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Loss Analysis of High Power Stirling-Type Pulse Tube Cryocooler

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Abstract. For the purpose of cooling high-temperature superconductor (HTS) devices, such as superconductor motors, superconducting magnetic energy storage (SMES) and current fault limiters, cryocoolers should be compact in size, light-weight, and have high efficiency and reliability. In order to meet the demand of HTS devices world-wide, the cryocooler needs to have COP efficiency >0.1. We have developed a high power Stirling-type pulse tube cryocooler (SPTC) with an in-line expander. The experimental results were reported in June 2012[1]. The cooling capacity was 210 W at 77 K and the minimum temperature was 37 K when the compressor input power was 3.8 kW. Accordingly, the COP was about 0.055. To further improve the efficiency, the energy losses in the cryocooler were analyzed. The experimental results and the numerical calculation results are reported in this paper.

1. Introduction

Recent technical innovation of high-temperature superconductor (HTS) devices has brought about the necessity for high efficiency cryocoolers. Pulse tube cryocoolers with 100~1000 W at 77 K cooling capacity have been developed for HTS applications. In 2004, J. H. Zia et al. reported a commercial prototype Stirling-type pulse tube cryocooler (SPTC) with 200 W cooling capacity as civilian equipment for HTS electronics applications [2]. For the purpose of cooling HTS and semiconductor sensors, a small Stirling cryocooler had been developed. However, the cryocooler's piston seal showed signs of wear after running for an extended period of time. This affected its reliability by decreasing its cooling capacity. By contrast, pulse tube cryocoolers are have no moving parts in the expander, and are, therefore, more reliable than other regenerative cryocoolers (e.g., GM and Stirling cryocoolers), Sumitomo Heavy Industries has been developed a high-power SPTC to satisfy the specifications demanded from HTS devices including compact size, light weight, high efficiency and high reliability and the experimental results were reported in June 2012[1]. The cryocooler, with an in-line expander, has a cooling capacity of 210 W at 77 K when the compressor input power is 3.8 kW at the operating frequency of 49 Hz, a compressor efficiency of 74% and a COP of 0.055.

However, the cryocooler's COP must be >0.1 in order to meet the demand of HTS devices world-wide, so the prototype model needs to be further improved. We investigated the cryocooler's energy loss by analyzing the experimental results and the numerical calculation results to improve its efficiency for HTS devices.

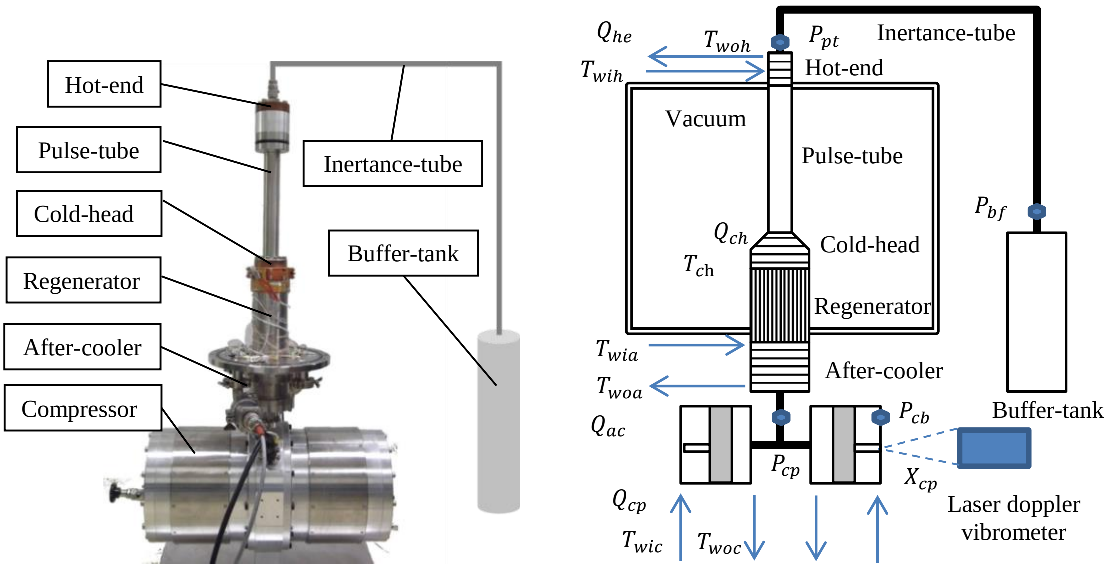

Figure 1. Photograph of prototype in-line pulse tube cryocooler and measurement points of pressure, piston displacement and temperature.

