Drag Reduction by Wavy Surfaces*

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Abstract

Recent studies on turbulent friction drag reduction by wavy surfaces are reviewed. Special focus is laid upon numerical studies of fully developed flow in a channel having wavy surfaces. Both the surfaces driven by the flow (*i.e.*, passive surfaces) and those driven by the external power (*i.e.*, active surfaces) are considered. In addition, the drag reduction by traveling wave-like blowing and suction, which is closely related to the wavy surfaces, is also discussed in detail.

Key words : Friction Drag, Compliant Surface, Wall Deformation, Peristalsis, Blowing and Suction

1. Introduction

Control of wall turbulence for skin friction drag reduction has been one of the central topics in turbulence research for the recent twenty years. Different types of control have been proposed, *e.g.*, passive control using structured roughness such as riblets⁽¹⁾, injection of polymer or surfactant⁽²⁾, active predetermined control using spanwise oscillation⁽³⁾, and active feedback control using micro-sensors/actuators⁽⁴⁾. All of these control methods have advantages and disadvantages, and are still competing each other.

For a fully-developed channel/pipe flow, there is a strict mathematical relationship between the skin friction drag and Reynolds stress⁽⁵⁾ (hereafter referred to as the FIK identity). For instance, the FIK identity for a fully-developed channel flow under a constant flow rate reads

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

where all the quantities are made dimensionless by using twice the bulk mean velocity $2U_b^*$ and the channel half-width δ^* , while the dimensional quantities are denoted by the superscript of *. The skin friction coefficient and the bulk Reynolds number are defined as $C_f = 2\tau_w^*/(\rho^* U_b^{*2})$ and $Re_b = 2U_b^* \delta^*/\nu^*$, where τ_w^* , ρ^* , and ν^* denote the wall shear, the fluid density and the kinematic viscosity, respectively. The FIK identity indicates that the skin friction coefficient is decomposed into two parts: one is the laminar contribution given by the well-known laminar solution, and the other is the turbulent contribution, which is proportional to the weighted integral of the Reynolds shear stress, $-\overline{u'v'}$. The weighting factor $(1-y)$, where y is the distance from the wall, indicates that the near-wall Reynolds shear stress is more responsible for the friction drag. Therefore, the target for friction drag reduction is very clear: all one has to do is to directly suppress Reynolds shear stress (or even make it negative) in the region near the wall such that the second term of Eq. (1) is reduced, possibly involving the indirect suppression of the Reynolds shear stress in the region far away from the wall⁽⁶⁾.

To achieve this simple objective, passive or active predetermined control using wavy surface is an attractive option, since it is expected at least easier to be implemented in practice than the feedback control. In the present paper, recent studies on friction drag reduction by wavy surfaces are reviewed. Since the effects of wavy surfaces in fully developed flows are less understood than that in transitional flows, special focus here is laid upon friction drag reduction in fully developed channel flows. For transition delay by wavy surfaces, readers are

Fig. 1 Schematic of typical compliant surfaces and their models. Redrawn by Fukagata *et al.*⁽⁸⁾ based on Gad-el-Hak⁽⁷⁾.

referred to review articles, *e.g.*, by Gad-el-Hak⁽⁷⁾. We consider surfaces driven by the flow (*i.e.*, passive surfaces) and those driven by external input (*i.e.*, active surfaces). In addition, the drag reduction by traveling wave-like blowing and suction, which is closely related to the wavy surfaces, is also discussed.

2. Passive surfaces

2.1. Compliant wall models

A soft surface, of which deformation is passively driven by the adjacent flow, is called compliant surface or compliant wall. Different models have been proposed for the compliant surfaces. Two popular models are the isotropic compliant wall model as shown in Fig. 1(b), which models Kramer's coating (Fig. 1(a)), and the anisotropic compliant surface model as shown in Fig. 1(d), which was introduced by Carpenter and Morris⁽⁹⁾ to represent Grosskreutz's compliant wall (Fig. 1(c)). Some other models, such as the homogeneous layer model, the double-layer compliant wall model, and the anisotropic fiber-composite compliant wall model, have been proposed (see, *e.g.*, Gad-el-Hak⁽⁷⁾ for details).

