Journal of Applied Fluid Mechanics, Vol. 8, No. 4, pp. 885-892, 2015.
Available online at www.jafmonline.net, ISSN 1735-3572, EISSN 1735-3645.
DOI: 10.18869/acadpub.jafm.67.223.23359

Experimental Study of Heat Transfer Enhancement in a Heated Tube Caused by Wire-Coil and Rings

S. Vahidifar† and M. Kahrom

Department of Mechanical Engineering, School of Engineering, Ferdowsi University of Mashhad, Mashhad , Khorasan Razavi, P.O. Box No. 91775- 1111, IRAN.

†*Corresponding Author Email: S_Vahidifar@yahoo.com*

(Received June 14, 2014; accepted September 1, 2014)

ABSTRACT

This study investigates heat transfer characteristics and the pressure drop of a horizontal double pipe heat exchanger with wire coil inserts. The amplification of convection heat transfer coefficient in the heat exchanger reduces the weight, size and cost of heat exchanger. One way of augmenting the heat transfer is to disturb the boundary layer. When an object is placed in a boundary layer, it affects the flow structure and alters the velocity and thermal profiles. The change is affected by the formation of jets and wakes in the boundary layer as it alters modifies transfer and friction coefficients on the wall. This paper studies the characteristics of the heat transfer and the pressure drop of a double pipe horizontal tube heat exchanger with an inserted wire coil and rings. Wire coil acts as a swirl flow, which increases turbulence and roughness whereas rings increase heat transfer as a promoter of turbulence and roughness. The experimental data sets were extracted from wire coils and rings tested within a geometrical range with a pitch of (P/D=1, 2, 4) and wire diameter of (d/D=0.05, 0.07, 0.11). For wire coil with d/D=0.11, P/D =1 and Reynolds number of 10000, the overall enhancement efficiency amounted to 128%.

Keywords: Enhancement of boundary layer; Ring insert; Convection heat transfer coefficient; Heat exchanger; Wire coil insert.

NOMENCLATURE

A	surface area of test tube, m^2
C	constant resistance thermal, m^2.k/w
C_{pa}	specific heat capacity of air, J/kg K
D	diameter of the tube, m
d	wire coil diameter, m
f	friction factor
h	average heat transfer coefficient, W/m^2 K
k	thermal conductivity, W/m. K
kW	kilo watt
L	length of the test tube, m
$\dot{m}$	air mass flow rate, kg/s
Nu	Nusselt number
ΔP	pressure drop, Pa
P	coil pitch, m
Re	Reynolds number
Q	heat transfer rate, W
R	thermal resistance, m^2.k/w
RTD	resistance Temperature Detectors
T	temperature, K
ΔT	temperature difference, K
ΔTm	log mean temperature difference, K
U	overall heat transfer coefficient,W/m^2 K
V	mean velocity in tube, m/s
$\dot{V}$	volumetric flow rate, m^3/s
ρ	fluid density, kg/m^3
η	overall enhancement efficiency
μ	dynamic viscosity, kg/m.s

Subscripts

i	inlet, inner
o	outlet, outer
a	air side
w	water side
ob	obstacle
s	smooth
t	thermal
h	hydraulic diameter
wa	water to air
m	mean temperature difference
f	fluid
pp	pumping power
pa	constant pressure condition for air

1. INTRODUCTION

Laminar-turbulent transition is an important problem in many wall-bounded flows of both scientific and engineering interests (Noro et al. 2013). Heat transfer is widely used in many applications in heat exchangers, chemical process, gas and oil production, air condition, automotive or food industries. Modern engineering and technology is moving constantly upward on the temperature scale and subsequently the concept of enhancement of heat transfer has derived the attention of many scientists and engineers in the past few decades (Shailendhra et al. 2011).

A general consideration in designing a heat exchanger is to make it compact, i.e., increase the overall heat transfer coefficient. Another consideration is reducing the temperature of driving forces which enhances the second law efficiency and decreases entropy generation. Over the past decades, several methods have developed and extensively applied to heat exchangers.

In industrial processes, more than 80% of energy is transferred in heat exchangers, which highlights the importance of enhancing heat transfer (He et al. 2012).

Generally, enhancement techniques can be classified into three broad categories:

(a) Active methods: active augmentation, which is studied extensively, requires some external power to achieve the desired flow modification. Given the complexity of this method, is rarely used in design purpose. Furthermore, it is not easy to provide external power in many applications.

(b) Passive methods: this method does not rely on external power, but the additional power is derived from the power of the fluid motion. Tube insert devices, which consist of twisted tape, wire coil, extended surface and wire mesh inserts, are the most important techniques of this group among which twisted tape and wire coils are widely used.

(c) Compound method: It is a hybrid method which draws on both active and passive methods. It has a complex design which limits its applications (Jafari et al. 2010).

