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Heat Transfer Enhancement Inside Elliptic Tube by Means of Rings Inserts

E. Z. Ibrahim[a],*

[a]Associate Professor, Mechanical Power Department, Faculty of Engineering, Zagazig University, Zagazig, Egypt.
*Corresponding author.

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Abstract

Heat transfer, friction factor and enhancement efficiency characteristics in an elliptic tube fitted with elliptic rings have been investigated experimentally. In the experiments, air was used as the tested fluid with a Reynolds number range of 10000 to 32464. The experimental results show a considerable increase in friction factor and heat transfer over the plain tube under the same operation conditions. Over the range investigated, the Nusselt numbers for both employed enhancement devices with different pitches are found to be higher than that of the plain tube. It was found that the best overall enhancement was achieved with pitch = 3d. The results obtained are correlated in the form of Nusselt number and friction factor as a function of Reynolds number and pitches. The results were compared with circular tubes have the same test conditions to show the difference between the circular and elliptic tubes.

Key words: Heat transfer enhancement; Elliptic tube

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INTRODUCTION

The development of high-performance thermal systems still has a considerable interest among the researchers to improve heat transfer. The conventional heat exchangers are improved by means of various augmentation techniques with emphasis on surface enhancements technique. Augmented surfaces can create one or more combinations of the following conditions that are favorable for the increase in heat transfer rate with an undesirable rise of friction: 1) disruption of the development of boundary layer and increase of the turbulence intensity, and 2) increase in heat transfer area. The heat transfer enhancement technology has been improved and widely used in heat exchanger applications; such as refrigeration, automotives, process industry, chemical industry, etc. One of the widely used heat transfer enhancement technique is inserting different shaped elements with different geometries in channel flow (Khaled, 2007; Shoji *et al*., 2003; Zimparov, 2001).

Yakut and Sahin (Yakut *et al*., 2004) studied the heat transfer and friction characteristics in a uniform heat flux fitted with conical-ring turbulators used to provide reverse/turbulent flows in each module of the conical rings. Therefore, significant improvement of heat transfer along the tube wall was achieved. Promvonge and Eiamsa-ard (Promvonge *et al*., 2006) investigated the enhancements of heat transfer in a uniform heat-flux circular tube fitted with conical-nozzles and swirl generator. In their research, the conical-nozzles were placed in a test tube with three different pitch ratios of conical-nozzles, apart from the snail mounted at the tube inlet. The use of the conical-nozzle in conjunction with the snail entrance led to maximum heat transfer rate up to 316% over the plain tube besides a substantial increase in pressure loss as well.

Eiamsa-ard and Promvonge (Eiamsa-ard *et al*., 2006) investigated the influences of V-nozzle inserts on heat transfer, flow friction, and enhancement efficiency characteristics in a uniform heat-flux tube with three pitch ratios. They found that the use of V-nozzles led to a considerable increase in heat transfer rate and a maximum gain of 1.19 on enhancement efficiency was obtained for the smallest pitch ratio used at low Reynolds number. This indicates the crucial effect of the reverse/recirculation flow can promote the heat transfer rate in tubes.

Smithberg and Landis (Smithberg *et al*., 1964) studied friction and forced convection heat transfer characteristics in tubes fitted with twisted tape swirl generators, and presented a correlation for predicting Nusselt number and friction factor. Date and Singham (Date *et al*., 1972), and Date (Date, 1974) reported the prediction of fully developed flow in a tube containing a twisted tape. Hong and Bergles (Hong *at al*., 1976) correlated heat transfer and pressure drop date for twisted tape inserts for uniform wall temperature conditions using water as working fluids in laminar flow. Eiamsa-ard, and Promvonge (Eiamsa-ard *at al*., 2005) reported the enhancement of the heat transfer in a tube with regularly spaced and full-length helical tape swirl generators, and concluded that the full-length helical tape with rod provides higher heat transfer rate than without rod by about 10%.

Chang *et al*. (Chang *et al*., 2007) presented an original experimental study on compound heat transfer enhancement in a tube fitted with serrated twisted tape. A set of empirical correlations that permits the evaluation of the Nusselt number and the fanning friction factor in the developed flow region for the tubes fitted with smooth and serrated twisted tapes was generated for engineering applications. Shaikh (Shaikh *et al*., 2007) studied the effect of baffle size and orientation on the heat transfer enhancement. The results showed that for the vertical baffle, an increase in the baffle height causes a substantial increase in the Nusselt number but the pressure loss is very significant. For the inclined baffles, the Nusselt number enhancement is almost independent of the baffle inclination angle. Naphon (Naphon, 2006) presented the heat transfer characteristics and pressure drop results of the horizontal double tubes with coil-wire insert. It was seen that the heat transfer rate and heat transfer coefficient depend directly on the mass flow rates of hot and cold water. They added that the effect of coil-wire insert on the enhancement of heat transfer tends to decrease as Reynolds number increases.

