Heat Transfer Enhancement using Synthetic Jet Actuators in Forced Convection Water Filled Micro-Channels

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Abstract: - The effect of a synthetic jet on heat transfer rates in forced convection of water in micro-channels has been studied numerically. Unsteady computations of laminar flow have been performed for a two-dimensional micro-channel, 200 μm high and 4.2 mm long, open at either end with the top surface hot and all other solid wall adiabatic. The orifice width was 50 μm, while the length of the orifice passage was 100 μm. The diaphragm was assumed to oscillate sinusoidally in time and parabolically in space. The effectiveness of the proposed cooling procedure has been evaluated by comparing the heat transfer rates with and without the synthetic jet. A maximum heat transfer enhancement of approximately 125% was achieved.

Key-Words: - synthetic jet, micro-channel, heat transfer, numerical simulations

1. Introduction

The increasing demands placed on micro-processors and other integrated electronic devices have resulted in higher and rapidly increasing rates of heat generation. As a consequence heat transfer has been and continues to be one of the limiting constraints on the design of very large scale integrated systems-on-chip.

Much research has been done improving the heat removal from chips [1], so that improved methods of heat removal from a single chip are becoming available. Traditionally, air has been the preferred fluid for cooling electronic components. However, the low heat transfer coefficient, coupled with a low specific heat capacity in addition to its low density makes air as a poor choice with heat fluxes going well beyond $1MW/m^2$. Upadhye and Kandlikar [2] demonstrated that direct cooling of an electronic chip using flowing water in micro-channels was possible and the pressure drop is a strong function of the channel geometry and that micro-channels must be optimized to match the given heat transfer and pressure drop requirements. Nicole *et al.* [3] showed experimentally that very high cooling rates could be achieved with flowing water as the coolant in silicon micro-channels. Thus, it is not only possible, but also desirable to pump a liquid such as water to provide cooling directly on an electronic chip.

Since the flow regime in micro channels is invariably laminar, the heat transfer rate is much lower than in turbulent flow and needs to be enhanced to dissipate the necessary large amount of heat. Because conventional jet impingement cooling requires a source of pressurised fluid and complex fluid packaging, the alternative of using synthetic jets in micro channels was suggested in our previous work [4]. Synthetic jets [5] are formed from an ingestion and expulsion of fluid through an orifice into a fluid-filled space. An oscillating diaphragm located in the cavity in the face opposite the orifice provides the motive power for the actuator. When "steady conditions" on the average over a cycle of oscillation of the diaphragm have been reached, there is a zero nett mass flow, but a non-zero nett momentum transfer from the actuator. Thus, under appropriate operating conditions this actuator provides a fluctuating flow away from the orifice into the surrounding medium, thereby creating a quasi-turbulent flow. As a result it may be possible to significantly increase the rate of heat transfer when this jet impinges on a hot surface.

Kercher *et al.* [6] showed that the performance of micro synthetic jet coolers is comparable to that of commercially available electronic cooling fans and that micro jet based cooling devices may be especially attractive in cooling "hot-spots".

We have established [7] that the introduction of synthetic jets into an air filled micro channels results in the creation the self sustaining vortices which remain essentially stationary in the main flow direction, but whose strength varies significantly during a cycle of the diaphragm. The vortices are

generated and reinforced during the expulsion phase of the cycle by the reflection of the synthetic jet from the solid surfaces of the channel; a mechanism entirely different from that of a synthetic jet in an infinite medium [8]. The non-dimensional parameters used in the analysis of synthetic jet actuators were the jet Reynolds number, $Re_j = \overline{V}_j d / \nu$, in which, $\overline{V}_j$, is the average orifice expulsion velocity, d is the orifice width, and ν is the kinematic viscosity, the Stokes number, $S = \sqrt{\omega\, d^2 / \nu}$, in which $\omega = 2\pi f$ is the angular velocity and f is the frequency of the diaphragm motion and the Strouhal number, $St = \omega d / \overline{V}_j$. The maximum heat transfer enhancement achieved with air was approximately 35%.

Since liquids are likely to be used as the cooling medium in the future [2], in this paper we investigate a two-dimensional channel in which water is flowing in order to establish the effectiveness of synthetic jets when liquids are used for cooling in micro-channels. The effectiveness of the proposed cooling procedure is evaluated by comparing the heat transfer rates with and without the synthetic jet and with channels handling air.

