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Authors
Smith, Hugh R.
Kennedy, Patrick B.

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Ernest O. Lawrence Radiation Laboratory

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Introduction

There are several classes of vacuum systems which cannot conveniently be baked but are superficially clean and must operate in the range of pressure from 10^{-5} to 10^{-7} mm Hg. Low-temperature traps and baffles are often used to minimize the pumping time to reach operating pressure as well as to assist in maintaining it. If these systems are not frequently cycled it has been observed[1] that the base pressure down to 5×10^{-7} mm Hg correlates rather closely with the nominal equilibrium value of the water-vapor partial pressure at the temperature of the cold trap or baffle. The partial pressure of water vapor drops below 10^{-6} mm Hg at approximately -166° F, and thus refrigeration of traps at this temperature and below is of particular interest. Liquid air or liquid nitrogen are the most commonly used coolants. However, the high cost of supplying them continuously has encouraged the development of ultra-low temperature refrigeration systems to cool the traps and baffles.

Ultra-low temperature is a relative term and has changed in meaning over the years as advances in refrigeration techniques have provided ever lower temperatures. As used in this paper ultra-low temperature applies to the temperature range between -130° F and -240° F. This range falls between the temperatures customarily obtained by ordinary two-stage refrigeration systems and those obtained by cryogenic methods at -240°F and below.

The applicable techniques of mechanical refrigeration for achieving and maintaining ultra-low temperatures are, in principle, well known. It is the purpose of this paper to describe practical methods of designing and operating mechanical refrigeration systems within the framework of commercially available components and refrigerants.

Types of Systems

The usual approach to the attainment of ultra-low temperatures by mechanical refrigeration is the cascade system. This employs a series of refrigerants of successively lower boiling points, each cycling in a closed system. Each refrigerant system is used to condense under pressure the gas of the next lower boiling point. The evaporator of the lowest boiling point refrigerant then handles the heat load. Most commercial units for ultra-low temperature employ this general system.

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

Figure 1 is a schematic drawing of a triple-cascade system using Refrigerants 12, 13, and 14 in single stages of compression in each cascade. A considerable range of operating temperatures can be obtained in a system using these three refrigerants depending on the relation of the capacities of the individual cascades. The unit shown in Figure 1 based out at -240° F on essentially no load, i. e., a thoroughly insulated evaporator coil.

Figure 2 is a schematic drawing of a double cascade with two stages of compression for each refrigerant. This unit employs Refrigerant 22 and Refrigerant 1150 (ethylene). The advantages of this system over that shown in Figure 1 are:
(a) simplification of the refrigerant controls, and (b) more compact construction since both compression stages of a cascade can be contained in a single compressor or closely coupled units. This system carries a load of approximately 2000 Btu/hour at -210° F. It is a typical installation for cooling the baffle over a 32-inch mercury diffusion pump. Figure 3 shows the compact arrangement of the compressors and components for the low-temperature cascade.

The large pressure tanks indicated in Figures 1 and 2 show one of the methods for maintaining reasonable off-cycle pressures in the low-boiling refrigerant cascades. The low-boiling refrigerants all have high saturated vapor pressures at room temperature. The additional volume provided by the expansion tank accommodates the refrigerant at a reasonable off-cycle equalized pressure. Another method is to charge the system at room temperature to a safe operating pressure. This is somewhat impractical for any but very small traps and baffles since the supply of condensed refrigerant is limited and only very small evaporator coils can be supplied.

Figure 4 represents schematically a mixed refrigerant system which was conceived by the authors in an attempt to overcome some of the inherent disadvantages of the cascade system. Briefly, a low-boiling refrigerant (which, in general, has a very high equilibrium pressure at room temperature) is mixed in a common system with a high-boiling refrigerant with which it might be paired in a cascade system. The mixed gases are compressed and pass to a standard water-cooled condenser where the high-boiling-point refrigerant is largely condensed out. The remaining gas mixture, rich in low-boiling-point refrigerant, proceeds through another condenser or stripper which is cooled by some of the condensate from the first condenser. This removes most of the remaining high-boiling-point refrigerant which falls back as liquid into the condenser as well as desuperheats the low-boiling-point refrigerant. The gas then passes to a final condenser which is cooled by the high-boiling condensate and is condensed there. This condensate is then used to refrigerate the evaporator coil from which it flows as gas back into a common suction line with the high-boiling gas. The "cascade" is thus contained in a single refrigeration system. This provides the advantages of simplified controls, single compressor units, and a large system volume available for expansion of the low-boiling refrigerant on the off-cycle. A less obvious but no less important advantage is the complete separation of compressor crankcase lubricating oil from the gaseous refrigerant. Although the major portion of the separation is accomplished in the oil separator, as will be described in a later section, any residual oil is removed completely in the stripping condenser.

Five of these units have been built using Refrigerant 12 and Refrigerant 1150 in a two-stage compression system. With these refrigerants, temperatures of from -170° F to -200° F are maintained in baffles for mercury pumps of various sizes. Figure 5 shows the final condenser, heat exchanger, and control assembly mounted on the face plate of a 32-inch mercury pump baffle. This closely coupled arrangement eliminates the necessity for runs of insulated line. Thus the compressor can be located remotely. The box around the assembly is sealed off and filled with powdered insulation to keep heat gains to a minimum.

The three types of systems described do not, of course, exhaust the possible combinations of refrigerants and compressors. They were chosen from experience as good examples of ultra-low temperature practice.

