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TECHNIQUES FOR REDUCING THERMAL CONDUCTION AND NATURAL CONVECTION HEAT LOSSES IN ANNULAR RECEIVER GEOMETRIES[1]

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Techniques for Reducing Thermal Conduction and Natural Convection Heat Losses in Annular Receiver Geometries

I. Introduction

An effective device for the collection of solar energy which has received widespread attention is the so called parabolic-cylindrical solar collector. In this design a circular receiver tube, with a suitable selective coating, is enclosed by a concentric glass envelope and situated along the focal line of a parabolic trough reflector. The heat transfer processes which occur in the annular space between the receiver tube and the glass envelope are important in determining the overall heat loss from the receiver tube. In typical high temperature receiver tube designs the rate of energy loss by combined thermal conduction and natural convection is of the same order of magnitude as that due to thermal radiation, and can amount to approximately 6% of the total rate at which energy is absorbed by the solar collector. The elimination of conduction and natural convection losses can significantly improve the performance of a large collector field.

In this paper, several techniques useful for the reduction of energy loss by thermal conduction and natural convection are considered. The receiver configuration chosen for study is typical of those used in the Solar Total Energy System at Sandia Laboratories. The receiver tube has a "black chrome" selective coating and is 2.54 cm in outside diameter. The inside diameter of the glass envelope is approximately 4.4 cm. Typical operating temperatures of the receiver tube and glass envelope are approximately 300°C and 100°C, respectively.

II. Conduction Heat Loss

In order to improve the overall efficiency of receiver designs, investigations into reducing the heat transfer through the annular space separating the glass and receiver tube have been undertaken. Of the three modes of heat transfer, the most significant heat loss savings can be accomplished by limiting conduction losses. Convection losses are negligible, so long as the annular space is properly sized. Radiation losses on the other hand, being primarily fixed by the receiver tube selective surface properties, are more difficult to reduce. Electroplating techiques have been optimized to provide high solar absorptivity ($\geq$ 0.95) with thermal emissivities ranging between 0.20 and 0.30 for temperatures around 590°K. In order to reduce radiative heat loss significantly, thermal emissivities below 0.15 are required. Such reductions do not appear possible without an accompanying reduction in solar absorptivity.

Attempts to limit heat transfer through the annular space will be discussed in the following sections. Techniques studied include (1) evacuation of the annulus gas, (2) oversizing the annular space, and (3) using gases other than air for the heat transfer medium.

A. Effect of Vacuum

A review of the literature on vacuum technology indicates that the thermal conductivity of a gas is a function of the mean free path of the gas molecules [1, 2, 3]. An expression relating the mean free path of a gas to the enclosure pressure and gas temperature is given by

$$\lambda = 2.331\ (10^{-20})\ T/P\delta^2, \qquad (1)$$

where T, P, and δ, the gas molecular diameter, are given in degrees Kelvin, millimeters of mercury and centimeters, respectively. For a given gas, the relative magnitudes of the molecular mean free path and the annulus gap determines whether the effective heat transfer coefficient for thermal conduction is (1) independent of annulus pressure, (2) a function of the annulus pressure, or (3) negligible. This is shown in Figure 1, which compares the conduction heat loss for a given annulus pressure with the conduction heat loss calculated for air at atmospheric pressure. Figure 1 was generated using dimensional data and expected operating temperatures for a Sandia Laboratories test receiver. The governing equation for the effective heat transfer coefficient for the annular space obtained from Reference 1, is given as

$$k_{ef} = \frac{k}{r_i Ln(r_o/r_i) + b\lambda(r_i/r_o + 1)}, \quad (2)$$

where all terms are defined in Table 1.

The reduction of receiver tube heat loss by evacuation has been analytically modeled and experimentally verified. Energy balances were made on the receiver tube and glass surfaces and were incorporated in a computer analysis which could (1) predict the resultant receiver assembly temperatures for known heat inputs or (2) predict the receiver assembly heat loss for a fixed receiver tube temperature. The analysis assumed steady state conditions and utilized correlations for the effects of pressure, wind, geometry, and temperature on the conduction and convection terms.

The Sandia Laboratories Phase IV-B receiver assembly design, schematically shown in Figure 2, was used for experimental verification. Heat input was provided by a Chromalox® resistance

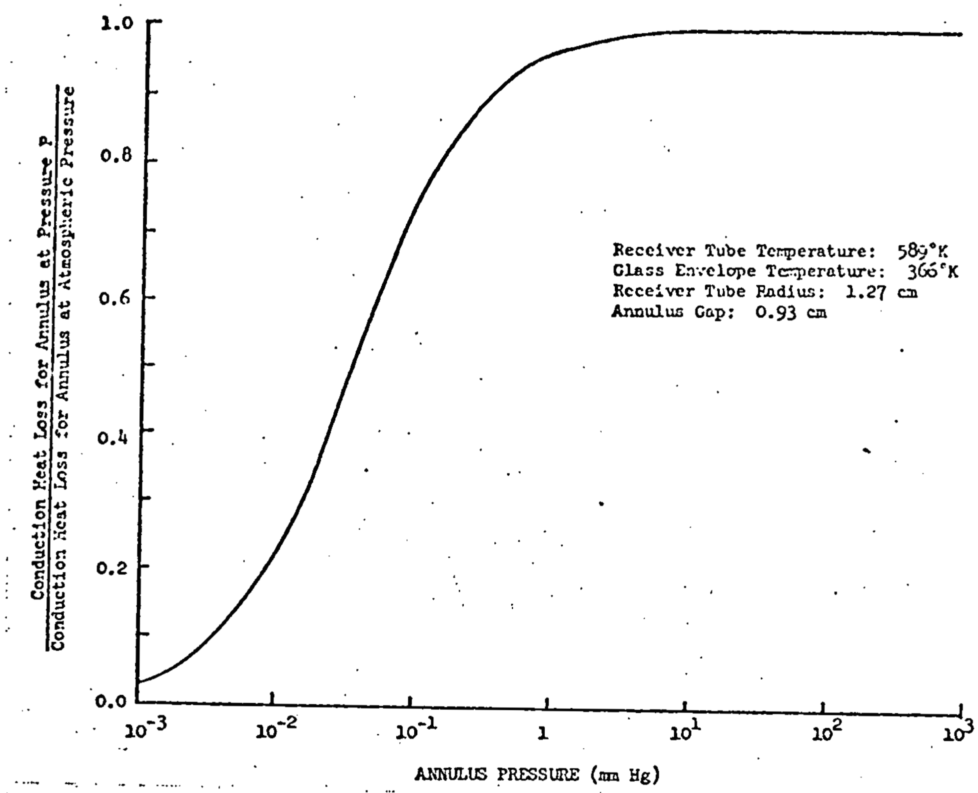


Figure 1. Effects of Pressure on Conduction Heat Loss Through an Annular Space

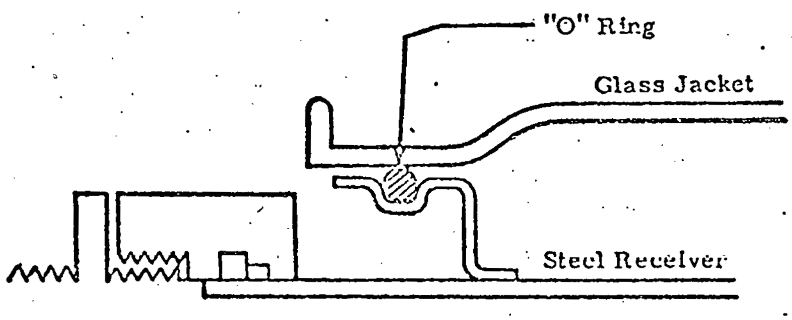


Figure 2. Sandia Laboratories Phase IV-B Proposed Receiver Design

heater element centered inside the receiver tube. Heat supplied to the receiver assembly was monitored using a standard resistor and voltmeter which were used to measure the electrical current. Input power was determined by measuring the voltage drop across the resistance heater and was maintained to within a ± 1.0 watt tolerance during testing. Receiver assembly temperatures were monitored using twenty-five chromel-alumel thermocouples. Locations of these thermocouples as well as dimensional data on the Phase IV-B test assembly are shown in Figure 3.

Annulus vacuums were monitored using a Pirani (thermocouple) Gauge for pressures below 1.0 mm Hg and a Wallace and Tiernan Pressure Gauge and Manometer for the higher pressures. All vacuums were maintained with a CEC Sampling Probe and Sargent-Welch single stage vacuum pump.

