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ENHANCING FORCED CONVECTION HEAT TRANSFER IN ANNULAR TUBES BY USING ALUMINUM MESH LAYERS

تحسين إنتقال الحرارة بالحمل القسري في أنابيب حلقية باستخدام طبقات شبكية من الألومنيوم

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خلاصة:

يتضمن هذا البحث دراسة عمليه للفقد في الضغط وانتقال الحرارة بالحمل القسري لتيار هوائي مضطرب يسري في فراغ حلقي أسطواني. الأنبوبة الداخلية مصنوعة من الألومنيوم بقطر خارجي 24.2 مم مسخنة من الداخل بفيض حراري منتظم. الأنبوبة الخارجية مصنوعة من البلاستيك (PVC) بقطر داخلي 48.1 مم ومعزولة من الخارج بطبقة من الصوف الزجاجي. تلف طبقات شبكية من ألياف الألومنيوم الرقيق " فلتر الهواء لجهاز تكييف الشباك " ملفوفة على السطح الخارجي للأنبوبة الداخلية بسمك متغير لدراسة تأثير سمك الطبقة على كل من فقد الضغط وانتقال الحرارة . يتغير رقم رينولدز خلال الدراسة بين حوالي 3000 آلي 20000 بينما تتغير نسبة سمك الطبقة الشبكية آلي سمك الفراغ الحلقي من 0.1925 إلى حوالي الواحد. وأوضحت الدراسة ان معامل انتقال الحرارة محسوبا على مساحة سطح الأنبوبة الداخلية يزداد بزيادة كل من رقم رينولدز وسمك الطبقات الشبكية وتتراوح الزيادة بين 80 إلى 270٪. بينما يزداد معامل الاحتكاك بزيادة سمك الطبقات الشبكية ونقصان رقم رينولدز وتتراوح الزيادة بين 114 آلي 10680٪.

وتم تحليل آداء المبادل الحراري على أساس ثبات قدرة الضخ ومساحة سطح انتقال الحرارة وتراوحت الزيادة في كمية الحرارة المنقولة "رقم نوسلت" بين 29 آلي 420٪. كما تم حساب الأداء على أساس ثبات كمية الحرارة ومساحة سطح انتقال الحرارة ووصل الانخفاض الأقصى في القدرة حوالي 76٪. وبدراسة فعالية المبادل الحراري لانتقال الحرارة ككل وجد أن الطبقات الشبكية ذات النسبة (1/δ) تساوى 0.1925 أكثر فاعلية من الطبقات الأخرى.

وقد تم وضع النتائج المعملية في معادلات تجريبية لإمكانية استخدامها في حساب معاملي انتقال الحرارة والاحتكاك.

ABSTRACT

An experimental investigation of heat transfer and pressure drop for turbulent flow of air in an annular tube of 48.1 mm outside diameter and 24.4 mm inside tube diameter are conducted. The inner tube, which is made of polished aluminum, heated under constant heat flux and winded by aluminum mesh-layers, while the outer tube is made of plastic (PVC) and insulated from outside with glass wool. Mesh-layers are used as an augmentative device with different thickness ($2.3 \le t \le 11.9$ mm). Reynolds number varied between 3000 and 20000 and Prandtle number was about 0.7. Friction factor are found to be 2.14 : 106.8 times the empty annular tube values, while Nusselt number are found to be 1.8: 3.7 times the empty annular tube at the same Reynolds number. The Nusselt numbers are found to be 1.29: 5.2 times the empty annular tube values based on constant pumping power and constant basic geometry. When compared with empty annular tube at constant heat duty and constant heat transfer area,

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a maximum reduction in pumping power of 76% can be achieved with augmented annular tubes. It is concluded that the use of this class of augmentative device is generally more effective for the case when mesh-layers thickness (t) is nearly equal to 2.3 mm (t/δ = 0.1925, where δ is annular gap thickness). Finally, correlations are developed for the prediction of friction factors and Nusselt number.

Introduction

Recently, convective heat transfer in porous media has an important application in geophysics, biomedical, agricultural engineering and many industrial applications These applications include the following: porous journal bearing, nuclear waste disposal, porous flat plate collectors, packed bed thermal storage, fibrous insulation, grain storage and drying, paper drying and food storage, heat exchangers, oil production, electronic cooling, heat pipes, filtration, and chemical reactors.

