# A Cold Ejector for Closed-Cycle Helium Refrigerators

D. L. Johnson
Radio Frequency and Microwave Subsystems Section

D. L. Daggett
San Jose State University, California

*This article presents the test results of an initial cold helium ejector design that can be installed on a closed-cycle refrigerator to provide refrigeration at temperatures below 4.2 K. The ejector, test apparatus, instrumentation, and test results are described. Tests were conducted both at room temperature and at cryogenic temperatures to provide operational experience with the ejector as well as for future use in the subsequent design of an ejector that will provide refrigeration at temperatures below 3 K.*

## I. Introduction

Ejectors have been used for many years near ambient temperature in refrigeration equipment, boilers, and chemical processing equipment. The principle of the ejector lies in the ability of a moving jet of primary fluid to entrain a secondary fluid and move it downstream. The ejector, then, is a simplified type of jet pump or compressor which pulls in the low-pressure stream and increases the pressure of that stream by mixing it with the high-pressure stream.

Rietdijk [1] first proposed the use of an ejector in cryogenic refrigerators. Through use of a cold ejector in place of a Joule–Thomson expansion valve in a closed-cycle helium refrigerator, subatmospheric pressure can be created in the volume over a liquid helium bath. The ejector pumps on the vapor over the bath, reducing the vapor pressure over the bath and thereby lowering the temperature of the bath. Since this is accomplished at cryogenic temperatures, low-pressure-drop (large-diameter) tubing and heat exchangers are not required to maintain the low pressures in the gas line between the bath and an external vacuum pump at ambient temperatures. Thus, a lower temperature can be obtained without necessitating the use of a vacuum pump or a room-temperature compressor at subatmospheric pressure. This eliminates air leakage, saves power, and permits the use of smaller heat exchangers within the system.

Other benefits of the ejector are that it is a simple, lightweight mechanical device that has no moving parts, does not require additional compressor power, and will not introduce any additional contaminants. By replacing the Joule–Thomson expansion valve with an ejector and using proper thermodynamic optimization, a potential increase in the overall system efficiency of a 4.2-K closed-cycle refrigerator can be realized [1], [2].

One of the authors (D. L. Daggett) recently received a 10-week NASA Summer Fellowship at JPL to work on a cold helium ejector for a 4.5-K closed-cycle helium refrigerator. His task was to perform a one-dimensional thermodynamic analysis for a cold helium ejector and to design a cold helium ejector which, when installed in a 2-watt, 4.5-K closed-cycle refrigerator (CCR), would produce 0.5 watt of refrigeration at 2.5 K. Another objective was to determine if the ejector circuitry could be retrofitted into an existing CCR by installing it in place of the Joule–Thomson valve, leaving all other components in the CCR intact.

The analysis had to take into consideration the geometry of the primary converging–diverging nozzle, the diameter of the nozzle throat, the length and angle of divergence of the nozzle diffuser, the length and diameter of the mixing chamber, and the length and angle of divergence of the ejector diffuser section. Any ejector design resulting from the analysis would likely take weeks to fabricate, as the small size of the nozzle presents special problems in fabrication.

In efforts to gain some early experimental experience with an ejector to aid in the design analysis, an ejector of approximate size and geometry was formulated from the refrigeration characteristics of the 2-watt CCR and from an averaging of the ejector designs discussed in the literature [3]–[8]. This ejector was fabricated and installed in the CCR within the 10-week NASA Summer Fellowship period to provide some test results to aid in the analysis.

The results of the initial ejector–CCR test are included in this introductory report. A forthcoming detailed report will describe at length the analysis of the cold ejector, the ejector design, and the ejector–CCR test results.

