HEAT TRANSFER ENHANCEMENT BY USING HOLED-PIN FINS IN A CONCENTRIC HEAT EXCHANGER

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ABSTRACT

The purpose of this investigation is to analyze the effects of the holed-pin fins which are located in the entrance of a concentric heat exchanger on heat transfer and pressure loss. The outer surface of the inner tube is maintained at constant temperature (~100 °C) by continuous contact with saturated water vapor introduced and filled into the annular space between the inner and outer tubes and discharged by a channel, like a teapot. The surface temperatures of the inner pipe, inlet and outlet temperatures of the air and pressure loss through the pipe are measured. The measurements are carried out for the Reynolds number ranged from 6000 to 12000. The design parameters are considered as the distance between the turbulators (s = 20, 30, 40 mm), hole diameter that is drilled on the pin fins (d = 6, 9, 12 mm) and the positions of the holes (top of the plate; namely *Type T* and center of the plate; namely *Type C*). Nusselt number and friction factor are found out based on the measurements. The comparison between the results of tabulated pipe and empty pipe, by means of Nusselt number and friction factor are done in order to show the effect of the turbulators in the concentric tube.

INTRODUCTION

In present study, arrays of rectangle plate fins with holes on each of them are inserted in a concentric tube which is used to enhance the heat transfer. As know, the pin fin inserted tubes have been used as one of the passive heat transfer enhancement techniques and are the most widely used tubes in several heat transfer applications. Before introducing the tabulated heat exchanger system in the present study a short blink to the literature will help to understand the effects of the usage of the fins on heat transfer and pressure drop.

Some experimental results for local heat transfer coefficients are presented for a converging channel with rib roughening elements with the cross-section being maintained square from inlet to exit by Abraham and Vedula [1]. Heat transfer and pressure loss in an air to water double pipe heat exchanger are experimentally investigated by Sheikholeslami et al [2].

NOMENCLATURE

A	[m^2]	Heat transfer surface area of the inner tube
C_P	[J/kg K]	Specific heat
D	[m]	Inner diameter of the inner pipe
d	[m]	Diameter of the hole on the plate
f	[-]	Friction factor
h_m	[W/m^{2}K]	Average heat transfer coefficient
k	[W/m K]	Thermal conductivity of air
$LMTD$	[K]	Logarithmic mean temperature difference
L	[m]	Length of the inner tube
Nu	[-]	Nusselt number
ΔP	[Pa]	Pressure loss
Pr	[-]	Prandtl number
Re	[-]	Reynolds number
s	[m]	Distance between two plates
ΔT_1	[K]	Difference of average wall temperature and inlet temperature
ΔT_2	[K]	Difference of average wall temperature and outlet temperature
T_w	[K]	Average surface temperature of the inner tube
T_o	[K]	Outlet temperature of the air
T_i	[K]	Inlet temperature of the air
V	[m/s]	Velocity of the air at the inlet

Special characters

ν	[m^2/s]	Kinematic viscosity of the air
ρ	[kg/m^3]	Density of the air
$\dot{V}$	[m^3/s]	Volumetric flow rate of air

Subscripts

i	inlet
o	outlet
m	mean
w	wall

A circular-ring and perforated circular-ring turbulators are placed in annular pipe. Results indicated that using perforated circular-ring turbulators lead to obtain lower heat transfer enhancement than the circular-ring turbulators. An experimental and numerical work are carried out to study the heat transfer enhancement in a heat exchanger square-duct fitted with oblique horseshoe baffles, by Skullong et al [3].

Promvonge [4] has studied the increase in heat transfer rate using conical ring with three different diameter ratios and placed with three different arrangements.

El Sayed et al [5] has investigated the effects of height, thickness, inter-fin spaces, number and tip-shroud clearance of fins on the heat transfer, fluid flow and pressure drop. Naik et al. [6] has proposed a design correlation which shows the distribution of optimal rib spacing for a wide range of rib geometries and operational conditions. Sahin et al [7] has searched the effects of the longitudinal and lateral separations of consecutively enlarged-contracted arranged fin pairs, widths of the fins, angle of attack, heights of fins and flow velocity on the heat and pressure drop characteristics by using the Taguchi method.

Arulprakasajothi et al [8] has carried out some analysis to show the effect of circular tube with conical strip inserts as turbulators in a laminar flow condition, using staggered and non-staggered conical strips. Cakmak and Yildiz [9] have performed a study to show the influence of the injectors with swirling flow generating on the heat transfer in the concentric heat exchanger.

