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Parametric Analysis of MEMS – Based Heat Exchanger with Different Test Fluids

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Authors' contributions

This work was carried out in collaboration between all authors. Author NJC designed the study, performed the finite element analysis and wrote the first draft of the manuscript and managed literature searches. Author MM revised the analyses of the study and reviewed the draft manuscript carefully. Author MAN supervised over all research and finalized the manuscript. All authors read and approved the final manuscript.

Research Article

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ABSTRACT

The latest generations of micro-scale systems are becoming more challenging to fit into designs. These chips are squeezing into smaller and smaller spaces with very little place for heat to escape. Therefore, MEMS heat exchanger has become popular in many practical applications although improvement of heat transfer characteristics is a key issue for the users as well as researchers. In the present study it is suggested that instead of using conventional working fluids, the micro sized hot structures can be cooled with an effective coolant which can be a good substitute of the conventional fluids. Ammonia has shown the highest outlet mean temperature during the study. The analysis is conducted using commercial finite element package to determine outlet mean temperature that is then used for further calculation of effectiveness, heat transfer coefficient and friction factor.

Keywords: MEMS heat exchanger; working fluids; friction factor; effectiveness.

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1. INTRODUCTION

Microelectromechanical systems (MEMS) is the technology of the very small, and merges at the nano-scale into nanoelectromechanical systems (NEMS) and nanotechnology. Heat exchangers have been mostly used in many engineering applications especially when fans and fins are not enough for convective cooling. The introduction of MEMS heat exchanger has opened up a new era of heat transfer in micro scale systems. These heat exchangers are becoming more popular in chemical, electronic and aerospace industries for superior heat exchange properties, compact design and low inner volume.

Various research works are available in literature regarding MEMS technology. A computer code was developed by Yuen and Hsu [1] in order to interpret the performance characteristics of the heat exchanger and to serve as a design tool for the refrigerator. Results of numerical simulation showed that the micro-channel heat exchanger was effective in generating a uniform liquid fraction over the whole base surface. Thus the design objective of generating a uniform temperature surface was met. Hardt et al. [2] found that channels equipped with micro fins allowed for a rapid exchange of heat. Such designs exhibited a potential to construct very compact heat exchangers and lent themselves as components of heat-exchanger reaction systems. The numerical results clearly showed a superior performance of the reactor containing microstructured heat- exchanger channels compared to unstructured channels. Okabe et al. [3] aimed to optimize a micro heat exchanger using Multi-Objective Optimisation (MOO). The physical phenomenon in micro heat exchanger is multidisciplinary and involves conjugate heat transfer. In order to solve the conjugate heat transfer problem, they used a commercial computational fluid dynamics (CFD) solver called CFD-ACE+ developed by Computational Fluids Dynamics Research Corporation in the USA. This CFD solver was interfaced with their in-house developed evolutionary algorithms.

Hossain and Islam [4] solved two-dimensional Navier -Stokes and energy equations numerically for unsteady laminar flow in periodic wavy (sinusoidal and triangular) channels. They found that beyond the critical Reynolds number the flow became self sustained. The influence of critical system variables on heat transfer rates and pumping power was assessed by Chandratilleke et al. [5]. They developed numerical models and determined the optimal parametric combinations for thermoelectric applications. The study showed that with appropriate selection of operating parameters, micro-heat exchanger design offered thermal resistances of the order 0.01-0.02 K W^{-1} and low fluid pump powers.

A series of numerical model of the fluid flow and heat transfer in counter-flow heat exchangers with oblique wavy walls were developed by Morimoto et al. [6] to determine the optimal design of recuperators. They found that with the oblique angle of 50-60 degree, significant heat transfer enhancement can be achieved at the cost of relatively small pressure loss. Mébrouk et al. [7] numerically investigated the convective heat transfer and fluid flow in a horizontal wavy enclosure. In their solution domain, the bottom wall was varied with a sinusoidal function while the top and the two side walls were flat. Their numerical results showed that the flow and the heat transfer were strongly affected by the amplitude of the sinusoidal profile.

Eiamsa-ard and Promvonge [8] had demonstrated the influence of the helical tape insert on the heat transfer and pressure drop characteristics. Their experimental study showed that the increment of heat transfer and pressure drop were strongly influenced by turbulence/swirling motion induced by the helical-tape and wavy-surfaced wall. The maximum increment of heat transfer rate and pressure drop were found about 2.67 and 22.3 times higher than those of the plain tube for the given flow range. Davis et al. [9] used an experimental setup to test the efficacy of the waterblocks in a typical CPU cooling environment. They found that water-cooling solutions had overcome both conduction resistance & airflow limitations, and therefore allowed another generation of decreased die size as well as increased CPU power.

