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OPTIMUM GEOMETRY OF MEMS HEAT EXCHANGER FOR HEAT TRANSFER ENHANCEMENT

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ABSTRACT
The study is based on an analysis of MEMS heat exchanger of three different geometries: wavy, triangular and rectangular using water, air and different types of refrigerants as test fluids. The problem is solved using finite element method. The aim of this analysis is to evaluate the performance of MEMS heat exchanger for different geometries and to obtain an optimum design for better heat transfer enhancement. It is apparent from this work that rectangular surface heat exchanger shows the best performance for heat transfer enhancement technique in comparison to other geometries. Moreover, it is also easy to manufacture. Therefore, the rectangular surface may be used instead of other configurations of heat transfer surfaces. In this analysis, emphasis is given on increasing heat transfer area of MEMS heat exchanger by reducing the pitch which shows that 0.475 mm is the optimum pitch as further decrease of pitch length does not have any significant effect on the effectiveness and heat transfer.

Keywords: model, MEMS heat exchanger, pitch, outlet mean temperature.

INTRODUCTION
The introduction of MEMS heat exchanger has established a new era in the history of heat transfer in micro scale systems. The use of MEMS-fabricated micro-channels for heat absorption and removal from microelectronic devices is thermally efficient due to the large surface area available for heat exchange. Heat exchangers are mostly used in processes to remove heat when convective cooling from fans and fins are not enough. Having the advantage of superior heat exchange properties, compact design at industrial throughput and low inner volume, the micro heat exchangers are becoming increasingly popular in the fields of chemical, electronic and aerospace industries.

Fast development of microelectronic circuits led the modern researchers to carry out researches in the fluid flow and heat transfer characteristics in micro channels. In 1998, Yuen and Hsu [1] designed and fabricated a micro-channel heat exchanger, the important basic component of a microminiature Joule-Thomson cryogenic refrigerator. Preliminary experiments showed that the heat exchanger was mechanically robust and had excellent thermal performance characteristics. Hardt *et al.,* [2] found that channels equipped with micro fins allowed for a rapid exchange of heat. Okabe *et al.,* [3] built up the optimization flow for Micro Heat Exchanger with a commercial multi-physics solver and pointed out the necessary functionalities of commercial solvers to be used in the field of evolutionary computation. Hossain and Islam [4] solved two-dimensional Navier-Stokes and energy equations numerically for unsteady laminar flow in periodic wavy (sinusoidal and triangular) channels. The flow in the channels had been observed to be steady up to a critical Reynolds number. Chandratilleke *et al.,* [5] developed simulation models for the micro-heat exchanger to assess the influence of critical system variables on heat transfer rates and pumping power, and to ascertain optimal parametric combinations for thermoelectric applications. McCandless *et al.,* [6] and Li [7] showed cooling performance of micro-machined MEMS Heat Exchanger in their work. Morimoto *et al.,* [8] made a series of numerical simulation of the flow and heat transfer in modeled counter-flow heat exchangers with oblique wavy walls for optimal shape design of recuperators. Mébrouk *et al.,* [9] reported a numerical investigation of natural convection and fluid flow in a horizontal wavy enclosed. Eiamsa-ard and Promvonge [10] performed an experimental study on a helical-tape insert in a circular wavy-surfaced tube using hot air as the working fluid. Davis *et al.,* [11] discussed the major design issues for commercial CPU coolers in their study.

Although many research works regarding micro heat exchangers are available and successful applications have been reported, the variation of performance with different geometries had rarely been carried out. In the present research, a two dimensional model is developed for investigation of MEMS heat exchanger with various surface geometries.

NUMERICAL MODEL
The heat exchanger is modeled by stacking several square plates on top of each other whilst leaving a gap of 1 mm in between. The length of the plates is maintained as 9.5 mm. Water, air and different types of refrigerants (ammonia, Freon 113, Freon 11) are used as coolant to transfer heat which circulates through gaps between the walls. Walls are considered as isothermal surfaces. Figure-1 depicts the cross section of the fluid flow field which is considered for the present analysis. It is assumed that fluid enters through the surfaces with a definite temperature. Heat is removed from the heated wall and transferred to the coolant. At the outlet, velocity weighted mean temperature is obtained. The analysis is done by varying the pitch for each geometry.

