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HEAT TRANSFER IN PULSATILE FLOW THROUGH SQUARE MICROCHANNELS WITH WAVY WALLS

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ABSTRACT

A three-dimensional numerical analysis is performed to understand the effect of axial wall conduction on conjugate heat transfer during single phase pulsatile flow in a square microchannel with wavy walls. A square microchannel (with wavy vertical walls) of length of 30 mm, and cross-section 0.4 × 0.4 mm^2 *is considered for the numerical simulation. The wavelength and amplitude of the vertical wavy shaped channel walls are 12 mm and 0.2 mm respectively. The working fluid is taken as water which enters the channel at 300 K, and constant heat flux boundary condition is imposed at the entire bottom surface of the substrate on which the microchannel was carved. All the remaining surfaces of the substrate exposed to surrounding are kept insulated. The velocity at the inlet of channel is the combination of a fixed component of velocity and fluctuating component of velocity which varies sinusoidally, thus causing pulsatile velocity at the inlet (amplitude, A = 0.2) with variation of frequency from 2 Hz to 10 Hz. Simulations has been performed for constant flow Reynolds number (Re = 100) and substrate thickness below channel bed to channel height ratio (*$\delta_{sf} = 1$*) with varying solid to fluid conductivity ratio (*$k_{sf} = 0.344 - 715$*). It is observed that* k_{sf} *play a key role in controlling heat transfer due to axial wall conduction. Change of pulsation frequency corresponding to Womersley number (Wo = 1.414 - 3.163) does not affect local Nu. It is also found that overall Nu depends on Re and* δ_{sf} *at particular value of Wo. Overall Nu increases with* k_{sf} *up to an optimum value, and then decreases due to the effect of axial wall conduction.*

Keywords: pulsatile flow, microchannel, axial wall conduction, Womersley number, conjugate heat transfer

NOMENCLATURE

A	Amplitude of pulsation
C_p	Specific heat at constant pressure, J/kg-K
D_h	Hydraulic diameter of the channel, mm
f	Frequency of pulsation, Hz
h(z,t)	Instantaneous local heat transfer coefficient, W/m^2-K
k_s	Thermal conductivity of substrate wall, W/m-K
k_f	Thermal conductivity of working fluid, W/m-K
k_{sf}	Ratio of k_s and k_f
L	Length of the tube, m
L_c	Characteristic length (Lc = D), mm
M	Axial conduction number
Nu_{avg}	Overall Nusselt number
Nu(t)	Space average instantaneous Nusselt number
Nu(z)	Time average local Nusselt number
Nu(z, t)	Instantaneous local Nusselt number
Pe	Peclet number
Po	Poiseuille number
Pr	Prandtl number ($\mu c_p/k_f$)
Re	Reynold's number ($\rho U D_h/\mu$)
$\overline{q'}$	Heat flux experienced at the solid-fluid interface, W/m^2
q'	Wall heat flux applied at the bottom of solid substrate
q_z	Local heat flux experienced along the axial direction of the channel, W/m^2
T	Temperature, K
t	Time, s
U	Average velocity at the channel inlet, m/s
Wo	Womersley number ($L_c(\omega/\nu)^{0.5}$)

X	Axial coordinate

Greek Symbol

ν	Kinematic viscosity, m^2/s
ρ	Density, kg/m^3
δ_f	Inner radius of the tube, mm
δ_s	Thickness of the tube wall, mm
δ_{sf}	Ratio of δ_s and δ_f
μ	Dynamic viscosity, kg/m-s
ϕ	Non-dimensional heat flux
Θ	Non-dimensional temperature
ω	Angular frequency, rad/s
θ	Time period, s
Φ	Viscous dissipation function

INTRODUCTION

With developments in modern electronics, the need for higher and higher flux removal from unit surface area is increasing day by day. The cooling need has already exceeded the maximum heat removal capacity of natural convection from any electronic chip. Therefore there is growing demand for microscale heat transfer devices involving mini/microchannel systems [1]. Because of smaller passage dimension along with smaller flow rate, the flow mostly remain laminar. Theoretically, turbulent flow can remove higher heat transfer over laminar flow due to mixing and eddies. Mixing can be induced by using wavy walls of microchannels, which can improve the rate of heat transfer.

