NATURAL CONVECTION HEAT TRANSFER IN A VERTICAL CONCENTRIC ANNULUS

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ABSRACT

Experiments were carried out to study the local and average heat transfer by natural convection in a vertical concentric cylindrical annulus. The experimental setup consists of an annulus has a radius ratio of 0.555 and inner cylinder with a heated length 1.2m subjected to the constant heat flux while the outer cylinder is subjected to the ambient temperature. The investigation covers heat flux range from 58.2 W/m^2 to 274.31 W/m^2. Results show an increase in the natural convection as heat flux increases leads to an improve in the heat transfer process. An empirical equation of average Nusselt number as a function of Raylieh number was deduced .

الخلاصة

أجريت تجارب عملية لدراسة انتقال الحرارة الموقعي و المعدل بالحمل الحر بتجويف حلقي بين أسطوانتين متحدتي المركز بالوضع العمودي، نسبة نصف القطر لهما تعادل 0.555 و بطول 1.2 متر . سخنت الأسطوانة الداخلية تحت فيض حراري ثابت، بينما عرّضت الأسطوانة الخارجية إلى درجة حرارة الجو. تغطي الدراسة مدى للفيض الحراري يتراوح من 58.2 W/m^2 إلى 274.31 W/m^2 . بيّنت النتائج زيادة معدلات الحمل الحر بزيادة الفيض الحراري مما يؤدي إلى تحسين معامل انتقال الحرارة.

KEY WORDS

Heat Transfer, Natural Convection, Concentric Annulus

INTRODUCTION

The problem of natural convection heat transfer across a horizontal and vertical cylindrical annulus has received considerable attention in view of its fundamental importance germane to numerous engineering applications. As a result, extensive experimental and theoretical works dealing with the flow and associated heat transfer characteristics of natural convection in such configuration have been reported in the literature. Comprehensive reviews on natural convection in concentric and eccentric annuli are available **(Kueehn 1976 &1978, Van de Sande 1979, and Yao 1980)** and there is no need to repeat them. However, all of the previous studies are concerned with the horizontal annulus; little attention has been paid to annuli with a vertical position. **(Van de Sande and Hamer 1979)** have obtained empirical correlations for natural convection heat transfer in concentric and eccentric annuli of constant heat flux. **(Glakape et al. 1986)** presented a numerical solution for air in concentric and eccentric configurations with specified constant heat flux at the boundaries. **(Akeel 2005)** has presented an experimental and theoretical study for mixed convection heat transfer through concentric annuli. The lack of experimental data concerning the problem of natural convection in the vertical concentric annulus, and the practical importance of this problem in the industry applications, motivated the present work.

EXPERIMENTAL APPARATUS

The test section is shown diagrammatically in **Fig.1** and consists of 4 mm wall thickness, 50 mm outside diameter and 1.2 m long aluminum cylinder (K) located centrally in 5 mm thickness , 90 mm inside diameter and 1.2 m long aluminum cylinder (I), by fitting it at the test section inlet with the 20 mm inside diameter , 50 mm outside diameter and 15 mm long Teflon tube (N) and at the test section exit with the teflon piece (M). A ring (P) is used to hold and support the aluminum cylinder (K) with the teflon piece (N) centrally inside the settling chamber by adjustable screws (Q). The teflon was chosen because of its low thermal conductivity in order to reduce the heat loss from the aluminum cylinder ends. A well design Teflon bell mouth (H) is fitted at the annulus outer aluminum cylinder (I) and bolted inside the settling chamber (D). The inlet air temperature was measured by one thermocouple (J) located in the settling chamber (D) while the outlet bulk air temperature was measured by three thermocouples (Z) located in the test section exit . The local bulk air temperature was calculated by using a straight line interpolation between the measured inlet and outlet bulk air temperature.

