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HEAT TRANSFER IN AN ANNULAR GAP

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Abstract—Heat-transfer coefficients were measured for the flow in the annular space between an inner rotating cylinder and an outer stationary one, with superimposed axial flow. The problem was stimulated by the desire for additional information on the design of cooling systems for electric motors of high power density. The speeds of rotation were such as to include Taylor numbers up to about 10^6, and the range of Reynolds numbers based on the axial velocity components and the gap distance were extended to 7000. Experiments were performed at three different Prandtl numbers 2.5, 4.5 and 6.5.

NOMENCLATURE

b, gap width;
f, friction coefficient;
Nu_{co}, Nusselt number for pure conduction;
Nu_R, Nusselt number based on temperature difference between rotor and bulk fluid;
Nu_{RS}, Nusselt number based on temperature difference between rotor and stator;
P, geometric coefficient [see equation (2)];
q, heat-transfer rate per unit area per unit time;
r_1, radius of rotor;
r_2, radius of stator;
Re_a, Reynolds number based on axial velocity component;
Ta_m, modified Taylor number [equation (1)];
T_B, bulk temperature;
T_R, rotor surface temperature;
V_a, axial velocity component;
ν, kinematic viscosity;
Ω, angular speed of rotor.

INTRODUCTION

FOR MANY years research workers in fluid mechanics have been interested in types of flow found in an annulus between two concentric cylinders rotating with respect to one another. Since Taylor first predicted and then demonstrated the laminar instability which now bears his name, a great effort has been made to understand the flow characteristics that can exist under various conditions. The interest is based not only on the challenge to understand the basic fluid mechanics, but also on the fact that flow in between annuli occurs in many engineering applications. The present study was stimulated by questions arising from the design of electric motors of high power density. Such motors are designed to have a high power output for a given size and weight. The heat dissipation will be correspondingly high and active provisions for cooling will usually be required. Liquid coolants will often be selected, and the fluid may have to be pumped through the annular space between the rotor and the stator. The purpose of the present investigation is to study the heat transfer in such a flow, that is the heat transfer to the fluid in an annulus between a rotating inner cylinder and a stationary outer one with a flow component in the axial direction.

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PREVIOUS WORK

Relatively few studies have been performed on the combined effects of rotation and axial flow on the heat transfer between concentric cylinders, which is the principal subject of the present work. One of the earliest and most extensive investigations of this problem was carried out by Gazley [1]. His apparatus consisted of a modified electric motor, and the installation was designed so that it could be operated with a smooth or with a slotted rotor. The working fluid was air. A subsequent study was conducted by Becker and Kaye [2], also with air. Additional investigations include that by Tachibana and Fukui [3], of Kosterin and Finat'ev [4] and of Kuzay and Scott [5]. All of the studies were conducted with air at a single Prandtl number of about 0.75. The work on heat transfer in an annular gap in the absence of axial flow is much more extensive and includes that by Longobardo and Elrod [6], Tachibana, Fukui and Mitsumura [7], and Aoki, Nohira and Arai [8]. The latter work in particular covers a number of different Prandtl numbers as well as a fairly wide range of rotational speeds. The data are considered reliable and they are in good agreement with those of Becker and Kaye [2]. The data by Aoki *et al.* [8] have, therefore, been selected to serve as a reference to check the reliability of the information from the present test installation for the limiting case for which the axial flow is zero.

EXPERIMANTAL INSTALLATION

The test apparatus was to be designed for the purpose of measuring the heat-transfer coefficient from the rotor to the fluid in the annulus. The design also was to permit axial flow through the annulus. A sketch of the installation is shown in Fig. 1. The rotor is $4\frac{1}{16}$ in (10.3 cm) in diameter and 48 in (122 cm) long. It is held by two ball bearings and driven by a variable speed electric

