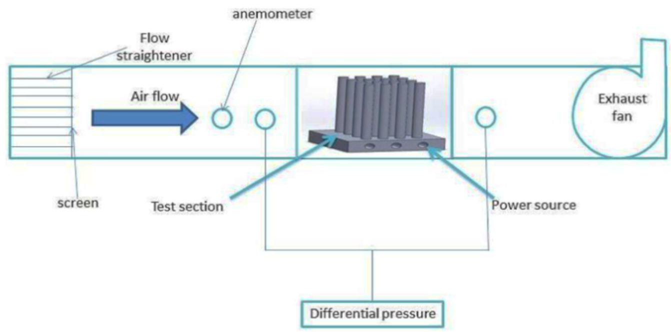
Pin fin array pressure loss and heat transfer were studied (Metzger, Fan, and Haley 1984) by examining the effect of pin fin design and array orientation. Heat transfer can be improved by using cylindrical pin fins with a staggered array direction, according to a study. Heat transfer rose by 20.0 percent and pressure loss increased by 100 percent when oblong pin fins were used instead of circular pin fins. Pin fin surface coefficients were likewise estimated to be around twice as high as those of the end wall. (Al-Taha 2018), It gets more difficult to flow when perforation sizes are reduced, and the average friction coefficient decreases. The maximum friction coefficient is found in a solid fin. In Nusselts, the average number of perforations decreases. In terms of heat transfer, the perforated fin has a higher average Nusselt number than the solid fin for each Reynolds number (11 percent of the nonperforated). According to research by (Hatem et al., 2020; Aldoori, 2021), perforated pin fins transfer more convective heat than solid pin fins. Thus, the perforated fins can be used in a wide range of situations where solid fins are commonly employed. Perforations increase the surface area, which raises the temperature differential. Forcing convection into the thermal performance of perforated pin fins has been studied (Tijani and Jaffri 2018). Because of this study, scientists now have a better grasp of how various characteristics, such as temperature distribution and pressure drop, affect the thermal performance of heat sinks. Channels of a rectangular shape were used to mimic a three-dimensional model. Perforated flat plates and pin fins have a 1 to 4% higher thermal efficiency than solid flat plates and pin fins.

The major purpose of this study is to explore and analyze the effects of Nusselt, Reynolds number, and pressure drop on the thermal performance of pierced pin fins based on the number of holes. There are also ways to improve the surface area and heat transfer coefficient of uniform pin fins when forced convection is used, and thermal boundary layer thickness. The fin body is cut with circular perforations. Perforated and augmented fins put on a board that generates heat and is subjected to forced convection are investigated in this study, as well as the ideal perforation that would result in the best fin performance. Furthermore, using one of the available CFD tools, the influence of a variety of parameters such as Nusselt, Reynolds number, and pressure drop, as well as perforation number, will be numerically calculated.

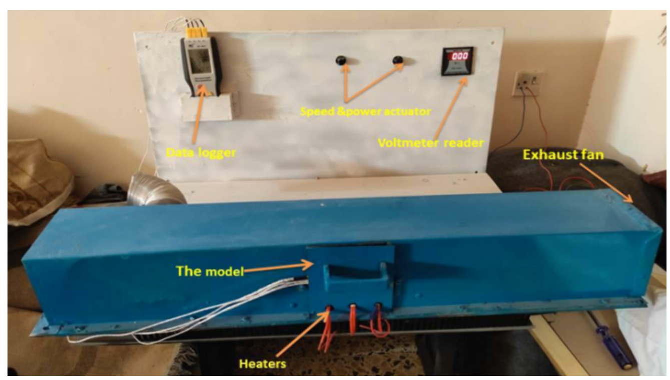
2. THE METHODOLOGY FOR THE RESEARCH

2.1 Measurements and test rig

The experiment model is a base with 16 fins made of 6061 Aluminum alloy, put within a sealed 1 m length of galvanized steel duct with a cross section of 0.2 m × 0.2 m with a variable speed fan (0–3 m/s) on one end and a pipe from which air is pulled on the other. An aluminum sheet plate was employed as a heat sink with a thickness of 20 mm and dimensions of 100×100 mm^2. Three side apertures (8 mm in diameter by 80 mm in length) at the bottom of the model base hold electric heaters, each with a capacity of 200 watts and controlled by a variable voltage source. In addition, the pressure difference across the test segment was measured using a digital manometer. The wind tunnel was built with all of the necessary parts, as shown in Figure 1.



a: The diagram of experimental work



b: a photograph of the equipment used

Figure 1: Parts of the device used in the experiment

To ensure the accuracy of the results, each experiment was repeated twice. A steady state was achieved after 2 hours of testing (Al-Doori, 2011). The highest accuracy limit for each experiment was obtained by determining standard deviations for each temperature point. The pressure drop and air velocity mean values for the cross-test rig were calculated for each experiment based on the data from each measurement.

2.2 The Investigation Procedures

It was necessary to measure the temperature fluctuation in the test rig produced by induced convection by inserting four thermocouples into a single fin as illustrated schematically in figure 1.Thermocouples were uniformly distributed across the fin's surface at four different locations. All trials reached steady state conditions within an hour and a half to two hours. The normal eccentricity of the two runs was calculated to estimate the exact boundary for each heat subject. Based on a single set of data from the thermistors installed on the fins, it appears that the temperature differential between the two base fins is minimal — less than 0.49 degrees Celsius. With thermal coupling, the average temperature of the inlet configuration was found to be 0.27 °C away from the free stream

DOI: 10.5098/hmt.19.8
ISSN: 2151-8629

temperature being measured. There is a unique identifier for every measurement.

