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Theoretical analysis of performance parameters of a microchannel heat exchanger

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Abstract The increase in energy demands in various industrial sectors has called for devices small in size with high heat transfer rates. Microchannel heat exchangers (MCHX) have thus been studied and applied in various fields such as thermal engineering, aerospace engineering, and nanoscale heat transfer. They have been a case of investigation due to their augmented thermal characteristics and low-pressure drop. The goal of the current investigation is to analyze analytically the thermohydraulic performance of the heat exchanger. Studies are done for various inlet conditions and flow conditions. At T_{hi} of 90°C, the effectiveness increased by about 22% for an increase in Re from 1000 to 5000 of the cold fluid. It was also observed that at Re = 5000 for the hot fluid, the heat recovered by the hot fluid increases by about 69% for an increase in the inlet temperature of the hot fluid from 50°C to 70°C.

Keywords Theoretical analysis, performance parameters, microchannel heat exchanger

Nomenclature

$\boldsymbol{m_h}$	mass flow rate of cold fluid	$\boldsymbol{T}$	Temperature [K]
m_c	mass flow rate of hot fluid	NTU	Number of Transfer Units
C_{max}	Maximum heat capacity of fluid [W/K]	t	Thickness of tube [mm]
C_{min}	Minimum heat capacity of fluid [W/K]	h	Heat transfer coefficient [W/m^2-K]
d_i	Inner Diameter of Inner Tube [mm]	R_t	Total Thermal Resistance [K/W]
d_o	Outer Diameter of Inner Tube [mm]	U	Overall Heat Transfer Coefficient [W/m^2-K]
D_i	Inner Diameter of Outer Tube [mm]	F	Correction Factor
D_o	Outer Diameter of Outer Tube [mm]	C_p	Specific Heat of Fluid [J-kg-K]
R_t	Total Thermal Resistance [K/W]	Q	Heat Transfer Rate [W]
U	Overall Heat Transfer Coefficient [W/m^2-K]	h	Heat transfer coefficient [W/m^2-K]

1. Introduction

Devices that allow thermal energy to be transferred between two or more fluids of varying temperatures are known as heat exchangers. The one heat exchanger that has been emerging due to its superlative characteristics is the microchannel heat exchanger (MCHX). Due to their versatility and enhanced thermal characteristics, micro-channel heat exchangers have grown in popularity over the years. Due to various factors such as their efficiency, heat transfer rate, more compact construction, and cheaper cost, they have increasingly been used in

HVAC&R. The thermo-hydraulic properties of the MCHX are investigated in this work. Many studies have looked into the performance characteristics of the MCHX, which are provided in this section.

In their study, Han et al. [1] described how microchannel heat exchangers have grown in popularity as a means of meeting energy demands. It is increasingly used in the fields of heating, ventilation, and air conditioning. The decrease in the number of scales, which improves fluid compressibility effects, has been ascribed to higher heat recovery. It was found that improving the microchannel structure will enhance the heat exchanger's thermo-hydraulic performance if the pressure loss and heat transfer characteristics are properly anticipated before constructing the microchannel. Deng et al. [2] developed and simulated an MCHX using engine coolant and refrigerant. For pressure drop, the extrapolation approach was employed, which not only produced excellent results but also reduced computation time. A study of microchannel heat exchangers and their prospective uses in energy conversion and utilization was provided by Khan and Fartaj [3]. With rising energy needs, space constraints, and material savings, it was claimed that tiny lightweight heat exchangers had pioneered the way ahead. They have a channel diameter of less than 1mm, allowing for a fast rate of heat transmission while still being light. The only thing that has to be demonstrated is whether or not traditional correlations hold in microchannels. Kwon et al. [4] studied cooling in a compact cross-flow MCHX. High-speed air movement through microchannels made this possible. The refrigerant mass flow ranged from 330 to 750 kg/m^2-s during the operations for $7500 < Re < 20,500$. When compared to commercial heat exchangers, the power density attained via these tests is above 100 W/cm^3. By adding air passages, the power densities recorded may be raised as needed. The main downside of the devices discussed in the paper is that they demand a lot of pressure. In a microheat exchanger used for microelectronic cooling, Nikkhah and Nakhjavani [5] conducted experiments to examine the heat transfer properties of a water-based nanofluid. Experiments were carried out with varying heat fluxes and nanofluid mass flow concentrations ranging from 0.1 to 0.3 percent. When compared to the base fluid, heat transfer increased by 40.1 percent at 0.3 percent mass concentration, which was attributed to an enhancement in the base fluid's thermal conductivity followed by an intensification of Brownian motion. The pressure drop was observed to rise as the viscosity of the working fluid increased.