Since compound method uses passive method, no additional external power is required. For this reason, it is widely used in several heat transfer applications, like heat recovery process, air condition, refrigeration and dairy process.

Wire coil inserts is superior to other enhancement techniques for several reasons:

(1) Lower cost.
(2) Easy installation and removal.
(3) Preservation of original smooth tube mechanical strength.

(4) Possibility of installation in an existing smooth tube heat exchanger (Garcia and et al. 2005).

When a wire coil, a twisted tape or other means of swirling flow are inserted into a flow field, they induce swirls into the flow field and produces periodic redevelopment of the boundary layer, which increases the effective heat transfer area and the turbulence intensity. The swirl induces tangential flow velocity component, which improves the mixture of flow between the tube core and the nearby wall region (Gunes et al. 2010).

Also Naphon (2006) studied the heat transfer characteristics and the pressure drop of the horizontal tubes with wire coiled insert. He also examined the effect of coil pitch and relevant parameters on heat transfer characteristics and pressure drop, showing that wire coil insert had a significant effect on the enhancement of the heat transfer, especially in the laminar flow region. Promvonge (2008) reported the thermal performance of a tube with square cross-sectioned coiled wire, comparing his experimental results with the performance of a circular cross-sectioned wire. The results revealed that under similar conditions, the overall enhancement provided by the square coiled wire insert was better than the circular one. Additionally, Promvonge (2007) examined the effect of snail entry on thermal enhancement of a tube fitted with circular and square cross-sectioned coiled wire.

The effect of different wire coiled geometries on pressure drop during the condensation of R-134 vapor inside a horizontal tube was experimentally investigated by Akhavan-Behabadi et al. (2008).

In another study, Behabadi et al. (2010) carried out an experimental study on the enhancement of heat transfer coefficient by coiled wire inserts during the engine oil is heated inside a horizontal tube. Moreover, the effects of Reynolds number and the wire coiled geometry on the heat transfer augmentation and fanning friction factor were studied. Finally, two empirical correlations were developed to predict the heat transfer enhancement of these coiled wire inserts. These correlations predicted the experimental Nusselt number in error band of ±20 percent.

Saeedinia et al. (2012) did an experimental study to investigate the heat transfer and pressure drop characteristics of CuO/Base oil nanofluid laminar flow in a smooth tube with different wire coil inserts under constant heat flux. The nanofluid is prepared by the dispersion of CuO nanoparticles in the base oil and stabilized by an ultrasonic device. Finally, two empirical correlations are developed for predicting the Nusselt number and the friction factor of the nanofluid flow which is inside coiled wires inserted tubes. These correlations predict the experimental data in an error band of ±20%.

Garcia et al. (2007) carried out an extensive experimental study on three wire coils with different pitches inserted in a smooth tube with laminar and transition regimes. Also, isothermal pressure drop tests and heat transfer experiments under uniform heat flux conditions were performed, which showed that in Reynolds numbers below 200, wire coils did not enhance heat transfer relative to a

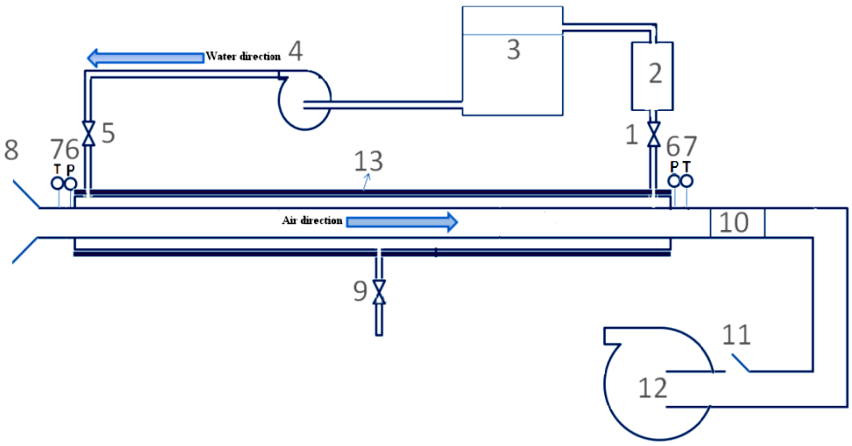

Fig. 1. Schematic of the experimental facility.

smooth tube. In Reynolds numbers between 200 and 1000, wire coils increase the heat transfer remarkably. In Reynolds numbers above Re = 1000-1300, the transition was from laminar to turbulent flow. In Reynolds number around 1000, wire coil inserts increase the heat transfer coefficient to eight times greater than the smooth tube.

Eiamsa-ard et al. (2012) experimentally studied the heat transfer and the pressure drop characteristics in a square duct fitted with tandem wire coil elements and tabulators in the turbulent regime. Therefore, the full-length coil was needed to be applied instead of tandem short-length coil to obtain higher heat transfer and performance, leading to more compact heat exchanger. The best operating regime for the wire coil tabulator was achieved in low Reynolds number where the thermal enhancement factor was about 1.33.