A review of the related literature shows that most of the previous studies treated the annular geometry completely filled with porous material and studied the natural convection heat transfer mode [1-4] Different numerical methods were used to study the operating parameters such as Rayleigh number, porosity, and aspect ratio on the heat transfer characteristics. Sultan and El-Shafei [5] investigated experimentally the local heat transfer of water flowing through vertical tube packed with spherical steel particles of different diameters The tube is heated electrically with a constant heat flux while Reynolds number is varied Chikh et al. [6] through using Darcy-Brinkman model, an analytical solution was obtained for forced convection in an annular duct partially filled with porous medium The inner tube is exposed to a constant heat flux and the outer is thermally insulated. The effect of the permeability, thermal conductivity, and thickness of the porous material are investigated Fowler and Bejan [7] investigated numerically the heat transfer, friction, and mechanical interaction between external laminar perpendicular flow and a solid surface covered by a layer of fibers They found that at porosities lower than a critical value, the fiber cover acts as insulation, while at higher porosities the fibers augmented the heat transfer El-Nimr and Alkom [8] presented a numerical solution of transient, developing, forced convection flow in concentric annuli partially filled with porous substrates The effect of geometry, solid matrixes metal and the fluid on the hydrodynamic and thermal behavior was investigated They found that porous substrates might improve Nusselt number by 1200% while keeping other flow and geometrical parameters fixed Peterson and Chang [9] through an experimental work studied two-phase heat dissipation in high conductivity porous channel heat sinks They used sintered copper particles inside a rectangular copper channel as a poroos material, subcooled water as a working fluid, flow rate is changed, and its subcooling is varied. Numerical computations were performed for laminar flow with high porosity media composed of small diameter silicon, carbon fibers aligned transverse to the streamwise flow specifically flow over a backward facing step by Martin [10] El-Kady [11,12] studied experimentally the effect of partially filling porous medium with localized discrete heat source attached to the bottom wall of a horizontal channel and operating parameters (Re, Gr, and q) on heat transfer characteristics.

From the above review it's concluded that there were a very little literature dealt with forced convection through annulus filled with porous material So, the aim of the present work is to investigate the enhancement of forced convection in an annulus partially filled with a porous aluminum mesh layers The effect of different parameters, such as the mesh layers thickness and operating parameters on Nusselt number are investigated

Experimental Apparatus

A schematic diagram of the experimental setup is shown in Fig.1-a It consists of a centrifugal blower (17) connected from its suction side to the test section outlet through a flexible connection (16) to avoid transmission of blower vibration to the test tube The inlet side of the test tube stabilizing section (4) is connected to a bell mouse connection (1) provided with fine screen (2) to ensure fairly uniform flow of air. A graduated opening (15) is used to control the airflow. The air velocity is measured in a section free of the porous media (4) by a hot-wire anemometer (3) in both X and Y direction and integrated to obtain the mean inlet velocity The test section consists of two-eoncentric tubes each of length 1980 mm, the outer tube (14) has an inside diameter of 48 1 mm was made of plastic (PVC) The inside tube of the annulus (7) is made of aluminum with outer diameter of 24 2 mm It consists of two sections: the heated section of length 980-cm, and the unheated one of length 1000-mm and is separated by a Teflon conneetion (8). It is plaeed inside the outer tube in a symmetrical position by the help of three groups eaeh consists of three fine threaded bolts (6) Each group consists of 3 bolts of 3 mm diameter at 120 ° to form a uniform annulus, one at both ends and the third one at the middle section. The unheated seetion is covered at the front end by a semispherical Teflon smooth end (5) to ensure no chock of air stream An entrance length (4) upstream of the test section with a length of approximately 40 hydraulic diameter is used for the redevelopment of the flow whieh is necessary for the pressure drop and heat transfer measurements Two static pressure taps (9,13) were provided at inlet and outlet from the heated section. Pressure drops are measured in the eases of porous and non-porous annulus by means of digital micro-manometer (10) with an aeeuracy of ±1 Pa. The heated section of the inner tube of the annulus is heated by a helical Nickel-Chrome wire (18) as shown in Fig 1.b It is placed in a glass tube (20) of diameter 12-mm The glass tube is filled with fine sand particles (19) and is closed at both ends by gypsum and inserted inside the inner tube of the annulus The gap between the glass wall and the inner surface of the aluminum tube is filled with aluminum powder (21) in order to insure uniform distribution of heat and prevent the heater from burning up. The inner tube of the annulus of the heated seetion is wounded with aluminum mesh layers (12) to study the effect of mesh-layers thickness on both pressure drop and heat transfer characteristics The test section was well insulated with fiber glass wool (11). The air temperature at inlet and outlet of the test section are measured by using two copper-constantan thermoeouples of type (J) and the air properties are calculated as a function of the average air temperature. The radiative heat losses from the porous media to the inner surface of the outside tube in the case of porous annulus tends to be zero because of the very small diffussivity of polished aluminum An autotransformer is used to control the heat input to the heat source, which was measured by an ammeter and voltmeter. The surface temperature of the inner tube of the annulus is measured by 7 copper-constantan thermocouples, and another 7 copper-eonstantan thermocouples are glued on the inside surface of the outer of the annulus tube corresponding to those fixed on the outer surface of the inside tube The temperature signals are then transferred to a digital temperature recorder (Yokogawa) of a sensitivity of 0 1 °C. The porosity, ε, was evaluated using the expression, $\varepsilon = (V - V_S)/V$, where V is the volume of the ehannel occupied by mesh-layers and V_S is the volume of mesh-layers (determined by dividing the weight of the mesh-layers by its density) This porosity took the value of 0 91 Nearly one hour was needed to reach the steady-state eondition, which is regarded, as the temperature readings with a time of about 20 minutes have no ehange

