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Experimental Investigation on the Effect a Rotational Shaft on the Thermal Behavior of a Circular Tube under Constant Heat Flux

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ABSTRACT

Active and passive methods are two main mechanisms of heat transfer improvement. The active methods use external forces to improve heat transfer. This investigation evaluates the thermal and frictional behavior of a circular tube containing a rotational shaft. Constant heat flux was exerted to the circular tube. The fluid inlet and outlet temperature as well as wall temperature of tubes were measured to calculate the hat transfer coefficient. The Re (Reynolds) number was between 800-2000. Also, the dimensionless rotational speed (Rs) had the values of 1,1.5, 2, 2.5 and 3. Results revealed that the rotational shaft could increase the Nu number. Up to %18. Also, the results showed that the rotational shaft could significantly increase the pressure drop and friction factor. The maximum increment of %78 was achieved for friction factor. It was revealed that the use of rotational shaft could be more efficient at low Re numbers and low dimensionless rotational speeds. Also, it was found that by the increment of Reynolds number and being in the transient regime the efficiency of the system would improve.

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NOMENCLATURE

L	Length tube (m)	$\bar{h}$	Average convective heat transfer coefficient (W/m^2-K)
P_{out}	Outlet pressure (Pa)	A	Heat transfer area (m^2)
P_{in}	Inlet pressure (Pa)	Nu	Nusselt number
ΔP	Pressure drop (Pa)	Nu_s	Nusselt number of smooth tube
D_H	Hydraulic diameter (m)	f	friction factor
Re	Reynolds number	f_s	friction factor of smooth tube
q	Heat transfer rate(W)	TEF	Thermal efficiency factor
$\dot{m}$	Mass flow rate (kg/s)	V	Mean velocity (m/s)
T_{out}	Outlet temperature (K)	k_f	Conductive heat transfer coefficient (W/m^2-K)
T_{in}	Inlet temperature (K)	C.B.R	Cost per benefit ratio
T_w	Wall temperature (K)	**Greek symbols**	
T_b	Mean bulk temperature (K)	ρ	Density (kg/m^3)
		μ	Dynamic viscosity (kg/m-s)

1. INTRODUCTION

In accordance to the importance of heat transfer augmentation at the efficiency improvement of various thermal units, numerous researchers have implemented investigations on the heat transfer improvement techniques. The thermal enhancements mechanisms include two main categories namely as active and passive methods [1-3]. The passive methods are those at which no external force is used in the heat transfer augmentation techniques. However, at the active methods, the heat transfer enhancement is occurred by means of external forces exerted on the system [4, 5]. At the present investigation the influence of a rotational shaft on the

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thermal and frictional behavior of a circular tube is experimentally investigated. Recently the use of a rotating elements in order to increase the heat transfer is significantly increased. This type heat transfer is commonly called as internal cooling technology and especially is used within the processes having a rotating shaft as like to internal cooling of turbines. At the following, some of the investigations which have used the rotational effect to increase the heat transfer are summarized.