Pitch can be defined as the distance between two adjacent peaks and it is calculated as linear length of the

ARPN Journal of Engineering and Applied Sciences

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plate divided by total number of peaks. The operating temperature range spans from 48 °C to 90 °C (321 K to 363 K). At this temperature range, water remains in liquid phase. Other working fluids such as ammonia, Freon 113 and Freon 11 remain in vapor phase.

Figure-2 shows the temperature distribution for wavy surface MEMS heat exchanger with water as working fluid. The low temperature region decreases as the surface area increases and the pitch length decreases from 0.95 to 0.11875 mm. It happens because, as the surface area increases, convection of heat is augmented and better heat is transferred from the wall to the fluid. Different geometries of MEMS heat exchangers investigated in this research are also shown in Figures 3 to 5. Outlet mean temperature and effectiveness increase as the pitch is reduced. This is because of the fact that the heat transfer area increases when pitch is reduced and fluid can absorb more heat from the surface.

GOVERNING EQUATIONS AND BOUNDARY CONDITIONS

The governing equations in this system are the incompressible Navier-Stokes equation (Eq. 1) and continuity equation (Eq. 2) accounting for the motion of the fluid

$$\rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=\rho g_x-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \quad (1)$$

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \quad (2)$$

and the convection and conduction equation (Eq. 3), without any heat sources, for the energy transport within the fluid

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}=\frac{\rho c_p}{k}\frac{\partial T}{\partial t} \quad (3)$$

For two dimensional analyses, Equations 1, 2 and 3 can be simplified in the following way,

$$\rho\left(\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=\rho g_x-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \quad (4)$$

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \quad (5)$$

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}=\frac{\rho c_p}{k}\frac{\partial T}{\partial t} \quad (6)$$

The results are obtained using FEMLAB by solving equations 4, 5 and 6 simultaneously in finite element method.

At the inlet, a parabolic velocity profile is specified. For fully developed laminar flow, velocity profile is shown in Eq. 7. Temperature is given as known constant value at the inlet.

$$u=\frac{3}{2}U_{av}\left[1-\left(\frac{x}{B}\right)^2\right] \quad (7)$$

Here u is the local velocity, U_{av} is the average velocity and B is one-half the distance of two parallel plates.

The walls have no-slip conditions for velocity and a specified temperature condition for energy balance:

$u = 0$

$T = T_w$

Finally, at the outlet or outflow boundary, normal flow (perpendicular flow) velocity conditions and convective flux heat transfer are assumed:

$u_t = 0$

$p = p_a$

$-k\nabla T = 0$ at normal flow direction

The geometry has been meshed with triangular elements. Size of the elements at the inlet and outlet boundaries is chosen to be finer than those of other boundaries. Two dimensional meshing of present model is shown in Figure-6. To investigate the influence of mesh size on the simulation, several trials have been made taking different numbers of elements. Assessments are conducted to obtain a fixed value of temperature. When the magnitude of outlet mean temperature becomes constant, the trial is stopped. The lowest value of the number of elements for this result has been considered as sufficient to evaluate heat transfer parameters correctly. For example, 9027 elements are taken for the wavy surface of 0.95 mm pitch to evaluate the heat transfer data correctly.

EQUATIONS FOR CALCULATING THE HEAT TRANSFER COEFFICIENT

Fluid is passed through the gap between the plates of MEMS heat exchanger. The mode of heat transfer is convection and steady state condition is assumed. So, heat transfer in the fluid can be expressed as

$$Q_{fluid}=\dot{m}\,C_p\,(T_o - T_i)$$

Where T_o is outlet mean temperature.

Again,

$$Q_{fluid}=Q_{conv}$$

$$Q_{conv}=hA\left(T_w-T_m\right)$$

Where,

$$T_m=\frac{\left(T_o+T_i\right)}{2}$$

T_i = inlet temperature of fluid = 321 K

VOL. 5, NO. 5, MAY 2010 ISSN 1819-6608
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T_w = wall temperature which is assumed to be 363 K (90 °C).
This value is chosen as below as the boiling point of water is 100 °C. At boiling temperature, water will start changing its phase and bubbles may begin to form which is undesirable for this study.