3. ANALYZING HEAT TRANSFER

Using continuity, momentum, and energy equations, the pin-fin model for a heat sink is utilized to control the turbulent regime, incompressible flow, and heat transfer. The fluid flow forces convection to solve the continuity, energy, and momentum equations. The heat transfer analysis of solid and perforated pin fins is also based on the following assumptions: (i) constant heat conduction within the pin fin; (ii) no heat generation within the pin fin body; (iii) uniform ambient temperature; (iv) uniform temperature at the pin fin base; (v) uniform heat transfer coefficient across the entire pin fin surface; and (vi) no heat losses and/or radiation between the pin fin and the surrounding environment (Yousfi, Sahel, and Mellal 2019).

As voltage and current values changed, so did how much heat energy was sent to base Q_b, which is called a "heat sink."

$$Q_{heater} = Q_b = IV \tag{1}$$

Where, I and V are the current of electrical and voltage, respectively, provided by the meters as shown in figure 1.

The electrical power supplied was predicted to have a direct effect on the pace of heat transfer from the heating element to the cooling medium. As a result, the heat sink projected area (Q) can be used to indicate convection heat transfer, as follows (Bergman et al. 2011):

$$h = \frac{Q}{A(T_{f.av.} - T_\infty)} \tag{2}$$

Where, A is the surface area of the fin, $T_{f,av}$ is the average temperature of the fin surface, and T_∞ is the temperature of the free stream of air. k_f, film air's thermal conductivity.

The heat transfer area for several geometrical configurations employed in this investigation is described by equations (Al-Damook et al. 2016), as shown in Figure 2:

The solid pin, $$A_{sp} = wl + N(\pi DH) \tag{3}$$

The perforated pin, $$A_{pp} = wl + \pi N[(DH) + (ndD) - (nd/2)] \tag{4}$$

The solid fin $$A_{sf} = 2HNL + (N-1)SL \tag{5}$$

The empirical constants are as follows:

$$\Delta p = p_{in} - p_{out} \tag{6}$$

Where P_{In} and P_{Out} are the pressures at the channel's intake and exit, respectively. The following formula is used to compute the thermal resistance (R_{th}).

$$R_{th} = \frac{T_w - T_{in}}{Q_b} \tag{7}$$

T_w is the average wall temperature, T_{In} is the temperature of the air intake, and Q_b denotes heat flux applied to the base plate.

The Reynolds number (Re) is written like this:

$$Re = \frac{uD_h}{\upsilon} \tag{8}$$

where, u the mean air inlet velocity, D_h and ν are the hydraulic diameter and the kinematic viscosity respectively.

The Nusselt number is a dimensionless number that describes how well a fluid transmits heat. It's the proportion of convective to conductive heat transfer across a boundary. In a heat transfer situation, the Nusselt number can also be used to determine whether there is more convection or conduction. It's an important consideration when evaluating heat In a heat transfer situation. The average number of Nusselt can be written as (Peles et al. 2005):

$$\overline{Nu} = \frac{hL}{k_{fl}} \tag{9}$$

For a smooth surface, the Nusselt number is stated as:

$$Nu_{sf} = 0.228Re^{0.731}Pr^{1/3} \text{ (For } 4000 \le \text{Re} \le 15000) \tag{10}$$

$$Nu = 0.683Re^{0.466}Pr^{1/3} \quad \text{(For } 4000 \le \text{Re} \le 40000\text{)} \tag{11}$$

The heat sink's thermal efficiency (η) can be computed as follows:

$$\eta = \frac{\overline{Nu}}{\Delta p} \tag{12}$$

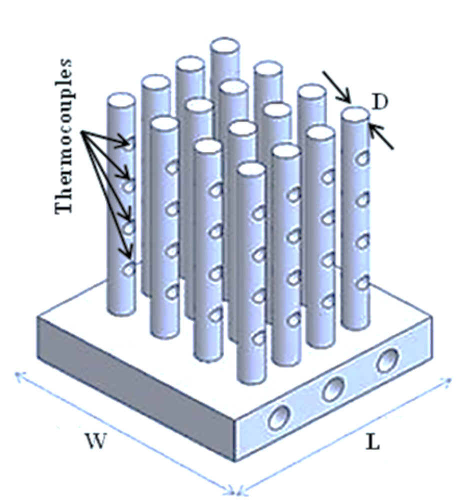
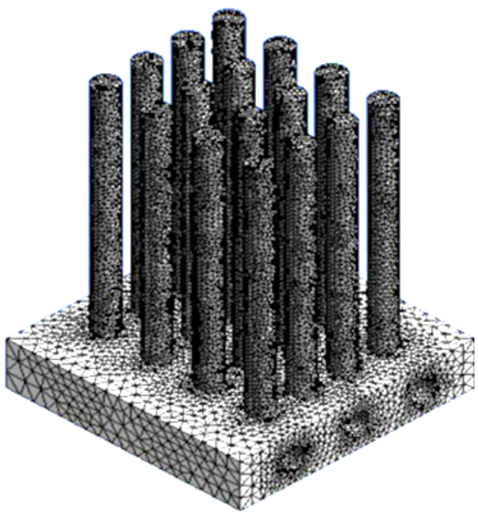


