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A novel concept for convective heat transfer enhancement

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Abstract—An analog between convection and conduction with heat sources is made to have a further understanding of the mechanism of convective heat transfer. There are three ways to raise the strength of heat sources/convection terms, and consequently to enhance the heat transfer: (a) increasing Reynolds and/or Prandtl number, (b) increasing the fullness of dimensionless velocity and/or temperature profiles, (c) increasing the included angle between the dimensionless velocity and temperature gradient vectors. Some approaches of heat transfer enhancement are suggested based on such a novel concept of heat transfer enhancement. © 1998 Published by Elsevier Science Ltd. All rights reserved.

1. INTRODUCTION

The well-known concepts of passive techniques of convective heat transfer enhancement [1–3] are to increase the heat transfer area and/or the convective heat transfer coefficient in terms of: (a) mixing the main flow and/or the flow in the wall region as by using rough surface, inserts, etc.; (b) reducing the flow boundary layer thickness by using offset strip fins, jet impingement, etc.; (c) creating the rotating and/or the secondary flow by using swirl flow device, duct rotation, etc.; (d) raising the turbulence intensity by using rough surface, turbulence promoter, etc.

The present novel concept of heat transfer enhancement results from a second look at the mechanism of convective heat transfer. With this concept, a number of heat transfer phenomena and conventional approaches of heat transfer enhancement may be known from a different point of view on the one hand, and some new approaches of heat transfer enhancement have been suggested on the other hand.

2. ANALOG OF CONVECTION AND CONDUCTION

In view of the fact that the convective heat transfer is essentially the conductive heat transfer under fluid motion [4], it is reasonable to study the analog between heat convection and heat conduction. Consider two cases, (a) the 2-D boundary-layer steady flow over a cold flat plate at zero incident angle, and (b) the 1-D steady-state heat conduction with heat sources between two parallel flat plates, as shown in Fig. 1. Their energy equations are written, respectively, for comparison as follows:

$$\rho C_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) \quad (1a)$$

$$-\dot{q}=\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right). \quad (1b)$$

It is easy to find that the convective term in the energy equation for the boundary layer flow corresponds to the heat source term in the heat conduction equation. It can be observed in Fig. 1 that the temperature gradients in the medium rises in the heat flow direction due to the presence of heat sources for both the conduction and convective problems. In addition, the temperature gradients reach their maximums at the cold boundaries. The integral of the equation (1a) over the thermal boundary layer thickness leads to

$$\int_0^{\delta_t}\rho C_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)\mathrm{d}y=-k\frac{\partial T}{\partial y} \quad (2)$$

which indicates the wall heat flux is equal to the overall strength of heat sources inside the thermal boundary layer. This implies that the convective heat transfer can be enhanced by raising the value of the integral of convection term (heat sources) over the thermal boundary layer.

3. POSSIBLE WAYS TO RAISING THE HEAT SOURCE STRENGTH

The equation (2) may be rewritten with the convection term in the vector form:

$$\rho C_p\int_0^{\delta_t}(\mathbf{U}\cdot\nabla T)\,\mathrm{d}y=-k\frac{\partial T}{\partial y} \quad (3)$$

where the thermophysical properties ρ, C_p are assumed constant.

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NOMENCLATURE

C_p	specific heat capacity at constant pressure	U	velocity vector
k	thermal conductivity	∇T	temperature gradient
Pr	Prandtl number	$\bar{\mathrm{U}}$	dimensionless velocity vector
$\dot{q}$	strength of heat source	U_∞	free-stream fluid velocity
q_w	wall heat flux	$\nabla \bar{T}$	dimensionless temperature gradient.
Re	Reynolds number		
T	temperature	Greek symbols	
T_w	wall temperature	β	included angle between fluid velocity and temperature gradient
T_∞	free-stream fluid temperature	δ_f	flow boundary layer thickness
u, v	velocity component along x- and y-coordinate, respectively	δ_t	thermal boundary layer thickness
		ρ	density.

