ROLE OF ENTROPY GENERATION AND FIELD SYNERGY ON HEAT TRANSFER FROM CONFINED WAVY WALL

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ABSTRACT

In the present study, three-dimensional numerical investigations are carried out to find the effect of confinement due to top wall on flow and heat transfer over sinusoidal wavy wall for a fixed Reynolds number (Re) of 500. For fixed amplitude and wavelength of the wavy wall, different values of mean channel height and varying shapes of top wall have been considered. The effect of introducing waviness in top wall with respect to a plane top wall has been studied for in-phase sinusoidal and out-of-phase sinusoidal configurations. Computations are carried out by using commercial software ANSYS Fluent 16.1. Flow and heat transfer characteristics are presented in terms of Nusselt number and friction factor. Influence of secondary flows developed inside the channel on heat transfer is evaluated by calculating secondary flow intensity and field synergy angle. Irreversibilities caused due to various confinements are analyzed by calculating entropy generation due to heat transfer and friction. The variation of entropy generation rate for different top wall configurations as well as channel heights have also been presented.

NOMENCLATURE

a	[mm]	Amplitude of the wave
D_H	[mm]	Hydraulic diameter of the channel
f	[-]	Friction factor
H	[mm]	Mean channel height
k	[W/mK]	Thermal conductivity
Nu	[-]	Nusselt number
Re	[-]	Reynolds number
S	[W/°K]	Entropy generation rate
U	[m/s]	X component of the velocity

Special characters

ρ	[kg/m^3]	Density of the working fluid
μ	[Ns/m^2]	Dynamic viscosity of the working fluid
θ	[°]	Field Synergy Angle
λ	[mm]	Wavelength of the sinusoidal wave

Subscripts

i	Inlet
v	Viscous
T	Thermal
o	Outlet
∞	Free stream

INTRODUCTION

Corrugated channels are considered to be efficient passive methods of heat transfer enhancement and are widely used in plate and compact heat exchangers. Vortices generated by corrugated surfaces enhance mixing in the flow through the channel. Literature confirms that the factors influencing flow and heat transfer in wavy channels are shape and smoothness of the geometry [1], aspect ratio of the channel [2], channel height [3], wave parameters [4] and phase difference between the waves of top and bottom walls [5].

Confinement of the channel has direct impact on flow and heat transfer characteristics. In general, confinement can be of two types, channel height and shape of the walls. Islamoglu and Parmaksizoglu [3] carried out experimental investigations to study the effect of channel height (H) on flow and heat transfer for $1200 \leq Re \leq 4000$. They considered two different channel heights and reported higher Nusselt number and friction factor for more height. Elshafei *et al*. [5] conducted experiments to study effect of channel spacing and phase shift in waviness on heat transfer and pressure drop for $3220 \leq Re \leq 9420$. They reported increased friction factor with an increase in channel spacing and phase shift. Moreover, at higher Reynolds number, effect of spacing variation on heat transfer and pressure drop is more pronounced than that of phase shift variation. Convective heat transfer is always accompanied by entropy generation, i.e. one way destruction of available work. Entropy generation or irreversibilities associated with convective heat transfer are due to heat transfer across finite temperature difference and due to fluid friction [6]. Ko and Cheng [7] carried out numerical study on how channel aspect ratio (W/H) and Reynolds number (Re) affect entropy generation in triangular wavy channels. They considered various channel aspect ratios, viz.1, 2 and 4 and different values of Re between 100-400. They reported lower entropy generation at higher Re for an aspect ratio of 1. Guo et al. [8] proposed Field Synergy Principle (FSP) to analyse the mechanism underlying heat transfer phenomenon. It proposes that the synergy between velocity and temperature gradient in a flow domain decides the magnitude of convective heat transfer. The objective of present work is to study the effect of confinement due to top surface on flow and heat transfer over a wavy wall. Effect of confinement on heat transfer is analysed with the help of Field Synergy Principle (FSP) [9]. Overall themo-hydraulic performance of the channel is evaluated by estimating entropy generation due to irreversibilities caused by heat transfer and pressure drop.

