THE CONVECTIVE HEAT TRANSFER OF FLUID FLOWING ACROSS A VERTICAL PLATE

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ABSTRACT

A numerical investigation was performed to determine the heat transfer to a fluid flowing across a vertical plate.laminar, inviscid and irrotational flow were assumed in the solution of the energy equation.

Temperature profiles, local and average Nusselt number are predicted for the Peclet number range from 30 to 100 and for Prandtle no.=0.72 (heat transfer in air).

A thinning of thermal layer accrued with increasing distance from the stagnation point, which contrasts with the conventional boundary layer. The average Nusselt no. decrease with increasing the angle of attack.

***Key words**: Forced convection; Heat transfer; Vertical plate*

INTRODUCTION

Heat transfer to fluids flowing normally or obliquely to plane surface are often encountered in practice. For example, the cooling effects of the wind on solar collector plates, walls and roofs of buildings, petrolium storage tanks and in a baffled exchanger.

Impinging flow on plane surfaces can be classified into two main categories. The first is when the flow is coming from Nozzle (1). The second is the uniform free stream flow across the surface (2), the present work is focused on the latter category .

The wedge solution of Merk (3) can not be applied to the present case simply because the flow bifurcates over two infinitely large surfaces.

Kang and Sparrow (4) performed a numerical solution for the heat transfer from an upstream facing rectangular blunt face situated in a uniform oncoming flow.

Igrashi (5-6) published a series of papers on the experimental heat transfer from a square prism. The heat transfer from the frontal face of the prism is analogous to the present study.

The present study includes a new attempt to solve the energy equation by direct substituting of the velocity component that obtained from a potential flow.

Governing Equation

The physical situation being investigation is presented in figure (1), which shows the oncoming flow attack on the vertical plate at different angles.

The problem involves the prediction of the heat transfer from the vertical plate by solving the steady, two-dimensional energy equation.

The fluid is assumed to be inviscid, irrotational incompressible and has constant thermo physical properties. Further more, compression work and viscous dissipation are assumed to be negligible in the energy equation.

Thus, the governing equations can be stated as:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right] \qquad (1)$$

The equation representing velocity components of flow were obtained by Hess (7) using the potential flow assumption are as follows:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left[\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right] \qquad (2)$$

$$v=U_\infty\left[\frac{\cos\beta}{2\pi}\log\left(\frac{(d+y)^2+x^2}{(d-y)^2+x^2}\right)+\sin\beta\right] \qquad (3)$$

The accompanying boundary conditions are: at the plate X=0; $T=T_w$, and in the oncoming flow for upstream of the body, $X=\infty$; $T=T_\infty$.

The Dimensionless Parameter

It is more convenient to re – arrange the energy equation so that each term is dimensionless in order to reduce the volume of the computational work by introducing the following dimensionless groups, where (θ) is dimensionless temperature. The boundary conditions becomes:

at X= 0, θ=1

at X=∞, θ=0

By substituting the dimensionless groups in the energy equation, the energy equation becomes dimensionless as follows: -

$$U\frac{\partial\theta}{\partial x}+V\frac{\partial\theta}{\partial y}=\frac{1}{pe}\left(\frac{\partial^2\theta}{\partial x^2}+\frac{\partial^2\theta}{\partial y^2}\right) \qquad (4)$$

Where, $U = u \backslash U\infty$, $V = v\backslash U\infty$, and $Pe = Re * Pr$

Computational Details

The dimensionless energy equation was solved numerically by a finite-difference scheme using the central difference approximation for an equal, regular and rectangular grid in each dimension.

Solution for a given Peclet number was achieved by iterative method (using Gause-seidel method) until convergence was reached (8).

All computations were performed on a 120×120 grid, and under-relaxation was necessary in the solution of the dimensionless energy equation.

The effect of the Peclet number on heat transfer was studying for (Pe) from 30 to 100 for different angle of attack.

