A STUDY OF A NEON CYCLE CRYOGENIC PUMPING SYSTEM FOR A LOW DENSITY PLASMA TUNNEL

by

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SUMMARY

This paper describes the development and testing of a large neon cycle cryogenic pump for use with a low density wind tunnel meant for both ionized and non-ionized flows.

The development work necessary in the choice of the refrigerant cycle and the production of the required hardware is described in some detail. Once the cryogenic pump was in operation it was thoroughly tested both as a refrigerator system and as a pump.

Experimental data on the thermodynamic properties of neon was obtained and found to be in excellent agreement with the available theoretical data. The refrigerator cycle was found to operate satisfactorily and as predicted. The pump itself was found to operate satisfactorily and as predicted over its entire designed range of operation. A pumping rate of 0.8 gr/sec N₂ was achieved while maintaining the test section pressure in the 1 - 4 x 10⁻³ torr region.
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I. INTRODUCTION

As low density flow studies are pushed into the region of higher Mach numbers and larger and more complex physical models and probes are to be tested, the research scientist is faced with the problem of pumping mass flows of low density gases at ever larger rates if he wants to obtain reasonably uniform flow conditions and long testing times. Although in theory oil booster pumps can be built as large as one wants, in practice the design of large boosters is complex and their construction and operation expensive. Oil diffusion pumps, although somewhat simpler in construction and cheaper to buy, have their optimum operating range in a region of pressure that is too low for use in wind tunnel operation.

A more promising solution to the problem of making fast low density pumps was engineered by Chuan (Ref. 1 and 2) at U.C.L.A. with the construction of his helium cooled cryogenic pump. Such a pump derives its pumping action from the use of a cooled surface that is sufficiently cold to first, condense the gas present and second, to give an adequately low vapour pressure of the condensate. The surface (called freeze-out surface) is kept cold by a continuous flow of refrigerant to it, creating a continuous pumping action (with certain provisos, as we shall see later). In their useful range of operation these pumps offer vastly improved pumping speeds at a smaller cost than oil booster pumps.

When it was decided several years ago to expand the low density testing facilities at U.T.I.A.S., French and Muntz (Ref. 3) indicated the possibility of using a neon cycle cryogenic pump for the new low density facility which was intended to be used mainly for the study of low density plasma flows.

This paper puts forth the reasons for this choice of pump and cooling cycle, along with a description of the development work that was necessary to put the system into operation. Special experiments were performed to measure the thermodynamic performance of the neon cooling cycle in order to provide comparisons with available theoretical data. Finally the operation of the entire system as a pump was studied and found to be completely satisfactory.

II. GENERAL CONSIDERATIONS

2.1 Definition of Performance Envelope

Work in low density aerodynamics has been performed for many years in the UTIAS low density wind tunnel. Because of the number of projects in this tunnel it became obvious that a second low density facility was needed. The basic requirement for this new tunnel is the simulation of phenomena encountered in upper atmospheric and ionospheric hypersonic flight. This implies:

(1) the possibility of producing low density ionized gas flows.

(2) mean free paths of the order of $\frac{1}{2}$" to permit the true simulation of free molecule flow over models of acceptable size (i.e. Langmuir probes, simple bodies, etc.).
In order to obtain these high Mach numbers and low density test conditions, conventional aerodynamic nozzles or free jets are used, and to allow for the large viscous effects relatively large dimensions of the flow fields are desired, leading to large mass flows.

Another most important factor in the design of the low density facility is the duration of the flow under the desired test condition. Because of the relatively long response time of instruments at low pressure conditions and the need to make more than one measurement under any one set of conditions, for good accuracy it was felt desirable that the proposed low density tunnel should permit testing times of no less than 5 minutes.

This therefore fixes the type of low density wind tunnel to one using fast vacuum pumps to achieve both the flow and the low density conditions.

At the inception of this project French and Muntz (Ref. 3) set about defining the performance envelope of the tunnel-vacuum pumps configuration. It was their aim to use a Plasmatron arc heater as a source of ionized gas. They found that one of the upper limits of operation of this device was a mass flow of 0.63 gram/sec air (or N₂). If the arc jet was to be used in the proposed tunnel, the pump would therefore have to be able to handle a mass flow of this magnitude.

As large mean free paths are desired, relatively low test section pressures are required. On Fig. 1 is plotted the required mass flow of 0.63 gr/sec N₂ as a volumetric flow at various pressures. The diameters that are required to pass this mass flow through a baffled pump opening are also indicated for a number of pressures. The pumping curves of two commercially available pumps are also shown. It is seen that at 10⁻⁴ torr a pump with an opening diameter of 4 meter is needed to pass the required volumetric flow. For reasons of economics it was therefore decided to limit the test section pressure to a minimum of 10⁻³ torr.

2.2 Choice of the Type of Pumping Systems

From Fig. 1, it is evident that oil diffusion pumps are not well suited for our purpose. Two types of pumps are therefore to be considered in more detail, i.e.

(1) the oil booster pump
(2) the cryogenic pump.

The choice between these two different types of pumping system rests for a large part on comparison of costs. A cost survey prepared by RARDE (Ref. 4) quotes figures for a system capable of pumping 10⁶ liters/sec at 10⁻³ torr.


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<th>Cryogenic pump based on helium refrigerator</th>
<th>Oil Booster Pumps</th>
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<td>Capital cost</td>
<td>$150,000</td>
<td>$1,500,000</td>
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<tr>
<td>Running costs</td>
<td>$3/hr.</td>
<td>$75/hr.</td>
</tr>
<tr>
<td>Floor area</td>
<td>200 ft²</td>
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In the case of booster pumps it must be remembered that the cost figures include the installation and use of the necessary electrical power and cooling water facilities. It is quite obvious that for these high pumping speeds cryogenic pumps are an obvious choice when only condensable gases have to be pumped.

2.3 Cooling of Gases by the Joule-Thomson Cycle

To achieve the very low temperatures required to cryopump gases, the refrigerant gas, circulating within the freeze-out surface, is cooled by some form of Joule-Thomson process. Detailed and lucid explanations of this process are given in numerous texts (two excellent ones, Refs. 5 and 6). A general description of the process is reproduced here from the above references.

In a Joule-Thomson cycle a gas is made to undergo an adiabatic continuous throttling process. Such a process is an isenthalpic one. By means of a compressor a constant high pressure is maintained on one side of a throttling valve and a constant lower pressure on the other side. For a given pressure and temperature on the upstream side of this valve, which fix the value of the enthalpy, the temperature of the gas on the downstream side of the valve, will be a function of the pressure to which the gas is expanded. Such a relationship between temperature and pressure is shown in Fig. 2. It is seen that when the gas is expanded to a sufficiently low pressure an appreciable drop in temperature can be obtained.

The numerical value of the slope of an isenthalpic curve on a T-P diagram at any point is called the Joule-Kelvin coefficient, denoted by \( \mu \). Thus

\[
\mu = \left( \frac{\partial T}{\partial P} \right)_h
\]

The locus of all points at which the Joule-Kelvin coefficient is zero i.e. the locus of the maxima of the isenthalpic curves, is known as the inversion curve, and has the general appearance shown in Fig. 3.

A Joule-Thomson expansion process which has initial and end states within the inversion curve envelope (positive \( \mu \)) (Fig. 3) will lower the temperature of the fluid while such a process occurring entirely outside the envelope (negative \( \mu \)) will raise the temperature of the fluid. The point where the inversion curve intersects the temperature axis, is known as the maximum inversion temperature. Expansion from a temperature above the maximum inversion temperature will always result in heating.
Now, consider the situation sketched in Fig. 3. If the fluid is expanded from state A to state B, the temperature of the fluid will rise. Further expansion to state C results in cooling to the original temperature and an end state D will lower the temperature below the initial temperature. Since the same net temperature drop is obtained by expanding the gas from either states C or A to state D, there is no reason for expending the energy required to compress the fluid to any pressure higher than the pressure at C.

Lowering the initial temperature results in operation in the portion of the temperature-pressure plane where the slopes of the isenthalps are greater and eventually where the isenthalps intersect the vapor pressure curve of the liquid-vapor equilibrium. The intersection of an isenthalp with this curve, as for instance point D', indicates that a fraction of the gas expanded into this region is liquified. Further reduction of the pressure will in general result in a change of liquifaction, but the process will always be such that the vapor pressure curve is followed.

Figure 4 shows the essential components of a Joule-Thomson refrigerator cycle. The Joule-Thomson refrigeration system is defined as a counter-flow regenerative heat exchanger connected in series with an expansion valve and an evaporator as shown schematically in Fig. 5. The compressor for the system operates at ambient temperature while the heat exchanger of Fig. 5 operates in the cryogenic temperature range so that in general a precooling heat exchanger or expansion engine between compressor and Joule-Thomson cycle is required. Intermediate heat exchangers and expansion engines are not considered as part of the Joule-Thomson system.

The Joule-Thomson system may be considered to be at a given uniform temperature. This will for instance be the case when a precoolant bath is in contact with, but outside of the system. The gas expansion at the valve acts upon starting the flow in the system as discussed previously. Thus appropriate precooling temperature and operating pressure choices must be made for each refrigerant to ensure a cooling effect, as explained above. Starting the refrigeration process consists of providing a quantity of compressed gas to point 1. The refrigeration made available by the expansion process is used to cool the mass of metal of the evaporator and the return stream side of the heat exchanger. The external heat load is withheld in order to expedite cooling. Cooling the heat exchanger return stream side, between points 4 and 5, results in cooling the inlet gas stream between points 1 and 2. For the ideal heat exchanger considered here, points 1 and 5 remain at the initial temperature while the temperature of points 2, 3 and 4 are depressed. As the temperature of point 2 is depressed, the temperature of points 3 and 4 also become colder due to the Joule-Thomson effect. Thus the refrigeration process is progressively lowering the temperature until an equilibrium condition is reached.

A possible equilibrium condition is shown by points 1, 2, 3, 4 and 5 on the temperature-entropy surface of Fig. 5. An external heat load has been applied to the evaporator between points 3 and 4 and is just equal to the refrigeration available. No refrigeration is available to further cool the heat exchanger. No condensation of liquid occurs; under this condition the term "evaporator" is a misnomer as refrigeration is obtained non-isothermally from the sensible heat of the gas. This condition is difficult to maintain because of the low thermal capacitance of the gas in the evaporator. Small variations in the heat load will rapidly change the system temperature.
The stable equilibrium condition at which the Joule-Thomson refrigeration process is normally operated is shown by points 1, 2', 3', 4' and 5 on the temperature-entropy surface of Fig. 5. Temperature depression at the expansion valve continues until liquid is condensed into the evaporator. Refrigeration is obtained isothermally by the evaporation of the liquid. A prerequisite for this type of operation is that the return stream pressure be less than the critical pressure since the temperature at which the evaporator will operate is fixed by the pressure as defined by the vapor pressure curve of the refrigerant. Operation with a one atmosphere return stream pressure will result in an evaporator temperature as given by the normal boiling temperature of the refrigerant.

This type of operation tends to be stable in temperature due to the presence of liquid and the magnitude of the heat of vaporization. The mass flow through the evaporator must be matched to the heat load for continuous stable operation as an excessive heat load will result in reducing the quantity of liquid rather than changing the system temperature. If corrective action is not taken, such as increasing the compressor flow rate, the evaporator will eventually go dry forcing the evaporator temperature to increase into the non-isothermal operating region. In contrast, an insufficient heat load will result in flooding of any type of evaporator. Thus, the refrigeration process must be controlled in either mode, but the case of isothermal operation is easier to control.

2.3.1 Choice of Refrigerant

If test section pressures as low as $10^{-3}$ torr (1 micron Hg) are required the condensing surface of the cryopump must be below about 300 K since the vapor pressure of air rises above $10^{-3}$ torr at approximately 320 K (Fig. 6) (Ref. 3). Three gases have boiling points that lie below 300 K; helium, hydrogen and neon (Ref. 7) (Table 1). They therefore constitute suitable refrigerants for the cryogenic pump. Each must be inspected closely to determine which one offers the best solution for cooling the freeze-out surface of the pump. Two criteria are of interest in this respect. The first is the specific refrigeration defined as the amount of refrigeration achieved per unit mass flow; the second is the refrigeration power in terms of the compressor power.

Helium

Because the maximum inversion temperature of helium lies between 25 and 600 K, the helium has to be precooled with the aid of liquid hydrogen, or the Joule-Thomson heat exchanger and expansion valve have to be preceded by an expansion engine in the cooling cycle. An expansion engine is a device that allows the gas to expand adiabatically against a retarding force and thus do external work. The retarding force is either provided by a turbine or a piston. By moving the turbine or piston, the gas loses some of its energy and hence is cooled.

The expansion engine - Joule-Thomson combination is normally used to liquefy helium in what is known as a Kapitza liquefier (Ref. 8). Because of the close machining tolerances required in turbine and expansion engine fabrication and the problem of operation at low temperatures, the cost of these liquefiers is usually high (A.D. Little, 300 watts helium liquefier, $35,000 - $45,000). One of the possible figure of merit for a refrigerating system is the number of SCFM (standard cubic feet per minute)
of refrigerant needed to produce 1 watt of refrigeration.

Using Ref. 6 and Fig. 5, for a temperature difference between entrance and exit of the Joule-Thomson cycle of

\[ T_1 - T_5 = 1^\circ K \]

and using a value for

\[ T_1 = 15^\circ K \]

(a standard value in the Kapitza liquefier (Ref. 8), which is presumably an optimum choice), this figure of merit is

1.66 SCFM/watt

If however, we use only the expansion engine to reduce the temperature of helium to its lowest possible value at 1 atmosphere pressure, i.e. \( T \approx 5^\circ K \) and let the gas warm up to \( 30^\circ K \) in the freeze-out surface (keeping the freeze-out surface at a maximum temperature of \( 30^\circ K \)) we need (Ref. 8)

7.70 SCFM/watt

If helium is to be used, it is seen that addition of the Joule-Thomson expansion process results in an important improvement.

**Hydrogen**

The maximum inversion temperature of hydrogen is \( 202^\circ K \). Liquid nitrogen could be used to precool the hydrogen to \( 80^\circ K \) before expansion in the Joule-Thomson process. For an approach temperature of \( 80^\circ K \) and

\[ T_1 - T_5 = 1^\circ K \]

we get (Ref. 6)

0.143 SCFM/watt

Under normal operation (Ref. 8) the boiling point of the liquid nitrogen precooling the hydrogen is reduced to its minimum of \( 64^\circ K \) (Fig. 7, Ref. 5) by depressing the ambient pressure in the liquid nitrogen heat exchanger. Under these conditions, the figure of merit for hydrogen is

0.107 SCFM/watt

Even though this indicates that on the basis of this particular figure of merit, hydrogen is a better refrigerating agent than helium, the safety hazards involved in handling it rendered it unsuitable for our use.

**Neon**

The thermodynamic properties of neon have been computed by Yendall (Ref. 9) and some useful data is represented graphically in Figs. 8, 9, 10, 11 and 12. Figure 8 shows the temperature-entropy diagram for neon, with a number of isobars and isenthalpic curves drawn. The isenthalpic relationship between temperature and pressure are shown on Fig. 9 for various values of the enthalpy. The cooling effect associated with the expansion
can be seen clearly on this figure. The final temperature upon expansion to a final pressure of 1 atm. is shown in Fig. 10 as a function of the initial temperature and pressure. It is seen on this figure that the boiling temperature of neon at 1 atm. can be reached, indicating that liquid neon will be formed.

The specific refrigeration obtained in expansions to a final pressure of 1 atm. are plotted on Fig. 11 as a function of initial temperature and pressure and the locus of the maxima is drawn on Fig. 12. It is of some interest to note that at the normal boiling point of nitrogen, i.e. 77°K, a slight extrapolation of the data in Fig. 11 indicates that a specific refrigeration of about 1.1 B.T.U./c.f. is reached at a pressure of about 230 atm. This value corresponds to

0.052 SCFM/watt

It is seen by the comparison shown in Table I that this value is superior to both hydrogen and helium. We may make therefore the following evaluation of the use of neon as a refrigerant:

(1) Neon has 3 times the specific cooling capacity of helium and twice that of hydrogen.

(2) Neon can achieve this higher refrigeration rate using a simple Joule-Thomson expansion process precooled by an atmospheric liquid nitrogen bath, thereby reducing the complexity and cost of a refrigeration system.

(3) Neon, being inert, presents no risks of explosions and requires no elaborate safety system.

(4) Because of its higher specific refrigeration capacity, the production of a given amount of refrigeration using neon will require a system 3 times as small as that of helium and twice as small as one using hydrogen. This will be reflected favourably in the capital cost of a neon system.

However, because of its high cost ($15.00 per atm. c.f.) neon must be contained in a system that is rigorously leak tight to avoid the loss of neon from the system and more importantly to eliminate entirely the leaking in of air, which would require the replacement of the neon charge. Essentially, this means that the high pressure (≈ 230 atm) side of the Joule-Thomson cycle (i.e. compressor, plumbing) must not have leaks with rates any larger than those encountered in vessels designed to operate at high vacuum (10^-4 - 10^-3 torr). Furthermore, a special control system was required to ensure that the pressure in the system at any point would always remain above atmospheric to avoid leaks into the system. It was anticipated that these requirements would cause some considerably difficulties. However, it was believed the problems could be solved, and the obvious advantages of neon as a refrigerant made the effort worthwhile. Also, since the price of neon has shown a steady decline since 1960 (the price at that time was $30.00 per atm. c.f.) it is hoped that in the near future, the cost will no longer cause any serious obstacle for the practical use of neon in refrigeration systems.

Because of these reasons of safety, economics, and efficiency it was decided to use neon as the refrigerant gas for the cryogenic pumping system of the proposed UTIAS Low Density Plasma Tunnel.
2.4 Theory of Cryopumps

If a surface at a temperature low enough to condense the gas present is introduced into an enclosure, the resultant pressure in the enclosure will be that of the vapor pressure of the condensate. As the condensate temperature is lowered to the triple point of the gas, freezing occurs.

In operation as a pump, the gas freezes on the cold surface as a solid layer and in a wind tunnel where the gas flows may be considerable, the pump is not strictly a continuous device because the thermal resistance of the condensate layer increases with thickness. Since the heat of fusion and other heat loads have to be transferred to the cold surface through this layer, the surface temperature of the condensate increases and the vapour pressure with it. Figure 6 shows the importance of maintaining the surface temperature of the condensate below approximately 300 K if one desires to operate at a minimum test section pressure of 1 micron Hg (10^{-3} torr). In the case of neon with a boiling temperature \( \approx \) 250 K at 1 atmosphere (Ref. 7) the pumping time will therefore depend strongly on the magnitude of the heat conductivity of the condensate layer.