In present study, the diameter of the holes on the rectangle plate fins which are in placed in an order, the distance between the plates, the location of the hole on the plate and Reynolds number have been the investigated parametric values. Heat transfer by means of Nusselt number and pressure drop by means of friction factor are the results of the experimental study.

EXPERIMENTAL SETUP

The main structure of the experimental setup was already explained in previously published papers of Eren et al [10, 11]. Schematic view of the setup is presented in Figure 1. The air is blowed by a fan and its speed is regulated by an inverter. An anemometer was used for measuring the velocity of the air. Pressure differences between the inlet and outlet of the exchanger are measured by an inclined manometer.

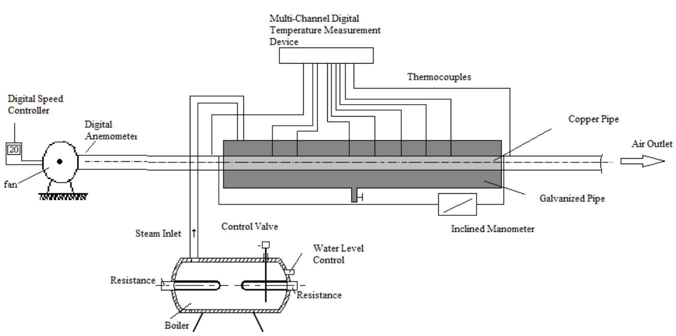
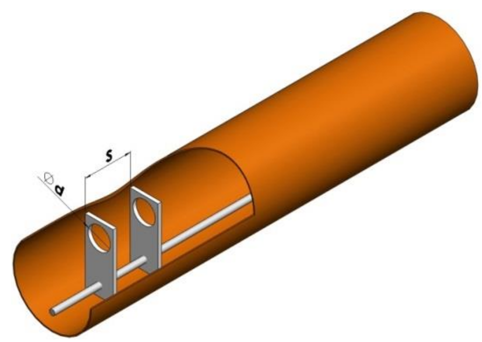


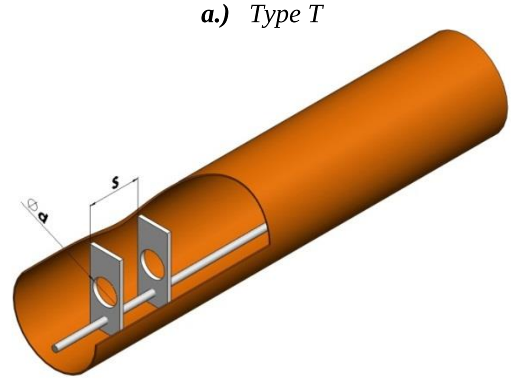
Figure 1 The schematic view of the setup

The heat exchanger is a classical type concentric tube of which outer surface of the inner tube is heated by the condensation of a stream of flowing water vapor. The inner and outer diameters of the inner tube are respectively 65 and 66 mm and the length of the tube is 900 mm. As an outer tube, a 1 mm-thickness of flat plate with dimensions of 700x1200 mm is rolled to obtain a circular tube with a diameter of 210 mm [10, 11].

The outer surface of the inner tube was kept at a constant temperature (100 °C) by continuous contact with saturated water vapor introduced and filled into the annular space between the inner and outer tube and discharged by a channel. The surface temperatures were measured by T-type thermocouples located every 50 mm distance along the tube length. The mean local wall temperature was determined by means of calculations based on the readings of the thermocouples. The inner and outer temperatures of air were also measured at certain points inside the tube. The pressure loss was maintained by an inclined manometer [10, 11].
The turbulators used in the experiments were illustrated in Figure 2. As seen from the figure rectangle plates each of 58x30 mm in dimension are located in the inner tube with a varied distance *s* = 20, 30, 40 mm are used in the experiments. On each plate a hole is drilled which has *d* = 6, 9, 12 mm diameter. The hole positioned on the plate in two ways; on top (*Type T*) and on the center (*Type C*).



a.) *Type T*



b.) *Type C*

Figure 2 Plate fins in the inner tube of the heat exchanger

DATA REDUCTION

All of the experiments are performed under the steady state conditions. Each run takes approximately 30 min. Heat transfer and pressure loss data are parameterized with the Reynolds number which is varied from 6000 to 12000. Re number is found as follows:

$$\mathrm{Re} = \frac{VD}{\upsilon} \tag{1}$$

where V is the velocity of the air at the inlet, ν is kinematic viscosity of the air, D is the inner diameter of inner tube.

The average Nusselt number is found by logarithmic mean temperature difference:

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \quad (2)$$

where ΔT_1 is the difference between mean wall temperature and inlet temperature of the air ($\Delta T_1 = T_w - T_i$), and ΔT_2 is the difference between mean wall temperature and outlet temperature of the air ($\Delta T_2 = T_w - T_o$). The mean wall temperature T_w was determined by means of calculations based on the readings of the thermocouples [10, 11].