Generally, the wall thickness in a microchannel is more compared to the channel cross-section dimensions. This causes axial back conduction effects in microchannel systems more prominently compared to conventional size channels. This effect also depends on the thermo-physical properties of the solid wall, working fluid and flow condition.

There are many types of flow conditions apart from uniform (steady) flow in engineering applications. In most of these cases, the flow is oscillating. Pulsating flow is varying transitionally in one direction i.e. vary periodically. Pulsatile flow is mainly characterised by two parameters (i) pulsating frequency or Womersley number (Wo) (ii) amplitude of oscillation. Pulsatile flow has many applications in different fields of engineering. Blood flow and respiration in human body, reciprocating engine, IC engine, pulse combustor, ramjet, pulsating heat pipe, etc. are few examples of it.

LITERATURE REVIEW

Uchida [2] studied the pulsating flow on steady laminar motion of fluid through a pipe of circular cross section. Due to pulsatile flow, velocity magnitude is maximum towards the wall instead of centre line of the pipe. They also obtained the velocity profile for pulsatile flow in pipe. The phase lag increases with frequency. Further, different researchers worked on pulsatile flow and obtained the annular effect, phase lag, and periodic axial fluctuations of temperature in pipes subjected to pulsating inlet condition.

Amir and Nourah [3] developed a correlation for heat transfer coefficient in steady and pulsatile flow of air through a rigid circular pipe. A critical value of 2.1×10^5 for dimensionless number was taken. They found that there was no improvement in Nu when Re < 2.1×10^5, but Nu enhanced significantly when Re > 2.1×10^5.

Siegel and Perlmutter [4] studied heat transfer for pulsating laminar flow through duct. They considered parallel plates having uniform wall temperature and constant wall heat flux separately. The flow through pipe was laminar and pulsatile. For uniform wall temperature condition, they found that Nu show periodic axial fluctuation.

Faghri et al. [5] analytically studied heat transfer in pipe with laminar pulsating inlet. They found that temperature field divided into the steady mean part and harmonic part due to the velocity pulsation. They obtained that due to the interaction between velocity and temperature oscillation creates an extra term for the enhancement of heat transfer.

Mehta and Khandekar [6] numerically studied the effect of flow pulsation in two types of flow domain (i) flow through axisymmetric circular pipe and (ii) flow through parallel plates in which flow pulsation is in transverse direction. They studied the effect of Reynolds number (Re), Prandtl number (Pr), amplitude ratio (A) and pulsation frequency (Womersley number, Wo) on time-averaged and instantaneous heat transfer and friction factor (Po). They observed that Nu did not vary in fully developed region due to flow pulsation compared with steady flow. Effect of Re, A and frequency of pulsation (Wo) were not appreciable for Nu enhancement but Pr play crucial role in heat transfer i.e. Pr value inversely proportional to time-averaged relative Nu. They obtained in case (i) or axial flow pulsation no enhancement of heat transfer but in case (ii) or transverse flow pulsation enhancement of time-averaged Nu in developing region for larger frequency (Wo) and smaller spacing ratio.

Maranzana et al. [7] studied both analytically and numerically to visualize the effect of axial wall heat conduction in mini/microchannel. They introduced a dimensionless number which signifies axial wall conduction known as 'axial conduction number (M)' defined as:

$$M = \frac{q'_{conductive}}{q'_{convective}} = k_s \frac{(\delta_s.\omega)/L}{(\rho.C_p.\delta_f.\omega.U_{av})} \qquad (1)$$

They concluded that when $M < 10^{-2}$, axial back conduction is neglected in the micro channel. M usually very low in case of macro channel because of low solid to fluid thickness ratio (δ_{sf}). They did the analysis based on assumption that the temperature difference between inlet and outlet of solid substrate (ΔT_s) and fluid domain (ΔT_f) in the axial direction are same. But the above assumption is not valid always.

To avoid the limitation in Maranzana et al. [7], Zhang et al. [8] numerically studied the effect of axial back conduction in thick circular microtube, outer wall subjected to uniform wall temperature. They consider both ΔT_s and ΔT_f effect. They found that solid to fluid conductivity ratio (k_{sf}) plays a vital role in case of heat transfer. They obtained maximum value of Nu was about 4.3. They conclude that axial heat conduction depend on several parameters like solid to fluid conductivity ratio (k_{sf}), solid to fluid thickness ratio (δ_{sf}), Reynolds number (Re) and Prandtl number (Pr).

