Study on the characteristics of flow resistance under the condition of low frequency flow fluctuation

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Abstract: The circulation of coolant in the cooling system of a nuclear reactor, under earthquake conditions, is a process of pulsating flow. When the fluctuating flow occurs in the loop, the fluid flow and heat transfer in the core are affected, making the study of flow fluctuation more important for practical applications.

Characteristics of flow resistance are investigated experimentally on a horizontal rectangular channel and a circular tube with a periodically fluctuating flow. The results are compared with the flow resistance characteristics of the steady state, showing that there is a relatively large difference between the results under a steady flow and a transient flow.

This article analyzes the influences of the flow fluctuation period and of the relative amplitude on the resistance characteristics of a circular tube flow. The results show that under the condition of flow fluctuation, a phase difference exists between flow rate and pressure drop. The results further show that the smaller the flow fluctuation period is, the larger the flow fluctuation relative amplitude will be, and the more influence the flow fluctuation will have on the fluid flow. Furthermore, an empirical correlation expressed as a function of the Reynolds number, the dimensionless frequency parameter, and the relative amplitude, is proposed for calculating a time-averaged friction coefficient.

Keywords: rectangular channel; circle tube; flow fluctuation; resistance characteristics

1 Introduction

Periodic fluctuations in the fluid flow create different characteristics of flow and heat transfer. In general, the flow fluctuations are divided into two categories: (1) pulsating flow, in which the periodically time-averaged velocity is non-zero, and (2) oscillating flow, in which the periodically time-averaged velocity is zero. Pulsating fluid flow occurs in many industrial products, such as ramjet, pump, fan (Kong Long[1]), and many others. In addition, the circulation of coolant in the cooling system of a nuclear reactor under earthquake conditions is also a process of pulsating flow and heat transfer.

Furthermore, the rectangular channel is widely used in aerospace, microelectronics cooling, compact heat systems as well as in other fields, for its advantages as regards compact structure, large heat transfer area, and design. The multitude of compact structures has resulted in a great demand for the development of efficient heat removal techniques, based on narrow rectangular channels, and which have broad application prospects.

Steady flow and heat transfer in narrow rectangular channels and in a circular tube have been studied by many investigators. B.X. Wang and X.F. Peng [2], X.F. Peng *et al.* [3] investigated the heat transfer in microchannels. W. Tong *et al.* [4], Mala and Li [5], Zheng Hui-fan *et al.* [6] and Jiang Jie *et al.* [7] studied characteristics of flow resistance in microtubes or microrectangular channels. It was found that the heat transfer and pressure drop in narrow rectangular channels were different from those in microchannels (Gopinath R. Warrier *et al.*[8] and Tai Wen-bo *et al.* [9]). However, a study on the pulsating flow does not exist to our knowledge. More important, a review of

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the public literature reveals that very limited data are currently available for modeling accurately the resistance characteristics of the flow in narrow rectangular channels under the condition of flow fluctuation.

In this paper, characteristics of flow resistance are investigated experimentally on a vertical rectangular channel with a periodically fluctuating flow. The test results are compared with those from a steady state. In addition, the influences of the flow fluctuation period, of the amplitude and of the average flow rate on the resistance characteristics of the vertical rectangular channel flow are analyzed in detail.

2 Experimental apparatus

2.1 Experimental set-up and test section

The experimental apparatus is illustrated in Fig. 1. The main components of the test apparatus are a manostat, an Electro Magnetic Flowmeter (EMF), a thermometer, a test section, a capacitive differential pressure transducer, and a pump with transducer.

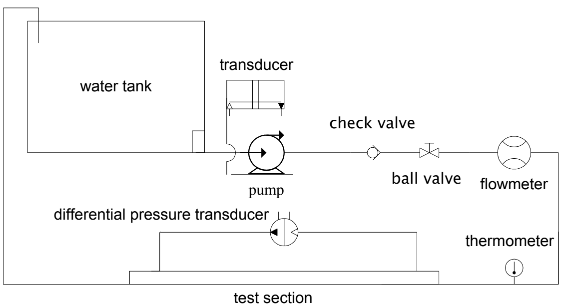

Fig. 1 Schematic of experimental apparatus.

