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Laminar heat transfer in the oil groove of a wet clutch

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Abstract—Numerical results for the analysis of heat transfer to laminar flow in radial grooves of wet clutch friction faces are presented. It is shown that a thin oil film intervening between the friction face and the metal separator plate has an important effect on the overall heat transfer coefficient between the metal plate and the oil flowing in the groove. Experiments carried out under continuously slipping conditions on an actual clutch are described and the measured overall heat transfer coefficients are compared to the results of the numerical analysis. The agreement between the numerical and the experimental results is within an r.m.s. deviation of 10.7%. The experimental error is estimated to be 10.1%. The good agreement allows the use of numerically calculated heat transfer coefficients for a complete thermal analysis of the wet clutch.

INTRODUCTION

IN THE OPERATION of an oil-cooled (wet) clutch, two rather distinct modes may be identified. In the engagement mode, the grooved friction plate and the metal separator plate slip with respect to each other for very short time intervals, typically from 0.2 to 2 s. The frictional heat generated during the engagement period is mostly absorbed by the metal separator plate with the oil flowing through the grooves playing a secondary role in heat removal. In the continuously slipping mode, the two plates may slip for periods of time as long as 10–20 s. In this latter mode, the temperatures at various points in the plates reach final steady state values in a few seconds, after which all of the generated heat must be removed by the oil flowing in the grooves of the friction plate. The value of the heat transfer coefficient between the metal separator plate and the groove oil will then determine the steady state temperature level in the clutch. Of the many groove patterns commonly used, the present study focuses on the radial groove pattern because of its simplicity.

Figure 1(a) shows a sector of a radially-grooved composite friction plate with two neighboring grooves and the land area between them where frictional heat is generated. Figure 1(b) shows a cross-sectional view of the metal separator plate and the friction plate, as well as their relative position during the continuously slipping mode. In particular, a thin film of oil is shown separating the two plates. The ratio of the film thickness to groove depth is generally less than 0.01. However, it will be shown in the following sections, that this film plays an important role in determining the heat transfer coefficient between the separator plate and the groove oil because of its screening action. Figure 2(b) shows the recirculating pattern in the groove. The top streamline is a dividing streamline separating the oil flowing in the grooves from the oil flowing in the film. The velocity distribution in the film over the land area preceding a groove corresponds to Couette flow. In the region directly above the dividing streamline, however, the velocity of the film oil is essentially uniform. The thickness of the film above the dividing streamline is, therefore, half of the film separating the two plates. Figure 2(a) shows a single groove with the top wall in motion and the other three walls at rest. For clarity, the film is not shown in Fig. 2(a). Cool oil enters the groove at the inside diameter and leaves at the outside diameter of the friction plate. Individual fluid particles follow paths similar to a helical path. The intense cross-stream recirculating flow enhances the heat transfer coefficient greatly as compared to the case in which the top wall is at rest.

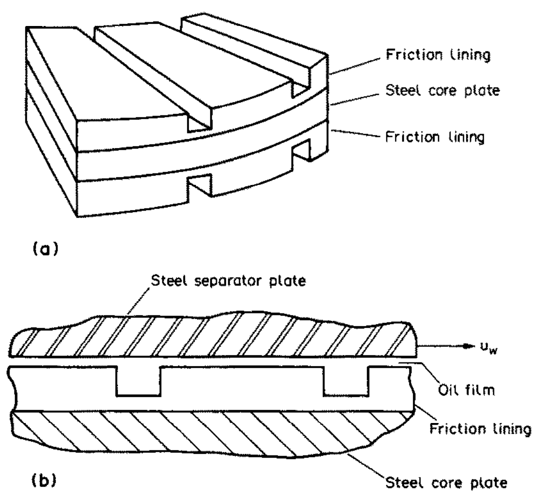

FIG. 1. (a) A segment of a wet clutch friction plate. (b) Cross section showing relative position of friction and separator plates.