Refrigeration Components and Procedures

As indicated previously, the principles of obtaining low temperatures with mechanical refrigeration are well known. The degree of success achieved as a practical matter is largely dependent on technique. The following sections will describe procedures and practices developed at this Laboratory over the past five years for achieving reliable operation of ultra-low temperature systems.

Many of the problems associated with the types of installations described in the preceding paragraphs are common to all of them. Probably the three most important are associated with system leak tightness, separation of the compressor crankcase oil from the gaseous refrigerant, and the control of the refrigerant flow.

Virtually all ultra-low temperature systems of a size appropriate for vacuum system application operate with the load evaporator and compressor suction at subatmospheric pressure. Any leakage from the atmosphere will be non-condensable and will gradually build up head pressures to the point that the system must be purged and recharged. Since the leakage is cumulative, the evaporator, suction-line piping, and compressor must be constructed to meet standards of leak tightness customarily employed in high-vacuum systems. It is unfortunate that typical refrigeration components are not produced to such standards. As a result, considerable effort is required in component modification and testing to ensure satisfactory service. Figure 6 is a typical specification outlining a system pressure and vacuum testing procedure.

The separation of compressor crankcase lubricating oil from the refrigerant as it leaves the compressor is not a unique problem with low-temperature systems. However, the results of ineffective separation can be considerably more serious owing to the total solidification of the entrained oil at the lower temperatures. This often results in blocking of the orifices in the refrigerant flow controls. The severity of this problem varies with the type of low-temperature refrigerant. It is much more severe with fluorocarbons than hydrocarbons. This seems to be related to the mutual solubility of the oil in the refrigerants and the freezing point of the resulting solutions and mixtures. The dissolving away of solidified lumps of oil by flowing liquid hydrocarbon refrigerants at low temperatures has been observed in a transparent section of tubing preceding an expansion valve. Such erosion did not occur under the same conditions with fluorocarbon refrigerants. Most of the oil is carried out of the compressor into the system during start-up when the high pressures and high gas velocities cause maximum oil entrainment. Available commercial oil separators are largely ineffective during this critical period when they actually should be most efficient. Commercial designs, in general, do not seem to be based on sound physical principles for the separation of entrained droplets from a flowing gas. Figure 7 indicates an improved configuration featuring a large volume for decreasing the gas velocity, a large effective surface area of copper-foil strip for oil condensation, and a porous metal filter with 10-micron openings over the outlet. This type of separator has proven to be quite effective in decreasing oil carryover.

Refrigeration control as a general subject has been extensively described in standard references.[2,3] Design of control systems for ultra-low temperature installations involves certain additional considerations, namely,

a. There are no refrigerant control devices manufactured specifically for ultra-low temperature operation; and

b. In refrigeration units of the capacity required for typical high-vacuum traps and baffles, very small flow rates of refrigerant must be accurately metered.

Figure 8 is a schematic drawing of a single stage of a cascade unit emphasizing the control elements typically used. The following paragraphs will elaborate on the characteristics of the various controls in relation to their operation on ultra-low temperature systems.

The metering device for liquid refrigerants may take many usable forms, but these vary widely in over-all function, degree of reliability, and complexity. Liquid-level float controls have operated satisfactorily but are cumbersome and require relatively large amounts of liquid refrigerant for actuation. The capillary tube is ideal for controlling very low flow rates. It also permits direct off-cycle pressure equalization, but is very susceptible to clogging with even small amounts of frozen oil and, of course, controls for only one heat load. The automatic expansion valve has excellent characteristics for system start-up but does not provide for off-cycle pressure equalization. Also the valve can only be applied to systems in which the evaporator-coil heat load and compressor capacity are carefully matched. Otherwise liquid refrigerant can flood all the way through the evaporator and suction piping to slug the compressor. A more complicated version of this type of valve, developed by R. Johns of this Laboratory, utilizes a charged bulb at the outlet of the evaporator coil to provide the pressure signal to actuate a sensitive electrical switch. The switch in turn operates a solenoid-actuated refrigerant metering valve. This valve eliminates the possibility of the liquid refrigerant flooding back. However, since the evaporator pressure measurement is related to temperature, additional control must be provided for start-up. The most satisfactory metering device has proven to be a standard thermostatic expansion valve with a special bulb installed at the evaporator outlet. Figure 9 illustrates the bulb construction. Very close superheat control is obtained because of the good heat transfer between the gaseous refrigerant and the temperature-sensitive bulb fluid. The bulb is charged during operation, so that when optimum control conditions have been reached it can be sealed off.

The evaporator pressure-regulating valve is used to maintain close pressure, and hence temperature, control on the evaporator. This permits initial installation of excess compressor capacity without affecting evaporator temperature. As compressors wear and lose volumetric efficiency over a period of time, the regulating valve will keep the baffle temperature constant. Most commercial valves of this type are designed for high volumetric throughput and relatively high suction pressures. For low-temperature use they must be modified by reducing the size of the flow orifice and decreasing the reference-pressure spring size. Appropriate modifications for several commercial valves are described by Austin.[4]

The crankcase pressure-regulating valve assists in start-up by limiting the compressor suction pressure to a preset maximum value. This is necessary on low-temperature systems because of the high off-cycle pressures of the low-temperature refrigerants. Typical start-up settings fall between 10 in. Hg vacuum and 10 psi gage.

