ENHANCEMENT OF NATURAL CONVECTION HEAT TRANSFER IN ANNULAR ENCLOSURES

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Abstract Natural convection heat transfer was investigated experimentally in a concentric–cylinders annulus. To enhance the rate of heat transfer, pin fins were installed on the surface of the inner cylinder. Two fin arrangements were employed, namely, staggered and in–line arrangements. The length of the fins varied from 2 to 8 cm and the Rayleigh number Ra changed in the range from 5×10^5 to 5×10^6 The heat transfer medium was air at atmospheric pressure. So, the results reported here are for the Prandtl number Pr $\cong 0.7$. It was found that, while the presence of fins usually decreases the heat transfer coefficient, the resulting increased surface area due to fins is so significant that the fins of both arrangements increase the rate of heat transfer, with the staggered one being more efficient. The normalized rate of heat transfer Q_i/Q_1 was found to be independent of Rayleigh number, and for each fin arrangement, correlations were obtained to express Q_i/Q_1 as a function of fins length. In the range of parameters considered in this work, the overall rate of heat transfer increased as much as 300%, corresponding to the longest fin of the staggered arrangement.

چکیده انتقال حرارت جابجائی آزاد در یک محفظه بسته متشکل از دو استوانه هم محور بصورت تجربی مورد بررسی قرار گرفته است. به منظور افزایش نرخ انتقال حرارت، تعدادی پره سوزنی روی سطح استوانه داخلی نصب گردید. در این پژوهش از دو آرایش یک در میان و خطی پره استفاده شده است. طول پره ها بین ۲ تا ۸ سانتیمتر و عدد ریلی در محدوده ۱۰۵×۵ تا ۱۰۶×۵ متغیر است. از آنجا که سیال مورد آزمایش هوا در فشار اتمسفر بوده است. نتایج گزارش شده، برای عدد پرانتل ۰/۷ است. نتایج بدست آمده نشان میدهد که بطور کلی وجود پره‌ها ضریب انتقال حرارت را کاهش میدهد، و افزایش سطح حاصل از پره‌ها بقدری حائز اهمیت است که در هر دو آرایش نرخ انتقال حرارت افزایش می‌یابد. آرایش یک در میان از کارآئی بهتری برخوردار است. همچنین معلوم شد که نرخ انتقال حرارت بصورت Q_i/Q_1 مستقل از عدد ریلی بوده و برای هر یک از دو آرایش پره‌ها رابطه‌ای برای بیان Q_i/Q_1 برحسب طول پره بدست آمده است در محدوده تغییرات پارامترهای این بررسی، افزایش نرخ کلی انتقال حرارت تا ۳۰۰ درصد مشاهده شده، که به بلندترین پره آرایش یک در میان مربوط بوده است.

INTRODUCTION

The phenomenon of natural convection in enclosures has been the subject of study for many years. In these efforts, the primary goal has been to gain a better understanding of heat transfer and temperature distribution as they relate to applications such as heat exchangers, natural circulation phenomenon in nuclear reactors, solar energy collectors, the thermal energy storage units, etc. In this connection, various geometries have been employed and the results are available in literature (for example see references [1-4]).

The present experimental study deals with natural convention heat transfer in an annular enclosure formed by two concentric vertical cylinders. The objective is to enhance heat transfer by increasing the contact surface area between the fluid inside the enclosure and the inner cylinder. This was accomplished by installing a large number of pin fins on the smooth surface of the inner cylinder.

The main motivation of the authors in conducting this work was its practical application to those thermal energy storage devices which benefit from the latent heat of a substance to store or release energy. To make the point clear, we refer to Figure 1 where a vertical smooth cylinder is situated in the solid-phase region of a two phase medium. Initially, the entire medium is at the liquid phase. However, as the cylinder is cooled by passing a cold fluid through it, heat is removed from the neighboring liquid and a thin layer of frozen material (solid phase) appears on the surface of the cold cylinder and grows with time.

At the first stage of freezing, the process is governed by the mechanism of heat conduction; but as the solidification process continues, the conduction thermal resistance increases, and heat transfer by natural convection becomes important. As seen in the figure, the buoyancy driven motion of the liquid phase substantially affects the shape of the frozen material, which has been quite symmetrical at the onset of freezing.

