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Field synergy principle of heat and mass transfer

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Simultaneous heat and mass transfer widely exists in nature and engineering, and it is of vital importance to enhance heat and mass transfer efficiency. In this paper, field synergy equation of heat and mass transfer is derived from its energy equation. Results show that the total transferred heat (including the conducted heat and the heat transferred by mass diffusion through the heat transfer interface) is determined by the values of fluid velocity and enthalpy gradient as well as the value of synergy angle α of velocity vector and enthalpy gradient field. Decreasing the value of α enhances the heat and mass transfer. This means the higher the synergy of velocity vector and enthalpy gradient field, the higher the total transferred heat. By the synergy principle of heat and mass transfer, some methods may be developed to improve the heat and mass transfer efficiency.

heat and mass transfer, energy equation, enthalpy gradient field, field synergy

There are many processes of momentum, heat and mass transfer simultaneously in engineering application, and they interact with each other. For instance, when hot air goes through wetted surface, heat transfers from hot air to the wetted surface by convective heat transfer; and the water vapor evaporated from the wetted surface carrying with its own enthalpy flows into the hot air. Under different evaporation rates, heat and mass transfer between hot air and wetted surface are different. Inaba et al.[1] and Feddaoui et al.[2] analyzed the influences of *Re* number, inlet liquid temperature and inlet liquid flow rate on heat and mass transfer efficiency between air and water through experimental and numerical research, respectively. Dowdy et al.[3] and Yan et al.[4,5] developed a series of apparatus and devices for enhancing heat and mass transfer efficiency between air and water. In liquid dehumidification conditioning, it is of vital importance to improve heat and mass transfer efficiency between liquid desiccant and air for the total efficiency of air conditioning system. Many researchers took up with heat and mass transfer of liquid dehumidification processes[6–8]. In aero-engine, it is a significant subject on how to cool aero-turbine blade effectively, and many scholars performed in-depth researches and analysis[9–12] on gas film cooling and internal cooling, pointing out the influences of different film hole geometries and angles as well as the heat transfer characteristics and gas dynamics of internal cooling passages on cooling efficiency. From the typical examples of researches on heat and mass transfer, people tend to change experimental methods or devices to enhance or decrease heat and mass transfer efficiency, whereas no physical mechanism has been discussed or identified.

To improve heat transfer performance, Guo[13] introduced field synergy principle of convective heat transfer, which reveals that the convective heat transfer intensity is related to not only temperature gradient, fluid velocity and properties, but also the included angle of the velocity field and the temperature gradient field. Then the field synergy principle of heat exchanger was put forward[14]. According to the field synergy principle of convective heat transfer, Chen et al.[15] analyzed the process of convective mass transfer and derived the field synergy equation of convective mass transfer applied in photocatalytic oxidation reactors. The filed synergy principle was developed to fluid flow and flow drag reduction[16]. However, no literature has been reported on

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how to enhance or decrease the transfer mechanism as to the process of heat and mass transfer simultaneously at present.

In this paper, we derived field synergy equation of heat and mass transfer within the two-dimensional enthalpy boundary by giving out the mathematic description of energy equation with heat and mass transfer for two-dimensional incompressible Newtonian fluid from the first law of thermodynamics, law of mass conservation, law of energy conservation and continuity equation, and enlarged it to the general form. Results show that the total heat flux with mass transport is affected by the value of fluid velocity rate, the value of enthalpy gradient and the value of included angle α between velocity vector and enthalpy gradient. Low value of α leads to high value of total transferred heat. Therefore it is important to reduce the value of α by enhancing the total heat transfer in the process of heat and mass transfer.

