Effect of Using Combined Square Nozzle & winglet with Helical Tape on Thermal Characteristics in Tube Heat Exchanger

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Abstract

Influence of combined square nozzle with helical tape inserted in a constant heat flux tube on heat transfer enhancement for turbulent airflow for Reynolds number ranging from 7000 to 14500 were investigated experimentally. Three different pitch ratios for square nozzle (PR = 5.8, 7.7 and 11.6) according to three different numbers of square nozzle (N = 3, 4 and 5) and constant pitch ratios for helical tape were used. The results observed that the Nusselt number and friction factor for combination with winglets were found to be up to 33.8 % and 21.4 %, respectively higher than nozzle alone for pitch ratio PR=5.8. The maximum value of thermal performance for using combination with winglets was about 1.351 for pitch ratio= 5.8. Nusselt number and friction factor correlations from experimental data were developed for turbulent flow.

Keywords: Heat transfer enhancement, thermal performance, turbulators, winglets.

1. Introduction

Thermal performance improvements needed in many industrial applications such as heat exchangers, solar air heater, thermal power generation etc. Therefore, heat transfer enhancement methods are often used in a heat exchanger to improve heat transfer coefficient [1-6]. Internal inserts in tubes is one of the most effective approaches for heat transfer enhancement without external power. These techniques lead to the generation of swirling flow which increases the turbulence that is responsible for enhancement heat transfer [7]. Numerous investigations using insertion of various turbulators configurations on the improvement of heat transfer were performed. Eimsa-ard et al. [8] studied the thermal characteristics through tube by using twisted tape elements. The results reveals that Nusselt number was significantly increased with twist ratio. Bhuiya et al. [9] studied experimentally the effect of triple helical tapes with different helix angles inserted in tube on heat transfer enhancement. The results showed that the heat transfer increased at the expense of increased blower power. the influence of circular cross sectional rings inserted in tube on the enhancement of heat transfer was studied a numerically by Ozceyhan et al. [10]. The results showed that the reduction of the spacing of the rings resulted in an increase in both Nusselt number and the friction factor. The effect of triple and quadruple twisted tapes on the improvement of heat transfer through tube was investigated by Zhang et al. [11]. The results show that the heat transfer rates and the friction factors in the tubes triple twisted tapes was higher than that of the plain tube. Influence of twin twisted tapes inserted through a helical-ribbed tube on heat transfer characteristics for turbulent flow with Reynolds

number ranging from 6000 to 60,000 was studied experimentally by Pongjet P. et. al [12]. The results showed that the twin twisted produces better thermal performance than the plain tube. This work is aimed to study the Influence of combined square nozzle and winglet with helical tape on thermal characteristics in tube heat exchanger.

2. Experimental Set up and Procedure
2.1. Experimental Set

The experimental rig of this work is consists of the heat transfer test section, high pressure blower, measurement device, augmentation turbulators. as shown in Fig. 1. The test section is made of an aluminum tube of length, inner diameter and thickness are 1350 mm, 45mm and 2.5 mm respectively. Flexible electrical wire was attached around the tube as an electric heater to heat the test tube at uniform heat flux condition. The variac transformer was used to control electrical power. The outer surface of the test section and heater coil were insulated by three layers of gypsum, glass fiber tape and asbestos tape to prevent heat transfer to surrounding. Eighteen thermocouples type-K with ±0.1 0C accuracy connected to measure surface temperatures. To measure air inlet and outlet temperatures two type-K thermocouples with ±0.1C accuracy were positioned at the entrance and exit of the test section the inlet air temperature was set to (20±1C), all thermocouples connected with selector switch and a digital reader. The orifice meter and inclined manometer were used to measure the air flow rate. six values of air flow rates were conducted to cover Reynolds number range from 7000–14500. Pressure drop through tube was measured using U-tube manometer.

2.2. Heat Transfer Augmentation Devices

In this work two types of inserts were used first square nozzle without winglets and with winglets. square nozzle inserts were made of aluminum with dimensions as shown in figure (3). Second type of inserts was helical tape having one types of configurations as shown in fig.3. The helical tape made of copper sheet was 1350 mm long, 3 mm tape width (w) and 1.2 mm thick (t) as shown in fig. (3) . all of the combination of square nozzle and helical tape inserts were located in

converging square nozzle arrangements with three different pitch lengths as shown in fig (3).

