Heat transfer enhancement in mini-channel using a fibrous porous medium

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Abstract
With the recent miniaturisation of electronic devices, the heat generation density of the devices has increased tremendously, requiring the improvement of the performance of the cooling system. In this study, for the high performance of water cooling devices, we investigated convective heat transfer in a mini-channel where porous media that were manufactured by randomly laminated and sintered fine aluminium wires were inserted as a heat exchanger. We conducted the convective heat transfer experiment by changing the flow state from laminar to turbulent using pure water. We also conducted a numerical analysis using a simple lattice model of porous media with the same wire diameter and porosity as in the experiments by using the software PHOENICS. From both results, we discussed the effect of the structure of the porous media on heat transfer. From the results of the calculated permeability of the porous media using the experimental data of pressure-drop, we confirmed that the flows with the porous media were in a non-Darcy flow state. As heat transfer characteristics, it was shown that the Nu number of the experimental results was larger than that of the computational fluid dynamics (CFD) analysis models. Regarding the effects of wire diameter and porosity, the smaller the wire diameter and porosity, the higher the Nu number. By considering the differences between experimental and analytical results, we proposed two sets of new combination of parameters considering the volume of the fibre of porous media and the surface area, which separately correlated well with relationships of the experimental and analytical results against the Nu number.

Keywords: Heat transfer enhancement, Forced convection, Porous medium, Mini-channel, Permeability

1. Introduction

With the recent miniaturisation and performance improvement of electronic devices, the heat generation density of the devices has significantly increased, requiring the improvement of the performance of the cooling system. Generally, it is critical for heat exchange techniques to increase the effective heat transfer area for convective heat transfer, enabling a large amount of heat transport compared to heat conduction. Therefore, in some cases, fins of various shapes are attached to the heat exchange section of machines that require cooling (Kitajo, 2005; The Japan Society of Mechanical Engineers, 2009). As an effective means for increasing the surface area in promoting heat transfer, using a metal-porous medium having high thermal conductivity is already considered. Metallic-porous media have a variety of shapes, such as those generated from foam metal melts and fibrous solid metals that are randomly laminated. Metallic-porous media are renowned for their effect of heat transfer enhancement because of a large surface area, they can maintain good heat conduction inside the solid phase, and could promote fluid mixing inside (Oda et al., 2003). Against this background, various studies were conducted using porous media as heat transfer accelerators (Kunugi et al., 2004; Leong and Jin, 2006). However, most of the studies are related to the heat transfer of metallic-porous media by foaming metal melts (Haack et al., 2001; Calmidi and Mahajan, 2000).

Few studies exist on heat transfer using fibrous porous media. Ichimiya et al. (Ichimiya et al., 2006) conducted a heat transfer flow experiment with a medium with a porosity close to one by inserting a thin copper wire into the entire flow path and evaluated the heat transfer performance to confirm the effect of metal-porous media with high porosity. Komatsu
et al. (Komatsu et al., 2009) filled a rectangular tube with a heat sink of aluminium fibre and conducted heat transfer performance experiments with porosity ranging from 0.8 to 0.99, and investigated the effects of fibre diameter and porosity in the heat transfer performance quantitatively. However, because most of those studies used air as the working fluid, there are few reports on the details of the improvement of heat transfer because of the porous media when water is used as the working fluid.

In this study, a porous medium made by intricately laminating and sintering fine aluminium wires was installed in a rectangular channel after the medium was directly welded to an aluminium block for heating. The heat transfer flow experiment using pure water as working fluid was performed in the range from laminar flow to turbulent flow. In this experiment, the heat transfer coefficient and pressure-drop were measured, and the relationship between them was investigated by calculating the permeability and Nu number. Furthermore, to verify and consider the experimental results, numerical thermal-fluid analysis with computational fluid dynamics (CFD) software was also performed using a porous medium of a simplified lattice structure different from the experiment. From both results, the effect of the structure of the porous media on heat transfer was studied.

Nomenclature

- $A$: coefficient of first-order velocity term ($=\mu/K$)
- $B$: coefficient of second-order velocity term ($=b/p$)
- $b$: coefficient
- $c$: coefficient ($=b/\sqrt{K}$)
- $cp$ [J/kg·K]: specific heat at constant pressure of fluid
- $Da_w$: the inverse Darcy number
- $d$ [m]: equivalent hydraulic diameter of flow path
- $dT/dx$ [K/m]: temperature gradient in the aluminium brock
- $d_w$ [m]: wire diameter of porous medium
- $f$: friction factor
- $h$ [W/m²·K]: heat transfer coefficient
- $K$ [m²]: permeability
- $k$ [W/m·K]: thermal conductivity of aluminium
- $ko$ [W/m·K]: thermal conductivity of porous medium
- in stagnant fluid
- $L$ [m]: length of porous medium
- $M$ [kg]: mass of porous medium
- $Nu$: Nusselt number based on $k_0$
- $Pr_c$: Prandtl number based on $k_0$
- $q$ [W/m²]: heat flux
- $Re$: Reynolds number based on channel diameter
- $Re_k$: Reynolds number based on permeability
- $S$ [m²]: total surface area of porous medium
- $T_i$ [°C]: inlet temperature of working fluid
- $T_w$ [°C]: heated surface temperature
- $V$ [m/s]: average flow velocity of working fluid
- $\Delta P$ [Pa]: pressure-drop
- $\mu$ [Pa·s]: viscosity of pure water
- $\nu$ [m²/s]: kinematic viscosity of pure water
- $\rho$ [kg/m³]: density of pure water
- $\rho_s$ [kg/m³]: density of structure body of porous medium

