CONVECTIVE HEAT TRANSFER AND FRICTION LOSS IN CYCLICALLY DIVERGING- CONVERGING ANNULAR TUBES

انتقال الحرارة بالحمل والفقد في الاحتكاك في أنابيب حلقية متباعدة متقاربة دورية المقطع

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الخلاصة : يتضمن هذا البحث دراسة عملية للسريان المضطرب وانتقال الحرارة للهواء في أنابيب حلقية متقاربة متباعدة ذات مقطع متكرر. الجهاز يتكون من مجموعة من الأنابيب الداخلية ذات مقاطع متقاربة متباعدة دورية وأنبوب خارجي ذو قطر داخلي ثابت . وقد تمت دراسة حالتين ، الأولى تسخين سطح الأنبوب الداخلي بفيض حراري منتظم بينما الأنبوب الخارجي معزول ، والحالة الثانية تسخين الأنبوب الخارجي بفيض حراري منتظم والأنبوب الداخلي معزول . الأنابيب الداخلية ذات طول كلي ثابت وقطر خارجي ثابت ، كل منها يتكون من أربعة أجزاء متشابهة ذات مقاطع متقاربة متباعدة ذات قطر عنق ثابت . وتم تغيير طول المقطع المتقارب والمقطع المتباعد مع المحافظة على طول جزء ثابت حيث تم إجراء التجارب على نسب مختلفة للطول المتقارب إلى الطول المتباعد (النسبة الطولية) وهي : 3.67 ، 1.8 ، 1.0 ، 0.56 و 0.27 مع رقم رينولدز من 3.1×10^3 إلى 3.3×10^4 ، ورقم برانتل 0.71.

وقد أظهرت النتائج زيادة معامل الاحتكاك ومعامل انتقال الحرارة في الأنبوب الداخلي ذو المقطع المتغير (المتقارب-المتباعد) عنه في الأنبوب الداخلي ذو المقطع الثابت . ففي حالة التسخين بفيض حراري منتظم داخلي كانت الزيادة في معامل انتقال الحرارة للأنبوب ذو النسبة الطولية 1.0 أعلى قيمة وتساوي 160% بالمقارنة مع الأنبوب ذو المقطع الثابت ، بينما في حالة التسخين بفيض حراري ثابت خارجي كانت الزيادة في معامل انتقال الحرارة للأنبوب ذو النسبة الطولية 1.0 أعلى قيمة وتساوي 117% بالمقارنة مع الأنبوب ذو المقطع الثابت . كما أظهرت النتائج زيادة معامل الاحتكاك في جميع حالات الأنبوب المتغير بنسبة تتراوح بين 67% و 650%، وكانت أقصى قيمة للزيادة في حالة الأنبوب ذو النسبة الطولية 0.27 وأقل قيمة في حالة الأنبوب ذو النسبة الطولية 3.67

وبمقارنة النتائج تبين أن أكبر معامل فعالية (مؤشر الكفاءة) للأنبوب ذو النسبة الطولية 1.0 والمسخن بفيض حراري من الداخل وذلك عند نفس سرعة السريان .

ABSTRACT

An experimental work for the flow and heat transfer in turbulent airflow in a cyclically diverging-converging annular tubes is presented. Two cases are presented in this work, where the inner tube is converging-diverging and the outer tube is of constant diameter. In the first case the inner tube is heated internally with an uniform heat flux, while, the outer tube is insulated tube. In the second case the outer tube is heated with an uniform heat flux while the inner tube is an insulated tube. Such type of section configuration is often used in compact heat exchangers to enhance the heat transfer and improve their thermal performance. In this work five test sections are investigated for each case. The inner tube of each case consists of four of identical convergent-divergent parts, and has a constant total length. Each convergent-divergent part has constant length, inlet and throat diameters, while the length of the convergent part to the length of the divergent part are varied as 3.67, 1.8, 1.0, 0.56, and 0.27. The experimental study is carried out in the Reynolds number range between 3.1×10^3 and 3.3×10^4 while the Prandtl number value is about 0.71

The results of the present work show that friction losses as well as heat transfer of cyclically diverging-converging annulus are higher than those of the constant cross section area one. For the internally heated case, an enhancement as much as 160% in the heat transfer coefficient is reported for the test section of 1.0 divergent-convergent length ratio. For the case of externally heated case an enhancement as much as 117% in heat transfer is reported for test section of 1.0 convergent divergent length ratio. The increase in friction factor for both cases of the cyclically diverging-converging annuli over that of the constant area one ranges from 75 to 650 %, with its maximum value at length ratio of 0.27 and minimum value at length ratio of 3.67.

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Comparative evaluation based on the efficiency index (effectiveness) shows that the annular tube having divergent convergent length ratio 1.0 performs better than the other tubes. It is concluded also that, the internally heated annular tubes perform better than the externally heated ones at the same flow velocity.

Key words: Forced convection in a divergent-convergent annulus.

1. INTRODUCTION

Whereas a constant cross-section area characterizes conventional ducts, there are numerous heat exchange applications involving ducts whose flow cross-section varies periodically along the stream-wise direction. The need for more efficient and compact heat exchanger has led to enhance heat transfer by the use of unconventional flow passages.