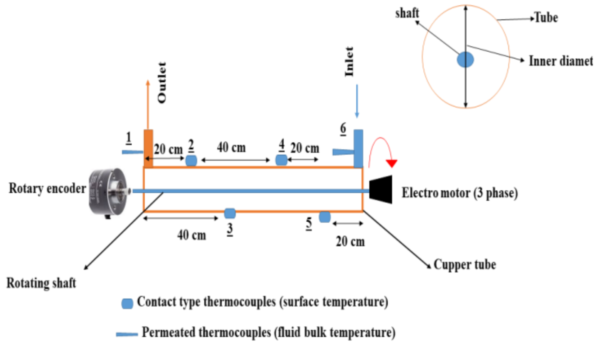
First time, White [6] reported that a rotational circular tube has better performance when compared to the stationary one. The results presented by White [6] revealed that at the higher rotational speeds, the pressure drop could be decreased up to 40%. Singaram et al. [6] and Chatterjee et al. [7] studied a half filled rotational circular tube. They proposed an empirical correlation for predicting the thickness of the film layer. Krivosheev et al. [8] probed the thermal performance of a half-filled rotational circular tube. The results revealed that as the rotational speed augments, the heat transfer rate raises. Chatterjeea et al. [9] investigated the thermal performance of a half-filled rotational circular tube. They provided an empirical correlation for the prediction of Nusselt number (Nu) inside a half-filled circular tube. Hussain and Hussein [10] implemented a numerical investigation to probe the mixed convective heat transfer within a heated square shaped enclosure at which a rotating circular shaped cylinder was running. In their study, the position of the rotating circular cylinder was the variant parameter. Park et al. [11] investigated the natural convective heat transfer within an enclosure including hot and cold rotating circular cylinder. The locating point of the hot and cold cylinder was the variant parameter which was probed by Park et al. [11]. Their results presented that as the rotating cylinder and walls of the enclosure gets closer, the heat transfer rate increases. Yoon et al. [12] numerically probed the influence of two rotating cylinder on the thermal management of a cubic. Kareem et al. [13] implemented a 3D simulation in order to study the mixed convection heat transfer within an enclosure. Selimefendigil and Öztop [14] conducted a comprehensive research to examine the effect of pulsating flow and the rotation of the cylinder on heat transfer of a heated channel. Their results revealed that the direction of the rotation and the Reynolds number (Re) could be effective on the heat transfer rate. Most of the experts studied the effect of rotational cylinder on the heat transfer, supposing that the cylinder has the circular surface. However, the triangular [15] elliptical [16] and rectangular [17] surfaces were of interest too.

Based on the above literature review and up to the author's best of knowledge, the majority of investigations have focused on the rotation of tube containing the working fluid. There are few studies that investigated the effect of a rotational element inserted in the tube. At the present paper the influence of the presence of a rotational circular shaft on the heat transfer of a circular tube under constant heat flux is studied. The Nu number, pressure drop, cost per benefit ratio (C.B.R.) and performance efficiency coefficient (PEC) factors are the parameters that are studied and investigated in this study.

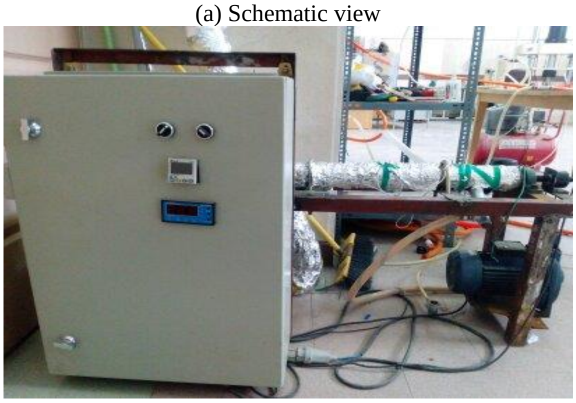
2. Experimental study

2. 1. Test Rig Definition

The schematic scene of the setup is shown in Figure 1. As is presented, the test rig is consisted of a circular tube which contains a rotating shaft. The geometrical properties of the circular tube and shaft are presented in Table 1. The circular cupper made tube was put under certain value of heat flux. The constant heat flux was produced via a heater



(a) Schematic view



(b) General view

Figure 1. (a) Schematic view (b) general view of the setup

TABLE 1. Geometrical properties of the circular tube and the circular rotating shaft

Element	Inner diameter (cm)	Width (mm)	Length (m)	Thickness (mm)
Circular tube	6.6	2	1	1
Circular shaft	1.5	-	1	-

wire. With the aim of producing fixed heat flux the amount of voltage exerted to the heater wire (100 Ω resistant) was regulated via a voltage regulator (2 kW Dimmer (DS 2000VA)). A 3 cm glass wool isolator was used to reduce the heat loss. The rotary shaft was rotated by means of a 3 phase electro motor. A reductant gear box (SAHAND-W090-15:1-90B5) was also installed between the electro motor and the coupling section to reduce the rotary speed of the electro motor and to produce the wanted speeds. Additionally, the electro motor was controlled by means of an inverter (DEG Drive- DGI 900). By means of the mentioned mechanisms the accurate rotational speed was adjusted. The amount of rotational speed was measured by means of a rotary encoder (HE50B-8-1024-3-T-24) which was coupled to the end of rotational shaft. The flow outlet and inlet temperature as well as surface temperature of the circular tube was recorded to evaluate heat transfer coefficient. Six K type thermocouples (as presented in Figure 1A) with a 12 channel data logger (Lurton BTM-4208SD) were utilized for recording the fluid and surface temperature. The pressure drop (pressure drop between the points 1and 6) was evaluated by means of manometer (Lurton PM-9100). The accuracy of the data logger and manometers were ± 0.5 °C and 1 $mbar$, respectively.