The pressure extremes of compressor operation can be controlled with standard commercial high-low pressure controls. In the case of a two-stage cascade, the operating suction pressure of the low-pressure stage may well be below the sensitive range of the control. In this instance the low-pressure control should be connected into the compressor intermediate and set to actuate at a pressure corresponding to the desired low-pressure cut-out point.

The refrigerant expansion tank with its associated controls is unique to low-temperature systems. It is not strictly a part of the primary control system but its functions during start-up and off-cycle justify a discussion of its operation here. As soon as the thermostat on the cascade condenser indicates that the temperature is low enough for condensation, the compressor starts up. Since the refrigerant throughout the system is warm, high compressor discharge pressures will result. The high-pressure control valve at the entrance to the expansion tank is set at a value somewhat less, say 25 psi, than the high-pressure cut-off on the compressor. As the pressure exceeds this set value the valve opens and the gas bleeds off into the expansion tank. The outlet of the tank is a capillary tube so that the gas return to the system suction line is very slow. As the system gradually approaches the operating condition, the pressure in the expansion tank will slowly come down to the operating suction pressure. The evacuated tank is then available for expansion of the high-pressure gas, should the system be shut off for any reason.

Some discretion must be exercised in the selection of compressors for low-temperature systems. The heats of compression of most low-boiling refrigerants are high. Unless the compressor has a water-cooled body or head, the temperature will reach levels at which the refrigerant decomposes. This results in excessive carbon formation on the valves, pistons, and head, causing eventual compressor failure. It is also helpful to bypass a small amount of liquid refrigerant through a capillary tube from the condenser into the compressor suction. This is particularly effective in cooling the compressor at low mass flow rates when the cooling effect of the main stream of refrigerant is negligible. Most compressors in sizes suitable for low-temperature application are not designed for subatmospheric use. As a result, the valve plate must often be modified with lighter valves and springs so that it will actuate with the compressor suction under vacuum. As a corollary, most of the compressor gasketing on the suction side must be replaced to achieve vacuum tightness.

Design and Operation

The preceding sections have described several alternative types of ultra-low temperature systems. Observations were made on the performance of components for these systems. The way in which this information is incorporated into an integrated design will be the subject of the following paragraphs.

The refrigerants are selected primarily on the basis of desired baffle temperature. Thus, the lowest temperature refrigerant is selected first and the higher temperature cascades are matched successively to it. The heat load at the desired baffle temperature is translated into cubic feet of gaseous refrigerant which has changed phase at that temperature. Tabulated data on the thermodyanmic properties of refrigerants provide this information. Compressors are then chosen for the volumetric displacement necessary to handle the gas at the suction pressure corresponding to the design temperature. Usually some excess displacement is provided as a safety factor. The heat exchangers, condensers, control devices, etc., are sized out by conventional methods based again on the flow rate of the refrigerant.

The assembly of the system is extremely important. The highest quality of workmanship typical of that associated with high-vacuum systems is necessary. Flare fittings should be avoided and all joints should be silver-soldered if possible. Figure 6 outlines the thorough testing procedures which must be observed.

The design of the baffle or trap to be cooled is primarily a high-vacuum problem. However, it must be compatible with sound refrigeration practice; the following principles should be kept in mind. The runs of evaporator coil should be under 50 feet in length to minimize pressure drop. Parallel coils connected to large liquid and suction headers have proven to be a very satisfactory construction. The large suction header prevents splash-over of liquid during start-up and provides a liquid reservoir which assists in maintaining a flooded evaporator at steady-state conditions. The baffle should be fabricated of high-conductivity material or material of thick section so that there are very small variations of temperature across the baffle surfaces between adjacent coils. The refrigerant connections into the vacuum system should be of re-entrant geometry to provide a long, low conductivity path to the vacuum wall. Otherwise the heat conduction at this point can be a serious addition to the refrigeration load. Thermocouples should be soldered to the evaporator both at the inlet and outlet. These help greatly in evaluating the system performance on start-up and are useful as continuous monitors of baffle or trap temperature.

Most of the ultra-low temperature refrigeration systems in use at this Laboratory refrigerate diffusion-pump baffles on large particle accelerators. It is important that they do not contribute to accelerator down-time. Experience has shown that few system difficulties occur precipitously, but instead develop over a period of time. Hence a systematic procedure has been set up for routinely checking system performance, and a small group of specially trained people is available for handling problems that arise. Baffle temperatures are continuously recorded, system pressures are read daily, and the compressor and accessories are inspected once a week. Any changes in operating conditions are obvious, and necessary remedial action can be planned to fit in with scheduled accelerator maintenance. Another point to be made here is that these rather complicated pieces of equipment do not and should not be expected to have the same reliability factor as a household refrigerator. The mental transition in accepting this fact may be eased by considering the analogy with a complex piece of electronic gear. No one expects such equipment to be as trouble-free as a household radio. As a result, maintenance time is accepted as a routine matter and skilled technicians are provided to service the equipment.

At a laboratory for nuclear physics research the development of ultra-low temperature refrigeration systems must be justified from an economic standpoint. Figure 10 compares the costs of the three different types of systems described in earlier sections with the liquid nitrogen coolant they replaced. It is clear that for semi-permanent installations, the economic advantages of the refrigeration systems are substantial. In slightly over a year, decreased operating costs compensate for the capital expenditure for the refrigeration equipment.