Different methods have been proposed to improve heat transfer characteristics, including passive methods that require no external power such as threaded surfaces, extended surfaces, swirl flow devices, and active methods that require external power such as surface or fluid vibration, fluid injection and suction. Fu, et al (1995a and b) tried to enhance the heat transfer rate of tube flow by inserting a coaxial inner tube into an outer-tube to deflect the fluid to the hot wall of the outer tube. It was found that, since the configuration of the inner tube is very simple, the accompanying pressure drop is not very serious and the results showed that this method does meet the goal of heat transfer enhancement. Garimella and Christensen (1993 and 1995c) studied the heat transfer and pressure drop of spirally floated annuli. They reported that Nusselt numbers were between 4 to 20 times the smooth annulus values in the low Reynolds number range, while turbulent enhancements were between 1.1 and 4. Gokhman and Kirpicov (1972) studied experimentally the turbulent motion of airflow along the converging-diverging duct. They reported that intensification of heat transfer was high and there was also a moderate increase in the pumping power. Araid et al. (1983) studied heat transfer and hydrodynamic resistance in a transformer oil plate heat exchanger type converging-diverging sides. They found that heat transfer intensification by a factor of 1.55 is achieved and the pumping power is increased by a factor of 0.35. Awad et al. (1986) studied experimentally the heat transfer and hydrodynamic resistance for airflow through a repeated converging-diverging duct. The converging part was of 5-mm length and the diverging one was of 25-mm length, and the slotted sides gab was equal to 20 mm. The results showed that heat transfer has increased by a factor of 1.5 and the pumping power by a factor of 0.238.

Rabie, et al (1989) investigated experimentally the turbulent flow and heat transfer of drag reducing additives in periodic convergent divergent pipe. They concluded that heat transfer and flow resistance are much higher in convergent divergent tubes than those of constant area pipe. They found that the lowest pressure drop and the lowest heat transfer occurs in the constant area tube and the highest values is that of the convergent divergent length ratio 1.0 with $\alpha=\beta=16°$

Farag et al (1999) experimentally investigated the heat transfer and pressure drop characteristics for annuli formed with spirally coiled and spirally grooved flutes on the outer wall of the inner tube. Air and water were used as the cold and hot fluids respectively. The water flows through the inner pipe and the air flows through the annular region. They concluded that Nusselt number and friction factor increase with spirally fluted and spirally grooved annuli over that of the smooth annulus. They concluded also that the Nusselt number of the spirally fluted annuli was higher than the corresponding smooth annulus values and grooved tube annuli.

Araid (2000b) investigated experimentally the air heat transfer and pressure drop in tubes inserted with different rod with different configurations. The rod inserts had a constant step equals 60-mm and the rod step ratio changes from 5:1 to 1:5 and the Reynolds number ranged between 18000 and 95000. The inner diameter of the tube, the outer and inner diameters of the rods were equal to 28, 12 and 6 mm respectively. It was concluded that the heat transfer enhancement for the effective case was about 58 % compared to the smooth tube, meanwhile, the mean friction losses increased by about 130 % in the same Reynolds range.

Sultan,A.A., and Sultan, G.I. (2000a) investigated experimentally the heat transfer and pressure drop of air flowing through horizontal annular tubes partially filled with aluminum mesh layers. The gap thickness of the annular tubes used was equal to 11.9 mm and the mesh layers thickness used were in the range (2.3 to 11.9 mm). The results show that friction losses and Nusselt number increase with layer thickness. They show that friction factors are 2.14 - 106.8 times the empty annular tube values, while Nusselt numbers are found to be 1.29 - 3.7 times the empty annular tube at the same Reynolds number.

The objective of the present work is to investigate experimentally, heat transfer and frictional loss characteristics of a turbulent flow in *cyclically diverging-converging* annular tubes. Constant heat flux modes of heat transfer are considered where the flow is heated by the inner surface of the annulus with the outer surface insulated in the first case. In the second case, the flow is heated by the outer surface of the annulus with the inner tube insulated. Five annular tubes of different divergent-convergent length ratio are used in each case.

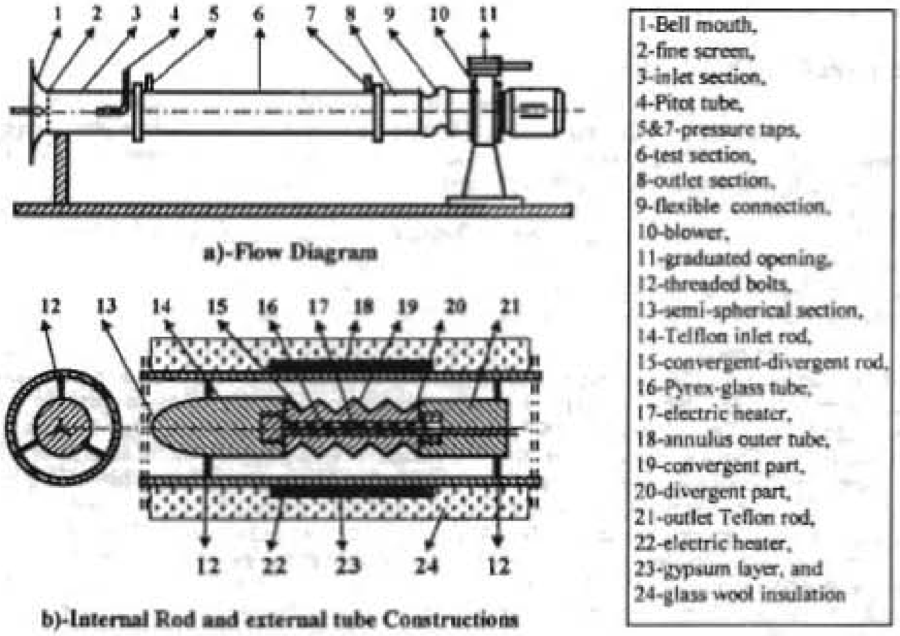
2. EXPERIMENTAL TEST RIG

A schematic diagram of the experimental test rig is illustrated in figure (1-a), while figure (1-b) shows the details of the test section. The test rig is basically an open air flow pipe system, consists of a centrifugal blower (10), test section (6), and bell mouth entrance section (3). Air is drawn by the blower through the bell mouth entrance section, provided with fine screen (2), to the annular pipe test section. The blower suction is connected to the test section through a flexible connection (9) to isolate blower vibration to avoid transmission of blower vibration to the test section (6). The bell mouth entrance length of approximately 50 hydraulic diameter ensures fully developed flow in the test section. The blower discharge outlet is provided with a variable area exit (11) to control the airflow through the system.