2.3. Experimental Procedure

The electric heater was first switched on and allowed to heat the test tube wall at uniform heat flux condition and the blower was then switched on. Firstly the experiment was carried out on the plain tube and then on different insertion turbulators. The air flow rate through the tube was set to a desired value. After reaching the stability condition, the inlet and outlet temperatures of air, pressure drop and the temperature variations along the wall of test section were recorded.

2.4. Data Reduction

The friction factor (f) is given by
$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\left(\frac{\rho u^2}{2}\right)} \qquad \ldots(1)$$

Heat absorbed by the air (Qair) is less by 5 % than the heat provided by the electrical heater (VI) due to the loss of heat to the surrounding
$$Q_{air} = Q_{cov} = \dot{m}C_p(T_o - T_i) = VI - Q_{loss} \qquad (2)$$
The heat transfer by convection from the test tube section can be evaluated as follows:
$$Q_{cov} = hA(\bar{T}_w - T_b) \qquad \ldots(3)$$

The average coefficient of heat transfer can be written as
$$h = \frac{\dot{m}C_p(T_o - T_i)}{A(\bar{T}_w - T_b)} \qquad \ldots(4)$$
Where: bulk mean temperature (T_b) is defined as:
$$T_b = \frac{(T_o + T_i)}{2} \qquad \ldots(5)$$
The average wall temperature is given by
$$\bar{T}_w = \sum T_w / 18 \qquad \ldots(6)$$
Nusselt number can be obtained by
$$Nu = \frac{hD}{k_a} \qquad \ldots(7)$$
The thermal performance (η) is defined as the Nusselt number ratio to friction factor ratio of insert to that of a plain tube [13]:
$$\eta = \frac{\frac{Nu_t}{Nu_p}}{\left(\frac{f_t}{f_p}\right)^{\frac{1}{3}}} \qquad \ldots(8)$$
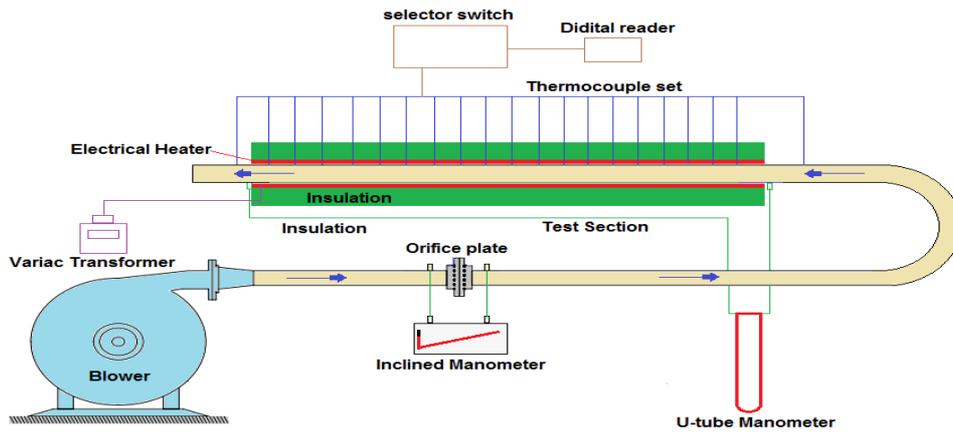
Fig. 1. Schematic diagram of experimental set.