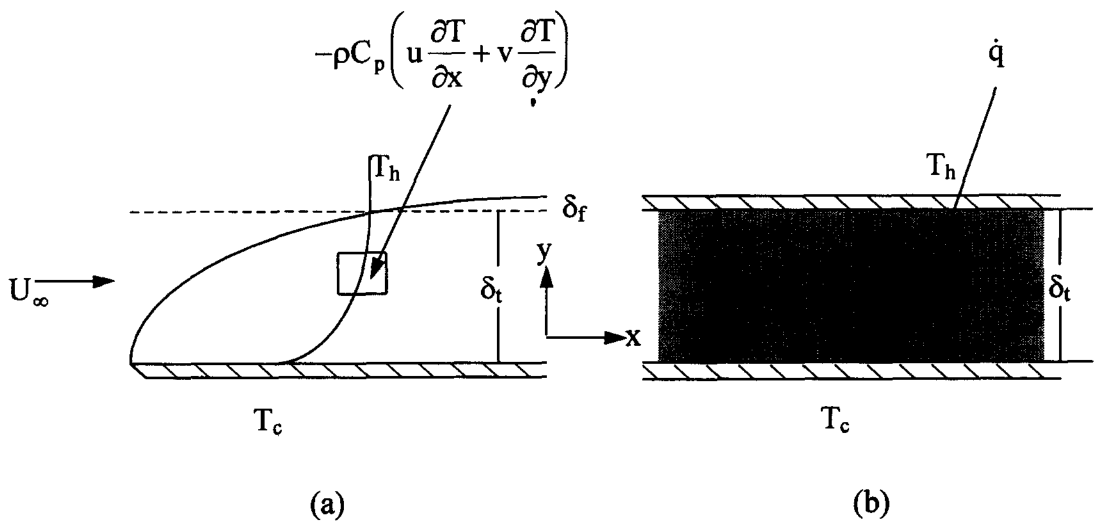


Fig. 1. Analog between (a) 2-D boundary layer flow and (b) 1-D heat conduction with heat sources.

The non-dimensionalization of equation (3) leads to

$$Re_x\, Pr \int_0^1 (\bar{\mathrm{U}}\cdot\nabla\bar{T})\,\mathrm{d}\bar{y} = Nu_x \quad (4)$$

where

$$\bar{\mathrm{U}} = \frac{\mathrm{U}}{U_\infty},\quad \nabla\bar{T} = \frac{\nabla T}{(T_\infty - T_w)/\delta_t},\quad \bar{y} = \frac{y}{\delta_t},\quad T_\infty > T_w.$$

It is evident that $Nu_x/(Re_x, Pr) \equiv St_x = I = \int_0^1(\bar{\mathrm{U}}\cdot\nabla\bar{T})\,\mathrm{d}\bar{y} \leqslant 1$ is dependent on the flow and temperature fields, and I must be a function of Reynolds and Prandtl numbers. In equation (4), $Re_x\, Pr(\bar{\mathrm{U}}\cdot\nabla\bar{T})$ represents the strength of the local dimensionless heat source in the thermal boundary layer flow. Hence, the various ways of increasing the overall strength of heat sources over the thermal boundary layer can be classified into three categories: (a) increasing Reynolds and Prandtl numbers; (b) increasing the fullness of dimensionless velocity and temperature profiles; (c) increasing the included angle between dimensionless velocity and temperature gradient vectors.

4. APPLICATIONS OF THE PRESENT NOVEL CONCEPT

4.1. *Further understanding of some phenomena of heat transfer/heat transfer enhancement*

Because it is well-known that Nusselt number increases with increasing the Reynolds number and Prandtl number, the present discussions are focused on the effects of the fullness of velocity and temperature profiles, and the included angle between two vectors on the heat transfer coefficient.

(a) The exact solution [5] of governing equations for the laminar flow past a wedge shows that Nusselt numbers become larger as the favorable pressure gradient goes up. Its physical mechanism lies in the

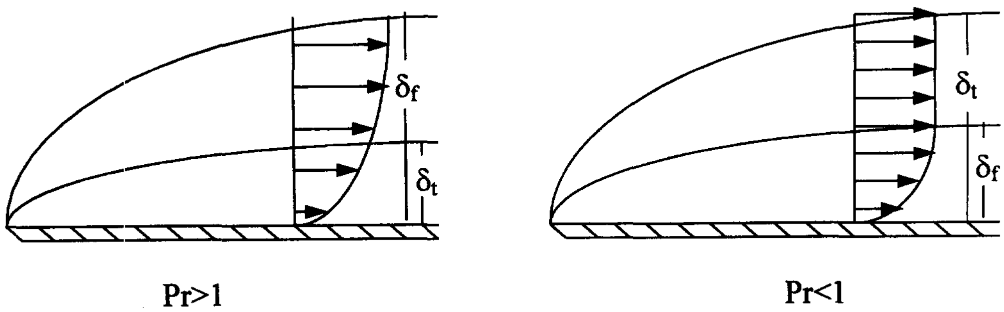


Fig. 2. The fullness of velocity profile in the flow boundary layer for $Pr > 1$ and $Pr < 1$.

fact that the favorable pressure gradient makes the velocity profile fuller, and consequently the strength of heat source higher in the boundary layer. It can be clarified from the value of I:

For the plat plate: $I = 0.322 Re_x^{-0.5} Pr^{-0.67}$, as $Pr \geqslant 0.7$

For the stagnation point: $I = 0.570 Re_x^{-0.5} Pr^{-0.67}$, as $Pr \geqslant 0.7$