Early experimental work involved maintaining a fixed receiver assembly heat loss while varying the annulus pressure. Variations in receiver tube coating properties necessitated bracketing the experimental data with analytical results calculated for receiver tube emissivities of 0.2 and 0.3 at 589°K. The receiver tube temperature, as expected from the results of Figure 1, is seen to be independent of annulus vacuum for pressures above 1 mm Hg in Figure 4. This is also shown in Figure 5, which provides heat loss data as a function of annulus pressure. Significant heat loss reductions of nearly 50% are seen to result if vacuums below 0.01 mm Hg can be maintained.

Based on the close agreement of results, the effect of wind on heat loss has also been analytically studied and is shown in Figure 6. Increasing the wind velocity increases the receiver heat loss in the annular space at atmospheric pressure by lowering the glass envelope temperature. By annulus evacuation, however, wind effects can be mimimized

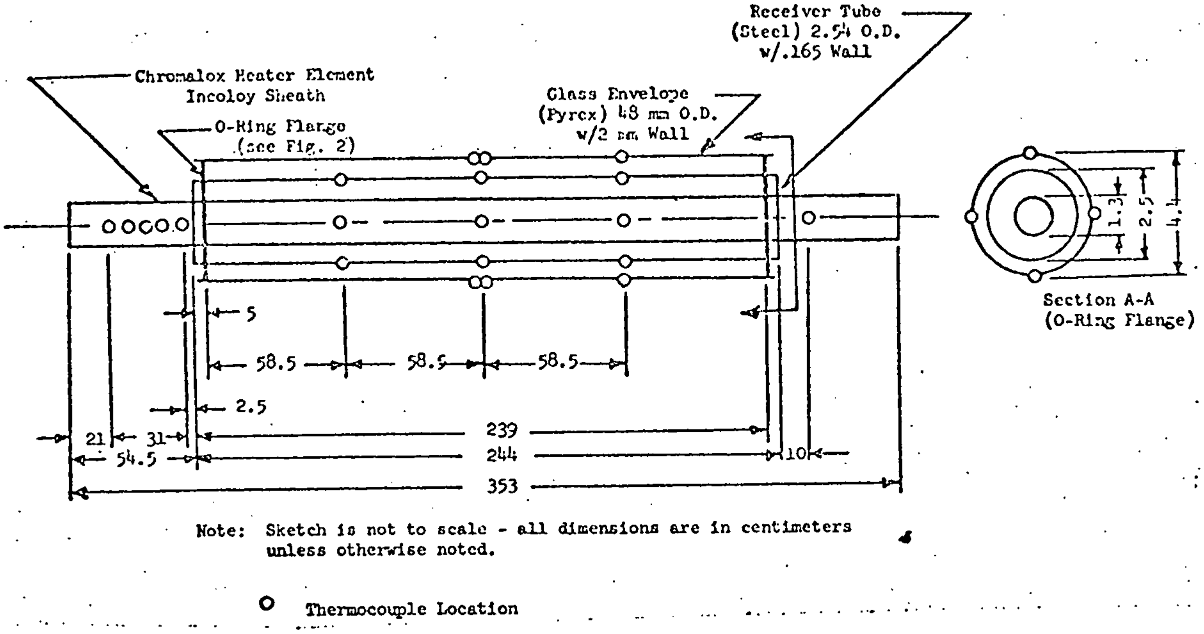


Figure 3. Schematic of Thermocouple Locations on the Phase IV-B Receiver

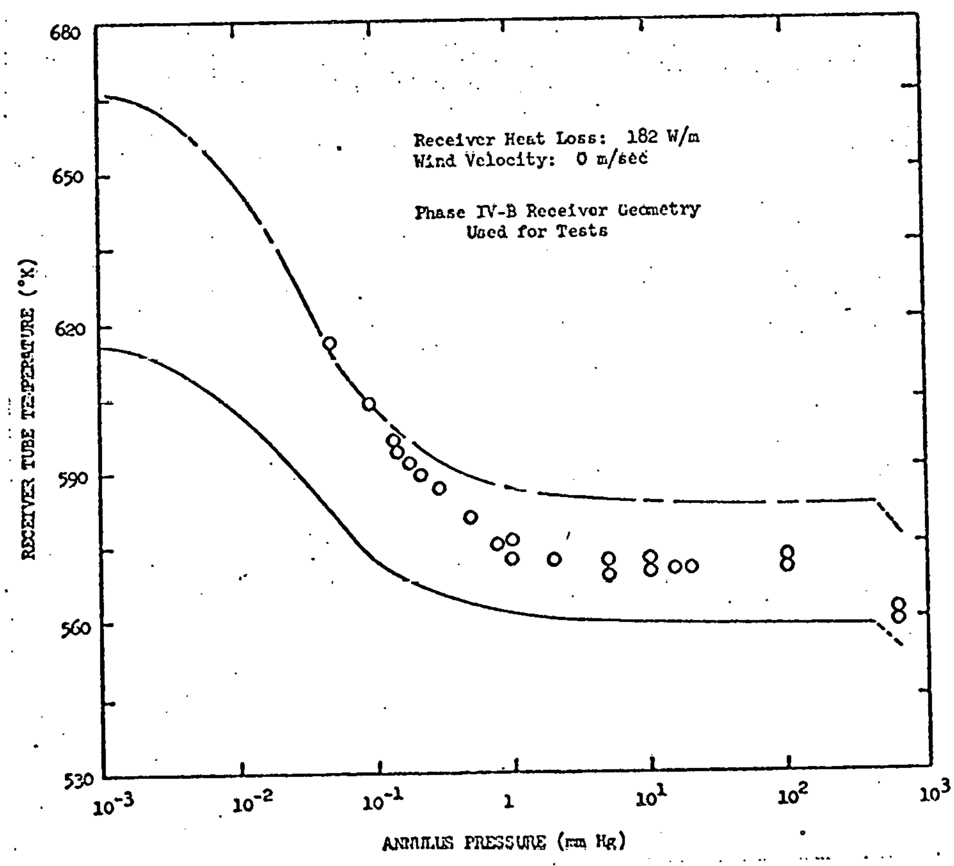


Figure 4. Receiver Tube Temperature as a Function of Annulus Pressure

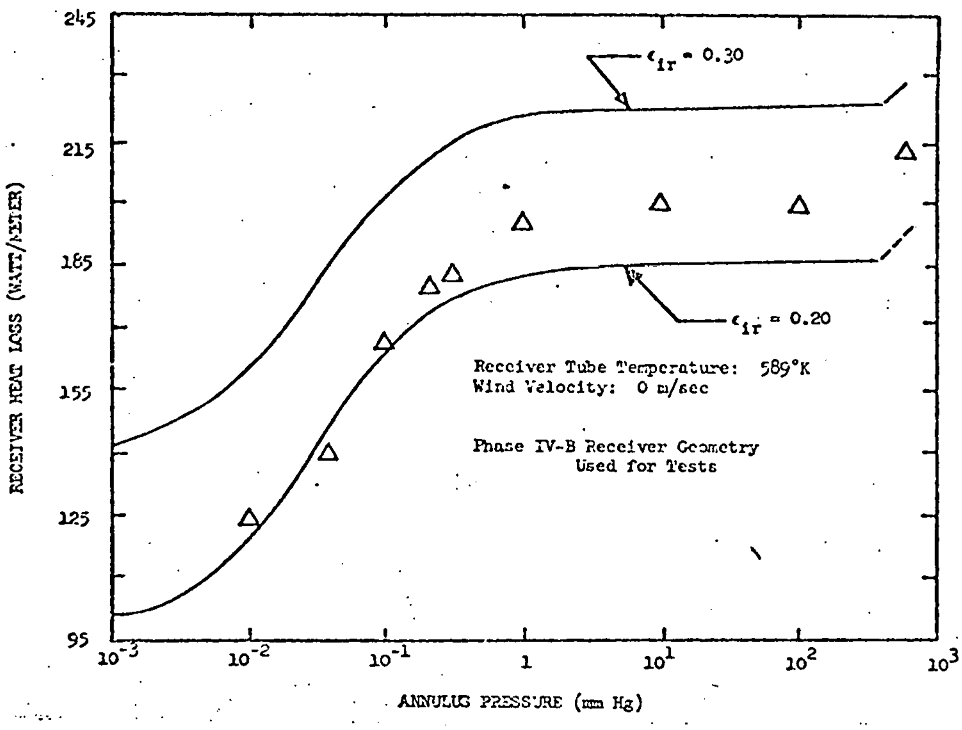


Figure 5. Effects of Pressure on Receiver Tube Heat Loss

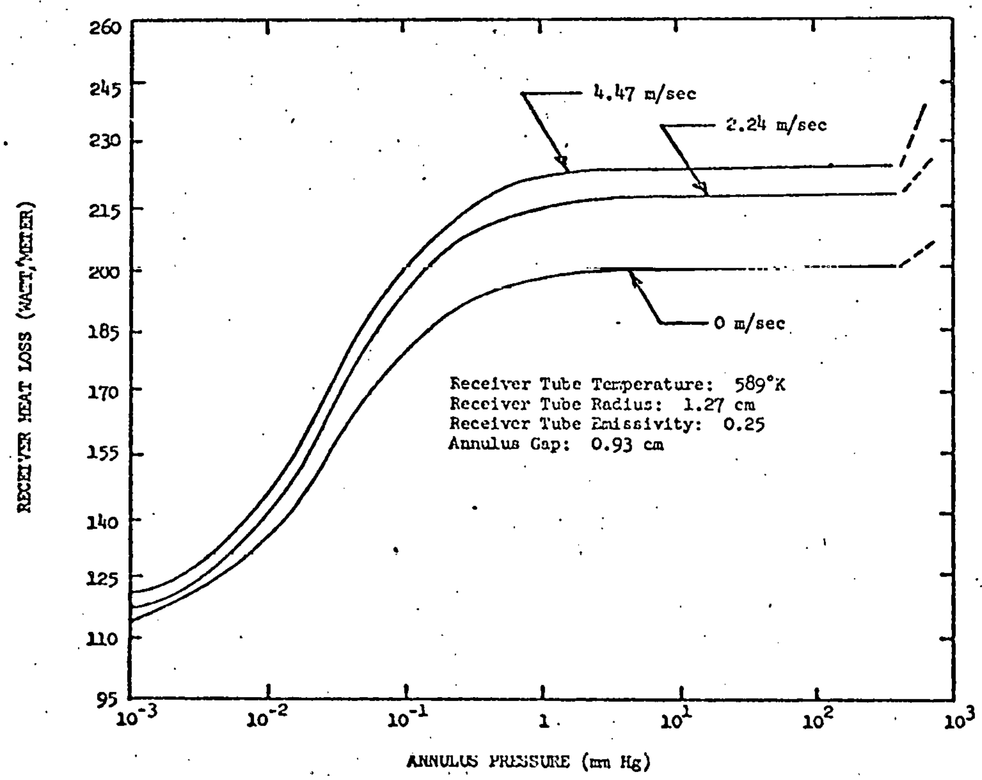


Figure 6. Heat Loss Reductions Resulting from Evacuation for Different Wind Conditions

since the dominating conduction losses are significantly reduced. The radiative loss, on the other hand, is such that changes in the glass temperature do not greatly increase the energy loss for a fixed receiver tube temperature.

B. Effect of Annulus Gap Sizing

