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Heat Transfer by Natural Convection in a Vertical Porous Layer*

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Heat transfer by natural convection in a vertical porous layer with horizontal heat flow has been studied analytically and experimentally. A boundary-layer analysis is extended to take account of both the vertical temperature gradient in the core of the porous layer and the apparent wall-film thermal resistance which is caused by a local increase in porosity in the vicinity of the wall. Comparison with experiment by use of porous layers composed of glass beads and water is afforded by overall heat-transfer coefficients and temperature distributions. Simple analytical expressions are obtained for the effect of the wall-film thermal resistance.

1. Introduction

Heat transfer by natural convection in a vertical porous layer with differentially heated side walls has recently come to receive attention due to its particular relevance to geothermal applications and thermal insulation.

The first experimental study was carried out by Schneider⁽¹⁾, in which some fundamental points such as the effects of the height ℓ on heat transfer, however, were left unknown.

A boundary-layer analysis⁽²⁾ clarified the basic heat-transfer characteristics by using the controlling nondimensional variable $Ra_g \cdot k/s\ell$, which at the same time suggested that the conduction regime, where the effect of convection is negligible, occurs when

$$Ra_g \cdot k/s\ell < 1 \sim 10$$

Experimental⁽³⁾⁽⁴⁾ and numerical⁽⁵⁾⁽⁶⁾ studies also are considered to suggest that the effects of the aspect ratio $A = \ell/s$ on heat transfer are reduced when using the above type of the nondimensional variable. Detailed examination of the numerical solutions, however, indicates that a change in the aspect ratio brings about a slight variation in both the critical Rayleigh number for the convection regime and the heat-transfer characteristics, although the effect seems to be saturated for $A > 6$. To explain these facts, a boundary-layer analysis is attempted in the present paper, taking account of the vertical temperature variation on the outside of the boundary layer.

When there is a large difference in thermal conductivities of porous-medium constituents with a variation of porosity near the wall, an excess thermal resistance is produced, which is expressed in terms of the apparent wall-film thermal resistance⁽⁷⁾. The effects of this thermal resistance on natural convection has not been fully considered, particularly when there exists a vertical temperature variation on the outside of the boundary layer.

The purpose of this paper is to study the heat-transfer characteristics of natural convection in vertical porous layers in more detail through an examination of such effects of the temperature variation in the undisturbed fluid and the wall-film thermal resistance owing to variable porosity.

2. Nomenclature

- A : aspect ratio = ℓ/s
- c_p : specific heat of fluid at constant pressure
- $D(m)$: constant, defined in Eq.(17)
- d_p : mean diameter of glass beads
- g : gravitational acceleration
- h : heat-transfer coefficient
- h_x : local heat-transfer coefficient
- h_m : mean heat-transfer coefficient
- h_w : apparent wall-film coefficient of heat transfer in porous medium with stagnant fluid⁽⁷⁾
- k : permeability = $\{\epsilon^3/150(1-\epsilon)^2\}d_p^2$
- ℓ : height of vertical layer
- m : nondimensional temperature gradient defined in Eq.(5)
- Nu : mean Nusselt number = $h_m s/\lambda$
- P : pressure
- Ra : Rayleigh number = $g\beta(T_w - T_r)\ell^3/(\nu\kappa)$
- Ra_g : Rayleigh number = $g\beta(T_h - T_c)s^3/(\nu\kappa)$
- $\tilde{R}$: modified Rayleigh number = $Ra_g k/(s\ell)$
- s : thickness of vertical layer
- T : temperature
- T_x : core temperature
- T_r : reference temperature, defined as T_x at $x = \ell/2$
- T_w : wall temperature

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- T_h : temperature of heating surface
 T_c : temperature of cooling surface
 T_{wa} : apparent wall temperature
 u, v : velocity components
 x, y : coordinates $(x^*, y^*) = (x/\ell, y/\ell)$
 β : cubical expansion coefficient
 κ : thermal diffusivity of porous medium⁽⁸⁾ $= \lambda / (\epsilon_p \rho)$
 λ : thermal conductivity of stagnant porous medium
 μ : viscosity of fluid
 ν : kinematic viscosity
 ρ : fluid density
 ρ_x : fluid density in core region
 ρ_r : ρ_x at $T = T_r$
 ϵ : porosity
 δ : boundary-layer thickness
 δ^* : dimensionless boundary-layer thickness $= \delta/\ell$
 η : dimensionless coordinate $= y/\delta$

3. Natural convection along vertical plate

First a preliminary analysis is made of natural convection heat transfer from a single vertical plate to a porous medium, which is later applied to the case of a vertical porous layer with differentially heated side walls.