Fig. 2. photo of experimental set up .
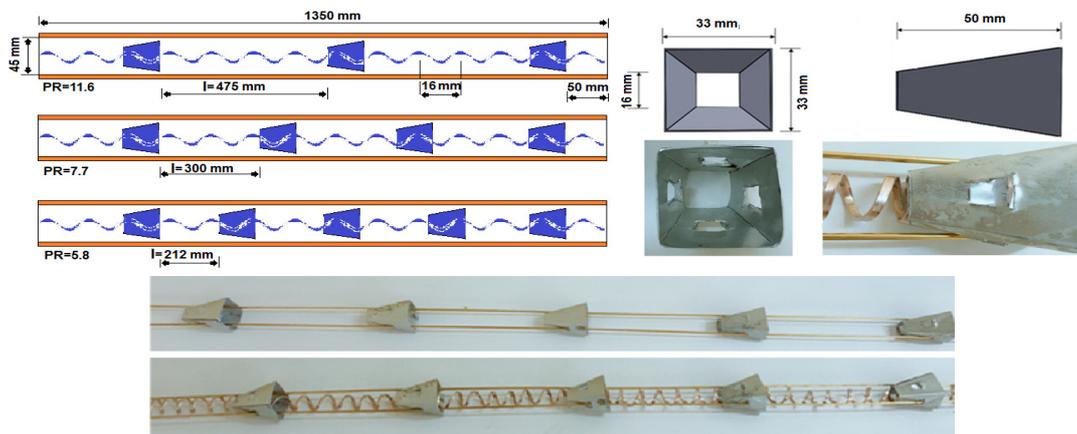
Fig. 3. Test tube with square nozzle and winglet alone and with helical tape.

3.   Results and Discussion

The experimental results of Nusselt number and friction factor for plain tube were validated by comparing with those obtained from Dittuse Boelter correlations and Blasius correlation [14].as shown in fig (4) and (5). The comparison shows a reasonable agreement deviation of (5.5 %) and (4.3%) for Nusselt number and friction factor.
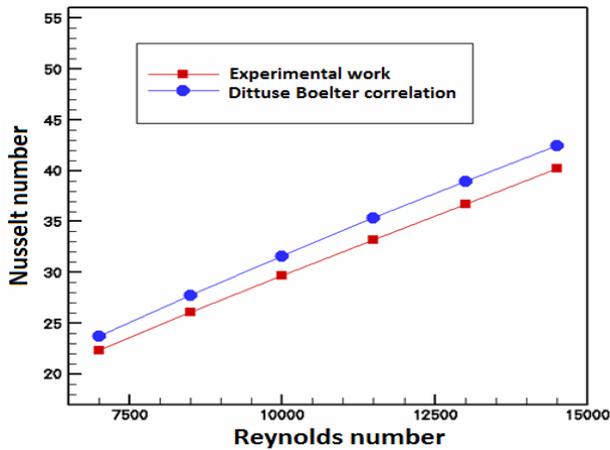
Fig. 4. Nusselt number versus Reynolds number for plain tube
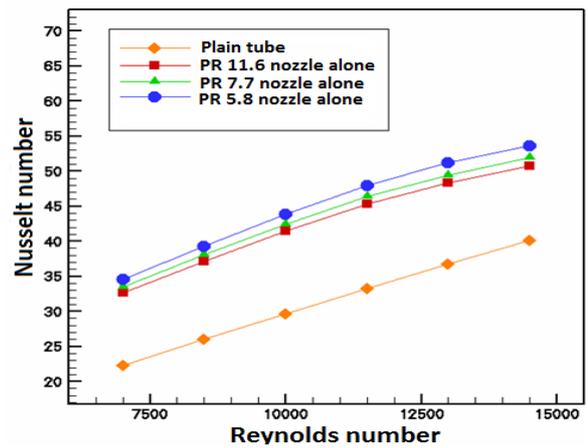
Fig. 6. Nusselt number versus Reynolds number for nozzel alone
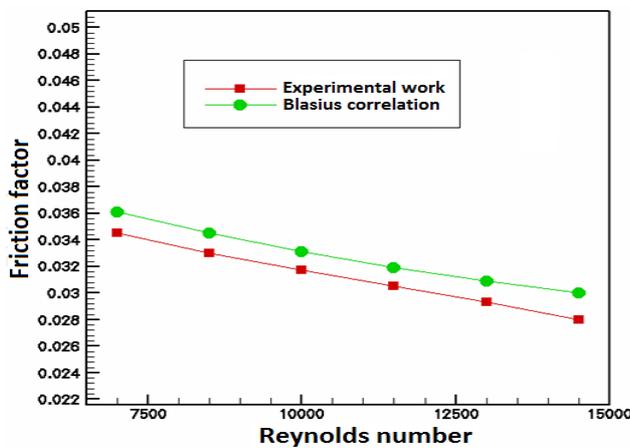
Fig. 5. Friction versus Reynolds number for plain tube
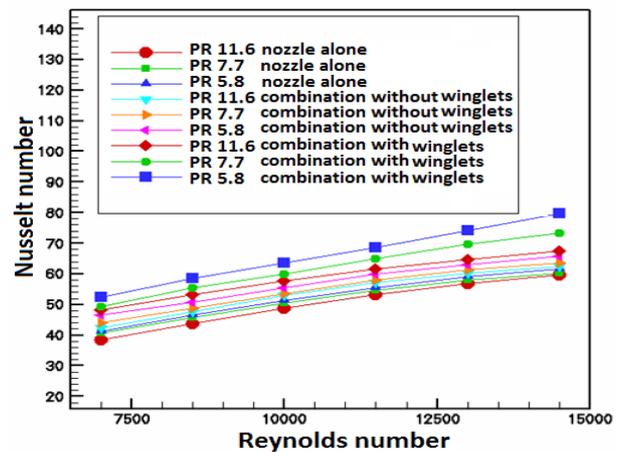
Fig. 7. Nusselt number versus Reynolds number for all cases

