Correlation of Pressure Drop in the Sintered Fibrous Porous Tube with Permeability and Friction Coefficient

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Abstract. In order to enhance the heat transfer performance of heat tubes, porous media are often sintered into them. This increases the heat transfer area due to the surface area being increased by the porous media. Our test apparatus consisted of a double tube structure, and the length was 150 mm. Previous experiments have shown its usefulness and revealed that a breakthrough in the heat exchanger has occurred. For example, air heated to 200 °C and water cooled to 1 °C flowed inside and outside the heat tube, respectively. In tests with conventional heat tubes, air was discharged at 150 °C through the test section. However, in tubes filled with 50 mm of porous medium, air was discharged at 5 °C. In other words, a conventional tube would need to be 100 times as long. We have already proposed a correlation that can accurately predict the heat transfer coefficient regardless of the pipe diameter and filling length. On this study is about understanding the mechanism of pressure drop and constructing an equation, and We have established a method for predicting the pressure drop of a porous body within approximately ±10%.

1. Introduction

It is important to recover waste heat in order to improve the energy efficiency of various systems. Natural gas demand is increasing due to high energy demand. Natural gas emits less NOx, SOx, and CO2 in comparison to coal or coal oil [1]. Liquified natural gas (LNG) is used extensively in Asia specifically to reduce cargo volume in transportation. LNG is liquefied by being cooled to -160 °C. When used, it is gasified by using sea water for warming. However, the method used to warm LNG means that the cryogenic energy (latent and sensible heat of the LNG) is transferred into the sea. Therefore, if devices could recover that energy, there will be some associated environmental and economic benefits.

A device that could be used to recover cryogenic energy is a thermoacoustic engine [2]. The thermoacoustic engine generates electricity by virtue of the temperature difference in the loop tube, which causes the working gas to oscillate thus spinning the turbine [3]. The temperature difference between sea water (dictated by the environmental temperature) and LNG (-160 °C) can thus be used to generate electricity and the LNG is regasified by the heat exchange through the thermoacoustic engine.
However, there is a problem associated with creating a temperature difference in conventional heat tubes due to the large thermal resistance inside the tubes on the gas side. Therefore, to reduce thermal resistance, a porous media that is termed a “sintered high porosity medium” is sintered into the heat tubes.

There are some methods used to enhance the heat transfer performance of heat tubes. One of them involves filling the heat tubes with porous media, thereby increasing the heat exchange surface area. This leads to an increased heat transfer coefficient. Porous media take various forms such as metal foams, fibers, and particles. Metal foam in particular is widely used as a porous medium in heat tubes. Gholamreza et al. [4] experimentally investigated the heat transfer and pressure drop in a metal foam-filled tube. They used R245fa as a working fluid and reported that the heat transfer coefficient increased by a factor of ten over that of the conventional, unfilled tube under Re = 8000 conditions. The pressure drop was notably higher in comparison with conventional tubes, and depended on the pore size of metal foam. In addition, other authors [5, 6] investigated the heat transfer enhancement and pressure drop associated with metal foam. The heat transfer performance could be improved by sintering the porous media onto the inner wall of the heat tube. Jiang et al. [7] investigated convective heat transfer in a sintered porous plate. When compared to an unfilled plate, the sintered porous plate had a local heat transfer rate that was higher by a factor of 15 for water, and by a factor of 30 for air. This indicated that it was more effective with air than water. What is more, the heat transfer coefficient of the sintered porous media plate was higher than that of a non-sintered plate, because the sintered plate had a higher effective thermal conductivity due to being sintered. There were many researches on foam and particle porous medium, whereas there were only a few studies that particularly investigate fibrous porous media. Thus, it is important for gaining a new knowledge to investigate fibrous porous media.

This present study investigated porous media that were termed “sintered high porosity media” that form is completely different from conventional porous media. They incorporated metal fiber-like wires intertwined with each other and sintered into the inner wall of a tube. Therefore, there was almost no thermal resistance between the wall and the porous media. It was expected that the performance of the heat exchanger would be enhanced due to the heat inside the tube being conveyed to the outside of the tube almost immediately.

