Heat transfer in fully developed turbulent channel flow with streamwise traveling wave-like wall deformation

Keisuke UCHINO*, Hiroya MAMORI** and Koji FUKAGATA*

* Department of Mechanical Engineering, Keio University
  Hiyoshi 3-14-1, Kohoku-ku, Yokohama 223-8522, Japan
  E-mail: fukagata@mech.keio.ac.jp

** Department of Mechanical Engineering, Tokyo University of Science
  Niijyuku 6-3-1, Katsushika-ku Tokyo 125-8585, Japan

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Abstract
The dissimilarity between the momentum and heat transfer due to streamwise traveling wave-like wall deformation in turbulent channel flows is investigated through direct numerical simulations. The flow rate is kept constant, and the bulk Reynolds number is $Re_b = 5600$. A constant temperature difference condition is imposed on the channel walls. The parametric study shows that the heat transfer is enhanced when the wave travels in the upstream direction. The maximum analogy factor is found to be 1.13, i.e., 13% enhancement of heat transfer under a given pressure gradient, when the wall deformation amplitude is large and the wall deformation period is short. An analysis using the identity equations for the drag and the heat transfer with a three component decomposition reveals that the random component plays an important role in the enhancement of the heat transfer.

Key words: Turbulent heat transfer, Wall deformation, Wall-bounded turbulent flow, Direct numerical simulation

1. Introduction
Heat transfer enhancement is an important issue for more efficient use of the limited energy resources. Examples of well-known heat transfer enhancement techniques are vortex generators, transverse ribs, and modification of wall surfaces (e.g., surface roughness). These techniques promote turbulence to enhance heat transfer while the fluid drag also increases because of the inherent similarity between the momentum and heat transfer, i.e., the Reynolds analogy (Reynolds, 1874), although in many applications it is preferable to enhance heat transfer without increasing the drag.

The similarity between the momentum transfer and the heat transfer exists in different levels, as reviewed by Kasagi et al. (2012). First, there is a similarity between the friction drag $\tau^*_w$ and the wall heat flux $q^*_w$, i.e.,

$$\tau^*_w = \mu^* \frac{\partial u^*_i}{\partial y^*_w},$$

$$q^*_w = \lambda^* \frac{\partial \theta^*}{\partial y^*_w},$$

where $\mu^*$ and $\lambda^*$ are the dynamic viscosity and the thermal conductivity, respectively; the subscript of $w$ means the wall value and the superscript * denotes the dimensional quantity. There is also a similarity between the Navier-Stokes and energy equations,

$$\frac{\partial u^*_i}{\partial t^*} = - \frac{\partial (u^*_i u^*_j)}{\partial x^*_j} + \mu^* \frac{\partial^2 u^*_i}{\partial x^*_j \partial x^*_j} - \frac{1}{\rho^*} \frac{\partial p^*}{\partial x^*_i},$$

$$\frac{\partial \theta^*}{\partial t^*} = - \frac{\partial (u^*_i \theta^*)}{\partial x^*_j} + \lambda^* \frac{\partial^2 \theta^*}{\partial x^*_j \partial x^*_j} + \frac{Q^*}{\rho^* c_p^*},$$
where \( u, \theta, \) and \( p \) denote the velocity, the temperature, and the pressure, respectively; \( \rho^* \) is the density, \( c_p^* \) is the specific heat at constant pressure, and \( Q \) is the heat source. Due to these similarities, the friction drag increases when the heat transfer increases in ordinary situations, and \textit{vice versa}.

Recently, Hasegawa and Kasagi (2011) and Yamamoto et al. (2013) demonstrated through direct numerical simulation (DNS) of a turbulent channel flow that this similarity can be broken by controlling the flow using the optimal and suboptimal control theories with wall blowing and suction even when the boundary conditions for velocity and temperature are also similar. Apart from these attempts using feedback control, Higashi et al. (2011) reported that such a dissimilar control effect can also be attained by an open-loop control, i.e., a traveling wave-like blowing and suction (Min et al., 2006; Lieu et al., 2010; Mamori et al., 2014), when the boundary conditions for velocity and temperature are dissimilar (i.e., the constant temperature difference condition). They performed a laminar analysis and DNS of a turbulent channel flow and concluded that the similarity is broken by the phase difference between the velocity and the temperature due to a viscous phase shift in the region near the walls.

