Transport Phenomenon of Simultaneously Developing Flow and Heat Transfer in Twisted Sinusoidal Wavy Microchannel under Pulsating Inlet Flow Condition

Sampad Gobinda Das\textsuperscript{a}, Suvanjan Bhattacharyya\textsuperscript{b}, Himadri Chattopadhyaya\textsuperscript{a}, and Ali Cemal Benim\textsuperscript{c}

\textsuperscript{a}Mechanical Engineering Department, Jadavpur University, Kolkata, West Bengal, India; \textsuperscript{b}Department of Mechanical Engineering, Birla Institute of Technology and Science Pilani, Pilani Campus, Vidyadhar Vihar, Pilani, Rajasthan, India; \textsuperscript{c}Center of Flow Simulation, Department of Mechanical and Process Engineering, Dusseldorf University of Applied Sciences, Dusseldorf, Germany

\textbf{ABSTRACT}

Heat transfer operations in microchannel are of special relevance in designing micro-devices involving accumulation of high heat flux. The current study explores simultaneously developing, unsteady laminar flow inside a twisted sinusoidal wavy microchannel and heat transfer involved. A square cross-sectioned channel is taken for the study. The channel is wavy in nature as well as twisted about the flow axis. The modified geometry helps in enhancing out of plane mixing of the fluid and formation of recirculation inside the flow. The evaluation is performed for a Reynolds number range of 1–100. A sinusoidal (varying with time) normal velocity component is employed at the inlet. The study was conducted over a range of pulsation amplitude and frequency. To solve the governing equations, a finite volume based method was employed. The Nusselt number data shows significant enhancement for the sinusoidal inlet velocity as compared to that of a steady case. In terms of pressure drop, the proposed design shows that at different frequencies and amplitudes, it can keep the pressure drop within admissible limits. Entropy generation is used as the measure of dissipated energy. Performance in terms of entropy generation is also reported.

\textbf{Introduction}

The evolution of science and technology brought us in an era where extensive use of energy enforcing us to look into new and more environment friendly roadways of producing and distributing energy. To accomplish the modern day demand, the thermal management of devices, which affects the efficiency of the system, is becoming point of attraction day by day. It is observed by many researchers that, reduction in channel dimensions, lead to raise in heat transfer (HT) coefficient, thus improving heat-sinking capability. Broad researches on microchannel heat exchangers (MCHE) has been done till date. With the enhancement in micro-fabrication technology, use of micro-fluidic systems is increasing day by day along different strides of science and technology. Microchannel heat sinks render an innovative cooling technique as they are capable of dispersing huge heat via a small area. Thus, microchannel heat exchangers can be considered as a potential replacement of conventional heat exchangers. In microchannel heat exchangers (HE), flow is generally laminar in nature and the heat transfer co-efficient is found to be proportional to velocity. They provide a great surface area to volume ratio, high convective HT coefficient, small mass and volume, while working with modest coolant inventory. These features furnish them very suitable for cooling devices such as high performance microprocessors, laser diode arrays, radars, and high-energy-laser mirrors. They are easy to maintain and offer increased latent heat capacity for microchannel evaporators. They offer upgraded energy efficiency, reaction speed growth and yield, safety, reliability, scalability, on-demand production, and excellent degree of process control. There are more applications in various fields of science and technology such as biotechnology, chemical engineering, electronic cooling technologies, sensing technologies etc.

