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EXPERIMENTAL INVESTIGATION OF USING NEW SHAPE AUGMENTATIONS FOR ENHANCING HEAT TRANSFER IN HEAT EXCHANGERS

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Abstract: The present investigation aims to study experimentally the effect of using basket twisted bars as a new shape of turbulators in order to achieve better heat transfer inside tube heat exchangers. In this study, air is used as a working fluid with six different values of flow rates (Reynolds number values vary from 6000 to 13500), while the value of heat flux is kept constant in this investigation. In addition, the effect of spacing ratio (*SR*) between these turbulators is taken into account. Therefore, 3, 4 and 5 pieces of these turbulators are distributed along the test section during experiments. The results show that the heat transfer rate for all cases tends to increase with rising Reynolds number value, while the friction factor shows downward behavior with the same Reynolds number value. In addition, it is found that (*SR*=4.2) gives maximum heat transfer rate with 115.9 % above plain tube case. Friction factor values increase by 313%, 235% and 193% for SR= 4.2, 6.4 and 11 respectively comparing with plain tube). The enhancement efficiency also increases when inserting baskets twisted bars with rates 127.5 %, 134.3 % and 139.8 % for *SR*= 11, 6.4 and 4.2 respectively.

Key Words: *Energy saving systems, Heat exchangers, turbulators.*

تحقيق عملي لبيان استخدام أدوات حشر جديدة الشكل لتحسين انتقال الحرارة في المبادلات الحرارية

الخلاصة: يهدف العمل الحالي الى دراسة عملية لبيان تأثير استخدام سلة قضبان ملتوية كأشكال جديدة من المضطربات وذلك لغرض تحقيق انتقال حرارة أفضل داخل المبادلات الحرارية الانبوبية. في هذه الدراسة، تم استخدام الهواء كمائع العمل بستة قيم مختلفة لمعدلات الجريان (قيم عدد رينولدز تتراوح من 6000 الى 13500)، بينما قيمة الفيض الحراري تم ابقائها ثابتة في هذا التحقيق. بالإضافة الى ذلك، تأثير نسبة المسافة (*SR*) بين قطع المضطربات تم أخذها بنظر الاعتبار. وبالتالي تم توزيع 3، 4 و5 قطع من المضطربات هذه على طول انبوب الاختبار خلال عمل التجارب. تُبين النتائج ان معدل انتقال الحرارة لكل الحالات يميل للازدياد مع ارتفاع قيمة عدد رينولدز، بينما معامل الاحتكاك يُبين سلوك تناقصي مع نفس قيمة عدد رينولدز. بالإضافة الى ذلك، وجد أن (*SR*=4.2) تعطي اعظم معدل للانتقال الحرارة بمقدار115.9% فوق حالة الأنبوب المستوي. قيم معامل الاحتكاك تزداد بمقدار 313%، 235% و193% لـ *SR* تساوي 4.2، 6.4 و11 بالتتابع مقارنة مع حالة الانبوب المستوي. كفاءة التحسين كذلك تزداد عند اضافة سلال القضبان الملتوية بمقدار127.5% ، 134.3% و 139.8% لـ *SR* تساوي 11، 6.4 و4.2 بالتتابع.

1. Introduction

The issues of rising energy prices and energy losses push the world to think seriously to find solutions for these issues like energy saving.

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Heat exchangers are considered one of many devices which are used for this purpose [1]. The efficiency of heat exchangers is improved by many technological ways. For example, using augmentations inside heat exchanger such as twisted tapes, coiled tubes, helical strip …etc., and this way is called Passive Technique [2]. In general, turbulators work on increasing turbulent fluctuations, and as a result of that the fluid will be mixed better and the thickness of thermal boundary layer will be reduced near the wall [3].

Given the importance of these techniques, many researchers pay their attention around these technologies and made many research. Thianpong et al [4] used twisted rings with different pitch and width ratios, to study their effects on heat transfer in tube heat exchanger. Karakaya and Durmuş [5] studied experimentally the effect of using conical spring turbulators with three cone angles (30^0, 45^0 and 60^0) in order to enhance heat transfer.