## II. Ejector Description

A schematic drawing of the ejector is shown in Fig. 1. The primary nozzle is a converging–diverging nozzle that acts to restrict the flow of gas in much the same way as the Joule–Thomson (J–T) valve in the CCR. The static high-pressure gas enters the ejector at the primary nozzle. As the gas moves through the converging portion of the nozzle, it is accelerated, converting the potential energy of the gas into kinetic energy. If the nozzle is properly designed for the gas flow, the gas reaches sonic velocity at the nozzle throat and continues to accelerate as the gas exits the nozzle. This high velocity jet is directed into a duct (called a mixing chamber) and induces a low-pressure secondary stream into the mixing chamber. The two streams exchange momentum in the mixing chamber and are discharged from the ejector at the end of the subsonic diffuser at an intermediate pressure.

The prototype ejector consisted of a non-adjustable primary converging–diverging nozzle assembly which was attached to a main ejector body (see Fig. 2). Precise concentric alignment of the primary nozzle with the mixing chamber is important in obtaining optimum performance. Thus, a positioning ring was installed around the primary nozzle to ensure concentric alignment with the ejector body. The nozzle was attached to the ejector by means of a stainless steel tube. This allowed some radial movement within the ejector body for the alignment ring to position the nozzle correctly. No attempt was made in this initial design to make the nozzle axial position an adjustable parameter. Rather, the fixed axial position of the nozzle with respect to the mixing chamber inlet was determined from calculations showing the required annular gap between the nozzle and the ejector body to produce a desired secondary flow rate. Differential thermal contractions in the ejector components had to be considered in the proper positioning of the nozzle for 4-K operation.

The ejector body and the nozzle were manufactured from brass. The primary and secondary inlets as well as the outlet area were machined with openings to allow stainless steel tubing to be inserted directly into the body and silver soldered in place. These tubes were used to plumb the ejector into the J–T circuit of the refrigerator. The two body parts of the ejector were soldered together with a low-temperature lead/tin solder for easy assembly and disassembly.

Manufacture of the components required precise machining as a result of the small dimensions and tight tolerances of the primary nozzle throat. Electro-discharge machining (EDM) was selected as the best way to fabricate all of the ejector components. A specially shaped electrode was manufactured to machine the converging portion of the primary nozzle. EDM was also used to form the 0.006-inch-diameter nozzle throat, the 0.030-inch-diameter mixing chamber, and the diffuser cones for both the nozzle and the ejector body.

## III. Refrigerator Description

Figure 3 shows a comparison between the closed-cycle refrigerator with a conventional Joule–Thomson circuit and the modified refrigerator using a cold ejector circuit. In the normal J–T cycle, compressed gas is cooled to below its inversion temperature and expanded through a small orifice. The J–T effect makes use of an isenthalpic expansion of the gas, transforming the high-level static pressure of the gas into kinetic energy as it passes through the small expansion valve orifice. Upon expansion, all of the kinetic energy is dissipated. The net effect is a decrease in the temperature of the fluid (or an increased condensation of the liquid).

Rietdijk [1] showed that it was possible to use part of this otherwise wasted kinetic energy to compress an amount of gas from a secondary loop in the ejector circuit. The high-pressure gas enters the ejector at the primary nozzle (Fig. 1) and is accelerated as it passes through the primary nozzle, causing the potential energy of the gas to be converted to kinetic energy. While the gas expands through the nozzle in a nearly isentropic fashion, a temperature decrease of the gas results. This high-velocity jet stream entrains a low-pressure secondary flow that enters the ejector at the suction port. The two streams exchange momentum in the mixing chamber and are discharged from the ejector at the end of the subsonic diffuser at an intermediate pressure. The discharge will exist as a two-phase fluid. The liquid portion is separated off and forms the secondary mass flow, passing through an optional heat exchanger and through a J–T expansion valve. The helium undergoes a slight isenthalpic expansion in the J–T expansion valve. The J–T valve is used to restrict the secondary mass flow. The liquid from this expansion is collected and used for refrigeration in a secondary pot. The low-temperature vapor from the evaporated liquid flows through the heat exchanger and into the suction side of the ejector to complete the cycle.

Note that in the J–T circuit of the conventional refrigerator, the temperature is dictated by the suction pressure of the compressor. In the ejector circuit, for the same pressure of the external compressor, a lower temperature can be achieved because of the additional pumping effect of the high velocity jet.