Procedure:

Determining the flow friction and heat transfer results in an empty annular tube and comparing them with the available correlations first standardized the experimental setup. Steady state values of friction factors and mean Nusselt numbers for uniform heat flux heating of air were then determined with each of the different mesh-layers thickness which is winded on the inner tube of the annular tube One can take into consideration that the minimum thickness of mesh layers used (t=2 3 mm) is restricted by the available in the market of this kind of mesh materials. While the maximum thickness (t =11.9 mm) is restricted by the annular gap thickness of the tested annulus. The characteristics of the augmented annular tubes are shown in table 1.

Table 1 Characteristics of tubes partially filled with mesh screens

Tube number	D_h (mm)	t (mm)	ε	L (mm)	t/δ
e	23.90	0.00	1.00	980	0.0000
1	8.840	2.30	0.987	"	0 1925
2	5 015	4 60	0.972	"	0 3849
3	3.210	7.10	0.954	"	0.5941
4	1. 702	11.9	0 910	"	0.9960

Flow and Heat Transfer Characteristics

Figure 1.e shows the detail cross-section of the augmented tube under test, and from whieh the following charaeteristics can be obtained:

The mean porosity ($\bar{\varepsilon}$) of the augmented annular tubes depends on the void fraction of mesh-layers material (ε), inner and outer diameters of the annular tubes (D_o, D_i), and the thickness of mesh layers (t). It can be calculated, with the aid of Fig 1-C and the above mentioned d .inition of porosity as follows:

$$\bar{\varepsilon} = 1 - 4\, t\, (1-\varepsilon)(t + D_i)/(D_o^2 - D_i^2) \quad (1)$$

The free cross-sectional area of the empty (A_{ce}) and augmented (A_{ca}) annular tubes can be calculated respectively as follows:

$$A_{ce} = \pi\, (D_o^2 - D_i^2)/4 \quad (2)$$

$$A_{ca} = \pi\ \bar{\varepsilon}\, (D_o^2 - D_i^2)/4 \quad (3)$$

The volumetric diameter defined as four times the volume for flow per unit length divided by the area of wetted surfaee per unit length was used as the hydraulic diameter For empty annular tube, it can be calculated as follows:

$$D_{he} = D_o - D_i \quad (4)$$

Based on the hydraulic diameter of the empty annular tube, Reynolds number of the empty (Re_{he}) and augmented (Re_a) annular tubes can be calculated from the following equation.

$$Re_{he} = Re_a = u\, D_{he}/\nu \qquad (5)$$

Correspondingly, Nusselt number of the empty (Nu_{he}) and augmented (Nu_a) annular tubes based on D_{he} can be written in the following form

$$Nu_{he} = Nu_a = h_i\, D_{he}/k \qquad (6)$$

Where h_i is the heat transfer coefficient based on the inside diameter of the annular tube and was calculated as follows

$$h_i = Q/[A_i\, (\Delta T_i)_m] \qquad (7)$$

Where

$$Q = I\ V \qquad (8)$$

$$A_i = \pi\, D_i\, L \qquad (9)$$

$$(\Delta T_i)_m = [(T_w - T_i) - (T_w - T_o)]/ Ln[(T_w - T_i)/(T_w - T_o)] \qquad (10)$$

and, $T_w = 1/7 \sum_{n=1}^{7} (T_{wn})$ (11)

Also, the friction factors of empty (f_{he}) and augmented (f_a) annular tubes can be calculated using the following correlation

$$f_{he} = f_a = \Delta P\, D_{he}/ (L\, \rho\, u^2/2) \qquad (12)$$

In view of the changing cross-section of the augmented tubes, the mean velocity u_g in the void space can be written as

$$u_g = u / \varepsilon \qquad (13)$$

And the hydraulic diameter of the augmented annular tubes based on mesh layers characteristics can be calculated with the aid of the volumetric diameter defined above, from the following relationship

$$D_{ha} = [4\ \bar{\varepsilon}\, (D_o - D_i)]/[4 + \bar{a}\, (D_o - D_i)(1 - \bar{\varepsilon})] \qquad (14)$$

Where ($\bar{a}$) is the surface area per unit volume of mesh material and is calculated from mesh layer characteristics and found to be ($\bar{a} = 22\times10^3\ m^{-1}$)