2. Porous media

In this study, we used porous media made by Mitsubishi Materials Corporation, which were manufactured using laminated and sintered fine aluminium wires. These fine aluminium wires were laminated complicately and the size of the pores in the porous medium was randomly configured. In this study, we experimented to investigate the effect of wire diameter and porosity on heat using four types of porous media, with wire diameters of 200 μm and 500 μm, and porosity of 75% and 85%. Hereinafter, we refer to the porosity of 75% and 85% as 200-75 and 200-85 when the wire diameter of the porous medium is 200 μm, and the porosity of 75% and 85% are referred to as 500-75, 500-85 when the wire diameter of the porous medium is 500 μm. Figures 1(a)-(d) shows each porous medium. Furthermore, Eq. (1) expresses the surface area of each porous medium $S$ [m²] by using the density of the structure body of the porous medium $\rho_s$ [kg / m³], the wire diameter of the porous medium $d_w$ [m], and the mass of the porous medium $M$[kg]. Table 1 shows the density and total surface area of the porous media.

$$S = \frac{M}{\rho_s} \times \frac{4}{\pi d_w} \times \pi d_w = \frac{4M}{d_w \rho_s}$$ (1)
3. Experiments
3.1 Experimental setup and method

Figure 2 shows the overall experimental setup used in this study. In the overall diagram of the experimental apparatus in Fig. 2, pure water was circulated by a thermostatic chamber (LAUDA RE206, manufactured by Ernst Hansen Co., Ltd.) and supplied into the test section. After passing through the test section, it was circulated again to the thermostatic chamber. A silicone hose connected the thermostatic chamber and the test section. Figure 3 shows a schematic view of the test section. As shown in Fig. 3 (a), a differential pressure gauge (GC50 manufactured by Nagano Keiki Co., Ltd.) measured the pressure-drop at 135 mm intervals, centred on the porous medium in the flow direction through a copper pipe. At the same time, as shown in Fig. 3 (b), thermocouples for measuring the temperature of three points in the aluminium block were installed using a connector for fixing the thermocouples, and the inlet temperature of the working fluid at the test section was measured. Figure 4 shows a detailed view of where the porous medium was set, which is indicated by the red frame in Fig. 3 (b). The porous medium has a cubic shape with a side of 10 mm, and one surface is welded to the aluminium block. The porous medium was inserted and fixed in an experimental channel having a square cross-section of 10 mm. A cartridge heater (E2A83, diameter 6.25 mm × height 50.8 mm, manufactured by Sakaguchi Electric Heat Co., Ltd.) was attached to the aluminium block. The experimental channel was made of heat-resistant resin POM (polyacetal resin; thermal conductivity 0.25 W/m-K) to minimise heat dissipation to the outside. As shown in Fig. 4, the sample of porous medium in this experiment was a small cube with one side of 10 mm, and the flow channel into which it was inserted, also had a small cross section of 10 mm × 10 mm. This is because the purpose of this research is to investigate cooling techniques for small heat-generating semiconductors such as CPUs. However, with the size of this sample, the flow could be influenced by the channel walls to some extent. Many of the previous studies cited in our later discussions adopted porous media having larger size than our samples. Thus, we would like to note the possibility that our results would include that kind of influence.

As an experimental method, a working fluid (pure water) maintained at 30 °C was provided from the thermostatic chamber into the test section, and 6 W was applied to the cartridge heater inserted in the aluminium block by a DC stabilised power supply. We measured the temperature of three points, installed at 5 mm intervals from a position 6.5 mm away from the heated surface of the porous medium in the aluminium block, the inlet temperature of the working fluid and the pressure difference before and after passing through the porous medium for 1 minute in the steady-state. The average value was calculated using a value of data logger (GL240, manufactured by GRAPHTEC). The temperature at three points in the aluminium block and the inlet temperature of the working fluid was measured with a K-type thermocouple (T-35 type, diameter 0.5 mm and diameter 1.0 mm, manufactured by Sakaguchi Electric Heat Co., Ltd.). The experiment was conducted by changing the Re number expressed in Eq. (2) using the equivalent hydraulic diameter

Table 1 The specifications of porous media used in this study.

| Samples  | Density [kg/m³] | Total surface area [m²] |
|----------|-----------------|-------------------------|
| 200-75   | 657             | 6.8 × 10⁻³              |
| 200-85   | 392             | 4.1 × 10⁻³              |
| 500-75   | 564             | 2.3 × 10⁻³              |
| 500-85   | 379             | 1.6 × 10⁻³              |

Fig. 1 The porous media used in this study (provided by the Mitsubishi material corporation). (a) Wire diameter 200 μm, porosity 75% (b) Wire diameter 200 μm, porosity 85% (c) Wire diameter 500 μm, porosity 75% (d) Wire diameter 500 μm, porosity 85%
of the rectangular channel as the representative length to approximately 700, 1200, 2200 and 4000. This operation was conducted with four types of porous media (200-75, 200-85, 500-75 and 500-85). Thus, we investigated the pressure-drop and the heat transfer characteristics of the intricately laminated aluminium fibrous porous media used in this study.

\[
Re = \frac{V d}{v}
\]

(2)

where, \(V \text{[m/s]}\) is the average flow velocity of the working fluid, \(d \text{[m]}\) is the equivalent hydraulic diameter of the flow path, and \(v \text{[m}^2\text{/s]}\) is the kinematic viscosity of pure water.