Fig. 2: A three-dimensional rendering showing the pin's dimensions and the thermocouples' placements along the pin

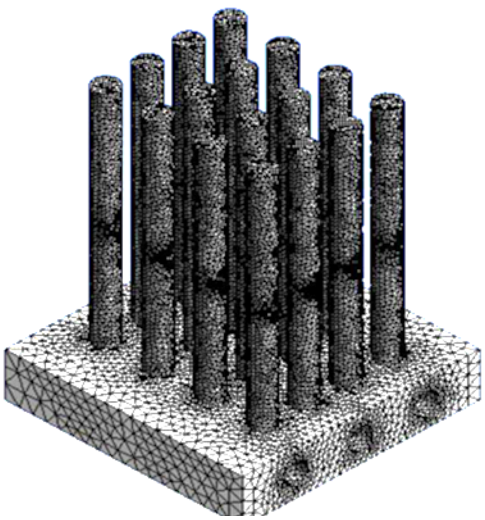
4. MODELLING OF COMPUTATIONAL AND GOVERNING EQUATIONS

Computational Fluid Dynamics (CFD) is the use of computational methods to predict fluid flow and heat transfer. In recent years, simulation methods have become more popular because they can help people predict how well they will do (Löhner 2008). Companies are facing a number of difficulties as a result of the difficulties in precisely estimating the performance of a new product. Only with insight offered by simulation techniques like CFD can the possibility of unanticipated performance difficulties be avoided (M. Ibrahim and Kasem 2021).

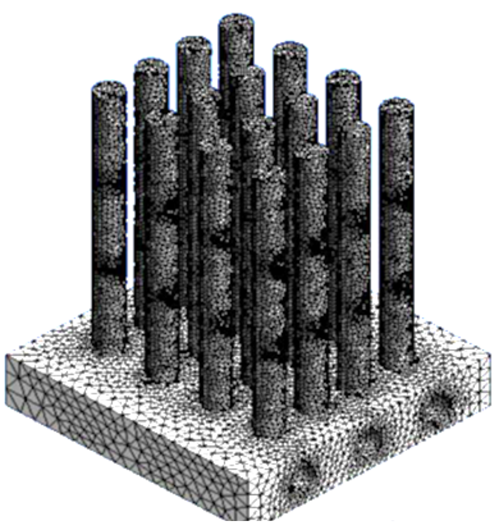
ANSYS was used to generate the model, with simulation approaches starting with the design of the base plate, fin, and tunnel shape. With the exception of the fin base, which is referred to as a base fin to transfer heat flux to the fin base, the mesh technique begins by designating the tunnel's inlet, outlet, and wall, while the fin body is referred to as an interface. The domain for these computations consists



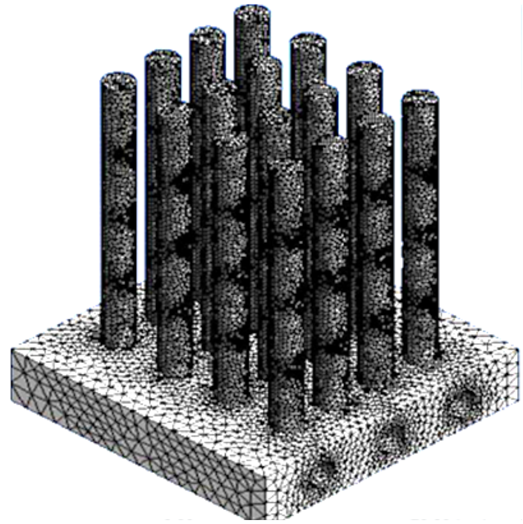
a. Non Perforated fin, N=0



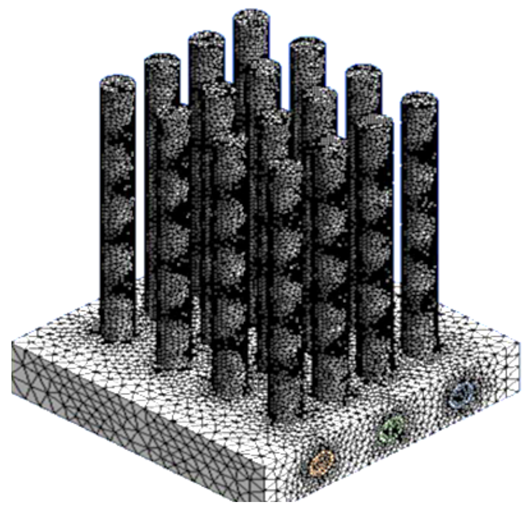
b. Perforated fin, N=1



c. Perforated fin, N=2



d. Perforated fin, N=3



e. Perforated fin, N=4

Fig. 3: Models of various fins mesh

of an entry region, an outlet level, and a popular online surface far enough away from the slab faces that the results are independent of border placements. The addition of a boundary layer above the base plane, on the other hand, can have a minor impact on the upstream plane. However, due to the irregular flow of the front edges of the boxes, such a space is required. Several tests were carried out using experimental data from the current investigation under the current settings. In order to achieve grid density independence in the response, grid point numbers in the three-dimensional region must be selected. Figure 3 depicts a typical grid with a high density near the plate. The flow field, pressure coefficient, and Nusselt number predictions were examined when the number of grids in all three dimensions was varied. Grid independence was found to be a significant issue. A lot of different grid distributions had to be looked at because there were a lot of steep corners and boundary layers.