Thus the Reynolds numbers for the augmented annular tubes, Re_{ha}, formed with the hydraulic diameter (D_{ha}) and the gap mean velocity u_g is

$$Re_{ha} = u_g\, D_{ha}/\nu \qquad (15)$$

Nusselt number and friction factor of the augmented annular tubes based on the hydraulic diameter (D_{ha}) can be evaluated as follows

$$Nu_{ha} = h_i\, D_{ha}/k \qquad (16)$$

$$f_{ha} = \Delta P\, D_{ha} / (L\, \rho\, u_v{}^2/2) \quad (17)$$

Results and Discussion

In order to standardize the experimental apparatus, test runs of the empty annular tube (smooth annulus without mesh layers) were conducted first and comparing it with the available correlations. Friction factor and mean Nusselt number for uniform heat flux were then determined with each of the different mesh layer thickness, which is winding around the inner tube of the annulus.

Empty Annular Tubes

The empty annular tube turbulent flow friction factors (f_e) measured for air, are plotted in Fig.2 versus Reynolds number, Re_e, and based on the hydraulic diameter of the empty annulus. The figure shows that f_e values are higher than the isothermal friction factor correlation given by Blasius by about 10%. The friction factor of the present study can be correlated with Re_e as follows:

$$f_{he} = 0.3277\ Re_{he}^{-0.2415} \quad (18)$$

in the range $10^4 < Re_{he} < 4\ 10^4$ with standard deviation of 4%

Results of heat transfer runs of empty annular tube of this study are presented in Fig.3 along with results of Ref [13]. As shown from the figure, the experimental Nusselt numbers are 5-7 % lower than those suggested by the previous references. The present data are correlated with Reynolds number with a standard deviation of 7% as follows

$$Nu_{he} = 0.01483\ Re_{he}^{0.8134} \quad (19)$$

All the observation mentioned above indicated that the experimental apparatus and the test procedure used in this study can generate reliable data

Augmented Annular Tubes

The experimental heat transfer and friction results are presented on the coordinates Nu_a versus Re_a and f versus Re_a. These coordinates were based on the hydraulic diameter of the empty annular tube rather than the augmented annular tube in order to directly show the change in Nu and f obtainable with augmented annular tubes over a comparable empty annular tube for a given mass velocity [14]

Turbulent flow friction factors in the augmented annular tube were found to be higher compared to the empty annular tube under the same operating conditions, as seen from Fig.4. The increase in friction factor ranged from 123 to 114% for tube 1 (partially filled with 2.3 mm mesh layer thickness) to as high as 10580 to 7940 % for tube 4 (completely filled with mesh layers 11.9 mm thickness). The other tubes produced friction factor variations intermediate between 114 and 10580%. Although, the turbulent flow friction factor in these tubes was found to be dependent on mesh layer thickness. it was not independent of Reynolds numbers at high flow rate

The inside heat transfer coefficient for the turbulent flow of air in one empty annular tube and for augmented annular tubes were analyzed in terms of Nu_a-Re_a relationship and Fig.5 shows this relation for all the augmented tubes. The Nusselt numbers of augmented annular tubes were higher compared to the empty annular tube at the same Reynolds number. The augmented annular tube completely filled with mesh layers (11.9-mm thickness), tube 5, produced a maximum improvement in Nusselt number of the order of 270%. Whereas tube 1 augmented by a mesh layer of thickness 2.3 mm, yielded an improvement of 80% only in Nusselt number, compared to the empty annular tube

To show the effect of mesh layers thickness on the heat transfer, Nusselt number is plotted in Fig.6 against mesh-layer thickness, represented by the term (t/δ) at different Reynolds number. The figure shows that, Nu increases with t/δ in the range 0.384[illegible]. This is due to the increase in effective thermal conductivity of the whole annulus. Where the effective thermal conductivity increases with t/δ. Also, it is shown from the figure that Nu at t/δ=0.1925 is higher than that at t/δ=0.3849. This increase, may be, is due to the increase in disturbance near the surface of the inner tube with respect to the decrease in effective thermal conductivity of the annular gap. The Nusselt number is found to increase with the increase of Reynolds number and the decrease of mesh-layers thickness, while friction factor increases with the decrease of Reynolds number and increase mesh-layers thickness

In view of changing cross-section of the augmented annular tube, the volumetric diameter, as defined above, is used as the hydraulic diameter D_{ha}, in the calculation of Nusselt number Nu_{ha}, friction factor f_{ha}, and Reynolds number Re_{ha}. The f_{ha}-Re_{ha} plot is shown in Fig.7. The f_{ha}-Re_{ha} plot, shown in Fig.7 reveals that friction factor in partially filled [illegible] tubes are still differ from that of the empty annular tube. This finding is also exists for the [illegible] plot shown in Fig.8, which indicates that there is an increase of Nu_{ha} for the augmented annular tube over that of the empty annular tube