We have already proposed a correlation that can accurately predict the heat transfer coefficient regardless of the pipe diameter and filling length.

On this study is about understanding the mechanism of pressure drop and constructing an equation.

2. Experimental method
2.1 Heat exchange rig

The experimental rig was composed of three sections for (1) air compression, (2) air heating, and (3) heat exchange. The experimental rig is shown Fig. 1. The red and blue arrows indicate dried air and cooled water, respectively.

(i) Air compression section

This section was composed primarily of a compressor, air dryer, air tank, filter, and flow regulator. The compressed air was dried with the air dryer before flowing on to an air-heating section through a filter, regulator, and Coriolis flow meter (0.1% of r.d.g.).

(ii) Air heating section

This section was composed of a rubber heater and hygrometer. The dried air was heated to 200 °C with a rubber heater controlled using a proportional–integral–derivative (P.I.D.) controller. The humidity was also confirmed to be 0% using a hygrometer (2% R.H.).
(iii) Heat exchange section

The amount of heat exchanged and the pressure drop were measured in the section shown in Fig. 2.

![Diagram of experimental apparatus with labels for cool air, hot air, isobutane, and porous media.](image)

Figure 1. Schematic diagram of experimental apparatus.

![Image of the heat exchange section with water in and out, porous part, and air flow.](image)

Figure 2. The heat exchange section.
It consisted of two tubes used as a double-tube heat exchanger. A heat transfer tube was placed within a polycarbonate tube, along its central axis. Air flowed inside the heat transfer tube and water kept at 1 °C by a constant temperature bath flowed around the heat tube in the same direction. The temperature and pressure of the air were measured using K-type thermocouples and digital differential pressure gages at the inlet and outlet of the test section. Similarly, the temperature of the water was measured using K-type thermocouples at the inlet and outlet of the test section. The pressure and the flow rate of the water were measured using a pressure gage and Coriolis flow meter, respectively. The physical properties were calculated using Refprop version 10 [13].

The experimental equipment was open to the air. Therefore, the amount of heat transfer was calculated using enthalpy and the flow rate in accordance with the following equation:

$$Q_{all} = \dot{m} (h_i - h_o)$$  \hspace{1cm} (1)

The test section was carefully insulated. However, there was some heat introduced into test section and a heat gain test was thus conducted. The test section is described in section 2.3.

### 2.2 Heat tubes containing “sintered high porosity medium”

In order to enhance the heat transfer performance inside the tube (gas side), “sintered high porosity media” were sintered into the heat tubes. This porous media took the form of metallic fibers which were arranged in an intricately intertwined structure (Fig. 3).

Thus, most of the air that flowed into the heat tube collided directly with the porous medium. In addition, the tube wall and porous medium were made from aluminum and sintered to each other; therefore, there was almost no thermal resistance between the two. Aluminum has a relatively high specific heat and thermal conductivity. For these reasons, the outside temperature of the heat tube was conveyed to the porous media almost instantly, increasing the possibility of a uniform temperature in the heat tube (gas side). The heat transfer was thus enhanced due to the heated air colliding with the cooled porous media. The specifications for the heat tubes containing porous medium are shown in Table 1.

### 2.3 Heat gain test

This test investigated the amount of intrusive heat absorbed due to the temperature differential between the environmental temperature and the heat tube’s temperature. Water and air were pumped into the double tube at the same temperature. This temperature was different to the environmental temperature and heat exchange occurred only between the heat tube and the environment. The amount of heat exchange was divided in accordance with the length of the heat tube before and after the test section. The measurement was repeated 5 times, and the amount of absorbed intrusive heat was calculated based on each temperature difference. Therefore, the amount of intrusive heat could be obtained using Eq. (2) where E is the intrusive heat coefficient, and ΔT is the temperature difference.
2.4 Uncertainties

There was no need to consider the heat gain in laminar flow, since the amount of intrusive heat was low. The measurement errors are listed in Table 2. In the case of turbulent flow, the error in the amount of heat transfer measured, $Q$, was within approximately 1% and was at most 2%. The measured masses $\dot{m}$ had associated errors of ± 0.1% of the readings according to the specification documentation for the digital scale used in the mass measurements. As for the measured temperature $T$, the error was ± 0.05 °C, which was estimated from the specification documentation. The error for the constant pressure specific heat $c_p$ was estimated to be ± 0.02 J (g K)$^{-1}$ based on the differences between the inlet and outlet measurements.