3.1 Effects of temperature variation on the outside of boundary layer

Steady, two-dimensional fluid flow is assumed and the Boussinesq approximation is adopted: fluid properties are independent of temperature except for the effect of density variation in modifying the action of gravity. The governing equations of continuity, energy and Darcy's law are

$$\partial u / \partial x + \partial v / \partial y = 0 \quad (1)$$

$$u \partial T / \partial x + v \partial T / \partial y = \kappa (\partial^2 T / \partial x^2 + \partial^2 T / \partial y^2) \quad (2)$$

$$0 = -\partial P / \partial x - (\mu / k) u - \rho g \quad (3)$$

$$0 = -\partial P / \partial y - (\mu / k) v \quad (4)$$

A boundary layer is assumed to be formed on a vertical plate of length ℓ maintained at uniform temperature. A previous boundary-layer analysis⁽²⁾ made by one of the authors is extended in the present paper to include the temperature variation of the ambient fluid on the outside of the boundary layer under the hypothesis of the linear relation:

$$(T_x - T_r) / (T_w - T_r) = m(x^* - 1/2) \quad (5)$$

where the nondimensional temperature gradient m is in the range $0 \leq m \leq 2$. In case of $m=0$ the ambient temperature T_x becomes uniform, while the other limit of $m=2$ affords the greatest temperature gradient in which T_x is equal to T_w at $x^*=1$.

The equation of thermal expansion is

$$\rho = \rho_r \{1 - \beta(T - T_r)\} \quad (6)$$

Substituting Eq.(6) into Eq.(3) and making use of the boundary layer assumptions, we obtain

$$u = kg\rho_r\beta(T - T_x)/\mu \quad (7)$$

This means that the velocity and thermal boundary-layer thickness are the same as denoted in the previous study⁽²⁾.

The energy equation is written in the integral form

$$\frac{d}{dx} \int_0^{\delta} u(T - T_x) dy + \frac{dT_x}{dx} \int_0^{\delta} u dy = -\kappa \left(\frac{\partial T}{\partial y} \right)_{y=0} \quad (8)$$

Similarities in temperature and velocity profiles are assumed⁽⁹⁾:

$$\theta \equiv (T - T_x) / (T_w - T_x) = (1 - \eta)^2 \quad (9)$$

where $\eta = y/\delta$.

Substituting Eqs.(5), (7) and (9) into Eq.(8), and putting

$$F \equiv (T_w - T_x) / (T_w - T_r) = 1 - m(x^* - 1/2), \quad (10)$$

we obtain the following differential equation

$$AF^2\delta^* \frac{d\delta^*}{dx^*} + \left\{ A \frac{dF^2}{dx^*} - CF \frac{dF^*}{dx^*} \right\} \delta^{*2} = -\frac{BF}{R_a k/\ell^2} \quad (11)$$

where

$$A = \int_0^1 \theta^2 d\eta = 1/5, \quad B = -(\delta\theta/\delta\eta)_{\eta=0} = 2, \quad C = \int_0^1 \theta d\eta = 1/3$$

with the boundary condition

$$x^* = 0 \quad \delta^* = 0$$

Equation(11), when solved for the boundary-layer thickness δ^* for $m \neq 0$, gives

$$\delta^* = \sqrt{\frac{30}{m} \left[\left\{ 1 - \frac{mx^*}{1 + (m/2)} \right\}^{-2/3} - 1 \right] (R_a k/\ell^2)^{-1/2}} \quad (12)$$

When mx^* is small, δ^* becomes

$$\delta^* = \sqrt{[20/[1 + (m/2)]] x^* (R_a k/\ell^2)^{-1/2}} \quad (13)$$

Further, in the limit of $m=0$, Eq.(13) reduces to the previous solution⁽²⁾.

