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DESIGN AND CONSTRUCTION OF A LIQUID HYDROGEN TEMPERATURE REFRIGERATION SYSTEM

BY D. B. CHELTON, J. W. DEAN, AND B. W. BIRMINGHAM

U. S. DEPARTMENT OF COMMERCE
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ABSTRACT

Maintaining a liquid hydrogen bubble chamber at 27°K has been achieved with an automatically controlled closed circuit hydrogen refrigeration system of 300 watts capacity. The system is sufficiently flexible to be used on other experimental apparatus requiring refrigeration at liquid hydrogen temperatures. Several control systems are discussed. Experimental evidence is compared to predicted performance for design operating conditions.

General design charts are developed that enable heat exchanger lengths and associated operating parameters to be determined for the pertinent heat exchanger configuration when employed in liquid hydrogen refrigerators of other capacities.

INTRODUCTION

The use of cryogenics in physics and engineering often requires continuous refrigeration of apparatus at low temperature [1,2,3]. An example of this, is the 15 inch diameter liquid hydrogen bubble chamber at the Lawrence Radiation Laboratory (LRL), Berkeley, California that requires refrigeration at approximately 27°K. The Cryogenic Engineering Laboratory of the National Bureau of Standards has developed an automatically controlled closed circuit hydrogen refrigeration system with a nominal capacity of 300 watts to meet the bubble chamber requirements. Previously the bubble chamber received refrigeration by the evaporation of liquid hydrogen from a reservoir, the liquid being supplied by a remotely located hydrogen liquefier.

A major design feature of the refrigerator is the use of the Collins type ribbon packed heat exchanger.* The choice of these commercially available heat exchangers alleviates much of the heat exchanger design and construction problem. The heat exchanger configuration, because of excessive pressure drop in the low pressure nitrogen passage, limits the refrigeration capacity to 300-400 watts for a single three way heat exchanger (see Figure 2). However, the application of multiple three way heat exchangers effectively increases the upper limit of refrigeration obtainable.

Design charts were developed that enable the determination of heat exchanger lengths and associated operating parameters for the heat exchangers when employed in liquid hydrogen temperature refrigerators. Charts of a general nature are given for hydrogen flow rates and liquid nitrogen required.

Refrigeration temperatures may be extended to that of liquid helium by using the hydrogen refrigerator to maintain a precoolant reservoir in an additional helium circuit [4]. The additional circuit may be used to produce helium liquefaction or refrigeration to 4.2°K, adding considerable flexibility to the apparatus without modification to the existing hydrogen system.

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* Manufactured by the Joy Manufacturing Company, Michigan City, Indiana.

BASIC CONSIDERATIONS

The desirable qualities of a closed circuit refrigeration system can be visualized by considering applicable economics, logistics and safety. The use of such a system is a more economical method of obtaining refrigeration than by evaporating liquid from a vented reservoir. The venting of the reservoir (supplied by a remotely located hydrogen liquefier) to the atmosphere results in the loss of the gas in addition to the loss of the sensible heat of the gas. In contrast, a refrigerator operates in a closed circuit and utilizes the sensible heat of the gas obtained from liquid evaporation. The sensible heat of hydrogen gas between saturation temperature and room temperature is equal to 8 times the latent heat of vaporization. The use of the sensible heat reduces the power required to operate a refrigerator to one-third of the power required to operate a liquefier for equal quantities of refrigeration.

Figure 1 shows the total power requirements of a refrigerator as compared to a liquefier based on theoretical performance of the final heat exchanger and $10^{\circ}K$ temperature difference at the top of the H_2 - H_2 - N_2 heat exchanger (see Figure 2). The power requirements have been calculated for a Hampson or Simple Linde cycle with $65^{\circ}K$ liquid nitrogen precooling. The refrigeration obtained from the liquefier is based entirely on evaporation of liquid. The power calculation assumes 65 percent isothermal compression efficiency and includes the work necessary to provide precooling. It is assumed that the nitrogen is supplied to the unit as saturated liquid at 2 atmospheres. The work to produce the nitrogen refrigeration was taken as 0.505 Kw-hr per liter of nitrogen [5, 6]. Although this may be optimistic for small scale procurement it serves as a basis for comparison.

For small capacity refrigeration requirements, operating power considerations may not be of major economic importance, since the power cost for both a refrigerator and a liquefier will be small in comparison to other operating costs. However, the power requirement influences the initial capital investment since the cost of a compressor is proportional to the compression power. The compressor for the 300 watt hydrogen refrigerator represents about 30 percent of the total system investment. The compressor for a liquefier of comparable refrigeration capacity would be considerably greater. The cost of purchasing liquid hydrogen as a source of refrigeration is usually more than predicted by power requirements since the costs must sustain a completely separate operation for the production and handling of the liquid. In addition to the increased costs of liquid, refrigeration logistics require substantial quantities of liquid to be

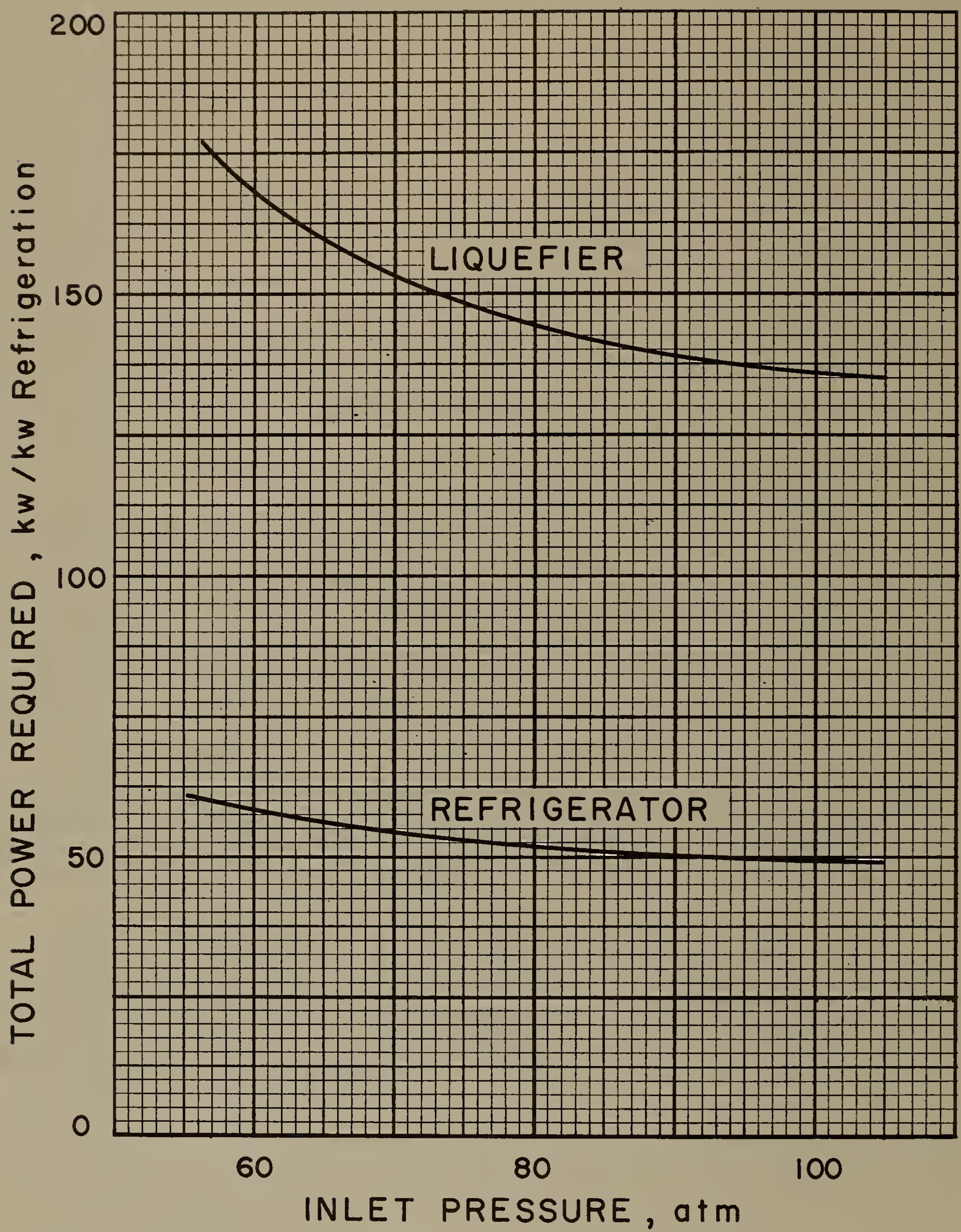

Figure 1. Power Requirements for Refrigerator and Liquefier

stored at the use site to insure uninterrupted operation. Losses thus incurred add to operating expenses.

The handling of a large number of small liquid hydrogen dewars presents a more formidable safety problem than a closed circuit system. Each handling increases the possibility of introducing contamination in the containers. Under some circumstances the contaminants may be accumulated in the refrigerated container. In addition, the stored containers required by logistics necessitate a separate area that must be suitably controlled.

REFRIGERATION SYSTEM

The liquid hydrogen bubble chamber at LRL must be maintained at operating temperature on a continuous basis for extended periods of time (perhaps two or three months.) Reliability is therefore a prime consideration, and for this reason the Hampson or Simple Linde cycle was chosen for the refrigerator. Although the thermodynamic efficiency is not great, the cycle has the advantage of simplicity and reliability since it avoids moving parts at low temperature.

A schematic arrangement of the refrigeration system is shown in Figure 2. Saturated hydrogen vapor at 2-10 psig from the evaporator passes to the low pressure side of the final heat exchanger. The gas is warmed in the final heat exchanger to approximately the temperature of the precoolant fluid by countercurrent exchange with the high pressure supply gas. The exit temperature is maintained, by preventing further heat transfer, until it enters the three-way (H_2 - H_2 - N_2) heat exchanger where it is paralleled with the evaporated nitrogen gas from the precoolant bath. By-pass valves are provided on both heat exchangers and transfer line to speed warm-up and cool-down procedures. The low pressure hydrogen and evaporated nitrogen in this exchanger is warmed from precoolant temperature to approximately ambient temperature by heat exchange with the entering high pressure hydrogen gas. As a result, the high pressure gas is cooled to slightly above the temperature of the precoolant. The high pressure hydrogen then exchanges heat with the precoolant liquid in the liquid-gas exchanger located in the precoolant tank. The tank is sized to sustain operation for a period of eight to ten hours at design conditions without additional transfer of liquid. Further cooling of the high pressure gas is performed in the final heat exchanger. Expansion results in approximately a 50-50 mixture of gas and liquid. The temperature difference between the high pressure stream and the low pressure stream at the warm end of the heat exchangers is represented as ΔT_1 and ΔT_2.