They used different arrangements (diverging, converging and converging-diverging) in their experiments. The results showed that the diverging arrangement with using cone angle 30^0 gives best heat transfer rate. The effect of using perforated nozzle-turbulator for improving heat transfer was studied by Mohammed et al [6].

Three different shapes (Square, Triangle and Circle) were used for making holes in nozzles-turbulators wall. They stated that the best heat transfer rate in term Nusselt number can be achieved by Triangle holes case with rate 253% compared with the plain tube case. Anvari et al [7] used conical rings in two arrangements (converging and diverging array), and studied their effect on heat transfer numerically and experimentally.

They found that diverging case improves efficiency more than converging case. Salih [8] studied experimentally enhancing heat transfer in tube fitted with discrete coils. Researcher used three spacing between turbulators (S/l= 0.22, 0.57 and 1.2) with different velocity flow. The results showed that the spacing S/l= 0.57 gives maximum heat transfer rate. Patil et al [9] investigated the effect on inserting cut corrugated twisted tape inside horizontal tube. They tested three twist ratios 8.33, 9.79 & 10.42 with multi values of Reynolds number ranged between 4000 to 9500.

They demonstrated that heat transfer efficiency increases by inserting corrugated twisted tape with rates varied from 18 to 52 percent. Shinde [10] studied experimentally the effect of inserting Screw tape which was made from different materials on enhancing heat transfer in double pipe heat exchanger. It was found that using aluminum screw tape improves heat transfer coefficient (Nu) with 15 to 20 % than M.S. screw tape. Chokphoemphun et al [11] presented experimental investigation of testing adding multiple twisted tapes (single, double, triple, and quadruple) inside horizontal tube. They stated that the maximum enhancement can be done by using quadruple type. Zang et al [12] analysed numerically the effect of using two spiral springs on improving thermal heat transfer.

The Authors reviled that using double spring increases heat transfer rate and gives considerable enhancing for thermal performance in tube heat exchanger. Although many efforts have been done to improve the heat transfer in heat exchangers by using different augmentations, it is worth noting that using basket twisted bars for this purpose did not investigate. Therefore, the present investigation aims to study the feasibility of using basket twisted bars with different spacing ratio (*SR*) for improving heat transfer in heat exchangers.

2. Apparatus Description

The experimental rig which is shown schematically in Fig.1 was built to perform the experiments. The test section was built from aluminum tube of dimensions {length (L) =1350 mm, inner and outer diameter 45 & 50 mm respectively}.

The test tube was heated by using flexible electrical wire to provide a uniform heat flux as boundary conditions. The outer surface of the test tube was insulated by two layers of gypsum and rubber to avoid heat losses. The temperature distribution along the tube wall was achieved by inserting eighteen thermocouples (type K) in the tube wall. Moreover, four thermocouples were inserted at the inlet and outlet of the test section to measure the bulk air temperature.

Fig.2 shows the basket twisted bars (which are made from iron) with all Dimensions. The cross section of twisted bars is square with length of (a= 6 mm). Four bars were used through the fabrication process, in which there ends were fixed together by welding. After that, the bars were twisted with pressure towards inside from the two ends until a basket of twisted bars is formed and the pressure stops when the size of the basket is commensurate with the internal diameter of the test tube.

Other facilities such as inclined manometers, blower, orifice meter, variac transformer, selector switch and digital thermometer were used to measure or specify flow rate, pressure drop, heat flux and temperature distribution along the test section.

