Experimental and numerical study on thermal-hydraulic performance of printed circuit heat exchanger for liquefied gas vaporization

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1 | INTRODUCTION

Printed circuit heat exchanger (PCHE) is a combination of plates that are photochemically etched form microchannel by diffusion soldering. Its heat transfer area density, that is, the ratio of the total heat transfer area on one side of the heat exchanger to the total volume of the heat exchanger, is high. In designing a new generation of small modular reactors, the PCHE was considered a potential intermediate heat exchanger for high-temperature applications; researches on its thermal-hydraulic performance have been widely carried out, both numerically and experimentally. For instance, Nikitin et al. studied the thermal-hydraulic performance of a zigzag PCHE in a supercritical CO₂ experimental loop, while Ngo et al. developed a numerical model and investigated the thermal-hydraulic performance of an S-shaped PCHE for hot water supplier numerically. Compared with existing hot water suppliers on the hot waterside, the S-shaped PCHE has a smaller volume and lower pressure drop.

Kim et al. compared the thermal-hydraulic performance of conventional zigzag channel PCHE and new PCHE with airfoil shape fins by numerical analysis; the results indicate that new PCHE has lower pressure loss being the heat transfer performance the same. Kim et al. carried out a hydraulic performance test of a newly developed PCHE with a longitudinal corrugated flow path using helium. Based on the experiment, numerical models of different hydraulic diameters and dip angles are established, and the correlation of friction...
of flow channels in a zigzag PCHE using supercritical CO₂ as a working fluid. Kim et al. tested the thermal-hydraulic performance of the PCHE under different conditions, both numerically and experimentally. The performance of PCHE using a KAIST helium test loop was investigated in the helium laminar region; the authors proposed a global $Nu$ correlation and Fanning factor correlation based on the experimental data. They also established a numerical model to obtain a local $Nu$ correlation. Afterward, the thermal-hydraulic performance was investigated, which was in a helium-water, mixture gas of helium and CO₂ by experimental tests and numerical simulations, separately. The $Nu$ correlation and Fanning factor correlation were developed based on the experimental and numerical data.

Yoon et al. developed an analysis code to evaluate the thermal performance design and total fee of crossflow PCHE. Through numerical analysis, Yoon et al. developed thermal-hydraulic correlations for an airfoil PCHE; the correlations were applicable within the range of Reynolds numbers from 0 to 15,000. Furthermore, they compared the total costs of four types of PCHEs (ie, straight, zigzag, S-shape, and airfoil), which were applied for the IHX in HTGRs and SFRs. Xu et al. assessed the effects of airfoil fin (AFF) arrangements on the thermal-hydraulic performance of an AFF PCHE. The staggered fin arrangement was more suitable and had a better performance in the AFF PCHE than parallel fin arrangement. Mylavarapu et al. tested the thermal-hydraulic characteristics of PCHE, which was used as high-temperature helium equipment. In their experiments, they controlled the mass flow between 15 and 49 kg/h, and increased the inlet pressure from 1.0 MPa to 2.7 MPa. The cold side and hot side inlet temperatures were 358-663 K and 481-1063 K, respectively. Based on the experiment, the pressure loss and heat transfer performance were analyzed. Also, a numerical model of semicircular ducts was established, and the numerical analysis of laminar flow with fluid dynamics development and fully developed hydrodynamics was performed. Jeon et al. investigated the thermal performance of a heterogeneous type PCHE numerically and analyzed the effects of the channel sizes for the operating condition. In most of these studies, the working fluids were supercritical carbon dioxide or helium and PCHE was operated at high or ultrahigh temperature to apply in ultrahigh temperature reactor.

The increasing use of natural gas (NG) as a safe and clean energy source pushes forward its demand. NG liquefaction is necessary for transporting large volumes of gas, and the liquefied NG (LNG) must be regasified to NG before use. Therefore, receiving terminals must be equipped with efficient and reliable heat exchangers, also known as “vaporizers.” However, conventional vaporizers (eg, ambient air vaporizer and open-rack vaporizer) have low efficiencies and huge volumes, and they cannot satisfy the heat exchanger requirements of vaporizers for LNG-fueled ships because of space limitation. The design of a high-performance compact vaporizer is thus of practical importance. In this regard, microchannel heat exchangers are promising vaporizers with excellent performance and small components, of which PCHE is the most representative example. However, the thermal-hydraulic performance of PCHE applied to the regasification of liquefied gas has rarely been investigated, which was operated at low temperature.