The thermal and hydraulic properties of a vapor venting two-phase MCHX were studied experimentally and computationally by David et al. [6]. Traditional non-venting heat exchangers working at varying heat fluxes were compared, with water fluxes also being varied. For most values of mass flow rate, the pumping power in the venting devices was found to be in good accord with the mathematical model and experimental findings. Copper venting device temperatures improved by the same amount as the reduction in saturation temperature. Zhang et al. [7] studied thermal characteristics in rough microchannels experimentally. A test system was constructed, and tests were undertaken to evaluate the impact of rough surfaces on microscale liquid flow. Roughness is essential even under laminar flow conditions, according to the findings. With an increase in Re, the effect of roughness on liquid flow becomes more prominent. Nu, which defines the thermal properties is dependent not only on Re but also on the geometry of the channel. The influence of microchannel performance on flow maldistribution was investigated computationally and experimentally by Joseph et al. [8]. The research aimed to improve the thermal properties of the system by creating local turbulences rather than using very tiny hydraulic diameters. The goal is to reduce the critical Re value by introducing perturbators, which will aid in reducing pressure loss and increasing the system's thermal efficiency. Wire net and S-shaped perturbators were compared and contrasted. Where the thermal efficiency reaches its maximum value, a discharge may be established. The flow rate deviation reduces as microchannel resistance increases, according to collector investigations. Fan and Luo [9] offered a literature assessment of microchannel heat exchangers' current advancements and applications. When compared to traditional heat exchangers, shrinking the channel size increases the heat exchange surface area per unit volume. The thermal characteristics are augmented due to the increased surface area for heat transmission. The heat exchanger's rapid reaction time also enables improved temperature control for a minor temperature differential between fluid flows. Yang et al. [10] developed, built, and tested three microchannel heat exchangers. For comparison, a straight channel was also created. The chevron channel heat exchanger was observed to have the lowest thermal properties with increased pumping power, almost 5 times greater than the other heat exchangers. All three heat exchangers outperformed the straight MCHX in terms of thermal performance.

Qu and Mudawar [11] investigated heat transmission and pumping power required in a heat sink with small channels using tests and numerical simulations. A grid of rectangular microchannels made up the heat sink. Higher Re was shown to be advantageous in lowering the outflow temperature as well as the temperatures within the heat sink. The numerical predictions and the experimental findings were in good accord. The traditional flow and energy equations were also shown to properly predict the heat transfer properties of an MCHX. Wang and Peng [12] investigated heat transfer characteristics in an MCHX experimentally. The working fluid was either water or methanol, and flow properties were studied in a rectangular microchannel. At around Re = 1000 – 1500, the fully formed turbulent convection regime was discovered to begin. is extremely unique and difficult to anticipate, and it is significantly influenced by several variables. The impact of fluid properties and geometrical factors on liquids flowing via MCHX was investigated experimentally by Peng and Peterson [13]. The liquid convection properties seen in normally sized channels are varied, according to the experimental data. Re, as well as the fluid properties and geometrical aspects, impact transition and laminar heat transfer. Peng and Peterson [14] thermal and hydraulic characteristics in microchannel structures experimentally. The geometrical parameters of the MCHX were discovered to affect laminar heat transmission. As Z approaches 0.5, the friction factor or flow resistance reaches a minimal value. Both the heat transmission and the pressure decrease may be calculated using empirical correlations. Liu and Garimella [15] studied flow characteristics in microchannels both experimentally and numerically. Experiments were performed at varying flows both in the laminar as well as turbulent domains. The decreased microchannel size was anticipated to have a substantial effect on the formation of turbulence, based on the calculated Kolmogorov length scales. The practical results matched the numerical calculations quite well.