The net heat is obtained by subtracting the heat loss from the gained heat occurred because of the water vapor on the surface. The outer surface of the test tube was well insulated and necessary precautions were taken to prevent leakages from the system. Thus, the convection heat transfer to the surrounding is neglected. Also the conductive heat loss through the thickness of copper tube, and radiation between the air and copper surface is neglected. The mentioned neglected heat rates are less than 0.5% of the total heat obtained by heating the surface by water vapor [10, 11].

Net heat is equal to the heat transfer rate occurred by logarithmic mean temperature difference, *LMTD*:

$$\dot{V}\rho C_p\left(T_o - T_i\right) = h_m A(LMTD)$$

$$h_m = \frac{(\dot{V}\rho)C_p(T_o - T_i)}{(\pi DL)LMTD} \quad (3)$$

where the fluid properties C_p, ρ and k are found based on bulk temperature of the air.

Then average Nusselt number can be calculated as follows:

$$Nu = \frac{h_m D}{k} \quad (4)$$

The measured pressure loss is used to calculate the friction factor:

$$\Delta P = f\frac{L}{D}\rho\frac{V^2}{2} \quad (5)$$

In order to compare the experimental results of the turbulated tube, a smooth tube flow is also tested. In addition, the well-known empirical formula of Dittus-Boelter and Blasius are used for comparisons [12].

$$Nu = 0.023\times(\mathrm{Re})^{0.8}\times(\mathrm{Pr})^{0.4} \quad (6)$$

$$f = 0.316\,\mathrm{Re}^{-0.25} \qquad (\mathrm{Re} \le 2x10^4) \quad (7)$$

where Prandtl number of the air is considered based on the average of inlet and outlet temperatures.

The well-known uncertainty analysis method introduced by Kelvin and McClintock [13] is used to estimate the total uncertainty of the parametric values; *Nu*, *Re* and *f*. The uncertainty of each variable is estimated by the help of instruments' catalogs. Then the total uncertainties for each independent variable were found for *Nu*, *f* and *Re* as **±13.75 %, ±10.1 % and ±3.5%** respectively.

RESULTS AND DISCUSSIONS

Always higher heat transfer with lower pressure loss is expected to be found in a heat exchanger application. As an intro, it can be said that, according to the results of the present study same result is obtained. The results of the tests are presented below.

The presentation begins with a display of the validation of the results for an empty smooth tube. The average Nusselt number for the smooth tube is experimentally obtained by using the equations from Eq. (2) to Eq. (4), and then the findings are compared to the results predicted by Dittus-Boelter empirical formula; Eq. (6). The maximum and minimum deviations between two results are respectively 16.5% and 2.8%.

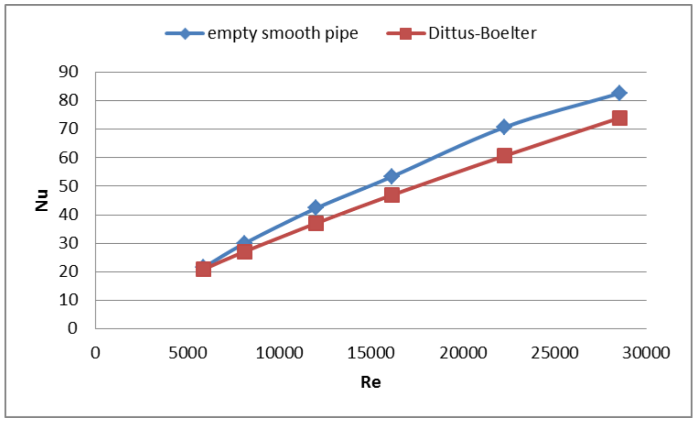


Figure 3 Nu number in the case of empty pipe

Similar graphical presentation is performed in Figure 4 for the friction factor. The results of the empty smooth pipe are compared to the results predicted from Blasius equation (Eq. 7). A good agreement is obtained as seen from the figure.

