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A Novel Joule-Thomson Refrigerator Driven by Cold Linear Compressor

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Abstract. The mechanical Joule-Thomson (JT) refrigerator operating in 2 K-class or 4 K-class temperature ranges serve as crucial equipment for space astronomical observation and deep space exploration. Current typical space-borne JT cryocoolers face multiple technical challenges due to their compressor operation at ambient temperatures, including excessive pressure ratio requirements, increased compressor stages, enlarged system volume/weight/power consumption, as well as complex configurations with multiple inefficient heat exchanger stages. This study analyzed the enthalpy of a typical helium throttling refrigeration cycle and found that there is an optimal pressure ratio for the throttling refrigeration. A quantitative comparison was made between operating at the optimal pressure ratio and operating at a typical 20 times pressure ratio, resulting in a 66 % increase in refrigeration capacity and a 28.1 % decrease in power consumption. This study innovatively proposes a novel solution by implementing cryogenic operation of the driver assembly and establishing a throttling refrigeration cycle with working fluid in sub-20 K temperature range. This approach fundamentally provides new possibilities for pressure ratio reduction in cryocooler systems.

1. Introduction

2 K-class or 4 K-class mechanical Joule-Thomson (JT) refrigerator is crucial to cool far infrared detector or extremely high sensitivity and resolution detector, such as SQUID（Superconducting quantum interference device）, SNSPD（superconducting nanowire single photon detector）,It is also commonly employed as a precooling stage for cryocoolers operating in lower temperature regimes, such as sorption refrigerators, dilution refrigerators, and adiabatic demagnetization refrigerators [1,2].

The typical spatial precooled JT refrigerator primarily consists of a precooler, compressor, throttle valve, heat exchangers, and evaporator, as illustrated in Figure 1. The refrigeration cycle comprises the following processes: Helium gas is first adiabatically compressed by the compressor to form a high-temperature, high-pressure gas (Process 5-1). Subsequently, the compressed gas undergoes precooling through thermal exchange with the returning working fluid in counter-flow heat exchangers CFHX3 and CFHX2, followed by two-stage precooling via the cold heads of the precooler (Process 1-a-b). The gas is further precooled in counter-flow heat exchanger CFHX1 (Process b-2). The working fluid then enters the throttle valve for isenthalpic expansion, achieving temperature reduction and generating a gas-liquid two-phase mixture (Process 2-3). This cooled mixture subsequently provides refrigeration capacity in the evaporator (Process 3-4). The return flow of working fluid precools the incoming high-pressure gas through the heat exchangers before ultimately returning to the compressor (Process 4-c-d-5), thereby completing the thermodynamic cycle [3]. The corresponding T-S diagram is shown in Figure 2.

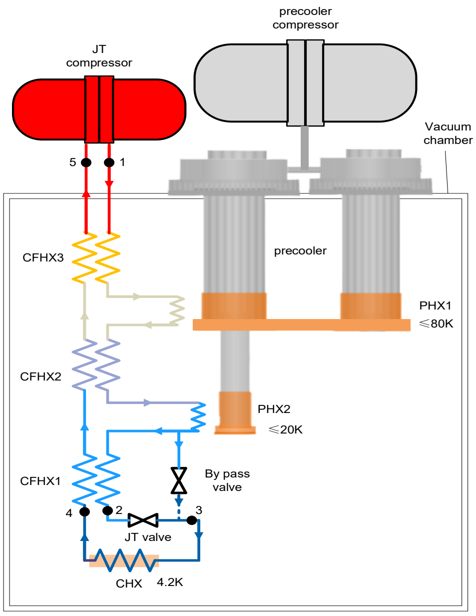

Figure 1. Diagram of a typical helium JTC.

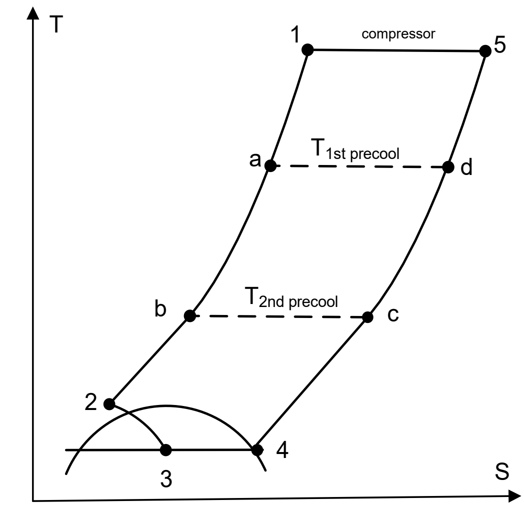

Figure 2. T-S diagram of a typical helium JTC.

