Development of a Miniature Pulse Tube Cryocooler of 2.5W at 65K for Telecommunication Applications*

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Abstract
The Fuji Electric Group has established main technologies with high reliability for use in Stirling cryocoolers for space satellite systems. For commercial applications, we also have developed and started selling a miniature pulse tube cryocooler from 2W to 3W at 70K with 100W electric power input. In the development of a new compressor, we introduce a moving magnet to a driving system to achieve greater compactness and higher efficiency in place of the moving coil that had about 70% efficiency. In addition, we adopted a coaxial pulse tube as an expander for compactness. This development is aimed at cooling a high-temperature superconductive (HTS) device in a wireless telecommunication system. The compressor requires total compression work of 75W with 90% efficiency for longer than 50,000 hours. Preliminary tests of each part of a moving magnet linear motor and a coaxial pulse tube have been completed. In the next phase, we have made a first-stage prototype compressor used by the new linear motor, and we have tested the new machine. Here we describe each test and combination test results of the cryocooler.

Key words: Cryocooler, Stirling, Pulse Tube, Linear Compressor, Moving Magnet

1. Introduction

Wireless communication has rapidly become widespread in recent times, which has brought about a shortage in the regions of electromagnetic wave frequency and created a need for measures to meet the situation. It is well known that one of these measures is the application of a superconducting filtering system. The system for a receiver is already in practical use, but one for a transmitter remains unexploited. Therefore, a project to develop a superconducting filtering system for a transmitter was begun in December 2005 and is scheduled to last until March 2008. The project is funded by a Japanese government institution, the Ministry of Public Management, Home Affairs, Posts and Telecommunications. A cryocooler for that system requires compactness, high efficiency and a long lifetime. The requirements are better performance than that provided by any previously developed cryocooler (1).
2. Cryocooler Designing

2.1 Motor Designing Concept

The target specifications for developing a compressor are shown in Table 1. This target was too high for a conventional moving coil motor to achieve, especially at an efficiency of 90%.

We started by developing a moving magnet linear motor with the potential of higher efficiency and greater compactness than that of current motors. To create higher performance and increase the generative force, a moving part needs to combine a main permanent magnet and two side magnets, as shown in Fig.1. The moving part becomes stable at the center of each leg of an outer yoke in this magnetic configuration because each magnetic circuit between a main magnet and one side magnet makes a loop through each leg of the outer yoke. If the moving part becomes displaced from a stable state to the left or right side, magnetic potential energy is stored. In other words, a force is caused in the opposite direction against that generated by the exciting current.

2.2 Motor Basic Equation

Fleming’s left-hand rule consists of defining an equivalent current for a generative force. Using an equivalent current of a permanent magnet, the generative force is described as follows:

\[ F = B_g \times 2I_M \times L \]  
(1)

where \( B_g \) is the magnetic flux density of the air gap between an outer yoke and an inner yoke, and \( L \) is the average peripheral length of a permanent magnet. This equation differs from a general \( BIL \) equation in that an equivalent current is twice. This is because the equivalent current exists on two side of a magnet. The equivalent current \( I_M \) is defined in the following equation:

\[ I_M = H_{CB} \times L_M = \frac{1}{\mu_0 \mu_r} \frac{B_r}{L_M} \]  
(2)

where \( H_{CB} \) is the magnetic coercive force, \( L_M \) is the magnetization direction thickness, \( B_r \) is the remanent magnetic flux density, \( \mu_r \) is the recoil relative permeability of a permanent magnet and \( \mu_0 \) is the permeability of the vacuum.

It is required that \( B_g \) or \( I_M \) is increased to bring about an increase in \( F \) without increasing the size of the motor, as is clear from Eq.(1). If you want to increase \( B_g \), you have to supply more current, but that leads to a decreased efficiency of the motor. There are two methods to increase \( I_M \). One is to use a stronger magnet. The other is to add two side magnets to a main magnet, which means that the number of hypothetical coils is increased. This method shows that an equivalent current is twice as great, as is shown in the following equation:

\[ F = B_g \times 4I_M \times L = 4B_g \frac{1}{\mu_0 \mu_r} \frac{B_r}{L_M} L \]  
(3)

Table 1  Target Specifications of the Compressor

<table>
<thead>
<tr>
<th>Type</th>
<th>Output Power</th>
<th>Motor Efficiency</th>
<th>Dimension</th>
<th>Lifetime</th>
</tr>
</thead>
<tbody>
<tr>
<td>Opposite Piston</td>
<td>75W</td>
<td>90%</td>
<td>Less than ( \phi \ 88 \times 195 ) mm</td>
<td>Over 50,000 hr</td>
</tr>
</tbody>
</table>
2.3 Pulse Tube Designing Concepts

The pulse tube expander was adapted to the coaxial return type under serious consideration of small size, light weight and simple installation to cooling objects. The target specifications are shown in Table 2, and an outline constitution of the expander is shown in Fig. 2.