2. General design

Figure 1 shows a photograph and a schematic of the prototype in-line pulse tube cryocooler and the significant points of measurement. Specifications are listed in Table 1. The cryocooler is an integral Stirling-type pulse tube and the compressor has opposed pistons driven by a linear motor. The total height of the expander and compressor is 1700 mm, the compressor outer diameter is 330 mm and its length is 670 mm. The compressor has a moving magnet-type motor with high efficiency, and the piston is guided by a flexure bearing, which keeps a several micrometer clearance between the piston and cylinder. The compressor is connected to the expander by connecting tubes of about 100 mm in length. The piston's position is monitored using a laser vibrometer. The expander is comprised of the after cooler, the regenerator, the cold head (low temperature heat exchanger), the pulse tube, the hot end (high temperature heat exchanger), the inertance tube and the buffer tank. The compressor, the after cooler and the hot end heat exchanger, are cooled by water. To improve the cooling capacity efficiency, the after cooler is a shell and tube type heat exchanger, while the regenerator is stacked with stainless-steel wire mesh of different mesh sizes. To reduce radiation loss, Super-Insulation (SI) is wrapped around the regenerator tube and the pulse tube in vacuum. To measure the pressure amplitudes, pressure transducers are mounted near the compressor discharge head, back space of the piston, the hot end of the pulse tube and the buffer tank. The mass flow rate through the phase-shifter can be estimated by the pressure at the hot end of the pulse tube. These measurements are used to calculate both the pressure-volume (P-V) work of the compressor (=work flow) and the expander. A PtCo sensor is used to measure the cold head temperature. To calculate the heat rejection, the inlet-outlet cooling water temperature of the after cooler and the hot end of the pulse tube are measured with Pt100 sensors.

3. Experimental results

We tested the performance of prototype cryocooler. Figure 2 shows a graph of the prototype in-line pulse tube cryocooler cooling capacity and COP vs input power. The initial gas pressure of the cryocooler is 2.1 MPa, operational frequency was 49 Hz, and electrical input power was 3.8 kW. We achieved a maximum cooling capacity of 210 W at 77 K and a COP of 0.055 at 16.0% Carnot (cold head temperature T_{ch}=77 K, hot side temperature T_h=320 K).

Table 1. Prototype SPTC Specifications.

Item	Value
Maximum cooling capacity	210 W at 77 K
COP	0.055
Maximum electrical input power	3.8 kW AC200 V
Operating frequency	49 Hz
Temperature of inlet water	20 ℃
Initial gas pressure (Helium)	2.1 MPa
Life time (at design)	50000 hour
Weight	150 kg
Pulse-tube	ID40 mm × L250 mm
Regenerator	ID90 mm × L100 mm
Inertance-tube	ID11 mm × L1800 mm
Buffer-tank volume	0.0035 m^3