French and Muntz (Ref. 3) did some small scale experiments in an attempt to predict the pumping time available using a neon cooled surface as a pump. Following Ref. 10 they idealized the condensing system to a one dimensional problem of a condensed phase at temperature \( T_s \) in contact with a condensing gas phase whose distant temperature, pressure and density are \( T_0, P_0 \) and \( \rho_0 \), and where the gas conditions a few mean free path from the condensate surface are designated by the subscript I.

Their theoretical analysis led them to an expression for the rate of condensation

\[
\omega = \frac{\sigma (P_I - P_s)}{\sqrt{2} \pi RT_s}
\]

where \( \omega \): mass of gas condensing per unit time  
\( \sigma \): condensation coefficient, approximately unity for most metals and larger than 0.9 for many materials, like the condensate itself (Ref. 10)

Consequently the above relation shows that there is necessary a driving pressure or pressure above the vapour pressure to cause the nonequilibrium process of net condensation to occur. Using the vapour pressure curve for nitrogen, insertion of typical numerical values shows that only a very slight overpressure is required for a reasonable mass flow. For example, for a condensing surface area of 180 sq. ft. and a mass flow of 0.63 gr/sec \( N_2 \), the driving pressure required would be about 10^{-4} torr. This area of 180 sq. ft. was therefore chosen as a minimum value, compatible with the requirement of 10^{-3} torr in the test section.

Using the pumping speeds and the thermal conductivity of the condensate measured in their small scale experiment in conjunction with this minimum freeze-out area, French and Muntz predicted the pumping times available for various mass flows before the full scale pump would reach a certain pressure level (Fig. 13). It is seen that on the basis of these predictions 5 minutes of testing would be available before the test section reaches
1 μ Hg \((10^{-3} \text{ torr})\) when a flow rate of 5 lbs/hr \(N_2\) \((0.63 \text{ gr/sec})\) is admitted to the pump. Longer testing times are available if lower mass flows are used and higher minimum test section pressure are admissible.

Theoretical expressions for the pumping speeds of cryopumps are given in Refs. 11, 12, and 13 among others. The most straightforward description is given here.

In free molecular flow, to interpret the pumping speed of cryogenic pumps in terms of the number of molecules striking the cryosurface, it is necessary to use a capture coefficient. The capture coefficient may be defined as the reciprocal of the average number of collisions a molecule makes before condensing. Experimentally the capture coefficient is defined as the ratio of the experimental pumping speed to the theoretical maximum speed;

\[
C = \frac{S}{\sqrt{\frac{RT}{2\pi M}}}
\]

where \(C\): capture coefficient
\(S\): measured pumping speed
\(\sqrt{\frac{RT}{2\pi M}}\): is the Maxwellian average speed of the molecules of the gas divided by 4. Kinetic theory indicates that the total number of collisions with the walls of a vessel per unit area per unit time is given by (Ref. 14)

\[
\frac{1}{4} n\bar{v}
\]

where \(n\): number density of molecules
\(\bar{v}\): Maxwellian average speed of the molecule. Hence \(\bar{v}/4\) represents the effective maximum pumping speed with which the gas will condense on the cold surface.

In continuum flow, the pumping speed also has a constant value \((S)\). This value is approximately 3 times that of the molecular flow and can be calculated by realizing that the mass flux density at the cold surface, which acts as an ideal orifice, will be limited to the maximum possible value; i.e. the value at the sonic condition. The pumping speed measured at the chamber pressure will thus be equal to

\[
S = C \sqrt{\frac{\gamma RT}{M}}
\]

where \(\sqrt{\frac{\gamma RT}{M}}\) is the speed of sound at the sonic condition.

In transition flow, the pumping speed cannot be specified as simply, but is expected to lie between these two extremes. In fact, it has been established (Ref. 11) that the pumping speed changes smoothly from the free-molecular flow pumping rate to the continuum region pumping rate for about a hundred fold pressure increase. The midpoint of the transition region occurs at a pressure where the mean free path is of about the same order of magnitude as the cryosurface dimensions. Reference 11 also gives experimental data confirming the free molecular and continuum limits for the pumping speed.
III. REFRIGERATION CYCLE

3.1 Capacity of Neon Cryopumps

As stated in Sec. II, it is desired to freeze-out 0.63 gr/sec \( \text{N}_2 \) (5 lb/hr) at the neon normal boiling point temperature (27.3\(^\circ\)K). The condensation load for freezing-out 5 lb/hr \( \text{N}_2 \) that is precooled to 80\(^\circ\)K by the intercooler (see Sec. 4.5.B) has been calculated on the basis of the data of Ref. 8, and found to be 190 watts. In these calculations account has been taken of the sensible heat of the gas and of the solid as well as the heat of fusion, vaporization and that associated with a phase change of the solid. The calculations are summarized in Table II.

It must be remembered, that the freeze-out surface of the cryopump must also bear the radiation load of the surrounding enclosure plus some assorted conduction loads. The freeze-out surface was therefore protected from the radiative heat load originating from the outer walls at ambient temperature by the introduction of a liquid nitrogen cooled shield. This reduces the radiative load to the freeze-out surface to an acceptable level. Section V of this report shows that these radiative and conductive heat loads add a constant load of about 40 watts, bringing the total cooling capacity requirement of the neon refrigeration system to about 230 watts.

At this point it must be mentioned that the economic advantages in the use of a neon system became even more attractive by the gift to the Institute of two National Bureau of Standards designed compressors of 21 SCFM capacity suitable for service at 2000 psi (135 atm). Although expansion from 2000 psi does not give maximum specific cooling (Fig. 11) or maximum liquid neon production (Fig. 14) (Ref. 15), it does permit operation at optimum conditions of compressor power as shown in Fig. 15 in which the required power in terms of liquid neon production rate is plotted (Ref. 15). The use of a lower pressure level also allows the use of lighter components in the neon pressure control plumbing (30% drop in pressure over that needed for maximum specific refrigeration).

From Fig. 11 we see that part of the refrigeration lost due to the lower pressure can be regained by reducing the boiling point of the nitrogen precooler to its minimum value of 64\(^\circ\)K. In this way it is possible to operate a refrigeration cycle that would furnish

\[ 0.971 \text{ B.T.U.}/\text{c.f.} \]

for a figure of merit of

\[ 0.058 \text{ SCFM/watt} \]

This represents an 11% loss in specific refrigeration as compared with the optimum value at a pressure of 230 atm. The figure of merit of

\[ 0.058 \text{ SCFM/watt} \]

is still 2.9 times better than helium and 1.8 times better than hydrogen although now we must depress the boiling point of the liquid nitrogen precooler as in the hydrogen case.
The expected saving in capital cost alone due to the available compressors, more than offset these minor losses in cooling efficiency and it was decided to build a 135 atm system with a 64°K precooler.

The Stearns-Roger Manufacturing Co. of Denver, Colorado was entrusted with the modification of the compressors for neon service along with the design of the pressure control system for the Joule-Thomson cycle and the dewar vessel of the cryogenic pump with its heat exchangers. A few months of testing the system as designed by Stearns-Roger showed that

1. The modified compressors were still unsuitable for neon use because of intolerable leak rates and contamination from the crankcase oil.

2. The design of the pressure control system of the Joule-Thomson cycle incorporated undesirable components. The suggested components (valves, gauges) were found to be impossible to seal adequately against leakage, some of them were selected incorrectly for the required flow rate; others acted as one-way valves when the gas had to flow in the reverse direction.

3. The liquid nitrogen radiation shield, which was incorporated in a Dewar type of construction to minimize the heat load on it (see Fig. 24), was found to receive a much higher heat load than was estimated by the manufacturer. This was the result of very poor emissivity properties of the walls. The consequences were particularly unacceptable since the functions of cooling the radiation shield and providing the initial temperature for the Joule-Thomson cycle were provided by a single liquid nitrogen bath. The large boil-off rate as a result of the heat load therefore would have necessitated the use of very large vacuum pumps to depress the boiling temperature of the liquid nitrogen.

4. The joints in the liquid nitrogen handling system within the evacuated pump chamber were unfit for service in a vacuum when cooled to liquid nitrogen temperature.

It was obvious that the compressors and pressure control piping had to be discarded and this was done. The dewar vessel of the cryogenic pump was too expensive to discard and it was believed that the radiation problem could be solved. The leaking joints in the liquid nitrogen handling system could be replaced. The separation of the two function of the liquid nitrogen bath was not possible without extensive modifications, but by reducing the heat load it was possible to use an existing vacuum pump to provide the lowering of the boiling temperature of the liquid nitrogen.

It was therefore decided to design a Joule-Thomson cooling cycle with a new compressor and new pressure control plumbing around the existing heat exchangers within the dewar vessel of the cryogenic pump. The heat exchangers were designed for 2000 psi service and therefore our new cycle had to operate at that pressure.

Once these design limitations were accepted, inspection of Fig. 11 shows that for a 64°K expansion from 135 atm to 1 atm, 0.058 SCFM/watt are required. As mentioned earlier the heat exchangers had been sized according to the N.B.S. compressors with a capacity of 21 SCFM. This corresponds to approximately 350 watts of refrigeration;
Further details of the Joule-Thomson cycle for the operating pressure of 135 atm. and an expansion to 1 atm. are given in Figs. 16 and 17. In Fig. 16, the isenthalpic expansion temperature is given as a function of the initial temperature and for comparison a curve of equal initial and expansion temperatures is drawn. The figure indicates that even without using the conventional Joule-Thomson exchanger, the expansion temperature is about 42°K for the working value of the initial temperature of 64°K. In Fig. 17, the temperature-enthalpy relationship is plotted for upstream pressure (135 atm) and pressure on the expanded side (1 atm.). Since with the use of the Joule-Thomson interchanger, the gas temperature at the entrance and exit of the cycle is about equal to the bath temperature of 64°K, Fig. 17 indicates the amount of heat absorbed by the gas in the cycle.

IV. DESIGN OF SYSTEM HARDWARE

Once the refrigeration rate requirements were settled it was then necessary to translate these design objectives into hardware capable of producing the desired results. This section contains a complete description of all the important components of the refrigeration cycle. In the production of the hardware, the necessity for near absolute leak tightness of all the industrial components caused severe difficulties. Some of the solutions adopted represent novel features and for this reason some of them are described in detail in the text.

A cross section of the neon cryogenic pump with its associated control plumbing is shown schematically on Fig. 18. The important sections of the pumping system are described in detail in the text. For reasons mentioned earlier the control plumbing (Fig. 19) maintains the pressure in the whole neon handling system above atmosphere. The functions of the individual components of the control plumbing are described in Sections 4.1 and 4.6. The freeze-out surface (Section 4.3, Figs. 21, 22, and 23) is completely enclosed in a LN\(_2\) cooled radiation shield (Section 4.4 and 4.5, Figs. 24, 25, 26 and 27). The LN\(_2\) handling system (Fig. 41) is described in detail in Section VI.

4.1 General Layout of the Pressure Control System

Figure 19 shows a simplified schematic layout of the pressure controls of the closed cycle neon refrigeration system, omitting the various heat exchangers. As was mentioned previously, it is necessary in order to have positive protection of the neon charge, that leaks into the system be entirely avoided. Even though much attention was paid to the leak tight construction, it was considered desirable to control the pressure such that even at the point of lowest pressure in the system, i.e. the suction side of the compressor, a slightly above atmospheric pressure be maintained. Difficulties in this respect can arise when the heat load onto the freeze-out surface is smaller than the available refrigeration. Net liquefaction will then result with a subsequent pressure drop in the system unless neon is supplied from storage. A balance can be obtained by diminishing the mass flow through the Joule-Thomson valve, but this requires some kind of by-pass between the high and low pressure line. Figure 19 shows such an arrangement.
Line 1 feeds the compressor with gas from the supply. Line 2 takes the gas at 2050 psi from the compressor to the Joule-Thomson expansion valve and freeze-out surface. The expanded gas returns to the compressor by line 1. Line 3 allows gas to go from the supply to line 1 and allows the surplus gas in line 2 to be pumped back into the supply when the Joule-Thomson valve is only partly opened.

Item (1) is a back pressure regulator which maintains the pressure in line 2 at a maximum value of 2050 psi. If the pressure in line 2 should rise above this value due to a throttling down of the Joule-Thomson valve, valve (1) opens and allows gas to enter line 3.

Item (2) is a one way valve that allows surplus gas in the system to be pumped back into the supply if the pressure of line (3) rises above that of the supply bottles.

Item (3) is a downstream pressure regulator that allows line 3 to be maintained at a pressure level adequate to supply the regulator (4).

Item (4) is a downstream pressure regulator that maintains line 1 at a pressure somewhat above atmosphere. This is necessary as any leakage of air into the neon cycle would cause blockage of the lines in the cold portion of the cycle as air solidifies at liquid neon temperatures. It would also spoil the neon charge in the system.

4.2 The Joule-Thomson Valve

The Joule-Thomson valve as designed and manufactured by Stearns-Roger is shown in Fig. 20. As can be seen it simply offers a continuously variable blockage to the gas flow and is designed specifically to minimize the heat loads.

4.3 The Freeze-Out Surface

As seen from Fig. 1, the diameter of the pump opening necessary to pass the required 0.63 gr/sec mass flow, (corresponding to a volumetric flow rate at $10^{-3}$ torr of 371,000 l/sec) is approximately 4 feet. Because of economic consideration it is advisable to keep the diameter of the pump to its minimum necessary value. The flow therefore was to be brought into contact with the freezeout surface across a plane whose diameter should be roughly 4 feet. Following Ref. 3, it was desired to have a freeze-out surface with a total surface area of approximately 180 square feet. The freeze-out surface must be designed in such a way that the gas entering the pump has free access to all of its parts; i.e. that no part of the freeze-out surface be shielded by another. This insures that the test section gas will be condensed with a minimum of pressure drop to reach all parts of the freeze-out surface.

During the initial design stages of the pump, more data about cryopumping surfaces became available. Whereas solid surfaces exhibited capture coefficient of the order of 0.6, finned cylinders with closely spaced fins were reported to have capture coefficients close to 1.00 (Ref. 11). It was therefore decided to form the freeze-out surface with banks of finned cylinders instead of refrigerated solid copper surfaces.
As we have seen in Section 2.4, there are two limiting speeds at which a given surface can pump gas corresponding to whether that surface is in the free molecular or continuum region of gas flow. It must be emphasized, that these two limiting speeds should serve only as broad limits as it is not sure that they apply correctly to all conditions of gas, temperature, geometry, etc.

It is not expected that the pressure in the pump chamber at typical running conditions will be low enough to produce free molecular flow to the freeze-out surface during pumping operations. If past experience is followed (Ref. 2) the pressure in the chamber during operations should be in the $10^{-4}$ torr range as a minimum, with $280^\circ K$ surface. From Refs. 16 and 17, using Sutherland's relation for the viscosity, the mean free path $\lambda$ of the gas in the chamber can be written as

$$\lambda = \frac{5.328 \times 10^{-3}}{P} \times \frac{T^2}{T + 210.6}$$

where $\lambda$ is in inches, $P$ in microns Hg and $T$ in degrees Rankine

At the lowest pressure of operation

$$P = 0.1 \text{ micron Hg (10}^{-4} \text{ torr)}$$

and assuming the gas to have a temperature of

$$T = 144^\circ R (80^\circ K),$$

which is the temperature of the gas leaving the liquid nitrogen baffle, (see Section 4.5B), we find

$$\lambda = 3.12 \text{ inches}$$

which puts the flow regime in the transition range at the lowest pressure. However, it is expected that under most operating modes, the pressure in the pump chamber may be considerably higher than $10^{-4}$ torr and the state of the flow will then be closer to continuum than to free molecular conditions. The maximum pumping speed for continuum flow (with a capture coefficient of 1.00) is

$$S = \sqrt{\frac{2RT}{M}}$$

(see Section 2.4)

which for $N_2$ at $80^\circ K$ leads to a value

$$S = 18.24 \text{ liter/sec.cm}^2 = 1.67 \times 10^4 \text{ liter/sec ft}^2$$

The total surface area was chosen at a value of 130 ft$^2$. This gives an effective pumping speed of $2 \times 10^6$ l/sec, as compared with the required value of 370,000 l/sec at $10^{-3}$ torr representing the maximum mass flow of 0.6 gr/sec $N_2$. This large value was chosen for two reasons.

(1) The pumping speeds of cryogenic surfaces are not well established quantities and at an acceptable extra cost we could build a reasonable safety factory in our pumping area.
(2) The larger area is obviously not wasted since the pumping time is dependent upon the temperature gradient across the thickness of the condensate. Hence a larger amount of gas can be deposited on the freeze-out surface before the thickness builds up enough to produce a surface temperature that prevents any further pumping.

To fulfill all of the above requirements the freeze-out surface was made of 6 equally spaced banks of finned thin-walled copper tubes inter-connected by a series of "U" bends. Figure 21 shows how each individual bank is constructed and Figs. 22 and 23 show how the banks are arranged with respect to each other and with respect to the pump opening. The six banks are arranged in parallel to minimize the pressure drop in the neon flow across the freeze-out surface. This was done to keep the neon boiling pressure, and hence the neon temperature, as low as possible. As mentioned earlier (Sec. 2.3.1) we were already limited to boiling the neon at a pressure slightly above atmospheric and the freeze-out surface was built in such a fashion as not to aggravate this situation. However, as is pointed out in Sec. VIII, this parallel arrangement did not distribute the neon evenly to the 6 vanes, with the result that some of the vanes did not pump as well as others.