The importance of natural convection on solid-liquid phase change was first shown in a series of papers by Sparrow and co-workers [5-7]. It is now a well-known fact that natural convection plays an important role on the thermal performance of the latent-heat type energy storage devices. To improve the performance of these devices, one may naturally think of increasing the heat transfer surface area by employing fins.

The present investigation employed pin fins to enhance heat transfer from the inner cylinder of a vertical annular enclosure. To simplify and to better control the problem, the phase change phenomenon was totally eliminated from the investigation, hoping that

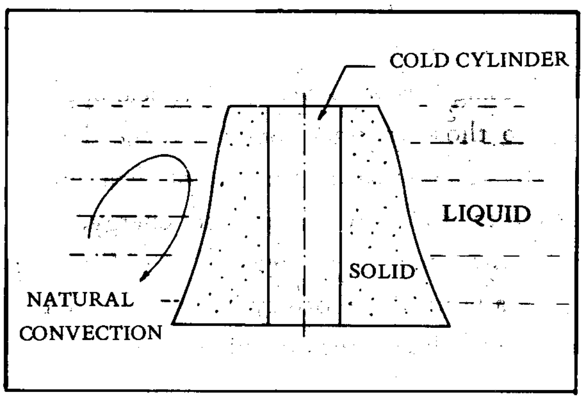

Figure 1. Freezing of a liquid

after gaining enough knowledge about the enhancement of single phase natural convection heat transfer in enclosures, the work would be continued in the future in a situation where the change of phase is also present. It is noteworthy that some researchers have reported studies where some kind of fins have been employed in a phase-change environment, but because of the complexity of the problem, the detailed information regarding the natural convection component of the total heat transfer has not been obtained.

The present experiments were carried out in a gaseous phase using air as the working fluid. The heat generated within the inner cylinder by electrical current was transferred to the outer cylinder of the enclosure by natural convection through air. Since for the temperature range considered in these experiments the Prandtl number Pr of air did not change significantly, the results reported here are for $Pr \cong 0.7$. The Rayleigh number Ra based on the radial gap size of the annulus ranged from 5×10^5 to 5×10^6. The fins were installed in either staggered or in-line arrangement, and in each case three different fin lengths l were employed, namely, l=2, 4, and 8 cm.

THE EXPERIMENTAL SETUP AND PROCEDURE

The main component of the experimental setup is the annular enclosure shown in Figure 2. As seen, the enclosure is made of two concentric copper cylinders. The diameter of the inner and outer cylinders are, respectively, equal to D_i= 20 mm and D_o=240 mm, resulting in the radial dimension of $R=(D_o-D_i)/2$=110 mm for the annulus.

The inner cylinder was fabricated from a solid copper rod. It was bored along the axis to a diameter of 12 mm and a Nickel-Chromium resistance wire (2 m long and 0.2 mm in diameter) was coiled and placed inside the cylinder to function as an electric heater. To prevent electrical contact between the resistance wire and the inner surface of the copper, a heat-resistant glass tube was inserted in between. Also, the thermal losses from the two ends of the cylinder were eliminated by rubber insulation. These arrangements enabled us to energize the inner cylinder by electrical current and to initiate a heat transfer process.

As mentioned earlier, the working fluid for heat transfer from the hot inner cylinder to the cold outer cylindrical vessel was air. It should be noted that the inside air was in pressure communication with the outside environment through the protective plastic tubing which carried the wires to the outside. Thus, the air could freely expand during a data run and was maintained at atmospheric pressure.

Temperature measurements were performed by four copper-constantan thermocouples. Three thermocouples were deployed along the surface of the inner cylinder with a distance of 50 mm apart from each other, and one was placed on the inner surface of the outer cylinder. In Figure 2, the location of thermocouples are identified by letters T, M, L and W. All thermocouple wires were stretched and directed along the surface of the annular enclosure while they remained attached to the walls. This arrangement ensured that the presence of the wires did not interfere with the free motion of air during a data run.

