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Experimental Investigation of Pump Control for Drag Reduction in Pulsating Turbulent Pipe Flow

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ABSTRACT

In this study, energy saving of pulsating turbulent pipe flow has been experimentally examined. The rotating speed of a centrifugal pump was periodically changed for generating the pulsating flow. The cycle-averaged friction Reynolds number is set to be 110 for comparison with the previous data of direct numerical simulation (DNS) by Sasou *et al*. (2007). It is found that the optimal values of the cycle period and the mean pressure gradient in the acceleration phase exist for increasing the energy saving and drag reduction rates. The maximum energy saving and drag reduction rates are respectively 58% and 63% when the cycle period normalized by the friction velocity and the radius is 10 and the mean pressure gradient in the acceleration phase is 4.1 times as large as the cycle-averaged pressure gradient. The maximum energy saving rate is 29% even if the energy recovery rate is 0% for the negative pressure gradient phase. The time variation of friction coefficient for these conditions is similar to that of the laminar flow, indicating that the laminar flow is maintained at all the phases.

INTRODUCTION

Development of efficient turbulence controls and prediction techniques for drag reduction is of a great importance from the view points of energy saving and the environmental impact mitigation. Investigation of relaminarization is subjected to effects of acceleration, suction/blowing and so on (Launder, 1964; Sreenivasan, 1982; Bewley *et al*., 2001).

There has been a considerable interest in the dynamics of pulsating and oscillating pipe flows because of those relevance to a number of engineering and biological systems (Lodahl *et al*., 1998; Sumida, 2003). Pulsating and oscillating flows have important effects on mass transfer rates, heat transfer rates and friction coefficients (Lodahl *et al.*, 1998; Nishimura *et al.*, 2000). Nevertheless, drag reduction techniques by pulsating turbulent flows have rarely been studied (e.g., Iguchi *et al.,* 1985; Takami *et al*., 2004). Recently, direct numerical simulation (DNS) of a pulsating turbulent channel flow is performed by Sasou *et al.* (2007) to examine the possibility of drag reduction and pumping power reduction. The time trace of the mean pressure gradient is square wave. Relaminarization of the pulsating turbulent flow is confirmed in a highly accelerated/decelerated flow with the suitable cycle period of more than 1000 wall units. The term of "relaminarization" is used for describing the phenomenon as a turbulent flow changes to a laminar flow.

The purpose of the present investigation is to optimize a time-varying mean pressure gradient to minimize the pumping power or skin friction by experiments. The effect of the mean pressure gradient and the bulk velocity upon the energy saving rate is also examined.

EXPERIMENTAL APPARATUS

A schematic drawing of the experimental apparatus is shown in Fig. 1. The test section is d = 20.1mm diameter and l = 2m long. The calibration of the diameter is conducted by using the relationship between the mean pressure gradient and the bulk velocity in the case of a steady turbulent flow in the pipe (see the data of the steady flow in Fig. 2). The test section is made in 1m (≈50d) downstream from a curved pipe, indicating that the inlet flow of the test section can be considered as the fully-developed turbulent flow without any influence of the upstream curved pipe. The working fluid is water. The pulsating flow is generated by adjusting the rotating speed of a centrifugal pump placed far from the test section. The rotating speed is periodically changed in time (see Fig. 5), and is decided as an obtained time trace of a mean pressure gradient imitates that of the DNS by Sasou *et al.* (2007).

The flow rate is measured by a magnetic flow meter (LF 600; Toshiba Co.) that is installed on the flow passage after the test section. The measurement of the pressure gradient is taken with a differential pressure gauge (EJX110J; YOKOGAWA Co.). Two pressure taps are 0.6mm inner diameter and locate on both the ends of the test section. The analog signals from the flow meter and the differential pressure gauge are sampled by a data acquisition system (MX100; YOKOGAWA Co.) with the sampling rate of 50 msec. Over 10 samples per cycle are taken in equal time intervals and over 20 cycles of data are totally collected on the disc of a computer in each experiment. The recorded data are phase-averaged. The temperature of the working fluid is measured by the thermocouple (type-K) that locates downstream of the centrifugal pump.