PROBLEM STATEMENT

Computational domain employed in present study has been shown in Fig.1 (a) in which sinusoidal wavy wall forms bottom wall of the domain. For fixed amplitude (a=3mm) and wavelength (λ=15 mm) of the wavy wall, three different values of channel height (H=10, 20, 30 mm) and varying shapes of top wall have been considered. Waviness has been introduced in top wall with respect to a plane top wall for both in-phase and out-of–phase configurations as shown in Figs. 1 (b), (c) and (d). SP, SIP, and SOP represent plane, in-phase sinusoidal and out-of-phase sinusoidal top wall configurations for fixed sinusoidal bottom wall. The wave parameters, viz. amplitude and wavelength, for sinusoidal top wall are same as those for the bottom wall. Isothermal top and bottom walls are considered here. All calculations are carried out for fixed Reynolds number of 500.

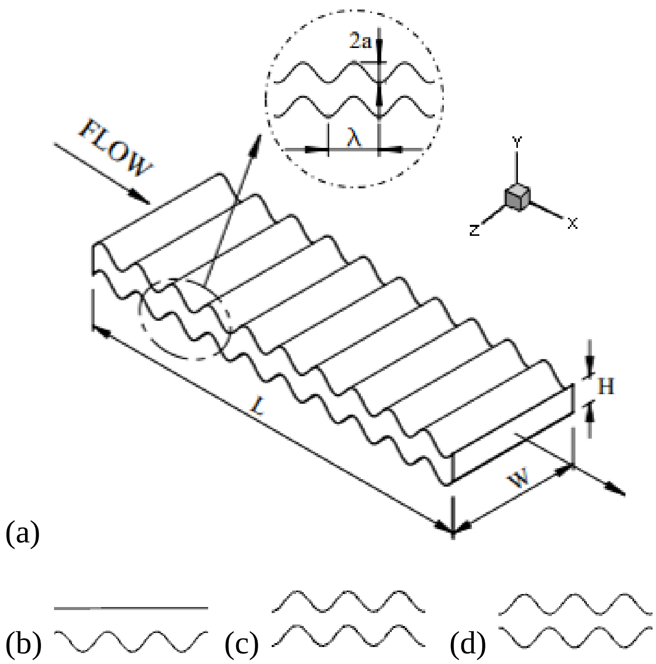


Figure 1 (a) Computation domain with wave parameters, schematic of the channel with top wall (b) plane (SP) (c) having in-phase (SIP) sinusoidal wave (d) having out-of-phase (SOP) wave.

NUMERICAL METHODS

Flow is assumed to be three-dimensional, steady, incompressible and laminar. Non-dimensionalised governing equations in tensor notation for mass, momentum and energy conservation can be summarized as follows.

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{1}$$

$$\frac{\partial (u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 u_i}{\partial x_j \partial x_j}\right) \tag{2}$$

$$\frac{\partial (u_j T^*)}{\partial x_j} = \frac{1}{\mathrm{Re}\,\mathrm{Pr}}\left(\frac{\partial^2 T^*}{\partial x_j \partial x_j}\right) \tag{3}$$

Here Re is given as

$$\mathrm{Re} = \frac{\rho U_\infty D_H}{\mu} \tag{4}$$

where ρ and μ represent density and dynamic viscosity of the working fluid at inlet temperature, D_H represents the hydraulic diameter of the channel which is given by $D_H=2H$, where H is the mean channel height between the top and bottom walls, and U_∞ is the characteristics velocity equal to free steam velocity. Moreover, Pr represents the Prandtl number. The non-dimensional temperature is given by

$$T^* = \frac{T - T_\infty}{T_W - T_\infty} \tag{5}$$

where T_W is the wall temperature and T_∞ is the free-stream temperature.

Boundary conditions imposed at inlet, outlet, side walls, top and bottom walls are as follows.

