A Closed-Cycle Refrigerator for Cooling Maser Amplifiers Below 4 Kelvin

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A helium refrigerator utilizing the Gifford-McMahon/Joule-Thomson (GM/JT) cycle was designed and tested to demonstrate the feasibility of using small closed-cycle refrigerators as an alternative to batch-filled cryostats for operating temperatures below 4 K. The systems could be used to cool low-noise microwave maser amplifiers located in large parabolic antennas. These antennas tilt vertically, making conventional liquid-filled dewars difficult to use. The system could also be used for a non-tilting beam waveguide antenna to reduce the helium consumption of a liquid helium cryostat. The prototype system is adjustable to provide 700 mW of cooling at 2.5 K to 3 W at 4.3 K. Performance of the unit is not significantly affected by physical orientation. The volume occupied by the refrigerator is less than 0.1 m^3. Two JT expansion stages are used to maximize cooling capacity per unit mass flow. The heat exchangers were designed to produce minimum pressure drop in the return gas stream. Pressure drop for the entire JT return circuit is less than 5 kpa at a mass flow of 0.06 g/sec when operating at 2.5 K.

I. Introduction

The DSN currently uses low-noise maser amplifiers operating in 4.5-K closed-cycle refrigerators on 70- and 34-m Cassegrainian antennas. Planetary spacecraft programs planned for the 1990s would greatly benefit from improved downlink performance from these antennas. One strategy for improving downlink performance is to further lower the noise temperature of the antenna systems. Two methods for reducing the system noise temperature are lowering the physical temperature of the maser well below the presently used 4.5 K, and cryogenically cooling the microwave components in the feed system between the feed horn and the maser.

New 34-m beam waveguide antennas are being implemented in the DSN that provide a large non-tipping location for the maser and feed components. For these

antennas, a batch-filled liquid helium cryostat provides a reliable method to accomplish both the above improvements. However, for the existing 70-m and 34-m antennas with feedcone-mounted masers, a new type of closed-cycle refrigerator that will operate at lower temperatures in a tilting environment is required. A similar closed-cycle refrigerator could also be used in conjunction with a liquid helium cryostat in the beam waveguide antennas to recondense the helium boil-off from the dewar and eliminate the need for manually refilling the system with liquid helium. This article describes the initial development of a refrigerator configuration that could satisfy these requirements.

One of the simplest approaches to providing closed-cycle cooling at temperatures below 4 K is to operate a conventional Joule-Thomson/Gifford-McMahon (JT/GM) cooler at a subatmospheric JT return pressure by pumping the gas returning from the JT circuit with a vacuum pump. The final temperature achieved using this configuration is a function of the pressure drop in the return path of the JT circuit heat exchangers and the capacity of the vacuum pump. The cooling capacity available at the final stage is limited by the JT mass flow, which is a function of the capacity of the GM expander and the efficiency of the heat exchangers. Unfortunately, the pressure drop in the heat exchangers also increases as the square of the mass flow, which limits the ultimate cooling capacity for a given operating temperature.

The refrigerator described in this article was designed specifically for operation at subatmospheric JT return pressures. A two-stage JT expansion was used to maximize the available cooling capacity per unit mass flow. The heat exchangers were designed to provide low pressure drop in the return path and to take maximum advantage of the GM engine capacity at each heat exchanger stage. Adjustable expansion valves are used to allow the cooling capacity and temperature to be varied from 700 mW at 2.5 K to 3 W at 4.4 K.

