# EXPERIMENTAL ANALYSIS AND VALIDATION OF STAGERRED, CIRCULAR, AND DROPPED PIN FINS IN 1- DIRECTIONAL AND 2- DIRECTIONAL FORCED CONVECTION

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*Abstract :* This experimental study investigates the variation and improvement of heat transfer in staggered pin fin heat sinks with both drop and circular shapes under a constant heat flux. A comparison was carried out using aluminum, stainless steel, and iron as materials for drop and circular-shaped pin fins. The pins, each with a diameter of 10 mm, were manufactured using Aluminum-Magnesium-Silicon alloy, stainless steel, and ferrous metals. The base plate was made of the same material, and the fins were fixed to it using threaded connections. Multiple tests were performed on each material to determine which offered better heat dissipation. Further, tests were conducted to evaluate which fin shape—circular or drop—was more efficient for maximizing thermal performance. Additional trials were also executed to determine whether one-directional or two-directional forced convection provided better heat transfer, helping to identify the most effective airflow method. The complete evaluation comparing drop and circular pin fins was carried out in this experimental work.

**Index Terms - Aluminum, Circular, CFD, Dropped, Iron Stainless-Steel**.

## IINTRODUCTION

Extended surfaces are utilized in heat exchangers to improve the rate of heat transfer between the surface and the surrounding fluid. Various fin types are used, including cylindrical, rectangular, annular, drop-shaped, tapered, or pin fins, which are typically mounted on either rectangular or cylindrical bases. Among these, pin fins are widely adopted as extended surface heat exchangers. A pin fin is generally a cylindrical or custom-shaped element extending perpendicularly from a surface, with cooling fluid flowing across it in a cross-flow pattern. Pin fins are defined by several parameters, including their shape, height, diameter, and the arrangement—either staggered or inline—relative to the flow direction. Fins serve a crucial purpose in improving thermal performance; by modifying their geometry, heat transfer can be maximized based on specific needs. These fins can be positioned both inside and outside heat exchanger units to enhance overall thermal efficiency.

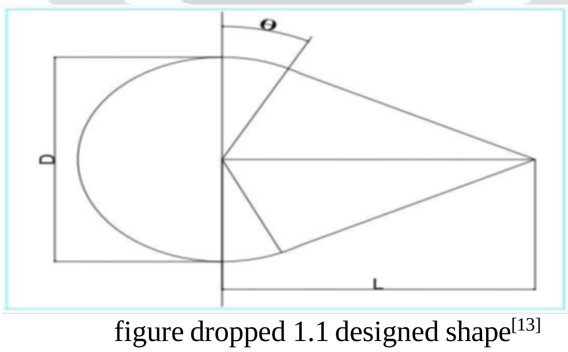



figure dropped 1.1 designed shape[13]

When the pin shape is modified from circular to drop-like, the wetted surface area exposed to the fluid flow increases, enhancing the overall heat transfer rate. The tail section in the drop design helps delay flow separation, which reduces both frictional drag and power consumption. The overlapping of fins generates a nozzle-like effect due to the tapered shape of the tail, accelerating the flow and directing it toward the next row of fins. This combination of fin interaction and speeding airflow, along with merging flow paths, results in higher turbulence—further improving heat transfer. However, carrying out experimental testing for various fin shapes is highly costly and time-consuming, as it involves expensive tools and fabrication processes. [13]

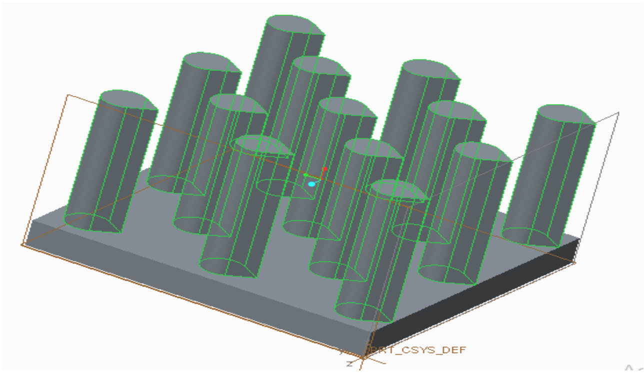



Figure 1.1 dropped shaped fins by using PTC Creo 3.0

## 2. EXPERIMENTAL SET-UP

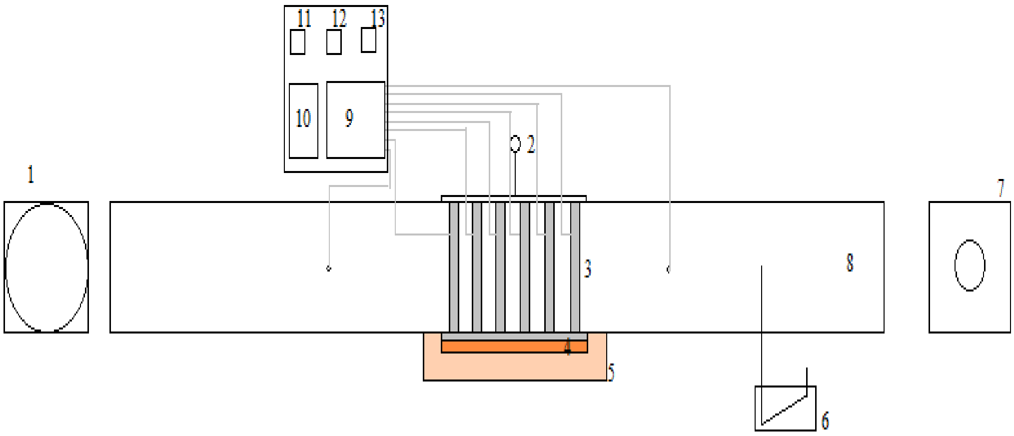



Figure 2.1 Experimental Set-up

| | | | |
|---|---|---|---|
| 1. | Blower | 8. | Duct |
| 2. | Small capacity blower | 9. | Dimmer stat |
| 3. | Pin fin array | 10. | Blower controller |
| 4. | Heater | 11. | Temperature indicator |
| 5. | Insulating box | 12. | Voltmeter |
| 6. | Manometer | 13. | Ammeter |
| 7. | Orifice meter | | |

