HEAT TRANSFER AND FRICTION IN ANNULAR TUBES CARRYING GAS FLOW WITH HELICALLY-TRIANGULAR-RIB ROUGHNESS ON THE INNER SURFACE

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خلاصة : يقدم هذا البحث دراسة تجريبية على خصائص الانتقال الحرارى الجبرى ومعامل الاحتكاك خلال الانفاق الحلقية الخشنة التى يسرى خلالها تيار هواء . ولقد تم تنفيذ مبادل حرارى حلقى يتكون من انبوبتين متمركزتين لهذا الغرض . الأنبوبة الخارجية للمبادل الحرارى مصنوعة من مادة البلاستيك وبقطر داخلى ٤٨ مم وسمك ٥مم ومعزولة من الخارج بالصوف الزجاجى ، أما الانبوبة الداخلية المركزية فمصنوعة من النحاس الاحمر بقطر خارجى ٢٢مم وسمك ٣مم . فى المبادل الحرارى تسخن الانبوبة المركزية بواسطة مرور تيار من بخار الماء المشبع عند الضغط الجوى بداخلها ، بينما يمر تيار من الهواء خلال الممر الحلقى وذلك بواسطة سحبة عن طريق شفاط هوائى . ولقد اجريت التجارب على خمس ممرات حلقية ، بدأت بالممر الحلقى الاملس للتأكد من مدى دقة النتائج وذلك بالمقارنة بنتائج الباحثين السابقين . يلى ذلك دراسة لمشكلة خلال أربع أنفاق حلقية خشنة اعدت بتشكيل السطح الخارجى للأنبوبة المركزية بأسنان مثلثة الشكل لولبية بعمق ٢مم ، وبزاوية لولب = ٥٩´ ، ٤٥´ ٢° ، ٣٢´ ٤° ، ١٧´ ٦° وبفراغات نسبية ٥٤ر٠ ، ٥١ر١ ، ٤٩ر٢ و ٤٧ر٣ على الترتيب ، وقد اجريت التجارب بحيث تغير رقم رينولد من ٣٩٠٠ الى ٤٤٠٠٠. ولقد أظهرت نتائج هذا البحث أنه كلما زادت الفراغات النسبية بين الأسنان اللولبية وكذلك كلما ذادت زاوية اللولب ذاد كل من معامل الانتقال الحرارى بنسبة تتراوح بين ٢٥٪ و ٩٧ ٪ ، بينما يزيد معامل الاحتكاك بنسبة تتراوح بين ٢٠٪ و ١٤٠ ٪ بالنسبة لنتائج الانبوب الحلقى الأملس . ولقد تم تحليل النتائج واستنتاج معادلة تجريبية بين رقم نوسلت ورقم رينولد وزاوية للولب وكذلك النسبة الفراغية للأسنان. وكذلك تم استنتاج معادلة مماثلة بالنسبة لمعامل الاحتكاك. وهذا البحث يفيد فى مجال المبادلات الحرارية المستخدمة فى كثير من التطبيقات الهندسية المختلفة.

ABSTRACT. This paper presents the results of an experimental investigation into the average heat transfer and friction characteristics of gas flow in annuli. The results are measured at five annular passages having a smooth outer tube with inside diameter of 48 mm, wall thickness of 5 mm and length of 3050 mm, and interchangeable inner tubes with total length of 2000 mm and the rough part 540 mm long. The results of the helical-triangular-rib surface geometry are measured for four helix angles (α = 59', 2°45', 4°32', and 6°17') all having a relative roughness size(e/d) of 0.1. The helical-rib relative spacings (c/e) equal to 0.54, 1.51, 2.49 and 3.47. In addition to the four rough tubes, a smooth tube geometry is tested to validate the experimental procedure. Air is used as a working fluid, encompassing flow range $3900 < Re < 44000$. The data are correlated in a form to permit performance prediction. The benefits of the roughness for heat exchanger applications are quantitatively established

INTRODUCTION

Investigations in the field of fluid flow and heat transfer have established that when a fluid flows over a surface a stagnant film adheres to the surface and acts as heat insulator. This film acts as a barrier to the flow of heat. This barrier which sticks to the wall can be reduced in effectiveness by different methods. One of the most efficient methods of turbulent convective heat transfer enhancement is artificial roughening of the heat-transfer surface. This is especially true for high power-consuming systems in which gas coolants are employed. Thus, the need for more efficient heat exchange devices has led to the development of a variety of unconventional internal flow passages to enhance the heat transfer coefficient. One such passage is the artificial rough annular tubes, in which fluid moves along an undulating path as it encounters the successive peaks and valleys.