### 3.2 Evaluation method

As one of the evaluations of the heat transfer characteristics of the porous media, we obtained the temperature gradient from the measured temperatures of the three points in the aluminium block, and the heat flux \(q\) was obtained from Fourier’s law, shown in Eq. (3). The heated surface temperature \(T_W\) was extrapolated from the temperature gradient in the aluminium block, and the heat transfer coefficient \(h\) was calculated from Eq. (4) using the heat flux and the inlet temperature of the working fluid.

\[
q = -k \frac{dT}{dx}
\]

(3)

\[
h = \frac{q}{T_W - T_L}
\]

(4)

where, \(q \text{[W/m}^2\text{]}\) is the heat flux, \(k \text{[W/m} \cdot \text{K]}\) is the thermal conductivity of aluminium, \(dT / dx \text{[K/m]}\) is the temperature gradient in the aluminium block, \(h \text{[W/m}^2\cdot \text{K]}\) is the heat transfer coefficient, \(T_W \text{[°C]}\) is the temperature of the heated surface, and \(T_L \text{[°C]}\) is the inlet temperature of the working fluid.

![Fig. 2 Schematic view of the experimental set up](image)
Fig. 3 Experimental set up of the test section. (a) Top view (b) Front view

Fig. 4 Schematic view of the porous medium section. (a) The overall of the porous medium section (b) The detail of the porous medium section.

4. Thermal-fluid analysis by computational fluid dynamics
4.1 Outline of thermal-fluid analysis

To verify the experimental results, we conducted a thermal-fluid analysis by CFD software using the finite volume method. Such complicated and irregular parts could have some effects on the pressure-drop characteristics and heat transfer characteristics because of the complicated structure of the porous media used in the experiment, as shown in Fig. 1. However, it is challenging to represent a complicated and random structure as is shown in Fig. 1 with a numerical model. There is also a method of reading the structure from the sample in Fig. 1 by CT scan and importing it into numerical calculation; however, this is extremely time-consuming, and the calculation results must be evaluated carefully. Therefore, here, we adopted the shape of a porous medium based on a simplified lattice model, as shown in Figs. 5 (a)–(d) by comparing it with the experimental results of porous media with complicated structures, and decided to consider
the influence of the irregularity of the lattice shape.

As a simplification method of the porous medium, a beam with a circular cross-section was constructed in only three directions in a cubic space of a certain length (for example, three coordinate axes in a Cartesian coordinate system). After the length of a side of the cubic space was decided from the porosity inside the cubic space and the cross-sectional diameter of the beam, the porous medium shape was constructed by stacking the cubic space with the decided length of a side in each direction of the Cartesian coordinate system. We used four types of simplified lattice models with the same wire diameter and porosity as those used in the experiment. The shape was a three-dimensional model, and the dimensions of each part were made as close as possible to the test channel used in the experiment. The calculation was performed by steady calculation, assuming that the flow is laminar in all cases without using turbulence models.

4.2 Outline of numerical model

For the CFD analysis, general-purpose thermal-fluid analysis software PHOENICS was used. PHOENICS adopts the structured grid method for generating the calculation grid, and a calculation area of \( x = 30 \text{ mm}, y = 5.6 \text{ mm}, \) and \( z = 12.4 \text{ mm} \) on the Cartesian coordinate system was set up at first, and a simulation model was built in.

Figure 6 shows a schematic diagram of the simulation model. As shown in the schematic diagram of the flow channel used in the experiment in Fig. 4, the shape of the porous medium was 10 mm wide \( \times \) 10 mm high. We used a model of a 8 mm \( \times \) 4.5 mm heater plate to make the heat input by the heater as similar as possible to the temperature distribution in the experiment, generated and reproduced 6 W. By using this basic form, we produced four types of analysis models, as shown in Fig. 7. These simulation models are constructed using a structured grid model and, as shown in Fig. 8, all the models have a total of 6,365,400 cells, with \( x = 300 \text{ cells}, y = 103 \text{ cells}, \) and \( z = 206 \text{ cells} \). In particular, the region where the porous medium is arranged is divided using a cubic calculation grid of a 50 \( \mu \text{m} \) width and a total of 4,000,000 cells, with \( x = 200 \text{ cells}, y = 100 \text{ cells}, \) and \( z = 200 \text{ cells} \). Therefore, even in the wire portion with a wire diameter of 200 \( \mu \text{m} \), calculation accuracy was ensured because approximately 3–4 calculation cells are included in the wire width direction. The boundary between the porous medium and the fluid is set by the cut-cell method, which can deal with the complex path shapes of the flow inside the porous medium.

