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Laminar Forced Convection and Heat Transfer Characteristics in a Square Channel Equipped with V-Wavy Surface

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Abstract: The numerical investigations on flow pattern and heat transfer characteristic in a heat exchanger channel equipped with V-wavy surface are examined. The influences of the flow attack angles (α = 15°, 20°, 25°, 30°, 35°, 40°, 45°, 50°, 55° and 60°) and wavy surface arrangements (V-tip pointing downstream called "V-Downstream", V-tip pointing upstream called "V-Upstream) are investigated for laminar regime, *Re* = 100-1200. The finite volume method with SIMPLE algorithm is used to solve the current research. The numerical result shows that the addition of the wavy surface in the heat exchanger channel can help to develop the heat transfer rate and thermal performance. The wavy surface generates the vortex flow through the test section that disturbs the thermal boundary layer on the heat transfer surface. The insertion of the wavy surface in the heating channel gives higher heat transfer rate around 1.5-10 times above the smooth channel depended on the flow attack angle, Reynolds number and wavy surface arrangement. In addition, the optimum thermal performance is around 3.0 at *Re* = 1200, α = 40° and V-Upstream.

Keywords: Heat Transfer Rate, Thermal Performance, Wavy Surface, Heat Exchanger, Finite Volume Method

Introduction

The effort to improve the thermal performance in various types of the heat exchangers is always seen in many industries. The thermal development can help to save production cost, thermal energy and also reduce the size of the heating system. The thermal development in the heat exchanger is divided into two methods; active and passive methods. The active method needs the additional power to enhance heat transfer rate and thermal performance, while the passive method is to generate vortex flow, swirling flow and to disturb the thermal boundary layer by turbulator or vortex generators. The passive method is more popular than the active method.

Many researchers attempt to create the appropriate vortex generator for various engineering works. For examples, the wavy surface (Duan *et al*., 2016; Khoshvaght-Aliabadi *et al*., 2016; Lotfi *et al*., 2016; Lu *et al*., 2017; Ranganayakulu *et al*., 2017; Xiao *et al*., 2017; Xu *et al*., 2015) is widely selected to augment heat transfer rate and performance in fin-and-tube heat exchanger. The wavy surface performs a better fluid mixing in the heating section that the reason for heat transfer augmentation. The amplitude, profile, flow attack angle, etc., of the wavy surface are important parameters that effect for heat transfer rate and thermal efficiency in the heat exchanger. The distinctive points of the wavy surface are as follows; easy for manufacture and maintenance, low pressure loss.

The V-rib/baffle (Abraham and Vedula, 2016; Deo *et al*., 2016; Fang *et al*., 2015; Jin *et al*., 2017; 2015; Kumar and Kim, 2016; Maithani and Saini, 2016; Promthaisong *et al*., 2016; Ravi and Saini, 2016; Singh and Ekkad, 2017) is another type of the vortex generators, which always use to increase heat transfer

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rate and thermal performance in tube/channel heat exchanger. The V-rib gives high heat transfer rate and thermal performance, but also performs enlarge pressure loss.

The new design of the vortex generators is the combination between wavy surface and V-rib called "V-wavy surface". The main aim of the new design is to support the generators production, maintenance and to remain the performance as the V-rib.

In the current research, the V-wavy surface is inserted in the middle of the heat exchanger channel to improve heat transfer rate and thermal efficiency. The influences of the flow attack angle (α = 15°, 20°, 25°, 30°, 35°, 40°, 45°, 50° and 60°), V-wavy surface arrangement (V-Downstream and V-Upstream) and Reynolds number (laminar flow, Re = 100-1200) on heat transfer, friction loss and thermal performance are investigated numerically. The numerical method can help to describe the flow and heat transfer mechanisms in the test section that is a way to improve the thermal system. The numerical results are reported in terms of flow visualizations, heat transfer characteristics and performance assessments.

Channel Geometry and Arrangement

Figure 1 presents a heat exchanger channel inserted with V-wavy surface. The square channel height, H, is equal to 0.05 m. The square profile of the wavy surface is fixed at $0.2H \times 0.2H$. The flow attack angle of the wavy surface is varied; α = 15°, 20°, 25°, 30°, 35°, 40°, 45°, 50°, 55° and 60°. The arrangement of the V-wavy surface is divided into two methods; V-tip pointing upstream called "V-Upstream" and V-tip pointing downstream called "V-Downstream". The periodic module of the present investigation is set around H. The laminar flow, Re = 100-1200, is considered for the present study.