The isotropic compliant wall is simply composed of a deformable plate, a rigid base, and the springs connecting these. Denoting the displacement in the wall-normal direction by $\eta(x, z, t)$, the governing equation of motion for the isotropic compliant wall is expressed as⁽¹⁰⁾

$$b\rho_m \frac{\partial^2 \eta}{\partial t^2} + D \frac{\partial \eta}{\partial t} + B \nabla^4 \eta - Eb \nabla^2 \eta + K \eta = f, \quad (2)$$

where b , ρ_m , E , and K are the thickness, the density, the elastic modulus of the membrane, and the spring stiffness, respectively, and $\nabla^2 = \partial^2 / \partial x^2 + \partial^2 / \partial z^2$. The flexural rigidity of the thin membrane, B , is given by

$$B = \frac{Eb^3}{12(1 - \nu_p^2)}, \quad (3)$$

where ν_p is the Poisson ratio. The compliant wall is driven by the wall pressure p_w and the normal stress on the wall σ_w . The forcing term is expressed by

$$f = (-p'_w + \sigma'_w), \quad (4)$$

where the prime denotes the fluctuating component.

On the other hand, the anisotropic compliant wall model has an inclined arm as shown in Fig. 1(d). The movement of the arm is restricted in $x - y$ plane parallel to the mean flow

direction. Assuming a small change in the arm angle, $\delta\theta$, from the equilibrium angle, θ , the governing equation for the surface can be written for a single variable, $\eta(x, z, t)$, as⁽⁹⁾

$$\eta = l\delta\theta, \quad (5)$$

where l is the arm length. The displacement (x'_w, y'_w, z'_w) and the velocity (u'_w, v'_w, w'_w) of the membrane are given by

$$x'_w = \eta \sin \theta, \quad y'_w = \eta \cos \theta, \quad z'_w = 0, \quad (6)$$

and

$$u'_w = \frac{\partial \eta}{\partial t} \sin \theta, \quad v'_w = \frac{\partial \eta}{\partial t} \cos \theta, \quad w'_w = 0, \quad (7)$$

respectively. The governing equation of motion for η is similar to that for the isotropic case, but it accounts for the inclination angle:

$$b\rho_m \frac{\partial^2 \eta}{\partial t^2} + D \frac{\partial \eta}{\partial t} + B \cos^2 \theta \nabla^4 \eta - Eb \sin^2 \theta \nabla^2 \eta + K\eta = f. \quad (8)$$

The driving force for the anisotropic compliant wall includes the shear stress τ'_w in addition to the pressure p'_w , the normal stress σ'_w , *i.e.*,

$$f = (-p'_w + \sigma'_w) \cos \theta + \tau'_w \sin \theta. \quad (9)$$

The unique feature of this anisotropic compliant wall model is that, as is clear from Eq. (7), the local Reynolds shear stress on the wall due to this surface motion is always negative when $0 < \theta < \pi/2$, *i.e.*,

$$-u'_w v'_w = -\frac{\sin(2\theta)}{2} \left(\frac{\partial \eta_f}{\partial t} \right)^2, \quad (10)$$

which is considered preferable for achieving drag reduction, as indicated by Eq. (1).

2.2. Monoharmonic analysis

For both isotropic and anisotropic compliant wall models, two-dimensional Fourier transform of the membrane equation of motion results in the equation of motion for the standard spring-mass-damper system, *i.e.*,

$$\frac{\partial^2 \hat{\eta}}{\partial t^2} + 2\zeta\omega_n \frac{\partial \hat{\eta}}{\partial t} + \omega_n^2 \hat{\eta} = \frac{\hat{f}}{b\rho_m}, \quad (11)$$

where the hat ($\hat{\cdot}$) denotes the Fourier coefficient and the normal stress term is neglected for simplicity. The natural angular frequency of the wavenumber mode k (with $k = \sqrt{k_x^2 + k_z^2}$) of the isotropic compliant wall model is given by⁽¹¹⁾

$$\omega_n = \sqrt{\frac{K_E + Bk^4 + Ebk^2}{b\rho_m}} \quad (12)$$

and that of the anisotropic compliant wall model is calculated to be⁽⁸⁾

$$\omega_n = \sqrt{\frac{K_E + Bk^4 \cos^2 \theta + Ebk^2 \sin^2 \theta}{b\rho_m}}. \quad (13)$$

The damping coefficient is given by

$$\zeta = \frac{D}{2b\rho_m\omega_n} \quad (14)$$

in both cases.