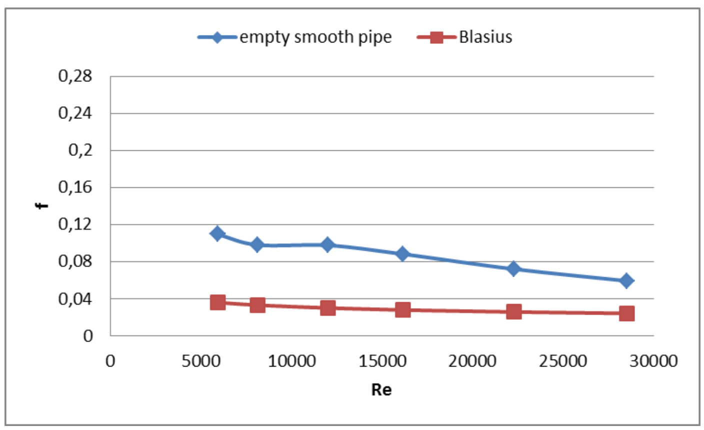


Figure 4 Friction factor in the case of empty pipe

The design parameters in the present study are the hole diameter drilled on the plate (*d*), space between the plates (*s*), the location of the hole on the plate (top and center, namely *Type T* and *Type C*). The effects of all mentioned parameters on heat transfer are presented in Figure 5, 6 and 7 respectively. It is evident from the Figure 5 that, *Nu* number increases while the diameter of the hole decreases. Increasing the distance between two plates results in augmentation of the heat transfer, as seen in Figure 6. Higher heat transfer occurs in the case of large distances. It can be observed from Figure 7 that, the location of the hole on the plate does not affect the heat transfer very much.

For all cases *Nu* number increases with increasing *Re* number, since the holed-pin fin turbulators interrupt the development of the boundary layer of the fluid flow and increase the degree of flow turbulence.

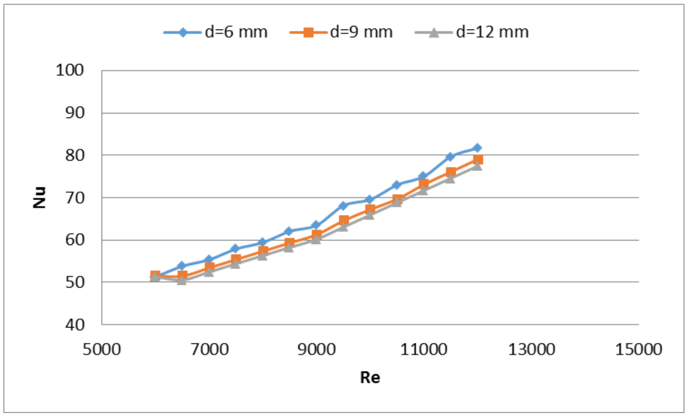


Figure 5 *Nu* versus *Re* with respect to hole diameter

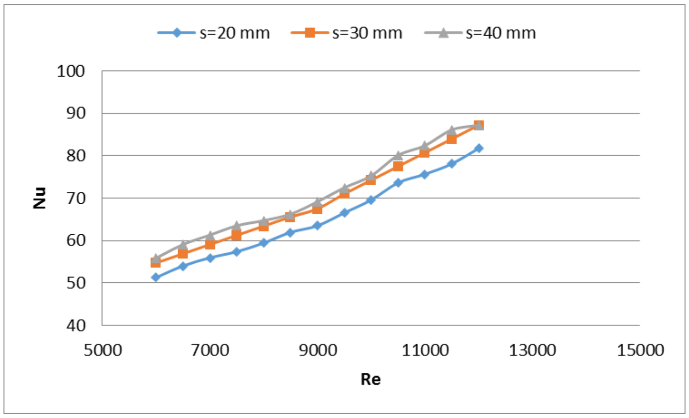


Figure 6 *Nu* versus *Re* with respect to plate distances

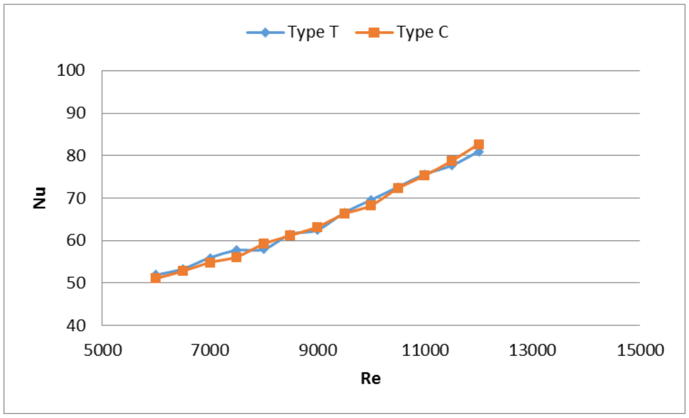


Figure 7 *Nu* versus *Re* with respect to location of hole on the plate

All turbulators used in the experiments cause some enhancement in *Nu* number meanwhile friction factor. In order to see the effects of those parameters on the friction factor, Figure 8, 9 and 10 has been drawn. In Figure 8, decreasing of hole diameter results in increasing of friction factor. Similarly in Figure 9, higher friction factor has been observed with higher distance. The effect of the location type on the friction factor is presented in Figure 10, and it is seen that for *Type T* the values are quite greater than the values for *Type C*.