Moharana et al. [9] experimentally and numerically studied axial conduction performance of hydrodynamically and thermally developing single phase flow in rectangular minichannel array subjected to constant wall heat flux at the bottom of substrate wall while other surface are adiabatic. They found that higher value of axial conduction number (M) lead axial wall conduction. The variation of Nu along axial direction affected by parameter M i.e. higher M lead to low Nu. They conclude that low conductive solid wall material reduce axial wall conduction effect.

Moharana and Khandekar [10] numerically studied axial back condition in microtube subjected to both uniform wall temperature and constant wall heat flux condition. They reported that k_{sf} play an important role in controlling axial back conduction. They found that for higher k_{sf}, the constant wall heat flux boundary condition look like constant wall temperature due to severe axial back conduction. They also found an optimum k_{sf} value for which average Nu is maximum in case of constant heat flux boundary condition, but it is not found in case of constant wall temperature boundary condition. In case of constant heat flux, average Nu increases up to an optimum value, and then decreases with an increase in k_{sf}. But in case of constant wall temperature, average Nu decreases with increase in k_{sf}. In both cases, higher δ_{sf} leads to higher axial back conduction.

Moharana et al. [11] numerically studied heat transfer in square microchannel subjected to constant wall heat flux at bottom of solid substrate. They concluded that conductivity ratio k_{sf} play key role in axial wall conduction similar to that by Moharana and Khandekar [10]. Larger k_{sf} value leads to decreased average Nu due to severe axial back conduction and small value of k_{sf} lead to a channel of zero wall thickness subjected to constant wall heat flux applied to one side of substrate and remaining are adiabatic. They also reported that for constant Re and δ_{sf}, there exist an optimum k_{sf} where the value of average Nu is maximum. They also reported that δ_{sf} is proportional with axial back conduction due to increase in thermal resistance.

Yadav et al. [12] numerically studied axial conduction in microtube using Helium as working fluid. They obtained optimum k_{sf} similar to that by [20] and [21] where average Nu is maximum. They also concluded that for higher wall thickness (δ_{sf}), average Nu is low and the average Nu increases with Helium flow rate.

Mishra and Moharana [13] numerically studied axial conduction in pulsatile flow through microtube subjected to sinusodially varying pulsatile flow velocity at inlet and constant wall heat flux at wall of the microtube. They carried out the simulation for wide range of conductivity ratio (k_{sf}) and frequency of pulsation (Womersley number, Wo) keeping thickness ratio (δ_{sf}), flow rate (Re) and amplitude (A) as constant. They reported that there exist an optimum value of k_{sf} for particular Wo. Heat transfer is very less in case of pulsation frequency. Average Nu is maximum at optimum k_{sf} and decreases by increasing beyond optimum value similar to Moharana et al. [11]. They reported that lower k_{sf} leads to increased heat transfer. They reported that by changing phase angle from 0 to 90, results decrease in fluid and wall temperature and increase in dimensionless heat flux. They obtained that effect of Wo is more inside the developing region and by increasing Wo, the value of time average relative Nu increases for all k_{sf} in axial direction and the values less than one indicate heat transfer is less compared to steady state.

Mohammed et al. [14] numerically studied heat transfer enhancement in wavy microchannel heat sink with water flow at Re 100 to 1000 for different amplitudes. From this they found that heat transfer performance better than the same cross-sectioned straight channel, pressure drop penalty was very small, friction factor and wall shear stress increase with the amplitude of wavy channel.

Sui et al. [15] experimentally studied heat transfer and friction flow in a wavy microchannel having a rectangular cross section. Here they considered wavy channel of fixed width of 205 μm, depth of 404 μm, the wavelength of 2.5 mm and waves were parallel. Experimental results are compared with a straight channel of the same cross section and length, and found that heat transfer enhanced in the wavy channel compared to the straight channel.

