Development of a linear compressor for compact 2 K Gifford-McMahon cryocoolers

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Abstract. Recently, a new, compact Gifford-McMahon (GM) cryocooler for cooling superconducting single photon detectors (SSPD) has been developed at Sumitomo Heavy Industries, Ltd. (SHI) [1, 2]. The objective is to reduce the total height of the expander by 33% relative to the existing RDK-101 GM expander and to reduce the total volume of the compressor unit by 50% relative to the existing CNA-11 compressor. In addition, considering the targeted cooling application, we set the design temperature targets of the first and the second stages to 1 W and 20 mW of heat load at 60 K and 2.3 K, respectively. Although optimization of the internal components is one way to miniaturize the volume of the compressor unit, major design changes are required because the volume of the adsorber and the oil separator is almost the same as the volume of the compressor capsule. Thus, one approach is to develop a non-lubricated compressor, such as a valved linear compressor. An experimental unit of a valved linear compressor was designed and built, and preliminary experiments were conducted. Under no-load condition, a low temperature of 2.19 K has been achieved. With 1 W and 14 mW heat load, the temperature is 48 K at the first stage and 2.3 K at the second stage, with an input power of about 1.2 KW. The detailed experimental results will be discussed in this paper.

Nomenclature
\( A \) Area
\( H \) Enthalpy flow
\( d \) Diameter
\( f \) Frequency
\( t \) Length
\( m \) Mass flow rate
\( M \) Mass
\( P \) Pressure
\( Q \) Heat flow
\( R_g \) Gas constant
\( T \) Temperature
\( v \) Velocity
\( V \) Volume
\( W \) P-V work
\( X \) Displacement of piston
\( C_r \) Reflection coefficient
\( \rho \) Density
\( \eta \) Viscosity
\( \gamma \) Specific heat ratio

subscripts
\( co \) Compressor
\( v \) Valve
* Boundary

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1. Introduction
Superconductors have several unique properties compared to conventional conductors under the superconducting phase transition temperature, for example, the Meissner effect and Josephson effects. Since those properties are crucial to superconducting electronic devices, cryocoolers with high reliability and adequate low temperature (below 2.3 K) are needed. A good example of superconducting electronic devices is superconducting single photon detectors (SSPD), which are now under development at National Institute of Information and Communications Technology, Japan [3, 4]. In such SSPD systems, commercially available SRDK-101D GM cryocoolers were used for cooling the detectors. However, in comparison with a similar semiconductor detector system, now the superconducting detector system, including the cryostat, is comparably larger.

In order to reduce the size of an SSPD system, a compact GM expander together with a low bottom-temperature of about 2.3 K, has been developed. The total height of the expander was reduced by 85 mm compared with a commercial 0.1W 4 K GM cryocooler [1, 2]. And then, considering the targeted cooling application, we set the design temperature targets of the first and the second stages under 1 W and 20 mW of heat load to be 60 K and 2.3 K, respectively.

In addition, it is required to reduce the total volume of the compressor unit by 50% relative to the existing CNA-11 compressor. Although optimization of the internal compositions is one of the ways to reduce the volume of the compressor unit, major design changes are indispensable because the adsorber and the oil separator have almost the same volume as the compressor capsule. Thus, it is considered that an effective way to reduce the total volume is to exclude these parts, i.e. to develop a non-lubricated compressor.

In 2004, Coery et al. reported that a 4 K GM cryocooler with a linear compressor (made by CFIC-Qdrive) for Air force applications was developed [5]. The size of the compressor was relatively large for cooling such a small superconducting electronic device. Recently, a new, compact valved non-lubricated linear compressor has been developed at SHI. The cooling performance of a 2 K GM expander operated by an experimental unit of the linear compressor was measured. The detailed experimental results will be reported in this paper.

2. Design of the valved linear compressor
In order to design a valved linear compressor for a 2 K GM cryocooler, the mass flow rate and the pressure ratio are set to be similar to the specification of a CNA-11 compressor. The design target of the linear compressor is shown in table 1. Furthermore, under these determined conditions, the movement of the compressor parts has been analyzed. And then the major components, such as the linear motor, the flexure bearing, the inlet and outlet valves, were designed.

2.1 Motion analysis model
The linear compressor moves complexly according to the load of the piston. Moreover, it is important to simulate the behavior of the valve. A motion analysis model of a valved linear compressor was developed to obtain vital information for designing the linear motor and the flexure bearing spring.

| First stage cooling capacity | 1 W at 60 K |
|----------------------------|-------------|
| Second stage cooling capacity | 20 mW at 2.3 K |
| Maximum electric input power | 1.2 kW AC100 V |
| Operating frequency | 40 ~ 70 Hz |
| Cooling temperature | Water 30 °C |
| Initial gas pressure | 1.3 ~ 2.0MPa |
| High / Low pressure | 2.25 / 0.85 MPa |
| Mass flow rate | > 0.8 g/sec |
| Compressor efficiency | > 60 % |
| Unit volume | < 35 L |