The concentric-rough-annulus tubes of the present experiments had a smooth outer tube with inside diameter of 48 mm, wall thickness of 5 mm and length of 3050 mm and interchangeable concentric inner tubes with total length of 2000 mm and the rough part 540 mm long. In the upstream of the rough part a smooth extension tube with length of 1040 mm is located to exist fully developed flow. This part has helical-triangular-rib surface geometry with four helix angles (α = 59` 2°45` , 4°32` and 6°17`) of helical-rib relative spacings (c/e) equal to 0.54, 1.51,2.49 and 3.47 respectively, all having an undulation height (e) of 2 mm and a relative roughness size (e/d) of 0.1. During the experiments, the annular tubes always have the inner surface at a constant temperature, different from the uniform entering temperature and the outer surface insulated. Air is used as a working fluid, encompassing flow range $3900 < Re < 44{,}000$. The inner tube is heated by saturated steam at atmospheric pressure. In addition to the four rough tubes, a smooth tube geometry is tested to validate the experimental procedure.

Some related work has been performed in the past regarding heat transfer in annuli. Rampf and Feurstein [1] employed an annulus having the inner surface only is roughened. The roughness elements different from that of the present investigation in which they consist of equally spaced rings of triangular cross-section. In their work, the test fluid was air and three different roughness heights were provided and each was tested at several different spacing. They reported that a maximum (heat transfer and friction) is reached for a spacing to height ratio which is very similar to that at which Geffroy et. al. [2] also report a maximum. The surface used by Geffroy et. al. was a sinusoidally corrugated one. They reported that the improvement ratio was slightly higher than a factor of two. The flow of water over triangular roughness is also made visible by dye streeks and a series of interesting pictures, illustrating vortex formation and fluid exchange. Their results show that the highest heat transfer values are maximum at space to height ratio of about 7.

The research performed by Dalle Donne and Meyer [3] represents another contribution to the literature on heat transfer and friction in annuli. Their work was largely concerned with artificial roughness used in nuclear reactors to improve the thermal performance of the fuel elements, in which gases are used as a heat transfer media. The experiments to measure the heat-transfer and friction coefficients of roughness are performed with single rods roughened with thread-type ribs of trapezoidal profile and contained in smooth tubes. They illustrated a transformation method based on the assumption that the velocity and temperature profiles normal to the rough surface can be described in turbulent flow by the universal laws of the wall :

$$u^+ = 2.5 \ln \frac{y}{e} + R(e^+) \qquad \ldots (1)$$

$$t^+ = 2.5 \ln \frac{y}{e} + G(e^+) \qquad \ldots (2)$$

The method has been applied to a geometrical configuration typical of a fuel element of a gas cooled fast reactor, and the agreement between the theoretical prediction and measurement was quite good. Their work was largely concerned with Reynolds-number upto 10^5. However, Firth and Meyer [4] made a comparison of the heat transfer and friction factor performance of four different types of artificially roughened surface. Each surface has near-optimum thermal performance for its own particular type of roughness. They concluded that, there is no advantage in using a transverse trapezoidal roughness. If a roughness is required with a low friction factor but without a reduction in rib height and it is also recommended that the best alternative is a helically ribbed surface.

Experiments were also conducted by Vilemas and Simonis [5] into the local heat transfer and friction of air flow in annuli with the inner rough tube for Reynolds numbers ranging from $5x10^3$ to $5x10^5$, relative height (e/d) from 0.0028-0.021 and relative spacing (c/e) approximatly equal to 10 of rectangular and rounded trapezoidal roughness elements. They reported that, in contrast to smooth tubes, the effect of variable physical properties of the gas on heat transfer in rough tubes depends on the relative height and Reynolds number and diminshes with the increase of these parameters.

