Al-Khwarizmi Engineering Journal,Vol. 13, No. 4, P.P. 41- 49 (2017)

Al-Khwarizmi Engineering Journal

Enhancement of Heat Exchanger Performance by Using Dimpled Tube

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(Received 9 February 2017; accepted 14 May 2017)

https://doi.org/10.22153/kej.2017.05.002

Abstract

The enhancement of heat exchanger performance was investigated using dimpled tubes tested at different Reynolds numbers, in the present work four types of dimpled tubes with a specified configuration manufactured, tested and then compared performance with the smooth tube and other passive techniques performance. Two dimpled arrangements along the tube were investigated, these are inline and staggered at constant pitch ratio X/d=4, the test results showed that Nusselts number (heat transfer) of the staggered array is higher than the inline array by 13%. The effect of different depths of the dimple (14.5 mm and 18.5 mm) has been also investigated; a tube with large dimple diameter enhanced the Nusselts number by about 25% for the range of Reynolds number between 4000- 20000. The overall enhancement ratio was used to differentiate the passive technique and comparison of different configurations for the technique itself, the result depicted that the 10 mm dimple diameter with a pitch ratio of X/d= 4 gives better performance than the static mixer and twisted tape techniques. The overall enhancement ratio of the 10 mm dimple diameter is varied from (1.21 to 1.65) that means good performance for a range of Reynolds numbers (4000-20000). This indicates that 21- 65 % of heat transfer area can be saved at the same pumping power compared with the smooth tube heat exchanger.

Keywords: *Dimple tube, heat transfer coefficient, Nusselts number, friction factor.*

1. Introduction

The techniques used in enhance the performance of heat exchangers can be classified into two main categories, passive and active techniques, in addition to a hybrid technique which includes two or more from each of passive and active technique [1]. A great deal of research has focused on various augmentation techniques with emphasis on rough surfaces, spiral ribs, transverse grooves, corrugated and spirally corrugated tubes, straight fins, and spiral and annular fins. In this investigation, augmented surface has been achieved with dimples strategically located in a pattern along the tube of a double-pipe heat exchanger with the increased area on the tube side. Augmented surfaces can create one or more combinations of the following conditions that are favourable for increasing the heat transfer coefficient with a consequent increase in the friction factor due to firstly the interruption of the development of the boundary layer and increase of the degree of turbulence, secondly the effective heat transfer area increased, and finally the generation of rotating and/or secondary flows. Kumar & Murugesan [2] considered and evaluated the heat transfer and pressure drop investigations of more than thirty published work from (2001 to 2012) of various twisted tapes placed in heat exchangers. The results showed that for modified twisted tape geometry, the heat transfer rate is higher with reasonable friction factor for both laminar and turbulent flow. M. Udaya Kumar et al. [3] reviewed the heat transfer enhancement techniques in square ducts of more than ten published works from years (2004 to 2013). They found that the heat transfer of square ducts was found considerably higher than the circular tube. This is mainly because the square duct has a high surface to volume ratio. The short length twisted tape in square and rectangular ducts performs worse than the full length twisted tape. However, regularly spaced twisted tapes perform significantly better than the full length twisted tapes. Watcharin et al.[4] studied experimentally the influences of the twisted tape insertion on the heat transfer and flow friction characteristics in a concentric double pipe heat exchanger. P. Murugesan et al. [5] investigated experimentally the heat transfer and friction factor characteristics of a circular tube fitted with plain twisted tapes and U–cut twisted tapes. Chinaruk Thianpong et al [6] performed an experimental study of fully developed turbulent flow in a dimpled tube in conjunction with a twisted tape. Khalil et. al. [7] investigated experimentally the heat transfer, pressure drop characteristics and efficiency enhancement for swirling flow through a sudden pipe expansion. P. Promvonge [8] conducted a set of experiments by inserting several conical rings as tabulators over a test tube. All the available enhancement techniques shown by the literature were mainly effective on heat transfer at the tube side. With some applications, the tube shape had a dual effect on both sides with a great penalty of the pressure drop on both sides. On the other hand, most of the enhancement devices depicted in the literature required addition of materials that impact on manufacturing cost while in the present proposed tube, no additional materials will be used. There is a shortage of the available data in the literature for the effect for dimpled tube geometrical parameters on the heat transfer and pressure drop in turbulent flow. The present work will cover the main geometrical parameters, such as dimple arrangement (staggered, inline), dimple diameter, and numbers of dimples. The relationship between the thermal and hydraulic performance must also be considered, by means of the overall enhancement ratio concept.