Figure 1. Analysis model of a valved linear compressor.
Figure 1 shows the motion analysis model. Assuming the working gas is an ideal gas, the state equation of gas, the mass and energy conservation equations for the compression space can be respectively written as,

$$ P_{co} = \rho_{co} R_{co} T_{co} \quad (1) $$

$$ \frac{dM_{co}}{dt} = m'_{out} - m'_{in} \quad (2) $$

$$ \frac{dQ_{co}}{dt} + m_{co} C_{co} T_{co} - m_{in} C_{co} T_{in} = \left( C_{co} P_{co} \frac{dV_{co}}{dt} + C_{co} V_{co} \frac{dP_{co}}{dt} \right) / R_{co} \quad (3) $$

According to the differential pressure and Lorentz force, the motion equation of the compressor piston is,

$$ i_{coil} B_{coil} = M_{co} \frac{d^2 X_{co}}{dt^2} + C_{co} \frac{dX_{co}}{dt} + K_{co} (X_{co} + X_{offset}) + A_{co} (P_H - P_L) \quad (4) $$

Taking the counter electromotive force caused by the movement of a lead in a magnetic field into account, the Killihoff expression in the electric circuit of the compressor motor is,

$$ \frac{dE_{coil}}{dt} = \ell_{coil} B_{coil} \frac{d^2 X_{co}}{dt^2} + L_{coil} \frac{d^2 X_{co}}{dt^2} + R_{coil} \frac{dH_{coil}}{dt} \frac{dH_{coil}}{dt} + \frac{k_{coil}}{C_{coil}} \quad (5) $$

where $i_{coil}$ is the current, $E_{coil}$ is the voltage, $\ell_{coil}$ is the coil wire length, $C_{coil}$ is the capacitance, $L_{coil}$ is the inductance, $R_{coil}$ is the resistance, $M_{co}$ is the mass of the moving part, $K_{co}$ is the spring constant, and $B_{coil}$ is the magnetic circuit induction. In addition, by applying the momentum equation, the valve motion is,

$$ 0 = M_{co} \frac{d^2 X_{co}}{dt^2} + C_{r} \frac{dX_{r}}{dt} + K_{r} X_{r} + A_{r} (P_{co} - P_H) \quad (6) $$

$$ \frac{dX_{r}}{dt} = -C_{r} \frac{dX_{r}}{dt} \quad (7) $$

where $C_r$ is the reflection coefficient of the inlet and the outlet valves. The amount of the gas leakage between the piston and the cylinder and the leakage through the valves was calculated by the following expression.

$$ m_{leak} = CV \sqrt{P_H^2 - P_L^2} \quad (8) $$

where $CV$ is the resistance coefficient, $P_H$ and $P_L$ is the high and the low pressure, respectively. The differential equations were integrated using the Runge-Kutta Gill method. The conditions for iteration convergence were that the relative discrepancies of the pressure and the piston displacement between two cycles were less than 0.1.

Figures 2 and 3 show the calculation results of the P-V work diagram, the pressure of the compression space and the displacement of the piston, the inlet and the outlet valves.
From figure 3, the neutral position of the piston is offset to the low pressure side by 3-4 mm due to the differential pressure of the upper and the lower pistons.

2.2 Design of motor and flexure bearing

The thrust of a linear motor is greatly affected by the air gap between the yoke and the magnet. Thus, it is possible to improve motor efficiency by reducing this gap. Then, to reduce the number of gaps, the yoke was designed to be integrated with the magnet.

Figure 4 and table 2 shows the model geometry and the calculation results of the transient response analysis of a 1AirGap linear motor (E-star Corp. Ltd.). The calculated motor efficiency was 0.81.

The volume efficiency of a linear compressor decreases as the leakage rate between the cylinder and the piston increases. Therefore, the clearance seal performance becomes very important since there is no oil between the piston and the cylinder in a non-lubricated linear compressor. To keep the clearance relatively small and without impact on a normal operation, the flexure bearing spring becomes indispensable. The design becomes difficult as the size of the flexure bearing spring decreases, because of a growing stress on the edge of the spring. Thus, a sub-spring was added as a stress reduction measure.

Figure 5 shows the calculation results of a flexure bearing with a sub-spring using contact stress analysis. The results show that it is possible to reduce stress by 5~10% when using this concept.

### Table 2. Calculation results of the transition response analysis of a 1AirGap linear motor.

| Parameter          | Value    |
|--------------------|----------|
| Operating frequency| 50 Hz    |
| Capacitor          | 0.22 mF  |
| Number of coil     | 190 turn |
| Voltage            | 95 V     |
| Current            | 14.4 A   |
| Force              | 708.6 N  |
| Current density    | 8.13 A/m²|
| Force constant     | 49.3 N/A |
| Copper loss        | 140.3 W  |
| Iron loss          | 123.0 W  |
| Power factor       | 0.99     |
| P-V work           | 1101.1 W |
| Input power        | 1364.4 W |
| Motor efficiency   | 0.81     |

**Figure 4.** Analysis model of a 1AirGap linear motor.

**Figure 5.** Calculation results of a flexure bearing with a sub-spring using contact stress analysis.

**Figure 6.** Cross section of an experimental unit of a valved linear compressor.
3. Evaluation of an experimental unit

Based on calculations, an experimental unit was designed, built and preliminary experiments were conducted.

