

# Effect on Performance of Microchannel Heat Sink by Employing Hexagonal Microchamber with Pin Fin Using Different Coolants through CFD Analysis

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**ABSTRACT:** Technological evolution in application of small-scale electronic devices have led to increase in cooling demand to accomplish this microchannel heat sink is decent option for removal of heat generated. In this numerical study by CFD analysis consequence of pin fin with hexagonal microchamber on performance of microchannel heat sink has been examined. Water, silicone oil, potassium formate solution and transformer oil were employed as coolant. Performance of MCHS was calculated in the basis of Nusselt no., pressure drop and thermal performance index at different values of Reynolds no. From this analysis it is found that use of hexagonal microchamber successfully lowers the pressure drop generated from application of pin fin.

**KEYWORDS:** Microchannel Heat Sink (MCHS), Heat transfer, Pressure drop, CFD analysis.

## I. INTRODUCTION

In this new era every month new technologically advanced electronic equipment especially small-scale devices are being introduced in market to fulfil rising demand of high-performance computers, reliable microcircuits of electrical appliances, thermal management of Li-ion batteries used in electric vehicles. These things generate tremendous amount heat when operated in summer seasons, unfavourable atmospheric condition and with high performance mode, from very compacted area. To dissipate heat from these devices various cooling techniques are discovered in past years. Many researches revealed that on reducing hydraulic diameter of pipe or any type of circular and no circular cross section heat transfer coefficient enhances. So, in order to increase heat transfer coefficient microchannel having hydraulic diameter in microns are used [1]. A conventional type microchannel consists rectangular cross section and straight channel made up of substrate material like copper, aluminium and silicon etc. every channel has inlet, outlet, side wall and bottom wall. Complete heat sink n no. of channels according to requirement. now a days for achieving better cooling performance it is essential to develop new model or have to make modification in conventional design. When fluid flows inside microchannels due to very small hydraulic diameter flow becomes laminar and laminar flow establishes velocity and temperature gradient between channel wall and fluid which do not allow fluid flowing in centre of channel to contact with wall. This phenomenon reduces heat transfer and non-uniform heating of coolant.

To overcome this problem obstacles, extended surfaces (Fins, Ribs) are employed in flowing path of fluid [2-4]. This technique initiates boundary layer redevelopment from every obstacle that allow hot fluid near channel wall and cold fluid at centre of channel to mix well and accomplish uniform temperature distribution [5]. Previous studies have modified microchannel heat sink design in various shape of cross section area and channel. They have found that modifying design helps in heat transfer intensification [6-7]. Design modification alter dimensions of channel, change total wetted area and manner in which fluid flow takes place. These days Nanofluid is better option for coolant it satisfies all coolant requirement of enhanced cooling. The thermo-physical properties of Nanofluid (NF) are varied according to percentage concentration of metal particle in solution of base fluid and metal oxide [8]. Although NF has

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great thermo physical properties but some studies revealed that NF is not adequate to use in MCHS due to heat flux form electronic device in practical situation is not uniform small rise in temperature can deteriorate surfactant property and it starts to disappear that results instability of particle in base fluid these particles adhere on channel wall and form a layer on it that reduces heat transfer [9].

## II. LITERATURE WORK

The innovative work of microchannels has done by Tuckerman and Peace [1] Most of researchers has done work on rectangular microchannel by etching as manufacturing technique. There are many experimental, numerical and analytical studies have been focused on utilization of Microchannel heat sink (MCHS) for removal of heat from small electronic devices.

Issam Mudawar and Weilin Qu [10] in 2001 have experimentally and numerically studied the heat transfer characteristics and pressure drop of a microchannel heat sink. The heat sink having array of rectangular microchannels cross section size of 231 wide X 713 deep in micrometre. Deionized water was used as the coolant and two heat fluxes, 100 W/cm<sup>2</sup> and 200 W/cm<sup>2</sup> applied relative to the platform space of the heat sink were tested. Results from experimental observation are compared and found in decent agreement with numerical prediction.