Although the traveling wave-like blowing and suction has demonstrated attractive control effects, its practical implementation is difficult. In that sense, a traveling wave-like wall deformation can be considered as a more practical alternative: in fact, an experimental study of the traveling wave-like wall deformation has been carried out by applying vibration to a natural rubber plate (Ishiwata et al., 2014). Hopfinger and Fukagata (2009) showed that a net flux is induced by the traveling wave-like wall deformation in the same direction as the wave in the absence of pressure gradient, indicating that the drag reduction by a traveling wave-like wall deformation could be achieved by a downstream wave. Following this finding, Nakaneishi et al. (2012) performed DNS of a turbulent channel flow controlled using a streamwise traveling wave-like wall deformation. As expected, the drag was found to decrease with the wave traveling in the downstream direction. In addition, the flow was stabilized and eventually relaminarized under several sets of parameters. In the relaminarized case, the maximum drag reduction rate was 69% and the maximum net energy saving rate was 65%.

In the present study, as an extension of the studies of Nakaneishi et al. (2012) and Higashi et al. (2011), we perform DNS of the turbulent channel flow to investigate whether the similarity between the momentum transfer and the heat transfer can be broken by the traveling wave-like wall deformation.

2. Numerical Procedure

Figure 1 shows the schematic of the channel flow with a traveling wave-like wall deformation. In order to deal with the wall deformation, a coordinate transformation from the Cartesian coordinates \((x_i, t)\) to the boundary-fitted coordinates \((\xi_i, \tau)\) similar to Kang and Choi (2002) is introduced, i.e.,

\[
\begin{align*}
    x_1 &= \xi_1, \\
    x_2 &= \xi_2(1 + \eta) + \eta_0, \\
    x_3 &= \xi_3, \\
    t &= \tau,
\end{align*}
\]

(5)

where \( \eta = (\eta_u - \eta_d)/2 \) and \( \eta_0 = (\eta_u + \eta_d)/2 \) with \( \eta_u \) and \( \eta_d \) denoting the wall-normal displacements of the upper and lower walls, respectively. The governing equations, i.e., the continuity, the Navier-Stokes, and the energy equations, on the \((\xi_i, \tau)\) coordinates read

\[
\begin{align*}
    \frac{\partial u_i}{\partial \xi_i} &= -S_i, \\
    \frac{\partial u_i}{\partial \tau} &= -\frac{\partial (u_i u_j)}{\partial \xi_j} - \frac{\partial p}{\partial \xi_i} + \frac{1}{Re_b} \frac{\partial^2 u_i}{\partial \xi_j \partial \xi_j} - \frac{dP}{d\xi_1} \delta_{i1} + S_i, \\
    \frac{\partial \theta}{\partial \tau} &= -\frac{\partial (u_i \theta)}{\partial \xi_j} + \frac{1}{PrRe_b} \frac{\partial^2 \theta}{\partial \xi_j \partial \xi_j} + Q + S_\theta,
\end{align*}
\]

(6)  (7)  (8)
The wall-normal velocity on the lower and the upper walls, $v_{\phi}(\xi_1, \xi_3, \tau) = u_2(\xi_1, 0, \xi_3, \tau)$ and $v_{\phi}(\xi_1, \xi_3, \tau) = u_2(\xi_1, 2, \xi_3, \tau)$, are given by

$$v_\phi = \frac{\partial \eta_\phi}{\partial \tau} = a \cos(k(\xi_1 - c\tau)), \quad v_\phi = -v_\phi,$$

where the velocity amplitude $a$, the wavenumber $k$, and the phase-speed of wall deformation $c$ are the parameters describing the wall motion. Accordingly, the displacement of the lower and the upper walls, $\eta_d$ and $\eta_u$, are given by

$$\eta_d = -\frac{a}{kc} \sin(k(\xi_1 - c\tau)), \quad \eta_u = -\eta_d.$$

Table 1  Dependency on the time step, the grid resolution, and the domain size.

