EXPERIMENTAL STUDY OF THE AUGMENTATION OF CONVECTION HEAT TRANSFER IN HORIZONTAL CIRCULAR TUBE USING TRIPPING WIRES INSERTS.

دراسة معملية لزيادة انتقال الحرارة بالحمل في أنبوبة أفقية
دائرية المقطع باستخدام سلك حلزوني داخلها

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الخلاصة ـ يتضمن هذا البحث دراسة عملية لانتقال الحرارة بالحمل الجبري ومفاقيد الضغط الناتج عن الاحتكاك لتيار هوائي يمر داخل أنبوبة نحاسية أفقية دائرية المقطع قطرها ٣ر١٣ / ٢٢مم داخلها ملف حلزوني من السلك قطر ٤ر١ مم بحيث تتغير قيمة زاوية الحلزون من ٢٦ر٤ : ٧٨ر٣٠ (عدد لفات الحلزون من ٧١٤ : ٨٩ لفة / متر) . السطح الخارجي للماسورة النحاسية يسخن باستخدام بخار تحت الضغط الجوى بدرجة تحميص من ٢ : ٣°م وتتغير سرعة الهواء داخل الماسورة لبعطى رقم رينولدز من ٦٠ : ٤٠٠٠ . لوحظ من النتائج أن الزيادة في معامل انتقال الحرارة تصل الى ٥٧٪ وقد أوضحت التجارب أيضا أن معامل انتقال الحرارة بالحمل دالة في رقم رينولدز في حين أن معامل الاحتكاك دالة في رقم رينولدز وزاوية الحلزون . وقد صيغت النتائج في معادلات تجريبية لامكانية حساب معاملي انتقال الحرارة والاحتكاك .

ABSTRACT- The work summarizes the results of the experimental investigation of convective heat transfer and pressure loss characteristics of air flow in horizontal circular copper tube of diameters 13 3/22 mm. The coil wire (with diameter of 1.4 mm) was inserted in the tested tube. The helix angle changed from 4.26° to 30.78°. The outer surface of the tube was heated by a little superheated steam (2-3°C) at one atmosphere. During the experimental work, Reynolds number changed from 60 to 4000 and Prandtl number was about 0.7. The enhancement of heat transfer for the tested tube with internal coils was higher than that of the bare tube by about 57%. However, a quantitevely increase in hydrodynamic resistance was noticed. Correlation of Nusslet number and friction coefficient has been made. It was found that the Nusslet number was strongly dependent nn Reynolds number, wherase the friction coefficient is dependant on helix angle as well as Reynolds number.

INTRODUCTION

The need for more efficient and compact heat exchanger devices leads to the development of a variety of unconventional internal flow passages to enhance the heat transfer coefficient. This is because of the availability of the world's limited material and energy resources and the ever increasing cost of energy over the past few years accelerated the research in the field of conversion of raw materials and reduction in energy usage for a given process. Use of rough surfaces is one of the several enhacement techniques reported by Bergles [1], Dipprey and Sabersky [2] and Nikuradse [3], through which it is possible to achieve a two fold objectives of obtaining the maximum heat transfer rate with a minimum frictional pressure drop. These devices can be employed either to increase the heat transfer rate or to reduce the pumping power or heat transfer area. Considerable work has been also reported on turbulence promoters, such as transverse rib-roughened tubes [4-6], spirally corrugated tubes [7-8] and converging diverging tubes [9-10]. However, very limited work has been published on the thermohydraulic performance of helical wire inserted tubes, especially for convective heat transfer applications. Sethumadhavan and Rao [11] investigated experimentally the heat transfer in a 25 mm inside diameter copper tube tightly fitted with helical-wire-coil inserts of varying pitch (p), helix angle (β) and wire diameter (e). They correlated their results in an imperical formula and they made an optimization study on the basis of maximization of the heat transfer rate and also minimization of pumping power and heat exchanger frontal area to identify the most efficient tube within the matrix of data. Sethumadhavan and Rao [11] study gives a considerable increase in heat transfer rate without a significant increase in friction power, as these tubes produce some helical flow at the periphery of flow, superimposed upon the main axial flow, and thus influence the velocity distribution, the turbulence level and the turbulent wall shear.The working fluid in their study was water and 50% glycerol and the heating miduim was water. Nag and Rao [12] studied the friction and heat transfer performance and developed suitable correlations for momentum and heat transfer roughness functions, based on friction and heat transfer similarity laws. The working fluid in their study was R-12 and the test tube was heated electrically

