Effects of arrangement and phase difference of oscillation of synthetic jets on heat transfer in micro-channels

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1. Introduction

In recent years, with the advancement of technology, electronic processors play an important role in human life and many researchers are working on the new methods to design these processors. One of the critical points in designing electronic processors for the systems known as micro electromechanical systems (MEMS), with low surface and high electric power, is very high heat generation due to complex calculations carried out by these processors. Therefore, cooling the processors and achieving uniformity of temperature distribution on their surfaces in order to avoid the formation of hot spots are the problems that could cause new technical challenges for the manufacturers of these systems.

As heat flux is higher than $500 \, \text{kW/m}^2$ (Mudawar [1]) in these systems, the traditional air-cooling methods would not provide enough cooling for the components. On the other hand, using fluid flow for cooling components would have limited capacity due to the fact that laminar flow regime in the channels designed for this purpose (Qu et al. [2]). Use of synthetic jet is a new solution and concept which has been recently proposed for cooling such systems and has managed to increase the cooling capability by several times greater than the conventional methods. Synthetic jet which includes a cavity and a membrane would increase cooling by suction and discharge of a fluid through the channel without adding any net mass to it. This increases the momentum and turbulence in the fluid inside the channel. Despite many studies which have been conducted on this field, the precise effect of characteristics of individual jets and mutual effects of these jets on the flow field and heat transfer in the channel have not been correctly known thus far. This is especially true when several synthetic jets exist in a cross-flow with the channel flow.

Lee et al. [3] studied the heat transfer for several channels with different geometries as well as various Reynolds numbers empirically and numerically and reported a difference of 5% between the empirical results and numerical solutions. Zhou and Zhang [4] studied effects of the interaction of a circular synthetic jet and a boundary layer formed by the cross-flow over a flat plate numerically using three-dimensional simulation. In their study, the position of vortices caused by the synthetic jet outflow via changing the micro-channel cross-flow variations was studied. Chandratilleke et al. [5] studied the effects of synthetic jet in a micro-channel using a two-dimensional nu-

merical method. They showed that use of a synthetic jet in a micro-channel increased the heat transfer rate in the order of 4.3 times greater than the case without the synthetic jet.

Chaudhari et al. [6] studied the effects of multiple orifice single-cavity synthetic jet flow on heat transfer from their front hot wall empirically. They performed the test with different orifice arrangements and reported that the maximum heat transfer in this test was 12 times greater than that of the free convection heat transfer and 30% higher than that of the single orifice jet. Shan et al. [7] studied the effects of variation in frequency and distance of a piston synthetic jet outlet from its front hot wall, to which a constant heat flux was applied empirically. They stated that the cooling effect of the synthetic jet was relatively stronger at higher frequencies and also the orifice-to-wall distance played an important role in the cooling process.

Fang et al. [8] studied the cooling effect of synthetic jet in a micro-channel with the length of 26mm empirically and reported that operation of the synthetic jet in a micro-channel for cooling purpose increased the heat transfer rate from 40% to 50% as compared to the case of no synthetic jet.

Lee et al. [9] simulated the effects of a synthetic jet by considering one and two synthetic jets in a three-dimensional micro-channel numerically. They reported a significant increase in heat transfer as compared to case of no synthetic jet by changing the heat flux applied to the micro-channel, oscillation amplitude of synthetic jet, and its frequency. Greco et al. [10] studied the flow field for two synthetic jets empirically. They found that, at the distance in which ratio of axes to the outlet diameter was 1.1, effects of outflows became very strong, while with an increase in distance, the flows of the two jets had no effects on each other and their behavior was similar to that of a single synthetic jet in the flow field.

Jing-zhou et al. [11] studied the interaction of a synthetic jet in micro-channel cross-flow by changing the oscillation frequency along with orifice shape empirically and found that the frequency variations had a direct effect on the outlet velocity, while the orifice shape had a negligible effect on the velocity and significant effect on the interaction of synthetic jet and micro-channel cross-flow. Xu et al. [12] studied vortex formation caused by the outflow from the synthetic jet empirically. They stated that both the velocity in the direction of streamlines before the impinging and the radial velocity after the impinging had a

self-similarity behavior.