II. Joule-Thomson Circuit Capacity

The refrigeration capacity of the final stage of a conventional Joule-Thomson refrigerator can be approximated by doing an enthalpy balance analysis of a system consisting of the final heat exchanger, the JT expansion valve, and the final heat station (see Fig. 1). If the capacity of the refrigerator is considered to be the constant temperature capacity (i.e., the the maximum heat load that can be applied without warming the refrigerator above the saturation temperature), the fluid at the outlet of the heat station can be considered to be saturated vapor. In this case the refrigeration capacity is equal to the heat of vaporization of the liquid fraction of the flow through the JT valve. The constant temperature capacity is given by the following:

$$q = h_3 - h_2 \quad \text{or} \quad q = h_{\text{sat}} - h_2 \tag{1}$$

where h_{sat} is the enthalpy of saturated vapor at the final heat exchanger return side inlet and h_2 is the enthalpy of the fluid exiting the JT valve. If the JT expansion is isenthalpic then h_2 is equal to the value of the enthalpy leaving the cold side of the heat exchanger. Then, h_2 can be expressed as follows:

$$h_2 = h_1 - q_{\text{he}} = h_1 - (h_4 - h_{\text{sat}}) \tag{2}$$

Where h_1 is the enthalpy of the fluid at the inlet of the heat exchanger, q_{he} is the heat transferred from the warm stream to the cold stream in the exchanger, and h_4 is the enthalpy at the outlet of the cold return of the heat exchanger. Combining Eqs. (1) and (2) gives the following:

$$q = h_4 - h_1 \tag{3}$$

Heat exchanger efficiency is defined by the following:

$$e = \frac{q_{\text{he}}}{q_{\text{max}}} = h_{\text{out}} - \frac{h_{\text{in}}}{h_{\text{out}'}} - h_{\text{in}} \tag{4}$$

where q_{he} is the actual heat exchange between the warm and cold streams in the heat exchanger and q_{max} is the maximum possible heat transfer with no thermal resistance between the warm and cold stream. If the heat capacities of the supply stream and the return stream are different, the heat exchanger is "pinched." This means that the heat transfer between the two streams is limited to the smaller of the two enthalpy changes. The primed enthalpy value indicates the fluid properties that would be achieved if the thermal resistance in the heat exchanger is zero.

In the case of operation near 4 K (100-kpa return pressure) a JT supply pressure can be chosen (1.6 Mpa) so that the heat exchanger pinch that occurs in any of the

stages is small. Equation (4) can be written in terms of the enthalpy change as follows:

$$h_4 = e_{h4'} \tag{5}$$

Substituting this expression into Eq. (3) yields the following:

$$q = e_{h4'} - h_1 \tag{6}$$

This shows that the final capacity of the Joule-Thomson stage is a function of the enthalpy of the fluid entering the final heat exchanger and the heat exchanger efficiency.

At temperatures below 4 K there is no single operating point for the JT supply pressure that will prevent an imbalance of the heat capacity between the supply and return gas in either the final heat exchanger or the second-stage heat exchanger. If the enthalpy change ($h_4 - h_{sat}$) of the final stage return stream is smaller than the enthalpy change ($h_{2'} - h_1$) in the supply stream, the temperature of the fluid entering the expansion valve will be higher. This will result in a smaller fraction of the JT flow being liquified and therefore reduce the capacity. A pinch in the second-stage heat exchanger upstream of the GM second stage will increase the loading on the GM stage and increase the temperature of the gas entering the final stage.

Collins [1] suggested using multiple stages of JT expansion to optimize the heat transfer in the JT heat exchangers as a method to avoid this problem. The heat capacity in the warm and cold streams in the heat exchangers can be balanced more closely by expanding the fluid in several steps along the final heat exchanger. This process results in the enthalpy of the fluid being lowered to the minimum level possible before the final expansion valve. This causes the largest possible fraction of the flow through the final expansion to be liquefied and yields the maximum final stage cooling capacity. Figure 2 shows the schematic of a refrigerator using two stages of JT expansion. Figure 3 represents the cycle on a temperature-entropy diagram. The dotted line indicates the path for a single JT expansion. The area under the constant temperature line at 2.5 K represents the cooling capacity of the refrigerator. The shaded area between the final expansion end points represents the cooling capacity gained by using two expansions.