1. Experimental Equipment and Data Reduction

The schematic diagram and photo of test rig used in this study is shown in figure (1), the test rig consists of an arrangement of force convection open loop flow system. This system contains air blower, piping, instrumentation and test section. The blower is connected to a carbon steel pipe of 50 mm diameter and 2 m length. Orifice plate intermediates this pipe is to measure the air flow rate through the system. Two pressure taps were placed upstream and downstream the orifice flange to measure the pressure difference, using water manometer. Temperature tap located upstream of orifice plate was used for measuring the air blowing temperature. To achieve a fully developed flow at the test section entrance, a tube with 1m length having the same test tube diameter been connected with the test tube by a special coupling. The test section is one of the main components in an experimental test facility which consisted of a tube in tube heat exchanger in cross flow configuration. The test section contains a tube of 35 mm diameter surrounded by a jacket pipe of 75 mm inner diameter with 1m long, and is insulated by 5 cm rock wool. In the present work, a constant wall temperature technique was used by applying saturated steam on the outer wall of test tube. The condensation heat transfer coefficients are approximately 100 times greater than the air convective heat transfer inside the tube, therefore the wall temperature will be more likely equal to the condensation temperature. This means that no set of thermocouples is required to be placed on the tube wall. The steam is supplied from a boiler available at the laboratory, and pressure regulator valve was used to obtain constant saturated steam pressure and temperature at annulus of test tube. The jacket is instrumented with pressure, temperature gauge, drain and vent valve to get rid of the non-condensable gases and to drain the condensate steam during the test period. Temperature and pressure taps are placed at the inlet and the exit of the test tube to measure the temperature and pressure of air across the tested section. The test section is a shell and tube configuration, where steam flows in shell side, while air flows inside the tube. The shell side is made from carbon steel tube of 75 mm inner diameter, outer diameter 82 mm and 1200 mm length. The shell ends are welded with the special flange that machined to produce sealing housing, where sealing cap can be pushed through by screw bolts. The flanges have a central bore hole with a diameter equals to the test tube diameter, where the test tube can be slide through aligning the tube to be in the shell center. Teflon rope of 10 mm is packed in the flange housing and warped round the test tube, while sealing cup is forced by the four bolts to compress the teflon rope on the test tube and on the inner ring of the flange housing to produce tight sealing and to maintain the saturation steam pressure and temperature. The shell is insulated by fiber glass of 76 mm thickness. Five holes of (13 mm) are drilled on the shell surface to connect the Following apparatus:-

- Steam valve of ½ inch.
- Pressure gage: rang of 0-10 bar with increments of 0.2 bar.
- Condensate drain valve size ½ inch.
- Vent valve size ½ inch.
- Thermocouple well.

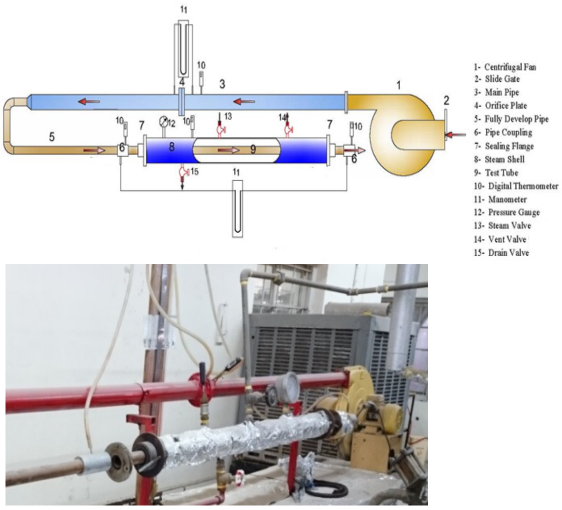

Fig. 1. Schematic diagram and photo of the test rig.