4.4 The Radiation Shield

If the freeze-out surface described in Section 4.3 is enclosed in a container whose walls are at 300°K, the total radiation load to the freeze-out surface when it is at 28°K, will be approximately 3000 watts. Such a large heat load is obviously undesirable and the 28°K surface must therefore be shielded from all surfaces at 300°K. To this end a liquid nitrogen cooled copper dewar (Figs. 24 and 25) surrounds the freeze-out surface on all sides except the top opening through which the gas to be pumped enters. Across the top, a LN2 cooled copper baffle with vanes at 45° every 3 inches blocks most of the radiation from the tunnel without impeding the mass flow to any appreciable extent (for detailed calculations see Appendix I and III). This LN2 cooled baffle also precools the test section gas to approximately 80°K (The cooling efficiency of the baffle is discussed in Appendix II). To further reduce the radiation to the freeze-out surface the LN2 cooled copper shield is lined with aluminum foil. This was necessary because the copper of the LN2 shield is very rough and has a dull appearance. The literature (Ref. 18) gives an emissivity of 0.6 for such copper, whereas the emissivity of the aluminum foil is estimated to be about 0.03, clearly resulting in an improvement. For the same reason of poor emissivity ($\epsilon_{Cu}= 0.6$) of the exterior copper wall of the liquid N2 radiation shield, the radiation load originating from the steel outer structural wall at 300°K, which also has a poor emissivity, was considerably reduced by filling the space in between these two walls with Perlite and by evacuating this space to a pressure of at most $5 \times 10^{-2}$ torr.
4.5 Intercooler Vanes

The intercooler vanes (Figs. 26 and 27) serve three purposes. They provide an opening in the radiation shield through which the gas can enter the pump cavity and be removed from the system. The vanes must also precool the entering gas to diminish the heat load on the freeze-out surface as much as possible. While allowing the gas to enter the pump chamber the intercooler vane must also block the 300°K radiation from the tunnel walls from seeing the 28°K freeze-out surface. The copper vanes of the intercooler were painted dull black to reduce the amount of radiation reflected off them from the 300°K surfaces above the intercooler to the freeze-out surface.

4.5.A Efficiency as a Radiation Shield

The efficiency of the intercooler vanes as a radiation shield is analysed in detail in Section V and Appendix III. It is seen that only a maximum radiation load of 32 watts can find its way to the freeze-out surface. If the intercooler were not present, 181 watts of radiation would fall on the freeze-out surface (Appendix III).

4.5.B Efficiency as a Heat Transfer Device

In Appendix II the intercooler is treated as a system of parallel ducts of uniform surface temperature through which the gas must flow before entering the pump chamber. The amount of heat transfer occurring between the vanes and flowing gas is calculated and from this an idea of the gas exit temperature is obtained. It is found that even under the worse possible flow conditions (that of a fully developed pipe flow toward the intercooler) the exit temperature of the gas is within 9°K of the intercooler vanes temperature. This situation is considered to be most satisfactory.

4.5.C Pressure Drop Across Intercooler

The pressure drop across the intercooler is calculated in Appendix I. It is seen that a pressure difference of 0.5 x 10^-3 torr is more than sufficient to pass the maximum design mass flow as can be seen from Fig. 28, which shows the results of the calculations of Appendix II, for the flow rate at an assumed cryopump pressure of 1 μ Hg.

4.6 Compressor and Control Plumbing

4.6.A Compressor

Because of the purity required of the neon working gas a standard compressor using various forms of piston rings as a sealant between crankcase oil and compressed gas was judged inadequate. Accordingly a Corblin diaphragm compressor was purchased. In this machine the gas is compressed by the pulsating action of metallic diaphragms. These are made to pulsate by the reciprocating motion of crank-mounted pistons acting on a fixed oil charge between piston head and diaphragms. Except for a catastrophic rupture of the metallic diaphragms, there is no way in which the crankcase oil can contaminate the working gas. However, the volumetric discharge
of the compressor was found to be dependent on having the proper amount of oil between diaphragms and piston. It is therefore important to ensure that the compressor is primed properly before each operation of the pump if the maximum refrigeration capacity of the system is desired.

4.6.8 Control Plumbing

As mentioned previously the neon undergoes a Joule-Thomson expansion; the gas cools itself down and eventually becomes liquefied. The gas undergoing this Joule-Thomson process is contained in a closed cycle system maintained at a positive pressure with respect to atmosphere.

Heat Exchangers

Beside the Joule-Thomson heat exchanger described in Sec. 2.3, a number of other heat exchangers are needed to increase the efficiency of the cycle. Figure 29 and Table III give the general arrangement of these heat exchangers.

Heat exchanger No. 3 is the Joule-Thomson heat exchanger described in detail in Sec. 2.3. It must be noted that there is an expansion valve allowing this heat exchanger to be bypassed for precooling the freeze-out surface. This allows for fast cool down of the freeze-out surface because if the Joule-Thomson heat exchanger was in the cycle, the warm vapor coming out of the freeze-out surface (location 7 on Fig. 29) would warm up the neon gas before it got to the Joule-Thomson expansion valve.

Heat exchanger No. 2 is located in the LN2 bath and is used to precool the neon before it enters the Joule-Thomson heat exchanger.

Heat exchanger No. 1 salvages some of the cooling potential of the cold ($\approx 66^\circ$K at 8 on Fig. 29) neon return vapor, and so reduces the heat-load on the liquid N2 bath.

Heat exchanger No. 4 salvages some of the cooling potential of the cold ($650^\circ$K) nitrogen vapor boiled off the LN2 trap again helping to keep the liquid nitrogen consumption down. This heat exchanger could not be built very efficiently, as the line carrying the LN2 vapor must be fairly large to permit the vacuum pump to handle the large volumetric boil off from the bath.

Components

The plumbing system and compressor are depicted in greater detail in Figs. 18, 30 and 31. All the piping of the system (excluding the compressor itself) is of stainless steel. The valves are quarter-turn ball valves with Teflon seals and seats. Most of the pipe joints are heli-arc welded except for a few places where "O" ring seals of one type or another had to be used. These "O" ring seals had to be used on components such as valves or gauges that could not be bought in a welding configuration.

As mentioned before (Section 2.3) neon is a rather expensive gas. The total neon charge in the refrigeration system is worth approximately $6,000.00 (this includes the reserve in the storage bottles). It is therefore imperative to reduce leakage from the system to a minimum. The whole
A neon handling system was built and tested as a high vacuum system. It was leak tested using a helium leak detector and no individual leaks larger than $10^{-6}$ STD. c.c/sec were tolerated. This figure was chosen because such a leak rate from a 2000 psi service line would cause the loss of an amount of neon worth less than $0.50 per year per leak at the present prices.

A general description of how the basic elements of the pressure control system function has already been given in Section 4.1 where the function of items 1 to 4 on Fig. 30 is explained. The function of the additional elements shown on Fig. 30 follows here.

Item (5) is a pressure reducing valve which reduces the pressure in the low pressure return line and so maintains the suction pressure of the compressor at approximately 0.5 psig. The low suction pressure allows the compressor to be started under a practically no load condition. This pressure reducing valve also serves to isolate the line pressure fluctuations caused by the compressor strokes from the rest of the suction line.

Item (6) are low pressure storage bottles. When the system is shut down and the compressor stopped, all the high pressure gas in the cycle (except the supply gas) is dumped into this storage volume to be stored at reduced pressure ($\approx 30$ psig). This insures that any leak in the plumbing will lose gas at a rate corresponding to a 30 psig differential pressure instead of a 2000 psig differential.

Because of the need to make each component as leak tight as possible, some commercial components had to be extensively redesigned and others modified, to insure their leak tightness.

Extensive modifications had to be made to pressure reducing valves, which contained an atmosphere-sensing rubber diaphragm. These are the valves labelled 4 and 5 on Fig. 30.

Figure 32 shows a cross section of the valves used for items 4 and 5. Since these valves were found to leak at just about every one of their joints they were a serious problem. Figure 33 depicts the modifications that rendered these pressure reducing valves suitable for our use.

Item (1) Since the original method of clamping the rubber diaphragm between the two halves of the valve body proved most unsatisfactory (large leak rate) another clamping method had to be devised. To this end a clamping plate with "0" ring seals was manufactured. The "0" ring seals serve to seal the two halves of the valve body. The diaphragm is clamped between two metal rings with mild concentric corrugations on their mating faces. The bolts holding the two metal rings together are spaced every inch or so, along the circumference of the rings. This prevents any warping of the mating surfaces. A positive seal was achieved by generously covering the mating faces of the rings with thick vacuum wax before clamping the diaphragm.

Item (2) In the event of the rupture of the rubber diaphragm, the spring balancing the gas pressure on the diaphragm will cause the valve to lock in the open position, allowing the gas in the system to rush out. When the valve opens, the valve piston moves to the left.
To prevent the loss of gas in the event of a diaphragm rupture, a holder with "O" ring seals was machined and inserted in the entrance to the passage leading to the diaphragm chamber of the valve. When the diaphragm breaks and the piston moves completely back, its back side will seal against the "O" ring of the holder at the entrance of the passage. The force of the spring on the lever arm of the piston will maintain a positive seal on the "O" ring. This feature was tested and found to operate satisfactorily.

Items 3 and 4: "O" ring seals were used to replace ordinary gaskets at these points.

Item 5: The rubber diaphragm was found to be too weak to permit the evacuation of the neon side of these pressure reducing valves. Since all air had to be purged from the system before neon was admitted, it was imperative that a way be found to allow evacuation of these valves. The valve arrangement shown as item 5 on Fig. 33 solved the problem by permitting the evacuation of both sides of the rubber diaphragm simultaneously.

Because of the low impedance they present to the gas flow and their fast and positive action, quarter-turn ball valves were used throughout the neon handling system. Since a clean system with leak free components was desired, stainless steel valves with Teflon seats and seals were used wherever possible. Unfortunately this type of valve was available only with pipe thread joints for high pressure service. This would make the installation and removal of these valves from the system extremely difficult. Also, it was found to be nearly impossible to seal these pipe thread joints adequately and with any degree of permanence. Figures 34 and 35 show the "O" ring and double flange arrangement that was eventually adopted with excellent results: the seal proved permanent and valve replacement is easy.

4.6.C Placement of the System

Figure 36 shows the physical location of the cryogenic pump and its control plumbing. The plumbing was located in a pit to save floor space.

4.7 Test Section Pressure Control Valve

Since cryopumps are constant mass flow devices independent of the pressure at which they operate, it will be necessary to use pressure control valves between the pump chamber and the tunnel to control the test section pressure. Figure 37 shows the test section attached to the wind tunnel and indicates the location of these valves which are gate valves of 16" diameter. Appendix IV lists the calculations used to predict the possible throughput of these valves which has been plotted on Fig. 28 in addition to the throughput of the intercooler. Figure 37 shows that the two 16" diameter valves are mounted on a dished head of 4.5 foot diameter that covers completely the entrance to the pump chamber. This 4.5 foot cover is a valve that can be moved up and down within the Tee section of the wind tunnel and acts as the main shut-off valve for the cryopump. It allows the test section to be brought up to atmospheric pressure to permit model changes, etc., while keeping the cryogenic surface under vacuum.
4.8 Remote Control Operation of the Cycle

Because the pressure control plumbing is located below floor level and the tunnel test section is at floor level it was felt desirable to have the capability to control the whole cycle of operation of the cryogenic pump remotely.

A quick glance at Fig. 31 will show that most of the valves are operated by electric motors. The majority of the remaining, less important, valves have limit switches to indicate an open or closed condition.

Provisions have been made for two electric control panels from which this electric circuitry can be monitored and operated. One of these control panels is located on the same level as the compressor and plumbing (Fig. 36). The other will eventually be located close to the test section of the wind tunnel. This arrangement allows the operation of the pump from these two locations.

The control panel located near the test section will also contain electric pressure metering devices (the plumbing system is equipped with three potentiometric pressure transducers).

Under normal operation all the lights of the control panel are green. It is therefore possible to detect any abnormalities at a glance.

4.9 Controlled Heat Load to the Pump

In order to make quantitative measurements of the amount of refrigeration available from the neon gas expanded through a Joule-Thomson plug at various temperatures, a known heat load had to be applied to the pump.

For this purpose heating elements (Fig. 38) were sealed in air-tight containers and attached (Fig. 39) to the 6 pipes carrying the liquid neon from the manifold header to the copper vanes. The heaters had to be sealed in air tight containers because they were of the type that uses tightly packed powder to conduct the heat away from the filament to the casing of the heater elements. Operating in a vacuum the heat conductivity between the filament and casing might have been reduced enough to cause the filament to overheat and burn out. In the experiments their power input was controlled to give balanced conditions as discussed in Sec. 2.3. The mass flow of neon through the J.T. cycle was measured by a fully enclosed rotameter flow meter (Fig. 40).

V. RADIATION AND CONDUCTION HEAT LEAKS TO THE FREEZE-OUT SURFACE

It is a matter of great importance to reduce to a minimum the radiation leaks to the freeze-out surface since the copper vanes and the condensate have an emissivity in the region of 0.6 to 0.9 (Refs. 18, 19 and 2). As indicated previously this is the primary reason why the freeze-out surface was surrounded with a LN₂ shield. To further reduce the radiation losses the vertical sides of the LN₂ shield were lined with aluminum foil (\( \varepsilon = 0.03 \), Ref. 20). This was made necessary because the LN₂ shield was manufactured of dull copper (\( \varepsilon \approx 0.6 \), Ref. 18). Appendix III gives the calculation of the
heat leaks to the freeze-out surface from this foil covered shield and other sources.

The largest of these other radiation sources is the $300^\circ K$ tunnel walls that emit through the intercooler openings and strike the $28^\circ K$ freeze-out surface. A rigorous calculation of these losses is also given in Appendix III.

To this radiation going through the intercooler slots must be added to the radiation reflected off the blackened intercooler. Reference 10 gives an emissivity of approximately 0.74 for the black glyptal paint used. Hence approximately 30% of the incident radiation is reflected. Of this, one can assume that about half is reflected to the freeze-out surface, whereas the remainder is reflected back to the tunnel walls. This estimate is based on the assumption that the radiation from the $300^\circ K$ surface is reflected only once before entering the pump cavity.

Since the two 16" diameter valves can pass a mass flow of 0.63 gr/sec $N_2$ under most conditions (Fig. 28) it appears that the 4.5 foot diameter valve will remain on top of the pump cavity for most operating conditions. This, as the calculations depicted in Appendix III and the results tabulated in Table IV show, results in a large decrease in radiative losses. This is brought about because the underside of the 4.5 foot diameter valve (Fig. 37) can be lined with aluminium foil ($\varepsilon = 0.03$). The emissivity of the $300^\circ K$ tunnel wall is approximately 0.5 (white paint). The tunnel walls cannot be lined with aluminium foil as they would reflect the light from the test section arc jet into the pump chamber.

Gas conduction losses within the pump chamber are minimal. Reference 11 states that the capture coefficient of the chosen type of freeze-out surface (finned cylinders) is close to 1.00. This means that practically all molecules are condensed upon hitting the cold surface. Under these conditions convection heat transfer is dominant and conduction may be ignored.

The losses due to conduction along supporting structures are insignificant as shown in Appendix III.

VI. LIQUID NITROGEN REFRIGERATION SYSTEM

6.1 Reduced Pressure Boiling Capability of LN$_2$ System

As pointed out in Sec. 2.3.1, the amount of cooling obtainable from the Joule-Thomson expansion of neon is critically dependent on the expansion temperature (Fig. 11). For the pressure of 135 atm. in our system it is necessary to maintain the gas entering the Joule-Thomson heat exchanger at $64^\circ K$ for optimum performance. This requires a reduced pressure boiling capability of the liquid nitrogen handling system. Figure 41 shows the pump arrangement that permits the LN$_2$ to be maintained at 2.1 psia (corresponding to a temperature of $64^\circ K$) (Fig. 7). As mentioned in Sec. 3.1 our present arrangement is less than ideal since the reduced pressure boiling capability is really only needed for the precooler bath of the neon cycle, and the present arrangement requires that the nitrogen that is used to cool the whole copper shield be maintained at $64^\circ K$. An additional
problem existed because the pumps used to depress the boiling point of the liquid nitrogen bath are normally used as backing for oil booster pumps. They normally work at about 0.5 mm Hg. of inlet pressure and the higher power of sustained operation at a pressure of about 50 mm Hg. at the pump inlet required additional cooling by provision of forced circulation of the oil through a water cooled interchanger.

6.2 Consumption of the Liquid Nitrogen System

The liquid nitrogen system must provide enough cooling to maintain the copper shield at 64°K despite the radiation and conduction load. It must also cool down the test gas from the test section to approximately 80°K through the intercooler and it must cool the incoming neon to 64°K.

If one assumes an $\epsilon \approx 0.6$ for oxidized copper (Ref. 18) the radiative load of the 300°K steel structural shell upon the liquid nitrogen cooled copper shield amounts to 3000 watts. To reduce this load and thereby the LN$_2$ consumption, Perlite was introduced in the dewar annular space (Fig. 24). Perlite evacuated to below 50 micron Hg, was quoted to have an effective conductivity of 0.02 - 0.03 B.T.U. - in/hr-ft$^2$-°F in Ref. 21. This value would reduce the load on the copper shell to approximately 70 watts. It must be pointed out that the manufacturer of Perlite and Ref. 22 state lower values for the conductivity of Perlite under vacuum. However, the more pessimistic value of Ref. 21 was used in our calculations.

The precooling of the mass flow to the pump to 80°K, the conduction from the 300°K steel shell and the cooling down of neon gas to 64°K are considered in detail in Appendix V. The resulting consumption of liquid nitrogen is given in Table V. It is seen that consumption rates between 21 and 12 liters per hour are predicted.

It is also necessary to cool down the cryopump from room temperature to 64°K at the start of each run. Table VI lists the structural members and fixes at 309 liters the liquid nitrogen required.