Navin Raja Kuppasamy et. al. [11] in 2014 they have conducted numerical study on microchannel heat sink with inclined passage. In this study they introduced inclined passage between adjacent channel wall in alternate orientation. This analysis considered effectiveness, Nusselt no. pressure drop as performance parameter and found that introducing slant passage divert flow between channels due to which mixing of fluid streams takes place and results in greater heat transfer. It also reduces boundary layer thickness. The overall performance increase 146% and thermal resistance reduced by 76.8% as related to simple MCHS. Generally, rise in heat transfer cost high pressure drop but using slanted passage pressure drop falls by 8%.

Vikas Yadav et. al. [12] in 2016 they have performed three dimensional CFD investigation on extended surface (fin) rectangular microchannels and conventional plain rectangular microchannel and compare each other. To optimize design parametric variation are also carried out. Three unique arrangements of prolonged surface microchannel. It was found that Case I has good heat transfer characteristics than Case II and Case III (At low Re no.) the average heat transfer augmentation is about 160% with acceptable pressure drop. Performance of finned MCHS enhances with increase in fin height and decrease in pitch of fin. Optimized values are no. of fin > 6, pitch of fin = 4.05 mm and diameter of fin = 0.08 mm.

H. Kamal and A. Dewan [13] in 2017 they performed numerical study on computationally prepared model of MCHS with high aspect ratio to observe the effect of geometrical features on MCHS and establish optimal design configuration to maximize heat transfer. They have also studied inferences of multichannel effect by introducing transverse microchamber. Width of the transverse microchamber is also optimized to get max. heat removal. Results showed that increase in average Nusselt no. is approximately 5% and reduction in pressure drop is approximately 3% to 7% for whole microchannel heat sink as compared to conventional MCHS design.

Ihsan Ali Ghani et. al. [14] in 2017 a numerical study was carried out on microchannel heat sink with rectangular ribs and secondary inclined passage in alternate configuration. The consequences were observed in the form of Nusselt no. and friction factor for Reynolds no, ranging from 100 to 500. To do so performance was evaluated by comparing new design of MCHS with microchannel having rectangular ribs and MCHS its secondary inclined passage (channel). They used computational FLUENT 14 software for analysis. The parameters of new MCHS design were also optimized and evaluated at Re no. 500, channel width ratio = 0.666, ratio of rib width to channel width = 0.5 and angle = 45 degree with performance factor of 1.98.

Varghese Vinu And Pa Abdul Samad [15] in 2018 a numerical analysis has been done on 3D numerical model to analyse the effect of various dimension of ellipsoidal rib between lateral microchamber on heat dissipation and pressure drop parameters. They considered conjugate heat transfer, entrance effect and viscous heating with thermophysical properties dependent on temperature. Main purpose of this study was to enhance heat transfer by optimizing ellipsoidal rib parameters surrounded by transverse microchamber. Outcomes from analysis showed that there was 25-46% rise in Nusselt no. for MCHS with ellipsoidal ribs and 20-47% increase in friction factor compared to simple straight MCHS for Re. no. varied from 132 to 664. Employing ellipsoidal rib in transfer microchamber improved heat transfer coefficient and optimized dimensions of rib are 0.4 mm and 0.125 mm.

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After carefully studying previous literature work it is concluded that microchannel heat sink with liquid cooling is one of best heat transfer technique that can be used to extract high heat flux.

Main purpose of this CFD study is to reduce pressure drop generated by fins in upstream direction inside MCHS with the application of Hexagonal microchamber surrounded by pin fin and to analyse hydrothermally application of different coolant fluid except water and Nanofluid. The reason behind study of using different coolant liquid which is being used commercially is to achieve high performance of MCHS without use of water.

### III. MODEL DESCRIPTION

Model taken from MCHS used by W. Qu and I. Mudawar [10] Experimental work. Physical model made up of copper substrate of size 44.8 mm X 9.8 mm with a total of 21 channels are considered for present work. The geometrical parameters of rectangular micro channel heat sink are channel width 0.231 mm ( $W_{ch}$ ), Channel Height 0.713 mm ( $H_{ch}$ ). Due to symmetric condition and to lessen the computational time, single unit of microchannel heat sink is considered for CFD model.

