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International Journal of Thermal Sciences 48 (2009) 188–194

International Journal of Thermal Sciences

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Theoretical analysis of heat and mass transfer to fluids flowing across a flat plate

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Received 15 August 2007; received in revised form 28 February 2008; accepted 9 March 2008

Available online 14 April 2008

Abstract

Analytical solutions for heat and mass transfer to fluids flowing across an isothermal flat plate were obtained. Laminar, inviscid and irrotational flow were assumed in the solution of the energy equation. The derived new relations were comparable with the experimental data and the solutions of other investigators.

Keywords: Heat convection; Mass transfer; Flat plate; Cross flow; Boundary layer; Solar collectors

1. Introduction

Heat transfer to fluids flowing normally or obliquely to plane surfaces are often encountered in practice. For example, the vertical take-offs of helicopters and rockets create a downwash with considerable cooling or heating respectively. The cooling effects of the wind on solar collector plates, walls, roofs of buildings and petroleum storage tanks are additional examples of applications.

Impinging flow on plane surfaces can be classified into two main categories. The first is when the flow is coming from nozzles (e.g., Kendoush [1]). The second is the uniform free stream flow across the surface either normally or obliquely. The present work is focused on the latter category.

The wedge solution of Merk [2] cannot be applied to the present case simply because the flow bifurcates over two infinitely large surfaces. Kang and Sparrow [3] performed a numerical solution for the heat and mass transfer from an upstream-facing rectangular blunt face situated in a uniform oncoming flow. Face-average Nusselt and Sherwood numbers range from 5×10^3 to 5×10^4.

Igarashi [4–7] published a series of papers on the experimental heat transfer from a square prism, a rectangular cylinder and a flat plate to an air stream. The flow pattern and heat transfer across the frontal face of the prism and the rectangular cylinder are analogous to the present problem.

Ramachandran, Chen and Armaly [8] performed a numerical solution of the Navier–Stokes and the energy equations for the mixed convection in stagnation flows adjacent to vertical surfaces.

Zhang et al. [9] performed numerical solutions for the convective heat transfer for both inline and staggered arrangements of parallel-plate fin heat exchangers. The inline and staggered arrangements were found to increase the rate of heat transfer in addition to the friction factor.

It can be deduced from the literature that the only theoretical work on the present problem was limited to the numerical solution of the conservation equations of heat, mass and momentum. The motive behind the present study is to provide analytical solutions for predicting the convective heat or mass transfer from surfaces in cross flow.

2. Analytic solution

Consider a flat plate of length $2d$ and unit width facing a uniform flow with the plate length lying along the x-axis as shown in Fig. 1. The free stream makes an angle β with the z-axis. Assume laminar, incompressible, irrotational and inviscid flow. The velocity components of the flow on the side facing the oncoming flow ($z = 0-$) are given by Hess [10] as follows

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doi:10.1016/j.ijthermalsci.2008.03.010

Nomenclature

C	specific heat at constant pressure
d	half length of plate
$\langle h\rangle$	average heat transfer coefficient
h	local heat transfer coefficient
k	thermal conductivity of fluid
L	length of plate or four times plate area divided by plate perimeter
$\langle Nu\rangle$	average Nusselt number $(2dh/k)$
Nu^*	average Nusselt number (hd/k)
Nu_L	average Nusselt number (hL/k)
Pr	Prandtl number (ν/α)
Pe	Peclet number $(2dU/\alpha)$
q	heat flux
Re^*	Reynolds number (dU/ν)
Re_L	Reynolds number (LU/ν)
Sc	Schmidt number (ν/D)
St	Stanton number $(h/\rho CU)$
T	temperature
$T_{w,0}$	temperature of plate surface and free stream fluid, respectively
U	mainstream velocity
$V_{x,z}$	velocity components in the x- and z-directions, respectively
x	Cartesian coordinate along the plate
z	coordinate normal to the plate

Greek symbols

α	thermal diffusivity
β	angle of attack
δ	thickness of the thermal boundary layer
θ	dimensionless temperature (Eq. (5))
ν	kinematic viscosity of fluid