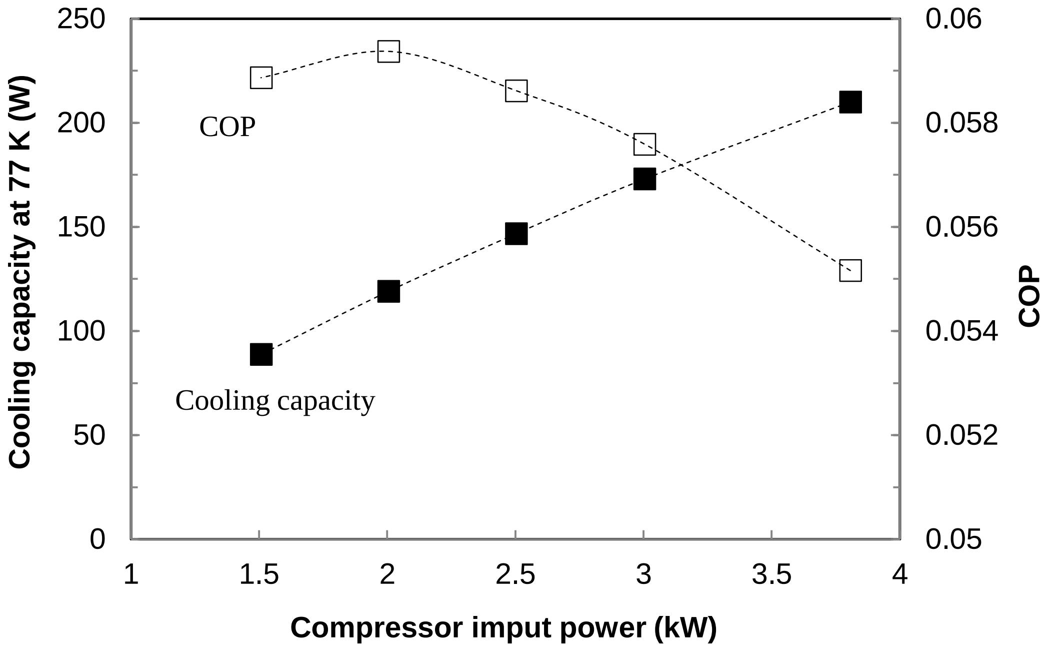

Figure 2. Measured cooling capacity and COP vs compressor input power.

4. Energy flow analysis

These experimental results differed greatly from the target cryocooler efficiency of COP=0.1 for the practical use of HTS. Therefore we studied energy flow and loss analysis in the SPTC expander to improve the efficiency of cooling capacity. Because the pulse tube cryocooler does not have moving parts in the expansion space, the expansion volume amplitude cannot be measured and the work flow of the pulse tube in the low temperature position cannot be calculated. Therefore, we think that the work flow in high temperature position is approximately equal to that of low temperature and have calculated the work flow in the pulse tube from the pressure difference between the hot end and the buffer tank. We also considered that it is possible to calculate the work flow using the differences in pressure in the regenerator, but this is difficult to do, because the regenerator matrixes used hybrid mesh and their temperature distribution is non-linear. We considered that the momentum equation of the inertance tube is equivalent to the LR circuit and the gas displacement in the inertance tube is calculated by the measured pressure amplitude of the hot end, P_{pt}, and the buffer tank, P_{bf}, respectively.

The gas displacement in the inertance tube, X_{it}, is calculated from equations 1-2[3]:

$$P_{pt} - P_{bf} = \left(L_{it} \cdot \frac{d^2X_{it}}{dt^2} + R_{it} \cdot \frac{dX_{it}}{dt}\right) \tag{1}$$

$$L_{it} = \rho_{it} \cdot \ell_i, \quad R_{it} = f_o \cdot \frac{8\pi\eta\ell_i}{A_{it}} \tag{2}$$

where L_{it} and R_{it} are the inductance, and the resistance factors of gas, respectively. f_o is frequency of operation, A_{it} is the cross-section area of the inertance tube, ℓ_i is the length of the inertance tube, and ρ_{it} is the density of gas in the inertance tube. η is the viscosity of working gas. Based on X_{it}, we calculated the pulse tube work flow, W_{pt}:

$$X_{pt} = \frac{A_{it} \cdot X_{it}}{A_{pt}} \tag{3}$$

$$W_{pt} = f_o \cdot \oint P_{pt} \cdot A_{pt} \cdot dX_{pt} \tag{4}$$

where A_{pt} is a cross-section area of the pulse tube, and X_{pt} is the gas displacement of the pulse tube.

Figure 3 shows a graph of experimental date regarding the piston displacement and the compressor pressure, the hot end and the buffer tank. These values are used to calculate the work flow of the pulse tube, W_{pt}. The enthalpy flow *'H'* is defined as the sum of work flow *'W'* and heat flow *'Q'*. It is shown in equation 5:

$$H = W + Q \tag{5}$$

The after cooler and the hot end heat exchange rate are calculated by using the heat capacity, C_w, the flow rate, G_{ex}, the inlet and the outlet temperatures, T_{win}, and, T_{wout}, of the cooling water:

$$\delta Q_{ex} = C_w \cdot G_{ex} \cdot (T_{wout} - T_{win}) \tag{6}$$

The SPTC is operated at 3.8 kW of electric input. The heat rejected, δQ_{cp}, in the compressor from losses is measured at 668 W, the back space loss of compressor, W_{cb}, is measured at 140 W. and the compressor work flow, W_{cp}, which generates and flows into the expander is 2883 W. Each heat exchange rate in the expander is measured at the heat exchangers, the rate of after cooler, δQ_{ac}, is 2609 W and the hot end, δQ_{he}, is 549 W. Based on the energy flow diagram, the enthalpy flow in the regenerator is calculated by subtracting the heat extracted at the aftercooler from the compressor work. This enthalpy flow is considered as a regenerator heat loss, which includes static loss, heat conduction loss of the regenerator wall and radiation. The enthalpy flow, H_{re}, calculated from equation 7:

$$H_{re} = W_{cp} - \delta Q_{ac} \tag{7}$$

The enthalpy flow in the regenerator, H_{re}, is 274 W. The work flow of the pulse tube, W_{pt}, is calculated from the experimental results as 598 W. In the same way, the rejected heat of the hot end is equal to the enthalpy flow in the pulse tube. The heat flow of the pulse tube, Q_{pt}, which is considered to be the heat loss of the pulse tube, is calculated from equation 8:

$$Q_{pt} = W_{pt} - H_{pt}(= \delta Q_{he}) \tag{8}$$

As a result, Q_{pt} is 49 W. Figure 4 shows the energy flow diagram for each result. At the cold head, the heat change rate of the cold head, δQ_{ch}, which is 210 W, is absorbed and then increased enthalpy flow into the pulse tube isenthalpical. Finally, the work flow of the pulse tube that flows into the inertance tube is dissipated by friction.

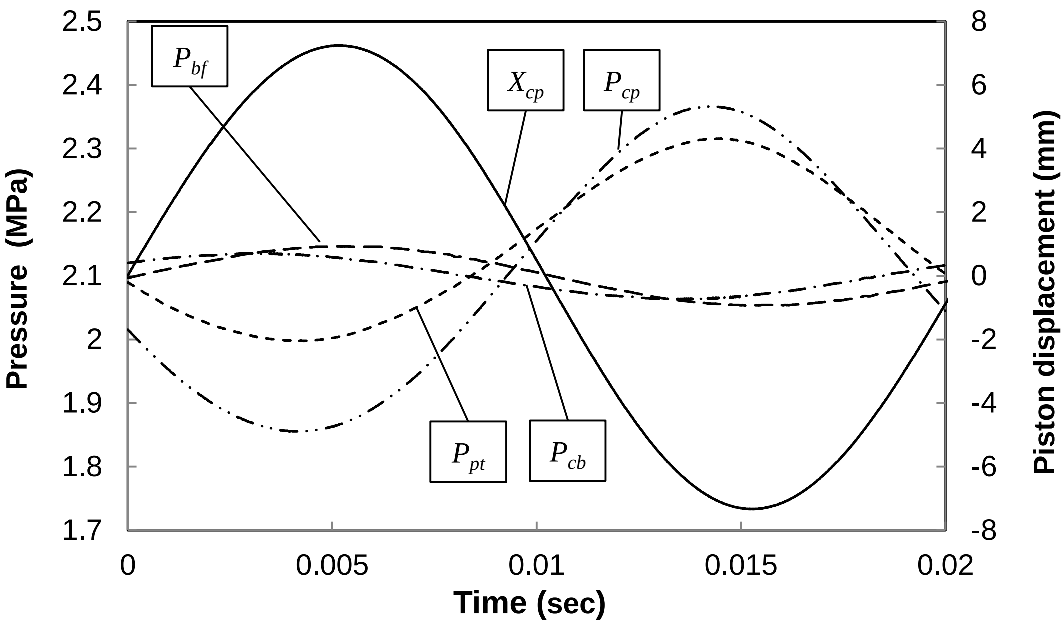

Figure 3. Pressure fluctuation and piston displacement.

