# Experimental Study on the Effect of Spiral Spring Inserts in a Circular Tube in Case of Forced Convection Heat Transfer.

M. Shalaby
*Mechanical Power Engineering Department., Faculty of Engineering., El-Mansoura University., El-Mansoura., Egypt.*, m.25jan@yahoo.com

Ahmed Sultan
*Lecturer at Mechanical Power Engineering Department., Faculty of Engineering., El-Mansoura University., El-Mansoura., Egypt.*

F. Araid
*Mechanical Power Engineering Department., Faculty of Engineering., El-Mansoura University., El-Mansoura., Egypt.*

Follow this and additional works at: https://mej.researchcommons.org/home

# EXPERIMENTAL STUDY ON THE EFFECT OF SPIRAL
# SPRING INSERTS IN A CIRCULAR TUBE IN CASE OF
# FORCED CONVECTION HEAT TRANSFER

دراسة تجريبية لتأثير ملف حلزوني في أنبوبة دائربـــة
على انتقال الحرارة بالحمل القـــــــــرى

By

## SHALABY, M. A., SULTAN, A. A. AND ARAID, F. F.

Mechanical Power Engineering Department, Faculty of
Engineering, El-Mansoura University, El-Mansoura, Egypt.

الخلاصة ـ ـ يشتمل هذا البحث على دراسة معملية لانتقال الحرارة بالحمل القسرى ومقاديـد
الفضط النـاتج عن احتكاك تيار هوائي بمر داخل أنبوبة نحاسية أفقية دائرية القطـــــــع
قطرها ٢٢/١٣مم بداخلها ملف حلزوني من السلك النحاسي بقطر ٤رامم بخطوة ثابتة حيـــــث
تكون نسبة الخطوة الى قطر السلك النحاسي تساوى ٣ ويتغير القطر المتوسط للمل ـــــــف
الحلزوني من ٤را الى ٤رامم . السطح الخارجى للماسورة النحاسة يسخن باستخدام بخـــــار
مشبع عند الفضط الجوى بدرجة تحميص من ٢ الى ٣م° وتتغير سرعة الهواء داخل الأنبوبــــة
لتبعطى رقم رينولد من ٦٠ ـ ٤٠٠٠ وقد توضحت الدراسة الى أن انتقال الحرارة فـــــــــى
الأثابب ذات الملف الحلزونى أعلى منها فى حالة الأثابيب العادية وتتراوح هـــــــــذه
الزيادة من ٢٥٪ الى ٤٥٪ بالمقارنة بالأثابيب العادية . وأن مقاديد الفضط تؤدى الــــى
زيادة ملحوظة فى قدرة الضخ .

ABSTRACT- Results are presented from experimental investigations of convective heat transfer and friction loss in a 13.5 mm inside diameter copper tube in case of air flow. This tube is fitted with helical-wire- coil inserts of varying diameter (Dc) ranges between 1.4 and 9.4 mm, constant pitch to diameter ratio (P/d=3) and constant wire diameter (d = 1.4mm). The outer surface of the test tube is heated by a little superheated steam (2-3 °C) at one atmosphere. During this work, Reynolds number varied between 60 and 4000 and Prandtl number was about 0.7. The heat transfer enhancement for the tested tube with internal coils was higher than that of the bare one by value ranges between 25% and 45%. The pressure loss results show a quantitevely increase in the pumping power.

## INTRODUCTION

The availabilty of the world's limited material and energy resources and the ever increasing cost of energy over the past few years aceelerated the research in the field of conversion of raw materials and reduction in energy usage for a given proeess. In the field of the heat transfer, this necessitated the development of compact and more efficient heat transfer equipment. In the previous work of the same authors [1], the work was taken up with the possible aim of having better heat transfer results in a gas to gas condenser by swirling the flow inside eopper tube.Their work summerized the result of the experimental investigation of convective heat transfer and pressure loss of air flow in a horizlntal 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 atmospheric pressure superheated by 2-3°C. In their 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.

