Studies on the development and efficiency improvement of a 1.5 W at 25 K two stage pulse tube cooler

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Abstract. A high frequency two stage pulse tube cooler (PTC) has been designed using Sage software for a JT-PTC hybrid helium recondensation system. The cold end of the first stage regenerator is anchored at 80 K using a liquid nitrogen supply. Such a thermally coupled design simplifies the design without the need for considering flow distribution between the stages and ensures that entire flow is available to produce cooling power at 25 K. The pulse tube cooler is designed for maximal utilization of the 900 W PV power provided by the Pressure Wave Generator (PWG). With the first prototype, a no load temperature of 40.4 K was achieved at a filling pressure of 24.1 bar. The effect of filling pressure on the acoustic matching of the PTC and PWG was investigated. It is observed that filling pressure has a significant effect on the PWG piston stroke amplitude. Using phasor analysis, it is shown that the phase relationship at different sections of the two stage PTC is detrimentally affected by the pulse tube volume. A scheme for achieving advantageous phasing by reducing the pulse tube volume is proposed. This involves maintaining the hot heat exchanger, inertance tube and buffer volume at 80 K. With the modification it is shown that a beneficial phase reversal across second stage Regenerator is achieved. The method is currently under experimental investigation.

1. Introduction

Hybrid cryocoolers which can provide refrigeration in the temperature range of 4-10 K are being studied with much interest [1][2]. Our work is focussed on developing a JT-Pulse tube hybrid cryocooler which can provide 0.5 W at 4.2 K, or an equivalent helium liquefaction rate of 17 liters per day. Such a cryocooler can be deployed with devices such as cryostats to achieve zero helium boil off. To achieve the design goal, the pulse tube cooler needs to produce 1.5 W at 25 K. This paper describes the research that has been carried till date to develop the two stage pulse cooler.

2. Thermodynamical analysis of two stage pulse tube cooler

A two stage pulse tube cooler is thermodynamically different from a single stage cooler as it engages multiple heat sources or sinks at different temperatures. This is illustrated in Figure 1. A single stage cooler operates between the temperatures $T_C$ and $T_H$ whereas a two stage cooler can operate with two refrigeration temperatures $T_{C1}$ and $T_{C2}$ as in a gas coupled cooler. It can
also operate with two heat rejection temperatures \( T_{H1} \) and \( T_{H2} \) as in a thermally coupled cooler. Using the notation of Figure 1a,

\[
COP_{I} = \frac{Q_{C}}{W} \tag{1}
\]

The overall coefficient of performance of a gas coupled cooler is

\[
COP_{II, overall} = \frac{Q_{C2}}{W} \tag{2}
\]

The coefficients of performance of the first stage and the second stage in the gas coupled cooler are

\[
COP_{II, first} = \frac{Q_{C1}}{W_{1}} \quad COP_{II, second} = \frac{Q_{C2}}{W_{2}} \tag{3}
\]

From Figure 1 (b) and the definitions in the equations 1 - 3

\[
W = W_{1} + W_{2} \tag{4}
\]

\[
\frac{Q_{C2}}{COP_{II, overall}} = \frac{Q_{C1}}{COP_{II, first}} + \frac{Q_{C2}}{COP_{II, second}} \tag{5}
\]

In eq. 5, \( COP_{II, second} \) will be equal to \( COP_{I} \) because both the heat engine source and sink temperatures are the same. Therefore

\[
\frac{1}{COP_{II, overall}} = \frac{Q_{C1}/Q_{C2}}{COP_{II, first}} + \frac{1}{COP_{I}} \tag{6}
\]

Because the first term on the right hand side is always positive

\[
COP_{II, overall} < COP_{I} \tag{7}
\]

i.e. the coefficient of performance of a two stage cooler is always less than that of a single stage cooler. The same result can be derived for the thermally coupled two stage pulse tube
cooler by noting that a refrigerator at $T_{C1}$ is required to provide a heat sink at $T_{C1}$. However, in a practical pulse tube cooler, the regenerator losses will be high in a single stage cooler as compared to a two stage cooler. Thus a two stage cooler is preferred over a single stage for low temperature refrigeration.

3. Staging method

Staging method refers to the arrangement of different stages of a multistage pulse tube cooler. A two stage thermally coupled design was chosen in order that the entire PV power generated by the Pressure Wave Generator (PWG) is channeled to the cold heat exchanger. This is not the case in a gas coupled cooler where some of the PV power flows into the first stage cold heat exchanger. Thus the thermally coupled design has the potential to produce a lower no load temperature and higher cooling power. But there will be a trade-off due to the second stage regenerator losses. The staging method is shown schematically in Figure 2.