Berighton and Jones suggested that the turbulent friction factor is independent of radius ratio (r_o/R_i) for their data for $0.0625 < r_o/R_i < 0.562$. The original paper is not available. As such, details of the paper as reported by Kays and Perkins in Ref. [6] are given.

For the purposes of evaluating the level of heat-transfer enhancement associated with the rough annular-tube flow, the Nusselt numbers of the present experiments will be compared to those of a smooth annular tube as represented in [6].

APPARATUS AND PROCEDURE

A schematic diagram of the experimental apparatus is sketched in Fig.1. Air is the working fluid. It is drawn into an adiabatic entrance length of a plastic tube (1) by a downstream blower (2). The flow rate is controlled by a valve (3).

The air velocity (u_o) is measured at the centre of the tube (1) with the help of a Prandtl-Pitot tube (4) and a micromanometer (5). The accuracy of measurement of the velocity is estimated to be $\pm$ 1.5%. The Pitot tube being situated 40 cm upstream from the annuli (6) and 140 cm upstream from the rough annular test section (7). The air velocity is also measured by the probe of the hot wire-anemometer (8) located 65 cm upstream from the annuli. The difference in velocity values by the two methods is $\pm$ 1%. The uniformity of velocity across the tunnel tube (1) is also checked. The variation is less than the estimated accurcy of measurement.

The temperature of the internal tube heated surface is measured by means of 16 copper-constantan thermocouples (9) made from 30 gauge wires and welded into the helical-rib surface geometry of the copper tube (10). The sixteen thermocouples are divided equally into four groups, each of them is welded along the test copper tube surface and each group is spaced at 90° , to check for possible eccentricities in the annulus. The position of thermocouples in respect of the roughness ribs is so chosen that possible local temperature differences on the tube surface are eliminated by average temperature of the copper test tube in the annuli is calculated as follows

$$\bar{t}_{cw} = \left(\sum_{i=1}^{n} t_{ci} \right) / n \qquad \ldots (3)$$

where n is the total number of the thermocouples used to measure the inner tube surface temperature = 16

The outside plastic tube of the annulus is insulated by a 50 mm thick glasswool slab surrounded by a layer of asbestos tape of about 5mm thick. Nine copper-constantan thermocouples (11) are welded to the outer surface of the plastic tube at the part of the annuli which surround the copper rough test section. The nine thermocouples are divided equally into three groups, each of them is welded along a line on the surface of the plastic tube and each group is spaced at 120°. The average outer surface temperature of the portion of the plastic tube surrounds the rough copper tube is calculated as follows

