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Dissimilar control of turbulent friction drag and heat transfer based on the implication of stress and flux identity equations

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Abstract: We present identity equations relating the wall heat flux and turbulent heat flux under different thermal boundary conditions. Similarly to the identity equation for the skin friction (Fukagata et al., 2002), the Nusselt number can be decomposed into different contributions. By utilizing the difference between the identity equation for heat flux and that for skin friction, a strategy for the dissimilar control of momentum and heat is proposed. The effects of the control strategy are examined by means of direct numerical simulation.

Due to the similarity between momentum and heat transport, simultaneous achievement of skin friction reduction and heat transfer enhancement is usually difficult and no general control strategy has been established.

Recently, an identity equation relating the skin friction and the Reynolds shear stress has been presented (Fukagata et al., 2002). For a fully developed channel flow, for instance, it reads

$$C_f = \frac{12}{Re_b} + 24\int_0^1 (1-y)(-\overline{u'v'})\,dy\,. \tag{1}$$

The length, velocity, and time are nondimensionalized based on the channel half-width, δ^* and twice the bulk mean velocity, $2U_b^*$ (the subscript of $*$ denotes dimensional variables), and $Re_b = 2U_b^*\delta^*/\nu^*$ is the bulk Reynolds number. This identity equation implies that the suppression of near-wall Reynolds shear stress is primarily important for drag reduction. This implication has been verified, e.g., by a suboptimal control (Fukagata & Kasagi, 2004).

In the present study, a similar exact mathematical relationship is derived of the turbulent heat flux and the Nusselt number. A strategy for the above-mentioned dissimilar control is proposed by clarifying the quantitative relationship between them and by utilizing its difference from Eq. (1). Validity of the proposed control strategy is examined by DNS with idealized feedback control.

We consider fully developed turbulent channel flows at a constant flow rate. Two different thermal boundary conditions are considered:

1. constant, but different temperatures on two walls (constant temperature difference condition: CTD);
2. constant heat flux on two walls (isoflux wall condition: IFW).

In both cases, temperature fluctuation is assumed to be zero on the walls.

The identity equation on wall and turbulent heat fluxes can be derived by relevant integrations of the temperature transport equation. See, Fukagata et al. (2005) for details of derivation process.

For the CTD case, the identity equation reads

$$Nu = 4 + 4Re_b Pr\int_0^1 (-\overline{v'\vartheta'})dy\,. \tag{2}$$

The dimensionless temperature, ϑ, is defined as

$$\vartheta = (T^* - T^*|_{y=1})/\Delta T^* \tag{3}$$

and ΔT^* is half of the wall temperature difference. The first term in the right hand side is the laminar contribution, which is identical to heat conduction, and the second term is the turbulent contribution. The turbulent contribution is a simple integration of turbulent heat flux, $(-\overline{v'\vartheta'})$, and is different from that in Eq. (1), which has a weighting of $(1-y)$. This difference suggests that simultaneous achievement of drag reduction and heat transfer augmentation is made possible by suppressing the near-wall turbulence and enhancing turbulence in the central region.

For the IFW case, the identity equation reads

$$\begin{aligned}\frac{1}{Nu} &= \frac{17}{140} - \frac{1}{4}\int_0^1 (1-\phi)(-\overline{v'\theta'})dy \\ &\quad - \frac{1}{4}\int_0^1 [(y^3-3y^2+2)\phi_T - \phi_T^2]dy\,.\end{aligned} \tag{4}$$