The test section, shown in figure (1-b), consists of two-concentric tubes. The inside tube of the annulus is 1100 mm length and consists of three sections; one heated and two unheated connected to each other with the aid of threaded parts. The two unheated sections (14 &21) are made of Teflon rod of 38.8mm outer diameter with 680mm and 140 mm length respectively. The leading edge of the first section (14) is a semispherical smooth end (13) to ensure smooth air entrance to the test section. The heated section (15) is 280mm length, made of an aluminum rod of 38.8mm outer diameter and heated electrically under uniform heat flux condition. The heater is made of helical Nickel-Chromel wire (17), fixed in a glass tube (16) of 12mm diameter. The glass tube is filled with fine sand particles and is closed at both ends by gypsum and inserted inside the heated section. The gap between the glass wall and the inner surface of the aluminum tube is filled with aluminum powder. Fine sand particles and aluminum powder are used in order to insure uniform distribution of heat and prevent the heater from burning up. The inside rod of the annulus is fixed inside the outer tube in a symmetrical position by the help of two groups of fine threaded studs (12) one at both ends of the rod. Each group consists of 3 bolts of 3-mm diameter distributed over the tube periphery with 120^o between each other to form a uniform annulus. The outer surface of the heated section of the inner tube is of cyclically varying cross section area. It consists of four identical convergent- divergent conical sections (19 and 20) with a maximum diameter of 38.8 mm and a minimum diameter of 26 mm. The outer tube (18) has a 1100mm length and 52.4mm inside diameter, consists of three parts. The first and the third parts are unheated tubes made of plastic (PVC). While, the main (second) part is made of steel with the same inner and outer diameters of the plastic tubes and having a length of 280 mm. The three parts are connected to each other as shown in Fig (1-a) with the aid of two plastic sleeves using special glue. The main part is heated electrically under uniform heat flux condition. The heater is made of Nickel-Chrome wire (22), which is wounded around the steel tube over a thin mica cylindrical sheet of 0.5mm thickness. The mica is chosen because of its high thermal conductivity and low electrical conductivity. The heater is then fixed with the aid of gypsum layer (23). The tube is then insulated with a 80mm thickness fiber glass wool insulation (24).

In order to estimate the heat lost to the surrounding, four groups of thermocouples are distributed inside the glass wool insulation with 10, 95, 180 and 265 mm longitudinal distances from the beginning of the heated tube and having a peripheral angles of 45^o between each other. Each group consists of two thermocouples having radial distances of 70 mm and 90 mm from the tube centerline.

Air velocity distribution is measured at section (3) by a calibrated Pitot-tube (4) and is integrated to give the mean velocity at inlet. Two static pressure taps (5,7) are provided at inlet and outlet from the heated section. Pressure drops are measured by a digital micro-manometer with an accuracy of ±1 Pa. The air temperature at inlet and outlet of the test section are measured by two copper-constantan thermocouples of type (J) and the air properties are calculated as a function of the average air temperature. An electric auto-transformer is used to control the power supplied to the test section electric heater, which is measured by an ammeter and voltmeter. The surface temperatures of the inner and outer tubes of the annulus are measured by 6 copper-constantan thermocouples each. The temperature signals are then transferred to a digital temperature recorder of a sensitivity of 0.1°C.



a)-Flow Diagram

b)-Internal Rod and external tube Constructions

Figure (1) Details of the Experimental Set-Up

3. FLOW AND HEAT TRANSFER CHARACTERISTICS

Figure 2 shows detailed cross-section of one of the *cyclically diverging-converging* annular sections under test. It is characterized by the half angle of convergence "α" at inlet, the half angle of divergence "β" at outlet. There are four divergent- convergent sections with length "L_{cy}" of 70 mm each. *Cyclically diverging-converging* annular tubes considered throughout this work, has the different geometrical data shown in table (1).

The wetted area of the constant cross section area annulus ($A_{c.w}$) and *cyclically diverging-converging* one ($A_{v.w}$) can be calculated respectively with the aid of figure (2) as follows (see Appendix A):

- For constant cross section area annulus:

$$A_{c.w} = \pi(D+d_o)(4L_{cy}) \tag{1}$$

- For variable cross section area annulus:

$$A_{v,w} = 4\pi D L_{cy} + \pi(d_o + d_i)\left[\sqrt{(d_o - d_i)^2 + 4L_{co}^{\;2}} + \sqrt{(d_o - d_i)^2 + 4L_d^{\;2}}\right] \quad (2)$$

Table (1) Characteristics of the Cyclic diverging-converging Annuli

L_r	α°	β°	L_{cy} mm	L_d mm	L_{co} mm
3.67	6° 38'	23° 4'	70	55	15
1.80	8° 05'	14° 22'	70	45	25
1.00	10° 22'	10° 22'	70	35	35
0.56	14° 22'	8° 05'	70	25	45
0.27	23° 4'	6° 38'	70	15	55

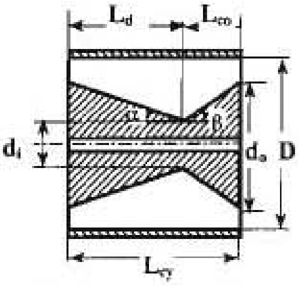


Figure (2) Construction of divergent–convergent test section (four of them in series)