RESULTS AND DISCUSSION

The behavior of thermal layer on the vertical plate can be examined with the aid of figure (2-3) at a fixed Peclet number it is observed that the thermal layer is thickest at the stagnation point and becomes thinner with increasing distance from the stagnation point. Correspondingly, the temperature gradient at the wall is least at the stagnation point and grows larger with distance. This is in contrast with conventional boundary layer where temperature gradient decreases away from the stagnation point. This is clearly reflects the very rapid acceleration of the flow due to the presence of the corner at the extremity of the plate.

The temperature profiles of (Pe=30) with different angle of attack are plotted in figure (4-5). It can be observed that when the angle of attack becomes more than zero, the thermal layer becomes more thinner near the lower side than the upper side because the stagnation move towards the lower side of the plate.

Average Nusselt number

The average Nusselt number is calculated according to the following steps :-

1. First we obtain an equation, which represent the local Nusselt number by analyzing the data obtained by Quasi Newton method. The equation is a 4 th order, as follows:

$$\overline{Nu} = a + b\bar{y} + c\bar{y}^{-2} + d\bar{y}^{-3} + e\bar{y}^{-4}$$

Where a , b , c , d , e are constants .

2 . The local Nusselt number then integrated over the plate to obtain the average Nusselt number as :

$$Nu^{\bullet} = \frac{\int_{-d}^{+d} Nu\, dy}{\int_{-d}^{+d} dy}$$

The numerically determined average Nusselt number is plotted figure (6). Results are presented for the range of Reynolds numbers between 40 and 140.

As shown from the figure, the average Nusselt number increases with increasing Reynold numbers, because the thermal layer becomes thinner with increasing Reynolds number

Local Nusselt number

The local Nusselt number is obtained as:

$$q = -k\,(dt \backslash dx)\,y = h\,(T\infty - Tw)$$

By substituting the derivatives we obtain :

$$Nu = \left(\frac{d\theta}{d\overline{x}}\right)_{\overline{y}}$$

Figure (7) shows the local Nusselt number at angle of attack (Beta=0) and for Pe=30 & Pe=100, as shown the local Nusselt number increases with increasing Peclet number.

The variation of local Nusselt number with the angle of attack for Pe=30 is shown in figure (8), it is observed that when the angle of attack becomes more than zero, the local Nusselt number is higher at the lower side than the upper side because the stagnation point moves toward the lower side so the thermal layer becomes more thinner than the upper side.

Variation of the Average Nusselt Number with Angle of Attack

Figure (9) shows the variation of average Nusselt number with angle of attack for Pe=30, the figure shows that the average Nusselt number decreases with increasing the angle attack because of the same reason mentioned above in temperature profile.

CONCLUSIONS

1. The temperature profiles show that the thermal layer is sensitive to the Reynolds number. In particular the thermal layer thickness decrease with an increase in Reynolds number .
2. The temperature gradient at the plate is least at the stagnation point (Beta=0) and grows larger with the increase of the distance from stagnation point, which in contrast with the conventional boundary layer because of the presence of the corner.
3. Average Nusselt number increase with increasing Reynolds number and decrease with increasing the angle of attack for fixed Reynolds number.

NOMENCLATURE

T	Temperature	°C
X	Co-ordinate in the direction of oncoming flow	
Y	Co-ordinate in transverse of oncoming flow	
$\overline{X}$	Dimensionless X co-ordinate	
$\overline{Y}$	Dimensionless Y co-ordinate	
u	Velocity component in X direction	m/s
y	Velocity component in Y direction	m/s
U	Dimensionless velocity in X direction	
V	Dimensionless velocity in Y direction	
U_∞	Free stream velocity	m/s
d	Half-height of the plate	M
Pe	Peclet number	
Re	Reynolds number	
Nu	Nusselt number	
β	Angle of attack	Degree
α	Thermal diffusivity	m^2/s
θ	Reduced temperature	
h	Heat transfer coefficient	$W/m^{2\circ}C$

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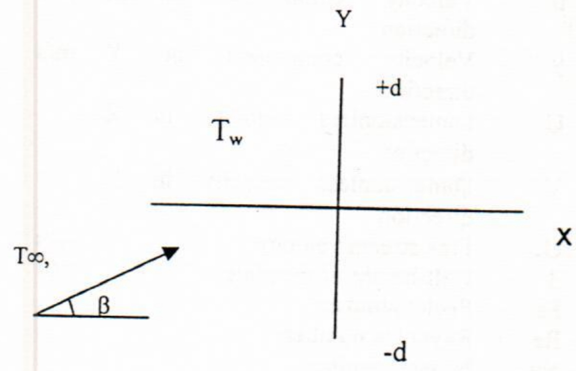