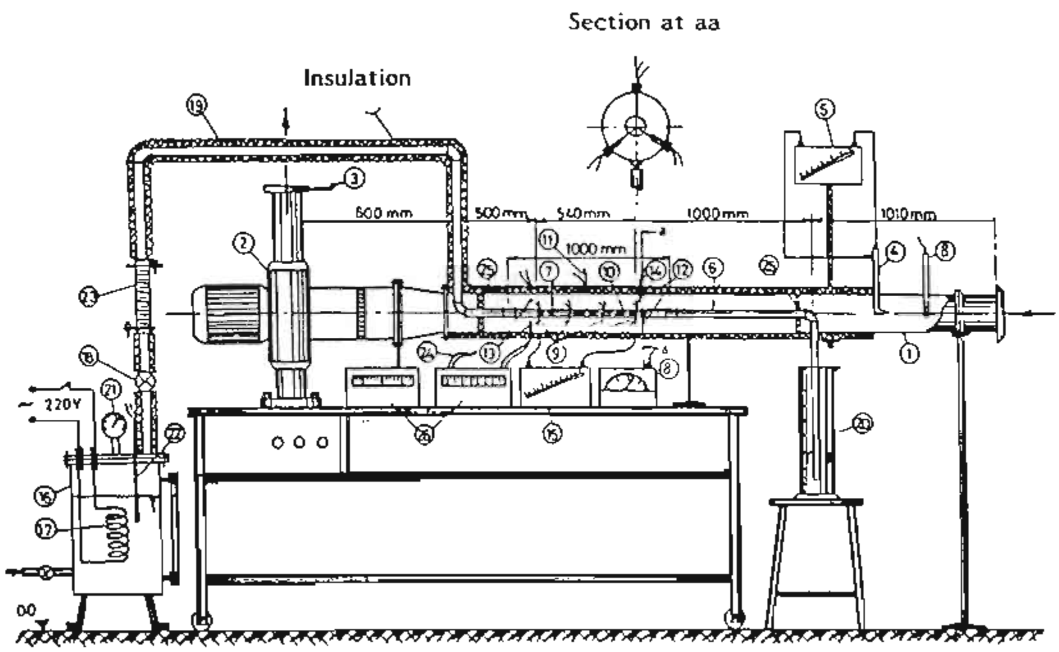

Fig. 1 Apparatus

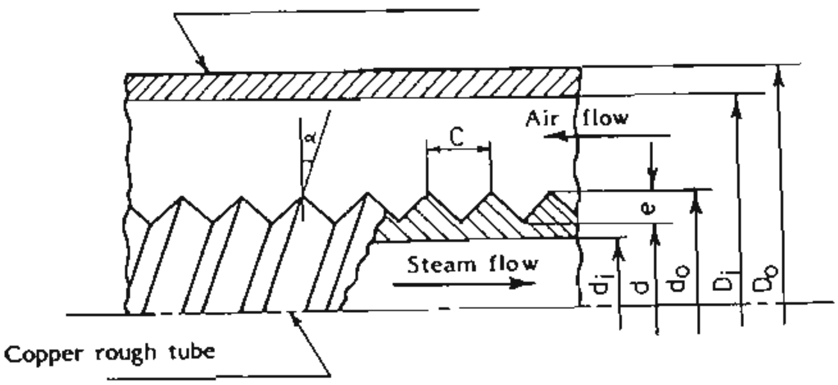

Fig.2. Configuration of the annular rough tube

$$\bar{t}_{pw} = \left(\sum_{i=1}^{k} t_{pi} \right) / k \qquad \ldots (4)$$

where k is the total number of thermocouples used to measure the outer surface temperature of the plastic tube=9. This helps in calculating the heat loss by conduction across the wall of the plastic tube.

The air temperatures at the rough annulus enterance and exit are measured by means of six thermocouples. The six thermocouples are divided equally into two groups (12 and 13), each group has three thermocouples and each of them is spaced at 120° on the periphery of the plastic tube. The thermocouples are inserted inside the annuli across the plastic wall by means of small diameter guide-tubes (14) to adjust their hot junctions in the annuli at 3, 7 and 10.5 mm far from the inner surface of the plastic tube. However, the maximum diviation in the measured temperature by each thermocoupl does not exceed 0.5°C. Then, the air inlet ($\bar{t}_1$) and outlet ($\bar{t}_2$) temperatures are calculated as the average value of the three measured values at each location. The air bulk temperature (t_b) is calculated as the average of the six measured values.

The static pressure difference along the annular rough tube is measured by means of the inclined water manometer (15) to an accuracy of about 5%. A saturated steam generated in an electric boiler (16) by means of an electric heater (17). The steam generated flows from the boiler to the rough copper tube of the annuli through the handle-valve (18) and the steam line (19) which is thermally insulated by glass wool of 50 mm thickness. The condensate is collected in the measuring cylinder (20). For measuring boiler pressure a calibrated pressure gauge (21) is used and the boiler temperature is sensed by the thermocouple (22). The boiler system is connected to the steam line (19) by means of an elastic tube (23) to vanish the effect of vibration. The ambient temperature is also measured by means of the thermocouple prob (24).

