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ABSTRACT

A simple mechanical refrigeration system is described for application to vacuum systems requiring baffle or trap temperatures down to -140°F. The system as described is used to refrigerate the baffles for a 32-inch mercury diffusion pump. At a typical operating condition it will handle a total heat load consisting of 1140 Btu/hour at -132°F and 750 Btu/hour at -37°F.

The system has certain advantages over the common presently applied multirefrigerant cascade technique for attaining temperatures down to -140°F. Only a single refrigerant is used in 2 stages of compression, hence the liquid and suction lines need not be insulated and may be as long as necessary within the limits dictated by line pressure drop. Further, the system is not critically charged and ample quantities of condensate are available for large volume flooded baffles and traps. Standard refrigeration components are used throughout so that the system is easily adaptable for scale-up or scale-down.

Performance data are given for a range of temperature from -120°F to -158°F. A cost analysis is also included to compare the system with CO_2-trichloroethylene refrigeration that was replaced.

A SIMPLE TWO-STAGE MECHANICAL REFRIGERATION SYSTEM FOR COLD TRAPS AND BAFFLES*

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The continuing growth of application of high vacuum in science and industry has emphasized the need for reliable low-cost sources of low temperature for the control of migration of pump fluids and system volatiles. In many specialized vacuum systems, such as certain types of nuclear particle accelerators and mass spectrometers, for example, the use of ultra-low-temperature (-108°F and below) traps and baffles is mandatory. Volatiles that freeze on the cold surfaces of these traps and baffles cannot migrate or raise the system pressure. Thus, lower pressures and cleaner systems are attainable. The classical approach in providing refrigeration for the traps has been the use of either liquid nitrogen (or liquid air) or a coolant composed of solid CO_2 and a suitable circulating medium such as trichloroethylene. In a large laboratory with a number of installations requiring low temperature, the costs of supplying merely the cooling medium can be a major item in the budget. The development of efficient, reliable mechanical refrigeration units was undertaken at the University of California Radiation Laboratory to eliminate the inconvenience and operating expenses inherent in the use of CO_2

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*Work done under the auspices of the U.S. Atomic Energy Commission

and liquid nitrogen. This paper describes the simple two-stage refrigeration system that was developed for installation where CO_2 would customarily be used.

The design specifications were established to fulfill the following conditions:

1. The mechanical refrigeration unit must justify its installation from an economic standpoint.

2. The unit should perform reliably with normal refrigeration maintenance.

3. The unit should perform efficiently, i.e., should carry a substantial heat load, at its design operating temperature of $-120^{o}F$.

4. The system should be flexible in its application: it should be adaptable to both large and small heat loads and to both high- and low-volume evaporators (baffles and traps).

The attainment of the design temperature of $-120^{o}F$ falls in the usual range of application of multirefrigerant cascade systems. Here two or more refrigerants are used in completely separate systems; the evaporator of the high-temperature refrigerant is the condenser of the low-temperature refrigerant. Although simple in concept, cascade systems can become quite complicated in practice, and have certain undesirable features for general application. Presently used low-temperature refrigerants have high saturated vapor pressures at room temperature. To accommodate this characteristic a cascade system must be critically charged, i.e., charged at room temperature with sufficiently small amounts of refrigerant to give safe pressures, or the low-temperature cascade (low side) must be capable of containing the high pressure safely during shutdown periods. Critically charging severely limits the evaporator (baffle) volume and heat-load capacity of a system, since only very small amounts of condensate are available at the expansion device at any one time. A high-pressure low side further complicates the multiple controls required in any cascade system, since the separate control systems must now be interlocked for safety. The cascade condenser, low-side suction, and low-side liquid line will run at low temperatures, $-40^{\circ}F$ and below, and must be thoroughly insulated. This puts a premium on locating the condensing unit close to the evaporator, as even with the best insulation practice, heat pickup in the lines can be excessive in long runs. Not only does the heat leak here cut down efficiency but also -- and more important -- the vapor generated can "vapor lock" the controlling expansion device. The typical cascade condenser also has a temperature overlap between refrigerants of up to $15^{\circ}F$, and this represents an inefficiency or "lost refrigerating effect."