The local heat flux q_x and the local heat-transfer coefficient h_x are calculated from

$$q_x \equiv h_x(T_w - T_x) = -\lambda \left(\frac{\partial T}{\partial y} \right)_{y=0} = \frac{B\lambda}{\ell\delta^*} (T_w - T_x) \quad (14)$$

The mean heat-transfer coefficient h_m is defined by the equation

Fig.1 Single vertical plate in porous medium

Fig.2 Apparent wall temperature

$$\begin{aligned} h_m l / \lambda &= \int_0^1 (h_x l / \lambda) (T_w - T_x) / (T_w - T_r) dx^* \\ &= \int_0^1 (B / \delta^*) \left(1 + \frac{m}{2} - mx^* \right) dx^* \end{aligned} \quad (15)$$

Performing an integration, we obtain the mean Nusselt number in the form

$$h_m l / \lambda = D(m) (R_a k / l^2)^{1/2} \quad (16)$$

where

$$D(m) = \begin{cases} 0.894 & \text{for } m = 0 \\ 1.18 & \text{for } m = 1 \\ 1.52 & \text{for } m = 2 \end{cases} \quad (17)$$

With increasing m , the temperature difference $(T_w - T_r)$ increases and the boundary-layer thickness decreases in the lower half of the vertical plate, where the contribution to the overall heat transfer is large, thus resulting in an increase in the mean heat-transfer coefficient.

The previous study⁽²⁾ for $m = 0$ indicates that there exists no appreciable difference between the integral and the exact numerical solutions, the latter giving $D(0) = 0.887$. Thus, although the present integral solution is based on a simple profile, Eq.(9), $D(m)$ in Eq.(17) is considered to be close enough to that of the exact solution.

Recently, Minkowycz and Cheng⁽¹⁰⁾ made an analysis of natural convection heat transfer from a vertical cylinder to a porous medium, which gives a limiting value $D(0) = 0.888$ as in the exact solution⁽²⁾ for the flat plate case.

3.2 Effects of wall-film thermal resistance

When the porosity near the wall differs from that of the interior region of a porous medium, in which the nonuniform distribution of the constituent solids is caused by the existence of the wall, a difference in thermal conductivities of porous-medium constituents brings about an increase in the thermal resistance in the vicinity of the wall. This excess thermal resistance can be treated as an apparent surface resistance⁽⁷⁾. Thus, it is natural that we consider the effects of the surface resistance on heat transfer.

Let $T_{w\alpha}$ be the apparent wall temperature extrapolated from the temperature profile in the interior region of porous media, as shown in Fig.2. The wall-film heat-transfer coefficient h_w , which is the reciprocal of the surface resistance for stagnant porous media, is defined as

$$-\lambda (\partial T / \partial y)_{y=0} = h_w (T_w - T_{w\alpha}) \quad (18)$$

where $(\partial T / \partial y)_{y=0}$ is a gradient of the extrapolated temperature and h_w can be estimated from the study of Ofuchi and Kunii⁽⁷⁾.

The apparent wall temperature, when approximated to be uniform over the vertical plate, is related to the mean heat flux $q_m = h_w (T_w - T_{w\alpha})$. Putting h_m' as the mean heat-transfer coefficient of a vertical plate of temperature $T_{w\alpha}$ and using Eq.(16), we obtain

$$\frac{h_m l}{\lambda} = D(m) \left(\frac{g \beta (T_{w\alpha} - T_r) l^3}{\nu \kappa} \frac{k}{l^2} \right)^{1/2} \quad (19)$$

Taking into account the wall-film thermal resistance, the mean heat-transfer coefficient h_m is defined as

$$q_m = h_m (T_w - T_r) = h_w (T_w - T_{w\alpha}) = h_m' (T_{w\alpha} - T_r) \quad (20)$$

Thus

$$\begin{aligned} q_m &= \frac{\lambda}{l} D(m) \left[\frac{g \beta (T_w - T_r - q_m / h_w) l^3}{\nu \kappa} \frac{k}{l^2} \right]^{1/2} (T_w - T_r - \frac{q_m}{h_w}) \\ &= \frac{\lambda}{l} D(m) \left[\frac{g \beta (T_w - T_r) l^3}{\nu \kappa} \frac{k}{l^2} \right]^{1/2} (T_w - T_r) \left(1 - \frac{h_m}{h_w} \right)^{3/2} \end{aligned} \quad (21)$$

which gives the mean heat-transfer coefficient:

$$\frac{h_m l}{\lambda} = D(m) (R_a k / l^2)^{1/2} \left(1 - \frac{h_m l / \lambda}{h_w l / \lambda} \right)^{3/2} \quad (22)$$

An additional remark is made on the rather rough assumption of constant apparent wall temperature $T_{w\alpha}$. Eq.(22) for $m = 0$ becomes

$$\frac{h_m l}{\lambda} = 0.894 (R_a k / l^2)^{1/2} \left(1 - \frac{h_m l / \lambda}{h_w l / \lambda} \right)^{3/2} \quad (23)$$

Here it is noted that the previous numerical results⁽²⁾ for $m = 0$, which consider the variation of $T_{w\alpha}$ with x , are well described by a simple correlating equation (23).