The volumetric hydraulic diameter is defined as four the volume for flow divided by the wetted surface area. For constant and variable cross section area annular tubes, it can be calculated as follows (see Appendix A):

$$Dh_c = D - d_o \quad (3)$$

$$Dh_v = \frac{4\pi D^2 - \frac{4\pi}{3}(d_o^{\,2} + d_o d_i + d_i^{\,2})}{4\pi D + \pi(d_o + d_i)\left[\sqrt{(d_o - d_i)^2 + 4L_{co}^{\;2}} + \sqrt{(d_o - d_i)^2 + 4L_d^{\;2}}\right] / L_{cy}} \quad (4)$$

Based on the hydraulic diameter of the constant cross section area annulus, the Reynolds number and Nusselt number of the two heating cases (Re_c) ,($Nu_{c,i}$) and ($Nu_{c,o}$) may be calculated from the following relations:

$$Re_c = \frac{u_c Dh_c}{\nu} \quad (5)$$

$$Nu_{c,i} = \frac{h_{c,i} Dh_c}{k}, \qquad Nu_{c,o} = \frac{h_{c,o} Dh_c}{k} \quad (6)$$

Where $h_{c,i}$ and $h_{c,o}$ are the heat transfer coefficients based on the inside diameter of the constant cross section area annulus heated from inside and outside respectively and are calculated as follows:

$$h_{c,i} = \frac{Q}{A_{c,i}(\Delta T)_m}, \qquad h_{c,o} = \frac{Q}{A_{c,o}(\Delta T)_m} \quad (7)$$

Where:

$$Q = I \cdot V \cos\phi = I.V \qquad (where, \phi = 0) \quad (8)$$

$$A_{c,i} = 4.\pi.d_o.L_{cy}, \quad A_{c,o} = 4.\pi.D.L_{cy} \quad (9)$$

$$(\Delta T) = \frac{(T_w - T_i) - (T_w - T_o)}{ln\left[\frac{(T_w - T_i)}{(T_w - T_o)}\right]} \quad \text{and} \tag{10}$$

$$T_w = \frac{\sum_{n=1}^{6}(T_{w,n})}{6} \tag{11}$$

Also, the friction factor of constant cross section area annular tubes (f_c) may be calculated using the following definition:

$$f_c = \frac{2\Delta P(Dh_c)}{4L_{cy}\rho u_c^2} \tag{12}$$

In view of the changing cross-section in case of the *Cyclically diverging-converging* annulus, the mean velocity u_v in the annular gap can be written as (see Appendix A):

$$u_v = \frac{u_c(D^2 - d_o^2)}{\left[D^2 - (d_o^2 + d_o d_i + d_i^2)/3\right]} \tag{13}$$

Thus the Reynolds number for the variable cross section area annular tubes, Re_v, formed with the hydraulic diameter (Dh_v) and the gap mean velocity u_v is:

$$Re_v = \frac{u_v Dh_v}{\nu} \tag{14}$$

The Nusselt numbers for the internally and externally heating cases ($Nu_{v,i}$ and $Nu_{v,o}$) and the friction factor of the *Cyclically diverging-converging* annulus (f_v) can be evaluated as follows:

$$Nu_{v,i} = \frac{h_{v,i} Dh_v}{k}, \qquad Nu_{v,o} = \frac{h_{v,o} Dh_v}{k} \tag{15}$$

$$f_v = \frac{2\Delta P Dh_v}{4L_{cy}\rho u_v^2} \tag{16}$$

Where $h_{v,i}$ and $h_{v,o}$ are the heat transfer coefficients based on the inside and outside surfaces of the variable cross section area annulus respectively and are calculated as follows:

$$h_{v,i} = \frac{Q}{A_{v,i}(\Delta T)_m}, \qquad h_{v,o} = \frac{Q}{A_{v,o}(\Delta T)_m} \tag{17}$$

Where,

$$A_{v,i} = \pi(d_o + d_i)\left[\sqrt{(d_o - d_i)^2 + 4L_{co}^2} + \sqrt{(d_o - d_i)^2 + 4L_d^2}\right], \qquad A_{v,o} = A_{c,o} \tag{18}$$

4. RESULTS AND DISCUSSION

In order to verify the performance of the experimental test rig, the results of the constant area annular tube are compared with the available correlations. Friction factor and mean Nusselt number in case of the uniform heat flux were then determined for the variable cross section area annuli with different length ratios, i.e. different divergent and convergent angles (α and β)

4.1 Constant Area Annular Tube

Due to the shortage in the available experimental data for the flow in annular ducts, our experimental results of the friction factor are compared with the correlation developed by Robert (1984). He suggested that the use of the hydraulic diameter of the constant cross section area annulus in conjunction with any pipe flow relation will lead to accurate results for the friction factor of the flow through annular ducts.

The friction factors for the insidly heated annulus and the outsidly heated one of the turbulent flow in the constant cross section area annulus are plotted in figure (3) versus the Reynolds number along with the data of Miller (1978). The figure shows that the experimental friction factor values are in good agreement with those calculated from Miller's correlation in the considered range of Reynolds number. The friction factor of the present study may be correlated with Re_c as:

$$f_c = 0.395\, Re_c^{-0.275} \tag{19}$$

in the range $3.14x10^3 < Re_c < 3.3x10^4$ with a standard deviation of 5%.

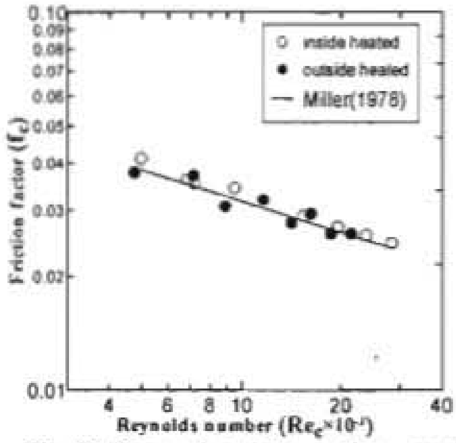


Fig. (3) Comparison between the present friction factor in the constant area annulus with those of Miller (1978).