In the present work the same authors of [1] investigate experimentally the effect of the coil diameter inserted in the copper smooth tube on the heat transfer

coefficient as well as on the pumping power used. Throughout the present experimental work, the results of the heat transfer and friction characteristics of air flowing through a horizontal tube with spiral spring inserts are obtained. The air flow is heated by a steam at atmospheric pressure superheated by 2-3°C. In this study the ratio between the coil pitch and its diameter (p/d) is constant and equal 3, in which at this value, the heat transfer coefficient is higher than the other values of (p/d) in [1]. The coil diameter changes from Dc=1.4 mm to 9.4 mm.

In fact the use of rough surfaces is one of the several enhancement techniques reported by Bergles [2], Dipprey and Sabersky [3] and Nikuradse [4], 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 tube [5-7], spirally corrugated tubes [8-9] and converging diverging tubes [10-11]. However, very limited work has been published on the thermohydraulic performance of helical wire inserted tubes, especially for convective heat transfer applications. Nag and Rao [12] studied the friction and heat transfer performance and developed suitable correlations for momentum and heat transfer roughtness functions, based on friction and heat transfer similarily laws. The working fluid in their study was R-12 and the test tube was heated electrically.

## TEST RIG AND OPERATING PROCEDER:

This investigation concerns the determination of heat transfer rates and friction losses in a tube with five helical wire coils for a wire diameter 1.4 mm and for five coil diameters Dc = 1.4, 4.6, 6.4, 7.8 and 9.4 mm. During this work the ptich to wire diameter ratio is constant and equal 3.

One smooth tube is 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 five helical wire coil diameters inserted in the copper tube, relative to a smooth tube.

Figure (1) 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 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. (1) 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 (o.15 mm diameter) (12) 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.
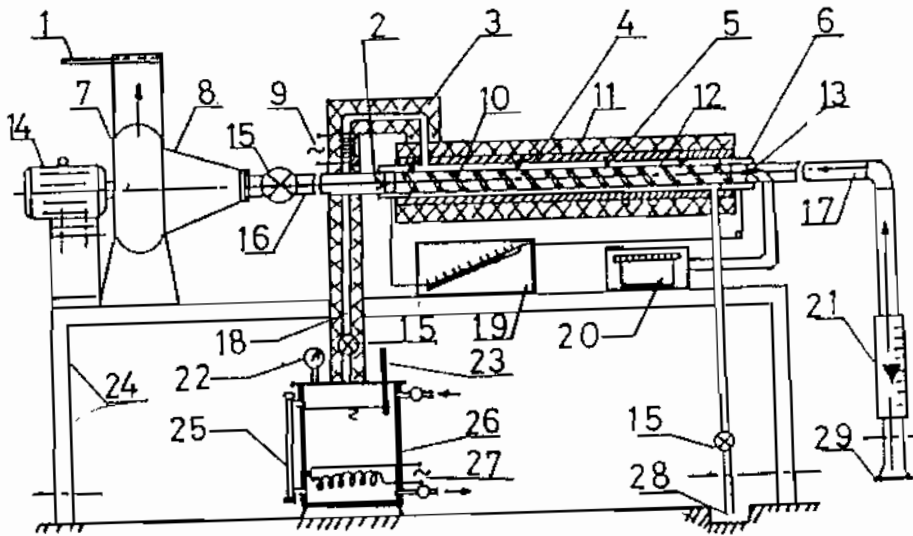
Fig.(1). Experimental test Rig.

1- Air Flow gate, 2,5 and 13 - Copper-Constantan Thermo-
couples , 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 -
Temperaturi Recorder, 21 - Rotameter, 22 - Tressure
gauge, 23 - Thermometer, 24 - Table, 25 - Water level
indicator, 26 - Electric boiler, 27 - Main heater, 28 - Drain
and 29 - Inlet collector.

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. (1). The augmentation coils have pitch to wire diameter ratio (P/d) equal 3, corresponding to helix angle $\beta$ equal 12.59° .

