Journal of Data Acquisition and Processing
ISSN: 1004-9037
https://sjcjycl.cn/
DOI: 10.5281/zenodo.98549781

NUMERICAL ANALYSIS OF CONVECTIVE HEAT TRANSFER IN MICROCHANNEL HEAT EXCHANGER

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ABSTRACT
The effects of property variations in single-phase forced convection heat transfer through circular Microchannels are investigated in the work. The thermal performance of 300 W microchannel heat exchanger is tested with microchannel diameters varying in the range from 0.5 mm to 1.5 mm. Thermal design of 300 W microchannel heat exchanger is carried out using effectiveness-NTU method. A computational analysis is then performed to confirm the design specifications of the heat exchanger. Fabrication of heat exchanger is carried out based on the design specifications and tested for parametric variations. Reduction in hydraulic diameter of microchannel enhances thermal performance of heat exchanger significantly. It is found that the overall size of the heat exchanger gets reduced by 50% for a 300 W heat exchanger. The study also suggests that microchannel of 1-mm diameter can be comfortably used for single-phase applications. Considering the high demand for heat transfer equipment, microchannel heat exchanger form one of the best suitable alternatives with respect to its weight and compact size. This increases the possibility of Microchannel Heat Exchanger's use in automobile and space applications.
Keywords: Thermal design, computational analysis, Microchannel heat exchanger, NTU, Parametric analysis

INTRODUCTION

The consumption of natural resources is a major concern for the society especially because of ever increasing population. This is a global issue which needs to be addressed by all the nations. One of the most important natural resources is the increasing use of energy. All sorts of activities in modern civilization consume some or the other form of energy. As of February 2023, India's installed renewable energy capacity (including large hydro) was over 174.53 Giga Watts, or roughly 42.5% of the nation's total capacity, up 396% over the previous 8.5 years, according to the World Energy Council Journal. Indian nation has witness the highest growth in renewable energy of 9.83% in the year 2022. Primary sources of energy have had the largest share of the energy market consumption and if this continues, consumption of fossil fuels will be peaking around 2030. The implication of this would be worldwide phase-out of coal in next 50 years at 2.5% per year, leaving huge reserves underground. One of the obvious choices under such circumstances would be effective utilization energy using a microchannel heat exchanger in specific is discussed in detail in the succeeding topics.

Over the past ten years, the growth of micro-fluidics, which is focused with micro-systems, has been particularly notable. Generally speaking, any devices with typical dimensions between 1 mm and 1 mm are referred to as micro-devices [1]. There are many different types of energy, including thermal energy, gravitational potential energy, solar energy, and nuclear energy. Engineering applications such as automotive applications, power generation, process industries, industrial plants, HVAC domains, and other areas require the conversion, transport, and utilisation of energy [1].

Efficient heat exchange devices (heat exchangers) play a vital role in many applications, which can save a significant amount of energy by improving heat transfer capabilities. This further reduces the size and cost of heat exchangers. Energy-efficient heat exchangers can reduce size and cost while saving a considerable amount of energy through improved power conversion efficiency. The development of small, affordable, and energy-efficient heat exchangers is the subject of research worldwide [2]. Modern automotive internal combustion engines generate a huge amount of heat. Consequently, metal temperatures around the combustion chamber can exceed 538°C and this heat needs to be dissipated efficiently. A third of the heat produced during combustion is transformed into energy that drives the car and its accessories. Through the exhaust system, a further third of the heat is released into the atmosphere. The cooling system, specifically the radiator, must remove the remaining third from the engine. This necessitates the use of smaller and lighter radiators with better performance and increased efficiency. High heat fluxes coupled with the need for smaller sizes make microchannel heat exchanger (MCHX) an obvious choice. Further studies have shown that single-phase flow remains unaffected for liquids as the hydraulic diameter of channels is reduced from 200 µm to 10 µm, however, this calls for a deeper understanding of MCHXs [3].

The applications of the microchannel are not limited to only mechanical engineering but have also been useful in the cooling of electronic components for different applications like automobile radiator or space applications. B Indulaxmi and G Madhu have done modeling and simulation of rectangular microchannels of size 10 mm x 4mm for thermal management of electronic chips using PI and PIC controls. Although the size of channel is more than microchannel, it improves the performance as compared other heat exchanger with large sized channels. It was concluded that the chip surface temperature dropped considerably even at low coolant velocities of 1.5 m/S in the microchannels.

