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DELAY IN RESPONSE OF TURBULENT HEAT TRANSFER AGAINST ACCELERATION OR DECELERATION OF FLOW IN A PIPE

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ABSTRACT

To predict the heat transfer enhancement as a result of application of a pulsating flow in a pipe, the response of turbulent heat transfer against the acceleration or deceleration of the flow in a pipe was investigated experimentally. The Reynolds numbers with the valve open (Re_{max}) and close (Re_{min}) were systematically varied in the range of 8,000 ≤ Re_{max} ≤ 34,000 and 700 ≤ Re_{min} ≤ 23,000, respectively. The spatiotemporal variation of the heat transfer in a pipe was measured using infrared thermography simultaneously with the temporal variation of the flow properties. Based on the experimental results, it was found that the delay of the heat transfer in response to the acceleration or deceleration can be characterized using the corresponding time lag Δt and first-order time constant τ. The values of Δt and τ can be expressed as non-dimensional forms of $\Delta t/(\nu/u_\tau^2)$ and $\tau/(R/u_\tau)$, respectively, where u_τ is wall friction velocity, ν is kinematic viscosity of fluid, and R is radius of a pipe.

INTRODUCTION

Although many studies have been carried out to enhance the turbulent heat transfer by pulsating the flow (e.g. Dec et al., 1992; Ishino et al., 1996), there are many conflicting results, and thus, a common understanding has not been reached as to what conditions the heat transfer is enhanced or suppressed. The main reason for this is considered to be due to the lack of data of the heat transfer variation at each phase of the pulsation.

A technique to measure the spatiotemporal variation of the heat transfer in a water pipe flow has been developed using high-speed infrared thermography (Nakamura et al., 2017) and the possibility of heat transfer enhancement has been explored by applying a pulsed rectangular wave to the flow (Shiibara et al., 2017). As a result, it was demonstrated that the heat transfer can be effectively enhanced by applying a quasi-rectangular flow pulsation with the Reynolds number oscillating between Re = 0 and 8,000. In addition, the results indicated that the enhancement or suppression of the heat transfer is closely related to the difference in the characteristics of the time-response of the heat transfer between the flow acceleration and deceleration phases. Thus, if the time response of the heat transfer at the acceleration and deceleration phases can be modelled separately, it will be possible to predict the heat transfer enhancement quantitatively according to the pulsating conditions.

In this study, the time response of the heat transfer was investigated experimentally against flow acceleration or deceleration by measuring the spatiotemporal variation of the heat transfer in a pipe using infrared thermography. The temporal variation of the flow properties in the pipe was simultaneously measured. The experiments were performed by systematically changing the Reynolds numbers of the maximum and minimum flow rates to elucidate the parameters which dominate the time response of the heat transfer.

EXPERIMENT

Measurements were performed using the experimental setup shown in Fig. 1. In this setup, water in a head tank (1) is supplied to a horizontal circular pipe through a rectifier tank (3). The inner diameter of the circular pipe was D = 20 mm, and the length of the inlet region (from the pipe inlet to the upstream port of the differential manometer (7)) was 2,040 mm. A bell mouth is installed at the pipe inlet. The acceleration and deceleration of the flow were achieved by respectively opening and closing an air-actuating valve (12) (VDA10705, OMAL) installed downstream from the test section (8). The Reynolds number with the valve open was varied from Re_{max} = 8,000 to 34,000, and that with the valve close was varied from Re_{min} = 700 to 23,000. The Reynolds number was evaluated from the kinematic viscosity at the film temperature.

Figure 2 shows the test section for the heat transfer measurement. The circular duct was fabricated from an acrylic pipe of 280 mm in length, and an open section spanning a length of 80 mm and an angle of θ = −70˚ to 70˚ was cut into the downstream region of the pipe, as shown in Fig. 2(a) and (b). Titanium foil with a thickness of 40.6 μm was glued around the entire circumference of the inner surface of the pipe, including the removed section. To increase the emissivity of the test surface, the outer surface of the titanium foil was coated with black paint with a thickness of about 15 μm. The titanium foil was heated electrically using electrodes attached to both ends of the test section. The instantaneous temperature distribution and its temporal fluctuation on the test surface was measured using a high-speed infrared thermograph (IRT;

