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Effect of Using Various Longitudinal Fin Number In Finned Channel Heat Exchangers On Heat Flow Characteristics

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ABSTRACT

This paper presents the air-side overall heat transfer coefficient and pressure drop characteristics for finned channel with various longitudinal fins number. The computational fluid dynamics analysis with ANSYS Fluent software was applied for the analysis of the heat transfer. Numerical simulation was performed for the different longitudinal fins number of (4, 8 and 12), at air flow speeds of 3 m/s to 7 m/s and validated with available correlations. The numerical simulations results, with respect to the pressure drop and exchanged heat were verified by the correlations available in the literature and compared to the simulation results for plain channel. The results indicated that maximum enhancement in Nusselt number of 265%, 221% and 153% credited to the finned channel with 12, 8, and 4 fins respectively. Moreover, the increase in fins number led to significant improvement for both Nusselt number and Euler number. The results showed good agreement with the existing correlations with a maximum deviation of 10% and 12% for Nusselt number and Euler number respectively for all case.

Keywords:
Longitudinal finned-channel, Fins number, Heat transfer, Pressure drop

1. Introduction

The finned tube heat exchangers are widely used in different industrial sectors. In heat exchanger design, it is essential to consider the interactions between the flow distribution as well as the heat transfer within the fins. The principle of using fins in heat exchangers is to enhance the thermal performance [1-4]. Several investigations have been conducted to study the effect of utilizing longitudinal fins in heat exchangers [5-9]. Some of these investigations stated that the boundary layers developments affected by fin spacing. [10-13]. A research by Jameson [14] revealed that the fin spacing considered very effective parameter on the heat transfer coefficient with remarkable effect on pressure drop.

A numerical simulation was achieved for the needle fins and round fins in finned heat exchangers. Results show that the usage of needle fins led to enhance Nusselt number up to 30 % for the considered turbulent Reynolds number with minimizing the mass of heat exchange surfaces of 23,8

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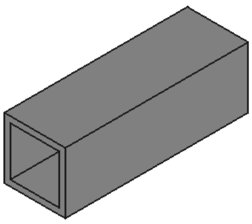
% [15]. Another study was conducted by Zhukauskas [16], of three flow regimes such as laminar, turbulent and separated flow in finned tube heat exchanger, moreover, correlations have been predicted for friction factor and Nusselt number for finned tube heat exchanger. According to the literature review, most of the investigations on finned tube heat exchanger are limited to experiments, furthermore, Jang *et al.,* [17] conducted the first numerical study of the annular finned tube.

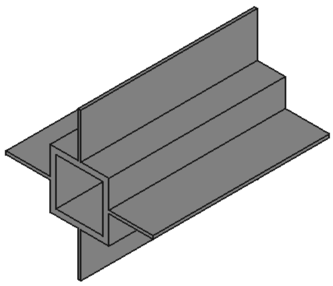
Nevertheless, the effect of using various extended surfaces configuration such as longitudinal and triangular fins in latent heat thermal energy storage system were numerically conducted. The finding revealed that the utilizing of external triangular fins provides higher enhancement about 18% comparing with longitudinal fins configuration.[18]. Additional numerical study has been conducted using longitudinal finned heat pipe in latent thermal energy storage system. The results indicate that thinner fins reduce the system cost, with take in consideration the limitation of the fin thickness to be welded on a heat pipe economically [19].

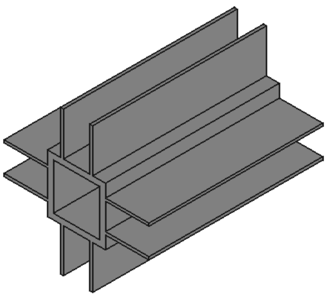
However, according to the published academic reports, the few of the conducted studies were discussed the influence of utilizing the longitudinal fins in heat exchangers. Therefore, the purpose of this study is to investigate the effect of using longitudinal fins with annular finned channel heat exchanger with various fins number on the heat transfer and flow characteristic under turbulent flow regime.

2. Numerical Approach

The proposed geometries of annular-longitudinal finned channel heat exchangers are shown in Figure 1. Furthermore, the geometrical dimensions of the studied models are presented in Table 1 and Figure 2 where the fin spacing accounted as the distance between two fin bases tangentially to the channel surface.