Several investigations were carried out to determine the effect of the coiled wire or the twisted tape elements on heat transfer and friction factor for a long time studied by Garcı´a *et al*. (Garcı´a *et al*., 2007) and Ozceyhan (Ozceyhan, 2005) respectively. This is because the wire coil or twisted tape inserted in the tube created swirling flows that modify the near wall velocity profile due to the various velocities distributions in the vortex core.

Kim *et al*. (Kim *et al*.,2001) investigated the flow pattern, void fraction and slug rise velocity on counter current two-phase flow in a vertical round tube with coiled wire inserts. They observed that the slug rise velocity and void fraction in a vertical round tube is higher for a coiled wire insert than that in a smooth tube. Ozceyhan *et al*. (Ozceyhan *et al*., 2001) conducted a numerical study for investigating the heat transfer enhancement in a tube with the circular cross sectional rings. The rings were inserted near the tube wall. Five different pitches were considered as p = d/2, p = d, p = 3d/2, p = 2d and p = 3d. A uniform heat flux was applied to the external surface of the tube and air was selected as working fluid. The results show that, Nusselt number increases and friction factor decreases with increasing Reynolds number and the highest Nusselt number and friction factor is obtained in the case of p = d/2. Heat transfer in flat tubes by means of helical screw-tape inserts was studied by Ibrahim (Ibrahim, 2011). The study showed that, the Nusslet number (Nu) and friction factor (f) decrease with the increase of spacer length or twist ratio for flat tube.

As mentioned above, the literature survey on investigations of different types of tube inserts indicates that these inserts generally attached on the tube walls in order to enhance the heat transfer and increase the effective heat transfer area. This work differs from those in the literature by attaching the elliptic ring inserts separated from the elliptic tube wall, thus the effective heat transfer area was not increased and heat transfer enhancement was achieved.

The objective of this paper is to study the heat transfer and flow characteristics in a horizontal elliptic tube with elliptic cross sectional ring inserts. The rings inserted near the tube wall and the heat transfer enhancement is investigated related with the effect of spacing between the rings. Five different spacings between the rings are considered as p = d/2, p = d, p = 3d/2, p = 2d and p = 3d. A uniform heat flux will be applied through the external surface of the elliptic tube and the air passes through it.

1. EXPERIMENTAL APPRATUS

The experiments were carried out in an open-loop experimental facility as shown in Figure 1. The loop consisted of a 3 kW blower, orifice meter to measure the flow rate, and the heat transfer test section. The outer surface of the elliptic tube is covered with an electric insulating tape on which nickel-chrome wire of 0.4 mm was uniformly wound to form the main heater as shown in Figure 2. The main heater is covered with an asbestos layer of 55 mm thickness, on which another nickel-chrome wire of 0.4 mm was uniformly wound to form a guard heater. The Guard heater is covered with a 40 mm thick asbestos layer. The elliptic tube ends are connected to transient sections to convert from circular to elliptic sections. These transient sections let good connections between the elliptic tube and the experiment loop. Two pairs of thermocouples are installed in the asbestos layer between the main heater and the guard heater. The thermocouples of each pair are fixed on the same radial line. The input to the guard heater is adjusted so that, at the steady state, the readings of the thermocouples of each pair became practically the same. Then all the heat generated by the main heater is flowing inward to the elliptic tube. The terminals of the wire are connected to a variac AC transformer, which is used to vary the voltage of the AC current passing through the heating coil. The test-section is elliptic tube and made of copper having a 43 mm major axis and 15 mm minor axis (hydraulic diameter = 27.72 mm) , 1250 mm long (L) and 1.5 mm thick (t) as shown in Figure 3. A copper elliptic rings have the same inner dimensions of the elliptic tube are fixed on the inside surface of the elliptic tube. The number of these rings depends on the required pitch for each case. The inner and outer temperatures of the bulk air were measured at certain points with a multi-channel temperature measurement unit in conjunction with the thermocouple type-K. Twenty thermocouples were tapped on the local wall of the tube and the thermocouples were placed around the tube to measure the circumferential temperature variation. The mean local wall temperature was determined by means of calculations based on the reading of type-K thermocouples.