The authors are indebted to John Muzinich and Robert Johns for valuable contributions in the construction and maintenance of this equipment.

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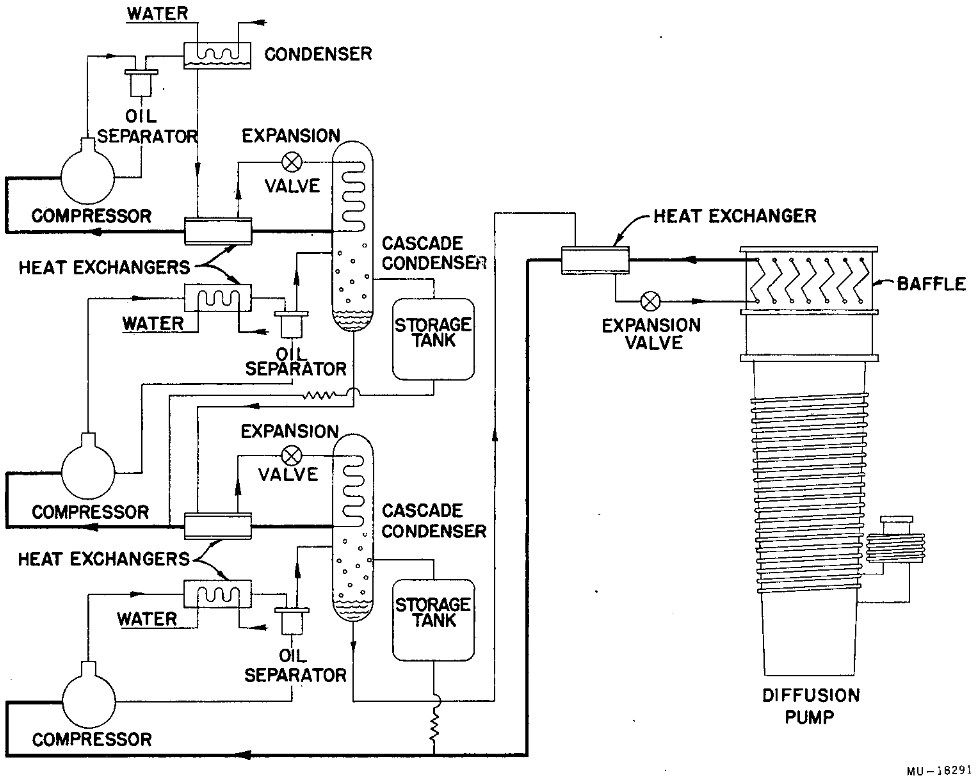
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STORAGE
TANK
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HEAT EXCHANGERS
CASCADE
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WATER
OIL
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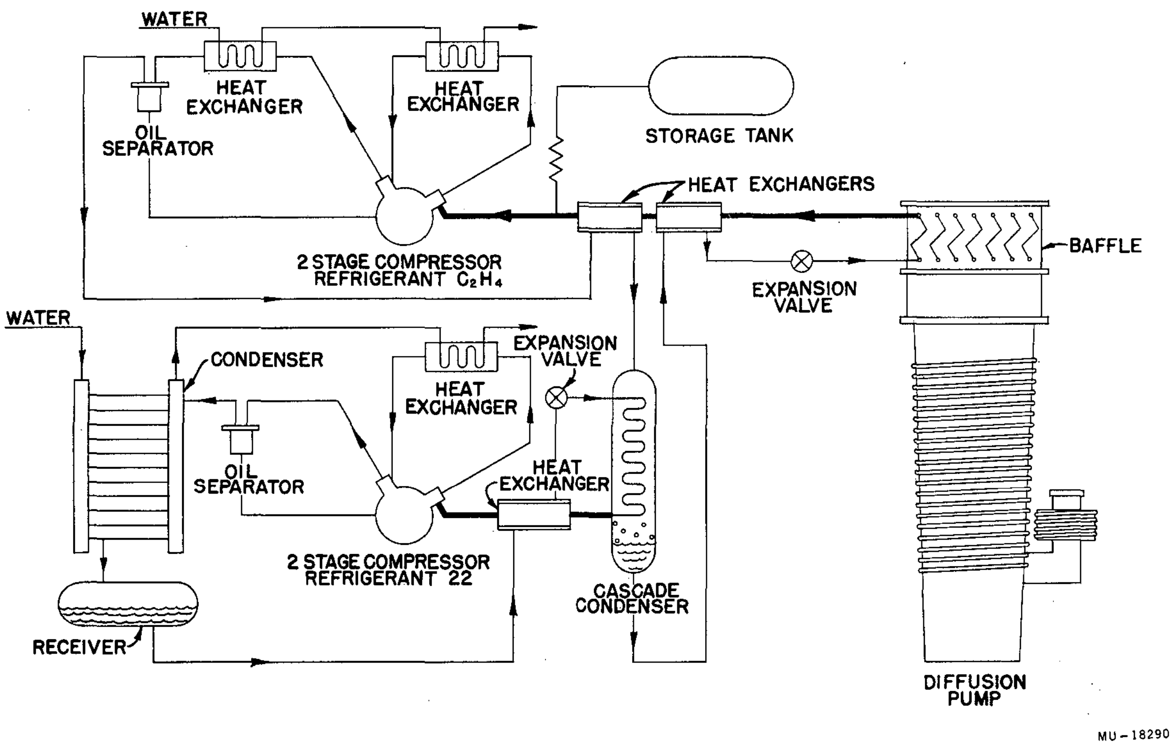
HEAT EXCHANGER
EXPANSION
VALVE
BAFFLE
DIFFUSION
PUMP
MU-18291