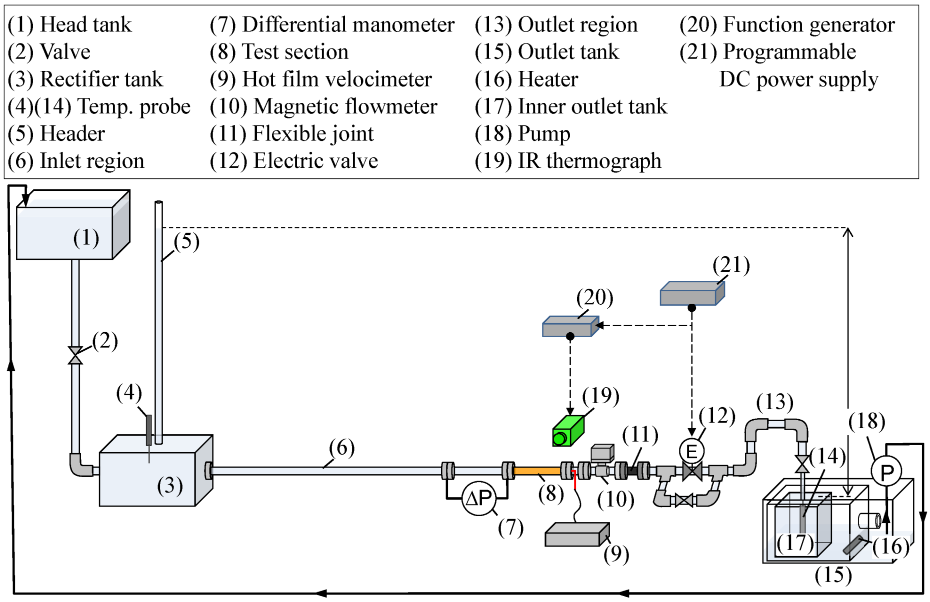
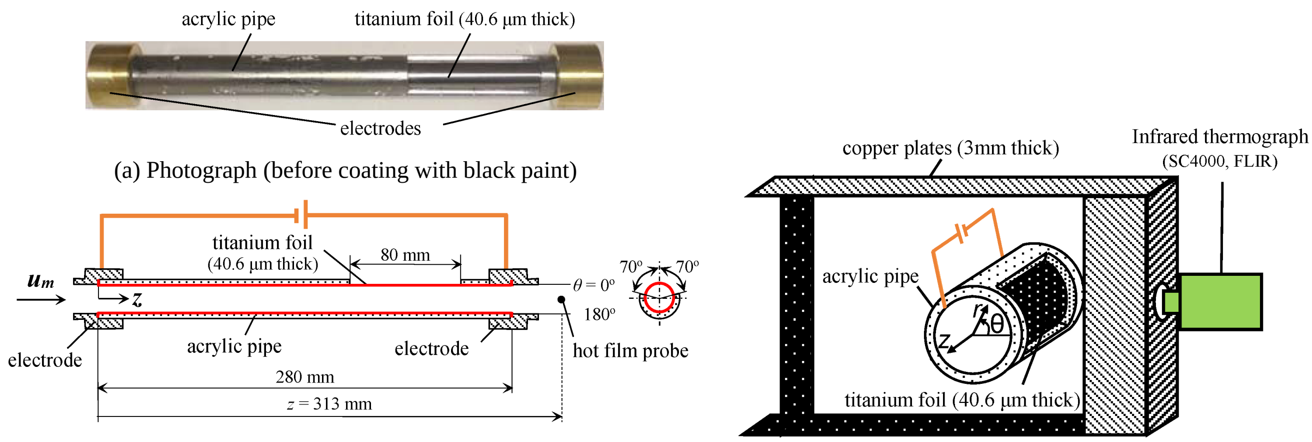


Figure 1. Overall view of the experimental setup.



(a) Photograph (before coating with black paint)

(b) Top-down cross-sectional view

(c) Schematic of measurement process

Figure 2. Test pipe for heat transfer measurement.