In the present study, considering the flammable and explosive properties of NG, liquid nitrogen was chosen as the working fluid. A PCHE with straight channels was first manufactured using selected laser sintering (SLS) technology, and a test system was set up to analyze the performance of PCHE used as a vaporizer for supercritical nitrogen. The research focused on the following main topics: (a) The heat transfer and pressure drop characteristics of PCHE were analyzed at different nitrogen pressures through experimental data. (b) Numerical solutions based on a 3-D numerical model of a single channel in the PCHE cold side were obtained using ANSYS Fluent software. Then, the numerical results were verified by experimental data. (c) The existing empirical correlations predicting the $Nu$ and $f$ were verified by experimental and numerical data.

## 2 | EXPERIMENTAL CONTENT

An experimental apparatus was built to test the thermal-hydraulic performance of PCHE for a vaporized process of supercritical nitrogen. Pressures and temperature at the inlet and outlet of PCHE both sides were measured. The heat transfers and pressure drop characteristics of the supercritical nitrogen vaporization process on the cold side of PCHE were the main concern.
2.1 Experimental facility and procedure

The experimental apparatus was composed of an open liquid nitrogen vaporized loop, a closed R22 loop, and a closed water loop (Figure 1). Liquid nitrogen from the low-pressure tank was pumped out by a cryogenic liquid pump and was pressurized to supercritical pressure \( P > 3.4 \text{ MPa} \). The supercritical nitrogen was forwarded through the cold side of PCHE after passing through a buff tank and a mass flowmeter. The supercritical nitrogen absorbed heat from the hot side and vaporized, and then, the resulting nitrogen gas was vented out. On the hot side, after the vacuum state was created, the closed loop was filled with R22 refrigerant gas. By shielding the electric pump, the R22 could circulate in the closed loop. Liquid R22 in the R22 tank was heated by hot water in a shell and tube heat exchanger with a horizontal floating plate after passing through a three-way control valve and a mass flowmeter. The heated R22 flowed to the hot side of PCHE and released heat to the cold side. For the closed water loop, a water pump was used to drive the water. An electric heater (5 kW) heated the water that was then pumped into the heat exchanger, transferring heat to R22; an electromagnetic flowmeter measured the water flow.

Different accuracies and ranges of pressure gauge transducers and thermo-sensors were used to ensure the accuracy of test data. The pressure gauge selected to acquire the data at the cold side of PCHE had a measuring range of 15 MPa and accuracy of \( \pm 0.2\% \) that mounted on the hot side of PCHE had a measuring range of 1 MPa and accuracy of \( \pm 0.25\% \). The pressure gauge for measuring the pressure of the water circuit had an accuracy of \( \pm 0.1\% \) over the entire range of 0.4 MPa. A Pt100 thermocouple was used to measure temperature. Because the temperature of liquid nitrogen increased from 113 K to 286 K as the heat was taken from the hot side, a temperature sensor with an accuracy of \( \pm 0.15\% \) at 173-573 K was mounted at the cold side inlet and that with an accuracy of \( \pm 0.3\% \) at 213-573 K was mounted at the cold side outlet. The temperature of the other loop was measured by a sensor with an accuracy of \( \pm 0.3 \) K at 223-573 K. Two mass flowmeters with different accuracy ranges of \( \pm 0.5\% \) were arranged on both cold and hot sides that on the cold side had a range of 120-1200 kg/h, while the other had a range of 70-700 kg/h. The accuracy of the electromagnetic flowmeter installed on the waterside was \( \pm 0.5\% \).

To reduce or to eliminate the heat loss attributable to convection and radiation, the whole experimental system was thermally insulated using nanoporous silica aerogel insulation material, aluminum foil, rubber insulation cotton, and aluminum zinc plate. The effect of heat loss on the analysis accuracy was investigated by theoretical and experimental analysis. First, theoretical analysis of heat loss was carried out, and the ratio of the theoretical heat loss to the total heat transfer of the PCHE was about 0.94%. Then, the heat balance experiment was performed. The quantity of heat released and absorbed of supercritical nitrogen and R22 were calculated by measuring their inlet and outlet temperatures and pressures. The result of the heat balance experiment proved the correctness of the theoretical analysis. Therefore, the effect of heat loss on the accuracy of the analysis in the experiment can be ignored.