The deionized water, pressurized by the pump, flows through the pipeline, and into the test section, after being throttled by the control valve. Subsequently, the water reflows through the pump. The pump controlled by the transducer runs in sine law with time, circulating the fluid in the pipe with periodical flow fluctuation. In addition, the signals of the flow, measured with the flowmeter, and the pressure drop, measured with the differential pressure transducer, are transmitted to the data acquisition system and recorded by the computer.

There are two test sections in the experiments. One is a narrow rectangular channel made of plexiglass, and the other is a galvanized circular tube, with an inside diameter (D) of 16.1**mm**, and a length of 6**m**.

The rectangular channel test section is illustrated in Fig. 2. Its length (L) is 2**m**. The width (b) of the rectangular channel is 40**mm**, and its height (a) is 3**mm**.

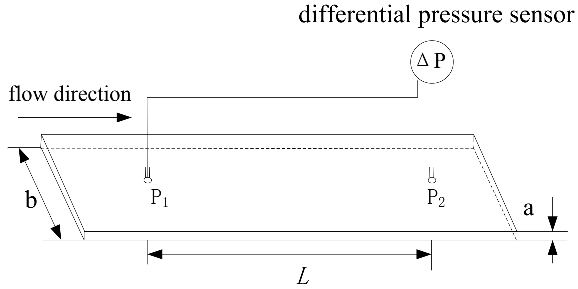

Fig .2 Schematic of test section.

The entrance lengths (L_e) of the test sections are shown in Table 1. Based on the correlation of Rohsenow *et al*. from 1973[10], the entrance lengths are long enough to ensure that a fully developed flow is achieved.

Table 1 Entrance lengths of test sections

Sections	Rectangular channel	Circular tube
L_e	400**mm**	800**mm**
D or D_h	5.6**mm**	16.1**mm**

2.2 Experimental conditions and uncertainty of the experimental apparatus

Experiments have been conducted over a range of fluctuation periods T and relative amplitudes A_r. They are shown in Table 2.

Table 2 Experimental conditions

Parameters	Value
T	7.5**s**~20**s**
A_r	0~0.8
Dimensionless frequency ω'	1.4~2.4
Water temperature	18~21℃
Experiment pressure	0.10~0.14**MPa**

The A_r can be set by changing the value of the amplitude of the velocity and of the time-averaged velocity. The T can be changed with the transducer that controls the pump. The water temperature will gradually increase due to the circulation. However, calculations showed that the effect of temperature change upon flow resistance could be ignored.

The official uncertainty of the EMF is ±0.5% within a range exceeding 2% of the maximum measurement flow rate. The official uncertainty of the differential pressure gage is ±0.2%, as given by the data logger.

2.3 Verification of the experimental system

To verify the correctness of the experimental apparatus, the characteristics of the flow resistance under a steady flow condition were first measured. The variation of the friction coefficient, for single-phase forced convection and as a function of the Reynold number, is shown in Fig. 3. The theoretical formulae listed in the textbook by Yu Ping-an *et al.* [11], Jing Si-rui and Zhang Ming-yuan [12] are also shown in Fig. 3. They are as follows:

$$\lambda = \frac{87}{Re_h} \quad \text{valid for } Re<2320,$$

$$\lambda = \frac{0.3164}{Re_h^{0.25}} \quad \text{valid for } 4000<Re<10000, \qquad (1)$$

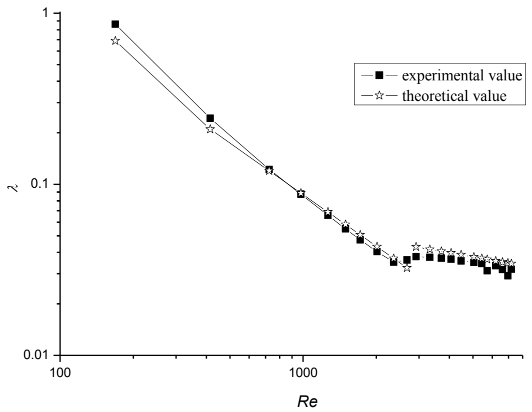

Fig. 3 Verification of experimental system with rectangular channel.