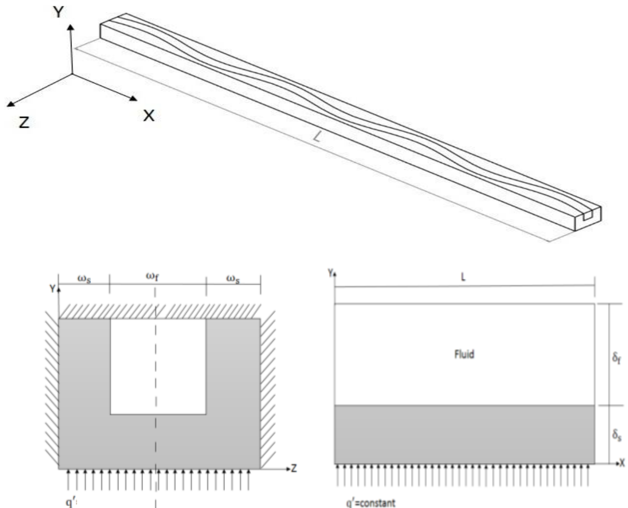
Gong et al. [16] numerically studied flow and heat transfer in a microchannel with wavy walls. They considered two types of wavy channels i.e. crests and troughs facing each other alternatively by phase angle 180º (serpentine channel) and crests and troughs facing each other (raccoon channel). It was found that in both the cases the heat transfer is more than straight channel.

Nandi and Chattopadhay [17] numerical studied developing flow in wavy microchannels (serpentine channel) under pulsating flow inlet. They analysed two dimensional wavy microchannel with constant wall temperature boundary condition and sinusoidal varying velocity at the inlet. From the simulation they found that heat transfer performance is more better in case of pulsatile inlet than that of steady flow in microchannel at different amplitude (0.2, 0.5, 0.8) and frequency (1, 5 & 10).

Nandi and Chattopadhay [18] numerically investigated developing flow and heat transfer in raccoon type microchannels under inlet pulsation. They found that heat transfer performance was better in case of pulsatile flow than steady for different amplitude (0.2, 0.5, 0.8) and frequency (1, 5, 10). Secondly, they found that heat transfer is better in raccoon type channel instead of serpentine channel.

PROBLEM FORMULATION

A parallel wavy wall microchannel having height of fluid (δ_f), thickness of solid substrate below fluid (δ_s), side wall thickness(ω_s) and length (L) is shown in the following diagram. In this case, water is used as working or flowing fluid in microchannel having inlet temperature at 300K and Pr=7. The pulsatile velocity varies sinusoidally with time leading to the development of pulsatile flow in the channel. The pulsatile velocity inlet (U_{in}) is having two components, one is fixed component (U_{av}), and the other is fluctuating component ($U_{av}.A.\sin(\omega t)$) that sinusoidally vary with time. The width (ω_f) and height of fluid (δ_f) in the fluid channel are constant at 0.4 mm, and the length of the channel (L) is 30 mm. So the hydraulic diameter (D_h) is 0.4 mm. The side view and front view of the computational domain is also shown in the diagram. The sinusoidal curve of pulsatile flow is shown below where the phase angles are indicated in degrees.



(a)

(b) (c)

FIGURE 1. (a) COMPUTATIONAL DOMAIN, (b) FRONT VIEW OF THE MICROCHANNEL, (c) SIDE VIEW OF THE MICROCHANNEL

Numerical simulations performed out under following assumptions (1) Single phase, laminar incompressible fluid flow (2)Thermo- physical properties of the working fluid are constant(3) Natural convection and radiation mode of heat transfer are neglected(4)Flow is pulsatile i.e. transient.The flow and heat transfer are governed by the continuity, Navier-Stokes and energy equations.

Boundary conditions

For liquid domain

$$\nabla.\vec{u} = 0 \tag{2}$$

$$\vec{u}.\nabla.\vec{u} = -\frac{1}{\rho}\nabla P + \frac{\mu}{\rho}\nabla^2\vec{u} \tag{3}$$

$$\vec{u}.\nabla.T = (k/\rho.C_p).\nabla^2 T \tag{4}$$

For solid domain

$$\nabla^2 T = 0 \tag{5}$$

$(\partial T/\partial z) = 0$; X-Y plane at $z = 0$and $z = 2\omega_s+\omega_f$ (6)

$(\partial T/\partial y) = 0$; X-Z plane at $y = (\delta_s+\delta_f)$ (7)

$(\partial T/\partial x) = 0$; Y-Z plane at $x = 0$ and $x = L$ (8)

q'= constant; X-Z plane at $y = 0$ (9)

$u = u_{av}(1 + A.\sin(\omega t))$; Y-Z plane at x = 0 (10)

$u = 0$; at solid liquid interface (11)

Grid independence test

In this case of pulsatile flow inlet, local Nusselt number [Nu (z,t)] is calculated for one phase angle 90 where Re = 100, frequency = 2 Hz and $\delta_{sf} = 1$ for three different grid sizes 14×105, 16×120 and 20×150. From the Fig. 2 increasing the grid size from 14×105 to 16×120, instantaneous local Nusselt number [Nu(z,t)] decreased by 0.4%, but by further increasing the grid size from 16×120 to 20×150, the percentage difference of [Nu(z,t)] was reduced to 3.12%. So that grid size '16 × 120' was used for simulation.

