Int. J. Heat Mass Transfer. Vol. 20, pp. 847–854. Pergamon Press 1977. Printed in Great Britain

HEAT TRANSFER IN THE ENTRANCE LENGTH OF A HORIZONTAL ROTATING TUBE

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(Received 5 May 1976 and in revised form 1 November 1976)

Abstract—An experimental study of heat transfer from steam condensing on the outside of a horizontal rotating tube through the tube wall to a laminar flow of cooling water on the inside of the tube is discussed. A brief description of the apparatus and data gathering technique are given. The data for zero rotation are compared to literature values to establish the credibility of the experimental results. The overall heat-transfer coefficient and a previously developed model for the steam-side coefficient are used to calculate the coolant-film coefficient at a variety of rotational speeds. The data are reduced to dimensionless form, and a general correlation is presented. For rotations up to 40 rev/min the cooling-side coefficient is slightly improved; however, at rotations above 40 rev/min a severe deterioration of the cooling-side coefficient is observed. A phenomena of fluid mechanics is postulated to account for the loss of heat transfer and supporting evidence from the fluid mechanics literature is brought forth.

NOMENCLATURE

c_p, heat capacity [J/kg K];
$h_{\ln}$, log mean cooling-side heat-transfer coefficient [W/m^2 K];
k, thermal conductivity [W/m K];
L, test section heat-transfer length [m];
T_1, inlet temperature [K];
T_2, outlet temperature [K];
T_w, wall temperature [K];
v, average axial velocity through tube [m/s];
w, mass rate of flow [kg/s];
ω, rotational speed [rad/s, s^{-1}];
μ, viscosity [Ns/m^2];
μ_w, viscosity in the film at the wall [Ns/m^2];
Γ, dimensionless swirl ratio, Re_R/Re_A;
ν, kinematic viscosity [m^2/s];
Re_A, axial Reynold's number, Dv/ν;
Re_R, rotational Reynold's number, $D^2\omega/\nu$;
Pr, Prandtl number, $c_p\mu/k$;
Gz, Graetz number, $Re_A \cdot Pr \cdot (2R_i/L)$;
$Nu_{\ln}$, logarithmic mean Nusselt number, $h_{\ln} D/k$.

INTRODUCTION

IN THE last decade interest in heat- and mass-transfer in or about rotating systems, such as disks and tubes, has increased greatly. Since heat- and mass-transfer are interrelated with fluid dynamics, it is necessary to understand this area as well because the resultant heat- and mass-transfer behavior is often controlled by the fluid dynamics of a system.

The problem of interest here is the transport of heat from the constant temperature wall of a horizontal tube rotating about its longitudinal axis to a laminar flow of cooling water through the inside of the tube. The entrance boundary condition is a uniform velocity profile in the fluid which emerges from a stationary tube and flows into a rotating tube. If the rotating tube is long enough, the uniform velocity profile will develop into a parabolic profile as described by Poiseiulle. In the case of a rotating tube the parabolic profile has superimposed upon it a solid body rotation at the end of the entrance length [1, 2].

Earlier workers studied associated problems for turbulent flows. Singer and Preckshot [3] studied the condensing-film coefficient on the outside of a rotating tube with a turbulent flow of cooling water on the inside of the tube. Cannon and Kays [4] studied the inside-film coefficient in a rotating tube with a turbulent flow of air as the moving-fluid medium.

The purpose of this study was twofold: (1) to provide some experimental data against which a theoretical numerical model could be verified, and (2) to develop a model for the fluid-mechanics and heat transfer of such a rotating flow. The remaining parts of this discussion are generally concerned with the first purpose; however, various portions of the model will be discussed where pertinent.

APPARATUS AND EXPERIMENTAL TECHNIQUE

The various components of the apparatus are shown in a flow diagram in Fig. 1. A boiler provided a constant supply of steam for condensation, and a secondary heat-transfer loop removed the heat from a recirculating supply of cooling water. Temperatures were measured with thermocouples and recorded using a potentiometer. Cooling water flow rates were measured utilizing an orifice and an electronic differential pressure cell. The ends of the rotating tube were

