10-20-2020

Experimental Study of Heat Transfer through a Rotating Heat Exchanger.

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Experimental study of heat transfer through a rotating heat exchanger

دراسه عملية لانتقال الحرارة خلال مبادل حراري دوار

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ملخص البحث

هذا البحث يدرس انتقال الحرارة بالحمل الجبري الرقائقي لماء ساخن ينساب خلال حيز حلقي اسطواني وينساب ماء بارد في الأنبوب الخارجي للحيز الحلقي. الأنبوب الداخلي يدور بسرعة تتراوح بين 350و1580 لفة/ دقيقة لتعطى رقم رينولدز مماسى يتراوح بين (200-1750). القطر الخارجي للأنبوب الحلقي ذو قيمة ثابتة تساوى 54,4 مم بينما يتغير القطر الداخلي لياخذ ثلاث قيم هي 21و26,5و33,5 مم وذلك للحصول على نسبة الأقطار0,386و0,487و0,62 على التوالى. ينساب الماء الساخن في الحيز الحلقي بسرعات تتغير من0,0042 إلى 0,52 م/ث لتسمح بتغير رقم رينولدز المحوري من 130 إلى 2300 ليغطى النطاق الرقائقي. يتم مرور الماء البارد في الأنبوب الخارجي للحيز الحلقي الذى يبرد عن طريق دائرة تبريد. تبرد المياه في خزان في حين أن الأنبوب الداخلي يدور ويمر حوله مياه ساخنة التي تسخن في خزان به سخانات كهربائية .تم في هذا البحث دراسة تأثير كل من نسبة الأقطار للحيز الحلقي ورقم رينولدز المحوري ورقم رينولدز المماسى على معاملات انتقال الحرارة رقم (نوسلت).

وقد بينت التجارب العملية في هذا البحث أن معامل انتقال الحرارة للحيز الحلقي الدوار كما للحيز الساكن يزداد بنقصان نسبة الأقطار لنفس رقم رينولدز المحوري والمماسى. كذلك اظهرت النتائج أن معامل انتقال الحرارة للحيز الحلقي الدوار أعلي من نظيره للحيز الحلقي الساكن عند نفس القيم لنسبة الأقطار ولرقم رينولدز المحوري .

بينت النتائج أيضا أن الزيادة فى معامل انتقال الحرارة بالحمل حوالى 200% عند رقم رينولدز مماسى حوالى (1744) ومحورى (1951) وسرعة دوران 1580 لفة/ دقيقة لنسبة أقطار 0,62 وان اقل زيادة لمعامل انتقال الحرارة حوالى 36% عند رقم رينولدز مماسى يساوى (205) ومحورى (1100) وسرعة دوران 350 لفة / دقيقة لنسبة اقطار 0,386 مقارنة بنتائج تلك الخاصة الحيز الحلقى الساكن عند نفس رقم رينولدز المحورى . تم صياغة النتائج فى صورة معادلات بين رقم نوسلت وكل من نسبة الاقطار ورقم رينولدز المحورى ورقم رينولدز المماسى للفراغ الحلقى الساكن والدوار.

Abstract

This paper presents an experimental study of laminar forced convection heat transfer for hot water flows through annular tube. The test section consisted of two annular spaces the inner of which for the tested hot water while the outer for the cooling water flow. The inner tube of the annulus is rotated with rotational speeds which varied from 350 to 1580 r.p.m. to give rotational Reynolds number in the range ($200 \leq Re_\omega \leq 1750$).The outer tube of the annulus have a constant diameter of 54.5mm but the diameter of inner tube of the annulus is equal to 21, 26.5 and 33.5 mm so as to give diameter ratios of 0.62, 0.487 and 0.386. Hot water flowed axially through the annular space with velocities that ranged from (0.0042 to 0.52m/s), to give axial Reynolds numbers in the range ($130 \leq Re \leq 2300$), to cover the laminar flow regime. The cold water that flowed through the outer annulus was cooled by a refrigerating circuit. In the present work the effects of radius ratio, axial Reynolds number and rotational Reynolds number on the heat transfer were investigated. The results show that. The rate of heat transfer of the rotational annular tubes is higher than that of the stationary one of the same radius ratio and axial Reynolds numbers. An increase of as much as 200% in the heat transfer is observed for annular tube of radius ratio equal to 0.62, Reynolds number of 1951 rotational Reynolds number of 1744 and rotational speed of 1580 r.p.m respectively, a decreases in heat transfer as much as 36% in the heat transfer is reported for annular tube of radius ratio (β) of 0.386, Reynolds number of 1100, and rotational Reynolds number of 205 and rotational speed of 350 r.p.m respectively. Two correlations were established between Nusselt number and axial Reynolds number. Rotational Reynolds number and radius ratio of the annulus for both stationary and rotational annular tubes.