1 Process of heat and mass transfer

Mass transfer is always in accordance with heat transfer, even in the isothermal process. This is caused by the enthalpy transfer in the process of mass transfer. Due to mass component transport, heat transfer in isothermal processes per unit time per unit area is

$$q_m = \sum_{i=1}^{m} N_i c_{pi} T, \tag{1}$$

where N_i is the mass transfer rate of component i, kg/(m^2·s); c_{pi} is the specific heat of component i, kJ/(kg·K); T is the temperature of component i. If temperature difference exists in the system, the total transferred heat q_t is

$$q_t = q_h + q_m, \tag{2}$$

where $q_h = -\lambda \dfrac{\mathrm{d}T}{\mathrm{d}y}$ if there only exists heat conduction in the system and λ is the heat conductivity, W/(m·K) or $q_h = h\Delta T$ if there exists heat convection in the system and h is the convective heat transfer coefficient, W/(m^2·K).

2 Mathematical descriptions of the heat and mass transfer process

For simplicity, we assume that (a) the flow is two-dimensional; (b) the flow is incompressible Newtonian fluid; (c) the fluid properties are constant; (d) the heat dissipation caused by viscosity can be negligible.

Figure 1 shows the schematic diagram of flow, heat transfer and mass transfer in laminar boundary layer. Take the micro-element for consideration as Figure 2 shows in Cartesian coordinate system, fluid flows in and out of the boundary, which is an open system. By use of the first law of thermodynamics, we have

$$\Phi = \frac{\partial U}{\partial \tau} + (q_m)_{\mathrm{out}} \left(h + \frac{1}{2}V^2 + gz \right)_{\mathrm{out}} - (q_m)_{\mathrm{in}} \left(h + \frac{1}{2}V^2 + gz \right)_{\mathrm{in}} + W_{\mathrm{net}}, \tag{3}$$

where q_m is mass flow rate, kg/s; h is the specific enthalpy of fluid, kJ/kg; V is the velocity of fluid; U is the internal energy of fluid; W_{net} is the net work done by the fluid, and the value of it is zero here. Neglecting the potential and kinetic energy change of the fluid, yields

$$\Phi = \frac{\partial U}{\partial \tau} + (q_m)_{\mathrm{out}} h_{\mathrm{out}} - (q_m)_{\mathrm{in}} h_{\mathrm{in}}. \tag{4}$$

Next the concrete expressions of each parameter in eq. (4) are derived.

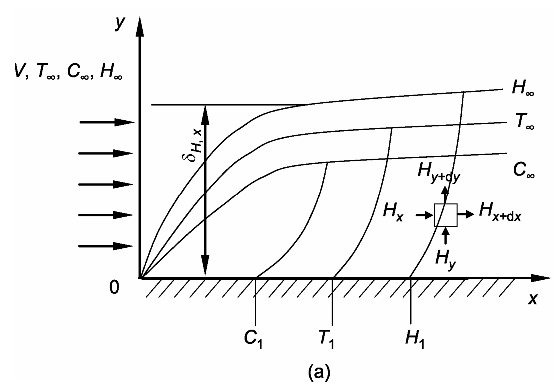

Figure 1 Schematic diagram of flow, heat transfer and mass transfer in the laminar boundary layer.

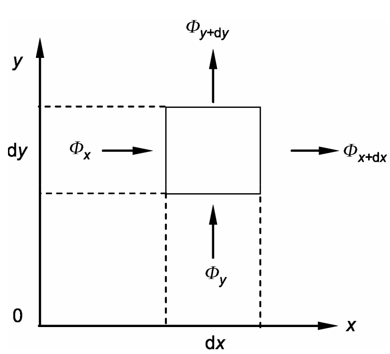

Figure 2 Schematic diagram of the micro-element.