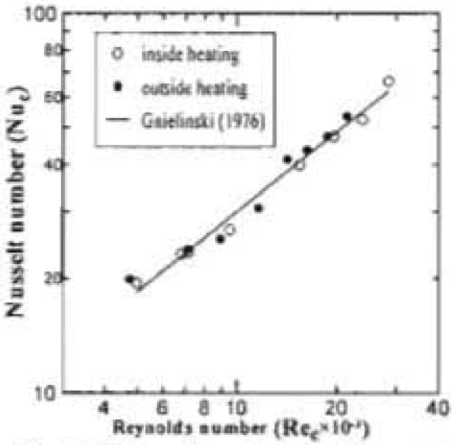


Fig. (4) Comparison between the present Nusselt number (Nu_c) in the constant area annulus with those of Gnielinski(1976).

To the author's knowledge, there is no correlation that relates Nusselt and Reynolds numbers for annulus. But, one can use any available equation that describes the relation between the Nusselt and Reynolds numbers for circular tube using the multipliers $[0.86(d_o/D)^{0.16}]$ and $[1-0.14(d_o/D)^{0.6}]$ suggested by Petokhov et al (1964) for annular tubes heated from inside and outside respectively. In these cases the hydraulic diameter of the constant area annuls must be used instead of tube diameter.

Figure (4) presents the results of the heat transfer for the constant cross section area annulus as a relation between the Nusselt and Reynolds numbers compared with the data of Gnielinski (1976) after using the above multipliers. As shown from figure (4), the experimental data of Nusselt numbers for the two present cases are in good agreement with those suggested by the previous reference. The present Nusselt data are correlated with the Reynolds number for the first and second cases with a standard deviation of 7% as follows:

$$Nu_{c,i} = 0.0475\, Re_c^{0.7} \tag{20}$$

These assess clearly demonstrate the reliability and shows the accuracy of the experimental test rig and results obtained from it.

4.2 Cyclically Diverging-Converging Annular Tubes

The experimental results of heat transfer and friction losses of the turbulent flow in variable cross section area annular tubes are presented as the Nusselt number and friction factor versus the Reynolds number. This satisfies the purpose of comparing the results of *cyclically diverging-converging* annular tubes with those of constant area one.

Figure (5) shows that the results of friction factors of the turbulent flow in *cyclically diverging-converging* annular tubes are higher than those of constant cross section area one. The increase in friction factor ranges from 75 to 380% for divergent convergent length ratio L_r=3.67 to as high as 572 to 650 % for length ratio L_r=0.27. The other annular tubes have a friction factor in between these two limits.

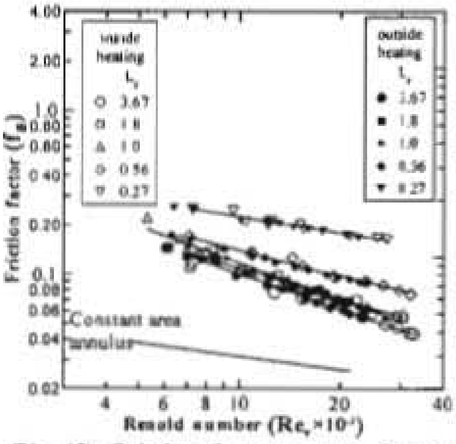


Fig. (5) Friction factor versus Reynolds number for cyclically diverging-converging annuli with the length ratio as a parameter

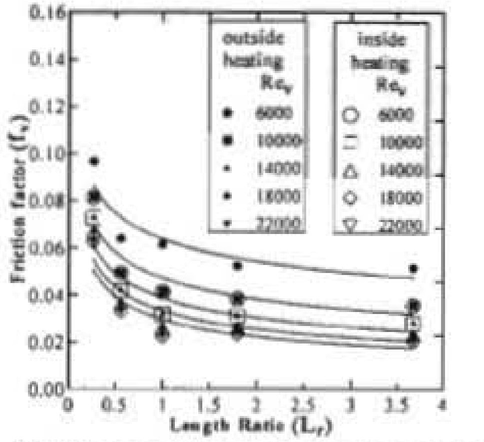


Fig. (6) Length ratio versus friction factor for cyclically diverging-converging annuli with the Reynolds number as a parameter.

Figure (6) presents the effect of convergent divergent length ratio on friction factor for different values of Reynolds number for the two cases studied in this work. It shows that, the friction factor increases with the decrease of length ratio i.e., with the decrease of the divergent part length. This may be attributed to the flow separation that is associated the decrease in the diffuser section length, resulting in increased frictional losses. Although, the turbulent flow friction factor in these tubes depends on length ratio, it is also dependent on the Reynolds numbers.

Figures (7 and 8) show the experimental results of heat transfer for the turbulent flow of air in five divergent-convergent annular tubes in comparison with those of the constant cross section area annulus for the two discussed cases. The results show that the Nusselt number for *cyclically diverging-converging* annular flows are higher than those of constant cross section area annulus. For the internal heated case it is seen from figure (7) that, the annular tube having divergent convergent length ratio of 1.0 achieves the highest improvement in Nusselt number, which is nearly as 160% of the constant area annulus. Whereas the annular tube having length ratio of 0.56 has the lowest improvement of about 14:82%, compared to the constant cross section area annular tube. For the externally heated case, as seen from figure (8), the length ratio of maximum heat transfer is 1 with a maximum increase of 117% compared to the straight annular tube. The increase in heat transfer for other length ratios ranges between 33% and 117%.