Nomenclature

A	surface area, m^2	u	Average velocity, m/s
A_c	cross surface area, m^2	k	thermal conductivity, W/m. k
C_p	Specific heat J/kg.k	L	Test section length, m
di	Annulus inner diameter, m	N	speed of rotating inner tube, r.p.m
do	Annulus outer diameter, m	T	Temperature, k
r_i	Annulus inner radius, m	m	Water mass flow rate, kg/s
r_o	Annulus outer radius, m	g	Gravity acceleration, m/s
D_h	Hydraulic diameter, m	h	Convection heat transfer coefficient, W/m2. K

Dimensionless numbers

Nu	Nusselt number (-)	Pr	Prantdl number (-)
Re	Axial Reynolds number (-)	Ta	Taylor number (-)
Re_ω	Rotational Reynolds number (-)		

Greek letters

ν	Kinematic viscosity, m^2/s	ω	Angular velocity, rad/s
μ	Dynamic viscosity, $N.S/m^2$	ρ	Density, kg/m^3
β	Radius ratio		

Subscripts

c	Cross-sectional area	o	outer
d	Based on annulus outer diameter	s	wall
h	Based on hydraulic diameter	wi	water inlet
i	Inner	wo	water outlet

1. Introduction

Heat transfer and fluid flow between concentric cylinders have been intensively investigated because of its basic importance in numerous engineering applications. Practical examples of forced convection between concentric cylinders include flow and heat transfer in double pipe heat exchangers, oil/gas drilling wells, nuclear fuel element cooling system, and plastic extruders, cooling of electric machines, turbo-machinery and journal bearings. Studying this type of flow is needed to improve the machines efficiency by optimizing the heat transfer characteristics and by minimizing the torque required to rotate the rotating parts.

Many engineering applications involve rotating machinery components. They include rotating membrane filters, co-axial rotating heat pipes, rotating extractors, cylindrical bearings, and rotating power transmission systems. A basic configuration for rotating machinery components is the annulus with a stationary outer cylinder and a rotating inner cylinder. The flow in such an annulus is called Taylor-Couette flow, in which the famous Taylor vortices damping effect of the imposed axial flow on the internal motions of the vortices. Excessive thermal stress can damage rotating machinery components [12]. This fact has motivated numerous interesting studies on

The mechanism of heat transfer in Taylor Couette flow or Taylor- Couette - Poiseuille flow.

1.1 Annuli with stationary cylinders:

Reynolds et al; [1] presented for the first time a general formulation of the fully developed forced convection problem in concentric annuli for arbitrarily prescribed wall temperatures or heat fluxes. Four fundamental thermal boundary conditions were established (first kind, second kind, third kind and fourth kind). Heaton et al; [2] solved the governing equations for the heat transfer in annular passages for simultaneously developing velocity and temperature profiles in laminar forced flow. Sparrow and Lin [3] provided the axial velocity distribution and the axial pressure drop for laminar forced flow in the entrance region of concentric annuli. El-Shaarawi and Alkam [4] used finite difference scheme to study the transient laminar forced convection in the entry region with initially step change in temperature at one of the annulus boundaries or a step change in temperature at both inlet cross section and one of annulus boundaries while the other boundary was kept adiabatic. They found that heating the outer boundary effect is much higher than heating the inner boundary. Yoo [5] studied numerically the fully developed mixed convection in an annulus with different walls temperatures with the outer cylinder colder than the inner cylinder. They found three types of flow patterns according to the number of eddy; they are two-eddy, one eddy, and no-eddy flows. Islam et al; [6] investigated numerically the steady laminar mixed convection in the entrance region of a horizontal concentric annulus with uniform heat flux at the inner wall and adiabatic outer wall. They found that Nusselt numbers of air at the entrance region are considerably greater than fully developed values. They found also that axial flow and temperature field development are strongly influenced by the buoyancy especially near the inlet

Section. Soares et al [7], Eraslan and Ozbelge [8] and Ilicali and Engez [9] studied the hydrodynamic and heat transfer characteristics of laminar flow inside concentric annuli; their focus was on the non-Newtonian fluids. Sutherland and Kays [10] performed an analysis of a hydro dynamically and thermally fully developed laminar flow in a concentric circular tube annulus with an arbitrarily prescribed heat flux around the periphery of either walls. Contrary to what they expected, they found that the wall temperature variation was very substantial. Lu et al; [11] studied experimentally the forced convection heat transfer of single-phase water in vertical and horizontal narrow annuli. They found that the heat transferred in a narrow annulus was not the same as that calculated by conventional formulae earmarked for internal flow in round tubes, and the transition from laminar to turbulent flow happened markedly earlier than in conventional round tubes.

