LAMINAR NATURAL CONVECTION IN HORIZONTAL HALF CYLINDRICAL GAPS

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ABSTRACT

A modification of the parabolic trough solar collectors is proposed where a flat rectangular collector instead of the conventional circular tube receiver was used. The space between the outer transparent cylinder and the rectangular collector forms a half cylindrical gap. The laminar natural convection in this gap is studied by a numerical solution of the simultaneous equations of mass, momentum, and energy based on the SIMPLE method where an irregular grid of 50X40 ($R \times \theta$) was used. The present study covers a wide range of the parameters affecting the average Nusselt number in the gap. The rotation angle, ω changes from 0° (horizontal plate) to 90° (vertical plate); the ratio of the cylinder diameter to flat plate length, RH changes from 1.33 to 4, and the Rayleigh number based on the flat plate length , Ra_H changes from 10^3 to 10^6. The different flow regimes are explained through the isothermal and stream lines for each of the studied parameters. Finally, a correlation is given for the average Nusselt number as a function of ω and Ra_H .

Keywords: Heat transfer, Natural convection, Half cylindrical gaps.

Nomenclatures

d_i	diameter of inner cylinder
d_{i*}	length scale
d_o	diameter of outer cylinder
h	local coefficient of heat transfer
$\bar{h}$	average coefficient of heat transfer
H	flat plate length
k	fluid thermal conductivity
$\overline{Nu}_H$	average Nusselt number based on plate length
P	dimensionless pressure
Pr	Prandtl number
q	heat flow per unit length
r	radius
R	dimensionless radius
Ra_H	Rayleigh number based on plate length
Ra_{di}	Rayleigh number based on inner diameter
RH	cylinder diameter / flat plate length
S_ϕ	linearized source term
T	temperature
V_R	dimensionless velocity in radial direction
V_θ	dimensionless velocity in tangential direction

Greek

α	thermal diffusivity
γ	radii ratio, do /di
Γ_ϕ	diffusion coefficient
θ	angular direction
ϑ	dimensionless temperature
ξ	temperature difference ratio, $(T_h - T_c) / T_c$
ρ	fluid density
τ	transmissivity
Φ	transported variable
Ψ	angle measured from horizontal axis
ω	angle of rotation

subscripts

c	cold
h	hot

1. INTRODUCTION

The parabolic cylinder concentrator which is known as the parabolic trough is generally used in solar energy conversion systems to obtain high temperatures in the range of 100 °C to 500 °C. The solar system shown in Figure (1) consists of a cylindrical parabolic reflector, glass-tube receiver (receiver tube surrounded by a glass tube envelope), storage tank, pump and connecting pipes. The incident radiation is directed from the parabolic reflector to a glass-tube receiver of a small area to increase the intensity of radiation and therefore high fluid temperatures can be obtained. Several studies were devoted to the design and performance of the parabolic trough concentrator. A simulation for the performance of the solar system was given by Ghazy [1]. He considered two types of receivers, the conventional copper black painted tube and glass-tube receivers. His results indicated that glass-tube receivers having transmissivity $\tau = 0.9$ in the short-wave region yield better performance than those with copper-tube receivers. A normal black painted or polished aluminum cavity was used by Sayed [2] as a black cavity bounding the absorber tube from above. This cavity resulted in a significant increase in the collector efficiency which reached 62 % at a mass flow rate of 0.033 kg/s.