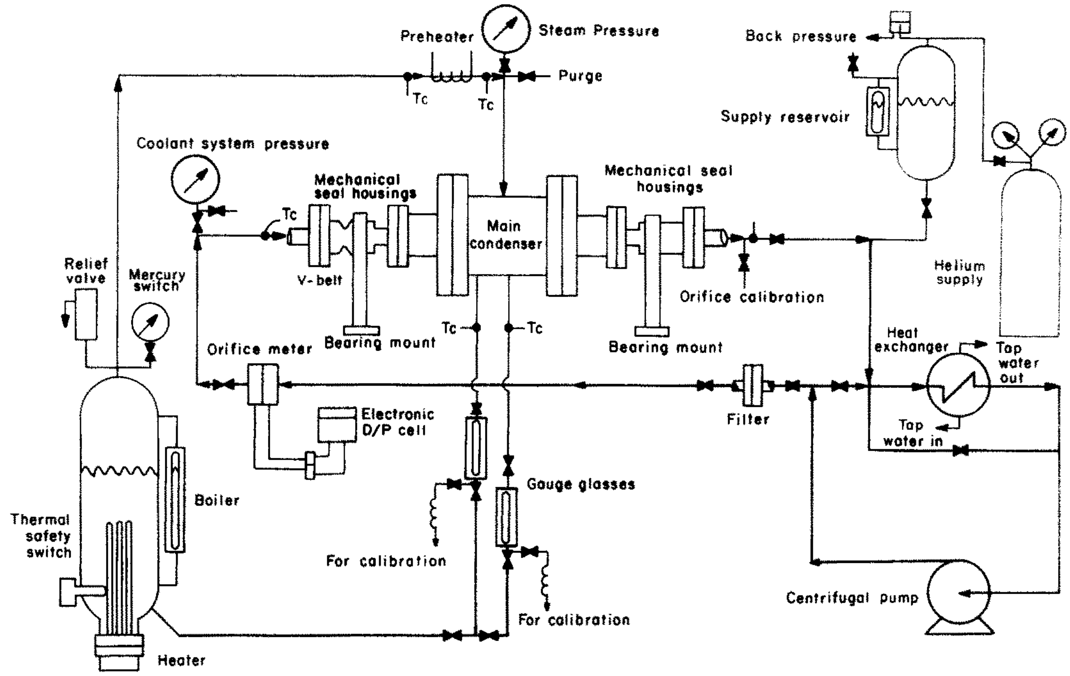

FIG. 1. Process flow sheet.

equipped with rotating mechanical shaft seals which kept the steam and cooling water isolated from the atmosphere. The rotating condenser tube (0.699 m long) was supported on both ends by precision ball bearings mounted in bearing supports. Rotation of the tube was accomplished by a system of electric motors and variable pitch sheaves. The drive mechanism provided nearly continuous variation of rotational speeds from 30 to 10 000 rev/min.

Of particular interest is the condensation chamber. The condensation chamber was constructed of a 0.305 m length of 0.203 m I.D. stainless-steel pipe. The end flanges were approx. 0.406 m O.D. and 0.025 m thick. The condenser tube was a red brass (85% copper, 15% zinc) tube with a 0.019 m I.D. and a 0.025 m O.D. Intermediate between the tube and the walls of the condensation chamber was the test section shroud. Figure 2 illustrates the relative positions of the various components. The condensation chamber had two condensate drains, one for the test section surrounded by the shroud and one for the length of tube extending on either side of the test section shroud. Since condensation near the end flanges was anticipated to occur at a different rate because of possible heat losses through the flanges, these end effects were eliminated by using the condensate rate from the test section only in the heat-transfer calculations. The shroud operated in an adiabatic fashion with no condensation occurring on the inside surface of the shroud. The condensate rate was obtained by a timed measurement in a calibrated gauge glass located below the condensation chamber.

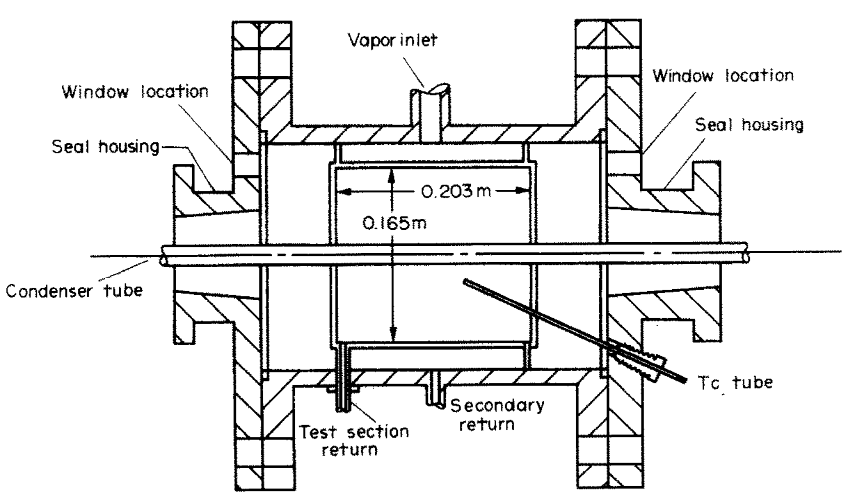

FIG. 2. Details of condensation chamber.