The inner cylinder was heated electrically using an electrical heater which consists of a nickel-chrome wire , wound as a coil spirals around solid teflon tube and is covered by a 2 mm thickness asbestos layer , and the space between the asbestos and the inner cylinder wall is fitted with a fine grade sand to avoid heat convection in it and to smooth out any irregularities in the heat flux. The hole apparatus is designed with a view to obtain a good concentricity of the core cylinder and the containing cylinder. The temperature of the outside surface of the inner cylinder was measured by seventeen asbestos sheath alumel-chromel (type K) thermocouples , arranged along the cylinder , the measuring heads of the thermocouples were made by fusing together the ends of two wires.

The thermocouples were fixed by drilling holes of 1.5 mm diameter in the cylinder wall and the ends of the holes chamfered by a 3 mm slug to locate the measuring junctions which were then fixed by a high temperature application Defcon adhesive . The excess adhesive was removed and the cylinder outer surface was cleaned carefully by fine grinding paper. All the thermocouples wires and heater terminals were taken out the test section through both teflon pieces (N,M) .

On the other hand , ten thermocouples (type K) were used to measure the inner surface temperature of the annulus outer cylinder (I). Thermocouples positions at the outer surface were located and then a 2 mm deep pits were drilled in which the thermocouples were fixed by Defcon adhesive. All thermocouples were used with leads , the thermocouple with lead and without lead were calibrated against the melting point of ice made from distilled water and the boiling points of several pure chemical substances. To determine the heat loss from the test section ends, two thermocouples were fixed in each teflon piece. The distance between these thermocouple was 12 mm. Knowing the thermal conductivity of the teflon , the ends condition could thus be calculated.

EXPERIMENTAL PROCEDURE

To carry out an experiment the following procedure was followed:

1- The electrical heater was switched on and the heater input power then adjusted to give the required heat flux.

2- The apparatus was left at least three hours to establish steady state condition. The thermocouples readings were measured every half an hour by means of the digital electronic multimeter until the reading became constant , a final reading was recorded. The input power to the heater could be increased to cover another run in a shorter period of time and to obtain steady state conditions for next heat flux.

3- During each test run , the following readings were recorded:

   a- The readings of the thermocouples in ˚C.

   b- The heater current in amperes.

   c- The heater voltage in volts.

DATA ANALYSIS

Simplified steps were used to analyze the heat transfer process by natural convection from the inner cylinder which was subjected to a uniform heat flux while the outer cylinder was subjected to the ambient temperature. The total input power supplied to the inner cylinder can be calculated:

$$Q_t = V'' \times I \tag{1}$$

The convection and radiation heat transferred from the inner cylinder is :

$$Q_{cr} = Q_t - Q_{cond} \tag{2}$$

where Q_{cond} is the conduction heat loss which was found experimentally equal to 5 % of the input power. The convection and radiation heat flux can be represented by:

$$q_{cr} = Q_{cr}/A \tag{3}$$

where:

$A = 2\pi\, r_1\, L$

The convection heat flux , which is used to calculate the local heat transfer coefficient is obtained after deduce the radiation heat flux from q_{cr} value. The local radiation heat flux can be calculated as follows:

$$q_r = F_{1-2}\, \varepsilon\, \sigma \left[\left((t_s)_z + 273 \right)^4 - \left((\overline{t_{s2}})_z + 273 \right)^4 \right] \tag{4}$$

where:

F_{1-2}= view factor between inner and outer cylinder ≈1

$(t_s)_z$ = local temperature of inner cylinder.

$\overline{(t_{s2})}_z$ = average temperature of outer cylinder.

ε = emissivity of the polished aluminum surface=0.09.

Hence the convection heat flux at any position is:

$$q = q_{cr} - q_r \tag{5}$$

The local heat transfer coefficient can be obtained as:

$$h_z = \frac{q}{(t_s)_z - (t_b)_z} \tag{6}$$

$(t_b)_z$ = Local bulk air temperature.

All the air properties were evaluated at the mean film air temperature (Keys 1966):

$$(t_f)_z = \frac{(t_s)_z + (t_b)_z}{2} \tag{7}$$

t_f = Local mean film air temperature.