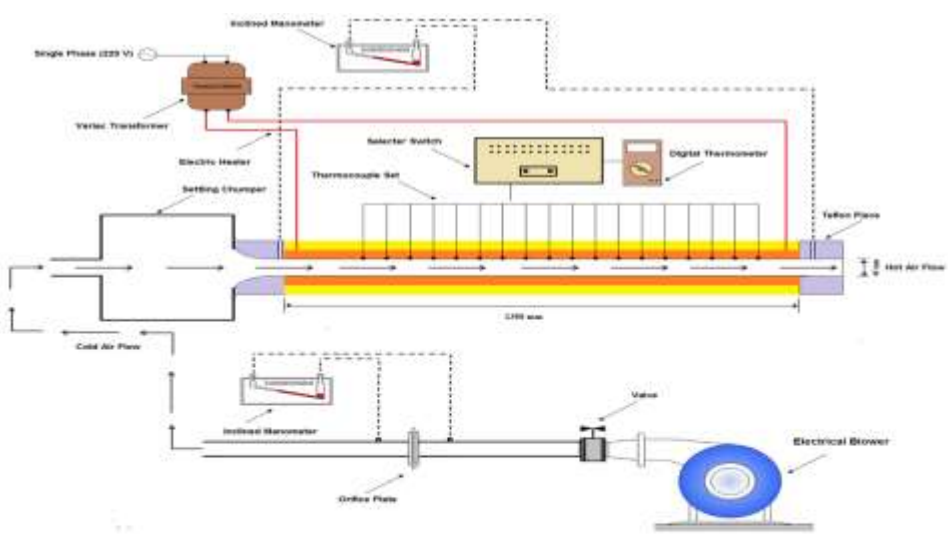
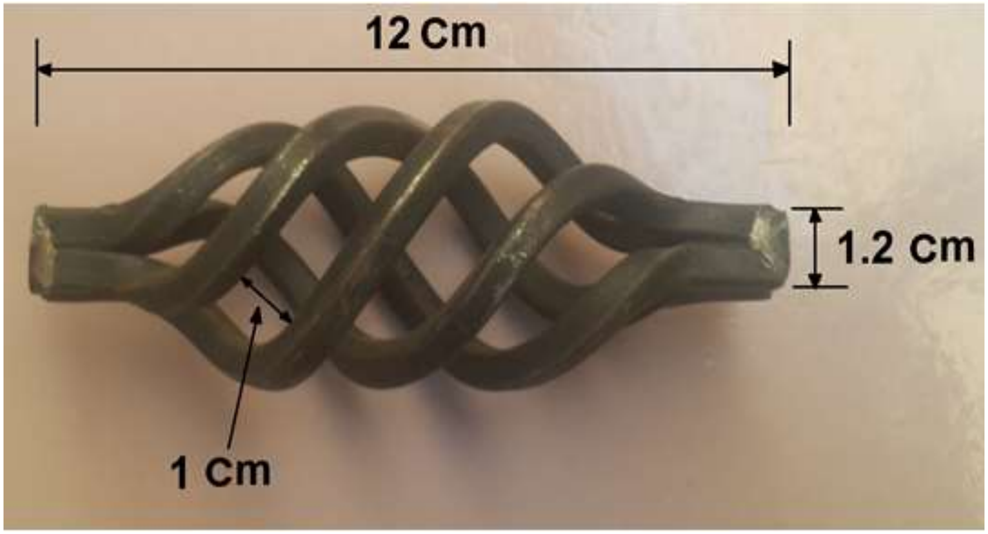


Figure 1. Schematic diagram of Experimental Rig.

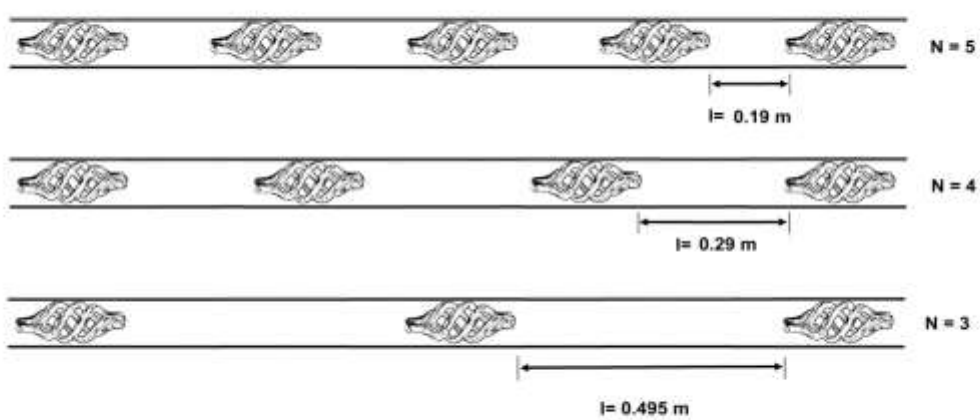


3. Study Plan

Two parameters are adopted in this work:

Reynolds number effect: Using air as a working fluid with six values of Reynolds number varied from 6000 up to 13500 in all the experiment runs.