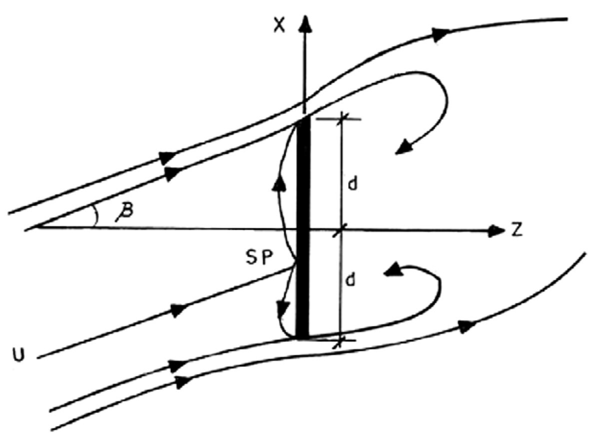

Fig. 1. The flat plate in the flow, "SP = Stagnation point".

$$V_z = 0 \tag{1}$$

$$V_x = U\left[\frac{\cos\beta}{\pi}\ln\left(\frac{1+x/d}{1-x/d}\right)+\sin\beta\right] \tag{2}$$

Heat is transferred from the plate, which is maintained at a constant temperature T_w to the flowing fluid which is at temperature T_0. The energy conservation equation without viscous dissipation and sources of heat is written in Cartesian coordinates as follows

$$V_x\frac{\partial T}{\partial x}+V_z\frac{\partial T}{\partial z}=\alpha\left(\frac{\partial^2 T}{\partial z^2}+\frac{\partial^2 T}{\partial x^2}\right) \tag{3}$$

Assuming that the heat conduction along the streamlines is much less than that across them, therefore

$$\frac{\partial^2 T}{\partial z^2}\gg\frac{\partial^2 T}{\partial x^2} \tag{4}$$

This assumption was used before by the author in calculating the heat convection to a circular disk facing a uniform flow (Kendoush [11]).

We shall define the reduced temperature θ as follows

$$\theta=(T_w-T)/(T_w-T_0) \tag{5}$$

Adopting the Pohlhausen method which was used earlier by the author [12], the temperature profile within the thermal boundary layer δ can be approximately represented by the first two terms of an infinite power series in z. Thus,

$$\theta=(3z/2\delta)-(z/\delta)^3/2 \tag{6}$$

Using θ instead of T in Eq. (3) and integrating Eq. (3) with respect to z between 0 and δ, also incorporating assumption (4) into Eq. (3) yields

$$\int_0^\delta V_x\frac{\partial\theta}{\partial x}\,dz+\int_0^\delta V_z\,d\theta=\alpha\int_0^\delta\frac{\partial}{\partial z}\left(\frac{\partial\theta}{\partial z}\right)dz \tag{7}$$

The continuity equation of the flow becomes the following after considering the flow field of Eqs. (1) and (2).

$$\frac{\partial V_z}{\partial z}+\frac{\partial V_x}{\partial x}=0 \tag{8}$$

Using the following boundary conditions at

$$z=0;\qquad V_z=0 \tag{9}$$

$$z=\delta;\qquad V_z=-\int_0^\delta\frac{\partial V_x}{\partial x}\,dz \tag{10}$$

and

$$z=\delta;\quad \theta=1 \tag{11}$$

Accordingly, Eq. (7) reduces to the following

$$\int_0^\delta V_x\frac{\partial\theta}{\partial x}\,dz+[V_z\theta]_0^\delta=-\alpha\left(\frac{\partial\theta}{\partial z}\right)_{z=0} \tag{12}$$

Substituting Eq. (10) into Eq. (12) yields

$$\int_0^\delta V_x \frac{\partial\theta}{\partial x}\,dz - \int_0^\delta \theta \frac{\partial V_x}{\partial x}\,dz = -\alpha\left(\frac{\partial\theta}{\partial z}\right)_{z=0} \tag{13}$$

This equation reduces to the following

$$\frac{\partial}{\partial x}\int_0^\delta (V_x\theta)\,dz = \alpha\left(\frac{\partial\theta}{\partial z}\right)_{z=0} \tag{14}$$