Inlet: $u = U_\infty$, $v = 0$, $w = 0$, $T = T_\infty$

Outlet: Pressure outlet $(p = p_\infty)$

Side walls: No-slip and adiabatic boundary condition

Top and bottom walls: No-slip and isothermal boundary conditions, $u = v = w = 0$, $T = T_w$

Commercial software ANSYS ICEM CFD 16.1 is used for generating non-uniform structured mesh. Computations are carried out with the help of commercial software ANSYS Fluent 16.1. SIMPLE scheme has been employed for solving governing equations. Convergence criterion set for mass, momentum, and energy equations are 10^{-6}. Detailed validations are carried out for flow and heat transfer through wavy channel by comparing the computed results with the experimental results of Ali and Ramadhyani [10]. Triangular wavy channel of in-phase arrangement with b/L$_c$ = 0.23 is considered and the results are presented in Fig.2.

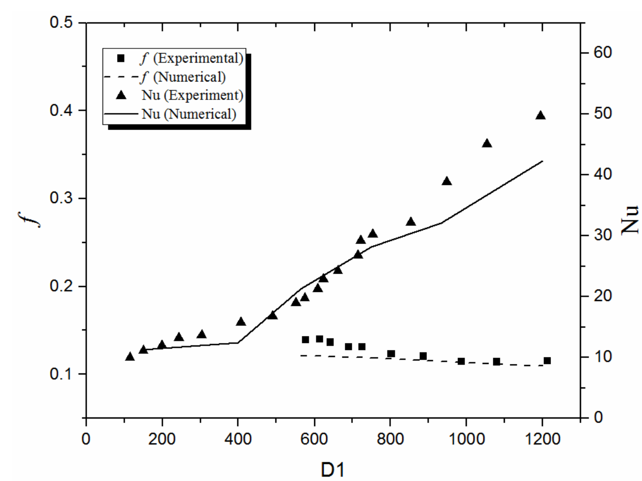


Figure 2 Comparison of computed values (a) Nu and (b) *f* with experimental data [10].

Maximum percentage difference between the experimental and the numerical results are 14% and 16% corresponding to Nusselt number and friction factor respectively. However, maximum uncertainties reported in the experimental results of [10] corresponding to Nu and *f* are 10% and 15 % respectively. Hence, the numerical results from present computations can be assumed to be in reasonably good agreement with the reported experimental results except for small deviation at higher values of Re considered. Table 1 presents the summary of grid independence study for the model SIP with a channel height of 20 mm. Percentage difference in values of Nu and *f* for a change of grid from 1 to 2 are 1.5 and 0.6 respectively while those for change of grid from 2 to 3 are 0.6 and 0.7. Accordingly, all the computations have been performed by considering grid 2.

Table 1 Grid independence for a channel height of 20 mm at Re = 500 (SIP configuration).

Grid number	Grid size (nodes)	Nusselt number (Nu)	Friction factor (*f*)
1	9.6×10^5	7.08	0.768
2	14.6×10^5	6.97	0.763
3	19.2×10^5	6.92	0.766

Even though 10 wavy modules are considered here, flow and heat transfer characteristics are similar except for two modules in first and last due to entry and exit effects. Hence, the flow and heat transfer results in the present work have been chosen corresponding to sixth module alone. Friction factor is calculated by the relation

$$f = \frac{\left(p_i - p_o\right) D_H}{0.5\times\rho\times U_\infty{}^2\times L} \tag{6}$$

where p_i and p_o are the mean pressure values at inlet and outlet of the wave module and *L* is the separation of outlet from inlet of the module and equals λ.

Nusselt number calculated for the cases present study corresponds to the bottom wall. Local Nusselt number is defined as

$$Nu_x = \frac{\partial T/\partial y\big|_{x=0}}{T_w - T_i}\frac{D_H}{k} \tag{7}$$

where *k* is the thermal conductivity of the fluid, T_i is the bulk mean temperature of the fluid entering in to the wave module.