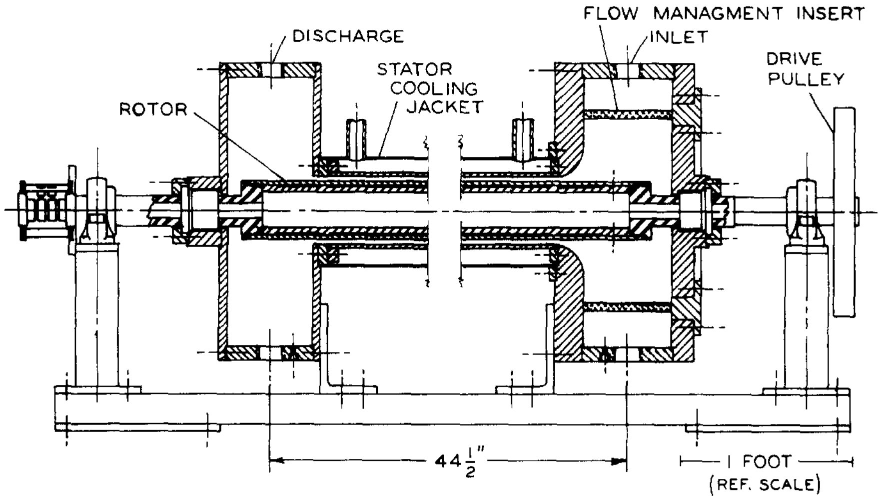

FIG. 1. Assembly drawing of the test section

motor through a pulley and belt transmission. The maximum speed of the rotor is about 160 rev/min. The stationary housing is mounted concentrically to leave an annular space of 0.22 in (0.55 cm). For axial flow, fluid is supplied through the right hand reservoir. This reservoir is provided with a set of screens to promote uniform flow and to reduce turbulence. The entrance to the annulus is rounded to prevent disturbances at this section. The flow is discharged through a simple reservoir at the end of the annular passage. The stationary housing is surrounded by a cooling jacket, so that the heat transfer from this stator may be controlled. The heat-transfer rate from the stator to the jacket is then computed from the flow rate and the temperature rise of the cooling fluid in the jacket.

The rotor itself consists of three separate sleeves. The surface of the outer one is in contact with the test fluid. Several thermocouples are attached to this sleeve at points just below the surface. A second sleeve containing several longitudinal grooves is placed inside the first sleeve to provide passages for the thermocouple wires. The third and innermost sleeve carries the heating tape which is wound around the outside of this sleeve. The tape is capable of delivering a maximum output of 4 kW, corresponding to about 15 kW/m^2. Pyrex was selected for the material of the third sleeve so as to reduce axial heat leakage toward the ends of the rotor. Two rotors were used in the course of the tests. The first, with the dimensions given above, has a smooth surface. The second one was provided with longitudinal slots to simulate the surface conditions of many actual rotors. There are a total of 30 equally spaced slots, 0.0625 in (0.16 cm) deep and 0.125 in (0.32 cm) wide with a diametral distance to the outer crests of 4.125 in (10.5 cm). The smooth rotor has been provided with a total of six thermocouples attached to the outermost sleeve within about 0.03 in of the surface. Two of these are located near the center (in an axial direction) of the rotor at positions of 180° apart. These thermocouples were used in the computation of the reported results. Four additional thermocouples are located at positions 4.5 in (11.4 cm) and 9 in (22.9 cm) upstream and downstream of the central position. These thermocouples are displaced circumferentially by 30° from each other and from the centrally located ones. By placing the thermocouples in this manner it was hoped to detect any possible abnormality in the flow or the heating pattern. Six thermocouples are also installed in the slotted rotor. Four of these are located centrally, again at positions displaced circumferentially by 180°. One set is located at the bottom and the other at the top of the slots. The remaining two thermocouples are located at 6 in (15.2 cm) upstream and downstream of the central location. The stator temperature was measured by thermocouples located at the outside wall of the stator. These thermocouples are separated by the thickness of the stator wall from the surface which is in contact with the test fluid. A correction, therefore, had to be applied to the readings of these thermocouples. This correction, however, was small as the stator wall was made of 0.25 in (0.64 cm) aluminum, and could be computed with confidence as the heat-transfer rates through the stator to the cooling jacket were known. A total of ten thermocouples were installed on the stator. Of these, four were located centrally at 90° from each other and at the same axial position of the corresponding ones for the rotor. The output from this set was used in the computation of the reported results. Additional details of the method of attachment of the thermocouples are given in [9]

To indicate the order of magnitude of the temperatures which occur in the experiment, it may be mentioned that the bulk temperature of the water ranged from about 70°F (21°C) to about 165°F (74°C) and a typical value for the difference between the rotor temperature and the bulk temperature was 15°F (8.3°C).

The power to the heating tape inside the rotor as well as the thermocouple signals were carried between the rotor and the outside through slip rings. The slip rings, which were obtained commercially, are made of coin silver with brushes of silver-graphite. They functioned very satisfactorily in the present application. Water was selected as a test fluid largely because it makes it conveniently possible to obtain a range of Prandtl numbers by just changing the temperature.