Substituting Eq. (6) into the above equation gives

$$\frac{\partial}{\partial x}\int_0^\delta V_x\left[\frac{3z}{2\delta} - \frac{z^3}{2\delta^3}\right]dz = \alpha\left(\frac{\partial\theta}{\partial z}\right)_{z=0} \tag{15}$$

Evaluating the integral and differentiating yields the following

$$\frac{5}{8\alpha}\left(\delta\frac{dV_x}{dx} + V_x\frac{d\delta}{dx}\right) = \left(\frac{\partial\theta}{\partial z}\right)_{z=0} \tag{16}$$

Substituting the differentials of Eq. (2) into this equation, yields the following

$$\frac{5\delta}{4\pi\alpha d}\left[\frac{U\cos\beta}{1-(x/d)^2}\right] + \frac{5}{8\alpha}\left[\frac{U\cos\beta}{\pi}\ln\frac{1+x/d}{1-x/d} + U\sin\beta\right]\frac{d\delta}{dx} = \left(\frac{\partial\theta}{\partial z}\right)_{z=0} \tag{17}$$

We may obtain the following equation from Eq. (6)

$$\left(\frac{\partial\theta}{\partial z}\right)_{z=0} = \frac{3}{2\delta} \tag{18}$$

Making the above two equations equal, yields the following first order non linear ordinary differential equation

$$\frac{5}{4\pi\alpha d}\left[\frac{U\cos\beta}{1-(x/d)^2}\right] - \frac{3}{2} + \frac{5}{8\alpha}\left[\frac{U\cos\beta}{\pi}\ln\frac{1+x/d}{1-x/d} + U\sin\beta\right]\frac{d\delta}{dx} = 0 \tag{19}$$

The solution of this equation is shown in Appendix A and is given as follows

$$\delta = \frac{d}{4}\left(\frac{\alpha\pi}{5Ud\cos\beta}\right)^{1/2}\left(4 - \left(\frac{x}{d}\right)^2\right) \tag{20}$$

or in dimensionless form

$$\frac{\delta}{d} = \frac{\sqrt{6}}{4}\left(\frac{\pi}{5Pe^*\cos\beta}\right)^{1/2}\left(4 - \left(\frac{x}{d}\right)^2\right) \tag{21}$$

where

$$Pe^* = Ud/\alpha \tag{22}$$

Eq. (21) represents the variation of the thickness of the thermal boundary layer along the flat plate. It confirms the fact that the higher the Reynolds (or *Pe*) number the thinner the boundary layer. There is no evidence of a singularity at the stagnation point (that is, at $\beta = x/d = 0$).

Hess [10] also derived the position of the stagnation point along the plate as follows (see Fig. 1)

$$x/d = -\tanh\left(\frac{1}{2}\pi\tan\beta\right) \tag{23}$$

Substituting the above equation into Eq. (21) gives the thickness of the thermal boundary layer at the stagnation point straightforwardly.

The heat flux from the plate is obtained as follows

$$q = -k\left[-(T_w - T_0)\left(\frac{\partial\theta}{\partial z}\right)\right]_{z=0} = h(T_w - T_0) \tag{24}$$

Therefore, the local Nusselt number becomes

$$Nu^*(x) = 3.0902(\cos\beta\, Pe^*)^{1/2}\left(4 - (x/d)^2\right)^{-1} \tag{25}$$

The average heat transfer coefficient is calculated from the following

$$\langle h\rangle = \frac{1}{2d}\int_{-1}^{1} h(x/d)\,d(x/d) \tag{26}$$

It should be noted that $\int_{-1}^{1}(4-(x/d)^2)^{-1}\,d(x/d) = 0.549$, therefore, we get the following for the average Nusselt number

$$\langle Nu^*\rangle = \frac{\langle h\rangle d}{k} = 0.848(Pe^*\cos\beta)^{1/2} \tag{27}$$

The most interesting case is the symmetric flow across the plate, for which $\beta = 0°$ in the above equations. Eq. (27) shows that when the fluid flows across the plate obliquely, a lesser rate of heat would transfer than the symmetric flow due to the term $\sqrt{\cos\beta}$.