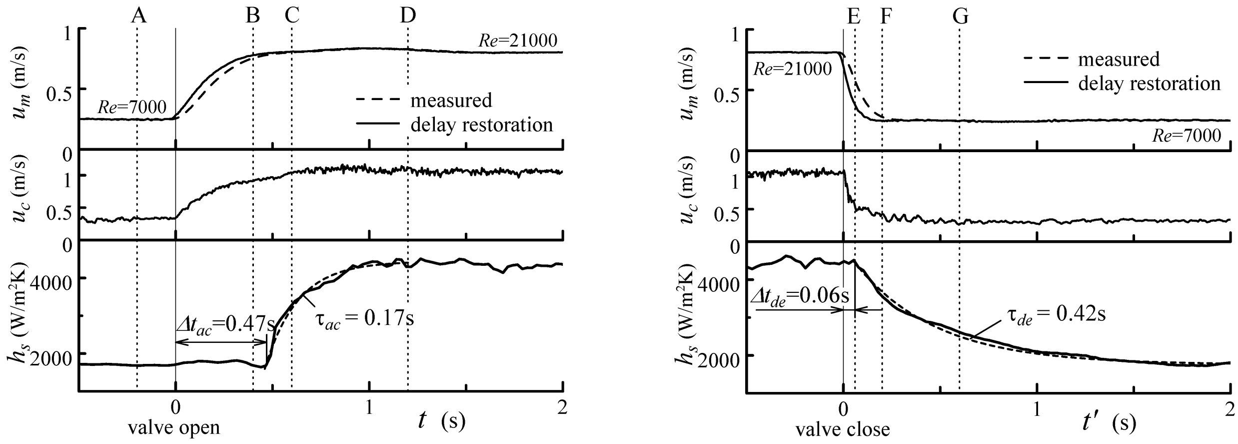
SC4000, FLIR). The instantaneous local heat transfer coefficient h for the water flow was evaluated as $h = \dot{q}_{cv}/(T_w - T_m)$, where T_w is the instantaneous local temperature of the heated surface measured using the IRT, T_m is the instantaneous mixed-mean temperature of the water flow, and $\dot{q}_{cv}$ is the instantaneous local convective heat flux of the water flow considering the heat losses, thermal diffusion, and thermal inertia of the heated surface (Nakamura et al., 2017).

MEASURED DATA

Acceleration and Deceleration Between Turbulent Flows

An example of the measured data between turbulent flows is shown in Figs. 3 and 4. Figure 3(a) and (b) shows the time series waveforms for the sudden acceleration and deceleration of the flow from Re = 7,000 to 21,000 and from Re = 21,000 to 7,000, respectively. From the top, the plots show the sectional-average velocity in the pipe, u_m, velocity fluctuation at the center of the pipe, u_c, and the surface-average heat transfer coefficient h_s. The sectional-average velocity u_m was obtained from the flow rate measured using electromagnetic flowmeter (MGR11A, Azbil). Since there is a delay in the output of the flowmeter, the delay was restored by finite difference analysis as shown by a solid line. The velocity fluctuation u_c was measured by a hot-film probe (55R14, Dantec) located at z = 313 mm, which was about 65-115 mm downstream of the heat-transfer measurement surface. Figure 4 shows the instantaneous distributions of the heat transfer coefficient corresponding to the times labelled A–G in Fig. 3.

Before the sudden acceleration (time A at Re = 7,000), the instantaneous heat transfer coefficient exhibits a structure extending in the flow direction (Fig. 4 (a)), indicative of a streaky turbulent structure. Shortly after the valve was opened (t = 0–0.47 s in Fig. 3(a)), the heat transfer coefficient remains low due to the laminarization effect although the Reynolds number increases to 21,000. The streaky structure is stretched in the flow direction as the flow velocity increases (Fig. 4(b),



(a) Sudden acceleration from Re = 7,000 to 21,000 (b) Sudden deceleration from Re = 21,000 to 7,000

Figure 3. Time series waveforms for flow acceleration and deceleration between turbulent flows.

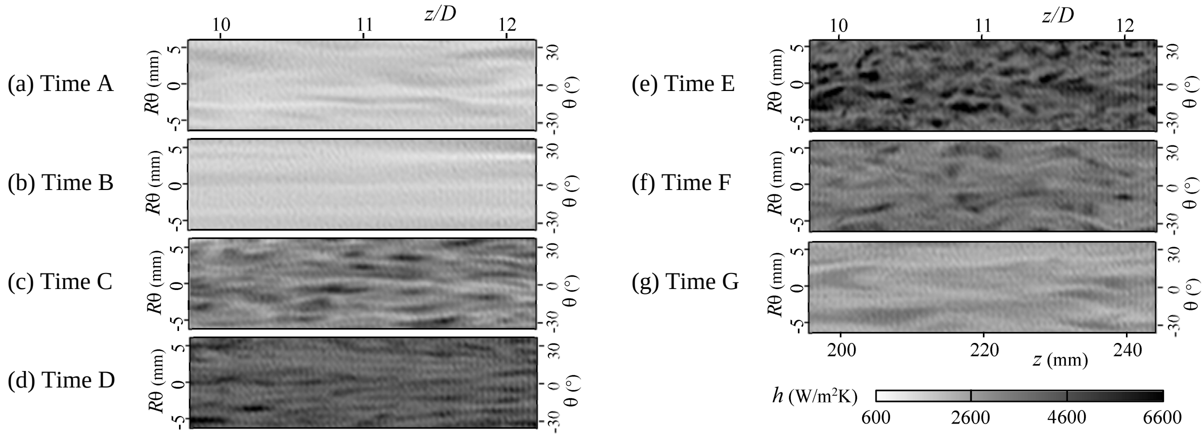


Figure 4. Instantaneous distributions of heat transfer coefficient corresponding to Fig. 3.

time B). After that (t > 0.47 s), the heat transfer coefficient increases rapidly to form a finer structure (Fig. 4(c), time C) due to a progress of flow turbulence. And ultimately reaches a state of steady turbulence of Re = 21,000 at approximately t = 1.2 s (Fig. 4(d), time D).

