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Heat transfer enhancement in sphere-packed pipes under high Reynolds number conditions

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ABSTRACT

Flow analysis in sphere-packed pipes (SPP) for different pipe to sphere diameter ratios was experimentally performed in order to clarify a relationship between the heat transfer and pressure drop characteristics. The experiments, using water as a working fluid, were carried out with $Re_D = 2000–33{,}000$ and $Pr = 5.0–6.0$. Experimental results of the pressure drop characteristics were compared with the Ergun's and Drag model correlations. Empirical correlations for the averaged Nusselt number are proposed, and SPP heat transfer performance is compared with that of the swirl flow. Furthermore, the applicability of the SPP system to the first wall cooling is also discussed from the temperature distribution aspect of the heating wall.

1. Introduction

In a recent design of the large helical device (LHD) type reactor, the "Force Free Helical Reactor" (FFHR) proposed at the national institute for fusion science [1], the first wall is expected to be exposed to a high heat load of almost 1.0 MW/m^2, which is removed by a high-temperature, molten salt "Flibe" flow. "Flibe" is a mixture of LiF and BeF_2 and has advantages for high heat capacity and the reduction of the MHD pressure drop due to its low electric conductivity. The Flibe blanket system, however, requires heat transfer enhancement under a high heat flux since Flibe is categorized as a high Prandtl number fluid. To overcome this issue, a sphere-packed pipe (SPP) has been proposed as a heat transfer enhancement technique for the high Prandtl number fluid.

The SPP is a passive heat transfer promoter whose matrix is composed of a number of metal spheres. The fluid is mixed well in the process of passing through the complicated flow channels, which leads to enhanced heat transfer. In addition, there is a heat transfer contribution from the heat conduction from the heating wall to each sphere, which is called the "fin effect". In addition, neutron multiplication can be expected by using Be for the SPP spheres. On the other hand, its complicated packing structure causes a relatively large pressure drop, which means a compromise must be made between the heat transfer enhancement and pumping power increase in order to optimize the design.

Varahasamy and Fand [2] investigated the heat transfer characteristics of SPP flows for water under a wide range of pipe to sphere diameter ratios. Okumura [3] clarified the flow structure of $D/d = 2.0$ by using a PIV system with a refractive index matching technique. The heat transfer performance of the high Prandtl number in SPP has been evaluated by using heat transfer salt (HTS) as the Flibe stimulant in a Tohoku-NIFS Thermofluid (TNT) loop at Tohoku University [4,5].

Although several papers have been published relating to forced-convection heat transfer in the SPPs, most studies [2–5] have been performed under relatively low Reynolds number conditions for the turbulent flow. Therefore, the objective of this study is to evaluate the pressure drop and the heat transfer characteristics of the SPP under high Reynolds number conditions for different diameter ratios of the pipe to the sphere. Moreover, empirical correlations are developed to predict the heat transfer performance in Flibe flows.

2. Experimental apparatus

Fig. 1 shows the schematic diagram of the experimental loop, where water is used as the working fluid. The fluid is introduced into the vertical test section and cooled by main and sub heat exchangers. A volume flow rate up to 70 [l/min] can be measured by the turbine flowmeter located at the top of the loop.

Details of the test section are shown in Fig. 2. The test section is made of stainless steel (SUS) pipe with an inner diameter D of 56 mm and thickness of 0.5 mm. Acrylic spheres are packed into the pipe, whose length L_p is 670 mm. Diameters of the spheres are 18.5, 25.9, 27.6 and 42.7 mm, which means the pipe to sphere diameter

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Nomenclature

C_P	specific heat at constant pressure
d	particle diameter packed into a pipe
D	pipe diameter
f_w	wall-modified friction factor $(1/M)(\Delta P/L_p)(d/\rho u^2)\{\varepsilon^3/(1-\varepsilon)\}$
f	a friction factor
H	axial 180° twist pitch
h	heat transfer coefficient
L_p	length of packed area
L_h	length of heated area
M	wall correction factor $1+2d/(3D(1-\varepsilon))$
Nu	Nusselt number hD/λ_f
ΔP	Pressure drop along the test section
Pr	Prandtl number $\mu C_P/\lambda_f$
p	Pumping power
Pr_{ef}	Prandtl number based on effective thermal conductivity $\mu C_P/\lambda_{ef}$
q	heat flux
Re_w	wall-modified Reynolds number $Re_d/\{M(1-\varepsilon)\}$
Re_d	Reynolds number based on a particle diameter ud/ν
Re_d	Reynolds number based on a pipe diameter uD/ν
T_b	bulk mixing temperature defined in Eq. (1)
T_{in}	inlet temperature of the working fluid
T_{out}	outlet temperature of the working fluid
T_w	inner wall temperature
u	inlet axial velocity
x	distance from the inlet