Fig. 2 shows the velocity profile along the plate from Eq. (2) where the stagnation point is seen at $-1 < x/d < -0.5$ for angle of attack $\beta = 30°$. The case of a flat plate moving in a stagnant fluid with a constant speed along the z-axis (Fig. 1) was analyzed numerically by Rodriguez-Rodriguez et al. [13]. Their velocity profile has some similarities with that of Fig. 2. Also seen in Fig. 2, is the variation of the thickness of the thermal boundary layer from Eq. (21) and the local *Nu* number distribution from Eq. (25).

The similarity between the mass and heat transfer conservation equations, particularly when viscous heating is ignored, allows us to use Eqs. (25) and (27) to calculate the rate of mass transfer from the plate simply by replacing *Nu* by *Sh* and *Pe* by *Re Sc* in the above mentioned equations.

3. Validation of the theory

Having derived the fundamental equations of local and average heat and mass transfer to fluids flowing across the flat plate, we are now in a position to validate these equations.

Igrashi and Mayumi [14] presented experimental data on the laminar flow heat transfer characteristics around a rectangular cylinder for Reynolds number range 2500–12800. The experiments were carried out in a low speed wind tunnel. The temperature of the free air stream was 20 °C ($Pr = 0.70996$). The present Eq. (25) is compared with the data of their Fig. 2 for normal flow across the width of the rectangular cylinder. For rational comparison Eq. (25) becomes the following

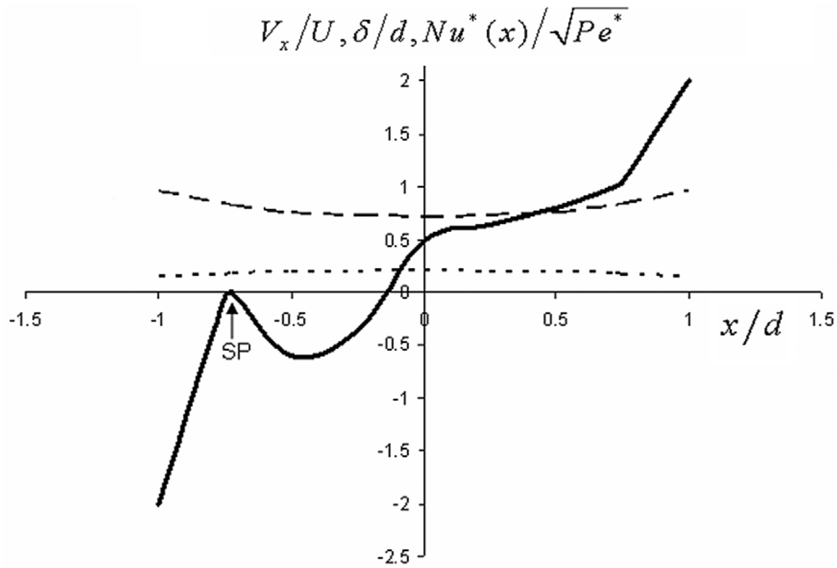

Fig. 2. The flow velocity profile along the plate V_x/U from Eq. (2) (———); the thermal boundary layer thickness, Eq. (21) (·········); and the local Nusselt number $Nu^*(x)/\sqrt{Pe^*}$ from Eq. (25) (— — — —). $Pe = 100$ and $\beta = 30°$.