2. 2. Experimental Procedure Table 2 provides different cases which are considered at the present study. As shown, four various flow rates of water flow were considered to examine the effect of Re number. At each constant water flow rate different rotational speeds were assumed too. The Rs = 0 is referred to the cases in which the circular shaft was stationary. The thermal energy produced by heater wire was kept constant during the tests. It should be noted that the initial data was noted after the system reached thermally stable condition. As is aware the repeatability and the uncertainty evaluation is urgent for any investigation. The tests were repeated for three times and then the average of these tests were used for the calculation of the investigated parameters. The uncertainty analysis was based on Moffat [18] method which was used based on reported literature [19, 20] to evaluate the uncertainty value. Table 3 presents the achieved values for uncertainty of studied parameters.

2. 3. Setup validation Furthermore to the uncertainty calculation which resulted in reasonable values, the results of the present study are compared to the existing data reported in the literature. The friction factor results were compared with the results of well-known Blasius equation. Also, the Nu number results were compared to those of the study conducted by Sun and Zeng [21]. As is presented in Figure 2, there are good agreement between the results of present study and those reported in literature. The maximum deviation was about 8% which was reasonable.

TABLE 2. various considered cases in this study

Water flow rates (l/min)	*Re*	*Rs*
2	788	0, 1,1.5, 2, 2.5, 3
3	1215	0, 1,1.5, 2, 2.5, 3
4	1519	0, 1,1.5, 2, 2.5, 3
5	1863	0, 1, 1.5, 2, 2.5, 3

TABLE 3. Uncertainty in investigated parameters

Parameter	Unit	Amount
Electrical current	mA	± 2
Electric resistence	**Ω**	± 3
Electric voltage	V	± 5
Water flow rate	lit/min	± 0.1
Water bulk temperature	℃	± 0.5
Surface temperature	℃	± 0.5
Nusselt number	%	± 9.46
Uncertainty in read values of tables ($\rho, c, \dots$	%	$\pm 0.1 - 0.15$

3. PARAMETER DEFINITION AND CALCULATION METHOD

3. 1. Calculation Process In this part the calculation process of various parameters evaluated in this study are presented. As stated before, a digital manometer was used for measuring the pressure drop. At the following equation used for calculating pressure drop is presented.

$$\Delta P = P_{in} - P_{out} \tag{1}$$

3. 1. 1. Calculation Method of the Overall Heat Transfer Coefficient The convective heat transfer coefficient was evaluating by following method:

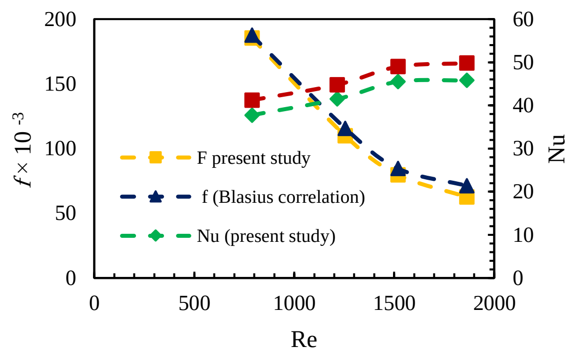


Figure 2. Comparison of the present results with existing results in the literature

As is aware, the thermal energy received by the working fluid (water flow) through the test rig is measured as follows:

$$\dot{q} = \dot{m}\, c_p (T_{out} - T_{in}) \tag{2}$$

In which the T_{in} and T_{out} are the inlet temperature and outlet temperature of the water flow, respectively.