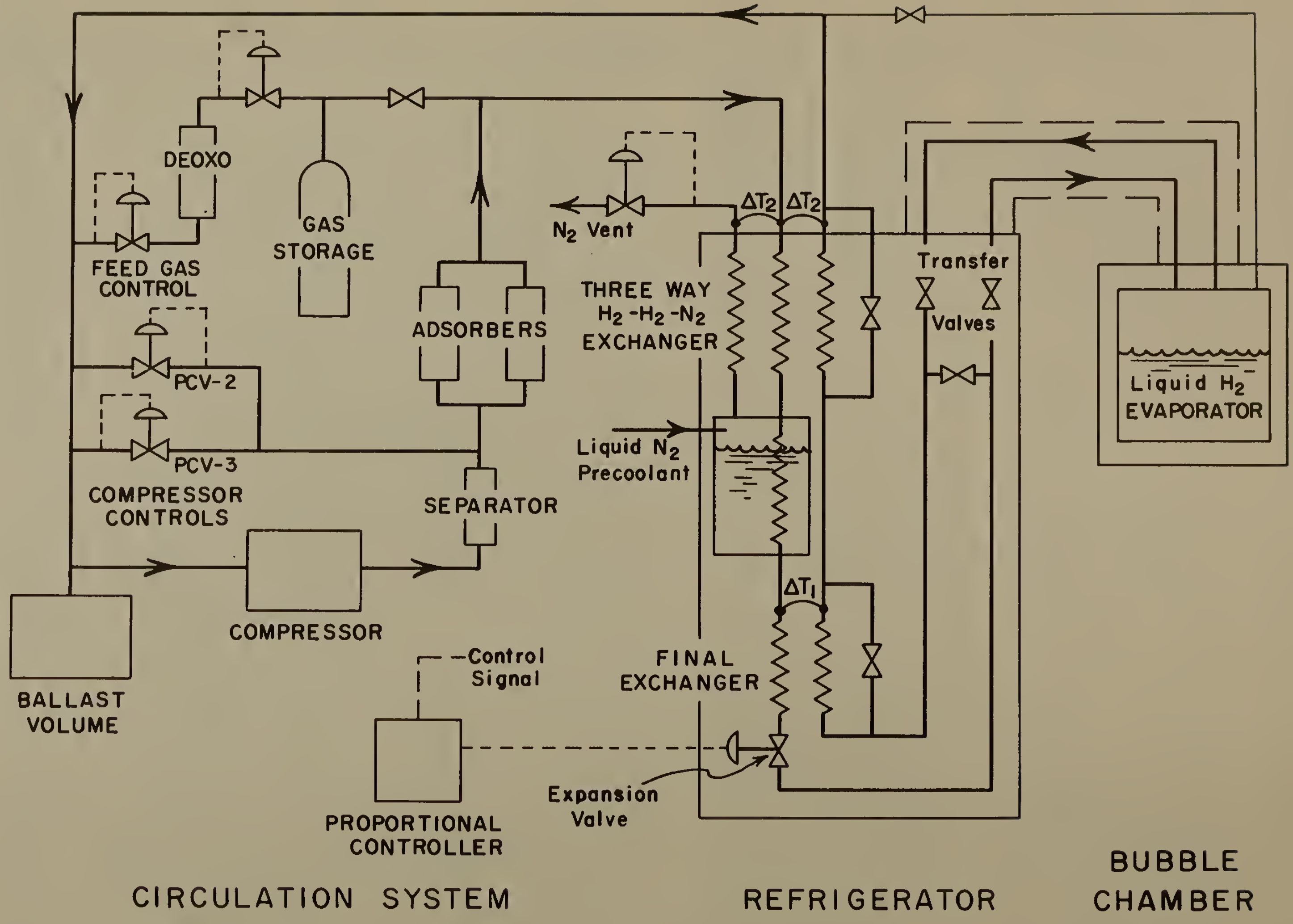

Figure 2. Schematic Arrangement of Refrigeration System

A small volume momentum chamber is located in the high pressure line prior to expansion. The purpose of the momentum chamber is to separate solid impurities that might precipitate out in the system. The chamber allows solids to collect before they have an opportunity to obstruct the expansion valve passage. The installation of such a device has been found desirable in the operation of other low temperature liquefiers and refrigerators.

The regulators associated with the circulation system shown in Figure 2 may be classified as feed gas and compressor controls. The prepurified hydrogen gas is stored in the high pressure (1800 psi) gas storage cylinders when the refrigerator and evaporator are at ambient temperature. This gas is drawn into the system through the feed gas controls as cool-down progresses. Enough gas is stored to fill the evaporator with liquid to the desired level. Pressure reduction from the storage cylinder to the compressor intake is done in two steps to allow the use of a standard commercial unit for removing oxygen impurity. These controls are valved out of the circuit after the refrigeration system is charged.

The compressor pressure control valves (PCV-2 and PCV-3) bypass the compressor when the discharge or intake pressure varies from their respective set points. The continuous operation of the compressor tends to reduce the intake pressure when the flow returning from the refrigerator is less than the compressor capacity, i.e., less than full refrigeration load. Control PCV-3 bypasses the flow from the compressor discharge and maintains the intake pressure at a fixed minimum. Control PCV-2 is not normally operative, but relieves the discharge pressure to the intake when a set point is exceeded.

High purity hydrogen is used to initially charge the refrigeration system. However, a purification train must be used in the main process gas stream to remove the impurities (oil and water vapor) introduced by the operation of the oil lubricated compressor. A micrometallic oil-water separator is located directly after the compressor aftercooler. Purification is accomplished by the use of an adsorption unit operating at ambient temperature. The adsorption unit consists of two cylinders containing 13X Linde Molecular Sieves ** having a relatively high affinity for oil vapor compared to other adsorbents. A valving arrangement allows flow through the cartridges either in series or parallel. Although normal operation is in series, parallel

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** Manufactured by the Linde Company, Tonawanda, New York.

operation permits continuous flow while a contaminated cartridge is replaced. The unit is designed to allow a minimum contact time of 20 seconds and a maximum superficial flow velocity of 10 feet per minute under design operating conditions. Design calculation, substantiated by initial operations, indicate cartridge operation will exceed 5-7 days without regeneration.

The evaporator of the refrigeration system shown in Figure 2 is part of the bubble chamber apparatus. No liquid hydrogen container is incorporated within the refrigerator proper. However, hydrogen circulation through the refrigerator may be accomplished independent of the evaporator by the use of the transfer valves. This arrangement is desirable when separate cool-down of the refrigerator is necessary. A more detailed view of the evaporator may be seen in Figure 3. The liquid hydrogen flask represents the evaporator of the closed circuit hydrogen refrigerator. The normal refrigeration load is produced by both the dynamics of the bubble chamber operation and the heat leak into the low temperature apparatus. The essential parts of the apparatus are surrounded by a liquid nitrogen temperature thermal radiation shield. The bubble chamber is initially filled by condensation of hydrogen gas within the coils located in the flask. For a given refrigeration requirement, the liquid hydrogen in the evaporator is maintained at a constant temperature. Temperature control of the bubble chamber is accomplished by adjusting the length of the conduction path between the chamber and the refrigerant. Adjustment is performed with the piston and control rod arrangement shown. Withdrawing the control rod (and piston) increases the distance between the chamber body and the refrigerant, thus reducing the heat transfer to the chamber.

Since the refrigeration system was designed to be incorporated with an existing evaporator, limited space was available for interconnections. The refrigerant and the gas from the evaporator are transferred through a high vacuum insulated two passage concentric transfer line. The essential parts of the transfer line assembly are shown in Figure 4. The vacuum jacket of the transfer line is a copper tube 1.375 inch outside diameter. The concentric arrangement consolidates the thermal heat influx by allowing only one hydrogen temperature tube to receive room temperature radiation. This reduces the overall static thermal heat influx to the unit-especially important where the length of the transfer line becomes appreciable. The thermal heat load into the transfer line is approximately 0.4 watts per foot of length.