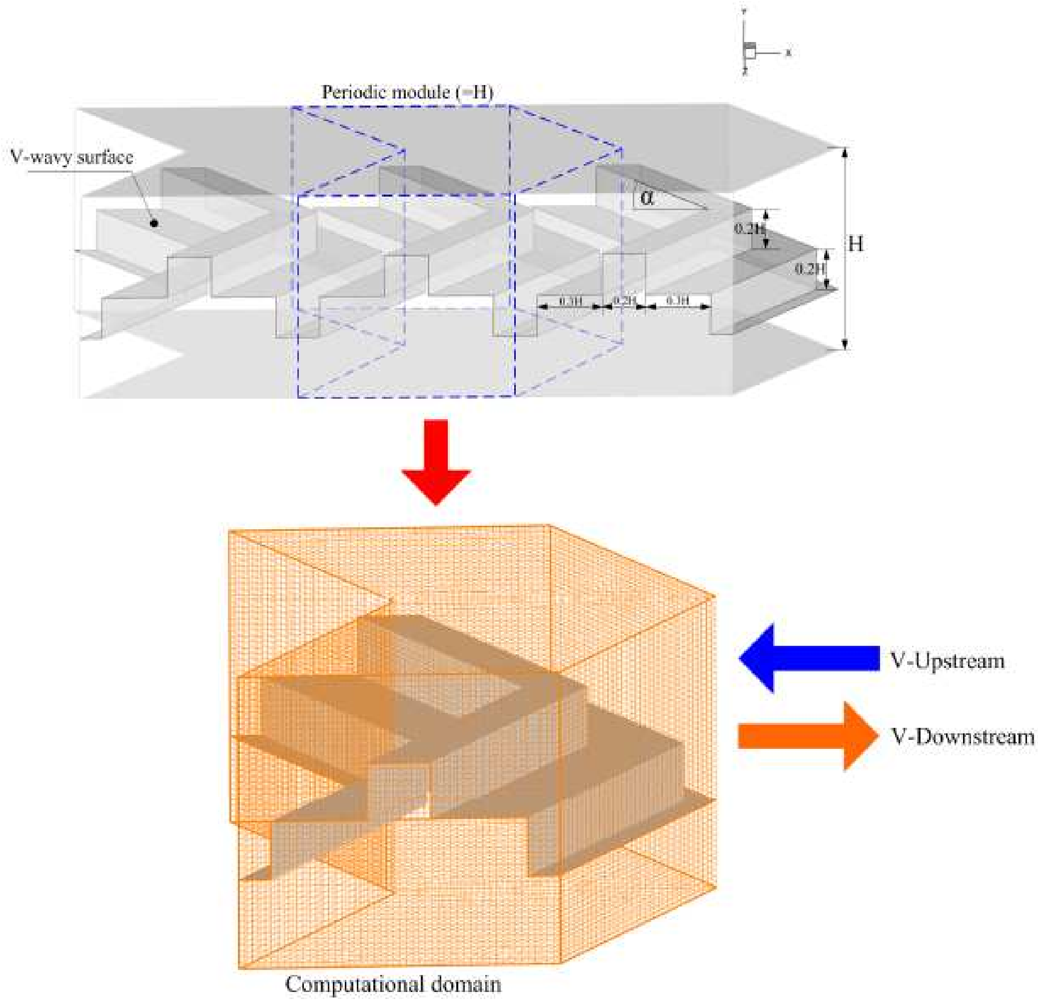

Fig. 1. Square channel inserted with V-wavy surface and computational domain

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Assumption and Boundary Condition

The current work is investigated under following assumptions:

- The thermal properties of the test fluid (air) stay constant at average bulk mean temperature.
- The Prandtl number of the test fluid is around 0.707.
- The flow and heat transfer are steady in three dimensions.
- The laminar flow with incompressible condition is considered for the present study.
- The force convection is regarded.
- The body force, viscous dissipation, radiation heat transfer and natural convection are unconsidered.

The computational domain is created with the boundary conditions as follows:

- The inlet and outlet of the domain are produced with periodic boundary.
- The channel walls are set with constant temperature around 310 K.
- No slip wall condition is used for all surfaces of the computational domain.
- The V-wavy surface is assumed as adiabatic wall (insulator).

Mathematical Foundation

The heating channel is governed by the continuity, the Navier-Stokes equations and the energy equation as Equation 1-3, respectively.

Continuity equation:

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

Momentum equation:

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] \tag{2}$$

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_j}\left(\Gamma\frac{\partial T}{\partial x_j}\right) \tag{3}$$

Γ is the thermal diffusivity, which equal to:

$$\Gamma = \frac{\mu}{\Pr} \tag{4}$$

The energy equation and governing equations are discretized by the QUICK scheme and power law scheme, respectively. The present investigation is answered by finite volume method. The SIMPLE algorithm is selected for the test. The solutions are measured to be converged when the normalized residual values are less than 10^{-5} for all variables, but less than 10^{-9} only for the energy equation. The important variables are Reynolds number, friction factor, local Nusselt number, average Nusselt number and thermal enhancement factor.

The Reynolds number is calculated by:

$$\mathrm{Re} = \rho\bar{u}D_h/\mu \tag{5}$$

The friction factor, f, is determined by pressure drop, Δp, through the periodic module, L:

$$f = \frac{(\Delta p/L)D_h}{\frac{1}{2}\rho\bar{u}^2} \tag{6}$$

The local heat transfer is written as:

$$Nu_x = \frac{h_x D_h}{k} \tag{7}$$

The average Nusselt number can be obtained by:

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

The Thermal Enhancement Factor (TEF) is considered by the augmentations on both heat transfer and friction factor at similar pumping power:

$$TEF = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = (Nu/Nu_0)/(f/f_0)^{1/3} \tag{9}$$

The Nusselt number and friction factor for the smooth square channel are represented by Nu_0 and f_0, respectively.