Figure (8) shows also that, the length ratio has a great effect on heat transfer at lower Reynolds numbers, which means that the divergent-convergent configurations may be more effective in the laminar flow regime. Increasing the Reynolds number fully developed turbulent flow occurs and the effect of divergent-convergent configurations in the development of boundary layer decreases. As a result, the increase in heat transfer decreases with Reynolds number and may be diminished at certain Reynolds number value.

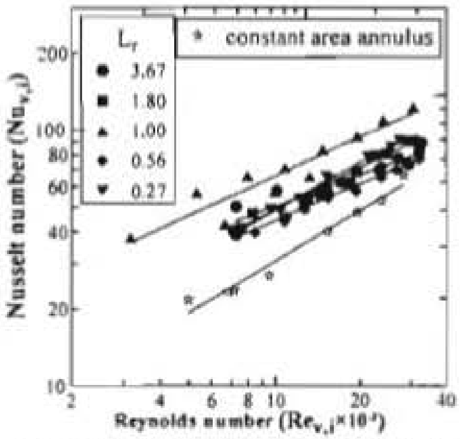


Fig. (7) Relation between Nusselt number and Reynolds number for cyclically diverging - converging annuli heated from inside for different length ratios.

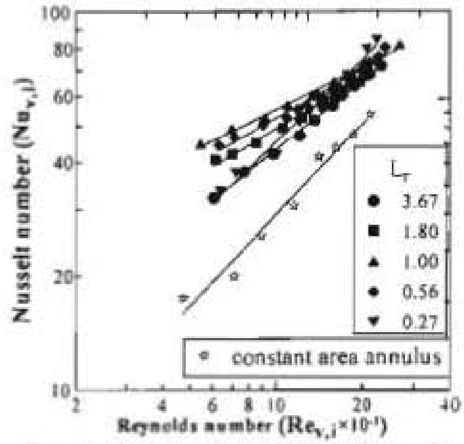


Fig. (8) Relation between Nusselt number and Reynolds number for cyclically diverging - converging annuli heated from outside for different length ratios.

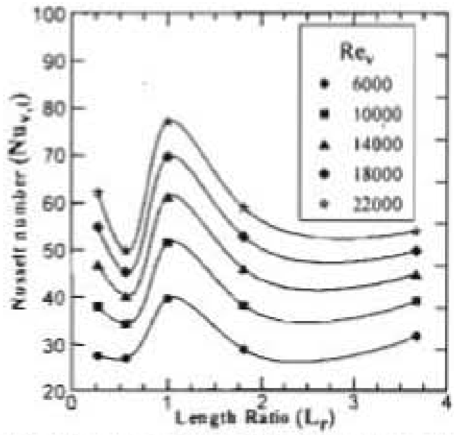


Fig. (9) Effect of length ratio on Nusselt number for cyclically diverging-converging annuli heated from inside for different Reynolds number.

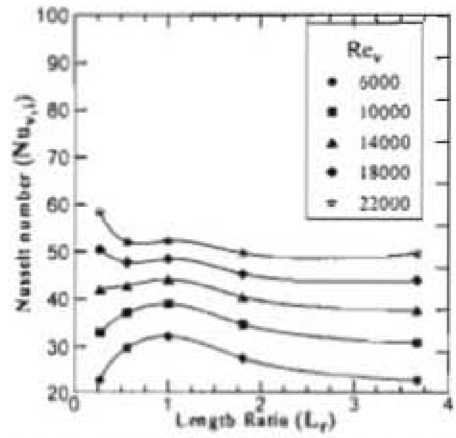


Fig. (10) Effect of length ratio on Nusselt number for cyclically diverging-converging annuli heated from outside for different Reynolds number.

To show the effect of length ratio on the heat transfer, the Nusselt number is plotted in figures (9and10) as a function of the length ratio at different Reynolds number. For the internal heated case, figure (9) shows that, Nusselt numbers increase with the decrease of length ratio up to maximum values at length ratio of 1 and then decrease with further decrease in length ratio. This may be due to the effect of the divergent and convergent angles (length ratios) on the development of the boundary layer near the surface of the inner tube. To prove this trend visualization work is needed which is beyond the scope of this work. The Figure shows also that, the Nusselt number at length ratio of 0.27 is higher than that at length ratio of 0.56 for Reynolds number higher than 6000, also the difference in Nusselt number values at these two length ratios increases with the Reynolds number. This may be explained by the flow separation occurs during the flow of air over the cyclically convergent-divergent shape of the inner tube of the annulus, which increases with the Reynolds number. For the externally heated case, Fig. (10) shows that the behavior of heat transfer depends not only on length ratio but also on Reynolds number i.e. flow regime. At Reynolds number approximately up to 16000 the Nusselt number increases with the decrease of length ratio up to a maximum value at length ratio of 1 and then decreases with further decrease in length ratio. In the Reynolds number region higher than 16000 the Nusselt number increases with the decrease of length ratio and has its maximum value at length ratio of 0.27. This may be also due to the effect of length ratio on the development of the boundary layer adjacent to the outer surface of the annulus.

5. OVERALL TUBE PERFORMANCE

Finally, it is very important to evaluate the enhancement process as a whole. Therefore, we have to 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 by Farag et al (1999) as follows:

$$\eta = (Nu_v / Nu_c)/(f_v / f_c) \tag{21}$$

From the experimental results of Nusselt numbers and friction factors, one may calculate the term (η) in case of *cyclically diverging-converging* annular tubes. These results are shown in figures (11and12) as a relation between η and Re_v, for the two cases. From figures (11and12), one may conclude that the enhancement is more efficient in case of annulus having divergent-convergent length ratio equal to 1.0. In general, that particular annular tube (L_r=1.0, and $\alpha=\beta=10^\circ\ 22'$), has the best performance compared to other convergent divergent sections considered in this work, since it has the highest heat transfer enhancement with moderate increase in friction losses.