1.2 Annuli with rotating cylinders: -

Becker and Kaye [12] analyzed the effect of the radial temperature gradient on the stability of Taylor-Couette flow. They found that heating the inner rotating cylinder stabilizes the flow, while heating the outer stationary cylinder destabilizes it. They also plotted the Nusselt number against the Taylor number. The Taylor number is defined as $Ta = \rho^2\omega^2\ [(r_o + r_i)/2](r_o - r_i)^3\ /\mu^2$ Their data showed that Nusselt numbers remained constant until the Taylor number exceeded a critical value, at which the Taylor vortices appear, beyond which their values increased with Taylor numbers. D. Simmers and Coney [13] studied experimentally the Fully-developed flow in a concentric annulus formed by a stationary outer cylinder, which may be heated isothermally, and a rotatable inner cylinder.

They found that the onset of Taylor vortex flow had a pronounced effect on the velocity profiles for all three directions and also heat transfer through the outer annular surface had a greater effect on the radial velocity profile than on the axial or tangential, but in the narrow gap case only. Aliean [14] experimentally investigated the overall heat transfer coefficient for a rotating cavity type heat exchanger. The heat exchanger mainly consisted of a rotating cylindrical cavity with axially through flow. Mollo et al. [15] used a mass transfer procedure to study experimentally the laminar flow in the entry region of a circular annulus with rotating inner walls. They got results of Sherwood number versus axial location and reported the critical Taylor number after which Taylor vortices were formed. Choi and Kim [16] studied numerically the fully developed mixed convection in an annulus with a heated rotating inner cylinder. They found that the effect of heating of inner cylinder postpones the formation of Taylor vortices when the rotation effect dominates the buoyancy effect. When the effect of buoyancy predominates over the effect of centrifugal force, the rotation of inner cylinder also stabilizes the two dimensional basic flows. When both rotation and buoyancy effects are important, the basic flow became unstable with respect to oscillatory disturbances. Abdel Ghaffar and sultan [17] experimentally investigated forced convection heat transfer from outer surface of annular tube with rotating inner tube.

The present work aimed at obtaining a solution for the laminar forced convection in the entry region of an annulus with rotating inner walls .The results presented are expected to be of practical value in the design and analysis of journal bearings, turbo-machinery, cooling of electric machines, heat exchange equipment and similar applications. The results clearly highlight the significant role of the annulus radius ratio, rotating speed, and fluid axial velocity on the fluid flow and heat transfer characteristics.

2. Experimental Test-Rig:-

The test rig shown in Fig. 2.1 consists of the test section (1), the cold water tank (2), refrigeration system (3), hot water system (4), temperature recorder (5). The components of the systems and details will be explained as follows:

2.1. The test section

The test section shown in fig. 2.2 is the main part of the test rig. It consists mainly of two annuli namely the inner annulus and the outer one. Each one of the annulus consists of two tubes where the inner tube of the outer annulus consists the outer tube of the inner annulus. The outer annulus is made of two galvanized steel tubes with inner and outer diameter of 54.5mm and 60.3mm for the inner tube and 68.08 and 88.9 mm, for outer one, respectively. The tubes of the outer annulus are assembled coaxially via two steel flanges by welding so that the inner diameter of the flanges equals the inner diameter of the inner tube of the outer annulus. Two copper tubes of 6mm each are welded on the outer periphery of the outer tube of the outer annulus at the beginning and end of it. These copper tubes are connected to the cooling circuit. The inner annulus as shown in Fig 2.2 consists of the inner tube of the outer annulus as its outer tube while the inner tube of it is made out of PVC of variable outer diameter of 21, 26.5 and 33.5 mm. The inner tube is belt driven by an electric motor of ½ Hp and of 1477 r.p.m. The surface temperature of the inner

tube of the outer annulus is measured by four thermocouples embedded within the tube wall at various axial locations as shown in fig 2.2. Additional thermocouple is used to measure the ambient room temperature. Two thermocouples are used to measure hot water inlet and outlet temperatures and two thermocouples, are used to measure cold water inlet and outlet temperatures, All thermocouples are of (k) type

2.2. Refrigeration system:

The refrigeration system is consisted mainly of an evaporator and condensing unit. The evaporator consists of (15m) total long copper tube of diameter (5/8") wounded inside a plastic cylindrical tank with approximate volume of 150 liters. The condensing unit type DKSJ-100 Copeland is consisted of a (1hp) semi- hermetic compressor working with refrigerant R-12, connected to an air cooled condenser. The circuit is provided also with a receiver and thermostatic expansion valve type TF-O.S with orifice no2 from Danfoss company to control the refrigerant passing through the evaporator of the system.