Heat transfer and flow properties in rectangular microchannels with varying aspect ratios were studied by Lee and Garimella [16]. The first focus is on laminar convective heat transport in the microchannel entry area. In a hydrodynamically created yet thermally evolving flow, computations were used to estimate stable, laminar heat transfer coefficients. Correlations were also postulated to predict the local and average Nu. In microchannel heat exchangers, Garyaev et al. [17] investigated the optimum thermal-hydraulic features ratio. An objective function was created that integrates the different aspects of a heat exchanger, such as channel number, length, and diameter. It was discovered that for a given number of microchannels and length of the channel, there is only one optimum channel diameter. Hetsroni et al. [18] wanted to show that traditional theory can accurately predict the laminar flow characteristics in microchannels. The transition occurs at accurate values of roughness, which is in perfect accord with flow data. The relationship between hydraulic diameter and channel length, as well as Re, are essential elements in determining how viscous energy dissipation affects flow characteristics. The frictional pressure decrease for liquid flows via square microchannels was studied by Judy et al. [19]. For channel diameters between 15 and 150 and $8 < Re < 2300$, pumping power is utilized to evaluate the friction factor. Due to their different polarity and viscosity qualities, several working fluids were employed, such as distilled water, methanol, and isopropanol. The mistakes that are dominated by the measurement of diameter account for the experimental uncertainty in the measurement of friction factor. A multi-nozzle MCHX was proposed by Tran et al. [20]. The impacts of different geometrical parameters on the MN-MCHXs' thermodynamic characteristics were numerically studied. This new heat exchanger outperformed the old one in terms of thermal performance, with a maximum thermodynamic performance of 124% greater.

Hasan et al. [21] examined the impact of channel geometry in a counterflow microchannel heat exchanger numerically. For square, rectangular, circular, and trapezoidal channels, simulations were conducted for $100 < Re < 1000$. The best thermal and hydrodynamic performance was achieved by the circular channel, which was followed by the square channel. Correlations to estimate the heat exchanger's efficiency were also suggested. It was also discovered that, regardless of the channel's characteristics, performance indices declined as Re increased. Al-Nimr et al. [22] used mathematical modeling to analyze the performance parameters of a micro-heat exchanger parallel plate. The study uses a mathematical model that incorporates both a continuous approach and the potential of boundary slip. The impact of a variety of dimensionless factors was investigated. It was discovered that when the Knudsen number rises, so does the velocity slip near the walls. To examine thermohydraulic performance in a rectangular-shaped MCHX, Dang et al. [23] used numerical simulations and experiments. It was also discovered that when the heat capacity of water rises, the pumping power drops. Dang and Teng [24] conducted experimental research to study the performance parameters of two cases: one in which the input temperature was varied on the hot side, and the other in which the discharges were varied on the cold side. Rather than the substrate thickness, the influence of hydraulic diameter is more evident. The heat flow rises as the hydraulic diameter decreases, and the pressure drops as well. Based on their performance, three models were identified: one with the lowest pressure drop, another with the maximum heat flow, and the third with the highest performance index. In their work, Mathew and Hegab [25] experimentally evaluated their previously established thermal model. The thermal properties of a parallel flow MCHX subjected to external heat load were studied using this model. The thermal model was found to be independent of the microchannel profile. The current study is therefore aimed at the theoretical analysis of performance parameters of the MCHX. Results drawn based on the present work are drawn in this paper.

2. Geometry

The design of heat exchangers is dependent on various factors such as working fluid, temperature range, allowable pressure drop, flow conditions, and type of application. Table 1 outlines the geometrical parameters of the MCHX.

Table 1: The below table outlines the geometrical dimensions of the MCHX.

Parameters	Magnitude (Unit)/ Description
Side of square tube	2 mm
Shell diameter	80 mm
Pitch	4.911 mm
Baffle spacing	27.333 mm
Tube material	aluminium
Number of tubes	140
Length of tube	364 mm
Number of passes	2
Number of baffles	10
Pitch to diameter ratio	1.642
Baffle cut	25%

3. Materials and Methodology

An analytical technique is adopted for the estimation of performance parameters. Working conditions are given in Table 2.

Table 2: Working conditions for the current study.

Parameters	Magnitude/ Description
Tube side and shell side fluid	Water
Inflow temperature of hot fluid	50°C, 70°C, and 90°C
Inflow temperature of cold fluid	30°C
Re on tube side	1000
Re on shell side	1000-5000

Given below is a detailed analysis of the method involved in estimating the thermal and hydraulic characteristics of the MCHX.