Chhanda et al. [10-11] conducted an analysis on the effects of different geometries of MEMS heat exchangers on heat transfer enhancement. They found that the critical pitch is 0.475 mm for the optimum effectiveness and heat transfer enhancement. Spann [12] presented a unique closed form mathematical solution for single-pass, two-fluid, parallel and counter flow microscale heat exchangers. The model included the effects of axial wall conduction, ambient thermal interaction at the axial exterior surface, and general end-wall boundary conditions. The author claimed that heat capacity rate ratio resulted in more energy transfer between the fluids, and thus enhanced the performance. White et al. [13] presented a numerical model for perforated plate heat exchangers which is used in cryosurgical probe. The numerical model was validated by experimentally testing several perforated plate heat exchangers that are fabricated using MEMS based manufacturing methods. According to the authors, the numerical model was able to accurately predict both the overall performance and the internal temperature distribution of perforated plate heat exchangers over a range of geometries and operating conditions.

In the present research, a finite element model is developed for the analysis of MEMS heat exchanger performance with various working fluids. The heat exchanger is modeled in such a way that several square plates are stacked on top of each other. Although changing the geometry of heat exchanger can be a way of increasing the heat transfer, it is obvious that it involves manufacturing costs and time. Rather it is more convenient to switch to an appropriate working fluid which can play a big role to improve heat transfer. Wang and Li [14] featured the status of natural working fluid research and application developments in China in their paper. According to their study, water and ammonia as refrigerants are promising technically and economically in the application of absorption and adsorption systems, and various commercialized products have been developed for utilization in industry as well as commercial building. Three groups of alternative absorption heat pumping processes with different natural working fluids were analyzed in the paper of Kotenko et al. [15]. Recent advancement on MEMS heat exchanger has been conducted by many researchers to improve heat transfer performance. Mohammed et al. [16,17] numerically investigated heat transfer and fluid flow characteristics in a square shaped microchannel heat exchanger using various types of nano-fluids. In their research, they used nanoparticles to improve heat transfer coefficient of the base fluids. Dang and Teng [18] showed that heat transfer rate can be increased by decreasing the size of channels in heat exchanger.

As the knowledge of effect of different working fluids on heat transfer behavior of MEMS heat exchanger is far from being complete, this research gives emphasis on evaluation of MEMS heat exchanger performance for different working fluids. It is evident that Freon 113 (R113) and Freon 11 (R11) have very good heat transfer coefficient. However, according to the Montreal Protocol in 1987, all CFCs including R113 should be abandoned by 2010 due to having high ozone-depleting characteristics. This inspires the present research to find a good substitute of CFCs, which has also good heat transfer coefficient. CFCs have been used in the paper to have direct comparison with other probable substitutes.

2. FINITE ELEMENT MODELING

In this analysis, a heat exchanger is considered where several square plates are stacked on top of each other with a gap of 1 mm in between. To simplify, a two-dimensional (2-D) model of the heat exchanger is developed where only a cross section between two plates is taken into consideration (shown in Fig. 1 where A_1 denotes the inlet area and A_2 denotes the surface area of convective heat transfer. The length of the plates is 9.5 mm. Fluid is used as coolant to transfer heat which circulates in gaps between the undulating walls.

The overall performance of the heat exchanger is calculated from the outlet mean temperature of the working fluids. Different fluids are used to evaluate their performances, such as: air, water, ammonia, R11 and R113 for a particular operating temperature range. The operating temperature range is considered from 48 to 90 ºC (321 K to 363 K). Within this temperature range, except water, other fluids such as ammonia, R113 and R11 remain in vapor phase at atmospheric pressure. The boiling point of R11 at atmospheric pressure is 23.77 ºC. Boiling point of R113 and ammonia is 47.6 ºC and -33.3 ºC respectively at atmospheric pressure [19]. Water remains in liquid phase (boiling point 100°C at atmospheric pressure). The temperature range is chosen such that working fluids can operate as single-phase fluid in the heat exchanger.