The entrance length of duct is 595 mm, test section 250 mm and downstream exit is 595 mm. The internal height and width of channel is 110mm and 160mm respectively. The front side of duct is constructed with acrylic material sheet for visibility of temperature sensor and all other side of duct is constructed with MS material sheet. The joints of MS sheet duct are sealed with rubber gasket.

There are two blowers named secondary and primary blower is used with controllable speed. Primary blower is used to blow the air in 'x' direction and it is act as main blower for assembly. The secondary blower is fitted perpendicular direction to the primary blower exactly on test plate section for blowing air in 'y' direction on test plate.

The staggered pin fin array is installed in test section. All fins are fitted on base plate by thread fitting and gap between threads is sealed by thermal grease. Three K-type thermocouple (Pencil shape) are installed on entrance length and exit length of duct to measure the average inlet and outlet air temperature respectively. Also, Six K-type thermocouple are installed on test plate to measure the temperature at various locations. A 1200-Watt plate type heater is mounted below the base plate. The gap between plate and heater is sealed by thermal grease. The inlet and outlet temperature are measured by six K-type thermocouples. [15]

## 3. PROBLEM IDENTIFICATION

1. The heat transfer rate is less in 1-directional heat flow model as well as in natural convection.
2. Due to circular shape of pin fins more materials are required which result in more cost.
3. The surface area of circular pin fin is less comparing to dropped shape of pin fin.
4. Heat Transfer rate is not uniform in 1-directional flow of fluid.

## 4. LITERATURE REVIEWS

[1] Manoj et al. conducted an experimental study for examine heat transfer behavior and enhancement in cylindrical pin fin heat sink subjected to constant heat flux. The fins was made from an Aluminum, Magnesium-Silicon alloy with 12 mm diameter, were mounted on a base plate of the few material using threading. From three tests determined suitable fin height, followed by two more for identifying the ideal spanwise and streamwise spacing (Sx and Sy). A final test on the selected configuration was performed under different air mass flow rates. The findings revealed that two-directional airflow significantly improved heat transfer uniformity and rate compared to one-directional airflow.

[2] R. Muthukumarn et al. carried out experimental research to analyze heat transfer and pressure drop in a horizontal wind tunnel with cylindrical, grooved, and perforated fins fixed on a base. Reynolds numbers from 2,000 to 25,000 and a zero clearance ratio (C/H = 0.0) were tested. Inline pin configurations with a constant spanwise pitch (Sx/d = 1.2) and varying streamwise pitch (Sy/d = 1.2, 2.4, 3.6) were used. Nusselt number and pressure drop were performance indicators. Grooved cylindrical fins yielded the highest heat transfer enhancement, matching prior findings.

[3] Manikandan C. et al. presented a CFD-based simulation study using ANSYS 15.0 to evaluate temperature and pressure drop in a perforated drop-shaped pin fin heat sink. Heat transfer increased by converting circular fins into staggered perforated drop shapes under constant heat flux. These drop fins, having the same cross-sectional area, caused delayed airflow through the fins, leading to greater thermal dissipation. Transient flow conditions were applied to ensure better simulation accuracy. The drop- shaped fins were found to outperform circular and rectangular fins in thermal performance.

[4] M.M. Sahu et al. developed a forced convection-based experimental model for pin fin heat sinks, validated by CFD. Circular pin fins in inline arrangements were assessed using empirical data for thermal resistance and heat transfer. Using aluminum fins (thermal conductivity 209 W/mK), a 6×8 staggered array with 48 pins was tested at an airflow velocity of 2.5 m/s. The base was heated with 40 W constant load. Pin height was fixed at 60 mm with a 6 mm diameter, while transverse pitch was varied (8 mm and 12 mm). CFD results supported experimental trends.

[5] Lei Chai et al. designed three micro pin fin heat sinks with drop shapes have different tail angles, optimized from conventional circular designs. The systems, tested experimentally using deionized water, showed that the 60° tail angle had the lowest flow resistance. The aerodynamic profile of the drop fins aided in smoothing flow distribution and delaying the laminar-to- turbulent transition. The performance varied with Reynolds number (200–1,000), and although a 30° tail angle improved heat transfer at the fin tail, flow disruption from downstream pins increased resistance under equal pumping power.