Also Eiamsa-ard et al. (2010) examined the heat transfer enhancement, friction factor and thermal performance factor in a tube equipped with combined devices inserted between the twisted tape and constant/periodically varying wire coil pitch ratio. The results of combined devices were compared with the result of each device separately, and the correlations of the Nusselt number and friction factor were developed for all studied parameters. Kim et al. (2008) studied the flow-induced vibration (FIV) in a two-phase flow with wire coil inserts at atmospheric pressure. The FIV correlation for a two phase flow with wire coil inserts was experimentally developed with a coefficient correlation of 0.956.

Kongkaitpaiboon et al. (2010) investigated convection heat transfer and pressure loss in a round tube fitted with circular rings, finding that depending on the operation conditions, the heat transfer rate was augmented by 57% to 195% compare to smooth tube. Akansu (2006) revealed that porous-ring insert in a tube on constant heat flux in Reynolds number 45000 had the maximum heat transfer rate for the specific pitch and thickness of the ring.

In this literature survey, most studies are generally focused on the effects of a specific variety of coil pitch and wire thickness with respect to wire coil and rings. In the present study, two different types of heat transfer enhancements, wire coil (swirl flow, turbulence promoter and roughness) and rings (turbulence promoter and roughness) are compared.

2. EXPERRIMENTAL APPARATUS AND SETUP

An experimental apparatus is used to study the heat transfer performance and friction factor in a tube with wire coil and rings inserts. The set-up consisted of a two-pipe heat exchanger and a measuring instrument. The schematic configuration of experimental equipments is shown in Fig. 1. it consists of two independent flow circuits.

The test tube with wire coil and rings insert is shown in Fig. 2. where P stands for helical and rings pitch d is wire coil and rings diameter and D is the tube inner diameter. Theses parameter can be arrayed in none-dimension form: dimensionless of wire diameter to pipe diameter (d/D) and pitch to wire diameter ratio (P/d).

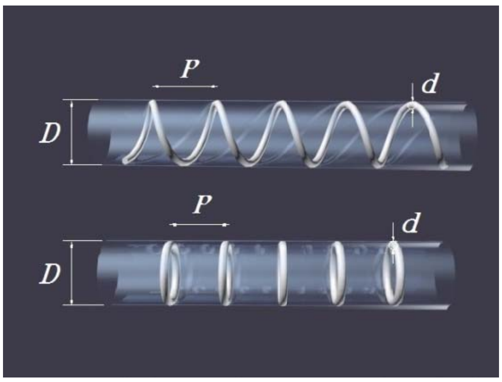

Fig. 2. Sketch of wire coiled and rings insert in smooth tube.

The wire coil was installed in the main circuit with flow of air with Pr=0.7. The secondary circuit used hot water to set the tank temperature to a constant value. Hot water was pumped from an open reservoir tank by a centrifugal pump. The air flow rate was measured by a turbine flow meter. The test section was a thin-walled, 165 cm long, steel tube with a wire coil and ring insert. The average inner (tube side) and outer (shell side) diameters of the tube exchanger were 7.6 cm and 12.7 cm, respectively.

Heat transfer experiments were carried out under constant temperature conditions. The warm water in the outer side was heated directly by the heater (2Kw). Electrical heater supplied and regulated warm water to 60°C by an electronic circuit. Test fluid inlet and outlet temperatures (Ti and To) were measured by submerged type RTDs (Resistance Temperature Detectors), calibrated with ± 0.2 °C deviation as compared to the post experiment condition. To measure the pressure drop in the air side, a low differential pressure transmitter (with a precision of ±1 pa) was employed.

A centrifugal fan (0.37kw) was used to convey air in the inner tube. To avoid the influence of obstacles as a Fin, a plastic layer was added to the wire coil and rings. Also, wire coil diameter and rings were 4, 6, 9 mm. Three different d/D ratios (d/D=0.05, 0.07, 0.11) and three different pitch ratios (P/D=1, 2 and 4) were considered in the experimental study. Air velocity in inner tube altered from 1 to 6 m/s as the air volume changed. Outer pipe was insulated by a 5cm thickness and the thermal conductivity of 0.034 W/m.K (glass wool) to minimize heat loses. Temperature differences were measured between the air inlet and outlet both with and without enhancement.

The uncertainties of experimental measurements were determined by the method introduced by Kline and McClintock (1953) in the present study, the highest uncertainties of Nusselt number, friction factor and Reynolds number were approximately ±7.9%, ±8.1% and ±5% respectively.

The Nusselt number and the friction factor were calculated for a smooth tube to validate the experimental method used prior to the experiments of the wire coil and rings inserts. The results of Nusselt number and friction factor for smooth tube were compared with the results of steady state flow correlations of Gnielinski (1976) and Petukhov (1970) obtained for the fully developed turbulent flow in circular tubes. Figs. 3. and 4.