6.3 Cool Down Time

Assuming that the Joule-Thomson heat exchanger is by-passed (Fig. 29) the neon is precooled to 64°K and circulated through the freeze-out surface. It will exit from the freeze-out surface at a temperature $T$, which is the temperature of the freeze-out surface and which is gradually approaching the temperature of the incoming stream of gas ($T_1$).
The cooling process, as described above, may be represented mathematically by
\[ \frac{\partial}{\partial t} m C_p (T - T_1) = -m C_v \frac{d}{dt} (T) \]
or
Refrigeration from the gas = Cooling down of the mass of metal

Since \( T_1 \) is a constant we may write
\[ \frac{\partial}{\partial t} m C_p (T - T_1) = -m C_v \frac{d}{dt} (T - T_1) \]

Rearranging and solving, we get
\[ \frac{m C_p}{m C_v} t = T - T_1 = A e^{t - t_0} \]

Now, at \( t = 0 \), \( T = 300^\circ K \)

Hence \( A = 300 - 64 = 236 \)

The cool down equation is then
\[ \frac{m C_p}{m C_v} t = T - T_1 = 236 e^{t - t_0} \]

To find the cool down time "t", we must estimate the time constant, making sure that we correct the values of the specific heats for the mean temperature existing during cool down. (i.e. \( \approx 180^\circ K \)). We have

\[ \frac{\partial}{\partial t} m C_p = 21 \frac{SCF}{min.} \times 0.25 \frac{BTU}{lbs \ C_p} \times 0.056 \frac{lbs}{c.f.} \]
\[ = 0.294 \frac{BTU}{min. ^\circ F} \]

\[ m C_v = [m C_v]_{Cu} + [m C_v]_{S.S.} \]
\[ = [230 \times 0.06] \times [50 \times 0.06] \]
\[ = 16.8 \frac{BTU}{^\circ F} \]

The equation for the cool down time is therefore
\[ T - T_1 = 236 e^{-0.0175t} \]

We consider that the freeze-out surface has cooled down when
\[ T - T_1 = 2 \]

The time required to reach this level is
\[ \frac{2}{236} = .0085 = e^{-0.0175t} \]
This gives \( t = 272 \) minutes 
or \( 4.5 \) hours

To this must be added the time necessary to cool down the 230 lbs of copper and 50 lbs of stainless steel of the freeze-out surface from \( 64^\circ\text{K} \) to \( 28^\circ\text{K} \) using the Joule-Thomson heat exchanger. The refrigeration rate of the Joule-Thomson cycle is a constant and from the data of Fig. 11, Sec. 2.1, this is equal to the specific refrigeration times the volume flow rate of the compressor, i.e.

\[
0.97 \frac{\text{BTU}}{\text{SCFM}} \times 20.3 \text{ SCFM} = 20.2 \frac{\text{BTU}}{\text{min}}
\]

We need \( \approx 600 \) BTU to cool down the freeze-out surface through the \( 36^\circ\text{K} \) of temperature from \( 64 \) to \( 28^\circ\text{K} \). Hence

\[
\frac{600 \text{ BTU}}{20.2 \text{ BTU/min}} \approx 0.5 \text{ hour} = 30 \text{ minutes}
\]

The total estimate time for cool down is then approximately 5 hours.

VII. PURPOSE OF THE EXPERIMENTAL PROGRAM

The purpose of the experimental program carried out on the cryogenic pumping system was

1. to determine the ability of the design to function as planned,
2. to determine the refrigeration capacity of the neon system under various operating conditions and to compare the results with the predicted capacity based on the theoretical neon thermodynamic data,
3. to determine the pumping speed and time of operation available with various mass flows before the pressure rise in the pump chamber requires the defrosting of the pump,
4. to determine the time required to defrost the pump,
5. to determine the liquid nitrogen consumption rate.

In the following subsections the experimental procedure to make the measurements will be described. The data obtained is given and discussed in Sec. VIII.

7.1 Liquid Nitrogen System

The amount of liquid nitrogen consumed was monitored by vacuum gauges located on the vacuum pumps used to depress the boiling point in the \( \text{LN}_2 \) trap. Thermocouples were also attached to the \( \text{LN}_2 \) cooled copper shield and its temperature history was followed. The consumption rate of \( \text{LN}_2 \) was calculated by using the manufacturer's data on the volume flow rate of the pump in conjunction with the measured pressures at the pump inlet.
The level of the liquid nitrogen within the trap was monitored by a differential pressure gauge. The trap was kept full with the help of this gauge and a manually operated valve on the LN₂ feed line.

7.2 Refrigeration Capacity of the Neon System

The experimental configuration to measure the refrigeration capacity of the neon system is depicted schematically in Fig. 42. Twelve 50 watt heating elements were attached to the freeze-out surface (Figs. 38 and 39) as described in Sec. 4.9. The power input was continuously variable from 0-600 watts by means of six powerstats, one for each set of two elements on a single vane. A voltmeter and ammeter arrangement allowed the determination of the power dissipated by each set of two heaters separately. A double check was available because the resistance at low temperature was also known.

A rotameter flow meter with a 2% accuracy and electrical readout was placed in the return line from the Joule-Thomson valve and measured the actual gas flow through this valve. Temperatures and pressures were continually monitored upstream of the meter to insure the accuracy of the mass flow reading. A photograph of the meter and associated instruments is shown in Fig. 40. The rotameter flow meter provided both a direct mechanical indication of the flow rate and an auxiliary electrical output. The relationship between the mechanical and electrical read out of the rotameter was found to be constant as shown by the calibration results of Fig. 43.

As mentioned in Sec. 2.3, the Joule-Thomson cycle is operated to produce liquid neon as the actual cooling agent in the freeze-out surface. This means that an excess cooling capacity will cause an accumulation of liquid neon in the evaporator (freeze-out surface), whereas an excess heat load will cause the evaporator to dry-up. The experiments were performed under balanced conditions by adjusting the neon flow rate to give equilibrium between heat input and refrigeration output to the evaporator.

This condition of equilibrium between the refrigeration furnished by the expanded neon and the heat input of the heaters was established when the pressure of the supply bottles of neon remained constant. A decreasing pressure in the supply bottles meant that some excess neon gas was being transformed into a liquid; hence too much refrigeration was furnished to the freeze-out surface to compensate for the fixed heat input of the heaters. In such a case the Joule-Thomson valve was then throttled down. If, on the other hand, the supply pressure was found to be increasing, then some of the pool of liquid neon inside the freeze-out surface was warming up due to a surplus of heat input to the vanes. In this case the Joule-Thomson valve opening was increased to augment the cooling capacity.

The supply bottles of neon were fitted with a potentiometric pressure transducer. Its resistance was monitored by a Wheatstone bridge and a galvanometer indicated any change in the pressure input to the transducer. The transducer had a manufacturer's rating of 3% accuracy. Since the galvanometer and Wheatstone bridge were of laboratory standard quality, the value of 3% can be taken as the accuracy of the bottle pressure measurement.
The temperature of each vane was monitored by Copper-Gold (2.1 At. % Cobalt) thermocouples immediately downstream of the neon manifold and at a point about 18" downstream of the heater elements, as shown on Fig. 42. A liquid nitrogen "hot" junction was used, the E.M.F. of the thermocouple was read by a high quality potentiometer-galvanometer arrangement and the temperature was calculated based on published calibration data for this kind of thermocouple combination. This arrangement measured the temperature with an estimated maximum error of approximately 5%. Another set of thermocouples placed on the LN$_2$ bath gave temperatures which could be checked against the liquid nitrogen temperatures inferred from the pressure above the liquid. Such checks indicated that this accuracy figure was a reasonable one.

7.3 Pump Performance

The mass flow of the gas introduced into the pump chamber above the intercooler was monitored by calibrated rotameters. The surface temperature of the freeze-out surface was followed by means of a commercial gas thermometer (Cryogenics Inc.) and numerous thermocouple stations. Cold cathode ionization gauges, Pirani and McLeod gauges monitored the pressure on top of the intercooler during each operation of the pump.

The gas was introduced in the pump chamber through an ½" diameter opening situated at the top of the vertical axis of the cryopump. The gas flow was broken up upon entry in the vacuum vessel by a baffle situated a few inches away from the exit of the gas inlet pipe. This baffle was approximately 2 feet above the center of the LN$_2$ cooled intercooler and radiation shield.

VIII. EXPERIMENTAL RESULTS AND DISCUSSION

8.1 Liquid Nitrogen System

8.1.A Initial Temperature of the Joule-Thomson Expansion and Cool Down Time

The pipe carrying the neon through the liquid nitrogen bath is submerged 28.5 cm. below the surface of the liquid. This creates an hydrostatic head equal to approximately 17 mm Hg., which added to the 110 mm Hg maintained on the surface of the liquid by the vacuum pumps brings the local value of the boiling point of the nitrogen to 65°K. Since liquid nitrogen freezes at 63°K, it was found impractical to reduce the pressure on top of the trap below the nominal 110 mm Hg since it was found that fluctuations in the control mechanism on occasion caused the pressure to fall so low that solidification of the liquid nitrogen occurred.

The thermocouple attached to the neon carrying pipe coming out of the liquid nitrogen heat exchanger registered between 65 and 67°K based on the standard calibration tables for the thermocouple. Since this thermocouple was not calibrated in situ and since the thermocouple was attached to the outside wall of the neon carrying pipe, this 1-2% discrepancy between the predicted and measured temperature must be considered as quite acceptable. The thermocouple results are shown on Fig. 44 which indicates a 1 to 2 degree fluctuation in the Joule-Thomson expansion temperature.
The temperature at the exit of the freeze-out surface is plotted as a function of time in Fig. 45. The time where the switch over from operation with the by-pass valve to that with the Joule-Thomson valve took place is indicated on the figure. It is seen that the total indicated cool down time is approximately 4 hours.

Since the measuring thermocouples are attached directly to the neon carrying pipe, it was believed that they measured a temperature that was lower than the average temperature of the rest of the vanes. This was found to be correct by observing the mass flow of neon for a period of time after the thermocouples had indicated a temperature of 28°K on the freeze-out surface. The mass flow rate of neon required to maintain the system at equilibrium (as explained in Sec. 8.2) is plotted on Fig. 46 as a function of the time after the thermocouples have indicated a 28°K temperature. It is seen that approximately another full hour elapses before the neon flow rate stabilizes.

This indicates a total cool down time of approximately 5 hours, in good agreement with the prediction of Sec. 6.3.

8.1.B Liquid Nitrogen Consumption

It was stated previously (Sec. 3.1) that a 400 scfm vacuum pump was necessary to pump the boil-off vapors from the LN$_2$ trap. Since the Institute has two 485 scfm Kinney pumps manifolded together, it was expedient to use both these pumps to evacuate the LN$_2$ bath. Under continuous suction of the pump with the maximum mass flow of gas to be pumped, the two Kinneys evacuated the liquid nitrogen vessel to the required 110 torr at a suction pressure at the pumps of approximately 10 torr. This difference in pressure exists across the control valve. Assuming that the pumping speed at 10 torr for the pumps is the nominal value for the pumps, we have a volume flow rate of

\[ 2 \times 485 \times 28.3 = 27,500 \text{ l/min} \]

where 28.3 is the conversion factor from cubic feet to liters. The corresponding mass flow at a suction pressure of 10 torr is then 27,500 x 10 torr l/min or

\[ 361 \text{ atm. l/min} \text{ of nitrogen gas} \]

This in turn corresponds to a consumption rate of

\[ 30.5 \text{ l/hr} \text{ of liquid nitrogen} \]

The pump performance has fallen to approximately 90% of its original value in the range from 10 - 20 torr, according to the manufacturers of the pumps. Hence the consumption is approximately 27 l/hr LN$_2$. This figure is in good agreement with the predictions of Sec. 6.2 tabulated in Table V.

The analysis of Sec. 6.2 does not take into account the heat conductivity of the gas when calculating the heat load on the freeze-out surface. When gas is admitted to the pump, the pressure on top of the intercooler rises to approximately $3 \times 10^{-3}$ torr from $10^{-5}$ torr when no test gas entered the pump. In the case when the pressure between the intercooler and
the dome of the pump is in the $10^{-3}$ torr region, the gas contained in this volume is in continuum flow with respect to the warm ($300^\circ$K) walls of the cryopump. Therefore the intercooler also has to dissipate conduction heat from the walls in this mode of operation. (In fact, the dome gets noticeably colder to the touch when gas is being admitted to the pump.)

Assuming with Dushman (Ref. 23) that the heat conductivity of a gas is independent of pressure as long as the pressure is higher than the range in which molecular flow occurs, we take as the value of "$k$", the heat conductivity constant for nitrogen gas,

$$k = 62.5 \times 10^{-6} \text{ cal/sec cm } ^{0}\text{K}$$

The intercooler and the dome of the pump presents each other with an area that is roughly 4.5 feet in diameter. The heat transfer between these two area would require an expenditure of 0.63 l/hr LN$_2$.

We see that the omission of this contribution to the heat load on the liquid nitrogen system was valid.

8.2 Refrigeration Capacity of Neon System

The refrigeration capacity of the neon system was determined by supplying a known heat load by electric heaters to the freeze-out structure and measuring the flow rate through the Joule-Thomson valve that gave balanced conditions as indicated by the fact that the neon pressure in the storage bottles remained constant. It was found almost immediately that it was impossible to supply equal power to the heaters on each of the six vanes. This is a consequence of the fact that the liquid neon produced in the Joule-Thomson expansion has to be distributed over the six vanes and quite obviously irregularities and asymmetries in the manifold must have resulted in a markedly uneven distribution of the liquid neon. The amount of power to each vane was therefore carefully adjusted to balance the neon supply to that vane, by observing the temperature difference as indicated by the thermocouples just upstream and downstream of each heater assembly. When too much power was dissipated in a particular vane, this showed as a drastic rise in downstream temperature and the power was then adjusted to keep the two temperatures about equal. After adjustment of the heaters and overall balancing of the power and the mass flow in the Joule-Thomson cycle, the power from the units could then be added to compare the total power with the mass flow. The maximum power input to each vane for a typical run at equilibrium conditions is shown in Fig. 47 and illustrates the vast difference that exists in the amount of neon carried by each vane in the supposedly symmetrical arrangement.

It was also found in the experiments that the thermal inertia of the freeze-out surface made the experiments cumbersome, because of the long times required before equilibrium conditions were reached. Of course, this same feature is a desirable one for use as a pump because it will make the pump insensitive to sudden changes in heat load.

Great confidence existed in the performance of the heater elements as their resistance was constant from run to run and within each run as shown in Fig. 48 which plots a typical voltage-current characteristics. Since the heaters were wrapped in cotton and then covered by many layers of
aluminum foil, there was no radiation losses from them.

8.2.A Neon Flow Rate Compared to the Electric Power Input

The results of these experiments are given in Fig. 49 where the neon mass flow rate is plotted against the electrical power input for two temperatures of the liquid N\textsubscript{2} precooling bath. The experimental points fit straight lines, the slope of which indicates the cooling power per unit mass flow. It is clearly seen that operation at the lower initial temperature resulted in a better specific refrigeration.

The intercepts of the two straight lines of Fig. 49 with the vertical axis, indicate that at zero power input a mass flow is required to balance the heat leaks to the freeze-out surface. From the intercepts, the magnitude of the heat leaks can be determined by converting the mass flow into power by using the values of the specific refrigeration as determined from the slopes of the lines.

Based on an intercept value of 2.5 SCFM for the curve at the initial temperature of 67°K, the heat leak to the freeze-out surface is approximately 37 watts, in good agreement with the prediction of 42 ± 4 watts of Appendix III.

The specific refrigeration as a function of initial temperature is plotted in Fig. 50, where the theoretical curve based on the data of Ref. 9 is compared with the two experimental points. It is seen that the actual performance was about 10% lower than the predicted theoretical one. This is a very reasonable figure since the theoretical curve assumes an ideal cycle, i.e., perfect heat exchanger with no losses through radiation or viscous effects.

Although the nominal capacity of the compressor at 135 atm. is 21 SCFM, the magnitude of the mass flow through the Joule-Thomson valve never reached this value. This was due to the sensitivity of the compressor performance on the proper priming of the compressor cylinders (as a matter of fact, variations was found from run to run). The largest mass flow measured in this series of experiments was a value of about 16 SCFM.

8.2.B Neon Flow Rate Compared to Test Gas Flow Rate

A second series of runs were made in which the heat load on the freeze-out system was provided by a gas load of nitrogen that condensed on the cold surfaces. In this case the system fulfilled its intended purpose of pumping gas. The system was again run under conditions where the neon mass flow was adjusted to balance the heat load and the results are plotted in Fig. 51 as a function of the mass flow of the pumped gas. Again this was done for the same two temperatures of the liquid N\textsubscript{2} precooling bath. The experiments cover the same range of neon mass flows as in the runs with the electrical heaters, but before the flow meter was installed some runs were made at a mass flow of 0.8 gr/sec. The effective cooling power to condense the N\textsubscript{2} on the freeze-out surfaces was obtained by using the data on the specific refrigeration from Fig. 49 and the data of Fig. 51. The data of Fig. 52 is the result of cross plotting the data of Fig. 49 and Fig. 51. in such a way that the neon consumption rate disappears from the relationship. From the slope of the curve of Fig. 52 it was found that the cooling power per unit mass of nitrogen was
36 watts/lbs \(N_2/\text{hr}\)

This value compares favourably with the predicted value of

38 watts/lbs \(N_2/\text{hr}\)
as given in Table II.

As was mentioned previously, a mass flow of 0.8 \(\text{gr/sec}\) of \(N_2\) could be pumped under balanced conditions, indicating that the pump was capable of somewhat higher flow rates than those in the experiments described in this section. Even though this value is lower than the nominal capacity of the compressor, the system proved capable of pumping a slightly higher mass flow (0.8 \(\text{gr/sec}\)) than the design value of 0.63 \(\text{gr/sec}\).

8.2 C Effect of Expansion Temperature

Figures 49, 50 and 51 show the worsening of the pump performance when the Joule-Thomson expansion temperature is raised from 67\(^\circ\)K to 80\(^\circ\)K. There is a 0.292 B.T.U./c.f. drop in the refrigeration capacity of the system. This is a 35\% worsening in performance which corresponds to the amount predicted by theory (Fig. 11 and 50). This illustrates the importance to depress the \(LN_2\) boiling point to the 66-67\(^\circ\)K region.

As seen from Fig. 44 there were fluctuations in the temperature of the precooler bath of a few degrees. Under the worst condition (67\(^\circ\)K), the deviation from the ideal 64\(^\circ\)K temperature causes a 4.5\% loss of cooling capacity from the ideal value of approximately 0.978 B.T.U. A shallower precooler bath would be able to retrieve about 1\(^\circ\)K of the 3\(^\circ\)K loss but the difficulty in keeping it full at all times would far out-weigh the 1-2\% gain in pump performance.