It can be seen from Fig. 3 that the measured values follow very closely the theoretical results, especially in the laminar flow region, implying that the experimental system is reliable and accurate for the purpose of conducting this research. Furthermore, the measured values under the steady flow condition are used as the reference in the following sections. The test friction coefficient for the circular tube is compared with the values obtained with the Blasius formula and the Laubatjef (Б.Н.Лобаев) formula. The test values are situated between the two empirical formula results, validating the reliability of conducting this research with this experimental system as shown in Fig.4.

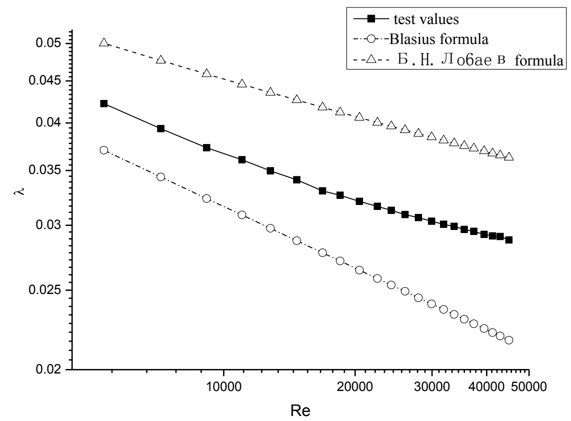

Fig. 4 Variation of experimental system with circle tube.

An illustrative group of the measured values, under the periodical flow fluctuation condition, is plotted in Fig. 5 (T=10s, A_v=1.302, B_v=0.624, A_r=0.48), showing that the pressure drop varies sinusoidally with time, in accordance with the set pulsation pattern of the flow. The delay attributed to the time constants of the flow meter and to the differential pressure transducer and inertia of the test fluid in the test loop was measured in a pre-experiment. This delay of about 1.3**s** has been deducted from the experimental results. However, there is still a phase lag of about 1.2**s** between the pressure drop and the velocity. This result confirms an earlier forecast for the pulsating flow obtained by Liu Ting-hao *et al.* [13]. Based on his theoretical analysis and experiments, a phase difference exists between the fluctuating curves of the velocity and the pressure drop. For a laminar flow, the phase difference depends only on the structure of the tube, the pulsating period, and the properties of the work medium. For a turbulent flow, the phase lag may depend on inertia and the acceleration process of the test fluid in the test section. However, further research is needed.

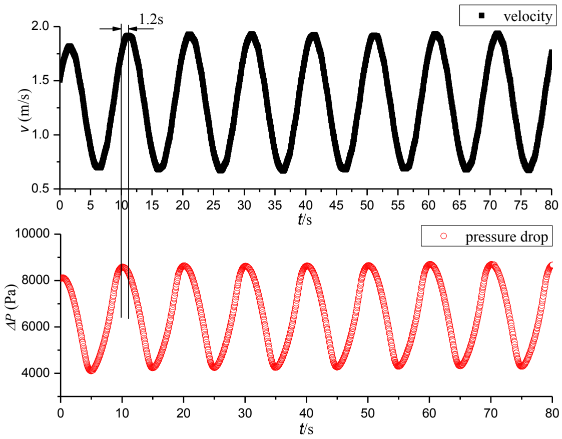

Fig. 5 Typical experimental phenomena.