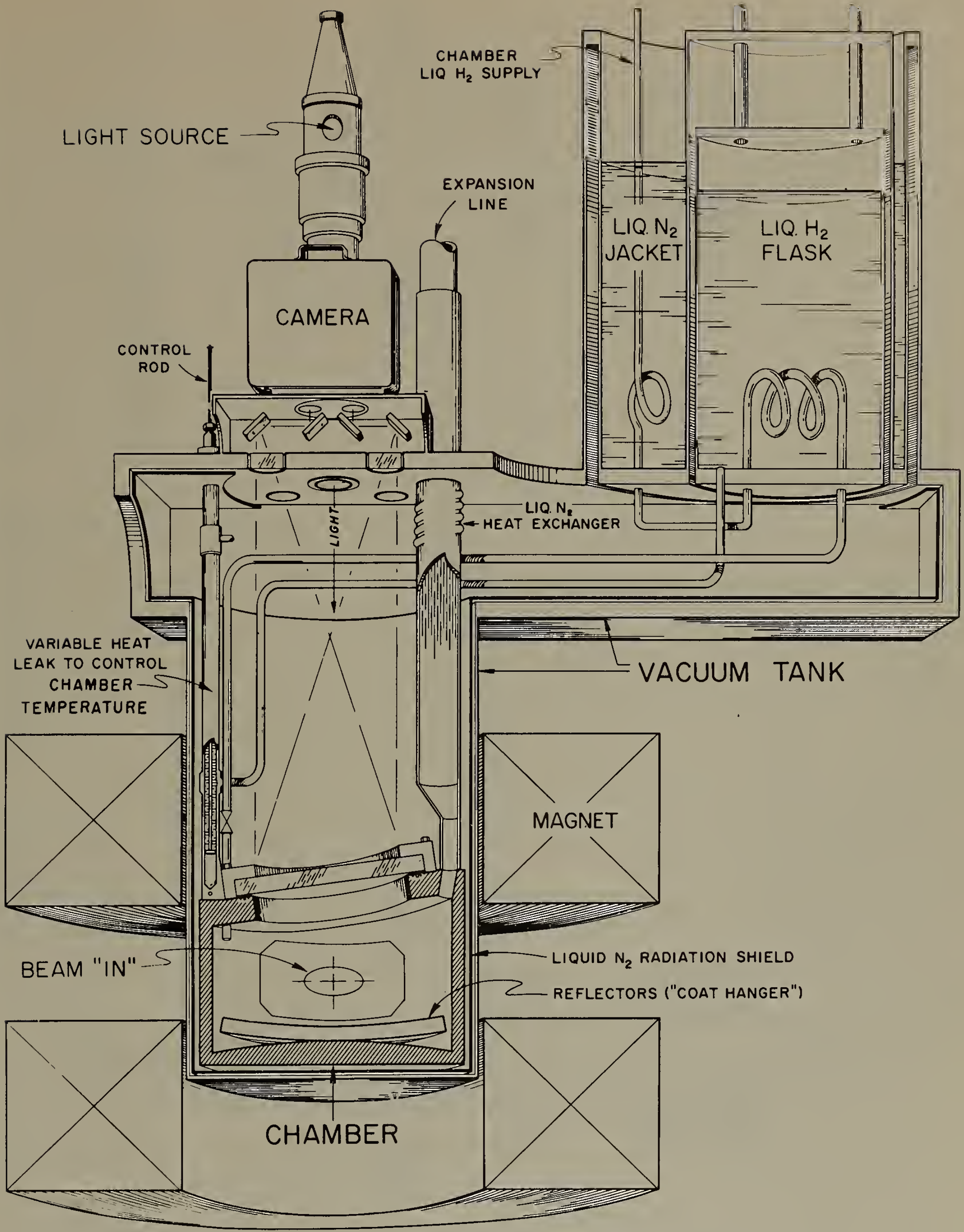


Figure 3. Cross Section View of 15—Inch Liquid Hydrogen Bubble Chamber

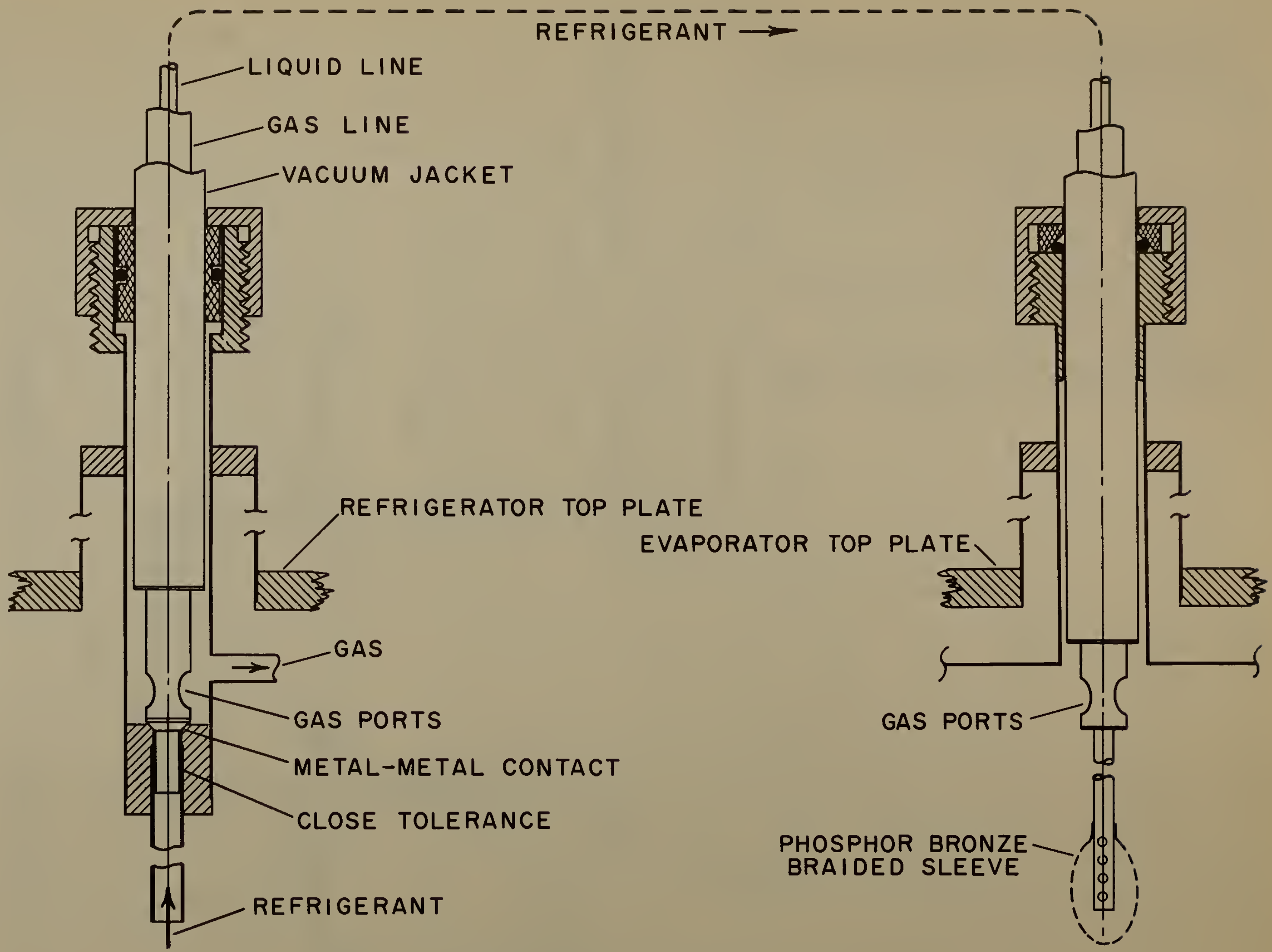


Figure 4. Refrigerator Transfer Line

Division of the flow on the refrigerator side of the transfer line is accomplished by a combination of a close tolerance and a metal-to-metal contact, as shown in the figure. An alternate procedure provides a polytetrafluoroethylene "O" ring on the inner liquid line to replace the metal-to-metal contact. The elastic seal compensates for surface irregularities. Slight leakage of liquid across the seal does not appreciably reduce the performance of the unit since the refrigeration of the liquid is utilized in the final heat exchanger to provide additional cooling of the high pressure gas. Excessive leakage, however, will reduce the efficiency of the heat exchanger to a point that counteracts this compensation. Calculations indicate that for a design $\Delta T_1 = 0.25^{\circ}K$ ($65^{\circ}K$ precooling), a 10 percent liquid leakage reduces the available refrigeration at the evaporator less than 1 percent. For a design $\Delta T_1 = 1.0^{\circ}K$, a 10 percent liquid leakage reduces the refrigeration 1.5 percent.

The evaporator side of the transfer line is provided with a phosphor bronze braided sleeve to reduce entrainment losses. The application of the metallic sleeve disperses the entering stream and allows the liquid to collect in the evaporator in a more quiescent state. This was found necessary since the evaporator receives a mixture of gas and liquid. With the fixed evaporator design it was found impractical to condense the entire stream in a submerged type heat exchanger. The application of the sleeve increased the refrigerator performance approximately 15 percent.

Figure 5 shows the entire refrigeration system during a performance test. The major component arrangement is similar to that shown in the schematic of Figure 2. The oil-lubricated compressor, capable of delivering 35 scfm of hydrogen at 1800 psi, is located at the left. A major portion of the circulation system is located within the panelled framework next to the compressor. The compressor controls, the feed gas controls, and the purification system are included. The high pressure gas storage cylinders are located behind the framework. Manual valves and gauges necessary for operations performed with the circulation system are on the attached panel. The second panelled framework, located at the right, contains the dewar with the refrigerator heat exchangers and the precoolant tank. Refrigerator gauges and manual valves are placed on this panel. An angle view of the refrigerator framework with the dewar removed is shown in Figure 6. The transfer line connects the refrigerator to the evaporator located at the far right. The evaporator shown, constructed for performance testing, consists of a 20 liter liquid hydrogen dewar with an electric heating element to simulate the refrigeration load normally provided by the bubble chamber.