Here, the dimensionless temperature, θ (which is different from ϑ above), is defined as

$$\theta = (T_w^* - T^*)/\Delta T_x^* \ , \qquad (5)$$

with $T_w^*(x)$ being the wall temperature, which linearly varies along the streamwise direction, and ΔT_x^* being the change of T_w^* over the streamwise distance of $\delta^*/2$ under the given wall heat flux. The first term in Eq. (4) corresponds to the heat transfer in a laminar channel flow with isoflux walls, i.e., $Nu = 140/17 \simeq 8.235$. The second term represents the contribution from the turbulent heat flux. This term is usually positive in a turbulent channel flow, which results in reduction of $1/Nu$ (i.e., increase of Nu) as compared to that of laminar flow. Here the quantity ϕ in the weighting factor denotes the partial flow rate, defined as

$$\phi(y) = 2\int_0^y \overline{u}(\eta) d\eta \ . \qquad (6)$$

The last contribution is determined solely by the velocity profile due to the presence of the Reynolds shear stress. Here, the quantity ϕ_T is its deviation of ϕ from that of the laminar flow, ϕ_L, i.e.,

$$\phi_T = \phi - \phi_L \ . \qquad (7)$$

The difference in the weighting factors, $(1-y)$ and $(1-\phi)$, suggests that the same control strategy as that proposed for CTD can be used. As compared to the CTD case, however, the difference in the weighting factors in the IFW case is much smaller (Fukagata et al., 2005). It implies stronger similarity between momentum and heat transports in the IFW case.

The proposed strategy is examined by means of DNS of channel flow at $Re_b = 3220$ (i.e., $Re_\tau = 110$ in uncontrolled flow) and $Pr = 0.71$. The opposition control scheme (Choi et al., 1994) is adopted for the suppression of near-wall Reynolds stress. The virtual detection plane is set at $y_d^+ = 10$. In addition, a virtual body force, i.e., $-\beta f(y)\theta'$, is added to the wall-normal momentum equation for the enhancement of turbulent heat-flux in the central region of the channel. Here, β is an amplitude coefficient and $f(y)$ is an envelope function.

Here, the CTD case is presented. According to the strategy above, the envelope function is set to have a value of unity in the central region away from the wall and zero near the wall: $f(y) = 1$ for $0.5 < y < 1.5$; $f(y) = 0$ for $0 < y < 0.5$ and $1.5 < y < 2$. The amplitude coefficient is set at $\beta = 7.4$ (nondimensionalized by using $2U_b^*$, δ^* and ΔT^*).

Figures 1 and 2 show the time traces of C_f and Nu in three cases: 1) without control, 2) with opposition control, 3) with opposition control and body force (present control). Both C_f and Nu decrease just after the onset of present control. After the initial transience, C_f returns to the level of uncontrolled flow, and Nu further increase to about 1.5 times of that of the uncontrolled flow.

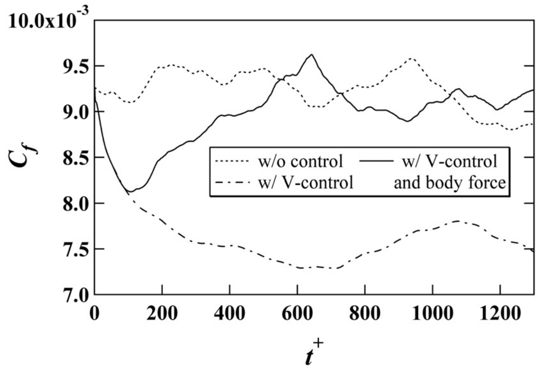

Figure 1: Time trace of C_f in the isothermal case.

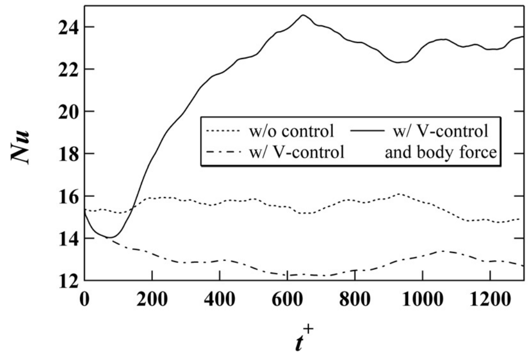

Figure 2: Time trace of Nu in the isothermal case.

The control strategy is also validated for the IFW case, although the effect is much smaller as expected from the identity equations.

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