2.3 Cold water system

The cold water system is consisted mainly of the evaporator of the refrigeration system, ¾ hp water pump and PVC plastic tubes. The evaporator submerged in side an aqua calcium chloride solution in order to permit lowering the temperature of the cooling water solution.

2.4 Hot water system

The hot water system consists of a cylindrical tank of volume 150 liters, provided by an electric heater of 2000 W power, and a thermostat to adjust the temperature of the hot water. The hot water is flowing inside the tested annular gap at a prescribed constant temperature.

3. Experimental Operation and Procedure

Three values can be varied independently in this system. The speed of rotation of the inner cylinder, the gap thickness (inner cylinder diameter), and the velocity of hot water inside the annular gap. For a given combination of these parameters, the outlet temperature of hot water will eventually, reach a certain fixed value, when this value is obtained, the system is in a state of equilibrium. This condition is ascertained by measuring water temperature at selected time intervals and observing any changes. When these changes become small (Within ± 0.1 C°), the steady state condition is assumed. For every case studied, the equilibrium time was approximately 30 minutes to be certain that the values recorded are representative at the steady state. This value is used in subsequent calculations.

Determining the heat transfer results in stationary annular tubes of different radius ratio and comparing them with the available correlation first standardized the experimental set up. Steady state values of mean Nusselt number for uniform temperature cooling of water then determined with each of the different radius ratio annular tubes for different speed of rotation of the inner tubes of the annuli. The characteristics of stationary annular tubes as well as rotational ones are shown in table (1).

Table 1 characteristic of annular tubes

Tube No	di mm	do mm	D_h mm	L mm	N rpm
1	33.5	54.4	6.3	550	0 -1580
2	26.5	54.4	21	550	0 -1580
3	21	54.4	33.3	550	0 -1580
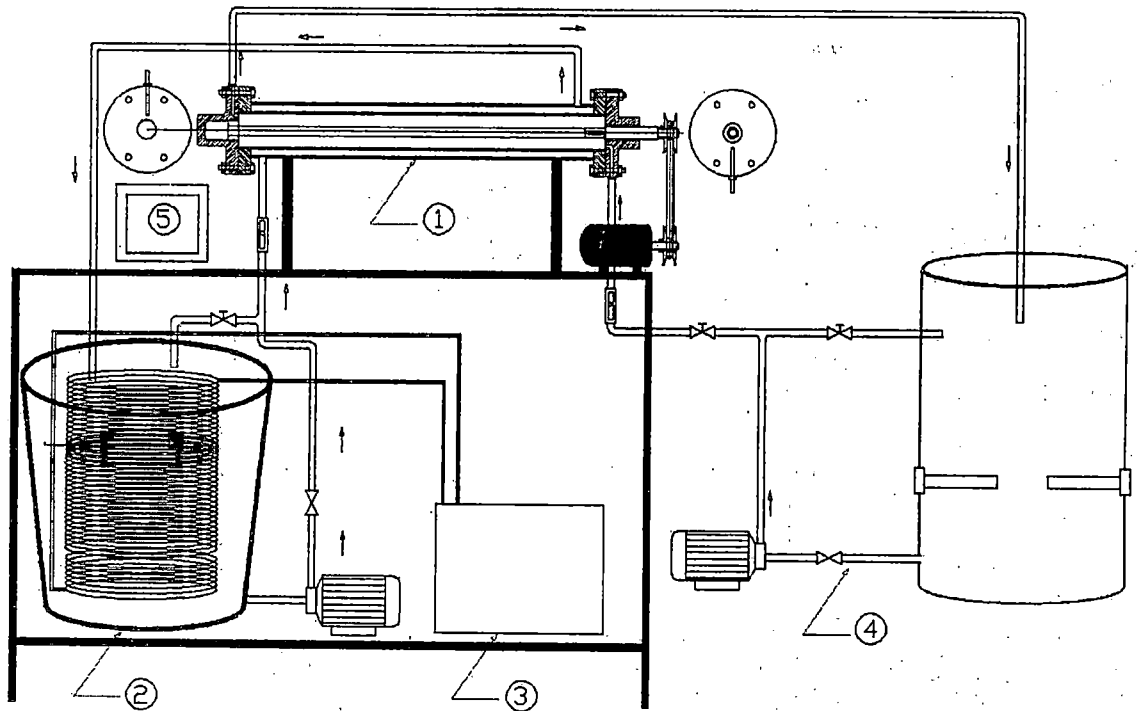
Figure 2.1. Experimental Test Rig