For each test the desired flow rate and speed of rotation was selected. The heating rate of the tape was then set as well as the flow rate through the outer jacket of the stator. The system was then allowed to come to equilibrium, and all measurements were made in the steady state. A detailed description of the installation is given in [9].

EXPERIMENTAL RESULTS

(a) *Descriptive parameters*

The results presented apply to a section normal to the axis of the rotor at a point midway between the inlet to and the discharge from the annulus. At this section errors due to axial gradients are at a minimum. The results are then given in terms of four principal parameters, the Prandtl number of the test fluid, a Reynolds number, a Taylor number, and a Nusselt number. The Prandtl number was determined simply from the average temperature of the fluid in the annulus at the measuring station. The Reynolds number, is defined here as

$$Re_a = \frac{2V_a b}{\nu}$$

that is in terms of the axial velocity component and the hydraulic diameter of the annulus. The modified Taylor number is given by

$$(Ta)_m = \frac{2\Omega^2 r_2^1 b^3}{\nu^2(r_1+r_2)}\left(\frac{1697}{\pi^4}P\right) \tag{1}$$

where P is a geometrical factor. This factor is equal to

$$P = 0.0571\left(1-0.652\frac{b}{r_1}\right)+0.00056\left(1-0.652\frac{b}{r_1}\right)^{-1} \tag{2}$$

The term in brackets in equation (1) tends to unity as b/r_1 tends towards values much smaller than 1.0 and then the ratio $2r_1/(r_1+r_2)$ also approaches unity. In that case the modified Taylor becomes equal to the Taylor number as usually defined. Taylor showed analytically and experimentally that the modified Taylor number accurately predicts the effect of the gap ratio b/r_1 on the onset of the vortex motion, at least as long as this ratio is no larger than the one which applies to the present experiments. As the present experiments were carried out with only one ratio b/r_1 they cannot give any indication whether or not all the effects of the gap ratio on the heat transfer are taken into account through the modified Taylor number. Nevertheless, the present results have been given in terms of this number with the expectation that, at least to a first approximation, the data will apply to a range of gap ratios. Both parameters Ta and Ta_m were introduced by Taylor. The names "Taylor number" and "modified Taylor number" were, of course, coined by subsequent authors. For brevity the modified Taylor number, Ta_m, will subsequently be simply called "Taylor number".

The heat-transfer results themselves are presented in terms of a Nusselt number. In the course of the analysis of the data several Nusselt numbers were considered based on the temperature difference between the fluid and the rotor, between the fluid and the stator and between the rotor and the stator respectively [9]. In the absence of axial flow, the Nusselt number based on the temperature difference between the rotor and the stator, Nu_{RS}, is the generally accepted parameter and is also being used for the present results. When there is axial flow, however, the quantity Nu_{RS} does not lend itself to a generalized representation and is quite dependent on such factors as the inlet temperature. In the presence of axial flow it was found that a more representative parameter was a Nusselt number based on the temperature difference between the local bulk temperature and the corresponding temperature at the rotor surface. This parameter was defined as

$$Nu_R = \frac{qb}{(T_R - T_B)k}$$

where the local heat-transfer rate was obtained from the power dissipated in the heating tape. The bulk temperature T_B was computed from the temperature of the water at the inlet to the annulus taking into account the heat added from the rotor as well as the heat transfer to the stator housing. The Nusselt number defined in this way (Nu_R) differs from the one generally used when $Re = 0$. For the limiting condition of zero axial flow the newly defined Nu_R exceeds Nu_{RS} by about a factor of 2. In the present paper Nu_{RS} appears only in Fig. 2, in which the results for $Re = 0$ are presented. The definition of the parameter Nu_R is really essential for a unified representation of the data taken in the presence of axial flow and the introduction of this parameter should not be regarded as an unnecessary complication of the nomenclature. Further discussion of this point is given in [9]. In addition, to reduce the dependence on the gap ratio, most of the heat-transfer results are given in terms of the ratio Nu_R/Nu_{co}, where the conduction Nusselt number is defined as

$$Nu_{co} = \frac{b/r_1}{\ln r_2/r_1}.$$

Finally, an estimate of the accuracy of the heat transfer results may be in order. Taking into account all of the measurements as well as geometrical uncertainties the values given for the Nusselt number are believed to be within $\pm$ 15%.

(b) *Results without axial flow*

Experiments for zero axial flow and the smooth rotor were conducted mainly in order to be able to compare present results with those of earlier investigators. In Fig. 2 the ratio of Nu_{RS}/Nu_{co} is plotted against the Taylor number. On this graph a line is drawn defined by the data of Aoki *et al.* [8]. These data, as mentioned earlier, are considered to be very reliable. The results of the present measurements are indicated by the open circles and the agreement between the two is thought to be very acceptable. This agreement lends credibility to the data obtained from the present test installation.