4.1. Procedures and numerical modeling

These fins are expected to be 100 mm in length, 100 mm in breadth, and 10 mm in thickness, where D = 10 mm, L = 100 mm, and Dp = 4 mm are the pin fin parameters. All perforated fin shapes may be determined by keeping the fin volume constant. Results from this study show that speeds between 1 to 3 m/s.

DOI: 10.5098/hmt.19.8

ISSN: 2151-8629

The following assumptions were made when utilizing time-averaging methods to solve conservation equations: **(a)** Newtonian fluid is the initial type of fluid. **(b)** Stable-state flow (steady state). **(c)** The flow field is defined by a three-dimensional turbulent flow (turbulent regime) **(d)** The tempo of the flow (incompressible). **(f)** A steady temperature on the surface. The effect of buoyancy was disregarded. **(g)** A single-phase flow. **(h)** Thermodynamic radiation and heat dissipation are not considered. **(j)** Reliable fluid properties with low viscosity dissipation Heat is transported by forced convection. **(k)** The forces exerted by the body are minimal.

Formulas for figuring out the average flow statistics in a 3-D forced convection (Kumari, Sajja, and Kishore 2021):

Equation of Continuity:

$$\frac{\partial u_i}{\partial X_i} = 0 \tag{13}$$

Equation of Momentum:

$$\frac{\partial}{\partial X_j}\left(\rho u_i u_j - \tau_{ij}\right) = -\frac{\partial P}{\partial X_i} \tag{14}$$

Equation of Stress:

$$\tau_{ij} = 2\mu S_{ij} - \frac{2}{3}\mu_t \frac{\partial u_k}{\partial X_k}\delta_{ij} \tag{15}$$

Turbulent viscosity:

$$\mu_t = \frac{\rho C_\mu k^2}{\varepsilon} \tag{16}$$

The distortion rate tensor, S_{ij} , and the Kronecker delta function, δ_{ij} can be

Consequently: $$S_{ij} = \frac{1}{2}\left(\frac{\partial u_i}{\partial X_j} + \frac{\partial u_j}{\partial X_i}\right) \tag{17}$$

$$\delta_{ij} = \begin{cases} 1 & i = j \\ 0 & i \neq j \end{cases}$$

Equation of Energy:

$$\rho C_p \frac{\partial (u_i T)}{\partial X_i} = \frac{\partial}{\partial X_i}\left[\frac{\partial T}{\partial X_i}(\lambda + \lambda_t)\right] \tag{18}$$

Fourier's equation is used to calculate heat transfer in the fin:

$$\frac{\partial^2 T}{\partial X^2} = 0 \tag{19}$$

RNG k − ε turbulent model; The RNG turbulence model which is proposed by (Yakhot et al. 1992) is more responsive to the effects of rapid strain and streamlines curvature, flow separation, reattachment and recirculation than the standard k − ε model. RNG k − ε model is for low Reynolds numbers which have been widely employed for flows containing separation and recirculation. The previous researchers provided in-depth information on this model.

The turbulent kinetic energy equation is used to get the following equation:

$$\frac{\partial \rho u_i k}{\partial X_i} = \frac{\partial}{\partial X_i}\left(\alpha_k \mu_{eff}\frac{\partial k}{\partial X_i}\right) + \mu_t S^2 - \rho\varepsilon \tag{20}$$

And the rate of turbulent energy dissipation equation is:

$$\frac{\partial \rho u_i \varepsilon}{\partial X_i} = \frac{\partial}{\partial X_i}\left(\alpha_\varepsilon \mu_{eff}\frac{\partial \varepsilon}{\partial X_i}\right) + C_{1\varepsilon}\mu_t S^2 \frac{\varepsilon}{k} - C^*_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{21}$$

$$C^*_{2\varepsilon} = C_{2\varepsilon} + \frac{C_\mu \eta^3 (1-\eta/\eta_\circ)}{1+\beta\eta^3} \tag{22}$$

Where $\eta = \frac{Sk}{\varepsilon}$, $\eta_\circ = 4.38$, $\beta = 0.012$

The following are the other experimental constants in the equations above:

$\alpha_k = \alpha_\varepsilon = 1.393$, $C_{1\varepsilon} = 1.42$, $C_{2\varepsilon} = 1.68$, $C_\mu = 0.0845$

The QUICK method (El-Sayed et al. 2004) is used to discretize the convective terms. The flow rate was first calculated using identical speed solutions with a predicted blast wave, pressure and velocity were altered in order to fill up the continuity equation. It continues until the sum of residuals from the continuity equations in each cell equals or falls below the threshold value for that cell. Solving the discretized equations one line at a time, iteratively the energy equation is used to estimate the temperature field in the calculation zone after the flow field has been computed initially. The energy equation is subject to the convergence condition.