When the thermally active cylinder was placed inside the cylindrical vessel, and the annular space was isolated from the outside environment by the cover plate, the entire enclosure was immersed in a water bath and kept in place by a number of strings. The water was at room temperature and the high thermal conductivity of copper ensured that the enclosure would have a uniform wall temperature.

In addition to the experiments performed on the bare inner cylinder, a series of experiments was carried out while a total number of 96 pin fins had been attached to the inner cylinder. A typical fin is shown in Figure 3.

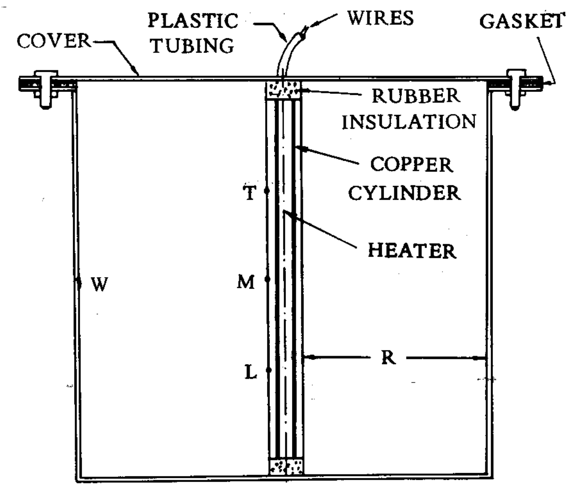

Figure 2. The annular enclosure

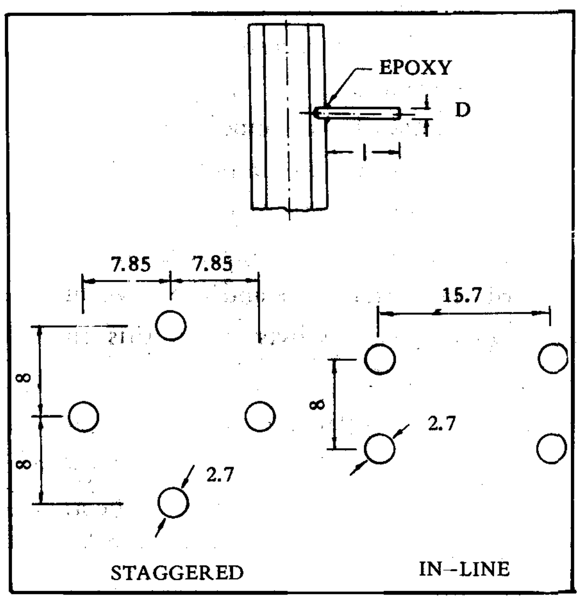

Figure 3. A typical fin and the fin arrangements (all dimensions are in mm)

The pin fins were made of copper rods with a diameter of d=2.7 mm, and were inserted into the respective holes on the wall of the inner cylinder. Epoxy cement was applied to the base of the fins to fix them in place. Three different fin lengths were used and for each fin length either a "staggered" or "in-line" arrangement was adopted. The fin distances in each arrangement are shown in the lower diagram of the figure.

To facilitate the identification of each case, the bare cylinder was assigned the number 1, while the finned cylinders were numbered 2 to 7. The fin arrangement and length for each case is shown in the following table:

Cylinder No.	1	2	3	4	5	6	7
Arrangement	bare	staggered			in-line		
Fin Length 1, cm	0	8	4	2	8	4	2

Later in the presentation of the results, we shall refer to these numbers for convenience.

The electrical power supplied to the resistance wire embeded in the inner cylinder was provided by a digital power supply which could adjust the voltage and current continuously from 0 to 40 volts and from 0 to 5 amperes, respectively. This instrument could resolve voltage and current to within 0.01 of the respective units.

The outputs of the copper-constantan thermocouples were read on a potentiameter with the resolution of 0.01 mv. Also, a digital thermometer was used to measure the temperature of the water bath and the environment. The smallest scale division on this thermometer was 0.1 C.