3 Data deduction

It is assumed that the fluid taken as the research object has a gradually adiabatic varied flow. The equation of fluid energy can be expressed as:

$$\frac{\partial}{\partial t}\iiint_{CV}\rho\left(u+\frac{v^2}{2}+gz\right)dV+\iint_{CS}\rho v_n\left(u+\frac{v^2}{2}+gz+\frac{p}{\rho}\right)dA=0 \quad (2)$$

Assuming $v_n = v$ at outflow cross-section A_2 and $v_n = -v$ at inflow cross-section A_1, and z=0, we get the following equations, by integrating the first and second sections in Eq. (2):

$$\frac{\partial}{\partial t}\iiint_{CV}\rho\left(u+\frac{v^2}{2}\right)dV=\iiint_{CV}\rho\frac{\partial}{\partial t}\left(\frac{v^2}{2}\right)dV$$
$$=\iiint_{CV}\rho v\frac{\partial v}{\partial t}dV=\rho v\frac{\partial v}{\partial t}V \quad (3)$$

$$\iint_{CS}\rho v_n\left(u+\frac{v^2}{2}+\frac{p}{\rho}\right)dA=\iint_{A_2}\rho gv\left(\frac{u}{g}+\frac{v^2}{2g}+\frac{p}{\rho g}\right)dA \quad (4)$$
$$-\iint_{A_1}\rho gv\left(\frac{u}{g}+\frac{v^2}{2g}+\frac{p}{\rho g}\right)dA$$

And in Eq. (4):

$$\iint_A \rho gv\left(\frac{p}{\rho g}\right)dA=pq_v \quad (5)$$

$$\iint_A \rho gvzdA=\rho gLq_v \quad (6)$$

$$\iint_A \rho gv\frac{v^2}{2g}dA=\frac{1}{A}\iint_A\left(\frac{v}{v_a}\right)^3 dA\left(\rho gq_v\frac{v_a^2}{2g}\right)=a\left(\rho gq_v\frac{v_a^2}{2g}\right)$$

Here $$a=\frac{1}{A}\iint_A\left(\frac{v}{v_a}\right)^3 dA \quad (7)$$

$$\frac{1}{\rho gq_v}\left(\iint_{A_2}\rho gv\frac{u}{g}dA-\iint_{A_1}\rho gv\frac{u}{g}dA\right) \quad (8)$$
$$=\frac{1}{\rho gq_v}\int_{q_V}\rho(u_2-u_1)dq_V=h_w$$

By substituting Eq. (5), (6), (7), and (8), Eq. (4) can be expressed as:

$$\iint_{CS}\rho v_n\left(u+\frac{v^2}{2}+\frac{p}{\rho}\right)dA=$$
$$\rho gq_V\left[\left(\frac{a_2v_{2a}^2}{2g}+\frac{p_2}{\rho g}+h_w\right)-\left(\frac{a_1v_{1a}^2}{2g}+\frac{p_1}{\rho g}\right)\right] \quad (9)$$

By substituting Eq. (3) and Eq. (9), Eq. (2) can be expressed as:

$$\frac{\partial}{\partial t}\iiint_{CV}\rho\left(u+\frac{v^2}{2}\right)dV+\iint_{CS}\rho v_n\left(u+\frac{v^2}{2}+\frac{p}{\rho}\right)dA \quad (10)$$
$$=\rho v\frac{\partial v}{\partial t}V+\rho gq_V\left(\frac{p_2-p_1}{\rho g}+h_w\right)=0$$

Using the Darcy-Weisbach Formula, $h_w=\lambda\frac{L}{D_h}\frac{v^2}{2g}$, the friction coefficient can be expressed as:

$$\lambda=\frac{2d\left[(p_1-p_2)q_V-\rho v\frac{\partial v}{\partial t}V\right]}{\rho q_V Lv^2} \quad (11)$$

The flow fluctuation in the experiment can be expressed in the form of a sine wave as a function of time:

$$q=A\sin\frac{2\pi}{T}t+B;\qquad v=\frac{1}{ab}\left(A\sin\frac{2\pi}{T}t+B\right)$$
$$\frac{\partial v}{\partial t}=\frac{2\pi A}{abT}\cos\frac{2\pi}{T}t \quad (12)$$