The friction Reynolds number is set to be Re_τ (= $u_\tau R/\nu$) = 110 that is based on the friction velocity u_τ [m/s], the radius R [m] and the kinematic viscosity ν [m^2/s]. The cycle period T [sec] is T^* = 0-30, where superscript of ( )* represents non-dimensionalization by u_τ and R. The pulsating flow in the pipe is driven by the periodic pressure gradient. The amplitude of the mean pressure gradient during acceleration phase standardized by $[-dp/dx]_T$ is α = 1-10. Here, $-dp/dx$ [N/m^3] is the pressure gradient and $[]_T$ the cycle-averaged mean value. Note that $[-dp^*/dx^*]_T = 2$ is formed in pipe flows and $[-dp^*/dx^*]_T = 1$ in channel flows. The flow direction is always in the same.

The drag reduction rate R_D [-] and the energy saving rate R_W [-] are evaluated and defined as:

$$C_f = \frac{\tau_w}{\frac{1}{2}\rho [u_b]_T^2}, \quad (1)$$

$$R_D = \frac{C_{f,\mathrm{Blasius}} - [C_f]_T}{C_{f,\mathrm{Blasius}}} \times 100, \quad (2)$$

$$W = \begin{cases} \dfrac{u_b\left(-\dfrac{dp}{dx}\right)}{\rho [u_b]_T^3} & (\text{for positive pressure gradient}), \\ R_E \dfrac{u_b\left(-\dfrac{dp}{dx}\right)}{\rho [u_b]_T^3} & (\text{for negative pressure gradient}), \end{cases} \quad (3)$$

$$R_W = \frac{W_{\mathrm{Blasius}} - [W]_T}{W_{\mathrm{Blasius}}} \times 100, \quad (4)$$

where C_f [-] is the fiction coefficient, ΔP [Pa] the differential pressure, u_b [m/s] the bulk velocity, ρ [kg/m^3] the density, τ_w [Pa] the wall shear stress, $()_{\mathrm{Blasius}}$ the value estimated by using the empirical formula of Blasius, W [-] the pumping power for making pulsating flow and R_E [%] the energy recovery rate. Note that the energy recovery rate of R_E = 100% implies that the pumping power in the deceleration phase is perfectly recovered and is reused to accelerate the flow. In order to obtain the friction coefficient, the wall shear stress is expressed as:

$$\tau_w = \frac{d}{4} \times \left(\frac{\Delta P}{l} - \rho \frac{du_b}{dt} \right) \quad (5)$$

from the Navier-Stokes equation.

RESULT AND DISCUSSION

Fig. 2 describes the dependence of the skin friction coefficient upon the bulk Reynolds number Re_b (= $u_b 2R/\nu$) of the steady flow. The present data correspond to the theoretical formula of the laminar flow when the bulk Reynolds number is 500 to 2000. It is also seen that the present data are in very good agreement with the Blasius equation for the bulk Reynolds number of 3000 to 30000. The bulk Reynolds number of the steady turbulent flow at Re_τ = 110 is about Re_b = 3012. Therefore, the target flow of the present study can be considered as a fully-developed turbulent flow.

ENERGY SAVING AND DRAG REDUCTION RATES

Fig. 3(a) depicts the dependence of the energy saving rate at the energy recovery rate of R_E = 100% upon the pressure gradient during the acceleration phase α and the cycle period T^*. Three points should be noted. First, the energy saving rate is positive at T^* = 0-15. The maximum energy saving rate is R_W = 58% and the maximum drag reduction rate is R_D = 63% (see Fig. 4(a)) at α = 4.1 and T^* = 10. Second, the optimum value of the cycle period exists at about T^* = 10 for the condition of constant α. This is in