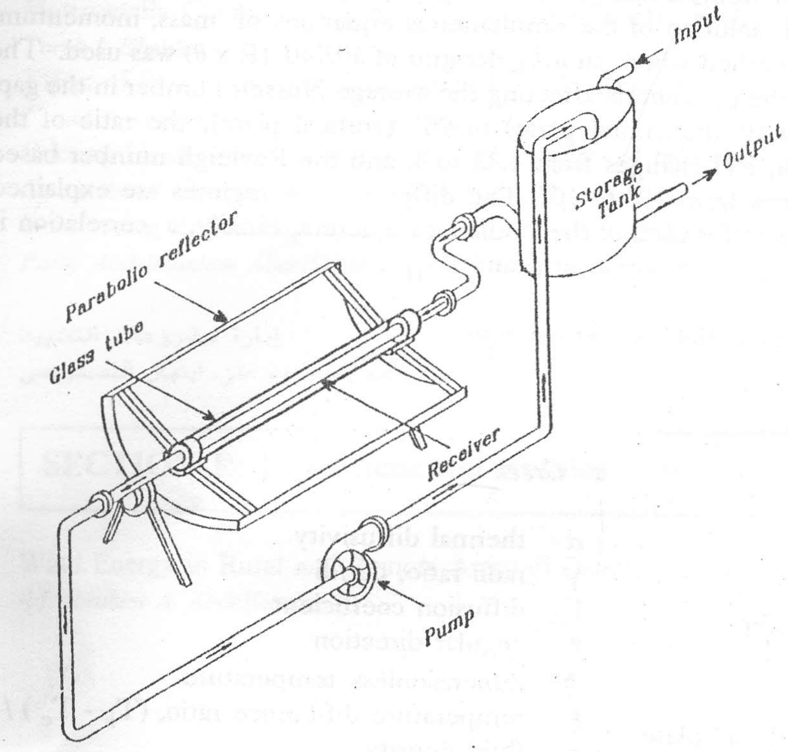

Figure 1. A sketch of the parabolic concentrating system.

Edenburn [3] analytically predicted the efficiency of cylindrical parabolic concentrators at different orientations and the results were in good agreement with measurements carried out at Sandia laboratories, USA .DEL Manufacturing Company at California, USA [4] designed parabolic trough concentrators for high performance and low cost. Each parabolic concentrator consisted of glass mirror segments sagged to the appropriate parabolic shape and back-silvered to obtain a high reflectivity (0.977). A parabolic trough solar collector with an innovative porous absorber receiver was investigated by Grald and Kuehn [5]. A finite difference method was used to study the effects of mass flow rate, acceptance angle, receiver dimensions, and material properties on the thermal efficiency. The numerical results showed a remarkable increase in thermal efficiency over commercially available parabolic trough concentrators.

Rabl [6] summarized more than three years of research on non-evacuated parabolic trough concentrators and reviewed measured performance data and critical design concentrations. Concentration in the upper portions of the practical range can provide good efficiency (40 - 50 %) in the 100-160°C temperature range with relatively frequent tilt adjustments (12 - 20 times per year). At low concentrations, performance will still be substantially better than that for a double glazed flat plate collector above about 70 °C and competitive below, while requiring only semi-annual adjustments for year round operation. In both cases, the cost saving associated with inexpensive reflectors and the optimal coupling to smaller and simple inexpensive absorbers can be as important as the improved thermal performance.

To improve the performance of the solar energy conversion system, we need to apply new developments on the system components. One of the main areas is the development of the tube receiver. The conventional receiver is usually made of two concentric cylinders. The outer cylinder is made of transparent material (usually glass or Pyrex) and the inner one is a black painted copper tube. This combination of the two tubes forms an annulus with a cold outer surface and a hot inner surface as shown in Figure (2-a). The reflected solar rays which come from the parabolic reflector impinge upon the outer surface of the glass tube. Due to the continuous tracking of the sun rays by the receiver, the incoming rays usually occupy a fixed angle on the annulus. On the other hand, the inner tube loses heat to the outer one by convection and radiation. To reduce this heat loss, it is proposed to :

a- Mirror the inner surface of the outer tube except that angle occupied by the reflected sun rays as shown in Figure (2-b). This is expected to reduce the radiation loss within the annulus.
b- Fill the annulus between the two concentric cylinders with an insulating material except that angle occupied by the reflected sun rays as shown in Figure (2-c).
c- Attach small fins to the inner tube of the previous case as shown in Figure (2-d). This will help in using a smaller diameter for the receiver tube and therefore will reduce the heat loss.
d- Use a flat rectangular collector as shown in Figure (2-e) instead of the circular tube. This may reduce the convection loss at the period of peak collection (noon time), when the flat receiver is horizontal. The present work deals with the convection losses in the cylindrical gap given in the receiver development of this case.