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 discussion will start with the heat transfer results to be followed by the hydrodynamic resistance results.

The test rig performance 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. Fig. (2) shows that the plotted results agreed within 7 percent with the results in Sukhatme [13], for the case of fully developed turbulent air flow. The well known correlation used for comparison was

$$Nu = 0.021 \ Re^{0.8} \ Pr_f^{0.43} \ (\frac{Pr_f}{Pr_w})^{0.25} \qquad \qquad \ldots (1)$$

In Fig. (2) Nusselt numbers are plotted as a function of Reynolds number for the tripping wire coils have diameters equal 1.4, 4.6, 6.4, 7.8 and 9.4 mm in the case of air flow through the test tube.

The data obtained are plotted, for each coil diameter, in Fig. (2) along with the corresponding data symbol. Each Nusselt number shown, has been corrected to its set-reference Prandtl number, by a $Pr^{-0.43} \ (\frac{Pr_f}{Pr_w})^{-0.25}$ dependance. The Figure, in general, shows that the results obtained at the coil diameters 1.4, 4.6, 6.4, 7.8 and 9.4 mm are higher than those obtained for the straight tube. One may observe that the results plotted in the Figure corresponding to the Reynolds number values between 50 and 700 have not a specified tend. However, the results obtained within the region of Reynolds number varies between 1000 and 4000 have, to some extend, a specified tend. For example, the results of the coil diameter 9.4 mm shows the highest value of the heat transfer rate. The Figure also, shows that the results plotted for the coil diameter equal 4.6 mm come below the results of 9.4 mm coil diameter and then followed by those obtained for the coil diameters 6.4, 7.8 and 1.4 mm. respectivily.

In fact one may also observe that, as the coil diameter is small enough by comparison to the test tube diameter, there is no remarkable effect on the heat transfer rates. On the other hand, when the coil surface becomes so close to the inner surface of the test tube, the heat transfer rate increases. This increase

In the heat transfer results may be occured due to the discontineuity of the boundary layer. It is also observed as that the , coil diameter decrease the heat transfer rate indicate a little decrease. This decrease in the heat transfer results may be occure due to the rundom generation of eddies beside the inner surface of the test tube. Fig. (2) also indicates that the heat transfer enhancement ranges between 25% and 45%. The figure also shows that the present set of results occupied the zoon between the solid line of the bare tube and the dashed line represents the results obtained in [1].

Fig. (3) shows the relation between friction losses coefficient and Reynolds number for a horizontal tube with different tripping wire. The tripping wires have coil diameters ranges between 1.4 and 9.4 mm. As expected the friction factor, in general, decreases with the increase of Reynolds number. On the other hand the friction factor for the tubes with inserted tripping wires shows a remarkable increase than those obtained by the smooth tube. Obviously, it is seen that the friction factor increases with the coil diameter. One may abserve that the slope of the friction coefficient lines decrease as the Reynolds number increases. This also indicates that the friction power has a remarkable increase with Reynolds number by comparison to the data obtained for the bare tube.


## CONCLUSION

The research work described here constitutes a comprehensive study of the fully developed turbulent heat transfer of a group of tripping wires. The tripping wires used as heat transfer augmentative device in the heating of air flow is carried out. The following conclusions can be drawn from the date obtained:

1- The tripping wires used in this experimental work, in general, increase the heat transfer coefficient with respect to the data obtained for the bare tube.

2- The results obtained show that the heat transfer enhancement has the higher values as the tripping coil diameter becomes so close to the inner surface of the test tube

3- The data obtained also shows that as the coil diameter decreases the variation of the heat transfer coefficient changes without a specified trend.

4- The friction coefficient decreases in general, with the Reynolds number increase in case of air flow

5- The coefficient of friction increases with the coil diameter.

6- The results obtained, also indicate that the slope of the friction coefficient curves decreases as the Reynolds number increases.