NOMENCLATURE

A	aspect ratio of the groove cross section, d/L
b	thickness of film between the dividing streamline and the moving wall
c	specific heat at constant pressure
c_o	specific heat of oil
D_i	inside diameter of friction plate
D_o	outside diameter of friction plate
d	depth of a groove
f_1	function defined by equation (19)
f_2	function defined by equation (22)
h_o	local overall coefficient of heat transfer from the moving wall to the groove oil
h_{rz}	local recirculation zone heat transfer coefficient
k	thermal conductivity of oil
L	width of a groove
$\dot{m}_{go}$	mass flow rate of oil in a single groove
$\dot{m}_o$	total mass flow rate of oil through clutch
N_g	number of grooves on one friction face
$\overline{Nu}_g$	mean groove Nusselt number
Nu_{rz}	local recirculation zone Nusselt number
n	distance normal to a wall
p	pressure
Pr	Prandtl number
Re_w	Reynolds number based on wall velocity and groove width
r	radial distance from center of clutch plate
r_i	inside radius of friction plate
r_o	outside radius of friction plate
T	temperature
T_b	bulk temperature of groove oil
T_{ds}	temperature along the dividing streamline
T_{dsa}	average temperature along the dividing streamline
$(\Delta T)_{LMTD}$	logarithmic mean temperature difference between the moving wall and the groove oil
u	velocity component in the x-direction
u_w	velocity of the moving wall
v	velocity component in the y-direction
w	velocity component in the z-direction
x	coordinate along width of groove
y	coordinate along depth of groove
z	coordinate along length of groove
z_{max}	length of groove.

Greek symbols

μ	dynamic viscosity
ν	kinematic viscosity
ρ	oil density
τ	torque exerted by friction plate
ω	angular velocity.

Subscripts

a	average
b	bulk
ds	dividing streamline
g	groove
i	inside
LMTD	logarithmic mean temperature difference
max	maximum
o	outside
oi	oil temperature at groove inlet
oo	oil temperature at groove outlet
rz	recirculation zone
w	moving wall
wi	wall temperature at inlet
wo	wall temperature at outlet.

Superscript

$\overline{}$	integrated value over the groove length.

The analysis of laminar flow and heat transfer in the system shown in Figs. 2(a) and (b) is the main focus of the present paper.

Patankar and Spalding [1] studied the problem of developing laminar flow and heat transfer in a square duct with no film separating the moving wall from the cavity. The main emphasis of their paper was to present a general method of solution of parabolic flows so that they present only a limited number of results for the square duct problem. Fully developed laminar flow and heat transfer for a square duct with one moving wall was studied by Abdel-Wahed *et al.* [2]. They presented fully-developed Nusselt numbers for $Pr = 0.7$ and 5 for a range of moving wall Reynolds numbers from 0 to 1000. Their results are for the case in which all four walls were maintained at a uniform temperature. One of the important findings of their paper was that for $Pr = 5$ the Nusselt number increases with the moving wall Reynolds number and is larger than that of a duct with stationary walls by a factor of about 5 at $Re_w = 1000$. This is due to the scrubbing action of the recirculating flow. In the present study, Reynolds numbers up to 2000, and Prandtl numbers between 40 and 600 are considered. In addition, the thermal boundary conditions of three insulated stationary walls and one moving wall with specified temperature are used. An approximate method to account for the effect of the thin oil film separating the moving wall from the recirculating zone is introduced. It will be shown that the calculated

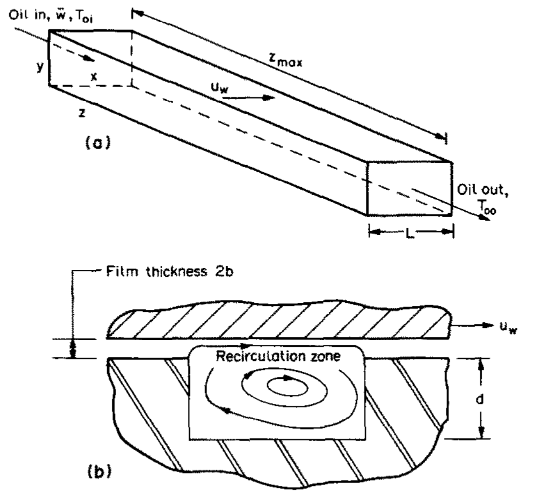

FIG. 2. (a) Geometry and coordinate system for a single groove. (b) Flow pattern and the relative position of film and recirculation zone.

heat transfer coefficients are in good agreement with experimental results obtained on an actual wet clutch operating under continuously slipping conditions.