## 5. MATERIAL SELECTION

The selected materials for experimentation have thermo physical properties which are given in tabular form

Table No.5.1 thermo physical properties of materials

| Sr No | Properties | Aluminium | Stainless-steel | Iron |
|---|---|---|---|---|
| 1 | Thermal conductivity (W/mk) | 237 | 15.1 | 80.2 |
| 2 | Specific heat, $C_p$ (J/KgK) | 903 | 480 | 447 |
| 3 | Density, $\rho$ ($Kg/m^3$) | 2702 | 8055 | 7870 |
| 4 | Melting points (K) | 933 | 1670 | 1810 |

## 6. DESIGN AND ANALYSIS

The design modeled pin fins for experimentation are shown in figure.4.1 In this experimentation three materials are chosen i.e. stainless-steel ,Aluminum and Iron with dropped and circular shaped geometries respectively

| | | |
|---|---|---|
|  |  |  |
| Aluminium circular | Aluminium Dropped | Stainless-steel circular |
|  |  |  |
| Stainless-Steel Dropped | Iron circular | Iron dropped shaped |

Figure 4.1 different geometries of fins

## 7. DATA REDUCTIONS

### 7.1 Heat transfer rate (Q) calculation

$Q=h^{*}A_{s}^{*}(\Delta T)$ (1)

Where,

Q - Heat transfer rate, KJ/s or watt

$h_s$ – Convective heat transfer coefficient, W/m$^2$ $^0$c

A - Area m$^2$

$\Delta T$ - Temperature difference, K

$C_p$ -Specific heat, KJ/Kg * K

m - Mass flow rate, Kg/s

### 7.2 Discharge of air through orifice ($Q_a$) Calculation

$$Q_a= C_d * \frac{\pi}{4} * d^2 * \sqrt{2 * g * \frac{H}{1000} * \frac{\rho_w}{\rho_a}} \quad (2)$$

Where,

$C_d$- Coefficient of discharge = 0.64

d - Diameter of orifice = 60mm = 0.06 m

$\rho_w$- Density of manometric liquid = 810 Kg/m$^3$

$\rho_a$- Density of manometric air = 1.225 Kg/m$^3$

$H_m$- Manometric reading=5mm

$$Q_a= 0.64 * \frac{\pi}{4} * 0.06^2 * \sqrt{2 * 9.81 * \frac{5}{1000} * \frac{810}{1.225}}$$

= 0.0145 Kg/m$^3$

### 7.3 Mass flow rate (m) calculation

$$m=\frac{Q_a}{V_a} \quad (3)$$

$V_a$ – Volume flow rate, Kg/m$^3$

From psychometric chart, At DBT=37$^0$C and WBT=29$^0$C the values of volume flow rate $V_a$=0.923 m$^3$/s

$m= \frac{0.0145}{0.923} = 0.0157$ kg/s

### 7.4 Coefficient of convective heat transfer (h) calculation

$h= m*C_p\{(T_{out}-T_{in})/A_s*[T_b-(T_{out}-T_{in})/2]\}$ (4)

Where,

| | |
|---|---|
| m - Mass flow rate = 0.0157 Kg/s $C_p$ - Specific heat = | 1.005 KJ/Kg0K |

$T_{in}$ = Inlet temperature $T_{out}$ = Outlet temperature

$T_b$= Average temperature of base plate $^0$C

### 7.5 Surface Area ($A_s$) Calculation

$A_s$= (b*L) + (π*r²*h) * N (For circular shape pin fin) (5)

= (0.15*0.25) + (π*0.01*0.07) * 20

= 0.0376 m²

$A_s$=(b*L) + ((π*d*H/2) +(L*b*H/2)) *N (For drop shaped pin fin)

= (015*0.25) + ((π*0.1*0.07/2) +(0.010*0.010*0.07/2))*20

= 0.0592 m²

Where,

b= Width of base plate=150mm=0.15m d= diameter of pin fin = 10mm=0.1m

L= Length of base plate = 250mm=0.25m

H=Height of pin fin =70mm=0.07m

N= Number of pin fins = 20

From Equation number (4)

h=85.641 W/m²k

From Equation number (1)

Q=85.41*0.0376*(29.8-29.3)

=1.61W

### 7.6 Nusselt Number Calculation (Nu)

Nu= h.L/K (6)

= 85.41*0.1875/0.025

= 642.30

Where,

h= Convective heat transfer coefficient (W/m$^{20}$c) L= Characteristic length (m)

= For Rectangular Duct

L=4(a*b)/2(a+b)=0.1875m K= Thermal conductivity (W/m$^0$c)=0.025(W/m$^0$c)

## 8 RESULT AND DISCUSSION

### 8.1 Result Table

### 8.1.1 for Aluminium

| Sr No | Material | Aluminium | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | Type of flow | 1D | | | | 2D | | | |
| | Shape of fins | Circular | | Dropped | | Circular | | Dropped | |
| | Dimmer stat | 50W | 100W | 50W | 100W | 50W | 100W | 50W | 100W |
| 1 | Actuacl heat transfer rate, Q | 1.61 | 2.25 | 4.86 | 5.89 | 2.18 | 3.34 | 10.43 | 11.36 |
| 2 | Convective heat transfer coefficient, h | 85.641 | 119.897 | 97.908 | 126.918 | 96.840 | 148.108 | 117.453 | 159.917 |
| 3 | Nusselt Number, Nu | 642.30 | 899.227 | 734.31 | 951.885 | 726.3 | 1110.81 | 880.897 | 1199.37 |