To enhance the heat transfer rate, various methods are employed. They can be broadly classified as active methods, and passive methods. Active techniques are
et al. [5] showed incongruity with conventional flow work on microchannel flow characteristics by Peng et al. on microchannel heat exchanger (MCHE). Experimental effects of frequency and pulsation amplitude on thermal performance of MCHE were investigated. They found augmented performance in microchannel diameters ranging from 133 to 367 \( m \). Meinhart et al. [6] organized particle image velocimetry (PIV) experiments, and measured velocity fields with order 1 \( \mu m \) spatial resolution. They reported ensemble averaged velocity profiles in a \( 30 \times 300 \mu m \) rectangular microchannel. Pfund et al. [7] studied flow visualization experimentally at low \( Re \) to check critical \( Re \) for transition regime. Morini and Spiga [8] experimented on the influence of viscous dissipation in liquid flow through microchannels. They found that this viscous dissipation predominates the flow at a hydraulic diameter less than 100 \( \mu m \). Experimental results from Xu et al. [9] on microchannel having hydraulic diameter of 344 \( \mu m \), exhibited well agreement with conventional theory. They investigated a large Reynolds number range, extending from quite low values up to the turbulent regime (20–4000). Liu and Garimella [10], conducted flow visualization experiments with Reynolds number (\( Re \)) ranging from 230 to 6500. Their test setup included channels with varying hydraulic diameters (244–974 \( \mu m \)). Using flow visualization technique, they gathered onset of turbulence and pressure drop data and compared results with numerical calculations. Friction factor (\( f \)) measurement of isopropyl alcohol and silicon oil flow along microchannels were presented by Pfahler et al. [11]. Their experimental results indicated concurrence with conventional theory for large channels. However, for small channels, their

surface vibration, electric field, flow pulsation and variable roughness structures. Passive techniques include geometric alteration of the flow passage and use of inserts, surface tempering, out of plane mixing and use of additives etc.

The area of microchannel heat transfer research has been explored largely till date. Tuckerman and Pease [1] works on trailblazer and received thermal resistance of around 0.09 °C/W as a result. Rectangular microchannel having depth of 302 and 50 \( \mu m \) width was their designed geometry. They employed constant heat flux of 790 W/cm² and water as the cooling medium. Kandikar et al. [2] summarized HE characteristics and the fluid flow behavior for micro- and minichannels. Although heat transfer enhancement using pulsatile velocity is being investigated since years [3], the effect of it on thermal performance of microchannel received attention only in recent times. Nandi and Chattopadhyay [4] studied and found augmented performance in microchannel HT because of inlet pulsation. It is also assessed the effect of frequency and pulsation amplitude on thermo-hydraulic performance of MCHE. Experimental work on microchannel flow characteristics by Peng et al. [5] showed incongruity with conventional flow theory. They worked on microchannels of hydraulic diameters ranging from 133 to 367 \( \mu m \). Meinhart

### Nomenclature

| Symbol | Description |
|--------|-------------|
| \( A \) | dimensionless wave amplitude |
| \( A_w \) | wave amplitude of channel geometry, \( m \) |
| \( b \) | breadth of the channel cross-section, \( m \) |
| \( D_h \) | hydraulic diameter, \( m \) |
| \( f \) | friction factor |
| \( f_r \) | frequency of velocity pulsations, Hz |
| \( HE \) | heat exchanger |
| \( h \) | heat transfer coefficient, \( W \ m^{-2} K^{-1} \) |
| \( HT \) | heat transfer |
| \( k \) | thermal conductivity, \( W \ m^{-1} K^{-1} \) |
| \( L_0 \) | thermal development length, \( m \) |
| \( L \) | length of the channel, \( m \) |
| \( MCHE \) | microchannel heat exchanger |
| \( N_{e,t} \) | augmentation entropy generation number |
| \( Nu \) | Nusselt number |
| \( p \) | pressure, Pa |
| \( PIV \) | particle image velocimetry |
| \( Pr \) | Prandtl number |
| \( q \) | heat flux, \( W \ m^{-2} \) |
| \( Q \) | heat flow rate, \( W \) |
| \( Re \) | Reynolds number |
| \( S \) | twisting pitch, \( m \) |
| \( S_{gen} \) | thermal entropy generation, \( W \ K^{-1} \) |
| \( S_{gen,0} \) | thermal entropy generation for steady-state flow, \( W \ K^{-1} \) |
| \( St \) | Strouhal number |
| \( T \) | temperature, K |
| \( TWSMC \) | twisted sinusoidal wavy microchannel |