2.1 The variation of internal energy

During the time interval of $\mathrm{d}\tau$, the temperature change of fluid in the micro-element is $\frac{\partial T}{\partial \tau}\mathrm{d}\tau$ and the concentration change is $\frac{\partial C}{\partial \tau}\mathrm{d}\tau$. Therefore the internal energy change is

$$\Delta U=\left(C+\frac{\partial C}{\partial \tau}\mathrm{d}\tau\right)\left(T+\frac{\partial T}{\partial \tau}\mathrm{d}\tau\right)c_p\,\mathrm{d}x\,\mathrm{d}y\,\mathrm{d}\tau$$
$$-CTc_p\,\mathrm{d}x\,\mathrm{d}y\,\mathrm{d}\tau=\frac{\partial H_v}{\partial \tau}\mathrm{d}x\,\mathrm{d}y\,\mathrm{d}\tau, \tag{5}$$

where C is the concentration of fluid, and H_v is the enthalpy per unit volume.

2.2 The variation of enthalpy

From x-coordination, the enthalpy input to the micro-element within $\mathrm{d}\tau$ interval is

$$H_x=Cc_p uT\,\mathrm{d}y\,\mathrm{d}\tau. \tag{6}$$

The enthalpy output of the micro-element within $\mathrm{d}\tau$ interval is

$$H_{x+\mathrm{d}x}=\left(C+\frac{\partial C}{\partial x}\mathrm{d}x\right)\left(u+\frac{\partial u}{\partial x}\mathrm{d}x\right)\left(T+\frac{\partial T}{\partial x}\mathrm{d}x\right)c_p\,\mathrm{d}y\,\mathrm{d}\tau. \tag{7}$$

Therefore, the net enthalpy input to the micro-element caused by fluid flow in x direction within $\mathrm{d}\tau$ interval is

$$H_{x+\mathrm{d}x}-H_x=\left(uT\frac{\partial C}{\partial x}+Cu\frac{\partial T}{\partial x}+CT\frac{\partial u}{\partial x}\right)c_p\,\mathrm{d}x\,\mathrm{d}y\,\mathrm{d}\tau. \tag{8}$$

In a similar way, the net enthalpy input to the micro-element caused by fluid flow in the y direction within $\mathrm{d}\tau$ interval is

$$H_{y+\mathrm{d}y}-H_y=\left(vT\frac{\partial C}{\partial y}+Cv\frac{\partial T}{\partial y}+CT\frac{\partial v}{\partial y}\right)c_p\,\mathrm{d}x\,\mathrm{d}y\,\mathrm{d}\tau. \tag{9}$$

Integrating eqs. (8) and (9) and the continuity equation $\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0$, we can obtain that the net enthalpy input to the micro-element caused by fluid flow within $\mathrm{d}\tau$ interval is

$$(q_m)_{\mathrm{out}}h_{\mathrm{out}}-(q_m)_{\mathrm{in}}h_{\mathrm{in}}=\left(u\frac{\partial H_v}{\partial x}+v\frac{\partial H_v}{\partial y}\right)\mathrm{d}x\,\mathrm{d}y. \tag{10}$$

2.3 The variation of total heat

According to Fourier heat conduction law and Fick's Law, the total transfer Φ_x and Φ_y caused by heat conduction and mass transport from x=x and y=y into the micro-element per unit time are as follows:

$$\Phi_x=-\lambda\frac{\partial T}{\partial x}\mathrm{d}y+N_x c_p T, \tag{11}$$

$$\Phi_y=-\lambda\frac{\partial T}{\partial y}\mathrm{d}x+N_y c_p T, \tag{12}$$

where $N_x=-D\frac{\partial C}{\partial x}\mathrm{d}y$ is the mass transfer rate into the micro-element from the x direction; D is the diffusion factor; $N_y=-D\frac{\partial C}{\partial y}\mathrm{d}x$ is the mass transfer rate into the micro-element from the y direction.