Immediately after the valve was closed (t' > 0 in Fig. 3(b)), the flow decelerates suddenly. At that moment (t' = 0–0.06 s), a mottled structure with a high heat transfer appears (Fig. 4(e), time E); this structure is considered to be caused by the crushing of the streaky structure in the flow direction. As this structure diffuses over time (Fig. 4(f) and (g), times F and G), the heat transfer coefficient decreases gradually, although the flow rate decreases rapidly. And ultimately, the heat transfer coefficient reaches a state of steady turbulence of Re = 7,000 at approximately t' = 2 s.

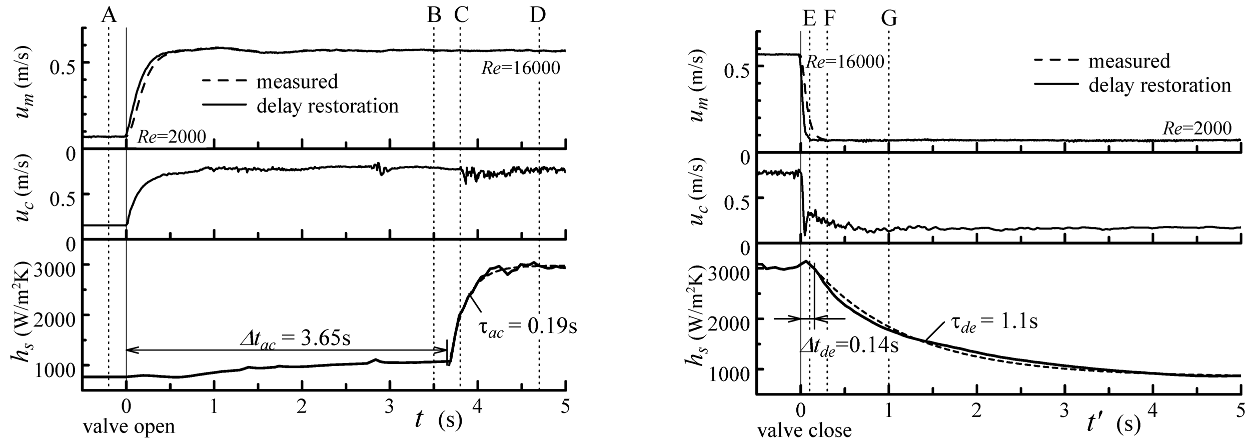
A total of 30 measurements were carried out between turbulent flows by systematically changing the combination of Reynolds numbers. As a result, the variation of the heat transfer with the acceleration and deceleration was qualitatively the same in each case while the delay of the heat transfer became shorten as the Reynolds number increased.

Acceleration and Deceleration Between Laminar and Turbulent Flow

Next, an example of the measured data between laminar and turbulent flow is shown in Figs. 5 and 6. Figure 5(a) and (b) shows the time series waveforms for the sudden acceleration and deceleration of the flow from Re = 2,000 to 16,000 and from Re = 16,000 to 2,000, respectively. Figure 6 shows the instantaneous distributions of the heat transfer coefficient corresponding to the times labelled A–G in Fig. 5.

Before the sudden acceleration (time A at Re = 2,000), the instantaneous heat transfer coefficient is uniform (Fig. 6 (a)) and not fluctuate in time since the flow is laminar. After the valve was opened (t > 0 in Fig. 5(a)), the heat transfer coefficient remains low keeping the state of laminar flow (Fig. 6(b), time B) although the Reynolds number increases to 16,000. The flow is sometimes disturbed (at t = 1 s and 2.8 s in Fig. 5(a)), but it does not grow into turbulent state. After that (t > 3.65 s), the heat transfer coefficient increases rapidly to form a finer structure due to the turbulent transition (Fig. 6(c), time C) and ultimately reaches a state of steady turbulence of Re = 16,000 at approximately t = 4.7 s (Fig. 6(d), time D). The turbulent transition in this experiment is considered to be triggered by a small protrusion, which existed at the inlet of the pipe.