Greek symbols

δ	thickness of twisted tape of a swirl pipe
ε	porosity
γ	twisted ratio H/D
λ_{ef}	effective thermal conductivity
λ_f	thermal conductivity of fluid
μ	viscosity of fluid
ν	dynamic viscosity of fluid μ/ρ
ρ	density of fluid

ratios D/d are 3.0, 2.2, 2.0 and 1.3, respectively. The porosity ε of the SPP is estimated using the following equations [6].

$$\varepsilon = \frac{0.151}{(D/d)-1} + 0.360 \text{ for } \frac{D}{d} \geq 2.033 \tag{1}$$

$$\varepsilon = -0.6649\left(\frac{D}{d}\right) + 1.8578 \text{ for } 1.866 \leq \frac{D}{d} < 2.033 \tag{2}$$

$$\varepsilon = 1 - \frac{2/3(d/D)^3}{\sqrt{(2d/D)-1}} \text{ for } 1 \leq \frac{D}{d} < 1.866 \tag{3}$$

The calculated porosities of the SPPs are shown in Table 1. In each experiment, spheres of the same diameter are packed in the pipe. A part of the packed area, whose length L_h is 600 mm, is uniformly heated via Joule heating, and the heat flux q can be can get as high as 47 kW/m^2. Two- and three-sheathed K-type thermocouples are inserted into the inlet and outlet of the flow channel, respectively. The bulk mixing temperature T_b is estimated using linear interpolation from the inlet temperature T_{in} and the outlet temperature T_{out}, which is given by

Table 1
Porosities of the SPPs

D/d	3	2.2	2	1.3
ε	0.4345	0.4899	0.5087	0.5921

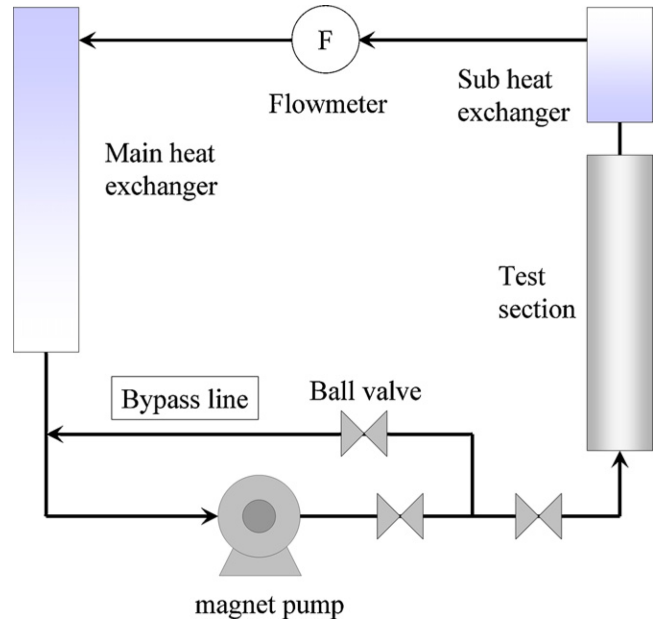

Fig. 1. Schematic diagram of overall loop.

$$T_b = T_{in} + (T_{out} - T_{in})\frac{x}{L_h} \tag{4}$$

where x is the downstream position from the heating entrance. K-type thermocouples 100 μm in diameter are fixed on the outer surface of the pipe with kapton tape at eight axial positions along the direction of flow, and at three angular positions at intervals of 22.5 °C in each axial position as shown in Fig. 2. Thus, there are a total of twenty-four thermocouples attached to the pipe. The first axial position of the thermocouples is located 175 mm downstream from the entrance of the heating area. The distance between each axial position is 50 mm. The inner wall temperature T_{wi} is calculated at each location by solving a one-dimensional steady heat conduction equation. The local heat transfer coefficient h_i is estimated using the following equation.

$$h_i = \frac{q}{T_{wi} - T_{bi}} \tag{5}$$

An averaged heat transfer coefficient h is calculated by taking the average of all the local heat transfer coefficients. An experimental Nusselt number is calculated using the following equation.

$$Nu = \frac{hD}{\lambda_f} \tag{6}$$

The test section is insulated in order to minimize heat loss. The pressure loss along the packed area is measured by a differential pressure gauge, which ranges from 1 to100 kPa.

3. Results and discussion

3.1. Pressure drop characteristics

Figs. 3–6 show the pressure drop of the SPP for D/d = 3.0, 2.2, 2.0 and 1.3, respectively compared with the Ergun's and Drag model correlations. The Ergun's correlation is given by the following equation [7].

$$\frac{\Delta P}{L_p} = 150\frac{(1-\varepsilon)^2}{\varepsilon^3}\frac{\mu u}{d^2} + 1.75\frac{1-\varepsilon}{\varepsilon^3}\frac{\rho u^2}{d} \tag{7}$$