EXPERIMENTAL DETERMINATION OF THE AVERAGE GROOVE NUSSELT NUMBER

Figure 3 shows a sketch of the main parts of the clutch test apparatus. The experimental set-up consisted of a radially-grooved composite friction plate, such as that shown in Fig. 1(a), sandwiched between two steel end plates. The friction plate had 72 equally spaced radial grooves per face. In each test run, the friction plate was set in motion at a constant angular velocity while the applied pressure forced the plates together to produce the desired torque. Oil was forced into the clutch cavity and after flowing through the 144 grooves of the composite plate exited at the outside diameter. Thermocouples were installed at the inside and outside diameters of the steel end plate with the junction very near the interface between that plate and the friction faces of the composite plate. Similarly, thermocouples were used to measure inlet and outlet temperature of the oil. For each test run, torque, angular velocity, oil flow rate, interface temperatures at i.d. and o.d. of the steel plate, and the oil inlet and outlet temperatures were measured after steady state thermal conditions were established. An energy balance yields

$$\tau\omega = m_o c_o (T_{oo} - T_{oi}). \tag{1}$$

All of the test runs used for comparison with the prediction of the theory described in the next section satisfied the energy balance indicated by equation (1) to within 5%. The relevant dimensions of the plates and the grooves were as follows:

$$D_o = 0.24\,\mathrm{m},\quad D_i = 0.16\,\mathrm{m},\quad L = 1.2\times 10^{-3}\,\mathrm{m}$$

$$d = 6\times 10^{-4}\,\mathrm{m},\quad z_{max} = 0.5(D_o - D_i) = 0.04\,\mathrm{m},$$

$$N_g = 72.$$

The oil flow rate for all tests was about 0.088 kg s^{-1}.

If we assume that the torque is due to viscous forces in a thin film separating the plates, the following equation can be used to calculate the film thickness b:

$$b = \frac{\omega}{\tau}\int_{r_i}^{r_o} \mu(T_w)(2\pi r - N_g L) r^2\,\mathrm{d}r. \tag{2}$$

In evaluating the integral in equation (2), the whole film is assumed to be at the separator plate temperature at that radial location. A linear distribution of T_w from T_{wi} to T_{wo} and the known viscosity temperature relation of the oil then allows evaluation of the integral and the film thickness b. Measurement of oil temperature at the inlet and outlet of the grooves made it possible to check the consistency of the data by performing an energy balance on the clutch plates and the oil rather than on the system within the housing. An external oil cooler was used to control the inlet temperature of the oil.

In the next section, a mathematical model capturing the essential elements of the heat transfer mechanisms

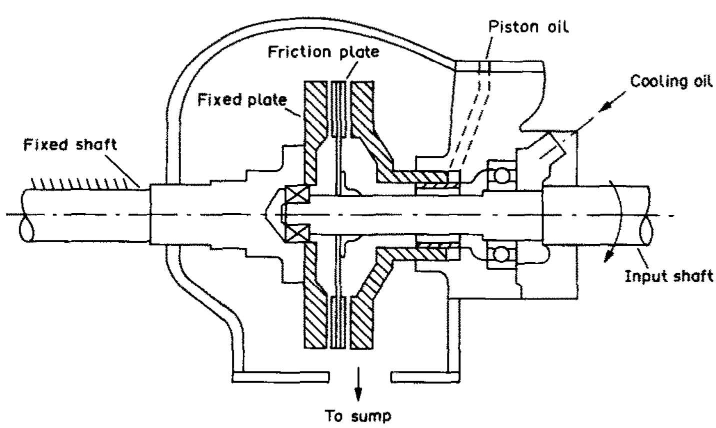

FIG. 3. Schematic diagram of wet clutch test apparatus.

of the friction pair used in the experiments is described. Following the mathematical analysis of the system, results of the numerical calculations are presented and the predicted and experimental groove heat transfer coefficients are compared.

