Development of high capacity split Stirling cryocooler for HTS

Kenta Yumoto, Kyosuke Nakano and Yoshikatsu Hiratsuka

Technology Research Center, Sumitomo Heavy Industries, Ltd.2-1-1, Yato-cho, Nishitokyo-city, Tokyo 188-8585 Japan

Abstract

Sumitomo Heavy Industries, Ltd. (SHI) developed a high-power Stirling-type pulse tube cryocooler for cooling high-temperature superconductor (HTS) devices, such as superconductor motors, superconducting magnetic energy storage (SMES), and fault current limiters. The experimental results of a prototype pulse tube cryocooler were reported in September 2013. For a U-type expander, the cooling capacity was 151 W at 70 K with a compressor input power of 4 kW. Correspondingly, the coefficient of performance (COP) was about 0.038. However, the efficiency of the cryocooler is required to be COP > 0.1 and it was found that, theoretically, it is difficult to further improve the efficiency of a pulse tube cryocooler because the work flow generated at the hot end of the pulse tube cannot be recovered. Therefore, it was decided to change the expander to a free-piston type from a pulse tube type. A prototype was developed and preliminary experiments were conducted. A cooling capacity of 120 W at 70 K with a compressor input power of 2.15 kW with corresponding COP of 0.056, was obtained. The detailed results are reported in this paper.

Keywords: High-temperature superconductor; Stirling cryocooler; Coefficient of performance; Compressor; Work;

1. Introduction

Technical improvement in the manufacturing process for high-temperature superconducting wires has made it possible to use superconducting technology for various fields such as energy, transportation, and medical. Superconducting motors, superconducting magnetic energy storage (SMES), and current fault limiters, etc. have been developed worldwide. For cooling high-temperature superconductor (HTS) devices, hundreds of watts of cooling capacity at 77 K are needed. Spoor (2013) reported on the experimental results of a commercial prototype...
Stirling-type pulse tube cryocooler (SPTC) with 1-kW cooling capacity at 80-K coaxial cold-head as civilian equipment for HTS electronics applications and Emery et al. (2013) reported on the development of a pulse tube cryocooler with a cooling capacity of 250 W at 77 K at a compressor electricity input power of 3.1 kW. This compressor is driven using the crank shaft method not a linear motor. In September 2013, the experimental results of a high-power Stirling-type pulse tube cryocooler was reported by Hiratsuka et al. (2013) and its cooling capacity was 151 W at 70 K with a compressor input power of 4 kW. Correspondingly, the coefficient of performance (COP) was about 0.038. However, the efficiency of the cryocooler is required to be COP > 0.1, and it was found that, theoretically, it is difficult to further improve the efficiency of a pulse tube cryocooler because the workflow generated from the hot-end of the pulse tube cannot be recovered. Therefore, we decided to change the expander to a free-piston type from pulse-tube type for higher efficiency. The target cooling capacity is 150 W at 70 K with a compressor input power of 2.15 kW COP of 0.07, and the cryocooler is used to cool a superconducting motor. We developed a prototype cryocooler and conducted preliminary experiments. The cooling capacity was 120 W at 70 K with a compressor input power of 2.15 kW COP of 0.056. Moreover, the cooling performance was affected by the difference in connecting tube diameter and the effect of anomalous flow of helium gas from the connecting tube to the expander.

2. General design

Figs. 1 and 2 show a 3-D diagram of a prototype unit and a schematic diagram of the points for measuring temperature, pressure, and displacement, respectively. A split-type Stirling cryocooler was selected for this development because the compressor and expander can be arranged independently. Helium gas is charged in the cryocooler and the initial gas pressure is 1.7 MPa. The compressor consists of a moving magnet-type motor and two opposed pistons driven by the linear motor. In this case, vibration from the pistons can cancel each other out. The pistons are guided by flexure bearings, which can maintain clearance of several micro meters between the pistons and cylinder. The expander consists of a displacer piston, cold-head, regenerator, and warm-end cooler. The displacer piston and regenerator are co-axially arranged. The regenerator is packed with stainless-steel screens. The warm-end cooler in the expander is a shell and tube type and is water-cooled. The displacer piston is also guided by flexure bearings, and a dynamic vibration absorber (DVA) is attached to suppress vibration from the displacer piston. The positions of the pistons in the compressor, \( X_{cp} \), and the displacer piston in the expander, \( X_{dp} \), are measured using laser vibrometers. The pressure at the compressor, at the hot and cold sides of the expander, \( P_{cp} \), \( P_{ea} \) and \( P_{ec} \), are measured using pressure sensors. The pressure-volume (\( P-V \)) work of each space in the cryocooler, \( W \), is calculated as follows.

\[
W = \int f \cdot P \cdot A dX
\]

where \( f \) is the operating frequency, \( P \) is the pressure in each space, \( A \) is the cross-sectional area of the pistons in the compressor or the displacer piston in the expander, and \( X \) is the position of the pistons in the compressor or that of the displacer piston in the expander.

The amount of heat dissipation in the warm-end cooler \( Q_{cl} \) is calculated as follows.

\[
Q_{cl} = C \cdot G_{cl} \cdot \Delta T
\]

where \( C \) is the specific heat of water, \( G_{cl} \) is the water-flow rate, and \( \Delta T \) is the temperature difference between the inlet and outlet water. Heaters and PtCo sensors are mounted on the cold-head to measure the cooling capacity at each temperature. Thermo couples are attached to the outer wall of the regenerator to measure the temperature distribution.