The local Nusselt number (Nu_z) then can be determine as:

$$Nu_z = \frac{h_z \quad D_h}{\kappa} \tag{8}$$

The average values of Nusselt number Nu_m can be calculated based on calculation of average inner surface temperature and average bulk air temperature as follows:

$$\overline{t}_s = \frac{1}{L} \int_{z=0}^{z=L} (t_s)_z \quad dz \tag{9}$$

$$\overline{t}_b = \frac{1}{L}\int_{z=0}^{z=L} (t_b)_z \quad dz \qquad (10)$$

$$\overline{t}_f = \frac{\overline{t}_z + \overline{t}_b}{2} \qquad (11)$$

$$Nu_m = \frac{q \quad D_h}{k(\overline{t}_z - \overline{t}_b)} \qquad (12)$$

The average values of the other parameters can be calculated as:

$$Gr_m = \frac{g \quad \beta \quad D_h^{\ 3} \quad \left(\overline{t}_z - \overline{t}_b\right)}{\nu^2} \qquad (13)$$

$$Pr_m = \frac{\mu \quad C_p}{k} \qquad (14)$$

$$Ra_m = Gr_m \, . \, Pr_m \qquad (15)$$

where:

$$\beta = 1 \big/ \left(273 + \overline{t}_f\right)$$

All the air physical properties ρ, μ, ν, and k were evaluated at the average mean film temperature ($\bar{t}_f$).

EXPERIMENTAL RESULTS

The variation of the inner cylinder surface temperature for different heat flux is shown in **Fig.2** . It is obvious that the surface temperature increases at the stage of entrance and attains a maximum point after which the surface temperature begins to decrease at high heat flux ($q \geq 181$ W/m^2) and be almost constant for small heat flux ($q \leq 152$ W/m^2). The rate of surface temperature rises at early stage is directly proportional to the wall heat flux because of the faster increasing of the thermal boundary layer as heat flux increase (i.e., increasing of buoyancy effect).

Fig.3 & 4 show the effect of heat flux on the local and average Nusselt number along the inner cylinder. It is clear that the results of higher heat flux for local and average Nusselt number are higher than that of lower heat flux. **Fig.3** shows also sharp decrease for the local Nusselt number values at the entrance of the annulus because the boundary layer thickness is zero and the natural convection is poor in this region, then increase downstream because of increasing of natural convection. The values of the mean Nusselt number are plotted in **Fig.5** in the form of log(Nu_m) against log(Ra) for the range of Ra from 0.68611×10^5 to 1.728559×10^5. All the points as can be seen are represented by linearization of the following equation.

$$Nu_m=2.31812\ Ra^{0.083188} \qquad (16)$$

CONCLUSIONS

1. The extent of the local mixing increases as the heat flux increases.
2. The heat transfer process improves as heat flux increase.
3. The effect of buoyancy is small at the annulus entrance and increase down stream.