8.3 Pump Performance

On Fig. 53 is plotted the pressure above the intercooler as a function of length of pumping time for three different mass flows admitted to the pump chamber. The performance of the complete system as a pump was investigated by admitting known continuous mass flows of nitrogen to the chamber above the intercooler and measuring the pressure in this chamber as a function of time. The results for these values of the mass flow are presented in Fig. 53. The highest mass flow run was of 10 hours duration at the same value of mass flow. The lowest mass flow run lasted 4.5 hours and then a slightly higher mass flow was admitted to give the data for the intermediate mass flow rate. The striking feature of these curves is the absence of a pressure rise with time as the condensate layer built up on the vanes. This must be credited to the extremely large surface area of the freeze-out surface. The nominal surface area of the star shaped freeze-out surface is 127 \(\text{ft}^2\) but the actual surface area of the six vanes is approximately 450 \(\text{ft}^2\) as a result of the finned type construction. Even taking this larger area fully into account, according to French and Muntz's prediction (Ref. 3)\(\text{there should have been an appreciable rise in pressure.}\)

As an example, their predictions as redrawn in Fig. 13 indicate that a pressure of \(10 \times 10^{-3}\) torr (10 micron Hg) should have been reached after 2\(\frac{1}{2}\) hours of running time at a mass flow similar to that of run No. 8 of Fig. 53. However, during run No. 8 (\(\dot{m} = 0.68 \text{ gr/sec}\)) an average pressure
above the intercooler of approximately $3.5 \times 10^{-3}$ torr, with about 8% fluctuation was measured and this value was maintained for 10 hours.

The calculations of French and Muntz were based on values of the thermal conductivity and density of the condensed phase as inferred from their data obtained with a scale experiment. The heat conductivities they determined appeared to be strongly dependent on the deposition rate. The value of the conductivity at the nominal deposition rate corresponding to our maximum mass flow was considerably lower than the one quoted by Bailey and Chuan (Ref. 1) of $4.0 \times 10^{-7}$ cal/sec cm°K. Since Bailey and Chuan's data was obtained at the temperature of liquid $H_2$ (about 20°K) and those of French and Muntz at about 29°K, the difference in these results might have been real. However, the running times encountered in the present experiments definitely imply a relatively high value of the thermal conductivity, which cannot be reconciled with the values of French and Muntz. On the other hand, if we assume a density of 1 gr/c.c. and use the conductivity given by Ref. 1 and further assume that the entire 450 ft$^2$ of the freeze-out surface is actively pumping, then after 10 hours running at a mass flow rate of 0.63 gr/sec (5 lb/hr) the condensate thickness would be 0.05 cm, corresponding to a surface temperature of 30°K and a vapor pressure of the condensate of $1 \times 10^{-5}$ torr. (See Appendix VI for calculations.) Even assuming that only half the surface area of the freeze-out surface is pumping, the surface temperature of the condensate is $\approx 33°K$ and the vapor pressure is only about $8 \times 10^{-4}$ torr. Such pressures are not inconsistent with our experimental run at 0.63 gr/sec (5 lbs/hr) (run # 8, Fig. 53).

It is clear from these experiments that the conductivity value of Bailey and Chuan is of the right order to explain the present results. Unfortunately, no adequate explanation for the results of French and Muntz can be proposed here.

In the smallest mass flow run (run No. 9, $m = 0.38$ gr/sec) (Fig. 53) an apparent rise of 13.6% in pressure can be noticed. After 10 hours, if this effect is real, this would cause a $0.5 \times 10^{-3}$ torr rise in tank pressure. However, a similar conclusion could have been drawn by using only the first 4.5 hours of the highest mass flow run; i.e. the fluctuation in pressure during the first 4.5 hours of run No. 8 is in such a fashion as to lead one to believe that the pressure above the intercooler is slowly rising, a conclusion that is proven false by the behaviour of the pressure during the remainder of the ten hours duration of this experimental run. Therefore, it appears reasonable that the observed effect is of the nature of a fluctuation and we therefore consider that the smallest mass flow run has stabilized at approximately $2.2 \times 10^{-3}$ torr.

Figure 54 shows a plot of the equilibrium pressure of the pump for various mass flows. This relationship applies only to our pump configuration as the effect of the pressure drop across the intercooler influences the 3 experimental points of Fig. 54. The precise explanation of these measured pressures is not quite clear, but several factors may have contributed to it.

(1) It must be pointed out that the computed (Appendix I) pressure drop across the intercooler is much smaller than these measured values since the calculations of Appendix I gave a pressure drop at the highest mass flow of about $0.5 \times 10^{-3}$ torr (0.5 μHg). However, this calculation was
based on the assumption of a uniform flow distribution across the inter-
cooler, which in fact may not have been realized. This would cause a
higher pressure drop, but the magnitude of it would be difficult to estimate.
Furthermore, the flow through the intercooler is in the transition regime
and the calculations are therefore somewhat uncertain. This situation is
further complicated by the fact that large temperature changes occur as the
gas flows through the intercooler.

(2) The temperature of the freeze-out surface is higher than anticipated.
This possibility can be ruled out, because quickly after shutting off the mass
flow, the pressure dropped to a very low value, indicating that the condens-
ate temperatures were of the right magnitude.

(3) The driving pressure associated with the interphase mass transfer
is higher than that calculated in Sec. 2.4. Since under our conditions, the
equilibrium vapour pressure is very much lower than the driving pressure, we
can calculate the pressure in the tank that is required to drive the mass
flow into the freeze-out surface by considering that all particles stick
ideally (i.e. captive coefficient = 1). As mentioned in Sec. 4.3, under
continuum conditions, the pumping speed of our freeze-out surface is $2 \times 10^6$
liters/sec. If we assume that only two vanes are pumping, the pumping speed
is $7 \times 10^5 \ell$/sec. In run number 8, Fig. 53, the mass flow of 0.63 gr/sec
corresponds to approximately $371,000 \mu \ell$/sec. We require only $0.5 \mu$ Hg
($3.7/7 \mu$ Hg) to drive this mass flow to the vanes. This driving pressure
therefore cannot explain the $3.5 \times 10^{-3}$ torr ($3.5 \mu$ Hg) in the pump at the
time.

(4) The conductance within the pump chamber may cause a pressure rise
at the pump entrance. Since there is evidence that not all vanes contribute
equally to the condensing process, an appreciable fraction of the gas may
have to flow across the inactive vanes to reach those that are working
properly, leading to pressure gradients. There will also be a pressure drop
associated with the gas flowing to the lower part of the active freeze-out
surface. These non ideal flow conditions create pressure differences which
lead to non-uniform condensation rates which have the overall effects of
raising the effective pressure in the pump chamber.

On the basis of the above considerations, the observed results
do not appear unreasonable. Furthermore, it is likely that the configuration
of the condensing surfaces and the unequal distribution of coolant will have
contributed to an appreciable extent. By a change in design, employing a
single coolant line and a more open configuration of the freeze-out surface
in the pump chamber, it should be possible to lower these pressure levels.

The defrosting time for the pump, after a 10 hour run, using
the by-pass line on the Joule-Thomson cycle and the heater elements attached
to the copper vanes, was about 2 - 3 hours. This, of course is dependent
upon the amount of gas condensed on the vanes.

A summary of the performance of the pump is given in Table

VII.
IX. CONCLUSIONS

A neon cycle cryogenic pump capable of pumping 0.80 gr/sec $\text{N}_2$ in the $1 - 4 \times 10^{-3}$ torr region was built. Although there existed evidence from preliminary experimental studies that the available pumping time would be of the order of minutes due to the low values of thermal conductivity of the condensate, the pumping time available was in fact 10 hours or more. During the course of testing, it was noticed that the refrigerant gas did not distribute itself evenly to the 6 parts of the freeze-out surface. This caused some of the parts of the freeze-out surface to have too little neon refrigerant flowing through them causing their temperature to rise with a resulting drop in their pumping speed. The fact that certain parts of the freeze-out surface pump at a lower rate than others, causes adverse test-gas flow conditions within the pump chamber, resulting in higher than anticipated test section pressures. The uneven distribution of neon to the 6 freeze-out banks could be remedied by a better design for the freeze-out surface (e.g. one using a single feed pipe instead of the 6 as used.) No sudden pressure rises during an experimental run were noticed. If these had occurred, they could have been attributed to the falling off of the condensate from the freeze-out vanes, as reported in some work (Ref. 25).

The pump was used as a calorimeter in order to obtain quantitative data on the efficiency of neon in a Joule-Thomson refrigeration cycle. The experimental data on the specific refrigeration available from neon are in excellent agreement with theory.

It was shown that a cryogenic pump built around a neon Joule-Thomson refrigeration cycle can commercially be constructed more cheaply than a comparable helium cryogenic pump. Now that the techniques have been evolved, the problem of containing the neon in a leak-tight system should not present undue problems to the builder.
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<td>Muntz, E.P.</td>
<td>A Review of Cryopumped Low Density Wind Tunnels, Royal Armament Research and Development Establishment, RARDE Memorandum (B) 56/63.</td>
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<tr>
<td></td>
<td>Klipping, G.</td>
<td>Investigation of Time Response and Outgassing Effects of Pressure Probes in Free Molecular Flow, University of Toronto, Institute for Aerospace Studies, UTIA TN No. 6, 1955.</td>
</tr>
<tr>
<td>14.</td>
<td>Sears, F.W.</td>
<td></td>
</tr>
<tr>
<td>15.</td>
<td>Lapin, A.</td>
<td></td>
</tr>
<tr>
<td>16.</td>
<td>Harris, E.L.</td>
<td></td>
</tr>
</tbody>
</table>
17. Kay and Laby

18. Moore, B.C.


19. Black, I.A.


20. Black, I.A.


21. Christiansen, R.M.


22. Christiansen, R.M.


23. Dushman, S.


24. Chuan, R.L.

A Study of the Condensation of Nitrogen Below the Triple Point, University of Southern California, Engineering Center, USCEC TN No. 56-201.

25. Mullen, L.O.


26. Lafrance, J.C.


27. Jakob, M.


28. McAdams, W.H.


<table>
<thead>
<tr>
<th>FLUID</th>
<th>H₂</th>
<th>He</th>
<th>Ne</th>
</tr>
</thead>
<tbody>
<tr>
<td>BOILING POINT (°K)</td>
<td>20.28</td>
<td>4.26</td>
<td>37.16</td>
</tr>
<tr>
<td>HEAT OF VAPORIZATION (BTU/lb)</td>
<td>193.0</td>
<td>8.8</td>
<td>37.1</td>
</tr>
<tr>
<td>HEAT OF VAPORIZATION (BTU/lb)</td>
<td>29.78</td>
<td>2.425</td>
<td>97.2</td>
</tr>
<tr>
<td>VOLUME NEEDED FOR 1490 BTU's (LITERS)</td>
<td>50</td>
<td>615</td>
<td>15</td>
</tr>
<tr>
<td>JOULE-THOMSON EXPANSION, FIGURE OF MERIT SCFM/Watt</td>
<td>0.107</td>
<td>1.660</td>
<td>0.052</td>
</tr>
</tbody>
</table>
## CONDENSATION LOAD (N₂)

### TABLE II

<table>
<thead>
<tr>
<th></th>
<th>Temperature $°K$</th>
<th>$C_p$ BTU/# $°K$</th>
<th>Transformation Ht. BTU/#</th>
<th>TOTAL BTU/#</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensible heat</td>
<td>80° - 63</td>
<td>0.433</td>
<td></td>
<td>7.3</td>
</tr>
<tr>
<td>Heat of vaporization</td>
<td>63</td>
<td></td>
<td>93</td>
<td>93.0</td>
</tr>
<tr>
<td>Heat of fusion</td>
<td>63</td>
<td></td>
<td>10.9</td>
<td>10.9</td>
</tr>
<tr>
<td>Sensible heat</td>
<td>63 - 35.6</td>
<td>0.432</td>
<td></td>
<td>11.8</td>
</tr>
<tr>
<td>Heat of transformation</td>
<td>35.6</td>
<td></td>
<td>3.52</td>
<td>3.5</td>
</tr>
<tr>
<td>Sensible heat</td>
<td>35.6 - 28</td>
<td>0.434</td>
<td></td>
<td>3.3</td>
</tr>
<tr>
<td><strong>TOTAL</strong></td>
<td></td>
<td></td>
<td></td>
<td>129.8</td>
</tr>
</tbody>
</table>

**HEAT LOAD FOR FREEZING OUT OF:**

- 5lbs./hr. $N_2$: 650 B.T.U./hr. 190 watts
- 6lbs./hr. $N_2$: 780 B.T.U./hr. 228 watts
**TABLE III**

<table>
<thead>
<tr>
<th>POINTS</th>
<th>TEMPERATURE</th>
<th>PRESSURE</th>
<th>ENTHALPY</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>300</td>
<td>2100</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>100</td>
<td>2000</td>
<td>20.2</td>
</tr>
<tr>
<td>3</td>
<td>75</td>
<td>2000</td>
<td>19.2</td>
</tr>
<tr>
<td>4</td>
<td>66</td>
<td>2000</td>
<td>19.0</td>
</tr>
<tr>
<td>5</td>
<td></td>
<td>2000</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>28</td>
<td>19</td>
<td>18.85</td>
</tr>
<tr>
<td>7</td>
<td>28</td>
<td>19</td>
<td>18.85</td>
</tr>
<tr>
<td>8</td>
<td>65</td>
<td>17</td>
<td>19.7</td>
</tr>
<tr>
<td>9</td>
<td>110</td>
<td>17</td>
<td>20.7</td>
</tr>
<tr>
<td>10</td>
<td>300</td>
<td>17</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>66</td>
<td></td>
<td>2.2</td>
</tr>
<tr>
<td>12</td>
<td>300</td>
<td></td>
<td>2.2</td>
</tr>
</tbody>
</table>
### TABLE IV

**HEAT LEAK TO FREEZE-OUT SURFACE**

<table>
<thead>
<tr>
<th></th>
<th>A WATTS</th>
<th>B WATTS</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>RADIATION FROM 72°K SURFACES</strong></td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td><strong>RADIATION FROM 300°K SURFACES THROUGH THE INTERCOOLER VANES</strong></td>
<td>4.8</td>
<td>32.1</td>
</tr>
<tr>
<td><strong>GAS CONDUCTION</strong></td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td><strong>SUPPORTS</strong></td>
<td>0.4</td>
<td>0.4</td>
</tr>
<tr>
<td><strong>TOTAL</strong></td>
<td>14.2</td>
<td>41.5</td>
</tr>
</tbody>
</table>

*A: HEAT LEAK WITH THE 4.5 ft. DIAMETER VALVE CLOSED.*

*B: HEAT LEAK WITH THE 4.5 ft. DIAMETER VALVE OPEN.*
HEAT LOAD IMPOSED UPON LN$_2$ SYSTEM

**TABLE V**

<table>
<thead>
<tr>
<th>Description</th>
<th>Maximum watts</th>
<th>Minimum watts</th>
</tr>
</thead>
<tbody>
<tr>
<td>RADIATION ON LN$_2$ COOLED SURFACES</td>
<td>251</td>
<td>54</td>
</tr>
<tr>
<td>COOLING OF 5 lbs/hr N$_2$ TO 80°K</td>
<td>141</td>
<td>141</td>
</tr>
<tr>
<td>COOLING OF NEON TO 65°K</td>
<td>404</td>
<td>200</td>
</tr>
<tr>
<td>SUPPORTS HEAT LEAK</td>
<td>135</td>
<td>135</td>
</tr>
<tr>
<td>TOTAL</td>
<td>931</td>
<td>530</td>
</tr>
</tbody>
</table>

TOTAL LN$_2$ REQUIRED (LITERS/hr.): 21.0 11.5
**LN₂ COOLDOWN LOAD**

**TABLE VI**

<table>
<thead>
<tr>
<th>UNIT</th>
<th>MTL.</th>
<th>WEIGHT (lbs.)</th>
<th>Unit heat removed 300°K - 65°K</th>
<th>Total Heat Removed B.T.U.</th>
<th>LN₂ required (l)</th>
</tr>
</thead>
<tbody>
<tr>
<td>CYLINDER</td>
<td>Cu</td>
<td>660</td>
<td>32 BTU/lbs</td>
<td>21,100</td>
<td>118</td>
</tr>
<tr>
<td>PIPE</td>
<td>Cu</td>
<td>83</td>
<td>32</td>
<td>2,660</td>
<td>21</td>
</tr>
<tr>
<td>HEAD</td>
<td>Cu</td>
<td>170</td>
<td>32</td>
<td>5,450</td>
<td>31</td>
</tr>
<tr>
<td>LN₂ BAFFLE</td>
<td>Cu</td>
<td>298</td>
<td>32</td>
<td>9,550</td>
<td>54</td>
</tr>
<tr>
<td>LN₂ BATH</td>
<td>S.S.</td>
<td>322</td>
<td>33.5</td>
<td>10,300</td>
<td>58</td>
</tr>
<tr>
<td>F.O. BANKS</td>
<td>Cu</td>
<td>230</td>
<td>32</td>
<td>7,730</td>
<td>29</td>
</tr>
<tr>
<td>TOTAL</td>
<td></td>
<td></td>
<td></td>
<td>311</td>
<td></td>
</tr>
<tr>
<td>FUNCTION</td>
<td>PREDICTED</td>
<td>MEASURED</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-----------------------------------------------</td>
<td>----------------</td>
<td>----------------</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>PUMPING TIME TO 10 μHg. @ 5 lbs./hr. N₂</td>
<td>45 min.</td>
<td>&gt;10 hrs.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>PUMPING SPEED</td>
<td>5 lbs./hr. N₂</td>
<td>6.1 lbs./hr N₂</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>LN₂ CONSUMPTION</td>
<td>20 L./hr.</td>
<td>≈ 30 L/hr</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>NEON COOLING CAPACITY</td>
<td>0.938 BTU/cf.</td>
<td>0.84 BTU/cf.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MAXIMUM REFRIGERATION AVAILABLE</td>
<td>350 watts</td>
<td>215 watts</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>WORSENING OF PERFORMANCE FOR 80°K JOULE-THOMSON EXPANSION</td>
<td>35%</td>
<td>35%</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>CONDENSATION LOAD (N₂)</td>
<td>38 W./lb./hr.</td>
<td>36.18 W./lb./hr.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>RADIATION LOAD ON THE FREEZE-OUT SURFACE</td>
<td>42 ± 4 watts</td>
<td>41.5 watts</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>COOL DOWN TIME</td>
<td>5 hours</td>
<td>5 hours</td>
<td></td>
<td></td>
<td></td>
</tr>
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</table>
TABLE VIII
CONVERSION FACTORS

<table>
<thead>
<tr>
<th>Conversion</th>
<th>Value</th>
</tr>
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<tbody>
<tr>
<td>1 lb. N₂/hr</td>
<td>0.126 gr/sec</td>
</tr>
<tr>
<td></td>
<td>0.097 atm. L/sec</td>
</tr>
<tr>
<td></td>
<td>74,225 microns L/sec</td>
</tr>
<tr>
<td>1 BTU/cf./min</td>
<td>17.58 watts/c.f.</td>
</tr>
<tr>
<td>10⁻³ torr</td>
<td>10⁻³ mm Hg.</td>
</tr>
<tr>
<td></td>
<td>1 micron Hg</td>
</tr>
<tr>
<td></td>
<td>1 μ Hg</td>
</tr>
</tbody>
</table>
FIGURE 1
PUMPING REQUIREMENTS

D: diameter of pump opening (with baffle) needed to pass 0.63 gr./sec. N₂ at P

D = 404 cm.