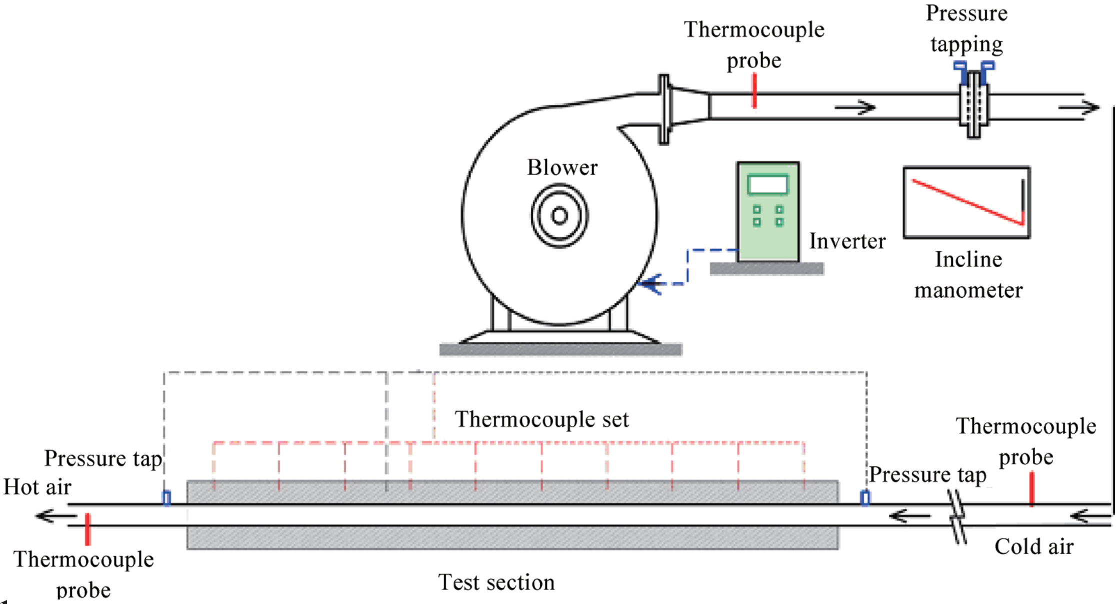

Figure 1
Schematic Diagram of Experimental Heat Transfer Set-up

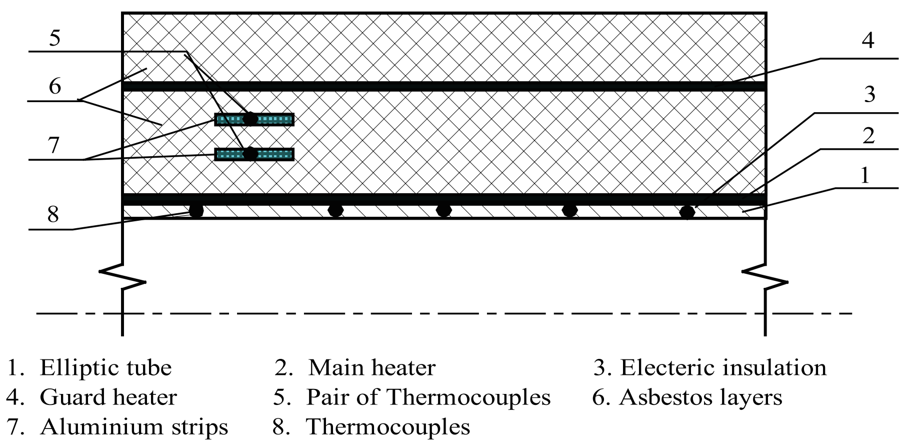

Figure 2
Heaters Arrangement

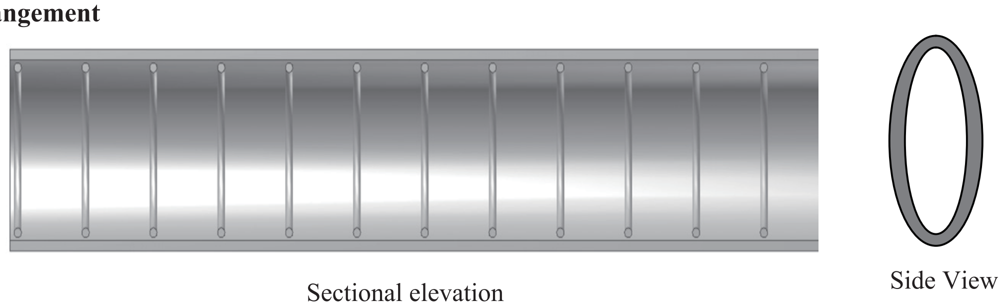

Figure 3
Test Section of Elliptic Tube Fitted with Rings

2. UNCERTAINTY ANALYSIS

The various characteristics of the flow, the Nusselt number, and the Reynolds numbers were based on the average of tube wall temperature and outlet air temperature. The local wall temperature, inlet and outlet air temperature, the pressure drop across the test section and airflow velocity were measured for the heated tube. The average Nusselt numbers were calculated and discussed where all fluid properties were determined at the overall bulk mean temperature. In order to quantify the uncertainties, the reduced data obtained experimentally were determined. The uncertainty in the data calculation was based on Reference [21]. The maximum uncertainties of non-dimensional parameters are ±4% for Reynolds number, ±12% for Nusselt number and ±13% for friction. The uncertainty in the axial velocity measurement was estimated to be less than ±5%, and pressure has a corresponding estimated uncertainty of ±4%, whereas the uncertainty in temperature measurement at the tube wall was about ± 0.6%. The experimental results were reproducible within these uncertainty ranges.

3. DATA REDUCTION

In the present work, air is used as the tested fluid and it is flowed through a uniform heat flux and insulated tube. The steady state of the heat transfer rate assumed to be equal to the heat loss from the test section, which can be expressed as:

$$Q_A = Q_C \quad (1)$$

Where:

$$Q_A = m\, C_{P\,A}\,(t_O - t_I) = VI \quad (2)$$