## IV. Refrigerator Configuration

The closed-cycle refrigeration system used in the tests described below is shown in Fig. 4. Before modification for the ejector circuit, the CCR with a fixed-orifice J–T valve provided 2 watts of refrigeration at 4.5 K with a mass flow rate of 2.6 scfm through the J–T circuit. The CCR uses a two-stage Gifford–McMahon expansion engine (CTI Model 350) to provide an intermediate refrigeration of approximately 25 watts at 60 K and 5 watts at 15 K. The heat exchangers and the J–T valve were built at JPL. The CCR uses a 5-hp Dunham–Bush compressor to provide 30 cfm of helium gas to the CCR. A detailed description of this refrigerator is presented in [9].

To install the cold ejector circuit, the 4-K station cold plate on which the maser is normally mounted was removed. In its place, a larger 4-K storage pot was installed which allowed ample volume for the phase separation of the fluid. The fixed-orifice J–T valve was removed, and the ejector was installed in its place. An adjustable J–T valve was installed to provide flow restriction in the secondary loop. The J–T adjustment is made using a Starret micrometer that could be adjusted from outside the cryostat. The micrometer was connected by a long, slender stainless steel rod to a finely tapered needle in the J–T valve body. A 10-cc volume copper pot was installed as the sub-4-K station in the secondary loop.

The ejector test measurements included static pressures, temperature, and mass flow at locations shown in Fig. 5. The pressures were measured at ambient temperature using Endevco Model 8530A pressure transducers connected by 0.063-inch-OD capillary tubing to the locations of interest. The intermediate temperatures of the CCR's expansion engine were measured using Lake Shore DT-500 silicon diodes. The low temperatures of the ejector circuit were measured using calibrated carbon glass resistors, also from Lake Shore Cryotronics. Small heater resistors were placed on the 4-K station and the secondary station to measure the refrigeration capacity of each station. A heater was also attached to the ejector inlet stream to permit variation of the temperature of the high pressure primary gas stream. The primary mass flow rate of the ejector circuit was measured on the return stream to the compressor at ambient temperature with a Hastings Linear Mass Flowmeter, Model NAHL-5, calibrated for helium. The flow rate for the secondary gas stream was measured during room temperature testing through the insertion of a small flowmeter in the secondary gas loop. When operating at liquid helium temperatures, the secondary gas stream could not be monitored: the cold secondary flow rate, however, can be estimated given knowledge of the temperature, pressure, and refrigeration capacity of the secondary loop.

## V. Results

The ejector was designed to operate with a primary pressure of 300 psia and a primary mass flow rate of 2.5 scfm, consistent with the operation of the 2-watt CCR with the J–T valve. The compressor return pressure was set to the same level for both configurations such that the ejector exit stream pressure would be 16–17 psia (operating temperature of 4.4 K). Unfortunately, the nozzle diameter was machined larger (0.006 inch) than was specified (0.005 inch), such that the mass flow rate through the nozzle at ambient temperatures was on the order of 0.6 scfm. This large flow rate was a clear indication that at 4 K the gas flow through the nozzle for the 300-psia operating pressure would be greater than that which the expansion engine and the heat exchangers in the circuit would be able to cool effectively. In fact, it would also tax the 5-hp helium compressor.

The low-temperature refrigeration capacity of the secondary loop of the ejector depends on the helium mass flow rate and on the vapor pressure of the secondary stream. A high flow rate yields a high capacity (heat of vaporization of the liquid), and the vapor pressure will determine the operating temperature of the secondary loop. Thus, the ejector operation requires a high rate of entrainment of the secondary stream by the primary stream to provide a large refrigeration capacity and to ensure a low secondary vapor pressure. Therefore, parameters of importance include the pressure $P_1$, the temperature $T_1$, and the mass flow rate $\dot{m}_1$ of the primary flow stream; the suction pressure $P_2$ and the mass flow rate $\dot{m}_2$ of the secondary stream; and the ejector discharge pressure $P_3$. For interpretation purposes, the data is often plotted as the ratio of the ejector discharge pressure to the secondary pressure $P_3/P_2$, and as the mass entrainment ratio $\dot{m}_2/\dot{m}_1$ (secondary mass flow rate over primary mass flow rate). Both $P_2$ and $\dot{m}_2$ can be varied by adjusting the position of the primary nozzle relative to the mixing chamber inlet and by adjusting the restriction of the J–T valve. In these tests, however, the nozzle position could not be adjusted; its fixed position was determined from calculations prior to assembly of the ejector.