The average heat transfer coefficient, h and the mean Nusselt number, Nu are estimated as follows:

$$h = \frac{\dot{m} C_p (T_o - T_i)}{A(T_w - T_m)}$$

i.e., $$h = \frac{\rho U_{av} A_1 C_p (T_o - T_i)}{A_2 (T_w - T_m)}$$

and

$$Nu = \frac{hD_h}{k}$$

Where D_h is the hydraulic diameter and is equal to (4 × Area)/Perimeter
Reynolds number is calculated as

$$Re = \frac{\rho U_{av} D_h}{\mu}$$

U_{av} is the mean velocity of fluid.
A_1 denotes area of fluid entry and A_2 represents the surface area of convection.

RESULTS AND DISCUSSIONS

Validation of the analysis

To justify the process, different outlet mean temperature of air has been attained in a wavy surface MEMS heat exchanger for a wide range of Reynolds number. Heat transfer coefficient and Nusselt numbers have been calculated. The values of Nusselt number is plotted against Reynolds number for comparison with other results available in the literature. Similar type of analysis done by Hardt *et al.,* [2] showed that the sine-shaped walls affect the flow pattern in such a way that a substantial heat transfer enhancement was achieved, reflected in Nusselt numbers of up to 32. Hossain *et al.,* [4] showed that Nusselt number increases with the increase in Reynolds number. Both these results obtained from the previous work along with the results of present study are plotted in the same graph on Figure-7. Good agreement is found with the results of Hossain *et al.,* whereas a slight deviation is seen while comparing with the result of Hardt *et al.,* The qualitative agreement of the results implies the validity of the present analysis.

Effect of geometry on heat transfer

Pitch is the distance between two closest peaks. In this study, it is varied from a value of 0.95 to 0.11875 mm. For all cases, average velocity of fluid has been maintained at 0.015 m/s [12]. It is seen from the study that as the number of peaks is increased on the surface, the surface area increases which plays an important role to enhance heat transfer from the wall surface to the fluid. Besides, the use of non-straight surface cause recirculation and reverse flow that enhances the fluctuations of fluid molecule which lead to even better convection heat transfer. As a result, the fluid in a heat exchanger with more peaks can remove more heat than that by the heat exchanger of the same length and amplitude with less number of peaks. Consequently, the mean temperature as well as the effectiveness gets higher in case of low pitch (more no. of peaks) heat exchanger. The heat exchanger effectiveness is defined as the ratio of actual heat transfer and maximum possible heat transfer.

The outlet mean temperature, effectiveness, heat transfer of fluids varies with geometry which is shown in Figures from 8 to 10. It is clear from Figure-8 that outlet mean temperature increases as pitch is reduced (surface area is increased). The rise of temperature is gradual in case of wavy enclosure whereas it is abrupt for the other surfaces. The ratio of A_1 and A_2 which is used to calculate heat transfer coefficient and then heat transfer (calculated in Watt) has the lowest value for rectangular surface. Area of fluid entry, A_1 is kept constant whereas surface area of convection A_2 is increased by reducing pitch (peak to peak distance). Thus, maximum value of A_2 is seen for rectangular surface in comparison to other surfaces and consequently much heat is removed from the surface of rectangular area. As effectiveness is related to temperature, it shows the same pattern of curve as temperature which is illustrated in Figure-9.

Heat transfer is calculated as the products of heat transfer coefficient, surface area of convection and temperature change. Heat transfer coefficient is interrelated with ratio of A_1 and A_2, properties of fluid and temperature. In Figure-10 heat transfer is plotted against pitch for all the surfaces. In every case, better heat transfer is achieved with reducing pitch. In case of wavy surface the rate of increment in heat transfer is not that remarkable as in triangular and rectangular surface.