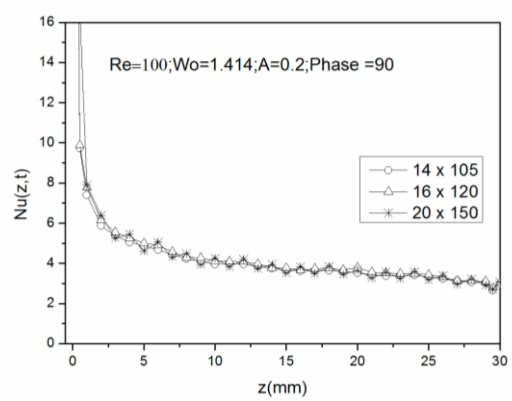


Fig. 2: Grid independence test

The governing differential equations are solved using commercial platform Ansys-Fluent®. For pressure discretization "standard" scheme was used. Second order upwind scheme was used for solving the momentum and energy equations. Convergence criteria for continuity and momentum equations is taken as 10^{-6} and for energy equation it is 10^{-8}.

Data reduction

The parameters are (a) local heat flux for a different phase angle for a cycle (b) local wall and bulk fluid temperature for a different phase angle for a particular cycle. These parameters help us to examine the effect of axial back conduction on local Nusselt number where conductivity ratio (k_{sf}) plays an important role.

Axial coordinate (x) in dimensionless form written as

$$z^* = \frac{x}{L} \tag{12}$$

The dimensionless heat flux at solid-fluid interface is written as

$$\phi = \frac{q_z}{\overline{q'}} \tag{13}$$

where q_z is the value of local heat flux at each axial location of z^*, $\overline{q'}$ is the ratio of applied heat flux at the bottom of solid substrate and net area of conjugate walls, given

$$(\overline{q'}) = (q').(2\omega_s + \omega_f)/(2\delta_f + \omega_f) \tag{14}$$

where, q' is the wall heat flux applied at the bottom of the solid substrate.

Dimensionless wall and bulk fluid temperature are given as

$$\Theta_w = \frac{T_{wz} - T_{fi}}{T_{fo} - T_{fi}} \tag{15}$$

$$\Theta_f = \frac{T_{fz} - T_{fi}}{T_{fo} - T_{fi}} \tag{16}$$

whereT_{wz} and T_{fz} are wall and average bulk fluid temperature at any location of z^* of the channel. T_{fi} and T_{fo} are the average bulk fluid temperature at the inlet and outlet respectively. The Instantaneous local Nusselt number is

$$Nu(z,t) = \frac{h(z,t).D_h}{k_f} \tag{17}$$

where $h(z,t)$ is the local heat transfer coefficient which is given by

$$h(z,t) = \frac{q_z}{T_w - T_f} \tag{18}$$

The space average local Nusselt number is given by

$$Nu(z,t) = \frac{1}{L}\int_0^L Nu(z,t)dz \tag{19}$$

The time average local Nusselt number is given by

$$Nu(z,t) = \frac{1}{\theta}\int_0^\theta Nu(z,t)dt \tag{20}$$

The overall Nusselt number is given by

$$Nu_{avg} = \frac{1}{L\theta}\int_0^{\theta}\int_0^{L} Nu(z,t)\,dt\,dz \tag{21}$$

Relative Nusselt number is given by

$$Nu_r = \frac{Nu_t}{Nu_s} \tag{22}$$

where Nu_t and Nu_s are transient and steady state Nusselt number respectively.