Mean velocity of flow, $u_m = \frac{\dot{m}_h}{\rho_h \times A_{tp}} = \frac{\dot{m}_h}{\rho_h \times (\frac{\pi}{4} \times {d_i}^2 \times N_t)}$

Reynolds Number, $Re = \frac{\rho \times u_m \times D_h}{\mu_h}$

Prandtl Number, $Pr = \frac{\mu_h \times {C_p}_h}{k_h}$

Friction factor, $f(turbulent) = (1.58lnRe - 3.28)^{-2}$

Nusselt's Number, $Nu_t = \frac{(f/2)Re_b Pr_b}{1.07+12.7\left(\frac{f}{2}\right)^{\frac{1}{2}}({Pr_b}^{2/3}-1)}$

Heat Transfer Coefficient on the inner side, $h_i = \frac{Nu_t \times k_h}{d_i}$

Tube-Side Pressure drop, $\Delta p_t = 4f\frac{LN_p}{d_i}\rho\frac{u_m{}^2}{2}$

Mass Velocity, $G_s = \frac{m_c}{A_s}$

Shell Side Heat Transfer Coefficient, $h_o = h_{id}J_cJ_lJ_bJ_sJ_r$

Shell Side Pressure Drop,

$\Delta p_s = [(N_b - 1)\Delta p_{bi}R_b + N_b\Delta p_{wi}]R_l + 2\Delta p_{bi}\left(1 + \frac{N_{cw}}{N_c}\right)R_bR_s$

Overall Heat Transfer Coefficient for clean surface, $U_c = \frac{1}{\frac{d_o}{d_i \times h_i} + \frac{d_o ln\,(\frac{d_o}{d_i})}{2k} + \frac{1}{h_o}}$

Number of Transfer Units, $NTU = \frac{UA}{C_{min}}$

Heat Capacity Ratio, $C^* = \frac{C_{min}}{C_{max}}$

Effectiveness, $\varepsilon = \frac{2}{1+C^*+(1+C^{*2})^{1/2}\frac{1+exp\,[-NTU(1+C^{*2})^{1/2}]}{1-exp\,[-NTU(1+C^{*2})^{1/2}]}}$

Outlet Temperature of Hot Fluid, $T_{h_o} = T_{h_i} - \frac{(\varepsilon \times C_{min}) \times (T_{h_i} - T_{c_i})}{C_{max}}$

Outlet Temperature of Cold Fluid, $T_{c_o} = T_{c_i} + ((\frac{C_h}{C_c}) \times (T_{h_i} - T_{h_o}))$

Heat Recovery, $Q = C_h \times (T_{h_i} - T_{h_o})$

4. Results and Discussions

The following section presents a summary of the results which demonstrates the influence of flow rate and inflow temperature of the hot fluid on the characteristics of the MCHX. Calculations are done for three inlet temperatures of the hot fluid and the Re of the cold fluid is kept constant for two values – 1000 and 5000. Based on these assumptions the following results have been derived.

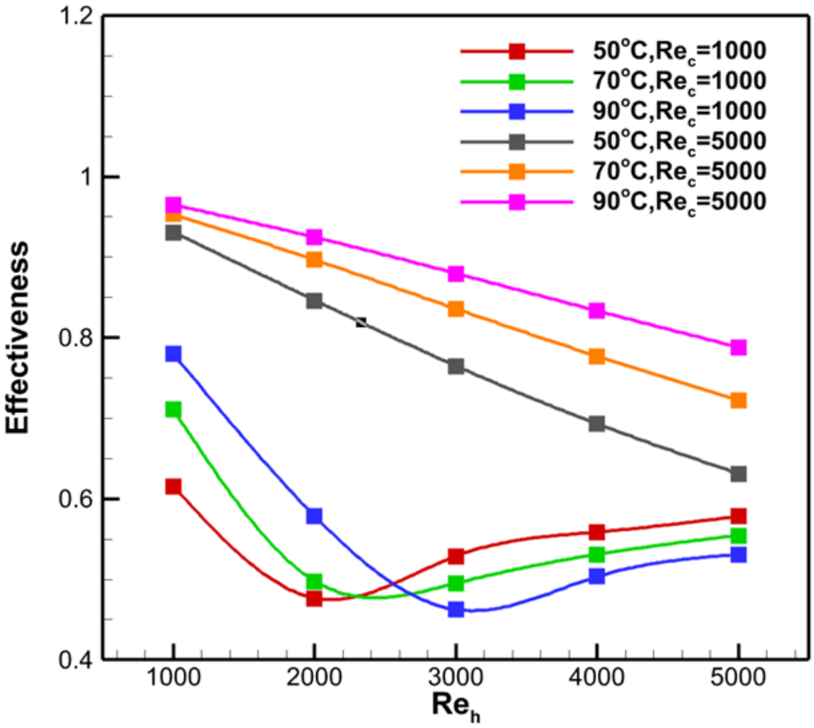

Figure 1: Effect of Re of hot fluid on the effectiveness of the heat exchanger at various inlet temperatures of the hot fluid.