The heat supplied by the electrical winding in the test tube is found to be 5-8% higher than the heat absorbed by the fluid for the thermal equilibrium test due to convection and radiation heat losses from the test section to the surroundings. Thus, only the heat transfer rate absorbed by the fluid taken for the internal convective heat transfer coefficient. The rate of heat transfer coefficient from the test section was calculated as:

$$h_C = Q_A / A\,(t_W - t_A) \quad (3)$$

Where:

$$t_A = (t_O + t_I) / 2 \quad (4)$$

$$\tilde{t}_W = \text{å}\; t_W / 20 \quad (5)$$

In which t_W is the local surface temperature at the outer wall of the inner tube. The average surface temperature t_W calculated from 20 points of t_W lined between the inlet and the exit of the test tube.

The average Nusselt number, Nu estimated as follows:

$$Nu = h_C\, d_I / k \quad (6)$$

The Reynolds number of the airflow inside the tube is given

$$Re = U\, d_I / n \quad (7)$$

The friction factor, can be written as

$$f = \Delta P / [(L/d_I)(r\, U^2/2)] \quad (8)$$

Where V is the mean air velocity in the tube. All of the thermo physical properties of the air are determined at the overall bulk air temperature from Equation (4).

The overall enhancement efficiency (η) can be written as mention in Reference [18] as follows:

$$h = (Nu_R / Nu_S)(f_S / f_R)^{1/3} \quad (9)$$

4. RESULTS AND DISCUSSIONS

The heat transfer and friction factor of the plain elliptic tube was compared with previous correlations obtained from other work (Incropera *et al*., 1996) for circular tube under similar conditions as shown in Figures 4 and 5 respectively. The circular plain tube data was found to be in good agreement with the previous correlations of Petukhov (Incropera *et al*., 1996) for both the Nusselt number and the friction factor within ± 12% and 10 % respectively.