When the annular enclosure was placed in the water bath and it was adjusted in a horizontal position using a beam level, the apparatus was ready for the experimentation. The power supply was subsequently turned on and was adjusted to the desired power. Depending on the adjusted power, it took 3 to 6 hours for the enclosure to reach the steady-state condition. During this warm-up period, the thermocouples' outputs were repeatedly read at 10 minute intervals. When two successive readings did not show any detectable difference, the steady-state condition had been reached. At this condition, the thermocouples, the voltage and current of the resistance wire, and water temperature were all read and recorded.

Basically, the same procedure was repeated for the "bare" and the "finned" inner cylinder cases, and the same quantities were recorded. The recorded data were then employed to determine the various heat transfer variables, as will be described in the next section.

DATA REDUCTION

The average heat transfer coefficient for either the bare or the finned inner cylinder was evaluated from

$$h_i = Q_i / A_i (T_i - T_o) \tag{1}$$

where A_i is the surface area for heat transfer, and T_i and T_o are the mean surface temperature of the inner and outer cylinders, respectively. In this equation, the rate of convective heat transfer Q_i was obtained from

$$Q_i = Q_{elec} - Q_{loss} \tag{2}$$

in which the electrical input Q_{elec} (=VI) was evaluated from the recorded resistance wire voltage and current, while the thermal losses $Q_{loss}(=Q_{rub} + Q_{rad})$ were calculated from the following correlations for the conduction losses Q_{rub} through the rubber insulations and the radiation losses Q_{rad} between the inner and outer cylinders.

$$Q_{rub} = k_{rub}\, A_{rub}\, \frac{T_i - T_\infty}{t_{rub}} \tag{3}$$

$$Q_{rad}=\sigma(T_i^4-T_o^4)/(\frac{1-\epsilon_i}{\epsilon_i A_i}+\frac{1}{A_i F_{i-o}}+\frac{1-\epsilon_o}{\epsilon_o A_o}) \qquad (4)$$

In these equations k_{rub}, A_{rub}, T_∞, t_{rub}, σ, A_i, and A_o are, respectively the thermal conductivity of rubber, conduction surface area of the rubber, water temperature, rubber thickness, Stefan-Boltzmann constant, and the surface area of the inner and outer cylinders. The emissivities ϵ_i and ϵ_o for slightly polished copper were taken as 0.1, and the radiation shape factor F_{i-o} between the cylinders was equal to unity.

The heat transfer results are converted to Nusselt numbers $Nu=h_iR/K$ and will be presented as a function of annulus Rayleigh number

$$Ra=\{g\beta\,(T_i-T_o)R^3/\nu^2\ \ Pr\} \qquad (5)$$

where R is the radial distance between the surfaces of the inner and outer cylinders. All thermophysical properties which appear in above equations were evaluated at the film temperature $T_f=(T_i+T_o)/2$.

RESULTS AND DISCUSSION

Heat transfer results are presented in Figure 4. The ordinate is the Nusselt number Nu based on the radial size of the annulus R, and the abscissa is the Rayleigh number Ra (Equation 5). Two sets of data points are shown in this figure. The first set is for the bare (no fin) inner cylinder (designated as cylinder no. 1), while the outer set is for the finned inner cylinder with staggered arrangement (cylinder no. 2, 3, and 4).

Attention is first turned to the bare cylinder results which are shown by open circles. It is clearly seen that, on a logarithmic scale, Nu varies linearly with Ra. It is noteworthy that natural convection results are normally correlated in the form $Nu=nRa^m$, where n and m are two constant values which are determined from experiment. The results of the present investigation can also be correlated in a similar way. Using the least-squares curve fitting method, the bare cylinder data are correlated as