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NOMENCLUTURE

A: inner cylinder surface area; m^2

D_h: hydraulic diameter=$2(r_2-r_1)$: m

I: current; Amp

κ : thermal conductivity; $W/m^2.°C$

L: annulus length; m

(Nu_m): mean Nusselt number

Q: convection heat loss; W

Q_t: total heat given; W

Q_{cr}: convection- radiation heat loss; W

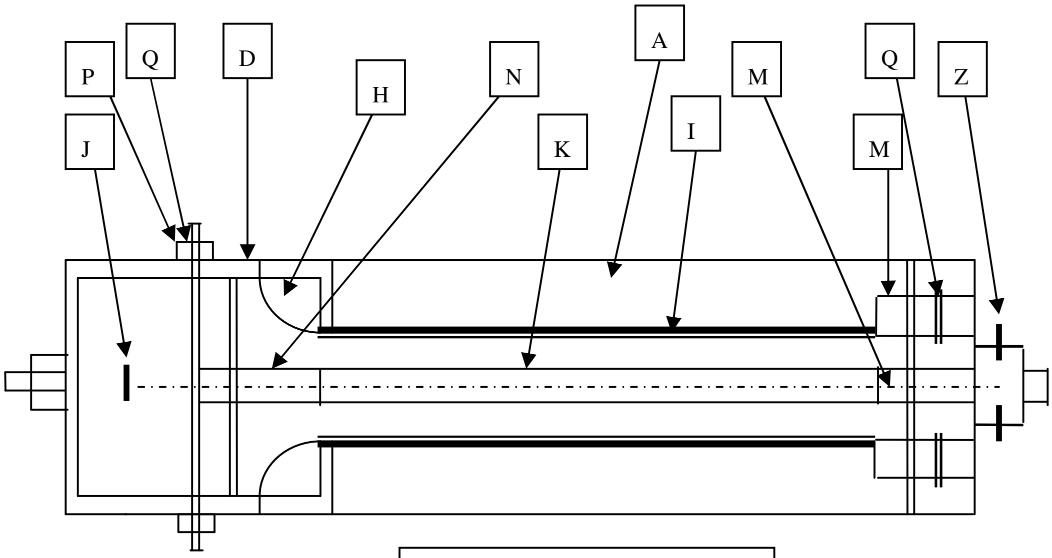
q_r: radiation heat flux; $W/m^2.°C$

q: convection heat flux; $W/m^2.°C$

r_1: outer radius of inner cylinder; m

r_2: inner radius of outer cylinder; m

V”: voltage; volt



A WOODEN BOARD

D SETTLING CHAMPER

H BELL MOUTH

I OUTER CYLINDER

K INNER CYLINDER

J THERMOCOUPLE

M,N TEFLON

P RING

Z THERMOCOUPLE

Q SCREW

Fig. 1: Diagram of Experimental Apparatus.