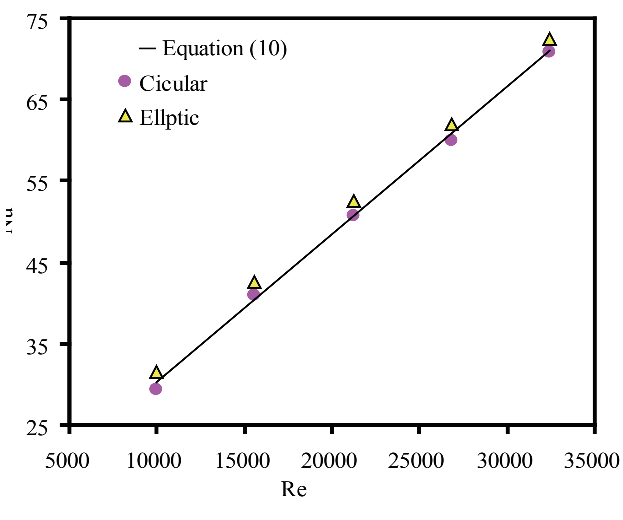

Figure 4
Variation of Nusslet Number with Reynolds Number

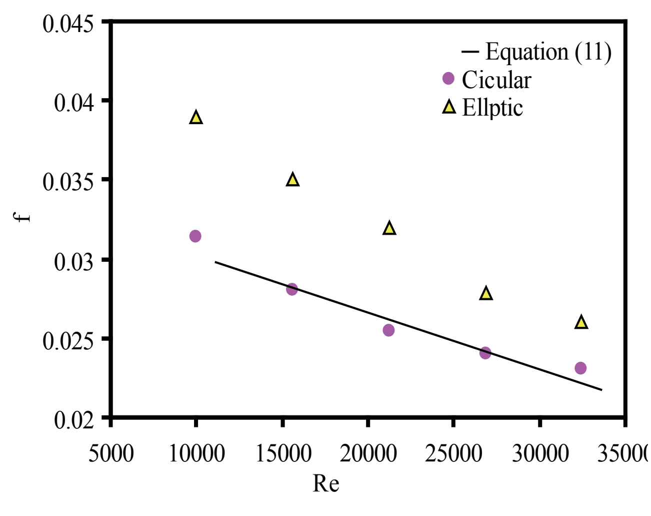

Figure 5
Variation of Friction Factor with Reynolds Number

Nusslet number correlation is as following for Reynolds number range ($10^4 \leq Re \leq 5\times10^6$):

$$Nu = (f\,Re\,Pr) / [1.07 + 12.7\,(f/8)\,(Pr^{(2/3)} - 1)] \quad (10)$$

The friction factor correlation is:

$$f = (0.79 LnRe - 1.64)^{-2} \text{ for } 3000 \leq Re \leq 5\times10^6 \quad (11)$$

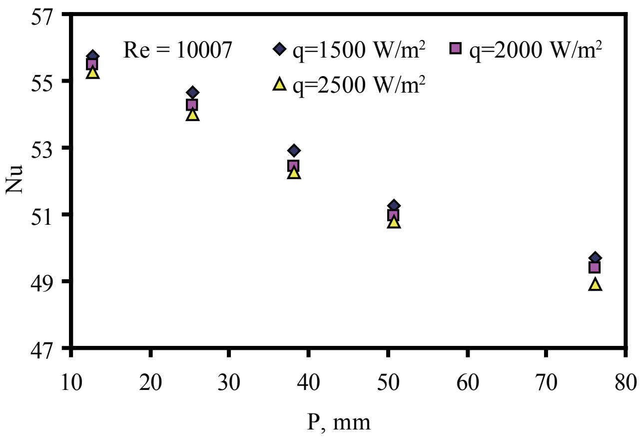

Figure 6
Variation of Nusslet Number with Ring Spacings at Different Heat Flux

Figure 6 and Figure 7 show the Nusslet number various with ring spacing and Reynolds number at varies heat fluxes respectively. It is seen that the maximum difference in Nu at different heat fluxes was less than 2%. Therefore the experimental results suggested a constant heat flux of 2500 W/m^2.

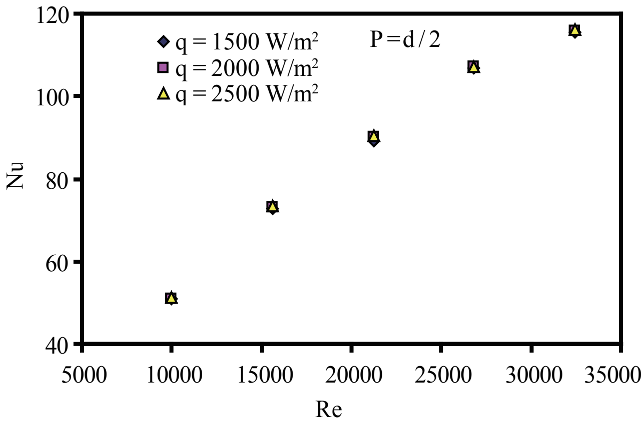

Figure 7
Variation of Nusslet Number with Reynolds Number for Different Heat Flux

The Nusselt numbers obtained under turbulent flow conditions for five different spacing between the rings was presented in Figure 8. The rings attached near the wall with a clearance of 0.2 mm. In this figure, the rings have a considerable heat transfer enhancements with a similar trend in comparison with the smooth tube. The Nusselt number increases with the rise of Reynolds number and the highest Nusselt number was obtained at spacing between the rings of P = d/2. This is because the clearance between the wall and the rings leads to higher velocities in the clearance gap, and consequently it breaks the boundary and the degree of flow turbulence increases. As a result, the heat transfer increases as the Nusselt number for the ring at P = d/2 is found to be about 66-92% more than that of the smooth elliptic tube.