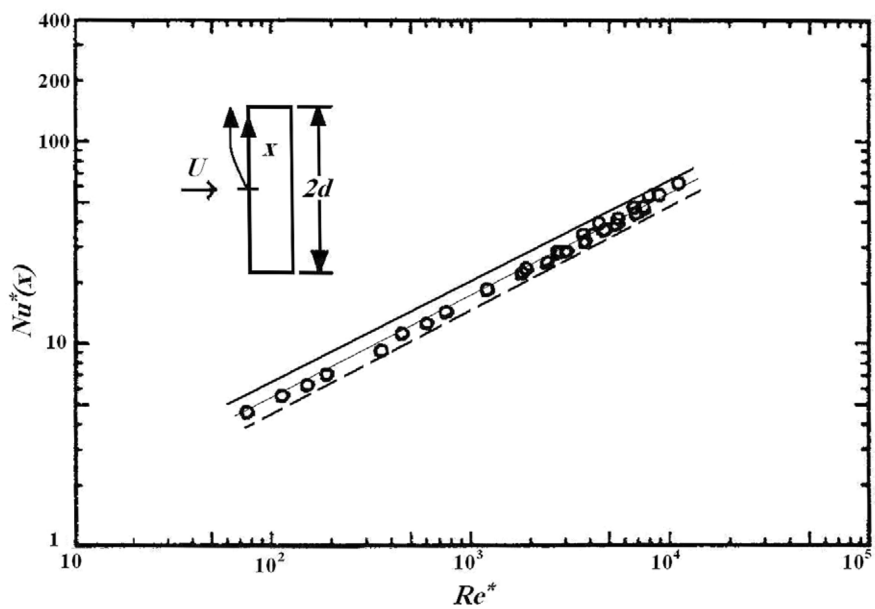

Fig. 3. Comparison of the experimental data of Igrashi and Mayumi [14]: (○ ○ ○) and the present solution: (Eq. 29) (– – –); Eq. (30) (——) and Eq. (31) (———) at $\beta = 0°$ and $Pr = 0.70996$.

$$Nu^*(x) = (3.0902)\left(\sqrt{0.70996}\right)\left(\frac{\sqrt{2}}{2}\right)Re^{0.5}\left(4 - \left(\frac{x}{d}\right)^2\right)^{-1} \tag{28}$$

Note that the parameter $\sqrt{2}/2$ is to rationalize the characteristic length in Nu and Re numbers, because Igrashi and Mayumi [14] used $2d$ as the characteristic length of Re and Nu numbers. For $x = 0$, this equation yields

$$Nu^*(x) = 0.46Re^{0.5} \tag{29}$$

For $x = 0.5d$, Eq. (28) gives

$$Nu^*(x) = 0.4909Re^{0.5} \tag{30}$$

and for $x = d$, Eq. (28) gives

$$Nu^*(x) = 0.6137Re^{0.5} \tag{31}$$

Fig. 3 shows good agreement between Eqs. (29)–(31) and the experimental data of Igrashi and Mayumi [14]. The lines of Eqs. (29)–(31) capture almost all the experimental data. Fig. 4 shows another comparison with the experimental data of Igrashi and Mayumi [14] for the 20° inclined front face DA of the rectangular cylinder. The experimental data of their Fig. 11b was compared with the present Eq. (27) as follows

$$\langle Nu^*\rangle = (0.848)(0.70992)^{0.5}\left(\sqrt{2}/2\right)(Re\cos 20)^{0.5} \tag{32}$$

The comparison of this equation with the experimental data of Igrashi and Mayumi [14] is shown in Fig. 4 where the

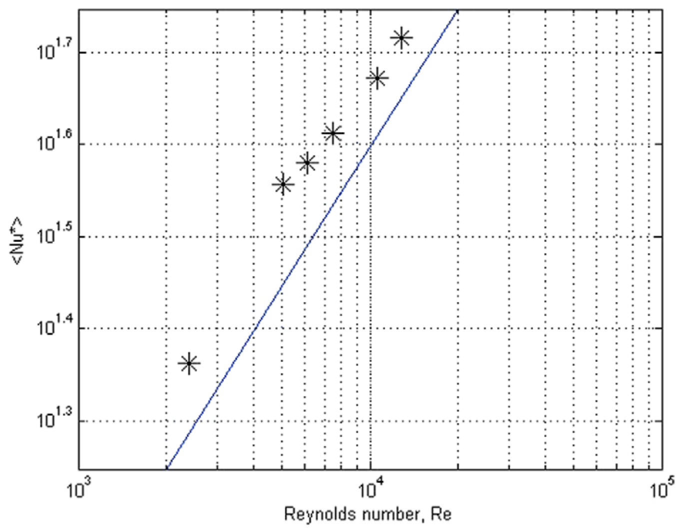

Fig. 4. Comparison between the experimental data of Igrashi and Mayumi [14]: (∗ ∗ ∗) and the present solution (Eq. (32)) (——) for the heat transfer from the inclined front face of a rectangular cylinder. $Pr = 0.70992$ and $\beta = 20°$.