For this reason, the precision of CFD solutions can only be as good as the physical models on which they are based. Besides the models and parameters above, the CFD simulations in this work have more boundary conditions as shown in table 1:

Table 1: Inlet and Wall tunnel Boundary Conditions

Parameters	**Values**
Inlet air temperature (ºC)	25
Heat flux (W/m^2)	1000, 1200, 1450, 1950, 2000,2500,2700, 3200
Inlet velocity (m/s)	3
Reynolds number	14000, 26000, 41000, 52000 and 67000
Wall thickness	2 mm
Heat generations rate	0
Convective heat transfer coefficient	From experimental result
Free stream temperature	25 ºC
Wall motion	Stationary
Shear Boundary Conditions	No slip
Wall Roughness Constant	0.15

4.2. The grid-independence

The grid-independence of numerical data is looked into and offered for all the fins in Table 2. The complexity of the flow and fluid properties define grid-independent solutions. Meshes of varying sizes are used to save time and effort while still producing accurate numerical results. A grid-independence evaluation of the numerical findings was carried out for the pin fins under investigation here. Grid-independent solutions rely on intricate engineering configurations, fluid complexity, and fluid characteristics. The present work's computational model was conducted on an HP desktop computer equipped with an Intel(R) Core(TM) i7-7700 CPU @ 3.60 GHz and 16.0 GB of RAM to reduce runtime and improve numerical accuracy, both of which are important concerns in CFD simulation. In addition, we studied a range of mesh sizes, from coarser to finer. Due to the fact that there was no correlation between the number of grid points in the circumferential direction and the predicted heat transfer coefficient (as shown in Table 2), all of the model runs were done with a grid of 180 × 95 × 36 (Model 4) to get the accuracy needed while keeping the simulation time to no more than 25 minutes per run.

Table 2: Study of grid independence

Model	Grids	Coefficient of heat transfer (W/m^2.^{o}K)
1	125×70×24	259.8
2	145×80×28	274.3
3	165×88×32	296.1
4	180×95×36	305.4
5	200×140×48	307.6

5. RESULTS

The following Pin fin heat sinks with perforated and solid pin fins are being investigated in terms of numbers and tests. To ensure the grid's independence, numerical results are tested on five different grid sizes. Using current experimental data, numerical findings are tested to ensure that they are accurate. Heat transfer rates in all fin models are also examined. Perforated and non-perforated pin fin temperature, pressure, and velocity streamlines are also presented. Horizontal rectangular fins with varied patterns of holes have been studied for their ability to promote heat transfer. There will be a comparison of perforated fin behavior to solid pin fin activity in order to study the increased heat dissipation rate. ΔP, Nu, Re, heat transfer coefficient ŋ and are affected by perforation.

In order to highlight the role of perforation in this investigation, a set power output of 600 watts and a constant air speed of 3 m/s were employed to achieve all of the results. From a solid case to unilateral perforation and up to quadruple perforation, several scenarios were explored in terms of the model, with comparisons of what happens in each situation to the ordinary case (solid pin fin).

Figure 4 shows the effect of perforation on pressure drop. Because of the increased vortex currents created by the decreasing pressure, which leads to device failure. There is a decrease in "pushing power" with an increase in the number of perforations, as seen in the figure above. This means there is a pressure drop value difference between a solid pin fin and four perforations, which has the lowest pressure drop value.

Perforation has a significant impact on the Nusselt number, as shown in the following figure 5. The Nusselt number increases as Reynolds increases. The greater the number of perforations in a pin fin, the greater the heat transfer surface area.

Figure 6 shows how perforations affect heat transfer performance. The best performance was achieved with four holes, while the worst was achieved with a solid pin fin. It's because perforation (Nu) increases with perforation and pressure drop reduces with perforation increase; so, performance is directly proportional to (Nu) and inversely proportional to (ΔP).