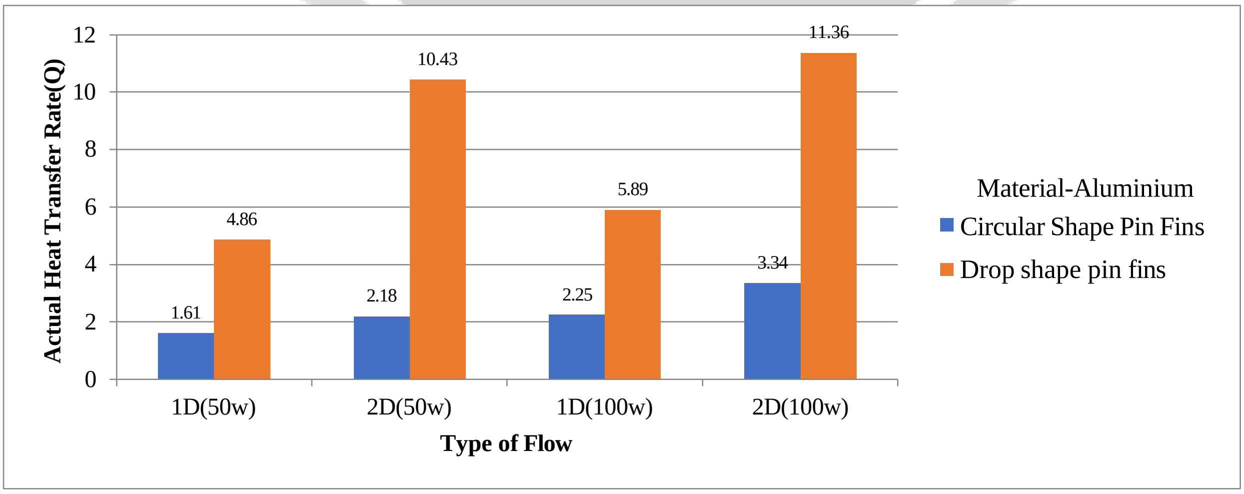
### 8.1.2 for Stainless Steel

| Sr No | Material | Stainless Steel | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | Type of flow | 1D | | | | 2D | | | |
| | Shape of fins | Circular | | Dropped | | Circular | | Dropped | |
| | Dimmer stat | 50W | 100W | 50W | 100W | 50W | 100W | 50W | 100W |
| 1 | Actual heat transfer rate, Q | 1.274 | 1.485 | 1.692 | 1.901 | 1.579 | 2.146 | 2.103 | 2.277 |
| 2 | Convective heat transfer coefficient, h | 36.490 | 64.560 | 39.004 | 76.151 | 51.384 | 96.199 | 59.228 | 104.910 |
| 3 | Nusselt Number, Nu | 273.67 | 484.2 | 292.53 | 571.13 | 385.38 | 721.49 | 444.21 | 786.825 |

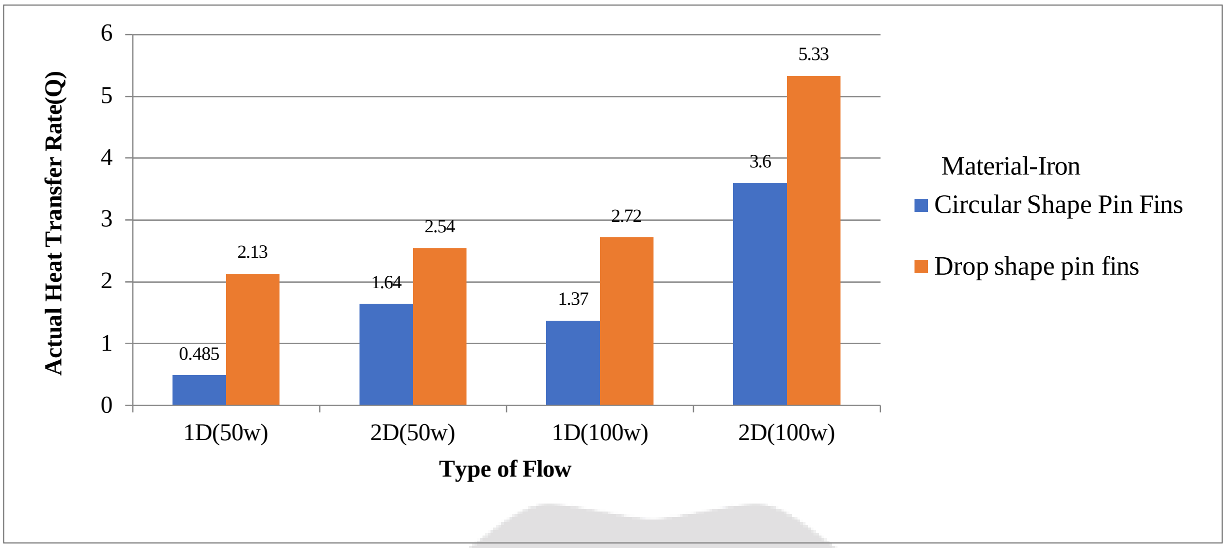
### 8.1.3 for Iron

| Sr No | Material | Iron | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | Type of flow | 1D | | | | 2D | | | |
| | Shape of fins | Circular | | Dropped | | Circular | | Dropped | |
| | Dimmer stat | 50W | 100W | 50W | 100W | 50W | 100W | 50W | 100W |
| 1 | Actual heat transfer rate, Q | 0.485 | 1.37 | 2.132 | 2.720 | 1.48 | 3.60 | 2.54 | 5.33 |
| 2 | Convective heat transfer coefficient, h | 64.560 | 81.220 | 72.034 | 91.906 | 79.177 | 119.897 | 85.977 | 128.669 |
| 3 | Nusselt Number, Nu | 484.2 | 609.15 | 540.25 | 689.29 | 593.82 | 899.22 | 644.82 | 965.01 |