In case of triangular surface in MEMS heat exchanger when pitch is reduced from 0.475 to 0.11875 mm, the value of heat transfer does not vary much while in case of rectangular surface, both the values of heat transfer and effectiveness [12] remain the same. It means that increasing the peaks has no effect on effectiveness and heat transfer behavior after 0.475 mm in case of triangular and rectangular surfaces. Figure-11 shows comparison of effectiveness of refrigerants in different geometries. It shows clearly that rectangle surface with Freon 11 as working fluid shows highest effectiveness compared to other combinations. In case of less number of peaks, the diminishing rate of heat transfer and effectiveness is higher in case of triangular surface than those of rectangular surface. For less number of peaks, wavy surface shows good performance in heat transfer and effectiveness than triangular surface. From this aspect, it can be concluded that rectangular surface can be considered instead of wavy surface for designing MEMS heat exchanger.

The flow field inside the enclosure of pitch 0.95 mm is presented in terms of streamlines in Figure-12. It is apparent that the streamlines do not vary from one instant to the next as steady flow is considered. It is also seen that vortices are formed close to the convex sections of the walls when pitch length is reduced from 0.95 mm to 0.475 mm. If pitch is reduced further, fluid can not enter the uneven sections, and vortices tend to be diminished. Therefore, the heat transfer is not enhanced as it is expected to be. So, 0.475 mm is considered as the optimum pitch for a 9.5 mm MEMS heat exchanger.

CONCLUSIONS

This study is an attempt to present the effect of different geometries on heat transfer characteristics under steady state condition in a MEMS heat exchanger. Various investigations have been conducted earlier using air as working fluid. Most of the researches have been based on wavy shape, comparison with conventional heat exchangers and effect of different parameters on heat transfer rates. So it has become essential to compare performance of MEMS heat exchanger for different geometries. Rectangular surface can be considered instead of wavy surface while designing MEMS heat exchanger as performance is better than wavy surface and also easier to manufacture. Decrease of pitch up to 0.475 mm improves effectiveness of micro heat exchanger. Further decrease of pitch of peaks increases surface area but causes no significant effect on heat transfer enhancement. The operating temperature range during the study spans from 48 °C to 90 °C (321 K to 363 K). At this temperature range, only water remains in liquid phase, other working fluids such as ammonia, Freon 113 and Freon 11 remain in vapor phase. Analysis for keeping the refrigerants in liquid phase may help the researchers to find better result of heat transfer behavior and thus a better combination of test fluid and geometry may come across for MEMS heat exchanger.

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Nomenclature

Roman symbols

A	Surface area (m^2)
A_1	Area of fluid entry (m^2)
A_2	Surface area of convection (m^2)
B	Half of the distance of two parallel plates (mm)
C_p	Specific heat at constant pressure (J/kg.K)
D_h	Hydraulic diameter (m)
g_x	Gravitational acceleration (m/s^2)
h	Convective heat transfer coefficient ($W/m^2.K$)
k	Thermal conductivity (W/m.K)
$\dot{m}$	Mass flow rate (kg/s)
Nu	Nusselt number
p	Pressure (Pa)
p_a	Atmospheric pressure (Pa)
Q_{conv}	Heat transfer due to convection (Watt)
Re	Reynolds number
T_m	Bulk temperature (K)
T_w	Wall Temperature (K)
T_i	Inlet Temperature (K)
T_o	Outlet mean temperature (K)
U_{av}	Average velocity (m/s)
u_t	Tangential component of local velocity (m/s)
u, v, w	Local velocity (m/s) in x, y and z direction respectively (m/s)

Greek symbols

μ	Dynamic viscosity (kg/m.s)
ρ	Density (kg/m^3)

Abbreviation

CPU	Central processing unit
MEMS	Microelectromechanical systems

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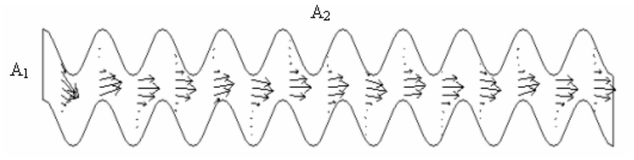

Figure-1. Cross section of the fluid flow field.