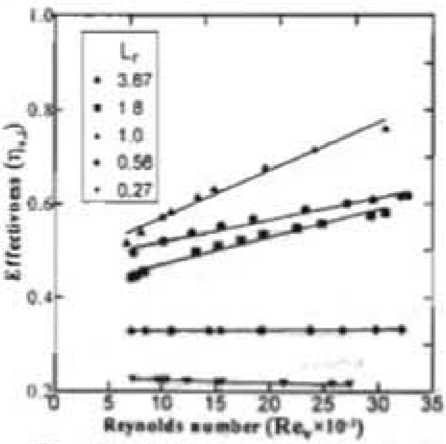


Fig. (11) Effectivness for cyclically diverging-converging annuli heated from inside versus Reynolds number.

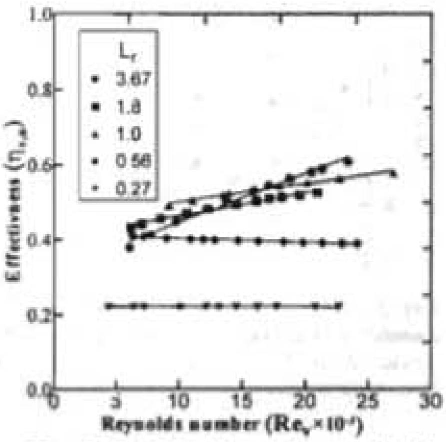


Fig. (12) Effectivness for cyclically diverging-converging annuli heated from outside versus Reynolds number.

6.CONCLUSIONS

In this work the heat transfer and friction loss of the turbulent flow in annular tubes having *cyclically diverging-converging* sections are investigated and compared with those of constant cross section area annular one for two cases. In the first case the annular tubes are heated from inside while, in the second case the annular tubes are heated from outside under constant heat flux condition with the other tube insulated. The following facts are concluded:

1. Both the friction factor and Nusselt number of the turbulent flow in *cyclically diverging-converging* annulus are higher than those of the flow in constant cross section area one.
2. Compared with the constant cross section area annular tube, for the internally heated case, an improvement as high as 160% in heat transfer has been achieved in *cyclically diverging-converging* annulus having divergent convergent length ratio 1.0. While the improvement is as low as 14% in the annulus that has length ratio of 0.56.
3. Nusselt numbers for the externally heated annuli are lower than those for the internally heated annuli.
4. The highest increase in friction losses in the *cyclically diverging-converging* annular tubes is estimated as 650 % for the tube having length ratio of 0.27 compared to the constant cross section area annulus. While the lowest increase is as low as 75% at length ratio of 3.67.
5. The *Cyclically diverging-converging* annular tube of L_r=1.0, $\alpha=\beta=10^\circ 22'$, has the best performance compared with the other divergent-convergent sections considered in this work, since it has the highest heat transfer enhancement with moderate increase in friction losses.

NOMENCLATURE

A	surface area, m^2	L	length, m
D	inner diameter of outer tube, m	L_r	length ratio (L_d/L_{co})
Dh	hydraulic diameter, m	Q	heat transfer rate, W
d	diameter of inner rod , m	T	temperature, K
h	heat transfer coefficient, W m^{-2} K^{-1}	u	Average air velocity, m/s
I	electric current, Amp.	V	electric volt, volt
k	thermal conductivity, W m^{-1} K^{-1}		

Dimensionless Groups

f	Friction factor	Pr	Prandtle number
Nu	Nusselt number	Re	Reynolds number

Greek Symbols

Δp	pressure drop, Pa	β	half angle of divergence
ΔT_i	Logarithmic mean temp. difference defined by eq.(10), K	ν	fluid kinematic viscosity, $m^2 s^{-1}$
		ρ	fluid density, kg m^{-3}
α	half angle of convergence	η	Effectiveness, Eq.(21)

Subscripts

c	constant cross section area annulus	i	inside, or inlet
co	convergent part	o	outside, or exit
cy	Cycle	v	Cyclically diverging-converging annuli
d	divergent part	w	Wetted, or wall

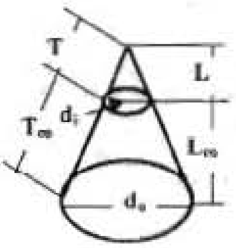
REFERENCES

- Araid, F. F., 2000b, " Convective Heat Transfer and Fluid Flow in Tubes with Different Rod Inserts", Proc. Of Mansoura 3rd International Engineering Conference, Faculty of Engineering, Mansoura University, Mansoura, Egypt, April, Vol. 2, pp. 233-242.
- Araid, F. F, Awad, M. M., and El-Hadik, A. A, 1983, "Heat Transfer and Hydrodynamic Resistance in Compacting Plate Heat Exchanger type Divergent-Convergent Surfaces", ASME, JSME, Thermal Engineering Joint Conference, Proceedings, Vol. 3, pp. 437-441.