Fig (6) shows the variation of Nusselt number versus Reynolds number for square nozzle alone with three different pitch ratios inserted in tube. The results show that the reducing of pitch ratios yield a significant increase in heat transfer due to higher turbulence of the flow between the square nozzles. For PR = 5.8, the increase in Nusselt number was in the range of 25.18–35.4 % more than the plain tube for the range of Reynolds numbers from 7000 to 14500.

The variations of Nusselt number with Reynolds number for combination of square nozzles and helical tape inserted with and without winglets are depicted in Fig (7). The results show considerable increase of heat transfer for the combination without winglets compared to nozzles alone for all pitch ratio due to turbulence of the flow as well as thin boundary layer which induce by the shape of nozzle and helical tape. The tube fitted with the combination without winglets leading to higher heat transfer than the tube with nozzle alone by around 17.6%, 12.8% and 9.5% for PR = 5.8, 7.7 and 11.6, respectively. Also the combination with winglets was more effect on heat transfer than that from combination without winglets. This is because fluid flow through winglets leads to the strengthening of vortices, which makes the flow more turbulent and in turn leads to improved heat transfer. The average Nusselt numbers for using the combination with winglets are found to be 17.5%,

10.2% and 7.3% more than for the tube with the combination without winglets or to be about 22.8 %, 17.7 % and 11.6 % more than for the tube with square nozzles alone for PR = 5.8, 7.7 and 11.6 respectively.

The relations of friction factor with Reynolds number value for nozzle alone and all turbulators are depicted in fig. (8) and fig. (9). In the fig. (8) it is observed that the increasing in friction factor values at PR 5.8, 7.7 and 11.6 are about 4.1, 3.9 and 3.5 times above the plain tube respectively. This is because the turbulators increase the surface area and creating more obstruction of the flow.