| Conditions used | $N_x \times N_y \times N_z$ | $L_x \times L_y \times L_z$ | Difference in $D$ | Difference in $H$ |
|----------------|-----------------------------|-----------------------------|-------------------|-------------------|
| Shorter time step | 0.00044 | 256 x 96 x 128 | $4\pi \times 2 \times \pi$ | -4.4 | 0.13 |
| Finer grid | 0.00044 | 512 x 96 x 128 | $4\pi \times 2 \times \pi$ | 1.2 | -1.0 |
| Larger domain | 0.00044 | 512 x 96 x 256 | $8\pi \times 2 \times 2\pi$ | -4.9 | -0.61 |
With the wave parameters of $a$, $k$, and $c$, the period of wall deformation $T$ is also computed as

$$T = \frac{2\pi}{kc}.$$ (20)

As for the thermal boundary conditions, we impose the constant temperature difference condition. Therefore, the heat source $Q$ is zero and the dimensionless upper and lower wall temperatures are set at $\theta_+ = 1$ and $\theta_- = -1$, respectively. The temperature $\theta$ is assumed to be a passive scalar. The drag coefficient $C_D$ and the Stanton number $St$ are computed as

$$C_D = \frac{(-dP/dx)\delta}{(1/2)\rho^*u_b^2}, \quad St = \frac{q^*_w}{\rho^*c_p^*u_b^2\Delta T^*},$$ (21)

where $\Delta T^*$ is the temperature difference between two walls. Note that the drag coefficient $C_D$ is identical to the skin friction drag coefficient $C_f = \tau_w^*/[(1/2)\rho^*u_b^2]$ in the case of plane channel flow, but it includes the pressure drag component in the case with wall deformation.

The present DNS code is based on that of Fukagata et al. (2006) and Nakanishi et al. (2012) and extended here to account for the heat transfer. The governing equations are spatially discretized on a staggered grid system with an energy-conservative second-order finite difference method (Kajishima, 1999; Ham et al., 2002). The low storage third-order Runge-Kutta/Crank-Nicolson scheme (Spalart et al., 1991) is used for the time integration together with a pressure correction procedure similar to SMAC method (Amsden and Harlow, 1970). All DNSs are performed under a constant flow rate and the bulk Reynolds number is set at $Re_b = 5600$, corresponding to the friction Reynolds number of $Re_f \approx 180$ in the uncontrolled (i.e., flat wall) case. The Prandtl number is set at $Pr = 1.0$ to maintain the similarity between Eqs. (7) and (8).

The size of the computational box is $4\pi \times 2 \times \pi$ in the streamwise ($\xi_1$), the wall-normal ($\xi_2$), and the spanwise ($\xi_3$) directions; the corresponding number of computational cells is $256 \times 96 \times 128$. The computational grid is uniform in the streamwise and spanwise directions, while non-uniform in the wall-normal direction. The spanwise and wall-normal grid spacings are $\Delta \xi^+_1 = 4.3$ and $\Delta \xi^+_3 = 0.93 - 5.9$, where the superscript of “+” denotes the wall units in the plain channel case. In the streamwise direction, the grid spacing is $\Delta \xi^+_1 = 8.7$, which corresponds to 64 grid points for one period of wave in the case where the largest dissimilarity is observed (i.e., Case 24). The computational time step is set to be $\Delta t^+ = 0.0044$. Under these conditions, a good agreement with Moser et al. (1999) has been confirmed for the uncontrolled flow.