(b) For the boundary layer flow with $Pr < 1$, the fullness of the velocity profile is greater than that for the cases of $Pr > 1$ because the thermal boundary layer is thicker than the flow boundary layer, as shown in Fig. 2. It turns out that the value of the integral $I = \int_0^1 (\mathbf{U} \cdot \nabla \bar{T})\, d\bar{y}$ increases with decreasing the Prandtl number. This explains why the Nusselt number is proportional to $Pr^{1/2}$, as $Pr \ll 1$, and to $Pr^{1/3}$, as $Pr \geqslant 0.6$ [6]. It is expected from this point of view that, the power index of Pr in the heat transfer correlation should have a continuous change from 0.5 to 0.333 as Pr varies from 0 to ∞.

(c) Consider a fully developed laminar flow in the channel composed of two parallel flat plates which are kept at different temperatures, T_h and T_c, respectively, as shown in Fig. 3. In this case, the velocity profile no longer changes in the flow direction, that is, the streamlines are parallel to the flat plates. The isotherms are parallel to the flat plates too for the developed thermal region. Consequently, the velocity and the temperature gradient are parallel to each other. As a result, the convection term/heat source term in the energy equation (1a) disappears and the convective problem then reduces to the conduction problem. The linear temperature distribution in the medium normal to the flow direction is an evidence for it. This means that the common belief that the fluid motion enhances the heat flux is not always true. In other words, not only the absolute value of $\mathbf{U}$ and ∇T vectors, but also their included angles of the $\mathbf{U}$ and ∇T vectors determines the convection effect on the heat transfer.

(d) For the fully developed laminar duct flow, $Nu = 4.36$ for the case of the isoflux thermal boundary condition, while $Nu = 3.66$ for the case of the isothermal boundary condition. The difference between these two cases can be well explained based on the concept of the included angle between the velocity and temperature gradient vectors. Numerical results indicate for the given Reynolds number that the included angles of the $\mathbf{U}$ and ∇T for the duct flow subject to the isoflux boundary condition are larger than that for the isothermal boundary condition, as shown in Figs. 4 and 5. It has been verified that the included angles are only the function of radial coordinate r for the fully developed duct flows. The included angle β will be 90° at the isothermal boundary, while it is larger at the isoflux boundary.

4.2. *Developing some new approaches of convective heat transfer enhancement*

Based on the concept that the heat transfer is enhanced as long as the included angle and/or the

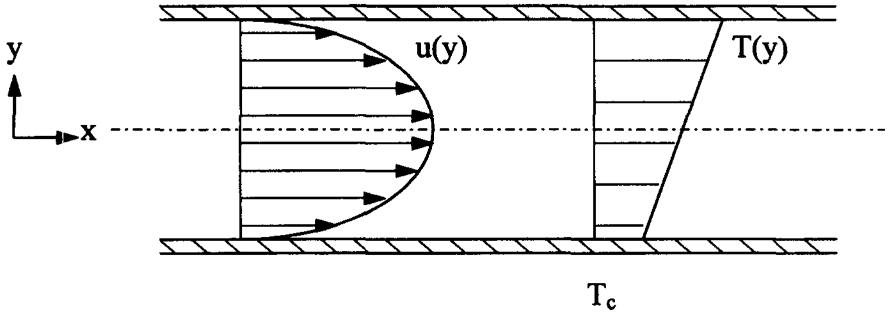
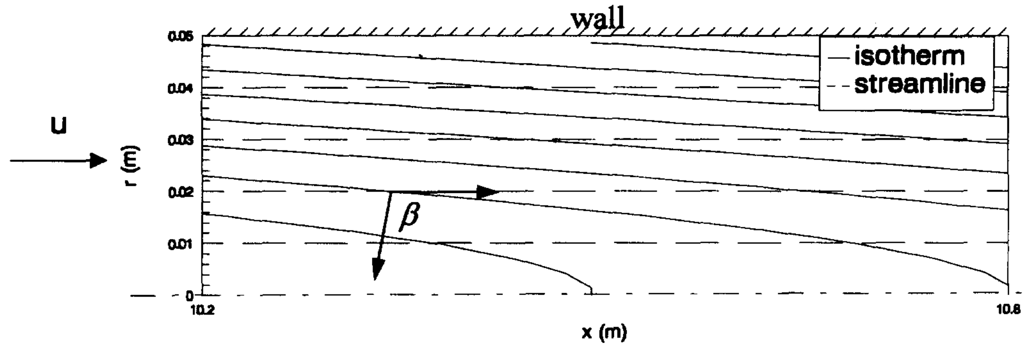
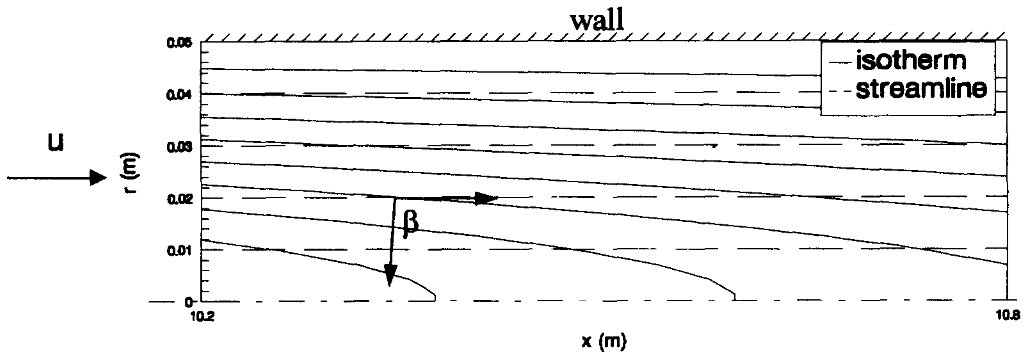


