An experimental investigation on parallel-plate finned heat exchanger working with cryogenic oscillating helium flow

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Abstract. The space-cycle averaged Nusselt number is used to characterize the heat transfer characteristics of cold end heat exchanger in cryocoolers working with cryogenic oscillating helium flow. An experimental setup for cryogenic oscillating flow heat transfer measurement was designed and established. According to the experimental data obtained, it is evident that low temperature is significantly disadvantageous for heat transfer in oscillating flow. Similar to the experimental results at room temperature, under different mass flow rates, the heat exchanger has its own corresponding optimal Reynolds number at certain temperature for maximized heat transfer capacity.

1. Introduction

Refrigeration mechanism of regenerative cryocoolers working from liquid nitrogen to liquid helium temperature zone has been widely studied [1, 2]. Among the components of a regenerative cryocooler, the cold end heat exchanger is essential to its performance optimization. For compressible gas, the thermal properties of helium change with temperature, especially dramatically in cryogenic environment. Meanwhile, the oscillating flow brings deviation for heat transfer characteristics of heat exchangers from conventional situations. Comprehensively speaking, there is obvious difference in the heat exchangers working with room temperature steady flow or with cryogenic oscillating flow. The traditional design methods for heat exchanger are not necessarily applicable for the cold end heat exchangers in regenerative cryocoolers.

Due to the significant difference between oscillating flow and steady flow, researchers have proposed a variety of methods to describe the heat transfer characteristics of oscillating flow heat exchangers. In 1967, Richardson [3] proposed that the experimental correlations in steady flow were approximated in one cycle integrated, which is called the time-averaged steady-flow equivalent approximation. Nika et al. [4] developed a representative method for the Nusselt number in complex domain based on classical thermoacoustic theory. In 1986, Gedeon [5] firstly proposed the characterization method of space-cycle averaged Nusselt number and verified the rationality of introducing the corresponding parameters in momentum and energy equations. This characterization method is especially operative in terms of experiments because it does not require accurate measurement of the phase difference between heat transfer temperature and heat flow on the wall. Tang et al. [6, 7] carried out experimental study on the heat exchanger characteristics of parallel-plate finned heat exchangers, focusing on the operating parameters, such as maximum Reynolds number.
Re_{max}, Valensi number Va, pressure ratio PR, as well as heat exchanger geometry, and a comprehensive heat transfer empirical correlation was then proposed [8]. The non-dimensional parameters discussed are defined as follows,

\[ Nu = \frac{h d_h}{\lambda} \]  \hspace{1cm} (1)

\[ Re_{max} = \frac{u_{max} d_h}{v} \]  \hspace{1cm} (2)

\[ Va = \frac{\omega d_h^2}{v} \]  \hspace{1cm} (3)

\[ PR = \frac{p_{m1} + p_{A1}}{p_{m1} - p_{A1}} \]  \hspace{1cm} (4)

where \( h \) is the heat transfer coefficient, \( d_h \) is the hydraulic diameter, \( u_{max} \) is the maximum flow velocity, \( v \) is the viscosity of fluid, \( \omega \) is the angular velocity, \( p_{m1} \) is the average working pressure, \( p_{A1} \) is the pressure amplitude measured by the sensor. Pressure ratio \( PR \) characterizes the compression expansion effect in the oscillating flow.

Due to strong temperature dependency, the thermophysical properties of working medium should be carefully studied for the heat transfer processes in the cryogenic environment, especially the helium flow in the heat exchangers of cryocoolers. Hands et al. [9] focused on the empirical correlation of thermal conductivity for helium below ambient temperature. Banjare [10] used CFD method to calculate the temperature and pressure distributions in a double-inlet G-M type pulse tube refrigerator working below liquid nitrogen temperature regime. Quan et al. [11] considered unsuitability of the heat transfer empirical correlations for the design of high-frequency pulse tube refrigerators, induced by cryogenic temperature.

However, there are still very few researches on the heat transfer characteristics in oscillating flow under cryogenic temperatures. In this study, an experimental setup operating at the temperatures as low as 30 K was established to investigate heat transfer characteristics of the cold end heat exchangers working with cryogenic oscillating flow. It is a follow-up research along the previous simulations and experiments at ambient temperature by the present group [6, 7].

2. Experimental setup

In order to extend the measurable temperature regime of the testing section to cryogenic zone and to simulate the actual working conditions of the cold end heat exchanger in cryocoolers, the design principle of experimental setup is based on energy conservation, that is, in the case where the oscillating “cold” fluid is heated by the “hot” wall of slit flow path, the heat capacity of the testing section is acquired by measuring the heat compensation controlled by electric heating wires in the steady state of heat transfer.