From the fig. (9) it can be observed that the friction factor increases with decreasing pitch ratio. Reducing the pitch ratio means increasing the number of nozzle and this leads to increase the blockage against flow stream and thus increasing in the friction factor. The results also showed the combination with winglets gives increase in friction factor at about 23.5%, 16.2% and 10.3 % higher than the combination without winglets and at about 29.4 %, 21.9 % and 16.5 % higher than the nozzle alone for PR 5.8, 7.7 and 11.6 respectively.
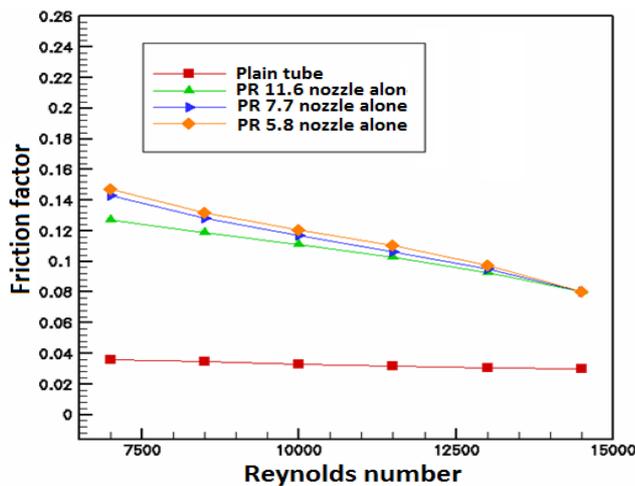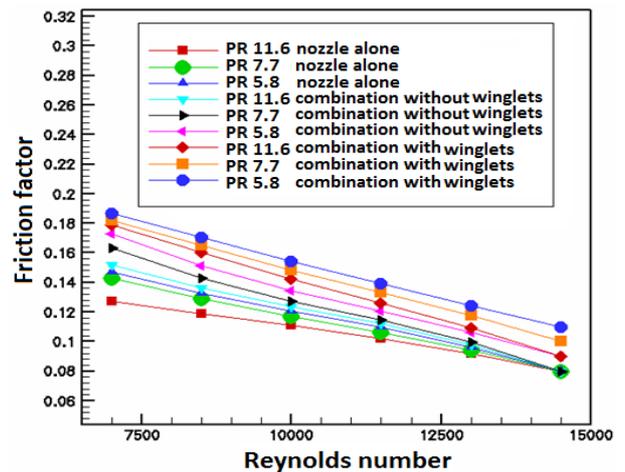
Fig. 9. Friction versus Reynolds number for all cases

The variation between the thermal performance factor and the Reynolds number for all turbulators used is shown in Fig.(10). It is observed that η shows a slight decrease with the increase of Reynolds number values but increase with reducing pitch ratio. From this figure, the thermal performance factor of combination is considerably higher than that of the nozzle alone. This indicates that the flow through combination leading to swirls intensity and make the mixing between both boundary and core regions more effective in turn leads to enhanced heat transfer. For the combination without winglets thermal performance factor are found to be 5.54 %, 4.1 % and 3.9 % higher than for the tube with nozzle alone for PR = 5.8, 7.7 and 11.6 respectively. The maximum η of about1.35 is achieved for combination with winglets for PR 5.8 and is higher than the combination without winglets around 10.5% while higher than the nozzle alone at around 15.5%.

Fig. 8. Friction versus Reynolds number for nozzel alone.
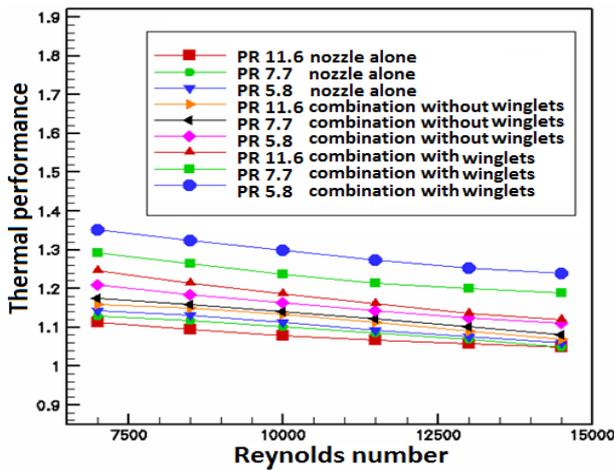
Fig. 10. thermal performance versus Reynolds number
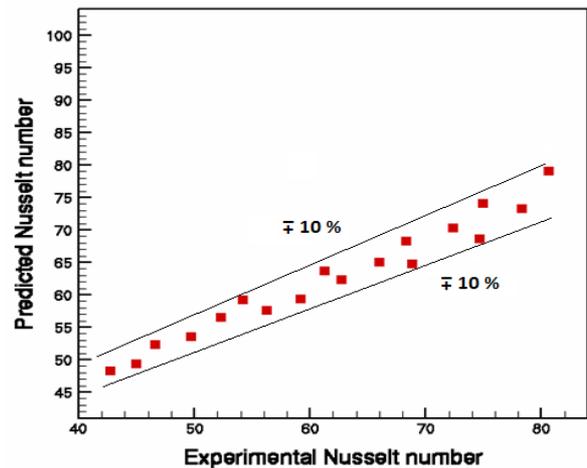
Fig. 12. predicted Nusselt number versus experimental results.