In preparation for an experiment the condensation chamber was first purged of non-condensables by alternately exposing the chamber to a supply of steam and then evacuating it using a vacuum pump. In any given individual experiment, a constant coolant flow rate was established and the inlet temperature was held as constant as possible. The condensate rate, inlet and outlet temperatures, and associated data were recorded, first at zero rev/min, and then at other rev/min. In general, this type of operation provided information at constant axial Reynold's numbers and constant Prandtl numbers. These dimensionless groups were evaluated at the bulk temperature of the fluid in the tube. The data were keypunched onto computer cards and the results systematically computed. The computer program had built into it temperature corrections for the orifice plates, condensation rate, and calibration corrections for thermocouples.

that they were supposed to follow that line, an explanation was sought.

The total length of the red brass condenser tube was 0.699 m; however, only 0.356 m of this tube were exposed to steam condensing on its outside surface. The thermal conductivity of this material is 159.3 W/m^2 K. Although the condenser and its associated hardware were heavily insulated to hold heat losses to a minimum, there was probably axial conduction of heat along the walls of the tube so that more than 0.356 m were effective in the heat-transfer process. Obviously, a 0.699 m length was not the effective length; however, this length was used in the calculations in Fig. 3 because the thermocouples were located at the ends of the rotating section of the tube. In order to test this hypothesis, the tube length was varied to see if a length between 0.356 and 0.699 m would allow the superposition of the data on the empirical line of Drew

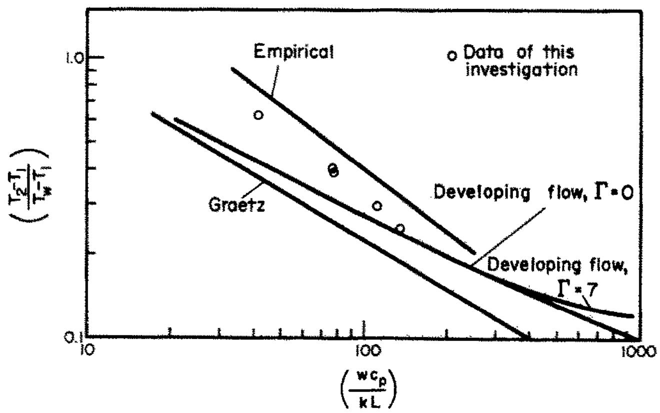

FIG. 3. Comparison of zero rotation data to existing theory.

EXPERIMENTAL RESULTS

Zero rotation

In order to ascertain the credibility of the apparatus for the determination of inside-film heat-transfer coefficients for the rotational configuration, data were taken at zero rotation and compared to the existing theory and correlations. Drew *et al.* [5] correlated the data of Holden and White for the dimensionless temperature rise of the cooling water $(T_2 - T_1)/(T_w - T_1)$ against the dimensionless Graetz number, wc_p/kL. The correlating line labeled as "empirical" is shown on Fig. 3. Also shown on this figure are: (1) Graetz's classical solution for a parabolic velocity profile for heat conduction only; (2) a numerical solution for heat transfer in a developing flow [2]; and (3) a numerical solution for heat transfer in a developing flow with rotation of the tube. The numerical solution is for a uniform velocity profile at the tube inlet developing to a parabolic profile at the end of the entrance length. The particular heat-transfer calculation shown in the figure for developing flow is for water entering the tube at 345.35 K and a tube wall temperature of 378.6 K.

One can see that the data do not fit exactly any of the correlative lines shown. They appear to follow the trend of the empirical line of Drew *et al.* [5]. Presuming *et al.* [5] in Fig. 3. A length of 0.508 m will cause the data of Fig. 3 to be shifted to the right so as to be coincidental with the empirical line. Therefore, an explanation for this deviation was found and the agreement of the data to an accepted correlation is considered adequate.

It should be mentioned that because of the heat leak due to axial conduction along the tube, all heat-transfer measurements were corrected by the subtraction of the heat loss determined at zero rotation of the tube and for zero flow rate of the cooling water. After applying the heat loss correction to the condensate rate the heat balance between the condensate and the cooling water generally agreed to within 5–20%. Heat balances for the lower rotation runs were always better than for the higher rotation runs.

With rotation

The preceding discussion demonstrates a method for checking the credibility of the heat-transfer measurements; however, because the rotating tube does not contain imbedded thermocouples in its wall, the inside-film coefficient had to be retrieved from the overall heat-flux measurement. A direct measure of the condensate rate was available from the shrouded test section, and from Singer's work [3] an expression for the condensing-film coefficient was available for the rev/min range 0 < rev/min < 260. The wall thickness and thermal conductivity of the rotating tube were known so the inside-film coefficient could be calculated. The nature of this calculation is iterative and, in the interest of brevity, is not included here. The reader may find the complete details of this calculation in reference [2]. Once the inside-film coefficient was determined, all of the dimensionless heat-transfer and fluid mechanics groups were evaluated.

flow the cooling-film coefficient is at least an order of magnitude smaller than the condensation-film coefficient. One ought to expect the heat transfer in this situation to be controlled by the cooling-film coefficient. The correlation for inside-film coefficients will be developed shortly.