Immediately after the valve was closed (t' > 0 in Fig. 5(b)), the flow decelerates suddenly. At that moment (t' = 0–0.14 s), a mottled structure with a high heat transfer appears (Fig. 6(e), time E) as in the case of deceleration between turbulent flows (Fig. 4 (e)). As this structure diffuses over time (Fig. 6(f) and (g), times F and G), the heat transfer coefficient decreases gradually, although the flow rate decreases rapidly. And



(a) Sudden acceleration from Re = 2,000 to 16,000

(b) Sudden deceleration from Re = 16,000 to 2,000

Figure 5. Time series waveforms for flow acceleration and deceleration between laminar and turbulent flow.

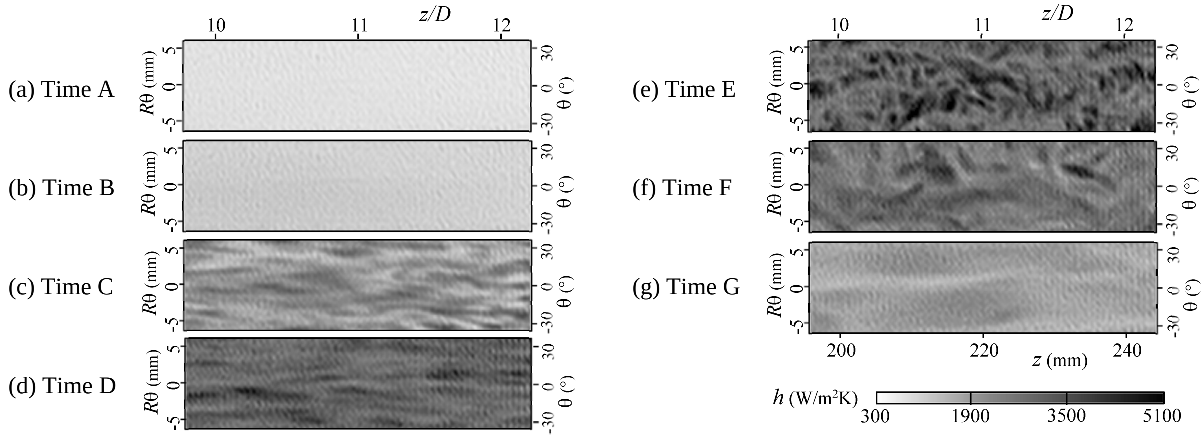
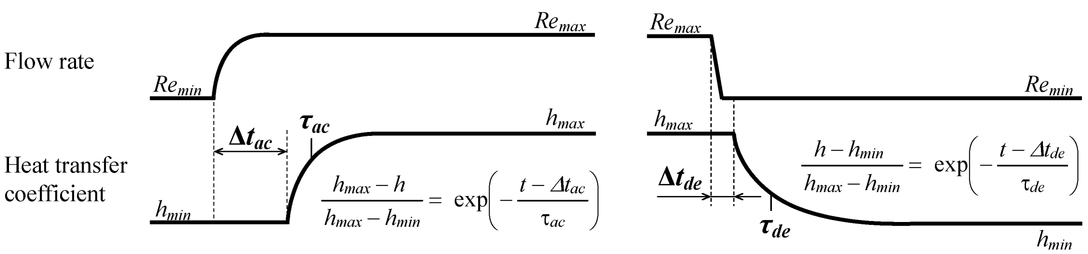


Figure 6. Instantaneous distributions of heat transfer coefficient corresponding to Fig. 5.



$$\frac{h_{max}-h}{h_{max}-h_{min}} = \exp\left(-\frac{t-\Delta t_{ac}}{\tau_{ac}}\right) \qquad \frac{h-h_{min}}{h_{max}-h_{min}} = \exp\left(-\frac{t-\Delta t_{de}}{\tau_{de}}\right)$$

Figure 7. Schematic of the time-responses of the heat transfer coefficient to the sudden acceleration and deceleration of the flow.

ultimately, the heat transfer coefficient becomes uniform reflecting the laminar state of Re = 2,000 at approximately t' = 5 s.