We used an aluminium alloy (A1011) as the thermophysical property value of the porous media, which is the same as the material of the porous media for the experiments. Table 2 shows the physical properties of this condition. As the working fluid, the value of pure water at 303.15 K, which is the temperature at the time of the experiment, was used, and Table 3 shows its physical properties. In the simulation, the flow in the channel was assumed as plane-symmetric at the centre of the porous medium when viewed from a plane perpendicular to the flow direction, and a plane-symmetric model was formed at the surface of the centre of the porous medium to reduce the calculation load. This is sufficiently reasonable regarding hydrodynamics considering the boundary conditions.

Fig. 5 The simplified porous media structures used in simulation. (a) Wire diameter 200 \( \mu \text{m} \), porosity 75% (b) Wire diameter 200 \( \mu \text{m} \), porosity 85% (c) Wire diameter 500 \( \mu \text{m} \), porosity 75% (d) Wire diameter 500 \( \mu \text{m} \), porosity 85%
Fig. 6 Schematic view of the flow path of the numerical model. (a) $x$–$z$ coordinate (b) $y$–$z$ coordinate

Fig. 7 The numerical model. (a) 200-75 (b) 200-85 (c) 500-75 (d) 500-85
Fig. 8 Meshes of the numerical model. (a) x–z coordinate (b) x–y coordinate (c) y–z coordinate

Table 2 Physical properties of aluminium in the thermal-fluid analysis model

| Property                  | Aluminium |
|---------------------------|-----------|
| Specific heat [J/kg·K]    | 896.0     |
| Thermal conductivity [W/m·K] | 204.0     |

Table 3 Physical properties of pure water in the thermal-fluid analysis model

| Property                  | Pure water |
|---------------------------|------------|
| Specific heat [J/kg·K]    | 4180       |
| Thermal conductivity [W/m·K] | 0.6150     |
| Density [kg/m³]           | 995.6      |
| Viscosity [μPa·s]         | 797.4      |

4.3 Boundary condition and computational condition

Regarding the four types of calculation models shown in Fig. 7, calculations were performed at a flow velocity of the fluid in the range of 0.0550 to 0.325 m/s. In the experimental apparatus, the length from the channel inlet to the test section was considered to ensure enough length for the inlet region until the flow fully develops. To reflect this in the simulation as well, we performed a flow analysis in the plane-symmetric of the x–z plane CFD calculation model with the same calculation grid shape and grid number on the y–z plane, as shown in Fig. 8, for a rectangular tube with a cross-section of 10 mm × 10 mm at each flow velocity (0.0550 to 0.325 m/s) in advance. Only the flow distribution was solved. The calculation area was x = 2000 mm, y = 5.6 mm, and z = 12.4 mm.

Then, the fully developed flow at the exit boundary was applied to the entrance flow boundary of the thermo-fluid analysis model. From the calculation results of the simulation, the evaluation was made based on the average value of the temperature of the heated surface (the end surface of the aluminium block on the porous medium side). We evaluated the pressure-drop of the working fluid before and after passing through the porous medium from the difference of the average value in the cross-section of the flow channel at 0.5 mm and 29.5 mm downstream from the inflow boundary in the flow direction. As shown in Fig. 3 (a), the pressure measurement was conducted at two measurement ports at the upstream side 67.5 mm and the downstream side 67.5 mm from the centre of the porous medium. However, in the
numerical model, setting at this position was challenging because of the large model size. Therefore, considering that the pressure-drop in the flow channel occurs predominantly in the porous medium and the loss outside the medium is small, in the numerical calculation, the position of collecting pressure-drop data was determined as described above.

5. Results and discussion

Figure 9 shows the relationship between the pressure gradient and the flow velocity. The vertical axis was expressed by the pressure gradient obtained by dividing the pressure-drop obtained from the experimental and analytical results by the length of the porous medium. The experimental and CFD results confirmed that the pressure-drop because of these porous media are not the Darcy flow that is linearly proportional to the flow velocity, but the non-Darcy flow that rises quadratically.

Under conditions where the Reynolds number \(Re_K\) (see Eq. (5)) inside the porous medium exceeds 1, the flow inside the porous medium becomes a non-Darcy flow, and flow resistance (Forchheimer resistance) proportional to the square of the velocity could occur (The Japan Society of Mechanical Engineers, 2009).

\[
Re_K = \frac{V \sqrt{K}}{\nu}
\]  

(5)

where, \(V\) [m/s] is the average flow velocity of the working fluid, \(K\) [m\(^2\)] is the permeability, and \(\nu\) [m\(^2\)/s] is the kinematic viscosity of pure water.