Numerical Validation

Figure 2 shows the verification of the smooth heat exchanger channel on heat transfer and friction loss. The comparison of the numerical scheme with the value from the correlation is also reported. As the figure, it is found that the difference of the numerical scheme has no effect for the numerical result on both flow and heat transfer. The deviations on the Nusselt number and friction loss are less than ±0.5%.

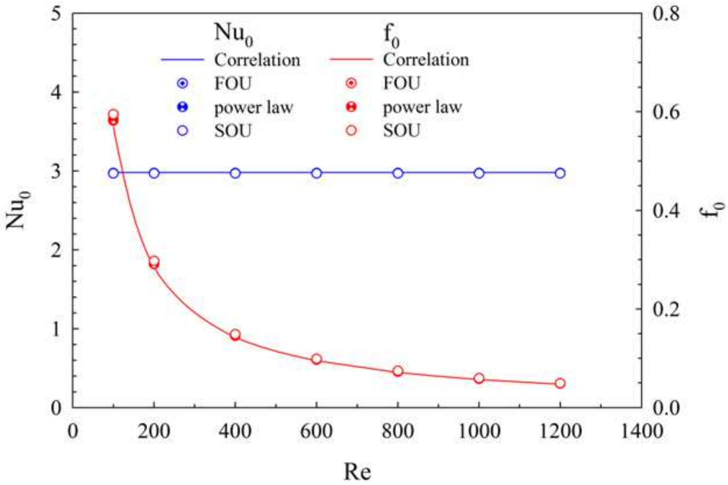
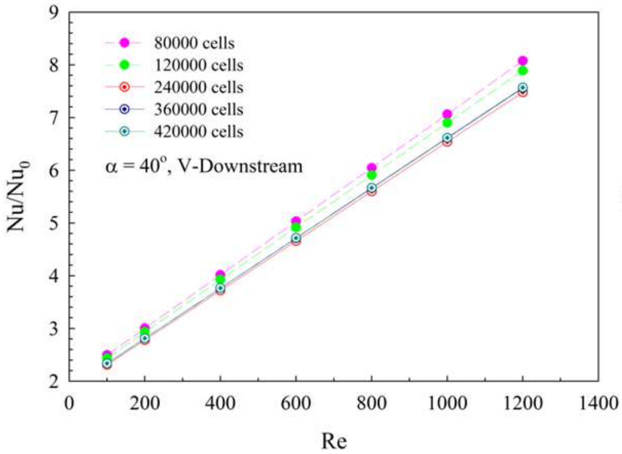
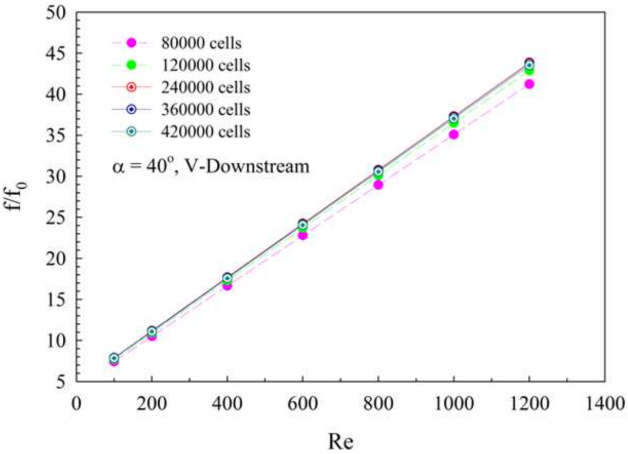


Fig. 2. Validation of the smooth square channel and numerical scheme comparison



(a)



(b)

Fig. 3. Grid independence for (a) Nu/Nu_0 and (b) f/f_0

Figure 3a and b plot the grid independence on heat transfer and friction loss for the computational domain of the heat exchanger channel inserted with V-wavy surface, respectively. The five different grid cells; 80000, 120000, 240000, 360000 and 420000, are compared. The numerical result shows that the increasing grid cell from 240000 to 360000 has no effect for friction factor and Nusselt number. Therefore, the grid around 240000 is applied for all cases of the present investigation. The optimum number of grid cell can help to save time for investigation and computer resource.

Numerical Result

Flow and Heat Transfer Mechanism

Figure 4a and b illustrate the tangential velocity vector in transverse planes (x/H = 0.5, 1.75, 3, 4.25 and 5.5) for the heat exchanger channel inserted with V-Downstream and V-Upstream wavy surfaces, respectively, at Re = 600 and α = 45°. The insertion of the wavy surface on both arrangements changes the flow structure in the square channel. The wavy surface can create the vortex flow through the test section. The vortex flow disturbs the thermal boundary layer on the heat transfer surface that helps to improve the heat transfer rate and thermal performance. The vortex flow in the channel heat exchanger is detected in all cases.

