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Quantitative measurement of spatio-temporal heat transfer to a turbulent water pipe flow

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ABSTRACT

A technique using high-speed infrared thermography was applied to measure the spatio-temporal heat transfer to a turbulent water flow in a horizontal circular pipe. The instantaneous distribution of the heat transfer coefficient and its temporal fluctuation was evaluated by solving inverse heat conduction equation of the heated thin-test-surface. As a result, it was demonstrated that the quantitative measurement, not only the time-averaged heat transfer, but also the statistics of the spatio-temporal fluctuation, was possible using this technique. In addition, a unique feature of the spatio-temporal heat transfer was clearly visualized for the turbulent pipe flow, which was dominated by the streaky structure similar to that for the turbulent boundary layer and the turbulent channel flow.

1. Introduction

Heat transfer in a pipe flow is essential in engineering application since it is the most fundamental element for heat exchange and heat transportation in various heat transfer equipment. Thus, numerous studies have been performed to investigate heat transfer in pipe flows. Among them, empirical correlations of the turbulent pipe flow, for example, by Dittus–Boelter, Petukhov (1970), Gnielinski (1976), are well-known, and have been frequently used to estimate heat transfer and to predict thermal performance and temperature of heat transfer devices. In these correlations, heat transfer, which is generally represented as Nusselt number, is treated as a variable that does not change with time. That is, the turbulent heat transfer has been treated as steady-state phenomenon in most cases in the previous studies.

In another aspect, the turbulent heat transfer is, by nature, nonuniform and unsteady, a fact reflected by the intermittent characteristics of the turbulent flow near a wall. This causes a wall-temperature fluctuation due to unsteady heat transfer between a fluid and a solid. This phenomenon, which is referred to as unsteady conjugate heat transfer, has believed to occur in numerous heat transfer devices that have been used in practice.

Unsteady conjugate heat transfer causes two problems from the viewpoint of the thermal design of devices. The first is the erroneous prediction of the total amount of heat transfer across the wall due to the nonlinear dependence between the wall heat flux and the heat transfer coefficient (Zudin, 2011; Mathie and Markides, 2013). If the prediction is made under the assumption of a constant wall-temperature, as is the conventional practice, the predicted heat flux should be different from the actual heat flux subject to temperature fluctuations. Although this effect leads to deterioration in the accuracy and reliability of the thermal design of equipment, it has rarely been considered in actual thermal design.

The second problem is high-cycle thermal fatigue, which has been reported as “thermal striping” in the piping systems of power plants (JSME standard 2003). This may be caused by repeated thermal stress in a solid wall if the amplitude of the thermal stress due to the wall temperature fluctuation exceeds the fatigue limit of the material.

In order to predict the unsteady conjugate heat transfer, it is important to understand the surface distribution of the heat transfer and its temporal fluctuation. However, the conventional measurements, that is, point measurements using such as thermocouples and heat flux sensors, accompanies difficulties to investigate the spatial distribution of heat transfer.

The present authors have developed the technique to measure the spatio-temporal heat transfer to turbulent air flows using infrared thermography (Nakamura, 2007; Nakamura and Yamada, 2013). This measurement is based on the principle that the instantaneous temperature distribution and its temporal fluctuation on a nearly isoflux wall (heated thin-foil) is reflected by the spatio-temporal fluctuation of the heat transfer to a turbulent flow. Although the temperature fluctuation on the foil attenuates in time and space due to a finite heat capacity and a lateral conduction, the quantitative heat transfer coefficient without attenuation can

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Nomenclature

D	inner diameter of pipe m
f	frequency Hz
f_{cut}	cutoff frequency of low-pass filtering Hz
h	heat transfer coefficient $= \dot{q}_{cv}/(T_w - T_m)$ W/(m^2·K)
h_m	mean heat transfer coefficient $=\overline{\dot{q}_{cv}}/(\overline{T_w} - T_m)$ W/(m^2·K)
k	wavenumber m^{-1}
k_{cut}	cutoff wavenumber of low-pass filtering m^{-1}
l	wavelength m
l_c	mean spacing of thermal streaks m
l_{cut}	cutoff wavelength of low-pass filtering$=1/k_{cut}$ m
Nu	Nusselt number$=h_m D/\lambda$
$\dot{q}$	heat flux W/m^2
R	inner radius of pipe m
Re	Reynolds number: $Re_D = u_m D/\nu, Re_\tau = u_\tau R/\nu$
T_{amb}	ambient wall temperature K or °C
T_m	mixed-mean temperature of the water K or °C
T_w	wall temperature on the test surface K or °C
t	time s
t_c	characteristic period of heat transfer fluctuation s
u_m	bulk mean velocity m/s
u_τ	wall-friction velocity m/s
z	streamwise coordinate
Δt	frame interval s
$\Delta(R\theta)$	pixel pitch in the circumferential direction m
Δz	pixel pitch in the streamwise direction m
δ	thickness m
ε_λ	spectral emissivity for infrared thermograph
λ	thermal conductivity W/(m·K)
ν	kinematic viscosity of fluid m^2/s
θ	circumferential coordinate

Subscripts

p, t	black paint, titanium foil

Other symbols

$(\overline{\ })$,$(\ )'$	averaged value, fluctuating value
$(\ )^+$	dimensionless value
$()_{\mathrm{rms}}$	root-mean-square value

be evaluated by solving the inverse heat conduction equation of the test surface. The result of this measurement revealed unique features of the spatio-temporal heat transfer to turbulent air flows that correlate with flow features in the near-wall region. In addition, the evaluated value of time-spatial variation of the heat transfer coefficient was confirmed to be reliable by comparing the statistics with the existing experimental and numerical results.