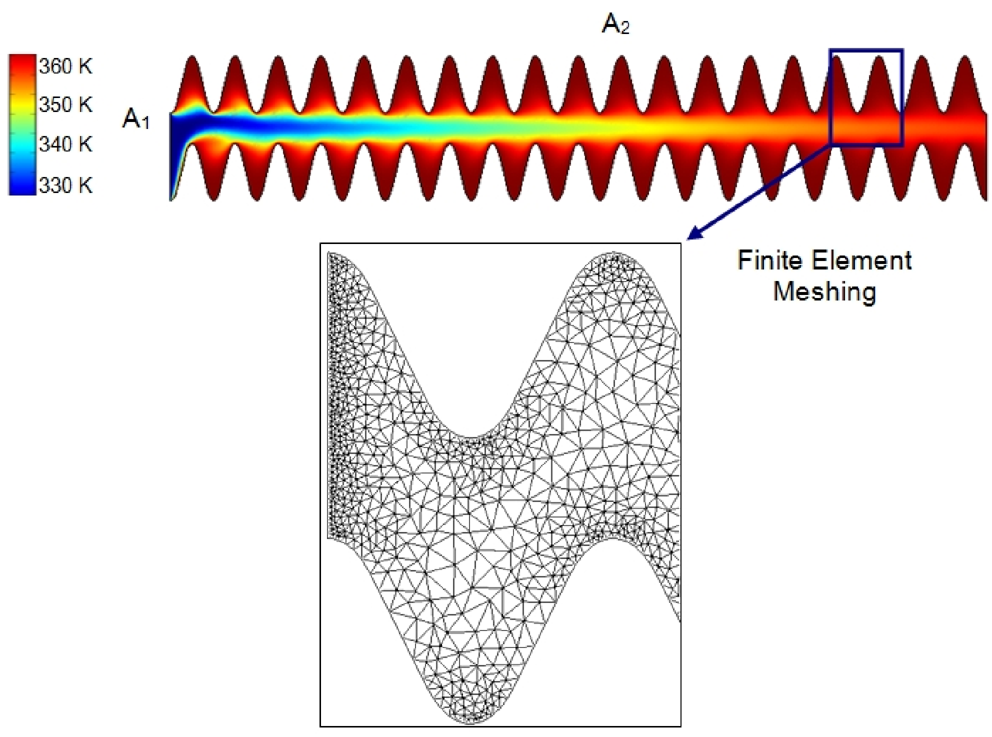

Fig. 1. Cross section of the model and typical temperature distribution for pitch 0.475 mm and water as test fluid

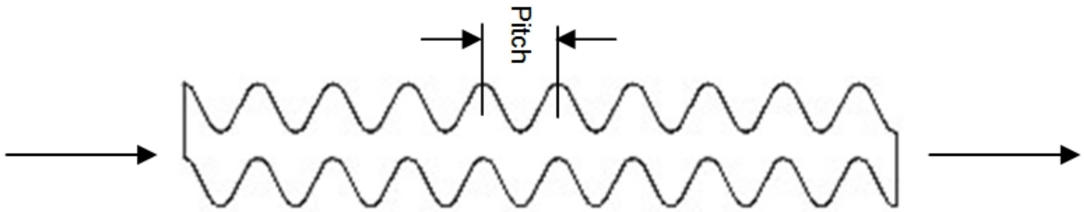

Fig. 2. Wavy surface of MEMS heat exchanger indicating pitch length

The analysis is done by varying the properties of fluids as well as the pitch of convective surface waviness. Pitch can be defined as the distance between two adjacent peaks as shown in Fig. 2. The performance of a MEMS heat exchanger varies for a particular fluid for different pitches. The initial length and gap between the two plates are maintained constant throughout the analysis while varying the pitch. In this study, the number of peaks per unit length has been increased. In other words, the pitch has been reduced from 0.95 mm to 0.11875 mm.

3. GOVERNING EQUATIONS

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

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

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

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

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

For two dimensional analysis Eq. 1, 2 and 3 can be simplified in the following way,

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

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

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

A finite element model is developed in FEMLAB 3.0 and the results are obtained by solving Eq. 4, 5 and 6 simultaneously.

Fluid properties vary with working temperature. For solving the governing equations, the values of different fluid properties are given as input parameters in the Subdomain Expressions dialog box of FEMLAB. Isothermal walls are considered as boundary conditions. At the inlet, a parabolic shape of velocity profile for developed fluid flow is specified. It is also specified that fluid enters the heat exchanger with a particular temperature. At the uneven surfaces no-slip condition for velocity and a specific temperature condition for the energy balance are considered as boundary condition. Finally, normal flow (perpendicular flow) velocity conditions and convective flux heat transfer are assumed at outlet. Heat is transferred from the heated wall to the coolant. At the outlet, velocity weighted mean temperature is obtained.

4. CALCULATION OF EFFECTIVENESS

To calculate the overall performance of the heat exchanger, the mean temperature at the outlet of the heat exchanger is to be attained. The velocity weighted mean temperature across a boundary is defined as,

$$T_o = \frac{\int_s Tuds}{\int_s uds} \tag{7}$$

Where ds is the boundary length element.