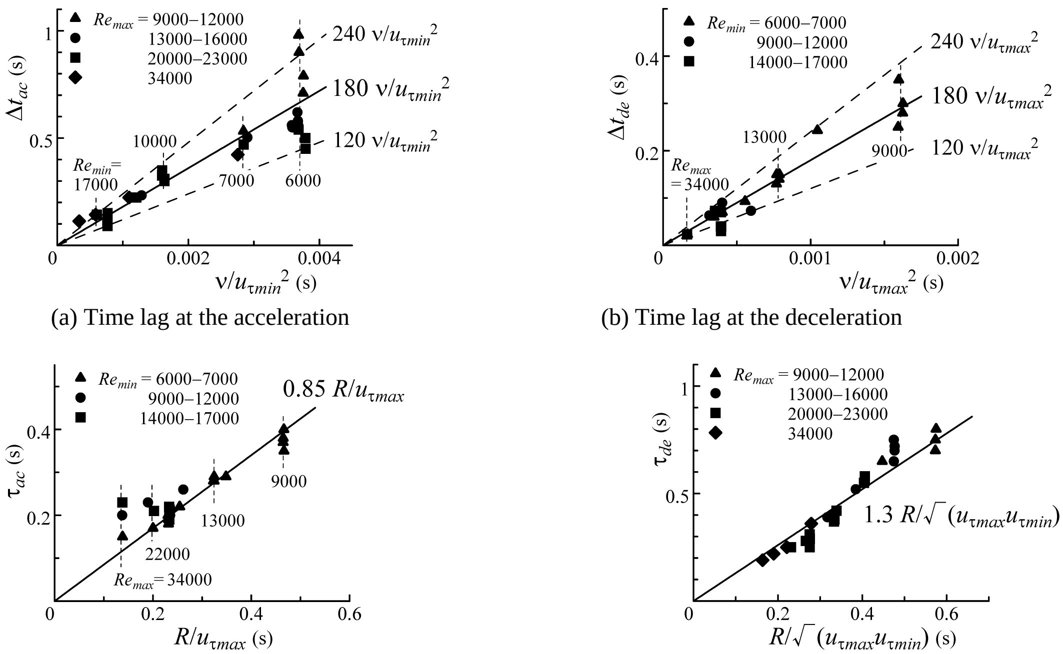
A total of 37 measurements were carried out between laminar and turbulent flow by systematically changing the combination of Reynolds numbers. As a result, the variation of the heat transfer with the acceleration and deceleration was qualitatively the same in each case. The delay of the heat transfer seems to be independent of the laminar Reynolds number Re_{min}, while the delay became shorten as the turbulent Reynolds number Re_{max} increases.

CHARACTERIZATION OF TIME RESPONSE

Modelling of the Delay

The delay of the heat transfer for both between turbulent flows (turbulent-turbulent case as shown in Fig. 3) and between laminar and turbulent flow (laminar-turbulent case as shown in Fig. 5) can be characterized using the time lag Δt and the first-order time constant τ. These trends are schematically shown in Fig. 7. In general, the acceleration time lag Δt_{ac} is considerably larger than the deceleration time lag Δt_{de}, and the acceleration time constant τ_{ac} is considerably smaller than the deceleration time constant τ_{de}. For example, in the case shown in Fig. 3, the acceleration time lag and time constant were estimated to be Δt_{ac} = 0.47 s and τ_{ac} = 0.17 s, and those for the deceleration were estimated to be Δt_{de} = 0.06 s and τ_{de} = 0.42 s.

These parameters of Δt_{ac}, τ_{ac} , Δt_{de}, and τ_{de} were formulated as a function of wall friction velocity $u_\tau = u_m\sqrt{\lambda/8}$, where