In the same way, the total transfer caused by heat conduction and mass transport from x=x+dx and y=y+dy out of the micro-element per unit time are as follows:

$$\Phi_{x+\mathrm{d}x}=\Phi_x+\frac{\partial \Phi}{\partial x}\mathrm{d}x=\Phi_x+\frac{\partial}{\partial x}\left(-\lambda\frac{\partial T}{\partial x}\mathrm{d}y+N_x c_p T\right)\mathrm{d}x, \tag{13}$$

$$\Phi_{y+\mathrm{d}y}=\Phi_y+\frac{\partial \Phi}{\partial y}\mathrm{d}y=\Phi_y+\frac{\partial}{\partial y}\left(-\lambda\frac{\partial T}{\partial y}\mathrm{d}x+N_y c_p T\right)\mathrm{d}y. \tag{14}$$

Therefore the net total heat which goes into the micro-element within $\mathrm{d}\tau$ interval is

$$\Phi\,\mathrm{d}\tau=\begin{bmatrix}\lambda\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)-c_p\frac{\partial(TN_x)}{\partial x}\\-c_p\frac{\partial(TN_y)}{\partial y}\end{bmatrix}\mathrm{d}x\,\mathrm{d}y\,\mathrm{d}\tau. \tag{15}$$

Integrating the above equations yields the energy differential equation of two-dimensional heat and mass transfer with constant properties and no heat source:

$$\frac{\partial H_v}{\partial \tau}+u\frac{\partial H_v}{\partial x}+v\frac{\partial H_v}{\partial y}$$
$$=\lambda\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)-c_p\frac{\partial(TN_x)}{\partial x}-c_p\frac{\partial(TN_y)}{\partial y}. \tag{16}$$

Considering that the temperature and concentration variations are very small along the x direction (neglect heat and mass transfer in the x direction) in the boundary layer, we can simplify eq. (16) as follows for steady state flow:

$$u\frac{\partial H_v}{\partial x}+v\frac{\partial H_v}{\partial y}=\lambda\frac{\partial^2 T}{\partial y^2}-c_p\frac{\partial(TN_y)}{\partial y}. \tag{17}$$

Integrating from both sides of eq. (17), we can obtain

$$\int_0^{\delta_{H,x}}\left(u\frac{\partial H_v}{\partial x}+v\frac{\partial H_v}{\partial y}\right)\mathrm{d}y=-\lambda\left.\frac{\partial T}{\partial y}\right|_w+N_y c_p T_w=q_w, \tag{18}$$

where $\delta_{H,x}$ is the enthalpy boundary width, which can be defined as the thin layer where enthalpy varies drastically along with the heat transfer surfaces (as shown in Figure 1). We adopt the place where the surplus enthalpy equals to 99% of the coming fluid as the outer boundary of $\delta_{H,x}$.

The right-hand side of eq. (18) is the total heat flux with mass transport. It consists of conducted heat and the heat caused by mass transport, the expressions of which are given by eqs. (2) and (1), respectively. Obviously, heat transfer can be enhanced when the direction of mass transfer heat flux is identical with that of heat conduction's. Otherwise, the total transferred heat will decrease.

The above analysis is based on two-dimensional laminar boundary questions, for more general questions, we have

$$u\frac{\partial H_v}{\partial x}+v\frac{\partial H_v}{\partial y}+w\frac{\partial H_v}{\partial z}=\frac{\partial}{\partial x}\left(\lambda\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\lambda\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(\lambda\frac{\partial T}{\partial z}\right)-c_p\left[\frac{\partial(TN_x)}{\partial x}+\frac{\partial(TN_y)}{\partial y}+\frac{\partial(TN_z)}{\partial z}\right]+\dot{q},\quad(19)$$

where $\dot{q}$ is the real internal heat source, which could be radiation heat, chemical reaction heat, electrical and magnetic heat, dispassion heat caused by viscosity and so on. By integration, eq. (19) can be rewritten as

$$\int_0^{\delta_{H,x}}\begin{Bmatrix}\left(u\frac{\partial H_v}{\partial x}+v\frac{\partial H_v}{\partial y}+w\frac{\partial H_v}{\partial z}\right)\\-\left[\frac{\partial}{\partial x}\left(\lambda\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial z}\left(\lambda\frac{\partial T}{\partial z}\right)\right]\\+c_p\left[\frac{\partial(TN_x)}{\partial x}+\frac{\partial(TN_z)}{\partial z}\right]+\dot{q}\end{Bmatrix}\mathrm{d}\,y=-\lambda\left.\frac{\partial T}{\partial y}\right|_w+N_yc_pT_w=q_w.\quad(20)$$

