Performance Study and Optimization of a New Type of 4K GM/PT Hybrid Refrigerator

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We recently proposed and designed a new type of 4K GM/PT hybrid refrigerator, where the upper warm stage is a typical GM refrigeration cycle, and the cold stage is a pulse tube refrigerator, on which is thermodynamically coupled the upper warm stage. Four different types of phase shifting assembly were analyzed for adjustment of phase shift in the pulse tube. We show that the volume ratio of the regenerator to pulse tube, the volume ratio of the cold auxiliary piston to pulse tube, and the initial phase angle between the cold auxiliary piston and main displacer are of three most important parameters. In this paper, we briefly summarize our previous work and further optimize the hybrid refrigerator with respect to the structure parameters and geometrical configuration.

INTRODUCTION

With the developments of rear earth regenerative materials, the performance of multi-stage GM and pulse tube refrigerators (PTRs) at 4.2K has been significantly improved [1-5]. Presently, the GM refrigerators can provide more than 3W at 4.2K with the COP up to $6 \times 10^{-4}$ [2]. However, vibrations and EMI caused by the moving displacer in the cold head of such coolers are of the severe problems for the users.

The absence of the displacer in the PTRs gives them many advantages over GM refrigerators. However, the COP of PTRs is still lower than $1.5 \times 10^{-4}$ [5], moreover, the cooling capacities at 40–45K on the 1st-stage are still smaller than that of GM refrigerators. Many applications, like MRI systems, require large cooling capacity of above 1W at 4.2K on the 2nd-stage cooling station simultaneously with 40–50W at 40K on the 1st-stage. In another words, it is often necessary to provide a high cooling capacity by the precooling stage in addition to a reasonable cooling capacity at 4.2K. This can be achieved by using a single-stage PTR with a precooling GM refrigerator. Gao et al. [6] proposed a hybrid GM/PTR, in which they connected the warm end of the pulse tube to the cold end of the precooling GM refrigerator. Liu et al. [7] numerically analyzed the cooling performance of this type of hybrid refrigerator. However, in their model they neglected the axial heat conduction of the gas and regenerative materials, and the pressure drop in the regenerator. So that it was impossible to obtain satisfactorily quantitative results.

To overcome the shortcomings of GM refrigerators and to improve the performance of PTRs, we recently proposed and designed a new type of 4K GM/PT hybrid refrigerator [8]. In this paper, we briefly summarize our previous work and further optimize the hybrid GM/PTR with respect to the structure parameters and geometrical configuration.

CONFIGURATION

The configuration of the new hybrid GM/PTR using one rotary valve connected to a compressor is shown in Figure 1, which has two stages. The upper warm stage of the refrigerator is a typical GM refrigeration cycle, on which the cold stage of a pulse tube cycle is thermodynamically coupled.
One can see that the new hybrid GM/PTR is not just a PTR precooled by a GM refrigerator. Only one rotary valve is integrated in the GM refrigerator and both stages are operated at a fixed frequency. Thus all gas passages are linked, and the pressure, temperature, and mass flow oscillations share the same frequency. The configuration is different from the hybrid refrigerator described in Ref. [7], where the pulse tube warm end was operated at the first stage cold temperature by use of a thermal bridge. In the present design, we introduced a cold auxiliary piston, which is directly linked to the main displacer, and the pulse tube warm end is connected to the auxiliary cold space. Therefore, analyses of either GM refrigerators or PTRs are not applicable to explain the work mechanisms of the new hybrid refrigerator. Particular attention will be focused on the thermodynamically couple between the upper GM refrigeration cycle and the cold stage of the pulse tube cycle and its phase shifter.

Four different types of phase shifters: (1) a cold auxiliary piston, (2) an orifice with reservoir encircled cold head of the upper stage, (3) the orifice with double-inlet, and (4): the cold auxiliary piston in combination with orifice and double-inlet, were proposed and analyzed. Type 1 is accomplished by means of a cold auxiliary piston directly connected to the main displacer of the upper stage of the GM refrigeration cycle (see Figure 1a). The cold auxiliary piston and main displacer are driven by the same rotary valve and their motions have the same phase. The warm end of the pulse tube and regenerator are connected respectively, to cold auxiliary piston and main displacer, thus the relation between the phases of the cold auxiliary piston motion and pressure in the pulse tube is reversed (180 degree) in type 1. The pulse tube cycle of the hybrid refrigerator can also operate in the orifice model (Type 2) or the double-inlet model (Type 3) located at cold head of the upper warm stage of the GM refrigeration cycle, as shown in Figure 1b. With the goal of enhancing the ability to adjust the phase angle to gain better performance of the pulse tube cycle, the above phase shifting concepts can be used either independently, or in combination with the cold auxiliary piston through skillful design. As a result, other new methods of phase shifters can be also incorporated for further improving the performance of the hybrid refrigerator.

