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ANALYSIS OF HEAT TRANSFER IN CIRCULAR TUBE USING Fe_3O_4-WATER NANO-FLUID WITH DIFFERENT FIN SHAPES

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ABSTRACT

Heat flow can be increased by using internal finned pipe and tape inserts. This article presents a numerical simulation analysis of constant heat flux and forced convection heat transfer under laminar condition with Fe_3O_4-water nanofluid. For a constant heat flux of 900 W/m^2 and modified T-shaped, longitudinal, and tapered internal fins, the convective heat transfer coefficient, nusselt number, and friction factor between pipe wall and working fluid are evaluated using numerical simulations. All the fins have constant height and constant cross-sectional area. Results indicate that T shaped fin provides higher heat transfer coefficient associated with convection when compared to other geometries, that's around 33% higher than plain tube. But this comes at a cost of increased friction factor. The highest friction factor noted for T shaped fin was 103.68% higher than the plain tube.

Keywords: Internal Fins, Convective Heat Transfer, Friction Factor, Heat Flux, Nano Fluid.

I. INTRODUCTION

Pollution and global warming are the new concerns humanity is facing in the modern world. As average temperature of earth keeps on increasing, mankind fears extinction of life as we know. So, countries are coming together to fight this so that we can pass a cleaner and more suitable environment to our next generation. So, to achieve this, countries have vowed to become net zero emitter of carbon in the coming 50 years. But here comes the challenge, what would be the replacement to fossil fuel? Answer to this question is solar energy, wind energy, hydro power plant, geothermal power, tidal energy etc. but each one has its own challenges associated with it.

Another way to reduce carbon foot print is to make already existing systems more efficient so that the energy demands will reduce and so with it the net carbon footprint. In this paper we will numerically asses the effects of different attachments in pipe which can increase the efficiency of flow heating without much increase in cost as well as energy requirements. The attachments considered are modified T shaped fin, longitudinal internal fins, tapered shape internal fins and working fluid considered is 0.4% concentration Fe_3O_4-water nano fluid.

II. METHODOLOGY

Objectives

Literature review shows the study of various designs of fins and other heat transfer enhancement techniques under different circumstances. It is found that T shaped internal fins are least considered fin shape while assessing the heat transfer enhancement techniques. Based on these following objectives of this study are proposed which are solved using numerical simulation:

1. To examine the impact of internal fins at low Reynolds numbers while maintaining constant cross-sectional area and height of the fin
2. Based on the number of fins and the mass flux, compare the convective heat transfer coefficient for longitudinal, tapered, and modified T-shaped internal fins.
3. Based on the number of fins and mass flux, compare the friction factor for longitudinal, tapered, and modified T-shaped internal fins.
4. To compare results against plain pipe or tube under similar working conditions

Problem Statement

A pipe with dimensions given in Table 1 is provided with a constant heat flux and laminar flow at inlet. Fe_3O_4-water was used as the working fluid in a numerical study for various fin shapes, and characteristics like the friction factor and nusselt number were determined for various scenarios.

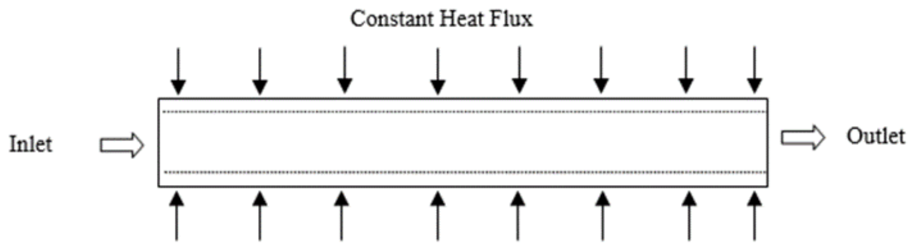

Figure 1: Side view of pipe model

Table 1. Numerical study parameters

Sr no.	Parameter	Value
1	Nominal/ Inner Diameter	9.52mm
2	Outer Diameter	10.02mm
3	Length of pipe	1500mm
4	Height of fin	1.5mm
5	Fin cross sectional area	0.75mm^2
6	Heat Flux on outer surface	900W/m^2

Calculations

1. Properties of nano fluid

It has been a practice since few years that to further increase the heat transfer characteristics, metal particles are scattered in base fluid, thus enhancing its capacities. Various properties required to define a nano fluid material in ANSYS were calculated from the following. Properties for Fe_3O_4 are given in Table 2.