The right-hand side of eq. (20) is the total heat flux through heat transfer surface and the left-hand side is the sum of all sources within the enthalpy boundary layer. The four items on the left-hand side of eq. (20) are convective source (heat transfer caused by fluid flow), heat conduction source (heat conduction parallel to heat transfer surface), mass transfer source (heat transfer caused by mass transport) and real heat source (including radiation heat, chemical reaction heat, electrical and magnetic heat, dispassion heat caused by viscosity and so on). According to eq. (20), we can recognize and explain some phenomena and methods of enhanced heat and mass transfer. Such as water-air system, air absorbing water vapor ability can be improved by water-jet evaporation; therefore, heat transfer can be enhanced. For liquid dehumidification system[17], dehumidifier goes through the mellapak column and enlarges the mass transfer area for the air to be handled; therefore, the total heat transfer in the process of dehumidification can be improved.

3 Field synergy equation of convective heat and mass transfer

Practically, when the dehumidifier absorbs the water vapor in the air, its dehumidification ability would decrease as the condensation heat would increase the temperature of the dehumidifier in the process of dehumidification. That is to say, heat transfer would suppress or even reduce mass transfer, and vice versa. This would lead to a low level of total heat flux on the surface of heat transfer. How to enhance the total heat flux relates to the field synergy of heat and mass transfer and fluid velocity. For simplification, we adopt two-dimensional enthalpy boundary and change the convective term of eq. (18) as follows:

$$\int_0^{\delta_{H,x}}\left(\boldsymbol{V}\cdot\nabla H_v\right)\mathrm{d}\,y=-\lambda\left.\frac{\partial T}{\partial y}\right|_w+N_yc_pT_w=q_w,\quad(21)$$

where $\boldsymbol{V}$ is the velocity vector of fluid flow. We set the included angle of velocity vector and enthalpy gradient to be α, and yields

$$\int_0^{\delta_{H,x}}\left(|\boldsymbol{V}|\cdot|\nabla H_v|\cos\alpha\right)\mathrm{d}\,y=-\lambda\left.\frac{\partial T}{\partial y}\right|_w+N_yc_pT_w=q_w.\quad(22)$$

Eq. (21) is the field synergy equation of heat and mass transfer within the two-dimensional enthalpy boundary. When there only exists heat conduction on the heat transfer surface, eq. (22) can be written as the field synergy equation of convective heat transfer in ref. [13]; when there only exists mass transfer on the heat transfer surface, eq. (22) can be written as the field synergy equation of convective mass transfer in ref. [15], proving that eq. (22) is right and is a comprehension and development of convective heat transfer and convective mass transfer field synergy equations.

From eq. (22), we can obtain that the total heat flux on the heat transfer surface is decided by three factors: (i) the value of fluid velocity rate. Under the same conditions, the higher the velocity rate, the higher the integral value of the right-hand side of eq. (22), so the higher of the total heat transfer; (ii) the value of enthalpy gradient. Under the same conditions, the higher the enthalpy gradient, the higher the total heat transfer; (iii) the included angle value of velocity vector and enthalpy gradient α. When α changes within 0—90°, the higher value of α, the lower the total heat transfer. Therefore it is advantageous to reduce the value of α by enhancing the total heat transfer in the process of heat and mass transfer.

4 Application

We take a very typical example to exhibit the great effect of α. Fluid of temperature T_2 flows through the heat transfer surface of temperature T_1 as Figure 3 shows. Component A goes through the surface transport mass with a rate of N_A and concentration of C_1 to the main fluid with concentration C_2 of component A.