4. Natural convection in vertical porous layer

The results of the preceding chapter for a single plate case are now applied to the case of a vertical porous layer with differentially heated side walls (Fig.3).

Hot and cold boundary layers are assumed to grow in a way which is similar to the behavior on a single plate in a porous medium. The temperature in the core between hot and cold boundary layers is assumed to vary linearly in the vertical direction according to Eq.(5). The reference temperature T_r is taken as the mean temperature of the isothermal side walls, i.e. $T_r = (T_h + T_c) / 2$. Using an approximation that apparent wall temperatures relating to wall-film thermal resistance are maintained at uniform temperatures, $T_{h\alpha}$ and $T_{c\alpha}$, on the heating and cooling surfaces, respectively, the mean heat flux

Fig.3 Vertical porous layer

through the vertical layer is expressed in terms of h_w as

$$q_m = h_w(T_h - T_{ha}) = h_w(T_c - T_{ca}) \quad (24)$$

Denoting by h_m the mean heat-transfer coefficient of the vertical layer based on the layer temperature difference $(T_h - T_c)$, and by h_m'' the mean heat-transfer coefficient of each side wall based on the temperature difference, $(T_{wa} - T_r)$ or $(T_r - T_{ca})$, we have

$$q_m = h_m(T_h - T_c) = h_m''(T_{ha} - T_r) = h_m''(T_r - T_{ca}) \quad (25)$$

where h_m'' is determined by substituting the apparent temperature difference into Eq.(16).

Thus we obtain the mean heat-transfer coefficient h_m :

$$\frac{h_m s}{\lambda} = \frac{D(m)}{2\sqrt{2}} \left[\frac{g\beta(T_h - T_c)s^3}{\nu\kappa} \frac{k}{sl} \right]^{1/2} \left(1 - \frac{h_m s}{\lambda} \right)^{3/2} \quad (26)$$

where $D(m)$ takes the same values as in Eq.(17) [$D(0)/2\sqrt{2} = 0.316$, $D(2)/2\sqrt{2} = 0.538$].

For porous media in the absence of the surface resistance, taking the limit of $h_w s/\lambda \rightarrow \infty$ yields

$$\frac{h_m s}{\lambda} = \frac{D(m)}{2\sqrt{2}} \left(Ra_{s,i} \frac{k}{sl} \right)^{1/2} \quad (27)$$

which reduces to the previous results⁽²⁾ in case of $m = 0$.

When $Ra_{s,i}$ is small and the thermal conduction dominates over the convection, we obtain the following equation

$$\begin{aligned} q &= h_w(T_h - T_{ha}) = \lambda(T_{ha} - T_{ca})/s \\ &= h_w(T_{ca} - T_c) = h_m(T_h - T_c) \end{aligned} \quad (28)$$

Then the mean heat-transfer coefficient h_m in the conduction regime is given by

$$\frac{h_m s}{\lambda} = \frac{1}{1 + 2/(h_w s/\lambda)} \quad (29)$$

For the case of $m = 2$, in which the core temperature T_x becomes equal to T_c at

Fig.4 Effects of wall-film thermal resistance on natural convection heat transfer of vertical porous layer

the bottom of the layer, and T_h at the top, respectively, Fig.4 indicates the effects of the wall-film thermal resistance on natural convection heat transfer of vertical layers according to Eqs.(26) and (29).

5. Experiment

The temperature gradient m can not be determined within the framework of the boundary layer analysis. Thus, an experimental examination of m is attempted.