$$Nu=1.301Ra^{0.237} \qquad (6)$$

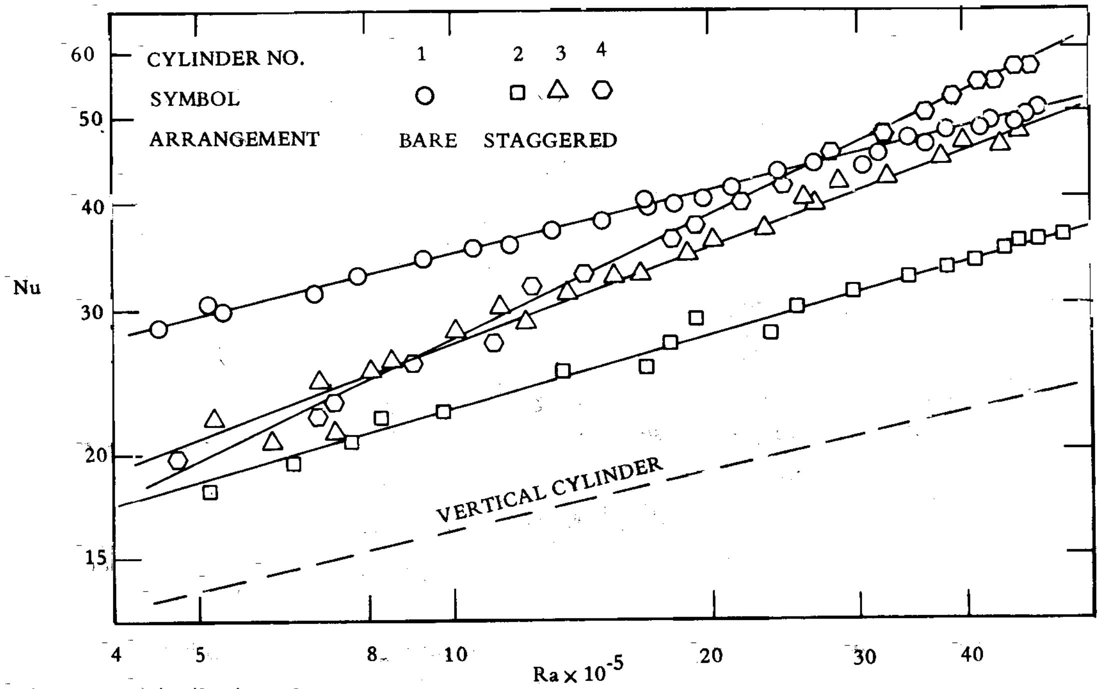

Figure 4. Distribution of Nu for the staggered arrangement

It is interesting to compare this result with the heat transfer result for an infinitely long vertical smooth cylinder situated in an infinite convective medium [8].

$$Nu=0.508Ra^{0.25} \tag{7}$$

To make the comparison meaningful, Nu and Ra in Equation 7 are defined in terms of R. Examination of Equations 6 and 7 indicates that the results of the present investigation are, on the average, 21.3% higher than those predicted by Equation 7. However, the exponents of Ra in these equations differ by only 5.2%. The high values of Nu in the present experiment are believed to be caused by the proximity of cold walls of the enclosure which strengthen the buoyancy forces and enhance the recirculating motion of air near the cylinder.

Vahl Davis and Thomas [1]have investigated natural convection between concentric vertical cylinders by a computational procedure. However, the range of parameters in their work is quite different from those employed in the present study and, therefore, no comparison is made here.

Now, attention is turned to the results for the finned cylinder which have been identified by the symbols corresponding to cylinder number 2, 3, and 4 in Figure 4. It is seen that, in general, the presence of fins have a decreasing effect on the heat transfer coefficient. This observation is severer at lower values of Ra. At higher Ra, as the length of the fins decreases, the heat transfer coefficient increases, and it may even increase above that of the bare cylinder.

The decreasing effect of fins on the heat transfer coefficient is apparently due to the fact that they present a larger resistance to the fluid motion and have a retarding effect on it. Moreover, the surface of the fins are colder, and the resulting reduced temperature difference between the solid boundary of fins and the surrounding fluid decreases the heat transfer. In any case, it should be emphasized that the main function of fins are to increase the surface area, and as it will be seen later in this section, the effect of the increased area may overwhelm the other factors, and the heat transfer is actually increased.

The solid lines passing through the data points in Figure 4 are the least-squares fits with the equation $Nu=nRa^m$, where the constant values (n, m) for cylinder number 2, 3, and 4 are,respectively, equal to (0.447, 0.284), (0.162, 0.370), and (0.041, 0.471). In general, it may be concluded that the Nu for the finned cylinders respond faster to a variation of Ra (exponents of Ra are higher than 0.25), but the values of Nu are between those for the bare cylinder annulus and the cylinder situated in an infinite domain (Equation 7).