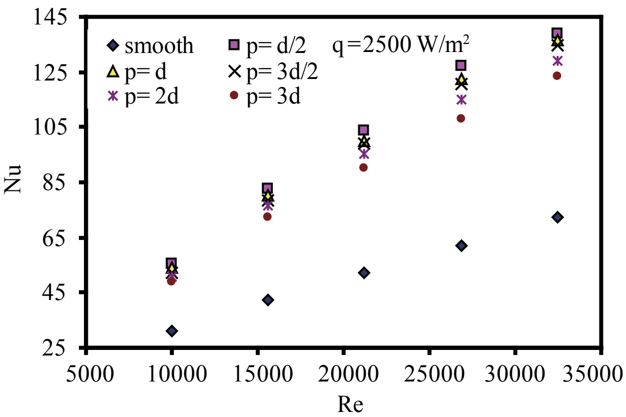

Figure 8
Variation of Nusslet Number with Reynolds Number for Different Ring Spacing

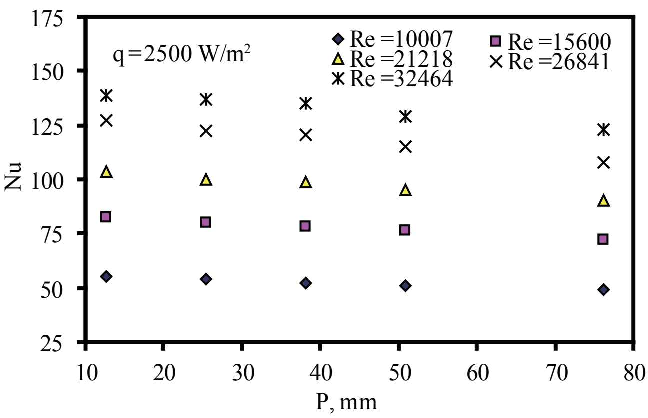

Figure 9
Variation of Nusslet Number with Ring Spacing for Different Reynolds Number

Figure 9 shows the variation of Nusselt number with increasing ring spacings for various Reynolds number. Since Nusselt number decreases with increases ring spacings, the maximum Nusselt number was obtained at the ring spacing (P = 12.7 mm) for all Reynolds number used. This is likely because of the higher turbulence and better contact surface area between fluid and heating wall surface.

For the smooth and ring elliptic tubes, the variation of the measured values of friction coefficient with Reynolds number is shown in Figure 10. It is seen that the friction coefficients of all ring inserted elliptic tubes have nearly the same trend, and it can be said that friction coefficient is independent of Reynolds number for ring elliptic tubes. The friction factor tends to decrease with increasing

Reynolds number and spacing between the rings. From the figure, it can be seen that, the highest friction factor is obtained at the spacing between rings of d/2. This may be attributed to the dissipation of the dynamic pressure of fluid due to higher surface area and the action caused by the reverse flow. Friction factor for the ring at P= d/2 is found to be about 240-359 % more than that of the smooth elliptic tube.

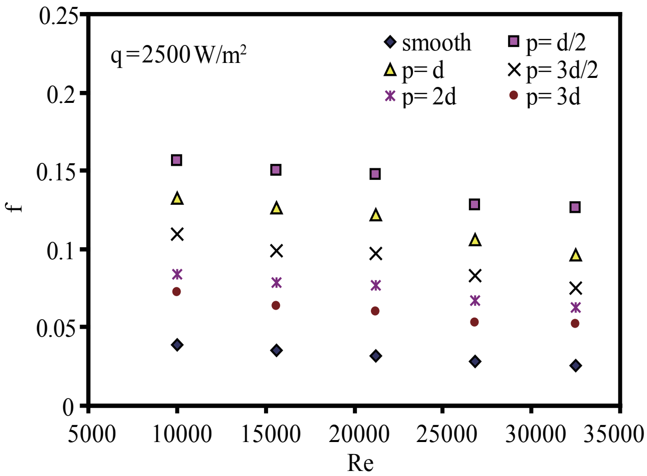

Figure 10
Variation of Friction Number with Reynolds Number for Varies Ring Spacing