MATHEMATICAL ANALYSIS

Before introducing the mathematical equations on which the analysis of the groove heat transfer problem is based, a description of certain simplifying assumptions will be given. The operating conditions of the clutch are specified in terms of torque τ, the angular slip velocity ω, the inlet oil temperature and flow rate, the physical properties of the oil and the clutch plates, and the geometric dimensions of the various components of the clutch. For a given set of operating conditions, the velocity of the moving wall varies linearly from $z = 0$ to z_{max} (see Fig. 2(a))

$$u_w = (r_i + z)\omega. \tag{3}$$

Similarly, the temperature of the moving wall will not be uniform along the length of the groove. It will have a distribution determined by a balance between the convective heat transfer to the groove oil and conduction within the separator plate. Because of the strong dependence of the oil viscosity on temperature and because, both u_w and T_w vary in the flow direction, the problem of groove flow and heat transfer is a fully three-dimensional parabolic type problem. This situation is further complicated by the existence of the thin oil film which must be included in the analysis. The solution of the groove heat transfer problem is only one step in an iterative process which includes the solution of the conduction equation in the solid plates of the clutch. If at every step of this iterative scheme one has to solve the fully three-dimensional groove problem, the amount of computations involved would become impractically larger. Even if one obtains such a solution, it would only be applicable to a given clutch with a specific set of operating conditions. What is needed is a solution of the groove oil heat transfer problem obtained independently from any specific operating conditions and cast in the form of a generally applicable correlation. Once such a correlation is available, it can be used in the appropriate step of an iterative scheme for the complete thermal analysis of the clutch.

The above discussion is the basis for making the following assumptions in the mathematical analysis of laminar flow and heat transfer in a single groove:

(1) Fully developed solutions for constant moving wall velocity and temperature, u_w and T_w, respectively, can be applied locally from $z = 0$ to z_{max} when u_w and T_w are variable. Properties will be assumed constant for each fully developed solution.

(2) The heat transfer coefficient for the transfer of heat across the dividing streamline is the same as that for a groove with zero film thickness.

(3) The resistance of the film intervening between the moving plate and the dividing streamline can be calculated separately and added to the resistance due to transfer of heat across the dividing streamline to obtain the overall resistance.

(4) For the calculation of the film resistance, the velocity in the x-direction in the film will be assumed uniform and equal to the local value of u_w. The temperature in the oil film at $x = 0$ will be assumed uniform and equal to the local value of T_w.

The above assumptions reduce the amount of computations by orders of magnitude. First, the three-dimensional flow will be reduced to a series of two-dimensional flows. Second, the time consuming velocity distribution calculation will be done only once in arriving at a groove heat transfer correlation for various Prandtl numbers.

The governing equations to be solved are as follows:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{4}$$

$$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{5}$$

$$\rho\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial p}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) \tag{6}$$

$$\text{at } x = 0 \quad u = 0, \quad v = 0, \quad 0 < y < d \tag{7}$$

$$\text{at } x = L \quad u = 0, \quad v = 0, \quad 0 < y < d \tag{8}$$

$$\text{at } y = 0 \quad u = 0, \quad v = 0, \quad 0 \leqslant x \leqslant L \tag{9}$$

$$\text{at } y = d \quad u = u_w, \quad v = 0, \quad 0 \leqslant x \leqslant L. \tag{10}$$

Define Re_w and A as follows:

$$Re_w = u_w L/\nu \tag{11}$$

$$A = d/L. \tag{12}$$

For each set of values of Re_w and A, u/u_w and v/u_w can be obtained by a numerical solution of equations (4)–(10) using the numerical scheme described in detail by Patankar [3].