$W_w$ (mm)	$W_{ch}$ (mm)	$H_{w1}$ (mm)	$H_{ch}$ (mm)	$H_{w2}$ (mm)	$H_{th}$ (mm)
0.118	0.231	12.7	0.713	5.637	3.175
$L_1$ (mm)	$L_2$ (mm)	$L_3$ (mm)	$L_4$ (mm)	$L$ (mm)	
5	11.588	11.588	11.588	44.764	

Table 1 Dimensions of model

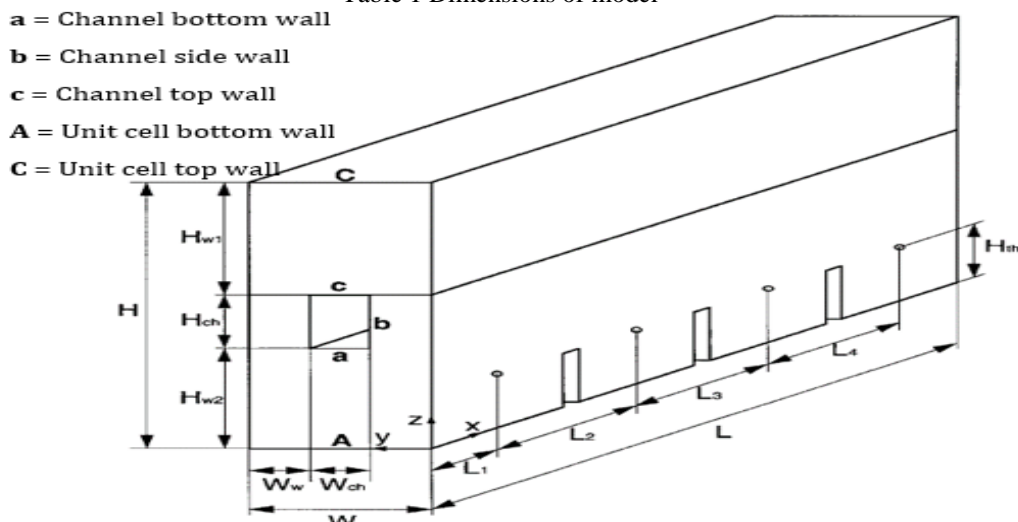


Fig 2 Physical model

### Computational fluid dynamics

Computational fluid dynamics is a subject that deals with computational approach to fluid dynamics with numerical resolution of the equations which convey the flow of the fluid and it deals with the equations of the fluid flow and also generic enough to be able to solve simultaneously together the equations that direct the energy transportation and as well the governing equations that determine the fluid flow pattern and how the fluid flow proceeds and energy transfer takes place; all these things can be undertaken together in an identical format. So, this outline enables us to deal with a very complex flow circumstances in reasonably fast time, such that for a particular set of conditions, a researcher will be capable to simulate and see how the flow is taking place and what kind of temperature distribution is there and what kind of pattern of flow and where vortex are formed, so that we can make changes to the parameters that are under control to modify the way that these things are happening.

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## Assumptions: -

Following assumption are made for simplifying Numerical Study in this work

1. Flow is Laminar and Steady.
2. Fluid is incompressible and Newtonian.
3. No slip boundary condition at wall.
4. Uniform heat flux throughout the bottom surface of MCHS (100 W/cm<sup>2</sup>).
5. There is no viscous dissipation.
6. Body force neglected.
7. Radiation Heat transfer is neglected.
8. Constant thermophysical properties for solid section.

In order to examine the effect of different coolant and proposed design performance values of Nusselt no., pressure drop and performance index are calculated using values of temperature, pressure and velocities at different location of channel.

## Governing equation and Parameter definition

Based on above mentioned assumptions required governing equation for analysis of heat transfer and flow characteristics of flowing fluid are as follows. By solving these equation velocity, temperature and pressure distribution can be computed.

### Continuity Equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0$$

Where u, v and w are the velocity components in x, y and z-directions respectively.

### Momentum Equation:

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = -\frac{1}{\rho_f} \frac{\partial p}{\partial x} + \frac{\mu_f}{\rho_f} \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right)$$

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} = -\frac{1}{\rho_f} \frac{\partial p}{\partial y} + \frac{\mu_f}{\rho_f} \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right)$$

$$u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = -\frac{1}{\rho_f} \frac{\partial p}{\partial z} + \frac{\mu_f}{\rho_f} \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right)$$

Where  $\rho_f$  and  $\mu_f$  are the density and dynamic viscosity of the working fluid, respectively, and p is the fluid pressure.