Fig. 1.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STORAGE TANK
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BAFFLE
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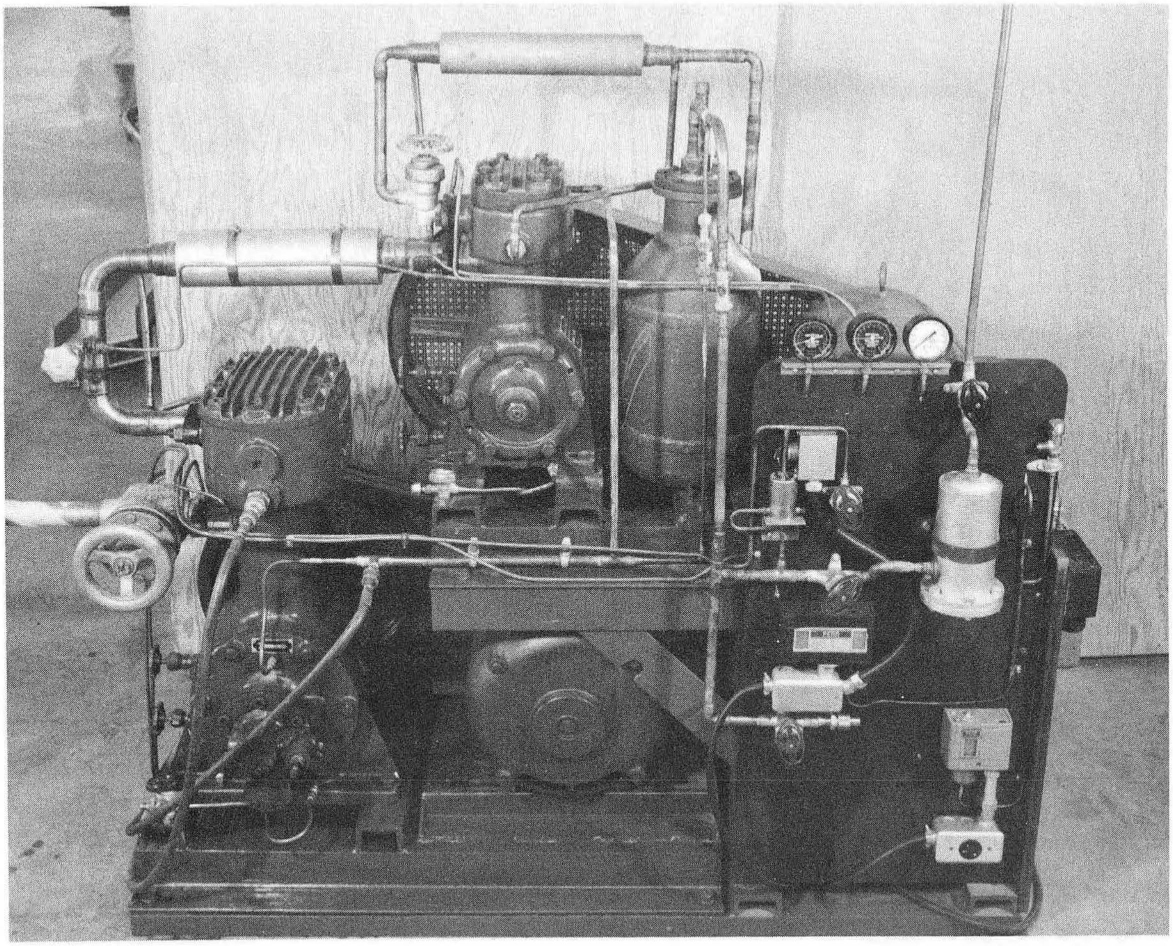
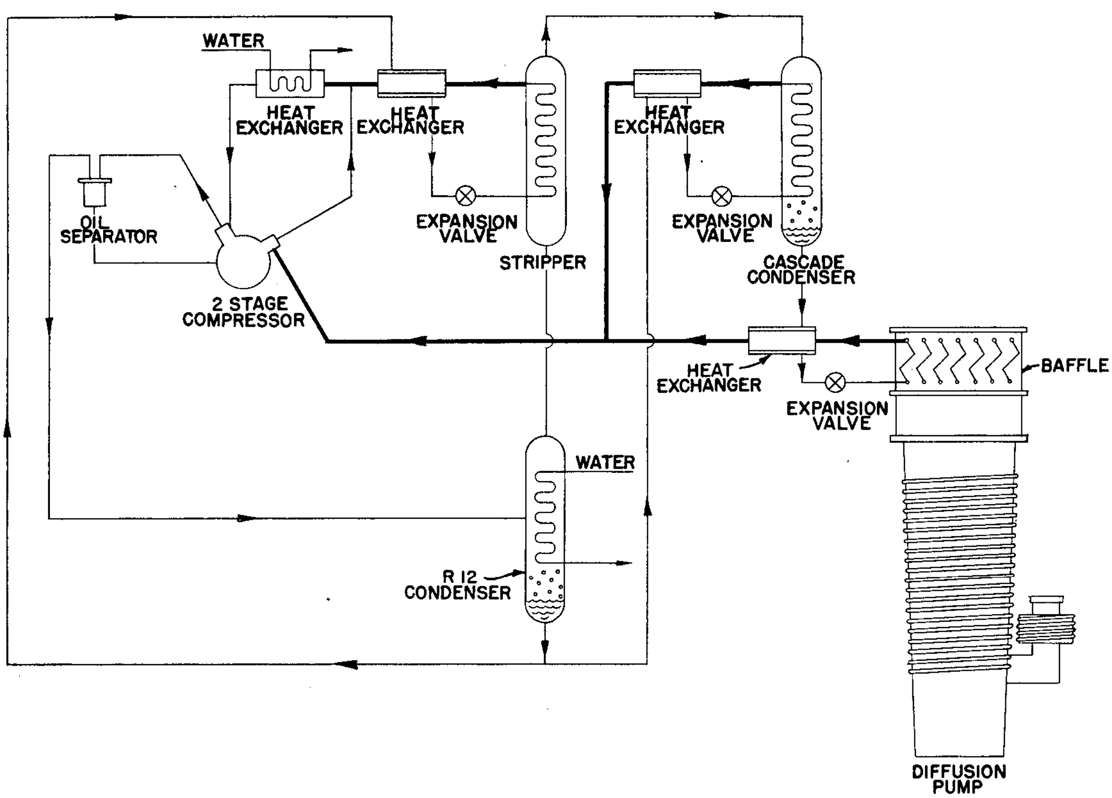
EXPANSION VALVE
WATER
CONDENSER
EXPANSION VALVE
HEAT EXCHANGER
OIL SEPARATOR
HEAT EXCHANGER
2 STAGE COMPRESSOR REFRIGERANT 22
CASCADE CONDENSER
RECEIVER
DIFFUSION PUMP
MU-18290