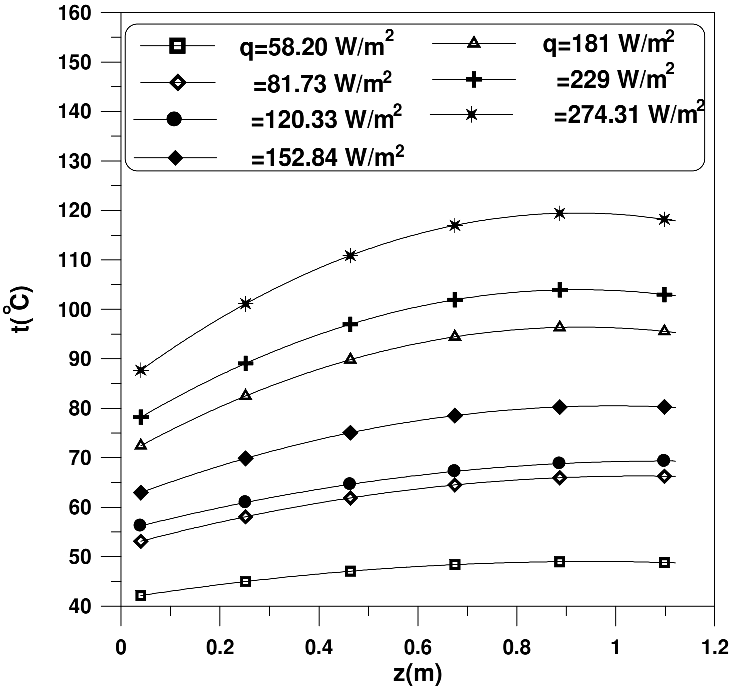


Fig.2: Variation of the Surface Temperature with the Axial Distance

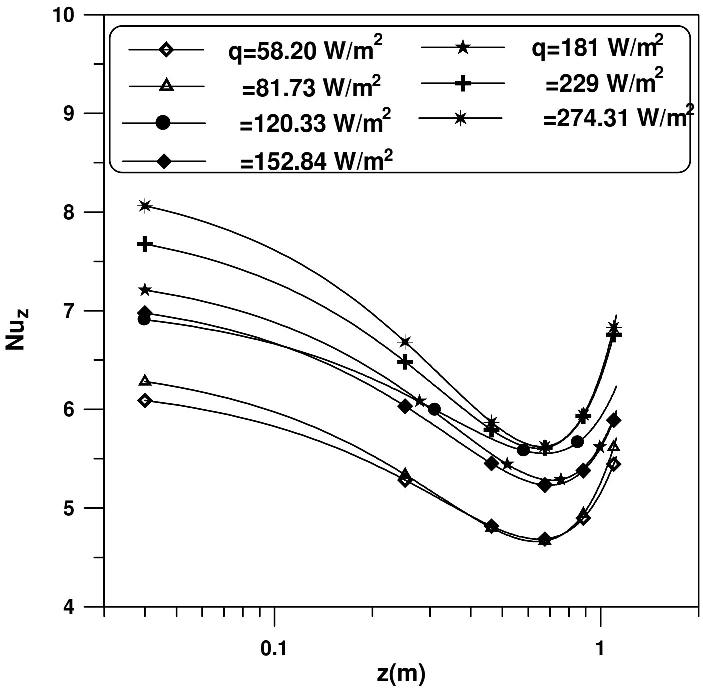


Fig.3: Local Nusselt Number Versus Axial Distance.

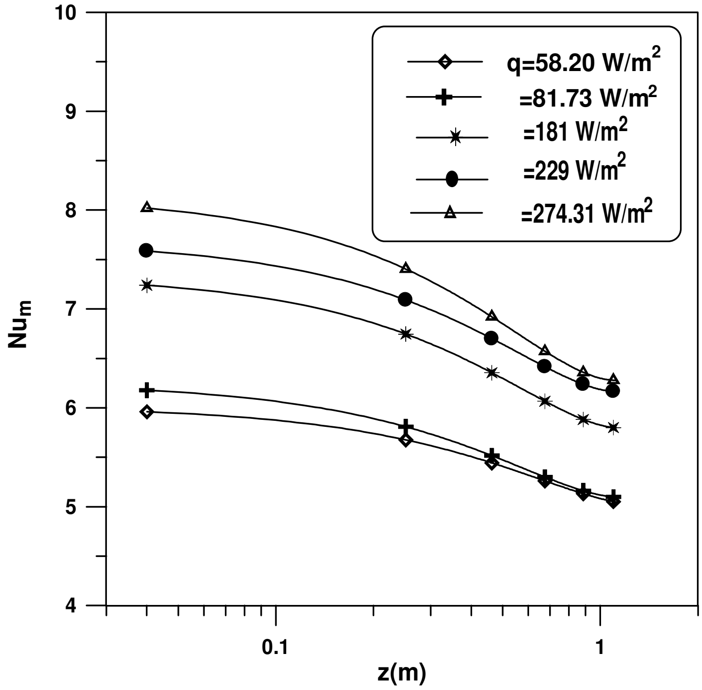


Fig.4: Average Nusselt Number Versus Axial Distance.

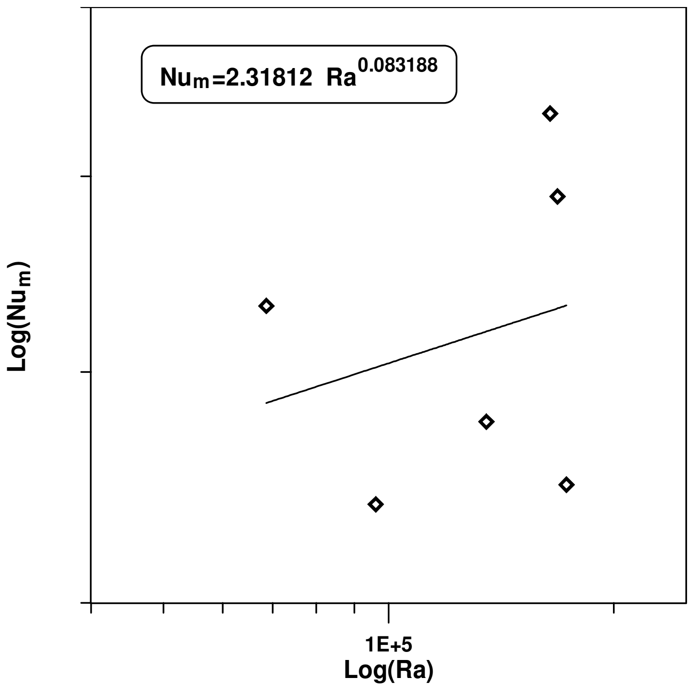


Fig.5: Logarithm Average Nusselt Number Versus log(Ra).