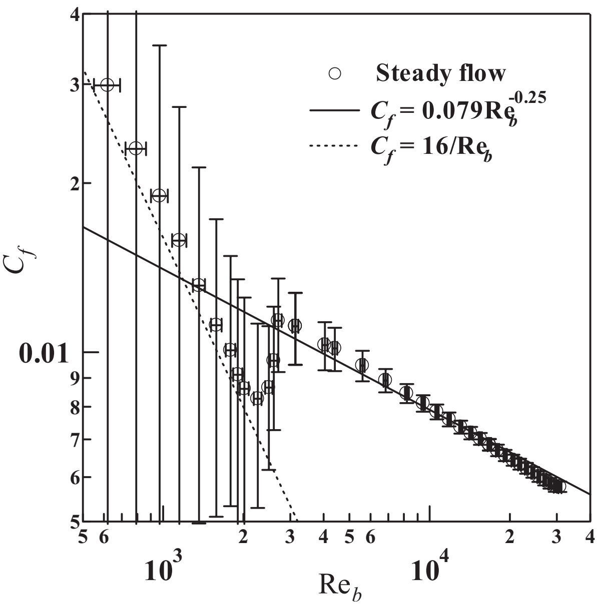

Fig. 2 Skin friction coefficient C_f for steady flow.

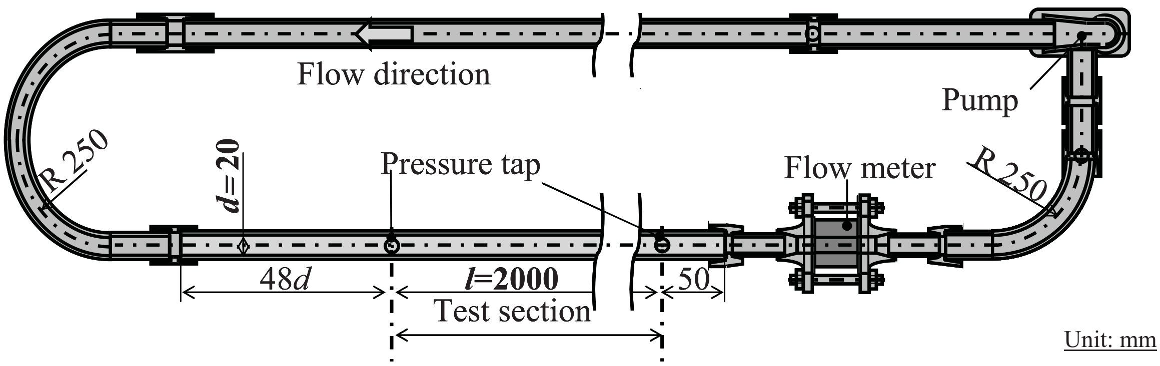

Fig. 1 Schematic of top-view experimental device.

good agreement with the optimum cycle period $T^* = 11.2$ of the DNS by Sasou *et al.* (2007). The optimum value of α is 2 to 4 for the condition of constant T^*. The energy saving rate decreases with the excessive amplitude of α. Thirdly, the energy saving rate of the present experiment (Fig. 3(a)) is lower than that of the DNS (Fig. 3(b)) for the same conditions of T^*, α and Re_τ. It should be noted that the energy saving rate is also affected by the difference of flow geometry (pipe/channel), the wave shape of the mean pressure gradient (see Fig. 8) and disturbance such as connecting parts upstream of the test section.

Fig. 4 displays the dependence of the drag reduction rate α and T^*. The positive drag reduction rate is confirmed at almost all the experimental conditions. The influence of T^* upon the drag reduction rate is similar to that on the energy saving rate. On the other hand, the drag reduction rate is increased with increasing α for the condition of constant T^*. This is mainly because the higher positive pressure gradient deteriorates turbulence in the acceleration phase and the higher adverse one makes slower laminar flow in the deceleration phase. When the cycle period over $T^* > 30$, the drag reduction rate is almost $R_D = 0$.