Fig. 2 Bode diagram of the compliant surfaces with the wall pressure p'_w as input and wall-normal velocity v_w as output: (a) Gain; (b) Phase.

From the classical control theory, the gain ($|\hat{G}(i\omega)|$) and phase delay ($\angle\hat{G}(i\omega)$) of the velocity ($\partial\eta/\partial t$) in response to the force pushing the surface ($-f$) can be found as

$$|\hat{G}(i\omega)| = \frac{\left(\frac{\omega}{\omega_n}\right)}{b\rho_m \sqrt{\left[1 - \left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left(2\zeta\frac{\omega}{\omega_n}\right)^2}} \quad (15)$$

and

$$\angle\hat{G}(i\omega) = \tan^{-1} \left[\frac{1 - \left(\frac{\omega}{\omega_n}\right)^2}{2\zeta\frac{\omega}{\omega_n}} \right], \quad (16)$$

respectively.

In order to design the membrane such that it responds and modifies the turbulent structure, the parameters involved in Eq. (2) or Eq. (8) should be chosen so that the characteristic frequency of membrane matches that of the turbulence. Typical scales of near-wall turbulence is, say, $k^+ \simeq 0.2$ in wavenumber and $\omega^+ \simeq 0.06$ in angular frequency (where the superscript of + denotes the wall unit), which correspond to the typical scales of quasi-streamwise vortex⁽⁴⁾.

A simple strategy for the choice of membrane parameters is to choose them such that the wall pressure and the wall-normal velocity induced by the membrane motion become in-phase, similarly to the opposition control by Choi *et al.*⁽¹²⁾ and its variant by Kang and Choi⁽¹³⁾ using surface deformation. In the case of isotropic compliant wall, for example, the relationship between the wall pressure p'_w (*i.e.*, input) and the wall-normal velocity $v_w = \partial\eta/\partial t$ (*i.e.*, output) is

$$\hat{v}_w = \hat{G}(i\omega)\hat{p}_w. \quad (17)$$

The in-phase relationship between v_w and p'_w is obtained when their phase difference is zero degree. According to the Bode diagram of this system shown in Fig. 2, however, there is

Fig. 3 Wall motion (magnified by 5 times in y direction), Reynolds shear stress (color on the wall), and vortical structure (white, $Q^+ = 0.02$) on solid and anisotropic compliant surfaces : (a) solid surface; (b) compliant surface with which 8% drag reduction was obtained. From Fukagata *et al.*⁽⁸⁾.

Fig. 4 Reynolds shear stress in solid wall case, drag reducing cases (Cases A1 and A2) and drag increasing case (Case B): (a) overview; (b) near the wall. From Fukagata *et al.*⁽⁸⁾.

no possibility for these two quantities to be in-phase. This was pointed out by Xu *et al.*⁽¹¹⁾. This analysis implies that in order to have a surface motion that reduces the friction drag, one should either consider a driving force other than the pressure fluctuations or design the membrane based on a strategy other than the opposition control.

2.3. Direct numerical simulation

Endo and Himeno⁽¹⁴⁾ performed direct numerical simulation (DNS) of turbulent channel flow with a simplified isotropic compliant surface as shown in Fig. 1(b). They neglected the membrane tension and flexural rigidity terms in Eq. (2) and considered only the spring and damping terms. The friction Reynolds number was $Re_\tau = 150$ and the spring and damping coefficients were chosen so as the membrane to respond to the near-wall coherent structure, as discussed above. The DNS was performed by using the body-fitted finite difference and initiated at time $t = 0$ with the velocity field in a solid channel. The deformation of surface was observed to be a wave of which streamwise wavelength was about 1200 wall units. About 3% drag reduction was obtained in the time period of $t^+ = 0 - 1000$.