To make the flow inside the pulse tube have as little turbulence as possible, we arranged the regenerator in the outer tube to straighten the flow at both ends of the pulse tube.

Table 2  Target Specifications of the Expander

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling Power</td>
<td>2.5W at 65K</td>
</tr>
<tr>
<td>Input Compression Work</td>
<td>80W</td>
</tr>
<tr>
<td>COP</td>
<td>3.0% at 65K</td>
</tr>
</tbody>
</table>

Fig. 1  Pattern Diagrams of a Moving Magnet

Fig. 2  Outline of a Coaxial Pulse Tube
3. Results and Discussion

3.1 Compressor

The system examination and the principle verification test of the linear motor were executed previously\(^2\). As a result, it was verified that the moving magnet type compressor of a side magnet system is effective for achieving small size and high efficiency. The first test model of the compressor, adopting this system, has been produced and executed the topic extraction. In the current phase of our study, the compressor itself was made on an experimental basis.

Figure 3 shows the external view of the compressor that has been tested and evaluated. It also shows the size of the motor, excluding the case of the compressor container. As the magnetic circuit of the stator, a small gap is formed between the outer yoke and the inner yoke, and the magnet as the moving part is inserted in the inner yoke. The flexure bearing, which has some spiral-shaped slits, is attached at both ends of the moving part; therefore, the moving part is held by the stator in order to control the possibility of its moving only in the axial direction. The outer yoke is formed by the laminated electromagnetic steel sheets arranged in a radial pattern, and the exiting coil is contained in the outer yoke. The moving part constitution is that the 4 neodymium magnets (ring shaped in- and outside lap monopole) are inserted into the nonmagnetic metal magnet holder. As for the magnetization direction of each magnet, the outer circle is arranged in a S-N-N-S pattern. The two peripheral N polar magnets of the main magnet are divided because of the restriction of the magnetizing length; there is no principle necessity for this to be the case.

Installing the standard expander, we acquired the unit characteristics of the compressor. The principal characteristics result of the compressor single unit is shown in Table 3. Concerning the pressure case and movable stroke, the experimental model has become larger. Next, the static characteristics are shown in Fig.4. The generative force and magnetic restoring force are shown, respectively, in Fig.4. In the figure, the measurement generative force \( F_{\text{all}} \) and the measurement restoring force \( F_r \) are both decreased almost linearly. The exciting force \( F_i \) showed 71.9N of the neutral position and satisfied the design goal of 70N. In addition, the falling rate of the exciting force \( F_i \) became low when a moving part is displaced; it was 12\% \((F_i=63.4 \text{ [N]})\) at the time of 4mm displacement. We also found that the non-linearity of the magnetic restoring force was improved. By the time it is a rated stroke (3.5mm), the magnetic restoring force was 26.9N and the spring constant was 7.69N/mm.

Higher rigidity of the support spring is necessary for the motor of this system; in fact, the magnetic force in the radial direction is large in comparison with the usual moving magnet type. In the simple model, the radial direction force that occurred according to the shift length of the magnet (the moving part) to the yoke was calculated with the three-dimensional magnetic field analysis. The section typical figure of the simple model is shown in Fig.5. This model was made as a simple cylindrical model; therefore, the inner and outer yoke are locked concentrically. The force in the X direction was calculated when the magnet was shifted in the X direction. The result is shown in Fig.6. As stated above, 1-MAG is composed of the main magnet only, and 2-MAG formed to be the side magnet type (the latest trial manufacture motor.)

In both conditions, it was indicated that the lower convex monotone also increases when the shift quantity increases. The magnetic force of 2-MAG became approximately 4.5 times larger than one of 1-MAG. The difference is quite large, since, for example, when there is a shift length of 0.1 mm, the radial direction magnetic force in 1-MAG is 7N while 2-MAG is 32N. That is, 2-MAG (side magnet type) can increase the generative force; however, radial direction magnetic force also increases. Therefore, careful handling is necessary when we use a side magnet type motor. In the case of a side magnet, it is necessary to strengthen the rigidity of its support spring because the radial direction
magnetic force is larger than the conventional type. Next, it is necessary to decrease the shift length of the magnet and cylinder as much as possible, to decrease the occurrence force itself. Based on that analysis result, we designed a flexure bearing.