2. Problem

2.1 Limitations of the typical JT refrigerator

In the typical JT refrigeration system illustrated in Figure 1, several critical issues persist.

First, the refrigeration cycle involving working fluid transitioning from ambient temperature to liquid helium temperatures and back requires multistage counterflow heat exchangers for cold energy recovery. The low heat exchange efficiency of these counterflow heat exchangers, particularly in high-temperature regions, significantly limits the system's refrigeration efficiency. Additionally, the multistage heat exchanger configuration results in excessive system volume and operational complexity.

Second, The JT refrigerator operating in the liquid helium temperature range exhibits a relatively slow cooling rate during the initial cooling phase. To achieve rapid temperature reduction, a bypass valve is typically implemented in the vicinity of the cold end in JT refrigeration systems. However, this configuration introduces additional system complexity and compromises the system's reliability [4].

Third, and most critically, the compressor operates at ambient temperature. Thermodynamic principles dictate that working fluid density inversely correlates with temperature under constant pressure. Between compressor outlet (P1 in figure1) and throttle valve inlet (P2), extreme temperature differences create substantial disparities in fluid density and heat capacity, leading to fundamentally distinct heat transfer mechanisms in high-temperature and cryogenic regions. Current technological limitations prevent direct measurement of actual heat transfer processes at cryogenic temperatures. In practical applications, engineers typically increase compressor pressure ratio to enhance working fluid density, thereby improving heat transfer rate and mass flow rate to achieve higher refrigeration capacity. However, this approach dramatically reduces refrigeration efficiency while imposing severe design challenges for compressors. For instance, 4 K JT refrigerators typically require suction pressures of 0.1 MPa with pressure ratios reaching approximately 20:1 for optimal performance. necessitating 2-3 series-connected compressor stages. This configuration substantially increases compressor quantity, physical dimensions, weight, and power consumption [5,6].

This raises fundamental questions: Does a theoretically optimal pressure ratio exist for JT refrigerators? Could refrigeration efficiency be improved under such optimal pressure ratio conditions?

2.2 Determination of optimal pressure ratio

The core of throttling refrigeration lies in utilizing the Joule-Thomson effect to generate cooling capacity. By neglecting heat transfer processes in intermediate precoolers and between heat exchangers as shown in Figure 1, the throttling process can be simplified to the fundamental JT refrigerator configuration illustrated in Figure 3, with its corresponding T-S diagram presented in Figure 4.

Under ideal conditions, the specific refrigeration capacity (per unit mass flow rate) provided by the evaporator in a basic JT refrigerator corresponds to the enthalpy difference between the inlet and outlet of the evaporator. Since the enthalpy remains constant during the throttling process, the refrigeration capacity of the throttling cycle can be expressed by Equation 1.

$$Q_e = q_m(h_4 - h_3) = q_m(h_4 - h_2) \tag{1}$$

Where Q_e is the refrigeration capacity, q_m is the mass flow, h_n is the enthalpy of the point n.

Below we present an analysis of enthalpy at the evaporator inlet and outlet.

First, enthalpy at evaporator outlet (P4 in Figure 3). The enthalpy of helium at cryogenic temperatures is a function of pressure and temperature. The evaporator contains saturated vapor at 0.1 MPa @ 4.2 K. While the outlet pressure matches the evaporator pressure (0.1 MPa), the temperature slightly increases to 4.3 K due to operational dynamics. Thus, the enthalpy at P4 is approximated as 0.1 MPa @ 4.3 K.

Second, enthalpy at evaporator inlet (P2 in Figure 3). The throttling process is isenthalpic, meaning the enthalpies at P3 (post-throttling) and P2 (pre-throttling) remain equal. Given the high thermal efficiency of Heat Exchanger 1, the temperature difference between P4 and P2 is minimal (<0.7 K). By establishing a pressure-enthalpy relationship at P2 (variable pressure, quasi-fixed temperature), Equation (1) demonstrates that the refrigeration capacity of the evaporator is proportional to the enthalpy difference between P4 and P2.