5.1 Experimental apparatus

The experimental apparatus is shown schematically in Fig.5. An annular space ① between two concentric copper cylinders heated from the inside is composed of both water and glass beads of 2.88 mm in mean diameter, forming a vertical porous layer

Fig.5 Experimental apparatus

with a height of 250 mm. Two kinds of inner cylinders ② of 180 mm and 280 mm in outer diameter with 5 mm in thickness are electrically heated by nichrome wires ③. A cooling chamber ④, through which water circulates, holds a cooling cylinder of 328 mm in inner diameter with 5 mm in thickness. The use of two kinds of inner cylinders yields aspect ratios, $A = \ell/s$, of 10.4 and 3.4. The heat loss through bakelite plates ⑤ and glass wools ⑥ is estimated by measuring the total heat flow at various temperature differences with porous layers of known thermal conductivities⁽¹¹⁾.

It takes about 10 hours for the system to attain a steady state, under which experimental temperature distributions are obtained for two vertical planes placed at 180° intervals around the vertical axis using calibrated copper-constantan thermocouples of 0.2 mm in mean diameter. Outlets ⑦⑧ for thermocouples and nichrome wires are shown in Fig.5. The temperature difference of the porous layer, $(T_h - T_c)$, is varied in the range of 3° to 40°.

The uniformity of the heating and cooling surfaces is nearly achieved such that the maximum variation of the local temperature difference of the porous layer amounts to about 10 percent of the mean temperature difference, $(T_h - T_c)$. Nondimensional temperatures shown in Figs.6 and 7 are based on the local temperature T_h .

The permeability of the porous medium is evaluated using the Blake-Kozeny equation

$$k = \{\varepsilon^3 / 150(1 - \varepsilon)^2\} d_p^2$$

Values of porosity ε in this experiment are $\varepsilon = 0.374$ for $A = \ell/s = 3.4$ and $\varepsilon = 0.45$ for $A = 10$, respectively.

Fig.6 Vertical variation of temperature in core of porous layer

5.2 Results and discussions

The vertical temperature distributions in the core region of the porous layer are shown in Figs.6 and 7, where the reversals of temperature gradients in Fig.6, however, are regarded as experimental errors due to small temperature differences. The results for the overall heat transfer are shown in Fig.8.

For $A = 10.4$ (Fig.6), the temperature gradient m is considered to approach the limiting value of $m = 2$ with an increase in $R \equiv Ra_g k / s \ell$. Fig.8 indicates that the heat-transfer characteristics are well correlated with the analysis for $m = 2$.

For $A = 3.4$ (Fig.7), although R is considerably increased, the values of m still remain to be about 1.3. Substitution of the corresponding value of $D(1.3) = 1.27$ [Table 1] into Eq.(27) results in a fairly good agreement with experiments, as shown in Fig.8. Thus, utilizing the temperature gradient m in the range where R is high, namely in the regime of the boundary-layer flow, we can see that the present analysis well describe the experimental results.

The estimation of R [$\equiv Ra_g k / s \ell = 2 Ra \cdot k / \ell^2 \cdot (s/\ell)^2$], for which the thickness of the boundary layer at the mid height $x^* = 1/2$ reaches half the layer thickness, suggests such high values of $R = 70$ to 80 for $m = 0$ to 2 as the limit of the applicability of the present analysis. However, the analysis is seen to be applicable over a wide range of R in Fig.8. Accordingly the interference of both boundary layers, which develop along both the heating and the cooling surfaces, is considered to be rather weak, thus exerting no appreciable influences on the flow near the vertical surfaces and the heat transfer [although the vertical temperature gradient in the core is formed by

Fig.7 Vertical variation of temperature in core of porous layer

Fig.8 Natural convection heat transfer of vertical porous layer in the absence of wall-film thermal resistance [$h_w s / \lambda \rightarrow \infty$]

the counter flow on both side walls]. The local temperature gradients at the top and bottom ends, namely at the leading edges of the boundary layers, do not seem to change excessively with an increase in R , which is also considered to be a reason for the assumption of constant m , leading to a simple correlation of heat transfer. Thus, such correspondences as $m=2$ to $A > 6$ and $m=0$ to $A=1$, as shown in Fig.8, reveal the nature of heat-transfer characteristics.

An increase in the temperature gradient is associated with an increase in heat transfer. In this regard, a close approach of m to its limiting value of $m=2$, as seen in Fig.6, is considered to bring about a saturation of the effects of A on the heat-transfer characteristics for $A > 6$.

For the case of $A < 1$, natural convection is suppressed owing to an increasing resistance to flow⁵. This regime is not dealt with in the present paper.