Spacing ratio (*SR*) between turbulators: Three spaces between turbulators have been chosen with (*SR*= 11, 6.4 and 4.2). Hence, distributing the turbulators by 3, 4 and 5 pieces along the test section as shown in Fig.3.



4. Experimental procedure

The experiments are performed with three distributions of turbulators and six Reynolds numbers (Re starts from 6000 then increases to 7500, 9000, 10500, 12000 and 13500), while the heat flux has been chosen as a constant boundary condition. All values of initial boundary conditions consists of temperature distribution, pressure drop and velocity of air flow are measured firstly. After that, the system is left for about 3 hours to reach the steady state conditions. Then, the pressure drop, outlet and inlet air temperatures and temperature distribution along the test tube wall are measured.

5. Data analysis

A MATLAB program is developed to analyze the experimental data and to perform calculations. In addition, the bulk mean temperature was used for calculating all fluid properties.
The friction factor (*f*) can be calculated from the following equation:

$$f = \frac{\Delta P}{\left(\frac{L}{d}\right)\left(\frac{\rho U^2}{2}\right)} \tag{1}$$

The actual heat input to the test section can be determined as follows:

$$Q = \frac{Q_{net} + Q_{conv.}}{2} \tag{2}$$

Where:

$$Q_{net} = \text{Input Voltage (V)} \times \text{Input Current (I)} - Q_{losses} \tag{3}$$

And

$$Q_{conv} = m^{\cdot}\, C_p\, (T_{b,out} - T_{b,in}) \tag{4}$$

Also, the convection heat transfer from the test section can be written as follows:

$$Q = \bar{h} A_s (\bar{T}\, wall - T_{bulk}) \tag{5}$$

Thus, the heat transfer coefficient can be written as follows:

$$\bar{h} = \frac{Q}{A_s (\bar{T}\, wall - T_{bulk})} \tag{6}$$

Where: T_{bulk} can be calculated from the following equation [8]:

$$T_{bulk} = (T_{b,out} + T_{b,in})/2 \tag{7}$$

The equation of average Nusselt number ($\overline{Nu}_d$) can be written as follows:

$$\overline{Nu}_d = \frac{\overline{h}.D_h}{K_a} \tag{8}$$

The enhancement efficiency ($\eta_{enhancement}$) which is used to estimate the thermal performance after adding turbulators can be calculated from the following equation [8]:

$$enhancement = \frac{\frac{Nu_t}{Nu_p}}{\left(\frac{ft}{fp}\right)^{\frac{1}{3}}} \tag{9}$$

6. Results and Discussion

The variations of average Nusselt number and friction factor against Reynolds number for plain tube case are displayed in Fig.4 and Fig.5. Values of average Nusselt number are compared with Mohammed et al [6] and Blasius which is cited in [8]. While, the friction factor values are compared with Mohammed et al [6] and Promvonge & Eiamsa-ard [13] in order to verify experimental results. It can be seen that the average deviation value in Fig.4 is about 4.4 - 5.6 %, while average deviation value reaches about 3-4% in Fig.5. Therefore, it can be said that the experimental results are acceptable.

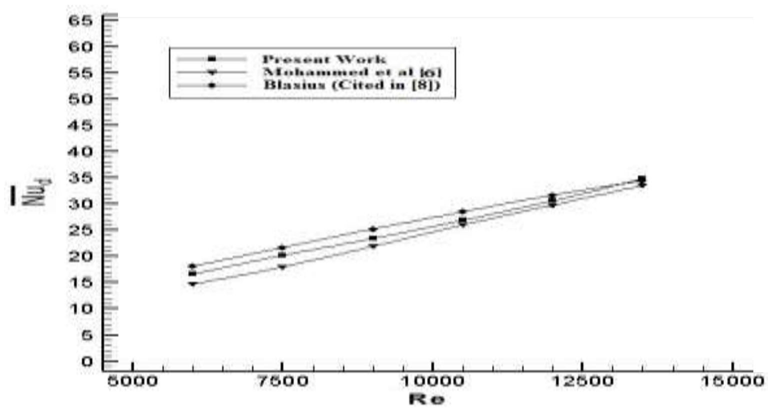


Figure 4. Validation of Nusselt Number for Plain Tube Case.