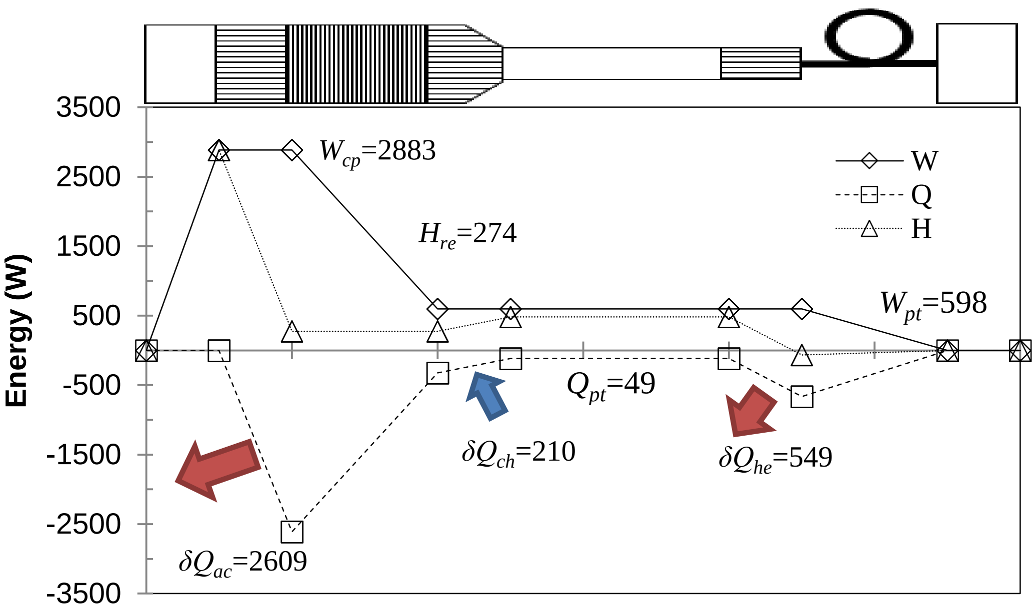

Figure 4. Energy flow diagram in SPTC

5. Numerical analysis model[4]

To confirm validity of this result and to understand the process occurring in the pulse tube cryocooler in detail and to design it, a numerical analysis model was developed. The SPTC was divided into a number of subsections, and each subsection was divided into control volumes (Figure 5 and Table 2).

Each subsystem can exchange work, heat, and mass with its surroundings through its section boundaries. For simplicity, expansion and compression in the buffer tank were considered to be adiabatic. Assuming the working gas is an ideal gas, the equation of state and the conservation of mass, momentum, and energy for the control volumes can be written as

$$P_j = \rho_j \cdot R_g \cdot T_j \tag{9}$$

$$M_t = \sum_{j-1} M_j \tag{10}$$

$$P_{j+1} = P_j - \Delta P_j \tag{11}$$

$$\dot{Q}_j + \dot{m}_j{}^* C_p T_j^* - \dot{m}_{j+1}{}^* C_p T_{j+1}^* = \frac{\left(C_p P_j \dot{V}_j + C_v V_j \dot{P}_j\right)}{R_g} \tag{12}$$

where ρ is the density of gas. P is the pressure. R_g is the gas constant. M_t is the total mass of gas. ΔP is the differential pressure. $\dot{Q}$ is rate of the heat transfer. $\dot{m}$ is the mass flow rate. V is the volume. C_p is the specific heat at constant pressure. C_v is the specific heat at constant volume. '*' means boundary.

The gas in the regenerator and the pulse tube space take the transfer of heat with the wall into consideration, and apply the energy equation to each space.

$$\dot{Q}_j = \frac{d\left(C_{wl} M_{wlj} T_{wlj}\right)}{dt} \tag{13}$$

Pressure and temperature in the buffer tank are shown from assumption by the next expression:

$$\frac{dP_{bf}}{dt} = \frac{dM_{bf}}{dt} \cdot \frac{\gamma R_g T^*}{V_{bf}} \tag{14}$$

$$\frac{dT_{bf}}{dt} = \left(1 - \frac{T_{bf}}{\gamma T^*}\right) \frac{T_{bf}}{P_{bf}} \cdot \frac{dP_{bf}}{dt} \tag{15}$$

Because the differential pressure and Lorentz force work, the motion equation of the compressor piston is shown in the piston as follows.

$$i_{coil}\, B_{co} \ell_{coil} = M_{mech} \frac{d^2 X_{cp}}{dt^2} + C_{coil} \frac{dX_{cp}}{dt} + K_{coil} X_{cp} + A_{cp}\left(P_{cp} - P_m\right) \tag{16}$$

where P_m is mean pressure of SPTC:

In addition, it is shown that the expression of Kirchhoff in the electric circuit takes into account the electromotive force by the lead that moves in the magnetic field into consideration by the next expression.