2. Mathematical and numerical model

A two-dimensional micro-channel 200 μm high and 4.2 mm long has been used. It is open at either end with the top surface hot and all the other walls adiabatic. Parts of the regions outside the channel have been included in the calculations as may be seen in Figure 1.

The transient incompressible flow generated by a constant pressure difference between the ends of the channel and by the synthetic jet actuator has been simulated using the commercial package, CFX-5.7. The conservation equations consisting of the equation of continuity

$$\nabla \cdot \tilde{U} = 0 , \tag{1}$$

the Navier-Stokes equation,

$$\rho\frac{\partial \tilde{U}}{\partial t} + \rho\nabla \cdot (\tilde{U}\tilde{U}) = -\nabla p + \mu\nabla^2\tilde{U} + \tilde{B} \tag{2}$$

and the energy equation,

$$\frac{\partial \rho h}{\partial t} + \nabla \cdot (\rho\tilde{V}h) - \nabla \cdot (k\nabla T) = 0 , \tag{3}$$

in which, ρ, $\tilde{V}$, p, μ, h, T, k, and t denote density, the velocity vector, the gauge pressure, dynamic viscosity, enthalpy, absolute temperature, thermal conductivity and time.
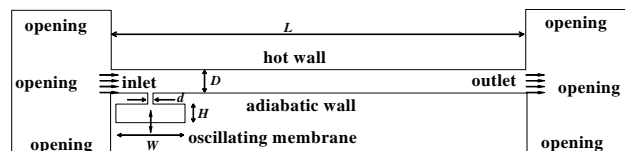

Figure 1. Sketch of computational domain.

The displacement of the membrane Y_m was assumed to be a parabola which varies sinusoidally in time, viz,

$$Y_m = A(1 - (2x/W)^2)\sin(2\pi f t) \tag{4}$$

in which A is the centreline amplitude, f is the frequency of oscillation and W is the width of the diaphragm.

A second order backward Euler differencing scheme was used for the transient term, whereas a second order upwind differencing scheme was employed for the advection terms in the Navier Stokes equation. At each time step of a cycle, the internal iterations were continued until the mass and momentum residuals had been reduced to 10^{-6}. 50 x 20 grid points in the stream-wise and transverse directions respectively were used in the orifice of the synthetic jet generator, so that mesh size was kept to 2.5 μm in the orifice. Outside the orifice grid was gradually expanded with maximum mesh size equal to 5 μm. The total number of mesh points was equal to 164316.

3. Results and discussion

To study the abilities of the synthetic jet to disrupt laminar flows in micro channels, an orifice 50 μm wide and 100 μm long was placed 1.2 mm downstream from the inlet. The width of the
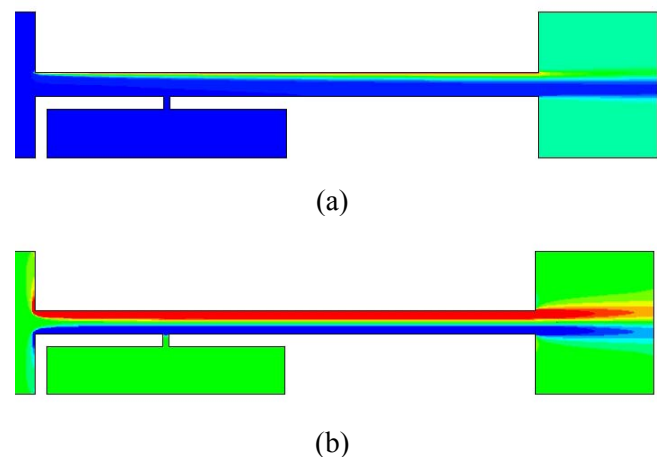

(a)

(b)

Figure 2. Temperature (a) and vorticity (b) contours in the case of steady flow.

diaphragm was 1 *mm* and the cavity depth was 250 μm. Since the inflow temperature and velocity distributions affect the heat transfer in the channel, external domains 1 *mm* long and 1.5 *mm* high were included at either end of the channel (Figure 1). A 750 *Pa* pressure was imposed on the inlet of the left external domain with the temperature set to 20º C. At the outlet of the right external domain, the pressure was set at zero with the absolute value of the reference pressure, p_{ref}, fixed at 100 *kPa* and the temperature set to 30º C. The upper wall of the channel was isothermal at 50º C.