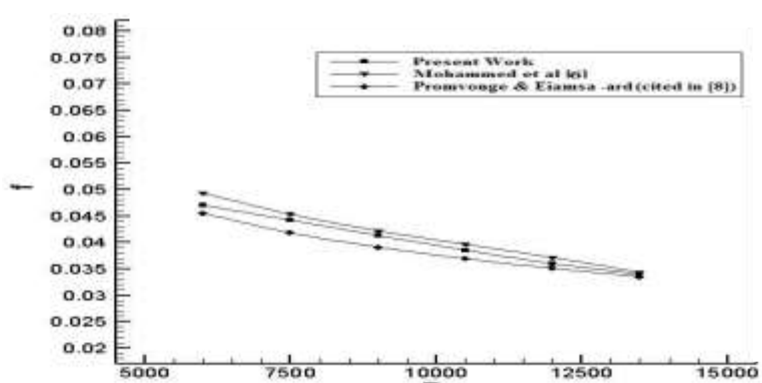


Figure 5. Validation of Friction Factor for Plain Tube Case.

Fig.6 presents the variation of average Nusselt number ($\overline{Nu}_d$) versus Reynolds number (Re) for different spacing ratio between turbulators. It can be observed that ($\overline{Nu}_d$) values show uptrend with increment (Re) due to increasing turbulence in flow as a result of rising Reynolds number, hence increasing heat convection [13]. Moreover, the ($\overline{Nu}_d$) values for all cases of using augmentations are higher than that of plain tube case with 115.9% for *SR*= 4.2, 97% for *SR*= 6.4 and 79% for *SR*= 11. This behavior can be attributed to that using turbulators increase turbulence intensity and reduce thickness of thermal boundary layer leading to raise convection [14].

In addition, it is found that the ($\overline{Nu}_d$) values show uptrend with decreasing (*SR*) for the same (Re). This result can be attributed to that decreasing spacing ratio make turbulence intensity stronger and thus increasing heat transfer.

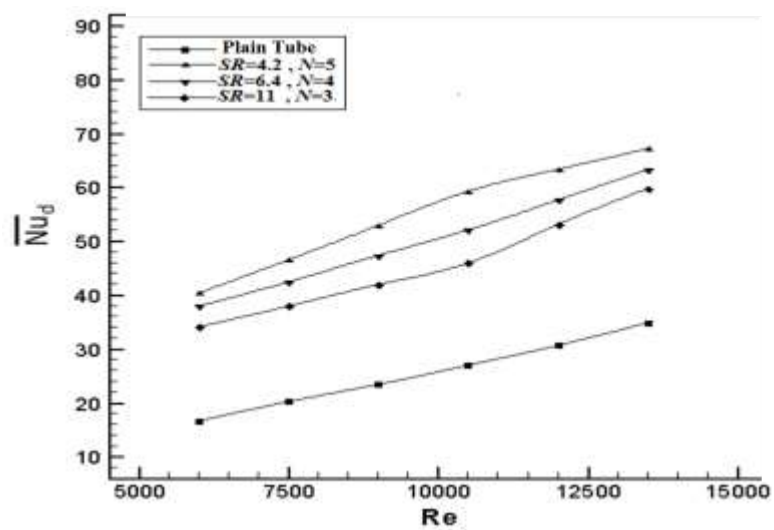


Figure 6. Variation of ($\overline{Nu}_d$) Versus (Re) for Different Spacing between Turbulators.

The influence of inserting basket twisted bars with different spacing ratio (*SR*) on the variation of friction factor (*f*) is shown in Fig.7. This figure shows that the friction factor value decreases with increasing (Re) value because of the inverse relationship between the friction factor and square velocity as shown in Eq.1 [15].

In addition, it is observed that the friction factor values rise significantly by adding these turbulators (*f* value comparing with plain tube increases by 313%, 235% and 193% for *SR*= 4.2, 6.4 and 11 respectively). This is due to increasing both of turbulence intensity as well as surface area leading to increase friction loss [16].