Optimum sizing of the annular space for operation at atmospheric pressure requires that the energy transferred across the gap be by thermal conduction and radiation heat transfer. Incorrect sizing could result in enhanced convective energy transport which would increase the net heat loss. Work to be discussed in a later section concerning natural convection has indicated that the effects of natural convection will be suppressed as long as the Rayleigh number is maintained below a value of 1000.

Experimental and analytical results shown in Figures 4 and 6 indicate that the Phase IV-B receiver sizing has not been optimized, since a heat loss reduction occurs when the annulus pressure is reduced below atmospheric pressure. By decreasing the pressure, the Rayleigh number is reduced below 1000 through lowering the annulus gas density. Based on these trends, the computer model was utilized to vary the gap spacing for a fixed receiver tube radius and temperature. From data presented in Reference 4, the effective conduction coefficient used for a particular spacing was generated using the following correlation.

$$K_{ef} = K \text{ for } N_{Ra} < 1000,$$

$$K_{ef} = 0.1558K\, N_{Ra}^{.2667} \text{ for } N_{Ra} > 1000 \qquad (3)$$

where K_{ef} is the effective thermal conductivity, K is the actual thermal conductivity evaluated at the mean temperature,

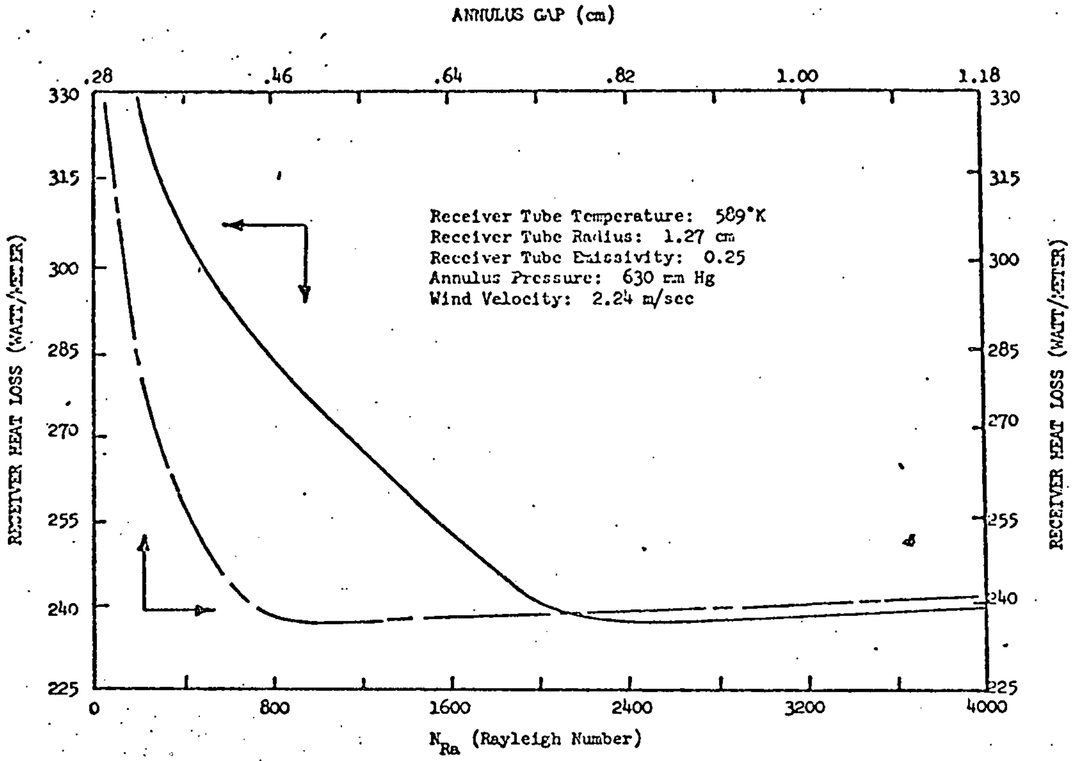


Figure 7. Annular Space Sizing Data for Atmospheric Pressure

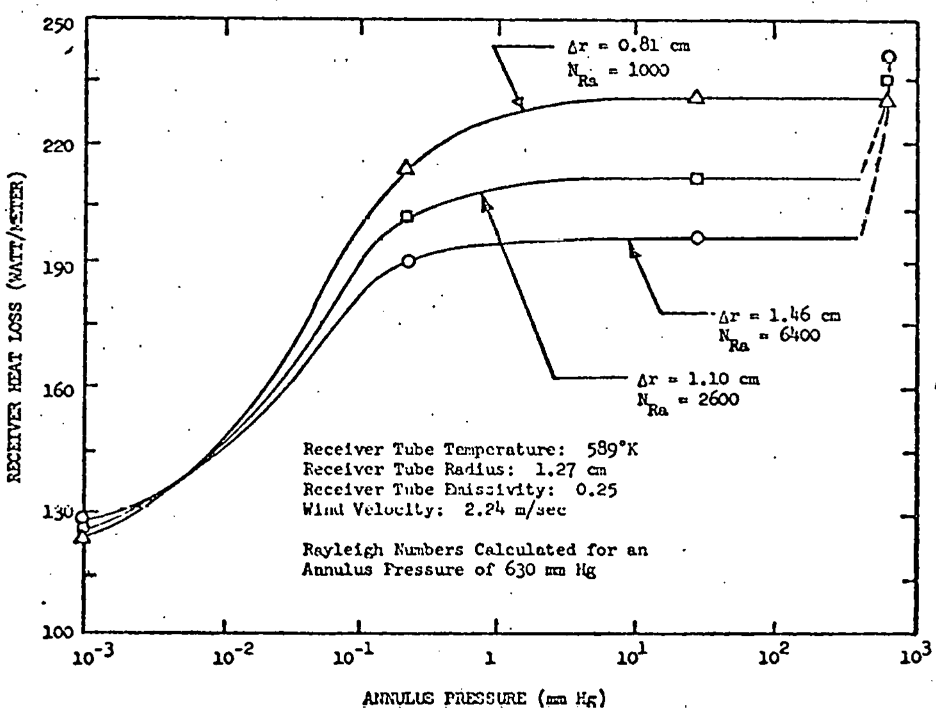


Figure 8. Receiver Annulus Oversizing for Reducing Heat Loss

and N_{Ra} is the Rayleigh number defined by

$$N_{Ra} = \rho g \beta \ell^3 \Delta T / \mu \alpha \ , \qquad (4)$$

where all symbols are defined in Table 1.