In order to determine the precision of data reproducibility, four runs were replicated in an independent experiment. The maximum deviation in inside-film coefficients in any pair of replicates was 5.28%, and the average was 3.1%.

Table 1. Range of experimental conditions

Series	Runs	Average Re_A	Average Pr	ΔT_{in}	rev/min
1	42C–51B	1564	2.32	54.35	0–160
2	52C–59B	1531	2.36	55.00	0–100
3A	64B–70B	1583	2.31	54.43	0–98
3B	71B–77A	1041	3.70	105.67	0–100
4	78C–86B	2262	2.36	57.97	0–175
5	89B–102B	916	2.21	49.15	0–175
6	103B–117B	3880	2.32	57.26	0–173
7	118C–129B	4048	2.24	56.52	0–180
8A	130A–140B	3909	1.96	34.57	0–185
8B	141A–153B	3516	3.56	103.94	0–190
9	154A–157A	3903	1.96	35.49	0–74
10B	171A–183A	1699	3.77	110.05	0–155

The experimental data represent coolant-film heat-transfer results for laminar flows in a rotating condenser tube. The logarithmic mean Nusselt number is presented in correlations involving axial Reynold's number Re_A, rotational Reynold's number Re_R, and Prandtl number Pr. The Nusselt number varies from 3 to 37 for axial Reynold's numbers varying from 900 to 4300, rotational Reynold's numbers varying from 0 to 4×10^4 (0 < rev/min < 175), and Prandtl numbers varying from 1.9 to 3.8. Experimental conditions are shown in Table 1.

Typical of the results for these experiments are those shown in raw data form on Fig. 4. A curve has been approximated by eye through each set of points to illustrate the general trend of the data. The effect of increased rotation is distinctive and, at the time of the original investigation, unexpected. The overall effect of increasing rotation is to cause first a slight improvement in the overall heat-transfer, and secondly, a severe degradation at higher rotational speeds.

The cause of this degradation could be in either of two places; in the condensate film or in the cooling water film. It is true that as the tube begins to rotate with a condensate film accumulating on the outside of the tube, the film grows thicker until a balance between centrifugal and gravitational forces exists. This condition occurs at a Froude number ($\omega^2 R_0/g$) of unity. For the condenser tube used in this experiment (0.025 m O.D.) the Froude number becomes unity at 190 rev/min, but the degradation in heat transfer begins to occur at about 40–50 rev/min. In addition, in laminar

An error analysis was also performed to assess the error which could be expected in the calculated inside-film heat-transfer coefficient. This error estimate in the inside-film heat-transfer coefficient was calculated to be 19.1% [2]. The major contributor to this error was the error of 13.9% estimated for the condensation-film coefficient [3]

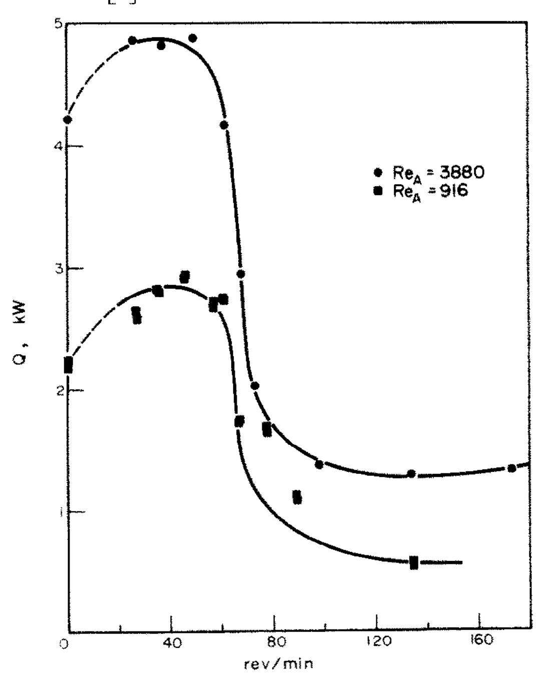

Fig. 4. Effect of rotation on heat transfer.