\[
Q_i = E_i \cdot \Delta T_i \tag{2}
\]

\[
Q_o = E_o \cdot \Delta T_o \tag{3}
\]

### Table 1. Experiment parameters

| Parameter                          | Value                   |
|------------------------------------|-------------------------|
| Test gas                           | Air (0 % R.H.)          |
| Refrigerant                        | Water (1 °C)            |
| Inner diameter [$D$]               | 12 mm, 18 mm            |
| Porous material fill length [$L$]  | 25 mm, 50 mm, 75 mm     |
| Full length                        | 280 mm                  |
| Heat exchanging section length     | 150 mm                  |
| Thickness                          | 1 mm                    |
| Porosity                           | 80%                     |

### Table 2. Measurement error (r.d.g.: reading, F.S.: Full scale)

| Measuring instrument                  | Measurement error                  |
|---------------------------------------|-------------------------------------|
| Thermocouple [°C]                     | ± 0.05                              |
| Hydrometer [%R.H.]                    | ± 2                                 |
| Digital pressure gauge [%]            | ± 0.25                              |
| Digital differential pressure gauge at double pipe section [kPa] | ± 0.015% of r.d.g. + 0.025% of F.S. (130 kPa) |
| Digital pressure gauge at double pipe section [kPa] | ± 0.02 % of r.d.g. + 5·10$^{-2}$ |
| Coriolis mass flow meter in air [kg/min] | ± 0.1 % of r.d.g. |
| Coriolis mass flow meter in water [kg/min] | ± 0.2 % of r.d.g. |
| Data logger                          |                                     |
| Thermocouple [mV]                    | ± (0.05 % of r.d.g. + 5·10$^{-3}$) |
| Digital pressure gauges [V]          | ± (0.05 % of r.d.g. + 2·10$^{-3}$) |
| Coriolis mass flow meters [V]        |                                     |
| Hydrometer [V]                       |                                     |
3. Heat transfer performance
3.1 Temperature difference of the air between the test section inlet and outlet

Figure 4 shows plots of the temperature at the inlet and outlet of the test section with a diameter \( D = 18 \text{ mm} \) as representative data.

As shown in this figure, the double tube structure is used to cool the outside cold water at 1\(^\circ\)C. The total length of the tube is 450 mm, but the filling length of the porous fiber is very small (25 mm, 50 mm, and 75 mm), which corresponds to the length of a thumb, in Fig.5.

That is to say, we have succeeded in reducing the length of the heat transfer tubes to one-hundredth. Conventional diameter tubes require a length of 3000 mm or more to cool down from 200 \(^\circ\)C to 30 \(^\circ\)C so a 450 mm heat-transfer tube comes out at 150 \(^\circ\)C, but a 25 mm filling tube comes out at 30 \(^\circ\)C, and a 50 mm filling tube comes out at 5 \(^\circ\)C.

In other words, we have succeeded in a breakthrough in heat transfer, where the heat transfer coefficient of air is greater than that of water forced convection.

![Thumb](image.png)

**Figure 5.** Example of porous fill length, thumb.
Figure 6. Correlation for heat transfer adapted to Porous tubes.

The goal of this study is to develop an equation for heat transfer coefficient and pressure drop for the design of a sintered-fiber heat exchanger, and we have already proposed a correlation that can accurately predict the heat transfer coefficient regardless of the pipe diameter and filling length, Fig. 6. So, this study’s main topic is to make the pressure drop correlation, following section.

4. Pressure drop
4.1 Accuracy of equipment

Measurements were performed on the conventional tube to calculate pressure drop and to confirm the accuracy of the equipment used to calculate the friction factor. Fig. 7 shows a plot of the experimental
data from the conventional tube along with results from calculations using equations for friction factor. The Hagen-Poiseuille equation and the Blasius equation were used in laminar flow and turbulent flow conditions, respectively. Fig. 7 shows the accuracy of the match between the experimental data and the results from these equations.