The permeability \(K\) represents the cross-sectional area of the vacant gap in the porous medium and the degree of ease of fluid flow inside the porous medium. Considering this, we decided to find the value of permeability \(K\) from the plot in Fig. 9, which shows the relationship between the gradient of the pressure-drop and flow velocity obtained from experiments and numerical calculations. It will be compared with the findings obtained from previous studies. The characteristics of the pressure gradient \(\Delta P/L\) [Pa/m] because of the porous medium is expressed by Eq. (6), known as Forchheimer’s equation (Beavers and Sparrow, 1969).
The permeability of the porous media used in the experiment was approximately 1/8 to 1/4 that of the simple lattice model. The porous media used in the experiment had a complicated structure, and irregular enlargement and reduction of the pore sizes existed. Moreover, the wires existed immediately after the fluid passed through the pores. On the other hand, in the simple lattice model, the shape and spacing of the pores between the structures are regular. Because of these structural differences, in the porous media used in the experiment, the fluid passed through a more complicated path than in the simple lattice model because the fluid immediately collided with another wire after passing through a pore.

Furthermore, when \( R_{ek} \) in Eq. (5) was calculated using the permeability \( K \) based on the experimental values in Table 4, it was in the range of 3.3 to 46 under the present experimental conditions. Similarly, when \( R_{ek} \) was calculated using the permeability \( K \) based on the analysis results of the simple lattice model in Table 4, it was in the range of 6.7 to 116. The results show that \( R_{ek} \) exceeded 1, and it was reasonable to assume a non-Darcy flow.

Next, based on the obtained pressure-drop \( \Delta P \), the characteristics of the porous media of this research will be discussed based on the findings of previous research (Beavers and Sparrow, 1969). Using the calculated permeability \( K \), Eq. (8) defines the friction factor \( f \) (Beavers and Sparrow, 1969).

\[
f = \frac{(-\Delta P/L)\sqrt{K}}{\rho V^2} \tag{8}
\]

By transforming Eq. (6) so that the left side is \( f \), we obtain Eq. (9). Figure 10 shows the relationship between the friction factor \( f \) in Eq. (9) and the Reynolds number \( R_{ek} \) using the permeability \( K \) defined in Eq. (5) for experimental values and numerical calculation results.

\[
f = \frac{1}{R_{ek}} + c \tag{9}
\]

The transformation from Eq. (6) to Eq. (9) shows that \( c \) is a constant represented by \( c = b\sqrt{K} \). Table 4 also shows the \( c \)-value calculated from the values of \( B (=b\rho) \) and \( K \) obtained so far.

As can be seen from Eq. (9), when the Reynolds number \( R_{ek} \) increases, the value of the friction factor \( f \) converges to a certain value of \( c \). Regarding this convergence value, Beavers and Sparrow (1969) investigated many porous media and found that data of porous foam media without free fibre ends made of nickel converge to an equation with a \( c \)-value of 0.074, as in Eq. (9). Paek et al. (2000) reported that the \( c \)-value of aluminium foam with a porosity of 89% to 96% converged to 0.105. Figure 10 shows that the data of the porous media used in the experiment converged on a curve that closely resembles each of the four types of porous media. Also, from Table 4, the \( c \)-value was in the range of 0.018 to 0.048, and it can be characterised by having a lower \( c \)-value than that of the porous media reported by Beavers and Sparrow (1969). It is known that the \( c \)-value changes depending on the structure of the porous medium and the porosity. Jiang et al. (2004) reported that the \( c \)-value was 0.45 for glass beads and steel particles with a porosity of 39% with a
particle diameter of 1.2 mm to 2.5 mm. On the other hand, Fig. 10 shows that the four types of porous media used in the CFD analysis were not well-coordinated, and in this $Re_K$ range, sufficient convergence is not yet seen. From Table 4, the $c$-value shows an equivalent or a relatively higher value than 0.074 for the case of 500-75. According to Hwang et al. (2002), in their flow experiment using an aluminium foam, a large amount of vortex flow was observed behind the downstream side of the porous medium with a porosity of 70%. It is reported that the $c$-value discussed here becomes larger such as 0.11. They also reported that, on the other hand, in the porous media with porosities of 80 and 90%, almost no vortices were found behind the porous media, and the $c$-values were as small as 0.068 and 0.065, respectively. According to the CFD analysis carried out by the authors, although detailed flow graphics are not shown here, in the case of 500-75, undulating flows and vortex flows were observed behind the porous medium. As reported by Hwang et al. (2002), it is considered that these flows caused the high $c$ value.

Considering from the report by Hunt and Tien (1988) that the lower the porosity of the porous medium, the higher the $c$-value, the author’s four CFD analysis results in Fig. 10 can be reasonably compared with the four experimental results. Although the $c$-values of the CFD analysis results were larger and more varied than the experimental results, it was qualitatively consistent with the findings of the past studies.

Thus, from the viewpoint of the friction factor and the $c$-value, the simplified porous media used in the CFD analysis belonged to different types than that of the porous media used in the experiment.