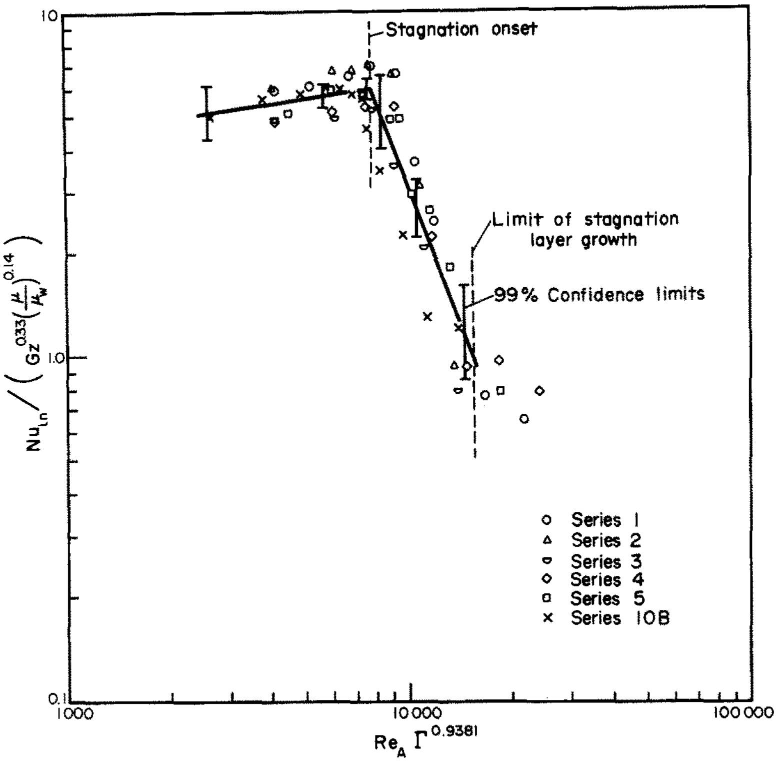

FIG. 5. Laminar flow heat transfer—a general correlation including rotational effects.

HEAT-TRANSFER CORRELATION AS FUNCTION OF ROTATION

The final correlation of the collant inside-film heat-transfer coefficient is illustrated in Fig. 5.

The ordinate of Fig. 5 is a logarithmic mean Nusselt number normalized using the dimensionless functionality of the RHS of an expression long accepted as a correlating equation for heat transfer in laminar flow at zero rotation.

$$Nu = 2N_{Gz}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}. \quad (1)$$

Although this expression is for an arithmetic averaged Nusselt number, it is essentially the same for a logarithmic averaged one, providing the temperature difference from one end of the tube is not less than one-half or greater than two times the temperature difference at the other end. In the case of these data, the prior constraint is met. The central idea was to plot an expression free from normal laminar flow dependences against a function of rotation so that only the rotational effect would be shown.

The function of rotation on the abscissa is quite another problem. Without this group the points at which the critical deterioration of the heat transfer occurs do not coincide along a horizontal scale. In fact, the central understanding of the fluid mechanics hinges on this point alone. Lavan *et al.* [1] have provided the explanation; however, in their fluid mechanics work the result is not nearly as dramatic as that reflected in these heat transfer results.

The theoretical study performed by Lavan *et al.* [1] utilized a numerical solution of the steady-state momentum, vorticity, and continuity equations to study developed flow emerging from: (1) a rotating tube and moving into a stationary one, and (2) a stationary tube and moving into a rotating one. Case 2, which is the problem of interest here, was solved for an already developed parabolic axial velocity profile present in the fluid as the boundary condition at the entrance to the rotating tube. As mentioned in purpose two of the introduction to this article, the original work [2] contained a numerical solution of the fluid mechanics and the heat transfer for a nearly identical problem patterned after Lavan's original work [7] with altered boundary conditions where necessary. Since the experimental boundary conditions, i.e. baffles, at the entrance of the rotating tube in this work presented a uniform velocity profile flow condition, the numerical solution modeled that condition instead of the condition used by Lavan *et al.* [1].

In both Cases 1 and 2 of the previous paragraph, Lavan *et al.* [2] found a very peculiar flow phenomena. In Case 2 the peculiarity manifested itself at specific ratios of angular to axial momentum, $\Gamma = Re_R/Re_A$, as a stagnation point near the tube wall in the proximity of the entrance to the rotating tube. The velocity and temperature profiles beyond this ratio were not able to be calculated because of the explicit nature of the finite difference mathematics. The derivative of the axial velocity profile takes on values of zero which causes instability in the numerical solution. Lavan *et al.* [7] have investigated the critical requirements of the formation of this condition. The complexities of the momentum-transfer and especially the centrifugal forces outward toward the wall cause a higher pressure near the wall than that which exists on the centerline of the tube at the same distance down the tube. This causes flow separation and associated stagnation at the wall. The stagnant film, of course, is the culprit in the deterioration of the heat transfer.

