Development Status of a High Cooling Capacity Single Stage GM Cryocooler

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ABSTRACT

Continuous improvement of high temperature superconducting (HTS) materials has lead to a thriving development of various applications including superconducting motor, superconducting power transmission cable and superconducting power generator. Those applications, in common, require a high-capacity and highly reliable cooling solution to keep the superconducting material at the proper temperature.

To meet such a requirement, a high capacity GM cryocooler which can provide about 650 W cooling power at 80 K[1] was developed. Since the physical size of this prototype unit is considerably larger compared with current available GM cryocoolers, parameters including regenerator size and valve timing should be redesigned carefully in order to get an optimized performance.

An unsteady-flow, 1-D simulation method was used to predict the behavior of the prototype cryocooler, and the result was used to form the guideline for the basic design. After the first prototype unit became available, the design was improved by feeding back the experiment results to the prototype units. In this paper, both the simulation method and experimental results will be reported.

INTRODUCTION

Since its invention in 1950s[2], Gifford-McMahon (GM) cryocooler is known as one of the most reliable commercial cryocooler for multiple applications including cryopumps, cooling of the radiation shield for an MRI, and cooling of infrared sensors for the military[3]. Most of the development of the GM cryocooler in last decade focused on increasing cooling performance near 4 K, performance optimization in the 4-30K temperature region and increasing the Carnot efficiency. Development of high heat capacity materials played an important role in this stage. In the 1990s, the development of rare earth materials made it possible for a GM cryocoolers to reach a temperatures near 4K. Recently, by utilizing a new rare earth material, Gd_2O_2S (GOS)[4], a prototype compact GM cryocooler which can reach 2.1K under no-load conditions was developed[5-6].

On the other hand, rapid growth of high temperature superconducting (HTS) material applications brings new challenges to the current cryocoolers. In contrast to the 4K region applications, HTS applications often require much more cooling power with relatively high cooling temperature of 80K is acceptable. Applications including HTS superconducting motors, power transmission lines and power generators are usually considered to consume 10^2 W to 10^4 W cooling power (per case). Most of these applications are currently using direct liquid nitrogen cooling or Turbo-Brayton cryocoolers. Though having the potential to significantly reduce the cost and space requirement, current commercial GM cryocoolers lack the suitable cooling power which is crucial in HTS applications. In the current market, even the largest GM cryocooler (scotch-yoke driven) can only provide about 100 W at 30K (with a 7 kW compressor unit)[7].

To fill the gap for high cooling capacity, a single stage GM cryocooler prototype unit is developed. After the design optimization of valve size, valve timing and regenerator material arrangement, a typical cooling capacity of about 650 W at 80K was achieved with an input power of about 13 kW. In this paper, the latest experiment results and optimization method will be described.

GENERAL DESIGN ASPECTS

A typical single stage, scotch-yoke driven GM type cryocooler design was chosen to be the start point of our development. As shown in Figure 1, the pressure wave is generated by a typical rotary valve. Pressure gauges which were used to monitor the compression and expansion room pressure are only utilized in the experiment. A 200 W brushless electric motor with controllable rotation speed is used to drive the motion of scotch-yoke. Figure 2 shows the more realistic external view of the current prototype unit. The physical size of this unit is about 680 mm in height and 640 mm in width.

The rotary valve is designed to be changeable so the operation timing and opening area can be easily optimized. Also the displacer is designed to be reopened for the arrangement of metal mesh inside to be changed.

In order to roughly determine the physical size of the expansion room, displacer and to achieve loss data for further optimization, a 1-D transient simulation was performed to show the principle performance prediction.

SIMULATION METHOD AND RESULTS

Simulation Method Description

A one-dimensional, transient, finite volume method simulation analysis was carried out to estimate the main loss in the prototype cryocooler. The compressibility of helium is taken into count by the continuity equation:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{U}) = 0 \tag{1}$$

where ρ represents the density of helium, and $\vec{U}$ is the velocity vector.