From Eq. (8), this friction factor $f$ originally means the ratio of the contribution to the pressure-drop is because of the static pressure $ΔP / L$, and the contribution to the pressure-drop because of dynamic pressure $ρV^2 / \sqrt{K}$. It can be understood that the smaller the friction factor $f$, the larger the contribution of the dynamic pressure in the porous medium, inferring that the heat dispersion effect by the flow is higher. From this point of view, since the friction factors $f$ of the porous media used in this experiment are smaller than those reported by Beavers and Sparrow (1969), their heat dispersion effect could be relatively strong.

| Table 4 Coefficients of A and B of Equation (7), and $K$, $c$ |
|-----------------|-----------------|-----------------|-----------------|
| Sample          | $A (= \mu / K)$ | $B (= b\rho)$  | $K [\times 10^9 \text{ m}^2]$ | $c (= b\sqrt{K})$ |
| 200-75 Experiment | $3.45 \times 10^5$ | $1.00 \times 10^6$ | 2.31            | 0.048             |
| 200-85 Experiment | $1.94 \times 10^5$ | $6.54 \times 10^5$ | 4.10            | 0.042             |
| 500-75 Experiment | $1.40 \times 10^5$ | $2.51 \times 10^5$ | 5.70            | 0.019             |
| 500-85 Experiment | $6.87 \times 10^4$ | $1.68 \times 10^5$ | 11.6            | 0.018             |
| 200-75 Analysis | $8.28 \times 10^4$ | $1.00 \times 10^6$ | 9.64            | 0.099             |
| 200-85 Analysis | $2.68 \times 10^4$ | $5.34 \times 10^5$ | 29.8            | 0.093             |
| 500-75 Analysis | $1.75 \times 10^4$ | $7.46 \times 10^5$ | 45.4            | 0.16              |
| 500-85 Analysis | $9.84 \times 10^3$ | $2.11 \times 10^5$ | 81.0            | 0.060             |
Figure 11 shows the relationship between the heat transfer coefficient calculated using Eq. (4) as an evaluation of the heat transfer characteristics and the flow velocity. From Fig. 11, it was confirmed that the heat transfer coefficient tended to increase as the flow velocity increased for all porous media. Rough rankings of the heat transfer coefficients of the porous media used in the experiment were highest with 200-75 at $9.89 \times 10^3 - 1.96 \times 10^4$ [W/m²·K], then with 500-75 at $8.31 \times 10^3 - 2.31 \times 10^4$ [W/m²·K], and with 200-85 at $7.09 \times 10^3 - 1.71 \times 10^4$ [W/m²·K], and with 500-85 at $6.99 \times 10^3 - 1.82 \times 10^4$ [W/m²·K]. The same rough rankings of the heat transfer coefficients of the simple lattice model used in the CFD analysis were the highest with 200-75 at $8.82 \times 10^3 - 1.76 \times 10^4$ [W/m²·K], then with 500-75 at $6.45 \times 10^3 - 1.43 \times 10^4$ [W/m²·K], and with 200-85 at $5.94 \times 10^3 - 1.33 \times 10^4$ [W/m²·K], and with 500-85 at $3.92 \times 10^3 - 8.57 \times 10^3$ [W/m²·K]. From this, the heat transfer coefficient of the porous media used in the experiment tended to be higher than that of the simple lattice model used in the CFD analysis.

According to a previous study (Hunt and Tien, 1988), Fig. 12 shows the heat transfer characteristics of the porous media by Eqs. (10) and (11). Here, the $Nu$ number is based on the thermal conductivity ($k_0$) when the fluid is stagnant inside the porous medium, we adopted the product of $Re_\kappa \cdot Pr_e$ obtained by the average Reynolds number $Re_\kappa$ based on the permeability $K$, and the Prandtl number $Pr_e$ based on the stagnant thermal conductivity $k_0$.

$$Nu = \frac{hL}{k_0} \quad (10)$$

$$Pr_e = \frac{\mu c_p}{k_0} \quad (11)$$

Here, $h$ [W/m²·K] is the heat transfer coefficient, $L$ [m] is the length of the porous medium, and $k_0$ [W/m·K] is the effective thermal conductivity when the fluid is stagnant inside the porous medium. $\mu$ [Pa·s] is the viscosity coefficient of pure water, and $c_p$ [J/kg·K] is the specific heat of the constant pressure of pure water. The feature of this arrangement (Hunt and Tien, 1988) is that the effective thermal conductivity of the porous medium is used for the $Nu$ and the $Pr$ numbers to study the effect of simple heat conduction on the metal part. The $k_0$ of the simple lattice model was a value obtained by CFD analysis using the Fourier law from the temperature gradient inside the porous medium in the simulation when water was stagnated inside. For the stagnant thermal conductivities of the porous media used in the experiment, the effective thermal conductivities were measured, not when the internal fluid was water, but when it was air instead.
Originally, the inside fluid should have been water, but a reasonable value was not obtained because of sealing problems. However, from the CFD models, the difference in the value of \( k_0 \) caused by the difference between the case of water inside and the case of air inside was approximately 5% to 10%. Therefore, the experimental value was substituted with the value when the inside fluid was air. From the above, Table 5 shows the calculation results of stagnant thermal conductivity in the experiment and CFD analysis. Table 6 shows the total surface area of the porous media in the experiment calculated by Eq. (1) for later discussions. The surface area of 3D-CAD models was used for the total surface area of the simple lattice model. Table 6 shows that the porous media used in the experiment had a surface area that is approximately 2–3 times that of the simple lattice model.