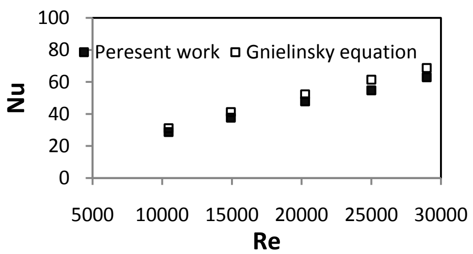

Fig. 3. Verification test of the smooth tube for Nusselt number.

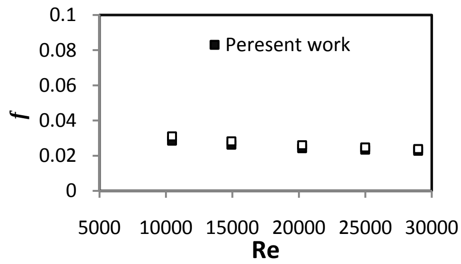

Fig. 4. Verification test of the smooth tube for friction factor.

show compare the results of the present smooth tube and the correlations of Gnielinski and Petukhov. As can be seen, there is a good agreement between the results for the present smooth tube and the correlations offered in the literature. These results verify the experimental set-up and measurement technique.

3. DATA AND ASSUMPTION

To calculate the air-side heat transfer coefficient, heat transfer correlations were used. It should be noted that the rate of heat loss from heat exchanger to ambient was negligible. For the air-side, we have:

$$Q_{\mathrm{a}} = \dot{m}c_{pa}\Delta T \tag{1}$$

Also, we have heat transfer between air and water:

$$Q_{wa} = UA_i\Delta T_m \tag{2}$$

Given that $Q_a = Q_{wa}$ we have:

$$U = \frac{\dot{m}c_{pa}\Delta T}{A_i\Delta T_m} \tag{3}$$

On the other hand:

$$U = \frac{1}{R_t} \tag{4}$$

With negligible of fouling factor for two fluid total thermal resistances equal:

$$R_{\mathrm{t}} = \frac{\mathrm{A_i}}{\mathrm{h_o A_o}} + \frac{\mathrm{A_i ln}\frac{\mathrm{D_o}}{\mathrm{D_i}}}{2\pi\mathrm{Lk}} + \frac{1}{\mathrm{h_i}} \tag{5}$$

To calculate the heat transfer coefficient of water-side relations, the reform Gnielinsky (2011) based on the hydraulic diameter $(\mathrm{D_h = D_O - D_i})$ was used and the total amount of calculations was kept constant. The constant C is defined as follows:

$$C = \frac{\mathrm{A_i}}{\mathrm{h_o A_o}} + \frac{\mathrm{A_i ln}\frac{\mathrm{D_o}}{\mathrm{D_i}}}{2\pi\mathrm{Lk}} \tag{6}$$

Thus, by combining Eq. (3), (4), (5), (6), the amount of heat transfer coefficient on the tube side will be:

$$\mathrm{h_i} = \frac{\dot{\mathrm{m}}\,\mathrm{C_{Pa}}\Delta\mathrm{T}}{(\mathrm{A_i}\Delta\mathrm{T_m} - \mathrm{C}\dot{\mathrm{m}}\mathrm{C_{Pa}}\Delta\mathrm{T})} \tag{7}$$

Also, we define the following dimensionless Nusselt number:

$$Nu = \frac{hD}{k_{\mathrm{f}}} \tag{8}$$

The Reynolds number:

$$Re = \frac{\rho VD}{\mu} \tag{9}$$

And the friction factor:

$$f = \frac{\Delta P}{\frac{1}{2}\frac{L}{D}\rho V^2} \quad (10)$$

According to constant pumping power evaluation criteria, we have (Webb 1981):

$$(\dot{V}\Delta P)_s = (\dot{V}\Delta P)_{ob} \quad (11)$$

Thus, the relationship between the friction factor and Reynolds number can be given as follows:

$$(f\ \mathrm{Re}^3)_s = (f\ \mathrm{Re}^3)_{ob} \quad (12)$$

The overall enhancement efficiency is expressed as the ratio of h_{ob}, That is, the ratio of an enhanced tube with wire coiled insert to the similar smooth tube, h_S at a constant pumping power is introduced by Webb (1981):

$$\eta = \left(\frac{h_{ob}}{h_s}\right)_{pp} = \left(\frac{Nu_{ob}}{Nu_s}\right)_{pp} = \left(\frac{Nu_{ob}}{Nu_s}\right)\left(\frac{f_s}{f_{ob}}\right)^{1/3} \quad (13)$$

4. EXPERIMENTAL RESULTS AND DISCUSSION

The variations of Nusselt number relative to Reynolds number for three different pitch ratios (P/D=1, 2 and 4) of the wire with d/D=0.11 are shown in Fig. 5. As can be seen, the heat transfer increases as the pitch ratio drops. The maximum heat transfer is achieved for the lowest pitch ratio (P/D=1) for rings. Given the turbulence intensity and flow pathway of this pitch ratio, it is greater and longer than other pitches.