To look into inconsistencies in previously published results for the pressure drop in microchannels, M. J. Kohl et al. conducted experiments. Channel hydraulic sizes ranging from 25 to 100 microns were used to build straight channel test sections with integrated pressure sensors. When all of the data from different researchers is taken together, the data appears to be scattered both above and below the analytical predictions. These problems were largely due to external pressure measurements and evaporation in the mass flow measuring devices.

It is evident that a lot of work has been done to study MCHXs and it is still in progress. Facts available from the open literature show that MCHXs are widely used in air-conditioners. The effect of fouling on microchannel heat sink performances has also been studied. The effect of influencing factors like channel shape, wall roughness, channel size ratio, fluid parameters, etc on MCHX performance has been done by some researchers. Different materials have also been tested like aluminum in place of conventional copper microchannel tubes. But an in-depth study of MCHXs for single-phase applications, like in the case of radiator, needs to be studied further in detail. The objective of current research is a parametric analysis of heat transfer and fluid flow in MCHX for the required heat duty.

PARAMETRIC VARIABLES

The present research work is associated with two competing objectives namely the parametric analysis and estimation of size. It is essential to consider these variables over the entire possible range of values to evaluate the performance of the heat exchanger as depicted in Table-1.

Table 1. Range for operating variables.

Sr. No	Parameter	Operating Range
1	Inlet temperature of fluid (T_{hi})	80ºC- 90ºC
2	Outlet temperature of fluid (T_{ho})	70ºC - 80ºC
3	Inlet temperature of air (T_{ci})	20ºC - 40ºC
4	Pressure of the system	1.1 bar -1.5 bar
5	Mass flow rate inside microchannel tubes ($\dot{m}$)	0.05kg/sec - 0.20kg/sec
6	Velocity fins side	10 m/s – 35 m/s

Increased effectiveness and a decrease in pressure result from adding more channels. The optimum thermal and hydraulic performance among different channel forms is provided by circular channels [10]. In light of this, the current study is conducted using the following geometry represented in Table 2.

Table 2. Geometry of MCHX

Sr. No	Geometric Parameter	Value/Range
1	Microchannel hydraulic diameter	0.5 – 1 mm
2	Type of fins on Airside	Plain fins
3	Fin spacing	0.5 mm

Methods

If the heat balance equation can be utilised to determine all four temperatures and the mass flow rates of both fluids, the LMTD Method, Kern Method, or Bell-Delaware Method can be comfortably employed for the design of Hx. The LMTD approach cannot be applied in the heat exchanger under study since only the entrance temperatures of both streams and the mass flow rate of just one stream are available. Because of this, the Effectiveness-NTU (Number of Transfer Units) approach is employed. The detailed methodology has been described in Figure 1. The preliminary size of the heat exchanger would be the size determined using the Effectiveness-NTU technique. Through computational analysis, the estimated size can be confirmed, and after confirmation, resizing can be done.