Measurements were also made at low values of Ta_m,

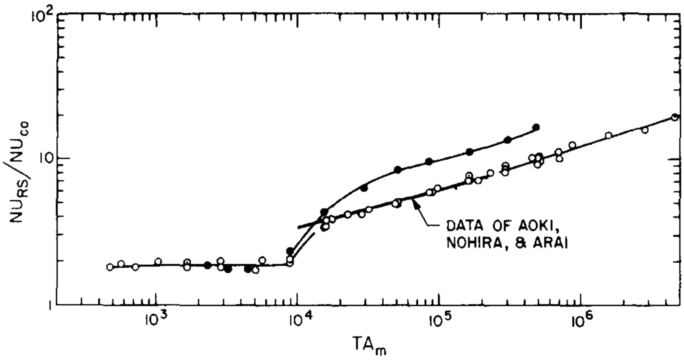

FIG. 2. Heat-transfer coefficients vs modified Taylor number at a Prandtl number of 4.5 and a Reynolds number of zero. ⊙, smooth; ●, slotted.

before the onset of vortices. Generally, simple Couette flow and a constant Nusselt number equal to Nu_{co} would be expected in this range. The observed Nusselt number is indeed constant, however, it definitely exceeds the value Nu_{co}. The difference may be explained by free convection. The existence of free convection has been confirmed in the present series of experiments by circumferential temperature measurements and the fact that free convection may account for the magnitude of the observed Nusselt number has been verified by comparison with data from earlier investigations by Kraussold [10].

Free convection exerts another important influence on the results in that it affects the onset of the vortex motion. For the present conditions it postpones the onset to a Taylor number of about 10^4 compared to a value of about 2000 which exists in the absence of free convection.

Also shown on Fig. 2 are the data for the slotted rotor. At low Taylor numbers before the onset of vortices the slots seem to have little effect and free convection still dominates the heat-transfer process. At the higher Taylor numbers the slots definitely improve the heat-transfer process. The increase in Nu_{RS} is significant, 40–50%, but not spectacular. One point of caution should be raised when comparing the results of the slotted rotor to those of the smooth one. The Taylor number for the slotted rotor was based on the average width, b, of the annular gap, that is the distance from the stator wall to an imaginary cylindrical surface passing through the points at one half of the slot depths [$b = 0.22$ in (0.55 cm)]. This selection is in a sense, arbitrary and a different selection would have the effect of translating the data parallel to the abscissa. Furthermore, the surface area of this imaginary cylinder [$d_{ave} = 4\frac{1}{16}$ in (10.3 cm)] was taken for the computation of the heat transfer per unit area.

(c) *Results with axial flow*

The main purpose of the project was to study the heat transfer to the fluid in the annulus in the presence of axial flow and for a range of Prandtl numbers. Experiments were conducted at three different Prandtl numbers ($Pr = 6.5$, 4.5 and 2.5) and for axial Reynolds numbers, Re_a, up to 7000. This is, perhaps, the first set of data for axial flow at Prandtl numbers different from that for air. The Reynolds numbers were selected so as to include the range which seems of most interest for industrial applications, and Taylor numbers were extended to fairly high values ($> 10^6$) for several operating conditions. In addition all of the experiments were conducted in a single installation which facilitates the detection of trends with changes in the independent variables.

The sets taken with the smooth rotor at Re_a of 800, 2700 and 7000 are shown in Figs. 3, 4 and 5 respectively. At $Re_a = 800$, Fig. 3, the flow is believed to be laminar before the onset of vortices and for most of the range for which vortex motion exists. The Nusselt number increases with the Prandtl numbers as usual. For most of the range a multiplier of $Pr^{0.4}$ would represent the effect of the Prandtl number fairly well, but there is not sufficient information to recommend a general empirical relation at this time. The onset of the vortices occurs at a Taylor number (Ta_m) of about 3×10^4, as compared to about 2000 which is the critical value for $Re_a = 0$ ideally obtained in the absence of free convection. This kind of change is in agreement with the findings of earlier observers (Becker and Kaye [2]) who showed that the Taylor number at onset, also called the critical Taylor number, steadily increased with Re_a, until it reaches about 3×10^4 for a Reynolds number (Re_a) of about 10^3. There was little further change for still higher Reynolds numbers and in fact some decrease in the critical Taylor was observed for $Re_a > 2000$. It may also be noted that at low Taylor numbers before the onset of vortices, rotation does not affect the Nusselt numbers and the heat transfer is determined by the axial Reynolds number.