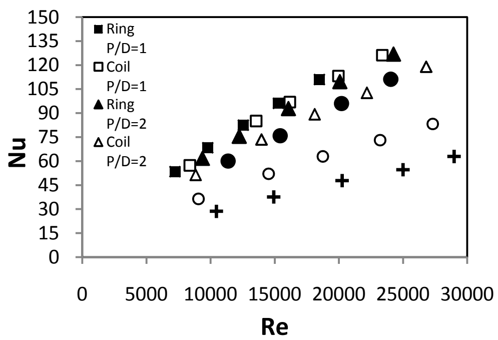

Fig. 5. Variation of Nusselt number with Reynolds number for different pitch ratios (d/D=0.11).

The Nusselt number increase for rings was in the range of 130-140%, 120-130% and 100-110% , which was more than that of the smooth tube depending on the values of Reynolds number for P/D=1, 2 and 4 respectively. The variation of friction factor relative to Reynolds number for three pitch ratios (P/D=1, 2 and 4) of the wire with d/D=0.11 is shown in Fig. 6.

Heat transfer ratio (Nu_{ob}/Nu_s) and Reynolds number are illustrated in Fig. 7. It must be noted that for a net energy gain, i.e., an effective heat transfer, the heat transfer ratio must be greater than unit.

As shown in Fig. 7. the heat transfer increase generated by wire coil and rings inserts is remarkable.

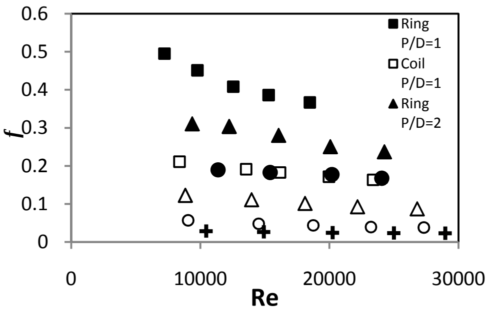

Fig. 6. Variation of friction factor with Reynolds number for different pitch ratios (d/D=0.11).

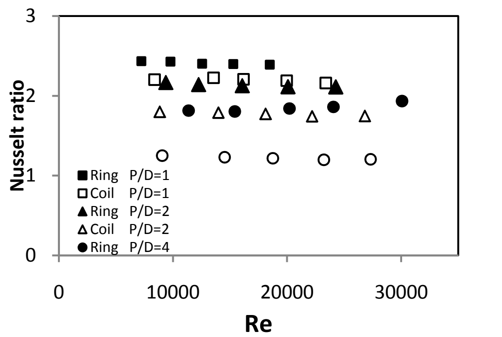

Fig. 7. Variation of Nusselt number ratio relative to Reynolds (d/D=0.11).

The Nusselt number ratio tends to decrease as the Reynolds number increases for all cases; however, the heat transfer ratio does not significantly change the value of Reynolds number. The heat transfer ratio rises with the decline of pitch ratio and the increase of wire thickness. According to Fig. 7. the rings with P/D=1 provides the maximum Nusselt ratio. This can be attributed to the reverse flow provided by the wire with d/D=0.11 and P/D=1, which improves the convection of the tube wall by mounting the mean velocity and temperature gradient and reducing the cross section of the flow field.

As shown in Fig. 7., for the same wire thickness, rings with P/D=2 are more effective in heat transfer than wire coils.

The friction factor decreases as the Reynolds number and pitch ratio rise. Low pitch ratio is equal to the increased number of coils. Thus, the utmost friction factor was observed for the case of P/D=1 because increased number of coils disturbed the entire flow field and produced more friction. In the rings of all pitch ratios, the friction factor increased significantly compared to the smooth tube because the upper surface area and the reverse flow led to the dissipation of dynamic pressure of the flow.

As seen in Fig. 8. mean increase in the friction factor of the wire coiled inserts with pitch ratios of P/D=1, 2 and 4 are approximately 6.9-7.2, 3.8-4.35

and 1.4-2.1 times greater than that of the smooth tube respectively. The mean increase in the friction factor of ring inserts with pitch ratios of P/D=1, 2 and 4 are approximately 14.5-17.5, 9.3-12 and 6.4-7.6 times greater than that of the plain tube, respectively.