Figure 5 presents the relationship between the upstream pressure of the throttle valve (P2 in Figure 3) as the abscissa and the enthalpy as the ordinate. The curves correspond to eight temperature conditions ranging from 4.6 K to 6 K at location P2, demonstrating the existence of a minimum enthalpy value. This critical point indicates the optimal operational state where the evaporator achieves maximum refrigeration capacity, with the corresponding pressure ratio being defined as the optimal pressure ratio. Table 1 systematically summarizes the quantitative correspondence between temperatures at P2 and the optimal pressure ratios under these thermodynamic conditions. In practical applications, the temperature difference between locations P2 and P4 is generally limited to ≤0.7 K. Consequently, when the temperature at P4 reaches 4.3 K, the corresponding temperature at P2 does not exceed 5 K, resulting in a theoretical optimal pressure ratio of ≤3.5. This value is significantly lower than the approximately 20:1 pressure ratio typically required for throttle-cycle refrigeration systems operating in the liquid helium temperature range.

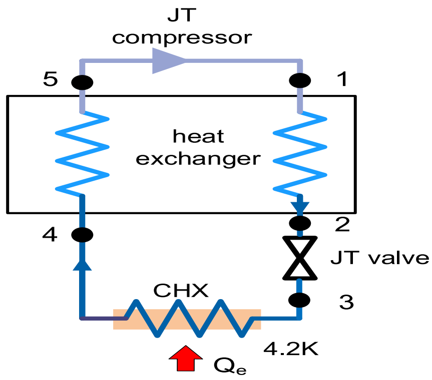

Figure 3. Diagram of a fundamental helium JTC.

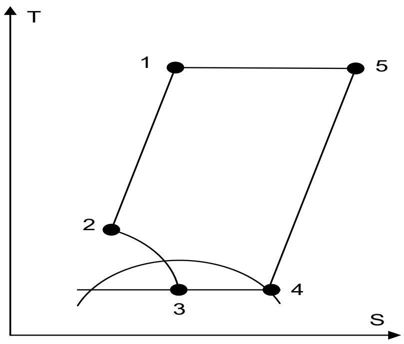

Figure 4. T-S diagram of a fundamental JTC.

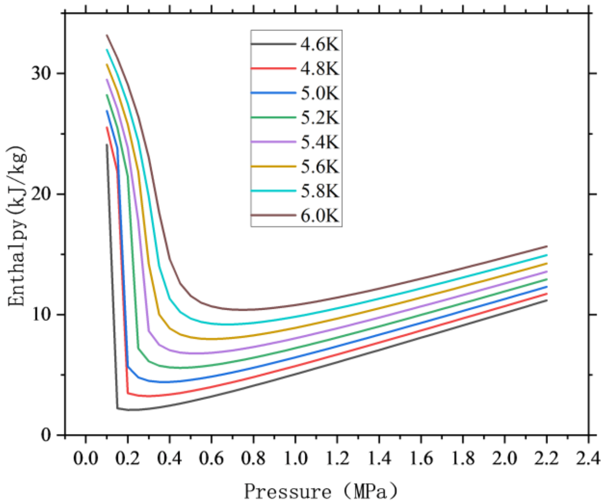

Figure 5. Diagram of enthalpy versus pressure at the throttle valve inlet.

Table 1. Optimal pressure ratio versus temperature at the throttle valve inlet.

P2 Temperature (K)	4.6	4.8	5.0	5.2	5.4	5.6	5.8	6.0
Optimal Pressure Ratio	2	3	3.5	4.5	5	6	6.5	7.5

2.3 Comparison of Cooling Capacity and Compressor Power Consumption

Excessively high pressure theoretically results in a reduction of cooling capacity and an increase in compressor power consumption. Here, we proceed to calculate and compare the cooling capacity and compressor power consumption of a JT refrigerator under two scenarios: ambient-temperature compressors operating at pressure ratios of 20:1 and 3.5:1, respectively, assuming constant flow rate.