Figure 5a and b report the temperature distribution in transverse planes for the heat exchanger channel inserted with the V-Downstream and V-Upstream wavy surface, respectively, at Re = 600 and α = 45°. The installation of the wavy surface in the heat exchanger channel performs a better mixing of the air flow. The cold fluid distributes from the center of the channel, while the hot fluid near the channel wall performs thinner. The improvement of

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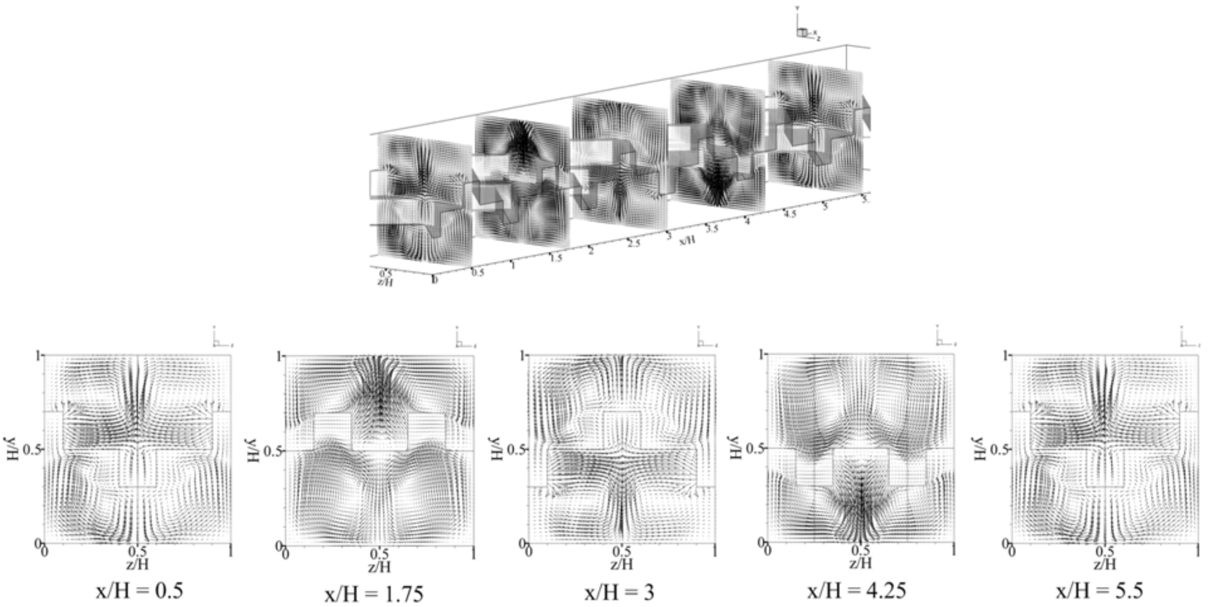
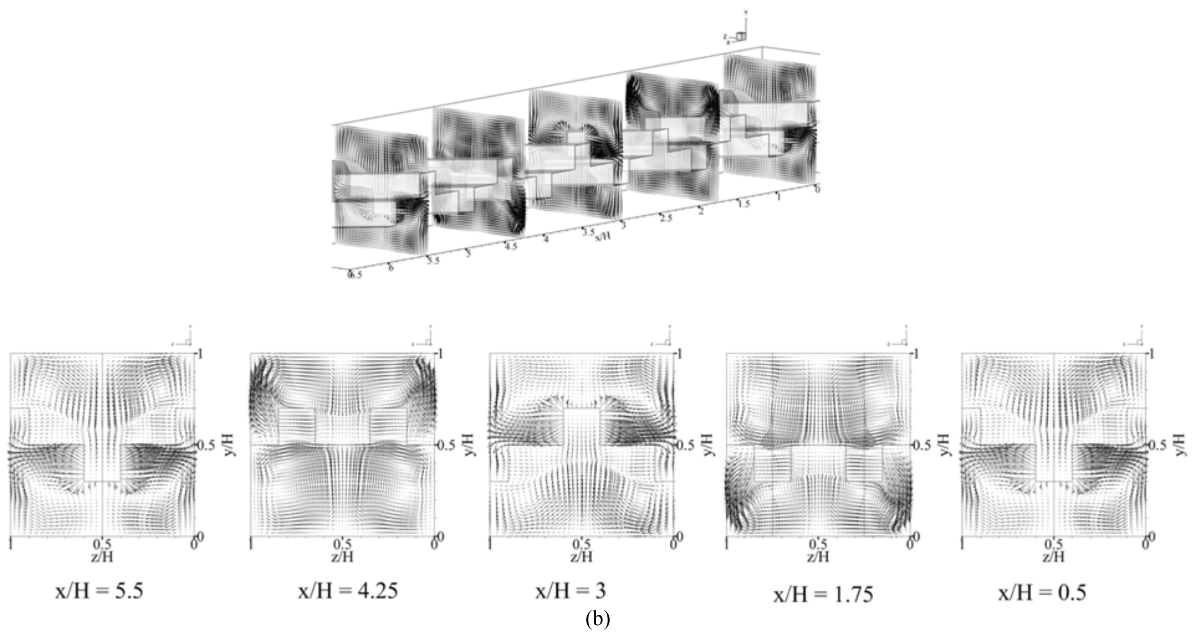
the heat transfer rate and thermal performance is due to the better fluid mixing.