In the present experimental program, four annular passages of varying helical-triangular-rib sizes on the outer surface of the inner copper tube are used and compared with a standard smooth annular tube. The smooth annular tube is used for standardising the experimental setup and also to compare the enhancement obtained in both heat transfer coefficient and friction factor. All the enhanced rough inner tubes are fabricated from plain copper tube of 22 mm outside diameter and 3 mm wall thickness, and are threaded along their active lengths. The thread shape is triangular in cross section and is 2 mm height (e). The length of the active threaded part equals to 540 mm and the pitches of threads are 1.07, 3.02, 4.98 and 6.93 mm, and the corresponding helical angles of the shaped threads are varying from 0.98° to 6.29°. Fig.2 shows the configuration and basic geometric characteristics of the annular rough tube. The outer smooth tube of the annuli is made from plastic material and has 48 mm inside diameter, 5 mm thickness and 3050 mm length. The concentric position of the annular tube is obtained by means of two honeycomb wooden discks (25) as shown in Fig.1 .

The air measured velocity at the inlet of the plastic tube is denoted by u_o and hence, the rate of air flow rate can be calculated by the following correlation

$$m' = \rho_i \; u_o \; A_o \qquad \ldots (5)$$

The average velocity in annular tube can be calculated from the continuity equation as follows

$$\bar{u} = \frac{u_o \; D_i^2}{D_e \; (D_i + d_m)} \qquad \ldots (6)$$

where $d_m = (d_o + d) / 2$

The coefficient of friction (ξ) can also be calculated from the following correlation

$$\xi = \frac{2\ \Delta P\ D_e}{\bar{\rho}\ \bar{u}^2\ L} \qquad \ldots (7)$$

The heat gained by the mass flow rate of air flowing through the annular heated rough tube is calculated as follows

$$\dot{Q} = \dot{m}\ cp\ (\bar{t}_2 - \bar{t}_1) \qquad \ldots (8)$$

Then, the heat transfer coefficient can be calculated from the Newton's law for heat flow between the threaded copper tube surface and the air flow as follows

$$h = \frac{\dot{Q}}{\pi\ d_m\ L\ (\bar{t}_{cw} - t_b)} \qquad \ldots (9)$$

The characteristic length used in the determination of the Nusselt number values and the Reynolds number values is the annular tube equivalent diameter (D_e).

All thermocouples used are connected to two 24-point self switching temperature recorders (26) having a full scale of 200°C. The thermocouple readings are recorded at intervals of 15 minutes during the experiment. The thermal steady state is assumed to be satisfied when the thermocouple readings of the air inlet, air outlet and the threaded surface of the copper tube are constant, and the change of all temperatures with time becomes negligible. The average time required to reach a thermal steady state was about 2.5 hours.

RESULTS, DISCUSSION AND COMPARISON

The presentation of the measured data and the enhancement obtained will begin with the heat transfer results, to be followed by the friction factors.

Heat Transfer. Before starting to evaluate the characteristics of the surface with regularly spaced roughness. The flow takes place through an annulus formed by two smooth cylinders and the flow direction is parallel to the axis, i.e. the apparatus is tested for its plain annuli performance at Reynolds numbers from 4400 to 44000 with nominal inlet air temperature of 32°C and the temperature factor ($\bar{t}_{cw}/t_b$) was about 2.3. Figure 3 shows that the present experimental results agree within ± 5% with the results obtained in [5-7] for smooth annuli, for the fully developed Nusselt numbers. The results of [5] were for annuli with diameter ratio (D_i/d_o) equal to 2.38, the results of [6] were for D_i/d_o = 2.0, and the results of [7] were for D_i/d_o = 1.4 and their dotted line is drawn from the following correlation

$$Nu = 0.017\ Re^{0.8}\ Pr_f^{0.4}\ (Pr_f\ /\ Pr_w)^{0.25}\ (D_i\ /\ d_o)^{0.18}\ , \qquad \ldots (10)$$

while the present results are obtained for diameter ratio equal to 2.18. One may observe that the heat transfer results have a little dependence on the diameter ratio.