The effectiveness of the proposed cooling strategy was evaluated by comparing the heat transfer rates with the synthetic jet operating with that obtained with a steady flow generated when a pressure difference of 750 *Pa* only acts between the ends of the channel. A pressure difference of 750 *Pa* between the ends of the channel was chosen in order to obtain the channel Reynolds number equal to ~110 as it was in the case of air [4] in order to compare heat transfer rates with channels handling air. The channel Reynolds number is defined as $Re_{ch} = \bar{U}_{ch}D/\nu$ where $\bar{U}_{ch}$ is the average velocity in the channel, which in this case was 0.49 *m/s*, *D* is the channel height and ν is the kinematic viscosity. Temperature and vorticity contours generated in this steady flow may be seen in Figure 2. The horizontal contours of vorticity indicate a laminar, parabolic flow with no mixing. The average heat transfer rate from the hot upper wall in the case of steady flow of water was 4.03×10^5 *W/m²* whilst for the air in steady case it was equal to 7.28×10^3 *W/m²* [4].

After obtaining the steady flow results, the synthetic jet was switched on by simulating the displacement of the membrane with amplitude of 40 μm at a frequency of 560 *kHz*. These forcing membrane parameters were chosen to produce the synthetic jet with Reynolds and Stokes numbers similar to those obtained for air in [4]. By actuating with these forcing membrane parameters, synthetic jet with Reynolds numbers of 134 and Stokes number equal to 3.2 was produced.

Figures 3-4 show variation of the average heat transfer rates from the heated surface for air and water filled micro-channels for one cycle of actuation. The non-dimensional time is given by the ratio of the physical time to the period of oscillation. It can be seen that in the case of water substantially larger increase in the heat transfer can be achieved comparing with air. When synthetic jet is switched on the improvement in heat transfer is up to 100% comparing with steady flow whilst for air we obtained maximum 35% increase.
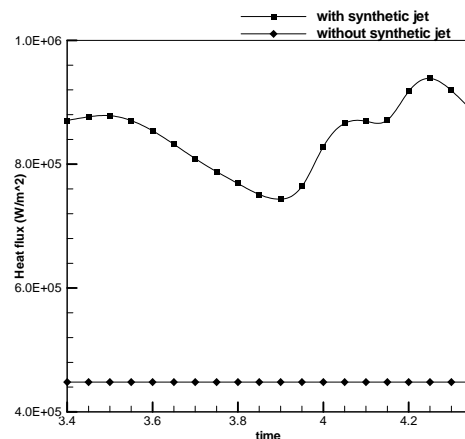

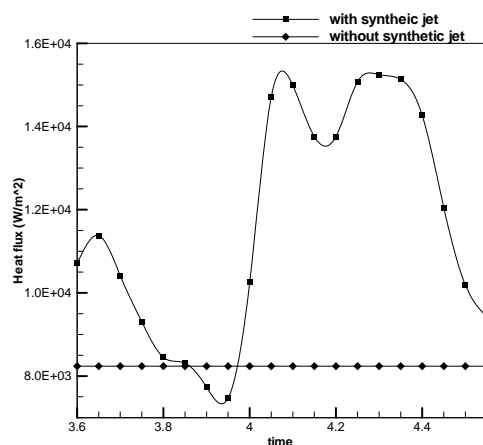Figure 3. The heat transfer rates for channel filled with water

Figure 4. The heat transfer rates for channel filled with air

As we have proved that cooling rates achieved with flowing water as the coolant are much higher then with air our further investigation will focus on the effect of synthetic jets *Re* number on the heat transfer in water filled micro-channels. By actuating with the displacement of the membrane with amplitudes of 20, 60 and 80 μm synthetic jets with Reynolds numbers of 67, 201 and 270 were produced in addition to previous results obtained for Re_j=134.