The average heat transfer coefficient of the test rig could be calculated via the following equation.

$$\bar{h} = \frac{\dot{q}}{A\left(\frac{\sum_{j=1}^{4} T_w}{4} - T_b\right)} \tag{3}$$

where the T_b is calculated as the average of T_{out} and T_{in}.

3. 2. Parameter Definition

3. 2. 1. Friction Factor The friction factor is the dimensionless form of the pressure drop and was calculated by the following equation.

$$f = \frac{\Delta P}{0.5\left(\frac{L}{D_h}\right)\rho V^2} \tag{4}$$

3. 2. 2. Nu Number The Nu number denotes the potential of the heat transfer and is the dimensionless form of the heat transfer coefficient. The Nusselt number could be calculated as follows.

$$Nu = \frac{\bar{h} \times D_h}{k_f} \tag{5}$$

In which the D_h and the k_f are the hydraulic diameter and the conductive heat transfer coefficient of the fluid flow.

3. 2. 3. Cost Per Benefit Ratio Factor (C.B.R) The Cost per benefit ratio (C.B.R) factor is defined as the increment percentage of the pressure drop per the increment percentage of the Nusselt (Nu) number. This Parameter is a criterion which could be feasibly used with engineers if they want to know that how much heat transfer improvement could provide any technique that is used.

$$C.B.R. = \frac{\%\Delta P}{\%Nu} \tag{6}$$

3. 2. 4. Performance Efficiency Coefficient (PEC) Analysis The performance efficiency coefficient (PEC) is another criterion for evaluating the heat transfer enhancement techniques. This criterion was used widely by other experts [21, 22] to analyze the efficiency of any heat transfer augmentation method. This parameter helps engineers to choose the proper method for heat transfer augmentation. Since this parameter considers both the effect of friction factor and Nu number, it could be used for utilizing the heat transfer enhancement techniques. Performance efficiency coefficient (PEC) is defined as the ratio of Nu number of the improved tube to that of the smooth tube at the constant pumping power and is defined as below:

$$TEF = \frac{\left(\frac{Nu}{Nu_s}\right)}{\left(\frac{f}{f_s}\right)^{\frac{1}{3}}} \tag{7}$$

In which the Nu_s and f_s respectively, are the Nu number and friction factor of base tube.

3. 2. 5. Dimensionless Rotational Speed The dimensionless rotational speed is calculated as follows:

$$R_S = \frac{\omega D_H}{2V} \tag{8}$$

4. RESULTS AND DISCUSSION

4. 1. Pressure Drop and Friction Factor Analysis

Figure 3 presents the behavior of pressure drop versus dimensionless rotational speed (Rs). As stated before, the pressure drop was recorded via a digital manometer. By looking at Figure 3, it could be concluded that, as the Rs and Re number increase, the pressure drop increases too. It could be explained that as the Rs and Re number increase the flow regime tends to transpass from laminar regime to the turbulent regime. Actually, the increment of Rs causes to generation of swirling flows which are perpendicular to the axial flow. The interactions between axial flow and the swirling flows created by rotational shaft lead in generation of more vortices and results in the pressure drop increment.

Figure 4 presents the variation of friction factor versus the Re number and the dimensionless rotational speed (Rs). As is shown, the friction factor diminishes with the increment of Re number. Although the pressure drop increases with increment of Re number (which is of first power and is at the nominator of Equation (4)) but the velocity amount increase too (which is of second power and is at the denominator). Since the effect of increment of velocity amount is dominant to the