Figure 7 shows that the heat loss is minimized for an annular space of 0.81 cm. (For the same test conditions, the Phase IV-B gap of 0.93 cm results in a Rayleigh number of 1550.) Despite the discrepancy in sizing, trends from Figure 7 point out that oversizing the gap results in minimal increased heat loss compared to that obtained from reducing the gap size to maintain the Rayleigh number below 1000.

Further investigation into varying the annular space to reduce heat loss resulted in the data summarized in Figure 8. By maintaining annulus pressures below 200-300 mm Hg and oversizing the gap, heat loss savings of between 15 and 30 W/m may be obtained over that lost by a receiver design sized to eliminate convection heat transfer at atmospheric pressure. Since the reduction of heat loss by 30 W/m for the "correctly" sized annular space necessitates using vacuums below 0.1 mm Hg, it can be noted that oversizing may allow for energy loss savings without requiring hard vacuum systems.

C. Effect of Gases other than Air

Utilization of gases other than air in the receiver annulus should reduce the conduction heat loss so long as (1) the gas thermal conductivity is less than that of air and (2) the effective Rayleigh number is similar to that of air for a given geometry. This is shown in Figure 9, which compares the use of argon, air and carbon dioxide in the annulus. Although carbon dioxide has a lower thermal conductivity than air (0.031 W/m - °K compared to 0.048 W/m - °K at 477°K) its other physical properties necessitate small gap spacings to minimize natural convection heat loss. The insulating effect of the lower thermal conductivity is thus lost because the conduction gap must be reduced.

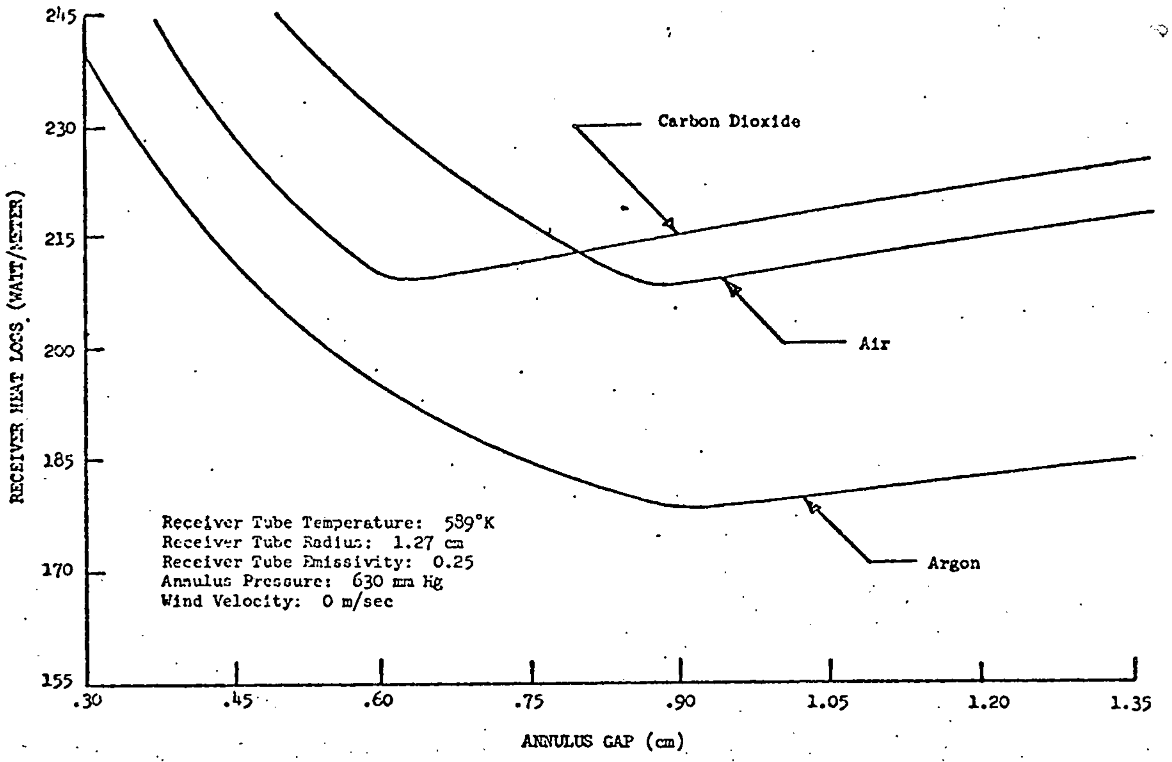


Figure 9. Annulus Sizing Data for Alternate Gases

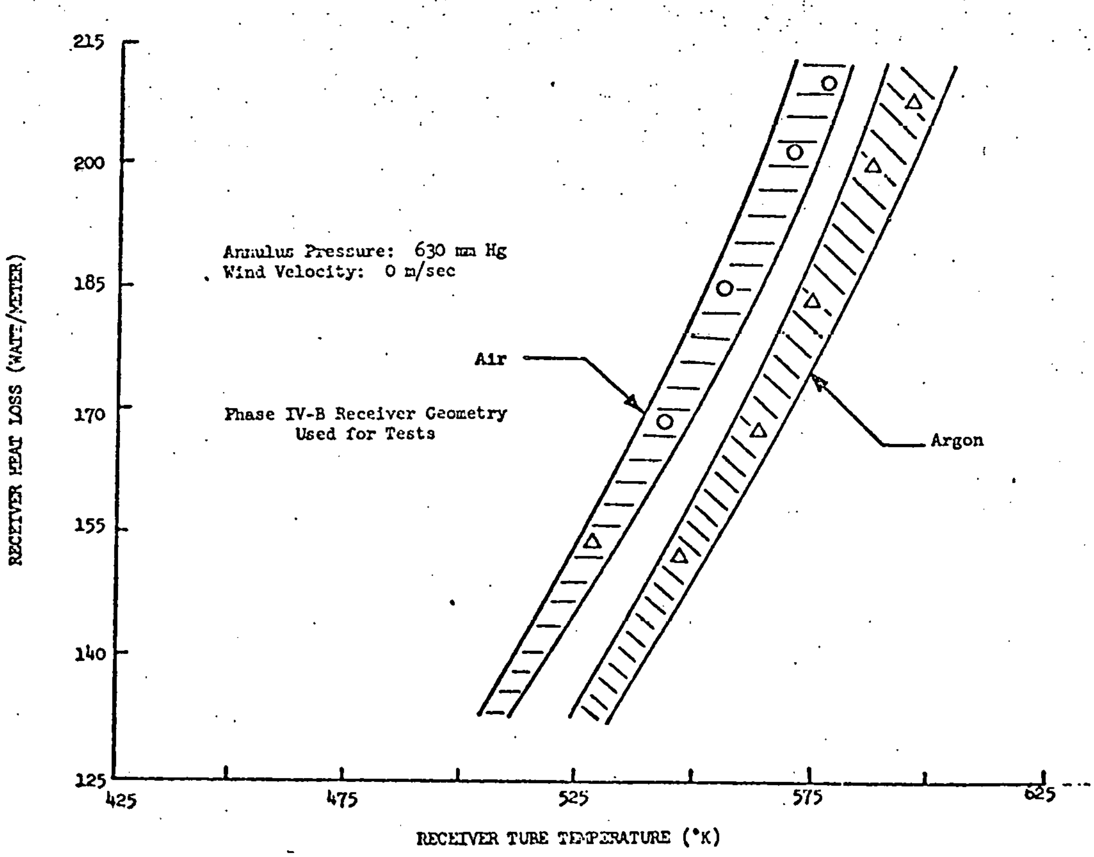


Figure 10. Heat Loss Versus Receiver Tube Temperature for Argon and Air

Experimental verification of the savings associated with using gases other than air has been accomplished using argon in the Phase IV-B test receiver. The choice of this gas was predicated on its abundance and also upon the fact that the optimal annulus gap for argon for limiting convective heat loss was close to the experimental annulus sizing. Data taken on both air and argon are shown in Figure 10, which compares receiver tube temperature with receiver heat loss. For a given receiver tube operating temperature, heat loss savings of 25 to 30 W/m may be realized by replacing air with argon. Such savings are comparable to oversizing the annulus and maintaining a partial vacuum of 100-300 mm Hg. The advantage gained using argon gas to reduce the heat loss over the aforementioned technique is that, due to the similarity in sizing the gap for air or argon, a complete loss of argon and replacement with air will still minimize the heat loss. As can be seen from Figure 8, an increase in heat loss will result in the event of loss of vacuum for the oversized geometry.