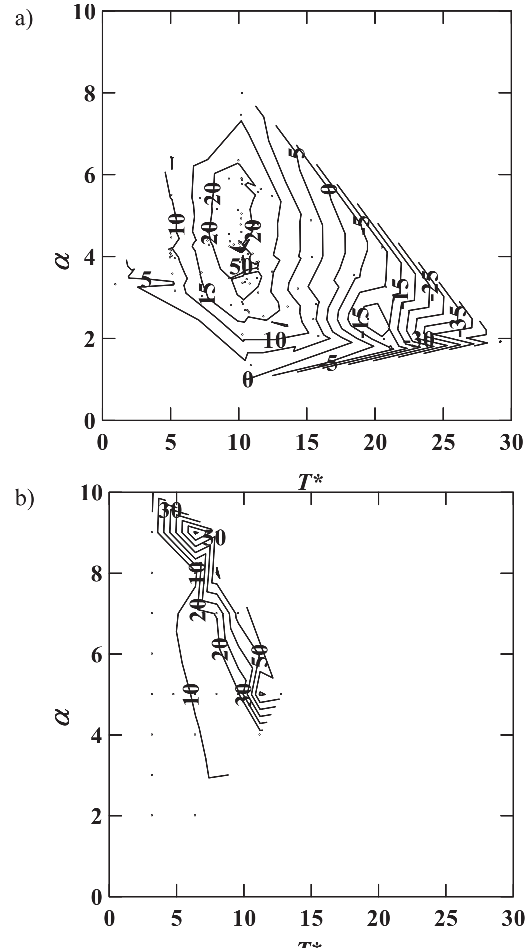

Fig. 3 Dependence of energy saving rate at the energy recovery rate of 100% for the negative pressure gradient phase upon pressure gradient in the acceleration period α and cycle period T^* at Re_τ=110; (a) present results, (b) results from DNS by sasou *et al.* (2007). Each dat represents each datum point.

DETAIL ANALYSIS FOR HIGHEST ENERGY-SAVING-RATE CASE

Fig. 5 shows the rotating speed of the centrifugal pump when the energy saving rate is highest $R_W = 58\%$ with the experimental conditions of $T^* = 10$ and $\alpha = 4.1$. The rotating speed is linearly increased/decreased in the first half of the acceleration/deceleration phases to generate the large amplitude of the pressure gradient.

Fig. 6 describes the phase-averaged skin friction coefficient versus the bulk Reynolds number. The phase-averaged friction coefficient is defied as:

$$C_f(t) = \frac{\dfrac{1}{N}\displaystyle\sum_{i=1}^{N}\tau_w(t+T\cdot i)}{\dfrac{1}{2}\rho\left(\dfrac{1}{N}\displaystyle\sum_{i=1}^{N}u_b(t+T\cdot i)\right)^2}, \tag{6}$$

where N is the collected number of cycle. The skin friction coefficient of the present experimental result is respectively decreased/increased in the acceleration/deceleration phases. The time trace roughly locates on the theoretical formula of

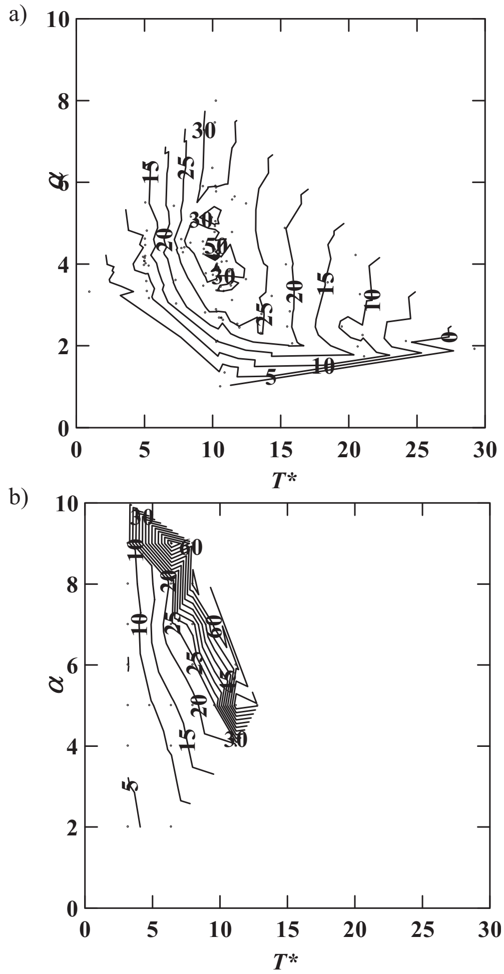

Fig. 4 Dependence of drag reduction rate upon pressure gradient in the acceleration period α and cycle period T^* at Re_τ=110; (a) present results, (b) results from DNS by sasou *et al.* (2007). Each dat represents each datum point.

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the laminar flow, indicating that the laminar flow is maintained at all the phases. It should be noted that the bulk Reynolds number becomes smaller than the transition Reynolds number (see Fig. 2) in the end of the deceleration phase. Accordingly, it may be expected that the trubulent flow field becomes laminar by itself in that phase.