2.1. Dimpled Tube

All types of dimpled tubes manufactured from copper of (1600 mm) length and inner diameter of 35 mm, shown in figure (2), were examined in the present work, also a smooth tube of (35 mm) inner diameter and (1600 mm) was used as a reference tube. The geometrical parameters, dimpled arrangement and dimensions of all dimple tubes are specified in Table (1) and figure (2). Manufacturing process was developed to obtain these tubes using a CNC machine to obtain dimples with a specified configuration.

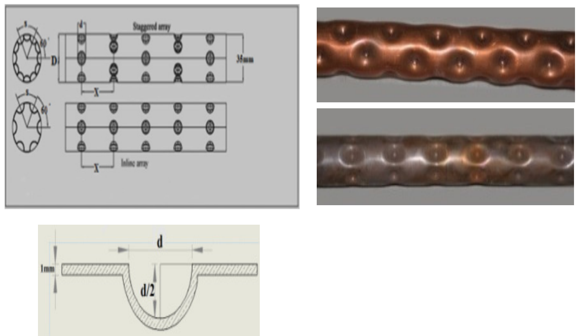

Fig. 2. Geometrical figure and photo of dimple tubes.

Table 1,
Test tubes configurations.

Dimpled Arrangement	Distribution Angle	d mm	X mm	d/D	X/d	No. of dimples
Inline pipe	60	10	40	0.285	4	174
Staggered pipe	60	10	40	0.285	4	162
Staggered pipe	120	14.5	58	0.414	4	63
Staggered pipe	120	18.5	74	0.529	4	48

2.2. Data Acquisition

The main object of the present experimental work is to evaluate the convective heat transfer coefficient then Nasselts number and pressure drop across the tested tubes. According to the first law of thermodynamic for incompressible flow with constant specific heat, the heat transfer to the air can be calculated:

$$Q_a = \dot{m}_a \, Cp_a \, (T_{a\,out} - T_{a\,in}) \quad \ldots (1)$$

The volume and the mass flow rate through the orifice plat are evaluated by measuring the pressure drop across the orifice, the upstream temperature and using the following equations [9]:

$$\dot{V} = C_D \, E \, A_o \sqrt{\frac{2\,\Delta P}{\rho_a}} \quad \ldots (2)$$

Where, C_D is the discharge coefficient, A_o is the cross-sectional area of the orifice plat, ΔP is the pressure drop across the orifice plat and E is the velocity approach coefficient.[9,10]

$$C_D = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^8 + 0.0029\beta^{2.5}\left[\frac{10^6}{Re_D}\right]^{0.75} + 0.09l_1\beta^4(1-\beta^4)^{-1} - 0.0337l_2\beta^3 \quad \ldots (3)$$

$$E = (1-\beta^4)^{-1/2} \quad \ldots (4)$$

Where, β is the orifice to tube diameter (d/D),and l_1 & l_2 are constant [10].
The mass flow rate is calculated from the following equation:

$$\dot{m}_a = \dot{V}\,\rho_a \qquad \dots (5)$$

In the heat transfer and fluid flow, it is essential to know the flow regime whether it is laminar, or turbulent ,which can be represented by the dimensionless parameter known as Reynolds Number used as criterion, and Re should be greater than 4000 for turbulent flow.

$$Re_d = \frac{\rho\, U\, D}{\mu} \qquad \dots (6)$$

The air density ρ and the dynamic viscosity μ are evaluated based on the air bulk temperature. The air velocity through the test tube can be calculated from continuity equation:

$$U = \frac{\dot{m}_a}{\frac{\pi}{4} D^2\, \rho} \qquad \dots (7)$$

To express the convective heat transfer from the tube wall to the air, the newton's law for cooling is applied. [11]

$$Q_w = h_a\, A_w\, (T_w - T_b) \qquad \dots (8)$$

The system is thermally balanced $(Q_w = Q_a)$, i.e, the heat gained by the air is the heat released by the tube wall. By combining equations (1) and (8), the convective heat transfer coefficient can be evaluated.