The heat exchanger effectiveness is defined as

$$\varepsilon = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}} \tag{8}$$

To determine the maximum possible heat transfer for the heat exchanger, maximum possible temperature difference is necessary to be calculated. The maximum temperature difference could be attained if either hot or cold fluid undergoes a temperature change equal to the temperature difference between the hot wall and the entering cold fluid. The fluid which undergoes this maximum temperature difference is one having the minimum value of $\dot{m}c$ because for the energy balance, the energy received by one fluid is to be equal to that given up by other fluid; so, maximum possible heat transfer is expressed as

$$q_{max} = (\dot{m}c)_{min}(T_{h_{inlet}} - T_{c_{inlet}}) = (\dot{m}c)_{min}(T_{hot\ wall} - T_{inlet\ fluid}) \tag{9}$$

The minimum fluid may be either the hot or the cold fluid, depending on the mass-flow rates and specific heats.

For a parallel flow heat exchanger where cold fluid is the minimum fluid, effectiveness can be expressed as,

$$\varepsilon = \frac{\dot{m}_c c_c (T_{c_2} - T_{c_1})}{\dot{m}_c c_c (T_{h_1} - T_{c_1})}$$

i.e., $$\varepsilon = \frac{(T_{c_2} - T_{c_1})}{(T_{h_1} - T_{c_1})} \qquad (10)$$

Where, T_{c_2} is outlet mean temperature, which can also be expressed as T_o

T_{c_1} = 321 K (48 ºC) (assumed)

Besides water and air, three refrigerants are used as working fluids during this analysis. Among them, R113 shows the highest boiling point of 47.6 ºC. R11 and ammonia remain at vapor phase whereas water is liquid at this temperature. So if the temperature range is chosen as 48 ºC to 90 ºC, then all working fluids remain at single phase. Thus this analysis is carried out considering only single phase flow. After 90 ºC, water starts changing its phase and formation of bubbles may start. It is undesirable as cavitation is not considered in the present study.

T_{h_1} = 363 K, where, T_{h_1} is heated wall temperature

Therefore, the expression becomes

$$\varepsilon = \frac{T_o - 321}{363 - 321}$$

i.e., $$\varepsilon = \frac{T_o - 321}{42} \qquad (11)$$

5. CALCULATION OF THE HEAT TRANSFER PARAMETERS

The outlet mean temperature is also utilized to calculate heat transfer parameters such as Nusselt number and Reynolds number.

Nusselt number, $$Nu = \frac{hD_h}{k} \qquad (12)$$

Where h is averaged heat transfer coefficient, D_h is the hydraulic diameter and k is the thermal conductivity of fluid.

The averaged heat transfer coefficient, h is determined as [11]:

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

i.e., $$h = \frac{\rho U_{av} A_1 C_p (T_o - T_i)}{A_2 (T_w - T_m)} \tag{13}$$

T_o is outlet mean temperature,

$$T_m = \frac{(T_o + T_i)}{2} \tag{14}$$

T_w = wall temperature which is assumed to be 363 K (90 ºC)
T_i = inlet temperature of fluid = 321 K

Reynolds number is calculated as

$$Re = \frac{\rho U_{av} D_h}{\mu} \tag{15}$$

ρ is the density, $\dot{m}$ is the flow rate and U_{av} is the mean velocity of fluid.

6. RESULTS AND DISCUSSION

6.1 Validation of the Analysis

For numerical validation, air has been used as a working fluid and flowed through the gap between two surfaces. Convective heat transfer is the principle mode of heat transfer in air. For the analysis, steady state condition is assumed. The geometry has been a wavy surface of pitch 0.95 mm and the velocity of air has been varied to obtain a desirable range of values for Reynolds number. The aim is to maintain laminar region in the flow field. For the comparison with other results available in the literature, variation of Nusselt number with Reynolds number is shown in Fig. 3. Hardt et al. [2] noticed that the flow pattern had been affected by the sine-shaped walls and a substantial heat transfer enhancement was achieved. Another analysis done by Hossain et al. [4] showed that Nusselt number increased with the increasing Reynolds number (Re). Steady flow gave modest increase in Nusselt number but unsteady flow gave rapid increase due to better mixing of core and near the wall fluids, but the rate of increase again slowed down as Re was increased more.