Fig. 1 shows the variation in the effectiveness of the heat exchanger with variation in the discharge of the hot fluid. Analysis has been done for three inlet temperatures of the hot fluid with the Re of the cold being minimum (1000) and maximum (5000). Fig. 1 also shows the variation in effectiveness under different flow conditions. When the Re of the cold fluid is minimum i.e., 1000, we observe that there is a drop in effectiveness. This is because in the initial phase, the heat capacity of the cold fluid dominates the hot fluid and after the inflection point the heat capacity of the hot fluid dominates. A different trend is observed when the Re of the cold fluid is maximum. Initially, the Re of the hot fluid is maximum and then it starts to increase. So, at first the value of C^* is maximum and then follows a downward trend. We can also observe that for maximum effectiveness at a particular value of mass flow rate of the cold fluid is given by the highest inlet temperature of the hot fluid as shown in Fig. 1. The same can also be observed when the Re of the cold fluid is the lowest.

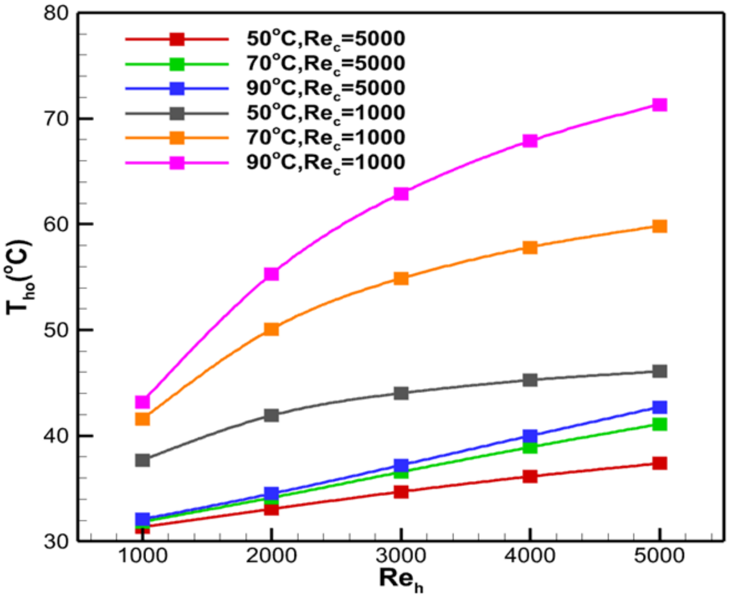

Figure 2: Effect of hot fluid flow on outlet temperature of the hot fluid at various inlet temperatures of the hot fluid

The variation in outlet temperature of the hot fluid for a different Re of the cold fluid (1000, 5000) for different inlet temperatures of the hot fluid is shown in Fig. 2. It can be observed that when the Re of the cold fluid is maximum, a higher outlet temperature of the hot fluid is observed for a higher inlet temperature of the hot fluid. Similar trends can be observed when the Re of the cold fluid is maximum. A higher outlet temperature of the hot fluid is observed at a higher inlet temperature of the hot fluid. Fig. 2 also demonstrates that at a particular Re of the hot fluid, the minimum outlet temperature of the hot fluid is observed when the Re of the cold fluid is maximum as the maximum heat that can be transferred increases proportionally.