Surface-averaged Nusselt number is given as

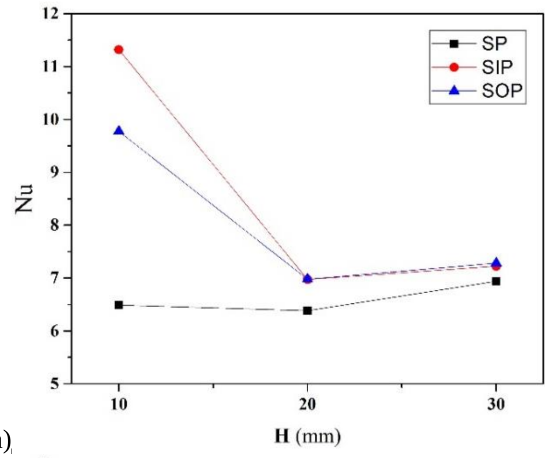
$$Nu = \frac{1}{A}\int Nu_x dA \tag{8}$$

RESULTS AND DISCUSSION

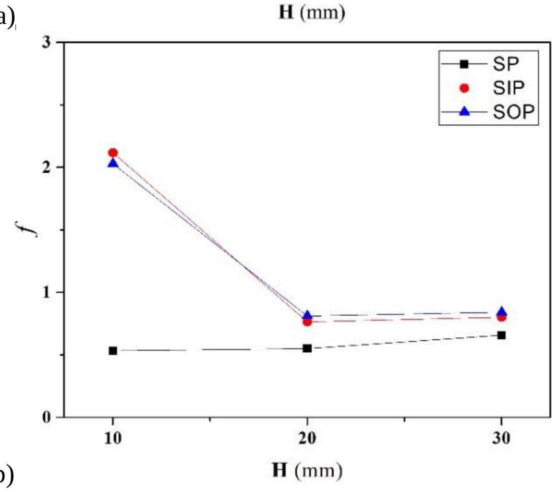
Wavy channels are introduced in heat exchangers to increase the heat transfer. However, gain in heat transfer is accompanied by increased loss of mechanical energy in terms of pressure drop. Following sections discuss effect of top wall geometry and channel height on flow and heat transfer characteristics and entropy generation in wavy channels.

Flow and heat transfer characteristics

Figure 3 shows the variation of (a) Nu and (b) *f* with channel height for three different top wall geometries. Nusselt number is found to increase with increase in channel height only for model SP. For other models, such as SIP and SOP, minimum Nu is observed at intermediate values of channel height which increases with increase in height beyond that corresponding to minimum Nusselt number. SIP geometry shows higher heat transfer coefficient in all the cases followed by SOP and SP. However, difference between the values of Nu is found to decrease with increase in channel height. For higher channel heights, Nu values for SIP and SOP are nearly same. This shows that effect of nature of top wall confinement becomes less significant with increase in channel height. Friction factor is found to increase with increase in channel height only for model SP. For other models, such as SIP and SOP, minimum friction factor is observed at intermediate values of channel height which increases with increase in height beyond that corresponding to minimum friction factor. Similar to Nu, nature of top wall has less influence on friction factor for higher values of channel heights.



(a)



(b)

Figure 3 Variation of Nu and *f* with channel height for various top wall geometries.

Introduction of waviness in the channel walls creates secondary flows for which the flow direction is different from main flow direction. These secondary flows may appear in the form of longitudinal and spanwise vortices. Ali and Ramadhyani [10] observed presence of longitudinal vortices in the wavy channel in their flow visualization studies even at low Re values of 500. The secondary flow plays an important role on heat transfer enhancement in wavy channels. Song and Wang [11] introduced a non-dimensional number, called 'Secondary Flow Intensity' (Se) which represents the ratio of inertia forces to viscous forces that are introduced by secondary flows in the domain. Mathematically, it can be expressed in the form

$$Se = \frac{\rho D_H{}^2}{\mu} \iiint_\Omega \left|\omega^n\right| \mathrm{dV} \Big/ \iiint_\Omega \mathrm{dV} \tag{9}$$

