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CONJUGATE HEAT TRANSFER ANALYSIS IN A CRYOGENIC MICROCHANNEL HEAT EXCHANGER

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Abstract - *Printed Circuit Heat Exchanger is highly integrated plate type compact heat exchangers in cryogenic application. The area density ($\beta = A_{HT} / V$) i.e. heat transfer area per unit volume of a printed circuit heat exchanger is greater 2500m²/m³ and hydraulic diameter is less than 1mm.A Counter flow rectangular micro-channel (40 mm × 1.6 mm × 1.2 mm) printed circuit heat exchanger model is designed and simulated using commercial ANSYS FLUENT package. Because of length and viscous nature the fluid flow through the channel is laminar and thermally fully developed. The performance is investigated numerically with helium at cryogenic temperature. The performance is affected by axial conduction at low Reynolds number. The Nusselt number (Nu), heat flux, dimensionless form of the fluid temperature and wall temperature, effectiveness, axial conduction are determined for different Reynolds number Re less than 100 with varying material i.e. wall to fluid thermal conductivity ratio for hot channel and hot channel. The dimensionless axial conduction parameter (λ) is calculated by using Kroeger'equation.*

Key Words: **PCHE; Micro-channel; Effectiveness; Reynolds number; Nusselt**

1. INTRODUCTION

Printed Circuit Heat Exchanger (PCHE) is a plate type compact heat exchanger with micro-channels. PCHE was invented in 1980 in Australia, and subsequently incorporated into refrigerator in 1985, by Heatric (UK). Printed Circuit Heat Exchanger manufacturing process is followed by chemical etching and chemical bonding. Micro-channels for fluid flow are constructed by chemical etching on the metal plates according to different configuration then plates are stacked alternately and assembled by diffusion bonding.

Due to compact size, high efficiency, large heat transfer area PCHE are used in cryogenic refrigeration and liquefaction systems.

Kim et al. [1] performed a numerical analysis on a three dimensional zigzag channel PCHE model to investigate characteristics of heat transfer and pressure drop of a supercritical CO_2 .The validation was compared with the experimental data and found that numerical data with 10% variation. A new PCHE airfoil model was designed. On comparison found that area density was same with the zig-zag channel and in case of pressure drop it was reduced to one-twentieth of the zig-zag channel.

Tsuzuki et al. [2] had done a 3D simulation of an s-shape and sine curve channel PCHE recuperators of CO_2. It is observed that s-shape and sine shape channel had one-fifth pressure drop as compared to the zig-zag channel.

Mylavarapu et al. [3] had done an investigation on various high temperatures up to 900°C and pressure up to 3 Mpa. High-temperature helium test facility was for validation. Result found that alloy 617 and 230 are most leading material for high temperature PCHE. Two PCHE are fabricated from alloy 617 each having ten hot plates and ten cold plates with 12 flow channels in each plate. The design and fabrication of PCHE are focused in the CFD study.

Kim et al. [4] developed a PCHE having longitudinal corrugation flow channels to evaluate the hydraulic performance in low Reynolds number region (Re ≤ 150).the validation is compared with the measured data obtained experimental data.

Kim et al. [5] investigated both numerically and experimentally the hydraulic and thermal performance of a PCHE in a helium water condition with vertical and horizontal settings. Numerical simulation was done at high Reynolds number of 2500. The results shown that the fanning factor of helium and water range was less than 0.97% and 0.66%.

Baek et al. [6] had done the experimental investigation to determine the thermal and hydraulic performance of a PCHE with multiple corrugated longitudinal micro-channels working under cryogenic temperature. Thermal performance is affected by axial conduction in low Reynolds number. To increase the effectiveness, the design of PCHE was modified to reduce the axial conduction.

2. METHODOLOGY

Effectiveness is a parameter used to estimate the thermal performance of heat exchanger. It is defined as ratio of actual heat transfer to maximum possible heat transfer

$$\epsilon = \frac{Q_{actual}}{Q_{max}} \quad (1)$$

$$\varepsilon = \frac{C_{Phot}\left(T_{hot,in} - T_{hot,out}\right)}{C_{Pmin}\left(T_{hot,in} - T_{cold,out}\right)} \quad (2)$$

The mass flow rates are the same for inlet and outlet. The heat capacity of fluid is assumed to be constant.