The system described here avoids these difficulties and uses only one refrigerant. It retains its simplicity, largely owing to the thermodynamic characteristics of a new refrigerant, Kulene 131 (Bromotrifluoromethane). Reference 1 shows data for

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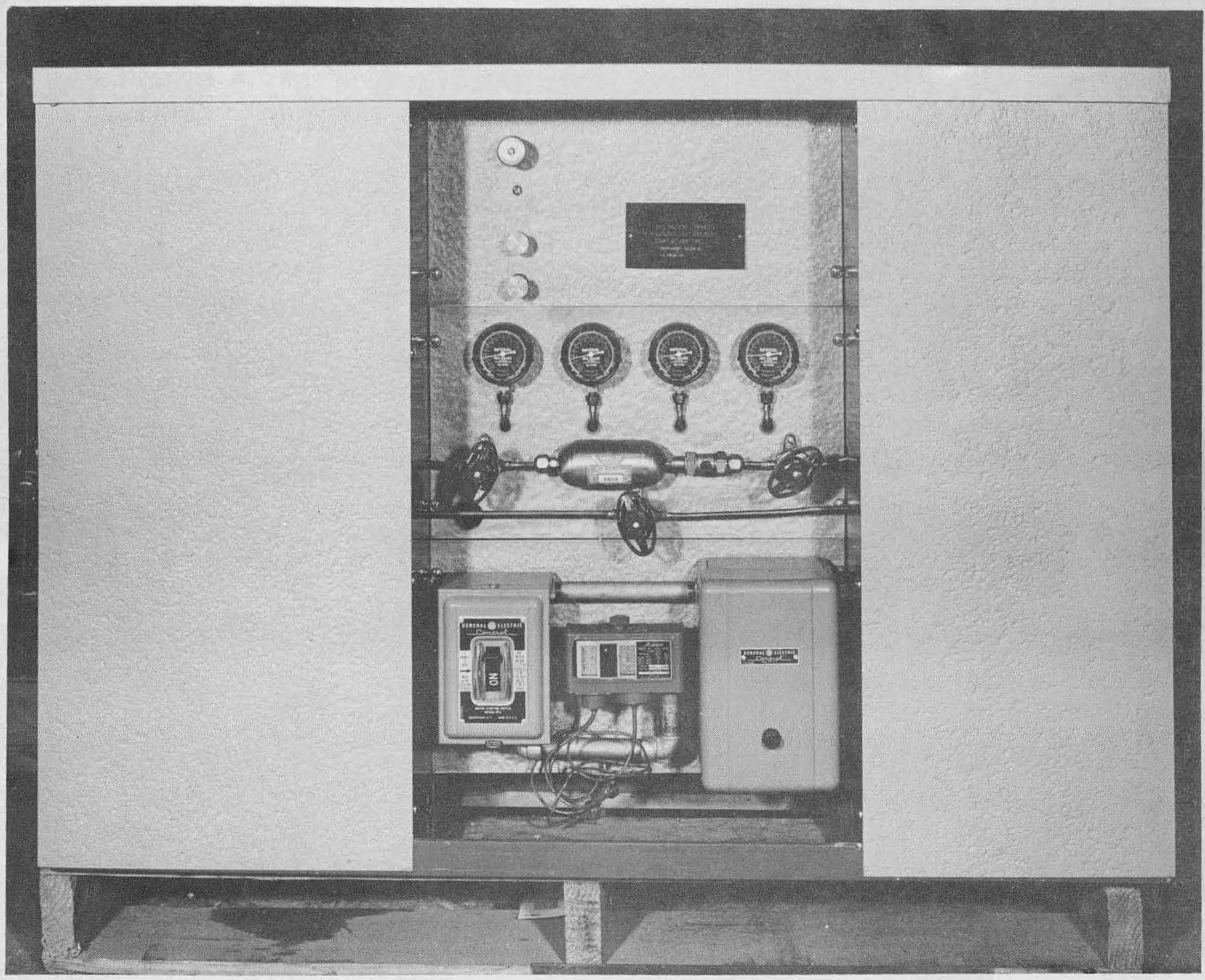
[1]Kulene 131, A New Low Temperature Refrigerant. Eston Chemicals Division, Americal Potash & Chemical Corporation, Los Angeles, California.