There have two energy equations that related to the present study such as

Energy equation for fluid region,

$$u \frac{\partial T_f}{\partial x} + v \frac{\partial T_f}{\partial y} + w \frac{\partial T_f}{\partial z} = \frac{k_f}{\rho_f C_{P_f}} \left( \frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2} + \frac{\partial^2 T_f}{\partial z^2} \right)$$

Governing equation for heat sink is given by,

$$0 = k_s \left( \frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} \right)$$

Where  $T_f$ ,  $T_s$ ,  $k_f$ ,  $k_s$  and  $C_{P_f}$  are the fluid's temperature, solid's temperature, fluid thermal conductivity, solid thermal conductivity and fluid specific heat, respectively.

The following parameters were considered in order to calculate the overall thermal performance of microchannel heat sink.

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Reynolds number at inlet is defined as

$$Re = \frac{\rho_f u d_h}{\mu_f}$$

Where  $u$  is inlet velocity and  $d_h$  is hydraulic diameter of microchannel which is defined as

$$d_h = \frac{2W_{ch}H_{ch}}{(W_{ch} + H_{ch})}$$

Where  $W_{ch}$  and  $H_{ch}$  are microchannel width and height respectively.

The overall Nusselt number is calculated as:

$$Nu = \frac{q_{\Gamma} d_h}{k_f (T_{\Gamma} - T_{fm})}$$

Where  $T_{\Gamma}$ ,  $T_{fm}$  and  $q_{\Gamma}$  are mean temperature of interface (channel wall), volume average bulk fluid temperature and heat flux at interface respectively.

$$q_{\Gamma} = \frac{q A_b}{A_{sur}}$$

Where  $A_b$  is bottom wall area,  $A_{sur}$  is the contact surface area of fluid and solid and  $q$  is heat flux applied.

## IV. METHODOLOGY

Ansys fluent 18.1 was used to simulate the models. Grid independence test was performed for getting independent grid size. There are two types of model were developed one is rectangular straight MCHS with water as coolant taken as reference and second is MCHS with hexagonal microchamber and pin fin. Numerical study done by Yadav Y. et al. [12] has optimized the size of fin inside microchannel. We have taken optimized design from literature of pin fin having following dimensions

$d_f$ (fin dia. In mm)	0.08
$h_f$ (fin height in mm)	0.713
P (pitch in mm)	4.05
No. of fin	6

Table 3 fin dimensions

### Solution method

The simulations were performed using Laminar and energy model. The SIMPLE procedure was used to deal with pressure-velocity coupling. The governing conservative continuity, momentum and energy equations was solved using the Finite Volume Method. The second-order upwind scheme was employed for solution of momentum and energy equations. The solution was assumed to be converged when residual criteria  $1 \times 10^{-4}$  is used for continuity equation,  $1 \times 10^{-6}$  for the velocity in x, y and z direction and for energy equation  $1 \times 10^{-7}$ .

**Boundary condition**

Boundary condition is a condition for thermal and hydrodynamic boundary applied on 3D geometrical model prepared by design modular in Ansys fluent.

Location	Condition
At Inlet, $x = 0$	$u_f = u$ (uniform velocity and calculated by using Reynold's no.), $v = w = 0$ $T_f = T_{in} = 288.15K$ (For water), $-k_s \left(\frac{\partial T_s}{\partial x}\right) = 0$ (For Solid)
At Outlet, $x = L = 44.8 \text{ mm}$	$p_f = p_{out} = 1 \text{ atm}$
	$-k_f \left(\frac{\partial T_f}{\partial x}\right) = 0$ (For water), $-k_s \left(\frac{\partial T_s}{\partial x}\right) = 0$ (For solid)
At fluid-solid interface	No slip and no penetration $u = v = w = 0$ $-k_s \left(\frac{\partial T_s}{\partial n}\right) = -k_f \left(\frac{\partial T_f}{\partial n}\right)$ $n$ is the normal to wall
At Top wall, $y = 6.35 \text{ mm}$	$u = v = w = 0$ ; $-k_s \left(\frac{\partial T_s}{\partial y}\right) = 0$
At Bottom wall $y = 0$	$-k_s \left(\frac{\partial T_s}{\partial y}\right) = q = 100 \text{ W/cm}^2$

Table 4 Boundary condition [12]

After setting all conditions and values of parameters, initialization of solution is done by hybrid initialization for getting initial set of solution for governing equations. After that calculation is started by taking 500 iterations for convergence of solution.