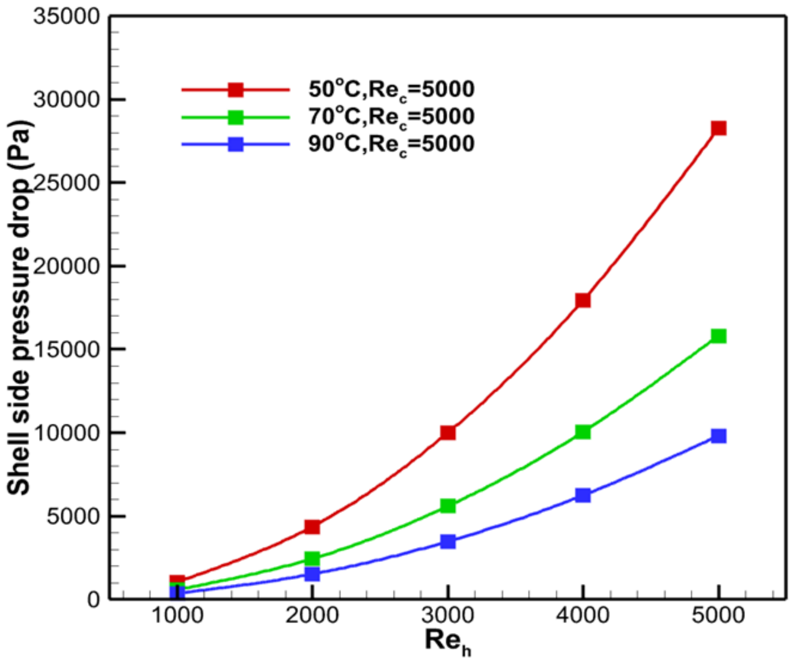

Figure 3: Effect of hot fluid flow on shell side pressure drop at various inlet temperatures of the hot fluid for the maximum flow rate of cold fluid.

The influence of the mass flow rate of the hot fluid on the shell side pressure drop at the minimum and maximum flow rate of the cold fluid is shown in Fig. 3 and Fig. 4. We can observe that with the rise in Re of the hot fluid the pumping power across the shell side augments. This trend can be observed for the lowest and highest Re of the cold fluid i.e., 1000 and 5000. At any particular value of Re of the hot fluid, the highest pressure drop is observed for the condition where the inlet temperature of the hot fluid is highest.

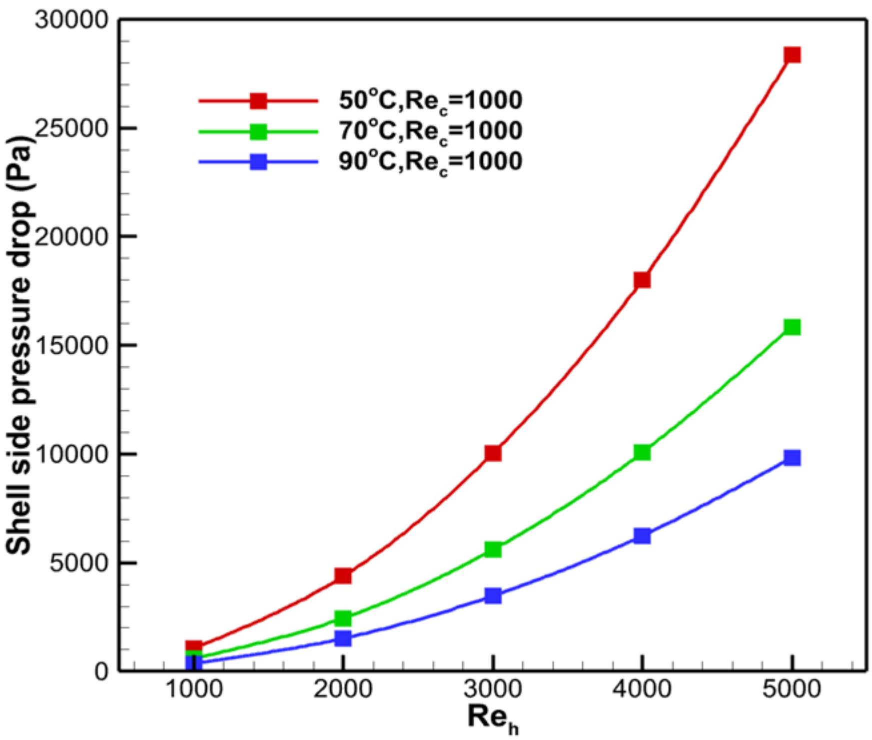

Figure 4: Effect of hot fluid flow on shell side pressure drop at various inlet temperatures of the hot fluid for the minimum flow rate of cold fluid.