The present work was taken up, therefore, with the possible aim of achieving better heat transfer results in a gas to gas condenser by swirling the flow inside it. This study summerized the results of the experimental investigation of convective heat transfer and pressure loss of air flow in a horizontal circular copper tube of diameter 13.5/22 mm. the coil wire of 1.4 mm diameter is inserted inside the test tube and adjacent to the inner surface. The outer surface of the tube was heated by a steam at the atmospheric pressure superheated by 2-3°C in this study the ratio between the coil pitch and its diameter (p/d) varying from 1.0 to 8.0, Reynolds number changing from 60 to 4000 and Prandtl number was about 0.7 and the tube length is 980 mm.

EXPERIMENTAL TEST RIG AND OPERATING PROCEDURE

This study concerns the determination of heat transfer rates and friction losses of ten helical-wire coils for a wire diameter 1.4 mm and for ten ptich to wire diameter, p/d=1, 1.2, 1.4, 1.6, 1.8, 2.0 ,3.0,4.0,6.0and 8.0 samples of which are shown in fig. (1).

One smooth tube was used to standardize the experimental set-up and also to evaluate the increase in the friction factor and the tube side heat transfer coefficient in ten helical wire coils inserted in the copper tube, relative to a smooth tube

The characteristic parameters, which define the roughness geometry of the ten helical wire-coils inserted in the copper tube are given in Table 1

Table 1 : characteristics of helical coils

P mm	1.4	1.68	1.96	2.24	2.52	2.8	4.2	5.6	8.4	11.2
p/d	1	1.2	1.4	1.6	1.8	2	3	4	6	8
β°	4.26	5.11	5.95	6.79	7.63	8.47	12.59	16.59	24.08	30.78
T /m	714	595	510	446	397	357	238	178	119	89
Data symbol	○	●	△	▲	□	■	◇	◆	▽	▼

Figure (2) shows a schematic diagram of the experimental test-rig. The actual test section consisted of a 980 mm long double pipe heat exchanger, the inner tube of which was either the smooth tube or the same with one of the inserted helical-wire-coils under test. The outer tube of the test rig was 50 mm inside diameter galvanized iron

Fig. (I). Sample of the used augmentation coils.

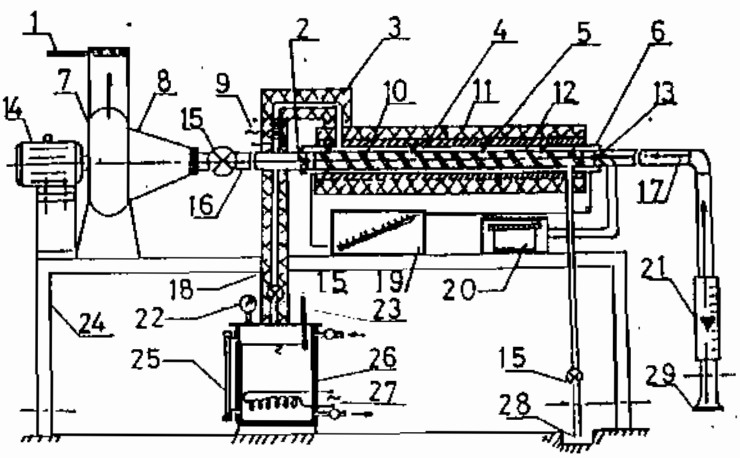

Fig.(2). Experimental Test Rig.