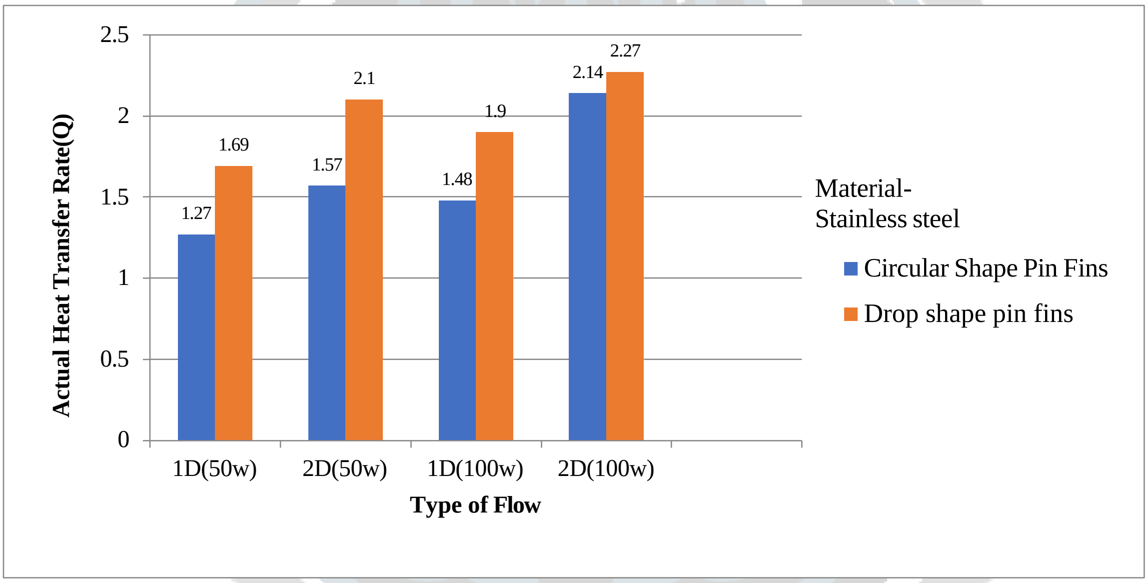
### 8.2 Graphs for Actual Heat Transfer Rate
### 8.2.1 for Aluminium