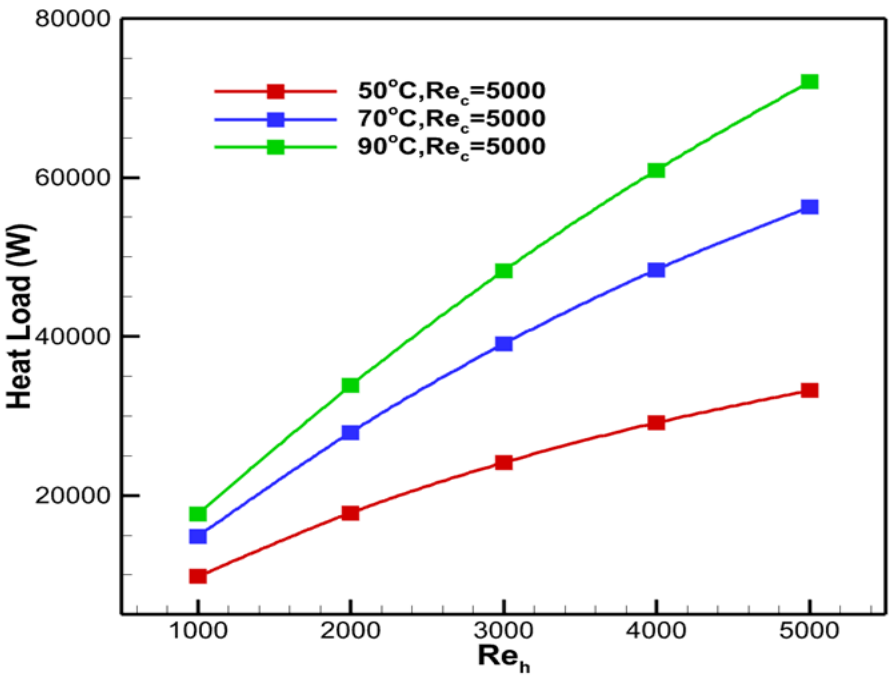

Figure.5: Effect of hot fluid flow on the heat recovery of the heat exchanger at various inlet temperatures of the hot fluid for the maximum flow rate of cold fluid.

Fig. 5 and Fig. 6 show the variation in heat load of the heat exchanger with the constant mass flow rate of the cold fluid (5000 and 1000) for varying inlet temperatures of the hot fluid. It can be observed that increasing the mass flow rate of the hot fluid leads to higher heat recovery. This is because heat recovery is directly proportional to the mass flow rate of the fluid. At a constant inlet temperature of the hot fluid, greater heat is recovered for the fluid with a higher Re of the cold fluid. At constant flow rates of the hot and cold fluid greater heat is recovered as the inlet temperature of the hot fluid increases. This is because the temperature difference across which the heat has to be transferred increases.

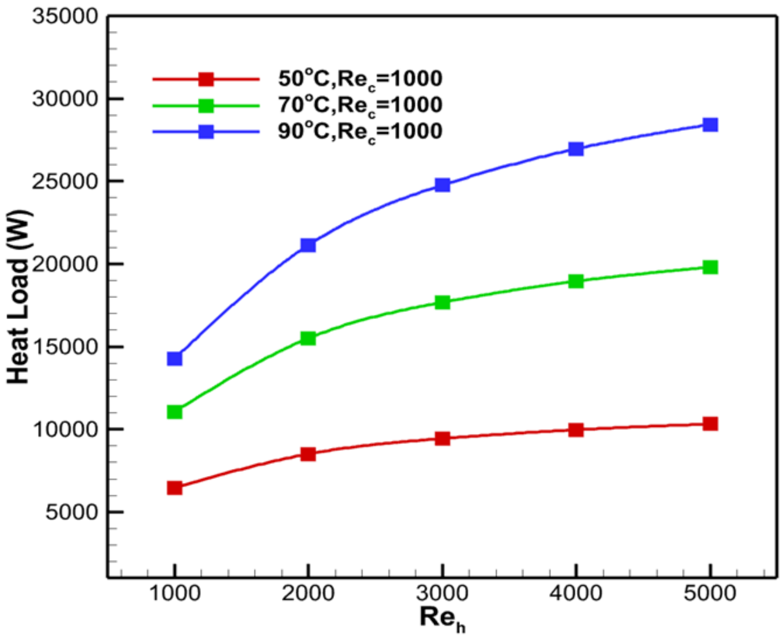

Figure 6: Effect of hot fluid flow on the heat recovery of the heat exchanger at various inlet temperatures of the hot fluid for the minimum flow rate of cold fluid.