In order to assess the numerical uncertainty, we performed the numerical simulation in the Case 24 using a finer mesh, $\Delta \xi^+_1 = 4.3$, a shorter time step, $\Delta t^+ = 0.0022$, or a larger domain, $L_x \times L_y \times L_z = 8\pi \times 2 \times 2\pi$. We examine the difference in the drag normalized by that in the uncontrolled flow, $D$, and the heat transfer normalized by that in the

\[ \text{Table 2} \quad \text{Primary wave parameters (amplitude $a$, wavenumber $k$, and phase-speed $c$), corresponding deformation amplitude $\eta^*$ and period $T^*$ in wall units, and resultant quantities (normalized drag $D$, normalized heat transfer $H$, and analogy factor $A$).} \]

| Case | $a$ | $k$ | $c$ | $\eta^*$ | $T^*$ | $D$ | $H$ | $A$ |
|------|-----|-----|-----|--------|------|----|----|----|
| 1    | 0.05| 2   | 1   | 4.5    | 17.8 | 0.93| 0.93| 0.98|
| 2    | 0.15| 2   | 1   | 13.4   | 17.8 | -  | -  | -  |
| 3    | 0.15| 4   | 1   | 6.7    | 8.9  | 0.38| 0.24| 0.63|
| 4    | 0.20| 1   | 1   | 35.6   | 35.5 | 0.92| 0.89| 0.97|
| 5    | 0.20| 2   | 1   | 17.8   | 17.8 | -  | -  | -  |
| 6    | 0.20| 4   | 1   | 8.9    | 8.9  | -  | -  | -  |
| 7    | 0.20| 4   | 3   | 3.0    | 3.0  | 1.96| 1.22| 0.62|
| 8    | 0.01| 1   | -1  | 1.8    | 35.5 | 1.01| 1.00| 0.99|
| 9    | 0.03| 1   | -1  | 5.3    | 35.5 | 1.02| 1.02| 1.00|
| 10   | 0.05| 1   | -1  | 8.9    | 35.5 | 1.00| 1.02| 1.02|
| 11   | 0.05| 2   | -1  | 4.5    | 17.8 | 1.02| 1.04| 1.02|
| 12   | 0.05| 2   | -2  | 2.2    | 8.9  | 1.07| 1.02| 0.95|
| 13   | 0.10| 1   | -1  | 17.8  | 35.5 | 1.35| 1.35| 1.01|
| 14   | 0.10| 2   | -1  | 8.9   | 17.8 | 1.33| 1.28| 0.96|
| 15   | 0.10| 1   | -3  | 5.9   | 11.8 | 1.08| 1.10| 1.03|
| 16   | 0.10| 2   | -2  | 4.5   | 8.9  | 1.17| 1.14| 0.98|
| 17   | 0.15| 1   | -3  | 8.9   | 11.8 | 1.20| 1.21| 1.01|
| 18   | 0.15| 2   | -2  | 6.7   | 8.9  | 1.27| 1.31| 1.03|
| 19   | 0.20| 2   | -1  | 17.8  | 11.8 | 1.90| 1.96| 1.03|
| 20   | 0.20| 2   | -2  | 8.9   | 8.9  | 1.46| 1.60| 1.10|
| 21   | 0.20| 2   | -3  | 5.9   | 5.9  | 1.26| 1.37| 1.08|
| 22   | 0.25| 2   | -3  | 7.4   | 5.9  | 1.38| 1.47| 1.07|
| 23   | 0.30| 2   | -2  | 13.4  | 8.9  | 1.87| 2.06| 1.10|
| 24   | 0.30| 2   | -3  | 8.9   | 5.9  | 1.51| 1.70| 1.13|
uncontrolled flow, $H$, defined as
\[ D = \frac{C_D}{C_{D0}}, \quad H = \frac{St}{St_0}. \]  
(22)

The subscript of “0” denotes those in the uncontrolled flow.

Table 1 shows the differences in $D$ and $H$ from those obtained using the numerical conditions used for the rest of the paper. In all cases, the total integration time is 1150 wall unit time and the average is taken from 800 wall unit time after the statistically steady state is achieved. As compared to the reference case, the differences due to the grid resolution are about 1% and those due to the time resolution and the domain size are less than 5%.