(a) Case 1


(b) Case 2


(c) Case 3

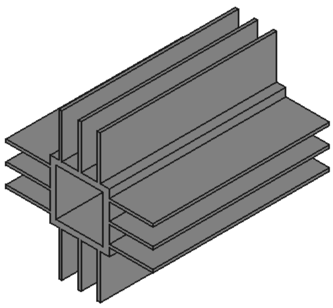

(d) Case 4

Fig. 1. Annular finned channel with longitudinal fins

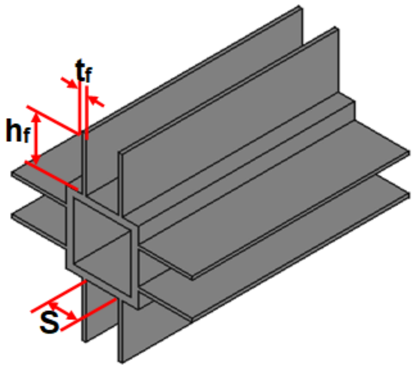


Fig. 2. Geometrical description of Longitudinal finned channel

Table 1
Geometrical dimensions of the studied models

Parameters	Case 1	Case 2	Case 3	Case 4
Channel length	60	60	60	60
Square channel rib, a	20	20	20	20
Number of fins, n	0	4	8	12
Fin height, h_f	0	15	15	15
Fin thickness, t_f	0	1	1	1
Fin base spacing, S	0	19	9	5.66
Fin pitch, S_f=S+ t_f	0	20	10	6.66

2.1 Governing Equations

For the proposed geometrical shape of the annular finned channel heat exchanger with a considered range of Reynolds number of (2,500, 5,000 and 10,000), the three dimensional, unsteady, incompressible and turbulent flow have been assumed to formulate the governing equations of continuity, momentum, and energy equations as following [20]
Continuity equation

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

Momentum equation:

$$\rho \frac{\Delta u_i}{\Delta t} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_i}{\partial x_i}\right)\right] + \frac{\partial}{\partial x_j}\left(-\rho\overline{u_i' u_j'}\right) \tag{2}$$

where

$$-\rho\overline{u_i' u_j'} = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\left(\rho k + \mu_t \frac{\partial u_i}{\partial x_i}\right)\delta_{ij} \tag{3}$$

The transport Eqs. 4 and 5 have been employed using RNG k-ε turbulent model in this numerical investigation.

$$\rho \frac{\Delta k}{\Delta t} = \frac{\partial}{\partial x_i}\left[\alpha_p \mu_{eff} \frac{\partial k}{\partial x_i}\right] + \mu_t S^2 - \rho\varepsilon \tag{4}$$

$$\rho \frac{\Delta \varepsilon}{\Delta t} = \frac{\partial}{\partial x_i}\left[\alpha_p \mu_{eff} \frac{\partial \varepsilon}{\partial x_i}\right] + C_{1\epsilon} \frac{\varepsilon}{k} \mu_t S^2 - C_{2\epsilon}\rho \frac{\varepsilon^2}{k} - R \tag{5}$$

$$R = \frac{C_\mu \rho \eta^3 (1-\eta_o)}{1+\beta\eta^3} \frac{\varepsilon^2}{k} \tag{6}$$

Meanwhile, μ_t is turbulent viscosity and C_μ=0.0845.

$$\mu_t = \rho \times C_\mu \times \frac{k^2}{\varepsilon} \tag{7}$$

The ($C_{1\varepsilon}$ =1.42 and $C_{2\varepsilon}$ =1.68) are model constants and derived analytically by the RNG theory [20]. Additionally, ß=0.012, η= Sk/ε , and η_0= 4.38, while, S is strain rate magnitude.
Energy equation

$$\frac{\partial}{\partial t}(\rho E) + \frac{\partial}{\partial x_i}(u_i(\rho E + p)) = \frac{\partial}{\partial x_i}\left(k_{eff} \frac{\partial T}{\partial x_i}\right) \tag{8}$$

where: *E* is total energy, while K_{eff} is effective conductivity represented with
$k_{eff} = k + k_t$

The equation of the energy transport has been applied within the fins (solid region) and defined as in Eq. 9 with neglect the radiation effects and the buoyancy.

$$\frac{\partial}{\partial t}(\rho c_p T) = \frac{\partial}{\partial x_i}\left(k_s \frac{\partial T}{\partial x_i}\right) \tag{9}$$

2.2 Boundary Conditions and Grid Independence

The numerical simulation has been carried out for four different geometries of finned square channels varies with fins number of (4, 8 and 12 mm) under turbulent flow regime with uniform velocity and temperature (T_{in}=300 K). The heat conduction in the fin body and heat convection from the fin to the surrounding are considered. Meanwhile, at the inner channel wall, a constant temperature (T_w =373 K) is assigned and all components of the velocity are set to be zero.