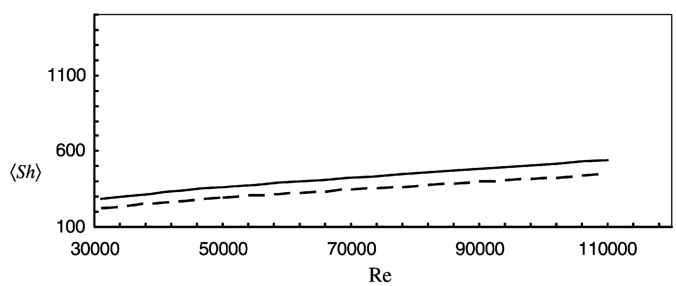

Fig. 5. Comparison between the present solution (Eq. (33)) (———) and the experimental correlation (Eq. (34)) of Chyu and Natarajan [15] (— — —) for the mass transfer. $Sc = 1.87$ and $\beta = 0°$.

discrepancy between theory and experiment is 16%, however, the present theory supports the conclusion of Igrashi and Mayumi [14] that their *Nu* number is proportional to $Re^{0.5}$.

Chyu and Natarajan [15] performed experiments to examine the mass transfer from the surface of a cube. The mass transfer version of Eq. (27), is the following

$$\langle Sh\rangle = 0.848(Re\,Sc\,\cos\beta)^{1/2} \tag{33}$$

This equation provided fair description of the experimental data of Chyu and Natarajan [15] as shown in Fig. 5. The experimental data of Chyu and Natarajan [15] were correlated by the following equation

$$\langle Sh\rangle = 0.868(Re)^{0.538} \tag{34}$$

Closer agreement would have been obtained if *Sc* number was less than unity instead of the present value of 1.87 because the present assumption of an inviscid flow in the energy equation restricts the present theory to low Prandtl and Schmidt number fluids.

It is believed that forced convection at the stagnation point of a sphere and a flat plate would be comparable provided that both bodies are situated in a uniform flow, and both are at isothermal states. Accordingly, Fig. 6 represents a fair comparison between Eq. (25) (with $Pr = 0.7$ and $\beta = x/d = 0$) and the experimental and theoretical results of the indicated investigators [16–24].

In the solar collector applications, a comparison was made with the mass transfer data of Sparrow et al. [25] who used rectangular cassettes containing naphthalene to form a smooth flat surface. These cassettes were placed in a wind tunnel and subjected to air speeds, at various angles of attack, from 4.5 to 24 m/s, corresponding to values of Reynolds numbers from 2×10^4 to 9×10^4. It should be noted that *Re* number $<10^6$ is the limit of laminar flow over a flat plate (White [26]). Sparrow et al. [25] obtained the following correlations by using the analogy between mass transfer and heat transfer

$$\langle Nu_L\rangle = 0.86Re_L^{1/2}Pr^{1/3} \tag{35}$$

Here the characteristic length in Nu_L and Re_L is four times the plate area divided by the plate perimeter. For an air temperature of 313 K, this equation gives

$$\langle h\rangle = 5.1(U/L)^{1/2}\ (\mathrm{W/m^2\,K}) \tag{36}$$

For the plate dimensions of 1.81 m × 0.89 m × 0.002 m; the value of L becomes equal to 1.193 m, hence Eq. (36) becomes

$$\langle h\rangle = 4.7\sqrt{U}\ (\mathrm{W/m^2\,K}) \tag{37}$$

The present equation (27) becomes the following for an air temperature of 313 K, $\cos\beta = 1$ and $d = 1.81/2 = 0.905$ m

$$\langle h\rangle = 4.927\sqrt{U}\ (\mathrm{W/m^2\,K}) \tag{38}$$

Fig. 7 shows an excellent agreement between this equation and the experimental correlation of Sparrow et al. [25].