Empirical correlations

$$f = 27Re^{-0.56}PR^{-0.04} \qquad \dots(9)$$
$$Nu = 0.0802Re^{0.76}Pr^{0.4}PR^{-0.124} \qquad \dots(10)$$

where $7000 < Re < 14500$

and $PR = 5.8$ to 11.6

friction factor and Nusselt number correlations by equations (9), (10), were compared with experimental data as shown in figure (11) and (12). The results showed that the maximum deviation is 8% for experimental friction factor data and 10% for experimental Nusselt number data.
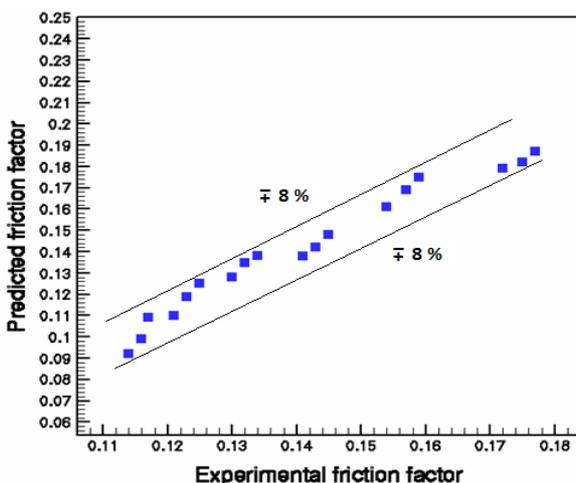
Fig. 11. predicted friction factor versus experimental results

4. Conclusion

The following can be concluded as:

1. The values of both Nusselt number and friction factor in tube with combination of nozzle and helical tape with and without winglets were higher than the values in the plain tube.
2. The value of the Nusselt number of the tube inserted in the nozzle alone for pitch ratio 5.8 is increased by approximately 5.6 % compare with its value for pitch ratio 11.6.
3. The Nusselt number for the tube in which the combination with winglets inserted for pitch ratio 5.88 is 17.5 % higher than the combination without winglets and 33.8% higher than the nozzle alone for the same pitch ratio.
4. The maximum thermal performance is about 1.351 for pitch ratio 5.8 in the case of the combination with winglets and this value is10.5 % higher than the combination without winglets and 15.5% higher than the case of the nozzle alone for the same pitch ratio.

Notation

A	Surface area, m^2
Cp	Specific heat j/kg K
D	Test tube inner diameter, m
h	coefficient of heat transfer w$/$(m^2 K)
I	current, A
L	Tube length, m
K	Thermal conductivity, (W/m K)
m	mass flow rate
Δp	pressure drop, Pa
PR	pitch ratio

Pr   Prandtl number Pr= (μ cp)/k
Q   heat transfer rate, W
Re   Reynolds number Re= (ρ v d)/μ
T   Temperature, K
u   working fluid velocity m/s
V   voltage, V

Greek Symbols

ρ   Density, (kg/m^3)
η   thermal performance

Subscripts

i   inlet
o   outlet
p   plain
t   Turbulators
w   wall
a   air

5. References

[1] S. Eiamsa-ard, K.Wongcharee, P.Eiamsa-ard,C.Thianpong, Heat transfer enhancement in a tube using delta-winglet twisted tape inserts, Applied Thermal Engineering 30 (2010) 310–318

[2] S.W. Chang, K.W. Yu, M.H. Lu, Heat transfer in tubes fitted with single, twin and triple twisted tapes, J. Exp. Heat Transfer 18 (2005) 279–294.

[3] Peng L, Nianben Z, Feng S, Zhichun L, Wei L, an experimental and numerical study on the laminar heat transfer and flow characteristics of a circular tube fitted with multiple conical strips, Int. J of Heat and Mass Transfer 117 (2018) 691–709.