This experimental model has not yet attained the optimum operational frequency of 50Hz, because resonance of the machine vibrating system is not taken into consideration. To assure the resonance of the machine vibrating system simply, an operational frequency test was executed. The result is shown in Fig.7. We found that both the compression efficiency and the power factor reach their highest values in the 65Hz vicinity. Based on 65Hz, the highest compression efficiency of 83% and the highest power factor of 94% are obtained. Principally, a causal relation doesn’t exist between mechanical resonance and electromagnetic resonance; therefore, we could view this attainment of the highest values of compressed efficiency and power factor at 65Hz as a coincidence. When only the simple unit efficiency of the compressor is pursued, although there is no problem with the frequency of 65Hz, there is a tendency for the frequency to increase while the expander decreases. A future challenge will be to find a compatible design for both mechanical resonance and electromagnetic resonance at a lower frequency than that of the present condition.

Table 3  Achievements of the Compressor

<table>
<thead>
<tr>
<th>Achievement</th>
<th>Rating Generative Force</th>
<th>Compression Efficiency</th>
<th>Dimension</th>
<th>Piston Stroke</th>
</tr>
</thead>
<tbody>
<tr>
<td>Achievement</td>
<td>72N</td>
<td>83%</td>
<td>(\phi 78 \times \phi 176) mm (Excepting Cases)</td>
<td>Rate : (\pm 3.5) mm (Max (\pm 5.0) mm)</td>
</tr>
</tbody>
</table>

Fig. 3  Compressor Appearance

Fig. 4  Static Characteristic Curve of Generative Force and Magnetic Spring
Fig. 5  Analysis Model of a Moving Magnet

Fig. 6  Distribution of Magnetic Force in the Radial Direction

Fig. 7  Characteristics of Compression Efficiency vs. Driving Frequency
3.2 Expander

To realize a small and lightweight expander, we proved the possibility of actualization by the principle verification test of the coaxial return configuration. This time, we designed the principal element of the expander, and we determined its fundamental characteristics during an experiment. The optimization of this data and component were assured, and then the first experimental model of the pulse tube was produced and the efficiency was verified experimentally. Using our knowledge of the expander’s fundamental characteristics, we completed the extraction of technical details.

The expander is comprised of a pulse tube, a regenerator, some heat exchangers and a phase shifter which is consist of an inertance tube and a buffer tank. In a preliminary test using the coaxial pulse tube as the experimental basis, it was possible to verify the form principally, but the cooling power continued to be inferior to the U-return type. To solve this matter, we first addressed the dysfunction of the straightener located at the regenerator and at the end of the pulse tube. The simulation of the flow at the pulse tube end was executed, and the result suggested that an eddy occurs near the hot end and the cold end in the pulse tube. Therefore, we re-modified the design of this straightener and then re-designed the element after the verification of the flow behavior in advance through another simulation. Next, we re-produced the experimental sample. Figure 8 shows the entire photograph when executing performance evaluation. Figure 9 shows the experimental result after improving the hot and cold side straighteners. As this figure, the compression work has shown the relationship of the cooling power for the cold head with 80 W PV-work. As for the improved the cold side straightener section, the cooling power was improved 8%. In addition, the hot side straightener showed a 27% improvement in cooling power (preliminary test model standard) (3).

To improve cooling efficiency, we attempted to optimize the constitution parameters. With the advanced simulation examination, the typical conditions that would improve the cooling efficiency were calculated and compared to the previous modification, and then we appraised the results. Namely, ① the diameter of the regenerator was increased, ② the length of the regenerator was shortened, and ③ the pulse tube was lengthened. For those three conditions, the experimental sample was produced, and cooling efficiency was verified. The result is shown in Fig.10.

The sample with the larger-diameter regenerator (condition ①) showed improved cooling efficiency. However, cooling efficiency remained the same under condition ③ and decreased under condition ②. As the primary factor, we may consider the dysfunction of the regenerator which is supposed to extend to total length of a regenerator. We therefore continue to work on improving the gas flow of the regenerator, and this part of the project is on schedule.