One of the earliest experimental studies of forced convective heat transfer that was utilized later by the solar collector designers was that of Jurges, as described by McAdams [27]. The air speed was measured at the center of the tunnel and the following empirical correlation was obtained

$$h = 3.8U + 5.7\ (\mathrm{W/m^2\,K}) \tag{39}$$

Watmuff et al. [28] suggested that this equation may include free convection and radiation effects and for this reason they proposed the following expression which has been widely used for flat plate solar collectors

$$h = 3U + 2.8\ (\mathrm{W/m^2\,K}) \tag{40}$$

Sartori [29] presented multiple empirical equations from the literature and analyzed them for the purpose of their use as design equations. For the case of the inclined solar collectors the following equations were suggested by Sartori [29]

$$h = 3.83U^{0.5}(2d)^{-0.5}\ (\mathrm{W/m^2\,K}) \tag{41}$$

for laminar flow and

$$h = 5.74U^{0.8}(2d)^{-0.2} - 16.46(2d)^{-1}\ (\mathrm{W/m^2\,K}) \tag{42}$$

for a mixed boundary layer flow.

These equations were originally based on heat convection to fluids flowing along the surface of the flat plate. Sartori [29] believed that these equations can be applied to heat convection

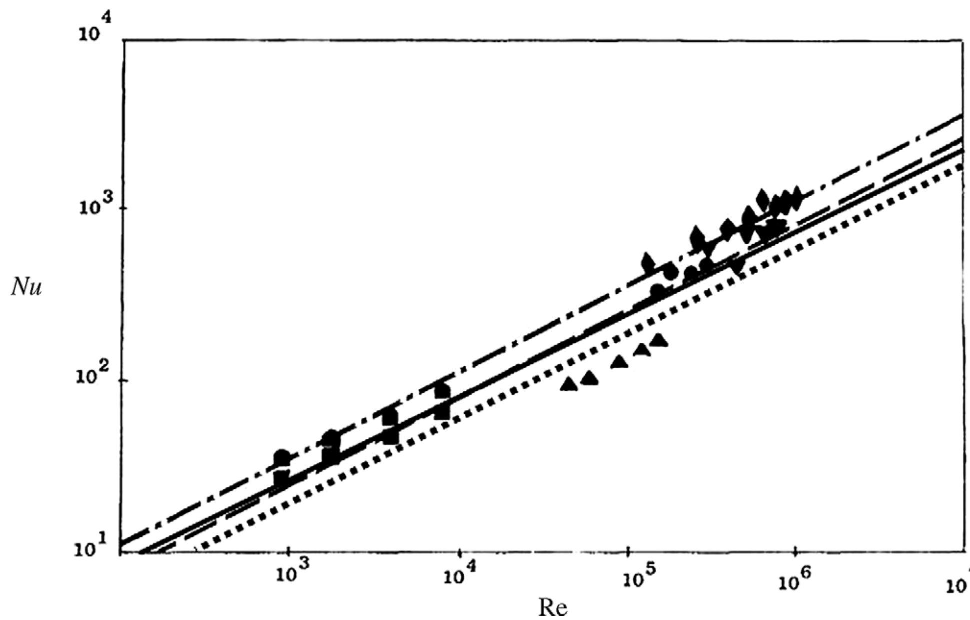

Fig. 6. Comparison of the heat transfer at the stagnation point. Kendoush [16] (— • — • —); Present Eq. (25) (———); Sibulkin [17] (— — —); Eckert [18] (- - - - -); Short [19] ■; Cary [20] ▲; Winkler and Danberg [21] ●; Korobkin and Gruenwald [22] ▼; Lautman and Droege [23] ◆ and Sato and Sage [24] ☗. $Pr = 0.7$ and $\beta = x/d = 0$.

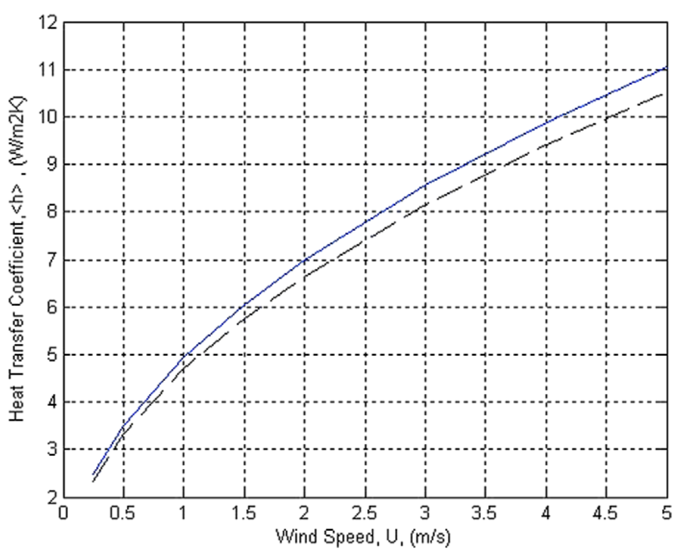

Fig. 7. Comparison with the experimental correlation of Sparrow et al. [25]: Eq. (35), (— — —) and the present solution: Eq. (36), (———).