Performance Analysis

The relations of the Nu/Nu_0 with Re at various flow attack angles for the heat exchanger channel installed with V-Downstream and V-Upstream wavy surfaces are plotted as Figure 6a and b, respectively. In general, the Nu/Nu_0 increases when enhancing the Reynolds number for both arrangements. The installation of the wavy surface in the heat exchanger channel provides higher heat transfer rate than the smooth square channel ($Nu/Nu_0 > 1$) in all cases. The heat transfer rate is around 2-9 and 1.5-10 times above the smooth channel for the wavy surface with V-Downstream and V-Upstream, respectively.

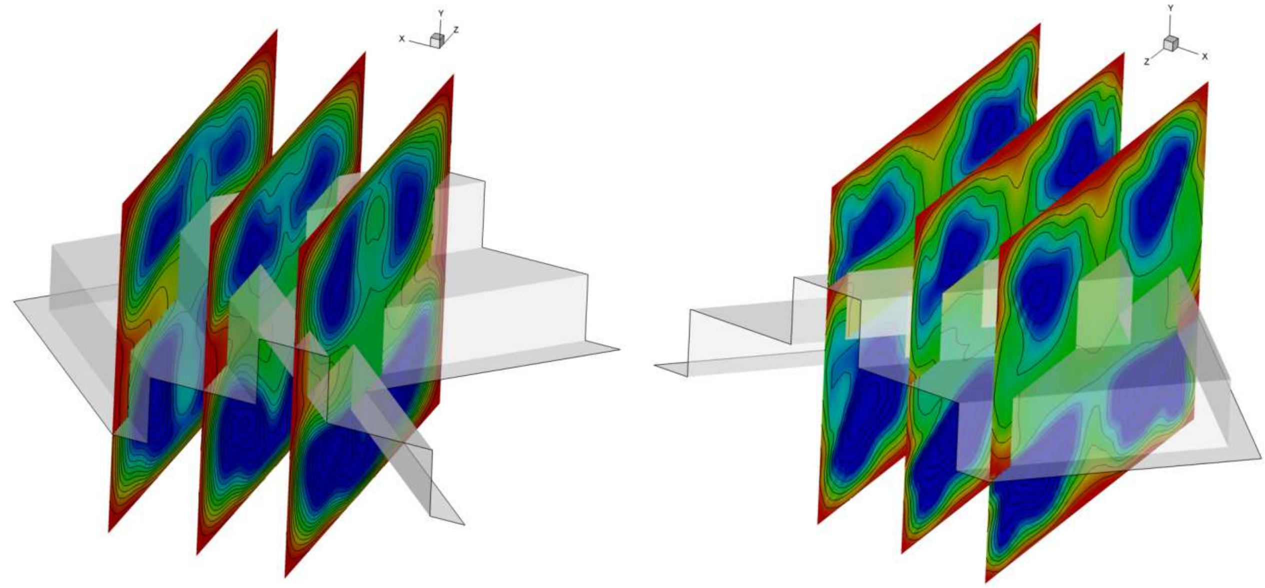


(a)



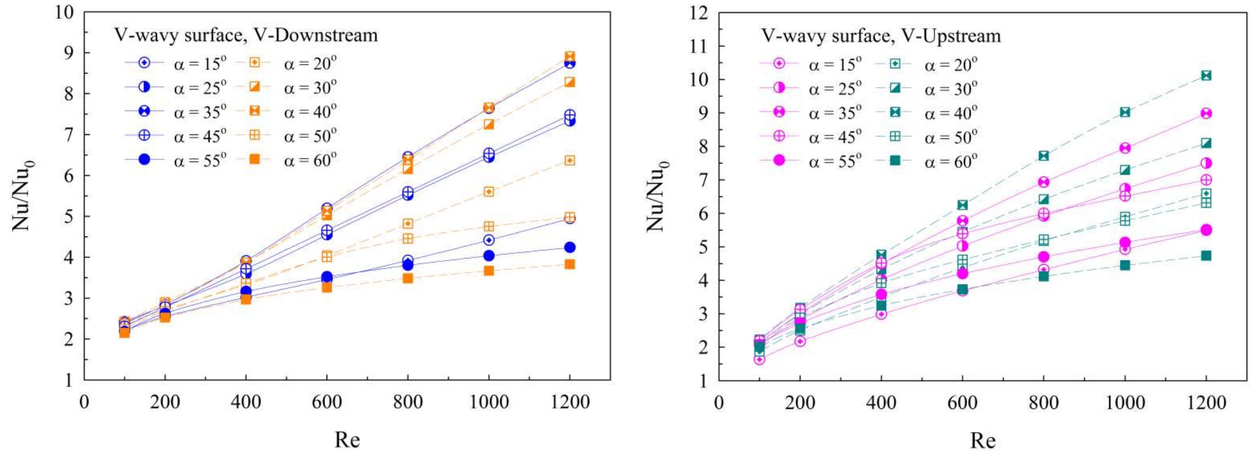
(b)