Correlations

As expected Nusselt number and friction factor were found to depend on Reynolds number and mesh layers thickness. An attempt was made using the present experimental results as well as the experiences of previous investigators, to correlate the friction factors and Nusselt numbers in terms of Reynolds number, mesh layers thickness to gap thickness ratio (t/δ) Prandtle number and diameter ratio of annular tube. The following correlations were obtained

Friction Factor Correlation

$$f_a = 9.303\, Re_a^{n1}\, f(t/\delta) \qquad (20)$$

Where

$$n_1 = 0.0947 - 0.981\,(t/\delta) - 0.545\,(t/\delta)^2$$
$$f(t/\delta) = 78.413\,(t/\delta)^3 - 51.225\,(t/\delta)^2 + 12.635\,(t/\delta) - 1$$

$$3000 \le Re_a \le 10000, \text{ and } 0.1925 \le (t/\delta) \le 1.0$$

The above correlation predicts the values of friction factor, which agrees with experimental results within ±10% as shown in Fig (9).

Heat Transfer Correlation

$$Nu_a = 0.0356\, Re_a^{0.81}\, Pr_f^{0.4}\, (D_o/D_i)^{0.16}\, N(t/\delta) \tag{21}$$

Where:

$N(t/\delta) = 1 - 1.457\,(t/\delta) + 2.101\,(t/\delta)^2$

$3000 \le Re_a \le 20000$, and $0.1925 \le (t/\delta) \le 1.0$, and $D_o/D_i = 1.988$

This correlation was found to be within ± 4% of the experimental data values, as shown in Fig (10).

Overall Tube Performance

~~Bergles~~ et al.[15] outlined several practical criteria for evaluation of the performance of augmented tubes, relative to a smooth tube, and performance ratios based on all criteria have been worked out for the tube side fluid. No published information is available on the application of these criteria for turbulent flow heating of fluids in annular tubes partially filled with porous material (aluminum mesh-layers). In the present work expressions for the performance ratios R_3, and R_4 have been determined for criteria 3, and 4

Criterion 3:

This criterion aims at improving the heat duty for the case of constant pumping power and for constant basic geometry of the exchanger. The performance ratio R_3 is given by:

$$R_3 = (Nu_a/Nu_e)_{Di,\, Do,\, L,\, P,\, Ti,\, \Delta Tm} \tag{22}$$

For equal pumping power in smooth and augmented tubes

$$A_{ce}\, f_e\, Re_e^3 = A_{ca}\, f_a\, Re_a^3 \tag{23}$$

For turbulent flow, the friction factor f_e in a smooth annulus is given by

$$f_e = 0.3277\, Re_e^{-0.2415} \tag{24}$$

On substituting for f_e in eq.(24), and on simplification, we get.

$$Re_e = (3.0516\, f_a\, Re_a^3\, A_{ca}/A_{ce})^{0.3625} \tag{25}$$

Knowing Re_e, the corresponding h_e for the smooth annulus is obtained and R_3 then becomes:

$$R_3 = h_a \text{ at } Re_a / h_e \text{ at } Re_e = Nu_a \text{ at } Re_a / Nu_e \text{ at } Re_e \tag{26}$$

Criterion 4:

This criterion aims at a reduction in the pumping power for equal heat duty and equal surface area. The performance ratio R_4 is given by:

$$R_4 = (P_a/P_e)_{Q,\, Di,\, Do,\, Li,\, \Delta Tm} \tag{27}$$

For equal heat duty and equal heat transfer area

$$Q_a/Q_e = [illegible] \tag{28}$$

$$Q_e = Nu_e \, \Delta T_m \, A \, k / D_{he} \tag{29}$$

$$Q_a = Nu_a \, \Delta T_m \, A \, k / D_{ha} \tag{30}$$

$$Q_a/Q_e = Nu_a/Nu_e = 1.0 \tag{31}$$

$$Nu_a = Nu_e \tag{32}$$

For turbulent flow, Nusselt number Nu_e in a smooth annulus is given by

$$Nu_e = 0.01483 \, Re_e^{0.8145} \tag{33}$$

On substituting for Nu_e in eq (32) we get,

$$Re_e = (67.431 \, Nu_a)^{1.228} \tag{34}$$

Knowing the values of Nu_a, the equivalent smooth tube Reynolds number Re_e can be calculated.