The only two parameters that could be adjusted in the tests were the primary pressure $P_1$ and the J–T valve restriction. The primary pressure was varied from atmospheric pressure (14.5 psia) for calibrating the pressure sensors to a high-pressure 310 psia. The high pressure is representative of the upper pressure limit of an operating helium compressor for the CCR. The primary mass flow rate through the nozzle was linear with respect to the primary pressure; the proportionality constant is dependent on the nozzle throat diameter. At 312 psia, the primary mass flow rate was 0.55 scfm for the room temperature measurements. The J–T restriction in the secondary loop was controlled with micrometer adjustment of the J–T valve. The micrometer could be adjusted from a fully restricted state (a 0.108 setting) to fully open (a 0.400 or larger setting).

Figure 6 shows the relationship between the mass flow rate and the suction pressure for the secondary stream as a function of the primary stream flow rate. For a given primary flow rate, the position along the curve gives an indication of the J–T restriction, with a fully restricted valve represented as the lower left end of the curve. From this figure, it is clear that a high primary stream flow rate is required if both a high secondary flow rate and a low suction pressure are desired.

Figure 7 shows the relationship between the pressure ratio and the mass entrainment ratio for several primary mass flow rates. It may be observed from the figure that there is a practical limit in the size of the primary that will optimize both the suction pressure and the entrainment ratio. For low entrainment ratios, the pressure ratio increases with increasing primary flow rates. As the entrainment ratio increases, however, this effect becomes less and less pronounced until finally this relation is reversed, with the pressure ratio decreasing with increasing primary flow rates. This implies that for a given secondary mass flow rate, there is a maximum pressure ratio that may be achieved by varying primary flow rates. To what degree this room temperature relationship also holds at cryogenic temperatures is uncertain, as it was not possible to vary the parameters sufficiently at cryogenic temperatures to measure this effect. However, it is understood that the efficiency of the nozzle and ejector design will have a large effect on the quality of the data.

Sample data sets for the ejector operating at cryogenic temperatures are shown in Table 1. The large flow rate of the primary stream made it difficult to vary the primary pressure, the heat loads, and the J–T micrometer positions (flow restrictions) to any great degree. The primary stream pressure could not be increased to much above 230 psia without warming the ejector. Likewise, the J–T valve could just barely be opened. Opening the J–T micrometer above 0.150 brought the temperature difference between $T_2$ and $T_3$ to near zero. The pressure ratios as a function of the primary pressure for the cold data are very consistent with the pressure ratios for the room temperature data. These data have been plotted in Fig. 8, along with the room temperature data, for two J–T flow restrictions. This correlation may prove useful in providing a dimensional analysis by which an ejector's cryogenic performance may be predicted by room temperature tests.

## VI. Conclusions

The test results of this first ejector design have been very helpful in furthering an understanding of the ejector operation at ambient and cryogenic temperatures. The warm temperature tests clearly showed the ability of the primary jet to pump the secondary gas stream, with pressure ratios reaching 6.5 in this first design. Both the pressure ratio and the entrainment ratio are very dependent upon the primary stream mass flow rate and the J–T restriction. At large J–T restrictions, the pressure ratio of the streams at cryogenic temperatures was quite consistent with the room temperature pressure ratios. This fact could aid in the performance of preliminary room temperature tests of an ejector design to anticipate cryogenic temperature performance.