Figure 6 illustrates the combinations of swirl ratio, Γ, required for flow separation in the developed flow problem. The lower solid line and the curved, dashed line were taken from Lavan *et al.* [1]. The solid line is for an analytical solution and the curved line is for Lavan's numerical solution. At very low axial Reynold's numbers the analytical and numerical solutions are quite similar; however, as the axial Reynold's number increases the two techniques yield different results. At the top of Fig. 6 are two additional lines which correlate the data at the onset of stagnation, and the limit of the stagnation layer growth from all of the individual sets of our experimental data when graphed as log mean Nusselt number vs Γ on log–log coordinates. The boundary conditions on the velocity profile at the inlet to the rotating tube are different for the Lavan case and for the developing flow experiment described here; the data taken from the heat-transfer experiments nevertheless do seem to be in a logical region if a mechanism similar to that described by Lavan is assumed to be valid. The data are plotted in this fashion and subjected to a least squares analysis to develop an equation for the onset of stagnation.

$$Re_A \Gamma^{0.9381} = 7757. \qquad (2)$$

This dimensionless group was used to correlate the data in Fig. 5.

The inclusion of the four data points for axial Reynold's numbers greater than 2100 in the correlation for the onset of stagnation (on Fig. 6) may be somewhat questionable since the correlation is for laminar flow only. However, it does not appear that the removal of the four data points in question would affect the least squares regression to any great degree.

Since the data for the limit of growth of the stagnation layer appear to follow the same trend as the data for stagnation onset, a parallel line was drawn through the three data points which could definitely be considered to be representative of laminar-flow conditions. The following relationship describes the limit of stagnation-layer growth.

$$Re_A \Gamma^{0.9381} = 14\,925. \qquad (3)$$

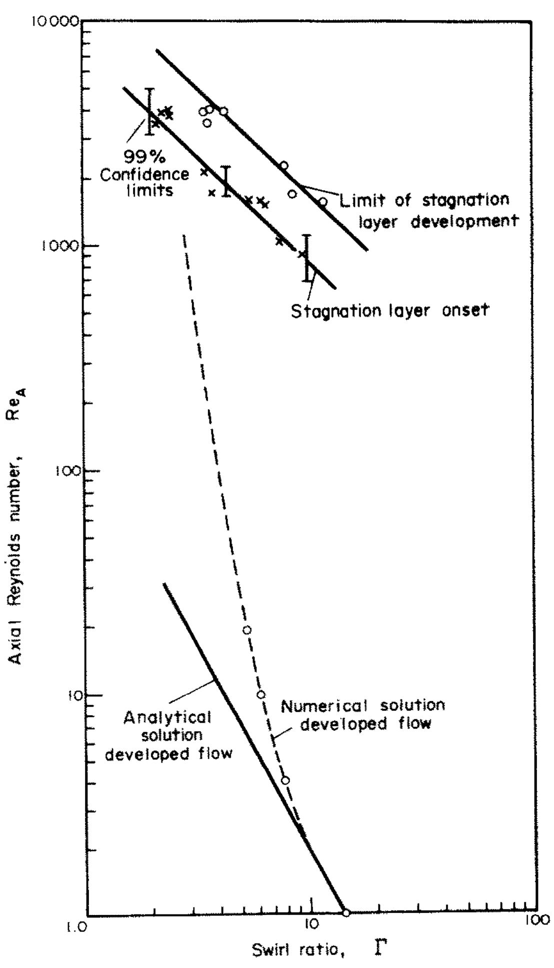

Fig. 6. Swirl ratios required for stagnation.

This growth limit relationship merely demonstrates that the stagnation layer cannot grow unfettered, i.e. the preservation of mass flow through the pipe eventually must prevail. A calculation of the order of magnitude of the thickness of this layer was made. The layer was considered as a fouling coefficient in the overall heat-transfer coefficient expression which would cause the degradation from the breakover point to the stagnation layer growth limit point on Fig. 5. Comparing the data from run 42C (zero rev/min) and run 47C (80 rev/min) in Series 1, on Table 1, a stagnant film of water 0.001 m thick was sufficient to cause the degradation.

The vertical dashed lines on Fig. 5 illustrate the onset of stagnation and the limit of stagnation layer growth.

Initially, at low rotational speeds, the heat transfer is enhanced 15 or 20%. This increase in heat transfer may be observed on Fig. 5 prior to the onset of stagnation. When the uniform axial velocity profile is used as the inlet boundary condition, it must be deformed to a parabolic profile at the end of the tube's entrance length. In non-rotational flow a radial circulation of fluid accomplishes this task, pumping fluid from the region of the tube near the wall toward the centerline of the pipe. In a rotating tube the same effect is evident but is enhanced by the additional momentum exchange provided by the angular rotation. Figure 7 illustrates this increased radial circulation effect for a cross-section of the tube at $Z^* = Z/D$ of 0.0225, and two swirl ratios, $\Gamma = 0$ and 7. Here $U^* = 2U/W_{avg}$ is the average of the axial velocity profile and $R^* = r/R$ where R is the inside tube radius. The improvement in heat transfer resulting from this radial convection may be observed on Fig. 3 for developing flow and $\Gamma = 7$.