Fig. 3. Fully developed laminar flow between two parallel plates at different temperature.



(a) isoflux boundary condition



(b) isothermal boundary condition

Fig. 4. Isotherms and streamlines of fully developed duct flows.

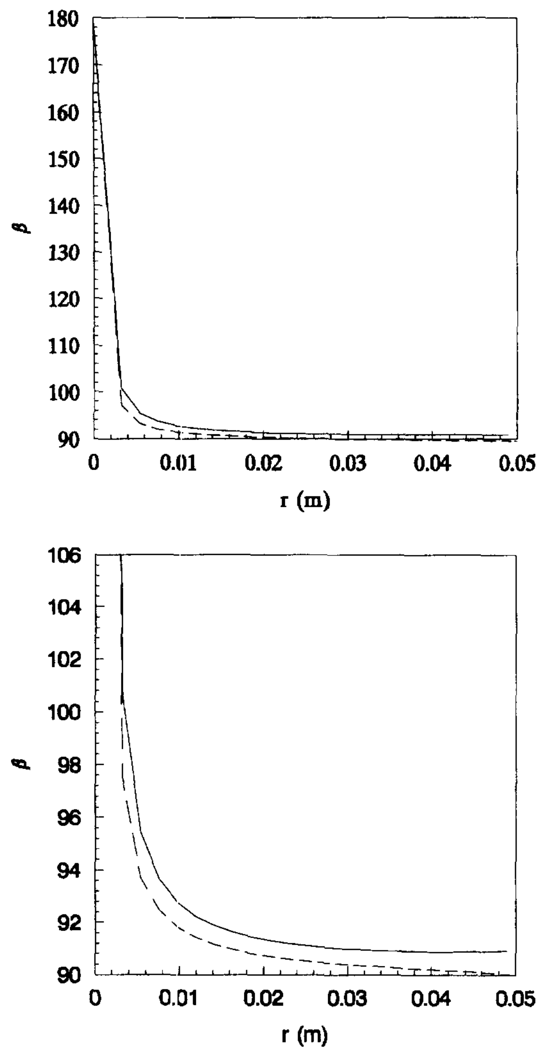


Fig. 5. Included angles of the velocity and temperature gradient vectors.

profile fullness increase, it is expected to develop some new approaches of heat transfer enhancement.

(a) The convective heat transfer could be enhanced by using the constracting duct.

The duct constraction will induce a favorable pressure gradient, which leads to the fluid acceleration and consequently to the increase of the velocity profile fullness. The heat transfer is enhanced therefore.

(b) The convective heat transfer can be enhanced by changing the thermal boundary condition.

As mentioned above, the Nusselt number of duct flow is larger at isoflux boundary condition than at isothermal boundary condition due to its favorable included angle between the velocity and temperature gradient vectors. If the wall heat flux increases along the flow direction, the included angle and the consequent Nusselt number of duct flow will be larger than that for the isoflux boundary condition.

(c) The convective heat transfer could be enhanced by using special inserts.

The inserts can be specially designed for the purpose majorly to increase the included angle between **U** and ∇T vectors, rather than to promote turbulence, so that the heat transfer is considerably enhanced with as little pressure loss as possible.

5. CONCLUDING REMARKS

(1) It is reasonable and useful to regard the convective heat transfer problem as a conduction problem with heat sources. With such an analog we not only can have a deeper understanding of convective heat transfer, but also develop some new approaches of heat transfer enhancement.

(2) There are three possible ways to raise the strength of heat sources/convection terms, and consequently to enhance the heat transfer. One among them is to increase the included angle of the **U**–∇T vectors, with which some approaches of heat transfer enhancement are suggested.

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