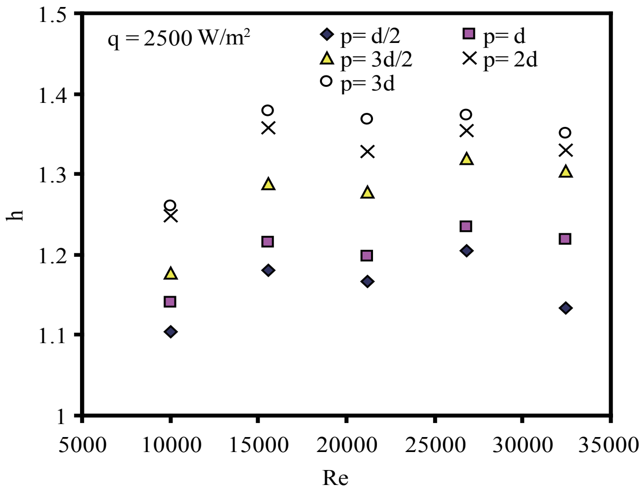

Figure 11
Overall Enhancement Ratio with Reynolds Number for Varies Ring Spacing

Heat transfer enhancement obtained leads to increasing the pressure drop caused by tube insertions. Therefore, a performance analysis is important for the evaluation of the net energy gain to determine if the method employed to increases the heat transfer is effective from energy point of view or not. The variation between the enhancement efficiency and Reynolds numbers at varies ring spacing is shown in Figure 11. From the figure, it can be seen that the overall enhancement ratio increases with the increase in ring spacings. It is clear from Figure 11 that for all cases, the overall enhancement ratio is greater than unity. The best overall enhancement achieved for the spacing between rings of 3d. It can be seen from the figure, that the spacing is not efficient in terms of overall performance because the fluid friction dominates the heat transfer.

5. COMPARISON WITH THE PREVIOUS WORK

Comparisons between the Nusselt number and the friction factor obtained from the present work with another research (Ozceyhan *et al.*, 2008) are showed in Figures 12 and 13, respectively. In the figures, the Nusslet number and friction factor for elliptic tube with inserted ring increases than a circular tube at P = d/2. These increases for Nusslet number are "between" 18.5% to 23.4%, where the friction factor is higher by about 8% to 13%.

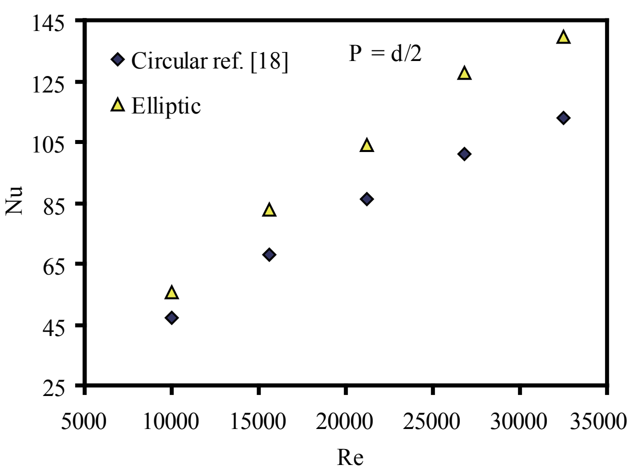

Figure 12
Comparison Between the Present Results of Nu - Re with That of Reference [18]

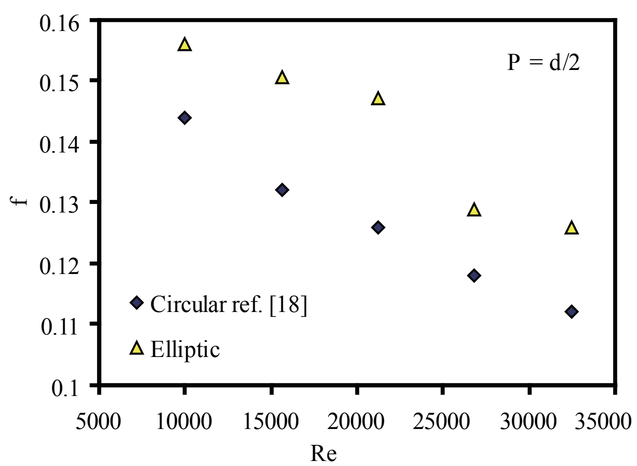

Figure 13
Comparison Between the Present Results of *f*-Re with That of Reference [18]

6. CORRELATION OF THE RESULTS

The general correlation of the Nu as a function of Re and P/L of the experimental results are expressed as following:

$$Nu = a\mathrm{Re}^{b}(P/L)^{c} \quad (12)$$

Where a, b and c are constants.

The experimental data is fitted to get the constants and the following correlation can be obtained:

$$Nu = 0.033\mathrm{Re}^{0.795}(P/L)^{-0.073} \quad (13)$$

$10000 \leq \mathrm{Re} \leq 32464$ and $0.15 \leq P/L \leq 0.9$

The general correlation of the f as a function of Re and P/L of the experimental results is expressed as following:

$$f = c\mathrm{Re}^{m}(P/L)^{n} \quad (14)$$

The experimental data is fitted to get the constants and the following correlation can be obtained:

$$f = 0.776\mathrm{Re}^{-0.261}(P/L)^{-0.493} \quad (15)$$

$10000 \leq \mathrm{Re} \leq 32464$ and $0.15 \leq P/L \leq 0.9$

The calculated Nusselt number (Nu_{cal}) and (f_{cal}) from Equations (13) and (15) is plotted versus experimental Nusselt number (Nu_{exp}) and (f_{exp}) in Figures 14 and 15. As shown from this figure the maximum deviation between the experimental data and the correlations is ±5% and ±10% respectively.