Fig. 2.



ZN-2266

Fig. 3.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HEAT EXCHANGER
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EXPANSION VALVE
EXPANSION VALVE
STRIPPER
CASCADE CONDENSER
2 STAGE COMPRESSOR
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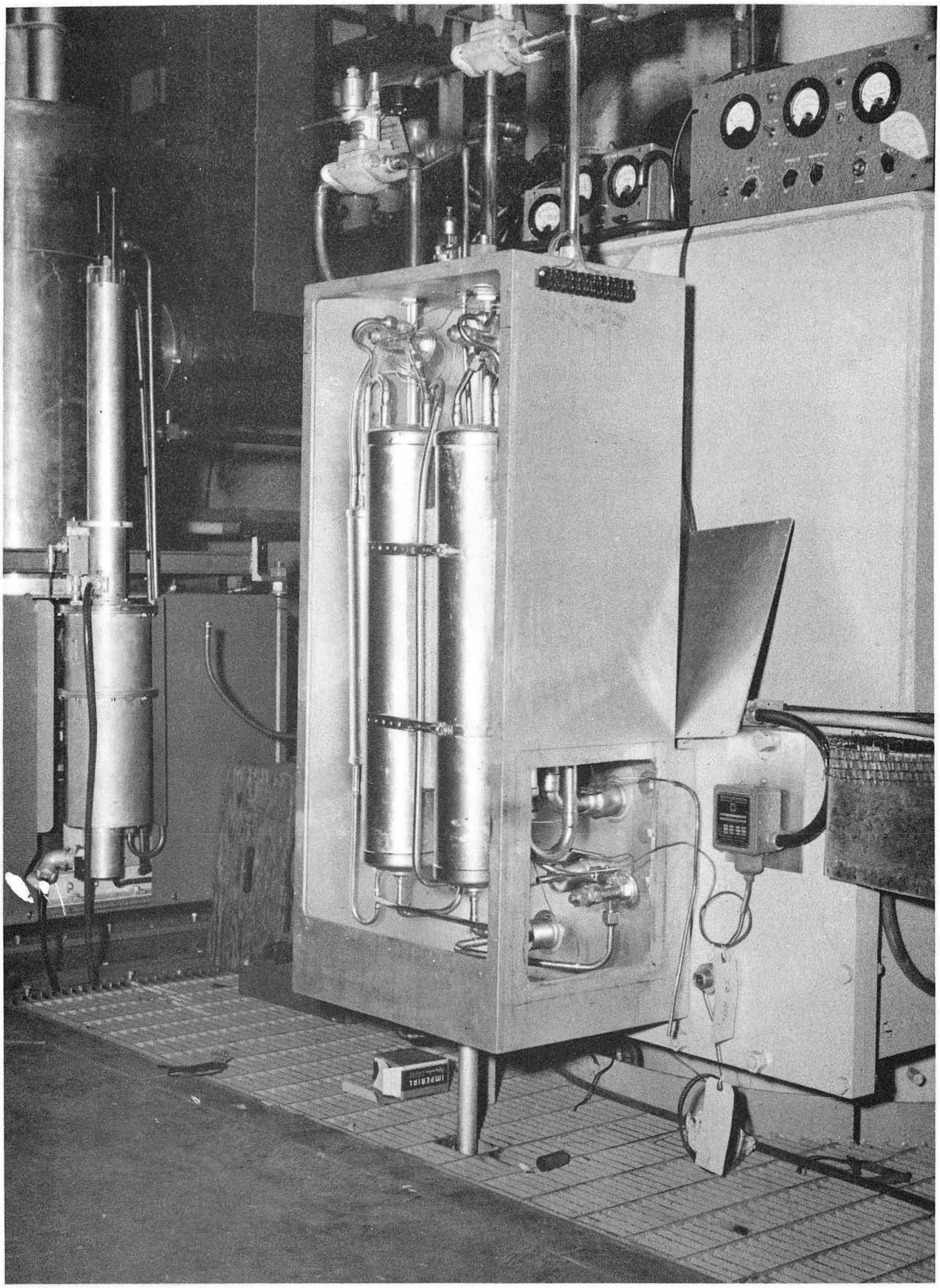
BAFFLE
EXPANSION VALVE
WATER
R 12 CONDENSER
DIFFUSION PUMP
MU-18293

Fig. 4.