Literature survey reveals that although several theoretical and experimental studies were given for the natural convection in complete horizontal annuli, none was found for the present case. Kuehn and Goldstein [7] presented a comprehensive review of the investigations up to 1976. A numerical study of the turbulent natural convection in a complete annulus of a radius ratio of 2.6 was given by Farouk and Güceri [8] in the Rayleigh number range from 10^6 to 10^7. Cho et al [9] used a bi-polar coordinates system to investigate the local and overall heat transfer between concentric and eccentric horizontal cylinders for a range of Rayleigh numbers less than $5x10^4$ based on the difference of radii. They found that a very small eccentricity gives an overall thermal behaviour similar to that of the exactly concentric cylinders. Mahony et al [10] and Hessami et al [11] investigated the effect of variable properties on natural convection in a horizontal annulus. They found that the Boussinesq approximation is valid for a temperature ratio $\xi = (T_h - T_c)/T_c$ less than .1, but can be used for a ratio up to 0.2 with reasonable accuracy in the calculated heat transfer. They also found that the Boussinesq approximation slightly overestimates the tangential velocity and the temperature gradient near the hot inner cylinder. Hessami et al [12] used air, glycerin and mercury, and showed that glycerin is more sensitive to the constant properties assumption, while air had not been significantly affected by this assumption. This is due to the large radii ratio they used ($\gamma = d_o/d_i = 11$). Finally their experimental data have been correlated with some other data from the literature for smaller values of γ. They also showed that the heat transfer from the inner cylinder should be almost the same as that in an infinite medium when $\gamma \geq 10$.

The various flow patterns in horizontal concentric annuli were investigated both numerically and experimentally by Rao et al [13] with two- and three-dimensional calculations over a range of Rayleigh numbers and radii ratio. The transition to the multicellular flow pattern which has been observed by Powe et al [14] is confirmed, and an oscillating flow is generated at the upper part of the annulus as Rayleigh number was increased.