[4] L.Wang, B. Sunden, Performance comparison of some tube inserts, Int. Commun. Heat Mass Transfer 29 (2002) 45–56.

[5] Khalid M. S, Ali I. S, Ahmed A. T, Mohamed M. A E, CFD modelling of entropy generation in turbulent pipe flow: Effects of temperature difference and swirl intensityApplied Thermal Engineering 100 (2016) 999–1006

[6] S.W. Chang, Y. Zheng, Enhanced heat transfer with swirl duct under rolling and pitching environment, J. Ship Res. 46 (2002) 149–166.

[7] Chou X. T, Wai L. M., Yew M. H.,Boon Th. T, The role of inserts in forced convection heat transfer augmentation, Int.Commun. Heat and Mass Transfer 39 (2012) 1138–1145.

[8] Eimsa-ard S, Thianpong C, Promvonge P. Experimental investigation of heat transfer and flow friction in a circular tube fitted with regularly spaced twisted tape elements. IntCommun Heat Mass Transfer 2006;33: 1225–33.

[9] M.M.K. Bhuiyaa, J.U. Ahamed, M.S.U. Chowdhury, M.A.R. Sarkar, B. Salam, R. Saidur, H.H. Masjuki, M.A. Kalam, Heat transfer enhancement and development of correlation for turbulentflow through a tube with triple helical tape inserts.Int.Commun. in Heat and Mass Transfer 39 (2012) 94–101

[10] Veysel O, Sibel G, Orhan B, Necdet A, Heat transfer enhancement in a tube using circular cross sectional rings separated from wall. Applied Energy 85 (2008) 988–1001.

[11] X. Zhang, Z. Liu, W. Liu, Numerical studies on heat transfer andflowcharac- teristics for laminar flow in a tube with multiple regularly spaced twistedtapes, Int. J. Therm. Sci. 58 (2012) 157-167

[12] Pongjet P, Somsak P, Monsak P, Chinaruk T, Heat transfer augmentation in a helical-ribbed tube with double twisted tape inserts Int.Commun.in Heat and Mass Transfer 39 (2012) 953–959.

[13] Sombat T, Yingyong K, Sompol S, Pongjet P, Heat transfer enhancement in tubular heat exchanger with double V-ribbed twisted-tapes.Thermal Engineering 7 (2016) 14–24

[14] F.P. Incropera, P.D. Witt, T.L. Bergman, A.S. Lavine, Fundamentals of Heat and Mass Transfer, John-Wiley & Sons, 2006.

تأثير استخدام المركب لفوهة مربعة ذو جنيح مع شريط حلزوني على الخصائص الحرارية في أنبوب مبادل حراري

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الخلاصة

تمت دراسة عملية تأثير اتحاد فوهة مربعة ذات جنيح مع شريط حلزوني مدرج في أنبوب ذي تدفق حراري ثابت على تحسين نقل الحرارة لتدفق هواء مضطرب ولعدد رينولدز يتراوح من ٧٠٠٠ إلى ١٤٥٠٠.حيث تم استخدام ثلاث نسب خطوة مختلفة للفوهة المربعة وهي (٥,٨، ٧,٧ و١١,٦) على وفق ثلاثة أعداد مختلفة من الفوهات المربعة وهي (٥، ٤ و٣) واستخدامت نسبة ثابتة واحدة للشريط الحلزونية. ولاحظت النتائج أن عدد ناسلت وعامل الاحتكاك للاتحاد مع الفوهة ذات الجنيح وجد أنه يصل إلى ٣٣,٨% و ٢١,٤% على التوالي أعلى من الفوهة وحدها لنسبة خطوة هي (٥,٨). كانت القيمة القصوى للأداء الحراري لاستخدام الاتحاد مع الفوهة ذات الجنيح حوالي ١,٣٥١ لنسبة خطوة هي (٥,٨). تم تطوير علاقات تجريبية لعدد ناسلت ومعامل الأحتكاك للنتائج التجريبية ولجريان مضطرب.

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