The experimental facility required precooling processes to attain a steady-state condition. The outer surface temperature of PCHE was measured at eight different locations. The actual experiment began only when the average temperature of external PCHE surface was below 113 K. After precooling, it was necessary to run for another 20 minutes when the test obtained the steady state at a given pressure.

2.2 Printed circuit heat exchanger

The PCHE was manufactured by SLS technology using AISI 316L stainless steel (Figure 2). It is a cross flow heat exchanger featuring the orthogonal arrangement of flow channels. The single channel has a semicircular cross section (inner diameter of 1.5 mm) and the flow channel is a straight

![FIGURE 1 Schematic of test apparatus. Key: FR, flow rate meter, P, pressure transmitter, and T, thermo-sensors](image-url)
The core dimensions of test PCHE is $520 \times 26 \times 54 \text{ mm}^3 (L \times H \times W)$ with 3380 channels on the hot side and 169 channels on the cold side. Table 1 summarizes the main geometry data of PCHE.

According to experimental pressure requirements, the cold side of the PCHE designed in such a way should withstand a routine operating pressure of 10.8 MPa. Therefore, a pressure test was operated before the experimental system was designed, and the experimental results demonstrated that the cold side of PCHE endured pressure up to 15 MPa.

### 2.3 | Ranges of experimental parameters

In this test, a total of 104 experimental points for the liquid nitrogen inlet temperature and pressure were analyzed. The supercritical nitrogen inlet temperature varied between 113 K and 128 K, inlet pressure was from 4.5 MPa to 6 MPa, and the mass flow rate was 299.94 kg/h.

### 2.4 | Experimental data reduction

Temperature, mass flow rate, and pressure were measured on the hot and cold sides of the PCHE as described in Section 2.1, so the overall heat transfer coefficient and Fanning friction factor can be estimated from the experimental data.

The overall heat transfer coefficient $U$ is defined as:

$$U = \frac{Q}{\psi A_{\text{cold}} \Delta T_m}$$  

(1)

where

$$Q = m_{\text{cold}} (h_{\text{cold,out}} - h_{\text{cold,in}})$$  

(2)

$A_{\text{cold}}$ is heat transfer area of cold side (Table 1), $\psi$ is correction factor, and $\Delta T_m$ is log-mean temperature difference (LMTD) given by:

$$\Delta T_m = \frac{Q}{\sum \frac{\Delta Q_i}{\Delta T_i}} = \frac{Q}{\Delta Q_{\text{cool}} \Delta T_{\text{cool}} + \Delta Q_{\text{con}} \Delta T_{\text{con}}}$$  

(3)

where $\Delta Q_{\text{cool}}$ and $\Delta Q_{\text{con}}$ is heat load when the R22 occurs cooling and condensation in the PCHE hot channel, respectively; $\Delta T_i$ is LMTD. For R22 cooling, LMTD is given by:

$$\Delta T_{\text{cool}} = \frac{(T_{\text{Lin}} - T_{\text{Lout}}) - (T_{\text{RSa}} - T_{L1})}{\ln \frac{T_{\text{RSa}} - T_{\text{Lout}}}{T_{\text{RSa}} - T_{L1}}}$$  

(4)

where $T_{\text{Lin}}$ is inlet temperature of R22, $T_{\text{RSa}}$ is R22 saturation temperature, $T_{\text{Lout}}$ is outlet temperature of supercritical nitrogen, and $T_{L1}$ is ultimate temperature at the R22 cooling process.

For R22 condensation:

$$\Delta T_{\text{con}} = \frac{T_{L1} - T_{\text{Lin}}}{\ln \frac{T_{\text{RSa}} - T_{\text{Lin}}}{T_{\text{RSa}} - T_{L1}}}$$  

(5)

where $T_{\text{Lin}}$ is inlet temperature of supercritical nitrogen.

The effectiveness of heat exchanger $\varepsilon$ is defined as:

$$\varepsilon = \frac{Q}{Q_{\text{max}}} = \frac{(T_{\text{cold,in}} - T_{\text{hot,out}})}{(T_{\text{hot,in}} - T_{\text{hot,out}})}$$  

(6)
The $\varepsilon$-NTU method\textsuperscript{21} is one of the commonly used heat exchanger thermal design methods that assumes the effectiveness of the crossflow PCHE is a function of the NTU. The relationship between effectiveness and NTU\textsuperscript{22} is expressed as follows:

$$
\varepsilon = 1 - \exp \left[ \frac{1}{C^*} \left( \text{NTU} \right)^{0.22} \left( \exp \left( -C^* \left( \text{NTU} \right)^{0.78} \right) - 1 \right) \right]
$$

(7)

The pressures drop was measured on the PCHE cold side. The overall pressure loss during testing originated from the inlet and outlet header loss, the friction loss in the channel and the acceleration loss due to the density change between inlet and outlet.