1. Test section.   3. Refrigeration system   5. Temperature recorder
2. Cold water system   4. Hot water system
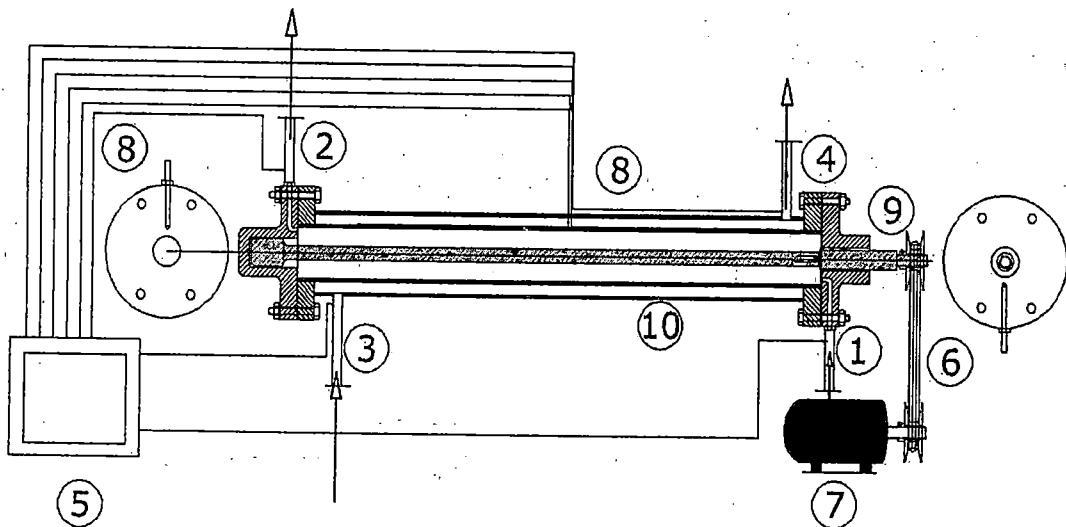
Figure 2.2. Test section

1. Inlet hot water   5. Temperature recorders   9. Outer section of rotated inner tube
2. Outlet hot water   6. V-belt dual groove belt pulley system   10. Test section (outer annulus)
3. Inlet cold water   7. Electric motor
4. Outlet cold water   8. Thermocouples location

4. Data Reduction

From the detailed constructions of the annular tubes described in table 1, the annulus cross sectional area can be set as:

$$A_C = \frac{\pi}{4}(D^2 - d^2) \qquad (1)$$

The volumetric diameter defined as four times the volume for flow divided by the area of the wetted surface is used as the hydraulic diameter, D_h for the case of comparison with the available literature. The hydraulic diameter can be expressed as:

$$D_h = (D - d) \qquad (2)$$

Based on hydraulic diameter as a characteristic length, Reynolds number of the flow through the annulus Re can be is given:

$$R_e = \frac{u\,D_h}{v} \qquad (3)$$

Nusselt number of the annular tube Nu based on the hydraulic diameter can be written as:

$$Nu = \frac{h\,D_h}{k} \qquad (4)$$

The heat transfer coefficient through the outer surface of the annular tube can be calculated from the heat balance equation described as.

$$Q = m\,c_p(T_{hwo} - T_{hwi})$$
$$= h\,A\,(LMTD) \qquad (5)$$

Where LMTD is the logarithmic mean temperature difference defined as:

$$LMTD = \frac{(T_{hwi}-T_w)-(T_{hwo}-T_w)}{\ln[\,(T_{hwi}-T_{wo})/(T_{hwo}-T_w)]} \qquad (6)$$

The effect of rotational speed of the inner tube may be accounted for by the rotational Reynolds number Re_ω based on hydraulic diameter and the inner diameter of the annular tube as a characteristic length. This rotational Reynolds number can be described as:

$$Re_\omega = \omega \times d\,\frac{D_h}{2v} \qquad (7)$$

Where

$$\omega = \pi\frac{N}{60} \qquad (8)$$

5. Results and discussion

Laminar forced convection of water flowing through annular tubes having rotational inner tube is studied experimentally. The outer tube of the annuls is subjected to a cold stream of water flowing through the outer annulus while the inner wall is insulated. The effect of axial Reynolds number Re, rotational Reynolds number, Re_ω and radii ratios of the annular tube d/D on heat transfer rates are discussed in the previous sup section. Tests are performed at of different radii ratio (0.386, 0.487, and 0.62) while both axial and rotational Reynolds number are varied over the range of 130 up to 2300, and 200 up to 1750, respectively.