The results of friction ratio for the whole set of wire coils and rings are shown in Fig. 8. Rings with d/D=0.11 have the maximum friction ratio (11.2-11.8). According to Fig. 8., the pressure drop for all case is higher than wire coils. When the ring diameter increases, a) the vortex and wake power increases and b) the cross section area of pipe reduces, the velocity increases and thus the pressure drop rises. By comparing the friction factor of d/D=0.11 for rings and wire coils, the amount of friction factor for ring will be almost twice the wire coil. Fig. 8. indicates that the amount of pressure drop for rings with d/D=0.11 and P/d=4 are equal to wire coil with d/D=0.11 and P/d=1.

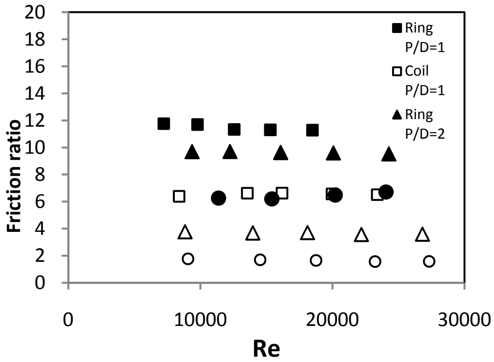

Fig. 8. Variation of friction factor ratio relative to Reynolds (d/D=0.11).

Heat transfer ratio (Nu_{ob}/Nu_s), friction factor penalty (f_{ob}/f_s) and overall enhancement efficiency $[(N_{ob}/Nu_s)/(f_s/f_{ob})^{1/3})]$ are used to evaluate the effect of wire coiled and rings inserts.

The variation of overall enhancement efficiency plotted against Reynolds number for all cases has been shown in Fig. 9. Thus, a performance study is necessary to evaluate the net energy gain required for making a conclusion. If the technique used for increasing the heat transfer is efficient in terms of energy, then the enhancement technique will be optimal. The pumping power has been kept constant in the comparison to determine the net gain. As shown in this figure, wire coiled insert is more valuable than the smooth tube. The overall enhancement efficiency tends to decrease with any variation in the Reynolds number, which greater than 1 for all Reynolds numbers. It is obvious from Fig. 9. that wire coils with P/D=1 and P/D=2 have respectively the maximum efficiency in terms of heat transfer and friction, rings with P/D=1, given the lack of swirl fluid flow, are come after them. As P/D increases, the influence of the swirl flow decreases to the extent that almost no difference is observed in the overall enhancement between wire coil and rings of P/D=4. In this case, the overall enhancement efficiency increase is caused by the turbulence promoter and roughness. In P/D=1, 2 for wire coil, there are some fluid flow phenomena such as swirl, turbulence promoter and roughness. Also, in P/D=1, 2 for rings, there are fluid flow phenomena such as turbulence promoter and roughness.

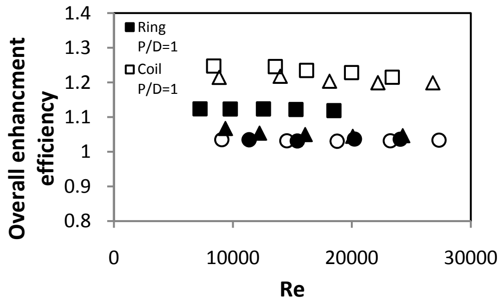

Fig. 9. Variation of overall heat transfer efficiency relative to Reynolds numbers (d/D=0.11).

Figs. 10. and 11. show the effect of d/D on the heat transfer and friction factor ratio under turbulent flow conditions for wire coil and rings in a constant pitch ratio (P/D=1), It is obvious that the heat transfer rate and friction factor provided by the tube with wire coil and rings is higher than that of smooth tubes. Rings tabulators disturb the development of the boundary layer of the fluid flow and increase the degree of flow turbulence. Obviously, rings provide higher heat transfer than wire coils because the turbulence intensity and the wake derived from rings are greater and longer than that of wire coil.

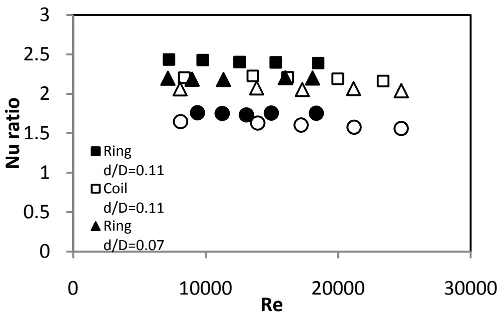

Fig. 10. Variation of Nusselt ratio relative to Reynolds number for different d/D ratios (P/D=1).