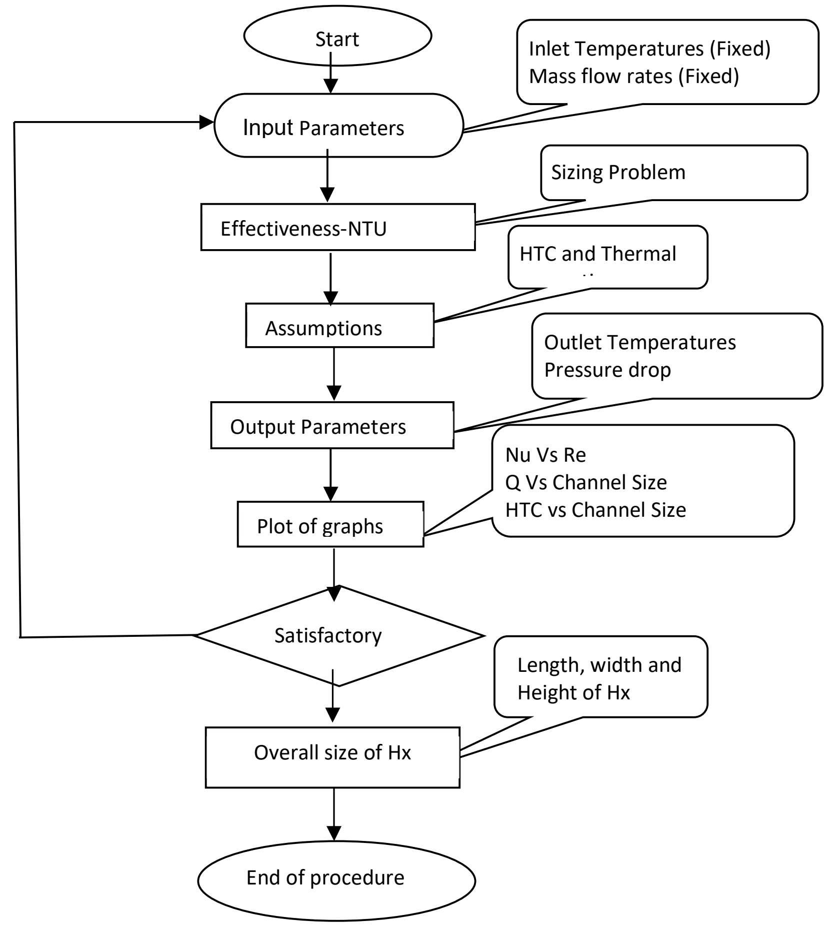


Figure 1. The methodology of mathematical modelling.

The analytical process that was used to design the heat exchanger is predicated on a few fundamental premises, including the denial of heat transfer between adjacent microchannels, the assumption that air only absorbs heat from coolant flowing through microchannels, and the uniformity of coolant liquid pressure distribution on the microchannel side. A system of equations utilising energy balance and e-NTU relations is employed based on the aforementioned presumptions.

The projected output operating parameters include pressure drop and temperature, total heat transfer rate, air-side outlet air temperature, and microchannel operating pressure and temperature. The analytical process does not alter the geometrical dimensions in order to support the comparison.

Table 3. The Impact of Hydraulic Diameter on Variables

Case	**d_i (m)**	**Ui (W/m^2K)**	**Re**	**Nu**	**Pump (W)**
1	0.00025	152200	8933	64	248962
2	0.0005	75200	4467	36	4789
3	0.00075	49700	2978	25	480
4	0.001	36900	2233	20	94
5	0.00125	29300	1787	17	27

6	0.002	17900	1117	12	2
7	0.0025	14100	893	10	0.55119

Based on the information in Table 3, a graph has been made to illustrate how changes in channel diameter effect other factors including overall HTC, Re, Nu, and pumping power as depicted in Figure 2. The plot of HTC and pumping power have been integrated in one graph in order to better understand the relative influence of channel size on Hx performance.

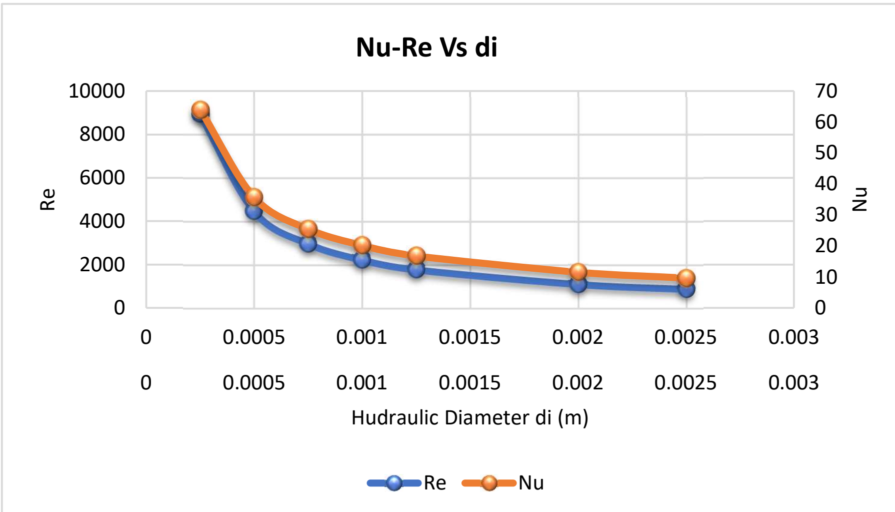


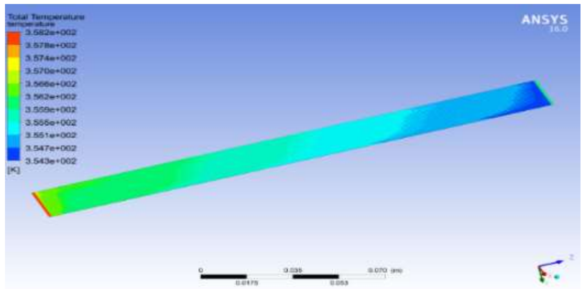
Figure 2: Graph of Re-Nu versus Hydraulic Diameter