Knowing P_e at Re_e and P_a at Re_a, the performance ratio R_4 can be found as follows

$$R_4 = P_a \text{ at } Re_a / P_e \text{ at } Re_e \tag{35}$$

Fig 11 shows the variation of the performance ratio R_3, with equivalent smooth tube Reynolds number Re_e. The augmented tube 1 of mesh-layer thickness of [illegible] mm shows an improvement of 421% in heat transfer at Re_e nearly equal 10^3 compared to [illegible]% only at Re_e nearly equal to 3000. While tube 4, fully filled of mesh layers (11.9 mm thickness) shows the least improvement of only 29 to 45%, over the entire range of Re_e. An important observation about these tubes is that, R_3 improves greatly as mesh layers thickness [illegible] all the augmented tubes studied. Performance ratio based on criterions 3 was generally found to increase with an increase in Re_e until they reach maximum values at certain Re_e values between 10^3 to $2x10^4$, and then decrease with further increase in Re_e. A similar trend was also observed by [16] of tube roughened by helically coiled ribbons.

The performance ratio based on criterion 4 is plotted in Fig 12, for tubes 1, 2, 3, and 4. This figure shows that a maximum reduction of 76% can be obtained in pumping power using tube 1 while there is a maximum increase in pumping power of 104% for tube [illegible] (partially filled with mesh layer of 2.1 mm thickness). Again the change in pumping power (R_4) is found to be dependent on Re_e in all the augmented annular tubes used in the present study. The change in pumping power increases with the increase of Reynolds number Re_e until they reach minimum values at certain Reynolds number Re_e, $2x10^4 < Re_e < 2.5x10^4$, and then decrease with further increase in Re_e.

Finally, it is very important to evaluate the enhancement process as a whole. Therefore, we must calculate the effectiveness (efficiency index) of the process (η), which defines the ratio between the rate of increase in Nusselt number and the rate of increase in friction factor according to [17] as follows:

$$\eta = (Nu_a/Nu_e)/(f_a/f_e) \tag{36}$$

From the experimental results for Nusselt numbers and friction factors, it is easy to calculate the term (η) in case of augmented annular tubes. These results are shown in Fig 13 as a relation between η and Re_{hc}, from which it is evident to conclude that the enhancement is more efficient in case of tube 1, partially filled with 2 3 mm mesh layer thickness In general, from the above discussion, it is observed that tube 1, with minimum mesh layer thickness, perform better than those with mesh layer thickness larger than 2.3 mm.

Conclusions

The following conclusions can be drawn from results of this investigation.

1 Friction factor increases with the increase of mesh layers thickness and the decrease of Reynolds nomber
2. Nusselt number increases with both Reynolds number and mesh layers thickness.
3 Compared with empty annular tube at constant pumping power and basic geometry, an improvement as high as 420% was obtained in the heat duty using mesh-layers of 2 3 mm thickness (t/δ = 0.1925). While the improvement was as low as 29% in heat capacity using mesh layers of 11 9 mm thickness ($t/\delta \cong 1$) in the entire range of Reynolds number.
4 On the basis of constant heat duty and constant heat transfer area, augmented tube with minimum mesh layers thickuess (t/δ = 0.1925) performs the best over the entire range of Reynolds number. A reduction in pumping power of about 54:76 % can be achieved using mesh-layers of thickness 2.3 mm (t/δ =0.1925). While a maximum increase in pumping power of about 104% was achieved with mesh layers of 7.1 mm thickness (t/δ =0.5941) over the entire range of Reynolds numher
5. The use of this class of augmentation devices are generally more effective for the case when mesh layers thiekness is nearly equal to 2.3 mm (t/δ =0.1925).
6 There are cases, however, when the size of the heat exchanger can not be changed, but its heat transfer must be increased. Under this condition, the use of mesh layers was found to be desirable provided that the high fluid pressure drop is tolerable

NOMENCLATURE

$\bar{a}$	specific area per volume of mesh-layers material, m^{-1}	Q	heat transfer rate, W
A	surface area, m^2	t	mesh-layers thickness, m
A_c	cross-section area, m^2	T	temperature, K
D	annular tube diameter, m	$(\Delta Ti)_m$	logarithmic mean temp difference defined by eq (10), K
h	heat transfer coefficient, W m^{-2} K^{-1}	u	air mean velocity in annular tube free of mesh-layers, m/s
I	electric current, Amp.	u_v	air mean velocity in the void space, m/s
k	thermal conductivity, W m^{-1} K^{-1}		
L	tube length, m		
P	pumping power, W		
Δp	pressure drop, Pa		

Dimensionless Groups

f	friction factor
Nu	Nusselt number
Pr	Prandtle number
Re	Reynolds number
R_3	thermal performance ratio, defined in eqn (26)
R_4	thermal performance ratio, defined in eqn (35)

Greek Symbols

δ	annular gap thickness $(D_o - D_i)/2$, m
ε	porosity of mesh-layers material
$\bar{\varepsilon}$	mean porosity of the augmented annular tube eqn.(1)
η	effectiveness, eqn.(36)
ν	fluid kinematic viscosity, $m^2 s^{-1}$
ρ	fluid density, kg m^{-3}