### Greek symbols

| Symbol | Description |
|--------|-------------|
| \( \Delta p \) | pressure drop, Pa |
| \( E \) | offset of channel cross section center from main axis, \( m \) |
| \( \eta \) | thermal performance enhancement factor |
| \( \mu \) | dynamic viscosity, \( kg \ m^{-1} s^{-1} \) |
| \( \rho \) | density, \( kg \ m^{-3} \) |
| \( \lambda \) | wavelength, \( m \) |

### Subscripts

| Symbol | Description |
|--------|-------------|
| \( 0 \) | steady case |
| \( B \) | fluid bulk quantity |
| \( i \) | inlet value |
| \( o \) | outlet value |
| \( w \) | wall |
result was much below from the theoretical value. Klank et al. [12] used micro-PIV to investigate three-dimensional flow behavior of micro-cell sorters and mixing structures. Experiments on smooth trapezoidal silicon microchannels having different aspect ratio were done by Wu and Cheng [13]. Their result indicated that microchannels having same hydraulic diameter but of different cross-sections can exhibit dissimilar friction factor results. If the flow is enforced to go past a curved path, recirculation zones can be formed, which may intensify fluid mixing and thus, improving HT.

Joshi and Wei [14] investigated the laminar flow for slightly tapered silicon microchannels with hydraulic diameter of 53–112 μm. They used micro-PIV for the purpose. The position of the maximum velocity was deviating from the mid-plane along the depth direction as per their findings. They indicated the wall taper as the reason behind it. They also had done numerical simulations to examine the influence of the sidewall angle on flow and HT. Mohammed et al. [15] tested friction factor and HT along a wavy microchannel. They found that with rise in wave amplitude, the friction penalty also increases. The influence of roughness on microchannel HT was investigated by Croce et al. [16]. An array of three-dimensional conical peaks spread over smooth surfaces of a plane microchannel was employed to increase the roughness. They considered different heights and different arrangements of the peaks on a range of Reynolds numbers. They found that the pressure drop may remarkably be affected by the surface roughness. Valencia [17] experimented with flow in the turbulent region. He [17] reported a thermal performance augmentation of 30%, coupled with drastic increase in the friction factor. Arkilik and Schmidt [18] performed analytical as well as experimental study of gas through microchannel. They [18] employed first order slip velocity boundary condition. They found both compressibility and rarefied effects. Numerical simulation of Patankar et al. [19] for ducts with periodically varying cross-section displays that due to the varying cross section of the channel, huge recirculation zones are created. Mala and Li [20] tested circular micro-tubes with diameters ranging from 50 to 254 μm and of high relative roughness. They found an increased friction factor in the laminar regime with increase in the relative roughness. Tu and Hjik [21] performed friction factor experiments on microchannels. Their results were compatible with the laminar theory. However, one of the microchannels they investigated had a major surface roughness which exhibited significant increase in friction factor. Avci and Aydin [22] worked on an open ended vertical parallel plate microchannel with asymmetric wall heating. Velocity slip and temperature jump at wall were also taken into account. They presented the exact analytical results for fully developed mixed convective heat transfer.

Chai et al. [23] investigated both experimentally and numerically, flow and HT in a silicon heat sink with 10 parallel trapezoidal microchannels. The channels were of hydraulic diameter 127.4 μm. From the velocity distribution of the numerical models, they found that the velocity develops a little more rapidly in this case. This causes the pressure drop to be a bit lower than that of the neglected entrance and exit plenum. They also found that the thermal resistance decreases with increase in the heating power at the substrate, when pumping power is kept constant.

Islamoglu and Parmakasizoglu [24] examined HT in a periodically corrugated duct both numerically and experimentally. They used a commercial computational fluid dynamics code for the purpose and compared the results with experimental outcome as well as with the pre-established experimental correlations. Their numerical result showed good concurrence with the experimental data and with the previous experimental correlations. At $Re = 400$, they found more than 200% enhancement in thermal performance.