**V. RESULTS & DISCUSSION**

**Result of Numerical analysis of MCHS**

Firstly, numerical analysis of rectangular straight microchannel heat sink by using water, inlet temperature of 288.15K as a coolant flowing through channel at different inlet velocities.

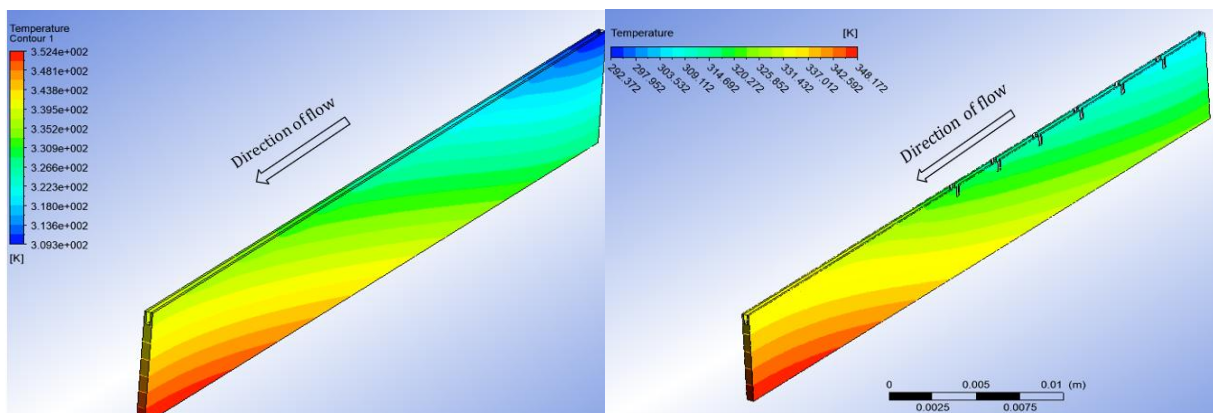


Fig 7 (a) Temperature Contour of MCHS (b) Temperature contour of MCHS combined with hexagonal microchamber and pin fin At Re no. 400.

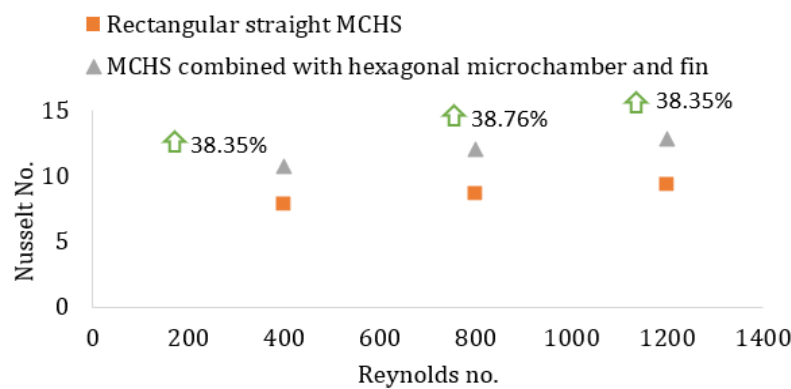
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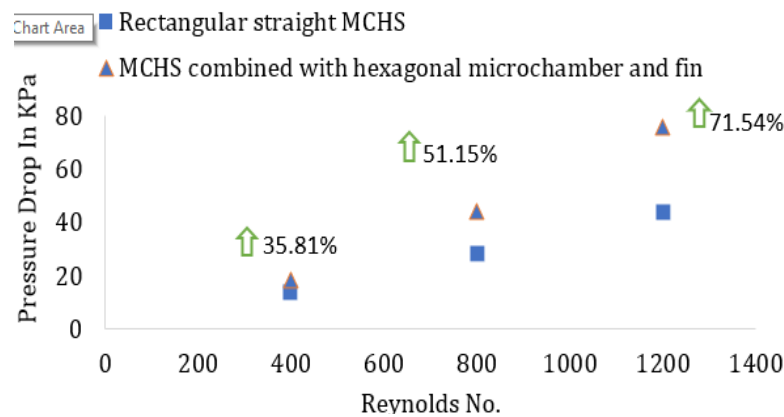
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After simulation values of Pressure drop, average interface temperature between fluid and channel wall, bulk fluid mean temperature and outlet temperature are noted down and values of performance parameter were evaluated and compared with reference model.