Once the solution for u and v has been obtained, the fully developed distribution of the velocity w can be obtained from the following equation:

$$\rho\left(u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y}\right) = -\frac{\partial p}{\partial z} + \mu\left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2 w}{\partial y^2}\right) \tag{13}$$

$$w = 0 \text{ on all four walls.} \tag{14}$$

For the solution of the temperature equation, we assume that the fully developed velocity distribution defined by the functions $u(x, y)$, $v(x, y)$, and $w(x, y)$ is established at $z = 0$ and remains unchanged for values of $z > 0$. The temperature distribution is then obtained from the following equations:

$$\rho c\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z}\right) = k\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{15}$$

$$\text{at } y = d, \quad T = T_w \quad \text{for } 0 \leqslant x \leqslant L \tag{16}$$

$$\frac{\partial T}{\partial n} = 0 \text{ on the other three walls} \tag{17}$$

$$\text{at } z = 0 \quad T = T_{oi} \quad \text{for } 0 < x < L, \quad 0 < y < d. \tag{18}$$

The axial conduction term has been neglected in equation (15). In the solution of equations (15)–(18), an additional parameter, the Prandtl number $Pr = \mu c/k$, is introduced. Using the method described by Patankar [3], the solution is obtained in the form

$$\frac{T - T_{oi}}{T_w - T_{oi}} = f_1\left(\frac{x}{L}, \frac{y}{L}, \frac{z}{L}, Re_w, A, Pr\right). \tag{19}$$

The bulk oil temperature is now calculated from

$$T_b = \frac{\int_0^d \int_0^L wT \, dx \, dy}{\int_0^d \int_0^L w \, dx \, dy}. \tag{20}$$

For zero film thickness, define the groove Nusselt number as follows:

$$Nu_{rz} = \frac{\int_0^L \left.\frac{\partial T}{\partial y}\right|_{y=d} dx}{(T_w - T_b)}. \tag{21}$$

It was found during the course of the present study that for $Re_w > 200$, Nu_{rz} reaches its final fully developed value to within 5% after $z/L > 1$. This fully developed value depends only on the three parameters Re_w, A, and Pr

$$Nu_{rz} = f_2(Re_w, A, Pr). \tag{22}$$

The function defined by equation (22) need be calculated only once. Assuming that Nu_{rz} is already available, the film resistance is accounted for as described below. Figure 4 shows the geometry and the coordinate system for the film. The energy equation for the film subject to our stated assumptions is

$$\rho c u_w \frac{\partial T}{\partial x} = k \frac{\partial^2 T}{\partial y^2} \tag{23}$$

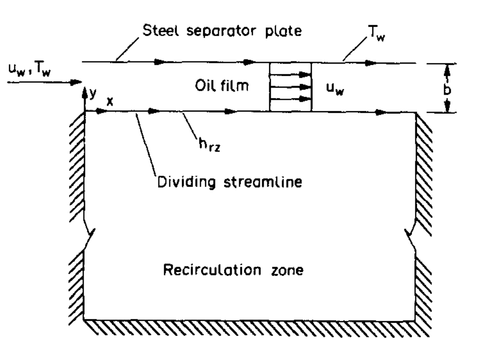

FIG. 4. Geometry and coordinate system for analysis of the film.

$$\text{at } x = 0 \quad T = T_w \quad \text{for } 0 \leqslant y \leqslant b \tag{24}$$

$$\text{at } y = 0 \quad k\frac{\partial T}{\partial y} = h_{rz}(T_{ds} - T_b) \tag{25}$$

$$\text{at } y = b \quad T = T_w \quad \text{for } 0 \leqslant x \leqslant L. \tag{26}$$