7- The operating conditions affect the pumping power more than the heat transfer coefficient.


## NOMENCLATURE

$C_p$  Specific heat of the fluid flow at constant pressure, [KJ/Kg. °C]

$D_c$  Coil outer diameter, [m].

$D_i$  inside     diameter of the bare tube, [m]

$d$  Coil wire diameter, [m]

$f$  Friction factor

$h$  convective heat transfer coefficient, [W/m². °C]

$k$  thermal conductivity of tube, [W/m. °C]

$L$  tube length, [m]

$Nu$  Nusselt number based on the inside diameter of the bare tube, (h Di/k).

$P_r$  Prandtl number, ($\mu$ cp/k)
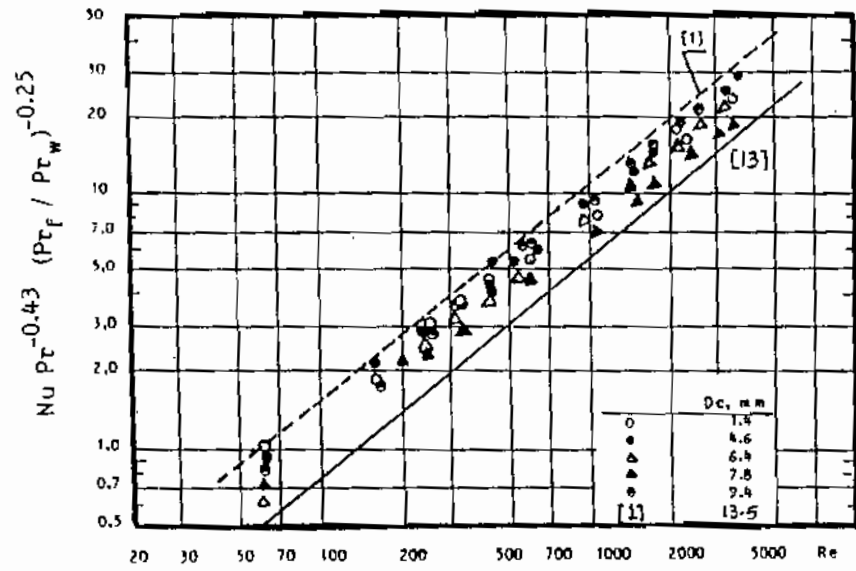
$P$  Pitch of the tripping wire coil, [m]

Fig. (2) Relation between Nusselt number and Free Streem
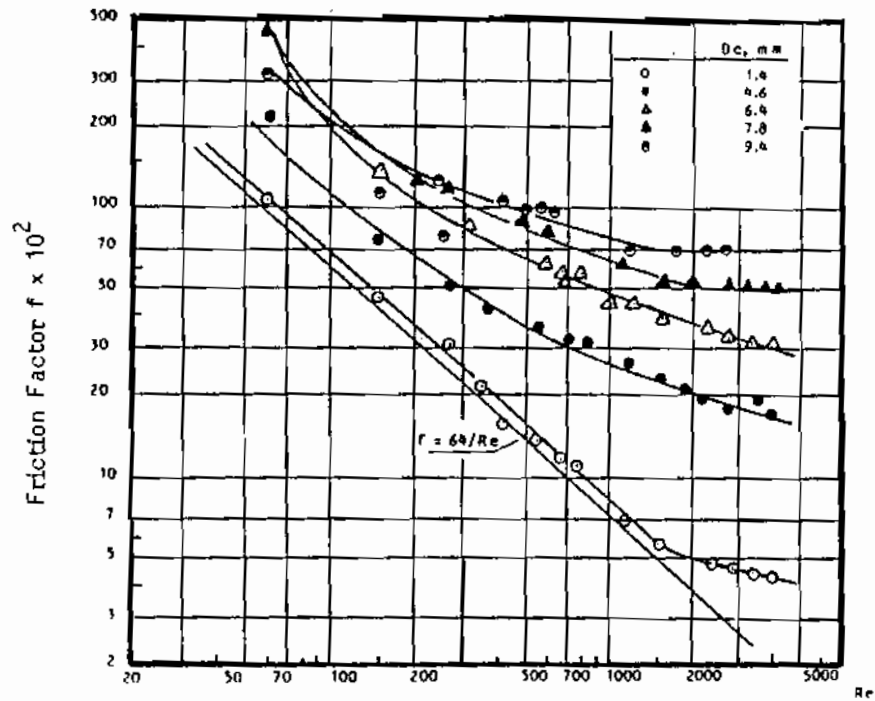Reynolds number with different coil diameters.