As temperature and concentration variations along x and z axes direction can be neglected, the total heat into the micro-element can be divided into two parts: (i) heat conduction q_1 ($q_1 = -\lambda \frac{\mathrm{d}T}{\mathrm{d}y}$) caused by temperature gradient; (ii) heat transfer q_2 ($q_2 = N_A c_{PA}(T - T_0)$) caused by the transport of component A.

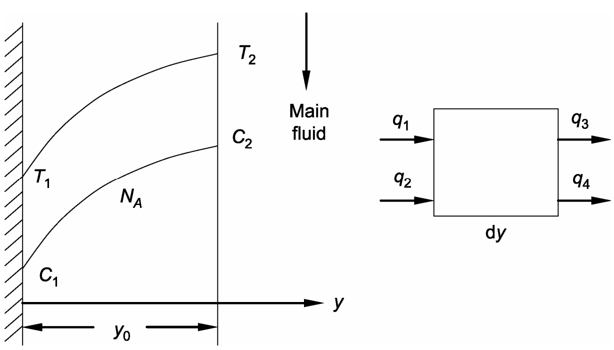

Figure 3 Schematic diagram of heat and mass transfer.

At a steady state, the heat flux which goes into the micro-element equals that flowing out of the element, so we have

$$\lambda \frac{\mathrm{d}^2 T}{\mathrm{d}y^2} - N_A c_{PA} \frac{\mathrm{d}T}{\mathrm{d}y} = 0. \tag{23}$$

We set $h = \lambda / y_0$, $\frac{N_A c_{PA}}{h} = C_0$, and substitute the boundary conditions, when y=0, T=T_1 (T_1 is the wall temperature); when y=y_0, T=T_2 (T_2 is the temperature of the main fluid), therefore the temperature distribution in the thin film is

$$T(y) = T_1 + (T_2 - T_1)\frac{\mathrm{e}^{\frac{C_0}{y_0}y} - 1}{\mathrm{e}^{C_0} - 1}, \tag{24}$$

where C_0 is called as Ackermann modification coefficient, which implies the effect of mass transport rate and its direction on heat transfer. According to the temperature distribution, we can obtain the conducted heat on the heat transfer surface as

$$q_1 = -\lambda \left.\frac{\mathrm{d}T}{\mathrm{d}y}\right|_{y=0} = -\frac{\lambda}{y_0}(T_2 - T_1)\frac{C_0}{\mathrm{e}^{C_0} - 1} = h(T_1 - T_2)\frac{C_0}{\mathrm{e}^{C_0} - 1}. \tag{25}$$

And the total heat flux is

$$q_t = q_1 + q_2 = h(T_1 - T_2)\frac{C_0}{\mathrm{e}^{C_0} - 1} + hC_0(T_1 - T_2) = h(T_1 - T_2)\frac{C_0}{1 - \mathrm{e}^{-C_0}}. \tag{26}$$

At a steady state, the mass transport rate $N_A = -D\frac{\partial C}{\partial y}$ from the wall to the main fluid is a constant, which means

$$D\frac{\partial^2 C}{\partial y^2} = 0. \tag{27}$$

By substituting the boundary conditions, when y=0, C=C_1 (C_1 is the concentration on the wall); when y=y_0, C=C_2 (C_2 is the concentration of main fluid). By solving the differential eq. (27), we have the concentration distribution in the thin film:

$$C = C_1 + (C_2 - C_1)y/y_0. \tag{28}$$

Then the enthalpy distribution in the thin film is obtained as

$$H_v = CTc_p = c_p\left[C_1 + (C_2 - C_1)y/y_0\right]\left[T_1 + (T_2 - T_1)\frac{\mathrm{e}^{\frac{C_0}{y_0}y} - 1}{\mathrm{e}^{C_0} - 1}\right]. \tag{29}$$

Based on the above equations, the curves of the included angle α are obtained with the relationship of boundary depth y as Figure 4 shows, and the curve indicating the relationship between α and the dimensionless total heat flux $\frac{q_t}{h(T_1 - T_2)}$ on the wall as Figure 5 shows (Here we set T_1<T_2 and C_1>C_2).