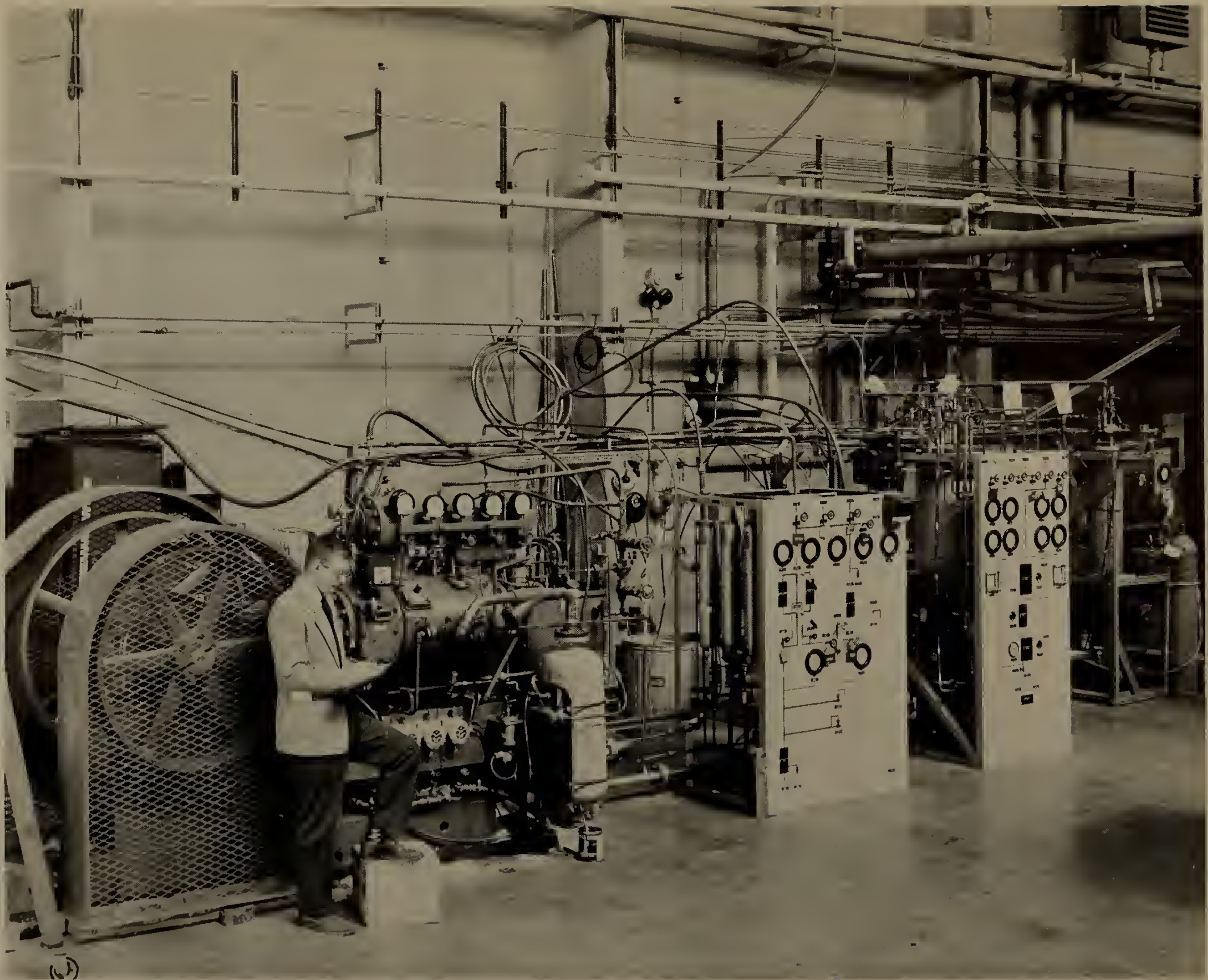


Figure 5. View of Refrigeration System

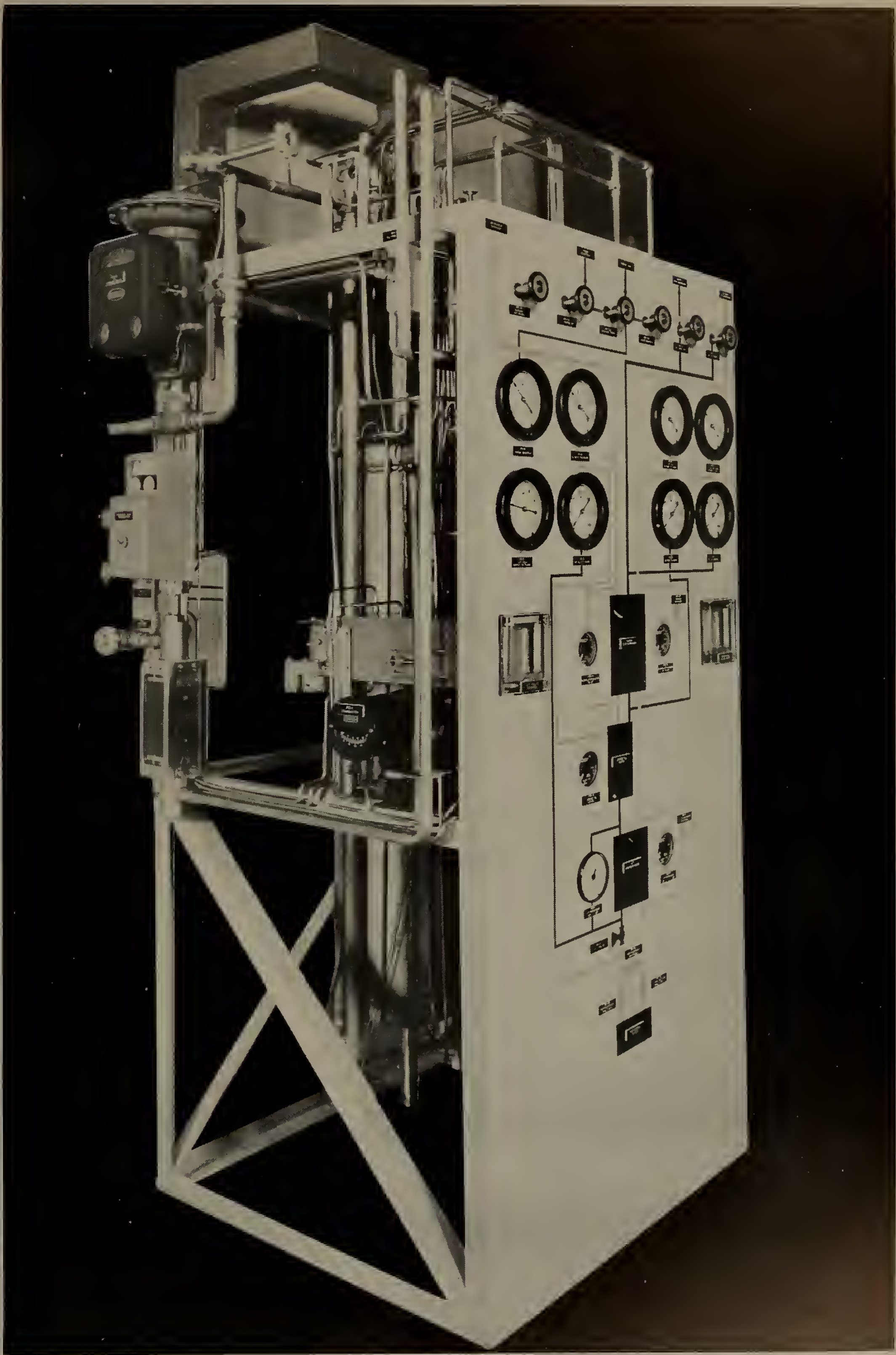


Figure 6. View of Refrigerator Panel Board

An overall view of the bubble chamber in operation prior to the refrigerator construction is shown in Figure 7. The refrigerator is placed adjacent to the bubble chamber. However, the congested test area requires that the circulation system be placed some distance away. A small portion of the Bevatron magnet is visible in the upper left corner. A nuclear radiation shield of heavy concrete separates the Bevatron from the experimental area. The particle beam is extracted from the accelerator and is directed through appropriate passages in the concrete shielding into the bubble chamber. The concrete bunker in the background normally encloses experimental apparatus, such as the bubble chamber, that is effected by stray nuclear fields. These stray fields reduce the clarity of event photographs and add to the difficulty of analysis.

REFRIGERATION CONTROL

Two basic parameters may be varied to control the quantity of refrigeration at the evaporator. From an energy balance application of the First Law of Thermodynamics, the controlling parameters are found to be the mass flow rate and the enthalpy difference at the top of the final heat exchanger. This may be expressed as follows:

$$Q = m\Delta H$$

The variation of these basic parameters presents many possibilities of refrigeration control. The mass flow rate is affected by the pressure and aperture size of the expansion valve, while the enthalpy difference of the final heat exchanger is affected by pressure (high and low), precooling temperature and heat exchanger efficiency. Of these variables and combinations thereof, three methods of refrigeration control were selected that could appropriately achieve the desired results. Control by these methods has been considered by others and has here been adopted for our particular requirements.[7,8]

The quantity of refrigeration may be controlled by adjusting the flow rate through the evaporator while maintaining a constant high pressure at the expansion valve. The control is performed by positioning the expansion valve in accordance with refrigeration demands. The position of the expansion valve regulates the required hydrogen flow while any excess flow from the compressor is by-passed to the compressor intake by PCV-3. The control signal must be proportional to the change in the refrigeration required and the process variable must be returned to an initial condition by the corrective action of the expansion valve. Stable operation requires other system

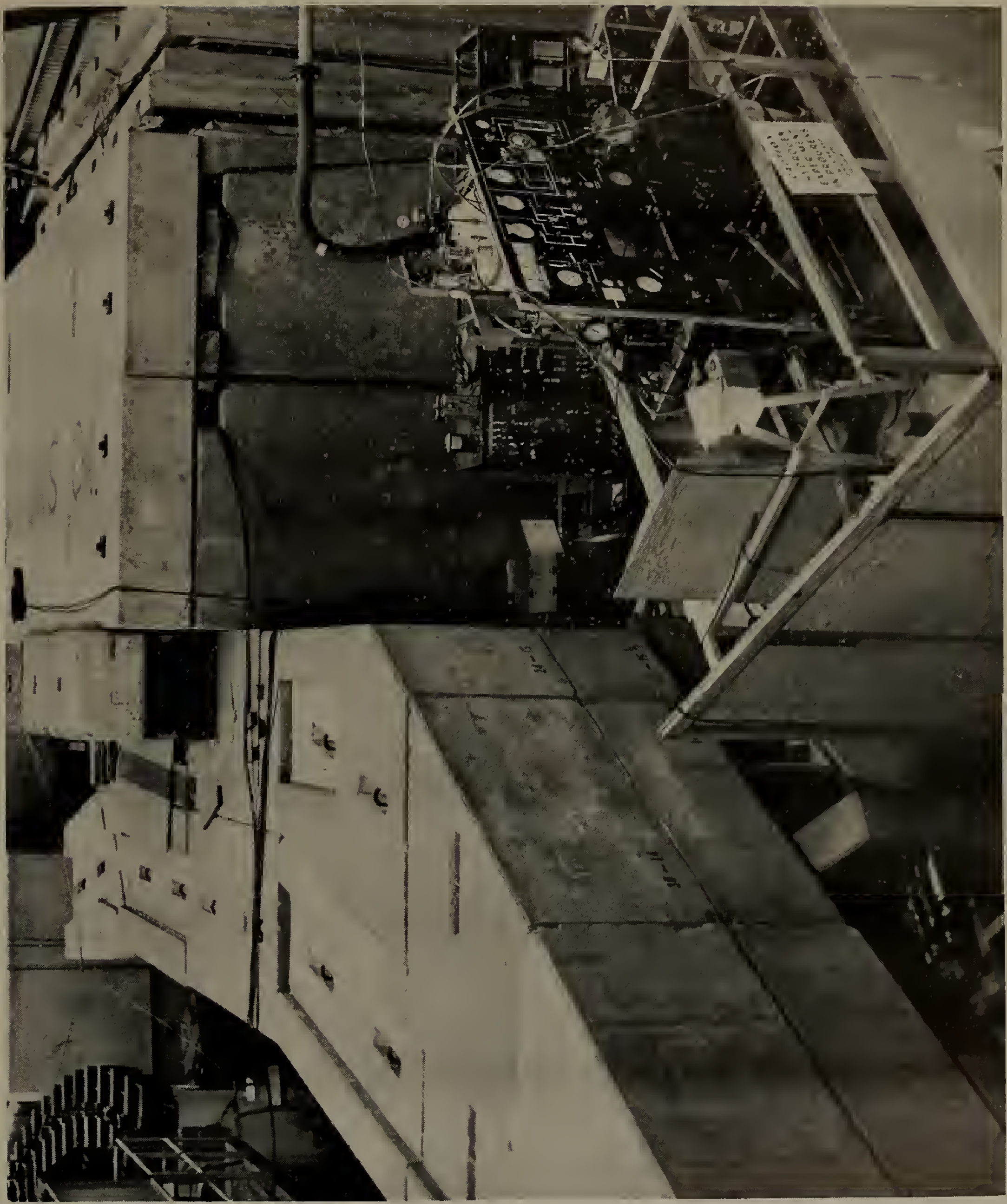


Figure 7. Bubble Chamber Installation at the Bevatron

variables be held constant and appropriate measures have been taken to control these variables. For example, a constant precoolant temperature is assured by a pressure regulator on the precoolant tank.