- It is observed that the HTC and subsequently overall HTC decreases with an increase in microchannel diameter which is attributed to an increase in flow velocity causing a proportionate decrease in Re number.
- There is also a reduction in pumping power required with an increase in microchannel diameter. This reduction in pumping power requirement is almost proportionate to the decrease in overall HTC.
- However, reduction in pumping power requirement cannot compensate for a reduction in overall HTC and therefore a golden mean is required to be found out.
- It is observed that the critical value of the microchannel diameter is around 0.001m. Further increase in diameter causes a minimal decrease in pumping power whereas the decreasing rate is still considered and therefore further reduction in overall HTC cannot be compensated by overall HTC.
- To reduce any chance of pressure variation at inlet because of its size, it has been decided to have two passes of microchannels.

NUMERICAL ANALYSIS

The purpose of using computational analysis here is not to validate the analytical results. However, it is worth testing to see whether both analytical and simulation are well agreeing with each other. Analytical solutions were possible using simplifying assumptions that may not reflect the facts, whereas the computational solution makes it possible to obtain realistic solutions without the need for simplifying assumptions. Additionally, it becomes easier to make a parametric variation using computational analysis.

Results are obtained for the first pass and second pass separately, the temperature drop in the first pass is found to be of 5^0C at inlet temp of 85^0C and The temperature drop in the second pass is found to be of 4^0C at inlet temp of 80^0C. The total temperature drop is found to be 9^0C considering both passes of fluids. This can be understood with ease from Figure 4. Thus the overall temperature drop of fluid would be 9^0C when the inlet temperature of the fluid is 85^0C indicating that the temperature at the fluid outlet would be 76^0C.



It is observed that the value of the heat transfer coefficient remains almost constant over the entire length of microchannels. Indicative of the fact that the properties of air do not change over the entire length and that fluid properties also remain almost constant throughout their flow in microchannels due to temperature drop of 9^0C only. The results of the computational analysis are finally tabulated as follows. It is noteworthy here that the value of temperature at the outlet as expected. The mass flow rate indicated in the table is for a single microchannel. Thus the total value of mass flow rate for 20 microchannels in a flat tube and 13 number of flat tubes is found to be 1.3 kg/S. It also noteworthy that pressure drop is within expected limits.

COMPARISON OF RESULTS

To confirm the size of MCHX needed for its fabrication, it is worth comparing the results of computational analysis with the results of the preliminary estimation of size using the Effectiveness-NTU method. Table 4 shows a quantitative comparison between the results of the preliminary estimation of size and the results obtained by computational analysis.

Table 4. Comparison of Results

Sr. No.	Variable	Analytical	Computational
1	Mass flow rate	0.005 kg/S	0.005kg/S
2	Inlet temp	85^0C	85^0C
3	Outlet Temp	74^0C	76^0C
4	Pressure drop	1.77 atm	1.2 atm
5	Pumping power	0.6085	0.6755
6	HTC over	39473	36000 W/m^{2}K

It is discovered that the outcomes of the computational analysis are in perfect accord with those of the analytical technique. It is important to note that the goal of using computational analysis is not to validate the conclusions reached through the analytical process. Computational analysis is used to verify that the conclusions reached throughout the analytical process can be trusted. The final size of the heat exchanger will be determined by the computational analysis' second goal, which is to complete the thermal and mechanical design of the MCHX using the results. This will allow the heat exchanger to be manufactured and used for more research. The following subject will provide further information on the specific steps involved in the thermal design of MCHX.

RESIZING OF MCHX

The results of the computational analysis are found to be in full agreement with the results obtained by the analytical procedure. This forms a strong base to go ahead with the thermal design of MCHX. The rerating of MCHX can now be done using Kern Method.