RESULT AND DISCUSSION

Heat flux experienced at the solid-fluid interface at low k_{sf} show a drastic change i.e. heat flux is more than the applied heat flux value. The effect of axial wall conduction is increased proportional with k_{sf} value which is overlap with zero wall thickness wall at k_{sf} = 13.1.The value of heat flux at solid-fluid interface decreased at the outlet of the channel with increasing k_{sf} because by increasing k_{sf} value thermal resistance of the substrate channel decreases leads to decrease in heat flux as Fig. 3(c-d). By increasing phase angle from 0 to 90 show that dimensionless heat flux at the solid-fluid interface increases slightly and decreases slightly by further increase from 90 to 270. The above results found because as per pulsatile velocity inlet velocity magnitude is maximum at phase angle 90 and minimum at 270.

For ideal condition flow through a circular tube subjected to constant wall heat flux boundary condition on outer wall results variation of bulk fluid temperature and wall temperature variation along the length of tube is linear in fully developed region i.e. difference between wall and bulk fluid temperature is constant in fully developed region. At low k_{sf} Fig. 4(a) it is observed that the difference between fluid and wall temperature gradually increases towards the end of the channel due to the wavy wall. In case of high k_{sf} value, the conventional theory does not valid because of the larger temperature of an outlet than the inlet of working fluid lead to conduction in the opposite direction to fluid flow. The wall temperature increases by the changing phase angle from 0 to 90 same as [16], but the variation is not found in the case of bulk fluid temperature that does not vary with phase angle. Again by the changing phase angle from 90 to 270 show that wall temperature decreases from initial value i.e. at phase angle 0. It is obtained that by increasing k_{sf} the wall temperature value decreases.

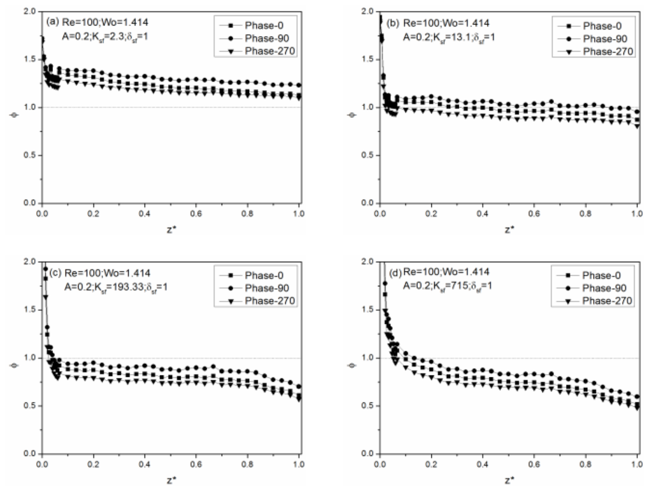


Fig. 3: Dimensionless heat flux variation in axial direction for different k_{sf} values

From Fig. 5, it is observed that when phase angle change from 0 to 90 the local Nusselt number value increases slightly and further decreases slightly by the changing phase angle from 90 to 270.The above result is observed because at phase angle 90 magnitude of velocity is maximum that causes proper mixing of fluid leads to increase heat transfer and at phase angle 270 magnitude of velocity is minimum leads to lower heat transfer.

Form Fig. 6, it is observed that for all k_{sf} the average Nusselt number vary periodically same as the sinusoidally varying pulsatile velocity inlet i.e. Nu(t) increases by increase phase angle from 0 to 180 (upper part of cycle) and further decrease by increasing phase angle from 180 to 360(lower part of cycle). It is also obtained that by increasing k_{sf} from 2.3 to 13.1 Nu(t) value increases which is later decreases by further increase in k_{sf} from 13.1 to 715 due to conjugate heat transfer or axial wall conduction effect.

It is obtained that $Nu_r(z)$ value is less compared to the steady state in developing zone due to pulsation effect that gradually increases for all k_{sf} toward developed zone. For low $k_{sf} = 2.3$ in Fig. 7(a) is observed that by increasing frequency relative Nu increases along the length. It is also observed that for all k_{sf} at f = 2 Hz relative Nu maintain a constant value. In Fig. 7(b) found that relative Nu maintain constant along the length for all frequency except f = 10 Hz, because of the high value of frequency backflow of higher temperature fluid occur leads to axial back conduction. For higher k_{sf} in Fig. 7(d) the value of relative Nu maintain constant along the channel for all Wo, which increases in outlet with an increase in Wo.