Fig. (1), Fluid attack a vertical plate at different angle

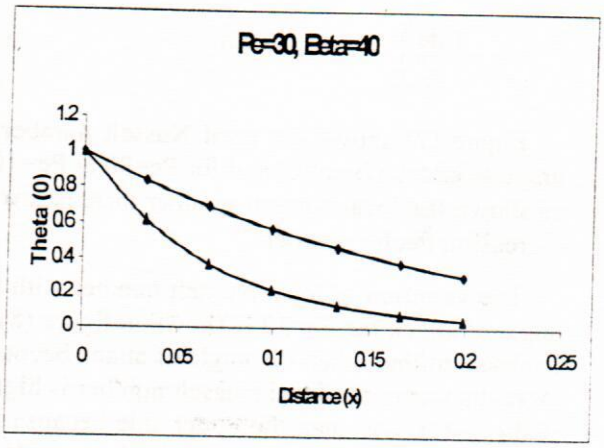

Fig. (4) Variation of temperature profile with angle of attack

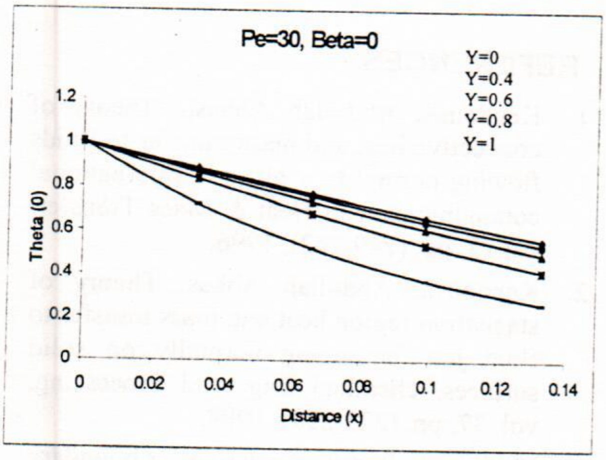

Fig. (2), Temperature profile

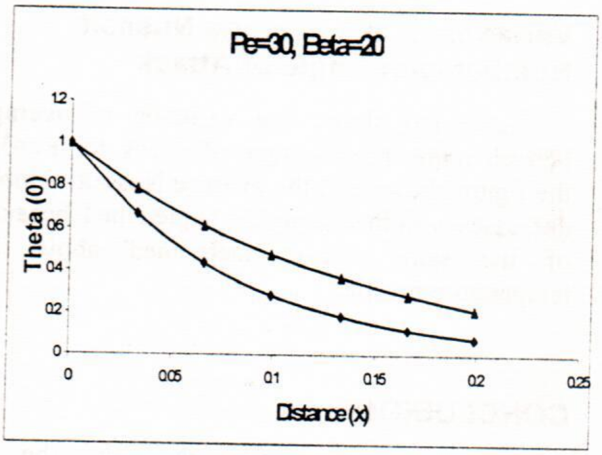

Fig. (5) Variation of temperature profile with angle of attack

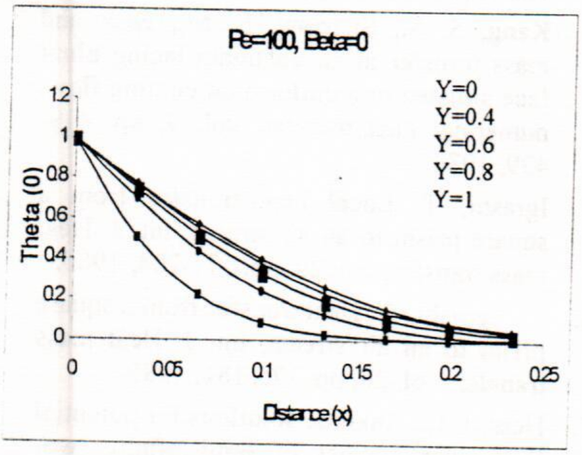