1- Air flow gate, 2,5 and 13 - Copper constantan thermocouples , 3 Thermal insulation, 4 - Heat exchanger, 6 - Tephlon flange, 7 - Air fan, 8 - Fan connection, 9 - Electric reheater, 10 - Augmentation coil, 11 - Test section, 12 - Tested tube, 14 - Electric motor, 15 control valves, 16 - Outlet stabilizing tube, 17 - Inlet stabilizing tube, 18 - Steam pipe, 19 - Micro - manometer, 20 - Temperature recorder, 21 - Rotameter, 22 - Pressure gauge, 23 - Thermometer, 24 - Table, 25 - Water level indicator, 26 - Electric boiler, 27 - Main heater, 28 - Drain and 29 - Inlet collector.

pipe, fitted with a tube of an insulated material and insulated with a glass wool of 50 mm thickness, having two ports one of which the superheated steam goes in and the second one for the outlet of the condensate as well as the uncondensing steam.

The experimental test rig Fig.(2) was an open loop in which air from the laboratory room was drawn through the system by a downstream blower (7). The flow rate was controlled by a valve (15) and measured by a standard rotameter (21). The pressure difference between the inlet and outlet of the tested tube (12) was measured by an inclined water manometer (19), to an accuracy of 5%. Slightly superheated steam at atmospheric pressure was used as a heating medium. The steam was generated in an electric boiler (26) and superheated by means of an electric heater (9) . The steam flowed from the boiler to the heat exchanger (4) , through the steam line (18) which was thermally insulated by glass wool (3) of 50 mm thickness. The condensate was discharged in the drain(28) . The temperature of the air flow at the test section inlet and outlet was measured by copper constantan thermocouples (0.15 mm diameter) (2) and(13) . The temperature of the inner surface of the test tube was measured by thermocouples at four points (5) . All the thermocouples used were connected to a 12-point self switching temperature recorder (20) , having a full scale of 200° C. For measuring pressure and temperature in the boiler (26) , a calibrated pressure gauge (22) and a mercury thermometer (23) with scale divisions of 0.1°C, were used. The test tube (12) has inside it an augmentation coil (10) as shown in Fig.(2) The augmentation coils have prich to wire diameter ratio (p/d) ranges between 1 to 8. Corresponding to a range of helix angle β between 4.26° to 30.78°

There were two unheated straight tubes(17) and (16), each of length 750 mm. These tubes were located before and after the test section (8), in order to stabilize the fluid flow. The two stabilizing tubes were connected with the test tube by two teflon pieces(6). in order to avoid the back conduction effect.

The air flow velocity in the test tube was calculated on the basis of the bare tube diameter. The physical properties of air were taken at the mean flow stream temperature, which was calculated as the difference between the average surface temperature of the tube wall and the logarithmic mean temperature difference.

The heat gained by the working fluid was calculated from the change of enthalpy of the air.

RESULTS, DISCUSSION AND CONCLUSIONS.

The discusion will begiu with the heat transfer results to be followed by the hydrodynamic resistance results.

The performance of the test rig was first checked by studying the heat transfer in the test tube without the augmentation coil. The test tube has outer diameter 22 mm and inner diameter 13.5 mm The results obtained agreed within 7 percent with results in [13], for case of fully developed turbulent air flow (Fig.3). The well known correlation used for comparison was

$$Nu = 0.021 \; R_e^{0.8} \; Pr_f^{0.43} \left(\frac{Pr_f}{Pr_w} \right)^{0.25} \qquad \ldots (1)$$

In Fig.(3) Nusselt numbers are plotted as a function of Reynolds number for the ten values of the tripping wire helix angle in case of air flow through the test tube.