5.1. Validation of the experimental results

Before reporting the main results of the present work, mention will be made of relevant auxiliary experiments. To demonstrate the validity of the experimental apparatus, heat transfer experiments were performed with stationary annular tube of radius ratio 0.487 in order to compare with heat transfer results in the literature. The relation between Nusselt number and Reynolds number based on the hydraulic diameter of the annular tube with radius ratio of 0.487 is presented in figure (3) as well as a straight line representing the correlation of sultan et al; [13]. It is concluded from the figure that the present experimental results for stationary annular tube with radius ratio of 0.48 are in fair agreement with the literature.
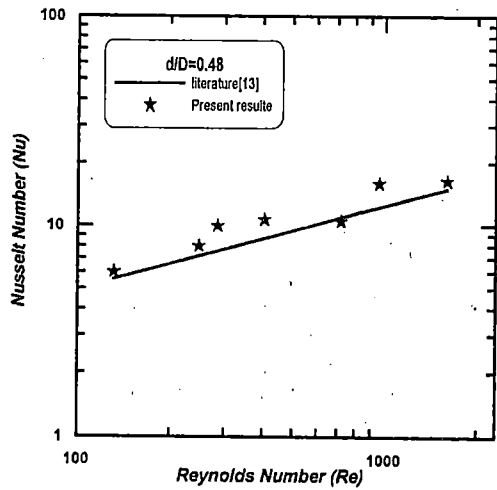
Fig.3:Comparison between Present heat transfers
Results and that reported in literature[13]

5.2 Stationary annular tube results

Experimental results for stationary annular tubes of radii ratios of 0.62, 0.487 and 0.386 are presented in figure 4. It can be seen that Nusselt number increases With Reynolds number. It is also noticed that Nusselt number decreases with the increase of radius ratio. In order to show the effect of diameter ratios on Nusselt number, elimination of Reynolds number is done via the complex term, $Nu/Re^{1/3}$.
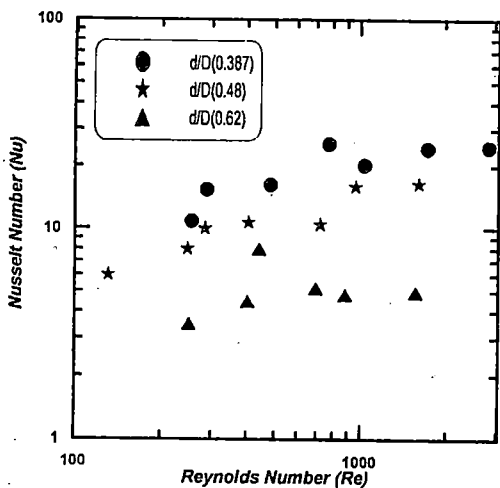
Fig.4:Relation between Nusselt Number
and Reynolds Numbers for different
radii ratios for stationary annulus

Figure 5 shows the relation between $Nu/Re^{1/3}$ and the radii ratios. It is seen from the figure that Nusselt number decreases with the increase in radii ratios at constant Reynolds number values.
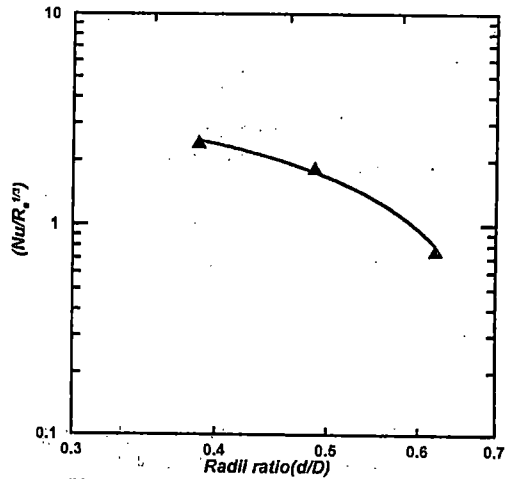

Fig.5:Relation between avarge Nusselt Number
and different radii ratio at stationary annulus

5.3 Stationary annular tubes correlation

An attempt is made to correlate the data of stationary annular tube of different radius ratio. A successive least square fit of the data collected for stationary annular tube of different radius ratios gave the following correlation:

$$Nu = 4.695\ Re^{1/3}\{1 - 1.397(d/D)\} \qquad (9)$$

In the following operating ranges: $130 \leq Re \leq 2300$ and $0.386 \leq d/D \leq 0.62$

This correlation predicts the values of Nusselt number, which agree with the measured one within ± 15% as shown in figure 6.