These initial test results have also been of benefit in the ongoing thermodynamic analysis of an "optimized" ejector design. The flow rate through the ejector nozzle was larger than predicted, and thus the new ejector nozzle throat will be made slightly smaller than the calculations indicate. A new ejector nozzle having a 0.004-inch-diameter throat is currently being fabricated.

# References


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**Table 1. Cold ejector performance data**

| $P_1$, psia | $T_1$, K | $P_2$, psia | $T_2$, K | $P_3$, psia | $T_3$, K | $\dot{m}_1$, scfm | $\dot{Q}_2$, mW | J-T setting | $P_3/P_2$ |
|---|---|---|---|---|---|---|---|---|---|
| 177 | 7.1 | 12.7 | 4.09 | 22.2 | 4.68 | 4.51 | 0 | 0.125 | 1.75 |
| 182 | 7.7 | 12.1 | 4.03 | 21.8 | 4.65 | 4.23 | 0 | 0.115 | 1.80 |
| 201 | 7.7 | 10.9 | 3.95 | 22.5 | 4.70 | 4.66 | 0 | 0.115 | 2.07 |
| 228 | 7.7 | 9.33 | 3.79 | 23.3 | 4.75 | 5.10 | 0 | 0.115 | 2.50 |
| 231 | 7.8 | 9.47 | 3.84 | 23.7 | 4.75 | 5.19 | 0 | 0.115 | 2.50 |
| 201 | 7.7 | 11.0 | 3.97 | 22.4 | 4.68 | 4.64 | 200 | 0.115 | 2.04 |
| 202 | 7.5 | 13.0 | 4.12 | 22.5 | 4.70 | 4.70 | 100 | 0.150 | 1.73 |