Micro-pin fin configurations with different fin shapes were tested numerically by Izci et al. [25] with $Re$ ranging between 20–120. They compared their results with previous works and received well agreement. They concluded that the rectangular shaped pin fin is furnishing best results in terms of convective HT coefficient and $Nu$ values. They claimed its larger HT area and efficiency of flow separation as the reason for it. However, the rectangular shape was also exhibiting the maximum pressure drop penalty. Overall, they indicated the cone shaped pin fin as the most efficient one in terms of thermal performance index.

Lu et al. [26] experimented on Al–Cu microchannel heat exchangers for fluid dynamics and HT characteristics. They used ethylene glycol and water as the working fluid. Their average Nusselt number data in the laminar zone showed a well agreement with the known laminar flow correlations.

Experiment on aluminum and copper base microchannel heat exchangers was conducted by Parida et al. [27]. To evaluate the overall cooling performance of the test setup, they used infrared thermography. Their findings indicate that the copper micro
heat exchanger yields better a thermal performance in terms of time constant, i.e. found to be in the range of 1–2 s.

Yeh and Yang [28] carried out experiments on eight micro-heat exchangers with rectangular, airfoil shaped and shuttle typed strips. Strip length, strip type and strip arrangements were considered as parameters for indicating the overall performance of device. An inspection of the previous work shows that the past investigations were majorly concentrated on modifications of the flow path geometry. However, as result, only moderate heat transfer augmentations could be achieved for relatively high pressure drops. The number of studies that employ the pulsation phenomenon is rather small. The specialty of the present contribution is that both techniques are applied in a combined manner. This means that the passage geometry is modified by twisting the channel to perturb out of plane mixing, while imposing pulsations onto the flow rate. Thus, twisted channels with fully three-dimensional geometries for unsteady flow, i.e. for pulsating flow rate, are computationally investigated in the present study. The focus of the study has been to achieve a heat transfer enhancement by a combined application of the both measures.

**Numerical modelling**

In this work, analysis is performed for a microchannel where the channel cross section traces a twisted sinusoidal curve around a straight line, referred to as “main axis” thereafter, as depicted in Figure 1. Denoting the space coordinate along the main axis by \( x \), the offset of the center of the channel cross section from the main axis is given as

\[
e = A_w \cos \left( \frac{2\pi}{\lambda} x \right) \quad (1)
\]

The cross section of the channel is square-shaped with 0.50 mm breadth. While the wave amplitude \( A_w \) was fixed at 1 mm, the total length was 25 cm, which is 50 times the hydraulic diameter. The pitch of the channel \( S \) was 5 cm with wavelength \( \lambda \) at 2.5 cm.

For analysis purpose, we consider unsteady, laminar, incompressible flow with constant material properties. Water is used as the working fluid. As typical application of such channels does not include high temperature, radiation [29] is neglected and fluid properties were assumed to be temperature invariant. Basic governing equations involved are as provided below [30]:

**Continuity equation**

\[
\frac{\partial u_i}{\partial x_i} = 0 \quad (2)
\]

**Momentum equation**

\[
\frac{\partial u_i}{\partial t} + u_j \frac{\partial u_i}{\partial x_j} = - \frac{\partial p}{\partial x_i} + \frac{1}{Re} \nabla^2 u_i \quad (3)
\]

**Energy equation**

\[
\frac{\partial T}{\partial t} + u_i \frac{\partial T}{\partial x_i} = \frac{1}{RePr} \nabla^2 T \quad (4)
\]

where \( u_i, p \) and \( T \) denote the velocity vector, pressure and temperature, respectively.
For the microchannel walls, Dirichlet type boundary condition is used. No slip condition on the walls implies zero velocity and a fixed temperature was prescribed imposing isothermal conditions on boundary. For the inlet velocity, a sinusoidal time variation is assumed in the form

\[ u_{in} = u_m \left( 1 + A \sin(2\pi f_r t) \right) \] (5)

In the above equation, \( u_{in} \) is the bulk inlet velocity, which varies around its time-averaged value (\( u_m \)) sinusoidally with amplitude of \( A \) and a frequency of \( f_r \). The latter is related to Strouhal number (\( St \)) through the relationship \( St = f_r D_h/um \). It may be mentioned here that similar inlet condition is reported in literature, e.g. in Nandi and Chattopadhyay [4, 31]. As the outlet of the domain is free exit, pressure outlet boundary condition, i.e. a zero gauge pressure was prescribed. The Reynolds number of flow is defined by \( Re = \rho u_m D_h/\mu \).