Examples of the computed instantaneous vorticity contours at different instants in the fourteenth cycle may be seen in Figures 5-6 for Re_j= 134 and Re_j= 270 correspondingly. The vorticity contours in Figures 5a, 6a occur at time *t*=14, when the diaphragm is moving at its maximum velocity towards the orifice; thereby resulting in the maximum expulsion velocity of the jet. At this stage for Re_j=134 the water from the orifice has reached the upper wall of the channel leading to a disturbance of the boundary layer. For Re_j=270 the impinging jet has been already reflected
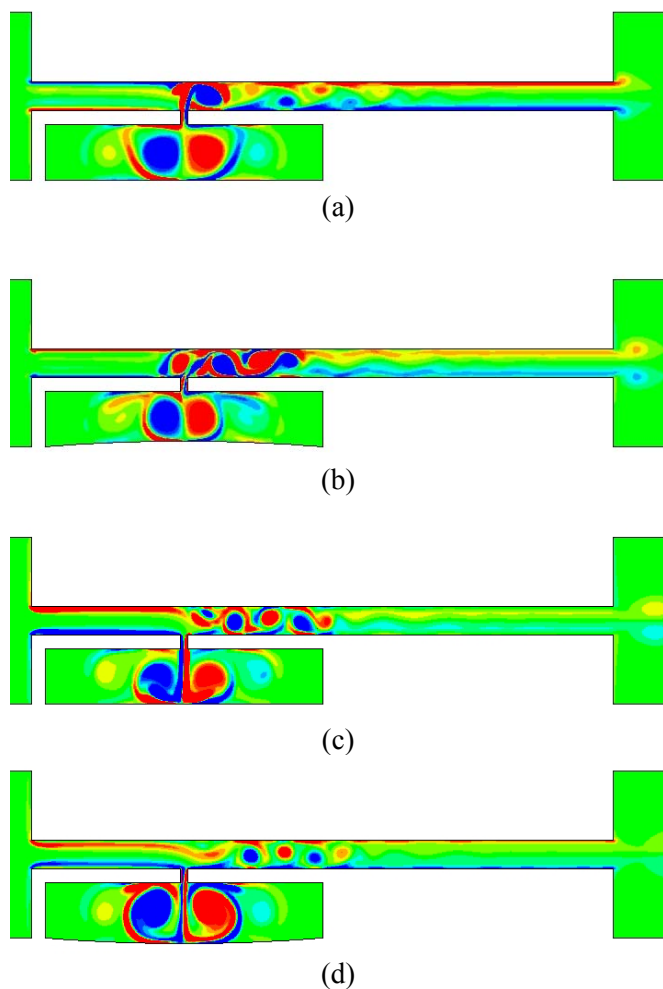
(a)

(b)

(c)

(d)

Figure 5. Vorticity contours at (a) t=14.0, (b) t=14.25, (c) t=14.5 and (d) t=14.75 for Re_j=134.
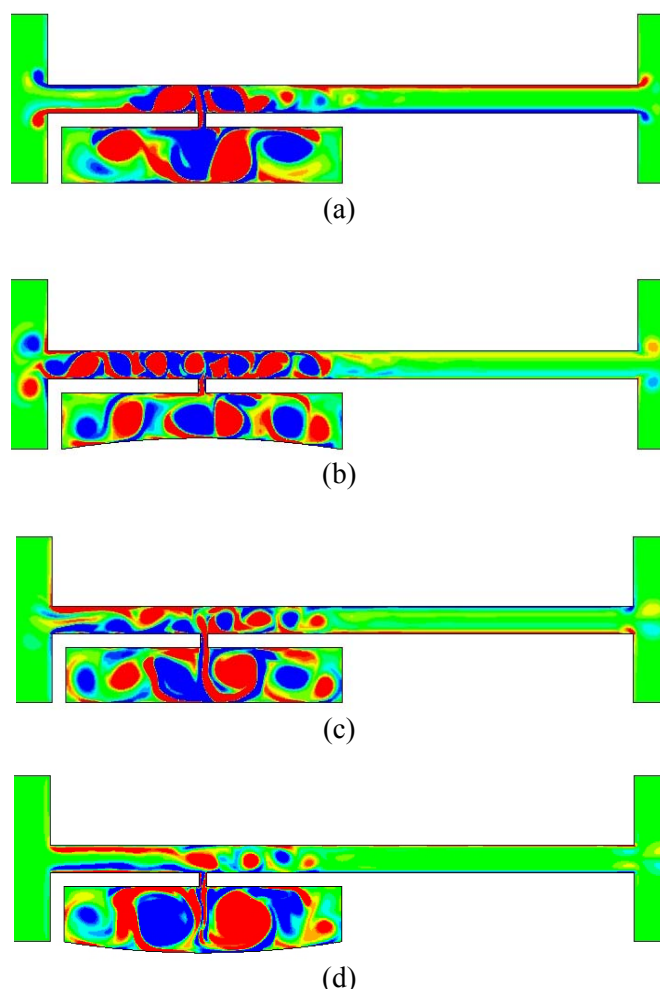
Figure 6. Vorticity contours at (a) t=14.0, (b) t=14.25, (c) t=14.5 and (d) t=14.75 for Re_j=270.

from the upper surface forming a counter clockwise vortex between the two streams. By the end of expulsion stage in both cases (Figures 5b, 6b), this reflection is repeated a number of times so that a series of geared vortices in the manner similar to what was observed for air [4] are created which further disrupt the boundary layer. It can be seen that with increasing of Re number jet spreads into larger area in the upstream direction during the expulsion stage and affects not only the flow downstream from the orifice but also the upstream flow. This is obviously a result of the increasing the jet to cross-flow velocity ratio.