The pressure difference measured in the experiment and shown in Fig. 2 can be expressed as follows:

$$\Delta p=P_1-P_2 \quad (13)$$

By substituting Eq. (12) and (13) into Eq. (11), the final formula of the friction coefficient is :

$$\lambda = \frac{2D_h\left[\Delta p - \rho L\dfrac{\partial v}{\partial t}\right]}{\rho L v^2} = \frac{2D_h\left[\Delta p - \rho L\dfrac{2A\pi}{abT}\cos\dfrac{2\pi}{T}t\right]}{\rho L v^2} \quad (14)$$

In the experimental procedure, the pressure difference, the temperature of the fluid, and the flow rate are recorded with a computer, and thus the instantaneous friction coefficient λ, under the condition of periodical flow fluctuation, can be obtained. The λ in the following part are all calculated with Eq. (14). It seems that the acceleration of the test fluid played an important effect on the friction factor.

4 Results and discussion

A total of two sets of experiments were performed to obtain the pressure drop characteristics for a single-phase forced convection in a small rectangular channel and a circular tube. The test conditions are shown in Table 2. The experimental results obtained are discussed in the following sections.

4.1 The transient friction coefficient and the steady flow friction coefficient

The processes determined by the friction factor and the Reynolds number in one period are plotted in Figs. 6 and 7. They imply that when the flow pulsates in the rectangular channel, one obtains a quasi-annular curve. For reference, the red line is the friction factor under a steady flow, in the same channel and with the same Reynolds number range. As it can be observed, the flow resistance behaves very differently under the condition of periodical flow fluctuation. This difference results from the change in the velocity distribution engendered by the acceleration of the test fluid. Indeed, it is common knowledge that the velocity distribution has a strong impact on the friction coefficient.

Furthermore, it can be observed that the quasi-annular curve in the laminar flow is moony, asymmetrical. But with a turbulent flow, it is a symmetrical ellipse. This shows that there will be two friction factor values for the same Reynolds number: one is larger, with a negative acceleration; the other is smaller, with a positive acceleration. The causes of the annular curves are the phase lag and the change in the velocity distribution in the present study. Furthermore, they both appear after the fluid flow in the accelerating state. This suggests that the acceleration of the fluid is the main cause of the annular curves. However, further work needs to be performed in order to explicit the details of how the acceleration of the fluid impacts the phase lag and the velocity distribution.

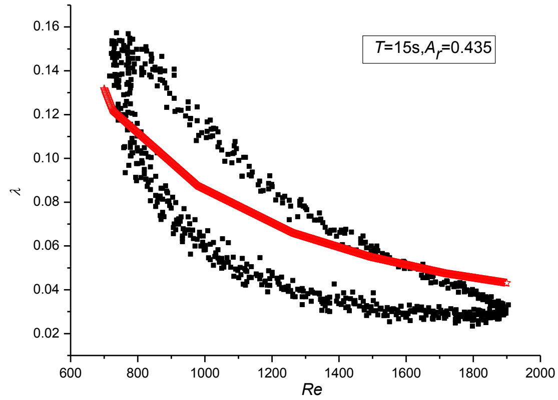

Fig. 6 Annular curve in the laminar flow.

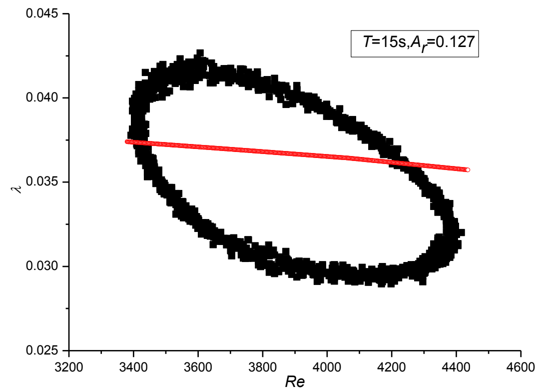

Fig. 7 Annular curve in the turbulent flow.

The friction coefficient and the Reynolds number obtained in a circular tube are plotted in Fig. 8. The quasi-annular curve is similar with that in a rectangular channel, which is due mainly to the small difference in the velocity distribution in different channels.