Nusselt ratio decreases as the Reynolds number increases, which in optimal condition is 2.48 times greater than the smooth tube. Under this condition, the maximum value for wire coil is 2.18 times greater than the plain tube. Nusselt ratio rises as thickness of wire and a ring is increase. This can be attributed to the stronger turbulence intensity, increased heat transfer area and higher mixing induced by larger wires as compared to that of small wires. The variation of Nusselt number relative to the Reynolds number for different wire diameter has been shown in Fig.10. According to Fig. 11, the Nusselt number ratio for the wire coils with d/D=0.11 is approximately 2.16-2.2.times greater than the smooth tube, depending on Reynolds number. For d/D=0.07 and 0.05, the Nusselt ratios of 2 and 1.56-1.64 have been reported respectively. As shown in Fig. 10, the Nusselt number ratio for the ring with d/D=0. 11 are approximately 2.3-2.4.times greater than the plain tube depending on Reynolds number. Nusselt ratio for d/D=0.07 and 0.05 are calculated 2.19 and 1.75, respectively. The friction factor decreases as the Reynolds number increases, as presented in Fig. 11. It is obvious that smaller wires (d/D=0.05 and d/D=0.07) alleviate the pressure drop. The friction factor for wire coils with d/D=0.11, 0.07 and 0.05 are averagely 6.3-6.5, 5.1-5.2 and 1.5-1.7 times greater than those of the smooth tube respectively.

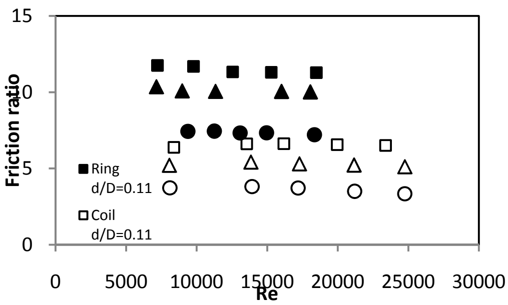

Fig. 11. Variation of friction ratio relative to Reynolds number for different d/D ratios (P/D=1).

According to Fig. 11., the friction ratios of rings for wires with d/D=0.11, 0.07 and 0.05 are respectively 11.2-11.8, 9.5-9.7 and 6.2-6.9.

In Fig. 12, the variation of overall enhancement efficiency is plotted against Reynolds number. Heat transfer enhancement is provided by the rising friction loss caused by the rings and wire coiled inserts. Thus, a performance study is necessary to evaluate the net energy gain to make a decision. The comparison is made based on the same pumping power to determine the net ultimate gain. For all case, the experimental data are compared by keeping constant the pumping power.

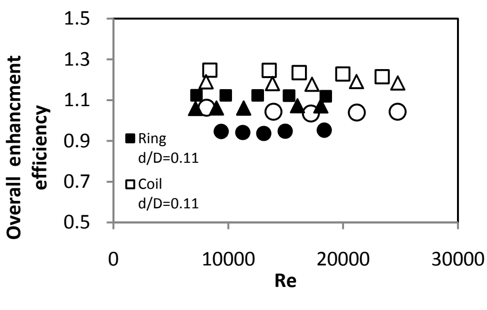

Fig. 12. Variation of overall heat transfer efficiency relative to Reynolds number.

As shown in Fig. 12, the enhancement factors are generally above unity. Wire coils with d/D=0.11 and P/d=1 has the maximum efficiency. Overall, the maximum enhancement is 128%. Wire coil with d/D=0.07 and P/d=1 and ring with d/D=0.11 and P/d=1 are in the second and the third order in terms of maximum efficiency. The overall enhancement efficiency for ring and wire coils with d/D=0.07 and d/D=0.05 are equal. The overall enhancement efficiency for Ring with d/D=0.05 is less than unit because the friction factor is more effective than the heat transfer.

5. CONCLUSION

The present experimental study focused on the analysis of the heat transfer and pressure drop between the wire coil and rings insert in smooth tube with Re=5000-25000 and Pr=0.7.

The circular cross-sectioned wire coil and rings were inserted in the tube. Wire coil and rings inserts tube produced remarkable increase in both heat transfer and pressure drop in comparison with the smooth tube relative to the pitches and wire thickness.

Wire coil acts as a swirl flow, with rotating flow being superimposed upon the central core flow, which generates centrifugal force. In most liquids in which the density is reduced relative to the temperature, the centrifugal force transfers the heated fluid from the boundary layer towards the tube axis, which increases the heat transfer. Rings acts as a turbulence promoter with the flow turbulence level being amplified by the separation and reattachment mechanism. Besides, when rings are in contact with the tube wall, they act as roughness elements and disturb the existing laminar sub layer.

Depending on the flow condition, wire geometry and pitch, the heat transfer rate increases by one or both of the earlier mechanisms. Nusselt ratio for rings 2.3-2.4 over plain tube is obtained. However, the overall enhancement efficiency of 128% is reported for rings in best condition. With respect to the heat transfer and friction, wire coils display superior performance than rings.

ACKNOWLEDGEMENTS

The author would like to express his gratitude to the faculty of Engineering of Ferdowsi University of Mashhad for their support during this research.

REFERENCES

Akansu, S. O. (2006). Heat transfers and pressure drops for porous-ring tabulators in a circular pipe. *International journal Applied Energy* 83, 280 - 298.