In this work, we applied this technique to measure the turbulent heat transfer in a pipe flow. Water was used as the fluid since the liquid flow has major impact on the unsteady conjugate heat transfer. Once this measurement is possible, this technique will be applied to measure the heat transfer for various pipe flows, such as separated flow, junction flow, pulsating flow, and others, which may result in the above mentioned problems.

2. Experimental setup

Fig. 1 shows the experimental setup. The water flow in the horizontal circular pipe was driven by the head difference. The inner diameter of the horizontal pipe was $D = 20.4$ mm and the length of the inlet region was $L = 1040$ mm ($L/D = 51$). A bell mouth was set upstream of the inlet region. The mass flow rate of water Q was measured by measuring the weight of the water overflowing from the inner outlet tank with a precision balance. The mean velocity in the pipe, $u_m = Q/(\rho\pi D^2/4)$, ranged from 0.05 to 1.3 m/s, resulting in Reynolds number Re_D ranging from 1000 to 39,000. The water in the outlet tank was heated by a heater with a thermostat up to about 30 °C, and was circulated by pumping the water from the outlet tank to the head tank until before data acquisition. The heating of water improves the accuracy of the infrared measurement because the temperature increase results in an increase in the emissive power from the test surface. The operation of the pump was stopped during data acquisition so as not be affected by vibration of the pump. During data acquisition, the water level in the header was controlled to be constant by adjusting the valve between the head tank and the rectifier tank.

The velocity distribution across the pipe at the test section (fabricated from glass) was measured using a laser Doppler velocimeter by setting a rectangular shaped water jacket with optical windows around the pipe (Shiibara et al., 2013a). The result showed that the flow was laminar following Hagen–Poiseuille flow at $Re_D \approx 1000$, and the flow became turbulent following logarithmic law at $Re_D > 8000$. Although the flow was unstable for $3000 < Re_D < 8000$, for which both the laminar and the turbulent states appeared irregularly in time, the flow became completely turbulent state when a turbulence promoter (a circular ring of $\phi 1$ wire) was attached at the inlet of the circular pipe.

The spatio-temporal fluctuation of the heat transfer was measured using a high-speed infrared thermography. Fig. 2 shows the test section for the heat transfer measurement. The circular duct in the test section was fabricated from an acrylic pipe of 280 mm in length, the middle section of which was cut out in a semicircular shape over a length of 220 mm. On the inner surface of the pipe, 22 µm-thick titanium foil was glued around the entire circumference, including the removed section. In order to increase the infrared emissivity of the test surface, the outer surface of the titanium foil was coated with black paint (approximately 20 µm in thickness). The titanium foil was heated electrically (electric current ~ 100 A) so that the temperature difference between the foil and the water flow was about 10 K at the measurement position ($z = 200$–240 mm). Here, z is the streamwise distance from the starting position of heating.

Since the heat capacity of the test surface (titanium foil coated with black paint) was very low along the removed section of the acrylic pipe, the wall temperature fluctuated according to the turbulent heat transfer to the water flow. The instantaneous distribution and its fluctuation of the temperature on the test surface was measured using a high-speed infrared thermograph (SC4000, FLIR; IRT hereafter), which can obtain thermal-images of 420 frames per second with a full resolution of 320 × 256 pixels. In this experiment, the frame rate increased up to 800 Hz with a reduction of pixels to 320 × 128. The thermal images of 8192 frames were acquired in each run. The catalog value of the noise-equivalent temperature difference (NETD) of IRT used here was 0.018 K at the room temperature.

The spectral emissive power detected by IRT, U_{IRT}, can be expressed as follows.

$$U_{IRT} = \varepsilon_\lambda f(T_w) + (1 - \varepsilon_\lambda) f(T_{amb}) \quad (1)$$

Here, ε_λ is the spectral emissivity of the test surface at the spectral range of IRT ($\lambda = 3 - 5$ µm), $f(T)$ is the calibration function of IRT for a blackbody, T_w and T_{amb} are wall temperatures on the test surface and ambient, respectively. The right-hand-side terms of Eq. (1) represent the emissive power from the test surface and ambient, respectively. In order to make uniform the emissive power from surroundings, the test section was shielded by copper plates coated with black paint, as shown in Fig. 2(b).

The spectral emissivity, ε_λ, was estimated using the equivalent surface ($\delta_t = 22$ µm-thick titanium foil coated with