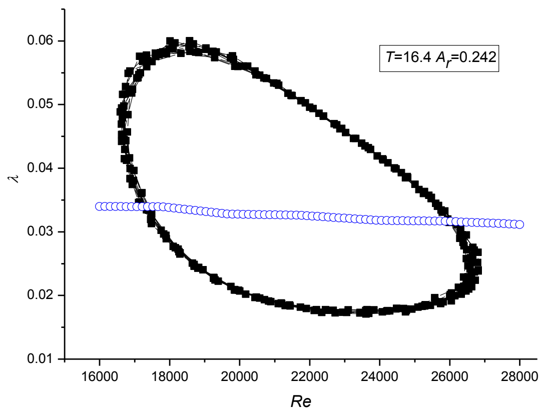

Fig. 8 The transient friction coefficient and the steady flow friction coefficient in a circular tube.

4.2 The time-averaged friction coefficient

The time-averaged friction coefficient under different flow fluctuation conditions is shown in Figs 9 and 10. It can be seen that the time-averaged friction coefficient increases gradually as the relative amplitude varies from 0 to 0.9, but it decreases at the same relative amplitude magnitude as the pulsating period grows larger. In addition, the time-averaged friction coefficient in a rectangular channel increases sharply when the A_r exceeds a value of approximately 0.1. There is a critical value of A_r in the characteristics of the change of the time-averaged friction coefficient. Furthermore, the critical relative amplitude value decreases as the period becomes smaller. The time-averaged friction coefficient in a circular tube has the analogical characteristic, as shown in Fig. 9.

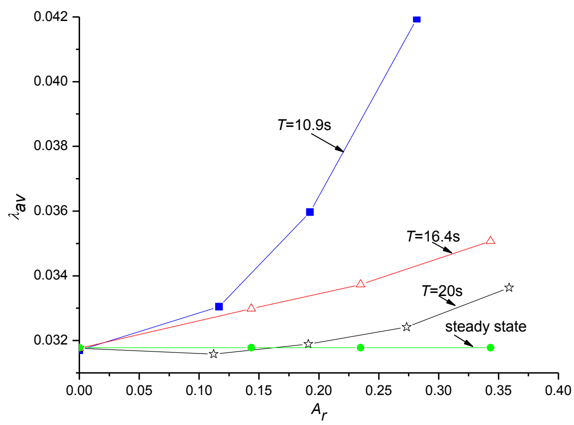

Fig. 9 Time-averaged friction coefficient under different cases in a circular tube.

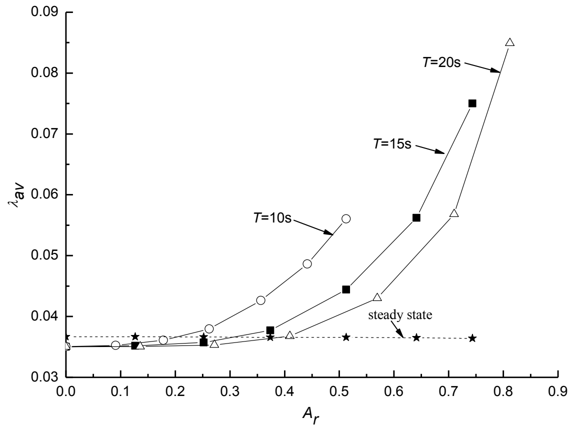

Fig. 10 Time-averaged friction coefficient under different cases in a rectangular channel.

This result confirms earlier forecasts for the pulsating flow obtained by Melda *et al* [14]. Introduced by Ohmi *et al* [15], the pulsatile flow can be divided into quasi-steady, intermediate, and inertia dominant regions, using the influence of the dimensionless frequency w' on the flow structure. Experiments showed that the relative amplitude may be another dimensionless parameter to classify the pulsatile flows. However, more studies and experiments are needed.