$$h_a = \frac{\dot{m}_a\, Cp_a\, (T_{out} - T_{in})}{A_w\, (T_{wi} - T_b)} \qquad \dots (9)$$

Where, T_b is the bulk air temperature and can be evaluated as follow:

$$T_b = \frac{T_{in} + T_{out}}{2} \qquad \dots (10)$$

The inner wall temperature $T_{wi,}$ can be evaluated from the measured heat flow through the test tube and the saturation steam temperature using the following equation: [12]

$$Q = \frac{2\,\pi\, L\, (T_s - T_{wi})}{\frac{\ln(^{ro}/_{ri})}{k} + \frac{1}{ro\, h_c}} \qquad \dots (11)$$

Where h_c , is the condensation heat transfer coefficient on the horizontal tube and can be calculated from the following equation:[12]

$$h_c = 0.725 \left[\frac{\rho_l(\rho_l - \rho_v)\, g\, h_{fg}\, k^3}{\mu_l\, D_o (T_s - T_{wo})}\right]^{0.25} \qquad \dots (12)$$

Heat transfer represented by dimensionless parameter Nussalts number is calculated by the following equation:[12]

$$Nu = \frac{h_a\, D}{k_a} \qquad \dots (13)$$

To validate the present experimental test apparatus, the test was performed on a smooth tube and then results compared with well-known equations of Dittus-Boelter, and Blasins equation for heat transfer and friction factor inside a tube for a turbulent flow at constant wall temperature [13].

$$Nu = 0.023\, R_e^{\,0.8}\, P_r^{\,0.3} \qquad \dots (14)$$

$$Nu = 0.012(Re^{0.87} - 280)Pr^{0.4} \qquad \dots (15)$$

$$f = \frac{0.316}{Re^{0.25}} \qquad \dots (16)$$

The friction factor (f) can be calculated by measuring the pressure upstream and downstream of the tested tube and according to the equation:[12]

$$f = \frac{2\, \Delta P\, D}{L\, \rho\, U^2} \qquad \dots (17)$$

Δp represents the pressure drop across the test section and is evaluated from the following equation:[9]

$$\Delta P = \rho\; g\, h \qquad \dots (18)$$

The data obtained from the smooth tube were compared with Blasins equation for smooth pipe, and the results are presented. The overall enhancement ratio is defined as the ratio of heat transfer enhancement ratio to the friction factor ratio; this parameter was derived according to using same pumping power for different configurations. [14].

$$\boldsymbol{\eta = \left(\frac{Nue}{Nup}\right) \Big/ \left(\frac{fe}{fp}\right)^{0.3} \geq 1} \qquad \dots (\mathbf{19})$$

2. Results and Discussion

The experimental investigation imposes a study of dimpled arrangement (staggered, inline), dimple diameter, and number of dimples. The relationship between the thermal and hydraulic performance must also be considered. Heat transfer and pressure drop for all test tubes were evaluated and presented as a dimensionless value by Nusselts number and friction factor in the range of Reynolds number from 4000 to 20000. The overall enhancement ratio (η) of dimpled tube compared with others passive techniques were discussed.

3.1. Verification of Smooth Tube Results

In order to verify the experimental results obtained from the present test facility for heat transfer and pressure drop, experimental tests were performed on a smooth tube of 1.6 m length (1.2 m in test section). The experimental results were compared with the results obtained by the well-known correlations under a similar condition to evaluate the validity of the smooth tube. Comparison of Nusselt number is shown in figure (3). Obviously, the experimental results of heat transfer are in good agreement with empirical correlations developed by Dittus–Boelter [13]. It is noted that the average deviation in Nusselts number was approximately of 5%, and the maximum deviation of 13%.