III. Heat-Exchanger Return-Path Pressure Drop

Return-path pressure drop is a critical parameter in the design of JT refrigerators because it determines the pressure and therefore the temperature of the helium bath in the final stage. Designing heat exchangers for these refrigerators requires a compromise between heat exchange and pressure drop. Factors that improve heat exchange (small cross-sectional area and long path lengths) tend to increase pressure drop.

The pressure drop for heat exchangers with flow across tubes is given by Hogan [2] to be the following:

$$\Delta P = f \frac{L}{D_e} \frac{M^2}{A^2 \rho} \tag{7}$$

where f is the friction factor, which is a function of the Reynolds number that involves the tube size and the fluid properties in the heat exchanger; L is the heat exchanger length, and D_e is the hydraulic diameter of the flow passage; M is the JT mass flow rate, A is the passage cross-sectional area, and ρ is the fluid density. The mass flow squared dependence of pressure drop shows the need to keep the JT flow as low as possible. Table 1 shows the calculated pressure drop for the three heat exchangers in the JT circuit for a nominal mass flow rate of 0.08 g/sec. The pressure drop in the first-stage exchanger is the largest contributor to pressure drop by a factor of 10.

IV. Hardware Description

A prototype closed-cycle refrigerator was built to study the feasibility of using the multiple JT valve concept to obtain temperatures of 2 K and below. The device features a two-stage JT expansion with adjustable expansion valves to optimize cooling capacity as a function of mass flow. The heat exchangers were designed to provide minimum pressure drop and still provide adequate heat transfer. Many of the components used in the fabrication of the prototype such as the final cold stage and the thermal switch used to aid in the cooldown of the system were adopted from existing JPL designs and are described by Higa and Weibe [3].

The entire refrigerator is less than 70 cm long and 20 cm wide and occupies a volume of less than 0.1 m^3. The JT circuit is contained in a cylindrical vacuum housing

that is 20 cm in diameter and 40 cm long. The complete assembly, shown in Fig. 4, weighs 20 kg.

A. Gifford-McMahon Expander

The refrigerator uses a Varian two-stage GM expander designed for cryo-pump applications. The device was designed to provide cooling at 20 K and 80 K with nominal cooling capacities of 5 W and 35 W respectively. The measured cooling capacity of the device as a function of temperature using a JPL 5-hp compressor is plotted in Figs. 5 and 6. The JT circuit heat sinks were bolted to the displacer cylinder flanges rather than soldered directly to the cylinder.

B. Heat Exchangers

The heat exchangers were adapted from those developed for the JPL prototype 2-W, 4-K refrigerator [4, 5]. They consist of a helix of corrugated bronze tubing wound on a Micarta mandrel. This assembly is housed in a tightly fitting stainless steel outer tube. The high pressure fluid travels through the inside diameter of the bronze tubing and the low pressure stream returns over the outside of the tube in the annulus between the inner mandrel and the outer tube in a cross-flow arrangement. The sizes and measured efficiencies of the heat exchangers are tabulated in Table 2.

C. Compressor and Vacuum Pump

A JPL 5-hp two-stage compressor was combined with a 21-l/sec direct-drive vacuum pump (Leybold-Heraeus S 65 BC) to provide the proper operating pressures for the refrigerator. A schematic diagram of the system is shown in Fig. 7. The helium compressor will supply the GM expander with 2.0-Mpa helium and maintain a 700-kpa return pressure. The JT circuit is supplied with 1.0- to 2.0-Mpa gas. Gas returning from the JT circuit is evacuated by the vacuum pump and then returned to the compressor at 110 kpa. An oil separator returns any vacuum pump oil vapor to the pump. The pressure at the vacuum pump inlet can be maintained below 50 torr for any flow rate below 0.25 g/sec, which is the highest practical flow rate for the refrigerator.

V. Performance

The adjustable expansion valves allowed comparisons of the refrigerator's performance using single and double JT expansions. Final-stage heat capacity is plotted as a function of temperature for both cases in Fig. 8. Using a two-stage expansion results in a 30-percent increase in cooling capacity at 2.5 K. These measurements were taken with a fixed mass flow of 0.06 g/sec. Supply pressure for the single-stage measurement was 1.6 Mpa. For the two-stage measurement, the high pressure was 1.6 Mpa for the first stage and 600 kpa for the second stage.