4.3 Fitting formula

Following the test data, a correlation as a function of the Reynolds number, the dimensionless frequency parameter, and the relative amplitude has been sought for the time-averaged friction coefficient determined by the periodical fluctuation. The correlation is expressed as:

$$\lambda_{av} = \lambda_{st} + Ce^{A_r/\tau} \tag{15}$$

Where,

$$C = -0.00211 + 0.00153\sqrt{w'},$$

$$\tau = -0.74221 + 0.99088\sqrt{w'} - 0.26819w'.$$

The λ_{st} is a steady component of the total friction coefficient. It is assumed that the λ_{st} for the pulsating transition to a pulsating turbulence flow remains under the form of the Blasius formula for a steady turbulent flow.

$$\lambda_{st} = 0.23Re^{-0.23} \tag{16}$$

Introduced by Ohmi *et al* [15], w' is the dimensionless frequency of flow fluctuation, which is used to describe the pulsating characteristic of the unsteady flow.

The performance of the correlation is shown in Fig. 11. Here the ring symbol stands for the data from a rectangular channel, while the pentagram stands for the data from a circular tube. The concordance is obtained with a mean absolute error of 10%.

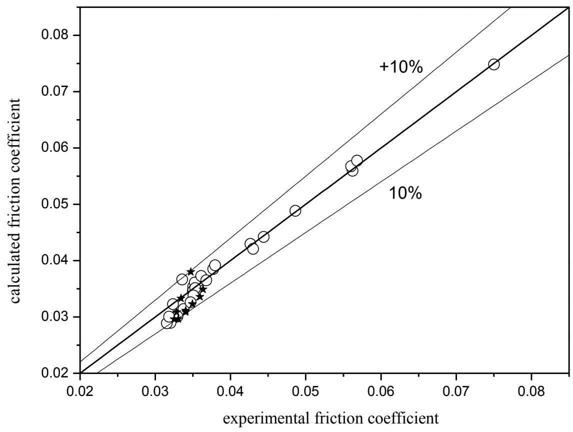

Fig. 11 Performance of the correlation for predicting the magnitude of the friction coefficient.

5 Conclusions

Characteristics of the flow resistance were investigated experimentally in a rectangular channel and in a circular tube with a periodically fluctuating flow. The results from the present experiments suggest that the flow rate lags behind the pressure drop due to the acceleration of the fluid in a pulsating flow, which can also be deduced from the theory. Compared with the friction coefficient in a steady flow, the instantaneous friction coefficient has two values for the same Reynolds number. The larger one occurs with a negative acceleration, and the smaller one with a positive acceleration.

In the present experiments, the time-averaged friction coefficient under a low relative amplitude and under the condition of a large fluctuation period is found to increase with an increasing relative amplitude, and to decrease with an increasing pulsating period. The experiments suggested that the magnitude of the time-averaged friction coefficient is a function of the relative amplitude, of the Reynolds number and of the dimensionless frequency, which have been fitted under the form of Eq. (15). Limited by the experimental context, this empirical formula was obtained with one aspect ratio and one circular tube. The verification for other aspect ratios will be accomplished through further work.

Nomenclature

A_v	Amplitude of the velocity.
B_v	Time-averaged velocity.
A_r	Relative amplitude, the ratio of A_v and B_v
D_h	Hydraulic diameter, $D_h = \dfrac{2ab}{a+b}$.
g	Acceleration of gravity.
T	Period, s.
t	Time, s.
ΔP	Pressure drop at the effective length of the test section, Pa.
Re	Instantaneous Reynolds number;
Re_h	Hydraulic diameter Reynolds number; $\dfrac{\rho v D_h}{\mu}$
Re_{av}	Time-averaged Reynolds number
L	The effective length of the test section.
q_V	Volume flow.
v	Velocity, m/s.
u	Internal Energy of fluid.
λ	Instantaneous friction coefficient.
λ_{av}	Time-averaged friction coefficient.
μ	Dynamic viscosity
ρ	Density
ω'	Dimensionless frequency, $\dfrac{\pi\rho D_h^2}{2T\mu}$.
$\sqrt{\omega'}$	Womersley number

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