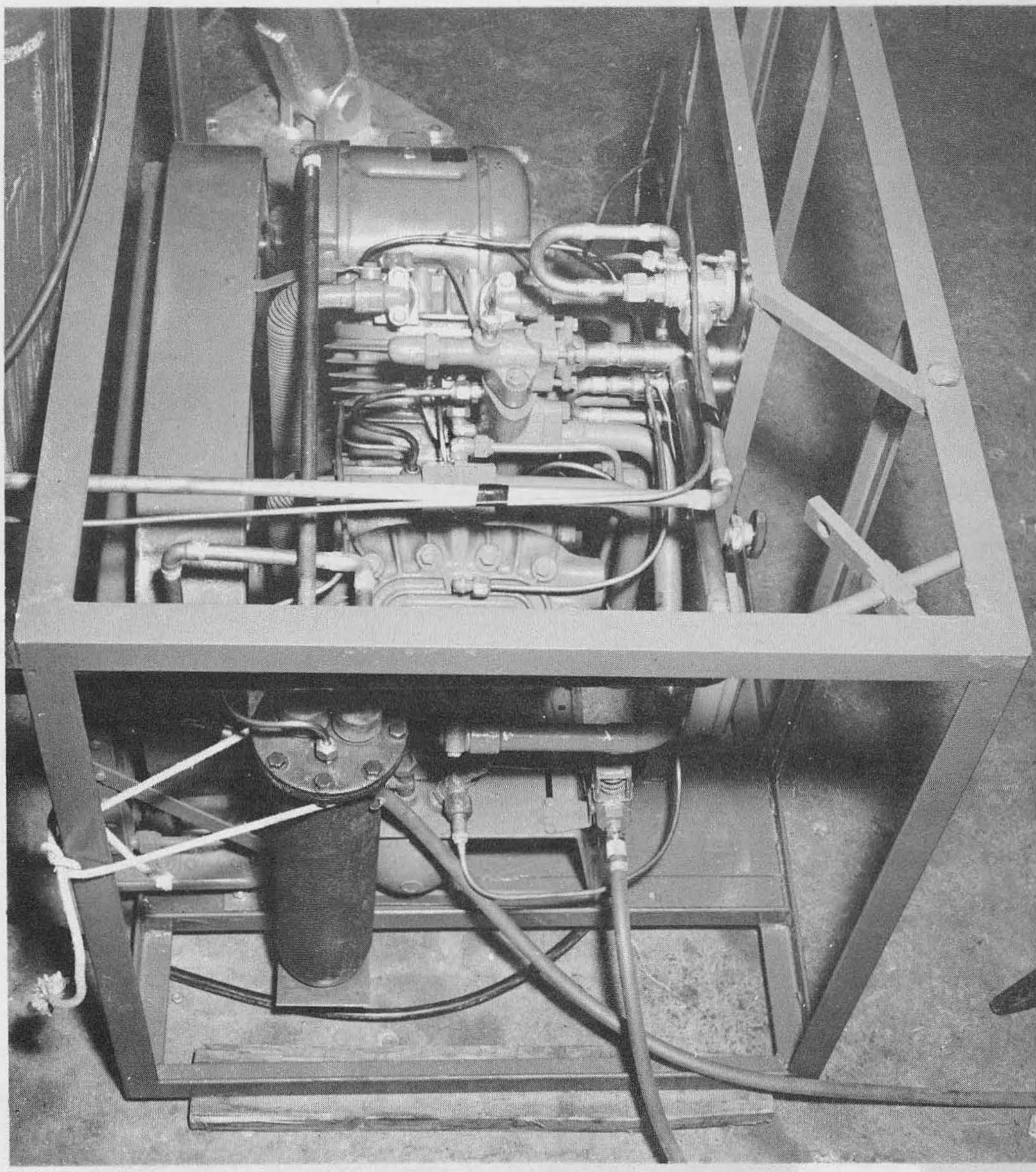
only the high-temperature properties, down to -40°F, of the refrigerant. These data were extrapolated on the basis of paralleling the properties of similarly constituted compounds. The extrapolated properties indicated the suitability of Kulene 131 for compound compression operation down at least to the design operating temperature of -120°F. Subsequent operation confirmed the accuracy of the extrapolation down to -158°F.