Variation of Se with channel height for different configurations is similar to that of the Nu and *f* variations. In narrow channels, generated vortices are convected and diffused rapidly due to the presence of channel wall [10] which is dominant in SIP and SOP configurations. Even though the strength of the secondary flow is much weaker than the main flow, these flows have a significant role in enhancing heat transfer in channel flows. These secondary flows move fresh fluid from walls to the center portion of the channel which leads to improved mixing in the flow. Influence of secondary flows on heat transfer can be analysed by computing the synergy between velocity and temperature gradient in the flow domain. This synergy can be

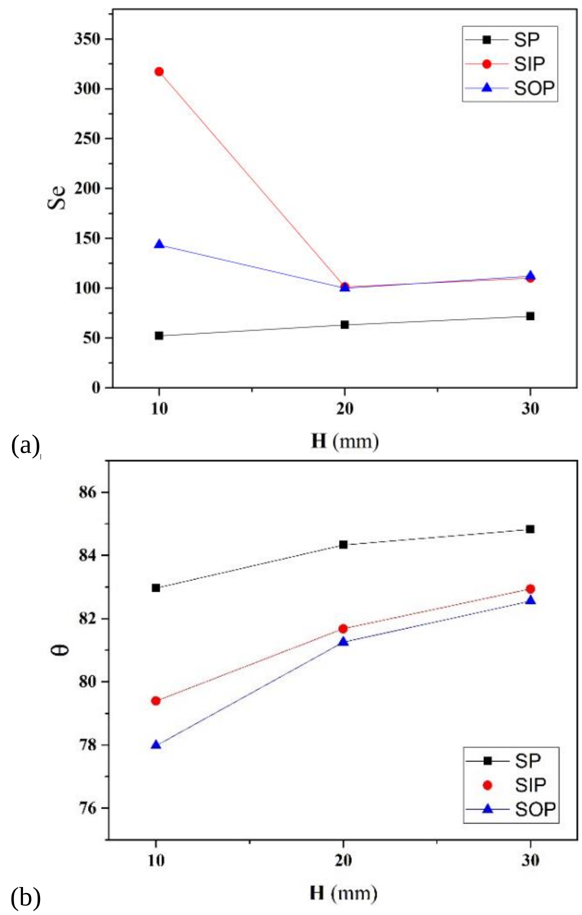


Figure 4 Variation of (a) intensity of secondary flow (Se) (b) FSA (θ) with channel height for different top wall geometries.

represented with the help of 'Field Synergy Angle' (FSA) [12] which is instantaneous angle between velocity and temperature gradient at any point in the flow and at any time. It is given by

$$\theta = \cos^{-1}\left(\frac{\vec{V}\cdot\overrightarrow{\nabla T}}{\left|\vec{V}\right|\times\left|\overrightarrow{\nabla T}\right|}\right) \tag{10}$$

where $\vec{V}$ and $\overrightarrow{\nabla T}$ represent velocity vector and temperature gradient vector respectively. Figures 4(a) and (b) show the variations of Se and FSA with channel height for different top wall geometries. The values of Se presented in Fig. 4(a) defined in Eqn. (9) are the volume weighted average values. FSA presented in the Fig. 4(b) is the volume weighted average value of FSA in the wavy module considered. Wavy channel with lower channel height shows better synergy for all configurations. Synergy between velocity and temperature gradient is found to decrease with increase in channel height. Out of the three configurations studied, SOP shows best synergy followed by SIP and SP. This may be due to presence of stronger secondary flows in the domain. Hence, the enhancement in heat transfer can be attributed to the improved synergy between velocity and temperature gradients in the domain.