The skin friction coefficient of the DNS, on the other hand, is respectively increased/decreased in the early phase of the acceleration and the other phases. The difference between the experimental and the DNS results is discussed below with Figs. 7 and 8. Note that the drag reduction rate of the DNS is about 70% at the same conditions (same α and T^*) which is higher than that of the experimental result. In order to obtain much higher energy saving rate in the experiment, the disturbance in the experimental device should be reduced.

Fig. 7(a) depicts the time traces of skin friction coefficient, the numerator and the denominator of Eq. (6) for the present experiment. In the acceleration phase, the denominator $1/2(u_b^*)^2$ is increased about 6 times (from 100 to 600), whereas the numerator τ_w is only increased about 4 times (from 0.5 to 2.0). Hence, the friction coefficient is decreased in the acceleration phase. In the deceleration phase, vice versa.

In the early phase of the acceleration for the DNS (see the period of $0 < T^* < 1$ in Fig. 7(b)), on the other hand, the numerator is rapidly increased about 5 times (from 0.2 to 1.0). The denominator only increases about 1.3 times (from 350 to 450). This phenomenon directly leads to the contradictory variation of skin friction coefficient in comparison with that of the present experiment. The same holds for the deceleration phase.

The difference between Figs. 7(a) and (b) is mainly due to the differentia of the time variation of the pumping force, $-dp^*/dx^*$, as displayed in Fig. 8. The force of the present study gradually rises and falls owing to the restriction of the actual pump performance. Consequently, the unsteady term, du_b^*/dt^*, tends to follow the change of the pumping force. As shown in Eq. (5), the difference between them equals to the numerator of Eq. (6), which slowly varies. Meanwhile, the wave shape of the force for the DNS is ideally set to be a square. The unsteady term cannot obey the change of the force. The differential between them becomes larger than that of the present experiment, directly leading to the difference of the skin friction coefficient shown in Fig. 6.

Fig. 9 describes the energy saving rate as a function of the energy recovery rate at T^* = 10 and α = 4.1. The maximum energy saving rate is R_W = 58% with the energy recovery rate of R_E = 100% as discussed with Fig. 3(a). Naturally, from a viewpoint of applying the present pulsating control to real engineering systems, an energy

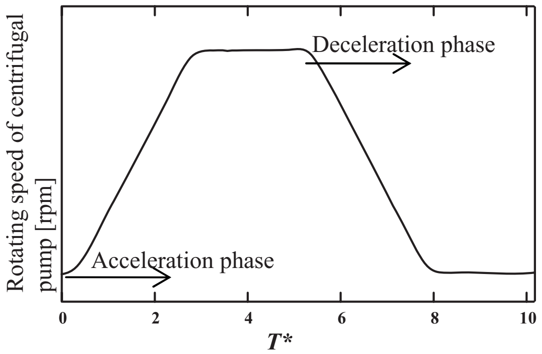


Fig. 5 Examples of time trace of rotating speed of centrifugal pump for best experimental condition.

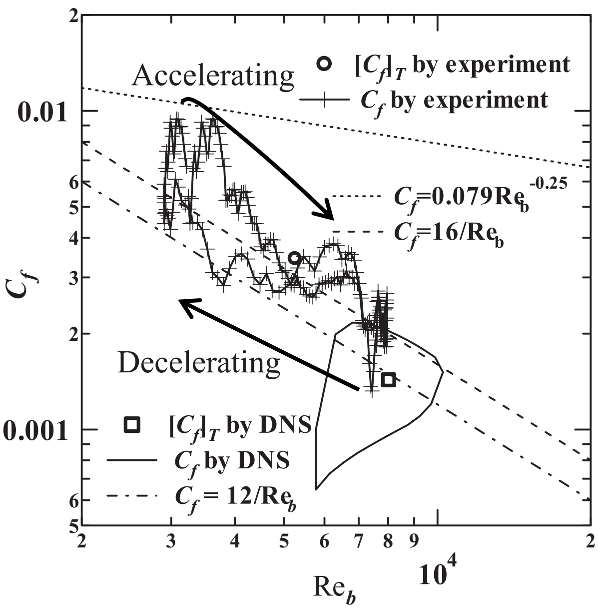
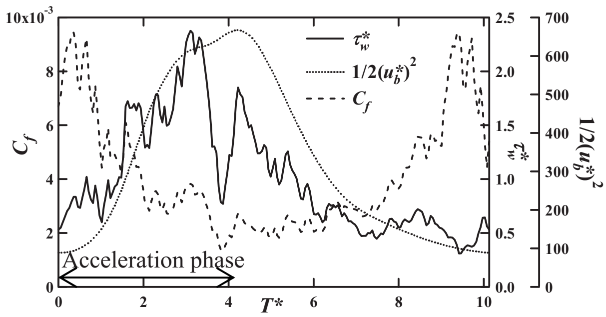