5.4 Rotational annular tube results

Attention will now be turned to the heat transfer results of rotational annular tubes. In this connection the sequence in which the experiments were performed was chosen in order to facilitate the presentation of results. For a specified value of radius ratio (d/D),
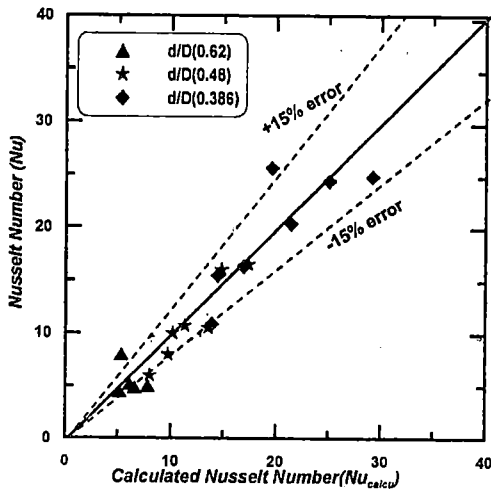
Fig.6:Comparison between the proposed correlation
and Exp-data for stationary annulus

Successive experiments are carried out with different values of rotational Reynolds number, Re_ω one at a time. The axial Reynolds number, Re is changed for each radii ratio and rotational Reynolds number in order to cover its specified range each time. Therefore three groups of experiments were performed and discussed in the following section.

The heat transfer coefficients of the outer surface of the annular tubes in case of laminar flow of water through the annular space were analyzed in terms of Nu-Re relationships. Figures (7-9) show the variation of Nusselt numbers, Nu with Reynolds number, Re for annular tubes of radius ratios of 0.62, 0.487 and 0.386, respectively, at different rotational speeds of 1580,657,571,491,400 and 350 r.p.m, respectively, Figure 7 shows that for a radius ratio of 0.62 the rotational speed of nearly 1580 r.p.m (Re_ω=1744 nearly) produce a maximum increase in Nusselt number of the order of 200 %, whereas the rotational speed of 571.5 r.p.m (Re_ω=460 nearly) yielded an increase of only 66 % in Nusselt number, compared to the results of the stationary annular tube of the same radius ratio. The same trend can be seen from figures 8 and 9 with an increase in Nusselt number of 130% and 175%, and an

Increase in Nusselt number of 75% and 36% at radii ratios of 0.487 and 0.386 respectively in comparison with the results of stationary annular tubes with the same radius ratios.
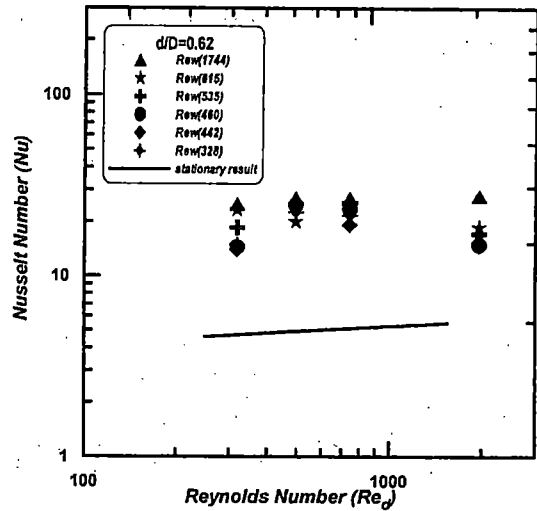

Fig.7:Relation between Nusselt Numberand Reynolds Number at fixed radius ratio of 0.62 and different rotational Reynolds Number.

The effect of rotational speed (rotational Reynolds number) on the heat transfer rate can be shown from figures 10 and 11. It is seen from Fig 10 for axial Reynolds number of 806 that Nusselt number increases with the increases rotational Reynolds number and with the decrease of diameter ratio. The same trend is also shown in Fig 11 for an axial Reynolds number of 537.

5.5 Rotational annular tubes correlation

An attempt is made to correlate the data of rotational annular tube of different radius ratio.

A successive least square fit of the present. Experimental data of rotational annular tube of different radii ratios gave the following correlation:

$$Nu = 4.61 Re^{1/3} Re_\omega^{0.155}\{1-1.397(d/D)\} \quad (10)$$

in the following operating ranges:
$130 \leq Re_d \leq 2300$,   and
$299 \leq Re_\omega \leq 1750$ and

$0.386 \leq d/D \leq 0.62$

The above correlation predicts the values of Nusselt number, which agree with the experimental results within $\pm$ 26% as shown in figure 12.
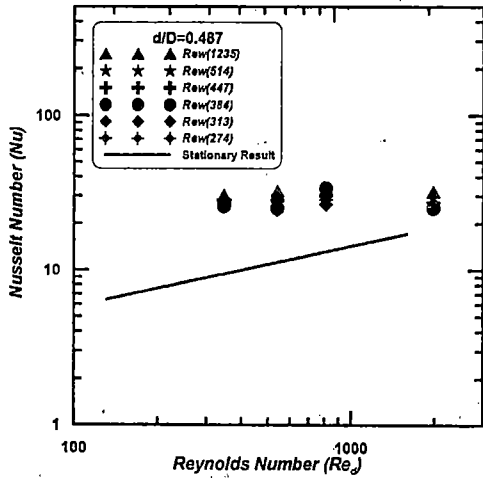