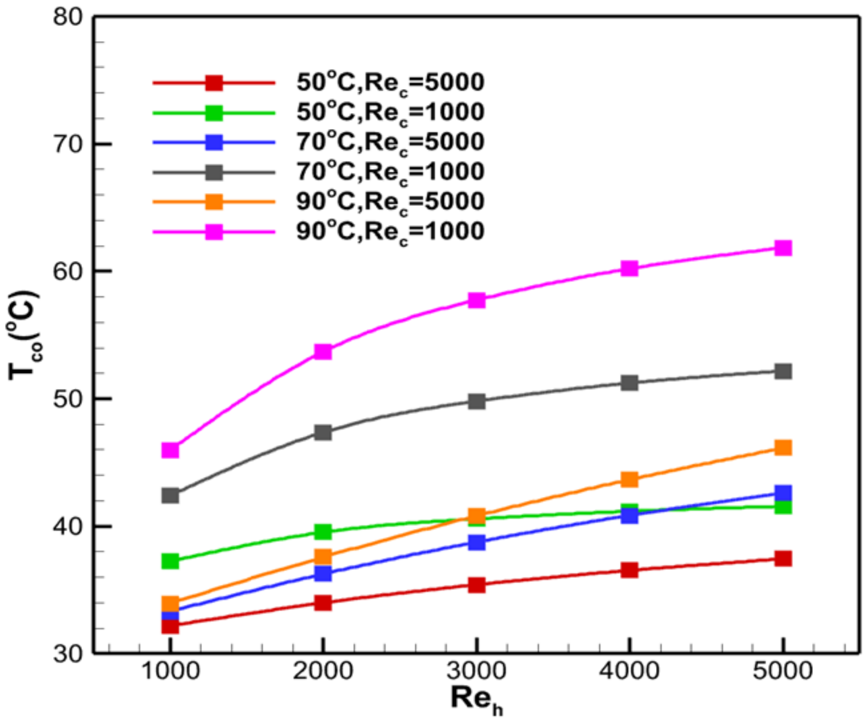

Figure 7: Effect of hot fluid flow on outlet temperature of the cold fluid at various inlet temperatures of the hot fluid

Fig. 7 shows the effect of the hot fluid flow rate on the outlet temperature of the cold fluid at different inlet temperatures of the hot fluid. It can be observed that at a particular value of the flow rate of the cold fluid, the outlet temperature of the cold fluid increases as the Re of the hot fluid increases. It can also be seen that at a constant value of the Re of the hot fluid and a constant value of the inlet temperature of the hot fluid, a higher outlet temperature of the cold fluid is seen for a lower value of the Re of the cold fluid. This is because as the mass flow rate of the cold fluid increases, the total heat that has to be transferred also increases and so does the temperature drop across which it has to be transferred. A lower value of the outlet temperature of the cold fluid indicates that the maximum possible heat that can be transferred has been achieved.

5. Conclusion

The present study focused on investigating the performance parameters of a microchannel heat exchanger at various inlet temperatures of the hot fluid at various flow rates of the hot and cold fluid. A theoretical model was developed for the analysis of these parameters. The effect of the flow rate of the hot fluid and the inlet temperature of the hot fluid on these parameters is investigated theoretically.

- For a lower value of flow rate of the cold fluid at any inlet temperature of the hot fluid, effectiveness shows a downward trend, reaches a minimum point, and then rises. At a higher value of flow rate of the cold fluid effectiveness continuously decreases from a maximum value with rising in the discharge of the hot fluid. Maximum effectiveness, in any case, is achieved with the maximum inlet temperature of the hot fluid. Just by increasing the flow rate of the cold fluid from Re = 1000 to Re = 5000 at 90°C, there is an increase in the effectiveness of the heat exchanger by about 22% at a constant Re of 1000 for the hot fluid.
- The final temperature of the hot fluid increases with increasing Re of the hot fluid. At a particular value of the Re of the hot fluid and at a particular inlet temperature of the hot fluid, a lower value of outlet temperature of the hot fluid can be seen for a higher value of the Re of the cold fluid.

- The graphs of shell side pressure drop indicate that as Re of the hot fluid increases, the pressure drop across the shell side also exponentially increases. At a constant value of the Re of the hot fluid, the higher-pressure drop is experienced for fluid with a lower inlet temperature of the hot fluid.
- With the rising inlet temperature of the hot fluid, the heat recovered by the MCHX also increases. At Re = 5000 for the hot fluid, the heat recovered by the hot fluid increases by about 69% for an increase in the inlet temperature of the hot fluid from 50°C to 70°C.
- With respect to the outlet temperature of the cold fluid, a higher value of outlet temperature is observed for a lower value of Re of the cold fluid. Outlet temperature also increases with an increase in Re of the hot fluid.

Disclosure Statement

On behalf of all the authors, the corresponding author states no conflict of interest.

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