Furthermore, grid independent study has been approved with a various number of grid elements as shown in Table 2. The relative errors in the pressure drop and heat transfer coefficient between proposed mesh grids should be less than 5% as reported by [21]. Nevertheless, the applicable element number 1,200,187 cells are utilized to formulate the mesh of the computational domains. The finite volume method with (RNG) renormalization group theory-based K-ε turbulent model is applied to estimate the heat transfer and flow characteristics.

Table 2
Grid independence

Number of Elements	h, Case-2	h, Case-3	h, Case-4	ΔP, Case-2	ΔP, Case-3	ΔP, Case-4
34,562	150.85	166.48	195.16	1584.37	1630.72	2083.67
103,689	134.59	150.91	183.91	1476.04	1481.04	1903.28
362,883	120. 41	147.53	178.47	1383.92	1320.83	1852.71
1,200,187	112.83	136.47	174.29	1243.76	1265.61	1815.33
1,552,460	111.74	135.92	173.84	1239.85	1264.93	1814.87
1,705,112	111.68	135.87	173.72	1239.85	1264.88	1814.75

2.3 Data Reduction

The rate of heat transfer of the heat exchanger is estimated for the air-side based on Eq. 10 where the (H_{in}) and (H_{out}) refer to the rates of the enthalpy flow of the computational domain inlet and outlet section respectively.

$$\dot{Q} = \dot{H}_{out} - \dot{H}_{in} \tag{10}$$

Then the heat transfer coefficient (*h*) is estimated using Eq. 11 where the η is the fin efficiency, A_f is the area of fin surface and A_t is area of tube surface without fin.

$$h = \frac{\dot{Q}}{(A_t+\eta A_f)\theta} \tag{11}$$

where *Θ* is the (LMTD) log mean temperature difference,

$$\theta = \frac{T_{in}-T_{out}}{ln\frac{T_{in}-T_w}{T_{out}-T_w}} \tag{12}$$

$$\eta = \frac{\tanh(\psi m h_f)}{\psi m h_f} \tag{13}$$

where

$$m = \sqrt{\frac{2h}{k_f t_f}}\ , \psi = 1.0 + 0.35ln\left(1.0 + 2.0\frac{h_f}{d}\right)$$

Additionally, Nusselt number and Euler number are estimated based on Eqs. 14 and 15 respectively.

$$Nu = \frac{hd}{k_a} \tag{14}$$

$$Eu = \frac{\Delta P}{\rho u_{max}^2} \tag{15}$$

3. Numerical Results Validation

In order to validate and evaluate the current numerical results of the pressure drop represented with Euler number and heat transfer represented with Nusselt number, comparison with the available correlations in the literature has been achieved. Figure 3.a presents a comparison of the Nusselt number with Schmidt correlations of [22] Eq. 16 and VDI [23] Eq. 17.

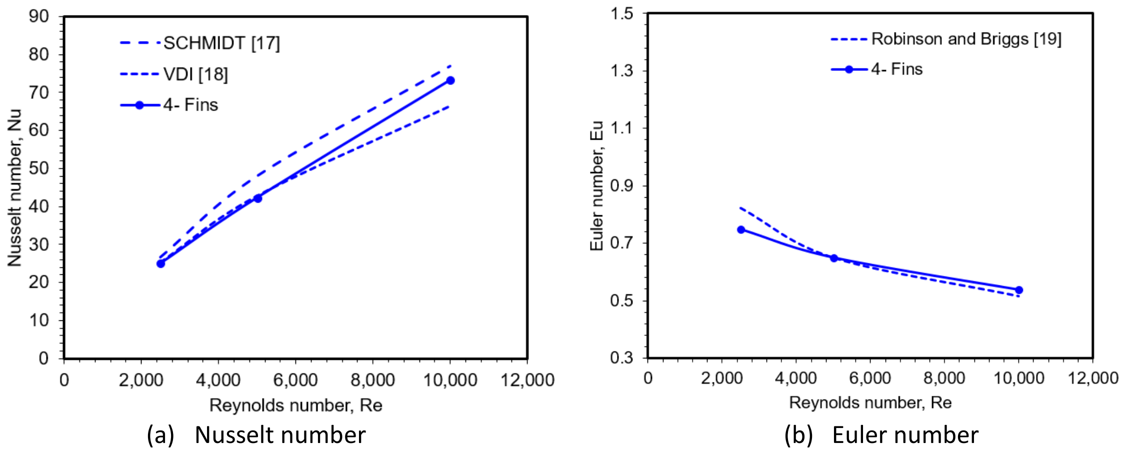
It can be observed that the Nusselt number for all fins number increased with increasing the fins number and Reynolds number. Furthermore, the results show good agreement with the correlations with maximum deviations of 10%.