The pressure loss through the connecting tube \( W_{ct} \) is calculated as follows.

\[
W_{ct} = \frac{1}{2} \cdot \left( P_{cp} + P_{ea} \right) \cdot d \cdot \left( V_{cp} - V_{ea} \right)
\]

where \( V_{cp} \) is the volume of the space in the compressor, and \( V_{ea} \) is the volume at the hot side of the expander.
3. Experimental results and discussions

Table 1 lists the operation conditions and experimental results. The compressor $P-V$ work was 1898 W, the expander cold side P-V work was 231 W, and the measured heat dissipations at the compressor cooler and the warm-end cooler were 222 W and 1744 W, respectively. The calculated pressure loss through the connecting tube was 174 W. Fig. 3 shows the efficiency and cooling capacity at 70 K with respect to the compressor input power, and Fig. 4 shows the cooling capacity and COP with respect to the cold-head temperature.

| Item                                | Value                |
|-------------------------------------|----------------------|
| Cold-head Temperature               | 70 K                 |
| Operating frequency                 | 48.5 Hz              |
| Initial gas pressure                | 1.7 MPa              |
| Input power                         | 2.15 kW              |
| Cooling water inlet temperature     | 20 ºC                |
| Water flow rate for compressor cooler | 0.5 l/min           |
| Water flow rate for expander cooler | 9.9 l/min            |
| COP                                 | 0.056                |
| Cooling capacity at 70 K            | 120 W                |
| Compressor P-V work                 | 1898 W               |
| Expander cold side P-V work         | 231 W                |
| Heat dissipation in compressor cooler | 222 W            |
| Heat dissipation in expander cooler | 1744 W               |
| Pressure loss through connecting tube | 174 W                |
3.1 Temperature distribution of regenerator

Spoor (2013) reported on the effect of anomalous flow of gas by the side-entry connecting tube. The deviation of flow changes the temperature distribution within the regenerator and causes a decline in cooling capacity. To measure temperature distribution at the regenerator, thermo couples are attached to the outer wall of the regenerator cylinder. They are attached 20 and 35 mm from the low-temperature side of regenerator and arranged every 90 degrees in the direction of the circumference (Fig. 5). Fig. 6 shows the experimental results when the cold-head temperature was 70 K. The temperature distribution of the regenerator wall is also shown. Line A is the temperature of the connecting tube side, and line C is the opposite side. The temperature of the connecting tube side was the higher of the two measure points in the cylinder axis direction. These experimental results are different to those from Spoor’s experiment. The reason of this difference may be due to the difference in the flow path and existence of the displacer. The temperature distribution was not symmetry to the connecting tube axis. The temperature distribution in the circumference direction indicates a potential gas flow deviation within the regenerator. In the future, this anomalous flow within the regenerator will be investigated using fluid analysis and gas flow controlled for improving cooling capacity.
3.2 The effect of the connecting tube configuration

We investigated the effect of the connecting tube configuration on cooling performance. Fig. 7 shows the cooling capacity at 70 K with a compressor input power of 2.15 kW and pressure loss through the connecting tube with respect to the inner diameter of the tube. The tube was a 500-mm-long straight stainless-steel tube. The pressure loss decreased and the cooling capacity increased as the inner diameter increased. However, the increase in cooling capacity became relatively smaller when the inner diameter was larger than 25 mm. Moreover, the lozenge plots in Fig. 7 correspond to the data with a straight flexible tube of the same length as a stainless-steel tube, and there was no effect of the different tubes on cooling performance. Moreover, we examined the effect on the cooling capacity when the tube was bent. Fig. 8 shows photograph of the cryocooler with flexible connecting tubes having 90-degree bends, and Table 2 shows the cooling capacity with bent connecting tubes. Even if a bent connecting tube is used, there is no significant difference in cooling capacity.

3.3 Evaluation of vibration suppression with dynamic vibration absorber

The expander receives force from the displacer and generates vibration on the cold-head. For specific application of this cryocooler, such a vibration should be suppressed as much as possible. Fig. 9 shows the strokes of the expander’s vibration with and without a dynamic vibration absorber (DVA). Vibration of the expander at the operation frequency can be reduced to about half by adjusting the natural frequency of the dynamic vibration absorber.
4. Conclusion

We developed a prototype split Stirling cryocooler and obtained a cooling capacity of 120 W at 70 K with an input power of 2.15 kW COP of 0.056. The target of our development was to obtain a cooling capacity of 150 W at 70 K and a COP of 0.07. We plan to analyze the losses in the cryocooler to further improve the COP. High mechanical reliability and easy maintenance are required while cryocoolers are actually used for cooling HTS devices. Furthermore, for cooling a superconductivity motor, it will be necessary to develop a cryocooler that can be used under severe conditions, such as high vibration and high ambient temperature. Such issues will be considered as a part of our future development plan.

Acknowledgements

This work was supported by Strategic Innovation Program for Energy Conservation Technologies Project of the New Energy and Industrial Technology Development Organization (NEDO) of Japan and a joint research with Sumitomo Electric Industries, Ltd.

References

Spoor, P. S., 2013. Anomalous Temperature and Amplitude-dependent performance characteristic of 1000W/80K Coldfinger, Cryogenic Engineering Conference and Cryogenic Materials Conference in 2013, Anchorage, Alaska, USA, pp. 1405-1409
Emery, N., Caughley, A., Meier, J., Nation, M., Tanchon, J., Trollier, T. and A.Ravex, A., 2013. Large Co-axial pulse Tube Preliminary Results, Cryogenic Engineering Conference and Cryogenic Materials Conference in 2013, Anchorage, Alaska, USA, pp. 1399-1404
Hiratsuka, Y., Nakano, K. and Kato, 2013. T., Recent Development Status of Stirling Type Pulse Tube Cryocooler for HTS, Journal of Physics:Conf. ser. 507