Figure 12 shows that the data of the porous media in the experiment showed a higher \( Nu \) number than the data of the simple lattice model of CFD overall, and that, as the value of the horizontal axis \( Re_k \cdot Pr_e \) increased, the \( Nu \) number increased at a large rate. The \( Nu \) number here means, as defined in Eq. (10), how much the heat transfer was accelerated when a fluid flowed, as compared with when the fluid inside the porous medium was stagnant. Thus, in the porous media used in the experiment, the permeability \( K \) was smaller than that of the simple lattice model (Table 4), the flow inside the porous medium was subdivided and complicated, and the surface area was 2–3 times larger (Table 6). Therefore, it is considered that the heat transfer enhancement because of the inside flow appeared significantly in the experiment.

Next, when comparing the \( Nu \) numbers in each of the experimental data and the data of the simple lattice model, the \( Nu \) number became larger when the wire diameter was smaller, and the porosity was smaller at the same value of \( Re_k \cdot Pr_e \). The smaller the porosity, the smaller the permeability, and the larger the surface area, can be understood by the same mechanism as mentioned above.

![Fig. 11 Relationship between heat transfer coefficient and flow velocity](image-url)
Furthermore, when the value of \(Re_K \cdot Pr_e\) became large, the \(Nu\) number did not have a linear relationship, but it tended to approach a certain value as the \(Nu\) number increased. According to Hunt and Tien (1988), as the \(Re\) number increases, the \(Nu\) number is directly proportional to the flow velocity and the number of the half-power of \(L/K^{1/2}\) defined in Eq. (12). The latter number is defined as the inverse Darcy number \(Da_L\).

\[
Da_L = \frac{L}{\sqrt{Re}} \tag{12}
\]

Equation (12) represents the length of the porous medium regarding the square root of the average cross-sectional area of the pore, that is, the length of the porous medium regarding the average width of the pore, and indicates the ratio of the pore of the porous medium to the entire length. Similar to Fig. 12, Fig. 13 shows the relationship between the \(Nu\) number and the heat transfer characteristics using the values of \(Re_K \cdot Pr_e \cdot Da_L^{1/2}\). According to Hunt and Tien (1988),

Table 5 Stagnant (effective) thermal conductivity of porous media

| Sample   | Experiment [W/m \cdot K] | CFD analysis [W/m \cdot K] |
|----------|--------------------------|----------------------------|
| 200-75   | 7.0                      | 20                         |
| 200-85   | 5.9                      | 12                         |
| 500-75   | 8.8                      | 25                         |
| 500-85   | 5.8                      | 13                         |

Table 6 Total surface area of porous media

| Sample   | Experiment \([\times 10^{-3} m^2]\) | CFD analysis \([\times 10^{-3} m^2]\) |
|----------|--------------------------------------|--------------------------------------|
| 200-75   | 6.8                                  | 2.0                                  |
| 200-85   | 4.1                                  | 1.3                                  |
| 500-75   | 2.3                                  | 0.89                                 |
| 500-85   | 1.3                                  | 0.53                                 |

Fig. 12 Relationship between the special Nusselt number and function of the Reynolds number – Prandtl number based on permeability
these are grouped into a linear relationship by the Nu numbers and $Re_k \cdot Pr_e \cdot Da_l^{1/2}$. However, in this study, the cohesion of the data was not good, and the linearity did not appear strongly. Furthermore, different groups were shown in the data of the porous media used in the experiment and the simple lattice model. The porous media studied by Hunt and Tien (1988) had a high porosity of 94% or more among the fibrous porous media. Where the porosity is as small as 75%, as in the present study, the data could not correlate well with the previous study. As a study using lattice-shaped porous media, Tian et al. (2004) conducted a convection heat transfer experiment on porous media with a porosity of 68 to 80% made of copper with square-shaped and diamond-shaped pores. They reported that in porous media using copper, which has a high thermal conductivity, the Nu number indicated the optimum porosity in considering the solid heat conduction through the wire metal fibres, and the forced convection of the fluid. Their study suggests a possible separation of the contributions of internal heat conduction and forced convection heat transfer.

The surface area of the porous media in the experiments in this study were about 2 to 3 times larger than that of the simplified lattice models, and the permeability in the experiments was 1/8 to 1/4 smaller than in lattice models. Therefore, it is considered that the contribution of solid heat conduction was relatively larger than that of forced convection in the experiments. On the other hand, the simplified lattice models were considered to have a larger contribution of forced convection than heat conduction in the porous media in the experiments. Therefore, we decided to treat the two sets of data separately for the correlations. About the difference between the two sample groups sorted by using the inverse Darcy number (Fig. 13), it was considered that the solid heat conduction through the metal fibres was the key factor in the experiment and that the convective heat transfer from the lattice surface area was that in the analysis.

Figure 14(a) shows the data of porous media in the experiment when Nu number and solid heat conduction was taken into account by adopting $Re_k \cdot Pr_e \cdot Da_l^{1/2} \cdot (S \cdot d_s / L^3)$. Here, the contribution of heat conduction in the solid was expressed explicitly in terms of $(S \cdot d_s / L^3)$, which means the volume ratio of the solid part in the porous medium. On the other hand, Fig. 14(b) shows a graph in which the numerical analysis results were correlated by adopting $Re_k \cdot Pr_e \cdot Da_l^{1/2} \cdot (S / L^2)$ in consideration of the difference in Nu number and surface area. Here, convective heat transfer from the surface was expressed explicitly by the term $(S / L^2)$, which means the surface area (per unit cross-sectional area) of the structure. As a result, the correlation was improved in the experimental data and the numerical analysis data, respectively, and linear tendency was obtained with respect to the horizontal axis within the range of the given conditions.