Fig. 6 Phase-averaged friction coefficient C_f. Present result at Re_τ=110, T^*=10 and α = 4.1; result from DNS by Sasou *et al.* (2007) at Re_τ=110, T^*=11.2 and α = 5.

(a)



(b)

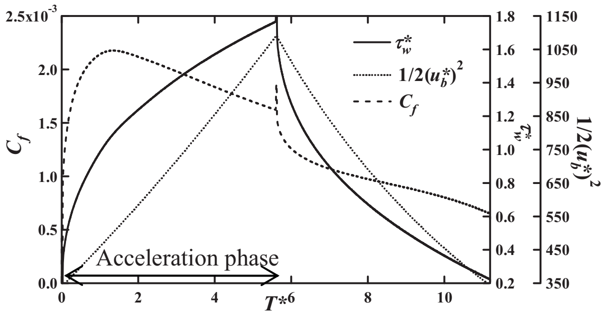


Fig. 7 Time trace of skin friction coefficient C_f, numerator τ_w and denominator $1/2(u_b^*)^2$ of Eq. (6). (a) present result at Re_τ=110, T^*=10 and α=4.1; (b) DNS result at Re_τ=110, T^*=11.2 and α=5 by Sasou *et al.* (2007)

recovery system must be added. As for the energy recovery rate of R_E = 0%, on the other hand, none of new facilities needs for real engineering applications. The energy saving rate of R_W = 29% can be achieved for the case of R_E = 0%, which is the half of the maximum rate.

SUMMARY AND CONCLUSION

The effect of a pulsating turbulent pipe flow upon the energy saving and drag reduction rates are experimentally investigated by controlling the rotating speed of a centrifugal pump. The Reynolds number Re_τ = 110 is same as that of the DNS by Sasou *et al.* (2007) for comparison and the following conclusions are remarked:

1. The maximum energy saving and drag reduction rates are respectively R_W = 58% and R_D = 63% for the case with the mean time pressure gradient during the acceleration term α = 4.1 and the cycle period T^* = 10. Moreover, the energy saving rate is R_W = 29% even though the energy recovery rate is R_E = 0%. It is found through the evaluation of the skin friction coefficient that a laminar flow is maintained at all the phases.
2. The cycle period T^* = 10 and the mean pressure gradient during the acceleration term α = 4 are optimum values for increasing the energy saving rate. The overall trend of the drag reduction rate is same as that of the energy saving one. These are similar to the results of the DNS by Sasou *et al.* (2007).
3. The skin friction coefficient respectively increases/ decreases on the deceleration/acceleration phases for all the drag-reduction cases mainly owing to the gradient change of the pumping force, $-dp^*/dx^*$.