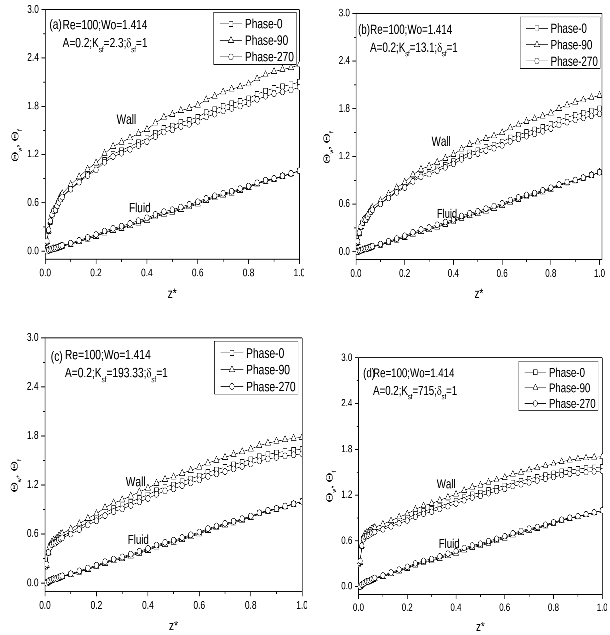


Fig. 4: Dimensionless bulk fluid and wall temperature variation in axial direction for different k_{sf} values.

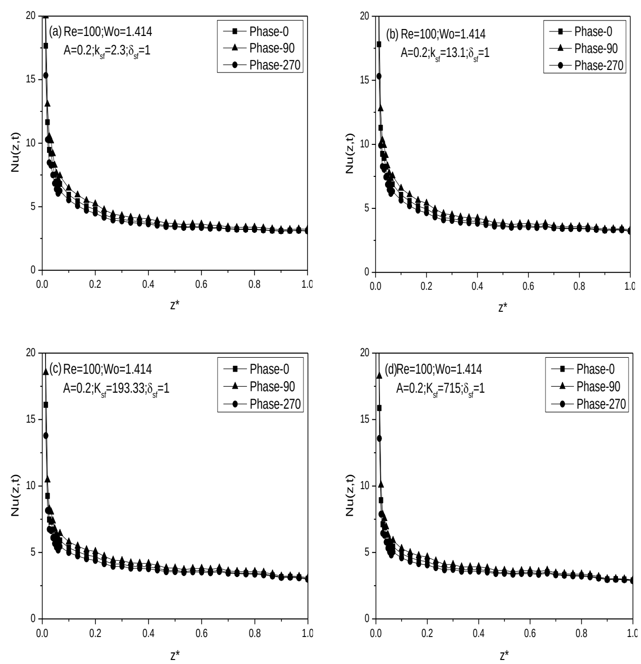


Fig. 5: Local Nusselt number variation in axial direction for different k_{sf} values.

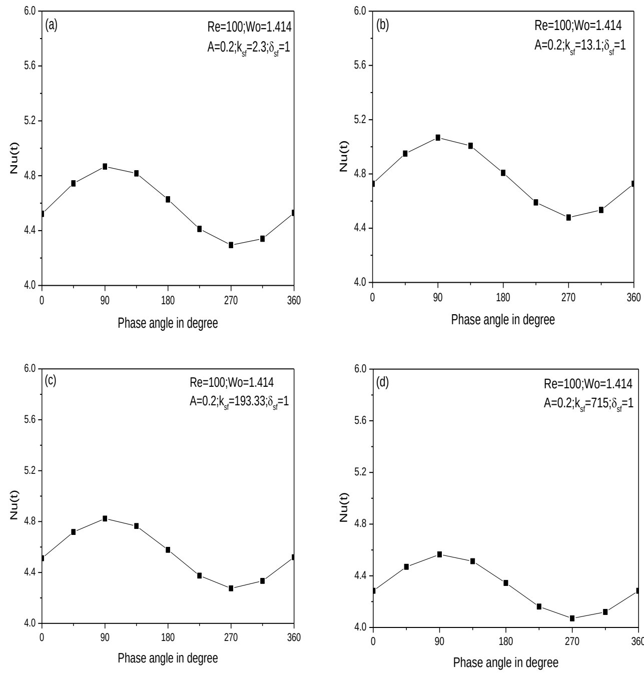


Fig. 6: Variation of space average local Nusselt number [Nu(t)] in axial direction for different k_{sf} values.