Graph 5 Comparison of Nusselt no. using both designs



Graph 6 Comparison of pressure drop using both designs.

**Thermal Performance Index** The application of Hexagonal microchamber and pin fin in microchannel heat sink has decent heat transfer characteristics compared to rect. straight micro channel with rise in pressure drop. Hence a non-dimensional parameter namely thermal performance index (TPI) is defined to study the overall performance of Hexagonal microchamber with fin heat sink by comparing with un finned microchannel heat sink (Rectangular straight MCHS). Thermal performance index is defined as

$$\eta = \frac{Nu/Nu_0}{\left(\frac{\Delta p}{\Delta p_0}\right)^{\frac{1}{3}}}$$

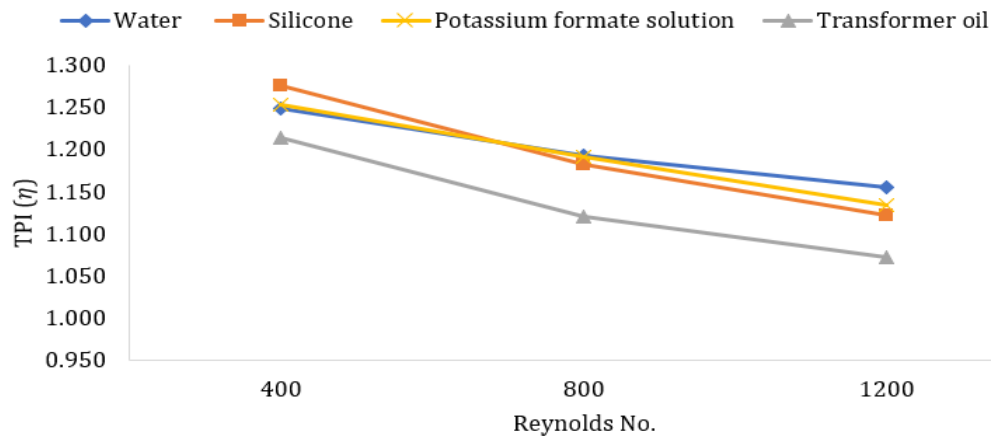
Where  $Nu$ ,  $Nu_0$ ,  $\Delta p$  and  $\Delta p_0$  are Nusselt no., Nusselt no. of reference design (RSMCHS) and reference fluid (water), pressure drop and pressure drop of reference design (RSMCHS) and reference fluid (water) respectively. If the TPI value is greater than 1, means the enhancement of heat transfer performance obtained by the enhanced MCHS is within acceptable pressure drop.

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Graph 7 TPI v/s Reynolds no. for different coolants.

## VI. CONCLUSION

In this paper the various coolants namely Silicone oil, potassium formate solution and Transformer oil at various Reynolds number at 400, 800 and 1200 were analysed on the basis of overall Nusselt number, pressure drop and performance index

- MCHS combined with hexagonal microchamber and fin gives 38.35 % increase in Nusselt no. and 35.81% rise in pressure drop at lower Re no. values (smaller than 800)
- To compare the performance of both the designs thermal performance index was introduced and it shows that silicone oil is best suits for lower Reynolds no. ( $Re < 800$ ) used with fin and hexagonal microchamber MCHS model.
- So, in this work we can say MCHS combined with hexagonal microchamber and fin when used with silicone oil perform 24% better at  $Re\ no. < 800$ .

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