ZN-2267

Fig. 5.

PRESSURE AND VACUUM TESTING PROCEDURE
FOR LOW TEMPERATURE REFRIGERATION SYSTEM

1. Charge system to full Freon-12 cylinder pressure and leak check with flame type halide detector.

2. Retest for leaks with electronic halide detector. When tight vent Freon-12.

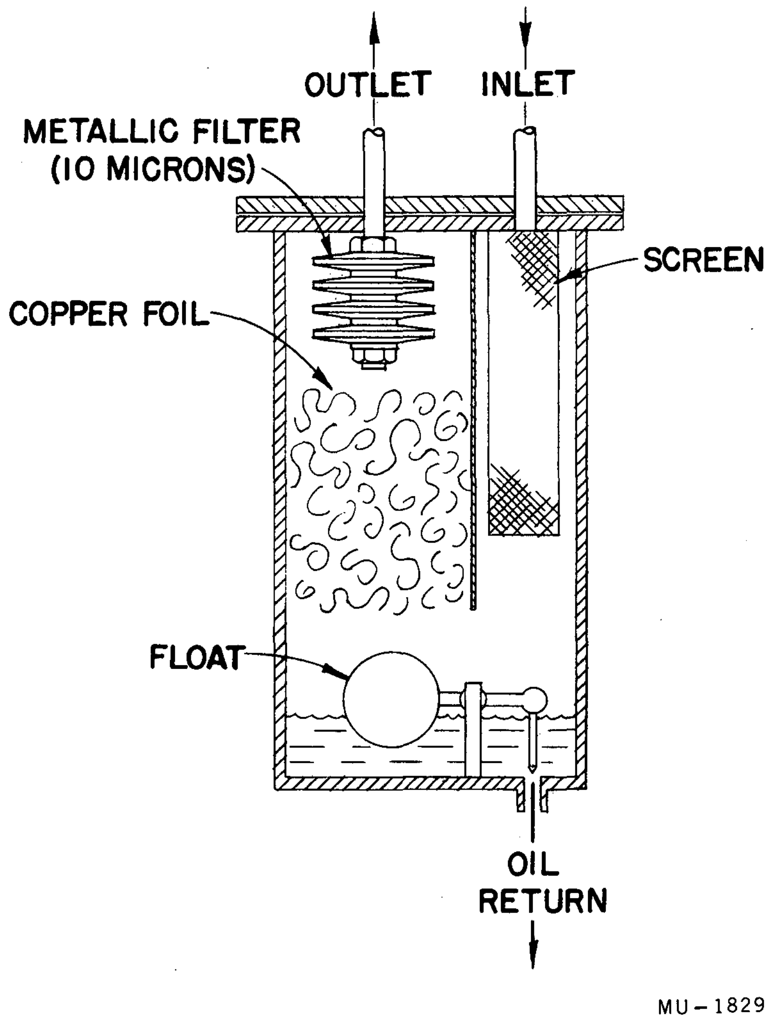
3. Install drier cartridge.

4. Heat drier and evacuate system to at least 100 microns Hg.

5. Leak check with helium sensitive mass spectrometer leak detector.

MU-18312

Fig. 6.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INLET
METALLIC FILTER
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SCREEN
COPPER FOIL
FLOAT
OIL
RETURN
MU-18294

Fig. 7.