In summary, reduction of the conduction heat loss in the annular space can be accomplished through (1) evacuation, (2) oversizing the annular space while maintaining the Rayleigh number below 1000 (partial vacuum) and (3) use of gases with low thermal conductivities. Figure 11 is provided to show the relative heat loss savings for each technique. Additional alternatives, such as charging the annular space with argon and evacuating and/or oversizing the gap are shown. Of the options, it appears that evacuation can best eliminate conduction heat loss. Problems, however, concerning the relative costs of each heat loss reduction scheme must be considered in selecting the best option for the receiver assembly.

III. Natural Convection in an Annulus

In the previous section, the classical experimental result for natural convection heat transfer between horizontal, concentric,

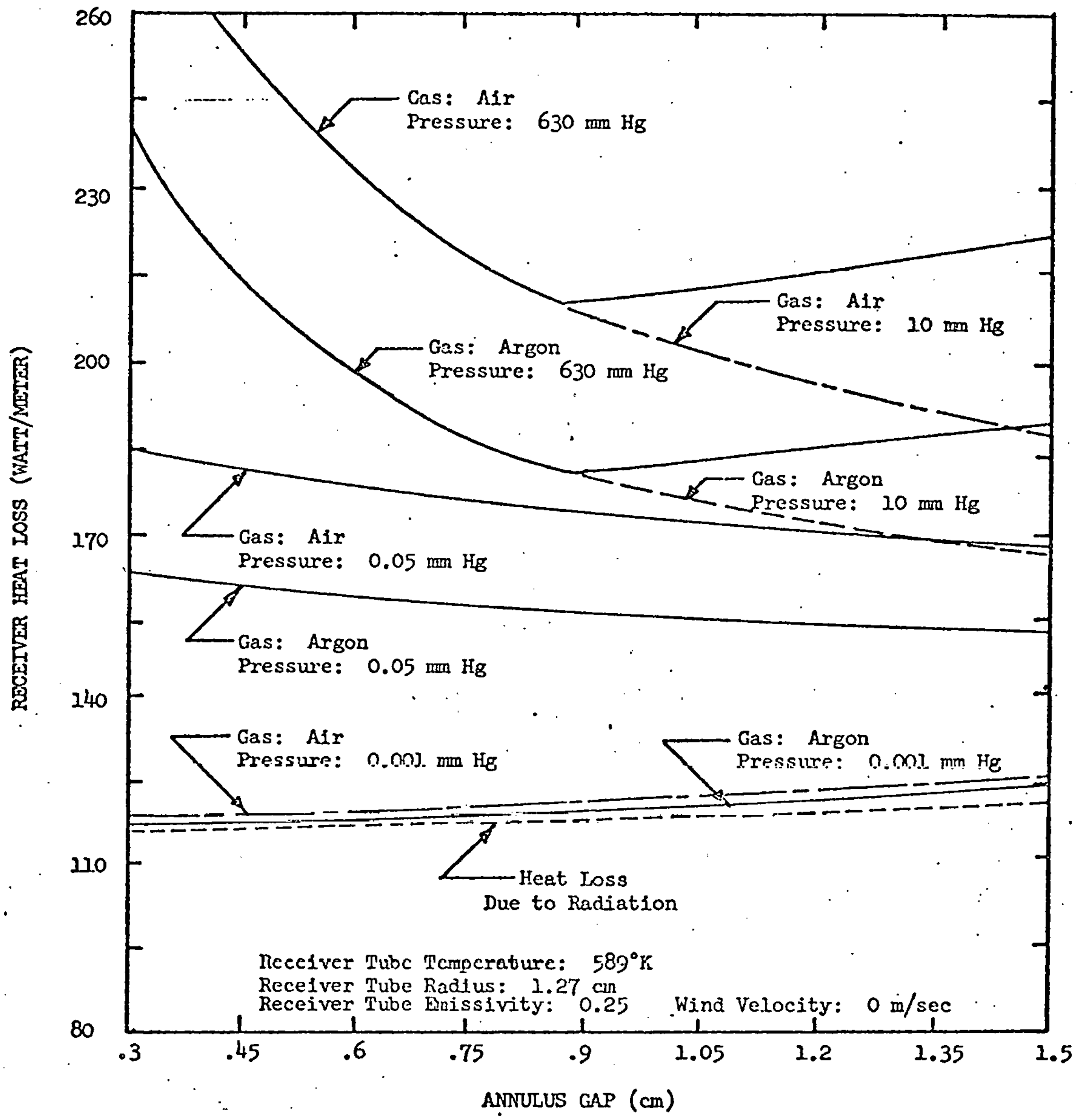


Figure 11. Summary Data on Heat Loss Reduction Techniques

circular cylinders, as originally presented by Kraussold [4], was used in the analysis of the heat transfer process between the receiver tube and glass envelope. Recently, Kuehn and Goldstein [5], have compiled a comprehensive review of the available experimental results for natural convection heat transfer between circular cylinders and proposed correlating equations using a conduction boundary-layer model. It is evident from this review that almost all the results available to date are, strictly speaking, valid only for horizontal, concentric, circular cylinders with uniform temperatures. Since the temperature distribution on a typical receiver tube is not uniform, it is important to determine the effect of this variation on the overall heat transfer process. Furthermore, it is also of importance to assess the effect of eccentricity since it is difficult to maintain precise alignment between the receiver tube and envelope due to deflections caused by gravity and differential thermal expansion.

Because of the scarcity of results for nonuniform temperature distributions and eccentric cylinders, a numerical analysis of the natural convection heat transfer process was performed in order to provide a better understanding of the effects of non-ideal situations. The results of this study are described in the remainder of this section.

A. Numerical Method

The partial differential equations which describe the heat transfer process are represented discretely through use of the Galerkin form of the finite element method; a technique which has been described in detail by several authors, e.g. Zienkiewicz [6], and will not be elaborated on here. The resulting methodology has been incorporated, by Gartling [7], into a user-oriented, finite element, computer program called NACHOS. This program makes use of an isoparametric mesh generator to allow complex boundaries to be modeled easily and accurately. The element library consists of an eight-node isoparametric quadrilateral and a six-node isoparametric triangle. Within each element, the velocity and temperature are approximated quadratically and the pressure approximated linearly.

The program is quite versatile and can be used for the analysis of both free and forced convection heat transfer as well as for isothermal flows. Both steady state and transient analyses can be performed. The flows must, however, be incompressible. Temperature dependent fluid properties can be easily incorporated into the analysis at the option of the user. This latter option was used in all the calculations discussed in the remainder of this section.

B. Heat Transfer Between Concentric Cylinders

In order to demonstrate that the results of the numerical analysis are compatible with existing experimental results, an initial series of calculations was performed for horizontal, concentric, circular cylinders with uniform temperatures. The temperatures of the inner and outer cylinders were held constant at 583°K and 333°K, respectively. These values were selected to correspond to the average operating conditions expected for the existing Sandia Laboratories Solar Total Energy collector field. The radius of the inner cylinder was held constant at 1.27 cm and the outer radius allowed to vary from 2.24 cm to 4.32 cm, producing radius ratios in the range 1.4 to 3.4. Rayleigh numbers ranging from approximately 300 to 97,000 were thus obtained with the Rayleigh number defined in the usual way as given by (4).