The dotted lines in Figs. 14(a) and 14(b) are approximate expressions for the experimental and analytical results, respectively. Equation (13) is shown below for the fibrous porous media of the experimental results, and Equation (14) is also shown below for the lattice porous media of the numerical analysis results.

$$Nu_{ex} = 0.13 \times Re_k \cdot Pr_e \cdot Da_l^{1/2} \cdot \frac{s \cdot d_s}{L^3} + 9.3$$

$$Nu_{ana} = 0.0046 \times Re_k \cdot Pr_e \cdot Da_l^{1/2} \cdot \frac{s}{L^2} + 2.9$$

It can be said that these formulas would express the characteristics of the heat transfer from the porous media in the experimental data and the numerical analysis. However, these are concise formulas that show trends within the present data of the experiment and numerical analysis. In addition, the constant terms (9.3 and 2.9) in the above equations have not been fully examined, and will need to be studied in the future, including their physical meanings.
Fig. 13 Relationship between the Nusselt and Reynolds number • Prandtl number • modified Darcy number

Fig. 14 Relationship between the Nusselt number and the new parameter suggested by the authors (a) the case of fibrous porous media which are the experimental results (b) the case of lattice porous media which are the numerical analysis results

6. Conclusions

In this study, we investigated heat transfer improvement by using fibrous metallic-porous media in a mini-channel flow. The porous media were made by laminating and sintering fine aluminium wires and directly welded to the end surface of the aluminium block. The aluminium block was inserted into the rectangular flow channel with the cartridge heater inserted. Then, a convection heat transfer experiment was performed when the state was changed from laminar to turbulent flow using pure water as the working fluid. In this experiment, we measured the heat transfer coefficient and the pressure-drop value and investigated the relationship between them by calculating the permeability and Nu number. Furthermore, to discuss the experimental results, we conducted a numerical thermal-fluid analysis by CFD using porous media of a lattice model with a simplified structure, which is different from the experimental porous media. From the results, the effect of promoting heat transfer given by the porous media was investigated. Consequently, the following
four points were found:

(1) Because of calculating the permeability from the average flow velocity and the pressure gradient of the porous media, the permeability of the porous media used in the experiment was smaller than that of the simple lattice model used in the CFD analysis. Compared with the structure of the simple lattice model, the porous media used in the experiment had a structure in which metal wires were laminated intricately and the pore size in the porous medium was randomly configured. The permeability became small because the fluid passed through a complicated path because of the presence of the wire material of the porous medium.

(2) From the relationship between friction factor $f$ and $Re_K$, it was found that the value of friction factor $f$ converges when $Re_K$ becomes large, which was smaller than that of the previous study. From this, it was found that the contribution of dynamic pressure in the porous medium was large, and it was speculated that the heat diffusion effect was increased because of the subdivision and complexity of the flow.

(3) From the relationship between the $Nu$ number and $Re_KPr$, it was confirmed that the $Nu$ number of the porous media used in the experiment was higher than that of the simple lattice model by CFD analysis. Regarding the effects of the wire diameter and porosity, the smaller the wire diameter and the lower the porosity, the higher the $Nu$ number and the higher the heat transfer promotion.

(4) The relationship between the $Nu$ number and $Re_KPr$, was developed from the variable $(Re_K Pr_s^\frac{1}{2} Da_l^2)$ on the horizontal axis using the inverse Darcy number in the previous study. As a result, although the data were well correlated to some extent, the experimental data and the numerical analysis data were separated vertically. Therefore, the experimental data were correlated by the new parameters $Re_K Pr_s^\frac{1}{2} Da_l^2 (S^* d_w/L^3)$ which took into account the heat conduction in solids. On the other hand, the numerical analysis results were correlated by the new parameter $Re_K Pr_s^\frac{1}{2} Da_l^2 (S^* L^2)$ considering the difference in surface area and convective heat transfer from the surface. As a result, the correlations were much improved in the experimental data and the numerical analysis data, respectively, and within the range of the given conditions, a linear tendency was obtained with respect to the horizontal axis, and respective equation were proposed.

From the above, the characteristics of the porous medium using the aluminium wire used in this experiment could somewhat be clarified by considering the larger contribution of the solid heat conduction. The porous medium of this research, therefore, was promoting heat transfer by effectively using the heat conduction of the metal part, as compared with the reported examples of the porous medium in the previous research. On the other hand, about the simplified lattice models in our CFD analysis, different characteristics were found and clarified by considering the lattice surface area for the convective heat transfer. However, to further promote heat transfer, it is conceivable to further reduce the porosity or the wire diameter. In that case, increasing the pressure-drop becomes large, and it is critical to be careful in its practical use. Because the improving contribution of heat conduction and the increase of pressure-drop are contradictory in principle, many numbers of porous media with various specifications must be investigated to obtain a more general tendency.

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