The heat transfer coefficients for the inner cylinder with in-line pin fins are shown in Figure 5. Examination of this figure reveals a trend similar to that discussed earlier. Again, the presence of the fins has reduced the heat transfer coefficients and the coefficients are lower than those of the staggered fins. Knowing that the number of fins (and, therefore, the total surface area) in both staggered and in-line arrangement is equal (96 fins in each case), we conclude that the staggered arrangement has superior heat transfer characteristics.

In this figure, the solid lines passing through the data points were obtained by least-squares method with (n, m)=(0.587, 0.258), (0.321, 0.316), and (0.044, 0.455) corresponding to cylinder numbers 5, 6, and 7, respectively.

A quantity of practical importance is the rate of heat transfer Q_i, which indicates the ability of an annular enclosure (or, practically

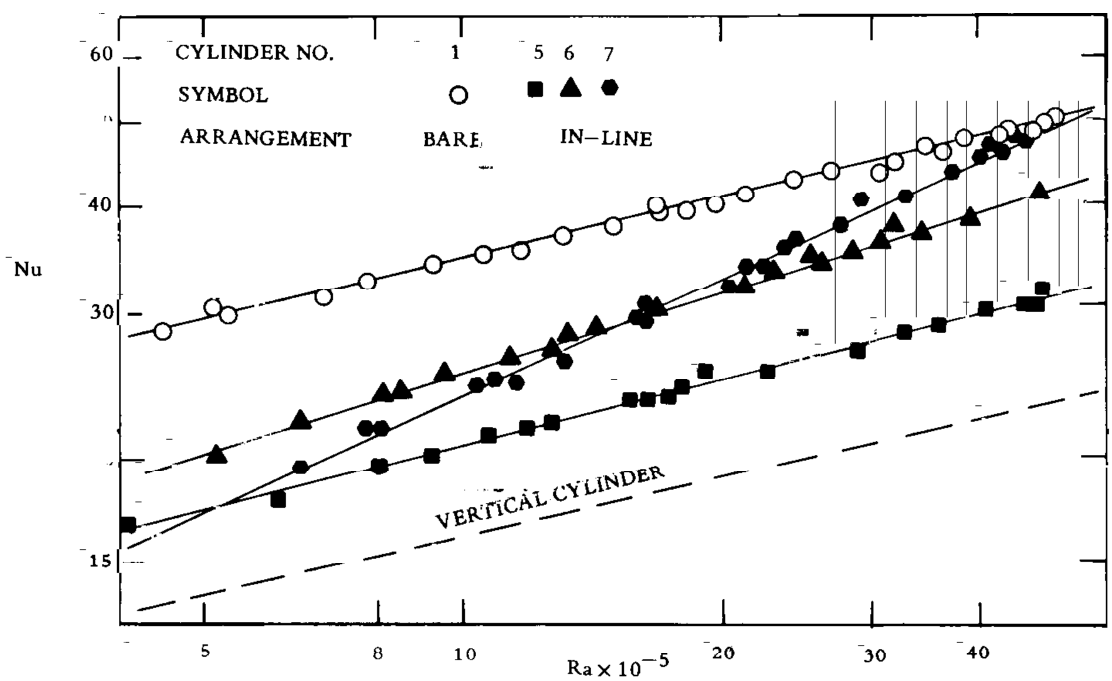

Figure 5. ***Distribution of Nu for the in-line arrangement***

speaking, an energy storage unit) to transfer thermal energy. The variation of this quantity with Ra for the bare and finned cylinders is shown in Figure 6. As seen, in all cases the rate of heat transfer increases when Ra is increased. It is also observed that Q_i depends on the fin length and the fin arrangement. The finned cylinders have higher rates of heat transfer, and as it was mentioned earlier, the staggered arrangement has a superior performance.

Although the rate of heat transfer Q_i is strongly dependent on Ra, when it is normalized in the form Q_i/Q_1 (Q_1 is the rate of heat transfer from the bare cylinder at the same Ra), it becomes nearly independent of Ra. This is shown in Figure 7. Note that the scale of the ordinate for each set of data in this figure is different. The horizontal solid lines seen in Figure 7 represent the arithmetic mean of the data points.