For constant heat transfer coefficient along the dividing streamline, an analytical solution is possible for the set of equations (23)–(26). This solution is given by Carslaw and Jaeger [4]. However, it was found more practical to solve this set of equations numerically. The value of the heat transfer coefficient h_{rz} is already defined by the given parameters Re_w, A, and Pr. The numerical solution of the set of equations (23)–(26) determines the variation of T_{ds} along the dividing streamline. The overall heat transfer coefficient from the moving wall to the oil flowing in the groove including the resistance of the film is now obtained from the equation

$$h_o = h_{rz}(T_{dsa} - T_b)/(T_w - T_b) \tag{27}$$

where T_{dsa}, the average temperature along the dividing streamline is defined as

$$T_{dsa} = \frac{1}{L}\int_0^L T_{ds} \, dx. \tag{28}$$

If one applies the heat transfer coefficient described by equation (27) locally from $z = 0$ to z_{max}, the bulk oil temperature will be obtained from the equation

$$\dot{m}_{go} c_o \frac{dT_b}{dz} = L h_o(z)[T_w(z) - T_b(z)]. \tag{29}$$

Equation (29) allows one to calculate the variation of the oil bulk temperature $T_b(z)$ from $z = 0$ to z_{max}. The average Nusselt number for the whole groove is now obtained from

$$\overline{Nu}_g = \frac{m_{go} C_o (T_{oo} - T_{oi})}{(\Delta T)_{LMTD} k z_{max}} \tag{30}$$

where

$$(\Delta T)_{LMTD} = \frac{(T_{wo} - T_{oo}) - (T_{wi} - T_{oi})}{\ln \frac{T_{wo} - T_{oo}}{T_{wi} - T_{oi}}}. \tag{31}$$

Equations (30) and (31) define the average groove Nusselt number, $\overline{Nu}_g$, in terms of quantities directly measured in the experimental work described in the previous section. After some numerical experimentation with fine and coarse as well as uniform and non-uniform grids, a 20×24 non-uniform grid was used to obtain the results reported in the next section.

RESULTS

From the experimentally measured total oil flow rate, the flow rate for a single groove is obtained from

$$\dot{m}_{go} = \dot{m}_o/(2N_g). \tag{32}$$

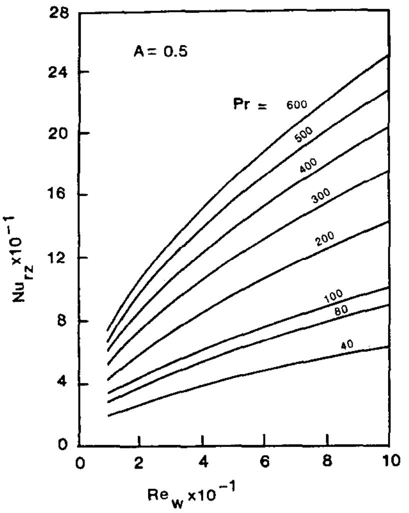

FIG. 5. Variation of recirculation zone Nusselt number with Re_w and Pr.

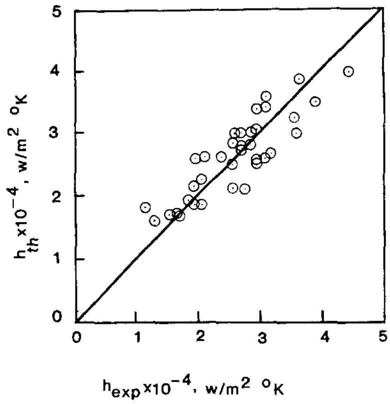

FIG. 7. Comparison of theoretical and experimental average groove heat transfer coefficients including the film resistance.

Equation (30), being expressed in terms of experimentally measured or known quantities, is then used to calculate the experimental value of $\overline{Nu}_g$.