The data for an axial Reynolds number of 2700 are given in Fig. 4, and it is seen that the transition to vortex flow occurs at about the same Taylor number as for $Re_a = 800$, which is in agreement with the statement just made. The flow in the annulus must now be assumed turbulent, but the general characteristics of the heat transfer are not changed. Before the onset of vortex motion the rotation has little effect on the heat transfer and again, the Nusselt number increases with the Prandtl number. The highest axial flow component was reached for $Re_a = 7000$ and the results are shown in Fig. 5. No data were taken beyond $Ta_m = 10^5$, because for the given heat source, the temperature

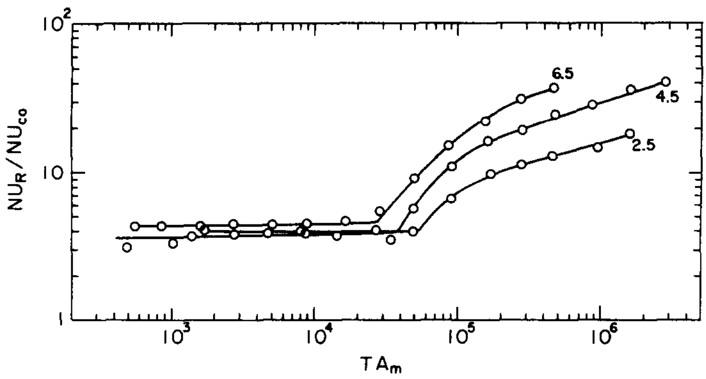

FIG. 3. Heat-transfer coefficients vs modified Taylor number at an axial Reynolds number of 800. Each curve is for a constant Prandtl number, which is indicated. The rotor is smooth.

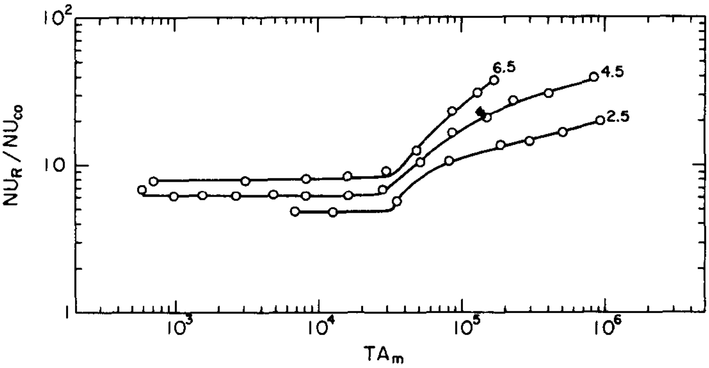

FIG. 4. Heat-transfer coefficients vs modified Taylor number at an axial flow Reynolds number of 2700. Each curve is for a constant Prandtl number, which is indicated. The rotor is smooth.

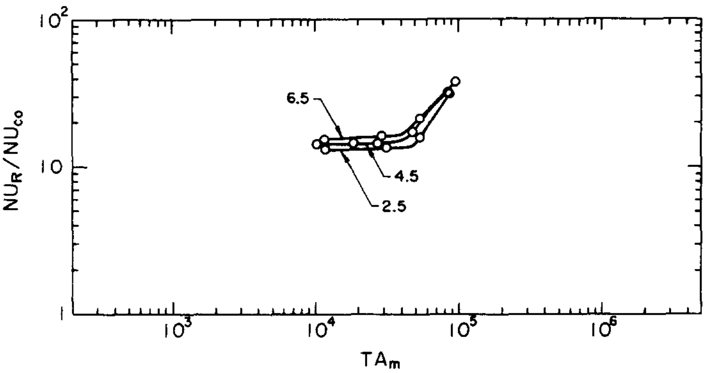

FIG. 5. Heat-transfer coefficients vs modified Taylor number at an axial flow Reynolds number of 7000. Each curve is for a constant Prandtl number which is indicated. The rotor is smooth.

difference between the rotor and the fluid became too small to be measured accurately. The axial flow improves the heat transfer for the whole range of Taylor numbers shown. Compared to the data at $Re_a = 800$, the Nusselt number is increased by a factor of about 6 and 4 before and after the onset of the vortex flow. In this highly turbulent range, the Prandtl number seems to be less of a factor. Although this may seem somewhat surprising, one has to realize that the interaction between the turbulent flow, the rotation introduced by the rotor, and the vortex structure is very complex and makes the visualization of the mechanism and the prediction of the results very difficult.