Tests performed with an additional heat load applied to the first GM engine stage show that the efficiency of the first stage heat exchanger could be reduced by 30 percent and still allow normal JT stage operation. Utilizing a less efficient but lower pressure drop perforated-plate-type heat exchanger in the first stage should allow the refrigerator to operate below 2 K.

VI. Conclusions

Tests of the prototype refrigerator confirm the feasibility of using a small GM/JT refrigerator for temperatures of 2.5 K and below. Two of the major factors involved with designing refrigerators for this application are minimizing the JT circuit return-path pressure drop and optimizing the refrigeration capacity per unit mass flow. Ninety percent of the pressure drop in the JT return path occurs in the first-stage (300–40 K) heat exchanger. This pressure drop can be lowered by using a less efficient plate-fin type device and allowing the first stage of the GM expander to absorb the excess heat. The heat capacity per unit mass flow can be improved by using a two-stage JT expansion.

Acknowledgments

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References

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Table 1. Calculated heat exchanger pressure drop

Heat exchanger	Temperature, K	Pressure drop, Kpa
1 stage	175	3.9
2 stage	20	0.24
3 stage	7	0.06

Table 2. Prototype heat exchanger efficiencies and sizes

Heat exchanger	Efficiency, percent	Length, cm	Diameter, cm
1 stage	90	25	3.75
2 stage	97	20	3.10
3 stage	98	35	3.10

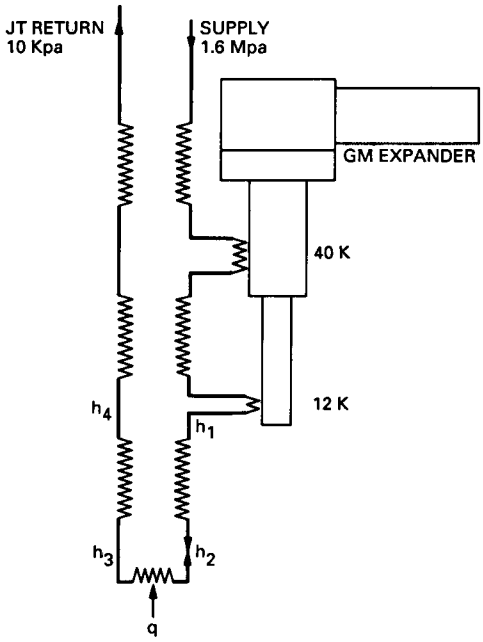

Fig. 1. Single-stage JT schematic diagram.

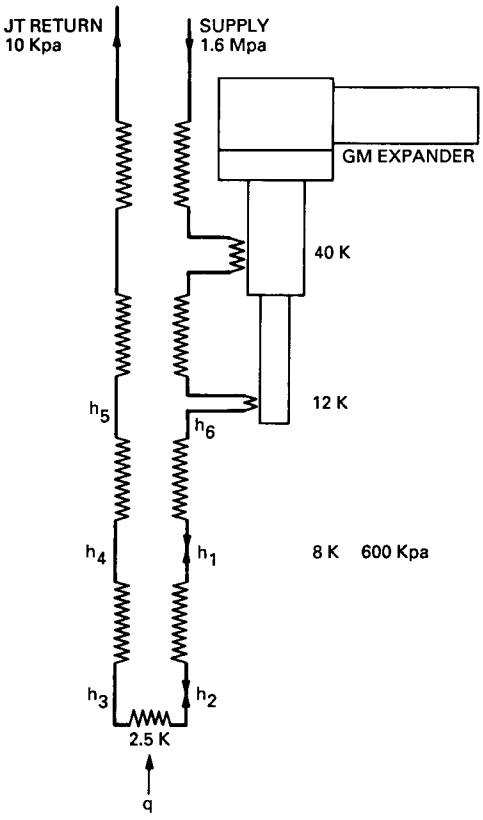

Fig. 2. Two-stage JT circuit schematic diagram.