3. Results and Discussion

3.1. Control performance

The normalized drag and heat transfer ($D$ and $H$) under different parameter sets are tabulated in Table 2, together with the analogy factor $A$ defined here as
\[ A = \frac{H}{D} = \frac{St/Sp}{C_D/C_{D0}}. \]  
(23)

When the analogy factor $A$ is greater than unity, the increment of the heat transfer is larger than that of the drag. The downstream wave-like wall deformation cases are Cases 1 to 7. The normalized drag and heat transfer slightly decrease in Cases 1 and 4, whereas these increase in Case 7. In Case 3, the drag and heat transfer also drastically decrease and the flow is relaminarized. In Cases 2, 5, and 6, the unstable relaminarization occurs (as discuss later). The values of $D$ and $H$ cannot be averaged in such cases and indicated as "-" in Table 2. In the upstream wave cases i.e., Cases 8 to 24, the normalized drag and heat transfer are greater than unity; namely, both the drag and the heat transfer are increased. The analogy factor greater than unity is also confirmed in several cases.

Figure 2 shows the time traces of the normalized drag and heat transfer in the downstream traveling wave cases, i.e., Cases 1, 3, 6, and 7. In these cases, the normalized drag and heat transfer exhibit similar changes. Case 1 shows slight decrease of the normalized drag and heat transfer. An excellent drag reduction effect and a convergence toward the laminar value (i.e., stable relaminarization) are observed in Case 3, while the heat transfer also decreases drastically in a similar manner. Since the turbulence is perfectly suppressed, the fluctuations in $D$ and $H$ are not observed. Note that such a relaminarization is observed not only in the wall deformation case, but also in the case of downstream traveling wave-like blowing and suction (Mamori et al., 2014). In contrast, the drag and heat transfer in Case 6 decrease similarly to those in the relaminarization case: they rapidly increase over the unity, and decrease again. This is the case of the unstable relaminarization: the turbulence recovery is due to the inflection-point instability of the mean velocity (Nakanishi et al., 2012). Case 7 shows the increase of both the normalized drag and heat transfer.

The time traces in the upstream traveling wave cases, i.e., Cases 14, 21, and 24, are shown in Fig. 3. The analogy factor is largest in Case 24 ($\eta^+ = 8.9, T^+ = 5.9$). Case 14 ($\eta^+ = 8.9, T^+ = 17.8$) has the same wall deformation amplitude as Case 24 with a longer deformation period. Case 21 ($\eta^+ = 5.9, T^+ = 5.9$) has the same wall deformation period as...
3.2. Contributions to drag and heat transfer

There are identity equations for the skin-friction coefficient (Fukagata et al., 2002) and the Stanton number (Kasagi et al., 2010), hereafter referred to as the FIK identities, which quantitatively decompose these dimensionless numbers into different contributions. In the fully developed plane channel flow under the constant flow rate and the constant temperature difference condition, the skin-friction coefficient and the Stanton number can be decomposed as

\[
C_f = \frac{12}{Re_b} + 12 \int_0^1 2(1-y)(-\bar{u'}v')dy, \tag{24}
\]

\[
St = \frac{2}{Re_b Pr} + \int_0^1 (-\bar{v'}\theta')dy, \tag{25}
\]

where the Reynolds shear stress (RSS) and the turbulent heat flux (THF) are denoted by \(-\bar{u'}v'\) and \(-\bar{v'}\theta'\), respectively.

The first terms in Eqs. (24) and (25) are the laminar contributions. The second terms are the turbulent contributions. The turbulent contribution for the skin-friction in Eq. (24) is the \(y\)-weighted integration of the RSS whereas the weighting factor for the THF in Eq. (25) is unity. Therefore, the simultaneous achievement of the skin-friction drag decrease and the heat transfer enhancement is possible, e.g., if both RSS and THF are enhanced in the central region of the channel.

In order to quantify modifications of \(C_D\) and St, we employ the three component decomposition for the FIK identities. The three component decomposition is defined as

\[
f(x, \xi_2, z, t) = (f)(\phi, \xi_2) + f''(x, \xi_2, z, t), \]

\[
= \bar{f}(\xi_2) + \tilde{f}(\phi, \xi_2) + f''(x, \xi_2, z, t),
\]

\[
= \bar{f}(\xi_2) + f''(x, \xi_2, z, t). \tag{26}
\]