Although the effect of wall-film thermal resistance is expressed by Eq.(26), the presence of the surface resistance may affect the preceding correspondence of m to A . When $h_w s / \lambda$ is large, however, the above correspondence will hold approximately. When $h_w s / \lambda$ is small, the change in the heat-transfer characteristics caused by the presence of the wall-film thermal resistance becomes large compared with that caused by the variation of m in the range 0 to 2. Therefore, as a first approximation, the effects of the excess thermal resistance near the vertical walls on the heat-transfer characteristics are estimated from Eq.(26) with the preceding correspondence of m to A .

6. Conclusions

- (1) Natural convection heat transfer

from a vertical wall to a porous medium is affected by the vertical temperature gradient m of the ambient fluid on the outside of the boundary layer, as seen in Eqs.(16) and (17).

(2) Heat transfer is reduced by the apparent wall-film thermal resistance owing to a variation of porosity in the vicinity of the vertical wall, as represented by Eq.(22). For porous media with uniform ambient temperature, Eq.(22) is simplified to Eq.(23) which can well describe the previous numerical results².

(3) For a porous layer with differentially heated side walls as shown in Fig.3, heat-transfer characteristics in the absence of wall-film thermal resistance, i.e. in the limit of $h_w s / \lambda \rightarrow \infty$, are expressed by Eq.(27)

$$\frac{h_m s}{\lambda} = \frac{D(m)}{2\sqrt{2}} \left(R_{a,s} \frac{k}{sl} \right)^{1/2}$$

with such correspondence of the temperature gradient m to the aspect ratio A as $m=2$ to $A > 6$ and $m=0$ to $A=1$. The saturation of the effects of A on the heat-transfer characteristics at about $A=6$ ⁽⁶⁾ is considered to be reflected in a linear temperature distribution which has an asymptotic limit for the temperature gradient of $m=2$.

(4) For a vertical porous layer with the apparent wall-film thermal resistance, the overall heat transfer is approximately expressed by Eq.(26)

$$\frac{h_m s}{\lambda} = \frac{D(m)}{2\sqrt{2}} \left(R_{a,s} \frac{k}{sl} \right)^{1/2} \left(1 - \frac{h_m s / \lambda}{h_w s / \lambda} \right)^{3/2}$$

with the same correspondence of m to A as in $h_w s / \lambda \rightarrow \infty$.

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Table 1 Calculated values of $D(m)$

m	$D(m)$	m	$D(m)$	m	$D(m)$
0.0	0.894	0.7	1.08	1.4	1.31
0.1	0.908	0.8	1.11	1.5	1.34
0.2	0.945	0.9	1.14	1.6	1.37
0.3	0.973	1.0	1.18	1.7	1.41
0.4	1.00	1.1	1.21	1.8	1.45
0.5	1.03	1.2	1.24	1.9	1.48
0.6	1.06	1.3	1.27	2.0	1.52

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Appendix I

Values of $D(m)$ are calculated from the following equation and shown in Table 1.

$$D(m) = \int_0^1 \sqrt{\frac{2m}{15}} \left(1 + \frac{m}{2} - mx^* \right) \times \left[\left(1 - \frac{mx^*}{1+(m/2)} \right)^{-2/3} - 1 \right]^{-1/2} dx^*$$

Appendix II

The analysis for porous layers between two flat plates has been compared with the experiment for porous layers between two concentric cylinders. The effects of curvature can be estimated by an analysis of Minkowycz and Cheng⁽¹⁰⁾ or by use of the concept of conduction film⁽¹²⁾. The latter yields the ratio of the mean Nusselt number of the outer surface of a cylinder to that of a flat plate, as in the previous study⁽²⁾,

$$\left(\frac{h_{nl}}{r} \right)_{cyl} / \left(\frac{h_{nl}}{\lambda} \right)_{fl} = \frac{\xi}{\ln(1+\xi)}$$

where $\xi = 1/[D(m)(r_o/l)(Rak/l^2)^{1/2}]$, r_o is the radius of the cylinder, and the subscripts cyl and fl refer to the cylinder and the flat plate, respectively.

The effects of curvature in the present experiment are estimated at 9

percent at $R=10$ and 6 percent at $R=20$ for $l/s=3.4$ (by putting $m=1.3$), and 4 percent at $R=10$ and 3 percent at $R=20$ for $l/s=10$ (by putting $m=2$). When R exceeds 20, the curvature has no appreciable influence on experiment.

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