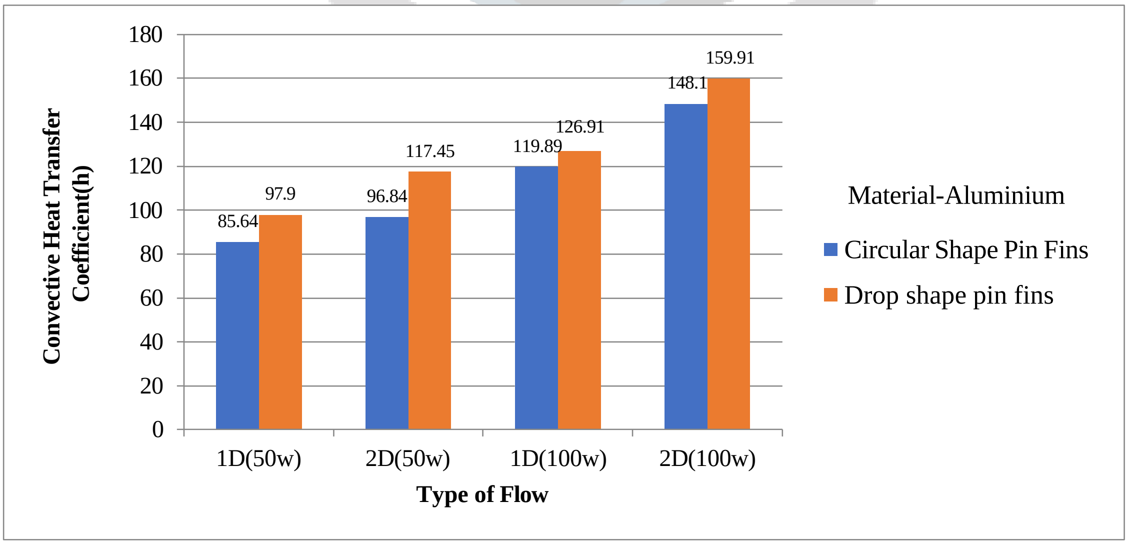
**8.2.2 for Iron**



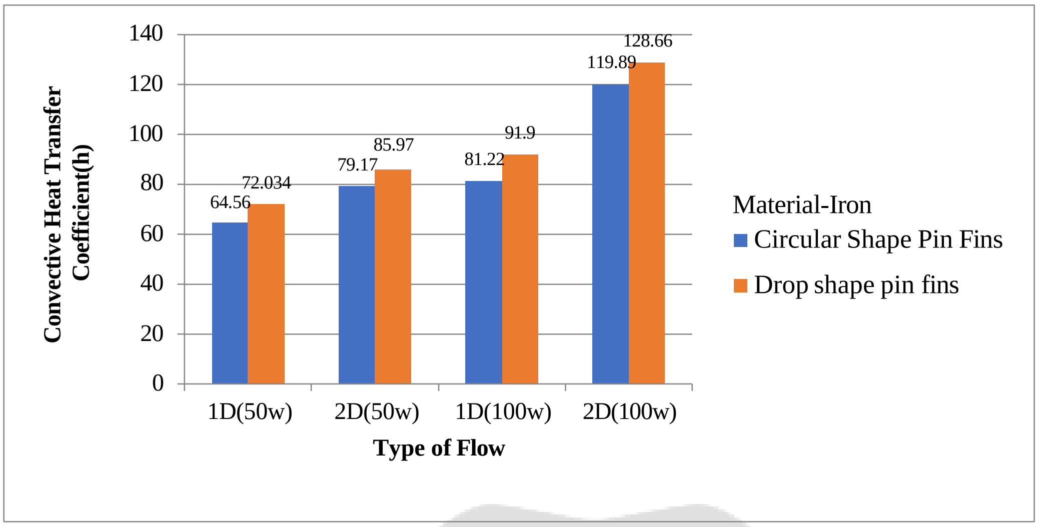

**8.2.3 For Stainless-Steel**



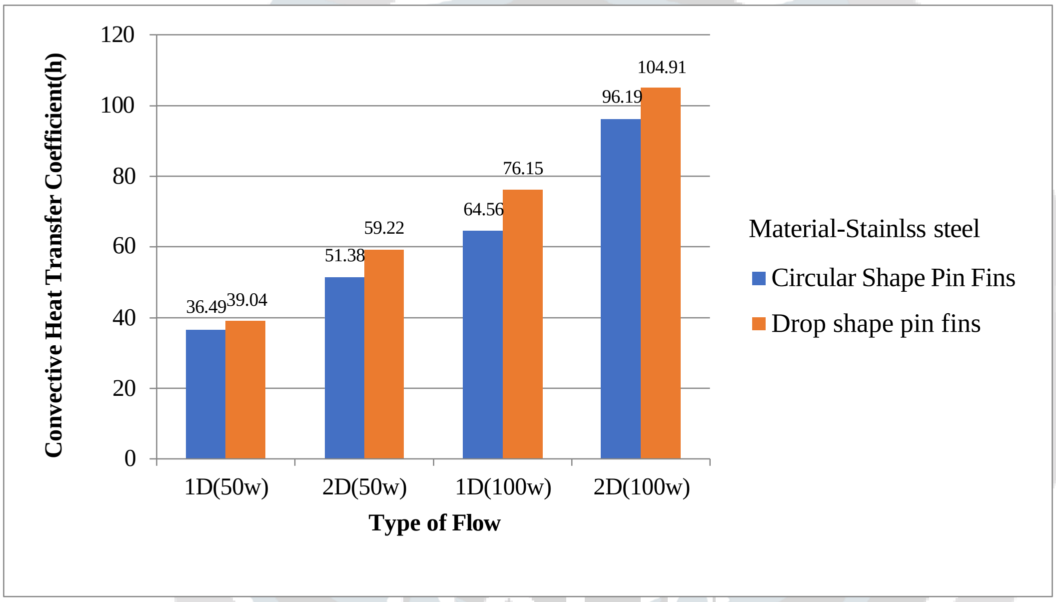

**8.3 Graphs for Convective Heat Transfer Coefficient (h)**
**8.3.1 for Aluminium**



**8.3.2 for