Using the mean values of Q_i/Q_1, the normalized rate of heat transfer now depends on the fin length and their arrangement. As demonstrated in Figure 8, Q_i/Q_1 increases with increasing the length of the fins 1. In the range of parameters considered in this

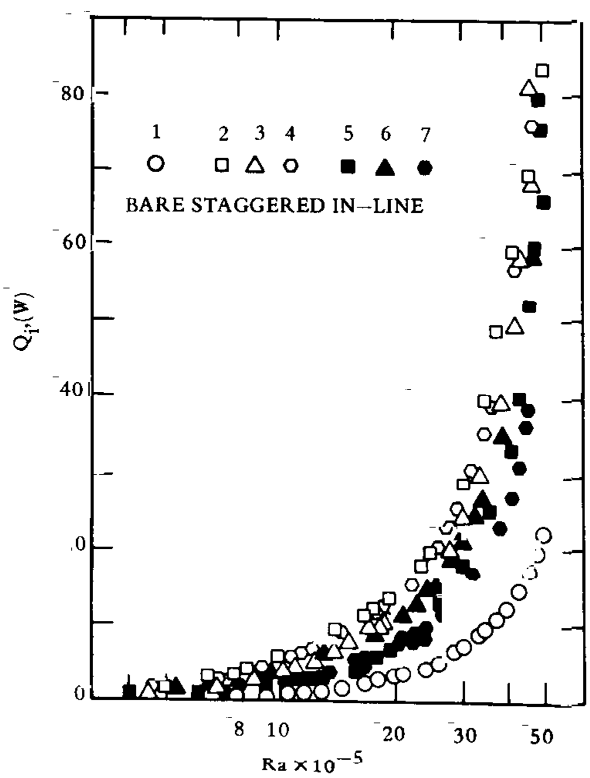

Figure 6. ***Rate of heat transfer for bare and finned cylinders***

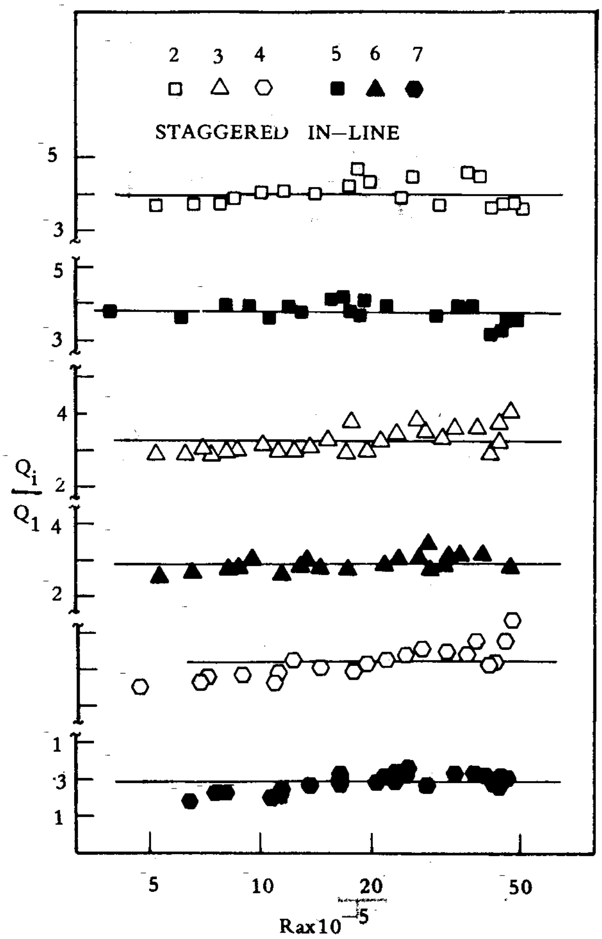

Figure 7. Normalized rate of heat transfer is independent of Rayleigh number

work, increases as large as 300% are seen. It is also observed that as 1 increases beyond a certain limit, the values of Q_i/Q_1 appear to approach asymptotically to a fixed value. In fact for very long fins, the fins are not efficient, and they can not increase the heat transfer beyond a certain value. In practice, the dimensions of the heat exchange device may put a limit to the length of the fins that can be employed in the device, and fins of infinite length are usually not encountered in reality.