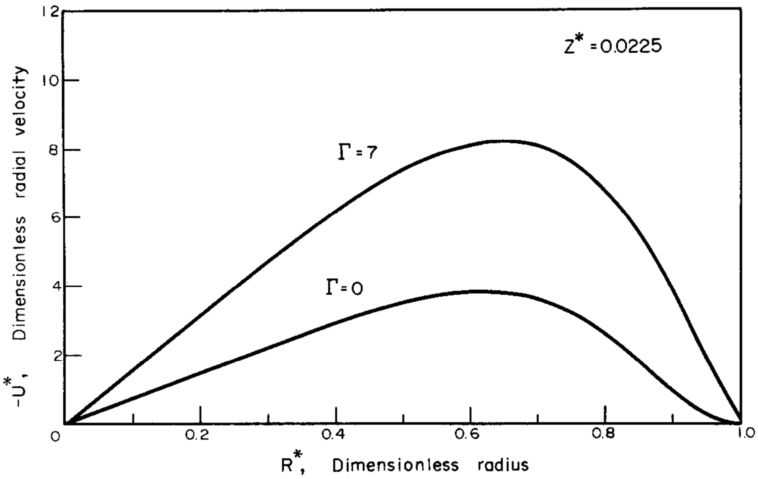

FIG. 7. Effect of tube rotation on radial convection.

FLOW STABILIZATION EFFECTS

Cannon and Kays [4] concluded that angular rotation stabilizes developing flow. They concluded that the transition axial Reynold's number increases upward from 2100 when angular rotation is superimposed. However, Nagib *et al.* [8] came to the opposite conclusion for developed flow. Nagib *et al.* [8] inserted thermistors in a fluid in a rotating tube and measured the root mean square of the fluctuating signal output from the thermistors. In addition, they offered dye streak data which, in general, supported the thermistor response data. The dye streak results suggested a somewhat higher transition axial Reynold's number than that from the thermistor response data. This is probably due to the thermistor probes causing some upset in the flow by their presence in the flow stream. Figure 8 shows some higher Reynold's number data. These data are graphed in the same fashion as those in Fig. 5; however, these data do not correlate using the same technique as the data which are unmistakably in the laminar flow region. Unfortunately, no data were taken between an axial Reynold's number of 2262 and 3516. It does appear that at $Re_A = 3516$ the flow is probably in transition since it does not correlate properly.

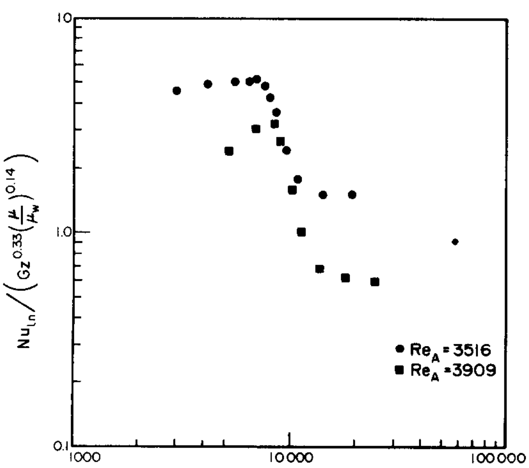

FIG. 8. Transition region heat transfer.

CONCLUSIONS

The following conclusions can be drawn:

(1) A general correlation has been developed for coolant inside-film heat-transfer coefficients for laminar flow in the developing flow region of a tube rotating about its longitudinal axis.

(2) At a critical swirl ratio, Γ, a stagnant film develops near the tube wall which causes severe degradation of the heat transfer in this rotating system. The critical ratio is described in this system by the relationship:

$$Re_A \Gamma^{0.9381} = 7757.$$

(3) The stagnant film cannot grow in an unrestricted fashion. This limit is described by a second relationship:

$$Re_A \Gamma^{0.9381} = 14925.$$

(4) The stabilizing influence of rotation on developing flow is difficult to ascertain from these results; however, the lack of a definitive conclusion is probably due to the fact that relatively low rotations were imposed on the tube in these experiments.

Acknowledgement—The authors are grateful to The National Science Foundation, the Shell Oil Company, Texaco Inc., and the American Chemical Society for supporting the junior author through various portions of this work.