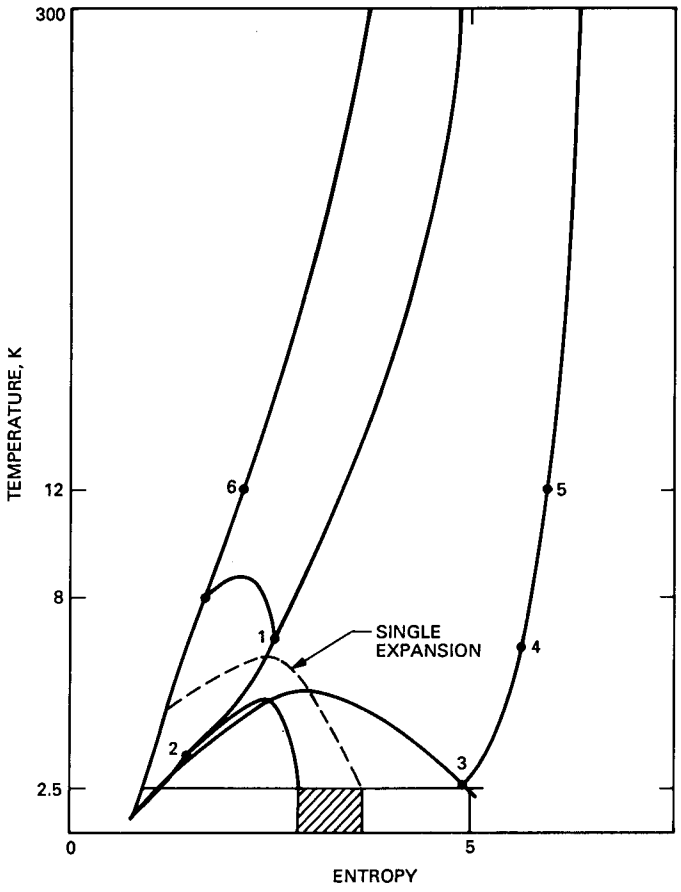

Fig. 3. Temperature-entropy diagram.