Fig. 4. Tangential velocity vector in transverse planes for the heat exchanger channel inserted with V-wavy surface of (a) V-Downstream and (b) V-Upstream at $Re = 600$ and $\alpha = 45°$



(a) (b)

Fig. 5. Temperature distribution in transverse planes for the heat exchanger channel inserted with V-wavy surface of (a) V-Downstream and (b) V-Upstream at $Re = 600$ and $\alpha = 45°$



(a) (b)

Fig. 6. Nu/Nu_0 Vs Re for the heat exchanger channel inserted with V-wavy surface of (a) V-Downstream and (b) V-Upstream

The variations of the f/f_0 with the Reynolds number at various flow attack angles for the square channel inserted with V-Downstream and V-Upstream wavy surfaces are depicted as Figure 7a and b, respectively. The present of the wavy surface in the channel heat exchanger not only increases in heat transfer rate, but also enhances the pressure loss. The insertion of the wavy surface provides higher friction loss than the smooth channel for both arrangements ($f/f_0 > 1$). The f/f_0 is around 6-45 and 6-43 for the wavy surface with V-Downstream and V-Upstream, respectively.

Figure 8a and b present the relations of the thermal enhancement factor with the Reynolds number at various flow attack angles for the square channel inserted with V-Downstream and V-Upstream wavy surfaces, respectively. The *TEF* increases when augmenting the Reynolds number, except from α = 55° and 60° of the V-Downstream wavy surface. Almost cases, the insertion of the wavy surface gives higher thermal performance than the smooth channel (*TEF* > 1). In the range investigates, the *TEF* is around 1.05-2.6 and 0.8-3.0 for V-Downstream and V-Upstream wavy surfaces, respectively.

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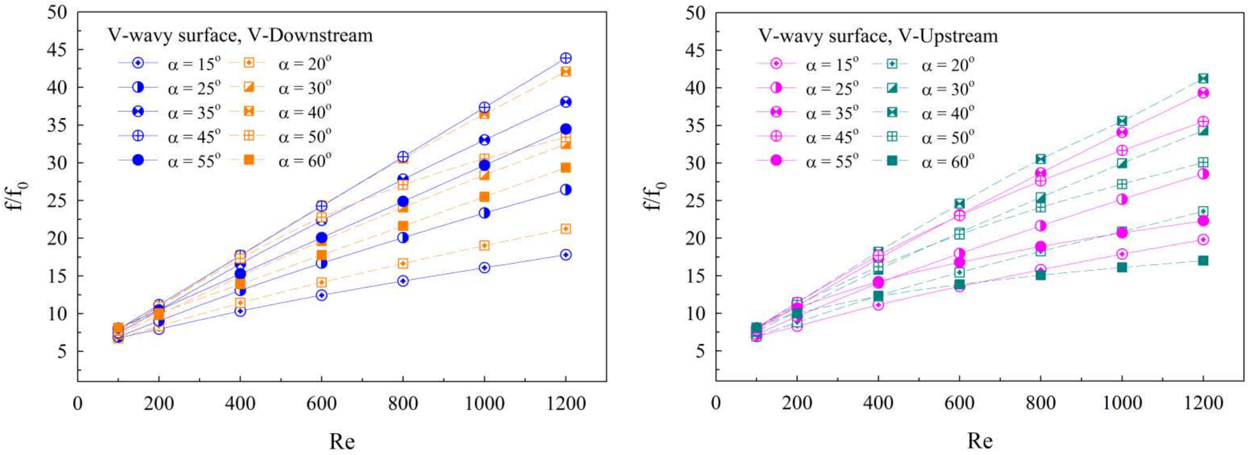
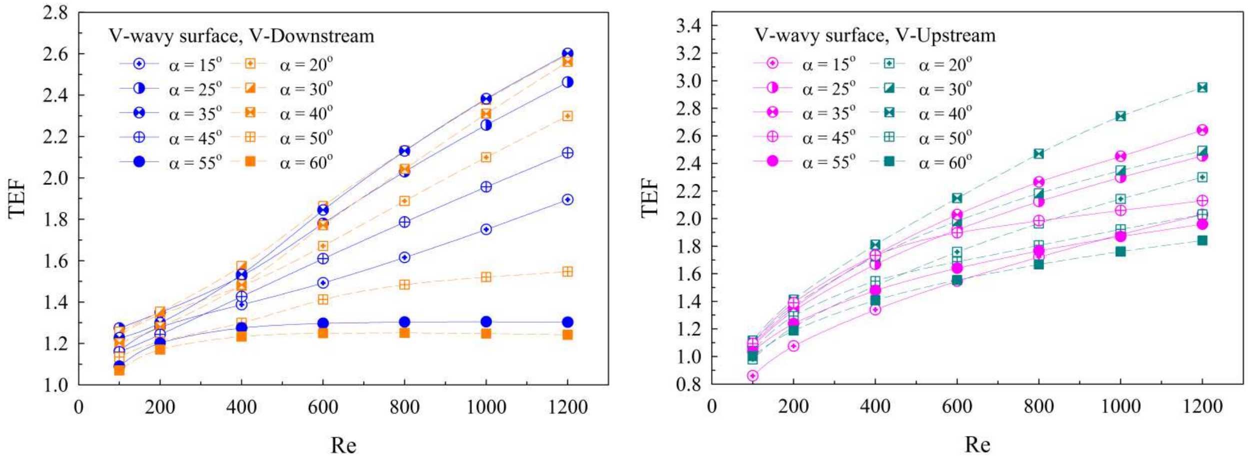


