Development of high-efficiency Stirling cryocoolers for high temperature superconducting motors

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Abstract. For wide spread high-temperature superconductor (HTS) devices, a cryocooler having COP of >0.1, with a compact size, light weight, high efficiency and high reliability is required. For practical use of superconductive devices, Sumitomo Heavy Industries, Ltd. (SHI) developed a high-efficiency Stirling type pulse tube cryocooler (STPC). The STPC had high reliability and low vibration. However, its efficiency was not enough to meet the demands of an HTS motor. To further improve the efficiency, we reconsidered the expander of cryocooler and developed a Stirling cryocooler (STC). Two prototype units of a compact, high-efficiency split Stirling cryocooler were designed, built and tested. With the second prototype unit, a cooling capacity of 151 W at 70 K and a minimum temperature of 33 K have been achieved with a compressor input power of 2.15 kW. Accordingly, COP of about 0.07 has been achieved. The detailed design of the prototype units and the experimental results will be reported in this paper.

1. Introduction

As a part of the New Energy and Industrial Technology Development Organization (NEDO) project, SHI has been developing a prototype HTS motor system in joint research with Sumitomo Electric Industries, Ltd. since 2013. This system is designed to be used for an electric vehicle, specifically a city bus.

In comparison with a conventional electric vehicle, it is expected that the power consumption can be reduced by 20% with an HTS motor system. The latter consists of an HTS motor, a cooling system, a vacuum chamber and a cryocooler. The HTS motor is cooled by a liquid nitrogen circulation system and kept at about 70 K. In order to re-cool liquid nitrogen, a compact and high-efficiency cryocooler is required. SHI has been developing high-efficiency cryocoolers since 2010. In the early stages of development, STPCs were developed. For an in-line type STPC, a cooling capacity of 175 W at 70 K, COP 0.046, was achieved with a compressor input power of 3.8 kW [1]. For a ‘U’ type STPC, a cooling capacity of 145 W at 70 K, COP 0.038, was achieved with the same amount of input power [2].

Figures 1 and 2 show the prototype units of these STPCs. Figure 3 shows the cooling capacity and COP with respect to the cold head temperature with a compressor input power of 3.8 kW. The ‘U’ type STPC was more compact than the in-line one, while the cooling capacity and COP were slightly degraded. As a matter of fact, the COPs of these two STPCs differ greatly from the target, COP 0.1. In order to improve the COP of the cryocooler, we studied energy flow and loss analysis in the expander of the STPC [3]. As a result, we found that an effective way to further improve the efficiency is to recover the energy of the pulse tube hot end, which is dissipated in the inertance tube. Therefore, we
decided to change the cryocooler from a pulse tube to a free-piston type Stirling to achieve the final target.

![Figure 1. Photo of an in-line type STPC.](image1)

![Figure 2. Photo of a ‘U’ type STPC.](image2)

![Figure 3. Comparison of the cooling capacity and COP of two STPCs.](image3)

2. General design of a split Stirling cryocooler
Figure 4 shows a 3-D model of a split STC. The compressor is the same as that used for the previous STPCs. A COP of 0.07 is set as the first target. A split Stirling cryocooler is selected for easier installation and lower vibration. The compressor is connected to the expander with a 750 mm U-tube. The displacer is guided by flexure bearings, which can maintain a clearance of several micro meters between the displacer and the cylinder. The regenerator is stacked with a few kinds of SUS wire meshes. The cooler is a shell and tube type heat exchanger and it is cooled by 30 °C water. On the expander, a dynamic vibration absorber is attached to suppress the vibration.

First, the cooling capacity and the energy flow rate were analyzed. Figure 5 shows schematic diagrams of the measurement points in the cryocooler. The strokes of the compressor pistons, \( X_{cp} \), and
the displacer, $X_{cp}$, are measured by laser vibrometer. The pressures in the compressor discharge head, $P_{cp}$, in the back spaces of the compressor pistons, $P_{cb}$, in the hot-end space of the expander, $P_{ea}$, and in the buffer of the expander, $P_{eb}$, are all measured by pressure sensors. Pressure-Volume (P-V) work of each space in the cryocooler is calculated by operating frequency, pressure, cross-sectional area of the compressor piston or the expander displacer, and the position of the compressor piston or the expander displacer.