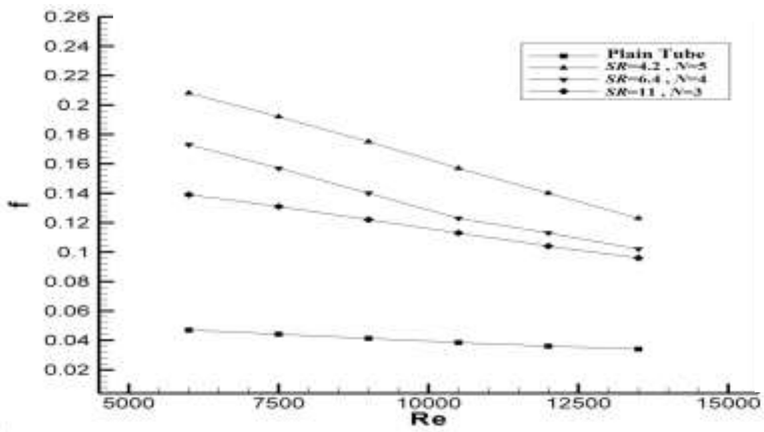


Figure 7. Variation of (*f*) Versus (Re) for Different Spacing between Turbulators.

The empirical correlations which are given in Table 1 are developed by relating both of average Nusselt number and friction factor with Reynolds number (Re) in plain tube case. While in case of using Basket twisted bars turbulators, the average Nusselt number and friction factor are related with Reynolds number (Re) and spacing ratio between turbulators (*SR*). In addition the experimental data are compared with predicted data within (±1.1% to ±2.9%) for Nusselt number and (±1.2% - ±3.1%) for friction factor.

Table 1. Empirical Correlations for Plain and Basket Twisted Bars Turbulators Cases.

No.	Case	Correlations of (Nu_d)	Correlations of (f)
1	Plain tube	$\overline{Nu}_d = 0.0075.Re^{0.9009}.\ Pr^{0.4}$	$f = 1.6194.Re^{-0.405}$
2	Basket twisted bars SR=4.2	$\overline{Nu}_d = 0.104.Re^{0.642}.\ Pr^{0.4}.SR^{0.3709}$	$f = 33.99.Re^{-0.64}.\ SR^{0.36}$
3	Basket twisted bars SR=6.4	$\overline{Nu}_d = 0.09.Re^{0.6291}.\ Pr^{0.4}.SR^{0.3579}$	$f = 29.803.Re^{-0.661}.\ SR^{0.339}$
4	Basket twisted bars SR=11	$\overline{Nu}_d = 0.0473.Re^{0.6797}.Pr^{0.4}.SR^{0.3203}$	$f = 2.0691.Re^{-0.457}.\ SR^{0.543}$

7. Thermal Performance Evaluation

The thermal performance evaluation of using basket twisted bars is presented in Fig. 8. This evaluation is very important because it is used to assess the possibility of using turbulators in practical applications. The evaluation is done by using Eq. 9 in term enhancement efficiency and under constant pumping power assumption. It can be observed that the enhancement efficiency value decreases with rising Re value. In addition, inserting turbulators with different spacing ratio (*SR*=11, 6.4 and 4.2) gives enhancement efficiency with rate more than unity. This indicates that enhancing heat transfer is more than the effect of increasing friction loss [17]. Fig. 9 shows a comparison between spacing ratio cases and their effect on enhancement efficiency. The comparison shows that the maximum enhancement is achieved by *SR*= 4.2 with rate 139.8 %.