Since the hydraulic diameter of headers is much larger than that of flow channel, the headers hardly undergo pressure loss. In other words, the header loss of inlet and outlet can be neglected. Thus, two pressure drop components were considered in the calculation of the pressure factor, that is, the friction loss and the acceleration loss:

$$
\Delta P_{\text{cold}} = \Delta P_{\text{acceleration}} + \Delta P_{\text{friction}}
$$

(8)

where $\Delta P_{\text{cold}}$ can be obtained by calculating the pressure measurement at the inlet and outlet of the PCHE cold end and the acceleration pressure drop is also calculated as:

$$
\Delta P_{\text{acceleration}} = \rho_{\text{cold,out}}v_{\text{cold,out}}^2 - \rho_{\text{cold,in}}v_{\text{cold,in}}^2
$$

(9)

where $\rho_{\text{cold,out}}$ is outlet density of supercritical nitrogen, $\rho_{\text{cold,in}}$ is inlet density of supercritical nitrogen, $v_{\text{cold,out}}$ is outlet velocity of supercritical nitrogen, and $v_{\text{cold,in}}$ is inlet velocity of supercritical nitrogen.

Given the channel of PCHE is a straight semicircular pipe, the frictional pressure drop is given as:

$$
\Delta P_{\text{friction}} = \frac{2f_{cold}L_{\text{cold}}}{D_h} \rho_b v_b^2
$$

(10)

where $D_h$ is hydraulic diameter, $L_{\text{cold}}$ is flow channel length of cold side, $\rho_b$ is average density of supercritical nitrogen, and $v_b$ is average velocity of supercritical nitrogen.

Accordingly, the Fanning factor is calculated as follows:

$$
f_{\text{cold}} = \frac{(\Delta P_{\text{cold}} - \Delta P_{\text{acceleration}})D_h}{2\rho_b v_b L_{\text{cold}}}
$$

(11)

Moreover, $\rho_b$ and $v_b$ of the supercritical nitrogen are calculated based on the average temperature $T_b$, which, in turn, can be calculated by:

$$
T_b = \frac{T_{\text{in}} + T_{\text{out}}}{2}
$$

(12)
where $T_{in}$ and $T_{out}$ are supercritical nitrogen temperatures at the inlet and outlet of the PCHE cold side, respectively.

All the experimental data were processed based on the information concerning the nitrogen thermo-physical properties of nitrogen from NIST Standard Reference Database.

2.5 | Experimental results and discussion

2.5.1 | Heat transfer coefficient and effectiveness

This test was carried out to assess the thermal-hydraulic performance of PCHE with the vaporized process of liquid nitrogen at different system pressure. Figure 4 shows the curves of the outlet temperature of PCHE cold side varying the inlet temperature. The outlet temperature of nitrogen is above 272 K and differs from each inlet indicating that the heat load transferred from the hot sides to the cold sides is different at each experimental point. Figure 5 shows the $Re_b$ trend with the $T_b$ of nitrogen. Since $Re_b$ ranges from 33 165 to 36 123, turbulent flow occurs at the PCHE cold side. The $Re_b$ slightly decreases as the bulk mean temperature of supercritical nitrogen increases from 192 K to 208 K for each pressure value, because pressure and temperature variations have a significant effect on the thermal properties of the supercritical nitrogen. In the proposed experiment, the mass flow rate was kept constant but the viscosity of nitrogen increased with rising temperature, which differed at different pressures. Meanwhile, the viscosity was smaller at lower pressure.

The overall heat transfer coefficient was calculated by using Equation (1) after measuring the temperatures on the cold and hot sides of the PCHE. As Figure 6 shows, the overall heat transfer coefficient did not change regularly with the increase of the $Re_b$. However, the overall heat transfer coefficient can attain relatively high values with the increasing nitrogen inlet pressure, indicating the high thermal performance of PCHE under these operational conditions. Also, heat transfer effectiveness of PCHE is high, close to 99% (Figure 7).

Besides, according to, it is known that a particular problem is the axial conduction effect when the compact microchannel heat exchanger is used in a cryogenic environment. In other words, the unneglectable temperature difference influences the thermal performance of the single heat exchanger significantly.