Figure 1 shows one of a series of units built for refrigerating the baffles of a 32-inch mercury diffusion pump. The unit is composed of a stock 3-hp 2-stage Freon-22 condensing unit mounted, with its accessories, in a frame for portability. A convenient central panel contains all hand valves, and pressure gages, motor controls, and gives access to the drier. Figure 2 shows the unit with paneling removed. Figure 3 is a schematic drawing of the over-all installation; including the diffusion pump and baffles. The temperatures and pressures indicated at various points represent a typical operating condition and illustrate the important features of the system. The bottom-stage suction, running at 26 inches Hg vacuum (2 psia), provides a baffle temperature of -132°F. The heat load measured at this condition is 1140 Btu per hour. The intermediate pressure is about 7 inches Hg vacuum (11.5 psia), and is used as the suction for maintaining -37°F on the lower baffle. An evaporator-regulating valve (ERV) is used to control the back pressure and hence the temperature of the lower baffle. The corresponding baffle heat load is 750 Btu per hour.



ZN-1403

Fig. 1



ZN-1402

Fig. 2

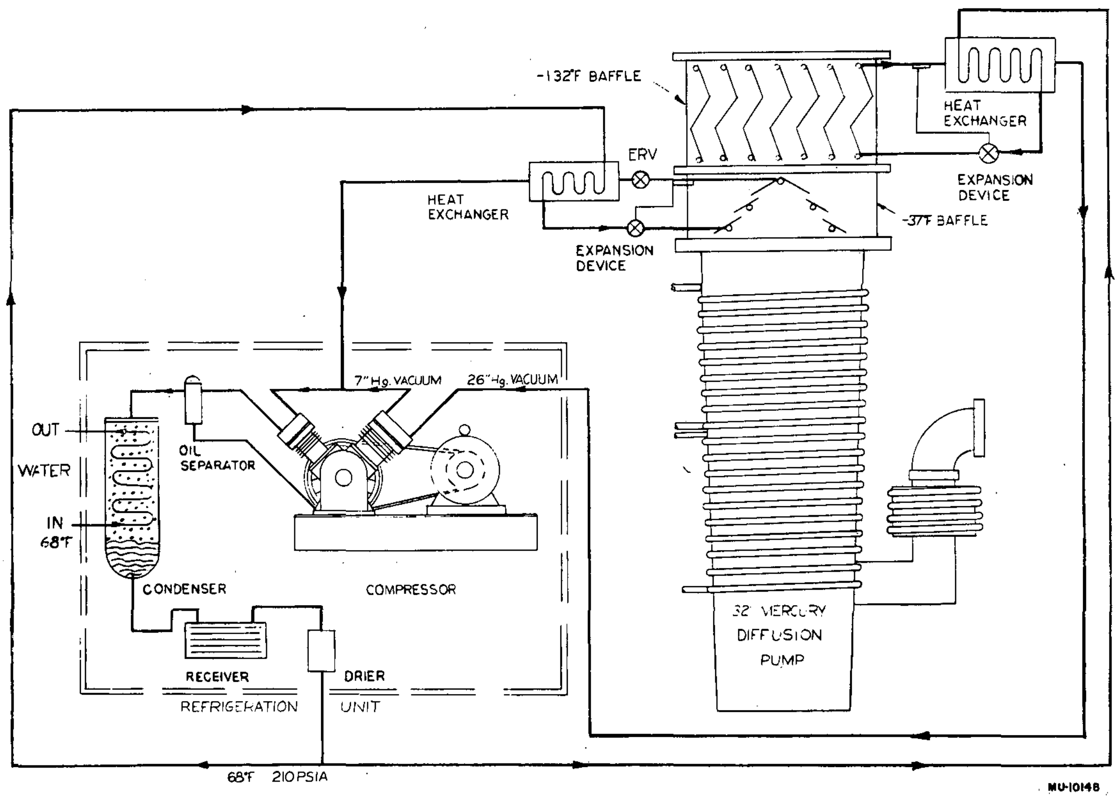

-132°F BAFFLE
HEAT EXCHANGER
EXPANSION DEVICE
ERV
HEAT EXCHANGER
EXPANSION DEVICE
-37°F BAFFLE
7" Hg. VACUUM
26" Hg. VACUUM
OUT
WATER
IN
68°F
OIL SEPARATOR
CONDENSER
COMPRESSOR
RECEIVER
DRIER
REFRIGERATION UNIT
DIFFUSION PUMP
68°F 210 PSIA
MU-10148

Fig. 3

The pumpdown time to reach typical equilibrium conditions is approximately 5 hours. However, a permissible operating temperature for limited periods, $-115^{\circ}F$, is attained in $1\frac{1}{2}$ hours. The long pumpdown reflects the considerable heat content of the typically large-volume baffles. As noted earlier in the paper, the system does not require critical charging and the baffles run flooded with a total system charge of approximately 35 pounds of Kulene.

Two types of expansion devices for controlling the refrigerant into the low-temperature ($-132^{\circ}F$) baffle have been used successfully. A modified float valve in which the entire float valve assembly and float housing are contained in an insulated evacuated jacket has been trouble-free in operation. A standard thermostatic expansion valve with a modified