The amount of heat dissipation at the expander cooler, $Q_{cl}$, and heat generation of the compressor, $Q_{cp}$, is calculated by specific heat of water, mass flow rate and $\Delta T$. The latter is the difference between the inlet and outlet water temperatures, which are measured with Pt100 sensors.

The pressure loss through the connecting tube, $W_{ct}$, is calculated as,

$$W_{ct} = f_o \cdot (1/2) \cdot f \cdot (P_{cp} + P_{ea}) \cdot d(V_{cp} - V_{ea})$$

(1)

where $V_{cp}$ is the volume of the compressor discharged head and $V_{ea}$ is the volume of the expander hot-end space.

In order to analyse the copper loss and the iron loss of the compressor, the current and coil resistance of the linear motor, $R_{cp}$, are measured with a wire-wound resistor. Then, the power consumption at a stalled state was measured while current was supplied. From the difference in power consumption between a normal state and a stalled state, the copper loss, $Q_{cc}$, and the iron loss, $Q_{ci}$, were calculated.

A heater and a PtCo sensor were mounted on the cold head for measuring the cooling capacity $Q_{ch}$. In addition, in order to reduce the radiation loss, a few layers of Super-Insulation (SI) were wrapped on the cold head.

![Figure 4. 3-D model of a split STC.](image)

![Figure 5. Schematic diagrams of measurement points in a STC.](image)
3. Experimental results of the first prototype unit of a split STC and energy flow analysis

The resonant frequency of the displacer is optimized to acquire an optimum performance by changing the weight of the displacer balancer or the spring constant.

The operating conditions and the experimental results of the first prototype unit are shown in Tables 1 and 2. A cooling capacity of 125 W at 70 K, COP 0.058 with an input power of 2.15 kW, was obtained. The compressor P-V work (work-flow) was 1780 W, the compressor back space P-V work was 81 W, the expander hot end space P-V work was 200 W, and the expander buffer P-V work was 0.8 W. The heat rejections in the compressor heat exchanger and the expander cooler were 260 W and 1780 W, respectively. The calculated pressure loss through the connecting tube was 60 W. The copper and iron losses of the compressor were 160 W and 80 W, respectively. Based on these results, the energy distribution in the prototype unit is shown in Table 3 and Figure 6 shows the energy flow diagrams.

The input power of 2150 W includes the compressor P-V work, copper loss, iron loss, back space P-V work, and mechanical loss of moving parts. The latter was about 50 W. The compressor efficiency was 0.83. The generated work-flow, \( W_{cp} \), decreased by 80 W in the connecting-tube and flew into the expander hot-end space. In the expander hot end space, \( W_{ea} \), which is the recovered work-flow by the displacer, was measured to be 200 W. The cooler rejected heat flow \( Q_{cl} \), and the difference between the sum of \( W_{cp} \), \( W_{ea} \) and \( -W_{ct} \), and \( Q_{cl} \) was calculated as \( H_{re} \). The latter is the expander heat loss, including the regenerator heat transfer loss, thermal conduction loss, displacer pumping gas loss and shuttle loss, which was 115 W. In addition, the cold end work flow, \( W_{ch} \), was carried out approximately 240 W from the energy balance.

**Table 1.** Operating conditions.

| Element                              | Symbol | Value  |
|--------------------------------------|--------|--------|
| Operating frequency                  | \( F_0 \) | 47.0 Hz |
| Temperature of cold head             | \( T_{ch} \) | 70 K    |
| Mean pressure                        | \( P_m \) | 1.7 MPa |
| Input power                          | \( W_{in} \) | 2150 W  |
| Temperature of cooling water         | \( T_{wi} \) | 30 °C   |
| Amount of water flow at cooler       | \( G_{cl} \) | 12.1 L/min |