Here, \(f\) represents any quantities and \((\cdot)\) denotes the phase-averaged component and the double prime denotes the random component. The phase \(\phi\) is defined as \(\phi = k(x - ct) - 2\pi n\), where \(n\) is an integer. The overbar denotes the mean component in the \(x - z\) plane and time, and the tilde denotes the periodic component. According to Eq. (26), the RSS and THF are decomposed as

\[
\bar{u'}v' = \bar{u'}v' + \tilde{u'}v', \tag{27}
\]
\[ \overline{\nu \theta} = \overline{\nu \theta}^{\theta} + \overline{\nu \theta}^{\nu}. \]  

Note that the correlation between the periodic and the random components are supposed to be zero, i.e., \( \overline{\nu \theta^j} = 0 \). Accordingly, the FIK identities (Eqs. (24) and (25)) are rewritten on the transformed coordinates as

\[
C_D = \frac{12}{Re_b} + \frac{12}{\nu} \int_0^1 (1 - \xi_2) (-\overline{\nu \theta^j}) d\xi_2 + \frac{12}{\nu} \int_0^1 (1 - \xi_2) (-\overline{\nu \theta^j}) d\xi_2 + 2 \int_0^1 (1 - \xi_2) (-\overline{\nu \theta^j}) d\xi_2 + 6 \int_0^1 (\xi_2 (\xi_2 - 2) \overline{S_1} d\xi_2), \tag{29}
\]

\[
St = \frac{2}{Re_b Pr} + \frac{1}{St} \int_0^1 (\nu \theta^j) d\xi_2 + \frac{1}{St} \int_0^1 (\nu \theta^j) d\xi_2 + \frac{1}{St} \int_0^1 (1 - \xi_2) \overline{S_1} d\xi_2, \tag{30}
\]

The right-hand-sides of these equations consist of four terms: the first term is the laminar contribution; the second is the random contribution; the third term is the periodic contribution; the fourth term is the contribution from the source term of the coordinate transformation. To simplify the discussion, the third and fourth terms are referred to as “the periodic component,” \( C_D^p \) and \( St^p \). The superscripts of \( l, r, p \) mean the laminar, random, and periodic contributions, respectively. Accordingly, the drag and heat transfer normalized by those in the uncontrolled flow are decomposed as

\[
D = \frac{C_D}{C_{Do}} = \frac{C_D^l + C_D^r + C_D^p}{C_{Do}} = D^l + D^r + D^p, \tag{31}
\]

\[
H = \frac{St}{St_0} = \frac{St^l + St^r + St^p}{St_0} = H^l + H^r + H^p. \tag{32}
\]

Figure 5 shows each term in Eqs. (31) and (32) in Case 14 (\( \eta^* = 8.9, T^* = 17.8 \)), Case 21 (\( \eta^* = 5.9, T^* = 5.9 \)), and Case 24 (\( \eta^* = 8.9, T^* = 5.9 \)). Since the simulations are performed under the constant flow rate condition, the laminar contribution terms are unchanged in all cases. In the uncontrolled flow, the random contribution is larger than that of the laminar contribution. In the controlled cases, too, the random contribution is the primary contributor, while the periodic contribution is small. The periodic contribution is negative except for \( D^p \) in Case 14. In Case 24, the amount of increase in \( H^r \) exceeds that in \( D^r \), resulting in the favorable dissimilarity between the drag and the heat transfer, i.e., \( A > 1 \). In contrast, in Case 14, the periodic components are positive for drag and negative for heat. This difference is likely to cause more enhancement of drag than heat to result in \( A < 1 \).