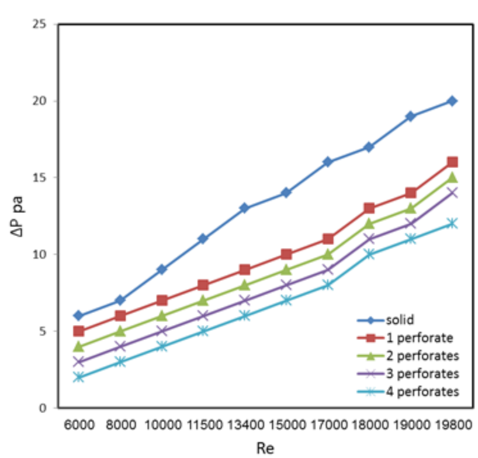


Fig. 4: The influence of perforations on pressure drop

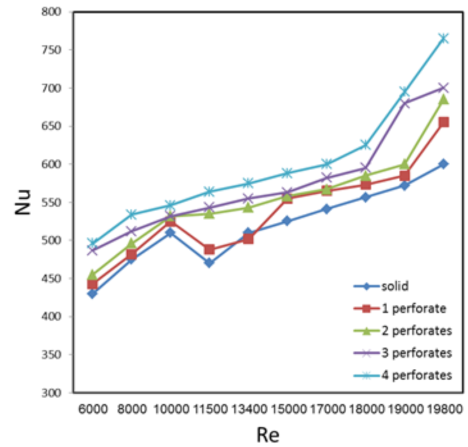


Fig. 5: The influence of perforations on Nusselt number

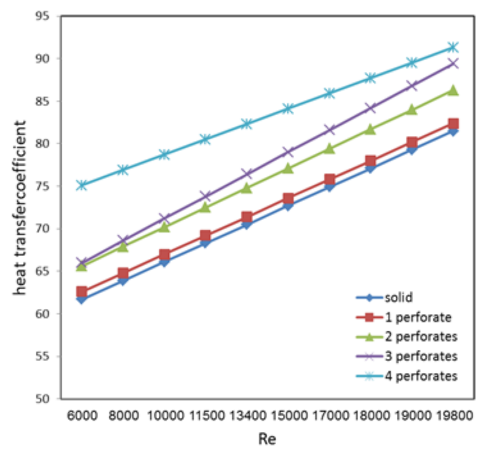


Fig. 6: perforation's significance on ŋ

Comparisons of current observed and computed Nu and ΔP versus Re are shown in Figure 7. The diamond symbol denotes measurements, while the solid line denotes computed values. In total, five alternative scenarios were examined: a solid, a hole, two holes, three holes, and four holes with a consistent diameter of perforation (4 mm). As can be seen, Nu and Re are expanding.

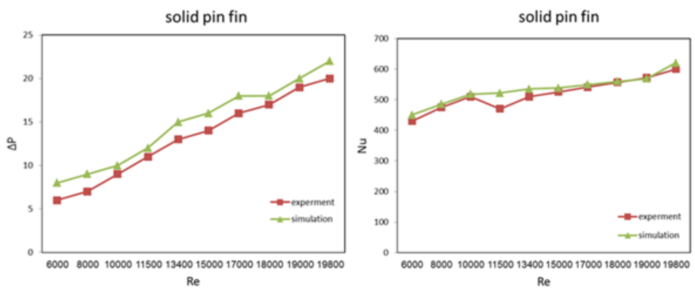
The simulated pressure decrease is greater than the actual experiment's. Unlike in the simulation, where the pins have complete contact with the top pin wall, inadequate contact may allow for bypass flow, resulting in a reduced pressure drop. Increased surface area may have facilitated faster heat transfer, which would have resulted in higher Nu at high Re in the experiment. It's possible that the turbulence model's shortcomings played an impact.

6. CONCLUSIONS

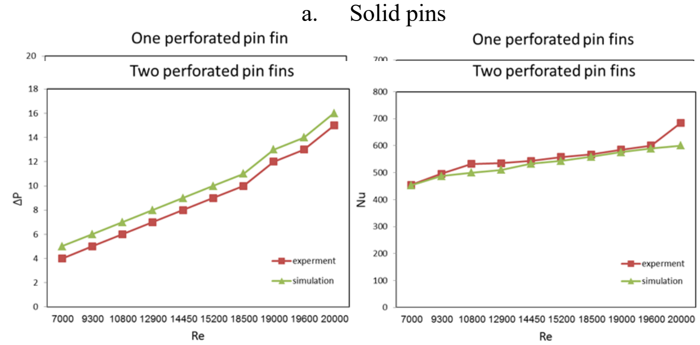
Forced convection heat transfer improves heat transfer in perforated pin fins. In this study, both analytical and theoretical methods were used to measure their heat transfer properties. Perforated pin fin heat sinks in various shapes and sizes are put to the test in terms of their efficiency in various applications. It was discovered via this research that:

- As the number of perforations increase, Nu rises. If the perforation is raised further, the thermal dissipation will be lower. Resulting in a decrease in vertical heat flow through pin fins and a change in wake shape due to the holes. The number of holes in the array of perforated pin fins should be taken into account when making it.
- ΔP across the heat sink decreases as the number of perforations grows. In every circumstance, perforated pin-fin arrays

outperform solid pin-fin arrays. For the same amount of heat transfer, perforated pin fins require less pumping power than solid pin fins.
- Optimum pin fin perforation can lead to the optimum overall system performance. Pin fins with four perforations perform best with the existing design.

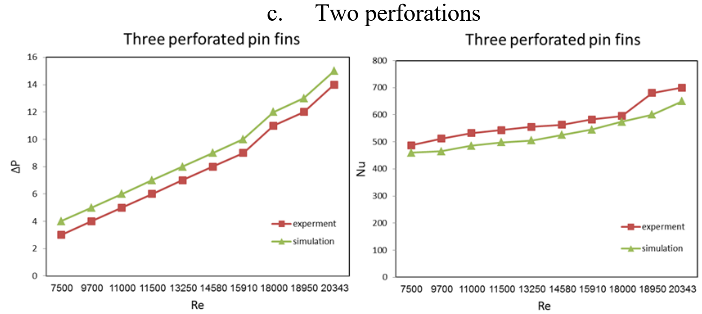


a. Solid pins

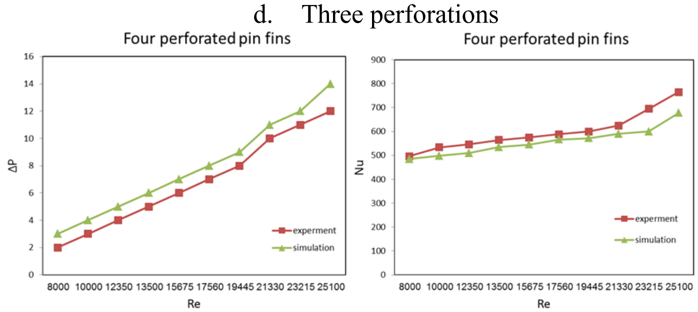


b. One perforation

c. Two perforations



d. Three perforations



e. Four perforations

Fig. 7: Comparisons of current observed and computed Nu and ΔP versus Re

ACKNOWLEDGEMENTS

The author would like to thank Tikrit University, especially the College of Petroleum Process Engineering.