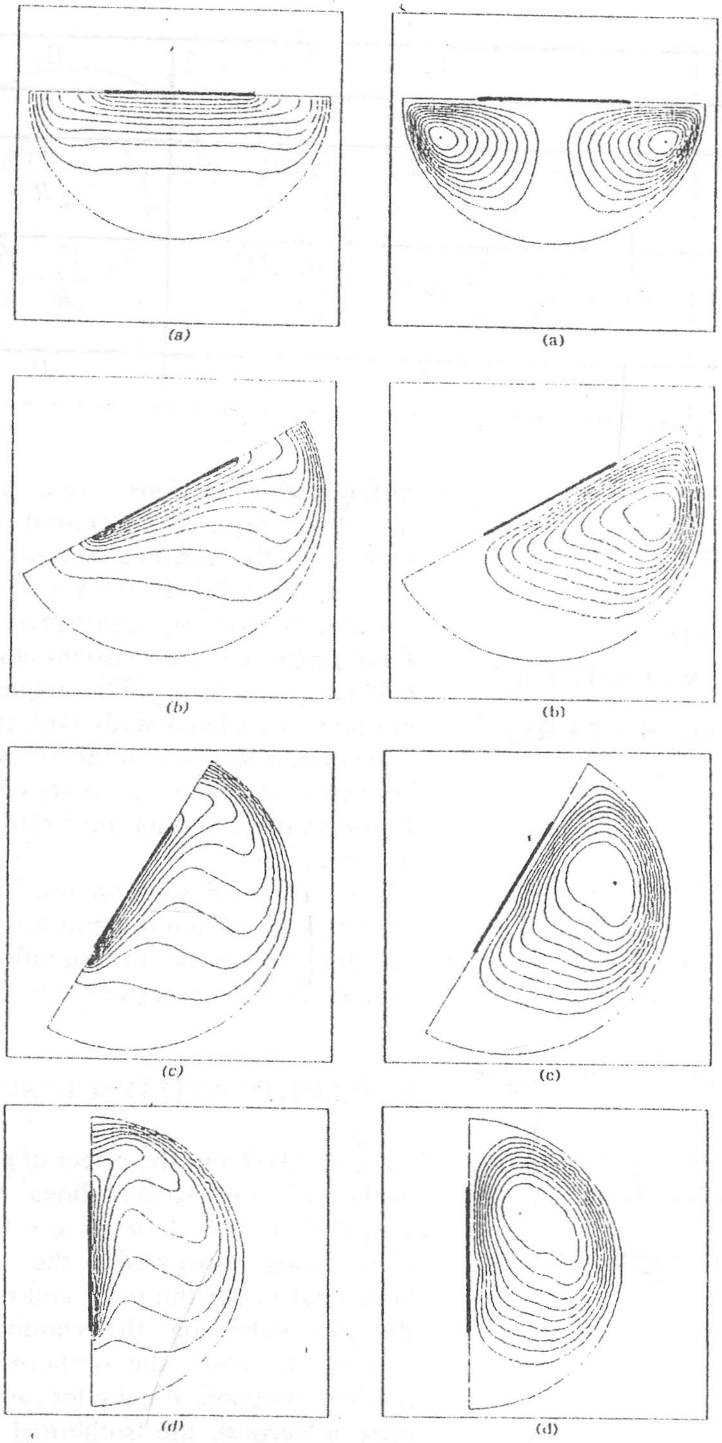

Figure (3) Effect of rotation angle on isothermal and stream lines

a) $\omega = 0°$ b) $\omega = 30°$ c) $\omega = 60°$ d) $\omega = 90°$

The stream lines for $\omega = 0°$ indicate two similar big cells with their centers near the far ends. When the plate rotates to the vertical position, the two cells merge into one big cell filling the whole domain.

Figure (4) shows the effect of RH on isothermal and stream lines for $\omega = 90°$, and $Ra_H = 10^5$. For RH = 4, the isothermal lines are dense in the region near the hot plate. The stream lines indicate almost stagnant cold air at the lower part of the cavity. As we increase the plate length (lower values of RH), the isotherms become more smooth and fill the cavity and the fluid moves in one big call as shown in the stream lines of case (c). The effect of rotation angle on the average Nusselt number for RH = 2, and $10^3 \leq Ra_H \leq 10^6$ is shown in Figure (5). For $Ra_H \leq 10^3$, the conduction regime prevails and no rotation effect takes place. As Ra_H increased, $\overline{Nu_H}$ was also increased and the rotation effect becomes more effective. For example, at $Ra_H = 10^6$, the value of $\overline{Nu_H}$ at $\omega = 90°$ is almost three folds of that at $\omega = 0°$.