D = 142 cm.

D = 50 cm.

LARGEST OIL DIFFUSION PUMP

LARGEST OIL BOOSTER PUMP

(UNBAFFLED)

Liter/sec.

D: diameter of pump opening (with baffle) needed to pass 0.63 gr./sec. N₂ at P

P (Torr.)
FIGURE 2

ISENTHALPIC STATES OF A GAS
FIGURE 3

ISENTHALP AND INVERSION CURVES

ISENTHALP CURVE

COOLING

HEATING

CRITICAL POINT

INVERSION CURVE

VAPOR PRESSURE CURVE
FIGURE - 4 -

JOULE-KELVIN EXPANSION CYCLE

HIGH PRESSURE

COOLER

LOW PRESSURE

COMPRESSOR

HEAT EXCHANGER

THROTTLE VALVE

LIQUEFIED GAS
Figure 5. Schematic of Refrigeration System
FIGURE 6

VAPOUR PRESSURE OF NITROGEN, ARGON AND AIR

T°K

NITROGEN

ARGON

AIR
EQUILIBRIUM VAPOR PRESSURE OF LIQUID NITROGEN

T.P. 63.18 K

FIGURE 7
FIGURE 8 - THERMODYNAMIC PROPERTIES OF NEON
FIGURE 9· THERMODYNAMIC PROPERTIES OF NEON
**FIGURE 10** ISENTHALPIC THROTTLING OF NEON TO 1 ATM.
FIGURE 11
REFRIGERATION AVAILABLE FROM NEON

THROTTLING TO 1 ATM. FROM

BTU/cf.

EXPANSION TEMPERATURE
FIGURE -12-

LOCUS OF MAXIMUM SPECIFIC REFRIGERATION

$P_{atm.}$ vs. $^\circ K$

EXPANSION TEMPERATURE
FIGURE 13

CRYOPUMP PERFORMANCE CURVES - TIME TO REACH VARIOUS PRESSURE LEVELS VERSUS INLET MASS FLOW.
FIGURE - 14 -

EXPANSION TEMPERATURE

66°K

72°K

77°K

INLET PRESSURE PSIA

NEON LIQUID PRODUCTION
FIGURE -15-

POWER REQUIREMENT FOR LIQUID NEON PRODUCTION

EXPANSION TEMPERATURE

INLET PRESSURE PSIA
ISENTHALPIC THROTTLING OF NEON FROM 135 TO 1 ATM.

FIGURE 16
FIGURE 17 - THERMODYNAMIC PROPERTIES OF NEON
FIGURE 18 PRESSURE CONTROL PLUMBING FOR THE NEON J.T. CYCLE
FIGURE -19-

CLOSED CYCLE NEON REFRIGERATION SYSTEM

LINE #1

NEON SUPPLY

COMPRESSOR

LINE #2

LINE #3

JOULE-THOMSON EXPANSION VALVE
FIGURE 20

TUNNEL WALL

LOW PRESSURE GAS

HIGH PRESSURE GAS

"O" RING SEALS

JOULE-KELVIN EXPANSION VALVE
FIGURE 21 SCHEMATIC OF FREEZE-OUT BANK
FIGURE 22 FREEZE-OUT SURFACE
COPPER SHIELD ARRANGEMENT

INTERCOOLER VANES

TUNNEL WALL

COPPER SHIELD

LN$_2$ CONTAINER

DEWAR SPACE (FILLED WITH PERLITE)

FREEZE-OUT SURFACE
FIGURE 25 LN$_2$ COOLED COPPER SHIELD
FIGURE 27 INTERCOOLER IN PLACE
FLOW THROUGH 4.5 FOOT DIAMETER VALVE

5 Lb/HR. AIR @ $P_{\mu}$

SONIC FLOW THROUGH THE TWO 15" D. VALVES

TRANSITION REGIME FLOW THROUGH THE 15" D. VALVES

FIGURE-28 FLOW THROUGH VARIOUS VALVES WITH $P_{cp} = 1/\mu$ OR LOWER
FIGURE 29 - SCHEMATIC OF J.T. CYCLE

300°K

N₂ VENT

HEATER

12

1

10

COMRESSOR

LN₂ BATH

2

3

2

5

EXPANSION VALVES

FREEZE-OUT SURFACE

HX 1

HX 2

HX 3

HX 4
LOW PRESSURE STORAGE BOTTLES

0.5 psi

NEON SUPPLY

COMPRESSOR

2050 psi

SV; SAFETY VALVE
PRV; PRESSURE REDUCING VALVE
BPR; BACK PRESSURE REGULATOR

FIGURE-30-Schematic of Pressure Control Plumber
FIGURE 31 PRESSURE CONTROL PLUMBING
THIS LINE PERMITS THE PURGING OF THE VALVE WITHOUT RUPTURING THE DIAPHRAGM

"O" RING SEAL

DIAPHRAGM CLAMPING PLATE WITH "O" RING SEALS

RUBBER DIAPHRAGM

"O" RING

"O" RING SEALS AGAINST THE BACKSIDE OF THE VALVE STEM IF THE DIAPHRAGM RUPTURES

FIGURE-33-MODIFICATION TO PRESSURE REDUCING VALVE
FIGURE 34 - "O" RING SEAL ON VALVE

DOUBLE "O" RING SEALS

QUARTER TURN BALL VALVE

DOUBLE FLANGE ARRANGEMENT

EIGHT BOLTS ON CIRCUMFERENCE

WELDED PIPE OR TUBING

PIPE THREADS

BOTH FACES OF VALVE ARE MACHINED FLAT
FIGURE 35 "O" RING SEAL ON VALVE
FIGURE 36 LAYOUT OF CRYOPUMP

- UNINSTALLED TEST SECTION
- CRYOPUMP DEWAR VESSEL
- CONTROL PANEL
- PRESSURE CONTROL PLUMBING
- NEON STORAGE BOTTLES
- FLOOR LEVEL

COMPRESSOR

21'
FIGURE 37

PLASMA TUNNEL

TEST SECTION

PLASMA NOZZLE

PRESSURE CONTROL VALVES

FREEZE-OUT SURFACE
FIGURE 38 - HEATER CARTRIDGE AND HOLDER
FIGURE 40 - FLOW METER AND THERMOMETER IN POSITION
CRYOPUMP SHIELD

LN2
TANK

PRESSURE REGULATING VALVE

HEATER

VACUUM PUMP

FIGURE 41 - LN2 HANDLING SYSTEM
FIGURE 42
SCHEMATIC OF INSTRUMENTATION TO MEASURE NEON COOLING CAPACITY
VANE # 1 HEATERS # 1-2
VANE # 4 HEATERS # 7-8

V O L T M E T E R
A M M E T E R
P O W E R S T A T
P R E S S U R E G A U G E
T E R M O M E T E R
T H E R M O C O U P L E S t a .
B. P. R.: Back Pressure Regulator
P. R. V.: Pressure Reducing Valve
F. O. B.: Freeze Out Bank

115-120 volts
FIGURE 43

FLOW METER

STABILITY OF MECHANICAL vs. ELECTRICAL READOUT.
FIGURE 44

JOULE-KELVIN EXPANSION TEMPERATURE

RUN # 5
RUN # 6
RUN # 7
RUN # 8
FIGURE 45

COOL DOWN TIME

Switched from J.T. Valve #2 to J.T. Valve #1. This brings HX. #3 into the cycle (fig.)
COOLING OF THE FREEZE-OUT SURFACE AFTER THE NEON-CARRYING PIPES HAVE REACHED 28 K.

RUN #8
FIGURE 47

VARIATION IN THE POWER ACCEPTED BY EACH VANE

WATTS

100
90
80
70
60
50
40
30
20
10

1 2 3 4 5 6

VANE #
FIGURE 48

STABILITY OF HEATER RESISTANCE DURING A 20 HOUR PERIOD.
FIGURE - 49 -

HEATER INPUT vs. NEON CONSUMPTION

<table>
<thead>
<tr>
<th>NEON SCFM</th>
<th>J.T. ≈ 67°K</th>
<th>J.T. ≈ 80°K</th>
</tr>
</thead>
<tbody>
<tr>
<td>• RUN # 5</td>
<td>+ RUN # 6</td>
<td>+ RUN # 7</td>
</tr>
<tr>
<td>○ RUN # 6</td>
<td></td>
<td>+ RUN # 7</td>
</tr>
<tr>
<td>△ RUN # 7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>■ RUN # 8</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

0.103 SCF/watt
0.548 B.T.U./cf.

0.068 SCF/watt
0.84 B.T.U./cf.
FIGURE 50

THEORETICAL CURVE

EXPERIMENTAL POINTS
(fig. 49)

EXPERIMENTAL EXPANSION TEMPERATURE

REFRIGERATION CAPACITY OF NEON
FIGURE 51

NEON CONSUMPTION vs. GAS LOAD

NEON SCFM

J. T. ≈ 67°K

RUN # 7

J. T. ≈ 80°K

RUN # 7

RUN # 8

NEON CONSUMPTION vs. GAS LOAD

ATM. L./sec. \( N_2 \)

gr./sec. \( N_2 \)
FIGURE - 52 -

REFRIGERATION CAPACITY NEEDED TO FREEZE OUT $N_2$

SLOPE: 358 watts/ATM. L./sec. $N_2$

WATTS

0  20  40  60  80  100  120  140  160  180  200  220

0.1 0.2 0.3 0.4 0.5

ATM. L./sec. $N_2$

$g/s$. sec. $N_2$
RUN # 8  $M = 0.5252 \text{ ATM L.} / \text{sec.} \text{ N}_2$ (0.68 gr./sec.)

RUN # 9  $M = 0.2625 \text{ ATM L.} / \text{sec.} \text{ N}_2$ (0.34 gr./sec.)

RUN # 9  $M = 0.3939 \text{ ATM L.} / \text{sec.} \text{ N}_2$ (0.51 gr./sec.)

$P_m$: PRESSURE ABOVE THE INTERCOOLER ($\text{Hg.}$) (i.e. IN THE TEST SECTION)
FIGURE - 54 -

OPERATING PRESSURE OF THE PUMP vs. MASS FLOW

$P_{\mu}$: PRESSURE ABOVE THE INTERCOOLER ($\mu$Hg.)

RUN # 8

RUN # 9

PRESSURE DROP DUE TO INTERCOOLER

ATM. L./sec. $N_2$

gr/sec. $N_2$
APPENDIX I

Pressure Drop Across the LN₂ Cooled Copper Baffle

The gas enters the pump chamber through the slots between the LN₂ cooled copper vanes of the intercooler of Figs. 26, and 27. Since the intercooler is circular the width of these slots varies from the center out (Fig. 26). For the purpose of calculations we consider the intercooler to be composed of 13 slots of equal widths (115 cm)

\[ b \approx 115 \text{ cm} \]
\[ a = 7.6 \text{ cm} \]
\[ \text{Depth} \approx 20 \text{ cm} \]

**Entrance conditions**
\[ T_2 = 300^0 \text{K} \]
\[ P_2 > 1 \mu \]

**Exit Conditions**
\[ T_1 = 80^0 \text{K} \]
\[ P_1 = 1 \mu \] (Ref. 2)

We must now find in what flow regime the gas flows through these slots of the intercooler in order to be able to predict the pressure drop necessary to drive the gas through the slots. Following the analysis of Ref. 23, the criteria for the 3 different flow regimes are

- **Viscous** \[ aP_\mu > 500 \]
- **Molecular** \[ aP_\mu < 5 \]
- **Transition** \[ 5 < aP_\mu < 500 \]

where \( a \): typical dimension of our duct
\[ P_\mu \]: pressure in microns Hg of the gas entering the duct.
In our case we have

\[ a \approx 7 \text{ cm} = 3 \text{ inches} \]

Mean value of "b" is 45" = 115 cm

Aspect ratios of opening if 45/3 = 15

By Ref. 26, even at low speeds an aspect ratio of 15 will not give free molecular flow below a Knudsen number of 15; i.e. mean free path of a gas must be at least 45", or a pressure well below 0.1 μ Hg inside the duct for free molecular flow to occur. Therefore we will not have free molecular flow through the duct opening. Our flow regimes are

\[ 5 < P_\mu < 500 \]

i.e.

\[ P_\mu < \frac{500}{7} = 71\mu \]

The gas therefore flows through the intercooler under two flow regimes depending on the entrance pressure (P_2)

- Transition  \( P_2 \leq 70\mu \)
- Viscous  \( P_2 > 70\mu \)

The total entrance area of the 13 slots of the intercooler is

\[ 13 \times 115 \times 7 = 10,500 \text{ cm}^2 \]

To render the problem amenable to the analysis of Ref. 23, we break this area into 7 x 7 cm square tubes. In this way we oversimplify the problem, and slightly overestimate the losses. We therefore need

\[ \frac{10,500}{49} = 210 \text{ such tubes} \]

The analysis of Ref. 23 is for circular tubes and by applying it directly to square tubes we will overestimate slightly the pressure drop across the intercooler but the analysis will nevertheless be close enough to reality to serve our purpose. Furthermore Ref. 2 states that for cryogenic pump operating with freeze-out surfaces in the 20 - 30°K region, the pressure in the pump chamber (P_1) will be slightly lower than 1 x 10^{-3} torr (1 micron Hg) under operating conditions. We therefore consider that the pressure in the pump chamber of our pump is 1 μ Hg. Under these conditions, the flow through the intercooler will become choked before viscous flow is established (at 70μ above the intercooler). Our analysis will therefore consider only two flow regimes.

1) Transition  \( P_2 < 70\mu \)
2) Choked flow
1) **Transition Flow** \((P_2 < 70\mu)\)

For slip flow (Ref. 23, p. 109)

\[
F = F_v \left[ 1 + 4 \left( \frac{2}{f} - 1 \right) \frac{L_a}{a} \right]
\]

and

\[
4 \left( \frac{2}{f} - 1 \right) = 6.793 \frac{1.2507 (a/L_a)}{1 + 3.95 (a/L_a)}
\]

i.e. \(f = F(a/L_a)\)

where \(\frac{dV}{dt}\) : volumetric flow rate across a plane

\(P\) : pressure at which it is measured

\(F = Q/P_2 - P_1\) = rate of flow per unit difference of pressure between the ends of the channel

\(\mu\) : micron Hg. (= 1.33 \text{ d/cm}^2)

\(\text{d/cm}^2\) : dyne/cm\(^2\)

\(\Delta P\) : pressure difference across the duct

\(a\) : typical dimensions of duct

\(L_a\) : mean free path of the gas

\(F_v\) : viscous conductance

For our particular configuration

\[a = 7 \text{ cm}\]

\[L_a = 5.09/P_2 \text{ cm for air at 25\degree C}\]

\[L_a = 5.09 \text{ cm}\]

\[P_2 = 1\mu\]

\[L_a = 0.1 \text{ cm}\]

\[P_2 \approx 50\mu\]

Then from Tables (Ref. 23)

\(f = 0.82\) for \(a/L_a = 1.4\), \(P_2 = 1\mu\)

\(f = 0.84\) for \(a/L_a = 70\), \(P_2 = 50\mu\)

and

\[F_v = \pi a^4/8n\ell P_a\], where

\(n\) : viscosity of the gas

\(P_a\) : \(P_2 + P_1/2\)

\(\ell\) : depth of the duct
Therefore
\[ F = \frac{w a^4}{8 n k} \left( \frac{P_2 + P_1}{2} \right) \left[ 1 + 4 \left( \frac{2}{f} - 1 \right) \frac{I_a}{a} \right] \]

Calculate \( F \) for air at 25°C, remembering that \( P_1 = 1 \mu \text{Hg} \) and from the data of Ref. 23, we have
\[ \left[ 1 + 4 \left( \frac{2}{f} - 1 \right) \frac{5}{7} \right] = 5.13 \]

with
\[ f = 0.82 \]

Therefore
\[ F = 25.1 \times 10^4 \left( \frac{P_2 + P_1}{2} \right) \left[ 5.13 \right] \]

or
\[ F = 1.29 \times 10^6 P_a \text{ c.c./sec slit} \]

The resulting flow rates as a function of pressure upstream of the slots are collected in the following table.

**TABLE I-1**

<table>
<thead>
<tr>
<th>( P_2 )</th>
<th>( P_2 )</th>
<th>( F )</th>
<th>( F )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \mu )</td>
<td>d/cm²</td>
<td>d/cm²</td>
<td>( \Delta P = 1 ) d/cm</td>
</tr>
<tr>
<td>1.25</td>
<td>1.66</td>
<td>1.45</td>
<td>1.87 ( \times 10^6 ) c.c/sec</td>
</tr>
<tr>
<td>1.50</td>
<td>1.99</td>
<td>1.66</td>
<td>2.14</td>
</tr>
<tr>
<td>1.75</td>
<td>2.33</td>
<td>1.83</td>
<td>2.36</td>
</tr>
<tr>
<td>2.00</td>
<td>2.66</td>
<td>1.99</td>
<td>2.57</td>
</tr>
<tr>
<td>2.25</td>
<td>2.99</td>
<td>2.16</td>
<td>2.79</td>
</tr>
<tr>
<td>2.50</td>
<td>3.33</td>
<td>2.33</td>
<td>3.01</td>
</tr>
<tr>
<td>2.75</td>
<td>3.66</td>
<td>2.50</td>
<td>3.23</td>
</tr>
<tr>
<td>3.00</td>
<td>3.99</td>
<td>2.66</td>
<td>3.41</td>
</tr>
</tbody>
</table>

2) - Flow Limit With Sonic Conditions Through the 4.5 Foot Diameter Opening

Area of opening is 15.9 ft²

\[ S = \sqrt{\gamma RT} \]

\[ S_{\text{air}} = 1128 \text{ ft/sec} \]

\[ V_{\text{air}} = 17,900 \text{ ft}^3/\text{sec} \]

\[ S_{\text{argon}} = 1061 \text{ ft/sec} \]

\[ V_{\text{argon}} = 16,900 \text{ ft}^3/\text{sec} \]

\[ S_{\text{nitrogen}} = 1177 \text{ ft/sec} \]

\[ V_{\text{nitrogen}} = 18,800 \text{ ft}^3/\text{sec} \]

These results are plotted on Fig. 28.
APPENDIX II

Efficiency of the LN2 Cooled Copper Baffle as a Heat Transfer Device

Forced heat convection in laminar flow through a tube or duct at uniform surface temperature is described in Ref. 27 and 28. Figure 22.7 of Ref. 27 gives \( h_a \frac{D_e}{k} \) as a function of \( \phi \), where

\[
\phi = \frac{C_p}{k} \rho u \frac{D_e^2}{L}
\]

\( k \): thermal conductivity of the gas
\( C_p \): heat capacity of the gas
\( L \): length of duct
\( h_a \): heat transfer/unit area/°K based on \( \frac{\Delta T_{in} + \Delta T_{out}}{2} \)
\( D_e \): effective diameter for a duct

\[
D_e = \frac{4a.b}{2a + 2b} \approx 2b \quad \text{for} \quad a \gg b
\]

\( \Delta T_{in} \): Temperature of the gas entering the duct minus the temperature of the walls of the duct.