The result of Endo and Himeno⁽¹⁴⁾ was soon disproved by Xu *et al.*⁽¹¹⁾. They performed DNS essentially similar to that of Endo and Himeno⁽¹⁴⁾ (to be strict, Xu *et al.*⁽¹¹⁾ took into account the membrane tension and flexural rigidity terms in Eq. (2), which was neglected in Endo and Himeno⁽¹⁴⁾), but no clear drag reduction effect was obtained. The major difference between Endo and Himeno⁽¹⁴⁾ and Xu *et al.*⁽¹¹⁾ is the integration time for sampling data, *i.e.*, $t^+ = 0 - 1000$ in the former and $t^+ = 500 - 3000$ in the latter. Together with the result of monoharmonic analysis discussed above, Xu *et al.*⁽¹¹⁾ concluded that the drag reduction observed by Endo and Himeno⁽¹⁴⁾ was merely a transitional one.

Fig. 5 Wall motion (no magnification in y direction), Reynolds shear stress (color on the wall), and vortical structure (white, $Q^+ = 0.02$) in the doubled computational domain. The parameters of compliant surface are the same as those of Fig. 3. From Fukagata *et al.*⁽⁸⁾.

In contrast to the studies introduced above, Fukagata *et al.*⁽⁸⁾ performed DNS of channel flow with anisotropic compliant surface as shown in Fig. 1(d). The DNS was performed using a finite difference code with a constant flow rate condition⁽¹⁵⁾ at the bulk Reynolds number of $Re_b = 3300$ (which corresponds to the friction Reynolds number of $Re_\tau \approx 110$ in the solid channel). They tried to optimize the parameter of compliant surface by using a stochastic optimization method, CMA-ES⁽¹⁶⁾. In order to enable 2000 runs of DNS required for the optimization study, the coupling between the surface and fluid field is done by the velocity coupling. Namely, the displacement of membrane was neglected on the assumption that its effect is small when the displacement is within the sublayer thickness⁽¹³⁾.

The maximum drag reduction rate attained in this optimization attempt was about 8%. The equilibrium arm angle in the drag reducing case was found to be about 60° , implying that the wall-shear stress (rather than the wall pressure) plays an important role for the surface motion. The resultant surface motion was found to be a wave traveling downstream as shown in Fig. 3 and the Reynolds shear stress on the wall was found to become negative due to the motion of surfaces restricted by the inclined arms. The natural frequency of the surface motion was found to be about $\omega^+ = 0.1$, suggesting that the membrane should be extremely soft in practical applications.

In order to examine the universality of the surface parameters obtained, DNS was performed also in a computational domain doubled in the streamwise direction as compared to that used optimization study. As shown in Fig. 5, however, the flow was destabilized and the drag was increased. It was found to be due to the development of longer waves and the excessively large wall-normal velocity induced thereby. This results suggests that the set of parameter obtained in the optimization study was unfortunately not universal.

3. Active surfaces

3.1. Actuation modes

Various actuation modes can be considered for active wavy surface. The wave can be either standing wave^{(17),(18)} or traveling wave. In the case of traveling wave, its direction can be either streamwise⁽¹⁹⁾ or spanwise⁽²⁰⁾. The input can be either surface deformation^{(17)–(20)}, blowing and suction^{(21),(22)}, or body force⁽²³⁾. In many of these studies, certain amount of drag reduction have been reported. In the followings, we focus on the streamwise traveling wave, which is related to the observation in the previous section.

3.2. Traveling wave-like blowing/suction

Based on implication of the FIK identity (1), Min *et al.*⁽²¹⁾ proposed a simple predetermined control method to make negative Reynolds shear stress in the region near the wall, *i.e.*, traveling wave-like blowing and suction. The local blowing/suction velocity from the walls,

Fig. 6 Normalized drag increment $\Delta D/a^2$ in linearized flow as function of wavespeed c and wavenumber k : (a) For upstream traveling wave ($c < 0$); (b) In wider range of parameters with color denotes $\Delta D/a^2$. From Mamori *et al.*⁽²²⁾.

Fig. 7 Velocity field induced by the upstream traveling wave-like blowing/suction ($k = 0.5$, $c = -1.5$) in linearized flow ($Re_c = 2000$), normalized by the blowing/suction amplitude: (a) wall-normal velocity v'/a ; (b) streamwise velocity, u'/a ; (c) product, $-u'v'/a^2$. From Mamori *et al.*⁽²²⁾.

v_w , is given by

$$v_w = a \cos(k(x - ct)), \quad (18)$$

where x and t denote the streamwise coordinate and the time, respectively. The parameters, a , k , and c , represent the amplitude, the wavenumber, and the wavespeed of the traveling wave, respectively.