Comparison of the friction factor (f) of present plain tube and the results obtained by Blasins correlation [13] is shown in figure (4), this figure shows the apparent deviation is in the range of Reynolds number. This was attributed to the pipe surface roughness. In the present work, the experimental data obtained from the smooth tube will be used as a reference for comparison with a dimpled tube, since all tested tubes are made from the same materials (copper) and same manufacturing process.

The following subsections will discuss the effect of the geometrical parameters on Nusselts number and friction factor. The comparison criteria of overall enhancement ratio for different types of dimpled tube will be discussed:

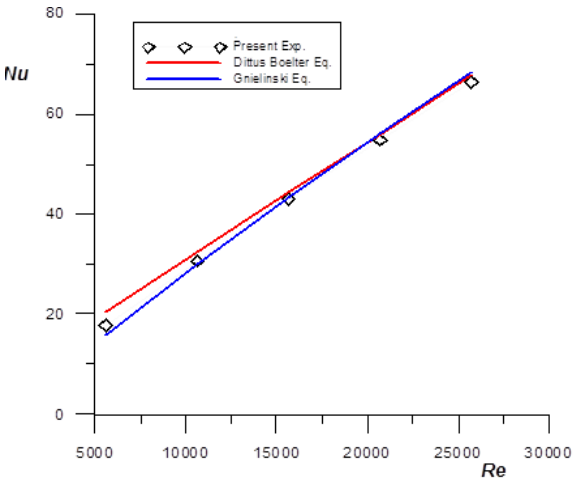

Fig. 3. Comparison of heat transfer of present experimental test of smooth tube with published empirical equations [13].

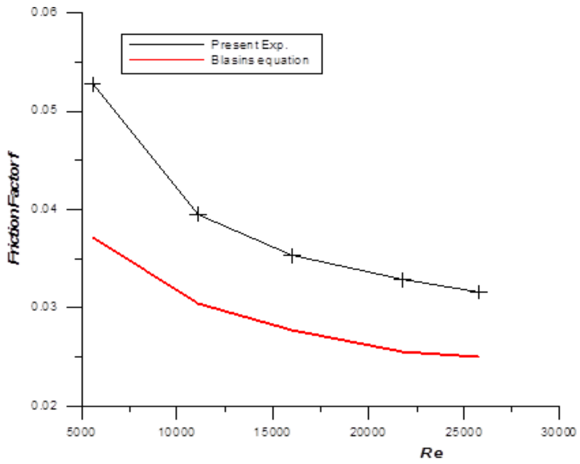

Fig. 4. Comparison of friction factor of the present test smooth tube with published Blasins equation [13].

3.2. Effect of Dimples Arrangement

Two dimpled arrangements along the tube were investigated, these are inline and staggered at constant pitch ratio X/d=4, all with approximately the same number of dimples, as shown in Table (1). The test results presented in figure (5) show that the heat transfer of the staggered array is higher than the inline array by 13%. This is due to more turbulence intensity and strong vortex formation induced in inline arrangement tube. It is clear that the Nusselts number increases with increasing Reynolds number as for the conventional turbulent flow in tube of approximately the same trend.

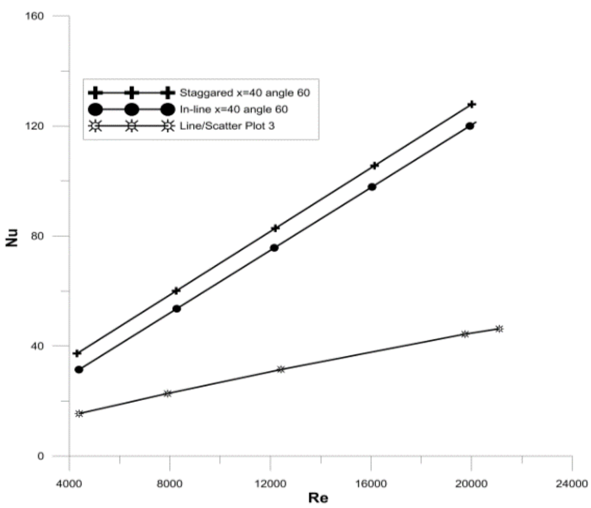

Fig. 5. Effect of dimple arrangement on heat transfer.