Fig.8:Relation between Nusselt Number and Reynolds Number at fixed radius ratio of 0.487 and different rotational Reynolds Number.
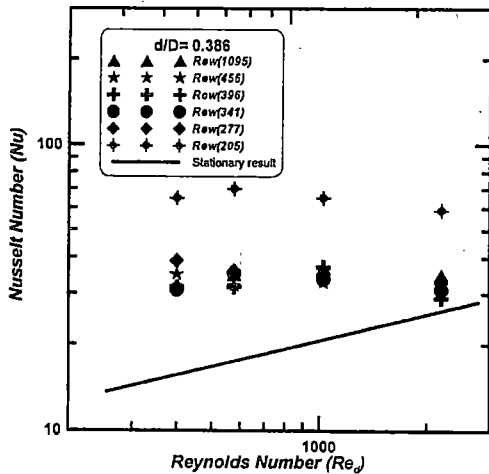

Fig.9:Relation between Nusselt Number and Reynolds Number at fixed radius ratio 0.386 and different rotational Reynolds Number.
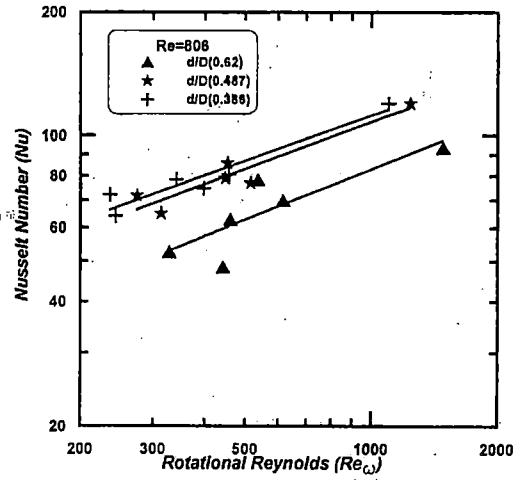

Fig.10:Relation between Nusselt Number and rotational Reynolds Number at fixed Reynolds Number 806 and different radii ratio.
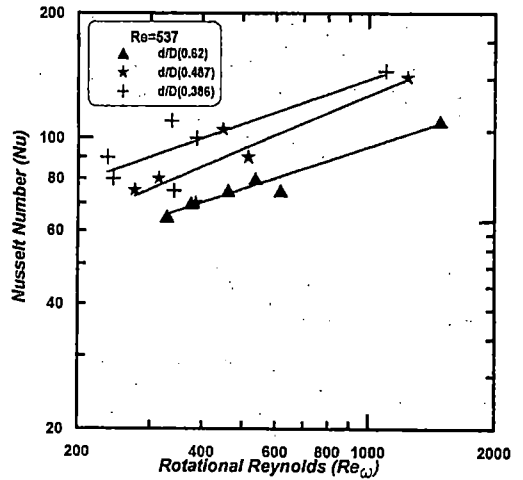

Fig.11:Relation between Nusselt Number and rotational Reynolds Number at fixed Reynolds Number 537 and different radii ratio.
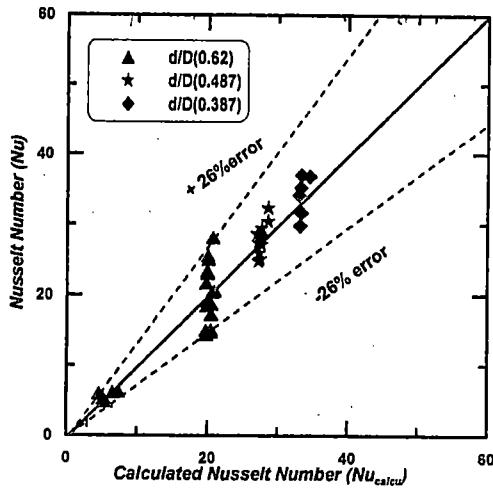
Fig.12:Comparison between the proposed correlation and Exp-data for rotating annulus

6. Conclusions

From the results of experimental investigations of stationary and rotating annuli, the following conclusions may be drawn:

1- Nusselt number increases with Reynolds number for both stationary and rotated annuli.

2- Nusselt number increases with the decrease of diameter ratio of the annulus for both stationary and rotating annulus.

3- Nusselt number increases with the increase of rotational Reynolds number at constant axial Reynolds number and constant diameter ratio.

4- The increase in Nusselt number ranges from 36% to 200% over that of stationary annulus. and the rotational annulus in the range of diameter ratio and axial Reynolds number

5- Two correlations are made to describe the relation between Nusselt numbers and axial Reynolds number, diameter ratio and rotational Reynolds number.

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