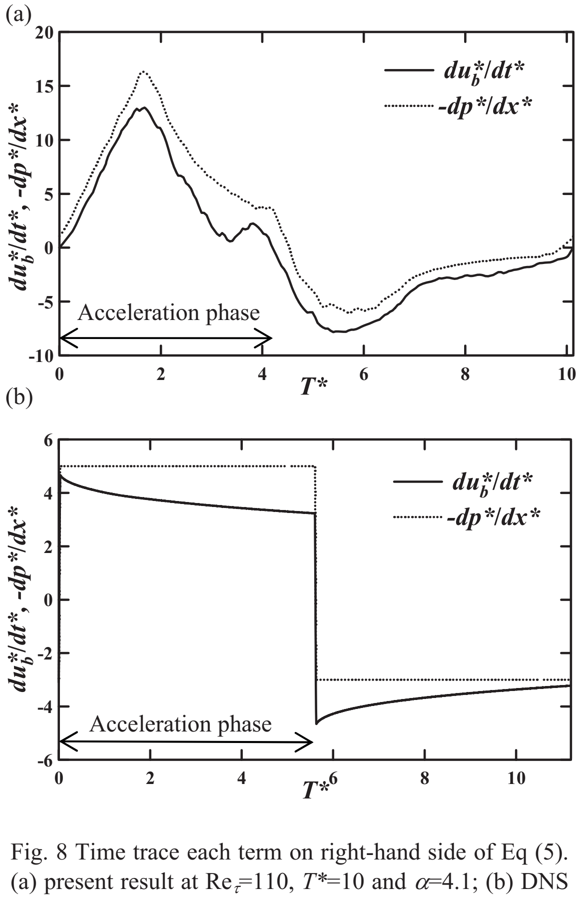


Fig. 8 Time trace each term on right-hand side of Eq (5). (a) present result at Re_τ=110, T^*=10 and α=4.1; (b) DNS result at Re_τ=110, T^*=11.2 and α=5 by Sasou *et al.* (2007)

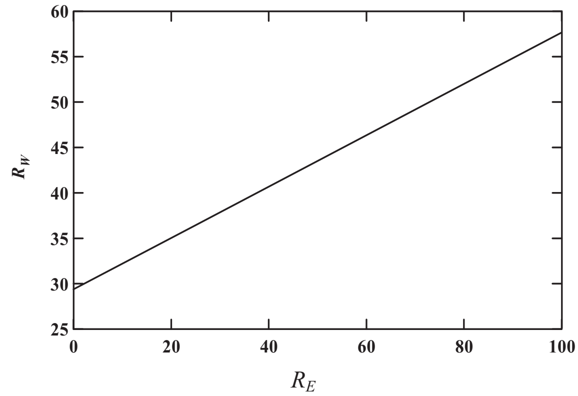


Fig. 9 Energy saving rate R_W versus energy recovery rate R_E at Re_τ=110, T^*=10 and α= 4.1.

REFERENCES

Bewley, T. R., Moin, P. and Temam, R., 2001, "DNS-based Predictive Control of Turbulence: an Optimal Benchmark for Feedback Algorithms", *J. Fluid Mech.,* Vol. 447, pp. 179-225.

Iguchi, M., Ohmi, M. and Nishizawa, K., 1985, "Relaminarization of Pulsatile Turbulent Flow in a Rectangular Duct", *Trans. of JSME,* Vol. B-51 (465), pp. 1384-1392, in Japanese.

Launder, B. E., 1964, "Laminarization of the Turbulent Boundary Layer in a Severe Acceleration", *Trans. ASME*, Ser. E, 86-4, pp. 707-708.

Lodahl, C. R., Sumer, B. M. and Fredøe, J., 1998, "Turbulent Combined Oscillatory Flow and Current in a Pipe", *J. Fluid Mech.*, Vol. 373, pp. 313-348.

Nishimura, T., Oka, N., Yoshinaka, Y. and Kunitsugu, K., 2000, "Influence of Imposed Osillatory Frequency on Mass Transfer Enhancement of Grooved Channels for Pulsatile Flow", *Int. J. of Heat and Mass Transfer*, Vol. 43, pp. 2365-2374.

Sasou, N., Iwamoto, K. and Kawamura, H., 2007 "Drag Reduction by Laminarization of Pulsating Turbulent Channel Flow", *Proc. 21st Japan Society of CFD Symposium*, pp. 213-218, in Japanese.

Sreenivasan, K. R., 1982, "Laminarescent, Relaminarizing and Retransitional Flows", *Acta Mechanica,* Vol. 44, pp. 1-48.

Sumida, M., 2003, "Pulsating Flow through a Tapered Curved Tube", *Trans. JSME,* Vol. B-69(678), pp. 98-105, in Japanese.

Takami, T., Ohsaki, T. and Miyanaga, H., "Wall Drag and Power Balance in Pulsating Turbulent Pipe Flow", *JSME Fluids Engineering Conference*, Vol. 2005, p. 125, in Japanese.