3.3. Effect of Dimples Diameter

From the test results presented in Figure (6) the effect of different depths of the dimple (14.5 mm and 18.5 mm), tube with 18.5 mm dimple diameter enhanced the Nusselts number by 25% compared with the tube of 14.5 mm dimple diameter for the range of Reynolds number between 4000-20000,this due to first, large dimple diameter makes the kinetic energy of the core flow dominant and much higher than the shear force induced by the dimpled surface. Second, the dimple depths is small, and the secondary flow is still within the limited space of the wall surface Therefore, in the present case, different depths of the dimple (14.5 mm and 18.5 mm) caused increasing in mixing between the secondary and core flow, the local heat transfer on tube wall will be improved.

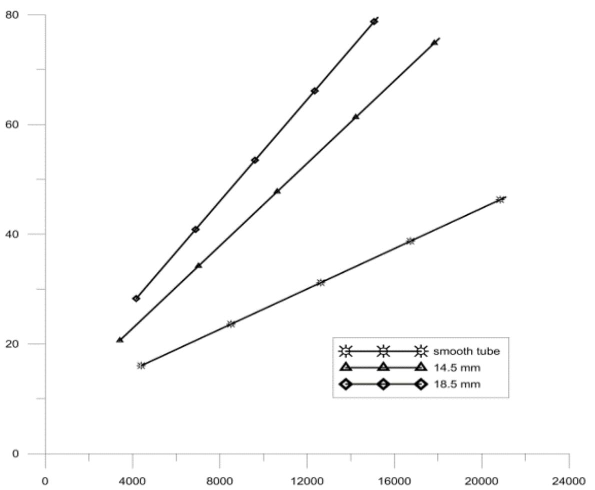

Fig. 6. Effect of dimple diameter on heat transfer.

2.4. Comparison with Other Augmented Surfaces

The overall enhancement ratio is used to differentiate the passive technique and comparison of different configurations for the technique itself. The overall enhancement ratio given in equation (19) is relation was based on the heat transfer and pressure drop of smooth tube; this parameter was derived according to using same pumping power for different configurations.[14] According to this relation, the overall enhancement ratio should be greater than unit. The greater value indicates better performance for that geometry, while values less than one indicate bad or worthless approach, also this parameter overcomes the friction factor variation. Figure (7) shows the comparison of the overall enhancement ratio for a present dimpled tube shape with staggered arrangements. The result depicted that the 10 mm dimple diameter with a pitch ratio of X/d= 4 gives better performance than the other techniques. The enhancement ratio of the 10 mm dimple diameter is varied from (1.21to1.65) for a range of Reynolds numbers (4000-20000). This indicates that 21-65 % of heat transfer area can be saved at the same pumping power compared with the smooth tube heat exchanger. The results showed that the 10 mm dimple diameter tube with a pitch ratio of X/d= 4 gives better performance than other passive techniques such as twisted tap and static mixer.

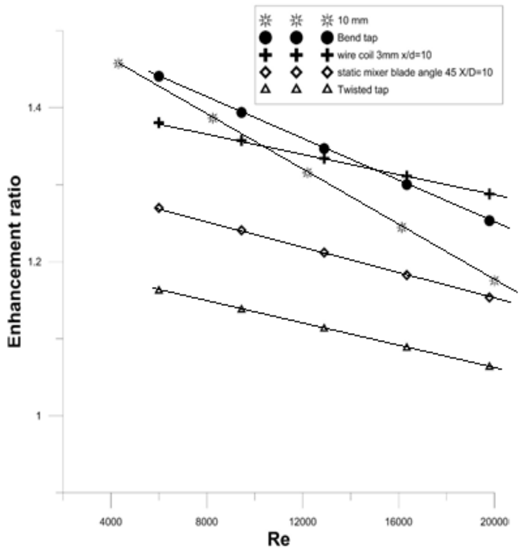

Fig. 7. Comparison of the overall enhancement ratio of the present dimple tube with other passive technique