Subscripts

a	augmented case based on ... diameter of the empty annular ...
e	equivalent empty annular ...
f	based on fluid properties
ha	based on hydraulic diameter of augmented annular tube
he	based on hydraulic diameter of empty annular tube
i	inside diameter, or inlet section
o	outside diameter, or exit section
w	wall

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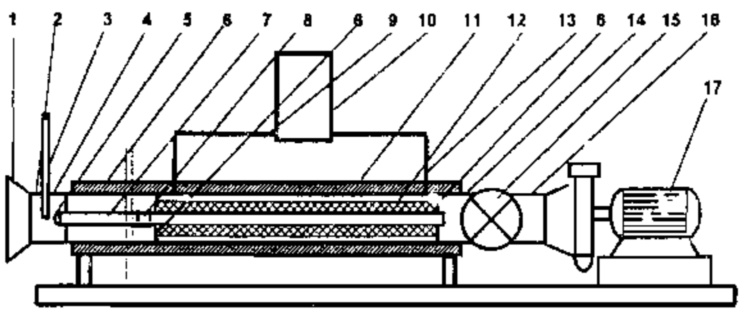
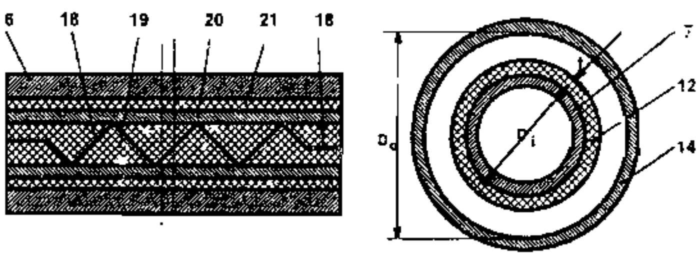


A)- Test rig



B)- Inner tube construction C)-Cross section view

Fig. (1) Details of the experimental set-up

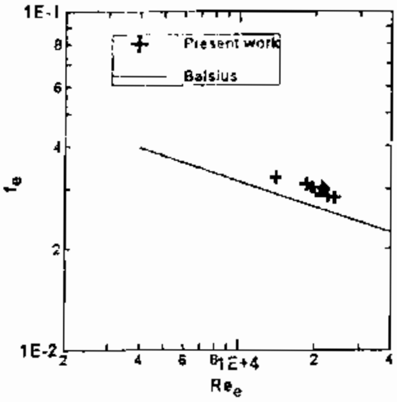


Fig (2) Friction factor versus Reynolds number for empty annular tube.

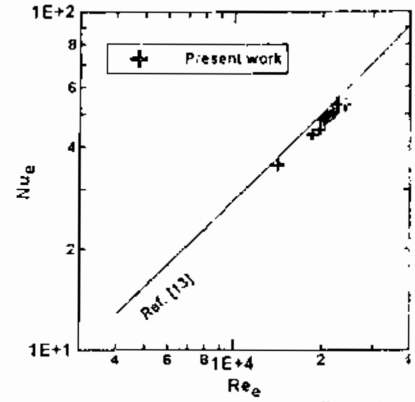


Fig (3) Nusselt number versus Reynolds number for empty annular tube

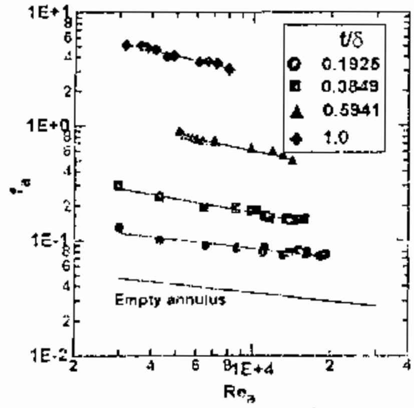


Fig (4) Friction factor versus Reynolds number for augmented annular tubes

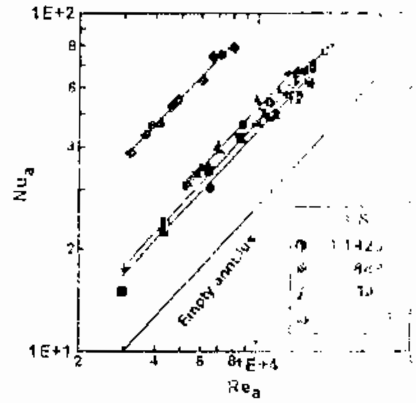


Fig (5) Nusselt number versus Reynolds number for augmented annular tubes

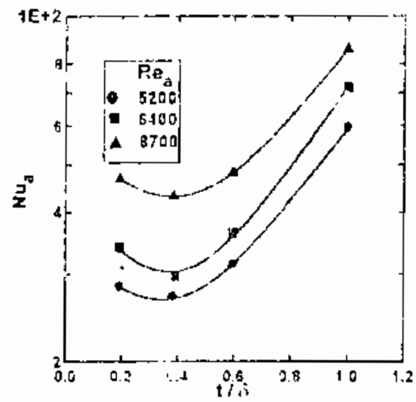


Fig (6) Augmented tube Nusselt number versus mesh layers thickness.