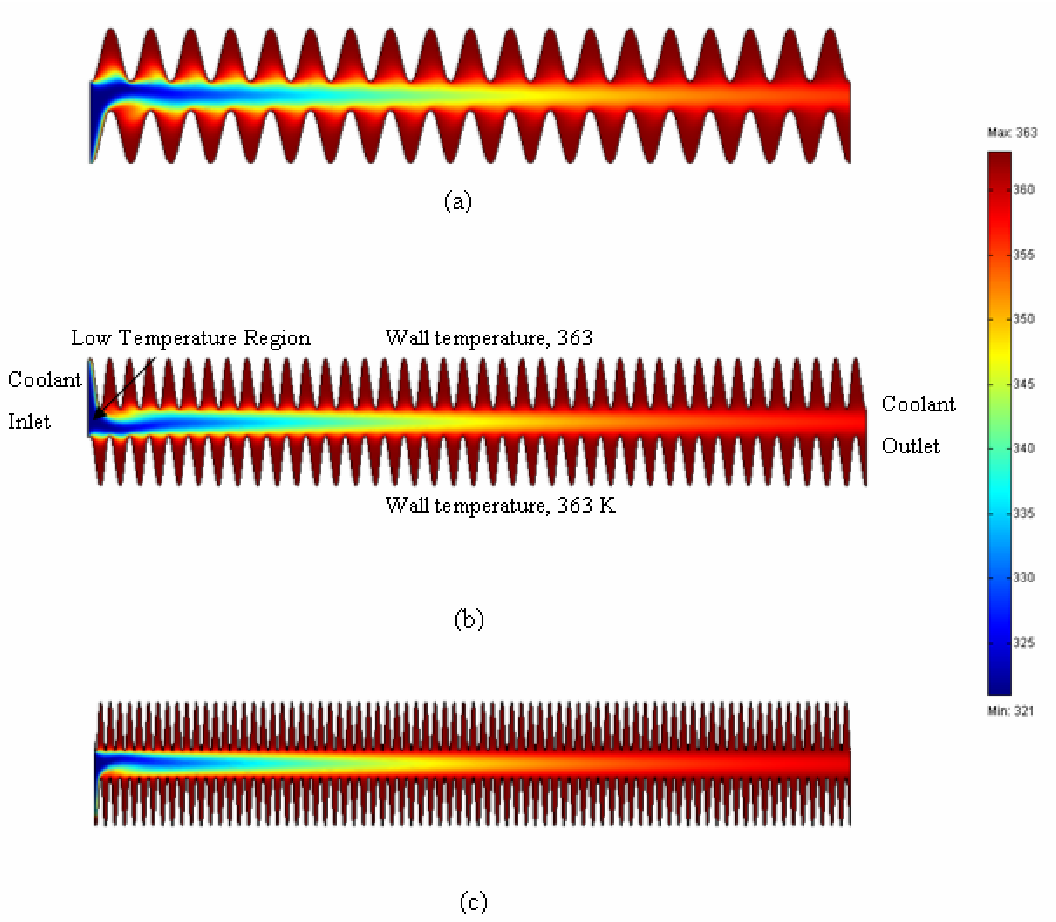

Figure-2. Temperature distributions in wavy surface MEMS heat exchanger for pitch (a) 0.475 mm (b) 0.2375 mm (c) 0.11875 mm for water.

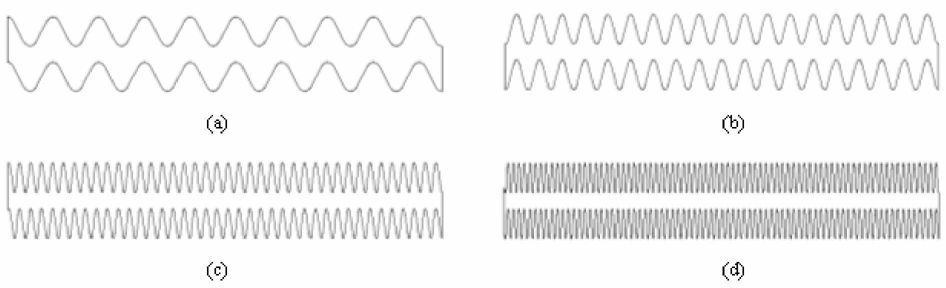

Figure-3. Wavy surface of pitch length (a) 0.95 mm (b) 0.475 mm (c) 0.2375 mm (d) 0.11875 mm.