Iron**

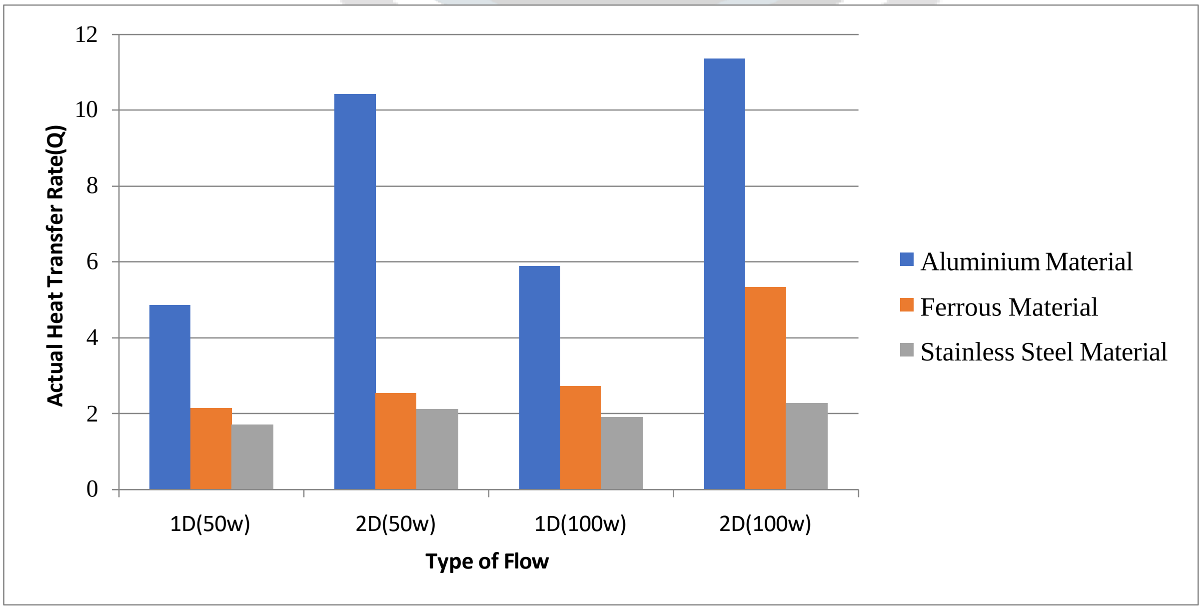



**8.3.3 For Stainless-Steel**

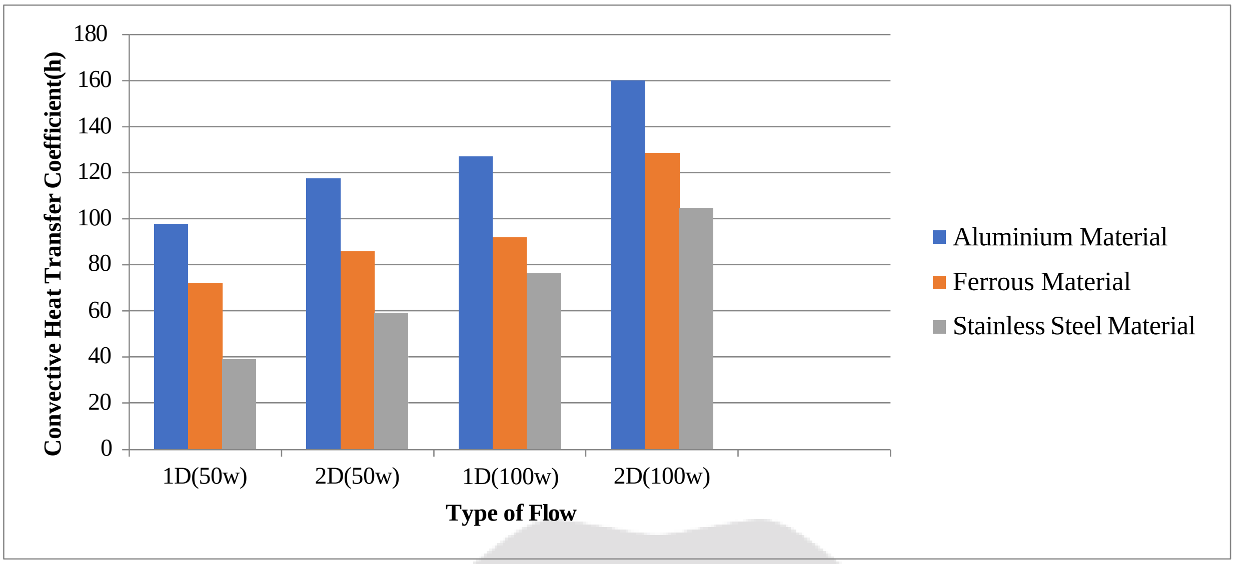



**8.4 Graphs for Dropped Shape pin fin**

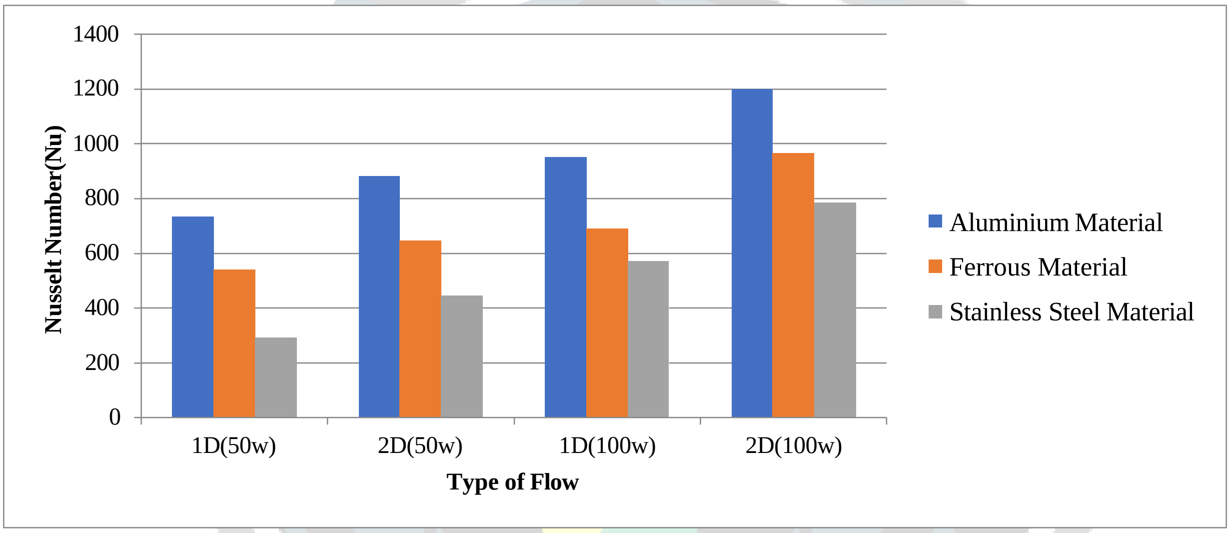
**6.4.1 Actual Heat Transfer Rate Vs Type of Flow for various materials**