The experimental setup is illustrated in Figure 1, which is mainly composed of a linear compressor, a vertical vacuum hood with a copper cooling shield, and the testing components integrated at the flange of the upper cover. The testing components mainly include a G-M refrigerator, a liquid nitrogen pre-cooling tank, regenerators, a cold end heat exchanger, a testing section, a thermal buffer tube, a needle valve, and a gas reservoir. They can be placed in either normal or cryogenic environment, according to the required working conditions.

The testing section was clamped by two flanges, which are connected with pipeline. A tapered section was designed to ensure uniformity of the flow inside heat exchanger. In the cryogenic environment, the testing section was sealed with indium wires and then bolted to avoid gas leakage. Considering the processing cost and the operating conditions of the measurement instruments, the gas-side temperature and pressure were measured. The illustration and picture of the testing section are shown in Figure 2.
3. Results and discussion

3.1. Heat leak correction

The electric heating power for thermal compensation is obtained as,

\[ Q_h = \eta_{ele} \times U \times I \]  \hspace{1cm} (5)

where \( \eta_{ele} \) is the electrothermal efficiency, \( U \) and \( I \) are the electric voltage and current, respectively.
The gas-side heat exchange is actually equal to the sum of compensational electric heating power and heat leakage. The heat loss corrections at different temperatures are shown in Figure 3. When the cooling process reached the set point, the heat loss from the surroundings was considered to balance the cooling capacity released by heat exchanger.

The inner wall temperature of heat exchanger \( t_{wi} \) obtains,

\[
t_{wi} = t_{wo} + \frac{Q_h}{2\pi\lambda_{Cu}} \frac{d_o}{d_i} \]

(6)

where \( t_{wo} \) is the average temperature on the heat exchanger outer wall. \( \lambda_{Cu} \) is the average thermal conductivity of copper in the temperature variation range. \( d_i \) and \( d_o \) are the inner and outer radius of the heat exchanger wall, respectively.

![Figure 3](https://example.com/figure3.png)

**Figure 3.** Heat loss rate revised at 75 K (left) and 35 K (right)

The average convective heat transfer coefficient \( h \) obtains,

\[
h = \frac{Q_c}{\sum_{i=0}^{n} A_{2i} \eta_{fin,i} (t_f - t_w)}
\]

(7)

where \( Q_c \) is the cold end heat transfer capacity, \( A_1 \) is the overall area of the heat exchanger fins, \( A_{2i} \) is the area on the \( i \)th fin, \( \eta_{fin,i} \) is the efficiency of the \( i \)th fin, \( t_f \) is the temperature of fluid, \( t_w \) is the wall temperature.

### 3.2. Heat transfer characteristics

The space-cycle averaged Nusselt number was used to characterize and summarize the heat transfer of oscillating helium flow in the parallel plate channels of heat exchanger. This experiment simulated the actual working conditions of the cold end heat exchanger in Stirling refrigerator, thermoacoustic refrigerator or pulse tube refrigerator.

A parallel-plate heat exchanger, with the hydraulic diameter of 1.5 mm and the flow path length of 20 mm, was tested. The pressure ratio at the inlet of heat exchanger was controlled to a typical value of 1.2.

Keeping the initial working pressure at 2 MPa and the pressure ratio of 1.2, the gas-side heat transfer coefficient in the oscillating flow is evaluated. The resulting coefficients are then converted into local Nusselt number, whose variation with the temperature and the normalized gas displacement amplitude is shown in Figure 4.

For the heat exchanger tested, it can be seen that when the gas displacement amplitude changes around the value of 0.5, the local Nusselt number dramatically increases, where the peak-to-peak displacement amplitude is equal to the fin length. The variation in Nusselt number is assumed to be caused by the increase in effective heat exchange area. During the entire oscillating period, some gas
parcels are only in contact with the heat exchanger. The heat of gas parcels delivered from one location to another on the same fin results in no effective heat transfer between fins. Only a small amount of parcels in contact with the heat exchanger fin can reach the adjacent fins and contribute to the effective heat transfer. When the gas displacement amplitude is increased, more gas parcels can participate in the actual heat transfer process, and then the calculated Nusselt number has higher value. As a result, Figure 4 takes on a symmetric wing-shaped pattern.

A series of experiments have also been carried out to study the variation of heat transfer Nusselt number $Nu$ with maximum Reynolds number $Re_{\text{max}}$, when the experimental pressure ratio and Valensi number $Va$ are fixed at 1.2 and 150, respectively. Figure 5 shows the obtained data in terms of Nusselt number varying with $Re_{\text{max}}$ and working temperature.