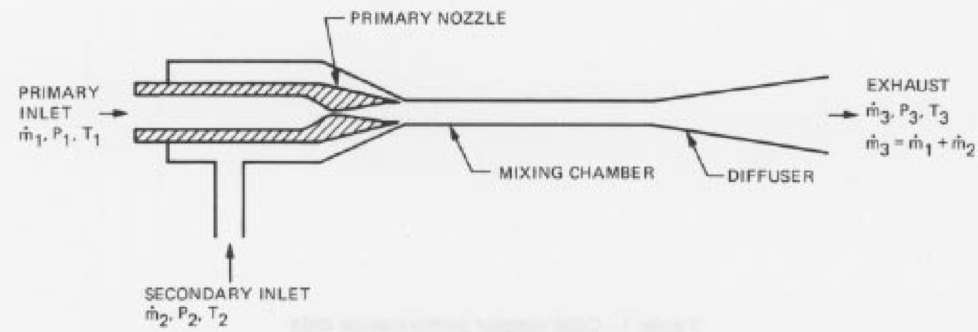


**Fig. 1. Schematic of ejector configuration**

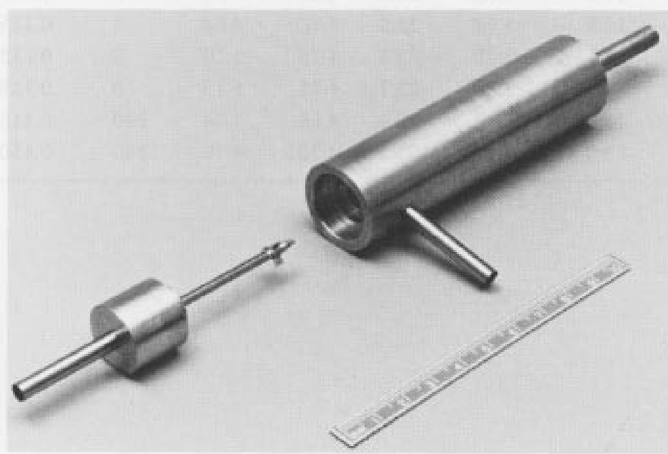

**Fig. 2. Expanded view of cold helium ejector**

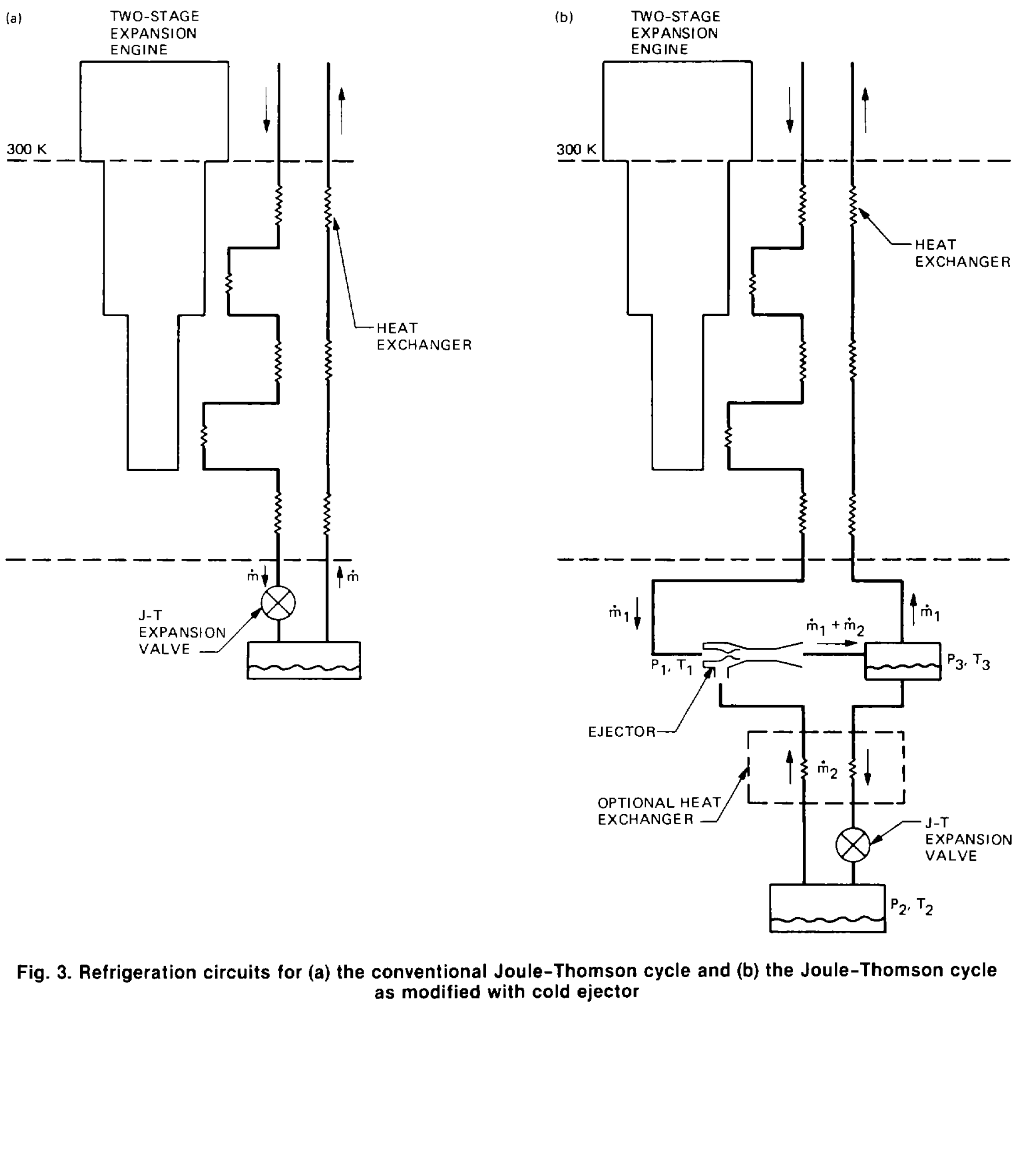


**Fig. 3. Refrigeration circuits for (a) the conventional Joule-Thomson cycle and (b) the Joule-Thomson cycle as modified with cold ejector**