The results for each p/d value are plotted in the figure along with the corresponding data symbol. Each Nusselt number shown, has been corrected to its set-reference Prandtl number, by a $Pr^{-0.43} \left(\frac{Pr_f}{Pr_w} \right)^{-0.25}$ - dependance. The straight lines plotted through the set of data in Fig.(3), were determined from a least squares fitting through all the data points. The enhancement of heat transfer coefficient provided by the tripping wire inserted inside the test tube is obviously seen in Fig.(3), by comparison with the heat transfer results for the test tube without the tripping wire. The heat transfer intensification is about 0.57%. This may be due to the effects of the increase of the turbulent intensity near by the inner surface of the test tube, i.e. the principal mechanism for heat transfer enhancement is due to the disruption of the laminar sublayer only. Thus one may expect that the operating conditions influence the flow friction more than the heat transfer. Heat transfer and flow friction characteristics of air flowing through a tube with a spiral spring insert in the heating mode operation do not seem to be available in the literature. Fig. (4) shows the relation between friction losses coefficient with Reynolds number for a horizontal tube with different tripping wire pitches in case of air flow through the test tube. One may observe that the friction factor, in general, decreases with the increase of Reynolds number, as expected for air. On the other hand the friction factor for the tubes with the inserted tripping wires shows a remarkable increase than those obtained by the smooth tube. Figure (5) shows that as the value of β increases the friction factor increases rapidly and reaches a maximum value nearly at $\beta = 8.47°$ (p / d = 2) and then slightly decreases with further increase in β . The variation in the flow friction factor may be due to the effects of the increase of surface area, the effects of the increase of the disturbance in the main core of the flow and the effects of the increase of the disturbance in the laminar sublayer of the boundary layer of the flow. It is also seen from Fig. (4) that when the flow rate increaes, or when Reynolds number increases, the flow friction decreases relatively, but the rate of the flow friction decrease is finished for high Reynolds number values ($Re > 2300$) Therefore one may conclude that, when the flow rate increases, the flow friction decrease relatively faster than the increase in the heat transfer coefficient. This indicates that the use of the spring insert will increase the tube side heat transfer coefficient. At the same time, the flow friction also increases as expected.

The basic data for the Nusselt number and the friction factor that have been presented here can be used as inputs to computations of the enhancement characteristics of swirl affected pipe flows. Such enhancement evaluations may be performed for a wide variety of constrains (e.g. fixed heat duty, fixed mass flow rate, fixed transfer surface area, fixed pumping power, etc). The actual execution of the enhancement evaluations is beyond the scope of this paper.

The present heat transfer, friction factor results could not be compared with those of the literature, because it was not possible to find a common ground for the characterization of the swirl.

CORRELATIONS

Finally an attempt is made to correlate the results obtained in the present such a correlation is quite useful from the designers stand point. The average Nusselt number is correlated with the other relevant governing parameters, namly Reynolds number, helix angle and prandtl number. The following correlation is obtained :

$$Nu = 0.033\ Re^{0.8}\ Pr_f^{0.43} \left(\frac{Pr_f}{Pr_w}\right)^{0.25} \quad \ldots (2)$$

where :

$60 \leq Re \leq 4000$, d/D = 0.104

$4.26 \leq \beta \leq 30.78°$,

The correlation (2) predicts values of Nu which agree with results to within $\pm$ 8% .

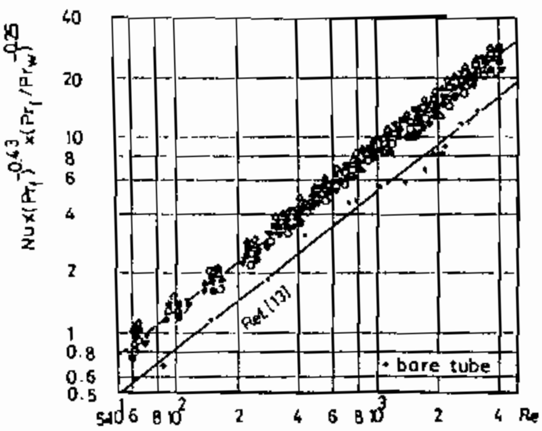

Fig.(3). The Variation of Nusselt number Versus Free Stream Reynolds number at Various helix angles.

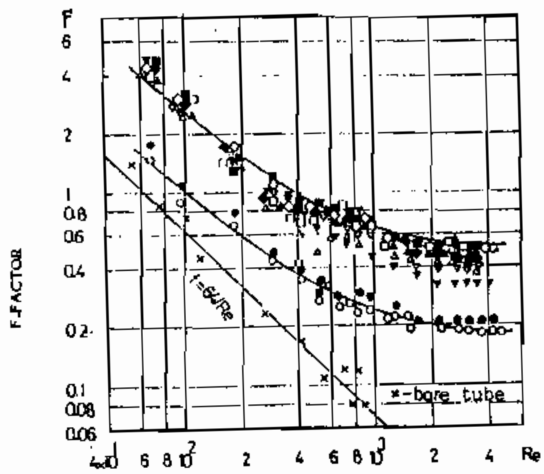

Fig.(4). The Variation of Flow Friction Factor Versus Free Stream Reynolds number at Various helix angles.