According to Kroeger's analytic solution the dimensionless axial conduction parameter λ is defined with heat exchanger length (L), cross sectional area of wall (Aw), thermal conductivity of wall (kw) and heat capacity rate (C) as the following equation

$$\lambda = \frac{K_w A_w}{L C_{min}} \quad (3)$$

Ineffectiveness is the parameter used to estimate the thermal performance (1-ε)

$$1 - \epsilon = \frac{1}{1 + NTU\left\{\frac{1+\lambda[\lambda NTU/(1+\lambda NTU)]^{1/2}}{1+\lambda NTU}\right\}} \quad (4)$$

3 PROBLEM DESCRIPTIONS

Numerical Analysis is performed using the Commercial Computational fluid dynamics CFD ANSYS FLUENT package to determine the governing equation for 3-D steady laminar flow and heat transfer in rectangular micro-channel printed circuit heat exchanger. The computational domain consists of three sub domain i.e. (a) the hot channels, (b) cold channels and (c) steel separators. The convective heat transfer in the cold and hot channels and heat conduction in the separators are calculated in the domains. The numerical models are constructed through grid dependency test.

The thermo-physical properties of the materials are independent of temperature. Metals from k_{sf} = 141.58 to 5061.5 are taken for analysis. Helium is taken as cryogenic working fluid .The thermo-physical properties i.e density, thermal conductivity and viscosity of Helium varies with temperature.

Assumptions Used In PCHE Modeling

- A steady-state condition is assumed.
- Assume to be uniform fluid flow.
- The outer surface of the metal plates that makes the PCHE are assumes to be adiabatic.
- Laminar flow Re≤100
- Pressure drop is negligible

4. NUMERICAL ANALYSIS

4.1. DIMENSIONS

Counter flow micro-channel heat exchanger = (40×1.6×1.2) mm^3

Fluid flow channels both hot and cold= (0.4×0.2) mm^2

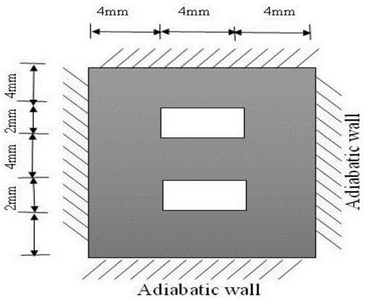

Fig.1. Numerical model of 3D PCHE

TABLE 1. Boundary conditions

Boundary Condition	Laminar flow
Velocity inlet	Velocity magnitude and Temperature
Pressure outlet	Gauge Pressure
Wall(outside)	Heat flux=0
Wall (fluid/solid interface)	Thermally coupled wall condition

TABLE 2. Velocity at Reynolds number 50

Properties	Temperature(K)	Mass flow rate (Kg/sec)	Velocity (m/sec)
$T_{hot,in}$	200	1.93854E-07	9.961661251
$T_{cold,in}$	100	1.0447E-07	2.696491757

TABLE 3. Velocity at Reynolds number 80

Properties	Temperature (K)	Mass flow rate(kg/sec)	Velocity (m/sec)
$T_{hot,in}$	200	3.10166E-07	15.938658
$T_{cold,in}$	100	1.88045E-07	4.314386811

4. RESULTS AND DISCUSSION

Figure.2 (a) represents in between 0.6 to 0.8 there is a transition point and between 0 to 0.7 temperature decreases according to k_{sf} and the deflection is more for low k_{sf}. In between 0.7 to 1.0 temperature reduces for high k_{sf}. Figure2.(b) represents the temperature of fluid varies from 1.0 to 0. Temperature increases in between 0 to 0.7 with the decrease of k_{sf}. In between 0.7 to 1.0 low k_{sf} has high temperature whereas high k_{sf} has low temperature variation

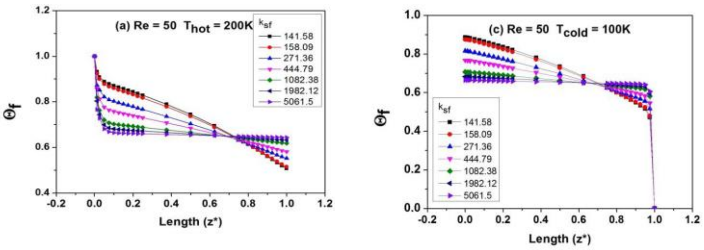

Fig.2. Variation of dimensionless fluid temperature along the length of micro channel heat exchanger as a function of Re and ksf, (a) hot channel ,(b) cold channel