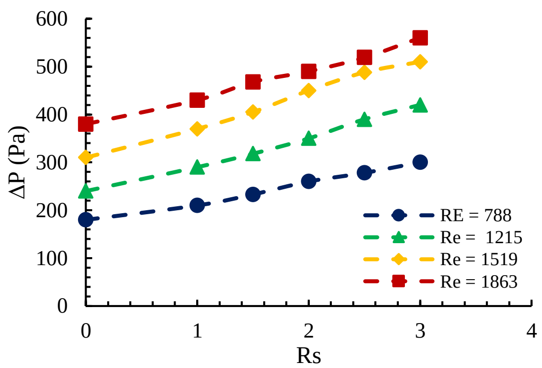
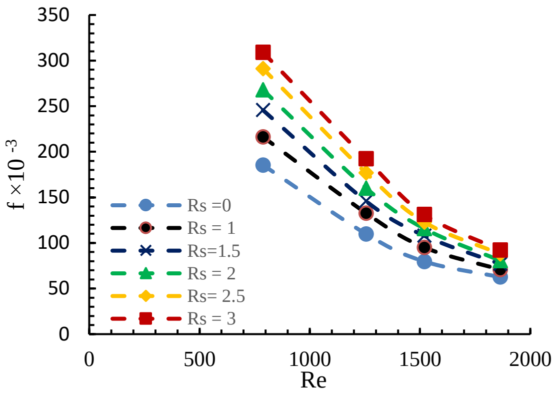
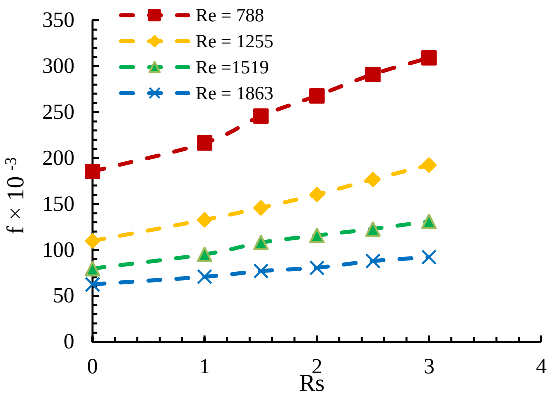


Figure 3. Pressure drop vs Rs



(a)



(b)

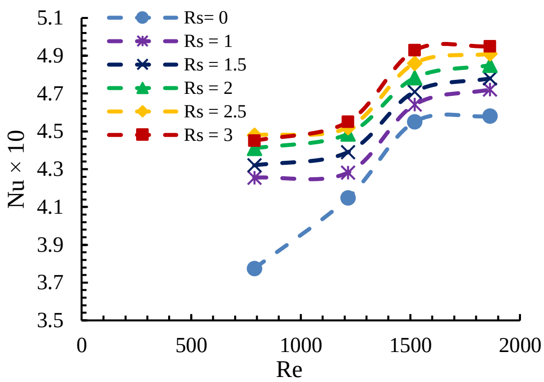
Figure 4. Friction factor coefficient vs (a) Re number (b) Rs

increment of pressure drop, as a result, the friction factor decrease. Looking at Figure 4B, it is seen that the increment of rotational speed (Rs) causes to the increment of friction factor. Indeed, the increment of Rs only increases the pressure drop and does not have effect on values of the axial velocity. So the axial velocity is independent to the amount of Rs. The maximum increment in friction factor due to increment of Rs was related to the Re = 1215 and was around 73%.

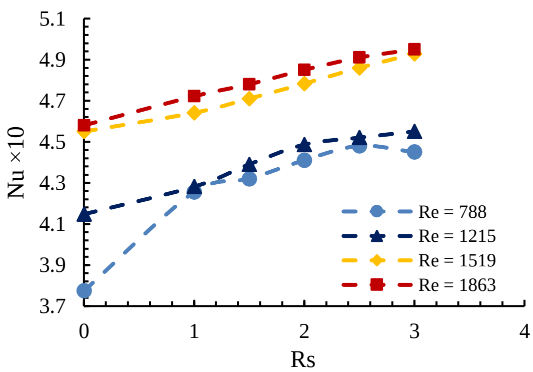
4. 2. Nusselt Number Analysis The Nusselt number (Nu) is a dimension less number and denotes the potential of heat transfer feasibility. The behavior of Nu number vs Re number and Rs is shown in Figure 5. Also, the Figure 6. presents the variation of Nu/Nu_0 vs Re number and the dimensionless rotational speed.