Yang and LiHao [13] studied the influence of orifice-to-wall distance on the vortex formation and its impinging to the wall empirically. They showed that variation of orifice-to-wall distance was effective for both strength and velocity of the formed vortices. Cadirci et al. [14] studied the impinging of outflow from a synthetic jet with the boundary layer formed on a flat plate and also investigated its effects on the separation of the boundary layer, displacement thickness and shape factor.

Despite all the studies about the effects of synthetic jet on heat transfer in micro-channels, the influence of interactions of synthetic jets in micro-channels are still unknown. In present study, effect of three synthetic jets, whose outflow interacted with the water cross-flow in a micro-channel, on the heat transfer increase and uniformity of temperature distribution on a hot surface is studied, and results are compared to the case of no synthetic jets in the micro-channel. A three-dimensional computational model was considered for two different arrangements of synthetic jets as well as two in-phase and π out-of-phase oscillatory conditions to better understand the effective characteristics in this process.

2. Governing equations

In the present study, the characteristics of flow and heat transfer are analyzed in a micro-channel, whose geometry is presented in Fig. 1.
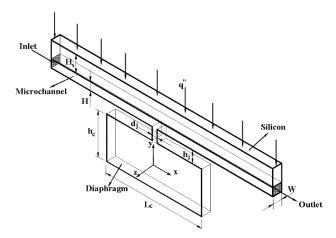

Fig. 1 Geometry of the micro-channel and synthetic jet

The governing equations for a three-dimensional incompressible turbulent fluid flow with constant properties include the continuity equation, the Navier-Stokes equations, and the energy equation are given as:

$$\nabla \boldsymbol{u} = 0 ; \tag{1}$$

$$\rho \frac{D\boldsymbol{u}}{Dt} = -\nabla p + \mu \nabla^2 \boldsymbol{u} ; \tag{2}$$

$$\rho c_v \frac{DT}{Dt} = k \nabla^2 T + \varphi; \quad \varphi = \tau_{ij} \frac{\partial u_i}{\partial x_j} , \tag{3}$$

where φ is the viscous dissipation term which is expressed in terms of unspecified viscous stress–tensor components τ_{ij}, ρ, c_v and k are the fluid density, the fluid specific heat and the fluid thermal conductivity, respectively.

Heat transfer equation in a solid (silicon) is written as:

$$\rho_s c_p \frac{\partial T}{\partial t} = k_s \nabla^2 T , \tag{4}$$

where ρ_s, c_p and k_s are the solid density, the solid specific heat and the solid thermal conductivity, respectively. Local Nusselt number based on channel hydraulic diameter is given as:

$$Nu_{D_H} = \frac{q''}{T_s - T_m} \frac{D_H}{k_l} , \tag{5}$$

where D_H is the channel hydraulic diameter and k_l is the fluid thermal conductivity. The local average temperature in any longitudinal section of the hot wall surface, T_s, is calculated as:

$$T_s = \frac{1}{W} \int_{-\frac{W}{2}}^{\frac{W}{2}} T \, dz . \tag{6}$$

The average temperature of the fluid in any section along the channel length, T_m, can be calculated using:

$$T_m = \frac{\int_A uT \, dA}{\int_A u \, dA} . \tag{7}$$

The standard deviation of the temperature which indicates the average temperature deviation of different regions on the hot surface from its average temperature is given as:

$$\sigma = \sqrt{\frac{\sum_{i=1}^{n} \left(T_s - T_{s\_ave} \right)^2}{n}} , \tag{8}$$

where n is the number of nodes along the hot wall and T_{s_ave} is the average temperature of the hot wall which can be calculated as

$$T_{s\_ave} = \frac{1}{LW} \int_0^L \int_{-\frac{W}{2}}^{\frac{W}{2}} T \, dz \, dx , \tag{9}$$

where L and W are length and width of the hot plate, respectively. In order to measure the amount of temperature uniformity improvement on the silicon hot surface, ratio of the average deviation during the jet operation to its value in the steady state is calculated as follows:

$$Relative \ Improvement = \frac{\sigma_{unsteady \ state} - \sigma_{Steady \ state}}{\sigma_{Steady \ state}} , \tag{10}$$

where unsteady and steady states refer to the states when synthetic jets are active and inactive, respectively.