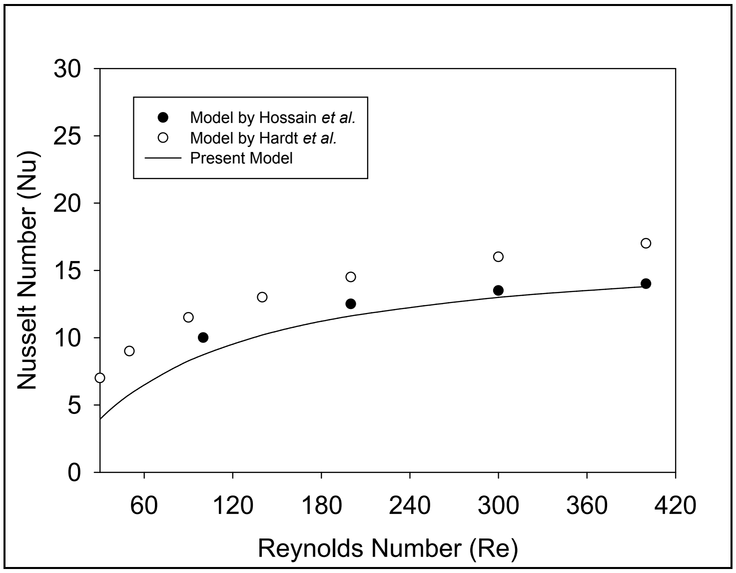

Fig. 3. Variation of Nusselt number (nu) with different Reynolds number (re) (pitch = 0.95 mm and air as working fluid)

Results available in the literature are plotted along with the results of present study in the same graph in Fig. 3 in order to have a direct comparison of the variation. Good agreement is found with the result of Hossain et al. [4]. A slight deviation is seen while comparing with the result of Hardt et al. [2]. The qualitative agreement of the results implies the validity of the present model.

6.2 Effect of Different Parameters with Reynolds Number

The performance of heat exchanger is evaluated using different working fluids in the present analysis. Emphasis has been given on fluid transport properties, viscosity, thermal conductivity and density which may vary considerably with temperature. Such property variations distort both velocity and temperature profiles, so all fluid properties are evaluated at the mean fluid temperature. The physical mechanism of viscosity involves momentum exchange. As laminar flow is considered here, molecules may move from one lamina to another, carrying with them a momentum corresponding to the velocity of the flow. There is a net momentum transport from high velocity regions to low velocity regions, thus creating a force in the direction of the flow. The rate at which momentum transfer takes place depends on the rate at which the molecules move across the fluid layers. In a gas, molecules will move about with some average speed proportional to the square root of the absolute temperature since, in the kinetic theory of gases, temperature is identified with the mean kinetic energy of a molecule. The faster the molecules move the more momentum they transport. So air is used as working fluid in most of the research works. In spite of this, air as working fluid causes low effectiveness of heat exchanger. Therefore, performance of other fluids must be investigated so that their performance can be compared with air.

Fig. 4 and 5 show the variation of effectiveness and Nusselt number with Reynolds number for different working fluids. The figures also indicate the optimum conditions for different working fluids. It is seen from the figures that effectiveness decreases with the increasing

Reynolds number. On the contrary, Nusselt number increases with the increasing Reynolds number. Therefore, optimum value of Reynolds number is considered here at the intersecting point of ascending and descending curves. The optimum points for water, air, ammonia, R11 and R113 are denoted as A, B, C, D and E respectively in Fig. 4 and 5. Optimum effectiveness and Nusselt number of water can be obtained at Reynolds number, Re = 88. Similarly, optimum Reynolds number for air, ammonia, R11 and R113 are found around 205, 160, 280 and 280 respectively. For the optimum Reynolds number of 280, R11 can operate with the highest effectiveness of heat exchanger, which is 79%. For ammonia effectiveness of heat exchanger can be attained upto 77% for the optimum Reynolds number. For R113, water and air, the value of effectiveness for optimum Reynolds number are 72%, 67% and 64% respectively.

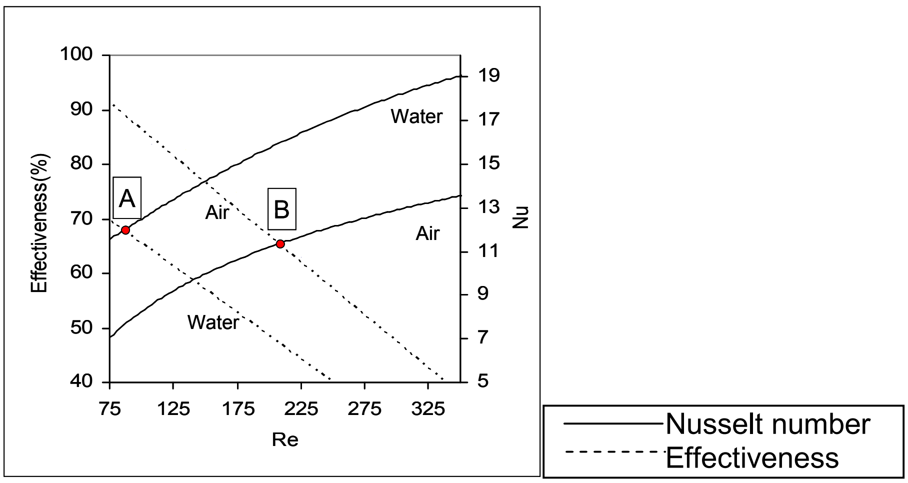

Fig. 4. Variation of effectiveness and Nusselt number with Reynolds number for wavy surface MEMS heat exchanger (pitch = 0.95 mm) with air and water as working fluid