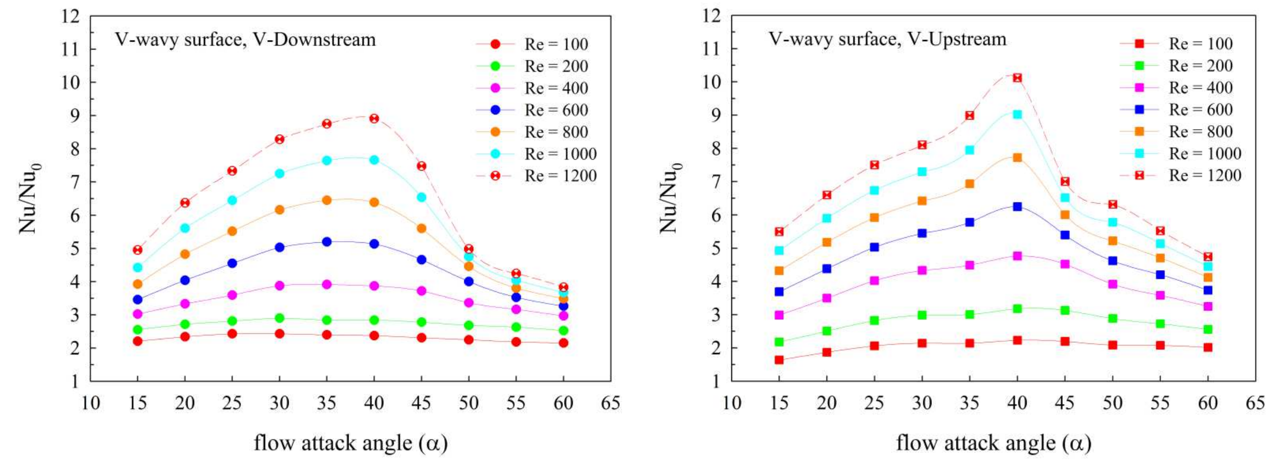
(a) (b)

Fig. 7. f/f_0 Vs *Re* for the heat exchanger channel inserted with V-wavy surface of (a) V-Downstream and (b) V-Upstream



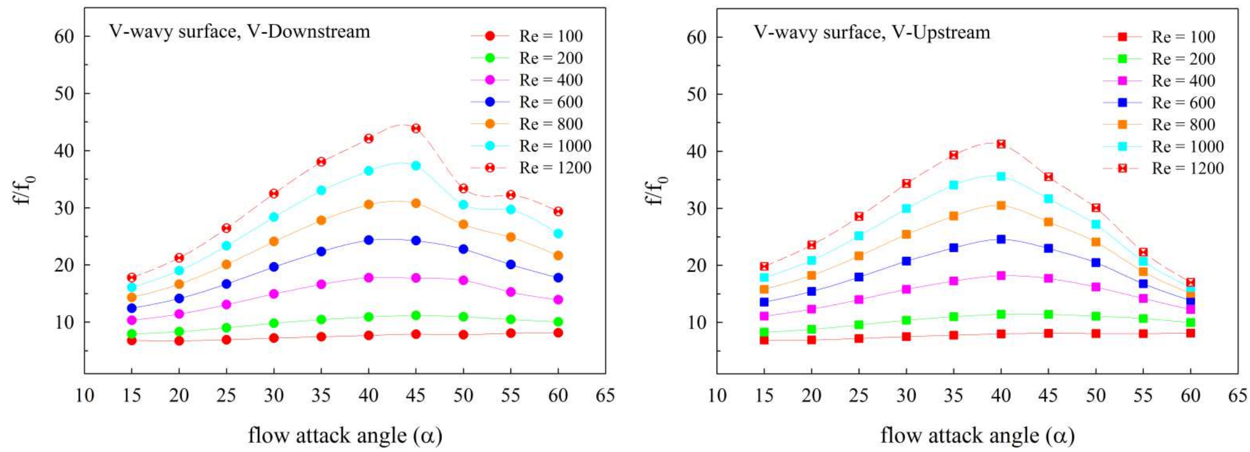
(a) (b)

Fig. 8. *TEF* Vs *Re* for the heat exchanger channel inserted with V-wavy surface of (a) V-Downstream and (b) V-Upstream