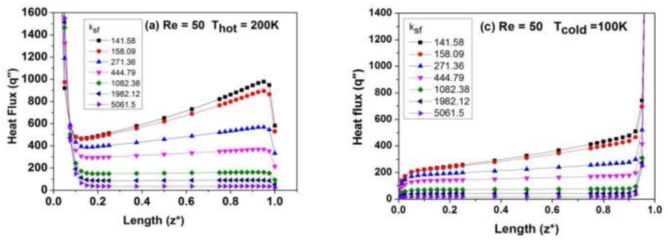

Fig. 3. Variation of heat flux along the length of micro channel heat exchanger as a function of Re and ksf, (a) hot channel (b) cold channel.

Fig. 3 (a) shows along the length of micro-channel heat exchanger from 0.1. Conjugate heat transfer is high for low k_{sf} and decreases with the increase of k_{sf} for Re = 50 in the hot channel. Fig.3 (b) shows conjugate heat flux is high at the inlet of the cold channel and decreases along the length towards the outlet. Low k_{sf} has high heat flux and decreases with the increase of k_{sf}.

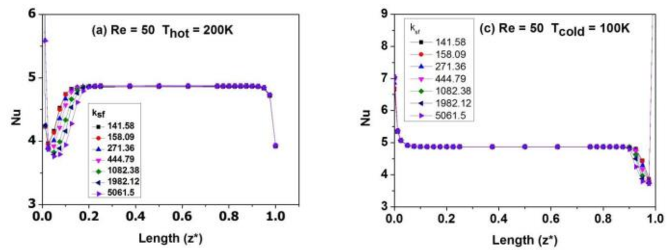

Fig.4. Variation of Nusselt number along the length of micro channel heat

exchanger as a function of Re and ksf, (a) hot channel (b) cold channel.

In the hot channel at Re = 50 as shown in Figure 4 (a) The Nusselt number is fluctuating in the developing zone from 0.1 to nearly 0.2 with a variation of $k_{sf.}$ and in developed zone remains constant In the cold channel as shown in Figure 4 (b) at Re = 50, Nusselt number is fluctuating in the developing zone with the variation of k_{sf}. and in developed zone remains constant

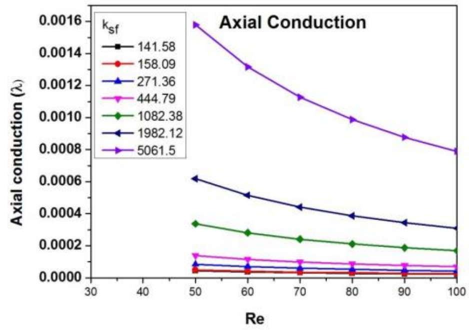

Fig. 5. Variation of axial conduction of the micro channel heat exchanger with a variation of Re as a function ksf.

Figure5. Shows the effect of axial conduction varying with Reynolds number. Axial conduction decreases with the increase in Reynolds number. The variation of axial conduction is directly proportional to k_{sf}.

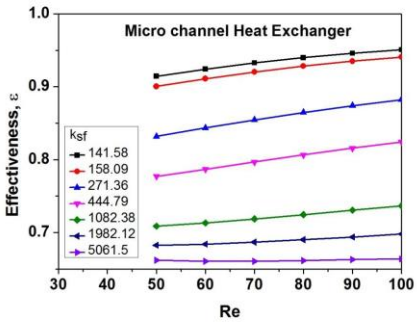

Fig. 6. Thermal performance of micro channel heat exchanger (ε) with variation of Re as a function of k_{sf}

Low k_{sf} has high effectiveness as shown in Fig.6.For high k_{sf} the effectiveness is low. As the Reynolds number increases the effectiveness of the micro-channel heat exchanger increases.

5. CONCLUSION

At low Reynolds number the effectiveness of the heat exchanger is low due to the effect of axial conduction. At high Reynolds number axial conduction diminishes so effectiveness increases. Low k_{sf} has highest effectiveness whereas high k_{sf} has highest axial conduction.

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