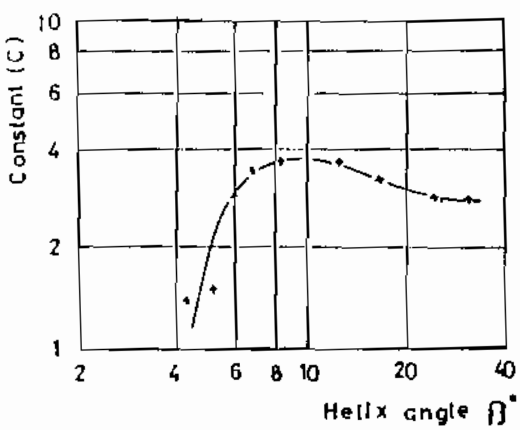

Fig.(5). The Variation of Constant C of Equation 3 Versus Helix Angle .

The friction factor (f) is correlated with other parameters.namly Reynolds number and helix angle (β). The correlation is as follows:

$$f = C\,(0.123 + 64/Re) \qquad \ldots(3)$$

where

$60 \leq Re \leq 4000$, $d/D = 0.104$,

$4.26 \leq \beta \leq 30.78°$

The numerical values of C are listed in table (2) and ploted in figure (5) as a function of helix angle (β). Equation (3) with values of C predicts the values of the data within $\pm$ 10% .

Table 2: The constant C for equation (3)

$\beta°$	4.26	5.11	5.95	6.79	7.63	8.47	12.59	16.59	24.08	30.78
C	1.884	2.044	3.945	4.770	5.121	5.179	5.131	4.498	3.921	3.914

CONCLUSION

Experimental investigation of a group of spiral coil used as heat transfer augmentative device in the heating of air flow was carried out. The following conclusions can be drawn from results of this investigation:

1- The spiral spring insert can Increase the tube side heat transfer coefficient significantly. The ratio of the heat transfer coefficient of the air flow in a tube with spring insert to that of a bare tube was about 1.57 in the range of Reynolds number from 60 to 4000 with the following correlation:

$$Nu_f = 0.033\, Re^{0.8}\, Pr_f^{0.43} \left(\frac{Pr_f}{Pr_w}\right)^{0.25},$$

$60 \leq Re \leq 4000 \quad \frac{d}{D} = 0.104, \; 4.26 \leq \beta \leq 30.78°$

2- The operating conditions influence the Flow Friction more than the heat transfer. As Reynolds number increase the friction factor decrease, but the rate of friction factor decrease is finished for high Reynolds number values ($Re > 2300$). The following correlation is used for the determination of flow friction as a function of Reynolds number and helix angle :

$$f = C\,(0.123 + 64/Re)$$

$60 \leq Re \leq 4000$, $d/D = 0.104$,

$4.26 \leq \beta \leq 30.78°$

The Constant for various helix angle is given in table (2).

NOMENCLATURE

- C_p specific heat of the fluid flow at constant pressure, (KJ/Kg.°c)
- D_i inside diameter of the bere tube, (m)
- d Coil wire diameter, (m)
- f Friction factor

h convecrive heat transfer coefficient, ($W/m^2 \cdot C$)
k thermal conductivety of tube, ($W/m\,°C$)
L length of tube, (m)
Nu Nusselt number based on the inside diameter of the bare tube, (h Di/k).

Pr Prandtl number, ($\mu\ c_p/k$)
P Pitch of the tripping wire coil, (m.)
Re Reynolds numnber based on the inside diameter of the bare tube, (V Di/u)

T temperature, (°C)
T/m number of turns per meter.
β helix angle, (°)
μ absolute viscosity of/fluid, (Pa.s)
ρ density of fluid, (kg/m^3)

Subscripts

f based on fluid
w based on wall
i inside

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