$$Nu = 0.45Re^{0.625}Pr^{\frac{1}{3}}\left(\frac{A}{A_t}\right)^{-0.375} \tag{16}$$

$$Nu = 0.38Re^{0.6}Pr^{\frac{1}{3}}\left(\frac{A}{A_t}\right)^{-0.15} \tag{17}$$

Figure 3(b) shows the validation of the Euler number results of the finned channel with four fins versus the correlation of Robinson and Briggs [24] Eq. 18, where the pressure drop is represented by Euler number. It is observed from the comparison of the presented results and the obtained results from Robinson and Briggs [24] that the Euler number increased with decrease Reynolds number and increase fins number. Nevertheless, the result shows that the maximum deviation is 12 % although the absence of the finning geometrical variables, for instance, the fins number, fin thickness, height, and spacing.

$$Eu = 18.93Re^{-0.316}\left(\frac{S_t}{d}\right)^{-0.927} n \tag{18}$$



(a) Nusselt number (b) Euler number

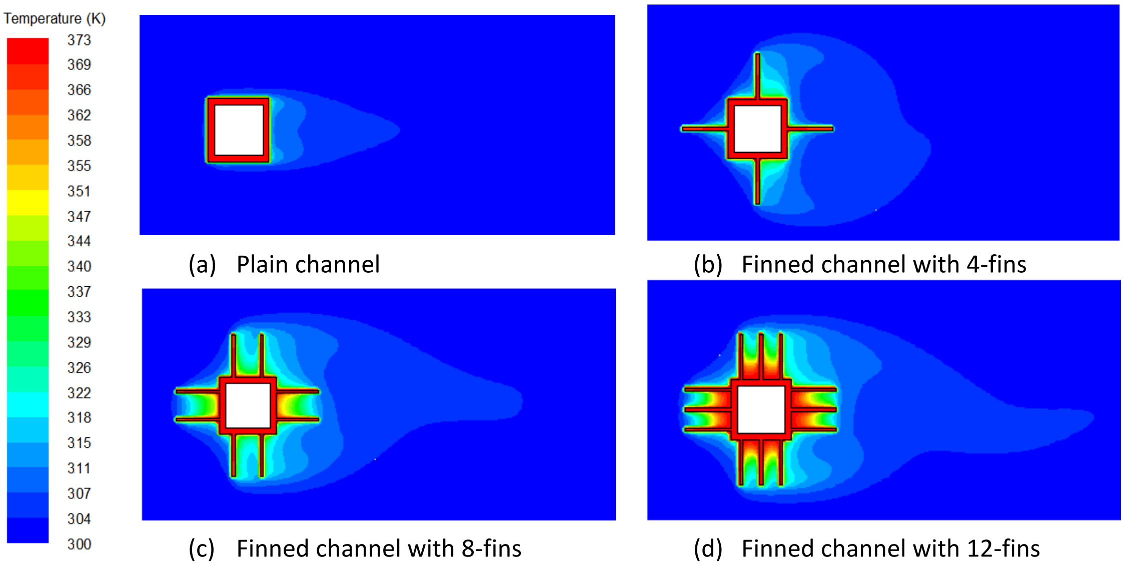
Fig. 3. The present results versus available correlations

4. Results and Discussion

4.1 The Visualization Results of Temperature Distribution and Flow Behaviour

With the intention to describe the behaviour of temperature distribution and clarify the heat transfer phenomena, its necessary to present the contour of temperature distribution as shown in Figure 4, for four cases such as plain channel, finned channel with four fins, eight fins and twelve fins under turbulent flow regime with Re=5,000. Figure 4(a) illustrated the temperature distribution over the plain channel, it obviously observed the weakness of temperature distribution and propagation.