The cooling capacity of the refrigeration system is calculated using Equation 1, with the premise that the mass flow rate is constant, and the temperature and pressure at the evaporator outlet, as well as the inlet temperature, are fixed. For the evaporator outlet (P4 in Figure 3), the pressure and temperature are set to 0.1 MPa@4.3 K, while the temperature upstream of the throttling valve (P2 in Figure 3) is fixed at 5 K, with pressures of 0.35 MPa and 2.0 MPa for the two cases. The corresponding enthalpy are determined by referencing thermodynamic tables.The relationship between the cooling capacities for pressure ratios of 20:1 and 3.5:1 is expressed in Equation 2. Assuming the cooling capacity at the 20:1 pressure ratio is defined as 100 %, the calculated cooling capacity at the 3.5:1 pressure ratio is 166 %. This indicates that, under identical flow rates, the compressor operating at a 3.5:1 pressure ratio generates 66 % more cooling capacity compared to the 20:1 pressure ratio scenario, showed in Table 2.

$$Q_{3.5} = Q_{20} \frac{h_{4(PR=3.5)} - h_{2(PR=3.5)}}{h_{4(PR=20)} - h_{2(PR=20)}} \tag{2}$$

Where Q_n is the cooling capacity at the n:1 pressure ratio, PR is the pressure ratio.

The theoretical power consumption of the compressor can be expressed by Equation 3. The calculation assumes a constant mass flow rate, with the compressor inlet pressure and temperature set to 0.1 MPa @ 293 K. The compression process is adiabatic, and the relationship between temperature and pressure follows Equation 4, where the specific heat ratio of helium γ is taken as 5/3. For pressure ratios of 20:1 and 3.5:1, the calculated outlet temperatures are 971 K and 484 K, respectively. The corresponding enthalpy are determined by referencing thermodynamic tables. The relationship between the power consumption of the compressor at these two pressure ratios is shown in Equation 5. Assuming the power consumption at the 20:1 pressure ratio is defined as 100 %, the calculated power consumption at the 3.5:1 pressure ratio is 28.1 % of the 20:1 case,, showed in Table 3.

$$W_{JT} = q_m(h_1 - h_5) \tag{3}$$

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \tag{4}$$

$$W_{3.5} = W_{20} \frac{h_{1(PR=3.5)} - h_{5(PR=3.5)}}{h_{1(PR=20)} - h_{5(PR=20)}} \tag{5}$$

Where W_{JT} is the power consumption of the compressor, γ is the specific heat ratio of helium.

Table 2. Comparison of cooling capacity.

Parameter	P4 (PR=20)	P2 (PR=20)	P4 (PR=3.5)	P2 (PR=3.5)
pressure@ Temperature	0.1 MPa @4.3 K	2.0 MPa @5.0 K	0.1 MPa @4.3 K	0.35 MPa @5.0 K
Enthalpy(kJ/kg)	21.7	11.3	21.7	4.4
Cooling Capacity	100 %	100 %	**166 %**	**166 %**

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Table 3. Comparison of compressor power consumption.

Parameter	P5 (PR=20)	P1 (PR=20)	P5 (PR=3.5)	P1 (PR=3.5)
pressure@ Temperature	0.1 MPa @293 K	2.0 MPa @971 K	0.1 MPa @293 K	0.35 MPa @484 K
Enthalpy(kJ/kg)	1527.1	5053.6	1527.1	2519.8
JT Compressor Power Consumption	100 %	100 %	**28.1 %**	**28.1 %**

3. Solution

The phenomenon wherein the actual pressure ratio significantly exceeds the theoretically optimal pressure ratio remains unexplored in dedicated international research. Primary contributing factors include substantial temperature gradient variations in the working fluid during the cycle, pronounced changes in material properties, multi-stage heat exchange processes, and the complexity of influencing parameters. Current technological limitations, particularly the absence of reliable low-temperature actuators, hinder systematic investigation.

This study innovatively proposes a cryogenic actuation system configuration where the actuator is maintained at low temperatures through flexible cryogenic links connected to the precooler's cold head. The precooler serves dual functions: (1) dissipating heat generated by the actuator and (2) subcooling the high-pressure working fluid compressed by the compressor to below 20 K. The subcooled high-pressure working fluid subsequently undergoes direct throttling refrigeration after passing through a cryogenic counter-flow heat exchanger. The schematic configuration of this integrated system is presented in Figure 6.