Baek et al. defined the dimensionless axial conduction parameter $\lambda$ to describe the axial conduction effect and analyze this detrimental effect; $\lambda$ is as follows:

$$\lambda = \frac{\kappa_w A_w}{L C_{min}}$$

(13)

where $\kappa_w$ is thermal conductivity of wall, $A_w$ is cross-sectional area of wall, $L$ is heat exchanger length, and $C_{min}$ is heat capacity rate. In Ref., the impact of the heat exchanger effectiveness can be ignored when $\lambda < 0.005$. In the present study, $\lambda$ was also calculated by measuring the experimental data and through Equation (13). Since the maximum value of $\lambda$ (0.000348) was much smaller than 0.005, the axial conduction could be neglected.

Based on the experimental data, the maximal power density was calculated to about 2.72 MW/m$^3$ and the compactness of the PCHE was approximately 1100 m$^2$/m$^3$ while the gasification volume with respect to the heat exchanger volume was 2864.882 m$^3$/h.
2.5.2 Pressure drop and friction factor

Figure 8 shows the pressure drop in the PCHE cold-side subject at different pressures varying the bulk mean temperature of nitrogen; the pressure drop keeps relatively high at a lower pressure because the increase in the Re, enhances the turbulence. Besides, the Fanning friction factor shows no variation with the increase of Re, (Figure 9). In order to analyze the Fanning friction factor on the cold side, the experimental f was compared with the Fanning friction factor using the Gnielinski correlation.25

The Gnielinski correlation is available for fully developed flows in circular pipe, and the f is as follows:

\[
f = \frac{1}{4} \left( \frac{1}{1.8 \log \text{Re} - 1.5} \right)^2, \quad 2300 \leq \text{Re} \leq 5 \times 10^6 \quad \text{and} \quad 0.5 \leq \text{Pr} \leq 2000
\]  

(14)

Figure 9 also shows the f calculated by the Gnielinski correlation varying with the Re. The experimental friction factor exceeds the calculated ones, with a maximum deviation of 14.6%. The main reason for the deviation may be the
measurement error of the experimental data and the different experimental conditions, such as the difference in the cross section of the pipe. Furthermore, the consideration of the header loss of inlet and outlet is different in this experiment and the Gnielinski correlation.

3 | NUMERICAL MODELS AND METHOD

The average convective heat transfer coefficient cannot be calculated because the internal pressure and temperature of the supercritical nitrogen in the cold side of the PCHE cannot be measured directly. The numerical simulation is available to obtain internal information of the PCHE channels. For numerical models, the selection of SST $\kappa - \omega$ model and the equations are described in detail. The internal information of the cold channel given by numerical data will be reliable if the 3-D numerical model is in good agreement with the experimental data.

Due to the complexity of the coupled heat transfer process of PCHE and since this study focuses on the heat transfer characteristics and flow characteristics of supercritical nitrogen in the cold side, the following assumptions are proposed. (a) The supercritical nitrogen and R22 flow in each hot and cold channel are assumed to be stable and evenly distributed, and the mass flow rate and temperature of each hot and cold channel are consistent. (b) Considering that the pressure drop in the channel is minimal compared with the high pressure of the system, the dependence of nitrogen density on the pressure in the channel was neglected. Based on these assumptions, the established numerical model could be reduced to a single channel of $2 \times 1.75$ mm using a constant heat flux at the top/down position and the insulated boundary condition at left/right position (Figure 10). The numerical model had the same length as the experimental PCHE (520 mm). The model of geometry and the required meshes for the solver were generated by ANSYS Gambit software. Approximately 2667600 meshes were finally generated.

ANSYS Fluent fluid dynamics computational software was used to model and run the simulation for the single channel of PCHE cold side. The energy, continuity, and momentum equations were discretized using the second-order upwind scheme; velocity and pressure were coupled through the semi-implicit method pressure linked equation (SIMPLE). The fluid inlet was set as mass flow inlet and the fluid outlet was chosen as pressure outlet according to experimental data. The density, viscosity, specific heat, and thermal conductivity of nitrogen were obtained from REFPROP. The thermal properties changed considerably and depended on temperature significantly, presented as a polynomial function of temperature. Table 2 shows the polynomial function of temperature at 6 MPa.