Table 2. Thermophysical properties of Fe_3O_4 [19]

Nanofluid	Ferric Oxide (Fe_3O_4)
Diameter	<50nm
Purity	99%
Density	5180 Kg/m^3
Specific Heat	104 J/Kg-K
Thermal Conductivity	17.65W/m-K

Density of Fe_3O_4-water was calculated using equation number 1: [3]

$$\rho_{eff} = \rho_b(1-\varphi_{np}) + \rho_{np}\varphi_{np} \qquad (1)$$

In this equation the functional density of nanofluid, density of water/base fluid, concentration of Fe_3O_4 and density of Fe_3O_4 is given by ρ_{eff}, ρ_b, φ_{np}, ρ_{np} respectively.

Using equation number 2, the functional thermal conductivity of Fe_3O_4-water was determined: [3]

$$k_{eff} = \frac{k_b\,[k_{np} + 2k_b - 2\varphi_{np}(k_b - k_{np})]}{k_{np} + 2k_b + \varphi_{np}(k_b - k_{np})} \qquad (2)$$

where functional thermal conductivity, water/base fluid's thermal conductivity, Fe_3O_4 particle's thermal conductivity of and concentration of Fe_3O_4 are given by k_{eff}, k_b, k_{np}, φ_{np} respectively.

Using equation 3, the functional specific heat of the Fe_3O_4-water was determined: [3]

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$$c_{peff} = \frac{(1-\varphi_{np}) \times \rho_b c_{pb} + (\varphi_{np} \times \rho_{np} c_{pnp})}{\rho_{eff}} \quad (3)$$

where functional specific heat, water/base fluid's specific heat, Fe_3O_4 particle's specific heat of and concentration of Fe_3O_4 are given by C_{peff}, C_{pb}, C_{pnp}, φ_{np} respectively.

Using equation 4 functional dynamic viscosity was determined: [3]

$$\mu_{eff} = (1 + 2.5\varphi_{np})\mu_b \quad (4)$$

where functional dynamic viscosity, dynamic viscosity of water/base fluid and concentration of Fe_3O_4 is given by μ_{eff}, μ_b, φ_{np} respectively.

2. Calculations for nusselt number and friction factor

Calculation for friction factor and nusselt number are carried out using below mentioned equation.

Darcy friction factor is calculated using equation number 5:

$$f = \frac{-2(dp/dx)D}{\rho U_m^2} \quad (5)$$

In this equation Darcy friction factor, hydraulic diameter, mean flow velocity and density of working fluid is given by f, D_h, U_m, ρ respectively. [7]

Convective heat transfer coefficient for constant heat flow per unit wall area is given by equation number 6:

$$h = \frac{\dot{Q}}{T_w - T_b} \quad (6)$$

In this equation the average wall-liquid interface temperature, heat flow on nominal diameter surface, the bulk mean temperature is given by T_w, Q, T_b respectively. [7]

Nusselt number is evaluated using equation number 7:

$$Nu = \frac{hD_O}{k_f} \quad (7)$$

Nusselt number, heat transfer coefficient associated with convection, nominal diameter, and fluid conductivity are all present in the equation and is given by Nu, h, D_o and k_f respectively [7]

III. MODELING AND ANALYSIS

To perform a simulation study, first models are created using SOLIDWORKS keeping height and cross-sectional area of fin constant. Figures 2,3,4 shows the cut section of isometric view of pipes. The geometry of modified T shaped fin contains an arc and taper on its extended surface.