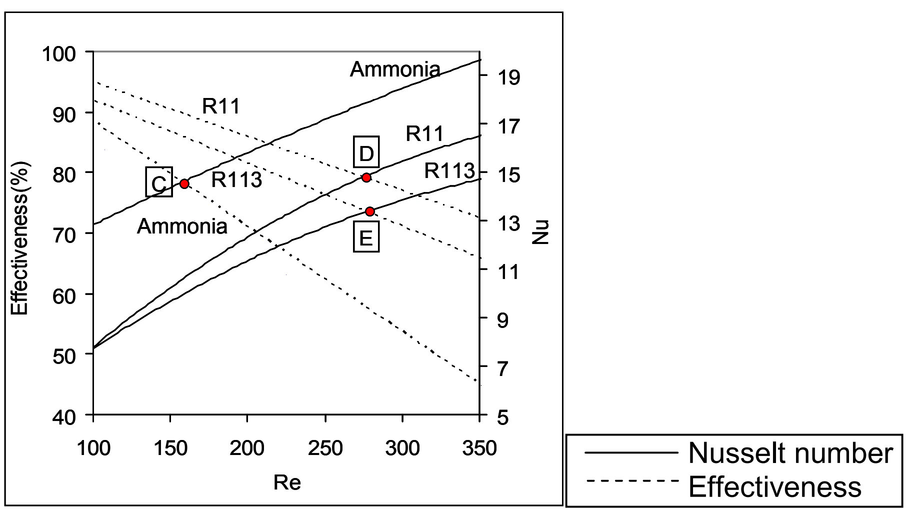

Fig. 5. Variation of effectiveness and Nusselt number with Reynolds number for wavy surface MEMS heat exchanger (pitch = 0.95 mm) with different working fluids

There is no denying the fact that the use of the uneven-surfaced wall cause increase in heat transfer area, which causes more heat transfer to the coolant. Furthermore, the recirculation and reverse flow due to uneven surface increase the fluctuations of fluid molecule, which lead to even better convection heat transfer. Thus for a wavy surfaced heat exchanger, the Nusselt number increases with the increase in Reynolds number. It is also evident from the Figs. 4 and 5 that with increase in Reynolds number, effectiveness decreases. It means that although the flow rate increases the effectiveness does not increase. It may happen due to the fact that heat exchanger is more effective at lower flow rate [20].

In addition to the effectiveness and heat transfer, frictional loss is another major concern for MEMS heat exchanger. In the case of fully developed laminar flow, it depends on the shape of the cross-section. It is characterized as the product of friction factor and Reynolds number which is also known as the Poiseuille number [21]

$$P_o = f.Re \tag{16}$$

and f = 4 $\times$ Fanning friction factor.

So, for two parallel plates, friction factor is calculated as

$$f = \frac{96}{Re_{opt}} \tag{17}$$

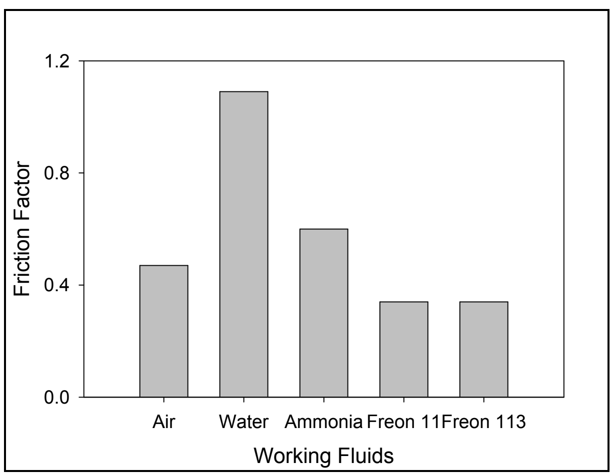

Fig. 6. Friction factor for different working fluids

Optimum Reynolds numbers for different fluids are determined from Fig. 4 and 5. For the optimum Reynolds number water shows the highest friction factor (greater than 1) which is shown in Fig. 6. The CFC refrigerants show the lowest friction factor of 0.34 whereas the value for ammonia is 0.6.

Lower friction causes lower energy consumption. Although CFC refrigerants show lowest friction, due to toxicity and unfriendly behavior with environment, CFC refrigerants are banned by the manufacturers. Thus, ammonia can be a very good substitution of CFC refrigerants as a coolant in microelectronic systems.