3.1 | CFD validation

The reliability of the proposed numerical model and the method must be assessed. The differences between
experimental and numerical data were compared by the following equation:

$$\text{Error} = \left| \frac{\text{CFD} - \text{Experiment}}{\text{Experiment}} \times 100\% \right|$$ \hspace{1cm} (15)

Hundred and four experimental data points of experimental conditions were used for numerical simulation, and the errors between numerical simulation data and experimental data were analyzed and summarized to confirm the correctness of the numerical method. The simulation condition was that the inlet conditions of cold side were consistent with the experiment and the heat flux applied on the upper and lower surfaces of the numerical simulation was calculated according to the actual heat flux of the experiment. Figures 11 and 12 show the outlet temperature and pressure drop differences between experimental and numerical data, respectively. The results highlight good consistency of the numerical and the experimental data.

The average difference between the numerical outlet temperature and the experimental data is 0.2345%, with a maximum of 1.254%. The average difference between the CFD results of the differential pressure and the experimental data is 2.375%, with the maximum difference is 5.6789%.
The consistency between CFD simulation and experimental measurements suggests that the correctness of the 3D numerical model is fully verified. Consequently, the numerical model, method, and data can be considered reliable.

### 3.2 Numerical results and discussion

The hydraulic performance of the single channel of the PCHE cold side has been calculated by numerical simulation. The coefficient of friction is obtained by numerical simulation and through Equation (11), respectively. Figure 13 shows the variation of Fanning friction factor with $\text{Re}_b$ in different inlet pressure; the Gnielinski correlation was also used to predict the $f$. The results show that the empirical friction factor correlation is well-validated against numerical values, with a maximum deviation of 5.12%.

The variation of average convective heat transfer coefficient can be calculated by the following equation.

$$
H = \frac{q}{T_w - T_{mw}}
$$

where $q$ is the heat flux, $T_w$ is area-averaged wall temperature, and $T_{mw}$ is mass-weighted average temperature of supercritical nitrogen, calculated by:

| TABLE 2 | Piecewise-polynomial functions at 6 MPa |
|------|---------------------------------|
| **100-140 K** |
| **Density** | $\rho = 6586.80192 - 154.44382T + 1.390229T^2 - 0.00433T^3$ |
| **Specific heat** | $C_p = -34972.79899 + 1134.1696T - 11.57553T^2 + 0.03944T^3$ |
| **Thermal conductivity** | $\kappa = 0.23565 - 2.39661 \times 10^{-4}T - 1.51519 \times 10^{-5}T^2 + 4.96296 \times 10^{-8}T^3$ |
| **Viscosity** | $\mu = 9.20938 \times 10^{-4} - 1.78064 \times 10^{-5}T + 1.27952 \times 10^{-7}T^2 - 3.33535 \times 10^{-10}T^3$ |
| **140-200 K** |
| **Density** | $\rho = 61544.16687 - 1366.7041T + 11.43075T^2 - 0.04253T^3 + 5.93151 \times 10^{-5}T^4$ |
| **Specific heat** | $C_p = 269355.00689 - 4401.41191T + 24.12557T^2 - 0.0441T^3$ |
| **Thermal conductivity** | $\kappa = 1.60019 - 0.02618T + 1.44562 \times 10^{-4}T^2 - 2.6559 \times 10^{-7}T^3$ |
| **Viscosity** | $\mu = 0.00411 - 9.27404 \times 10^{-5}T + 7.86179 \times 10^{-7}T^2 - 2.95654 \times 10^{-10}T^3$ |
| **200-350 K** |
| **Density** | $\rho = 537.73984 - 3.93407T + 0.01167T^2 - 1.2124 \times 10^{-5}T^3$ |
| **Specific heat** | $C_p = 5387.16265 - 40.50113T + 0.1309T^2 - 1.43693 \times 10^{-4}T^3$ |
| **Thermal conductivity** | $\kappa = 0.03415 - 0.000169217T + 7.47591 \times 10^{-7}T^2 - 8.10651 \times 10^{-10}T^3$ |
| **Viscosity** | $\mu = 1.32819 \times 10^{-4} - 2.71427 \times 10^{-5}T + 2.3346 \times 10^{-10}T^2 - 2.66866 \times 10^{-13}T^3$ |
Figure 12: Pressure difference on the cold side of PCHE (CFD vs Experiment) in different pressure.

Figure 13: Numerical Fanning friction factor varying with $Re_b$ of supercritical nitrogen in different pressure.

where $T_j$, $P_j$, and $V_j$ are the temperature, density, and volume of supercritical nitrogen in the $j$th cell, respectively.