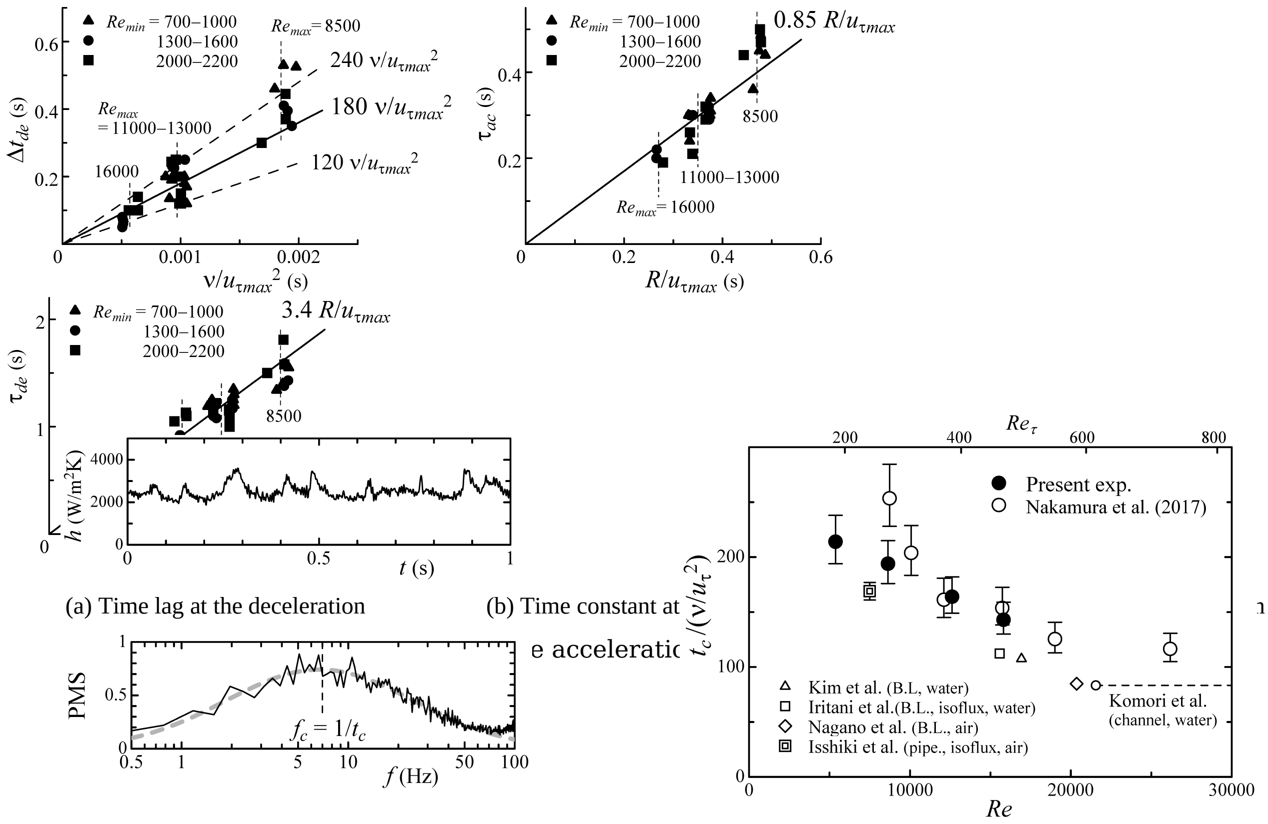
(a) Time lag at the acceleration

(b) Time lag at the deceleration

(c) Time constant at the acceleration

(d) Time constant at the deceleration

Figure 8. The delay of the heat transfer at the acceleration and the deceleration (turbulent-turbulent case).



(a) Time lag at the deceleration

(b) Time constant at

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(a) Temporal fluctuation of the local heat transfer coefficient

λ is pipe friction coefficient. The wall friction velocities at Re_{max} and Re_{min} are denoted as $u_{\tau max}$ and $u_{\tau min}$. To develop a dimensionless formula, the time lag Δt and the time constant τ were plotted against R/u_τ or ν/u_τ^2, which have dimensions of time, as shown in Figs. 8 and 9. Here, R is the pipe inner radius and ν is the kinematic viscosity of water.

Formulation of Time-Lag

Figures 8(a) and (b) show the time-lag at the acceleration Δt_{ac} and the deceleration Δt_{de} for the turbulent-turbulent case, and Fig. 9(a) shows that at the deceleration Δt_{de} for the laminar-turbulent case. The value Δt_{ac} for the laminar-turbulent case was not plotted here since it seems to depend on the surface roughness of the pipe used here. In other words, it may be possible to control the time lag Δt_{ac} for the laminar-turbulent case by the surface roughness of the pipe.

The time lag Δt tended to vary in proportion to ν/u_τ^2 rather than R/u_τ, although the plots are largely scattered. It is known that the characteristic time of turbulent vortical structure near the wall can be non-dimensionalized by ν/u_τ^2 (e.g. Kasagi et al., 1989). Namely, the time lag is considered to be related to the vortex structure near the wall.

The value Δt_{ac} for the turbulent-turbulent case can be approximated as non-dimensional form of $\Delta t_{ac}/(\nu/u_{\tau min}^2) \approx$ 120-240 and the value Δt_{de} for both the turbulent–turbulent and the laminar-turbulent cases can be approximated as $\Delta t_{de}/(\nu/u_{\tau max}^2) \approx$ 120-240. Here, the acceleration time-lag Δt_{ac} depends on $u_{\tau min}$, which is the wall friction velocity before the flow accelerates, and the deceleration time-lag Δt_{de} depends on $u_{\tau max}$, which is that before the flow decelerates. The proportional constants of both Δt_{ac} and Δt_{de} were similar as 120-240.

Interestingly, this time is similar to the characteristic period of the heat transfer fluctuation $t_c/(\nu/u_\tau^2) \approx$ 130-240 as shown in Fig. 11 at Re = 5000-16000. The value t_c was estimated here from the maximum of the pre-multiplied power spectrum (Fig. 10(b)), which is calculated from the temporal fluctuation of the local heat transfer coefficient (Fig. 10(a)). As shown in Fig. 11, the value t_c corresponds well to the mean bursting period reported in the previous researches. Assuming that the heat transfer fluctuation is mainly caused by the process from the formation to the collapse of the streaky structure, it can be considered that the t_c corresponds the value of mean length of the streaky structure divided by the advection velocity. From the above consideration, it is reasonable to suppose that the time lags are related to the length of the streaky structure which corresponds to the mean bursting period, although the mechanisms of it should be clarified in the future research.