highly sensitive bulb has also operated successfully and is considerably more compact and cheaper. In either case the expansion device is mounted directly on the baffle housing in an insulated assembly including the heat exchanger. This assembly is the only item in the system requiring insulation. The liquid line from the compressor is at ambient temperature and the suction line approaches ambient temperature as it emerges from the heat exchanger. This feature permits long uninsulated runs from the condensing unit without loss of efficiency due to heat leak. The only limitation is dictated by suction-line pressure drop; however, properly sized suction lines have been run up to 60 feet with little effect on system performance.

The typical conditions noted previously on the schematic represent normal operation of the particular installation. To determine the limits of performance of the unit for possible extension to other applications, a series of runs was made with various heat loads on the evaporator. Figure 4 shows a plot of the evaporator heat load in Btu per hour versus temperature. It can be seen that quite substantial heat loads, roughly equivalent to, say, a 20-inch diffusion-pump baffle, can be handled down to -145°F. Below this temperature, the capacity drops off rapidly and it becomes advantageous to use other refrigerants in a different type of system.

Increase or decrease of capacity of the system to match larger or smaller heat loads presents no special problems. The condensing unit herein described incorporates the smallest two-stage compressor known to the authors. However, small single compressors mounted in tandem afford a selection of stage displacements to meet any desired requirement. In this case individual oil separators will be required on each compressor with suitable oil-return lines to maintain proper crankcase levels. For an increase in capacity, a wide variety of multistaged compound compressors are available. A unit should be selected that includes a water-jacketed body and provision for an unloaded startup. It should be emphasized at this point that the construction of this or similar low-temperature systems for trouble-free operation requires a very high level of workmanship. It is particularly appropriate to discuss this with reference to vacuum