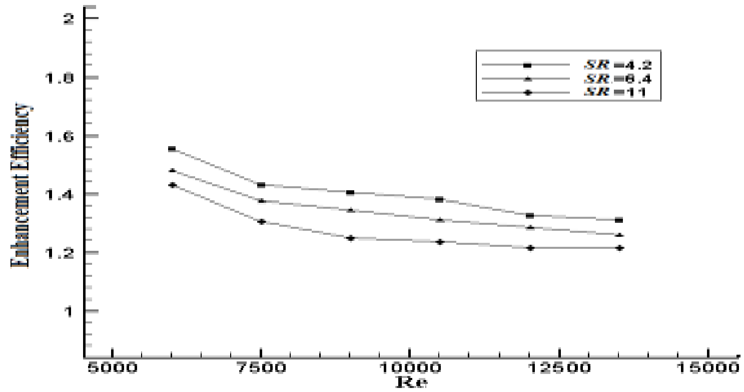
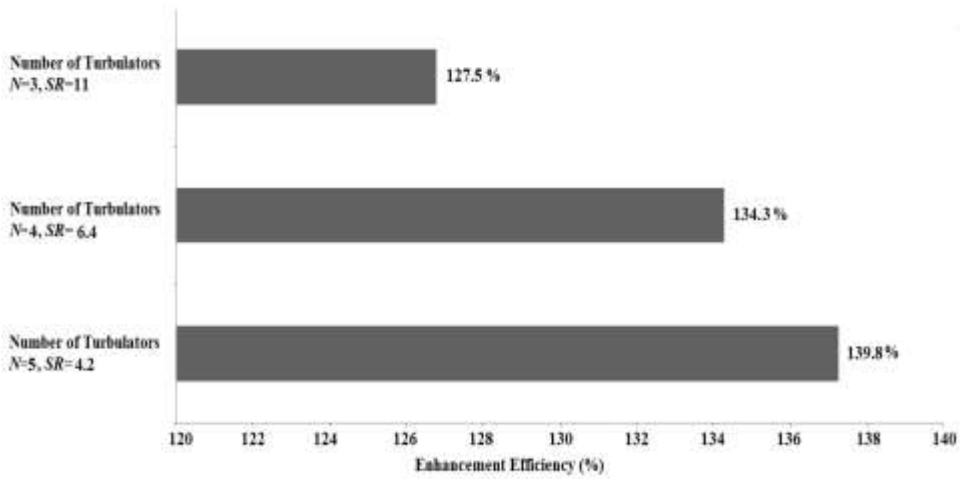


Figure 8. Enhancement Efficiency versus Reynolds Number.



8. Conclusions

According to the experimental results, the following points can be concluded:

1- Nusselt number has positive relationship with increasing Reynolds numbers, while the friction factor inversely proportional with increasing Reynolds numbers.
2- Heat transfer rate in terms of Nusselt number increases considerably when using Basket twisted bars turbulators.
3- Heat transfer rate increases with decreasing spacing ratio, in other words, reducing distance between turbulators leads to enhance heat transfer.
4- Using Basket twisted bars with *SR*= 4.2 provides maximum heat transfer rate with 115.9 % above plain tube case.
5- Enhancement efficiency value for all cases of using Basket twisted bars is more than unity. Therefore, it can be demonstrated that the feasibility of using these turbulators as heat transfer improvers.

Nomenclature

A_s Inner Surface Area of Test Section, m^2
a Cross Section Length of Twisted Bars, mm
C_p Specific Heat of Air, J/kg.K
d Inner Diameter of Tube, m
D_h Hydraulic Diameter, m
f Friction Factor
f_p Friction Factor of Plain tube Case
f_t Friction Factor of Adding Turbulators Case
h Coefficient of Heat Transfer, $W/m^2.K$
I Current, A
K_a Thermal Conductivity, W/m.K
L Length of Test Section, m
l Space between turbulators, m
$m^{\cdot}$ Mass Flow Rate, kg/s
N Number of Basket Twisted bars Turbulators
Nu_d Average Nusselt Number
Nu_p Nusselt Number of Plain tube Case
Nu_t Nusselt Number of Adding Turbulators Case
Pr Prandtl Number
SR Spacing Ratio between Turbulators = d/l
Q Actual Heat Input to the Test Section, W
Q_{net} Heat Supplied, W
$Q_{conv.}$ Convection Heat Transfer from the Test Section, W
Re Reynolds Number
$T_{bulk,in}$ Temperature of Air at the Test Section Entrance, K
$T_{bulk,out}$ Temperature of air at the Test Section Exit, K
T_{bulk} Bulk Temperature, K
T_{wall} Average Surface Temperature of test Section, K
U Velocity of Working Fluid, m/s
V Voltage, Volt
ΔP Pressure Drop, Pa
η Enhancement Efficiency
ρ Density, kg/m^3

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