REFERENCES

1. Z. Lavan, H. Nielsen and A. A. Fejer, Separation and flow reversal in swirling flows in circular ducts, *Physics Fluids* **12**(9), 1747 (September 1969).
2. J. E. McElhiney, Heat transfer to a flow in a rotating tube, Ph.D. Dissertation, University of Missouri (Columbia) (August 1969).
3. R. M. Singer and G. W. Preckshot, The condensation of vapor on a horizontal rotating cylinder, in *Proceedings of the* 1963 *Heat Transfer and Fluid Mechanics Institute*. Stanford University Press, Palo Alto (1963).

4. J. N. Cannon and W. M. Kays, Heat transfer to a fluid flowing inside a pipe rotating about its longitudinal axis, *J. Heat Transfer* **91C**, 135 (1969).
5. T. B. Drew, J. J. Hogan and W. H. McAdams, Heat transfer in streamline flow, *Ind. Engng Chem.* **23**, 936 (1931).
6. Z. Lavan and A. A. Fejer, Investigation of swirling flows in ducts, Defense Documentation Center Report AD 639 325, Cameron Station, Alexandria, VA (1966).
7. H. M. Nagib, Z. Lavan, A. A. Fejer and L. Wolf, Jr., Stability of pipe flow with superposed solid body rotation, *Physics Fluids* **14**, 766 (1971).

TRANSFERT THERMIQUE A L'ENTREE D'UN TUBE HORIZONTAL ET EN ROTATION

Résumé—On analyse une étude expérimentale de la condensation de vapeur d'eau à l'extérieur d'un tube horizontal en rotation et du transfert de chaleur à travers la paroi vers un écoulement laminaire d'eau froide à l'intérieur du tube. On donne une brève description du montage et de la technique de mesure. Pour juger de la valeur des résultats expérimentaux, les mesures pour une rotation nulle sont comparées à des données connues. On utilise le coefficient global de transfert et un modèle donné précédemment pour le coefficient de transfert côté vapeur afin de calculer le coefficient de convection interne à différentes vitesses de rotation. Pour des rotations inférieures à 40 tours/min, le coefficient de convection interne est légèrement augmenté; on observe une forte détérioration du coefficient pour des rotations supérieures à 40 tours/min. On postule un phénomène de mécanique des fluides pour tenir compte de la chute du transfert thermique et on produit une preuve tirée de documents relatifs à la mécanique des fluides.

DER WÄRMEÜBERGANG IM EINLAUFBEREICH EINES HORIZONTALEN, ROTIERENDEN ROHRES

Zusammenfassung—Es wird über eine experimentelle Untersuchung des Wärmeübergangs von an der Außenseite eines horizontalen, rotierenden Rohres kondensierendem Wasserdampf an im Innern laminar strömendes Kühlwasser berichtet. Die Versuchs- und die Datenerfassungsanlage werden kurz beschrieben. Zur Überprüfung der Meßgenauigkeit werden die Resultate für das nicht rotierende Rohr mit Angaben aus der Literatur verglichen. Aus den Wärmedurchgangskoeffizienten und einem vorher entwickelten Modell zur Berechnung des Wärmeübergangskoeffizienten auf der Kondensatseite wird der Wärmeübergangskoeffizient auf der Kühlwasserseite bei verschiedenen Rotationsgeschwindigkeiten berechnet. Die Daten werden in eine dimensionslose Form gebracht und es wird eine allgemeine Korrelation angegeben. Eine Steigerung der Drehzahlen bis zu $40\,\mathrm{min}^{-1}$ erbringt eine geringfügige Verbesserung des Wärmeübergangskoeffizienten auf der Kühlwasserseite; bei höheren Drehzahlen wird eine starke Beeinträchtigung des Wärmeübergangs beobachtet. Zur Erklärung dieser Verminderung des Wärmeübergangs wird ein strömungsmechanisches Phänomen postuliert, für das Beweise aus der Strömungsliteratur angeführt werden.

ТЕПЛООБМЕН НА НАЧАЛЬНОМ УЧАСТКЕ ГОРИЗОНТАЛЬНО РАСПОЛОЖЕННОЙ ВРАЩАЮЩЕЙСЯ ТРУБЫ

Аннотация — Рассматриваются результаты экспериментального исследования переноса тепла от пара, конденсирующегося на внешней стороне горизонтальной вращающейся трубы, через стенку трубы в ламинарный поток находящейся в ней охлаждающей воды. Приводится краткое описание экспериментального стенда и методики получения данных. С целью установления надежности экспериментальных результатов проведено сравнение данных, полученных при отсутствии вращения, с опубликованными данными. Общий коэффициент теплообмена и ранее разработанная модель переноса тепла со стороны пара используется для расчета коэффициента теплообмена пленки охлаждающей жидкости при различных скоростях вращения. Данные приводятся к безразмерному виду, и дается общее корреляционное соотношение. Постулируется положение из механики жидкости, объясняющее уменьшение (спад) интенсивности теплообмена и приводится в его пользу доказательство из литературы.