In order to calculate the predicted value of $\overline{Nu}_g$, first the film thickness b is computed using equation (2). Once this film thickness is known, the methods described previously are used to calculate the average groove Nusselt number. Figures 5 and 6 show the fully developed recirculation zone Nusselt numbers, Nu_{rz}, as a function of Re_w and Pr for an aspect ratio of 0.5. Figure 7 shows a comparison of the theoretical and experimental overall groove heat transfer coefficients. The theoretical value is calculated for the set of operating conditions in the corresponding experimental run. The experimental value, h_{exp}, was calculated by using equation (30) which is expressed in terms of directly measured quantities. The agreement between the theoretical and experimental values is within an r.m.s. deviation of $\pm 10\%$. The error in the experimental values was estimated to be about $\pm 10\%$. The agreement between the values predicted by the method described in this paper and the experimentally measured values is, therefore, within the accuracy of the experimental data.

Figure 8 shows a comparison of theoretical and experimental overall groove heat transfer coefficients if the film resistance were neglected in the theoretical calculations. The resulting theoretical coefficients are observed to be up to 130% above the experimental values. The screening effect of the film separates the hot moving wall from the cool groove oil and must, therefore, be included in a realistic analysis of the groove heat transfer problem. Similar phenomena might occur in heat transfer in grooves used for lubrication.

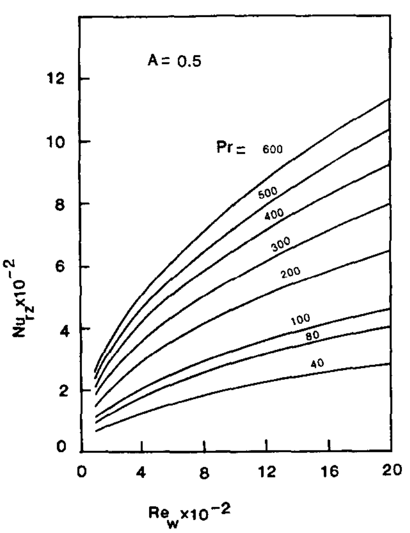

FIG. 6. Variation of recirculation zone Nusselt number with Re_w and Pr.

DISCUSSION AND CONCLUSIONS

The objective of the research described in this paper was to develop a rational method of estimating heat

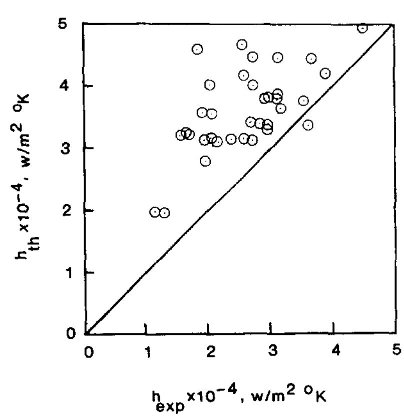

FIG. 8. Comparison of theoretical and experimental average groove heat transfer coefficients neglecting the film resistance.

transfer coefficients for the flow of heat from metal separator plates to the oil flowing in radial grooves of a paper-based friction plate. Certain simplifying assumptions were made deliberately in order to make the calculation of the heat transfer coefficients independent of the operating conditions of the clutch. In particular, using fully developed heat transfer coefficients on a local basis, makes the resulting heat transfer coefficients independent of the oil flow rate. The experiments designed to verify the mathematical model were described in a previous section of this paper. Since in the experiments the grooved friction plate was rotating and the metal separator plate was stationary, a question may arise whether neglecting the Coriolis forces would introduce significant errors in the numerically calculated results. The magnitude of the Coriolis force is proportional to the ratio of the groove width to the radial distance of the cross section under consideration from the center of the clutch plate. For the clutch plate used in this study, this ratio was 0.0125 resulting in a small effect due to Coriolis forces. To verify this assumption, calculations were made with and without the Coriolis force term at $Re_w = 2000$ and 100 at $Pr = 5$ and 500. The greatest effect of the Coriolis force was found to occur at the lower Prandtl number and was about 1.7%. Furthermore, inclusion of the Coriolis forces makes the results dependent on the magnitude of the oil flow rate. For typical clutch diameter, groove dimensions, and operating conditions, the Coriolis forces may be safely neglected in calculating the oil heat transfer coefficient. For plates of small diameter with large groove dimensions, these forces may have to be included.