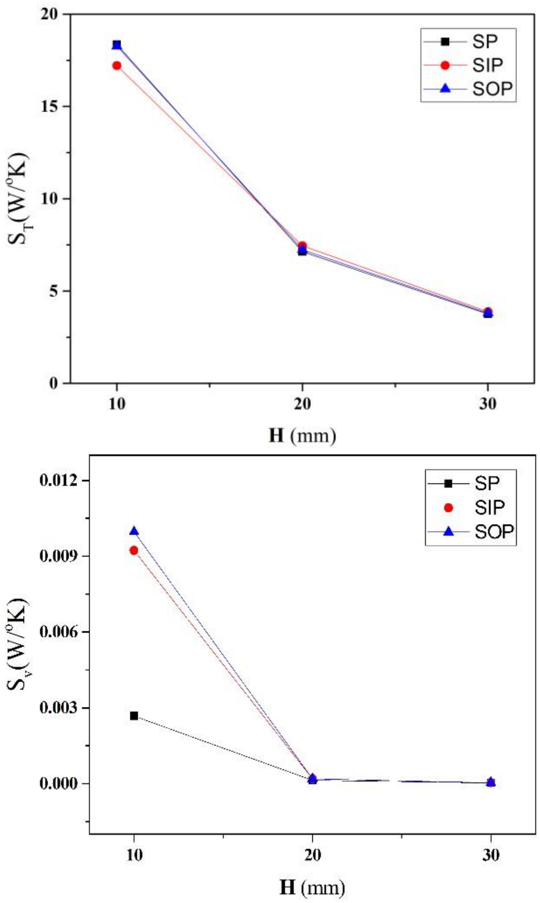
Entropy generation

Heat transfer across temperature gradient and flow with friction are the causes responsible for entropy generation in a channel flow [6]. This section focuses on irreversibility and functioning of entropy generation mechanism inside the confined wavy channel. Entropy generation rate accounting for thermal and friction effects are respectively given as

$$S_T{}''' = \frac{k}{T^2}\left(\frac{\partial T}{\partial x_j}\right)^2 \tag{11}$$

$$S_V{}''' = \frac{\mu}{T}\frac{\partial u_i}{\partial x_j}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \tag{12}$$

Entropy generation analysis helps us to assess the amount of exergy loss in convective heat transfer due to heat transfer and friction factor. Hence, estimation of entropy generation rate would give us an idea of thermodynamic performance of the system. Overall entropy generation inside the domain is calculated by taking volume weighted average value of both $S_T{}''$ and $S_V{}''$, which is given by S_T and S_v. Variation of S_T and S_V with channel height for different top wall configurations has been shown in Fig.5. Model SP corresponds to highest entropy generation both due to heat transfer and viscous effects followed by SIP and SOP. Entropy generation rate decreases with increase in channel height, which may be attributed to the fact that the narrow channels possess higher velocity as well as temperature gradients. It is inferred that the role of top wall geometry in entropy generation becomes significant only at lower channel heights. For higher values of channel height, entropy generation is independent of the nature of top confinement. For wavy channel with larger channel heights, effect of viscosity is dominant only near the walls whereas in rest of the domain it is dominated by the core flow. Moreover, temperature gradients are weaker compared to those in narrow channels. This may be the reason for lower values of entropy generation rates in channels with more height. It is clear from the figure that the entropy generation is dominated by heat transfer rather than the viscous irreversibility. Hence, entropy generation due to viscous irreversibility is less significant in deciding the total entropy generation, which is sum of entropy generations due to heat transfer and friction. Lower entropy generation is observed for channels having more height which represents a thermodynamically efficient system.



(a)

(b)

Figure 5 Variation of S_T and S_V with channel height for various top wall geometries.

CONCLUSION

Three-dimensional numerical investigations are carried out to study the role of entropy generation and field synergy on heat transfer from a confined wavy wall. A sinusoidal wavy wall is considered as bottom of the channel for three different top wall configurations, viz, plane, in-phase and out-of-phase sinusoidal shapes. SIP configuration shows higher heat transfer that increases with increase in channel height followed by SOP and SP configurations. Higher value of friction factor is observed for SIP configuration followed by SOP and SP configurations. Secondary flow generated inside the channel shows weak dependence on channel height for its higher values. For lower channel height, SIP shows higher intensity of secondary flow than that for SOP and SP. Better synergy between velocity and temperature gradient is observed for channels with smaller height. SOP is found to have better synergy followed by SIP and SP configurations. Entropy generation rates, both heat transfer and friction, are found to decrease with increase in channel height and nature of top wall plays significant role only for lower values of channel height. Entropy generation due to heat transfer is found to dominate over that due to friction for all the cases.

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