As is presented in Figure 5 the augmentation of both Re number and the dimensionless rotational speed cause to increment at Nu number. Indeed, that as the Re number and the Rs increases the more interactions occur between the axial flow and the swirling flows created by rotational shaft. These interactions cause to creation of eddies and vortexes. These phenomena lead in better mixing at the boundary layer. As the mixing phenomena improves the heat transfer coefficient and subsequently the Nu number increase too. Figure 6 presents the variation of Nu/Nu_0 vs Re number and Rs. As is presented the maximum increment was occurred at the lower Re numbers and the highest dimension less rotational speed. Actually, at the lower Re numbers the swirling flows could better effect on the boundary layer. As the Re number increase the axial flow is more potent than the swirling flow and could easily sweep the swirling flow with itself. Consequently, the swirling flow would have less effect on the boundary layer and less mixing phenomenon occurs at the boundary layer. Subsequently less increment at the heat transfer coefficient and the Nu number happens. Looking at Figure 6B, it could be seen that there is a deviation from trend at the highest Reynolds number (1863.98). It should be noted that the mentioned Reynolds number falls in a range that mostly in this range of Reynolds number the flow has transient regime. Within the transient regime the fluctuations in the flow intensify and the turbulence intensity of the flow is more than those related to lower Reynolds number. This increment in the turbulence intensity increase the heat transfer coefficient and consequently leads in increment in Nusselt number.

4. 3. Performance Efficiency Coefficient (PEC) Performance efficiency coefficient (PEC) is one of the



(a)



(b)

Figure 5. Nu number vs (a) Re number and (b) Rs

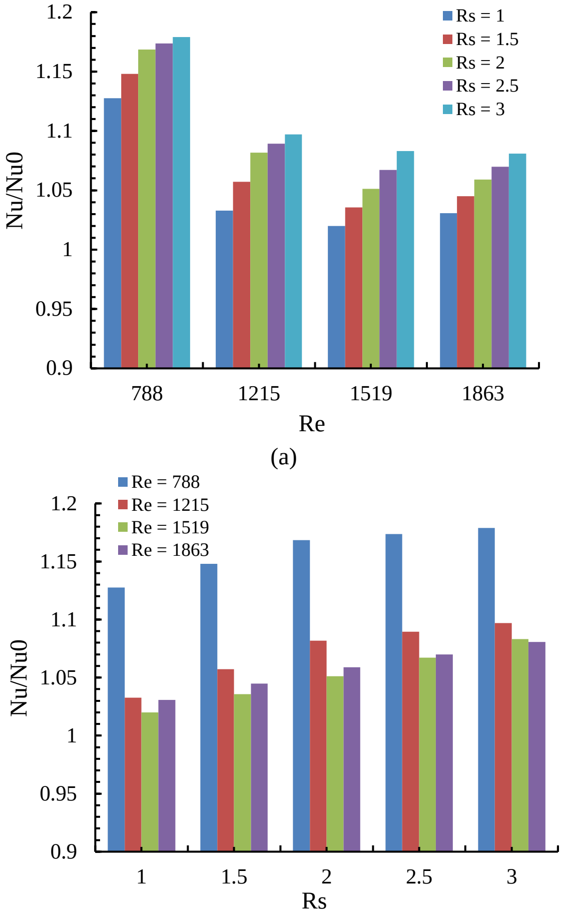


Figure 6. Nu/Nu_0 number vs A) Re number and B) Rs

very important parameters in the evaluation of heat transfer enhancement methods. Due to the simultaneous consideration of the influence of both friction factor and Nu number PEC is very useful in evaluating heat transfer enhancement techniques. Analyzing this parameter allows the engineers to choose the best heat transfer enhancement technique for a certain application. The variation of PEC vs dimensionless rotational speed is presented in Figure 7. As is shown by increment of the Rs the PEC decreases. This point denotes that the increment of friction factor is dominant to the increment of Nu number. The best amounts for the PEC is achieved for Re= 788 and Rs= 1. This point shows that at the lower Re numbers and rotational speeds the presented technique could be more effective than other cases. As shown in Figure 6, it could be seen that the thermal efficiency of the system in Reynolds number of 1863 against the trend. As mentioned before, this values of Reynolds number falls in the transient regime. In this regime the Nusselt number increases to the turbulent nature of the flow. Consequently, in this Reynolds number the thermal efficiency factor finds improved values.