The Figure shows also that the Nusselt number value, in rough annular channels, increases differently over the studied range of the Reynolds numbers ($3900 < Re < 44000$). The fully developed Nusselt numbers are displayed on log-log coordinates for five separate annular channels, as shown in Fig.3. In addition to the smooth tube geometry, the results of the helical-triangular-rib surface geometry are measured for four helix angles (α) equal to 59', 2° 45', 4° 32' and 6° 17' all having a relative roughness size (e/d) equal to 0.1 and the corresponding spacing ratios (c/e) are 0.54, 1.51, 2.49 and 3.47 respectively. It is seen that the Nusselt number value increases, in general, with the spacing ratio. The heat transfer results of the annuli

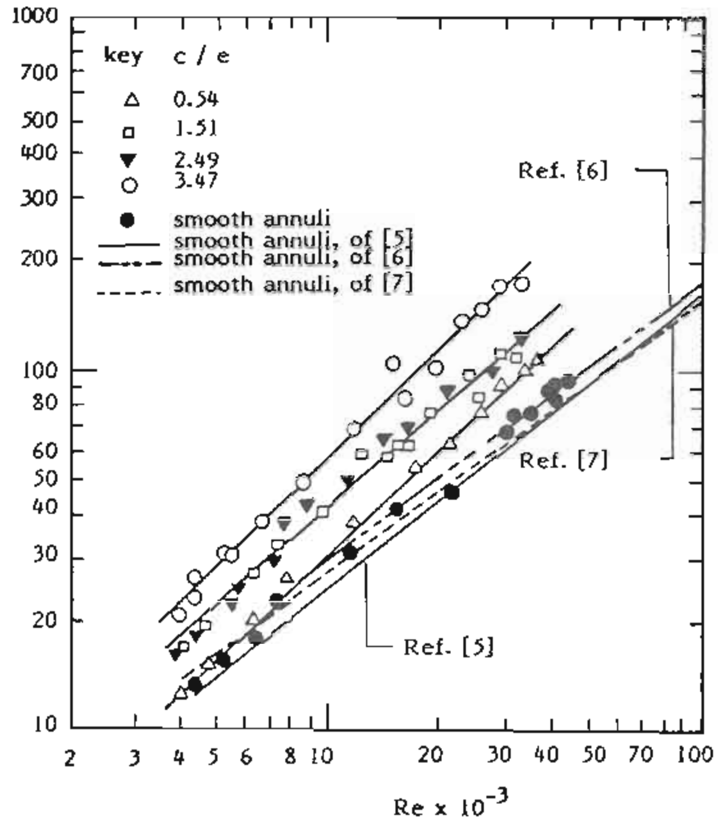

Fig.3. The effect of roughness on Nusselt number for fully developed turbulent flow through an annulus of inner surface heated and outer surface insulated.