Fig. (3) Relation between Friction Factor and Free Stream
Reynolds number with different coil diameters.

$R_e$　Reynolds number based on the inside diameter of the bare tube $[\frac{\rho v D_1}{\nu}]$

T　temperature, [°C]

$\mu$　absolute viscosity of fluid, [P a.s]

$\rho$　density of fluid, [kg/m$^3$]


## Subscripts

f　based on fluid

w　based on wall

i　inside


## REFERENCES

1- Sultan , A. A., Shalaby , M. A. and Araid , F. F. "Experimental Study of the Augmentation of Convection heat Transfer in Horizontal Circular Tube Using Tripping Wires Inserts" Mansoura Eng. Journal. (MEJ) Vol. 15, No. 1, June 1990.

2- Bergles, A. E. "Enhancement of heat transfer", Proc, 6th Int. Heat Transfer Conf., Vol. 6, p-89, Toronto. (1978).

3- Dipprey, D.F. and Sabersky, P.H. "Heat and momentum transfer in smooth and rough tubes", Int. J. Heat Mass Transfer, Vol. 6, p-329, (1963).

4- Nikuradse, J. "Gestz mapigkeit der turbulenten storomnng in glatten Rohren", Forschung-scheft xx, p-356, (1932).

5- Webb, B.J., Eckert, E.R.G. and Goldstein R.J. "Heat transfer and friction in tubes with repeated rib roughness", Int. J. Heat Mass Transfer, Vol. 14, pp. 601-617, (1971)

6- Dalla Donne, M. and Meyer, E. "Turbulent convective heat transfer from surfaces with two-dimentional rectangular ribs", Int. J. Heat Mass Transfer, Vol. 20, pp. 584-620, (1977).

7- Gee, D.L. and Weeb, R. L., "Forced convection heat transfer in helically rib-roughened tubes", Int. J. Heat Mass Transfer, Vol. 23, pp. 1127-1136, (1980).

8- Withers, J.G., Young, E. H. and Lampert, W. B., "Heat transfer characteristics of corrugated tubes", Chem. Engg. Prog. synp. ser. 174 (74). pp. 15-24, (1978).

9- Mehta, M. H. and Raja Rao, M. "Heat transfer and frictional characteristics of spirally enhanced tubes for horizontal condenser", Symp. on Enhanced Heat Transfer, pp. 11-21, 18th National Heat Transfer Conference, San Diego, (1979).

10- Araid, F. F., Shalaby, M. A., and Awad, M. M. "Convective heat transfer to viscous fluid flow in convergent-divergent tubes", 4th. IASTED Int. Symp. Modelling. Identification and control, M/C' 85, Grindewald, Switzerland pp. 55-60, (1985).

11- Souze Mendes, P. and Sparrow, E.M., "Periodically converging-diverging tubes and their turbulent heat transfcr, pressure drop, fluid flow, and enhancement characteristics", Trans., ASME, J. Heat Transfer, Vol. 106, pp. 55-63, (1984).

12 Nag, S. and Raja Rao, M. "Forced convection heat transfer in smooth tubes roughened by helically coild ribbons", Int. J. Heat Mass Transfer. Vol. 30, No. 7, pp-1541-1544, (1987).

13- Sukhatme, S. P. "A text book on heat transfer", Orient Longman Limitcd, (1983).