A finite volume based solver, ANSYS Fluent 19.2 was used for obtaining the solution of the equation set. Simulations were performed for different Reynolds numbers in the range 1–100, for different pulsation frequencies and amplitudes.

Because of the complex geometry, it was necessary to deploy an unstructured grid. However, the mesh was generated with uniform structure density distribution. A rigorous grid independence test regime was ensured as depicted in Table 1. Three major parameters, namely, the \( Nu \), \( f \) and performance parameter (\( \eta \)) were considered for evaluating grid independence. Table 1 shows that the grid with 872,250 nodes deliver results, which can be deemed sufficient, as further refinement does not change the output significantly. Figure 2 depicts a close view of the mesh.

Solutions were obtained using SIMPLE algorithm [19]. For momentum equations, QUICK scheme and for energy equation MUSCL scheme was chosen following Nandi and Chattopadhyay [4, 31]. For convergence, residual values at each time steps were ensured to be below \( 10^{-5} \) and \( 10^{-8} \) for the momentum and energy equations, respectively.

### Data reduction

Since, constant specific heat capacity of fluid is used to measure the energy absorbed, \( T_b \) (bulk temperature) could be assumed as the average of the temperatures at channel inlet and channel outlet, which in return is accounted for the calculation of fluid properties.

\[ T_b = \frac{T_i + T_o}{2} \] (6)

The convective heat transfer coefficient (\( h \)) is represented as follows [31],

\[ h = \frac{q_w}{T_w - T_b} \] (7)

The Nusselt number (\( Nu \)) is defined in terms of heat transfer coefficient and hydraulic diameter (\( D_h \)), given by

\[ Nu = \frac{h D_h}{k} \] (8)
The friction factor \( (f) \) is evaluated from,
\[
f = \frac{2 \Delta p}{\rho u_m^2}
\]
where \( u_m \) is the time averaged bulk velocity.

The thermal performance enhancement factor, \( \eta \) is given, as per Chang et al. [32]
\[
\eta = \frac{Nu}{Nu_0} \left( \frac{f}{f_0} \right)^{0.33}
\]
where the subscript 0 denotes the same channel but without pulsation at inlet.

The amount of entropy developed due to temperature in any irreversible process is defined as the entropy generation. This is a function of temperature. With decreasing thermal entropy generation, a high heat transfer rate can be brought. Here, as used by Jafari and Ghazali [33], the total entropy generation is given by,
\[
S_{\text{gen}} = Q \left[ \frac{1}{T_0} - \frac{1}{T_b} \right]
\]
where \( Q \) is the heat flux, \( T_0 \) and \( T_b \) are the inlet and bulk temperatures, respectively.

Augmentation entropy generation number \( (N_{S,e}) \) is the ratio of total entropy generation furnished by TSWMC with pulsating flow \( (S_{\text{gen}}) \) to the total entropy generation by TWSC with steady-state flow \( (S_{\text{gen,0}}) \). Thus, to exhibit better thermal performance it should always be less than unity. Lower values of entropy generation mean a better thermal performance for the proposed geometry. It can be written as;
\[
N_{S,e} = \frac{S_{\text{gen}}}{S_{\text{gen,0}}}
\]

**Results and discussion**

The dimensionless frequency (i.e. Strouhal number) and pulsation amplitude were kept within the range of \( 1 \leq St \leq 10 \) and \( 0.1 < A < 1 \) for the current study which corresponds to typical microchannel applications. Due to addition of periodic pulsation at inlet, the instantaneous axial velocity in the middle of the microchannel should show periodic change in nature. The observed velocity signal in Figure 3 confirms that.