The first method for refrigeration control maintains a constant high pressure by monitoring the pressure prior to the expansion valve (high pressure control). A variation in the required refrigeration will affect the compressor intake pressure. Since PCV-3 maintains the intake pressure above a fixed minimum, a reduced refrigeration requirement will cause PCV-3 to bypass gas from the compressor discharge as previously indicated. Consequently, the discharge pressure tends to be reduced. The pressure reduction tendency is indicated by an air signal from the pressure transmitter. The expansion valve controller receives the signal and closes the expansion valve proportionally. The latter action returns the discharge pressure to the reference point and reduces the hydrogen flow delivered to the refrigerator. An increased refrigeration demand, not to exceed the refrigeration capacity of the system, will produce a similar sequence of events and result in an increased hydrogen flow through the refrigerator while maintaining a constant compressor discharge pressure.

The second method for refrigeration control maintains a constant high pressure by monitoring the level of a low pressure gas holder (gas holder control). The gas holder control method requires slight modifications to the schematic shown in Figure 2. Control PCV-3 may be omitted and the ballast volume defined as a piston type gas holder. The use of a gas holder is advantageous for several reasons. The application of a gas holder introduces a time delay between an operational error and uncontrolled venting. The gas holder may also be used for a gas storage volume in refrigeration systems having small evaporator liquid capacities and be used to reduce compressor surge problems. A variation in the refrigeration demand results in a gas holder level displacement. The displacement is translated into the control signal for the expansion valve control, adjusting the refrigeration hydrogen flow rate as in the high pressure control system previously described. The expansion valve adjustment returns the gas holder level to the reference point and thus maintains constant operating pressures.

A third method of controlling the quantity of refrigeration regulates both the flow rate and the high pressure before expansion by maintaining a fixed expansion area (constant aperture control). Constant aperture control is possible with the arrangement shown schematically in Figure 2. The expansion valve is manually set to maintain the design operating pressure (1800 psi) at a full refrigeration demand. Further expansion valve adjustment is not required. A decrease in refrigeration demand tends to lower the compressor intake pressure, but since this pressure is to be kept constant, high pressure flow is bypassed. The division of flow results in less hydrogen to the refrigerator and a reduction in the discharge pressure since the expansion valve has a constant aperture. However, the reduction in pressure is not excessive since it is not proportional to the reduction in refrigeration demand. In fact, operation of the unit with the constant aperture control has indicated a discharge pressure reduction from 1800 psi to 1000 psi for a change in refrigeration demand from 300 watts to approximately 35 watts. An advantage of the system is that an expansion valve controller is no longer required. However, an additional pressure regulator is necessary to maintain design operating pressure in the adsorbers to prevent excessive velocity and a reduction in contact time.

HEAT EXCHANGER CONSIDERATIONS

The use of the Collins type heat exchanger tube establishes the gas phase heat exchanger configuration. Therefore, the design resolves into determining the required heat exchanger lengths and the associated pressure drops. The liquid-gas heat exchanger located in the precoolant tank consists simply of a coil of copper tubing. The heat transfer and pressure drop correlations of Trumpler and Dodge[9] were used in the design of the gas phase heat exchanger. These correlations were considered applicable since the present heat exchanger tubes were geometrically similar. The heat exchanger design procedures are presented in the appendix.

The effect of the temperature difference of the two fluids flowing throughout the length of the heat exchanger must be considered in the design. The location within an exchanger where the temperature of the inlet and return streams have a minimum temperature difference may be defined as a "pinch" location. This pinch condition usually occurs at the warm end of a counter-flow heat exchanger using fluids with constant specific heats. However, a variation of the specific heat with temperature may cause the pinch location to occur within the heat exchanger. This internal pinch condition occurs in some instances in counter-flow exchangers using hydrogen gas.

The pinch condition may be illustrated by plotting the temperature in the exchanger as a function of the total heat transferred. Such plots are usually called cooling curves. A typical cooling curve is shown in Figure 8. It is apparent by inspection that the log mean

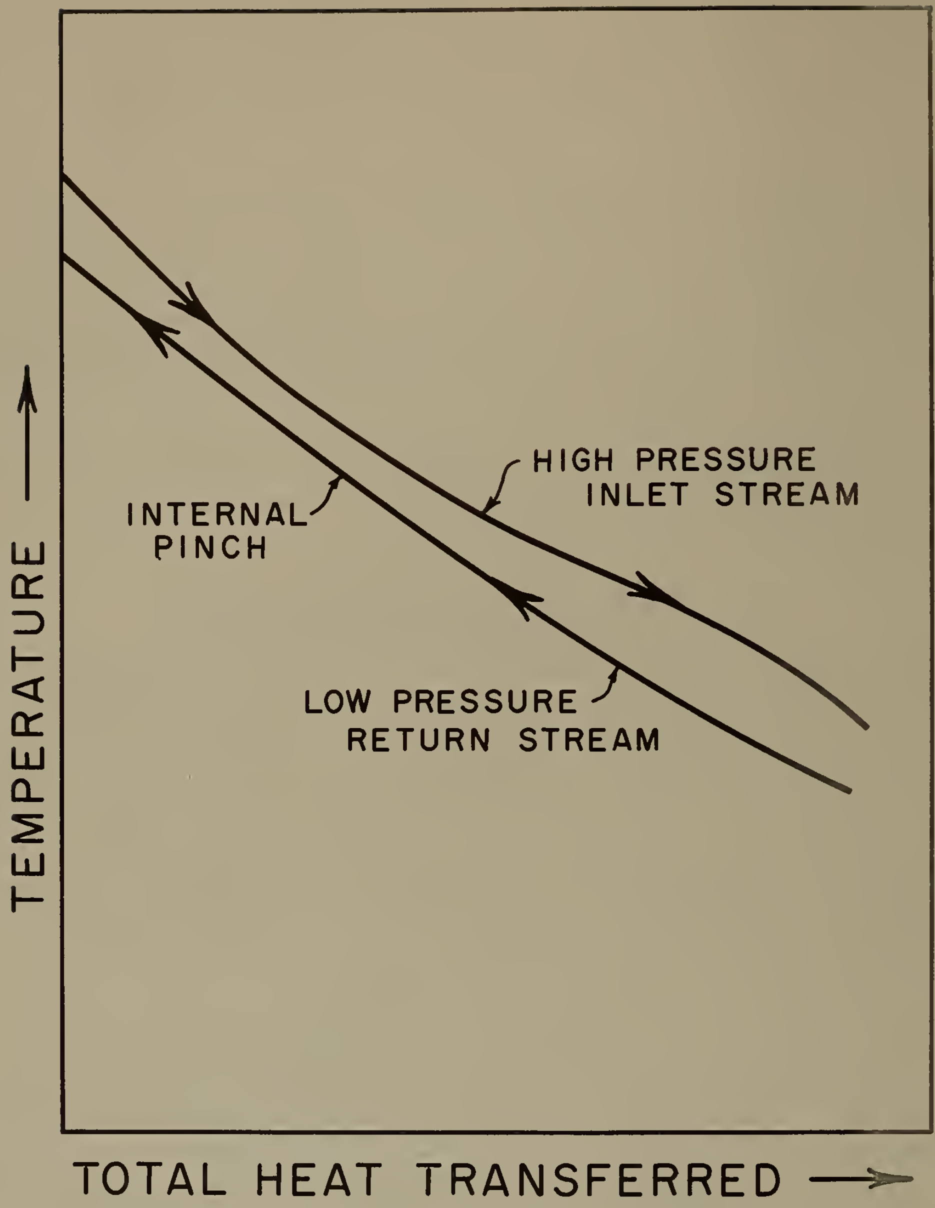


Figure 8. Typical Cooling Curve with Internal Pinch

temperature difference (LMTD), for use in the conventional heat transfer equations, cannot be applied over the entire length of the heat exchanger by considering only end point conditions. The LMTD can only be correctly applied by performing a number of stepwise calculations. The number of steps must be determined by examining the shape of the cooling curves and subdividing it into a series of straight sections. Average thermal properties of the fluids are used in each section.

Heat exchanger efficiency has a pronounced effect on the performance of a refrigeration system. Theoretical refrigeration, based on zero degree temperature difference at the warm end of the final exchanger,is a useful reference in the initial selection of the characteristics of a given system. Figure 9 indicates the theoretical refrigeration available. However, calculations indicate the quantity of refrigeration is reduced because of finite temperature differences by 5 percent per degree K with 65^{o}K precooling and 6.7 percent per degree K for 78^{o}K precooling.

The nominal size of 300 watts selected for the refrigerator was based on the dynamic load of the bubble chamber with sufficient additional capacity to facilitate the cool-down of the system and to condense liquid into the bubble chamber. The quantity of refrigeration resulted in the choice of a standard compressor having a capacity of 35 scfm (1 atm, 20^{o}C) of hydrogen at 1800 psig, with a precoolant temperature of 65^{o}K and reasonable heat exchanger efficiency. This flow rate results in a theoretical refrigeration capacity of about 260 watts when the precoolant bath is operated at atmospheric conditions (see Figure 9).