ARPN Journal of Engineering and Applied Sciences

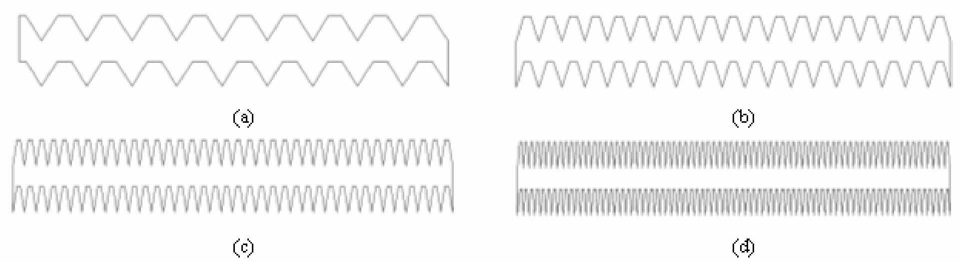

Figure-4. Triangle surface of pitch length (a) 0.95 mm (b) 0.475 mm (c) 0.2375 mm (d) 0.11875 mm.

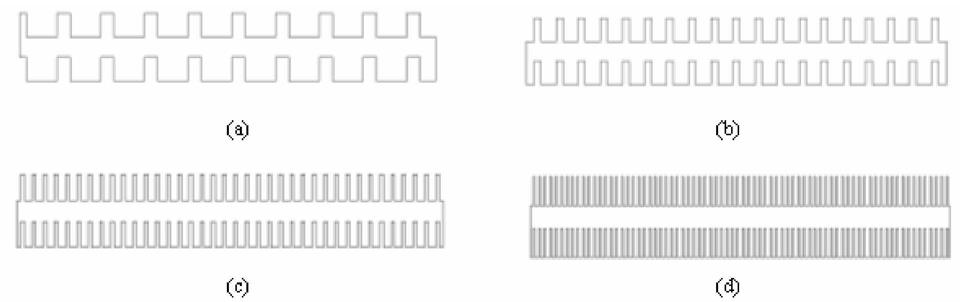

Figure-5. Rectangle surface of pitch length (a) 0.95 mm (b) 0.475 mm (c) 0.2375 mm (d) 0.11875 mm.

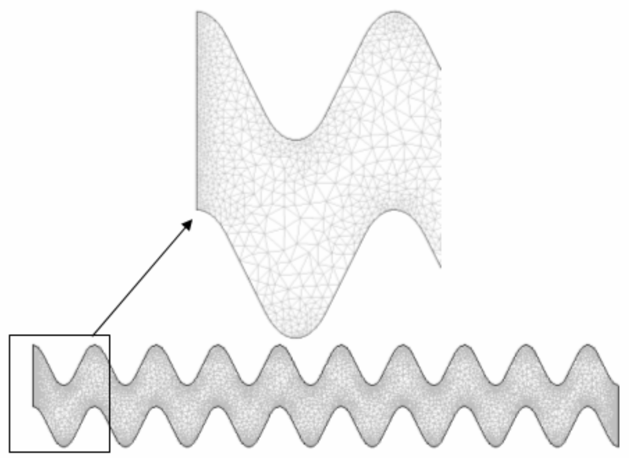

Figure-6. Two dimensional meshing of wavy surface MEMS heat exchanger of pitch 0.95 mm.

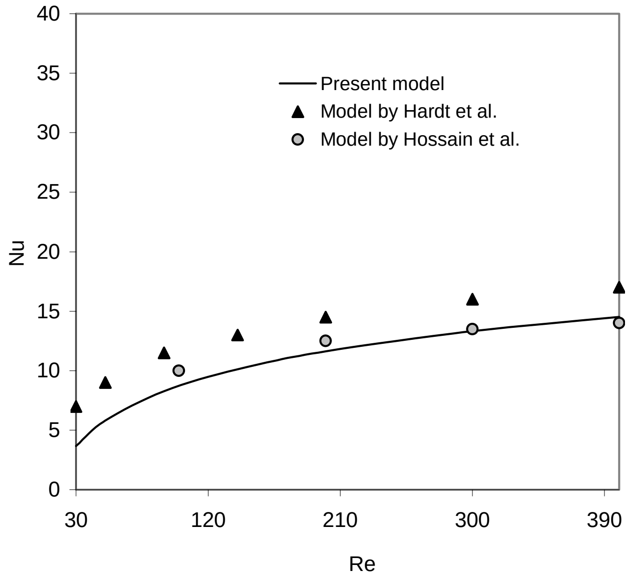

Figure-7. Variation of Nusselt number (Nu) with different Reynolds number (Re) for the pitch of 0.95 mm and air as working fluid.