Figures 4 and 5 depict the variation of time averaged \( Nu \) with \( Re \) for different \( St \) and \( A \). Figure 4 shows that lower values of \( Re \) and inlet pulsation amplitude, the \( Nu \) yields almost same result as of the steady one. This can be attributed to weak swirl generation inside the flow, at low \( Re \). At low Reynolds number, the viscous force subjugates over the effect of the wavy and twisted nature of the channel. Also, at low amplitude of pulsation, the flow cannot be disrupted. With increase in \( Re \) and amplitude of pulsation, Figures 4 and 5 display an accountable rise in the Nusselt
number result as compared to that of the steady case. At higher $Re$, the flow pulsation seizes over the viscous force. Subsequently, better mixing of fluid occurs and the modified contour of the channel geometry provides generation of recirculation zones. So, one can interpret that at higher $Re$ and amplitude of the pulsation of flow at inlet, the current model can provide better HT result. The current data is compared with previous well-established work of Nandi and Chattopadhyay [4] in Figure 5. The microchannel described in the earlier work involved a two-dimensional sinusoidal wavy channel while the present geometry is twisted and hence the flow encounters a more tortuous path here. As a consequence, larger enhancement of heat transfer is noted in this case—about 100% higher at low $Re$ and 10% higher at $Re$ of 100. Overall, the proposed geometry shows a significant gain in performance vis-à-vis a simple wavy channel as depicted in [4].

Thermal development length ($L_{th}$) is a parameter of importance in the field of heat transfer augmentation. This was inspected for a range of $St$ and $Re$. Figure 6 establishes the time-averaged dimensionless thermal development length ($L_{th}/D_h$) as a function of $Re$. With increased frequency, the development length is found to grow. The steady case however shows highest development length. This can be caused due to the flow pulsation employed at inlet, which incorporates

![Figure 7. Variation of $\Delta \rho$ with $Re$ for different amplitude and $St$.](image1)

![Figure 8. Variation of friction factor as a function of $A$ for different $St$, at $Re = 10$. Steady-state results of Mohammad et al. [15] and Nandi and Chattopadhyay [4] for $Re = 10$ are also shown for comparison.](image2)
Figure 9. (a) Variation of $S_{gen}$ with $Re$ for pulsating flow with different amplitudes at $St = 10$ and for steady-state flow, (b) variation of $S_{gen}$ with $Re$ for different values of amplitude and Strouhal number, (c) variation of augmentation entropy generation number with $Re$ for different amplitudes at $St = 10$. 
better mixing of the fluid situated near the wall with the core fluid. For a straight duct, the development length is given by \( L_{th}/D_h = 0.05Re^{Pr} [34] \), which leads to much lower values compared to the present configuration. As an example, at \( Re = 100 \) and \( Pr = 0.7 \), the straight duct has a development length of 3.5 [34]. But for the same parameters, \( L_{th} \) varies between 8.2 and 9.3 for the proposed geometry. This is expected to be due to tortuous passage of fluid flow.

Pressure drop is the penalty that has to be paid at the cost of enhanced heat transfer in enhancing techniques by employing wavy and twisted channel geometry. The narrow and fluctuating flow passage prevents the fluid mixing specially at low Reynolds number attributing high pressure drop. Figure 7 shows the fluctuation of the time-averaged \( \Delta p \) as a function of the amplitude at different frequencies. In Figure 8, it can clearly be seen that the time-averaged \( f \) of the steady-state case is larger compared to those of the cases with pulsations for the present geometry. It is depicted that with increase in frequency of the inlet flow pulsation (represented by Strouhal number), the friction factor decreases. However, with increase in the amplitude of the flow pulsation, an increasing trend for \( f \) is observed, which can be due to the larger quantity of fluid incorporated with higher amplitude. The dependence of \( f \) on \( A \) is observed to be more pronounced for higher values of \( St \).