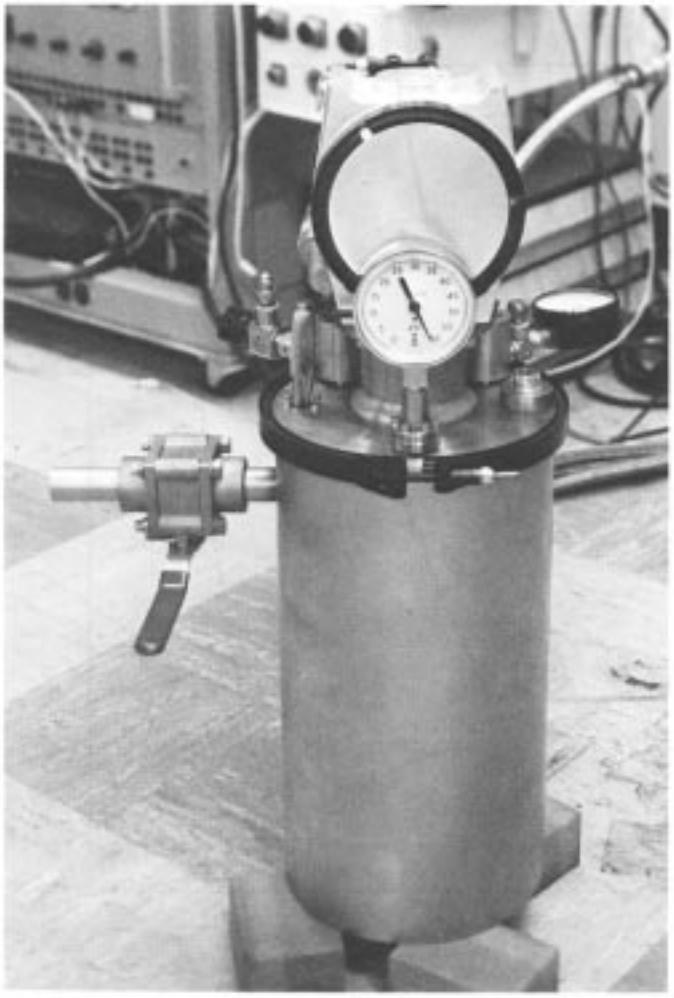

Fig. 4. Prototype refrigerator.

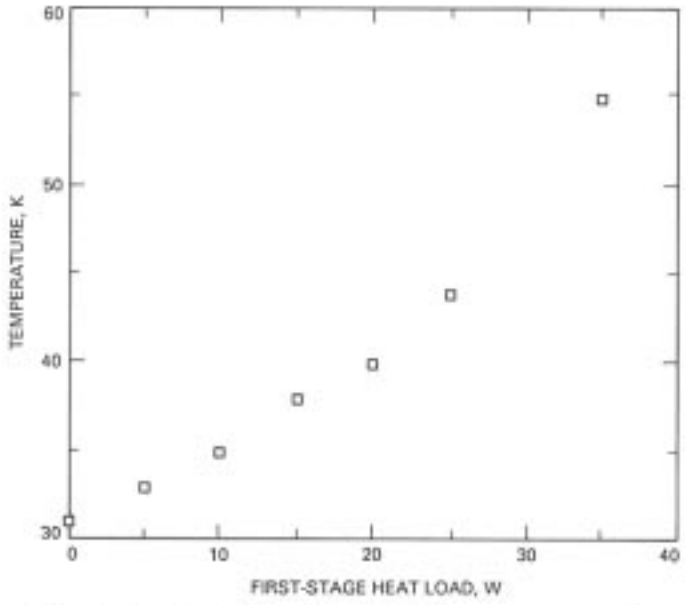

Fig. 5. Varian GM refrigerator first-stage capacity.

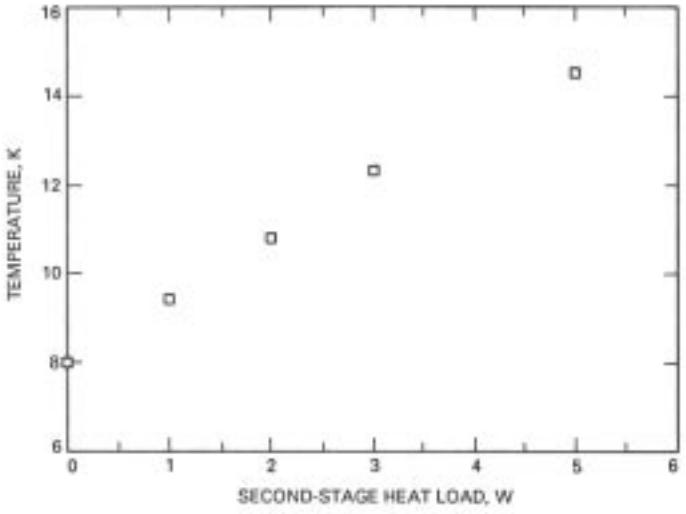

Fig. 6. Varian GM refrigerator second-stage capacity.