Advantages of the Novel Process Compared to Conventional Room-Temperature Compressor-Driven JT Refrigerators. First, low-temperature working fluid stability, The helium working fluid operates entirely within a cryogenic temperature range (20 K to 4.2 K), minimizing temperature gradient variations and stabilizing material properties; Second, reduced thermal losses and pressure drop, By eliminating the majority of heat exchangers, the system significantly reduces thermal losses and pressure drop, thereby enhancing overall refrigeration efficiency. Third, improved compression efficiency, The increased density of the working fluid at cryogenic temperatures enables higher compression efficiency, allowing for lightweight compressor design. Fourth, simplified system architecture: Removal of the bypass valve streamlines the process, reduces structural complexity, and improves operational reliability. Fifth, enhanced electromechanical efficiency, Reduced electrical resistance in cryogenic environments lowers Joule heating losses, improving motor efficiency. Last, most critically, the system operates near the theoretically optimal pressure ratio, resulting in higher cooling capacity, fewer compressors, reduced volume/weight, and significantly lower power consumption, collectively boosting system-wide efficiency.

Key Technical Challenges for Cryogenic Actuators include: First, Flexible Cryogenic Link Technology: a flexible cryogenic link, fabricated from materials with high elastic modulus and thermal conductivity, interconnects the cryogenic compressor and the precooler's cold head. This link provides mechanical support while ensuring efficient heat transfer from the compressor to the cold head. Second, Piston-Cylinder Clearance Sealing at Cryogenic Temperatures: integration of flexible springs with gas-bearing technology to maintain precise sealing under thermal contraction and mechanical stress. Third, Piston Surface Coating Technology: coatings for piston surfaces must balance wear resistance, lubrication, adhesion, and stability in extreme cryogenic environments. Fourth, Cryogenic Valve Design: conventional valve materials (e.g., high-carbon steel, stainless steel, or alloy steel) suffer from low-temperature embrittlement, leading to reduced fatigue life. Enhanced toughness can be achieved via specialized heat treatments, such as cryogenic treatment. Fifth, high-efficiency cryogenic moving-magnet linear motor technology: requires maintaining strong magnetic fields and minimizing thermal losses at cryogenic temperatures to ensure optimal electromechanical performance [7].

We are currently conducting pertinent research, having designed and fabricated a cryogenic DC linear compressor, and is now performing associated experimental testing.

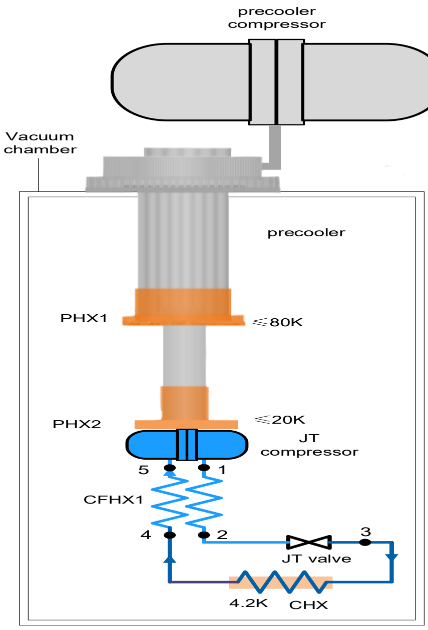

Figure 6. Diagram of novel JT refrigerator driven by cold linear compressor.

4. Conclusion

This paper analyses the refrigeration processes of current state-of-the-art space-borne throttling refrigerators, calculates the optimal pressure ratio for throttling refrigeration systems operating in the liquid helium temperature range, and identifies the critical issue wherein the actual pressure ratio significantly exceeds the theoretically optimal value. A comparative analysis is conducted between engineering-empirical and theoretical-optimal pressure ratios in terms of cooling capacity and compressor power consumption. Furthermore, this study proposes an innovative solution framework involving cryogenic linear compressor-driven throttling refrigeration. The novel throttling refrigeration system demonstrates potential to address the longstanding challenge of excessively high actual pressure ratios, thereby enabling operation closer to the theoretical optimum.

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