\( \Delta T_{out} \): Temperature of the gas leaving the duct minus the temperature of the walls of the duct.

For a flat duct with \( \phi \leq 6 \), following the analysis of Ref. 27, we may simplify the above relationship to

\[
h_a = \frac{1}{2} C_p (\rho u) \frac{D_e}{L}
\]

\[
= \frac{C_p}{L} (\rho u) \frac{b}{L}
\]

the heat balance equation is

\[
H = \frac{\Delta T_{in} + \Delta T_{out}}{2} = M C_p (\Delta T_{in} - \Delta T_{out})
\]

\( H \) = heat transfer/unit vane length = \( 2L h_a \)

\( M \) = mass flow/unit vane length = \( (\rho u)b \)

Therefore

\[
\Delta T_{in} + \Delta T_{out} = \frac{2M}{H} C_p (\Delta T_{in} - \Delta T_{out})
\]

\[
= \frac{\rho u}{L h_a} C_p (\Delta T_{in} - \Delta T_{out})
\]
Substitute for $h_a$ and get

$$\Delta T_{\text{in}} + \Delta T_{\text{out}} = \Delta T_{\text{in}} - \Delta T_{\text{out}}$$
or

$$\Delta T_{\text{out}} = 0 \text{ if } \varphi \leq 6$$

Thus, if $\varphi$ is kept below 6, heat transfer to the gas is complete. Now,

$$\varphi = \frac{c_p \rho u D_e^2}{k L} = \frac{c_p \rho u 4b^2}{k L}$$

The intercooler essentially consists of a number of parallel vanes.

Pitch 3"  Height 6"

Developed length 8" = 20.3 cm (=L)

Distance normal to vane $2\frac{1}{2}'' = 6.35$ cm (=b)

Mass flow 5 lbs/hr = .631 gr/sec. $N_2$ (planned maximum capacity of pump)

Flow area $\pi \times 26''^2 = \pi \times 66^2 = 13,700$ cm$^2$

Mass flow per unit area

$$\rho u = \frac{.631}{1.37 \times 10^4} = 4.6 \times 10^{-5} \text{ gr/cm}^2\text{sec}$$

From Ref. 29, for nitrogen we have

$$c_p = \frac{1.04 \text{ watts/sec}}{\text{gr}^0K} \text{ at } 80^0K, 0.01 \text{ atm}$$

$$k = 7.61 \times 10^{-5} \text{ watts/cm}^2\text{K} \text{ at } 80^0K, 1 \text{ atm}$$
Substitute these values of $\rho u, C_p, k, b$ and $L$ in the expression for $\varphi$; we get

$$\varphi = \frac{1.04 \times 4.6 \times 10^{-5} \times 4 \times (6.35)^2}{7.61 \times 10^{-5} \times 20.3} \approx 5$$

which indicates that the heat transfer from gas to vanes is practically complete. We next calculate the values of $\Delta T_{out}$ under various flow conditions.

The vanes are cooled by liquid nitrogen which is kept at $65^0K$. Since the heat transfer path through the vanes is fairly long, a certain temperature drop will have to be allowed. Take $10^0K$ for this drop and take as the uniform vane temperature $75^0K$.

The working gas enters at about room temperature conditions. To be conservative, use an inlet temperature of $400^0K$.

The gas enters the intercooler with a temperature of $400^0K$. A temperature boundary layer will diffuse outward from the vane walls into the fluid and because of the low Reynolds numbers, these boundary layers grow quickly and meet. Further length diminishes the temperature in the center between the vanes. This process is amenable to theoretical analysis after a velocity distribution has been assumed. For nitrogen, the values of $C$ and $k$ quoted above are used. The value of $k$ at $80^0K$ was used to be conservative, but also because the air immediately adjoining the wall will have this temperature.

Assuming uniform flow entering the entire intercooler, we can use the value of $\rho u$ derived above and find a corresponding value of $\varphi$;

$$\varphi = \frac{D_e \frac{u C_p D_e}{k L}}{2 x 6.35 \times 4.6 \times 10^{-5} \times 1.04 \times \frac{2 x 6.35}{7.61 \times 10^{-5}} \times \frac{20.3}{20.3}} = 7.98 \times 0.626 = 5.00$$

Corresponding to this value of $\varphi$, a value of $h_a \frac{D_e}{k}$ is found from Fig. 22.7, Ref. 27. We get

$$h_a \frac{D_e}{k} = 2.43$$

Hence

$$h_a = \frac{2.43 \times 7.61 \times 10^{-5}}{1.27 \times 10' \times 1.46 \times 10^{-5} \text{ watt cm}^{20K}} = 1.27 \times 10^{-5} \text{ watt cm}^{20K}$$

Mass flow per unit length of one vane spacing =

$$4.6 \times 10^{-5} \times 6.35 \times 1 = 2.92 \times 10^{-4} \text{ gr/sec.}$$

Heat transfer area per unit length = $1 \times 20.3 \times 2 = 40.6 \text{ cm}^2$
Total heat transfer per °K
\[ H = 1.46 \times 10^{-5} \times 4.06 \times 10' \]
\[ = 5.93 \times 10^{-4} \text{ watt/°K} \]

Now the following heat balance must apply
\[ H \left[ \frac{\Delta T_{\text{in}} + \Delta T_{\text{out}}}{2} \right] = \text{mass flow} \times C_p \times \left( \Delta T_{\text{in}} - \Delta T_{\text{out}} \right) \]
\[ 2.96 \times 10^{-4} \left( \Delta T_{\text{in}} + \Delta T_{\text{out}} \right) = 2.92 \times 10^{-4} \times 1.04 \left( \Delta T_{\text{in}} - \Delta T_{\text{out}} \right) \]
\[ 2.96 \left( \Delta T_{\text{in}} + \Delta T_{\text{out}} \right) = 3.04 \left( \Delta T_{\text{in}} - \Delta T_{\text{out}} \right) \]
\[ \Delta T_{\text{out}} + \Delta T_{\text{in}} \left[ \frac{3.04 - 2.96}{3.04 + 2.96} \right] = 0.013 \Delta T_{\text{in}} \]

Incoming temperature difference is
\[ \Delta T_{\text{in}} = (400 - 75) \text{°K} = 325 \text{°K} \]

Hence
\[ \Delta T_{\text{out}} = 4.225 \text{ °K}, \text{ for uniform flow across the intercooler.} \]

Assume the same geometry of vanes but consider a fully developed pipe flow towards the intercooler. This means that if the effect of the intercooler to make the distribution more uniform is neglected, then the maximum mass flow per unit area is twice the value in the first calculated case.

Hence
\[ \frac{D_e \rho u}{k} \frac{C_p}{L} \frac{D_e}{L} = 10 \]

and
\[ \frac{h_a D_e}{k} = 4.56 \text{ per unit length} \]

Total heat transfer per °K
\[ H = \frac{4.56 \times 7.61 \times 10^{-5}}{1.27 \times 10^{11}} x 4.06 \times 10' \]
\[ = 1.109 \times 10^{-3} \text{ watt/°K} \]

Total mass flow:
\[ 2 \times 2.92 \times 10^{-4} = 5.84 \times 10^{-4} \text{ gr/sec.} \]

Our equation becomes
\[ 5.55 \times 10^{-4} (\Delta T_{\text{in}} + \Delta T_{\text{out}}) = 5.84 \times 10^{-4} (\Delta T_{\text{in}} - \Delta T_{\text{out}}) \]

\[ \Delta T_{\text{out}} = \Delta T_{\text{in}} \left[ \frac{5.84 - 5.55}{5.84 + 5.55} \right] = 0.025 \Delta T_{\text{in}} \]

Hence

\[ \Delta T_{\text{out}} = 8.125^\circ K \]

Therefore, under the worst flow condition, the exit temperature of the gas is approximately \( 83^\circ K \) \((75^\circ + 8^\circ)\).
APPENDIX III

Radiative and Conductive Heat Leaks to the Freeze-Out Surface

As shown on Figs. 23 and 24 the 280 K freeze-out surface is surrounded by a LN$_2$ cooled copper shield whose vertical walls are covered with aluminium foil. The freeze-out surface will therefore suffer heat leaks from

1) Radiation from the foil covered walls of the LN$_2$ shield

2) Radiation from the top and bottom of the LN$_2$ cooled copper shield

3) Radiation originating on the 300 K steel tunnel walls and going through the intercooler slots.

4) Conduction heat leaks from the various structural supports and from the gas within the pump chamber.

All these losses must be estimated as they will affect the required size of the neon refrigeration system. For estimating the radiation losses we consider that the freeze-out surface and LN$_2$ cooled radiation shield are concentric cylinders sharing the same vertical axis. The radiative heat transfer between cylindrical bodies sharing the same axis is given by

$$Q_R = \frac{A_1 \sigma (T_2^{\frac{1}{4}} - T_1^{\frac{1}{4}})}{\frac{1}{\epsilon_1} + \frac{A_1}{A_2} (\frac{1}{\epsilon_2} - 1)}$$

where $Q_R$ is in watts,

$A = \text{area in square feet}$

$T = \text{temperature in degrees Kelvin}$

$\sigma = 0.533 \times 10^{-8} \text{ watts/ft}^2 \text{K}^4$

$\epsilon = \text{emissivity of the surfaces in question}$

We must now estimate the various losses separately.

1) Radiation From Foil Covered Walls of LN$_2$ Shield

$\epsilon_1 = 0.9$ (both the condensate and the dull copper of the freeze-out surface absorb heat well (Refs. 18, 19 and 2)

$\epsilon_2 = 0.03$ (aluminum foil, Ref. 20)

Here we consider the projected area of the freeze-out surface, i.e.

$$A_D = \pi DL + \frac{2\pi D^2}{4}$$

$$= 105 \text{ ft}^2 \text{ (see Fig. 21 for the dimensions of the freeze-out surface)}$$
\[ T_1 = 28^\circ K \] (neon boiling point temperature)
\[ T_2 = 72^\circ K \] (slightly depressed LN\(_2\) boiling point)

Hence
\[ Q_R = 0.342 \text{ watts} \]
= radiative load from the walls of the tank

2) Radiation From Bottom and Top of the Tank

\[ \varepsilon_1 \approx 0.9 \] (condensate and dull copper)
\[ \varepsilon_2 = 0.6 \] (oxidized copper, Ref. 18)
\[ T_1 = 28^\circ K \quad T_2 = 72^\circ K \]

Then
\[ Q_R = 4.5 \text{ watts} \]

Hence the total radiation load from the LN\(_2\) cooled walls is approximately 5 watts.

3) Radiation Going Through the Intercooler Slots

A.1) A rigorous calculation is done assuming that the 4.5 feet Dia. valve is not placed over the intercooler (Fig. 37, Sec. 4.7).

We must estimate how much of the 300\( ^\circ \)K surface is "seen" by the 28\( ^\circ \)K cavity.

From the diagram (Fig. A) we see that all surfaces from A' to B' will radiate at 45\(^\circ\) slant down to a portion of the 28\( ^\circ \)K cavity. Between C' and A' and B' and D' we have a slightly different angle.

We must define what portion of the intercooler vanes is illuminated by each portion of the 300\( ^\circ \)K areas (Fig. B) and what is the azimuth and elevation angles at which the 300\( ^\circ \)K surfaces radiate to these intercooler sections.

In Tables III-1 and III-2 we have grouped the vanes under 6 areas illuminated by various portions (I to VI, Fig. A) of the 300\( ^\circ \)K wall (Fig. A,B). The vanes of the intercooler are spaced in such a way that 300\( ^\circ \)K radiation from each of these six areas of the wall is free to pass through the intercooler slots and illuminate the freeze-out surface, either directly or by reflection off the aluminum foil lined LN\(_2\) copper shield. By summing over these 6 areas we will be able to find the total amount of radiation falling on the intercooler slots (and hence on the freeze-out surface) from the 300\( ^\circ \)K steel wall of the tunnel.
FIGURE A

TUNNEL WALL (300*K)

GEOMETRIC MODEL OF TUNNEL WITH INTERCOOLER IN POSITION

INTERCOOLER VANES (77*K)

LINES OF SIGHTS
## TABLE III-1

<table>
<thead>
<tr>
<th>Area</th>
<th>Vane</th>
<th>R</th>
<th>W</th>
</tr>
</thead>
<tbody>
<tr>
<td>IV</td>
<td></td>
<td>35&quot;</td>
<td>39&quot;</td>
</tr>
<tr>
<td>III</td>
<td></td>
<td>43</td>
<td>49</td>
</tr>
<tr>
<td>II</td>
<td></td>
<td>61</td>
<td>47</td>
</tr>
<tr>
<td>I</td>
<td></td>
<td>70</td>
<td>45</td>
</tr>
</tbody>
</table>

Area V and VI are special cases radiating to only a small portion of the intercooler. Table III-2 lists the azimuth and elevation angles for A300.

### TABLE III-2

<table>
<thead>
<tr>
<th>Area</th>
<th>Φ (azimuth)</th>
<th>θ' (elevation)</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>34°26'</td>
<td>16°10'</td>
</tr>
<tr>
<td>II</td>
<td>41°38'</td>
<td>14°2'</td>
</tr>
<tr>
<td>III</td>
<td>60°14'</td>
<td>11°51'</td>
</tr>
<tr>
<td>IV</td>
<td>56°46'</td>
<td>11°51'</td>
</tr>
<tr>
<td>V</td>
<td>23°30'</td>
<td>2°17'</td>
</tr>
<tr>
<td>VI</td>
<td>78°</td>
<td>2°52'</td>
</tr>
</tbody>
</table>

Since black-body radiation is isotropic one can deduce (Lambert's law)

$$q_{b\theta} = q_b(e) \frac{\cos \theta \, \cos \theta}{\pi} = \frac{\sigma T^4}{\pi}$$

= energy emitted per unit area of black body, per unit time, per unit solid angle in the direction θ.
The energy emitted through the shaded solid angle is then

\[ q_{b\theta}^{(e)} \sin \theta \sin \theta \cos \phi \]

per unit area of black solid surface.

Integrating the above expression within the confines of our illuminating angles \( \Phi \) and \( \theta' \), (Fig. B) gives the amount of radiation emitted from a black-body of unit area. This radiation falls on the intercooler vanes and because of our choice of \( \Phi \) and \( \theta' \) goes through the slots of the intercooler. Since \( \Phi \) and \( \theta \) varies with each one of the 6 areas (I to VI) we must sum over each of the individual contributions. The radiation from the 300 K surface, within the solid angles defined by \( \Phi \) an \( \theta \) is

\[
q_b^{(e)} = \frac{\sigma T^4}{\pi} \int_0^\Phi \int_0^\theta \cos \theta \sin \theta \sin \theta \cos \phi \, d\theta \, d\phi
\]

\[ = \sigma T^4 F_{\text{area}} \]

We evaluate this function \( F \) for each of the 6 areas, I to VI.

Area I

\[ q_b = \sigma T^4 F_I \]

\[
F_I = \frac{1}{\pi} \int_0^{0.600} \int_{45^\circ}^{85^\circ} \cos \theta \sin \theta \sin \theta \cos \phi \, d\theta \, d\phi
\]

\[ = 0.0266 \]
Area II

\[ F_{II} = \frac{1}{\pi} \int_{0}^{\pi} 0.727 \int_{45^\circ - 7^\circ 1'}^{45^\circ + 7^\circ 1'} \sin \theta \cos \theta d \theta d \phi \]

= 0.208

Area III

\[ F_{III} = \frac{1}{\pi} \int_{0}^{\pi} 1.051 \int_{45^\circ - 5^\circ 55'}^{45^\circ + 5^\circ 55'} \sin \theta \cos \theta d \theta d \phi \]

= 0.034

Area IV

\[ F_{IV} = \frac{1}{\pi} \int_{0}^{\pi} 0.990 \int_{45^\circ - 5^\circ 55'}^{45^\circ + 5^\circ 55'} \sin \theta \cos \theta d \theta d \phi \]

= 0.032

Area V

\[ F_{V} = \frac{1}{\pi} \int_{0}^{\pi} 0.410 \int_{45^\circ - 1^\circ 8'}^{45^\circ + 1^\circ 8'} \sin \theta \cos \theta d \theta d \phi \]

= 0.0026

Area VI

\[ F_{VI} = \frac{1}{\pi} \int_{0}^{\pi} 1.36 \int_{45^\circ - 1^\circ 26'}^{45^\circ + 1^\circ 26'} \sin \theta \cos \theta d \theta d \phi \]

= 0.0110

We must now estimate the size of the different areas that are emitting 300°K radiation to the 28°K surface. To do this we find the azimuth angle (Ω) subtended by the 300°K surface at the freeze-out surface through each slot of the intercooler.
here

\[ \theta_Z = 2 \arctan \frac{L/2}{D} \]

L = length of each copper vane across the diameter of the intercooler as measured on the intercooler. Fig.(25).

The azimuths are listed in Table III-3.