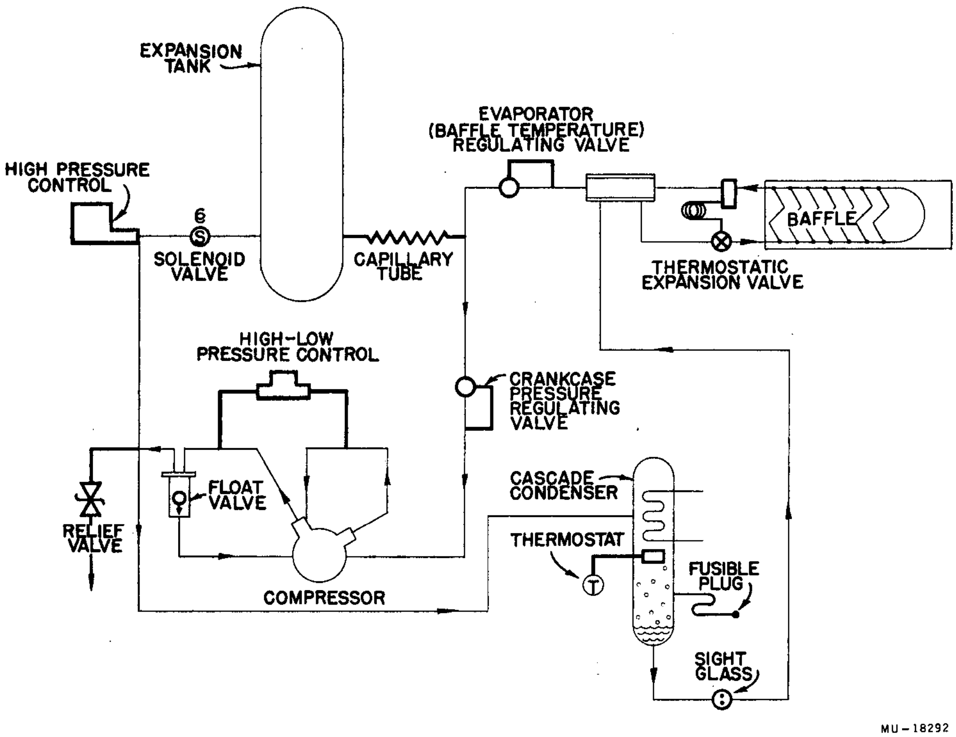
EXPANSION TANK
EVAPORATOR (BAFFLE TEMPERATURE) REGULATING VALVE
HIGH PRESSURE CONTROL
6
S
SOLENOID VALVE
CAPILLARY TUBE
BAFFLE
THERMOSTATIC EXPANSION VALVE
HIGH-LOW PRESSURE CONTROL
CRANKCASE PRESSURE REGULATING VALVE
CASCADE CONDENSER
FLOAT VALVE
RELIEF VALVE
THERMOSTAT
T
FUSIBLE PLUG
COMPRESSOR
SIGHT GLASS
MU-18292

Fig. 8.

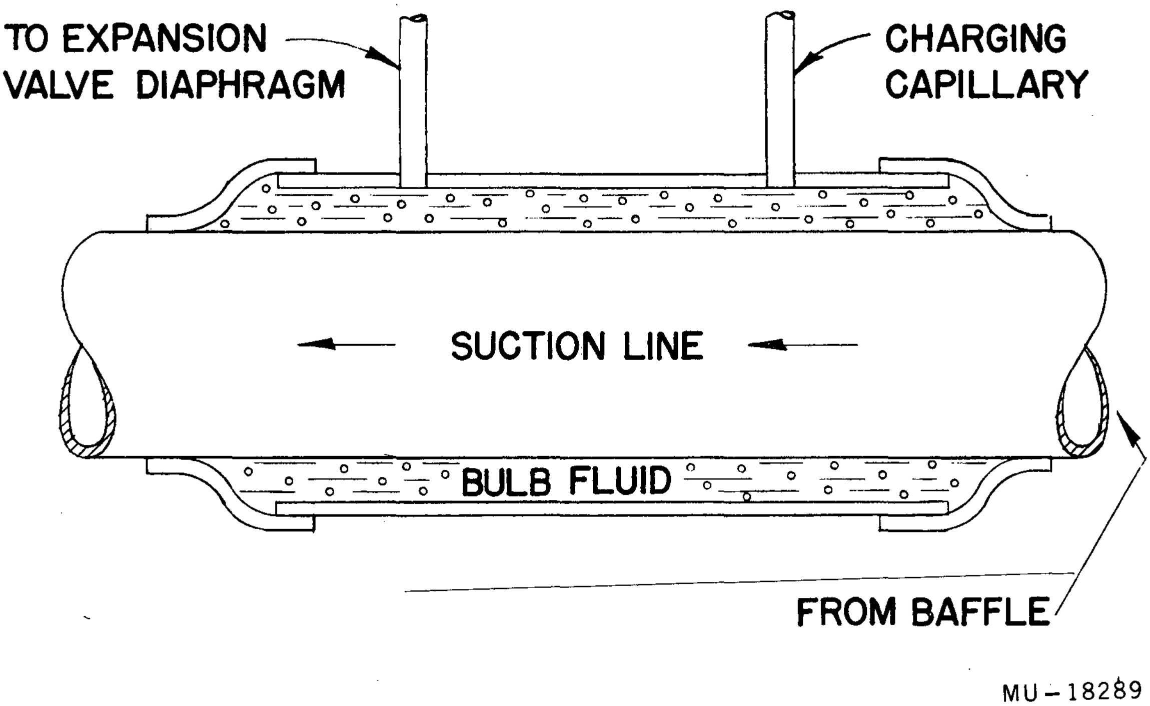
TO EXPANSION
VALVE DIAPHRAGM
CHARGING
CAPILLARY
SUCTION LINE
BULB FLUID
FROM BAFFLE
MU-18289

Fig. 9.

REFRIGERATION COST COMPARISON

	Compound Cascade	Cascade	Binary	Liquid Nitrogen
Operating Cost				
Maintenance	$ 330	$ 330	$ 330	$ 1,200 ①
Refrigerant	000	000	000	6,100 ②
Water	315 ③	315 ③	315 ③	000
Power	270 ④	270 ④	270 ④	000
Initial Installation Cost	6,800 ⑤	6,050 ⑤	7,300 ⑤	500 ⑤
Total Cost, Ist Year	7,715	6,965	8,215	7,800
Yearly Cost ⑥	1,595	1,520	1,645	7,350

① Servicing Trap or Baffle
② $.107/Liter, 25% Loss
③ $.0003/Gal.
④ $.0085 Kw/Hr.
⑤ Does Not Include Baffle
⑥ Operating Cost + 10% Depreciation

MU-18311

Fig. 10.