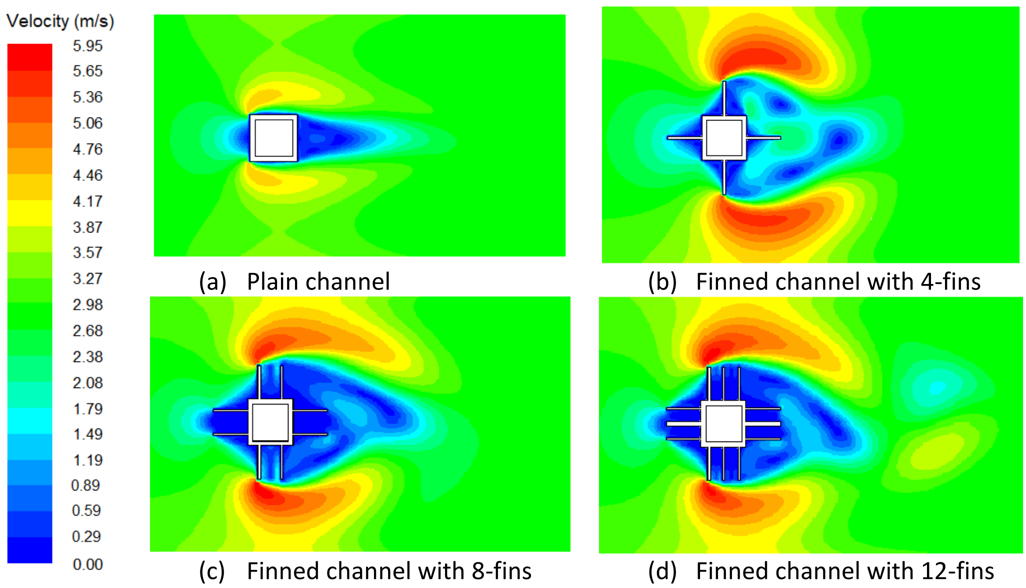
This can be attributed to the small contacted surface area of the channel with surrounding flow. Furthermore, Figure 4(b), 4(c), and 4(d) illustrate the temperature distribution over the finned channel with different fins number 4, 8, and 12 respectively. It can obviously observe the significant incremental in temperature distribution over fins surfaces especially between the fins and this clearly shown in Figure 4(b) and 4(c). The finding reveal that the heat transfer increased and enhanced with increase fins number due to increase surface contact area. Figure 5, shows the velocity contour for finned channels with different fins number to clarify the effect of fins number on the stream flow. Figure 5(a) displays the velocity contour over plain channel, while figures 5(b), (c), and (d) display the contours of finned channel with various fins number of (4, 8 and 12) respectively. It is clearly observed the effect of fins number on the flow intributions over the finned channel surfaces and increase the fluctuations near fins wall where with increase the fins number the contact surface increase as well and this lead to increase the turbulent intensity and reducing the boundary layers the enhancing the heat transfer.



(a) Plain channel (b) Finned channel with 4-fins

(c) Finned channel with 8-fins (d) Finned channel with 12-fins

Fig. 4. Temperature distribution contour at Re=5,000



(a) Plain channel (b) Finned channel with 4-fins

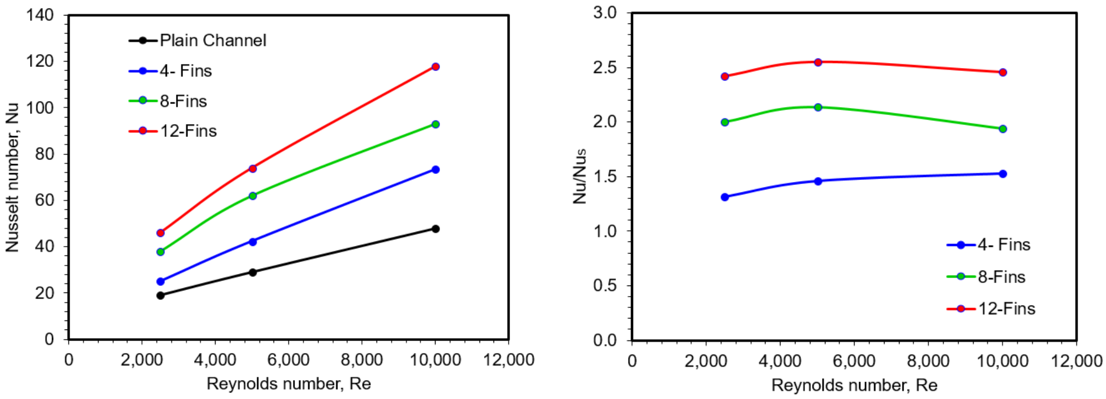
(c) Finned channel with 8-fins (d) Finned channel with 12-fins