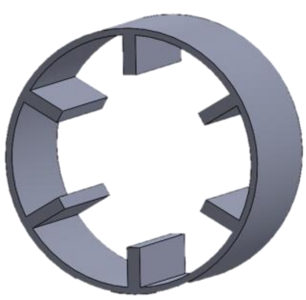

Figure 2: Longitudinal Fin

Figure 3: Tapered Fin

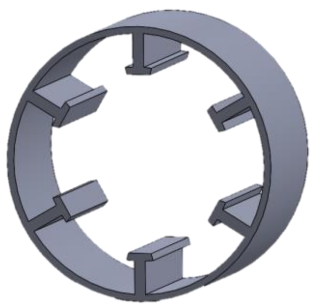

Figure 4: Modified T fin

After importing files to ANSYS FLUENT design modeler, the fluid domain and pipe wall domain is defined. After this mesh is generated. Triangular mesh is chosen because triangular mesh can cover difficult curves and corners of geometries well. Mesh method used to generate mesh is Sweep method with automatic source and target. Number of elements in each model exceeded 1000000 with good quality of mesh, based on parameters like skewness, orthogonality etc. Some meshes are shown in figure 5,6.

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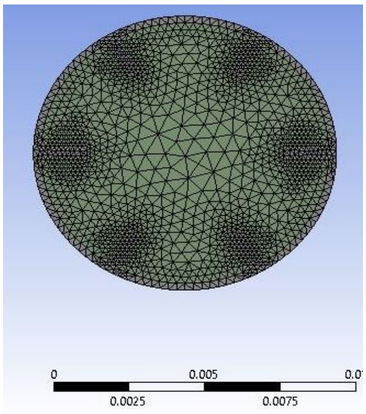

Figure 5: Longitudinal Fin Mesh

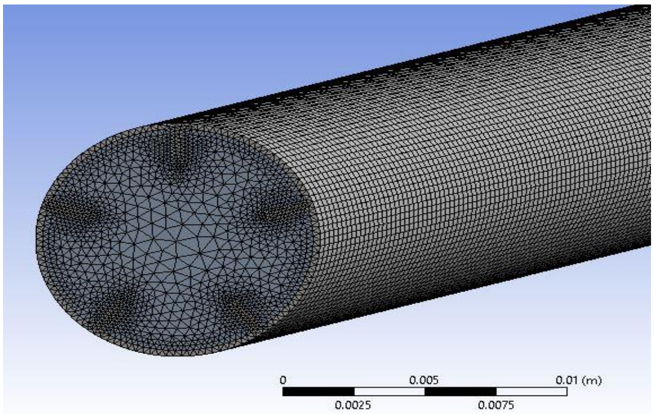

Figure 6: Isometric view of meshed pipe

To perform analysis in ANSYS FLUENT, the laminar with energy model is used. Constant heat flow per unit area boundary condition is provided at the outer wall for each and every case. For inlet, mass flow boundary condition is chosen and for outlet pressure outlet is chosen. SIMPLE scheme is selected with gradient as green-gauss node based. Pressure, momentum and energy equation are all set to second order upwind. Also, in this problem instead of relying on just residual monitor for deciding convergence, several other monitors are setup such as mass flow rate monitor, outlet temperature monitor and velocity monitor to ensure hydrodynamically developed flow. For every analysis an average of 450 iterations are required to converge below residual of 10^{-6}.

Numerical Validation

The nusselt number and friction factor for a constant flux condition in a plain circular pipe flow are known and results are used to validate the ANSYS numerical model.

Under conditions of continuous flux, the Nusselt number for developed circular pipe flow is given by:

$$Nu_L = \frac{h.L}{k_f} = 4.36 \quad (8)$$

where k is the nano-fluid's associated thermal conductivity, L is the hydraulic diameter/ characteristic length, and h is the heat transfer coefficient for convection. The Prandtl number of the working fluid must be greater than 0.6 for this finding to hold true for circular pipe flow under constant heat flux conditions.

Equation mentioned below provides the value of Darcy's friction factor in the developed regime for laminar flow:

$$f = \frac{64}{Re} \quad (9)$$

In this equation Reynolds number and Darcy friction factor are given by Re, f respectively.

First, a grid sensitivity test is carried out to make sure the model has enough elements for accurate computations. Figure 7's graph illustrates how the nusselt number varies with the amount of mesh elements.