Heat exchanger efficiency also influences the liquid nitrogen required as precoolant for a given system. Figures 10 and 11 show the liquid nitrogen consumption for various temperature differences at the warm end of the H_2 - H_2 - N_2 heat exchanger. It is assumed that the nitrogen is supplied to the unit as saturated liquid at 2 atmospheres. Finite temperature differences increase the precoolant consumption by 4-5 percent per degree K.

REFRIGERATOR PERFORMANCE

The actual refrigerator performance was determined for a number of operating conditions in an effort to verify the method of calculation. A summary of experimental evidence for the design operating condition is compared to design predictions in Table I. The refrigeration efficiency found in the experiments, considering only the region below the precoolant, indicates 97.4 percent of the design

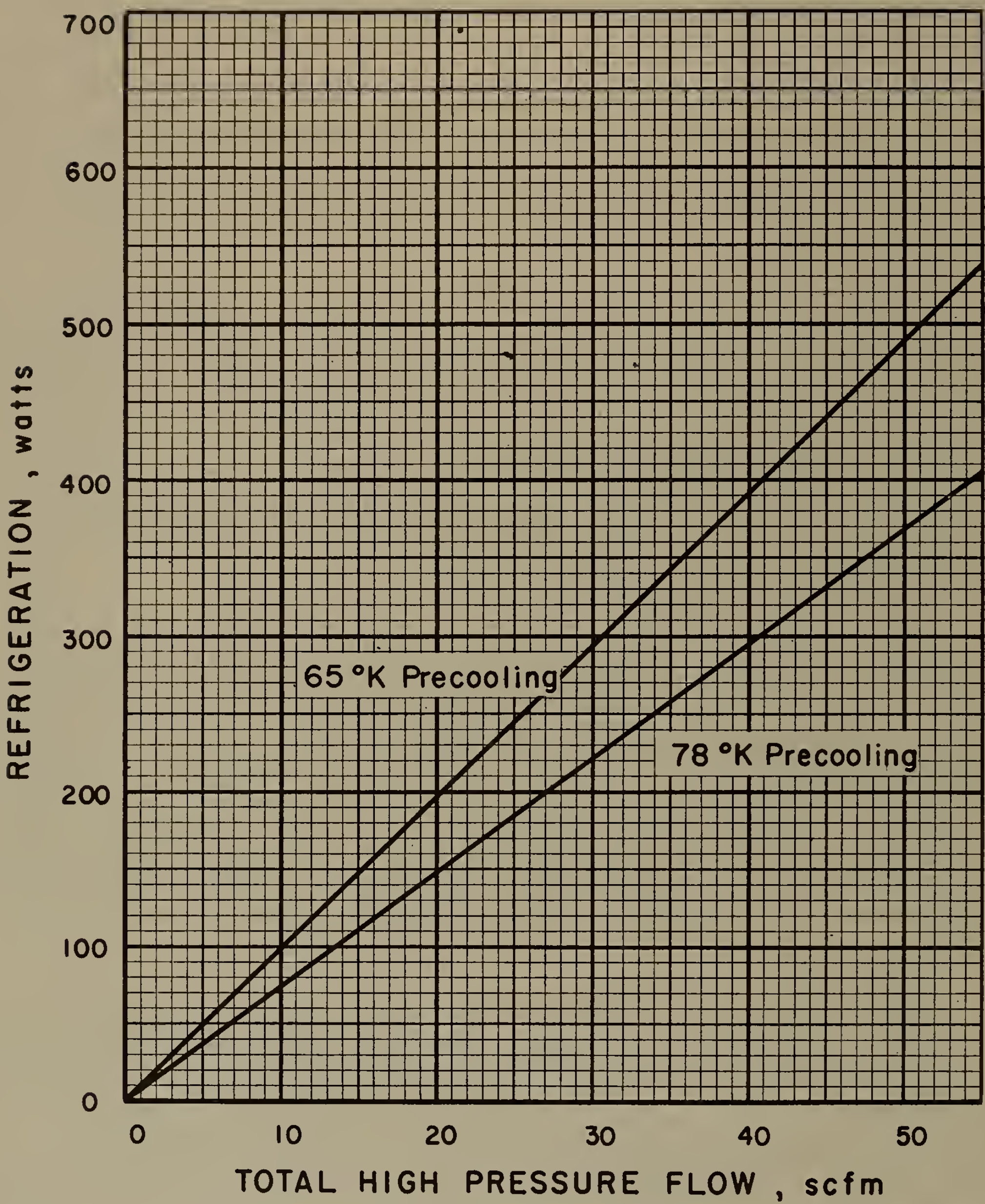


Figure 9. Theoretical Refrigeration

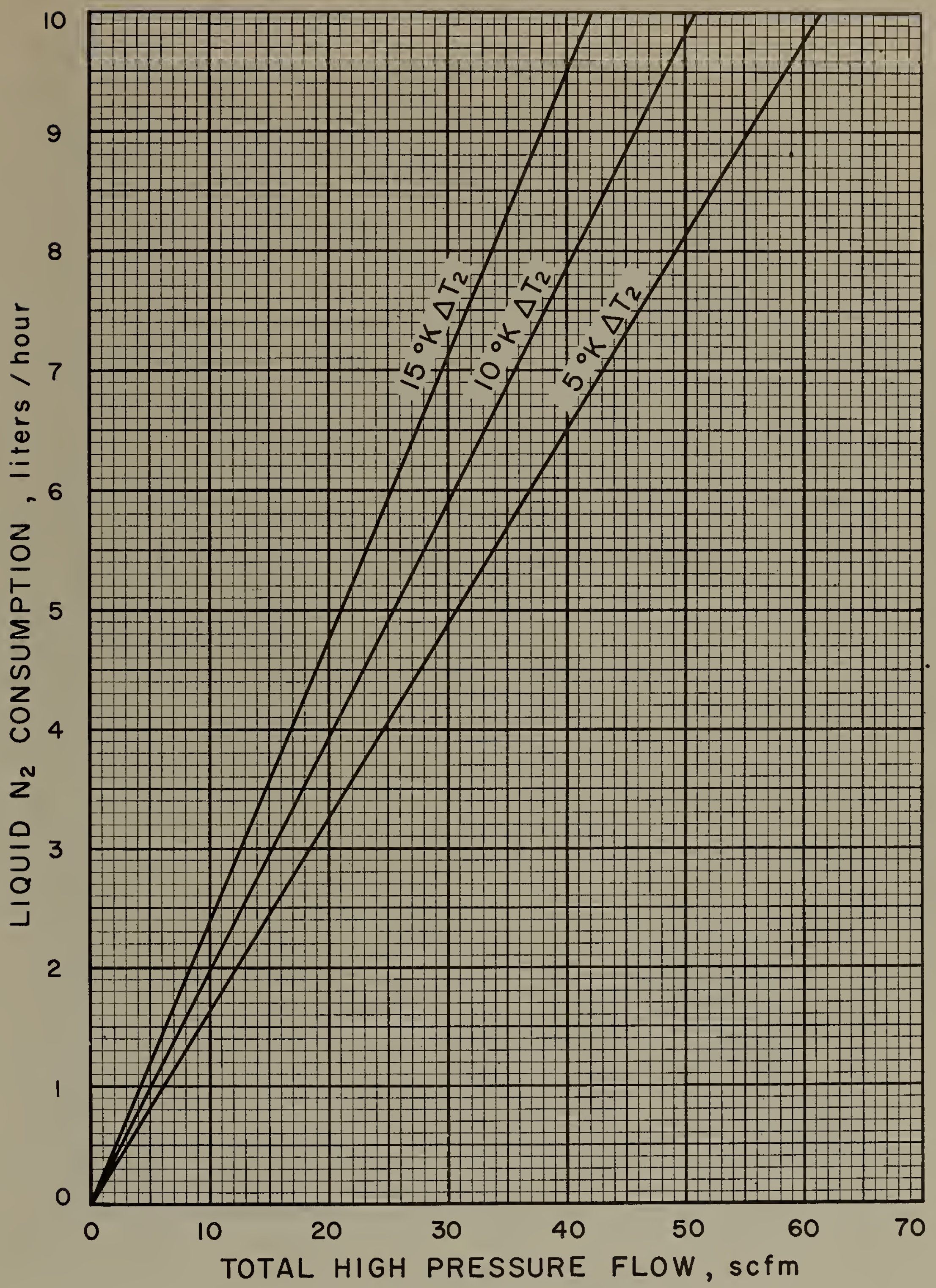


Figure 10. Liquid N_2 Consumption for 65°K Precooled Refrigerator

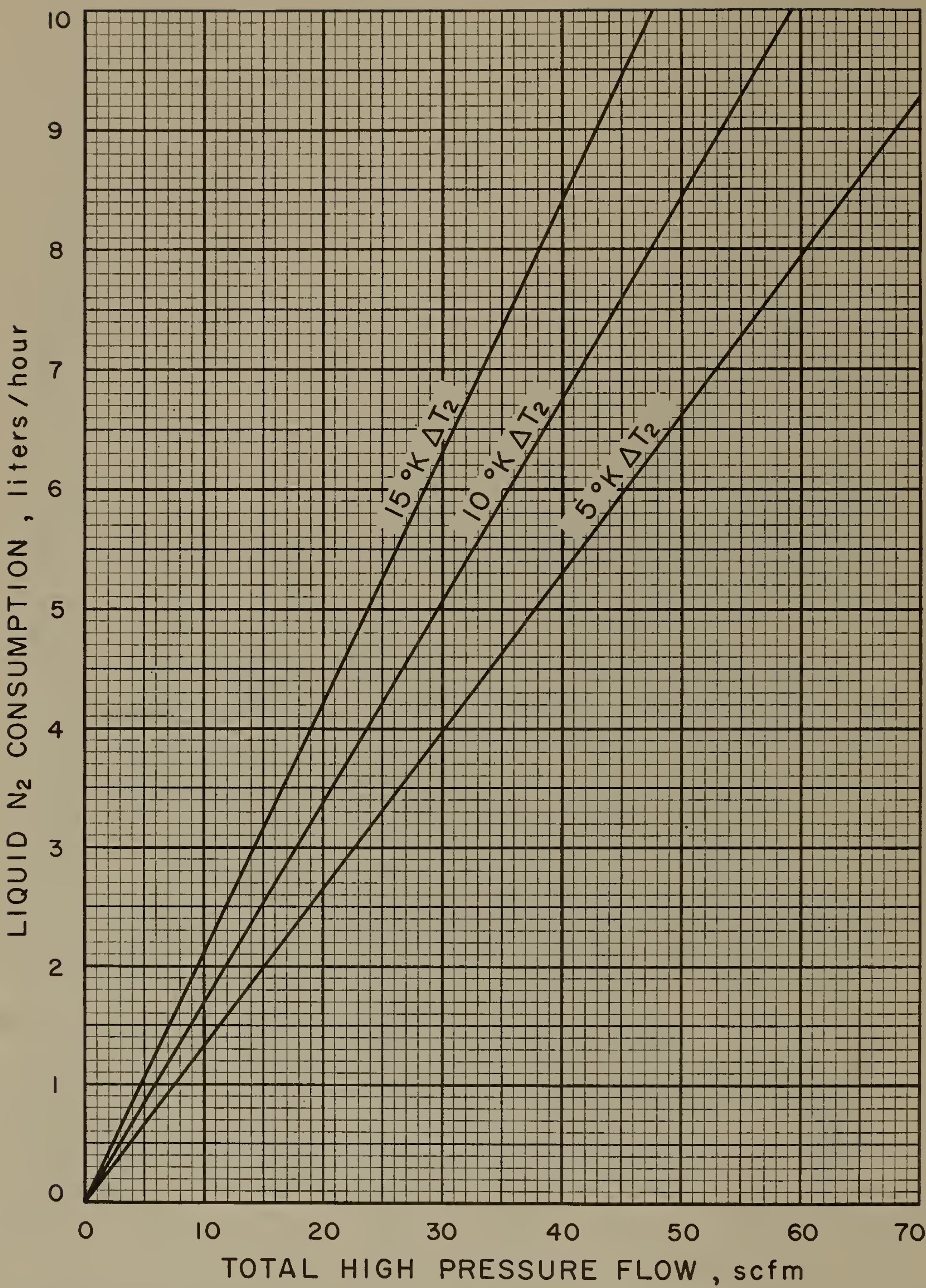


Figure 11. Liquid N_2 Consumption for 78°K Precooled Refrigerator

TABLE I

REFRIGERATOR PERFORMANCE

	Design	Experimental
Flow - scfm	35	35
Operating pressure - psig	1800	1800
Precoolant temperature - ^{o}K	65	65
Total refrigeration - watts	338	329
Refrigeration efficiency		
Percent of design	--	97.4
Percent of theoretical	98.4	96.0
Liquid nitrogen consumption - liters/hr	8.1	8.3
Temperature differences		
(warm end) - ^{o}K		
H_2 - H_2 - N_2 exchanger	14	14
Final exchanger	0.3	0.8
Low pressure H_2 pressure drop - psi		
H_2 - H_2 - N_2 exchanger	5.2	4.0
Final exchanger	0.55	0.51

prediction. The total refrigeration produced by the refrigerator must take into consideration the static heat influx to the system. This, of course, reduces the refrigeration available at the evaporator. Experiments indicate a static heat influx of 35 watts. This heat leak includes the static load on the evaporator and the 14 foot transfer line with appropriate fittings and connections. The large portion of the static heat leak is attributed to the evaporator used for the performance test since no effort was made to optimize the cryogenic design.

The comparisons shown in Table I indicate good agreement between design prediction and experimental results and justifies the design procedure used. The performance of the critical final heat exchanger results in a heat exchanger efficiency, based on temperatures, of 98.3 percent. This heat exchanger efficiency results in a favorable refrigerator performance.

The design equation used to predict pressure drop within the heat exchangers is known to be valid within ± 20 percent for a specified range of Reynolds number. The pressure drop comparisons shown in Table I are within the stated tolerance. However, the nitrogen flow in the H_2 - H_2 - N_2 heat exchanger is insufficient to create a Reynolds number in the turbulent region ($Re < 550$). As would therefore be expected, the pressure drop predictions in the nitrogen passage were found to be considerably below experimental results. In fact, at design flow rate, the pressure drop was found to be 2.5 times the pressure drop predicted by the design equation. The ratio of experimental pressure drop to predicted pressure drop increases at lower flow rates.

The increased nitrogen pressure drop must be considered in the attainment of the appropriate precooling temperatures at the desired flow rate. Refrigerators having hydrogen flow rates below 35 scfm are not materially affected by the increased nitrogen pressure drop since appropriate precooling temperatures can still be obtained. The design flow of 35 scfm appears to be the threshold for the attainment of 65 $^{\circ}$K in the precoolant tank using a single three way heat exchanger. Flow rates above this threshold require division of the low pressure nitrogen gas into multiple three way heat exchangers. The design charts for heat exchanger determination may be used by selecting the proper heat exchanger flow rate. For example, a total flow of 60 scfm would be divided into two parallel heat exchangers of 30 scfm each. The heat exchanger characteristics would therefore be selected from the design charts at 30 scfm.