The results may also be examined in the form of graphs for the Nusselt number ratio (Nu_R/Nu_{co}) vs the axial Reynolds number (Re_a) at constant Taylor numbers (Ta_m). Two such graphs have been prepared as Figs. 6 and 7. The first is prepared for a Taylor number of $Ta_m = 5 \times 10^3$. In this range and for the Reynolds numbers under consideration the heat transfer is insensitive to changes in Taylor number and no vortex has yet been established. It is seen that under these conditions the Nusselt number increases continuously with Re_a. Comparison with the data of Carpenter [11] which were taken without any rotation ($Ta_m = 0$) shows that in this range the motion of the

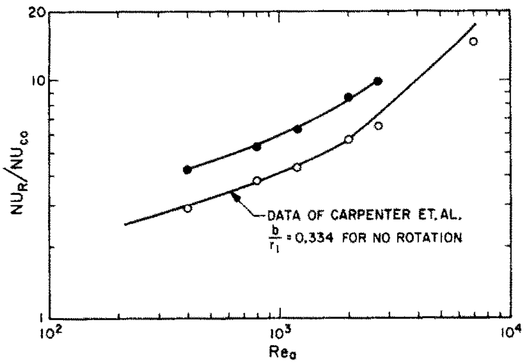

FIG. 6. Cross correlations of heat-transfer coefficient vs Reynolds number at a Prandtl number of 4.5 and a modified Taylor number of 5×10^3. ⊙, smooth; ●, slotted.

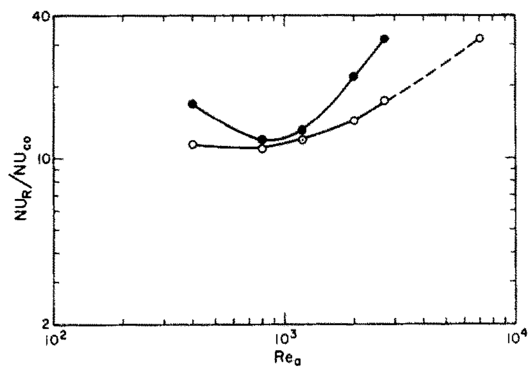

FIG. 7. Cross correlation of heat-transfer coefficient vs Reynolds number at a Prandtl number of 4.5 and a modified Taylor number of 10^5. ⊙, smooth; ●, slotted. (Dashed portion of curve obtained through extrapolation from $Ta_m = 0.9 \times 10^5$ to $Ta_m = 10^5$.)

rotor has no influence on the heat transfer at all, that is the heat-transfer mechanism is determined entirely by the axial flow. At the high Taylor number of $Ta_m = 10^5$ the results are quite different. For the range shown on Fig. 7 transition has occurred and vortex motion must be assumed to exist in the annulus. The data show that the axial flow has little effect on the heat transfer up to about $Re_a = 10^3$. Beyond that there seems to be an interaction between the axial flow and the vortex motion and the Nusselt number increases about 3.5 times as the Reynolds number goes from 10^3 to 8×10^3.

These two examples illustrate the heat-transfer characteristics for two uniform conditions, one is the absence of vortex motion, and one when such motion was taking place. Operating conditions may also exist where the Taylor number is such that vortex motion exists when the axial Reynolds number is low, but where this vortex motion subsides with higher Reynolds numbers. It is then actually possible to observe a decrease in heat transfer with increasing Reynolds number. This possibility will be discussed further in connection with he results of the slotted rotor.

(d) *Slotted rotor*

The effect of the slots on the heat-transfer characteristics is presented through direct comparison with the data for the smooth rotor. This comparison is given in Figs. 2, 8 and 9 in which the Nusselt number ratio is plotted against the Taylor number for three different axial Reynolds numbers. Each graph shows two curves, one for the smooth rotor and one for the slotted one, and all of the three graphs are for a Prandtl number of 4.5. The data for zero axial flow, which are shown in Fig. 2, have already been discussed in section (b). When there is some axial motion ($Re_a = 800$) the slots do bring about an improvement in heat transfer (Fig. 8) even before the transition to vortex flow and, as for the smooth rotor, the Nusselt number in this range is independent of the Taylor number. The flow transition, when compared to that of the rotor, is displaced slightly towards higher Taylor numbers, and for Taylor number beyond 10^5 the slotted rotor again shows an improved heat-transfer coefficient. For still higher axial flows (Fig. 9, $Re_a = 2700$) the slotted rotor performance is similar and provides an increased Nusselt number for all Taylor numbers. It may be pointed out that the flow transition for the slotted rotor is even reduced compared to the value for $Re_a = 800$. This observation, that the Taylor number for transition first increases with increasing axial flow and then decreases again has been made rather consistently for the slotted rotor at all operating conditions, and the fact that there is also some indication of this trend for the smooth rotor has been mentioned before. This interrelation between flow transition and axial flow rate incidentally may lead to the situation where an increase