The momentum equation is simplified to only consider the flow resistance caused by metal mesh in the displacer.

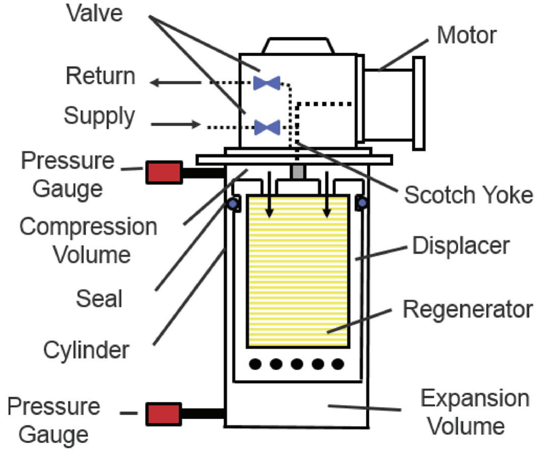

Figure 1. A simplified diagram of the prototype cryocooler.

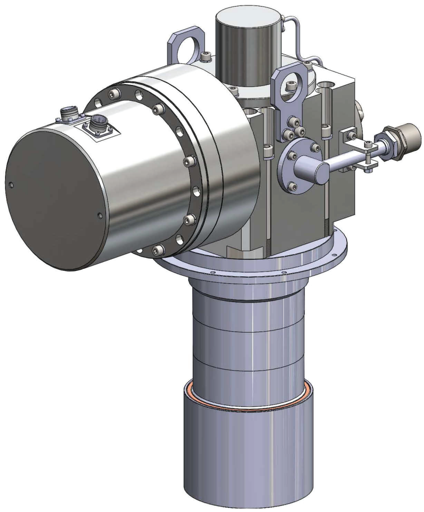

Figure 2. A 3-D drawing of the external view of current prototype unit.

$$\nabla p = -\frac{1}{2}\frac{f_r}{d_h}\rho\left|\vec{U}\right|^2\frac{\vec{U}}{|\vec{U}|} \tag{2}$$

In Eq. 2, p represents the pressure of helium under consideration. f_r represents the Darcy friction coefficient posed by metal mesh and d_h is the hydraulic diameter of the calculated area. The pressure drop occurring in other components such as valve and connection pipes is modeled by a simpler calculation.

The energy equation of helium is:

$$\frac{\partial(\rho h)}{\partial t} + \nabla\cdot(\rho\vec{U}h) + h_f(T - T_s) = 0 \tag{3}$$

where h is the enthalpy of helium and h_f stands for heat transfer coefficient between helium and metal mesh. T and T_s are the temperature of helium and mesh, respectively. The cooling capacity is calculated by a similar equation, except that the heat transfer is between the helium and the cold end heat exchanger.

Since the heat conduction loss through the wall of the cylinder, the shuttle loss and the radiation heat loss are not included in this calculation, additional analysis is needed to interpret the simulation results.

Analysis of Simulation Results

The simulation results for the cooling capacity and losses are summarized in Table 1. The cooling capacity is derived from the simulation results of the cooling power (as described in previous section), with additional consideration towards shuttle loss, thermal conduction loss through cylinder wall and radiation loss.

Regenerator loss is obviously the largest part among all losses. Therefore, reducing the regenerator loss should be one of the first priorities during the prototype development.

Pumping loss and shuttle loss, according to our experience, are sensitive to clearance size between the displacer and the expander cylinder. Actual loss in the experiment may be affected by dimension deviation, mechanical vibration and even surface abrasion. Since only a simplified model was used in simulation, the results of those terms may be different from the experiment.

Thermal conduction and heat radiation are also calculated separately from the simulation. Different from the pumping and the shuttle loss, these terms are considered to be rather consistent with the actual conditions. However, thermal conduction and radiation from ambient environment are usually difficult to reduce as long as the physical size is not significantly changed.

Table 1. Simulation Results of Cooling Capacity and Losses.