NOMENCLATURE

A	surface area (m^2)
C_P	specific heat capacity at constant pressure (J/kg °C)
D	Diameter of the pin fins (m)
D_h	hydraulic diameter (m)
d	Perforation diameter of the pin fins heat sink
H	Height of the fins (m)
h	heat transfer coefficient (W/m^2 °C)
I	electrical current (A)
k	thermal conductivity (W/m °C)
L	Heat sink length (m)
N	Number of the pin fins
Nu	average Nusselt number
n	Number of the perforations on each pin
P	Pressure (Pa)
Q	supplied heat rate (W)
Re	Reynolds number
S	spacing between perforations (m)
T	Temperature (°C or K)
u,v,w	flow velocity components in x, y, z directions, respectively (m/s)
V	Voltage (V)
W	Heat sink width (m)
x, y, z	Cartesian coordinates

Greek Symbols

ε	Effectiveness
η	Efficiency
μ	Viscosity (kg/m s)
ρ	Density (kg/m^3)
τ_w	average wall shear stress (Pa)

Subscripts

a	air
av	average
b	fin base
f	fin
fl	film
in	inlet
out	outlet
pp	perforated pin
sf	solid fin
sp	solid pin
x, y	transverse and longitudinal directions, respectively
∞	free stream

REFERENCES

Al-Damook, Amer, N Kapur, J L Summers, and H M Thompson. 2016. "Computational Design and Optimisation of Pin Fin Heat Sinks with Rectangular Perforations." *Applied Thermal Engineering* 105: 691–703. https://doi.org/10.1016/j.applthermaleng.2016.03.070

Al-Doori, WHAR. 2011. "Enhancement of Natural Convection Heat Transfer from the Rectangular Fins by Circular Perforations." *International Journal of Automotive and Mechanical Engineering* 4: 428–436. https://doi.org/10.15282/ijame.4.2011.5.0035

Al-Taha, Wadhah Hussein Abdulrazzaq. 2018. "Effect of the Circular

Perforations on the Heat Transfer Enhancement by the Forced Convection from the Rectangular Fins." *Diyala Journal of Engineering Sciences* 11 (3): 62–70. https://doi.org/10.24237/djes.2018.11310

Aldoori, Wadhah H. 2021. "The Effect of Fin Height on Forced Convection Heat Transfer from Rectangular Fin Array." *Materials Today: Proceedings*. https://doi.org/10.1016/j.matpr.2021.07.191

Aldoori, Wadhah Hussein, and Ahmed Hasan Ahmed. 2020. "The Effect of Varying Tube Diameters on Enhancement Heat Transfer by Forced Convection through a Horizontal Tube." *Journal of Advanced Research in Fluid Mechanics and Thermal Sciences* 66 (2): 158–67. https://www.akademiabaru.com/index.php/archives/article/view/788

AlEssa, Abdullah H, Ayman M Maqableh, and Shatha Ammourah. 2009. "Enhancement of Natural Convection Heat Transfer from a Fin by Rectangular Perforations with Aspect Ratio of Two." *International Journal of Physical Sciences* 4 (10): 540–547. https://academicjournals.org/journal/IJPS/article-full-text-pdf/54E68EA19486

Bahadure, Mr Saurabh D, and Mr G D Gosavi. 2014. "Enhancement of Natural Convection Heat Transfer from Perforated Fin." *International Journal of Engineering Research* 3 (9): 531–535. https://doi.org/10.17950/ijer/v3s9/903

Baruah, Monoj, Anupam Dewan, and Pinakeswar Mahanta. 2011. "Performance of Elliptical Pin Fin Heat Exchanger with Three Elliptical Perforations." *CFD Letters* 3 (2): 65–73. https://www.akademiabaru.com/index.php/archives/article/view/287

Bergman, Theodore L, Theodore L Bergman, Frank P Incropera, David P Dewitt, and Adrienne S Lavine. 2011. *Fundamentals of Heat and Mass Transfer*. John Wiley & Sons.