**Table 2.** Experimental results and loss analysis result.

| Element                                           | Symbol | Value  |
|---------------------------------------------------|--------|--------|
| Cooling capacity and COP at 70 K                  | \( Q_{ch} \) | 125 W / 0.058 |
| Compressor P-V work                               | \( W_{cp} \) | 1780 W  |
| Compressor back space P-V work                    | \( W_{cb} \) | 81 W    |
| Expander hot end P-V work                         | \( W_{ea} \) | 200 W   |
| Expander buffer P-V work                          | \( W_{eb} \) | 0.8 W   |
| Pressure ratio at expander hot end                 | \( P_{ea MAX} / P_{ea MIN} \) | 1.28 |
| Heat rejection at compressor heat exchanger       | \( \delta Q_{cp} \) | 252 W   |
| Heat rejection at expander cooler                  | \( \delta Q_{cl} \) | 1805 W  |
| Compressor copper loss                            | \( Q_{cc} \) | 160 W   |
| Compressor iron loss                              | \( Q_{ci} \) | 80 W    |
Table 3. Results of loss analysis

| Element                              | Symbol | Value |
|--------------------------------------|--------|-------|
| Compressor mechanical loss           | $Q_{cm}$ | 50 W  |
| Connecting tube pressure loss        | $W_{ct}$ | 60 W  |
| Heat loss of expander                | $H_{re}$ | 115 W |

Figure 6. The schematic diagrams of the energy flow in STC.

In summary, the first prototype unit did not reach our first target and further improvement was needed. Especially, both $W_{ea}$ and $W_{ch}$ were lower and the regenerator flow resistance loss was higher than what we expected. The reason is that the compressor was designed and optimized for an STPC, and thus the compressor resonance frequency deviated from its optimum. Therefore, it is considered that the mean pressure and the regenerator size should be optimized to match the compressor, and the pressure ratio was too low to obtain high work flow.

Because of time constraints on the project, only the expander was improved. Next, the cooling performance was further improved by optimizing the regenerator ratio, clearance of displacer, heat transfer coefficient of low temperature heat exchanger and by reducing the conduction loss.

4. Experimental results the second prototype unit

Next, the second prototype unit, in which the heat loss was reduced, will be reported. Figure 7 shows the efficiency and cooling capacity at 70 K with respect to the compressor input power. Figure 8 shows the cooling capacity and COP with respect to the cold head temperature. Figure 9 shows the cooling capacity and COP with respect to the cooling water temperature. Figure 10 shows the cooling capacity with respect to the cold head inclination. From the experimental results, it is obvious that total losses have been greatly reduced. As a result, a cooling capacity of 151 W at 70 K, corresponding COP 0.07, was achieved with an input power of 2.15 kW. A no-load temperature of 33 K was reached. Also, the effect of cooling water temperature and inclination on cooling capacity and COP was very small. Accordingly, the developed STC can withstand environmental conditions in a vehicle.
Figure 7. The efficiency and cooling capacity at 70 K with respect to the compressor input power.

Figure 8. The cooling capacity and COP with respect to the cold head temperature.
5. Vehicle test
The HTS motor system is characterized by high torque and efficiency at a low motor rotation. In order to take advantage of an HTS, the prototype HTS vehicle is supposed to be used in a city bus. The designed torque is 650 Nm and the designed power is 100 kW, respectively. The cryocooler is required to operate continuously 24 hours a day. The cryocooler is controlled by an inverter to change the output according to the load of the motor. The developed STC in this project was installed in an HTS motor system and a trial running on a test course was conducted. So far, no failure or performance degradation in the HTS vehicle environment was observed. The details of the system and results are going to be reported in the International Workshop on Cooling-system for HTS Applications (IWC-HTS) in Matsue, Japan.

The first target of the cryocooler has been achieved. However, there are still many problems and subjects to be solved before the HTS motor is ready for practical use. As for the cryocooler, more progress in, not only the cooling performance, but also the vibration proof, size minimization and reliability, are required, and need to be continuously researched.

6. Conclusion
Two prototype units of a split STC have been developed and the losses in a STC were analyzed. As a result, a cooling capacity of 151 W at 70 K, COP 0.07, was achieved with an input power of 2.15 kW. A prototype unit of the developed STC was installed in an HTS motor system of an electric vehicle, and a trial running test was conducted. The prototype unit operated without failure or obvious performance degradation. However, for typical use of an HTS motor, higher performance and reliability of the cryocooler, which can be used under severe conditions, such as vibration and temperature in a vehicle, is required. Based on these experimental results, in order to further improve the efficiency of a cryocooler, the size, settings and the impedances matching approaches of the compressor and expander, will be further optimized.
References

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