4.2 Pressure drop in tubes containing porous medium

Fig. 8 shows the pressure drop in the part of the tube containing porous medium per mm of length. The horizontal axis shows the Reynolds number average between the inlet and outlet. The pressure drop in the porous medium part alone was calculated by subtracting the pressure drop of the normal part of the tube from the porous medium part. Thereafter, it was divided by the length of the porous medium part. The pressure drop increased as the Reynolds number increased and depended only on tube diameter. The metal fibers of the porous medium all possessed the same form and were evenly distributed. Therefore, there was no bias in the arrangement of the metal fibers in the porous medium. In addition, even in the region where the Reynolds number was 2000 or less, the pressure drop increased continuously, following a similar curve. This indicated that the fluid impacted the porous media and exhibited an effect similar to turbulent flow, unlike the flow in conventional tubes.

4.3 Permeability and form coefficient of porous media

The pressure drop associated with porous metals classified as metal foams can be expressed as a quadratic function of the velocity of the fluid, the permeability, and the form coefficient. [7] [9] [14] [15] This Forchheimer-extended Darcy equation describes the pressure drop associated with metal foams at high fluid flow velocity.

\[
\frac{\Delta P}{L} = \frac{\mu}{K} u + C \rho u^2
\]  

(4)

The Darcy equation is normally used in regions where the Reynolds number is 10 or lower. However, in the case where the velocity of the fluid is increasing, the Forchheimer term; \(C \rho u^2\), is added because the influence of inertial resistance cannot be ignored. In Eq. (4), the left-hand side expresses the pressure drop per unit length, and the right-hand side represents viscous resistance and inertial resistance. \(\mu\) is the dynamic viscosity of the fluid, \(K\) is the permeability of the porous media, \(u\) is the velocity of fluid, \(C\) is the form coefficient of the porous media, and \(\rho\) is the density of the fluid.

![Figure 8. The relationship between Reynolds number and pressure drop for all tubes containing porous medium.](image-url)
Fig. 9 and Fig. 10 show plots of the experimental data for a tube of diameter $D = 12$ mm with an 18-mm long porous medium section. The horizontal axis shows the air velocity and the vertical axis

![Figure 9](image-url)

**Figure 9.** Measured pressure drop with respect to air velocity for tubes of diameter 12 mm containing various lengths of porous medium.

![Figure 10](image-url)

**Figure 10.** Measured pressure drop with respect to air velocity for tubes of diameter 18 mm containing various lengths of porous medium.
shows the pressure drop per unit length. Each tube with its associated length of porous medium is represented by a quadratic function.

Fig. 11 shows a quadratic approximation of the pressure drop in each tube containing porous medium. The correlation coefficients for each of the approximations was over 0.999. Despite the different tube diameters, the pressure drop at specific velocities was consistent. This indicated that the viscous and inertial resistance of the porous medium was almost identical, even though the tube diameter was different. In the other words, the air flowed through the porous medium using the same mechanism because the metal fibers were arranged uniformly in the tubes containing porous medium.

![Figure 11. The relationship between pressure drop and velocity.](image)

From the coefficients of this approximation along with the dynamic viscosity and density of the air, the permeability and form coefficient of the porous medium was calculated. Table 3 shows the $K$ and $C$ of each of the tubes. As can be seen, the values are almost identical. It was found that the metal fibers were evenly arranged without bias in their structural arrangement.

In general, the Reynolds number and friction factor of a metal foam are defined by using permeability as follows [6][10]:

$$Re_K = \frac{\rho u_0^2 K}{\mu}$$ (5)

| Inner diameter [mm] | Porous length [mm] | Permeability [m$^2$] | Form coefficient [m$^{-1}$] |
|---------------------|--------------------|----------------------|-----------------------------|
| 12                  | 25                 | $0.68 \times 10^{-8}$ | 1224                        |
|                     | 50                 | $1.06 \times 10^{-8}$ | 990                         |
|                     | 75                 | $1.30 \times 10^{-8}$ | 1042                        |
| 18                  | 25                 | $1.25 \times 10^{-8}$ | 1051                        |
|                     | 50                 | $1.28 \times 10^{-8}$ | 1018                        |
|                     | 75                 | $1.37 \times 10^{-8}$ | 1022                        |
Eq. (5) and Eq. (6) can be substituted into Eq. (4), allowing Eq. (7) to be obtained.