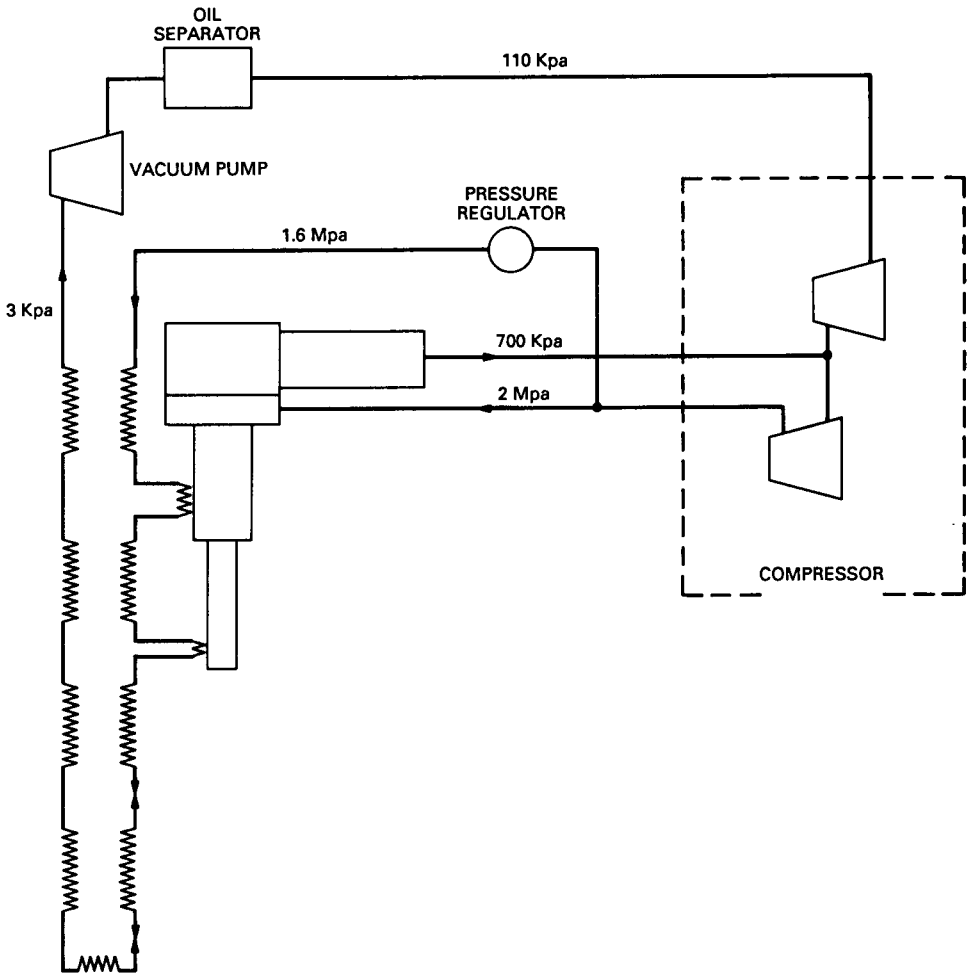

Fig. 7. Compressor and vacuum pump schematic diagram.

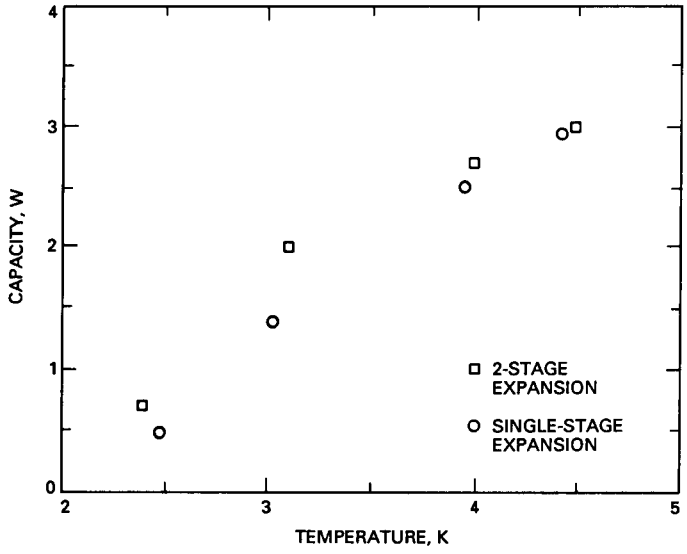

Fig. 8. Final-stage cooling capacity as a function of temperature.