Another observation made during the course of the present study, was that oil films encountered in clutch engagement are so thin that their thermal inertia may be neglected without introducing appreciable error. For the film thicknesses encountered in the present study, for example, simple addition of the conductive resistance of the film to $1/h_{rz}$ yields overall resistances within 0.5% of those calculated by the more accurate method presented in this paper.

In summary, the following conclusions may be drawn from the present study:

(1) Fully developed Nusselt numbers obtained for uniform velocity and temperature of the moving wall can be applied on a local basis to analyze heat transfer in a groove where the moving wall velocity and temperature vary along the length of the groove.

(2) The fully developed Nusselt numbers for the case where film thickness is zero can be combined with the film resistance to predict the overall groove heat transfer coefficient.

(3) The resistance of the film in many cases of practical interest is significant and cannot be neglected.

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TRANSFERT DE CHALEUR LAMINAIRE DANS LES RAINURES D'UN EMBRAYAGE MOUILLE A L'HUILE

Résumé—On présente les résultats de l'analyse du transfert thermique par écoulement dans les rainures radiales des faces de frottement d'un embrayage mouillé. On montre qu'un mince film d'huile intervenant entre la face frottante et le plateau métallique séparateur a un effet important sur le coefficient global de transfert thermique entre le plateau et l'huile s'écoulant dans la rainure. Des expériences avec des conditions de glissement continu sur un embrayage sont décrites et les coefficients de transfert thermique mesurés sont comparés aux résultats de l'analyse numérique. L'accord entre les résultats numériques et expérimentaux est à l'intérieur d'une déviation de 10,7%. L'erreur expérimentale est estimée à 10,1%. Ce bon accord autorise l'utilisation des coefficients de transfert thermique calculés numériquement pour l'analyse thermique complète de l'embrayage mouillé.

WÄRMEÜBERTRAGUNG BEI DER LAMINAREN ÖLSTRÖMUNG IN DEN VERTIEFUNGEN EINEN KUPPLUNGSSCHEIBE

Zusammenfassung—Die Wärmeübertragung bei der laminaren Strömung in den radialen Vertiefungen in den Reibflächen einer Kupplung wird numerisch untersucht. Dabei zeigt sich, daß ein zwischen der Reibfläche und der metallischen Trennscheibe liegender dünner Ölfilm einen erheblichen Einfluß auf den Wärmeübergangskoeffizienten zwischen der Metallplatte und dem fließenden Öl besitzt. Es werden Versuche an einer realen, gleichmäßig rutschenden Kupplung beschrieben. Ein Vergleich des gemessenen Wärmeübergangskoeffizienten mit numerischen Ergebnissen zeigt Abweichungen innerhalb 10,7%. Der Meßfehler wird mit 10,1% abgeschätzt. Die gute Übereinstimmung erlaubt den Einsatz von rechnerisch ermittelten Wärmeübergangskoeffizienten bei einer vollständigen thermischen Untersuchung einer Kupplung.

ЛАМИНАРНЫЙ ТЕПЛОПЕРЕНОС В КАНАВКЕ МУФТЫ СО СМАЗКОЙ

Аннотация—Представлены численные результаты анализа теплопереноса к ламинарному потоку в радиальных канавках фрикционных поверхностей муфты со смазкой. Показано, что тонкая пленка смазки, находящаяся между фрикционной поверхностью и металлической разделительной пластиной, оказывает существенное влияние на коэффициеит суммарного теплопеноса между пластиной и текущей по канавке смазкой. Описываются эксперименты, проведенные в условиях непрерывного скольжения по муфте, и проводится сравнение измеренных коэффициентов суммарного теплопереноса с результатами численного анализ. Численные и экспериментальные результаты согласуются с точностью до 10,7%, а погрешность экспериментов оценивается в 10,1%. Это позволяет использовать численные результаты для коэффициентов теплопереноса при анализе теплопереноса в муфте со смазкой.