### 8.4.2 Convective Heat Transfer Coefficient (h) Vs Type of Flow for various materials




### 8.4.3 Nusselt Number (Nu) Vs Type of Flow for various materials




## 9 CFD VALIDATION

### 9.1 Temperature contour for iron material in circular and dropped shape

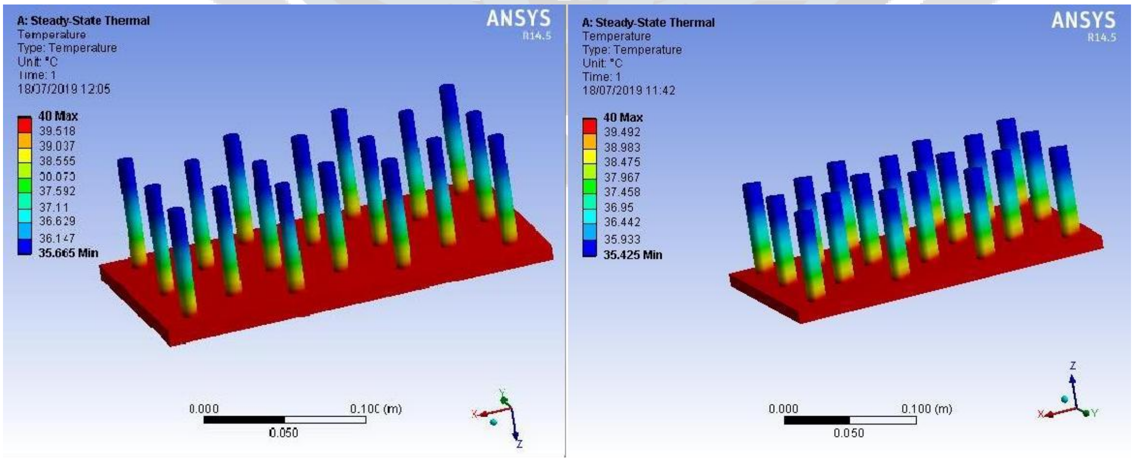



Figure No.9.1 Temperature contour (a) Circular shaped fins (b) Dropped shaped fins

The temperature contours shown above illustrate the static thermal distribution of iron pin fins with Circular and Drop shapes. It is evident from the figures that the Drop-shaped pin fins exhibit slightly improved heat dissipation compared to the Circular fins. Heat is more concentrated near the base of the fins and gradually decreases toward the tip. A rise in temperature within the fin structure can lead to potential damage, such as deformation or melting, and must be released promptly to prevent failure and maintain performance.

The temperature distribution shown in the contour plot clearly indicates that the thermal rise in the Drop-shaped pin fin array is marginally lower than in the other two fin shapes under consideration. The Drop-shaped fins enhance the interaction between the airflow and the fine surface, resulting in better thermal transfer than Circular pins.

maximum static temperature reaches 400 °C and the minimum is 35.6650 °C, while for Drop-shaped fins, the maximum is 400 °C and the minimum temperature drops to 35.4250 °C.

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**9.2 Validation of Data**

In this division outcomes from experiment and numerically are presented and compared. Results obtained for temperature of 40$^0$C and for constant mass flow rates of air in Set up. Thermocouples were used to measure temperature of fins at inlet, outlet and base plate. Computation results was obtained using ANSYS – CFD Post. The temperature contours shown in above figures clearly indicated that dropped shape fins of any material gives more heat transfer as compared to circular shape. This result presents and compares the findings from experimental testing and numerical simulations. The data was collected at a consistent base plate temperature of 400 °C and with a fixed air mass flow rate. Thermocouples were placed at the fin outlet, inlet and base to measure temperature readings. Numerical analysis was performed using ANSYS – CFD software. As illustrated in the temperature contour plots, drop-shaped fins—regardless of material—demonstrated better heat dissipation compared to circular fins. Discrepancies found between experimental and simulation outcomes may have resulted from various factors, including manufacturing tolerances, air leakage in the duct system, uneven heat distribution at the base. The results shown below are for a two-directional airflow setup, with ambient air maintained at 22 °C and the base plate held at 400 °C.
The variation in the experimental values may be caused due to many factors. Some of them are from fabrication errors, leakage of air from system, uneven heating etc. The experimental results with comparison of CFD results are shown in the following tables
The below table is 2- Directional flow and for 22$^0$C atmospheric air temperature, base plate temperature at 40$^0$C.

| **Sr. No** | **Dia** | **Aluminium** | | | | **Iron** | | | | **Stainless Steel** | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | **Circular** | | **Dropped** | | **Circular** | | **Dropped** | | **Circular** | | **Dropped** | |
| | | Exp. $T_{out}$ | CFD $T_{out}$ | Exp. $T_{out}$ | CFD $T_{out}$ | Exp. $T_{out}$ | CFD $T_{out}$ | Exp. $T_{out}$ | CFD $T_{out}$ | Exp. $T_{out}$ | CFD $T_{out}$ | Exp. $T_{out}$ | CFD $T_{out}$ |
| 1 | 50 | 32.12 | 37.33 | 32.71 | 37.52 | 29.86 | 33.52 | 27.2 | 33.21 | 25.23 | 26.67 | 24.20 | 26.42 |
| 2 | 100 | 32.90 | 38.33 | 35.6 | 38.22 | 31.21 | 35.66 | 30.11 | 35.43 | 26.11 | 27.48 | 25.88 | 27.14 |

**10.CONCLUSION**

1) Under single-direction airflow, the temperature of the pin fins increases along the downstream rows. Heat removal efficiency decreases row by row due to the rise in surrounding air temperature, causing the initial rows to show better heat transfer compared to those downstream.
2) With two-directional airflow, the temperature at the tips of all fins remains nearly uniform, thanks to the turbulence created by the intersecting air streams. The heat transfer distribution becomes more uniform, and the overall heat transfer rate improves significantly compared to single-direction flow.
3) The overall heat transfer is influenced by the shape of fin and surface area of the fins.
4) Drop-shaped fins mostly offer higher heat transfer and better performance compared to circular fins across all tested materials
5) Two-directional airflow enhances heat removal better than one-directional airflow.
6) In these materials the drop shape generates higher turbulence, increasing thermal transfer. Both the convective heat transfer coefficient and Nusselt number value are greater than those for the circular design.
7) The Nusselt number value for each material exceeded 100, it's indicating the flow regime is turbulent in all test cases.

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Siddiqui.
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