It can be seen from the results given in Figure 4 that working temperature has influence on working pressure with a fixed initial charging pressure. However, within the average pressure range observed experimentally, the maximum gap between Nusselt numbers is below 5%. Therefore, it can be considered that the working pressure, depending on initial charging pressure and working temperature, has little effect on the performance of the oscillating flow heat exchanger. More significantly, the Nusselt number increases with the rise in the maximum Reynolds number of oscillating flow.

3.3. Comparison with ambient temperature conditions

Comparison of the heat transfer performance at cryogenic or ambient temperature deserves much attention. The space-cycle averaged Nusselt number under identical other working conditions was obtained to have an intuitive perspective on the different heat transfer performance caused by the dramatic change in thermophysical properties due to temperature drop. Quantitative analysis should be introduced by non-dimensional indicators such as thermal penetration depth and Prandtl number, which can be considered as a function of temperature. Here, the data obtained from experiments are only presented in terms of temperature and Reynolds number for the convenience of comparison, as shown in Figure 6. It is evident that low temperature is significantly disadvantageous for the heat transfer in oscillating flow.

However, higher space-cycle averaged Nusselt number does not necessarily indicate higher heat transfer capacity for a certain heat exchanger. The actual heat transfer capacity of the tested section at the initial pressure of 1 MPa, the oscillating frequency of 50 Hz and the working temperature of 45 K is calculated with different Reynolds numbers, as shown in Figure 7. Similar to the results at ambient temperature, under different mass flow rates, the heat exchanger has its own optimal Reynolds number at a certain temperature. In this situation, the optimal Reynolds number stays around 500 to 1000.
4. Conclusions
An experimental setup for measuring the cryogenic oscillating flow heat transfer was designed and established to simulate actual operating environment of the cold end heat exchanger of a regenerative refrigerator. The temperature zone was set between 80 K and 30 K. The combination of liquid nitrogen precooling (first stage) and G-M refrigeration (second stage) was adopted to meet the requirement for experimental operation. The variations of space-cycle averaged Nusselt number with maximum Reynolds number and Valensi number showed similar trends as those in our previous study for the cases in room temperature zone. Declined Nusselt number and heat transfer capacity along with the temperature drop indicate that cryogenic environment brings no benefit for enhancing the heat transfer in oscillating helium flow. In the follow-up research, the quantitative non-dimensional criterion will be introduced as a key temperature indicator to describe the heat transfer characteristics and to realize optimal design of the cold end heat exchanger for cryocoolers.

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NOMENCLATURE

- $h$: heat transfer coefficient
- $d_h$: hydraulic diameter
- $d_i$: diameter of inner wall
- $u_{max}$: maximum flow velocity
- $U$: electric voltage
- $t_{wo}$: temperature at outer wall
- $A_{2j}$: area of the $i$th fin
- $Nu$: Nusselt number
- $Pr$: Prandtl number

Greek Letters

- $\omega$: angular velocity
- $\lambda_{Cu}$: thermal conductivity of copper
- $\eta_{ele}$: electrothermal efficiency
- $\eta_{fin,i}$: efficiency of the $i$th fin
**Subscripts**

- \( h \)  hydraulic parameter
- \( m \)  mean value
- \( \text{max} \)  maximum value
- \( w \)  wall
- \( \text{ele} \)  electric
- \( \text{fin} \)  fin

**References**

[1] Paek I, Braun J E, Mongeau L, 2005 *J. Acoust. Soc. Am.* **118**(4) 2271-2280

[2] Nsofor E C, Celik S, Wang X D, 2007 *Appl. Therm. Eng.* **27**(14-15) 2435-2442

[3] Richardson P D, 1967 *Appl. Mech. Rev.* **20** 201–217

[4] Nika P, Bailly Y, Guermeur F, 2005 *Int. J. Heat Mass Tran.* **48**(18) 3773-3792

[5] Gedeon D, 1986 *J. Heat Tran.* **108** 513-518

[6] Tang K, Yu J, Jin T, Wang Y P, Tang W T, Gan Z H, 2014 *Int. J. Heat Mass Tran.* **70**(3) 811-818

[7] Tang K, Yu J, Jin T, Gan Z H, 2013 *J. Zhejiang Univ-Sc. A* **14**(6) 427-434

[8] Huang J L, Liu M L, Jin T, 2017 *Appl. Sci-Basel.* **7**(117) 1-13

[9] Hands B A, Arp V D, 1981 *Cryogenics* **21**(12) 697-703

[10] Banjare Y P, Sahoo R K, Sarangi S K, 2010 *Cryogenics* **50** 271-280

[11] Quan J, Liu Y J, Liang J T, Wang J, 2012 *Cryocoolers* **17** 309-314.