An important feature of the staging method is that the intermediate heat exchanger is anchored at 80 K. In the preliminary stages of the study, a liquid nitrogen supply was used to achieve this. Eventually this will be replaced by an appropriate cooler capable of providing refrigeration at 80 K. The hot heat exchanger is anchored at 300 K. The Pressure Wave Generator (PWG) used for the study was CFIC make model 2S175W-X [3]. Its maximum power consumption is 1400 W at 50 Hz and can supply acoustic power of 900 W to a matched pulse tube cooler.

4. Cryocooler design

The two stage pulse tube cooler was designed using Sage [4]. The entire pulse tube cooler along with the PWG was modeled which enables the holistic design of the PTC taking into consideration the PWG characteristics. The PWG parameters, provided by CFIC, were input to the model. The main input parameters are the stroke and piston diameter. The design
operating pressure and frequency were chosen as 25 bar and 40 Hz. The components of the two-stage pulse tube cooler, like Regenerator 1, 2 (Figure 2) and the inertance tube were optimized for maximizing the cooling power at 25 K and the optimized dimensions are shown in Table 1. For an input stroke of ±6.5 mm, the model predicts a cooling power of 0.01 W at 25 K with an input PV power of 850 W. Even though the predicted cooling power is well below the design goal of 1.5 W at 25 K, experimental study of the design was undertaken to gain experience with such systems.

Table 1: Optimized dimensions of the Pulse Tube cooler obtained using Sage model.

| Component        | Diameter (mm) | Length (mm) | Filled with       |
|------------------|---------------|-------------|-------------------|
| Regenerator 1    | 37.7          | 36          | # 400 SS mesh     |
| Regenerator 2    | 37.7          | 94          | # 400 SS mesh     |
| Pulse tube       | 28.3          | 131         | -                 |
| Inertance tube   | 4.5           | 2924        | -                 |

The pulse tube cooler was fabricated and assembled as shown in Figure 3. Mylar multilayer insulation was wrapped on the two stage cooler for radiation shielding. A calibrated cernox sensor was used to measure the cold heat exchanger temperature. The cooler was enclosed in a vacuum jacket at a vacuum of 2E-4 mbar. The vacuum jacket also has an inbuilt liquid nitrogen thermal shield to cut-off the radiation losses.

5. Experimental Results

5.1. Effect of operating pressure on acoustic matching

The cooler was first tested near the design conditions at an average filling pressure of 24.1 bar. The combined resonance frequency of the PWG-PTC system was experimentally evaluated to be 39 Hz which is very close to the design frequency of 40 Hz. Hence, the cooler was tested at 39 Hz. A no load temperature of 40.4 K was achieved and the corresponding cool down curve is shown in figure 4. This is higher than the design no load temperature of 25 K. A piston stroke amplitude of 3.9 mm was measured from the pressure amplitude in the piston backspace. This is less than the design piston stroke amplitude of 6.5 mm.

Figure 4: Cool down curve at 24.1 bar.
The parasitic losses in the system due to wall conduction, convection and radiation were evaluated using the rate of temperature rise method [5] similar to the transient model analysis of pulse tube cooler [6]. The temperature of the cold heat exchanger was monitored after switching off the PWG and is shown in Figure 5. The gradient of the warmup data along with the mass of the cold heat exchanger and specific heat of cold heat exchanger material were used to calculate the overall parasitic loss. It was found to be 0.14 W at 41 K. These parasitic losses are low and may have caused the no load temperature to be higher by a few tenths of a Kelvin.

The PTC was also tested at the filling pressures of 19.8 bar and 16.0 bar. The resonance frequency was evaluated at each of these filling pressures and was found to be 39 Hz, equal to that at 24.1 bar. The cool down curves are shown in figures 6, 7. At filling pressures of 19.8 and 16.0 bar, no load temperatures of 41.0 and 42.3 K respectively were obtained. Table 2 shows other important parameters observed in the experiments.

**Table 2: Experimental observations during the pulse tube cooler testing.**

| Parameter                        | Experiment-1 | Experiment-2 | Experiment-3 |
|----------------------------------|--------------|--------------|--------------|
| Filling pressure (bar)           | 24.1         | 19.8         | 16.0         |
| Resonant operating frequency (Hz)| 39           | 39           | 39           |
| No load temperature (K)          | 40.4         | 41.0         | 42.3         |
| Input voltage (V)                | 46.1         | 47.5         | 47.7         |
| Input current (A)                | 16.0         | 16.0         | 16.0         |
| Input electric power (W)         | 737.6        | 760.0        | 763.2        |
| Pressure amplitude in PWG (bar)  | 1.3          | 1.3          | 1.3          |
| Piston stroke amplitude (mm)     | 3.9          | 4.2          | 5.4          |