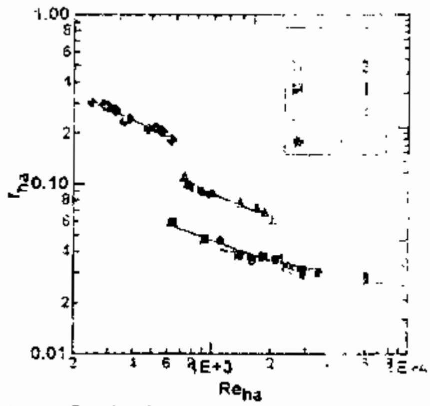


Fig (7) Friction factor versus Reynolds number for augmented annular tubes

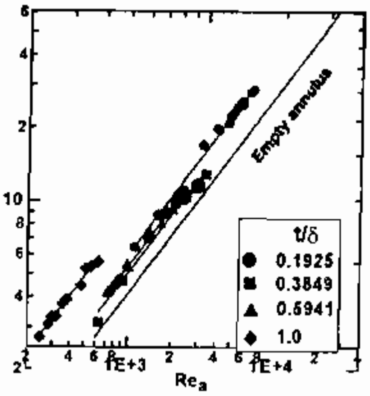


Fig (8) Nusselt number versus Reynolds number for augmented annular tubes

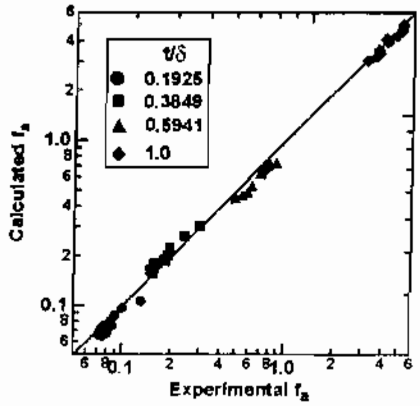


Fig (9) Calculated versus experimental friction factor for augmented annular tubes.

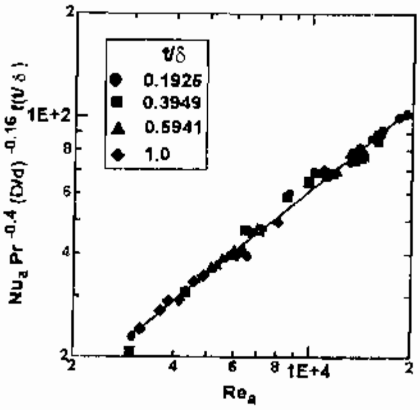


Fig (10) Representation of general equation of Nusselt number for augmented annular tubes.

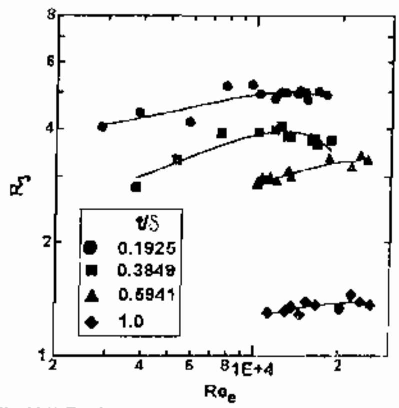


Fig (11) Performance ratio R_3 versus equivalent Reynolds number for augmented annular tubes

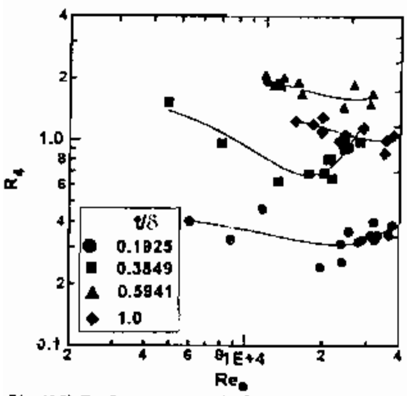


Fig (12) Performance ratio R_4 versus equivalent Reynolds number for augmented annular tubes

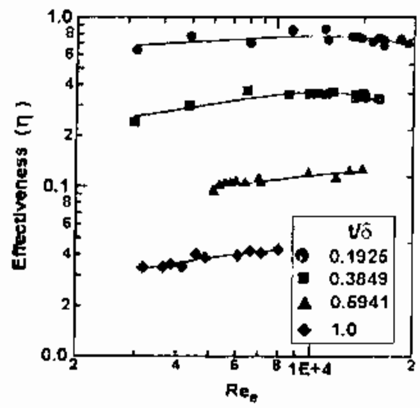


Fig (13) Augmented tubes effectiveness versus equivalent empty tube Reynolds number .