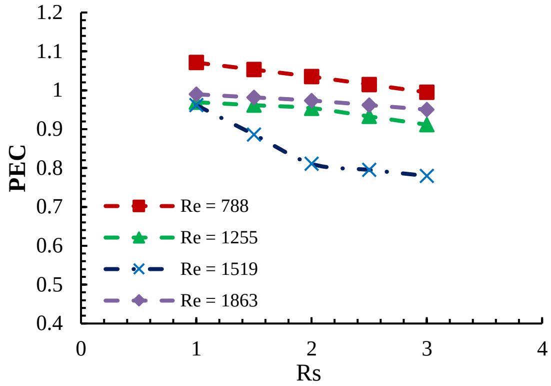


Figure 7. Variation of PEC vs Rs

4. 4. The Cost Per Benefit Ratio Factor

The C.B.R factor is the ratio of variation percentage of pressure drop per variation percentage of Nu number. This parameter is another important parameter which could be notified as an assessment criterion for heat transfer enhancement techniques. As is presented in Figure 8 the C.B.R factor increases with the increment of dimensionless rotational speed. The best values of C.B.R factor (lowest amounts) are related to lower amounts of Rs denoting this point that the rotational shaft could be more effective in low rotational speeds. However, at the low rotational speeds the C.B.R values are more than unite. This points out that the pressure drop increment is always dominant to the increment of Nu number.

5. CONCLUSIONS

Present study experimentally evaluated the influence of a rotational circular shaft on thermal behavior of a circular tube. The circular tube was put under fixed heat flux. The heat flux was produced via heater wire which was evenly twisted around the circular tube. The key results of this study are summarized as following.

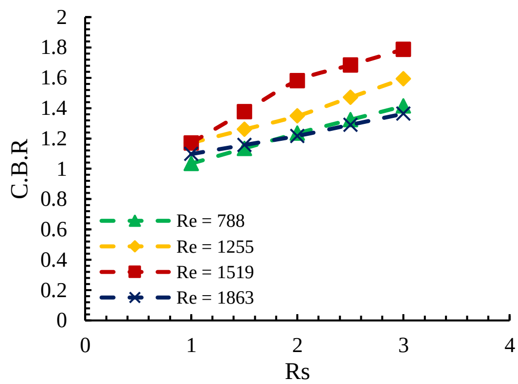


Figure 8. Variation of C.B.R vs Rs

The rotational shaft increases the Nu number.
The maximum augmentation was around 19%.
The rotational shaft could significantly increase the pressure drop and friction factor.
The maximum increment of 78% was achieved for friction factor.
From the view point of both PEC and C.B.R factors the use of rotational shaft to increase the thermal performance of the heat exchanger could be more efficient at low Re numbers and low dimensionless rotational speed. However, by the increment of Re number and reaching to the transient regime of the flow the behavior of the system differs and gets better.

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Persian Abstract

چکیده

روشهای فعال و غیرفعال دو مکانیسم اصلی بهبود انتقال حرارت هستند. روشهای فعال از نیروهای خارجی برای بهبود انتقال گرما استفاده می کنند. این تحقیق رفتار حرارتی و اصطکاکی یک لوله دایره ای حاوی شافت چرخشی را ارزیابی می کند. شار حرارتی ثابت بر روی لوله دایره ای اعمال شد. دمای ورودی و خروجی سیال و همچنین دمای دیواره لوله ها برای محاسبه ضریب انتقال کلاه اندازه گیری شد. تعداد Re (رینولدز) بین 800-2000 بود. همچنین ، سرعت چرخش بدون بعد (Rs) مقادیر 1،1.5 ، 2 ، 2.5 و 3 را نشان داد. نتایج نشان داد که شافت چرخشی می تواند تعداد Nu را افزایش دهد. تا٪ 18 همچنین ، نتایج نشان داد که شافت چرخشی می تواند به طور قابل توجهی افت فشار و ضریب اصطکاک را افزایش دهد. حداکثر افزایش 78٪ برای فاکتور اصطکاک حاصل شد. مشخص شد که استفاده از شافت چرخشی می تواند در تعداد کم Re و سرعت چرخش بدون بعد کم کاراتر باشد. همچنین ، مشخص شد که با افزایش تعداد رینولدز و قرار گرفتن در رژیم گذرا ، کارایی سیستم بهبود می یابد.