3. Flow field and boundary conditions

In this study, flow field is analyzed using three-dimensional numerical methods. To achieve more accurate

results, the double precision solver and structured mesh with a dense grid on the regions with the probability of high gradient of flow characteristics is used.

In this study, a micro-channel with the total length of $L = 10$ mm is simulated in a three-dimensional space, in which three synthetic jets with two different arrangements (as shown in Fig. 2) are considered along the micro-channel.
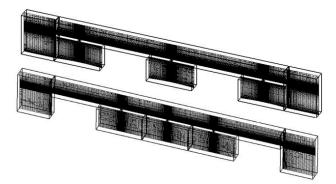

Fig. 2 Arrangement of the synthetic jets along the channel

Dimensions of all the synthetic jets, according to Lee et al. [9], are shown in Fig. 1. Slot width is $d_j = 50$ μm , slot height is $h_j = 100$ μm , channel height which is equal to the distance of jet outlet to the hot plate is $H = 200$ μm , channel width is $W = 200$ μm , length of the synthetic jet cavity is $L_c = 2$ mm , and height of synthetic jet cavity is $h_c = 1$ mm . Also, height for the silicon with heat flux on its outer surface is $H_s = 500$ μm and width equal to the channel width is considered for it.

Knudsen number for the water as the cooling fluid is calculated as 5×10^{-6} with respect to the smallest geometric characteristic as the outlet of synthetic jet. As a result, the continuity assumption for this flow field is established.

No-slip condition are considered at walls and all walls of the channel and synthetic jet are assumed to be adiabatic with the heat flux of equal to 600 kW/m², being applied to the silicon outer wall. The pressure difference of 750 Pa between inlet and outlet of the micro-channel causes the water flow through the channel. Inlet temperature of water is 293°K and all physical properties of water are assumed to be constant at the room temperature. To achieve accurate solutions in the laminar sublayer, the value of y^+ is approximately considered to be equal to one.

4. Numerical scheme

In order to analyze the flow field, SIMPLE algorithm was used for coupling the velocity and the pressure while the second order Upwind Method was used for the discretization of the convection terms in governing equations. In Accordance with numerous studies in this field, the four-equation SST turbulence model was used to analyze the turbulence in the flow field. After reaching the solution and convergence of the flow field in the steady state (no synthetic jets), the final solutions in this step were used as the initial solutions for the transient calculations. In

the transient calculations, at $t = 0$, the diaphragm began to oscillate from its stationary condition and a dynamic mesh was utilized to oscillate from a position with zero amplitude ($t^* = 0$) to the highest oscillation amplitude during the discharge stage ($t^* = 0.25$) according to:

$$Y = A \left[1 - \left(\frac{x}{\frac{L_c}{2}} \right)^2 \right] \left[1 - \left(\frac{z}{\frac{W}{2}} \right)^2 \right] sin\left(2\pi ft \right), \qquad (11)$$

where A and f are the oscillation amplitude and frequency, respectively. Also, t^* defined as:

$$t^* = \frac{t}{T} , \qquad (12)$$

is a dimensionless time parameter, where t indicates time and T represents the period of membrane oscillation.

In this study, synthetic jets were analyzed under two different arrangements and two in-phase and π-radian out-of-phase oscillation conditions. In the first oscillation condition, membrane of all of the three synthetic jets was in-phase and, in the second oscillation condition, actuator of the central jet oscillated π-radian out-of-phase with respect to two other jets.

In order to obtain the flow field characteristics, the continuity equation, the Navier-Stokes equations and the energy equation were solved. In order to ensure convergence of the energy equation, the average temperature variations of the silicon-fluid interface wall, were checked in each time step and the lack of change in the average temperature during several successive time steps was considered as a convergence criterion.

5. Results and discussion

5.1. Validation

In order to validate results of the present study, simulation results of the present work was compared to those of Lee et al. [9], and a good agreement was observed between the two. Dimensions of the synthetic jet considered in this work were similar to those considered by Lee et al. A synthetic jet operated in the middle of the micro-channel with length of 4 mm, interacting with the micro-channel cross-flow at the pressure difference of 750 Pa between its two ends.