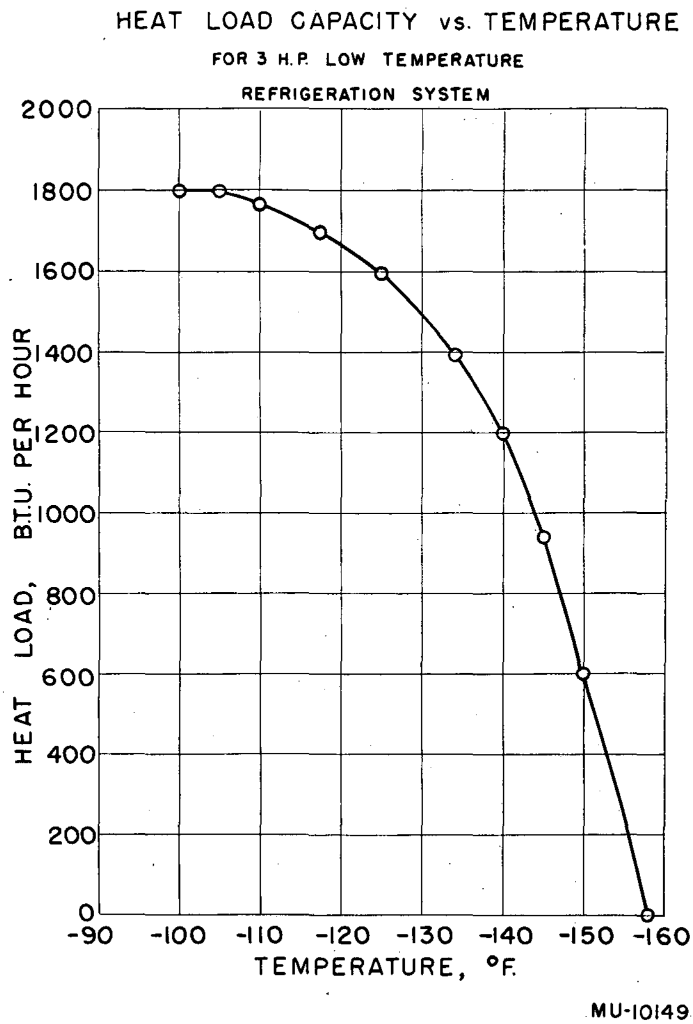

HEAT LOAD CAPACITY vs. TEMPERATURE
FOR 3 H.P. LOW TEMPERATURE
REFRIGERATION SYSTEM
2000
1800
1600
1400
1200
1000
800
600
400
200
0
HEAT LOAD, B.T.U. PER HOUR
-90
-100
-110
-120
-130
-140
-150
-160
TEMPERATURE, °F.
MU-10149

Fig. 4

technique; the same precautions, attention to cleanliness, dryness, and leaktightness are essential to subsequent successful operation.

Eleven of the units have been built and put into service with remarkably uniform performance. Several of them have more than 6 months' running time without servicing or attention. This has afforded an excellent opportunity for preparing a cost analysis comparing their performance with installations still utilizing a circulated CO_2 coolant for their baffle cooling.

Table I shows a cost comparison of the two systems based on a year's operation. Capital equipment is depreciated on the basis of a ten-year term of service, and it is assumed that condenser cooling water goes down the drain. The CO_2 consumption is a measured value and the efficiency of utilization of the material agrees closely with figures provided by the vendor for installations of this type.

Another way of making the cost comparison reveals that in 182 days the conserved operational costs of the CO_2 will pay for the entire capital expenditure for the mechanical refrigeration system.

Several other configurations of the system described, for application to small portable vacuum carts, are in the design stage. It is expected that the extension of the range of practical operation of compound compression systems will lead to considerably wider applications of mechanical refrigeration in high-vacuum technology.

COOLING SYSTEM YEARLY COST ANALYSIS

Heat Load: 1750 BTU/hr. @ -110°F

	Kulene Unit	Remarks	CO_2 Circulating System	Remarks
I. Operating Costs				
Power	$130	$.0085/KW hr.	$ 13.90	$.0085/KW hr.
Water	$315	$.0003/gallon	--------	-----------
Refrigerant	----	------------	$3820.00	$.035/lb.
Servicing & Maint.	$200	1/24 man year Daily gage readings, routine Ref'r. maintenance	$ 400.00	Servicing trap 24 hrs./day 1/12 man year
II. Capital Equipment Depreciation	$191	$1910 total installed cost	$ 22.30	$223 total installed cost
Total system yearly cost	$836		$4266.10	

TABLE 1

MU-10150

We are indebted to Robert Johns and Alfred Bledsoe for valuable contributions in construction and operation of the developmental refrigeration unit.

List of Figures for

"A Simple Two-Stage Mechanical Refrigeration System for Cold Traps and Baffles"