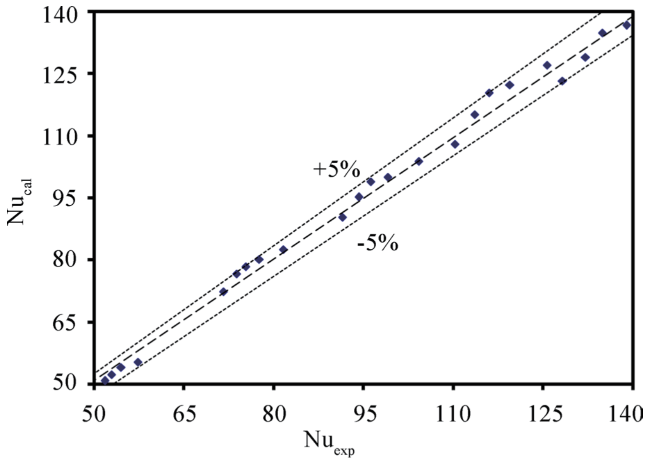

Figure 14
Nu_{cal} Against Nu_{exp}

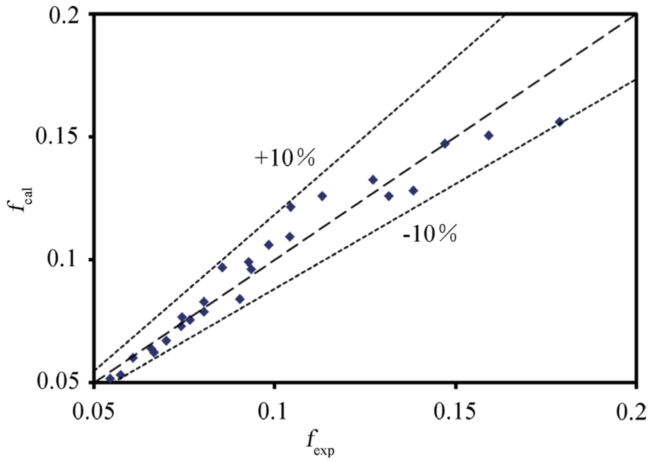

Figure 15
f_{cal} Against f_{exp}

7. CONCLUDING REMARKS

An experimental study was performed to determine the effect of attaching the rings near the wall and the elliptic ring spacing on the heat transfer and friction characteristics within the range of Reynolds number from 10000 to 32464 for a uniform heat flux in an elliptic tube. In this study, constant heat flux is used to evaluate the overall performance of five different spacing (p = d/2, d, 3d/2, 2d and 3d) and the following conclusions were derived:

a) For all cases, Nusselt number increases and friction factor decreases with increasing Reynolds number. The highest Nusselt number and friction factor is obtained at the ring spacing of p = d/2.
b) The Nusselt number and friction factor increase with decreasing ring spacing.
c) For the all Spacing between the rings, the overall enhancement ratios are higher than unity for all investigated cases.
d) The fluid friction dominates the heat transfer, so the use of these rings with a spacing of d/2 is not thermodynamically advantageous based on heat transfer enhancement.
e) The best overall enhancement is achieved for the ring spacing of 3d.
f) Correlations for the Nusselt number and the friction factor based on the present experimental data were introduced for practical use.
g) At P = d/2, the increase of Nusslet number is about 18.5% to 23.4%, while the friction factor is higher than a circular tube by about 8 % to 13 %.

NOMENCLATURE

A : Heat transfer surface area, m^2
Cp : Specific heat of air, J/kg K
d_I : Inside equivalent diameter of the test tube, m
f : Friction factor
h : Heat transfer coefficient, $W/m^2 K$
I : Current, A
k : Thermal conductivity of air, W/m K
L : Length of the test section, m
m : Mass flow rate, kg/s
Nu : Nusselt number
ΔP : Pressure drop, Pa
P : Spacing between the rings, m
Q : Heat transfer rate, W
Re : Reynolds number
t : Temperature, °C

t˜: Mean temperature, °C
U : Average axial velocity inside the test section, m/s
V : Voltage, V
ν : Kinematic viscosity, m^2/s
ρ : Density, kg/m^3
h : The overall enhancement efficiency

SUBSCRIPTS

A : Air
C : Convection
I : Inlet
O : Outlet
W : Wall
R : ring inserted tube
S : smooth tube

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