Figure 14 shows the variation of average convective heat transfer coefficient with $Re_b$. The average convective heat transfer rises with increasing $Re_b$ at each pressure because that turbulence intensity increases with the increase of $Re_b$, which enhances the average convective heat transfer performance at each pressure. Also, the inlet pressure of nitrogen induces elevation of the average convection heat transfer coefficient from 4.5 MPa to the default value of 6 MPa, probably because the density and thermal conductivity obviously increase with increased pressure. With the influence of increasing thermal conductivity, the near-wall fluid temperature gradient becomes smaller. Meanwhile, the $Re_b$ decreases
as the pressure increases since the viscosity is higher at higher pressure.

In this study, turbulent flow developed in cold channels. The numerical $Nu$ was compared with heat transfer correlations. To the author's knowledge, heat transfer correlations have been developed using various fluids under various conditions; the heat transfer correlations for the turbulent region are summarized in Table 3.

Although the well-known Dittus-Boelter and Gnielinski correlations have been widely used, the former has definite uncertainty and the latter is developed for fully developed flows. The Berbish correlation was developed for air flow inside a horizontal semicircular duct of 23 mm inner radius.

The $Nu$ obtained from the numerical data was compared with the empirical correlation (see Table 4).

From Table 4, the error for the heat transfer coefficient predicted by Berbish correlation that was developed for a semicircular pipe is higher than those from Dittus-Boelter and Gnielinski correlations. Although the Gnielinski correlation was developed for circular pipe, it is well-validated against numerical data with a maximum deviation of 14.68%. Given that the error in $Nu$ is high when the inlet pressure of nitrogen is of 6 MPa, that is to say the empirical correlations fail to predict the $Nu$ well at low $Re_b$. Nevertheless, the flow rate herein is also constant. The $Re_b$ changes slightly because the thermal property

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**TABLE 3** Currently available heat transfer coefficient correlations

| Authors         | Channel shape | Working fluid | $Nu$ correlation                     | Valid ranges                                      |
|-----------------|---------------|---------------|--------------------------------------|--------------------------------------------------|
| Dittus-Boelter  | Circular      | Any fluid     | $0.0243 Re^{0.8} Pr^{0.4}$          | $10^4 < Re < 1.2 \times 10^5$, $0.7 < Pr < 120$, $L/D > 60$ |
| Gnielinski      | Circular      | Any fluid     | $Nu = \frac{0.0228}{(Re-1000) Pr^{1/3}} \left(\frac{Pr}{Re} - 1\right)^{1/3}$ | $2300 < Re \leq 5 \times 10^6$ and $0.5 \leq Pr \leq 2000$ |
| Berbish         | Semicircular  | Air           | $0.0228 Re^{0.8}$                    | $8200 < Re < 5.8 \times 10^4$                   |

**TABLE 4** Numerical/empirical correlations for $Nu$ percent differences

| Dittus-Boelter       | Gnielinski           | Berbish           |
|----------------------|----------------------|-------------------|
| Average error        | Maximum error        | Average error     | Maximum error     | Average error     | Maximum error     |
| 4.5 MPa              | 6.301%               | 8.334%            | 2.447%            | 5.350%            | 19.012%           | 19.899%           |
| 5 MPa                | 11.164%              | 12.627%           | 5.956%            | 8.335%            | 22.491%           | 23.092%           |
| 5.5 MPa              | 12.391%              | 14.076%           | 7.733%            | 10.513%           | 22.978%           | 23.838%           |
| 6 MPa                | 16.415%              | 17.659%           | 12.487%           | 14.684%           | 25.604%           | 26.378%           |

**FIGURE 14** Average convective heat transfer coefficient varying with $Re_b$ of supercritical nitrogen in different pressure conditions.
of the supercritical nitrogen is susceptible to temperature and pressure. The heat flux of vibrational wall also may affect the predictive accuracy of empirical correlations. In order to assess the effect of heat transfer correlation, the $Nu$ was generated by changing the mass flow rate of fluids from $4.93 \times 10^{-4}$ kg/s to $1.693 \times 10^{-3}$ kg/s using the numerical model at the inlet nitrogen pressure of 6 MPa and compared with those using the empirical correlations.