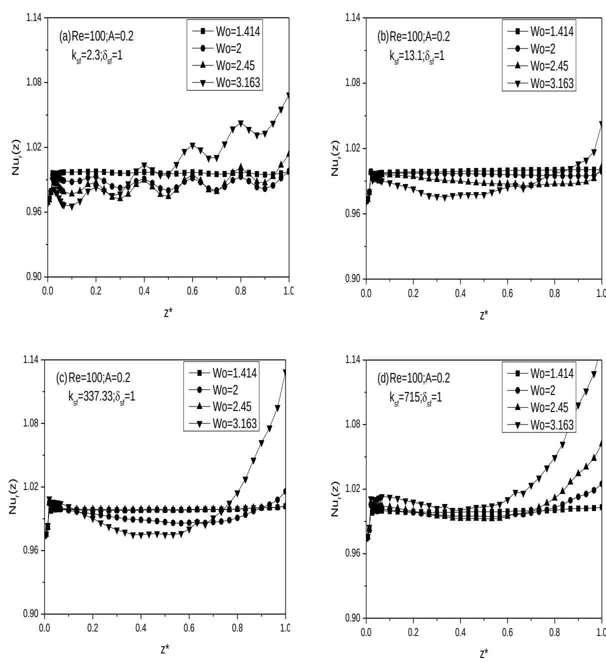


Fig. 7: Time average relative Nusselt number [$Nu_r(z)$] variation at different pulsation frequency.

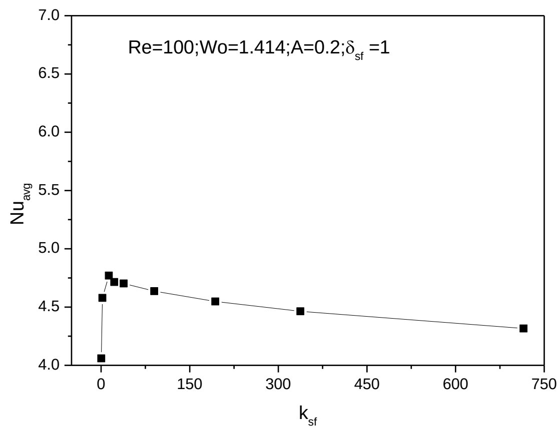


Fig. 8: Variation of overall Nusselt number with k_{sf}

Fig. 8 shows the optimum k_{sf} value at which Nu is maximum. Fig. 8 shows variation overall Nusselt number for all k_{sf} value from 0.344 to 715 at particular pulsating frequency, f = 2 Hz (Wo = 1.414). The value of Nu_{avg} increases rapidly up to an optimum value from low k_{sf} value 0.344, after reaching optimum value the Nu_{avg} starts decrease due to axial wall conduction in the solid substrate of the microchannel. It is obtained that there exist an optimum k_{sf} where Nu_{avg} is maximum. The Nu_{avg} variation look similar to Yadav et al. [12] where flow inlet is stedy but in Mishra & Moharana[13] flow is pulsatile.

CONCLUSION

A numerical simulation carried out to show the effect of pulsatile velocity inlet on axial wall conduction for laminar flow through in a microchannel subjected to constant wall heat flux boundary condition at bottom of solid substrate. Simulation done for wide range of conductivity ratio k_{sf} from 0.344 to 715, thickness ratio (δ_{sf} = 1) and Re = 100 while amplitude of pulsatile velocity fixed (A = 0.2). To observed the effect of pulsation the pulsation frequency vary i.e. f = 2, 4, 6 and 10 Hz corresponding Wo = 1.414, 2, 2.45 and 3.163 respectively. From the numerical simulation conclusions are observed as follows:

- By increasing k_{sf} value, the dimensionless wall temperature decreases and the profile did not similar to the conventional one due to conjugate heat transfer.
- The dimensionless heat flux value decrease towards outlet by an increase in k_{sf} due effect of conjugate heat transfer.
- The value of overall Nu is maximum at a moderate value of k_{sf} which is lower than that of Mishra and Moharana [16].

- By increasing the k_{sf} value beyond optimum value, the overall Nu decreases due to the effect of conjugate heat transfer leads to increase axial wall conduction.
- Nu_{avg} is function of Re, δ_{sf} and Wo.

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