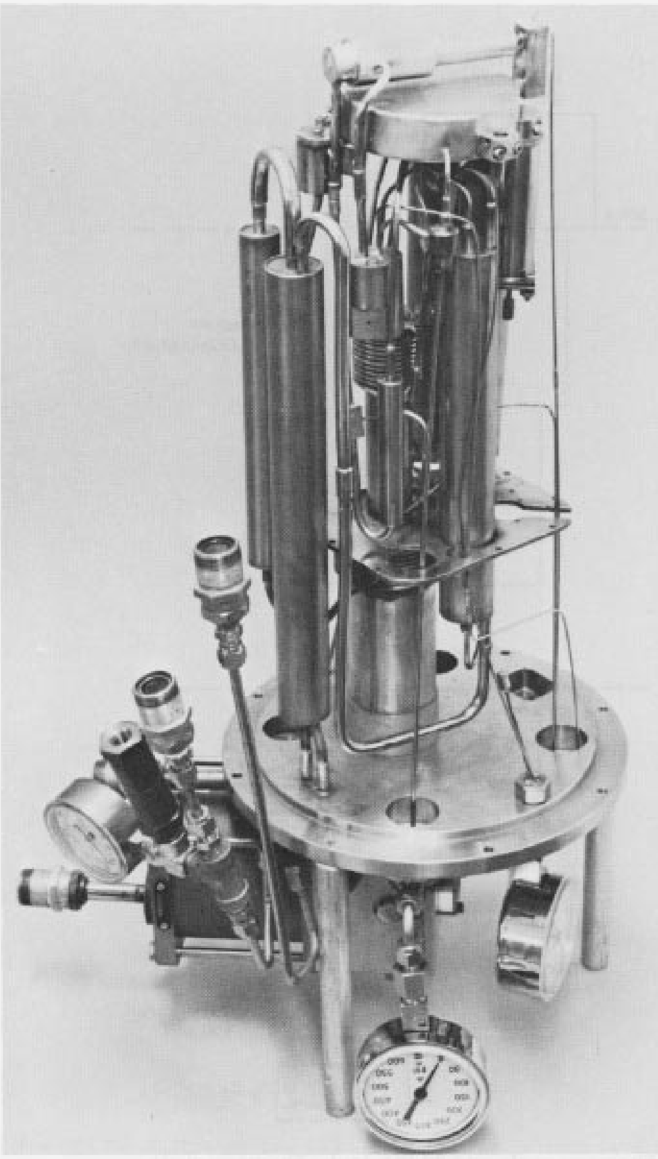

Fig. 4. Closed-cycle helium refrigerator modified with cold ejector for laboratory testing