Formulation of Time-Constant

Figures 8(c) and (d) show the time constants at the acceleration τ_{ac} and the deceleration τ_{de} for the turbulent-turbulent case, and Fig. 9(b) and (c) show those for the laminar-turbulent case. The time constant τ is plotted against R/u_τ since it tended to vary in proportion to R/u_τ rather than ν/u_τ^2.

The observation of the time series data of instantaneous heat transfer coefficient indicates that the acceleration time constant τ_{ac} corresponds to the time from the initiation of turbulent vortical structure of Re_{max} until it develops into steady turbulence. Namely, it is reasonable to consider that the value τ_{ac} is related to $u_{\tau max}$. As shown in Fig. 8(c) and Fig. 9(b), the value τ_{ac} can be approximated by a non-dimensional form as $\tau_{ac}/(R/u_{\tau max}) \approx 0.85$ for both the turbulent-turbulent and the laminar-turbulent cases. Note that the value τ_{ac} cannot be smaller than the flow acceleration time. Thus, as shown in Fig. 8(c), the value τ_{ac} cannot be decreased at $R/u_{\tau max} \leq 0.2$ (at $Re_{max} \geq$ 22,000).

The observation of the instantaneous heat transfer coefficient indicates that the deceleration time constant τ_{de} from the turbulent to laminar flow corresponds to the time from the formation of the mottled structure (Fig. 6(e)) of Re_{max} until it diffuses to the laminar state. Namely, it is reasonable to consider that the value of τ_{de} for the laminar-turbulent case is related to $u_{\tau max}$. As shown in Fig. 9(c), the value τ_{de} for the laminar-turbulent case can be approximated by a non-dimensional form as $\tau_{de}/(R/u_{\tau max}) \approx 3.4$, which is about four times that at the acceleration of $\tau_{ac}/(R/u_{\tau max}) \approx 0.85$.

In contrast, the value τ_{de} for the turbulent–turbulent case (Fig. 8(d)) is well proportional to $R/\sqrt{u_{\tau max}\, u_{\tau max}}$, rather than $R/u_{\tau max}$. The reason for this is considered to that the deceleration from Re_{max} (turbulent) to Re_{min} (turbulent) is affected by both the diffusion time of the mottled structure at Re_{max} (Fig. 4(e)) and the time to organize the turbulent vortical structure of Re_{min} (Fig. 4(a)). As shown in Fig. 8(d), the value τ_{de} for the turbulent-turbulent case can be well approximated by a non-dimensional form as $\tau_{de}/(R/\sqrt{u_{\tau max}\, u_{\tau max}}) \approx 1.3$.

Using these correlations, it is expected that the heat transfer enhancement or suppression of the pulsating flow consisting of sudden acceleration and deceleration can be predicted for any combination of Re_{max}, Re_{min}, and the pulsating conditions (pulsation cycle period and duty ratio).

CONCLUSION

The delay of the heat transfer in response to the sudden acceleration or deceleration of the flow in a pipe was investigated experimentally. As a result, it was found that the delay at the acceleration and the deceleration can be characterized using the corresponding time lag Δt and first-order time constant τ. The values of Δt and τ can be expressed as non-dimensional forms of $\Delta t/(\nu/u_\tau^2)$ and $\tau/(R/u_\tau)$, respectively, where u_τ is wall friction velocity, ν is kinematic viscosity of fluid, and R is radius of a pipe.

The time lag at the acceleration Δt_{ac} between the turbulent flows can be approximated as $\Delta t_{ac}/(\nu/u_{\tau min}^2) \approx$ 120-240, and that at the deceleration Δt_{de} for both the turbulent–turbulent and the laminar-turbulent cases can be approximated as $\Delta t_{de}/(\nu/u_{\tau max}^2) \approx$ 120-240, where $u_{\tau min}$ and $u_{\tau max}$ are wall friction velocities before the acceleration and before the deceleration, respectively. The time constant at the acceleration τ_{ac} can be approximated as $\tau_{ac}/(R/u_{\tau max}) \approx 0.85$ for both the turbulent–turbulent and the laminar-turbulent cases. The time constant at the deceleration τ_{de} from the turbulent to laminar flow can be approximated as $\tau_{de}/(R/u_{\tau max}) \approx 3.4$, and that between the turbulent flows can be approximated as $\tau_{de}/(R/\sqrt{u_{\tau max}\, u_{\tau max}}) \approx 1.3$.

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