The phase-averaged statistics in Case 24 are shown in Figs. 6 and 7. The channel is contracted and expanded at \( 0 < \phi/(\pi/2) < 2 \) and \( 2 < \phi/(\pi/2) < 4 \), respectively. Due to the wall deformation, the wall-normal velocity shows antisymmetric distribution as shown in Fig. 6(a). As shown in Fig. 6(b), the streamwise velocity increases and decreases in the contracted and expanded regions, respectively, which generates a net flux in the streamwise direction: called as the pumping effect (Hepfner and Fukagata, 2009). Figure 6(c) shows the temperature distribution. Slight difference is observed between the control and uncontrolled cases in the region near the wall.
Figure 7(a) shows the distribution of the product of the random velocity components, $\langle -u''v'' \rangle$, referred to as the random-RSS. The random-RSS in the controlled flow is larger than that in the uncontrolled flow, which corresponds to the increase of the random contribution to the skin-friction drag as shown in Fig. 5(a). The product of $v''$ and $\theta''$ is also increased as compared with the uncontrolled case, as shown in Fig. 7(b). Figures 7(c) and (d) show the periodic components. The positive and negative $\langle -e_u e_v \rangle$ alternate while the non-quadrature between $e_u$ and $e_v$ appears in the region near the wall. A significant amount of $\langle -e_v e_\theta \rangle$ also appears in the region near the wall, which creates the periodic contribution to the heat transfer similarly to that to the friction drag (Min et al., 2006; Mamori et al., 2010).

The upstream wave-like wall deformation or blowing and suction is known to work to destabilize the flow as was shown by DNS (Min et al., 2006; Nakanishi et al., 2012). According to the stability analysis of Lee et al. (2008), this destabilization by a relatively high-speed upstream wave is likely due to an enhanced transient growth of streamwise streak-type three-dimensional disturbance. The disturbance primarily grows to enhance both the Reynolds shear stress and the turbulent heat flux in the region near the wall, as shown in Fig. 7. Unlike the Reynolds shear stress, which should be anti-symmetric around the centerline (i.e., $y = 1$), however, the turbulent heat flux, which should be symmetric around the centerline, is also enhanced in the core region to enhance heat transfer more than drag as expected from Eqs. (24) and (25). Of course, however, this difference stems from the difference in the wall boundary conditions between the velocity and temperature fields.

It is worth noting that we have also performed DNS under the uniform heat generation (UHG) condition (not shown in the present paper; see Uchino (2015)), in which the heat source is set to equal to the mean pressure gradient and the wall temperature is imposed zero so that the Reynolds analogy strengthens as compared with the constant temperature difference condition. Unfortunately, the modification of heat transfer was found to be much smaller than that in the constant temperature difference condition, and no improvement in the analogy factor was confirmed. Considering the fact that Yamamoto et al. (2013) succeeded in increasing the analogy factor also in UHG case by using a streamwise traveling wave-like blowing and suction, the present disappointing result in UHG case is likely due to the slight difference between the blowing and suction and the wall-deformation (Höpfner and Fukagata, 2009), effects of which on the heat transfer should be investigated in details in the future.

4. Conclusions

Direct numerical simulations of the turbulent channel flows with the traveling wave-like wall deformation are performed. The Reynolds number is set to be $Re_b = 5600$ and the constant temperature difference condition is imposed.

For the downstream traveling wave, the skin-friction drag and the heat transfer decrease and the analogy factor is less than unity. The stable and unstable relaminarization phenomena are observed under some sets of parameters. For the upstream traveling wave, in contrast, both the skin-friction drag and heat transfer increase and the analogy factor is larger than unity. The maximum analogy factor is 1.13 when the wall deformation amplitude is $\eta^+ = 8.9$ and the wall deformation period is $T^+ = 5.9$. The analysis using the FIK identities reveals that the random components of velocity and...
Fig. 6 Periodic averaged fields in the Case 24: (a) wall-normal velocity; (b) streamwise velocity; (c) temperature. Solid line, phase averages at $\phi/(\pi/2) = 0, 1, 2, 3$; broken line, uncontrolled turbulent profile.

Fig. 7 Periodic averaged fields in the Case 24: (a) random-RSS; (b) random-THF; (c) periodic-RSS; (d) periodic-THF. Solid line, phase averages at $\phi/(\pi/2) = 0, 1, 2, 3$; broken line, uncontrolled turbulent profile.
temperature play the dominant role for the drag increase and the heat transfer enhancement.

The present parametric study also suggests a possibility for a larger analogy factor to be achieved in the range of $\eta^* > 8.9$ and $T^* < 5.9$, which is unfortunately difficult to study using the present numerical technique. A more comprehensive parametric study is therefore left as a future work, by improving the numerical method so as to be more robust against steeper deformation.

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