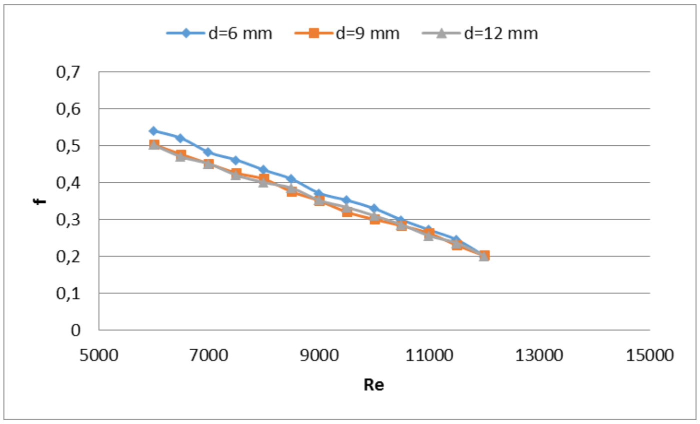


Figure 8 Friction factor versus *Re* with respect to hole diameter

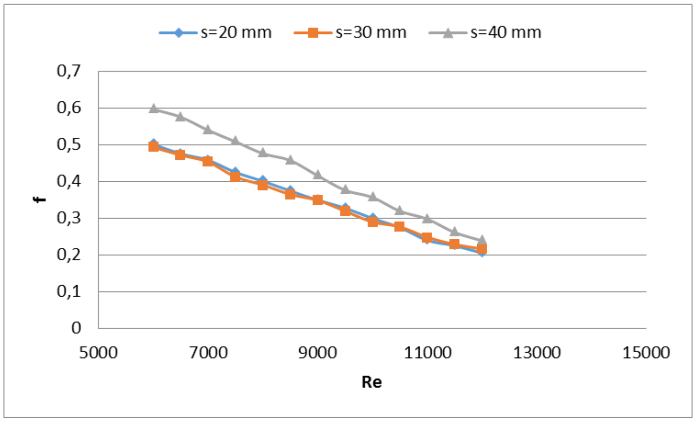


Figure 9 Friction factor versus *Re* with respect to various plate distances

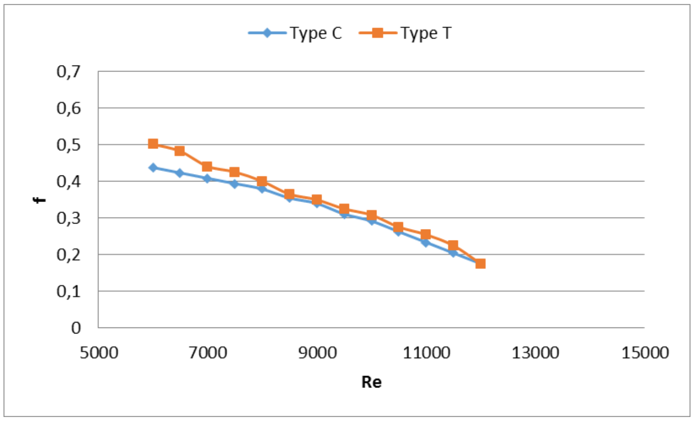


Figure 10 Friction factor versus *Re* with respect to location of hole on the plate

As mentioned above turbulators used in all cases cause an increment of friction factor, comparatively to the empty smooth pipe. Those increments can be attributed to the dissipation of the dynamic pressure of the fluid due to higher surface area and flow blockage of the holed-pin fin turbulators along the tube wall.

The friction factor decreases with increasing *Re* number, because friction factor inversely changes to the square of the air velocity.

CONCLUSIONS

In present study, plates with holes on them are located at the entrance of the inner tube of a heat exchanger. These plates behave like turbulators, and hence cause some heat transfer enhancement. Major results of the experiments are as follows:

- ✓ Turbulators not only produce more turbulence than the smooth tube but also increase heat transfer area.
- ✓ Highest heat transfer and friction factor are observed for the smallest diameter of the holes on the plates.
- ✓ Highest heat transfer and friction factor are observed for the highest distance between two plates.
- ✓ The location of the hole seems has no great importance on either heat transfer or friction factor.
- ✓ Heat transfer increases with increasing *Re*, however friction factor decreases with increasing *Re*.

The recommendation of the authors is to make an optimization analysis for diminishing the pressure loss.

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