\[ f_K = \frac{1}{Re_K} + C\sqrt{\frac{K}{\rho u^2}} \]  

(7)

Eq. (7) was used to categorize the experimental data. In Fig. 12, the horizontal and vertical axis represent the Reynolds number and friction factor based on permeability, respectively. Although the length of the porous filling and the tube diameter were different, the porous medium was arranged by using permeability. In addition, it was found that these plots for the porous medium adopted similar shapes because these were governed by one equation.

The last term of Eq. (7), \( C\sqrt{\frac{K}{\rho u^2}} \), was found to be 0.105 in this experiment. Fig. 12 shows a comparison between this study’s data and Peak et al. [10]. Peak et al. investigated a metal foam using water with a porosity of 89-96%, and the value of friction factor was almost identical to that of the present study.

However, it shows a very good agreement with our air data, but there is some variability and we have explored why.

### 4.4 Consider New Physical Mechanism

When passing through a porous body, the fluid velocity should change with the porosity \( \varepsilon \). In previous studies, there is no correlation that considers this mechanism including Paek et al. [10]. Since it is an in-pipe flow, the law of conservation of mass holds.

In other words, if the channel is narrowing and the conservation of mass law holds as shown Eq. (8), then the velocity should be increasing in the porous.

Therefore, it was decided to use the representative velocity of the fluid that considers the flow velocity variation by porosity, Eq. (9).

The law of conservation of mass

![Figure 12](image-url)
\[ \rho A u = (A \varepsilon) \times \rho \times \left( \frac{u}{\varepsilon} \right) \tag{8} \]
\[ \left( \frac{u}{\varepsilon} \right) \tag{9} \]

4.5 New Correlation is born

The results are shown in Fig.13. Independent of the four important parameters, inner diameter, porous filled length, porosity and fiber’s diameter. We find that the experimental values have very good accuracy.

By incorporating the velocity variation of the porosity, the pressure drop estimate was increased by about twice as much as before.

**Figure 13.** New correlation using the velocity in porous.

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**NOMENCLATURE**

- \( d \) : tube inner diameter, \( m \)
- \( D \) : tube outer diameter, \( m \)
- \( u \) : air velocity, \( m \cdot s^{-1} \)
- \( L \) : porous media length, \( m \)
- \( A \) : cross-sectional area, \( m^2 \)
- \( \text{Re} \) : Reynolds number
- \( Q \) : amount of heat transfer, \( W \)
- \( \dot{m} \) : mass flow, \( g \cdot s^{-1} \)
- \( h \) : specific enthalpy, \( J \cdot g^{-1} \)
$T$  air temperature  K, °C  
$Q_{sec}$  lost heat  W  
$E$  lost heat coefficient  W·K$^{-1}$  
$P$  pressure  Pa  
$U$  kinetic energy  W  
$c_p$  specific heat at constant pressure  J·(g·K)$^{-1}$  
$R_{all}$  heat resistance  W·K$^{-1}$  
$\alpha$  heat transfer coefficient  W·K$^{-1}$·m$^{-2}$  
$Q_{all}$  amount of heat transfer  W  
$k$  thermal conductivity  W·K$^{-1}$·m$^{-1}$  
$\Delta P$  pressure drop  Pa  
$C$  form coefficient  m$^{-1}$  
$Re_K$  Reynolds number based on permeability  -  
$K$  permeability  m$^{2}$  
$f_k$  friction factor based on permeability  -  

**GREEK SYMBOL**

ν  dynamic viscosity  m$^{2}$·s$^{-1}$  
ρ  density  g·m$^{-3}$  
μ  viscous coefficient  Pa·s  

**SUBSCRIPTS**

i  inlet  
o  outlet  
air  air side  
water  water side  
al  aluminum side  

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