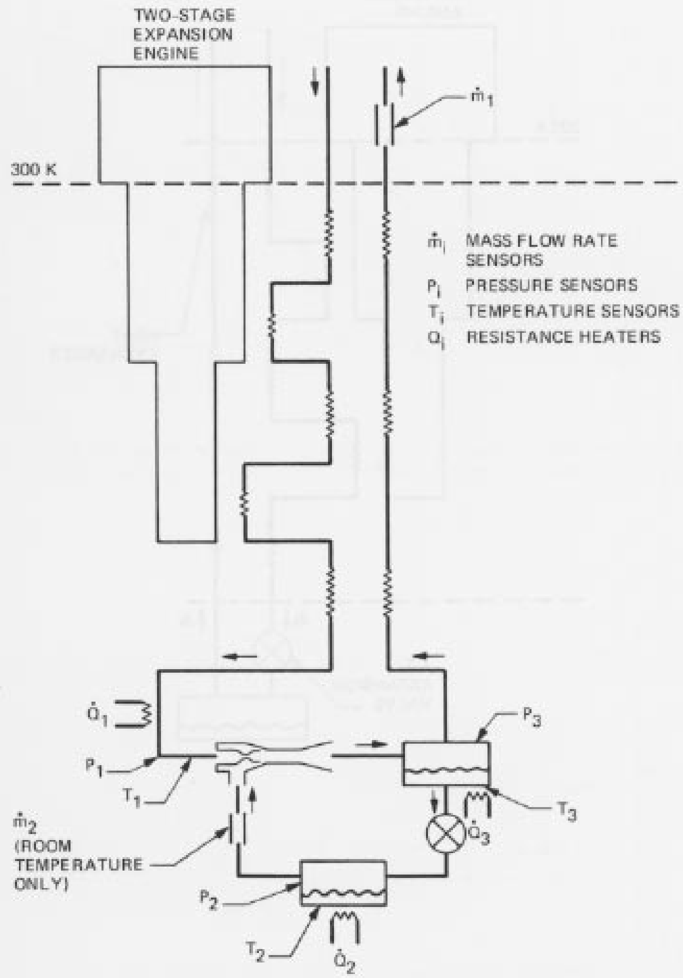


Fig. 5. Refrigerator-ejector schematic showing location of sensors

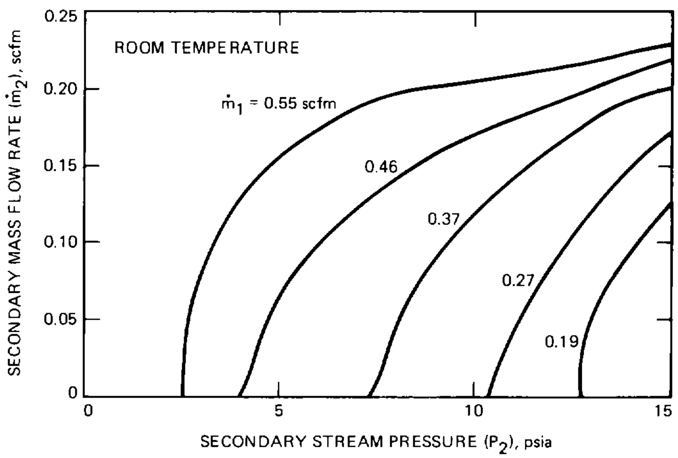


**Fig. 6. Secondary mass flow rate as a function of the secondary stream suction pressure for different primary mass flow rates**

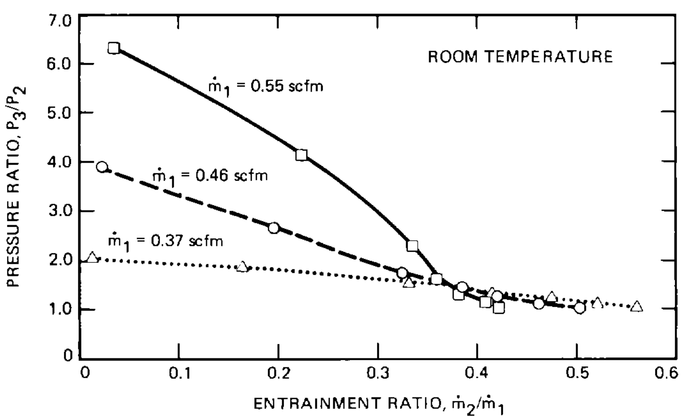


**Fig. 7. Pressure ratio as a function of the entrainment ratio for different primary mass flow rates**

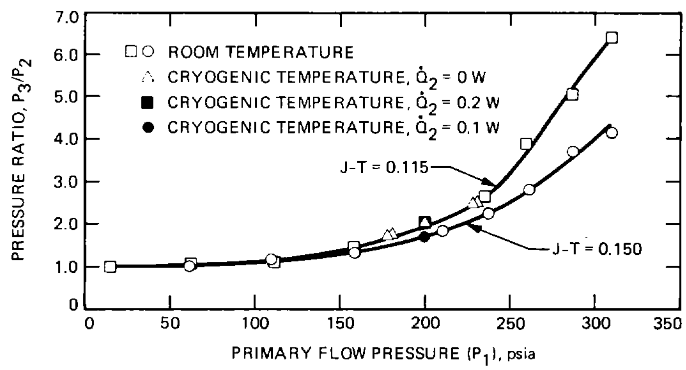


**Fig. 8. Comparison of room temperature and cryogenic temperature pressure ratios as a function of primary stream pressure for two different J-T flow restrictions**