THEORETICAL ANALYSIS

Theoretical analysis and optimization have been carried out by a computer simulation program, which is an extended version developed by the present author [9]. The model is based on 1-D, unsteady compressible gas flow and neglect geometric complexity in the regenerator. We here focus on the low temperature stage. The junction between the cold space of the upper stage and the low temperature regenerator is taken as the left boundary. The right boundary is the inner surface of the phase shifter (the
auxiliary piston or the orifice and reservoir) of the pulse tube. Each method of phase shifters can be embodied by properly dealing with the right boundary condition. For example, when both the auxiliary piston and the orifice are used, the right boundary is movable, and the width of the grid to the left of it is varied accordingly. In addition, the gas velocity through the boundary is determined by the orifice and reservoir. Otherwise, the length of the grid left to the right boundary is constant if the auxiliary piston is not employed, and the gas velocity is zero if the orifice is not applied. Details regarding the numerical constructions and implementations are given in Ref. [8, 9].

RESULTS AND DISCUSSIONS

The simulation program was first applied to analyze the cooling performance of the hybrid GM/PTR with four different types of phase shifters described above. Then it is used to optimize the geometrical configuration. The major design specifications are as follows: the 1st-stage temperature $T_1 = 40K$, the 2nd-stage $T_c = 4.2K$, the high and low pressures $p_H = 2.0MPa$ and $p_L = 0.6MPa$, respectively, and the operating frequency $f = 1Hz$. Without special note the main structure parameters are given in Table 1.

The regenerator is filled with 50% Er$_3$Ni and 50% ErNi$_{0.9}$Co$_{0.1}$ Grain, and its filling factor is 0.6. The cooling capacity as a function of the lowest temperature with four phase shifters is shown in Figure. It is obvious that the cooling performances with only the cold auxiliary piston or the orifice as its phase shifter are not as good as expected. The low temperatures with those phase shifters cannot reach 4.2K. Nevertheless, the performance can be greatly improved in the case when the orifice is combined with the double-inlet or the cold auxiliary piston is combined with the orifice and double-inlet. The best phase shifting assembly in our simulation is of the cold auxiliary piston combines with the orifice and double-inlet. In this case the refrigerator can provide a net cooling power of 0.85W at 4.2K.

Table 1 Main structure parameters of the hybrid GM/PTR

<table>
<thead>
<tr>
<th>Components</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Regenerator: Diameter</td>
<td>29 mm</td>
</tr>
<tr>
<td>Length</td>
<td>160 mm</td>
</tr>
<tr>
<td>Materials</td>
<td>Er$<em>3$Ni+ErNi$</em>{0.9}$Co$_{0.1}$</td>
</tr>
<tr>
<td>Pulse tube: Diameter</td>
<td>15 mm</td>
</tr>
<tr>
<td>Length</td>
<td>160 mm</td>
</tr>
<tr>
<td>Auxiliary piston: Diameter</td>
<td>22 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>25 mm</td>
</tr>
<tr>
<td>Gas buffer volume:</td>
<td>500 cm$^3$</td>
</tr>
</tbody>
</table>

Figure 2  Cooling capacity vs. temperature

Figure 3  Variations of gas temperature profile

Figure 4  Cooling capacity vs. volume ratio of the regenerator to pulse tube
Figure 5 Cooling capacity vs. volume ratio of the cold auxiliary piston to pulse tube

Figure 6 Cooling capacity vs. initial angle of the cold auxiliary piston

Figure 3 shows the transient gas temperature profiles along the regenerator length at three different crank angles of the rotary valve. A region of almost constant temperature appears near the cold end of the pulse tube. The temperature profiles are rather flat in the 40% of the regenerator length to the cold end. About 80–90% of the temperature drop is concentrated in the 40% of the regenerator length near the hot end.

The effects of volume ratio of the regenerator to pulse tube, volume ratio of the cold auxiliary piston to pulse tube, and initial phase angle between the cold auxiliary piston and main displacer, on the cooling capacity at 4.2K are illustrated in Figure 4–6. The cooling capacity increases rapidly with increasing volume ratio of the regenerator to pulse tube (keeping the length constant and altering the diameter of regenerator) (see Figure 4). After reaching a maximum value, the cooling capacity decreases slightly with increasing the volume ratio. The initial phase angle has considerable effects on the cooling capacity at small volume ratio. When the volume ratio becomes larger than 4 the effect is negligible. The influences on the cooling capacity of volume ratio of the cold auxiliary piston to pulse tube at different diameters of pulse tube are given in Figure 5. The refrigerator with the pulse tube diameter of 15mm can achieve a slightly larger refrigeration power than others. One can see (Figure 4 and 5) that the most suitable volume ratio of the regenerator to pulse tube is around 4-5 and the optimal volume ratio of the auxiliary piston to pulse tube is about 0.3-0.35. Figure 6 shows that the effects of the initial phase angle on the cooling capacity are slight in the range of 40–70 degree. From above figures, one can find that the volume ratio of regenerator to pulse tube, the volume ratio of cold auxiliary piston to pulse tube, and the initial phase angle between cold auxiliary piston and main displacer are of three most important parameters which need to be well designed for achieving the better performance of the hybrid refrigerator. It is worth noting that the optimum values of these parameters depend on the structure and size of the refrigerator.

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REFERENCES