During the suction stage the vortices persist but weaken as time progresses while the axial velocity is reduced, Figures 5c-6c and 5d-6d. It should be noted that disturbance in the case of Re_j=134 propagates further downstream then in the case of Re_j=270. This can be explained by the decrease of the mean through flow velocity in the channel from ~0.5 ms^{-1} in the case of Re_j=134 to ~0.2 ms^{-1} in the case of Re_j=270 so that vortices in the latest case can not move appreciably.

However, since they persist for the whole cycle, just as in the case of a synthetic jet in a stationary atmosphere, the boundary layer is disturbed for the whole cycle. It should be noted that while for Re_j=67 and 134 the velocity fluctuations repeat almost exactly after a transient of a couple of diaphragm cycles, with increasing Re_j, particularly at 270, the velocity, whilst retaining the fundamental frequency of the diaphragm oscillations, has an amplitude which appears to be chaotic.

The heat flux distribution along the surface at the time instants shown on Figures 5-6 is given on Figure 7. A large increase in the surface heat flux on the downstream side from the orifice (x $\cong$ 100μm) which can be seen on Figure 7a is caused by the impingement of the cold expelled fluid on the heated surface (see Figure 5a, 6a) at the stage when the velocity of this expelled fluid has the highest momentum. Due to larger spreading of the jet for Re_j=270 the area where instantaneous heat flux has a maximum value increases while the actual value of
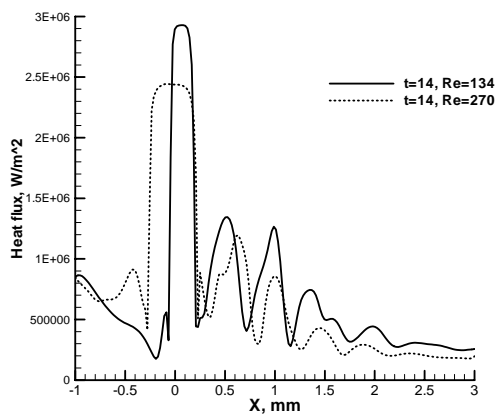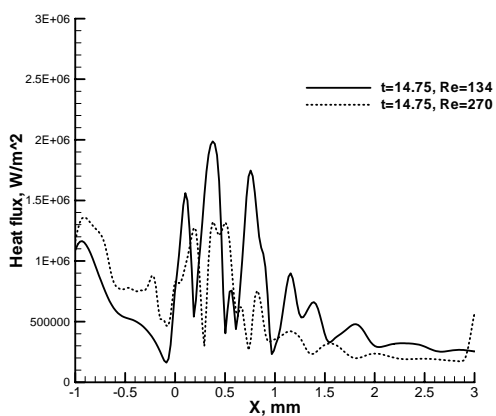
(a)
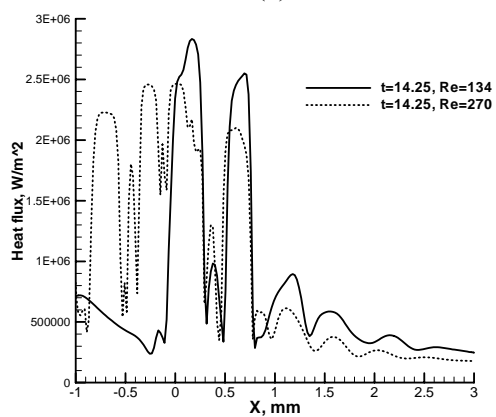
(b)
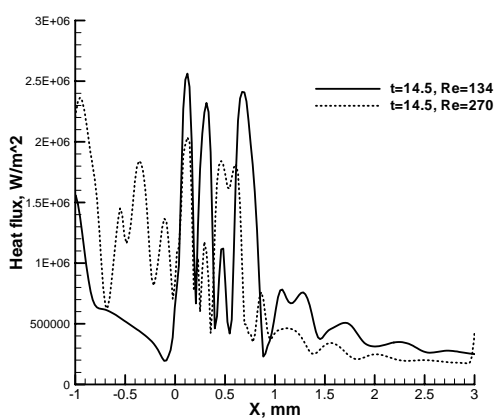
(c)