To facilitate the application of the results of the present investigation in practical situations, the mean values of Q_i/Q_1 seen in Figure 8 were correlated by the following qua-

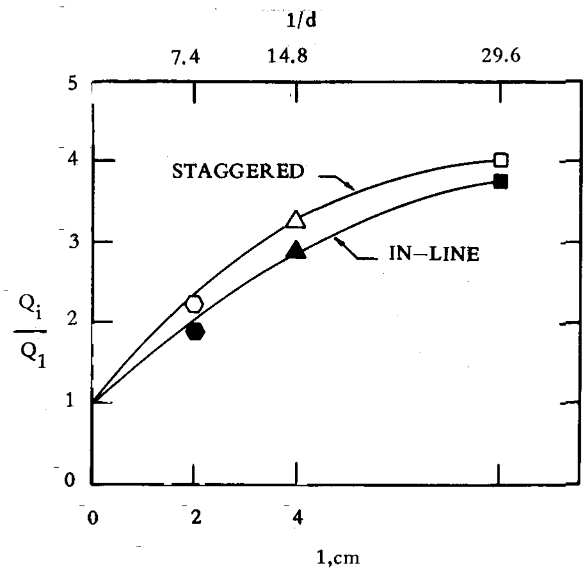

Figure 8. Normalized rate of heat transfer versus fin length

dratic equations

$$Q_i/Q_1 = -3.246 \times 10^{-3}\,(l/d)^2 + 0.199\,(l/d) + 0.977 \quad (8)$$

$$Q_i/Q_1 = -1.967 \times 10^{-3}\,(l/d)^2 + 0.154\,(l/d) + 0.959 \quad (9)$$

Equations 8 and 9 apply to the finned inner cylinder with staggered and in-line arrangements, respectively.

CONCLUDING REMARKS

The present experimental investigation has revealed the heat transfer characteristics of an annular enclosure consisting of two concentric cylinders. In a major part of this study, pin fins were attached on the surface of the inner cylinder in order to increase the overall rate of heat transfer.

Two types of fin arrangements were considered, namely, staggered and in-line arrangements. The experiments were carried out for three different fin lengths 1=2, 4, and 8 cm, while Rayleigh number varied in range from 5×10^5 to 5×10^6. It was found that:

1. The presence of fins increases the overall rate of heat transfer from the annulus. This increase is basically due to an increased surface area.

2. When the fins are present, the heat transfer coefficient for the inner cylinder is generally lower than that for the bare cylinder case.

3. Fins with staggered arrangement perform better than those with in-line arrangement.

4. As the lengths of the fins are increased, the overall rate of heat transfer improves. This improvement can be as large as 300% (see Figure 8).

ACKNOWLEDGMENTS

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NOMENCLATURE

A_i	surface area for heat transfer
A_o	surface area of the outer cylinder
A_{rub}	conduction surface area of the rubber
d	fin diameter
D_i	diameter of the inner cylinder
D_o	diameter of the outer cylinder
g	gravitational constant
h_i	average heat transfer coefficient for the cylinder number i
I	current
k	thermal conductivity of air
k_{rub}	thermal conductivity of rubber
l	length of fin
Nu	Nusselt number
Pr	Prandtl number
Q_1	rate of heat transfer from cylinder number 1 (bare cylinder)
Q_{elec}	electrical power dissipated in the annulus
Q_i	rate of heat transfer from cylinder number i
Q_{loss}	rate of thermal energy loss
Q_{rad}	rate of radiation loss from the inner cylinder
Q_{rub}	rate of conduction loss through the rubber insulation
R	radial size of the annulus=$(D_o - D_i/2$
Ra	Rayleigh number
T_f	film temperature
T_i	mean surface temperature of the inner cylinder
T_o	surface temperature of the outer cylinder
t_{rub}	rubber thickness
T_∞	water temperature
V	voltage