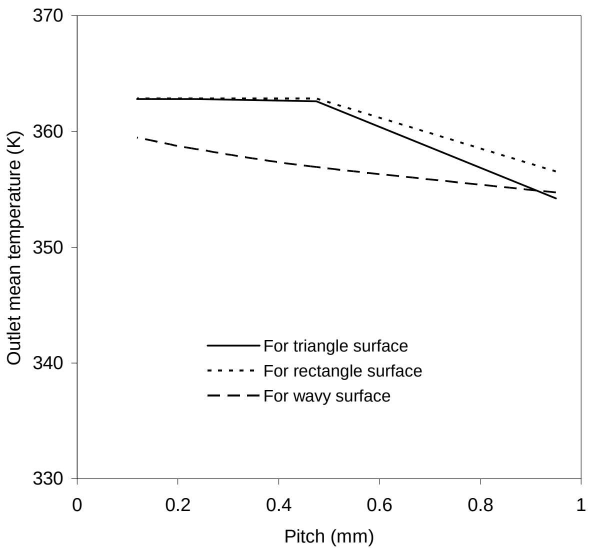

Figure-8. Variations of outlet mean temperature of MEMS heat exchanger of different geometries for water as working fluid (U_{av} = 0.015 m/s).

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ARPN Journal of Engineering and Applied Sciences

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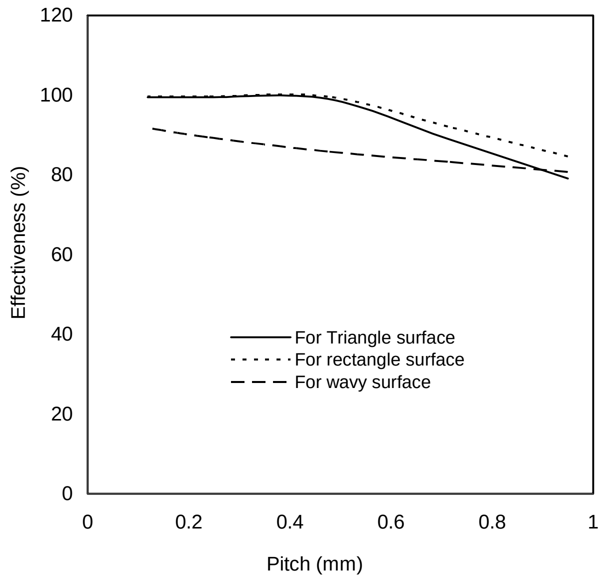

Figure-9. Comparison of effectiveness of MEMS heat exchanger of different geometries for water as working fluid (U_{av} = 0.015 m/s).

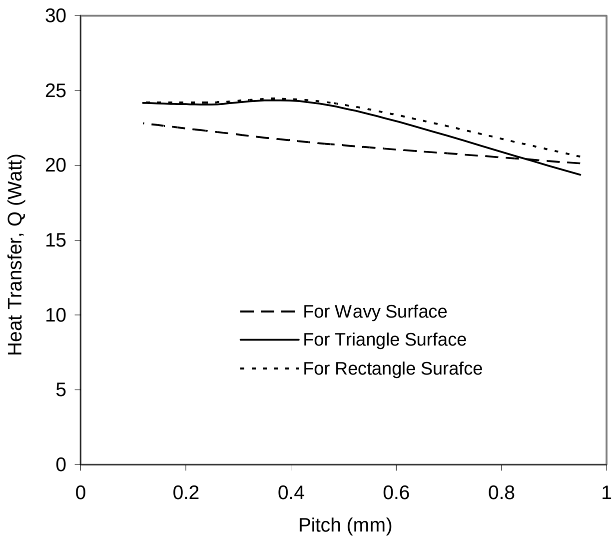

Figure-10. Comparison of heat transfer of MEMS heat exchanger of different geometries for water as working fluid (U_{av} = 0.015 m/s).