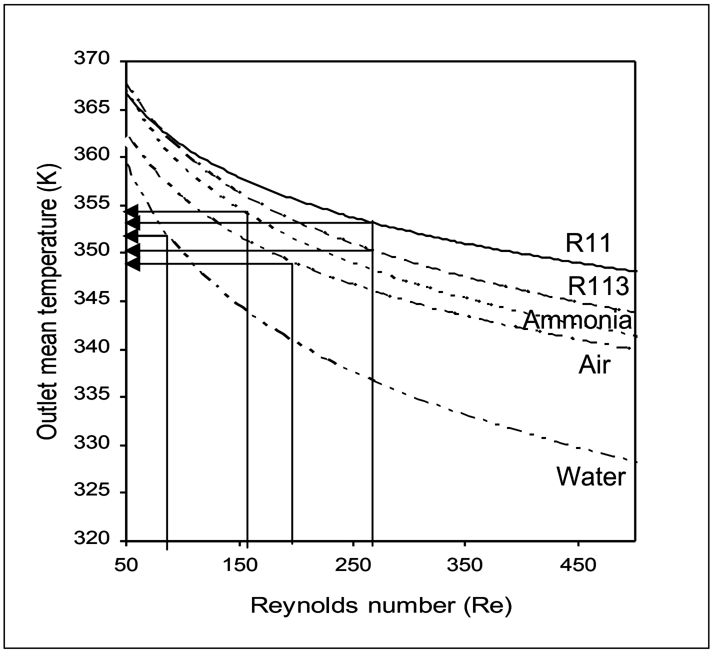

Fig. 7. Variation of outlet mean temperature with Reynolds number for MEMS heat exchanger (pitch = 0.95 mm) with various working fluids

Fig. 7 shows the decrease in outlet mean temperature with increase in Reynolds number. As velocity increases, fluid does not get enough time to absorb heat from its surroundings. As a result, outlet mean temperature does not increase as expected. The effectiveness has been determined using Eq. 11. For the optimum Reynolds number, the highest outlet mean temperature is exhibited by ammonia (354 K). Outlet mean temperature of R11, water, R113 and air are 353 K, 352 K, 350 K and 348 K respectively. It is also observed that effectiveness increases as the value of outlet mean temperature approaches the wall temperature. From this aspect, ammonia sets itself as the best option of an effective coolant.

6.3 Effect of Different Parameters with Pitch

In the present study, MEMS heat exchanger is modeled with different pitches to evaluate their performance. The number of peaks in wavy surface is increased without changing overall length of the plate. As a result, pitch is reduced from 0.95 to 0.11875 mm. For all cases, average velocity has been maintained at 0.015 m s^{-1}. Fig. 8 depicts the variation of outlet mean temperature with different pitches for wavy surface heat exchanger. It is seen that the outlet mean temperature decreases with the increase in pitch. Since the number of peaks increases in the wavy surface, the convective surface area also increases which enhances heat transfer from the wall surface to the fluid. As a result, in a heat exchanger with lower pitch, fluid can take away more heat in comparison to that with higher pitch of wavy surface. Consequently, the outlet mean temperature also increases in case of low pitch (more no. of peaks) heat exchanger.

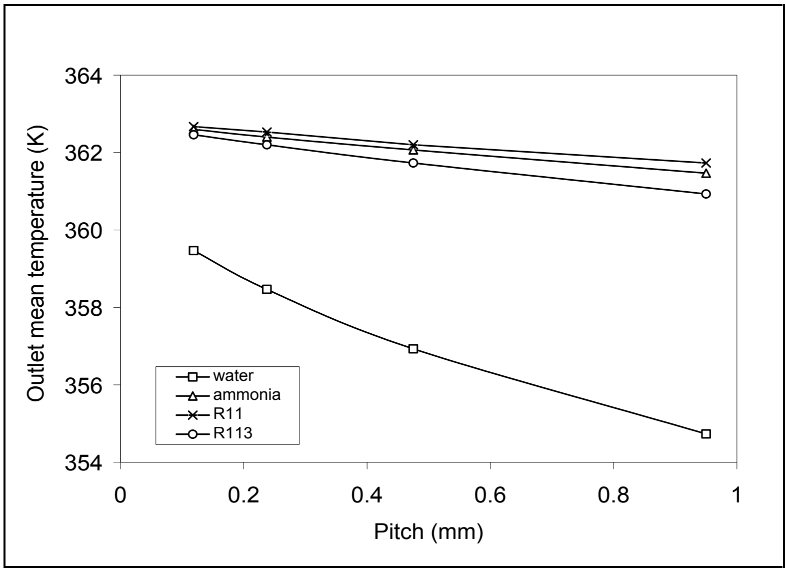

Fig. 8. Variation of outlet mean temperature with different pitches of wavy surface MEMS heat exchanger for various working fluids (U_{av} = 0.015 m/s)