From the preliminary test model, the improvement of the hot and cold side straighteners and constitution parameter made it possible to decide the constitution that satisfies our goal (COP3%) as a primary experimental model. For the next phase, we have targeted greater efficiency, so we will be pursuing improvement of the regenerator material and structure at both ends.

<table>
<thead>
<tr>
<th>Achievement</th>
<th>Achievement</th>
</tr>
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<tbody>
<tr>
<td>Cooling Power</td>
<td>2.5W at 65K</td>
</tr>
<tr>
<td>Input Compression Work</td>
<td>79.4W</td>
</tr>
<tr>
<td>COP</td>
<td>3.1% at 65K</td>
</tr>
</tbody>
</table>
Fig. 8  Pulse Tube Appearance

Fig. 9  Load Curve by Different Straighteners

Fig. 10  Comparison of Cooling Characteristic with Different Tube Dimensions
3.3 Combination Test

The combination appraisal has been executed for the moving magnet type compressor and the coaxial pulse tube that were made on an experimental basis. The appraisal test result and the topic of the secondary experimental model are described in the following paragraphs.

To obtain stable experimental data, we inserted a test cooler into an oven with a constant temperature of 25°C and then performed the test under the condition of water cooling heat dissipation (water temperature 25°C) for both the compressor and the expander. Under these conditions, total resonance between the expander and the compressor was not optimized, so we decided to optimize by simply changing the compressor frequency. The principal characteristic results in the achievement of the primary trial manufacture are shown in Table 5.

Under the present test conditions, a cooling power of 2.5W at 65K was achieved. Total COP (= cooling power / electric power) at the present stage is 2.0% which is shorter than the goal of 2.2%; however, we obtained the prospect of 2.3% COP if we can improve the production method of the compressor. Decreasing the electric power consumption and optimizing the miniaturization are future matters. In this trial manufacture, optimum frequency of the expander and the compressor are different. So a cooling test was executed in the optimum frequency vicinity of each element. The electric power consumption of 2.5W at 65K and the relationship to cooling power are shown in Fig.11. In the figure, the operational frequency with maximum efficiency was 63Hz, and the maximum efficiency was 2.0% (2.5W at 65K). The frequencies of the highest efficiency of each element are that the expander is 50Hz, and the compressor is 65Hz. This is the reason why there is the optimum frequency of this cooler exists in between the two frequencies. It meant that compressor efficiency is more influenced by the frequency. In this combination test, the quality of each element (especially the expander) is bad in comparison with the single unit test results. The main reason is probably the exacerbation of a leak in the clearance seal caused by the compressor, because of a production problem(4).

Next, we presume that the cooling efficiency is increased by a decrease in clearance seal leakage. The previously expressed seal leak loss of 13.3W was calculated as shown in Fig.12, and it shows the breakdown of the electric power consumption driven by optimum frequency 63Hz in the figure. According to the figure, the total COP of 2.0% in the current experimental model becomes 2.3% prospectively, when the seal leak is fixed. The improvement of the seal leak and the actualization are feasible, since the production process of the compressor can be changed easily and is the only requirement for the solution. Therefore, we consider that it is close to the achievement level of cooling efficiency under the present conditions. To achieve 2.5W at 65K by electric power consumption of 100W, matching of the expander and further efficiency improvement of each element will be necessary.

<table>
<thead>
<tr>
<th>Table 5  Achievements of the Cooler</th>
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<tbody>
<tr>
<td><strong>Achievement</strong></td>
</tr>
<tr>
<td>Cooling Power</td>
</tr>
<tr>
<td>Electric Power</td>
</tr>
<tr>
<td>COP of Cooler</td>
</tr>
<tr>
<td>Occupied Volume</td>
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</table>
4. Conclusion

We performed a single unit test of the moving magnet type compressor that uses a side magnet and a coaxial pulse tube expander and a combination test of both. We obtained the following conclusions:

(1) The side magnet type compressor satisfied the generative force 72N and design specification 70N, achieved a maximum single until efficiency of 83%.

(2) The coaxial pulse tube expander achieved a cooling power of 2.5W at 65K, and this satisfied the COP 3.1% target specification.

(3) In this experimental model, COP 2.0% was achieved, and the prospect that 2.3% could be achieved with improvement of the piston seal leak was obtained.

Our goal is further miniaturization in the future.
Acknowledgment

This work was supported in part by the Ministry of Internal Affairs and Communications (MIC) of Japan through a program whose title is "Research and development of fundamental technologies for advanced radio frequency spectrum sharing in mobile communication systems".

References