From Table 2, a non dependence of resonant operating frequency on the filling pressure indicates that the two stage PTC impedance is a weak function of filling pressure. This is valid for this specific cooler. In general, filling pressure has a significant effect on the resonance frequency on the PTC. The gas spring stiffness associated with this cooler is not significantly affected by filling pressure. For the total input current of 16.0 A, as the filling pressure is reduced from 24.1 bar to 16.0 bar the piston stroke amplitude increases from 3.9 to 5.4 mm. This results in more compression of the operating gas and increase in the input power from 737.6 to 763.2 W. The pressure amplitude in the PWG does not reflect this increase in compression as it is directly proportional to filling pressure. Thus the filling pressure has a strong effect on Piston stroke.
amplitude and hence PWG impedance. An ideal matching PWG and PTC is said to occur when the maximum piston stroke amplitude (in this case 6.5 mm) is obtained at the maximum value of current (in this case 16 A). The experimental results show that a better matching is to be achieved at 25 bar.

To compare the experimental data with Sage model predictions, the experimental data of Table 2 were input into the model. For the data of Experiment-1, the model predicted a no load temperature of 24 K. This is very different from the experimentally observed no load temperature of 40.4 K. The reasons for such a deviation could be the manifold; one of the reasons for low performance is investigated in the next section.

5.2. Effect of pulse tube volume on phasing

In order to further analyze the reasons for low performance, a phasor analysis of the design was done using the data from Sage and is shown in Figure 8. Inlet and outlet in Figure 8 refer to the direction from the PWG to the buffer. For a single stage PTC a phase reversal with a phase of $+30^\circ$ at the cold end of the regenerator is ideal [7], where a positive phase angle means that the pressure leads the mass flow. But the phasor diagram of the current design configuration does not show phase reversal. However the phase angle observed at the inertance tube inlet is $56.8^\circ$ which is close to $60^\circ$ recommended for the single stage pulse tube cooler.

These contrasting pictures point to the importance of pulse tube volume. In a two stage cooler, the pulse tube has to provide buffering action across a large temperature difference. Typically buffering has to be provided from the no load temperature to ambient temperature. This means that the pulse tube volume will be high. This being the case, the phase angle at the cold end of the Regenerator 2 will be pushed towards the negative side thus making it difficult to achieve the phase reversal. This indicates that the pulse tube volume should be optimized and kept to the minimum possible.

One way to reduce the volume of the pulse tube is to require of it a lesser buffering action by maintaining the warm heat exchanger at 80 K. This means that the pulse tube has to achieve temperature separation from no load temperature to 80 K instead of ambient temperature. This condition was simulated using sage software where the hot heat exchanger, inertance tube and the buffer were all maintained at 80 K [8]. The PTC components were optimized in such a condition and the optimized PTC dimensions are shown in Table 3.

A phasor diagram was constructed for this configuration and shown in Figure 9. A phase reversal is observed across Regenerator 2 which minimizes the regenerator losses. A cooling power of 2.1 W at 25 K is predicted for this configuration. This is sufficient to meet the required cooling power need of the helium recondensation system. Experimental investigation of this method is currently underway.
Table 3: Optimized dimensions of the two stage PTC, using Sage model, with the hot heat exchanger, inertance tube, buffer at 80K.

| Component           | Diameter (mm) | Length (mm) | Filled with          |
|---------------------|---------------|-------------|----------------------|
| Regenerator 1       | 37.7          | 24          | # 400 SS mesh        |
| Regenerator 2       | 37.7          | 44          | # 400 SS mesh        |
| Pulse tube          | 28.3          | 65          | -                    |
| Inertance tube      | 4.5           | 970         | -                    |

Figure 9: Phasor diagram of the optimized pulse tube configuration with the hot heat exchanger, inertance tube, buffer maintained at 80 K.

6. Conclusions

This work has analytically shown that a two stage pulse tube cooler is thermodynamically less efficient than a single stage pulse tube cooler. But a two stage PTC is required to reduce regenerator losses. To meet the cooling requirement of a proposed recondensation system, a two stage PTC was designed using Sage software. A no load temperature of 40.4 K achieved at 24.1 bar filling pressure with an input power of 750 W. A significant effect of filling pressure on acoustic matching is observed. The work also established the detrimental effect of pulse tube volume using phasor analysis. A high pulse tube volume causes the phase angle at the two stage regenerator to be pushed to the negative side where the pressure lags the mass flow. A method for achieving advantageous phasing is proposed which involves maintaining the hot heat exchanger, inertance tube and buffer at 80K. The predicted cooling power is 2.1 W at 25 K which can meet the cooling power requirement of the proposed helium recondensation system. Experimental work to investigate this method is underway.

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