<table>
<thead>
<tr>
<th>Vane</th>
<th>L</th>
<th>D</th>
<th>( \theta_Z ) (Radian)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>34</td>
<td>8&quot;</td>
<td>2.26</td>
</tr>
<tr>
<td>B</td>
<td>39</td>
<td>&quot;</td>
<td>2.34</td>
</tr>
<tr>
<td>C</td>
<td>44</td>
<td>&quot;</td>
<td>2.44</td>
</tr>
<tr>
<td>D</td>
<td>47</td>
<td>&quot;</td>
<td>2.48</td>
</tr>
<tr>
<td>E</td>
<td>49</td>
<td>&quot;</td>
<td>2.54</td>
</tr>
<tr>
<td>F</td>
<td>50</td>
<td>&quot;</td>
<td>2.54</td>
</tr>
<tr>
<td>G</td>
<td>51</td>
<td>&quot;</td>
<td>2.54</td>
</tr>
<tr>
<td>I</td>
<td>50</td>
<td>&quot;</td>
<td>2.54</td>
</tr>
<tr>
<td>J</td>
<td>49</td>
<td>&quot;</td>
<td>2.48</td>
</tr>
<tr>
<td>K</td>
<td>46</td>
<td>&quot;</td>
<td>2.42</td>
</tr>
<tr>
<td>L</td>
<td>43</td>
<td>&quot;</td>
<td>2.30</td>
</tr>
<tr>
<td>M</td>
<td>36</td>
<td>&quot;</td>
<td></td>
</tr>
</tbody>
</table>

From the azimuth angle \( \theta_Z \) and \( \bar{A} \) (Fig. D), the distance from the 28°K surface to the wall, we find the arc length (S) subtended at the wall by \( \theta_Z \). The height of each area is obtained from Fig. A. Table III-4 lists the mean values of S for each area and Table III-5 lists the total emitting area for each wall area I to VI.
The total radiation from the 300°K surface reaching the 28°K surface is given by
\[ Q = \sum_{i} a_i e^{-\epsilon_0 T^4} \sum_{i=1}^{VI} A_i F_i \]

where \( A_i \) is given in Table III-5 and \( F_i \) was calculated previously.

Hence
\[
\sum_{i=1}^{VI} A_i F_i = 5.6 (.0266 + 11.6 (.028) + 9.6 (.034) + 4(.032) + 64 (.0026) + 1(.011) = 1.053 \text{ ft}^2
\]

Here \( \sigma = 0.533 \times 10^{-8} \text{ watts/ft}^2 \text{ K}^4 \)
\( T_1 = 28^\circ\text{K} \quad T_2 = 300^\circ\text{K} \)

Then
\[
Q_{RT} = \epsilon (.533) \times 10^{-8} (81) \times 10^8 \times 1.053
\]
\[ Q = 45.4 \text{ watts} \]

if \( \epsilon = 0.5 \) (white paint on tunnel wall) then
\[ Q_{RT} \approx 28 \text{ watts} \]

A.2) To this radiation going through the intercooler slots must be added the one bouncing off the blackened intercooler. If the blackened intercooler surfaces had an emissivity of 1 (absorbs 100% of incoming radiation) then this would amount to 0 watts. But Ref. 20 gives an emissivity of \( \approx 0.74 \) for black glyptal. Hence \( \approx 30\% \) of the falling radiation is reflected. Of this one can assume that \( \approx 15\% \) is reflected to the freeze-out surface. (The other 15\% being reflected back to the tunnel walls.)

We assume that the radiation from the \( 300^\circ\text{K} \) surface, bounces off the vanes twice before entering the pump cavity. We must find the total radiation falling on the intercooler vanes and multiply it by the attenuation coefficient (15\% of 15\% or 0.0225\%) due to the double reflection off the vanes.

For the sake of calculation, we assume that the intercooler-tunnel wall combination can be represented by two 4.5 foot diameter surfaces about 2 feet apart with temperatures of \( 300^\circ\text{K} \) and \( 77^\circ\text{K} \) and emissivity of 0.5 and 0.74 respectively.

Simple calculations lead to 181 watts of radiation load on the \( \text{LN}_2 \) cooled intercooler vanes. The percentage of this total load that reaches the freeze-out surface is

\[ 181 \times 0.0225 \approx 4.1 \text{ watts} \]

Hence the total radiation falling on the freeze-out surface from the \( 300^\circ\text{K} \) walls is:
1. Radiation going directly through intercooler vanes: 28 watts
2. Radiation reflected off the intercooler vanes: 4.1 watts
3. Radiation from the LN2 cooled walls: 4.8 watts

**TOTAL 36.9 watts**

We see that the intercooler vanes block about 80% of the incident radiation from the 300°K surfaces.

**B)** It appears that the 4.5 foot diameter valve (Fig. 37) will always remain on top of the cryopump. (It can pass 5 lbs/hr. as shown in Section 4.7). This valve can be lined with aluminum foil. Calculations are made assuming that this cover is over the pump.

\[ \epsilon_1 \approx 0.9 \text{ (freeze-out surface)} \]
\[ \epsilon_2 = 0.03 \text{ aluminum foil} \]

\[ A_1 = 13.6 \text{ ft}^2 \]
\[ A_2 = 14.8 \text{ ft}^2 \]
\[ T_1 = 77^\circ \text{K} \]
\[ T_2 = 300^\circ \text{K} \]

We consider the simple case of two surfaces \( A_1, A_2 \) close together and in full view of each other. Simple calculations lead to a radiation load of 19 watts. But as seen from Section A, the intercooler allows approximately 20% of the radiation falling on it to pass through to the 28°K surface. Hence with the above arrangement, we have a heat load of

\[ 0.20 \times 19 = 3.8 \text{ watts} \]
on the freeze-out surface.

**4) Conduction Heat Leaks to the Freeze-Out Surface**

1) **Gas Conduction:**

\[ Q = KA \frac{\Delta T}{\Delta X} \]

\( K_{\text{air}} \) at 80°K = 0.0043 B.T.U./ft-hr-°R (Ref. 29)

\[ A = \log \text{mean area with projected area of freeze-out surface} \]

Then

\[ A = \frac{132-106}{\ln 132/106} = 117 \text{ ft}^2 \text{ (Fig. 21 and Section 4.3)} \]

\[ \Delta T = 72^\circ \text{K} - 28^\circ \text{K} = 44^\circ \text{K} \]

\[ \Delta X = 0.3 \text{ ft (mean distance between freeze-out surface and walls of tank)} \]
Substituting the above values in the expression for $Q$, we get

$$Q = 133 \text{ B.T.U./hr} = 39 \text{ watts}.$$

As mentioned in the text (Sec. 8.1) because of the high capture coefficient of the freeze-out surface, gas conduction heat loads are insignificant. We therefore assume that only about 10% of the above value is present. Hence $Q_c \approx 4 \text{ watts}$.

2) Support

A - Stainless steel pipe - negligible.
B - Teflon pads.

There are 4 pads supporting the freeze-out star. Assuming approximately 2 cm$^2$ of contact area, we have

$$Q = \frac{5.85 \times 10^{-4} \times 60 \times 2 \times 50^\circ C}{14.33 \times 0.62}$$

$$= 0.41 \text{ watts}$$

These heat leaks to the freeze-out surface are tabulated in Table IV of the text.
APPENDIX IV

Test Section Pressure Control Valves

Since cryogenic pumps are constant mass flow devices, we must control the test section pressure by means of throttling valves directly above the pump (Fig. 37). Two 15" diameter gate valves could be fitted in the available space. In order to find the throughput of these valves we must first determine the nature of the flow regime through them. Following Ref. 23 the criteria for the flow regime are

\[
\begin{align*}
\frac{dP}{\mu} &> 500, \text{ flow is viscous} \\
\frac{dP}{\mu} &< 5, \text{ flow is molecular} \\
5 &< \frac{dP}{\mu} < 500, \text{ flow is in transition regime}
\end{align*}
\]

here \( d = 15 \times 2.54 = 38 \text{ cm.} \) (diameter of opening)

\[
\frac{P}{\mu} = \text{pressure in micron Hg.}
\]

Hence if

\[
\frac{P}{\mu} > \frac{500}{38}
\]

i.e. if \( \frac{P}{\mu} > 13 \) microns Hg., flow is viscous and

if \( 1 \mu < \frac{P}{\mu} < 13 \), the flow is in the transition regime.

1) Consider viscous flow (i.e. \( P_2 > 13\mu \))

Following Ref. 23

\[
\frac{dV}{dt} = \frac{\mu a^4}{6n\ell} \ Pa \ (P_2 - P_1)
\]

\( P_2 = \) upstream pressure
\( P_1 = \) downstream pressure (\( \approx = 13\mu \))
\( Pa = P_2 + P_1/2 \)

\[
\frac{dV}{dt} = \frac{5 \times 7.87 \times 10^{-3}}{32 \times 24} \text{ atm. c.c./sec}
\]

\[
= 510 \text{ atm. c.c./sec}
\]

Properties of Air at 25°C (Ref. 29)

\( n = 1.845 \times 10^{-4} \) poise
\( a = 19 \text{ cm} \) (radius of valve opening)
\( \ell = 12 \times 2.54 \text{ cm} \) (length of passage formed by the valve body)
Then;

\[
\frac{u}{8nL} = 915 \times 10^4
\]

and

\[
P_2 \frac{dV}{dt} = \frac{\tau a^4}{8nL} \left( \frac{P_2 + P_1}{2} \right) (P_2 - P_1)
\]

Substituting the known values, and solving for \(P_2\), we get

\[
P_2 \approx 13\mu
\]

We postulated that \(P_1 = 13\mu\); then 5 lb/hr \(N_2\) mass flow will pass through the 15" diameter opening with practically no pressure drop across the valve under these flow conditions.

2) **Transition Region**

In this regime, the pressure upstream of the 15" diameter valve (\(P\)) is

\[1\mu < P < 13\mu\]

The viscous conductance corrected for slip is

\[
F = \frac{\tau a^4}{8nL} P_a \left( 1 + \frac{4\xi}{a} \right) \text{(Ref. 23)}
\]

\[
\xi = \frac{2-f}{f} \frac{n}{P_a} \left( \frac{R_0 T}{2n \mu} \right)^{\frac{1}{2}}
\]

here \(f \approx 0.84\) (Ref. 23)

\[
L_a = \frac{2''}{P} = \frac{6.76 \text{ cm}}{P_a (\text{dyne/cm}^2)}
\]

Then

\[
F = \frac{\tau a^4 P_a}{8nL} \left( 1 + 4 \left( \frac{2-f-1}{2} \right) \frac{L_a}{a} \right)
\]

= Flow rate per unit difference of pressure between the ends of the channel.

Substituting all the above values in the expression for \(F\), we get

\[
F = 457.5 \times 10^4 \left[ P_2 + 5.23 \right]
\]
### TABLE IV - 1

<table>
<thead>
<tr>
<th>$P_2$ (μ Hg)</th>
<th>$P_2$ d/cm²</th>
<th>$F$ ($\Delta P = l d/cm²$)</th>
<th>$F$ ($\Delta P = P_2 - P_1$)</th>
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<tr>
<td>1.25</td>
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<td>31.49 x $10^6$ cc/sec</td>
<td>.106 x $10^5$ l/sec</td>
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<td>42.14</td>
<td>1.1209</td>
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</table>

These results are plotted on Fig. 28.

### C. Sonic Limit of Flow

An upper limit is imposed upon the flow of gas through the valve opening by the establishment of sonic speed through the valve.

Area of valve opening is

$$\frac{\pi x 15^2}{4 x 144} = 1.22 \text{ ft}^2$$

The speed of sound is given by

$$S = \sqrt{\gamma RT}$$

- **Air:** $\gamma = 1.4$, $R = 1716 \text{ ft} \cdot \text{lb/slug} \cdot \text{O R}$
- **Argon:** $\gamma = 1.66$, $R = 1276$ "
- **$N_2$:** $\gamma = 1.47$, $R = 1778$ "

Hence:

- $S_{\text{air}}: = 1128 \text{ ft/sec}$
- $S_{A}: = 1061 \text{ ft/sec}$
- $S_{N_2}: = 1177 \text{ ft/sec}$

These results are plotted on Fig. 28.
APPENDIX V

Liquid Nitrogen Consumption

1. Radiation Heat Leak to 72°K Surface

The cryopump container is considered to be formed to two concentric cylinders (Fig. 24). The space in between is filled with "Perlite" powder insulation and evacuated to below 50 x 10^{-3} torr. The outer wall is at 300°K while the inner wall is maintained at 72°K or lower with liquid nitrogen. The conductivity of "Perlite" under these conditions is

\[ k = 0.02 - 0.03 \text{ BTU - in/hr-ft}^2\text{-°F} \] (Ref. 22)

\[ k = 1.1 \times 10^{-2} \text{ BTU - in/hr-°F} \] (Ref. 23)

which gives 70 watts and 35 watts of thermal load respectively. To this must be added the radiation falling on the intercooler which comes to 181 or 19 watts depending whether the 4.5 foot diameter valve, lined with aluminum foil, is in place or not (App. III). Hence the thermal (mostly radiation) load on the copper shields is

<table>
<thead>
<tr>
<th>Maximum</th>
<th>Minimum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q_R = 251 watts</td>
<td>54 watts</td>
</tr>
</tbody>
</table>

2. Cooling Down of Air to 80°K by Intercooler

The amount of refrigeration (Q) needed to cool down a mass M of N_2 (or air) is

\[ Q = \dot{M} C_p \Delta T \]

\[ M = 5 \text{ lb/hr} \]

\[ C_p = 0.44 \text{ BTU/lb °K} \]

\[ \Delta T = 300 - 80 = 220°K \]

Then

\[ Q = (5 \text{ lb/hr}) (0.44 \text{ BTU/lb °K}) 220°K = 485 \text{ BTU/hr} = 141 \text{ watts} \]

3. Support Heat Leak

A conical shaped S.S. support (Fig. 24) holds the copper shield within the steel structural shell. It's dimensions are 60" (average dia.) x 16" long x 3/16" thick

Now

\[ Q = \frac{KA\Delta T}{\Delta x} \]
K = 10.6 BTU/ft/hr\(^o\)K
ΔX = 16 in. = 1.33 ft
ΔT = 300-65 = 225\(^o\)K

\[ A = \pi Dt = \frac{(60)(0.187)}{144} = 0.245 \text{ ft}^2 \]

\[ Q = \frac{10.6 \times 0.245 \times 235}{1.33} = 460 \frac{\text{BTU}}{\text{hr}} = 135 \text{ watts} \]

4) Cooling Down of the Neon Gas to 66\(^o\)K by HX No. 2 (Fig. 29)

\[ h_1 = 19.3 \text{ BTU/c.f.} \quad h_2 = 18.85 \text{ BTU/c.f.} \]
\[ P_1 = 2000 \text{ psi} \quad P_2 = 2000 \text{ psi} \]
\[ T_1 = 75\(^o\)K \quad T_2 = 64\(^o\)K \]

We have 20.3 SCFM of neon flowing through the heat exchanger No. 2. The LN\(_2\) must supply an amount of refrigeration

\[ Q = \dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\dot{\do
APPENDIX VI

We try to estimate the surface temperature of the nitrogen condensate under certain pumping conditions. Reference 24 gives conductivity and density values for solid nitrogen

\[ k = 4.0 \times 10^{-4} \text{ cal/sec cm} \cdot \text{O} \cdot \text{K} \]
\[ \rho = 1 \text{ gr/cm}^3 \]

Consider the case when a mass flow of 0.5252 atm-L/sec was being condensed by the freeze-out surface.

\[ \dot{M} = 0.5252 \text{ atm-L/sec} \]
\[ P_{\text{tank}} = 3.5 \mu \text{Hg} \]

Assume that only 10% of the freeze-out surface is pumping.

Area of freeze-out surface = 0.10 x 450 ft\(^2\)
= 45 ft\(^2\)

The weight of nitrogen deposited over a 10 hour period is

\[ 0.5252 \times 1.25 \times 3600 \times 10 = 2.36 \times 10^4 \text{ grams} \]

This is deposited on an area of

\[ 45 \text{ ft}^2 \times 929.03 = 4.18 \times 10^4 \text{ cm}^2 \]

giving a thickness of

\[ \frac{2.36 \times 10^4}{4.18 \times 10^4 \times 1} = 0.564 \text{ cm} \]

Figure 48 shows that we need approximately 197 watts of refrigeration to pump 0.5252 atm-L/sec N\(_2\). Then, the temperature drop across the condensate layer is

\[ T = \frac{Q \cdot dx}{k \cdot \text{area}} \]
\[ = 27.8^o \text{K} \]

This would bring the surface temperature of the condensate to 55^oK which would require a tank pressure of a few hundred millimeters of Mercury. At the time the pressure in the pump chamber was 3.5 x 10\(^{-3}\) torr. We must therefore have much more than 10% of the freeze-out surface at 28^oK or we must have a thermal conductivity of the condensate that is vastly superior to the one quoted above.

There is indication that only 2 of the 6 vanes are pumping properly (Fig. 46). Under these conditions, the temperature drop across the condensate thickness is

\[ \frac{27.8^o \text{K} \times 0.1 \times 6}{2} = 8.3^o \text{K} \]
and the surface temperature of the condensate is 

\[ 27 + 8.3 = 35.3^\circ \text{K} \]

This corresponds to a vapor pressure of \( 6 \times 10^{-3} \) torr for the condensate (Ref. 13) which does not agree with the measured pressure of \( 3.5 \times 10^{-3} \) torr in the pump chamber. We therefore must have had more than two of the six vanes pumping gas if the value of \( \rho \) and \( k \) are as assumed in the Appendix.
A STUDY OF A NEON CYCLE CRYOGENIC PUMPING SYSTEM FOR A LOW DENSITY PLASMA TUNNEL

This paper describes the development and testing of a large neon cycle cryogenic pump for use with a low density wind tunnel meant for both ionized and non-ionized flows.

The development work necessary in the choice of the refrigerant cycle and the production of the required hardware is described in some detail. Once the cryogenic pump was in operation it was thoroughly tested both as a refrigerator system and as a pump.

Experimental data on the thermodynamic properties of neon was obtained and found to be in excellent agreement with the available theoretical data. The refrigerator cycle was found to operate satisfactorily and as predicted. The pump itself was found to operate satisfactorily and as predicted over its entire designed range of operation. A pumping rate of 0.8 gr/sec N₂ was achieved while maintaining the test section pressure in the 1 - 4 x 10⁻³ torr region.
### Key Words

<table>
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<th>LINK A</th>
<th>LINK B</th>
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<td>ROLE</td>
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**Cryogenic pumping**  
**Neon Joule-Thomson Cycle**

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