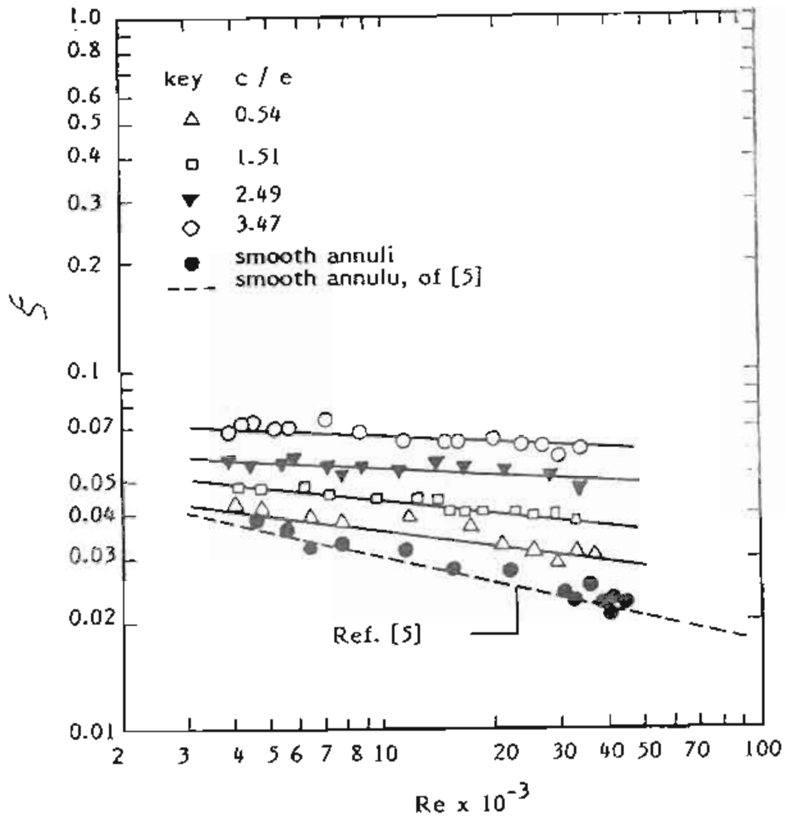

Fig.4. The friction of rough annular channels, compared with that for smooth annuli.

with the spacing ratio equal to 0.54 are of the same order of the results displayed for the smooth annuli for Re $<$ 8000, while the heat transfer enhancement increases with Reynolds number, for Re $>$ 8000, to be about 25% with respect to the smooth annuli. In general, it is possible to state that the thickness of the boundary layer decreases with viscosity, or more generally, it can be expected that high Reynolds numbers lead to a greater influence of surface roughness because of the thinner sublayers, and this will be noted, in the next section, for the friction coefficient. This conclussion comes from several exact solutions of the Nevier-Stokes equations presented in [8]. The Nusselt number values show a remarkable increase as the relative spacing increases to 1.51 and 2.49 and the displayed data for the two annulies appear more or less the same as shown in Fig.3. The solid line fits their points and shows that the heat transfer enhancement of the two channels is higher than the same obtained for the case of c/e = 0.54 by about 32%. The Figure shows also that the heat transfer results of the annuli has the relative spacing equal to 3.47 are displayed on the Figure top and the heat transfer enhancement is about 40% with respect to the data of the two channels of c/e equals to 1.51 and 2.49. Overall inspection of Fig.3 shows that the Nusselt number values, in general, increases with the Reynolds number and with the helix angle, for the whole set of channels. In all, 67 data points are obtained and in addition a number of experiments are repeated.

Friction Coefficient. The test rig is first checked by studying the friction coefficient of the smooth annuli. The results obtained agree very well with those given in [5] as shown in Fig.4. The coefficient of friction values are displayed, as shown in the figure, on log-log coordinates as a function of Reynolds number. It is seen that the friction in the rough annular channels increases with the relative spacing (c/e), i.e. with the damagement of streamlining of roughness elements. The rough annuli with (c/e) = 0.54 has the friction coefficient values higher than those of smooth annular channel by about 20%. As the relative spacing increases to have 1.51, 2.49 and 3.47 the incremental values in the coefficient of friction are about 50%, 90% and 140% respectively, with the respect to the smooth annuli. In fact, due to the surface of the annular channels is sufficiently rough no predominantly viscous region can exist, and the apparent shear forces are transmitted to the wall in the form of pressure drag on the irregularities, and the friction coefficient becomes virtually independent of the Reynolds number especially at high relative spacing values.

Most important from the point of view, the ability of the fluid to transport momentum, heat and mass transverse to the mean flow direction is greatly enhanced.

Correlations of Data. In addition, it would be a very valuable service to collect the present available information on the annuli and to present the results in a uniform fashion for the use of the designer. The heat transfer and friction data are correlated using accepted correlations for rough annular tubes. This will permit the data to be interpreted for a wide range of c/e, α and Re.

The average Nusselt number is correlated with the other relevant parameters of the test annular tubes as follow

$$Nu = 0.016\,(1 + \alpha \cdot c/e)^{1.86}\,Re^{0.83} \qquad \ldots (11)$$

The correlation (11) predicts values of Nu which agree with results to within ± 9%. In correlation (11), the exponent of 0.83 on Re clearly suggests the presence of a turbulent boundary layer flow. The correlation shows also that the Nusselt number is a significant function of the helix angle and the relative spacing.