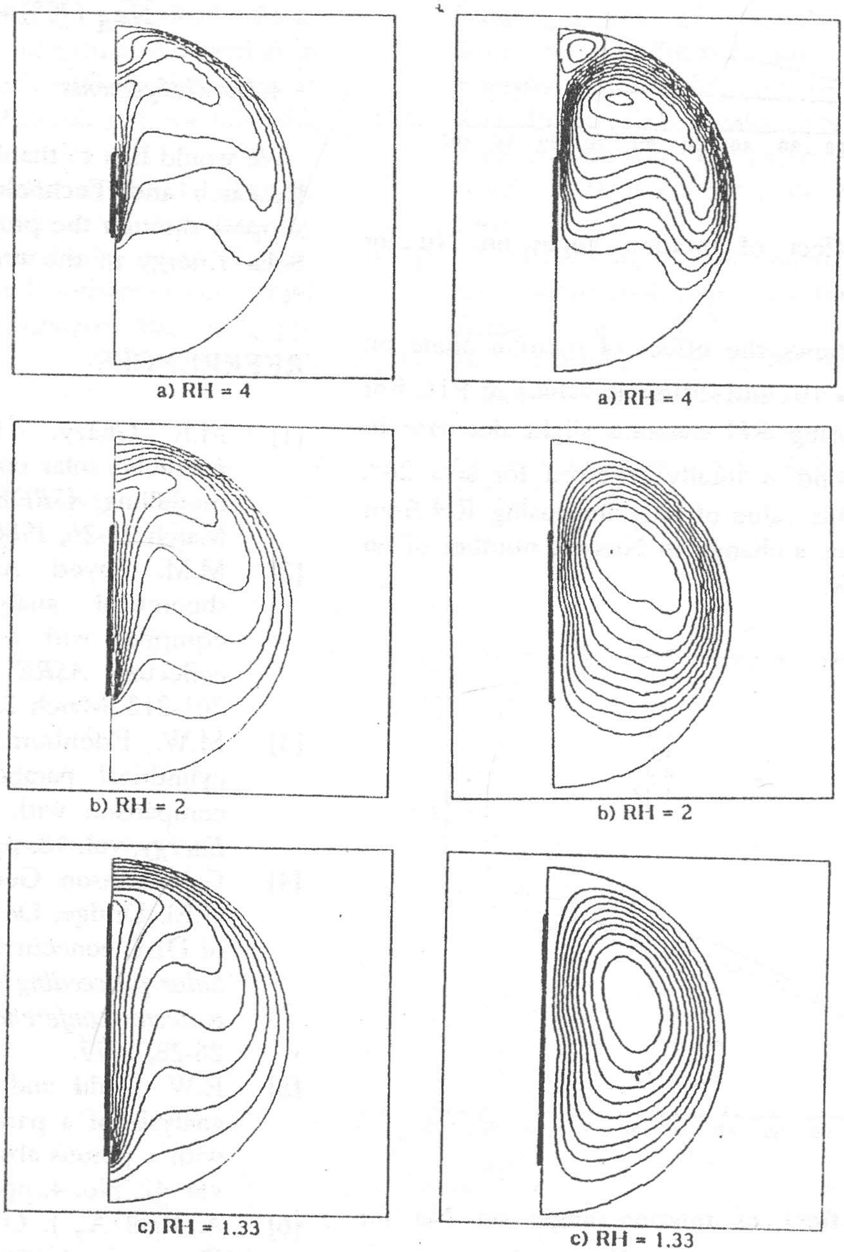

Figure 4. Effect of RH on isothermal and stream lines.

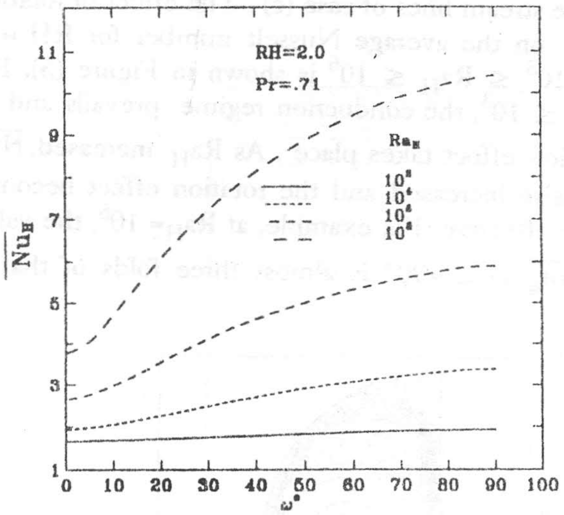

Figure 5. Effect of rotation angle on Nu for different Ra_H

Figure (6) shows the effect of rotation angle on $\overline{Nu_H}$ for $Ra_H = 10^5$ and different values of RH. For $\omega = 0^o$, increasing RH causes a slight decrease in $\overline{Nu_H}$. This trend is totally reversed for $\omega > 20°$. However, at this value of Ra_H, increasing RH from 1.33 to 4 causes a change in Nusselt number of no more than 15%.

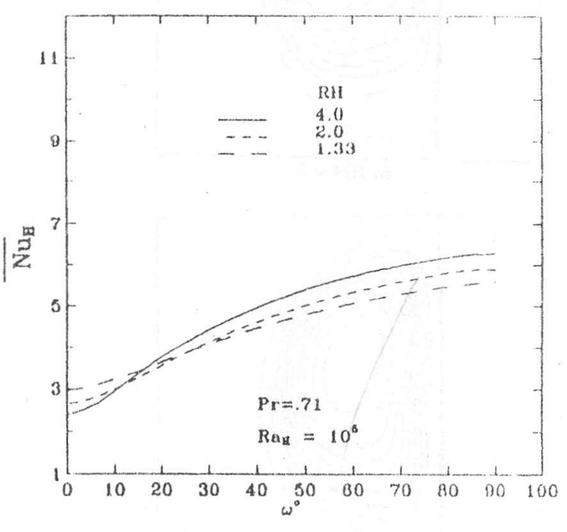

Figure 6. Effect of rotation angle on Nu for different RH.

5. CORRELATIONS

The data for RH = 2 were correlated as follows

$$\overline{Nu_H}(\omega) = \overline{Nu_H}(0°) + [\overline{Nu_H}(90°) - \overline{Nu_H}(0°)] . \sin\omega$$

where,

$$\overline{Nu_H}(0°) = 0.684\ Ra_H^{0.121}$$

$$\overline{Nu_H}(90°) = 0.358\ Ra_H^{0.243}$$

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