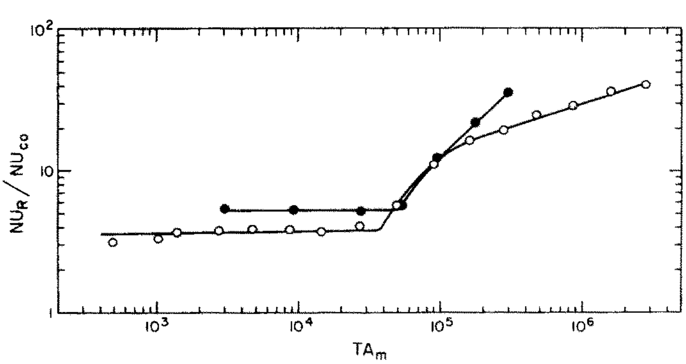

FIG. 8. Heat-transfer coefficient vs modified Taylor number at a Prandtl number of 4.5 and a Reynolds number of 800. ⊙, smooth; ●, slotted.

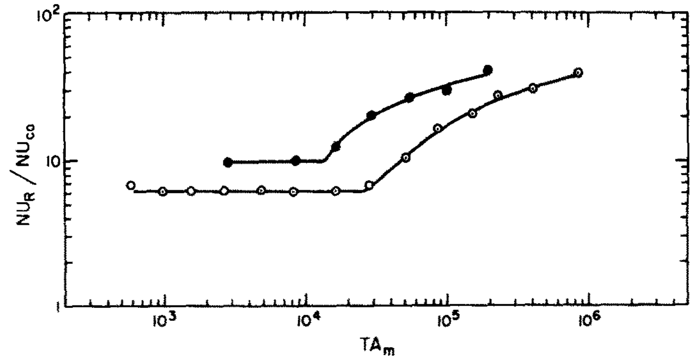

FIG. 9. Heat-transfer coefficient vs modified Taylor number at a Prandtl number of 4.5 and a Reynolds number of 2700. ⊙, smooth; ●, slotted.

in axial flow may actually cause a decrease in heat-transfer coefficient. This possibility is well illustrated by the data taken with the rough rotor at a Taylor number of 10^5 (Fig. 7).

(e) *Pressure drop*

A few measurements were made of the pressure drop between the inlet and the discharge reservoir. This information, it was thought, would be helpful in the design of a cooling system. The results are given in terms of a friction factor, f, in Figs. 10 and 11 for the smooth and slotted rotor respectively. The pressure drop on which f is based includes entrance and discharge losses as well as the higher shear in the first portion of the annulus before the flow is fully established. This explains why the friction coefficient is somewhat higher than that expected for the ideal case as indicated on the graph. It also may explain why the laminar–turbulent transition is smoother than usual. The speed of the rotor has no noticeable effect until the Taylor number reaches 10^5, and for $Ta_m = 10^6$ the friction is about 60–70% higher than that for pure axial motion. The slotted rotor leads to friction factors which are generally higher than those for the smooth rotor, but the curves exhibit the same trends. Again, the rotation does not have any definite effect on the friction until the Taylor number is increased beyond $Ta_m = 10^5$.

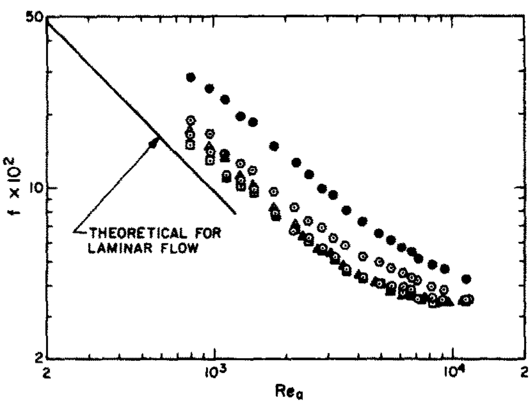

FIG. 10. Friction factor vs Reynolds number for the smooth rotor. ⊙, $Ta_m = 0$; ⊡, $Ta_m = 10^3$; △, $Ta_m = 10^4$; ⬭, $Ta_m = 10^5$; ●, $Ta_m = 10^6$.