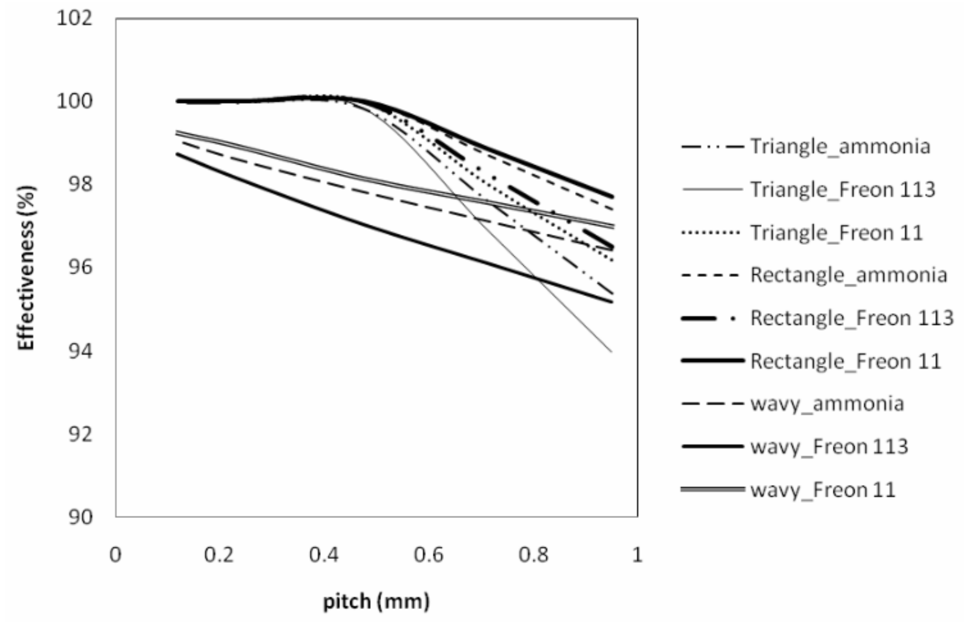

Figure-11. Comparison of effectiveness of MEMS heat exchanger of different geometries for various refrigerants as working fluids (U_{av} = 0.015 m/s).

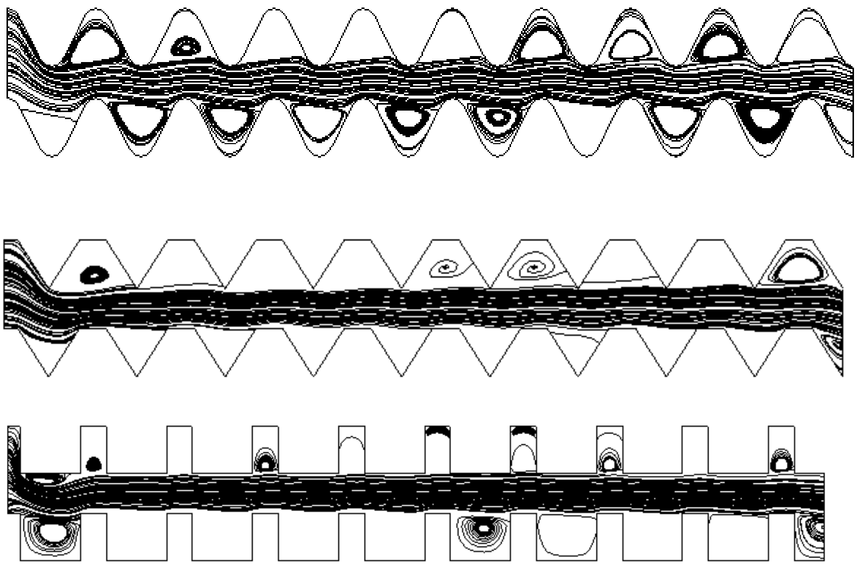

Figure-12. Flow patterns of water through different types of geometries of MEMS heat exchanger (pitch = 0.95 mm and U_{av} = 0.015 m/s).