$$E_{coil} = \ell_{coil} B_{co} \frac{dX_{cp}}{dt} + L_{coil} \frac{di_{coil}}{dt} + R_{coil} i_{coil} \tag{17}$$

where i_{coil} is current value, E_{coil} is voltage value, ℓ_{coil} is coil line length, C_{coil} is capacitance, L_{coil} is inductance, R_{coil} is resistance, M_{mech} is mass of moving part, K_{coil} is spring constant, and B_{co} is magnetic flux density:

The mass flow rate of the inertance tube, $\dot{m}_{it}$ will simplify the calculation as follows.

$$\dot{m}_{it} = \rho_{it} \cdot A_{it} \cdot \frac{dX_{it}}{dt} \tag{18}$$

The temperature, T_j, of enthalpy transferred across the interface of a control volume is supplied to the adjacent upstream control volume. For the matrix in the regenerator, the differential equations were integrated using the Runge-Kutta Gill method. The conditions for iteration convergence were that the relative discrepancies in temperature and pressure between two cycles were less than 10^{-5}, and that the energy reserve in the regenerator and pulse tube were less than 0.1W.

Table 2. Number of control volume.

Element	Connect -tube	After- cooler	Regenerator				Cold -head	Pulse -tube	Hot- end
			1	2	3	4			
Number	100	20	20	20	40	50	10	100	10

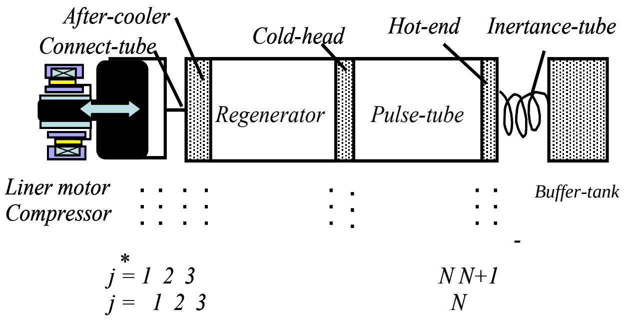

Figure 5. SPTC numerical model.

6. Result and discussion

Results of the energy distribution in the pulse tube cryocooler are compiled on Table 3. Figure 6 shows a graph of the experimental and numerical results that compare the cooling capacity and the work flow vs each cold head temperature. The simulated performance agrees roughly with the measured performance, indicating that the numerical analysis model accurately represents the operating cycle of the cryocooler, and that such models can be effectively used to design pulse tube cryocoolers. The loss ratio for the work flow of the pulse tube is shown in Figure 7. The heat loss of the regenerator accounts for 46% of the work flow of the pulse tube. And the friction loss of the regenerator, which is occurred when the work gas flows through the regenerator, is known as viscosity dissipation. It is difficult to measure a mass flow rate in the regenerator so the friction loss cannot be determined from the experimental result. So we predict the net work flow as a simplified. We considered that the net work flow, W_{ei}, eliminates the friction loss and is amplified by work flow of the pulse tube, W_{pt}, and the thermal ratio of hot-side temperature T_h (= compressor temperature T_{cp}) and cold head temperature, T_{ch}. From balance of W_{cp}, the energy loss of friction in regenerator, W_{fl}, is calculated. They are as follows in equation 19:

$$W_{fl} = W_{cp} - W_{ei} = W_{cp} - W_{pt} \cdot \frac{T_h}{T_{ch}} \tag{19}$$

As a result, W_{ei} is 2485 W, and W_{fl} is 397 W. Moreover, by analyzing the numerical result, the friction loss in the expander is 320 W, too. It is thought that both thermal loss and friction loss in the regenerator greatly contributed to cooling capacity reduction, and there is room for a capability improvement in a loss. To improve the efficiency of the SPTC, the regenerator loss must be reduced.

The pulse tube cryocooler has an inertance tube as the phase-shifter of work flow so that it cannot recover the work flow through inertance tube. On the other hand, the Stirling cryocooler can be more efficiency for the reason having the displacer so that it can recover the work flow that is dissipated energy in case of the pulse tube cryocooler. Therefore, we investigated the possibility of cooling capacity, Q_{st}, if the pulse tube cryocooler had a displacer piston as the phase-shifter and recovering work flow of the pulse tube [5]:

$$Q_{st} = \eta_{pv} \cdot \left(W_{cp} + W_{pt}\right) = \frac{Q_{ch}}{W_{cp}} \cdot \left(W_{cp} + W_{pt}\right) \tag{20}$$