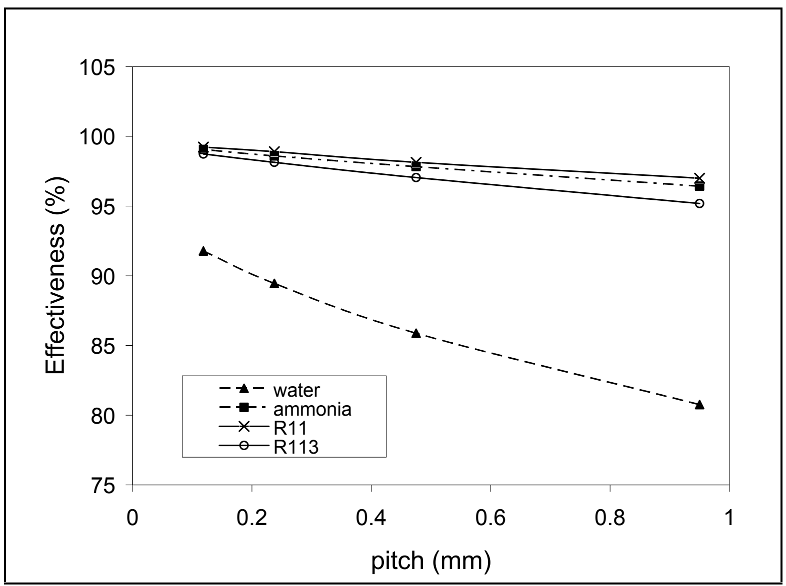

Fig. 9. Variation of effectiveness with different pitches of wavy surface MEMS heat exchanger for various working fluids (U_{av} = 0.015 m/s)

Another conspicuous point of this Fig. 8 is that water shows the lowest outlet mean temperature in comparison to other fluids which remain vapor phase in this particular temperature range. The reason may be the greater values of enthalpy of saturated vapor than that of liquid water. As effectiveness is related to outlet mean temperature, it shows the same trend as shown in Fig. 9.

7. CONCLUSION

The analysis is an attempt to propose a working fluid for MEMS heat exchanger which may have some merits over the conventional fluids- air and water. Air and water are the most widely used working fluids for having the advantage of easy availability and being economical. In spite of having these advantages there may be some other constraints (such as forming rust on metals by water, dust content in air) that can bar the way of proper heat transfer in micro sized heat exchangers. Again, there may be another issue of frictional loss through the heat exchanger that can correspondingly cause large pumping costs. So some heat transfer parameters and friction factors for different test fluids have been analyzed here to demonstrate a better assessment. It cannot be denied that R11 shows the highest effectiveness in comparison to other fluids. But for being toxic and not friendly to the environment, it has become essential to look for another fluid which can be used in the same applications but without the same environmental concerns [22]. Ammonia can be regarded as the better option in this regard. Its high effectiveness, low friction factor and high outlet mean temperature sets itself as the most preferable working fluid for MEMS heat exchanger. In addition, ammonia's density and limited range of flammability, engineering advances for refrigeration systems, and the most importantly its environment-friendly characteristics can be considered while designing heat exchangers. The ability of ammonia to absorb larger amounts of heat per volume makes it possible to use in smaller pipes and smaller components. In the present research, it can be concluded that better heat transfer enhancement can be achieved if the most appropriate working fluid can be used in the MEMS heat exchanger.

NOMENCLATURE

Roman symbols:

A	Area exposed to convective heat transfer
A_1	Area of fluid entry
A_2	Surface area of convection
C_p	Heat capacity
D_h	Hydraulic diameter
ds	Boundary length element
f	Friction factor
g	Gravitational constant
h	Heat transfer coefficient
k	Thermal conductivity
p	Fluid pressure
P_o	Poiseuille number
$\dot{m}$	Mass flow rate
Nu	Nusselt number
q	Heat transfer
Re	Reynolds number
T	Temperature
t	Time
u	Fluid motion along x-direction
U	Velocity of fluid
v	Fluid motion along y-direction
w	Fluid motion along z-direction
x	x-axis

y	y-axis
z	z-axis

Greek Symbols:

ρ	Density of fluid
ε	Effectiveness of heat exchanger
μ	Dynamic viscosity

Subscripts:

av	Averaged
c	Cold fluid
c_1	Incoming cold fluid
c_2	Outgoing cold fluid
h_1	Incoming hot fluid
h	Hydraulic
i	Inlet
m	Mean
max	Maximum
min	Minimum
o	Outlet
opt	Optimum
x	x-direction

Abbreviation:

CFD	Computational fluid dynamics
CPU	Central processing unit
CFC	Chlorofluorocarbon
MEMS	Microelectromechnical-system
MOO	Multi-objective optimisation
NEMS	Nanoelectromechanical-system
R11	Freon 11
R113	Freon 113

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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