Doori, Wadhah H Al. 2019. "Numerical Estimation of Pressure Drop and Heat Transfer Characteristics in Annular-finned Channel Heat Exchangers with Different Channel Configurations." *Heat Transfer—Asian Research* 48 (4): 1280–1291. https://doi.org/10.1002/htj.21432

Doori, Wadhah Hussein Abdulrazzaq Al. 2019. "Effect of Using Various Longitudinal Fin Number In Finned Channel Heat Exchangers On Heat Flow Characteristics." *Journal of Advanced Research in Fluid Mechanics and Thermal Sciences* 53 (1): 1–10. https://www.akademiabaru.com/submit/index.php/arfmts/article/view/2402

Ehteshum, Mehedi, Mohammad Ali, Md Quamrul Islam, and Muhsia Tabassum. 2015. "Thermal and Hydraulic Performance Analysis of Rectangular Fin Arrays with Perforation Size and Number." *Procedia Engineering* 105: 184–191. https://doi.org/10.1016/j.proeng.2015.05.054

El-Sayed, Saad A, Shamloul M Mohamed, Ahmed A Abdel-latif, and E Abouda Abdel-hamid. 2004. "Experimental Study of Heat Transfer and Fluid Flow in Longitudinal Rectangular-Fin Array Located in Different Orientations in Fluid Flow." *Experimental Thermal and Fluid Science* 29 (1): 113–128. https://doi.org/10.1016/j.expthermflusci.2004.02.006

Ganesh Murali, J, and Subrahmanya S Katte. 2008. "Experimental Investigation of Heat Transfer Enhancement in Radiating Pin Fin." *JJMIE* 2 (3). https://citeseerx.ist.psu.edu/viewdoc/download?doi=10.1.1.586.7878&rep=rep1&type=pdf

Ganorkar, A B, and V M Kriplani. 2012. "Experimental Study of Heat Transfer Rate by Using Lateral Perforated Fins in a Rectangular Channel." *MIT International Journal of Mechanical Engineering* 2 (2): 91–96. https://web.archive.org/web/20180422004549id_/http://mitpublications.org/yellow_images/1361593270_logo_4.pdf

Hatem, Mohanad, Hassan Abdellatif, and Wadhah Hussein. 2020. "Enhancement of Perforated Pin-Fins Heat Sink under Forced Convection." *International Research Journal of Engineering and Technology (IRJET)* 7 (10). https://www.irjet.net/archives/V7/i10/IRJET-V7I10245.pdf

Huang, Cheng-Hung, Yu-Chen Liu, and Herchang Ay. 2015. "The Design of Optimum Perforation Diameters for Pin Fin Array for Heat Transfer Enhancement." *International Journal of Heat and Mass Transfer* 84: 752–765. https://doi.org/10.1016/j.ijheatmasstransfer.2014.12.065

Ibrahim, Marwa, and Mohamed Kasem. 2021. "Numerical Thermal Study of Heat Transfer Enhancement in Laminar-turbulent Transition Flow Through Absorber Pipe of Parabolic Solar Trough Collector System." *Frontiers in Heat and Mass Transfer (FHMT)* 17. https://doi.org/10.5098/hmt.17.20

Ibrahim, Thamir K, Ahmed T Al-Sammarraie, Manar S M Al-Jethelah, Wadhah H Al-Doori, Mohammad Reza Salimpour, and Hai Tao. 2020. "The Impact of Square Shape Perforations on the Enhanced Heat Transfer from Fins: Experimental and Numerical Study." *International Journal of Thermal Sciences* 149: 106144. https://doi.org/10.1016/j.ijthermalsci.2019.106144

Ibrahim, Thamir K, Ahmed T Al-Sammarraie, Wadhah H Al-Taha, Mohammad Reza Salimpour, Manar Al-Jethelah, Ahmed N Abdalla, and Hai Tao. 2019. "Experimental and Numerical Investigation of Heat Transfer Augmentation in Heat Sinks Using Perforation Technique." *Applied Thermal Engineering* 160: 113974. https://doi.org/10.1016/j.applthermaleng.2019.113974

Karabacak, Rasim, and Gülay Yakar. 2011. "Forced Convection Heat Transfer and Pressure Drop for a Horizontal Cylinder with Vertically Attached Imperforate and Perforated Circular Fins." *Energy Conversion and Management* 52 (8–9): 2785–2793. https://doi.org/10.1016/j.enconman.2011.02.017

Kumari, D, Venkata Sajja, and P M Kishore. 2021. "Numerical Solution of the Effects of Heat and Mass Transfer on Unsteady MHD Free Convection Flow Past an Infinite Vertical Plate." *Frontiers in Heat and Mass Transfer (FHMT)* 16. https://doi.org/10.5098/hmt.16.24

Löhner, Rainald. 2008. *Applied Computational Fluid Dynamics Techniques: An Introduction Based on Finite Element Methods*. John Wiley & Sons.

Metzger, D E, C S Fan, and S W Haley. 1984. "Effects of Pin Shape and Array Orientation on Heat Transfer and Pressure Loss in Pin Fin Arrays." https://doi.org/10.1115/1.3239545

Peles, Yoav, Ali Koşar, Chandan Mishra, Chih-Jung Kuo, and Brandon Schneider. 2005. "Forced Convective Heat Transfer across a Pin Fin Micro Heat Sink." *International Journal of Heat and Mass Transfer* 48 (17): 3615–3627. https://doi.org/10.1016/j.ijheatmasstransfer.2005.03.017

Prasad, BVSSS, and AVSSKS Gupta. 1998. "Note on the Performance of an Optimal Straight Rectangular Fin with a Semicircular Cut at the Tip." *Heat Transfer Engineering* 19 (1): 53–58. https://doi.org/10.1080/01457639808939914

Ridouane, El Hassan, and Antonio Campo. 2008. "Heat Transfer Enhancement of Air Flowing across Grooved Channels: Joint Effects of Channel Height and Groove Depth." *Journal of Heat Transfer* 130 (2). https://doi.org/10.1115/1.2790022