(d)

Figure 7. Heat flux along the surface at (a) t=14.0, (b) t=14.25, (c) t=14.5 and (d) t=14.75 for Re_j=134 and Re_j=270.
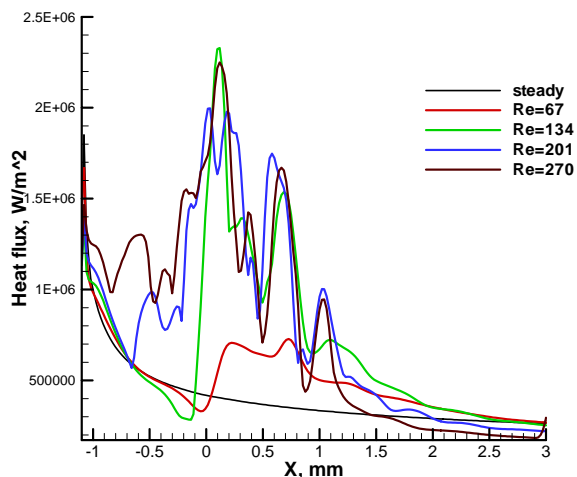

Figure 8. Variation of the time averaged heat flux along the top heated surface for different Re numbers
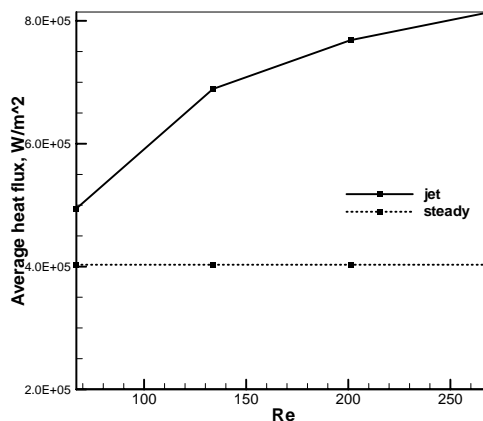

Figure 9. Variation of the averaged over cycle and over the top surface heat flux with Re number

this maximum instantaneous flux decreases. The most noticeable difference in the heat transfer rate can be seen on Figure 7b in the area upstream from the orifice where a series of created in the case of Re_j=270 geared vortices cause significant increase comparing with Re_j=134.

During the suction stage shown on Figure 7c, d fluctuations of the heat transfer decrease in the magnitude thus reflecting the weakening of the vortices. As in the case of Re_j=134 the disturbances propagate further downstream then in the case of Re_j=270 the local heat flux in this case is higher further downstream. Variation of the time averaged heat flux along the top heated surface for different Re numbers can be seen on Figure 8. The local increase in the time averaged surface heat flux on the downstream side from the orifice caused by the first impingement of the cold expelled fluid on the heated

surface for Re_j 134 is 560% and for Re_j 270 is 450% bigger than in the steady case.

This significant increase in the heat transfer rate can be used for cooling "hot-spots" [6]. Secondary maxima in surface heat flux can be related to the increase in heat transfer caused by disruption the boundary layer by the series of geared vortices created during expulsion stage (Figure 5b, 6b) and also the remnants of the jet, sustained by the weakening vortices during suction stage (Figure 5c-d, 6c-d). It should be noted that for $x>1.5$ *mm* a surface flux remains the same as in steady case or even decreases slightly at higher Re_j.

The variation in the heat transfer flux averaged over a cycle and over the whole hot surface (4.2 *mm*) is presented in Figure 9. It can be seen that the average heat transfer rates are substantially greater than the steady values. The decreasing gradient of the curve with increasing Re_j indicates that there may be a limiting value of the heat flux enhancement which can be achieved using this particular set of parameters. The maximum heat transfer enhancement in the case of Re_j 270 is approximately 125%; a very significant increase.

Conclusions

A two-dimensional channel in which water is flowing has been investigated numerically. We have shown that in case of introducing synthetic jet, the maximum local rate of heat transfer can be increased by 450% - 560% when averaged over a synthetic jet cycle, which may be especially important in cooling local hot spots. If averaged over the region of action of the jet as well as over a cycle the heat transfer rate has been increased by 100% whereas in air at the same jet Reynolds number the heat transfer was significantly less and could only be enhanced by 35%.

Acknowledgments

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