The correlation for friction in rough annular tubes is also obtained in the following form

$$\xi = 0.128\,(1 + \alpha \cdot c/e)^{2.26}\,Re^{-0.15} \qquad \ldots (12)$$

The correlation (12) predicts values of ξ which agree with results to within $\pm$ 8%.

In the two correlations α is in radians, and they are valid in the following ranges $3900 < Re < 44000$, $0.0 < \alpha < 6° \, 17'$ and $0.0 < c/e < 3.47$.

CONCLUDING REMARKS

In this study the focus is on the average heat transfer and the friction coefficients of air flow in helically-triangular-rib rough annulus channels. The artifical geometry of the inner tube surface has a relative spacing ranging between 0.0 and 3.47 and a helix angle varying from 0.0° to 6° 17'. The Reynolds number based on the equivalent diameter of the annuli is varied from 3900 to 44000.

The following conclusions may be drawn from the present analysis:

1- The average heat transfer coefficient and the coefficient of friction increases with the relative spacing and the helix angle.

2- The heat transfer results increase, in general, with Reynolds number.

3- The friction coefficient values are more or less decrease with Reynolds number, however it seems to be independent of Reynolds number at high relative spacing.

4- The correlations have been derived to calculate the heat transfer [Eqn. (11)] and friction [Eqn. (12)] in rough annular channels with allowance for the effect of the relative spacing and the helix angle.

NOMENCLATURE

A	cross section area of an annulus, mm^2
A_o	cross section area of the outer tube of the annuli, mm^2
c	roughness pitch, mm
c/e	relative spacing
Cp	specific heat at constant pressure, KJ /Kg.K
D_e	equivalent diameter of annulus, $D_i - d_o$, mm
D_i	inside diameter of the outer tube of the annuli, mm
D_i/d_o	diameter ratio
D_o	outside diameter of the outer tube of the annuli, mm
d	root diameter of the roughness, mm
d_i	inside diameter of the inner tube, mm
d_m	mean diameter of the threaded part, $(d_o + d)/2$, mm
d_o	outside diameter of the threaded tube, mm
e	height of the roughness ribs, mm
e^+	dimensionless height of roughness ribs, eu^*/ν
$G(e^+)$	dimensionless gas temperature at the tip of the ribs
h	heat transfer coefficient, $KW/m^2.K$
k	thermal conductivity, KW/m.K
L	length of the rough part, mm
m'	mass flow rate of air, Kg/sec
Nu	Nusselt number, h De/k
Pr	Prandte number, μ cp/k
ΔP	pressure difference, N/m^2
Q'	rate of heat transfer, W
Re	Reynolds number, $\bar{u}$ De/ν
$R(e^+)$	dimensionless air velocity at the tip of the ribs
R_i	inside raduis of the outer tube of the annuli, mm
r_o	outside raduis of the inner tube of the annuli, mm
r_o/R_i	raduis ratio of the annuli
t	temperature, K

t^+	dimensionless air temperature at the distance y from the rough wall
t_b	air bulk temperature, K
t_c	copper tube temperature, K
$\bar{t}_{cw}$	mean copper tube temperature, K
t_{cw}/t_b	temperature factor
t_p	plastic tube temperature, K
$\bar{t}$	mean plastic tube temperature, K
t_1	mean air temperature at the inlet of the rough annuli, K
t_2	mean air temperature at the exit of the rough annuli, K
u	air velocity, m/sec
u^+	dimensionless air velocity at the distance y from the rough wall, u/u^*
u^*	dynamic velocity, $\sqrt{\tau_w/\rho}$
$\bar{u}$	air mean velocity in the annuli, m/sec
u_o	air mean velocity before the annulus section, m/sec
y	radial distance from the wall of the considered point, cm
α	helix angle, deg
ξ	friction coefficient
μ	viscosity
ν	kinematic viscosity
ρ	air density

Subscripts

f	fluid properties
i	properties at the channel inlet
o	properties at the outer tube surface
w	wall condition

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