The thermodynamic properties of hydrogen presented by Woolley, Scott, and Brickwedde[11] have been used in the design of the refrigerator and in the determination of the refrigerator performance.

BIBLIOGRAPHY

1. R. L. Blumberg, J. D. Gow and A. J. Schwemin "The Development and Operation of the Ten-Inch Liquid Hydrogen Bubble Chamber," Proceedings of 1956 Cryogenic Engineering Conference, K. D. Timmerhaus, Editor, University of Colorado, Boulder, Colorado, Paper H-1 pp. 318-324 (1956)

2. B. W. Birmingham, D. B. Chelton, D. B. Mann, and H. P. Hernandez, "Cryogenic Engineering of Hydrogen Bubble Chambers," ASTM Bulletin, No. 240, pp. 34-39 (1959)

3. B. W. Birmingham, "A Mechanical Refrigeration Process for the No-Loss Storage of Liquid Hydrogen," Refrigerating Engineering. Vol. 65, No. 7, pp. 42-44, 86 (1957)

4. D. B. Chelton, J. W. Dean and T. R. Strobridge (to be published)

5. D. B. Chelton, J. Macinko and J. Dean, "Hydrogen Liquefaction by a Dual Pressure Process," Refrigerating Engineering, Vol. 65, No. 8, pp. 39-41, 62 (1957)

6. D. B. Chelton, J. Macinko and J. W. Dean, "Methods of Hydrogen Liquefaction," NBS Report 5520 (1957) (Unpublished)

7. A. G. Prodell and D. C. Colley, Columbia University, New York (private communication)

8. G. J. Eckman, University of California, Lawrence Radiation Laboratory, Berkeley, California (private communication)

9. P. R. Trumpler and B. F. Dodge, "The Design of Ribbon-Packed Exchangers for Low Temperature Air Separation Plants," Transactions of the American Institute Chemical Engineers, Vol. 42, No. 2, pp. 75-84, (1947)

10. W. H. McAdams, "Heat Transmission," McGraw Hill, New York, Second Edition, pp. 232, (1942)

11. H. W. Woolley, R. B. Scott, and F. G. Brickwedde "Compilation of the Thermal Properties of Hydrogen in Its Various Isotopic and Ortho-Para Modifications," Journal of Research National Bureau of Standards, 41, 379 (1948) RP 1932

APPENDIX - HEAT EXCHANGER DESIGN

The physical properties of the Collins type heat exchanger tubes used in the calculations of the gas phase heat exchangers are presented in Figure 12. The gas passages (tube annuli) are packed with a spiral wound ribbon to introduce a large number of fin surfaces in the gas streams. These fins are soldered to the tube walls to assure good thermal contact. The active surface area of each gas passage is the total fin and tube wall surface. The free cross sectional area of a passage is equivalent to the annular area less the area obstructed by the fins. The tabulated values in Figure 12 were calculated from the tube geometry.

Trumpler and Dodge[9] have determined the heat transfer correlation for the Collins type heat exchanger tubes. The standard dimensionless relationship for heat transfer in fluids flowing without phase change in turbulent motion through heated or cooled tubes is used.

$$Nu = m\,(Re)^{n}\,(Pr)^{1/3}$$

where Nu = Nusselt number

Re = Reynolds number

Pr = Prandtl number

The coefficient, m, was determined to be 0.118 and the exponent, n, to be 0.7. Experimental evidence shows that these values are applicable for Reynolds numbers ranging from 550 to 6000.

The heat transfer coefficients determined by the solution of the heat transfer correlation is subject to modification by the application of a fin efficiency term. The fin efficiency was calculated as described by McAdams[10]

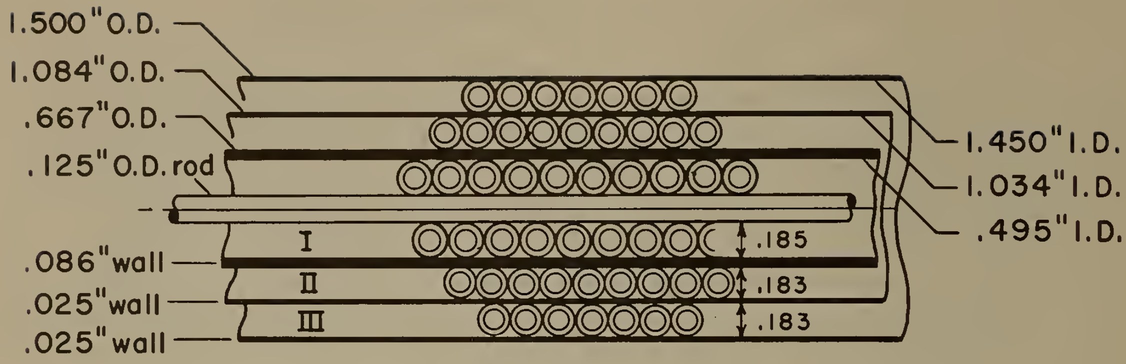
$$N = \frac{\tanh ax_f}{ax_f}$$

where x_f = fin length

$$a = \frac{h_f b_f}{K\,S_f}$$

h_f = calculated film coefficient

b_f = exposed perimeter of fin



Thickness of Ribbon Wall = .018
Width of Ribbon = .052

ANNULUS	NO. of FINS PER TURN	TOTAL ACTIVE SURFACE AREA ft^2/ft (b)	NO. of TURNS PER INCH	FREE AREA of CROSS SECTION ft^2 (S)	$D = \frac{4S}{b}$ ft
I	22	.66	5	.00090	.00548
II	42	1.4	5	.00277	.00792
III	64	2.1	5	.00402	.00765

Figure 12. Heat Exchanger Geometry

S_f = cross sectional area of fin

K = thermal conductivity

The result of these calculations is presented in Figure 13. Modified heat transfer coefficients were used at each chosen temperature level for the calculation of the heat exchanger length. A summary of the results of these calculations are graphically represented in Figures 14, 15, 16 and 17.