Item	Results (W)
PV power	853.4
Regenerator loss	91.5
Pumping loss (including pulse effect, minus means cooling effect)	-34.6
Shuttle loss	33.0
Thermal conduction loss	14.0
Radiation loss	2.4
Net cooling capacity	707.6

The calculated net cooling capacity of 707.6 W is considered to be a good starting point since it's considerably closer to the 650 W cooling capacity development target, and has a reasonable margin to cover the possible calculation error in future development.

EXPERIMENT SETUP AND RESULTS

Based on the principle design parameters derived from the simulation, a prototype unit was manufactured for further optimization. The test setup, optimization of valve timing and regenerator, and a typical cooling load-map and cool-down process will be described in the following sections.

Experiment Setup

Heat load is generated by 4 cartridge heaters attached to the cold head and the temperature on cold head is measured by a PtCO (Chino) sensor, as shown in Figure 3.

In order to provide sufficient helium flow to the expander, two helium compressor units (F70) were connected in parallel to the prototype unit. Typical total power consumption of compressor is about 7 kW x 2 = 14 kW. It should also be noted that F70 compressor itself is water-cooled and the power consumption of water chiller is not included here.

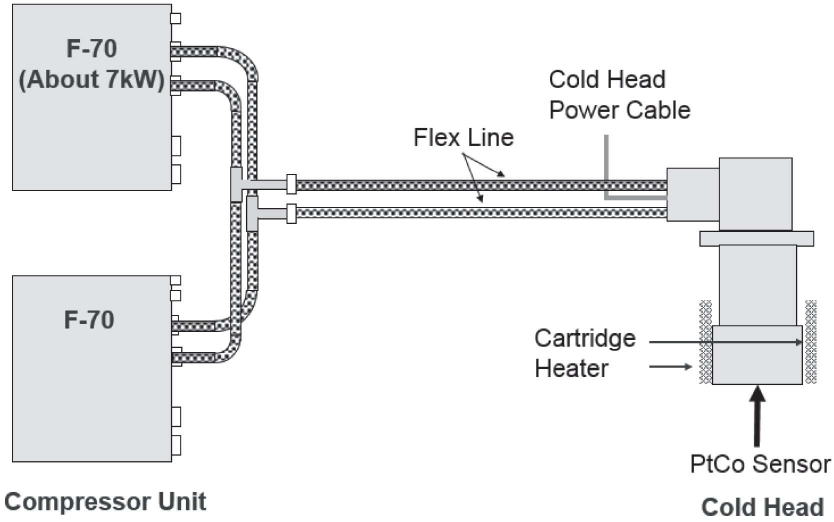

Figure 3. A schematic drawing of the experiment setup.

Valve Timing Optimization

Optimization value for the valve timing usually depends on the target temperature region, helium mass flow rate and characteristics of the compressor unit. Although there is already sufficient knowledge of compressor unit, the 80K temperature region is unfamiliar and we have no experience with the relatively large helium mass flow rate (compared with formal 4K and 20K GM cryocooler products). A comparison experiment was conducted to show the effect of expansion timing on the cooling capacity.

As shown in Table 2, the temperature of the cold head under 650 W heat load is only slightly improved with longer expansion timing, which means the expansion timing is already close to a local optimized point.

Regenerator Optimization

The regenerator optimization focused on the arrangement and amount of metal meshes (wire screens) filled in the displacer. Three kinds of copper and stainless meshes with different mesh size are used in this experiment. Considerable cooling capacity improvement was achieved by altering the amount of each kind of mesh, which is consistent with the simulation prediction described in previous section. Table 3 shows part of the results.

Cold head temperature under 650 W in the “Base line” case is different from the previous valve timing optimization experimental results because there were other minor modifications of the inner structure. Starting from the “Base line” case, increasing the amount of coarse meshes by 10% improved the cooling capacity by about 25-35 W. The best performance was almost exactly 650 W at 80K when the amount of coarse meshes is further increased.