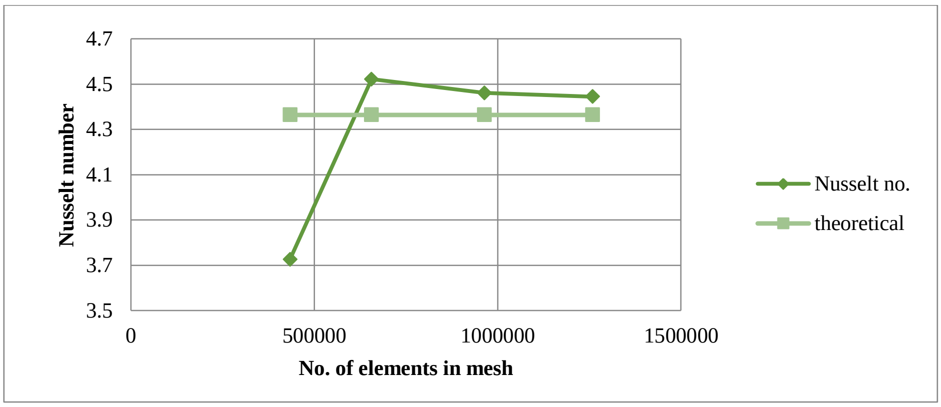

Figure 7: Grid sensitivity test for nusselt number

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From the grid sensitivity test the error in nusselt number calculation is found to be 1.58% at 963832 elements and 1.43% for 1259263 elements. Therefore, in view of error percentage the mesh elements for every calculation lied between 963832 and 1259263.

After this the numerical study on model is performed for Nusselt number under different Reynolds number. The maximum error encountered was 1.97% at Reynolds number of 300. The graph is shown in figure 8

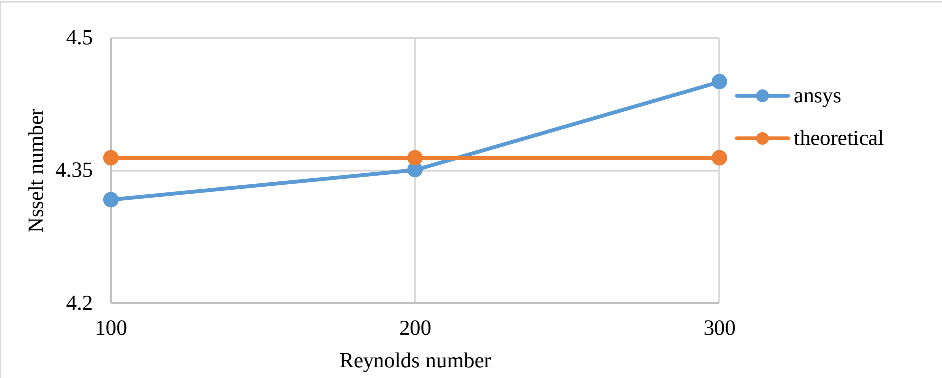

Figure 8: Nusselt number Vs. Reynolds number

Then a numerical study on model is performed for friction factor under different Reynolds number. The maximum error encountered is 2.56% at Reynolds number of 200. The graph is show in figure 9

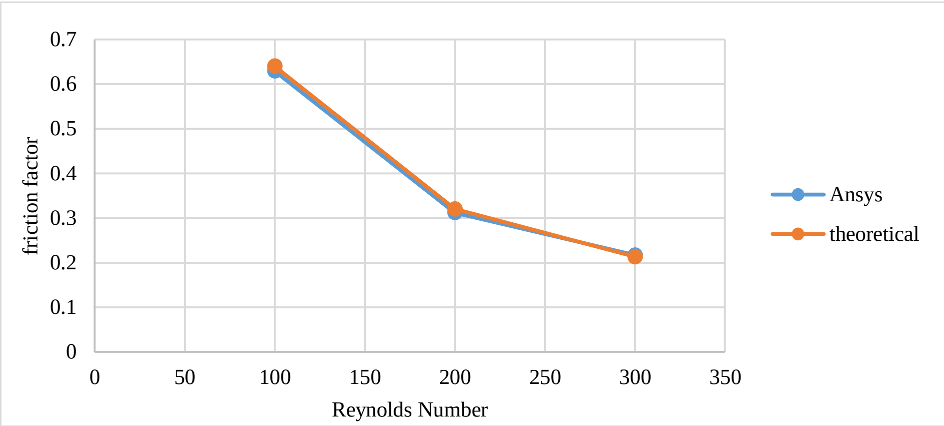

Figure 9: Friction factor Vs. Reynolds number

IV. RESULTS AND DISCUSSION

From the graph shown in figure 10, Nusselt ratio is found highest for modified T finned pipe. It was almost 33% higher than that of plain/un-finned tube. The graph shows that the Nusselt number and convective heat transfer coefficient both rise as the number of fins rises in each case considered.