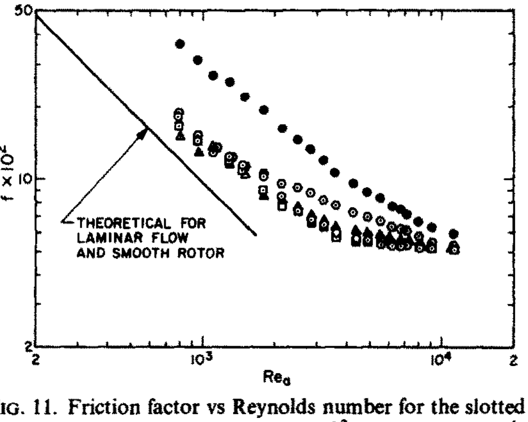

FIG. 11. Friction factor vs Reynolds number for the slotted rotor. ⊙, $Ta_m = 0$; ⊡, $Ta_m = 10^3$; △, $Ta_m = 10^4$; ⬭, $Ta_m = 10^5$; ●, $Ta_m = 10^6$.

The data for the frictional pressure loss in the axial direction have been given here as an aid in the design of a possible cooling system. The axial friction drop does not, of course, account for the entire power dissipated by friction as there is an important frictional torque on the rotor. No comparisons were made, therefore, between the heat transfer and the axial drop in friction as such comparisons were not believed to be meaningful.

CLOSING COMMENTS

The experiments which have been conducted provide information on the heat transfer in the annular space between a rotor and a stator in the presence of axial flow. The only previous work of this type was performed with air and the present investigation may be the first to provide data for Prandtl numbers higher than 1.0. For some test conditions it was also possible to extend the range of Taylor numbers beyond those previously investigated. The measurements with the slotted rotor provide some guidance as to the possible effect of the surface conditions of the rotor. The data show that the heat-transfer coefficients are generally increased as might be expected.

The feature that might deserve most emphasis has to do with the influence of various factors on the onset of the vortex motion. For the smooth rotor the Taylor number for the onset of vortex flow first increases with increasing axial flow, and after reaching a maximum seems to decrease slightly with further increases of the axial flow. This tendency towards decreasing critical Taylor numbers is very definite for the slotted rotor. Furthermore, it was also seen that free convection may have a strong influence on the transition point. As the onset of vortex flow has a major effect on the heat transfer, the point of onset plays an important role in predicting the heat transfer for the annular flow. It is hoped that the results of the present work may be of some help in predicting the performance for some specific application. In addition, and perhaps more importantly, they do stress the necessity of giving special attention to the flow transition when designing a cooling system for a rotor.

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TRANSFERT THERMIQUE DANS UN ESPACE ANNULAIRE

Résumé—On mesure les coefficients de transfert thermique dans un espace annulaire entre un cylindre tournant à l'intérieur d'un cylindre fixe, avec superposition d'un écoulement axial. Le problème a été stimulé par le désir d'une information supplémentaire en vue de la conception des systèmes de refroidissement des moteurs électriques à forte densité de puissance. Les vitesses de rotation sont telles qu'elles conduisent à des nombres de Taylor allant jusqu'à 10^6 et les nombres de Reynolds basés sur la vitesse axiale et la distance radiale entre cylindres varient jusqu'à 7000. Des expériences portent sur trois nombres de Prandtl différents 2,5, 4,5 et 6,5.

WÄRMEÜBERTRAGUNG IN EINEM RINGSPALT

Zusammenfassung—Für eine Strömung im Ringraum zwischen einem inneren rotierenden und einem äußeren stationären Zylinder mit überlagerter Axialströmung wurden Wärmeübertragungskoeffizienten gemessen. Das Problem wurde durch den Wunsch nach zusätzlicher Information zur Konstruktion von Kühlsystemen für Elektromotoren hoher Leistungsstärke aufgeworfen. Die Drehgeschwindigkeiten sollten Taylor–Zahlen bis zu 10^6 einschließen, und der Reynolds–Zahlen-Bereich auf der Basis der axialen Geschwindigkeitskomponenten und der Spaltbreite wurde auf 7000 ausgedehnt. Die Experimente wurden mit drei verschiedenen Prandtl–Zahlen 2,5; 4,5 und 6,5 durchgeführt.