Following the accepted standard, results of the analysis are presented in Figure 12 as a plot of the heat loss ratio versus the Rayleigh number , where the heat loss ratio is defined to be the ratio of the energy loss per unit length due to natural convection to that due to thermal

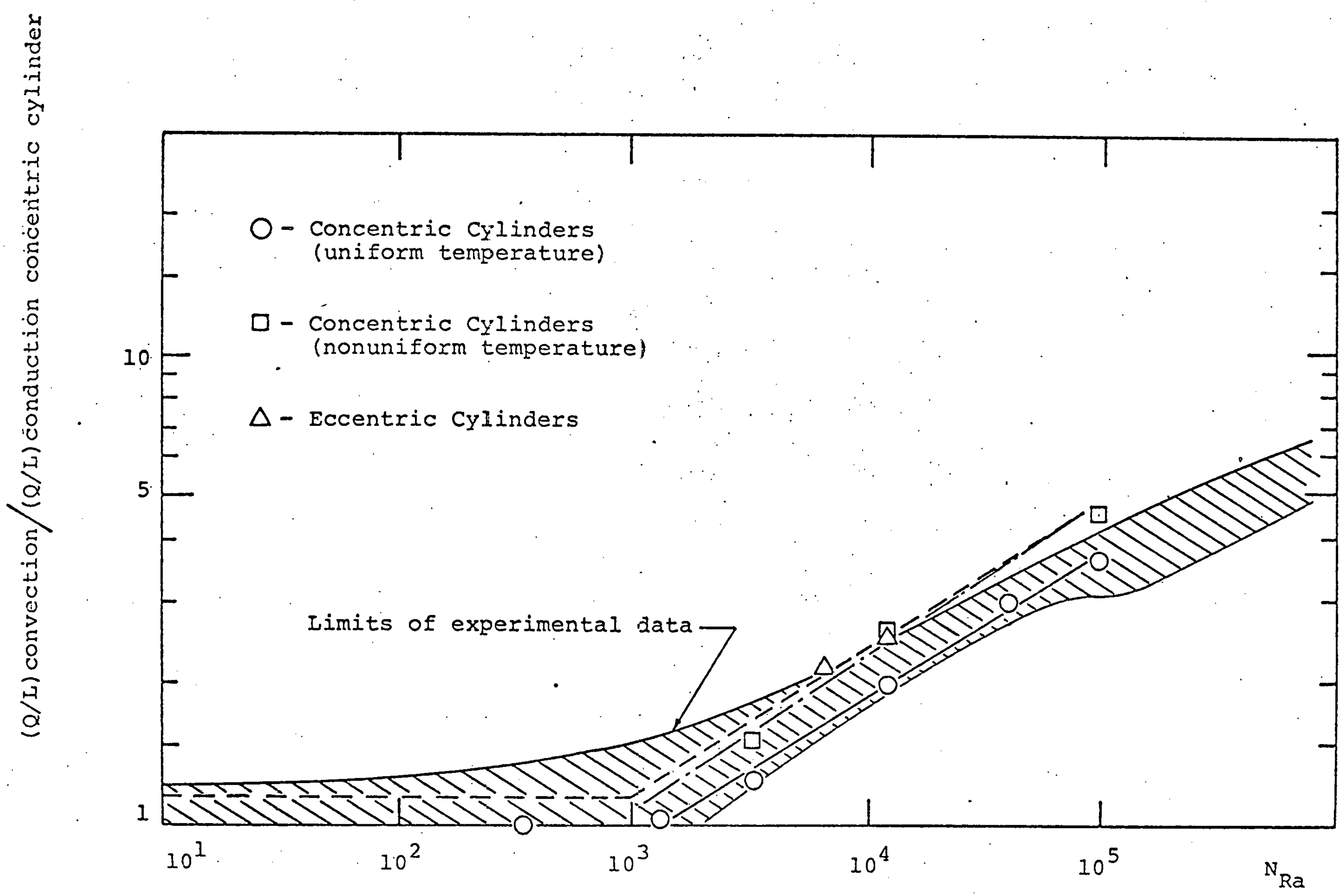


Figure 12. Natural Convection Heat Transfer in an Annulus - Comparison of Computed and Experimental Results.

conduction acting alone. In Figure 12, the cross-hatched area indicates the region occupied by the experimental data as compiled by Kuehn and Goldstein [5]. Typical streamlines and isotherms as computed for a Rayleigh number of approximately 12,000 are shown in Figure 13.

It is evident that, in general, there is rather good agreement between the computed and experimentally determined values of natural convection heat transfer rates. As expected, the heat loss is seen to initially deviate from that due to thermal conduction at a Rayleigh number of approximately 1000. The numerical method employed is apparently of sufficient accuracy to warrant its application to the study of the non-ideal situations described previously which have not, as yet, been extensively investigated.

C. Nonuniform Temperature Distribution

Two situations involving nonuniform temperature distributions on concentric, circular cylinders were selected for study. In both cases the temperature of the outer cylinder was held constant at 333°K and the temperature of the inner cylinder allowed to vary according to

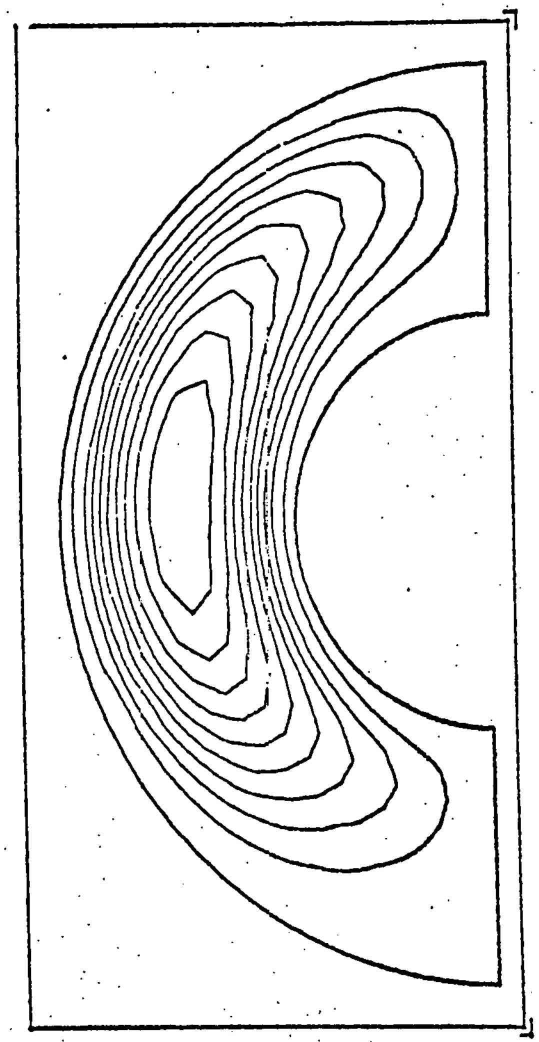
$$T = T_m \pm T' \cdot \cos\Theta \ , \tag{5}$$

where T_m is the mean temperature (533°K), T' the perturbation to the mean, and Θ is the angle measured from the bottom of the cylinder. A perturbation of 139°K was used in all calcuations, yielding a total variation of 278°K around the inner cylinder. The two cases studied are distinguished by the choice of sign in Equation (5); the positive sign corresponding to the occurrence of greatest temperature on the lower surface of the inner cylinder and vice versa. A variation of 278°K is far in excess of that encountered in conventional receiver tubes but, as will subsequently be shown, even this amount of nonuniformity has only a small effect on the natural convection process. In all calculations, the radius of the inner cylinder was held constant at 1.27 cm and the radius of the outer cylinder varied in order to vary the Rayleigh number. It should be noted that the average temperature upon which the Rayleigh number is based has the same value (458°K) as that associated with the uniform temperature case.