Figure 4 shows that with the increase of C_0 (increases the mass transfer), the included angle α decreases, so the total transferred heat can be increased and this is

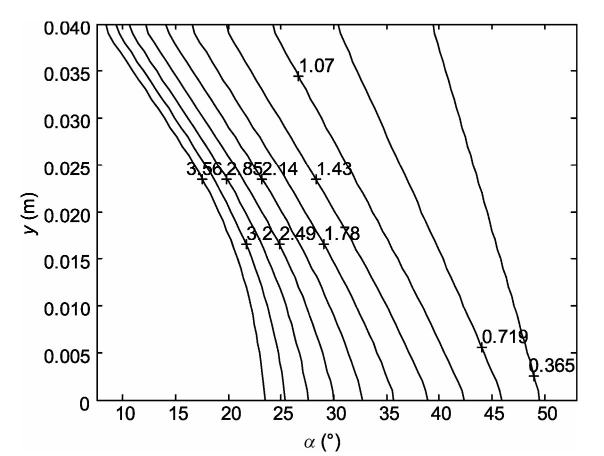

Figure 4 The relationship of boundary depth y versus α with different values of C_0.

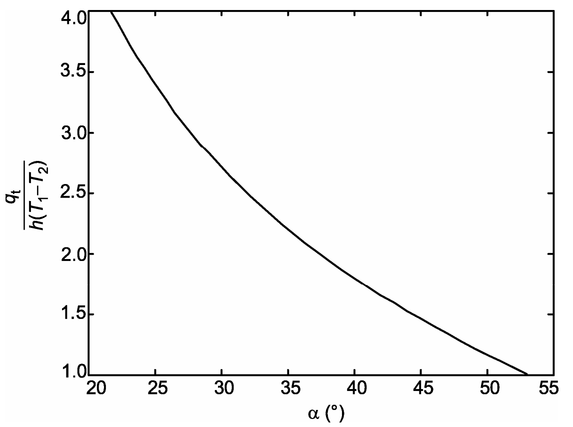

Figure 5 The relationship between $\frac{q_t}{h(T_1-T_2)}$ and α on the wall.

identical with eq. (26). Furthermore, α decreases with the increase of y versus different values of C_0.

Figure 5 shows that with the decrease of the included angle α of velocity vector and enthalpy gradient, the dimensionless total heat flux $\frac{q_t}{h(T_1-T_2)}$ on the wall increases, which is available for optimal design of the injection angle of the gas-film pores to take as much heat as possible using as less cooling gas as possible in aero-turbine blade cooling.

5 Conclusions

(i) We gave out the mathematic description of energy equation with heat and mass transfer for two-dimensional incompressible Newtonian fluid from the first law of thermodynamics, law of mass conservation, law of energy conservation and continuity equation. And it was enlarged to a general form. By the energy equation with heat and mass transfer, we derived the field synergy equation of heat and mass transfer.

(ii) The total heat flux with mass transport consists of heat conduct and the heat caused by mass transport. It is affected by three factors: (a) the value of fluid velocity rate. Under the same conditions, the higher the velocity rate, the higher of the total heat transfer; (b) the value of enthalpy gradient. Under the same conditions, the higher the enthalpy gradient, the higher the total heat transfer; (c) the included angle value of velocity vector and enthalpy gradient α. When α varies within 0—90°, the higher value of α, the lower the total heat transfer. Therefore it is important to reduce the value of α by enhancing the total heat transfer in the process of heat and mass transfer.

(iii) With the increase of C_0, the included angle α of velocity vector and enthalpy gradient decreases, while the total transferred heat can be increased. With the decrease of the included angle α, the dimensionless total heat flux $\frac{q_t}{h(T_1-T_2)}$ on the wall increases. This can be a guide for optimizing heat and mass transfer processes.

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