- Awad, M. M, Shalaby, M. A., and El-Shazly, S. A., 1986, "Compactness Effect in Air Plate Heat Exchangers of Convergent-Divergent Channels", Bulletin of the Faculty of Engineering, El-Mansoura University, Egypt, Vol. 11, No. 1, June.
- Bergess A.G., Blumen, A.R., and Taborelc, J. 1974," Performance Evaluation Criteria for Enhanced Heat Transfer Surfaces", ASME Trans. J.Heat Transfer, vol.11, pp.239-243.
- Farag, E. H, Hamed, M. H., and Abou-Taleb, F. Sh, 1999, "Turbulent Heat Transfer Enhancement I Annular Ducts", Proc. Of 3rd Int. Conference on Eng. Research, ICER-99, Faculty of Eng. & Technology, Suez Canal University. Port Said, Egypt November, Vol. 1, pp.193-206.
- Fu, W. S., and Tseng, C. C., 1995a, "Enhancement of Heat Transfer for a Tube with an Inner Tube Insertion", Int. J. Heat Mass Transfer, Vol. 38, No. 20.
- Fu, W. S., Tseng, C. C., and Huang, C. S., 1995b, "Experimental Study of the Heat Transfer Enhancement of an Outer Tube with an Inner-Tube Insertion", Int. J. Heat Mass Transfer, Vol. 38, No. 18, pp. 3443-3454.
- Garimella, S., and Christensen, R. N., 1993, "Experimental investigation of Fluid Flow Mechanisms in Annuli with Spirally Floated Inner Tubes", ASHRAE Trans- Actions, Vol. 99, Part 1, pp. 1205-1216.
- Garimella, S., and Christensen, R. N., 1995c, "Heat Transfer and Pressure Drop Characteristics of Spirally Floated Annuli Part II Heat Transfer", J. of Heat Transfer, Vol. 117, pp. 61-68.
- Gnielinski, V., 1976, "New Equation for Heat and Mass Transfer in Turbulent Pipe and Channel Flow", Int Chem. Eng., Vol. 16, pp. 359-368.
- Gokhman, A. A., and Kirpikov, V. A., 1972, "Intensification of Heat Transfer by Forming Pressure Gradient in Fluid, In Russian", Teplo I Massaperenos j., Vol. 1, Part 2, Minsik, P. 128.
- Miller, D. S., 1978, "Internal flow system", British Hydrodynamics Research Association (BHRA), Fluid Engineering, Bedford, U.K.
- Petokhov, B. S., and Roizen, I.I., 1964, "Generalized Relationship for Heat Transfer in a Turbulent Flow of Gas in Tubes of Annular Sections", High Temperature, (USSR), Vol. 3, pp. 65-68.
- Rabie, L. H., Tolba, M.A., Araid, F. F., and Awad, M. M.,1989, "Viscoelastic Fluid and Heat Transfer in Pipes of Periodically Varying Cross Section Area"
- Robert, D. Blovins. 1984, "Applied Fluid Dynamics", Hand Book, Van Nostrand Reinhold Company Inc., New York.
- Sultan, A. A. and Sultan, G. I., 2000a,"Enhancing forced Convection Heat Transfer in Annular Tubes by Using Aluminum Mesh Layers", Mansoura Engineering Journal (MEJ), Faculty of Engineering, Mansoura University, Mansoura, Egypt, Vol. 25, No. 4. December, pp. 15-28.

APPENDIX A

1-Flow volume calculations



Referring to the figure shown.

$L = d_i L_{co} / (d_o - d_i)$

The convergent part Volume:

$V_c = \pi d_o^2 (L + L_{co})/12 - \pi d_i^2/12$

Substituting for L from the above equation, we get:

$V_c = \pi L_{co}[d_o^2 + d_i^2 + d_o d_i]/12$

In the same way the divergent part volume can be calculated as,

$V_d = \pi L_d[d_o^2 + d_i^2 + d_o d_i]/12$

The length of the convergent divergent cycle can be set as, $L_{cy} = L_{co} + L_d$

The total volume of the convergent divergent cycle can be calculated as,

$V_{cy} = \pi L_{cy}[d_o^2 + d_i^2 + d_o d_i]/12$

The volume of the four cyclically convergent divergent sections can be written as,

$4\pi L_{cy}[d_o^2 + d_i^2 + d_o d_i]/12 = \pi L_{cy}[d_o^2 + d_i^2 + d_o d_i]/3$

Thus the flow volume can be expressed as: $V_f = \pi D^2 L_{cy} + \pi L_{cy}[d_o^2 + d_i^2 + d_o d_i]/3$

The mean velocity of flow can be calculated by, $u_c(\pi/4)(D^2 - d_o^2)4L_{cy} = u_v \times Flow\ volume$

From which, $u_v = u_c(D^2 - d_o^2)/\{D^2 + (d_o^2 + d_i^2 + d_o d_i)/3\}$

2-Witted area calculations:

Refering to the figure shown in the Appendix we get, $T_{co} + T = T(L + L_{co})/L$,

The surface area of the convergent part is $A_{co} = \pi d_o(T_{co} + T)/2 - \pi d_i T/2$

Subistituting for $(T_{co}+T)$ from the above equation. we get, $A_{co} = \pi T(d_o^2 - d_i^2)/(2d_i)$

From the above figure we get,

$$T = \sqrt{(d_i/2)^2 + L^2} = \frac{d_i}{2(d_o - d_i)}\sqrt{(d_o - d_i)^2 + 4L_{co}^2}$$

Substituting for T from the above equation, we get,

$$A_{co} = \frac{\pi(d_o + d_i)}{4}\sqrt{(d_o - d_i)^2 + 4L_{co}^2}$$

Also the surface area of divergent part can be written as,

$$A_d = \frac{\pi(d_o + d_i)}{4}\sqrt{(d_o - d_i)^2 + 4L_d^2}$$

The surface area of the four convergent-divergent cycles can be set as,

$$A_{v,i} = 4A_{cy} = \frac{\pi(d_o + d_i)}{4}\left\{\sqrt{(d_o - d_i)^2 + 4L_{co}^2} + \sqrt{(d_o - d_i)^2 + 4L_d^2}\right\}$$

The total surface (witted) area can be expressed as,

$$A_{v,w} = 4\pi DL_{cy} + \pi(d_o + d_i)\left[\sqrt{(d_o - d_i)^2 + 4L_{co}^2} + \sqrt{(d_o - d_i)^2 + 4L_d^2}\right]$$

The volumetric hydraulic diameter can be set as, $Hd_v = 4\ V_f / A_{v,w}$