Fig. (3), Temperature profile

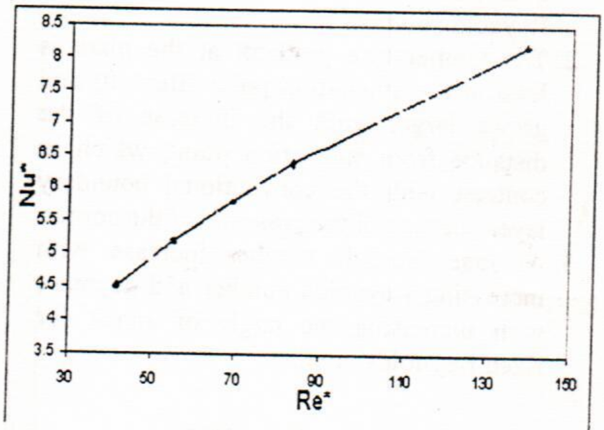

Fig. (6) Average Nusselt number

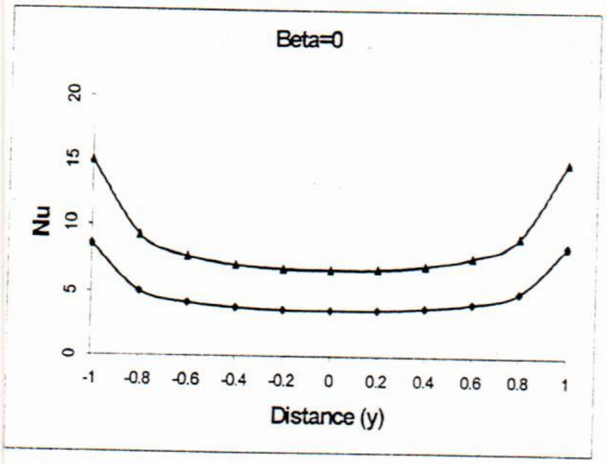

Fig. (7) Local Nusselt number

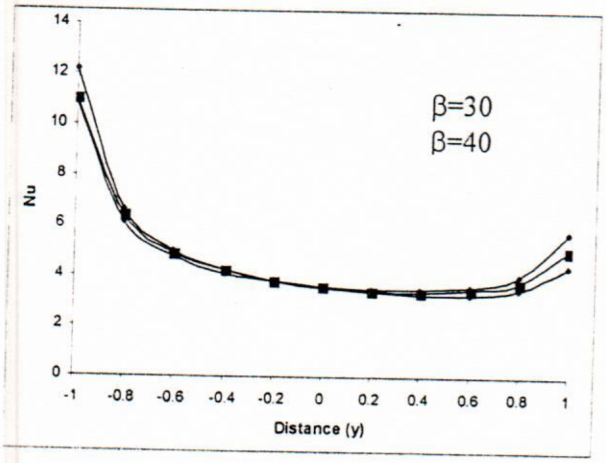

Fig. (8) Variation of local Nusselt number with angle of attack

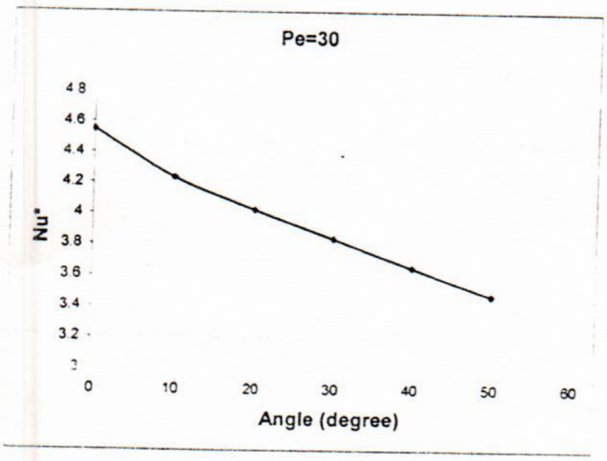

Fig. (9) Variation of average Nusselt number with angle of attck