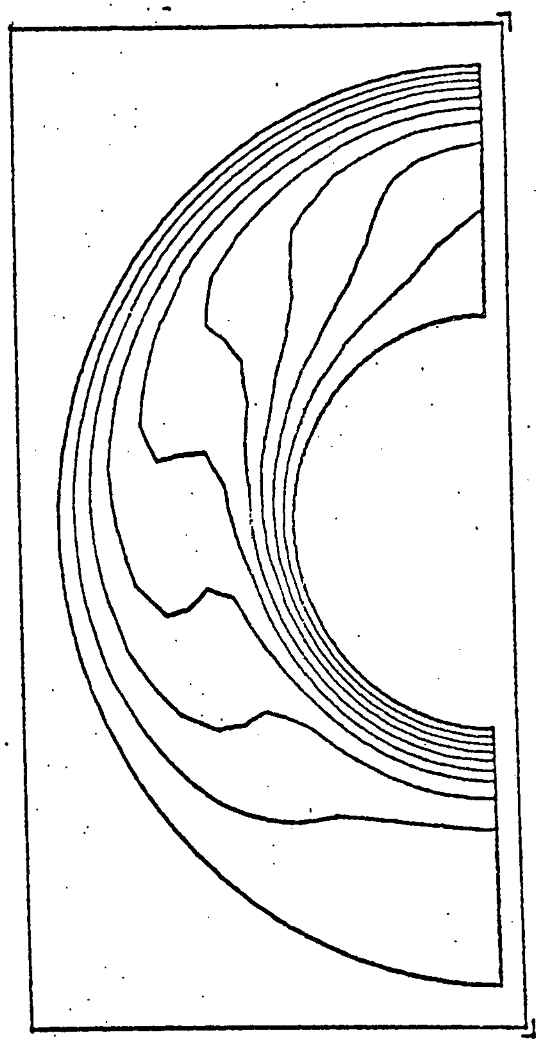
When the highest temperature occurred on the lower surface of the inner cylinder, the calculated results were virtually indistinguishable from those obtained with uniform wall temperatures as plotted in Figure 12. Consequently, the results for this nonuniform case are not plotted. It should, however, be noted that the higher temperature on the lower surface produced flow patterns which, in some instances, differed significantly from that illustrated in Figure 13. For Rayleigh numbers of approximately 12,000 and greater, a two cell flow pattern with one cell above the other and each spanning the entire gap between cylinders, was obtained. The altered flow pattern, however, produced no appreciable change in the overall heat transfer characteristics.

When the higher temperature occurred on the upper surface of the inner cylinder, the natural convection heat transfer rate was enchanced over that obtained with uniform temperatures, as shown in Figure 12. The line plotted in Figure 12 is tentative since only three points have been determined. Departure from the thermal conduction curve again occurs at a Rayleigh number of approximately 1000. Calculated streamlines and isotherms for a Rayleigh number of approximately 12,000 are presented in Figure 14.

From the results described in this section, it is evident that highly nonuniform temperature distributions are required in order to appreciably affect the natural convection process between concentric cylinders. Since the nonuniformity of

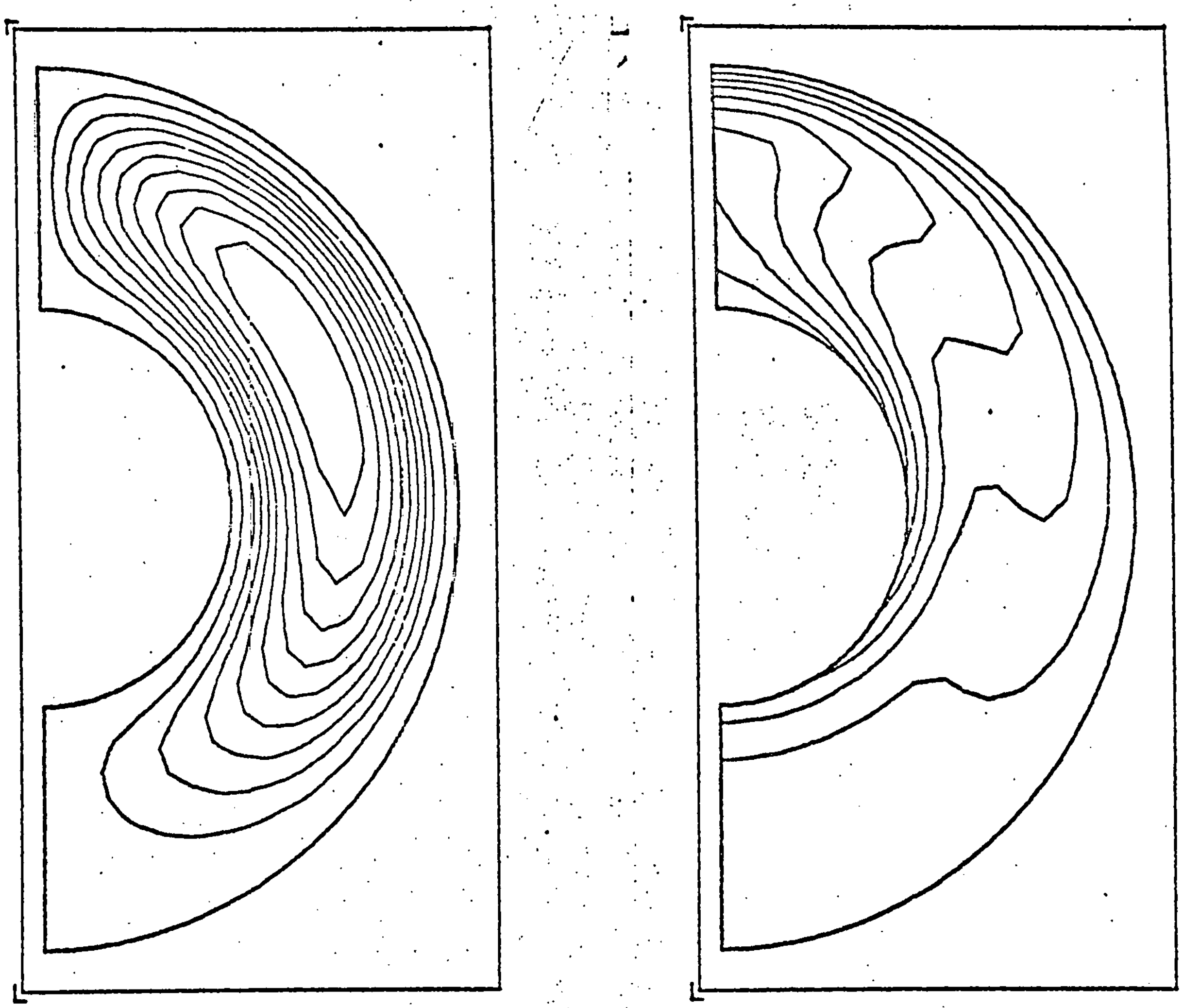


Streamlines



Isotherms

Figure 13. Streamlines and Isotherms for Natural Convection Between Concentric, Circular Cylinders - Uniform Temperature ($T_i = 583°K$, $T_o = 333°K$, $N_{Ra} = 12,142$)



Streamlines   Isotherms

Figure 14. Streamlines and Isotherms for Natural Convection Between Concentric, Circular Cylinders - Nonuniform Temperature ($444°K < T_i < 722°K$, $T_o = 333°K$, $N_{Ra} = 12,142$)

temperature in typical receiver geometries is substantially less than that used in the current study, it is anticipated that the effects of nonuniform temperature can be neglected in most instances.

D. Eccentric Cylinders

The geometry chosen for the study of eccentric cylinders consisted of inner and outer cylinders of radii 1.27 cm and 2.79 cm, with the inner cylinder displaced downward a distance equal to one-half the gap width yielding an eccentricity of 0.76 cm. Each cylinder was held at a uniform temperature. Although other geometries could be analyzed without difficulty, the selection of the case chosen for study was influenced by certain practical considerations. First, a downward displacement of the inner cylinder enhances the convection process more than an upward displacement. Secondly, an eccentricity of one-half the gap width is greater than that encountered in practical receiver designs. Finally, it was convenient to maintain symmetry about a vertical plane in order to simplify the numerical computations.

For proper interpretation of subsequent results, it should be recalled that the heat loss by thermal conduction between eccentric, circular cylinders with uniform wall temperatures is given by

$$Q/L = 2\pi K\Delta T/(\ln x + \sqrt{x^2-1}) \ , \qquad (6)$$

where

$$x = (r_o^2 + r_i^2 - \varepsilon^2)/2r_o ri \ , \qquad (7)$$

and all other terms are defined in Table 1. Upon comparison of (6) with the expression for thermal conduction between concentric cylinders, it can be concluded that the effective gap width for eccentric, circular cylinders is

$$\ell = \frac{r_o^2 + r_i^2 - \varepsilon^2 + \sqrt{\left[r_o^2 - (r_i+\varepsilon)^2\right]\left[r_o^2 - (r_i-\varepsilon)^2\right]}}{2r_i} - r_o \quad . \qquad (8)$$

This effective gap width is used as the length parameter in the definition of the Rayleigh number for eccentric cylinders. For the case under consideration, the effective gap width is 1.25 cm as compared with 1.52 cm for a concentric arrangement. Thus, one effect of eccentricity is to reduce the magnitude of the Rayleigh number.