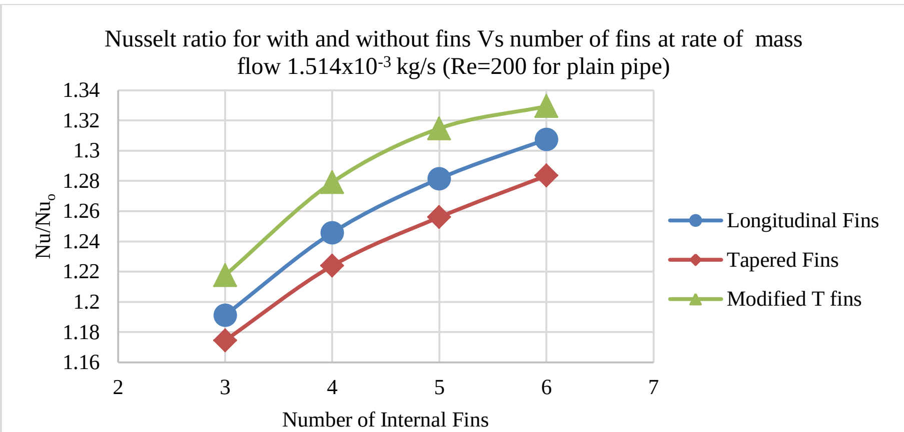

Figure 10: Graph of ratio of nusselt number with fins and un-finned surface Vs Number of fins

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From the next graph shown in figure 11, Modified T fin has the greatest Nusselt number among considered cases, i.e., high heat transfer coefficient associated with convection irrespective of number of fins involved when compared to others. It is observed that Nusselt number first decreases slightly and then keeps on increasing with increasing mass flow rate. Highest Nusselt number recorded is 5.841 for pipe with six internally T shaped finned tube.

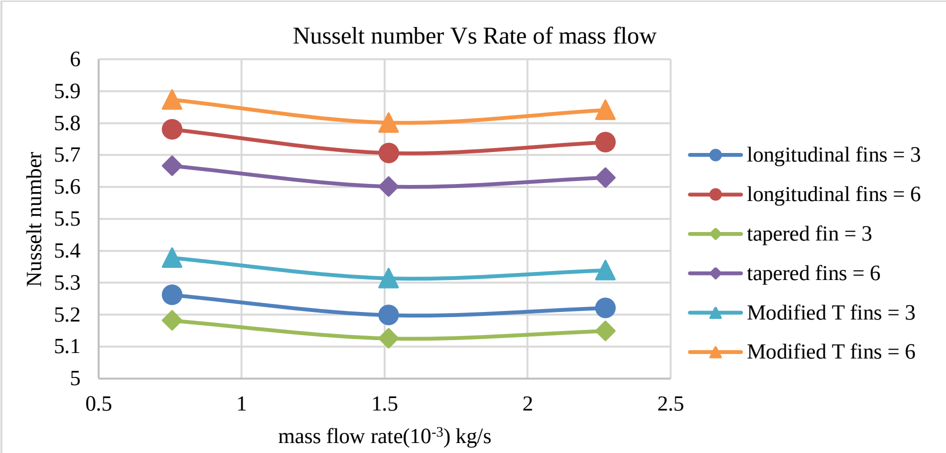

Figure 11: Graph of Nusselt number Vs Rate of mass flow

Previously we found that modified T shaped finned tube provide higher convective heat transfer coefficient than other. But as it can be seen from the graph shown in figure 12, that modified T finned tube also provides much higher friction factor too. The highest friction factor observed was 0.652357 i.e., 103.86% higher than plain tube and lowest was for tapered fins, which was 0.426573 i.e., around 33.3% higher than plain tube