It is palpable from the figure that the present work is providing better result in terms of friction factor than the other published results for configurations [4, 15] with some similarity to the present one. The comparisons to those results [4, 15] are presented for steady-state flow in Figure 8. The non-twisted two-dimensional wavy channel of Nandi and Chattopadhyay [4] delivers slightly higher values compared to the present geometry, whereas the non-twisted three-dimensional wavy channel of Mohammad et al. [15] exhibits even higher values. This comparison indicates the importance of the twisted geometry.

Figure 9(a) elaborates the variation of time-averaged thermal entropy generation, for TSWMC. The thermal entropy generation depends upon temperatures only. From the figure, it is clear that the thermal performance of a MCHE can be enhanced by employing sinusoidal pulsation at the inlet, thus minimizing the entropy generation. It can also be noted that with increase in pulsation amplitude the entropy generation result yields better performance. Figure 9(b) portrays the variation of time-averaged thermal entropy generation with \( Re \) at different amplitude and Strouhal number. It can be easily discerned from the figure that with increase in \( Re \), the entropy generation is showing a downward trend which is desirable. The figure also illustrates that the case with maximum Strouhal number and pulsation amplitude is showing...
the best result. To compare the performance of the TSWMC with plain channel, a dimensionless quantity named augmentation entropy generation number is used. It may be defined as the ratio of total entropy generation furnished by TSWMC to the total entropy generation by the plain channel. Figure 9(c) depicts the comparison of augmentation entropy generation number with \( Re \). The objective is to decrease the thermal entropy generation, hence, a augmentation entropy generation value less than unity would suggest better result. The figure displays that as an effect of inlet flow pulsation, the present case always showing values less than 1. It is also noteworthy that at the highest amplitude (i.e. 0.8) the channel is showing best result.

To assess the effect of inlet flow pulsation, a parameter called thermal performance enhancement factor (\( \eta \)), is used in the current study, as already mentioned above (Eq. (10)). By definition, \( \eta \) is a measure for the improvement of the heat transfer performance by the flow pulsations compared to the steady-state flow. In Figure 10(a), the time-averaged \( \eta \) is shown as a function of \( St \) at different Amplitude (\( A \)) and constant Reynolds number (\( Re = 10 \)). The figure shows that the present scheme offers \( \eta \) values larger than unity for all cases, showing that flow pulsations have a beneficial effect for all considered cases. One can also see that \( \eta \) increases with increasing \( St \) and increasing \( A \). For comparison, the pulsating flow results in the non-twisted wavy channel of Nandi and Chattopadhyay [4] are also included in the figure, for \( A = 0.8 \). These values are smaller compared to the present results for \( A = 0.8 \), indicating again, the important role of the twisted geometry.

In Figure 10(b), the time-averaged \( \eta \) is shown as a function of amplitude (\( A \)) at different \( Re \) and constant Strouhal number (\( St = 10 \)). The results are also compared with the previous work of Nandi and Chattopadhyay [4], for non-twisted wavy channel. Also from this perspective, one can see that pulsating flow always leads to a thermal performance improvement (\( \eta > 1 \)). This increases with increasing \( A \) and \( Re \). Compared for the same \( Re \), the present results exhibit higher \( \eta \) values compared to those of Nandi and Chattopadhyay [4], underlining again the important role of the twisted geometry.

Figure 10(c), shows \( \eta \) as a function of amplitude (\( A \)) at different \( St \) and constant \( Re \) (\( Re = 10 \)). It is also obvious from the figure that at amplitude 0.8, the present setup is providing better result than the previous well-established work of Nandi and Chattopadhyay [4]. At \( St = 10 \), and amplitude of 0.8, it is providing a \( \eta \) value even greater than 2.0. In general, increased amplitude and frequency of pulsation at inlet can provide improved thermal performance. This can be attributed to the fact that the superimposing of the pulsation at inlet with the main flow incorporates the required instability in the flow to enhance proper
mixing of fluid. Also, at high amplitude (A) of pulsation the wavy and twisted nature of the channel geometry also contributes to the mixing of fluid and recirculation zones generated. Much more decrease in hydrodynamic, as well as thermal boundary layer thickness, occurs. Hence, the present geometry shows even a much more thermal enhancement.