The results obtained for two different Rayleigh numbers are plotted in Figure 12. For the lowest Rayleigh number, the temperatures of the cylinders were held constant at the values previously used in the analysis of concentric cylinders discussed in Part B of this section (583°K, 333°K). Streamlines and isotherms for this case are presented in Figure 15. The effect of the reduced gap size is to reduce the Rayleigh number from 12,142 to 6,694. In the other instance, the temperature of the inner cylinder was increased and the temperature of the outer cylinder decreased, by equal amounts, in order to maintain the average temperature constant at 458°K, and Rayleigh number constant at 12,142. The former case is representative of the effect of eccentricity with constant cylinder wall temperature while the latter case is representative of the effect of eccentricity at constant Rayleigh number. The case of constant wall temperature is, of course, more representative of situations likely to be encountered in practical solar receiver designs.

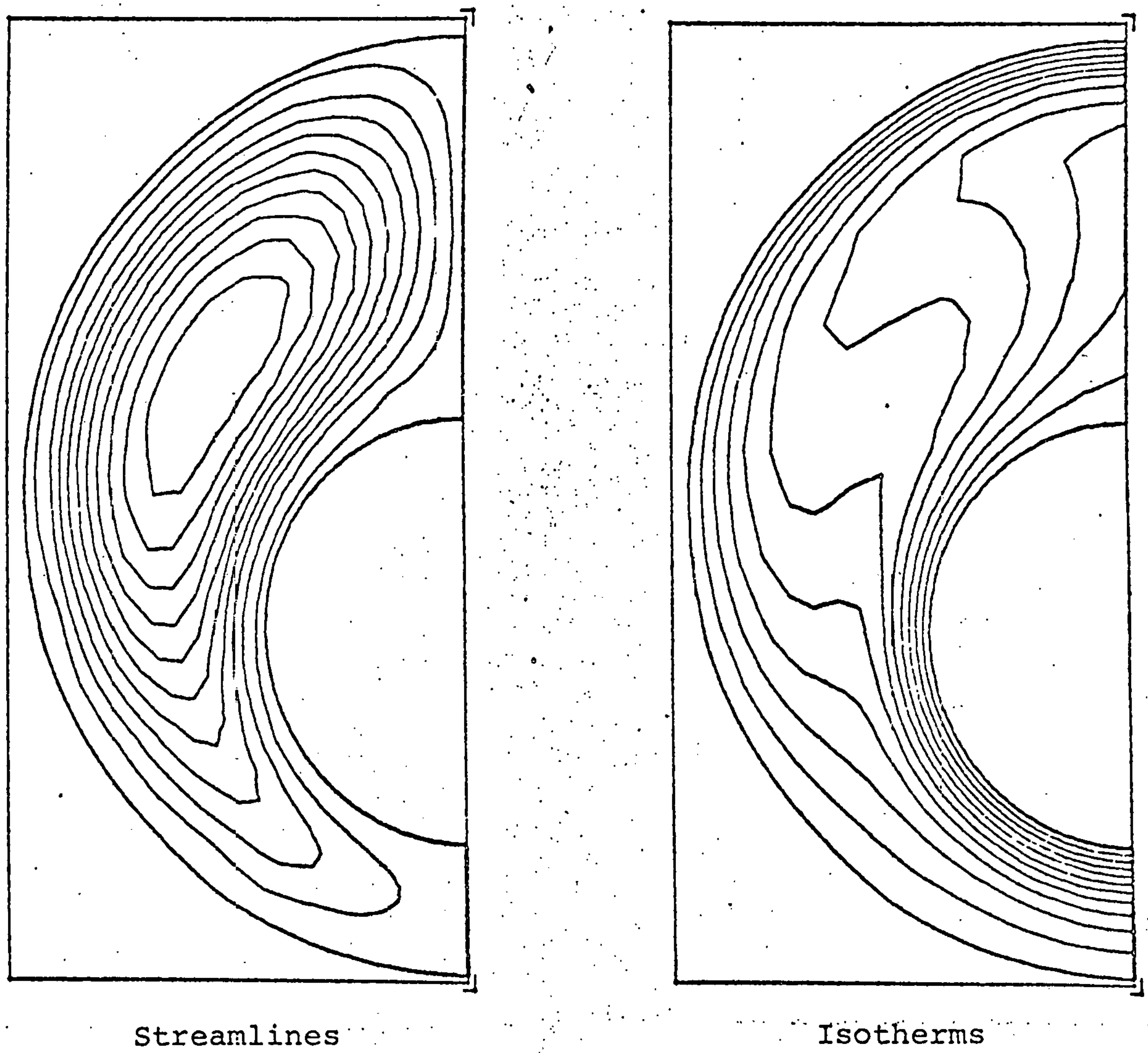


Figure 15. Streamlines and Isotherms for Natural Convection Between Eccentric, Circular Cylinders - Uniform Temperature (T_i = 583°K, T_o = 333°K, N_{Ra} = 6,694)

It should be noted that the ordinate in Figure 12 is the ration of heat transferred by natural convection to that transferred by thermal conduction for concentric cylinders. Hence, the conduction limit for the eccentric case studied is not unity but rather has a value of 1.15 due to the decrease in effective gap width. The dashed line in Figure 12 is, of course, only a tentative estimate of the general trend.

The single most important observation to be made from this analysis is that, for constant cylinder wall temperatures, a rather large increase in eccentricity causes only a slight increase in natural convection heat transfer. Although an eccentric arrangement results in an increase in heat transfer by thermal conduction, the reduced gap size causes a reduction in the Rayleigh number which tends to suppress natural convection. Hence, the slight increase in heat transfer is not unreasonable. The tentative trend indicated in Figure 12 still predicts departure from conduction-like behavior at a Rayleigh number of approximately 1000.

IV. Summary

It is well known that natural convection effects have a negligible influence on heat transfer in a uniformly heated annulus when the Rayleigh number, based on gap size, is less than approximately 1000. However, below this value, thermal conduction continues to be an important mode of heat transfer. The magnitude of this effect can be reduced if the pressure in the annulus is sufficiently reduced. Theoretical calculations showing the effects of a reduced annulus pressure were performed for a typical cylindrical solar collector and compared with measurements made on an electrically heated test section. The results demonstrate that it is possible to calculate, with reasonable accuracy, the heat loss and associated temperatures of the receiver tube and glass envelope as a function of the pressure in the annulus.

If it is not possible to maintain the pressure in the annulus low enough to effectively reduce conduction losses, then the energy loss, for a uniformly heated annulus, can be minimized by sizing the annulus so as to maintain a Rayleigh number near 1000 over the expected range of operating conditions. In order to assess the relative importance of nonuniform temperature distribution and eccentricity, a numerical study, using a state of the art finite element computer code, was performed for the natural convection process in an annulus. For uniform temperature, the numerical study produced flow patterns and associated heat transfer rates in good agreement with published experimental results. The numerical results also demonstrated that highly nonuniform temperature distributions or large eccentricities are required in order to appreciably infuence the natural convection process.

An interesting observation made during the course of the study was that a reduction in energy loss by thermal conduction of from 10 to 20% can be effected by maintaining a moderately low pressure of, say, 50 mm Hg in the annulus and adjusting the gap size to correspond to the maximum allowable Rayleigh number. This pressure is approximately two orders of magnitude greater than the pressure required to cause a significant reduction in the thermal conductivity of the air in the annulus. Similar reductions in energy loss by thermal conduction were shown to be feasible through the use of gases other than air in the annulus.

TABLE 1. SYMBOLS

b – Parameter in Eqa. (2)

g – Acceleration of gravity

K – Thermal conductivity

K_{er} – Effective thermal conductivity

L – Length of receiver tube

ℓ – Gap size

N_{Ra} – Rayleigh number

P – Pressure

Q – Heat loss

r_i – Outer radius of receiver tube

r_o – Inner radius of glass envelope

T – Temperature

T' – Temperature perturbation

Tm – Mean Temperature

Δr – $r_o - r_i$

ΔT – Temperature difference between cylinders

<u>Greek</u>

α – Thermal diffusivity

β – Coefficient of volumetric thermal expansion

δ – Molecular diameter

ε – Eccentricity

ε_{ir} – Thermal emissivity

θ – Angular position

λ – Mean free path

μ – Dynamic viscosity

ρ – Density

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