where η_{pv} is the efficiency rate of the cooling capacity, Q_{ch}, and the work flow of the compressor W_{cp}. As a result, Q_{st} is 254 W and the COP is 0.067 have been predicted. These results suggest that pulse tube cryocooler cannot achieve the final goal. But actually, by using the displacer, the flow rate of the working gas tends to reduce and it is possible to improve the cooling capacity a little. Equation 21 optimizes the cryocooler efficiency [6]:

$$\eta_{ch} = \frac{W_{ch}}{M_{ch}} \tag{21}$$

where η_{ch} which is an index of optimizing the efficiency of the cryocooler is the rate of the maximum mass flow in the cold head, M_{ch}, and the work flow of low temperature, W_{ch}. In Stirling cryocoolers, the amplitude of the cooling capacity can be increased by piston volume of a compressor and an expander. And it is necessary to increase the amount of gas that flows from the compressor to the expander. In pulse tube cryocoolers, it is necessary to increase the amount of the gas more than Stirling cryocoolers to raise the pressure ratio for the existence of the pulse tube volume. Increasing the amount of gas causes more loss in the regenerator. For this reason, we think that Stirling cryocoolers are one method to improve the efficiency of the cryocooler. From the above analysis, for improving efficiency of a pulse tube cryocooler, it's necessary to reduce the losses in the regenerator and to add the mechanism which can recover the work flow. Therefore, we considered that the change of the cryocooler type is estimated to achieve final target efficiency at the development of the cryocooler for HTS cooling. However the existence of moving parts in expander causes problems about vibration and reliability.

Table 3. Experimental and numerical results.

Symbol	Experimental (W)	Numerical (W)
W_{cp}	2883	2863
Q_{ac}	2609	2627
H_{re}	274	262
Q_{ch}	210	247
W_{pt}	598	570
Q_{he}	549	546
Q_{pt}	49	24

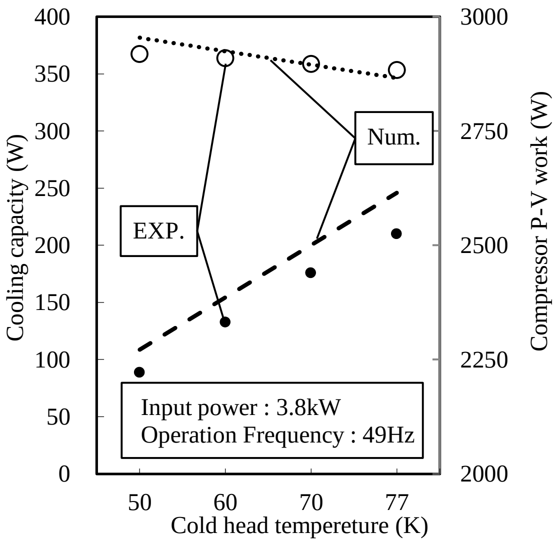

Figure 6. Experimental and numerical results that compare the cooling capacity and COP vs compressor input power.

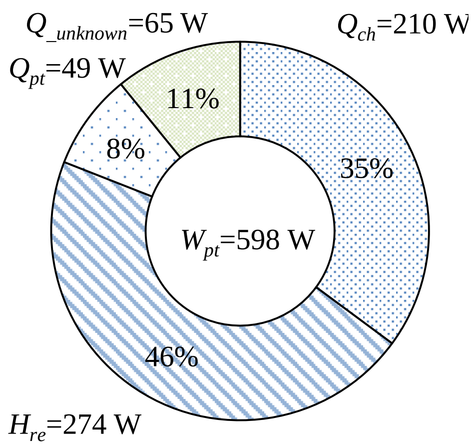

Figure 7. Percentage of cooling capacity and energy loss in work flow of pulse-tube.

7. Conclusion

We developed a Stirling-type pulse tube cryocooler with cooling capacity of 210 W at 77 K and a COP of 0.055. However, these results differ greatly from the target cryocooler efficiency of COP=0.1 for the practical use of HTS. Therefore, we studied energy flow and loss analysis in the expander of the SPTC to improve the cooling capacity efficiency. To understand the process occurring of the SPTC, we developed a numerical analysis model to confirm the validity of these results. By experimental and numerical results, we clarified the thermal loss and friction loss of the regenerator, which greatly reduces cooling performance. Additionally we concluded that there is room for changing the type of cryocooler to achieve the final target.

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