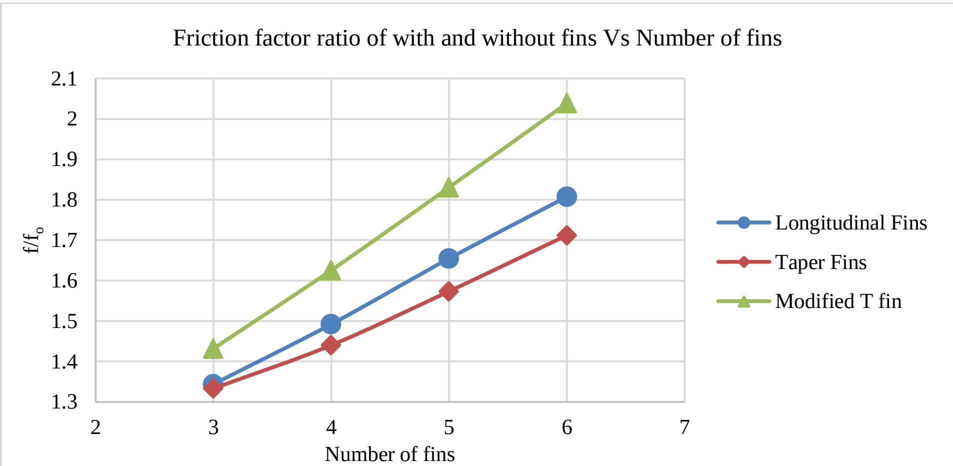

Figure 12: Graph of ratio of Friction factor with fins and un-finned surface Vs Number of fins at rate of mass flow 1.514x10-3 kg/s (Re=200 for plain pipe)

Previous results shows that T finned internal tube offer better convective heat transfer coefficient but that comes at an expense of increased friction factor. Here also we can see from the graph shown in figure 13, that T finned has higher friction factor and it decreases with increasing mass flow rate. And also, as number of fins increases so the friction factor.

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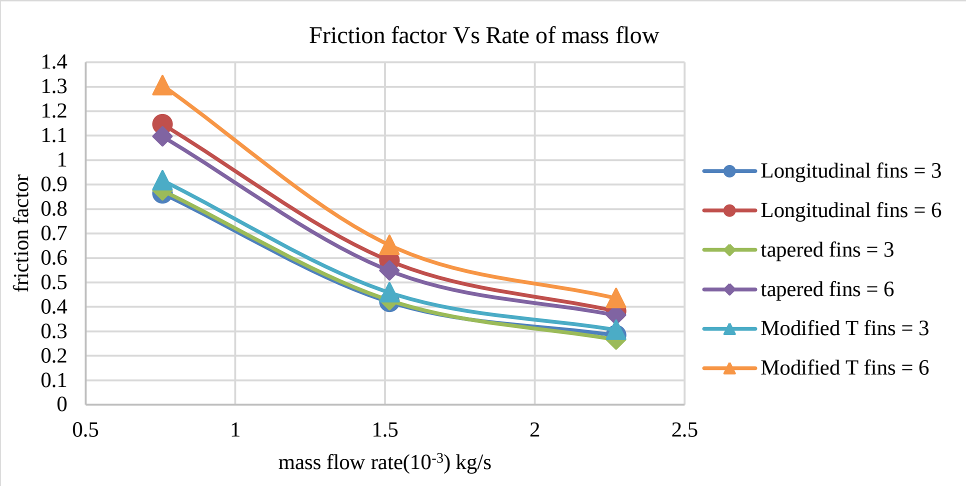

Figure 13: Graph of Friction factor Vs Rate of mass flow

V. CONCLUSION

Simulation study for different fin geometries is carried out. From the results we can conclude that modified T shaped fin provide higher convective heat transfer coefficient than tapered and longitudinal shaped internal fins. The highest value of convective heat transfer coefficient observed is 378.92 W/m^2K and as number of fins increases, heat transfer coefficient associated with convection increases. We also observed that on increasing mass flow from 0.7574 to 2.27 x 10^{-3} kg/s (Re_{equ} = 100 to 300 for plain pipe), the nusselt number first decreases slightly than increases. After this we saw results in friction factor and from that we can conclude that modified T shaped internal fins also provide with increased amount of friction factor. The highest recorded friction factor is 1.304823, and it also increases as the count of fins grows and lowers as the mass flow rate increases.

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