The oscillation equation for the membrane of the two synthetic jets was similar to Eq. (11), and the maximum oscillation amplitude in the central line was 40μm with the frequency of 560 Hz. Comparison of the profiles of inlet and outlet velocities of the channel and also variations of the circumferential Nusselt number over solid-fluid interface wall indicate a very good agreement between the results. Fig. 3 shows the velocity profiles at different times of a complete oscillation at the channel inlet. As noted, there is a very good agreement between the results of the present study and those reported by Lee et al. [9].

In order to ensure the accuracy of the heat transfer results, variation of the circumferential Nusselt number is

presented in Fig. 4 for both studies, and good agreement of the results indicated reliability of the methodology and the energy equation analysis of the flow field used in the present work.
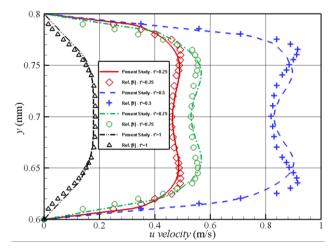

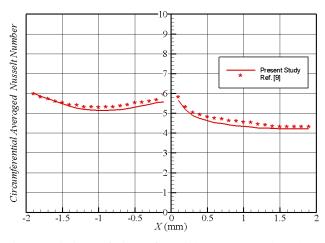Fig. 3 Inlet velocity profile of the micro-channel at different times

Fig. 4 Variations of circumferential Nusselt number along the micro-channel length

As indicated in the figure, due to thin boundary layer, Nusselt number in the micro-channel inlet is higher than other regions in the micro-channel, and the Nusselt number was reduced with the growth of the boundary layer. However, in the middle part of the channel there is an increase in Nusselt number due to the synthetic jet operation which causes turbulence in the boundary layer, and increased momentum of the flow field as a result.

5.2. Flow characteristics

When the central jet operates at its highest amplitude during the discharge stage with the oscillation amplitude of $A = 40$ μm and frequency of $f = 560$ Hz, streamlines are shown in the following figures for two different arrangements. In-phase or out-of-phase states of the synthetic jets are compared based on the central jet. In the in-phase state, all the three synthetic jets simultaneously operate in the suction and discharge stages, while in the out-of-phase state, the central jet has the π-radian phase difference with respect to the two side jets such that, when the central jet operates at its highest discharge amplitude, two

side jets are operating at the lowest suction amplitude, and vice versa.

Figs. 5-8 demonstrate vortices formation in the synthetic jets outlet and their movement in the two arrangements and two in-phase and out-of-phase states. In the in-phase state, movement of the vortices in the side jets is almost symmetric, while in out-of-phase state, their movement is almost similar.
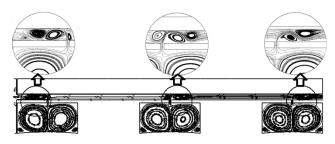

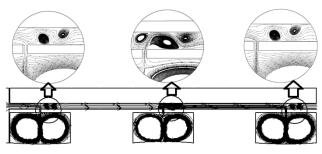Fig. 5 Streamlines of in-phase state for the first arrangement, $t^* = 0.25$

Fig. 6 Streamlines of out-of-phase state for the first arrangement, $t^* = 0.25$
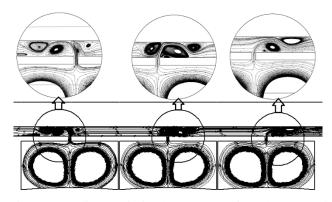

Fig. 7 Streamlines of in-phase state for the second arrangement, $t^* = 0.25$
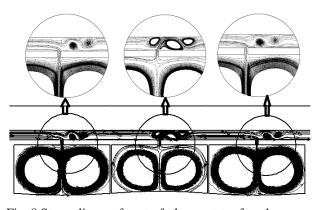

Fig. 8 Streamlines of out-of-phase state for the second arrangement, $t^* = 0.25$

Figs. 9 and 10 show temperature variations over the heated/cooled silicon-water interface along the micro-channel for different synthetic jets arrangements. As shown in these figures, a significant decrease in the temperature of the wall is observed compared to the case when the synthetic jets are absent. The temperature decrease for the first arrangement in the in-phase and π-radian out-of-phase cases is about 21% and 25%, respectively. For the second arrangement, the decrease in temperature is 24% and 28% for in-phase and π-radian out-of-phase cases, respectively.
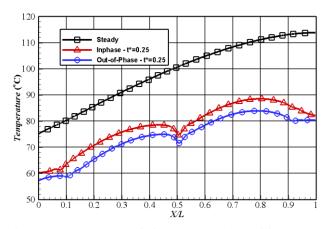