Figure 6 shows the cross section of an experimental unit of a valved linear compressor. To reduce the pressure drop loss as much as possible, the gas charging side was installed at the piston upper end. Reed valves were used as both the inlet and the outlet valves. The position of the piston was monitored using a laser vibrometer and pressure transducers were mounted within the compressor space and used for calculating the pressure-volume (P-V) work. To calculate the heat capacity, the inlet and the outlet cooling water temperatures of the heat exchanger were measured with Pt100 sensors.

Figure 7 shows an experimental unit of the valved linear compressor. Table 3 shows a comparison between the experiment and calculations. A P-V work $W_{co}$ of 860 W was obtained with an electric input power $W_{inp}$ of 1.2 kW. The latter exludes the power consumption of the cooling fan, the cooling circulating pump and the rotary valve motor. At a high / low pressure of 2.27 / 1.09 MPa and an operating frequency of 50 Hz, a mass flow rate $m_{re}$ of 0.89 g/s was obtained.

Accordingly, a volume efficiency $\varepsilon_{vol}$ of 0.64, an indicated efficiency $\varepsilon_{ind}$ of 0.88, a motor efficiency $\varepsilon_{mot}$ of 0.75, a mechanical efficiency $\varepsilon_{mech}$ of 0.95 and a compressor efficiency $\varepsilon_{co}$ of 0.4 were obtained.

$$\varepsilon_{co} = \varepsilon_{vol} \cdot \varepsilon_{ind} \cdot \varepsilon_{mot} \cdot \varepsilon_{mech} = \frac{m_{re} \cdot W_{th}}{m_{th} \cdot W_{co} \cdot W_{mot} \cdot W_{inp}} \quad (9)$$

where $m_{th}$ is the theoretical mass flow rate, $W_{th}$ is the theoretical work flow, $W_{co}$ is the measurement compressor work flow, $W_{mot}$ is the measurement motor work flow, and $W_{inp}$ is the compressor input power.

The compressor efficiency, compared to the target of 0.6, was relatively low. The low volume efficiency was caused by the off-set of the piston from a neutral position. In addition, the gas leakage between the piston and the cylinder was too large. The measured heat dissipation at the compressor cooler was 379 W. The copper loss, which can be estimated from the resistance and current, was 173 W, and the iron loss was 206 W. The measured iron loss was larger than the calculated value, thus the

| Item                      | Measurement | Calculation |
|---------------------------|-------------|-------------|
| Operating frequency       | 50 Hz       | 50 Hz       |
| High / Low pressure       | 2.27 / 1.09 MPa | 2.25 / 0.85 MPa |
| Discharge temperature     | 133.3 °C    | 125 °C      |
| Charge temperature        | 25 °C       | 28 °C       |
| Mass flow rate            | 0.89 g/sec  | 0.93 g/sec  |
| Piston stroke             | 13.1 mm     | 13.5 mm     |
| Input power               | 1.2 kW      | 1.2 kW      |
| P-V work                  | 0.86 kW     | 0.81 kW     |
| Ideal P-V work            | 0.76 kW     | 0.81 kW     |
| F-X work                  | 0.91 kW     | 0.99 kW     |
| Volume efficiency         | 0.64        | 0.87        |
| Indicated efficiency      | 0.88        | 0.99        |
| Motor efficiency          | 0.75        | 0.82        |
| Mechanical efficiency     | 0.95        | 0.95        |
| Compressor efficiency     | 0.40        | 0.62        |
motor efficiency was lower than the calculated efficiency. In general, the motor efficiency can be improved by reducing these losses.

| Item                  | With CNA-11 compressor unit | Measured results |
|-----------------------|------------------------------|------------------|
| First stage temperature with 1 W | 45 K                        | 48 K             |
| Second stage temperature at 2.3 K | 20 mW                       | 14 mW            |
| No-load second stage temperature | 2.18 K                      | 2.19 K           |

4. Performance of a 2 K GM cryocooler with the linear compressor

A 2 K GM expander was operated with an experimental linear compressor, and the measured cooling capacity is shown in table 4. As shown in table 4, with an input power of 1.2 KW, a cooling capacity of 1 W at 48 K at the first stage and 14 mW at 2.3 K at the second stage has been achieved with a linear compressor. That can be compared to 1 W at 45 K and 20 mW at 2.3 K achieved with a CNA-11 compressor.

Figure 8 shows the cool-down curves. As shown in figure 8, the second stage temperature reaches 2.3 K in about 3 hours.

5. Conclusion

An experimental unit of a valved non-lubricated linear compressor for a 2 K GM cryocooler, which can be used for cooling superconducting electronic devices, has been developed. Under no-load condition, a low temperature of 2.19 K has been achieved. With 1 W and 14 mW heat load, the temperature was 48 K at the first stage and 2.3 K at the second stage with an input power of about 1.2 KW. In the future, we plan to further improve the efficiency of the compressor and to reduce the size of the heat exchanger thus reducing the total volume of the compressor unit.

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