Figure 15A,B show the curves of $Nu$ and $f$ as a function of $Re_b$ within the range $21\,715 \leq Re_b \leq 37\,249$, respectively. The $Nu$ acquired from the numerical data was compared with those from the three empirical correlations. Since the $Nu$ calculated by Berbish correlation is too different from the result calculated by CFD, it does not apply to the prediction of the heat transfer coefficient. Meanwhile, the $Nu$ calculated by CFD differs from that using the Dittus-Boelter correlation by 3.245% on average and by 9.248% at most. Compared with the
Gnielinski correlation, the numerical results have an average error of 3.546%, with a maximum of 8.538%. Comparatively, the effect of the heat transfer correlation could be accurately evaluated by the Gnielinski correlation. Furthermore, Figure 15B plots the $f$ as a function of $Re_b$. With the increase of the $Re_b$, the friction coefficient decreases and is consistent with the downward trend of the Gnielinski correlation. Given the valid numerical data within $\pm 5\%$ maximum deviation, the present results are in good agreement with the empirical correlation results.

4 CONCLUSION

This paper investigated PCHE characteristics of the heat transfer and pressure drop through a nitrogen gasification experimental test loop. Also, the thermal-hydraulic characteristics of a single channel on the PCHE cold side were simulated by establishing a numerical model. The numerical results agreed well with the experimental data from the cold side. The experimental and numerical results of heat transfer and pressure drop performance of PCHE with the working fluid of supercritical nitrogen were analyzed, and the following main conclusions can be drawn:

1. The variation of outlet temperature, $Re_b$, and the overall heat transfer coefficient with the $Re_b$ were studied experimentally. The $Re_b$ slightly decreased as the bulk mean temperature of supercritical nitrogen increased from 192 K to 208 K at each pressure. The overall heat transfer coefficient mainly varied from 980 W/m$^2$ K to 1100 W/m$^2$ K and increased with the increase of inlet pressure. According to the experimental data, the effect of axial conduction was negligible. The maximal power density was about 2.72 MW/m$^3$, and the compactness of PCHE was approximately 1100 m$^2$/m$^3$.

2. The pressure drop and $f$ at the cold side of PCHE were also investigated experimentally. The pressure drop decreased with the increasing of the inlet pressure. In addition, the experimental friction factor exceeded the data calculated by the Gnielinski correlation with a maximum deviation of 14.6%. The empirical correlation of the $f$ was then validated in the present experiment.

3. The numerical $f$ and average convective heat transfer coefficient were analytically studied through the established numerical model. The empirical $f$ correlation was well-validated against numerical values; its maximum deviation was of 5.12%. The inlet pressure of nitrogen induced the increase of the average convective heat transfer coefficient from 4.5 MPa to 6 MPa. Moreover, the $f$ obtained by numerical data was more consistent with the empirical correlation than that using experimental data. In summary, the Gnielinski correlation best assessed the effect of heat transfer correlations.

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NOMENCLATURE

- $A$: Surface area of heat exchanger (m$^2$)
- $D_h$: Hydraulic diameter (m)
- $f$: Fanning friction factor
- $h$: Enthalpy (kJ/kg)
- $H$: Average convective heat transfer coefficient (W/m$^2$·K)
- $L$: Channel length in the PCHE (mm)
- $m$: Mass flow rate (kg/s)
- $Nu$: Nusselt number
- $P$: Pressure (MPa)
- $Pr$: Prandtl number
- $Q$: Heat load (W)
- $q$: Heat flux density (W/m$^2$)
- $Re$: Reynolds number
- $T$: Temperature (K)
- $U$: Overall heat transfer coefficient (W/m$^2$·K)
- $\Delta P$: Pressure drop (Pa)
- $N$: Velocity (m/s)
- $\rho$: Density (kg/m$^3$)

SUBSCRIPT

- $b$: Bulk mean
- $cold$: Refers to the cold side of PCHE
- $hot$: Refers to the hot side of PCHE
- $in$: Inlet
- $j$: The $j$th cell
- $min$: Minimum
- $mw$: Mass-weighted average
- $out$: Outlet
- $w$: Wall

ACRONYMS

- 3-D: Three dimensional
- AFF: Airfoil fin
- CFD: Computational Fluid Dynamics
- Eq.: Equation
- HTCR: High Temperature Gas-cooled Reactor
- IHX: Intermediate Heat Exchanger
- LMTD: Log-mean temperature difference
- PCHE: Printed Circuit Heat Exchanger
- SFR: Sodium-cooled fast reactors
- SIMPLE: Semi-Implicit Method Pressure-Linked Equation
SLS  Selected laser sintering

CONFLICT OF INTEREST
None declared.

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