Cooling Load-map and Cooling Down Process

A typical cooling load-map is shown in Figure 4. With all the optimization performed, a typical cooling temperature was 79.8K under 650 W heat load. The power consumption of the compressor at this point was about 13 kW (6.5kW x 2). The gap between this result and previous simulation analysis (707.6 W) is mainly caused by the calculation error in pumping loss, shuttle loss and lack of consideration to the characteristic of compressor unit.

A typical cooling down process is shown in Figure 5. The temperature on the cold head stage usually reaches its stable state within 1 hour under a no-load conditions. Minor modification including valve timing and regenerator material may affect the required cool-down time, but in principle the required time should be similar.

Table 2. Optimization Experiment Results for Valve Timing.

Item	Temperature under 650 W (K)
Case 1 (original timing based on simulation)	87.6
Case 2 (expansion timing + 1.1 %)	87.5
Case 3 (expansion timing + 3.3 %)	87.3

Table 3. Optimization Experiment Results for Regenerator Material.

Item	Temperature under 650 W (K)
Case 1 (Base line)	84.3
Case 2 (Portion of coarse meshes + 10 %)	82.3
Case 3 (Portion of coarse meshes + 20 %)	79.8

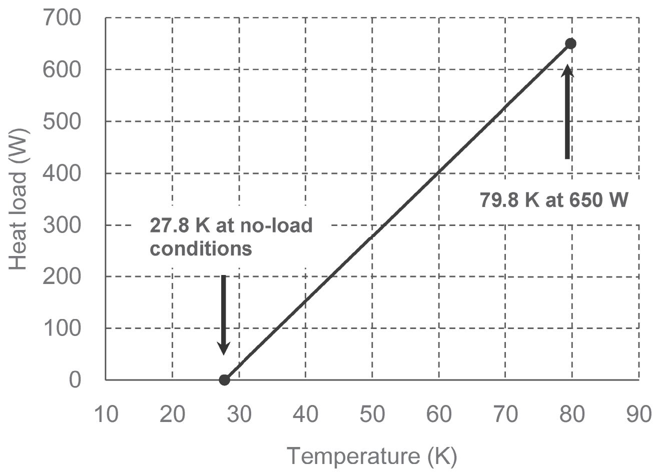

Figure 4. A typical cooling load-map.

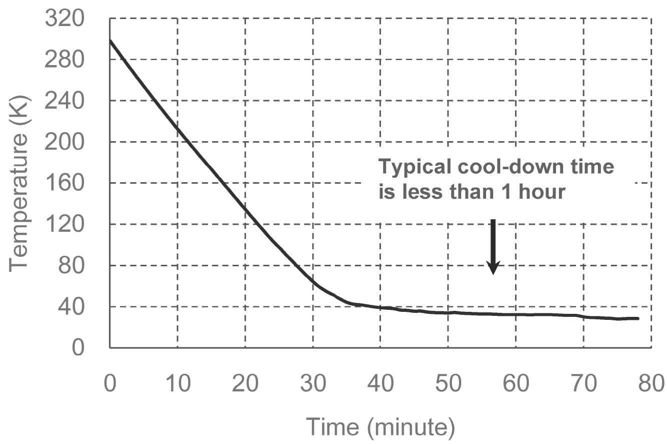

Figure 5. A typical cool-down process from room temperature.

CONCLUSION AND FUTURE WORK

A high capacity, single stage GM cryocooler prototype unit is developed to provide an alternative option in the 80K temperature region. The initial design was based on traditional scotch-yoke driven GM cryocooler structure with estimated parameters obtained from simulation method. After a prototype unit became available, further improvements were made mainly towards the valve timing and regenerator material based on experimental results.

The current prototype unit is about 680 mm x 640 mm in physical size. With 13~14 kW compressor power, a typical cooling capacity of 650 W at 80K was achieved. Typical cooling-down time from room temperature is under 1 hour.

As future work, more reliability and stability test will be taken to discover potential problem during long term operation.

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