3. Conclusions

An experimental study of fully developed turbulent flow in a spherical dimpled tube has been done. The influences of the dimpled arrangements and dimple diameter on the heat transfer rate and friction factor characteristics have also been investigated, and the following points can by highlighted:-

a. Staggered arrangement of dimpled tube manifested better enhancement of heat transfer by 13% than Inline arrangement at same dimples pitch ratio. At the same time, higher friction factors were recorded for staggered arrangement.

b. The large dimple diameter enhanced the Nusselts number by 25% compared with the small dimple diameter.
c. The heat-transfer coefficients of the enhanced dimpled tubes are a lot higher than the conventional smooth tubes and give only marginal drops in pressure. An important reason for the heat-transfer enhancement of the enhanced tubes is the modified rough surface, which causes increases in the turbulence mixing intensity in the flow field, and also by restricting the development of fluid boundary layers close to the heat transfer surfaces, which contributes to increases in heat-transfer operation.
d. The overall enhancement ratio of the 10 mm dimple diameter is varied from (1.21to1.65) for a range of Reynolds numbers (4000-20000) compared with the static mixer and twisted tape.

Notation

Latin Characters

Character	Description	Units
A	Tube surface area	m^2
C_D	Discharge coefficient	
C_p	Specific heat at constant pressure	kJ/kg K
D	Tube diameter	m
d	Dimple diameter	m
f	Friction factor	
H	Head	m
h	Convection heat transfer coefficient	W/m^2 K
L	Tube length	m
$\dot{m}$	mass flow rate	kg/s
Nu	Nusselt number	
Re	Reynolds number	
T	Temperature	K
U	Velocity	m/s
P	Pressure	N/m^2

Greek Symbols

Character	Description	Units
ρ	Fluid density	kg/m^3
μ	Dynamic viscosity	kg/m.s
β	Orifice to tube diameter ratio	

Subscripts

Symbol	Title
a	Air
b	bulk
s	steam

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تحسين اداء المبادل الحراري باستخدام الانابيب المندبة

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الخلاصة

تم في هذا البحث دراسة تحسين اداء المبادلات الحرارية باستخدام الانابيب المندبة وتم لهذا الغرض تصنيع اربع نماذج مختلفة الاشكال واختبارها ومقارنة ادائها الحراري مع الانبوب الاملس. تمت دراسة تاثير ترتيب الندب على طول الانبوب (متناظر , متخالف) عند (X/d=4) بينت النتائج العملية ان معدل انتقال الحرارة (عدد نصلت) للانابيب متخالفة الندب اكبر ب ١٣% من الانابيب متناظرة الندب. كذلك تم دراسة تاثير قطر الندبة (14.5mm,18.5mm) واظهرت النتائج تحسن عدد نصلت وبالتالي انتقال الحرارة بنسبة ٢٥% عند استخدام انبوب بقطر كبير لمدى عدد رينولدز يتراوح بين ٤٠٠٠-٢٠٠٠٠ . استخدمت نسبة التحسين الكلية للمقارنة بين التقنية الحالية والتقنيات السلبية المستخدمة في تحسين اداء المبادلات الحرارية وكذلك للمقارنة بين نماذج البحث الحالية اظهرت النتائج ان الانبوب ذا الندب بقطر 10mm وخطوة X/d=4 تعطي اداء افضل من الاساليب السلبية المختلفة لتحسين الاداء وكانت النتائج تشير الى ان الانابيب المزودة بالندب المتخالفة تعطي اداء افضل من اسلوب استخدام الخلّاط وكذلك من اسلوب استخدام الاجزاء الملتوية . تتراوح نسبة التحسين الكلية بين ١,٢١الى ١,٦٥ وهذا يعني اداء حرارياً جيداً عند مديا عدد رينولدز ٤٠٠٠-٢٠٠٠٠ اي تخفيضاً في مساحة انتقال الحرارة بنسبة ٢١-٦٥% لقدرة الضخ نفسها مقارنة مع استخدام الانبوب الاملس.