Conclusions

In present work, the impact of a sinusoidally pulsating flow rate in the proposed sinusoidal wavy and twisted microchannel is investigated by analyzing transient three-dimensional Navier–Stokes equations, for the laminar flow regime, for Reynolds numbers from 1 to 100. The inlet velocity was prescribed by superimposing a sinusoidal fluctuating component to the mean flow. Investigations were performed for a range of frequency and amplitude. As thermal wall boundary condition, constant temperature is maintained at the channel walls. Time averaged Nusselt numbers were obtained to assess the HT augmentation results. To compare the efficiency of the proposed scheme with previous well-established works, thermal performance enhancement factor (η) combining HT rate and pressure drop was also examined. The proposed geometry showed better results in terms of thermal performance and also with minimal pressure drop. The influence of flow rate pulsations on the thermal development length was also examined.

Notes on contributors

Sampad Gobinda Das is pursuing Ph.D. in Mechanical Engineering Department, from Jadavpur University, West Bengal, India. He has completed his Master’s degree in Mechanical Engineering from Jadavpur University, Kolkata, India. He received his B.Tech degree from Kalyani Government Engineering College, Kalyani, West Bengal, India in Mechanical Engineering. His research interest lies in CFD in fluid flow and heat transfer, specializing on laminar, turbulent, steady, unsteady separated flows and convective heat transfer enhancement.

Suvanjan Bhattacharyya is currently working as an Assistant Professor in the Department of Mechanical Engineering of Birla Institute of Technology and Science Pilani, Pilani Campus, Rajasthan, India. He completed his post-doctoral research from University of Pretoria, South Africa. He completed his Ph.D. in Mechanical Engineering from Jadavpur University, Kolkata, India. He received his Master’s degree from Indian Institute of Engineering, Science and Technology, India (Formerly, Bengal Engineering and Science University), on Heat-Power Engineering. His research interest lies in CFD in fluid flow and heat transfer, specializing on laminar, turbulent, steady, unsteady separated flows and convective heat transfer, experimental heat transfer enhancement. He is the author and coauthor of 85 papers in high ranked journals and prestigious conference proceedings with an H-index of 12. He has bagged best paper award in a number of International conferences as well. He is also on editorial boards of 11 international journals and reviewer of more than 25 prestigious journals.

Himadri Chattopadhyay is Professor of Mechanical Engineering of Jadavpur University, Kolkata, India. He has served at CSIR-Central Mechanical Engineering Research Institute, Durgapur as Scientist before joining the current faculty position. He was the Director of School of Bioscience and Engineering of Jadavpur University for two years. He has more than 220 papers in journals and conferences and edited two proceedings. He has more than 2500 citations with an H-index of 29. His research domain of interest includes thermo-fluid science, CFD, turbulence, heat transfer augmentation, jet impingement, bio-engineering, materials processing and energy. He is a recipient of DAAD fellowship and DAAD Re-invitation Fellowship. He is fellow of Institution of Engineers, India and West Bengal Academy of Science and Technology.

Ali Cemal Benim received his B.Sc. and M.Sc. in Mechanical Engineering from the Bosphorus University of Istanbul, Turkey, and Ph.D. degree from the University of Stuttgart, Germany in 1988. Following his Postdoctoral period at the University of Stuttgart, he joined ABB Turbo Systems Ltd. in Baden, Switzerland in 1990. He was the Manager of the Computational Flow and Combustion Modeling group. Since 1996, he is a Professor for Flow Simulation and Energy Technology at the Düsseldorf University of Applied Sciences, Germany.

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