Min *et al.*⁽²¹⁾ first examined the control effect by using the linear analysis. In fact, one can predict the drag increment (which is essentially the nonlinear effect) from the linear solution by substituting the first order solutions u' and v' into the FIK identity. Figure 6 shows the result of linear analysis, reproduced by Mamori *et al.*⁽²²⁾. For all wavenumbers k , the normalized drag increment $\Delta D/a^2$ (which corresponds to the second term of the FIK identity) is found to take negative value when the wave is traveling upward ($c < 0$). Since the base flow is laminar, this result indicates that the drag is less than that of the laminar flow at the same bulk Reynolds number. It should be noted, however, that such sublaminal drag does not mean that the net power required to drive the flow (*i.e.*, summation of pumping and actuation powers) is less than that for the laminar flow. In fact, it has been mathematically proved that the lower bound for net power in this case is that of the laminar Poiseuille flow^{(24), (25)}. Namely, the net power always increases when control is applied to laminar base flow. Since this actuation induces a unidirectional flow in the absence of mean flow, too, it can also be interpreted as pumping from walls⁽¹⁹⁾.

The mechanism of drag reduction (or pumping) by this traveling wave-like blowing/suction has been investigated in detail by analysing the phase relationship between the induced

Fig. 8 Vortex core (red) and shear-dominated region (blue) in turbulent channel flow at $Re_b = 5600$: (a) uncontrolled; (b) controlled by using upstream traveling wave-like blowing/suction. From Mamori⁽²⁶⁾.

Fig. 9 Reynolds shear stress. Dotted line, uncontrolled; solid line, the case of 44% drag reduction by upstream traveling wave-like blowing/suction. Red, viscous shear stress; black, Reynolds shear stress; blue, total shear stress. From Mamori⁽²⁶⁾.

streamwise and wall-normal velocity components⁽²²⁾. Figure 7 shows the velocity fields of u' and v' and their product $-u'v'$ corresponding to the local Reynolds shear stress in a drag reduction case. It is found that the velocity field is basically identical to that of the corresponding potential flow: u' and v' are in quadrature; thus no Reynolds shear stress is produced when spatially averaged. As can be observed in Fig. 7(b), however, one can find some distortion of u' distribution in thin regions near the walls due to the viscosity effect. This distortion leads to departure from quadrature and produces the nominally negative Reynolds shear stress. It was also found that the thickness of this layer scales similarly to that of Stokes' second problem.

The traveling wave-like blowing/suction is known to be also effective in turbulent channel flow⁽²¹⁾. Figure 8 shows a sample vortical structure in a turbulent channel flow at the bulk Reynolds number of $Re_b = 5600$, which corresponds to the friction Reynolds number of $Re_\tau \approx 180$ in the uncontrolled flow. It is obvious that less number of vortical structures are identified in the controlled case. As shown in Fig. 9, the near-wall Reynolds shear stress is reduced similarly to the laminar case introduced above and friction drag is reduced according to Eq. (1). In the case of turbulent base flow, the net power can also be reduced. As shown in Fig. 10, the gain G (*i.e.*, the ratio of drag reduction to the input power) of this control is much smaller than that in the case of feedback control. Plausible reasons for the small gain are that the upstream traveling wave destabilizes the flow while making negative Reynolds shear stress near the wall^{(32)–(34)}.

Fig. 10 Gain G and net energy saving rate S achieved by different active control schemes: opposition control and suboptimal control⁽²⁷⁾, temporally-periodic spanwise wall-oscillation control⁽²⁸⁾ streamwise traveling wave-like blowing/suction⁽²¹⁾, steady streamwise forcing⁽²⁹⁾, and spatially-periodic spanwise oscillation⁽³⁰⁾. From Kasagi *et al.*⁽³¹⁾.

It is worthwhile to note that traveling wave-like blowing/suction has sometimes been obtained as a result of feedback control. Koumoutsakos⁽³⁵⁾ designed the feedback controller so as to cancel the vorticity flux on the wall. In the case where the drag was decreased by 40%, a traveling wave-like structure was observed as the resultant actuation pattern. More recently, Kasagi *et al.*⁽³⁶⁾ performed suboptimal control⁽³⁷⁾ aiming at suppression of Reynolds shear stress and enhancement of turbulent heat flux using a formulation similar to that of Fukagata and Kasagi⁽³⁸⁾. The control input to maximize the ratio of heat transfer to friction drag was found to be a downstream traveling wave-like blowing/suction.