Fig. 5. Velocity contour at Re=5,000

4.2 Effect of Fins Number on Heat Transfer and Flow Characteristics

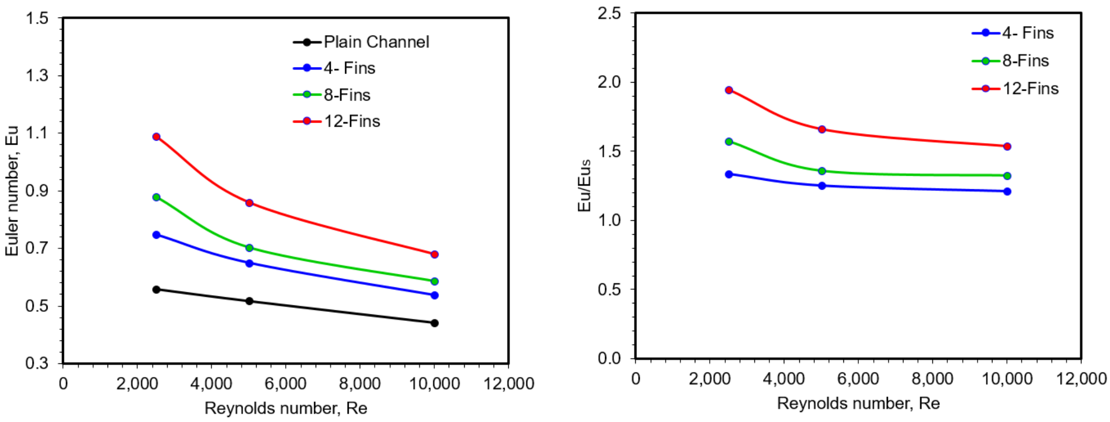
The effect of fins number on heat transfer is shown in Figure 6. The numerical results of Nusselt number versus Reynolds number are illustrated in Figure 6(a), the results confirm that the increase in stream velocity represented by Reynolds number lead to enhance Nusselt number significantly as well as the increase in fins number provides higher enhancement in Nusselt number. Moreover, Figure 6.b. show the Nusselt number ratio against Reynolds number with different fins number. The results confirm the significant effect of increasing fins number, where the maximum heat transfer enhancement represented by Nusselt number enhancement assigned to the finned channel with twelve fins followed by eight and four fins.

Similarly, Figure 7, displays the Euler number results against Reynolds number with different fins number and compared with the plain channel as shown in Figure 7(a) the result indicate that the Euler number increased with increase fins number and decreased with increasing Reynolds number. Meanwhile, Figure 7(b), presents the Euler number enhancement ratio against Reynolds number. The results demonstrate a significant enhancement of Euler number for the finned channel with twelve longitudinal fins followed by eight, and four fins. Therefore, the increase in fins number led to significant improvement for both Nusselt number and Euler number.



(a) Nusselt number results (b) Nusselt number enhancement

Fig. 6. Numerical Nusselt number results versus Reynolds number



(a) Euler number results (b) Euler number enhancement

Fig. 7. Numerical Euler number results versus Reynolds number

5. Conclusion

The numerical investigation on heat exchangers with an annular-finned channel successfully achieved with different fins number to investigate the pressure drop and the heat transfer characteristics. The results show significant incremental in temperature distribution over fins surfaces especially between the fins, the finding reveal that the heat transfer increased and enhanced with increase fins number due to increase surface contact area. Moreover, the increase in stream velocity represented by Reynolds number lead to enhance Nusselt number significantly as well as the increase in fins number provide higher enhancement in Nusselt number, where the maximum heat transfer enhancement represented by Nusselt number enhancement assigned to the finned channel with twelve fins followed by eight and four fins. Nevertheless, the results demonstrate a significant enhancement of Euler number for finned channel with twelve longitudinal fins followed by eight, and four fins. Therefore, the increase in fins number led to significant improvement for both Nusselt number and Euler number. A good agreement was observed in the comparison of the current finding and the existing correlations with a maximum deviation of 10% and 12% for Nusselt number and Euler number respectively for all case.

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