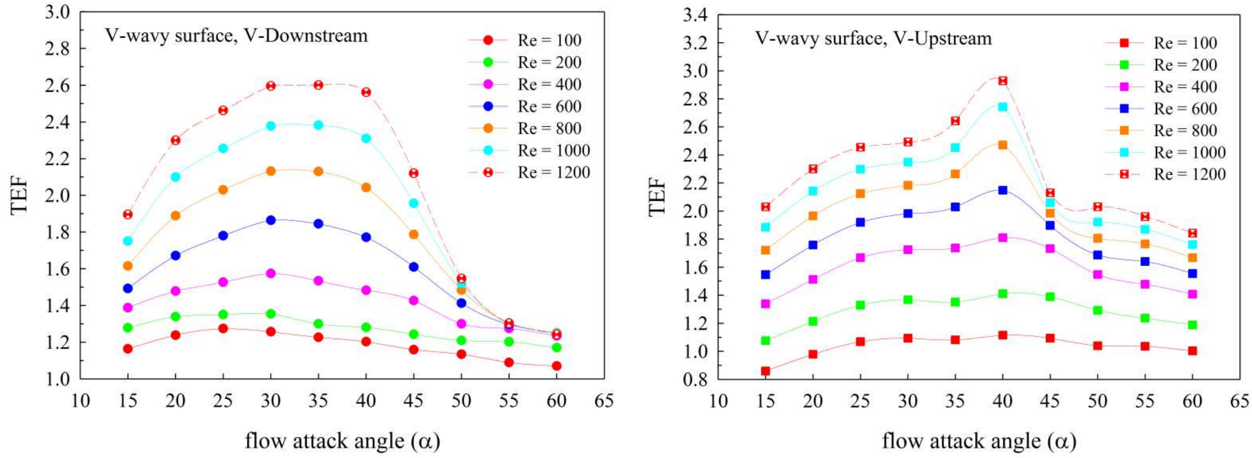
(a) (b)

Fig. 9. Nu/Nu_0 Vs α for the heat exchanger channel inserted with V-wavy surface of (a) V-Downstream and (b) V-Upstream



(a) (b)

Fig. 10. f/f_0 Vs α for the heat exchanger channel inserted with V-wavy surface of (a) V-Downstream and (b) V-Upstream



(a) (b)

Fig. 11. *TEF* Vs α for the heat exchanger channel inserted with V-wavy surface of (a) V-Downstream and (b) V-Upstream

Figure 9a and b report the variations of the Nu/Nu_0 with the flow attack angle at various Reynolds numbers for V-Downstream and V-Upstream wavy surfaces, respectively. In range $15° \le \alpha \le 40°$, the heat transfer rate increases when increasing the flow attack angle. The heat transfer rate decreases when $\alpha > 40°$. The peak of heat transfer rate is found at the flow attack angle of 40° on both arrangements.

Figure 10a and b present the relations of the f/f_0 with the flow attack angle at various Reynolds number for V-Downstream and V-Upstream, respectively. In range $15° \le \alpha \le 45°$ for V-Downstream, the f/f_0 increases when increasing the flow attack angle. The friction factor decreases when $\alpha > 45°$ for V-Downstream. For V-Upstream in range $15° \le \alpha \le 40°$, the f/f_0 increases when enhancing the flow attack angle. The friction loss reduces when $\alpha > 40°$ for V-Upstream. In conclusion, the peak of the friction factor in the heat exchanger channel inserted with V-Downstream and V-Upstream wavy surfaces is detected at 45° and 40°, respectively.

Figure 11a and b report the relations of the *TEF* with the flow attack angle for the heat exchanger channel inserted with V-Downstream and V-Upstream wavy surfaces, respectively. The optimum *TEF* is found at the flow attack angle of 30° and 40° for V-Downstream and V-Upstream wavy surfaces, respectively.

Conclusion

Numerical investigations on flow and heat transfer in a square channel inserted with V-wavy surface are reported. The influences of flow attack angle, V-wavy surface arrangement and Reynolds number on heat transfer and friction factor are studied. The major findings are concluded as follows;

The addition of the wavy surface can improve the heat transfer rate and thermal performance due to the wavy surface generates the vortex flow that disturbs the thermal boundary layer on the heat transfer surface.

The heat transfer rate enhances around 1.5-10 times higher than the smooth channel when inserting the wavy surface in the heating system. The insertion of the wavy surface gives the pressure loss around 6-45 times over the plain channel.

The peak of heat transfer rate is found at the flow attack angle of 40° for both arrangements, while the peak of the friction loss is detected at the flow attack angle of 40° and 50°, respectively, for V-Downstream and V-Upstream.

The optimum *TEF* is around 2.6 and 3.0 for 30° V-Downstream and 40° V-Upstream, respectively, at *Re* = 1200.

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Ethics

This article is original and contains unpublished material. The corresponding author confirms that all of the other authors have read and approved the manuscript and no ethical issues involved.

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Nomenclature

D_h	hydraulic diameter of the square channel, m
H	square channel height, m
f	friction factor
h	convective heat transfer coefficient, W m^{-2} K^{-1}
L	periodic length, m
Nu	Nusselt number
P	static pressure, Pa
Pr	Prandtl number
Re	Reynolds number, $(\rho u_0 D_h/\mu)$
T	temperature, K
TEF	thermal performance enhancement factor, $(Nu/Nu_0)/(f/f_0)^{1/3}$

Greek Letter

μ	dynamic viscosity, kg s^{-1} m^{-1}
ρ	density, kg m^{-3}
α	flow attack angle

Subscript

0	smooth duct
pp	pumping power