Fig. 9 Temperature variations over the silicon-water interface wall for the first arrangement of the synthetic jets
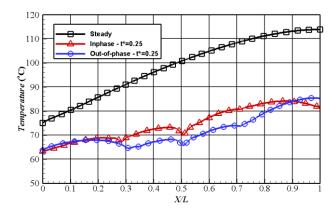

Fig. 10 Temperature variations over the silicon-water interface wall in the second arrangement of synthetic jets

Figs. 11 and 12 represent variations of Nusselt number over the silicon-water interface wall for two arrangements of synthetic jets. A significant increase in the Nusselt number during the synthetic jets operation compared to the case when the jets were not operating is clearly shown in the figures. The increase in the first arrangement is 64% and 82% for in-phase and out-of-phase, respectively, while in the second arrangement, it is 73% and 92% for in-phase and out-of-phase, respectively.

Effects of changes in the arrangements and difference of oscillation phases of the synthetic jets on temperature distribution uniformity improvement over the silicon-water interface have been evaluated. The results have shown that improvement in the first arrangement, compared to the case when the jet is not operating, is very close to each other and about 34% in both in-phase and out-of-

phase states, while in the second arrangement, it is 47% for both in-phase and out-phase states.
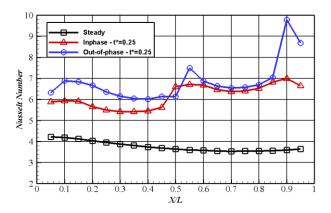

Fig. 11 Variations of Nusselt number over the silicon-water interface wall in the first arrangement of synthetic jets
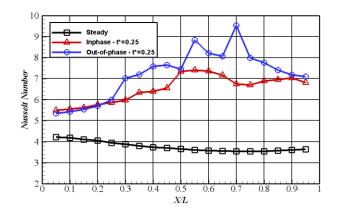

Fig. 12 Variations of Nusselt number over the silicon-water interface wall in the second arrangement of synthetic jets

6. Conclusion

In this study, a three-dimensional numerical simulation of flow and heat transfer was carried out in a micro-channel exposed to the operation of three synthetic jets with two arrangements and different oscillation phases. Synthetic jets interacted with the micro-channel cross-flow induced by applying pressure difference between its inlet and outlet. Results indicate decreased average temperature, increased Nusselt number, and improved temperature distribution uniformity over the solid-fluid interface wall compared to the case when the synthetic jets were not operating. Oscillation of the synthetic jets changes the flow velocity at the channel inlet and outlet. Operation effects of synthetic jets on heat transfer in the micro-channel inlet are negligible due to the lack of growth of boundary layer; therefore, appropriate arrangement of the synthetic jets at appropriate distance from the micro-channel inlet is very significant. In-phase and out-of-phase oscillations of the actuators of the synthetic jets are also effective in reducing temperature and improving temperature distribution uniformity over the solid-fluid interface wall.

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EFFECTS OF ARRANGEMENT AND PHASE DIFFERENCE OF OSCILLATION OF SYNTHETIC JETS ON HEAT TRANSFER IN MICRO-CHANNELS

S u m m a r y

In this study, effects of two different arrangements and π-radian phase difference in the oscillation of membranes of three synthetic jets, in a cross-flow of a micro-channel, on heat transfer and temperature distribution uniformity of a hot wall, located at certain distance from the synthetic jet outlet, were numerically studied by a three-dimensional model using dynamic mesh. Obtained results show that synthetic jets provide better cooling for the hot plate; however, the position and phase difference of the oscillation in the actuators of the synthetic jets are effective on the degree of cooling improvement. Depending on the position and operating conditions of the synthetic jets, reduction in the average temperature of the hot plate and increase in the Nusselt number vary from 21% to 28% and from 64% to 92% as compared to micro-channel without synthetic jet, respectively. In addition, operation of the synthetic jets improved temperature uniformity over the hot wall and improvement varies from 34% to 47% as compared to case of no synthetic jet in the micro-channel.

Keywords: Micro-channel, Synthetic jet, Numerical, Heat transfer.