3.3. Traveling wave-like deformation: peristalsis

Although the traveling wave-like blowing/suction is effective, it is difficult to fabricate such device in reality. An alternative option is to use the traveling wave-like deformation so as to induce the wall-normal velocity similar to the blowing/suction, as suggested in the concluding remarks of Min *et al.*⁽²¹⁾.

The traveling wave-like deformation is well-known by the term, peristalsis. Høpfner and Fukagata⁽¹⁹⁾ studied the difference between the peristalsis and the traveling wave-like blowing/suction in the absence of mean flow. As shown in Fig. 11 the traveling waves of wall deformation induce pumping in the forward direction (*i.e.*, the same direction as the wave propagation), while the traveling waves of blowing and suction induce pumping in the backward direction (*i.e.*, opposite direction as the wave). In both cases, fluid particles were observed to be entrained into a circular motion by the wall actuation. The pumping direction originates from a different viscous damping during the backward and forward motion of fluid particles along its circular trajectory. The supplementary movies of their paper⁽¹⁹⁾ (<http://journals.cambridge.org/fulltext.content/supplementary/S0022112009007629sup001/>) clearly demonstrate these motions.

The difference between the traveling wave-like blowing/suction and peristalsis was discussed also by using a weakly nonlinear analysis⁽¹⁹⁾. It was found that the governing equation and the boundary condition for these cases are exactly the same at the leading order. The difference between these lies in the second order boundary conditions, which is zero for blowing/suction, and non-zero for peristalsis. This additional effect acts opposite to the effect of Reynolds stress generated from the first order solutions, and induces a net flow in the forward direction.

The above findings on the difference between the peristalsis and blowing/suction suggest

Fig. 11 Velocity and pressure fields in the peristalsis and blowing/suction cases. The Reynolds number based on the wavespeed and channel half-width is 300, wave amplitude is $\phi = 0.5$, and the wavelength is $L = 6$ (made dimensionless by using the wavespeed and channel half-width). The pressure field is indicated by color: high pressure is red, low pressure is blue. From Heepffner and Fukagata⁽¹⁹⁾.

that the drag reduction in turbulent channel flow may be achieved by wave-like wall deformation traveling downstream, but not upstream. Preliminary DNS results by Nakanishi *et al.*⁽³⁹⁾ confirm that the drag is in fact reduced by the downstream traveling wave-like wall deformation. They also found that the turbulent flow at the bulk Reynolds number of $Re_b = 5600$ (*i.e.*, $Re_\tau \approx 180$ in uncontrolled flow) could be relaminarized under some sets of parameters.

4. Concluding remarks

In the present paper, recent numerical studies on friction drag reduction in fully developed channel flow by passive and active wavy surfaces have been reviewed.

According to the studies introduced here, drag reduction effect by passive compliant surfaces in fully developed channel flows is still questionable. At least, the monoharmonic analysis clearly indicates that the effects similar to the opposition control⁽¹²⁾ cannot be expected. Drag reduction may be obtained by streamwise traveling wave as observed in Fig. 3. The membrane should be extremely soft in order to have sufficient deformation, but too large deformation caused by the long wave would result in drag increase as observed in Fig. 5. One possibility to suppress the development of long wave may be to limit the wavelength by fixing the membrane at several streamwise locations. The other possibility is to use a membrane that has different properties (*e.g.*, elasticity) in streamwise and spanwise directions. Validity of such modification should be studied in the future.

On the other hand, the active surfaces have been effective in reducing friction drag. The main disadvantage is that the gain is on the order of 1 – 10, which means that the mechanical efficiency of the actuators must be on the order of 10 – 100% in order to have net energy saving in practice. The Reynolds number effect is also an important remaining issue. Assuming that the direct modification by control is limited in the near-wall region, the drag reduction rate is expected to have relatively mild dependency to the Reynolds number, as has been theoretically shown by Iwamoto *et al.*⁽⁴⁰⁾ for the case of feedback control. Similar investigation, however, should be made in terms of the net energy saving, since the gain of traveling wave control is so small that slight difference in drag reduction rate would crucially affect whether the net energy can be saved or not.

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