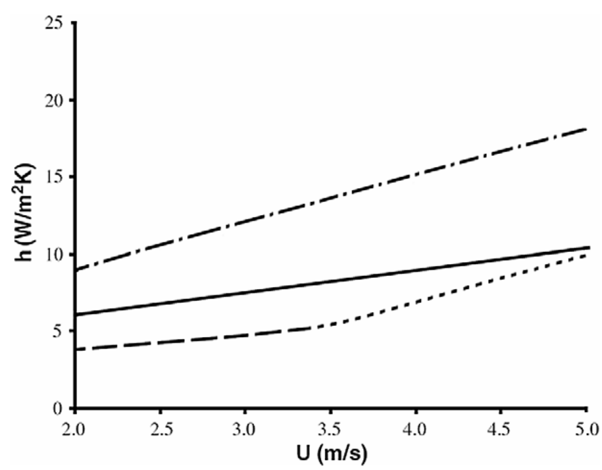

Fig. 8. The present solution Eq. (43) (———) is compared with Eq. (40) (— • — • —) of Watmuff et al. [28], Eq. (41) (— — —) and Eq. (42) (- - - - -) of Sartori [29] for the design of the inclined solar collector.

from inclined surfaces without specifying the angle of inclination. Using the present Eq. (27), this problem is solved as follows

$$h = 0.848k(\cos\beta U\,Pr/\upsilon)^{0.5}(d)^{-0.5}\ (\mathrm{W/m^2\,K}) \tag{43}$$

This equation was compared with Eqs. (40)–(42) presented by Watmuff et al. [28] and Sartori [28] in Fig. 8 for a value of $d = 1$ m, $\beta = 0$ and assumed $Pr = 0.7$. The present Eq. (43) lies between the two extremely bound equations of Watmuff et al. [28] and Sartori [29] but has the advantage of specifying the angle of inclination.

4. Conclusions

The new solution developed in this paper for the convective heat and mass transfer to fluids flowing normally or obliquely across a flat plate has been successful in predicting the rates of heat or mass transfer. The present analytical results were comparable with the available analytical and experimental results.

Acknowledgement

I wish to thank the University of Florida's Department of Mechanical and Aerospace Engineering's Thermal Science and Fluid Dynamics Group and the Institute of International Education's Scholar Rescue Fund for their continuing support of my work.

Appendix A. The solution of Eq. (19)

Eq. (19) does not match any of the standard forms of the first order ordinary differential equations. Accordingly, a series solution was adopted [30]. Assume the following as a general solution

$$\delta = A_0 + A_1 x + A_2 x^2 + \cdots A_n x^n \tag{A.1}$$

where $A_0, A_1, \ldots, A_n$ are constants to be determined later. The first three terms of this infinite series were considered. The logarithmic term of Eq. (15) was expanded in a Maclaurian's series as follows

$$\ln \frac{1 + x/d}{1 - x/d} = 2\left(\frac{x}{d} + \frac{1}{3}\left(\frac{x}{d}\right)^3 + \cdots\right) \tag{A.2}$$

Eqs. (A.1), its square value and its derivative together with Eq. (A.2) were substituted into Eq. (19). The resulted equation, which no longer contains δ is an identity in x. By equating coefficients of equal powers of x in this equation, we obtain the following relations for A_0, A_1 and A_2 after a lengthy algebra

$$A_0 = (6\pi\alpha d/5U\cos\beta)^{1/2} \tag{A.3}$$

$$A_1 = 0 \tag{A.4}$$

and

$$A_2 = -(3\pi\alpha)(6\pi\alpha d/5U\cos\beta)^{-1/2}/(10Ud\cos\beta)$$

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