The properties of the channel-side fluid i.e. ethylene glycol are determined at the bulk-mean temperature of 55.2^0C. Density ρ_h = 1113kg/m^3, Specific heat Cp = 2200 kJ/KgK, Dynamic viscosity µh = 0.000198 NS/m2, Thermal conductivity k = 0.615098 W/m^{2}K, and Prandtl Number Pr = 5.42. The hot fluid mass flow rate is as assumed in the analytical procedure and it is 0.8 kg/S. Since there are 20 microchannels in a flat tube and the number of flat tubes is 26, the total number of microchannels would be 520 and thus mass flow rate through a single microchannel is obtained to be 0.001538 kg/S. Considering 1 mm dia microchannel, the velocity of the fluid is obtained as 1.759 m/S. Using simple formula Reynold's Number, it is found to be 9889, thus indicating that the flow is in transition mode. Thus using following applicable correlation called Prandtl's correlation for Nusselt Number [15]

$$\mathrm{Nu} = \frac{\left(\frac{f}{2}\right)(\mathrm{Re.Pr})}{1+8.7\left(\frac{f}{2}\right)^{\frac{1}{2}}(\mathrm{Pr}-1)} \qquad (14)$$

RESULTS AND DISCUSSION

A graphical representation has been created to show how changes in channel diameter affect other variables including overall HTC, Re, Nu, and pumping power. To comprehend the relative impact of channel size on Hx performance, a plot of HTC and pumping power have been combined in the same graph. It has been found that as microchannel diameter is increased, the HTC and subsequently the overall HTC drop. This is related to an increase in flow velocity, which results in a commensurate fall in Re number.

It has been found that the microchannel diameter's critical value is approximately 0.001 m. While the decreasing rate is still taken into account, a further increase in diameter only slightly reduces pumping power, so a further decrease in overall HTC cannot be made up for by overall HTC. The amount of pumping power needed likewise decreases as microchannel diameter rises. The decrease in total HTC and the reduction in pumping power requirements are virtually proportional.

The Nusselt Number is discovered to be 61.9 using the aforementioned correlation. Given that the microchannel is only 1 mm in size, the number indicates an extremely high heat transfer coefficient. The figure obtained for the hot-fluid side's HTC 38 kW/m^{2}K is in excellent agreement with the result of computer analysis. Convective mode is discovered to have a heat transmission rate of 6.8 kW, which is significantly higher than the necessary heat duty of 3 kW. Additionally, it is important to note that the required pumping power is only 16 W, which is again well within operating limits. The outcomes show that MCHX can be manufactured using the findings from computational and analytical methods.

CONCLUSIONS

The results obtained from the computation analysis closely agree with the results of the analytical procedure, the agreement is within ±5%. Thus creating a strong base for the thermal and hydraulic design. The rerating done using thermal design exactly matches with the desired process conditions. The face of existing heat exchanger is 150 x 200 mm, whereas with the use of microchannels, it has been reduced to 100 x 200 mm. When microchannels are employed in place of conventional flat tubes then for the same rates of heat transfer, the size of heat exchanger is reduced by 40%.

LIST OF ABBREVIATIONS

Hx		Heat Exchanger
MCHX		Microchannel Heat Exchanger
LMTD		Log Mean Temperature Difference
k	[W/mK]	Thermal Conductivity
HTC	[W/m2K]	Heat Transfer Coefficient
U	[W/m2K]	Overall Heat Transfer Coefficient
ṁ	[kg/s]	Mass flow rate
D_h	[m]	Hydraulic Diameter
T	[C]	Temperature
Th	[C]	Temperature of hot fluid
Tc	[C]	Temperature of cold fluid
ΔT		Temperature drop
ΔP		Pressure drop
Nu		Nusselt Number
Pr		Prandtl Number
Re		Reyolds Number
f		Fanning Friction Factor
C_p		Specific Heat
ρ_h		Density of hot fluid

Declaration

Availability of data and material

Most of the data generated or analysed during this study are included in this published article. The datasets analyzed during the current study and not published in the article are available from the corresponding author on reasonable request.

Competing interests

The authors declare that they have no competing interests.

Funding

The author(s) received no financial support for the research, authorship, and/or publication of this article.

Authors' contributions

Conceptualization, data aggregation, research, technique, original draft writing, writing review, and editing were carried out by SHB. MML oversaw and handled the article writing tasks. Additionally, MML provided the tools required to finish the article. The final manuscript was read and approved by all writers.

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Acknowledgement

The authors would like to thank the MIT World Peace University Pune's School of Mechanical Engineering for its support. We also like to express our sincere gratitude to the Dean of the Faculty of Engineering and Technology for his invaluable advice and help with this initiative. Additionally, we would like to express our gratitude to the university for providing the tools, facilities, and technical assistance required for this study.

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