Akhavan-Behabadi, M. A., M. R. Salimpour and V. A. Pazouki (2008). Pressure drop increase of forced convective condensation inside horizontal coiled wire inserted tubes. *International Communications in Heat and Mass Transfer* 35, 1220–1226.

Akhavan-Behabadi, M. A., R. Kumar, M. R. Salimpour and R. Azimi (2010). Pressure drop and heat transfer augmentation due to coiled wire inserts during laminar flow of oil inside a horizontal tube. *International Journal of Thermal Sciences* 49, 373–379.

Eiamsa-ard, S., N. Koolnapadol and P. Promvonge (2012). Heat transfer behaviors in a square duct with tandem wire coil element insert. *Fluid Dynamics and Transport Phenomena Chinese Journal of Chemical Engineering* 20,.863 – 869.

Eiamsa-ard, S., P. Knivesrangsan, S. Chokphoemphun and P. Promvonge (2010). Influence of combined non-uniform wire coil and twisted tape inserts on thermal performance characteristics. *International Communications in Heat and Mass Transfer* 37, 850–856.

Garcia, A., J. P. Solano, P. G. Vicente and A. Viedma (2007). Enhancement of laminar and transitional flow heat transfer in tubes by means of wire coil inserts. *International Journal of Heat and Mass Transfer* 50, 3176–3189.

Garcia, A., P. G. Vicente and A. Viedma (2005). Experimental study of heat transfer enhancement with wire coil inserts in laminar-transition-turbulent regimes at different prandtl numbers. *International journal of Heat and Mass Transfer* 48, 4640–4651.

Gnielinski, V. (1976). New equations for heat and mass transfer in turbulent pipe flow and channel flow. *International Chemical Engineering*, 16, 359–368.

Gunes, S., V. Ozceyhan and O. Buyukalaca (2010). Heat transfer enhancement in a tube with equilateral triangle cross sectioned coiled wire inserts. *Experimental Thermal and Fluid Science* 34, 684–691.

He, Y., H. Zheng, Q. Zhang, Y. Tang and J. Zhang (2012). Analysis of enhanced heat transfer performance through Duct with Constant wall temperature. *Journal of Thermodynamics and Heat transfer* 26(3), 472 – 475.

Incropera, F., P. D. Dewitt (2011). *Introduction to Heat Transfers* 17rd ed. John Wiley& Sons.

Jafari Nasr, M. R., A. Habibi Khalaj and S. H. Mozaffari (2010). Modeling of heat transfer enhancement by wire coil inserts using artificial neural network analysis. *Journal of Applied Thermal Engineering* 30, 143–151.

Kim, D. H, S. H. Chang (2008). Flow-induced vibration in two-phase flow with wire coil inserts. *International Journal of Multiphase Flow* 34, 325 – 332.

Kline, S. J. and F. A. McClintock (1953). Describing uncertainties in single sample. *Experiments, Mechanical Engineering* 75, 385–387.

Kongkaitpaiboon, V., K. Nanan and S. Eiamsa-ard (2010). Experimental investigation of convective heat transfer and pressure loss in a round tube fitted with circular-ring tabulators. *International Communications in Heat and Mass Transfer* 37, 568 – 574.

Naphon, P. (2006). Effect of coil-wire insert on heat transfer enhancement and pressure drop of the horizontal concentric tubes. I*nternational Communications in Heat and Mass Transfer* 33, 753–763.

Noro, S., Y. Suzuki, M. Shigeta, S. Izawa and Y. Fukunishi (2013). Boundary Layer Receptivity to Localized Disturbances in Freestream Caused by a Vortex Ring Collision. *Journal of Applied Fluid Mechanics* 6(3), 425-433.

Petukhov, B. S. (1970). Heat transfer in turbulent pipe flow with variable physical properties. *Advances in Heat Transfer* 6, Academic Press, New York.

Promvonge, P. (2008). Thermal performance in circular tube fitted with coiled square wires. *Energy Conversion and Management* 49, 980–987.

Promvonge, P. (2008). Thermal enhancement in a round tube with snail entry and coiled-wire inserts. *International Communications in Heat and Mass Transfer* 35, 623–629.

Saeedinia, M., M. A., Akhavan-Behabadi and M., Nasr (2012). Experimental study on heat transfer and pressure drop of nanofluid flow in a horizontal coiled wire inserted tube under constant heat flux. *Experimental Thermal and Fluid Science* 36, 158–168.

Shailendhra, K. and S. P. AnjaliDevi (2011). On the Enhanced Heat Transfer in the Oscillatory Flow of Liquid Metals, *Journal of Applied Fluid Mechanics* 4, 57–62.

Webb, R. L. (1981). Performance evaluation criteria for use of enhanced heat transfer surfaces in heat exchanger design. International Journal of. Heat and Mass Transfer 24, 715–726.