The heat exchanger pressure drop was calculated with the aid of the correlation presented by Trumpler and Dodge.

$$\frac{\Delta P}{L} = \frac{0.005185 f\, G_s^{\,2}}{\rho\, De'}$$

G_s = mass velocity, - #/hr - ft^2

ρ = density - #/ft^3

De' = equivalent diameter -in

f = Fanning friction factor

ΔP = pressure drop -psi

L = length -ft

The density has been evaluated at the arithmetic mean temperature determined from the heat exchanger end point conditions. The Fanning friction factor was evaluated from the equation[9]

$$f = 0.476\,(Re)^{-0.2}$$

These equations may also be applied for Reynolds numbers ranging from 550 to 6000 as in the heat transfer equations. The calculated pressure drop is represented in Figures 18 and 19.

The pressure drop for the nitrogen passage is omitted since it was found that the Reynolds number of the evaporated nitrogen gas was lower than the recommended value. Therefore the pressure drop equation was not applicable. A method of estimating the pressure drop in the nitrogen passages is given in the section on Refrigerator Performance. The resulting pressure drop is high enough to limit the attainment of low precooling temperatures by pressure reduction. This situation may be remedied by a rearrangement of the heat exchanger above the precooling temperature level. The high pressure flow may be divided into two equal streams and directed through two parallel three way heat exchangers. After passing through the heat exchangers the flow would be rejoined before entering the gas-liquid heat exchanger in the precoolant tank. The multiple heat exchangers result in reducing the nitrogen pressure drop, making the attainment of a low precoolant temperature possible. However, the flow in the heat exchangers must be monitored and adjusted accordingly. A hydrogen flow rate of 35 scfm proved to be the threshold at which the attainment of a $65^{\circ}K$ precoolant temperature is possible. Flow rates greater than 35 scfm require the use of multiple three-way heat exchangers.

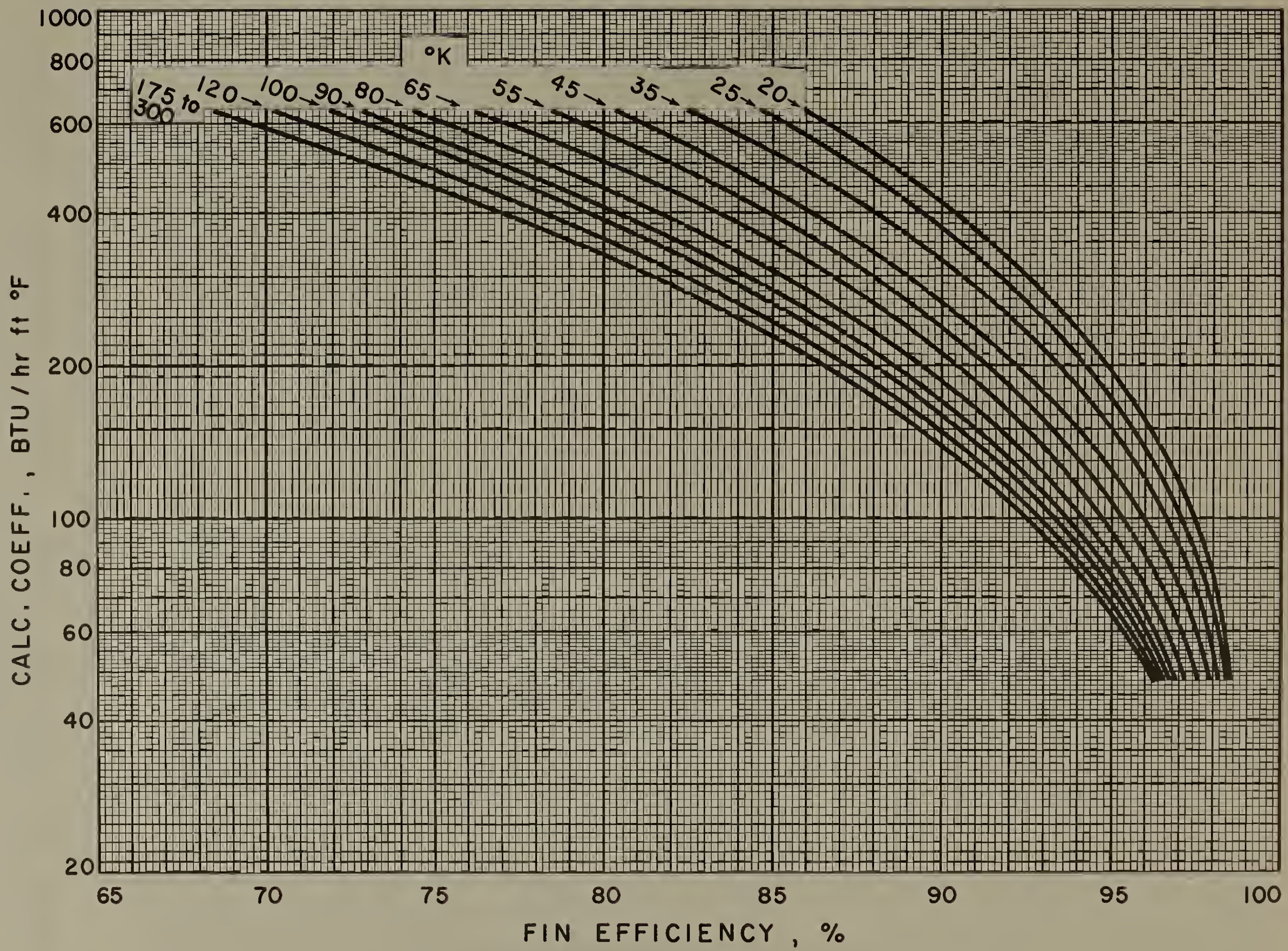


Figure 13. Heat Exchanger Fin Efficiency

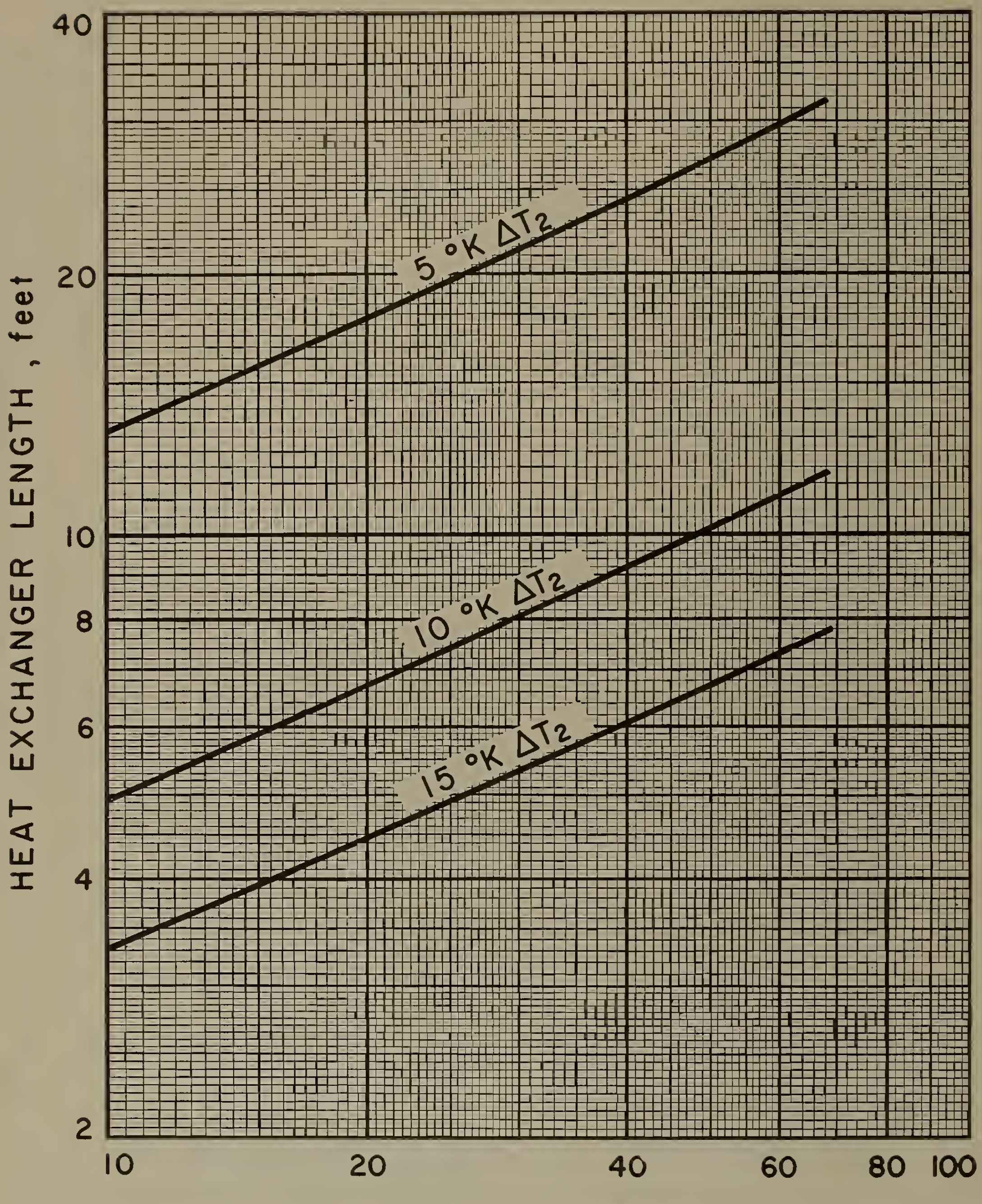


Figure 14. H_2 —H_2 —N_2 Heat Exchanger Length for 65°K Precooled Refrigerator

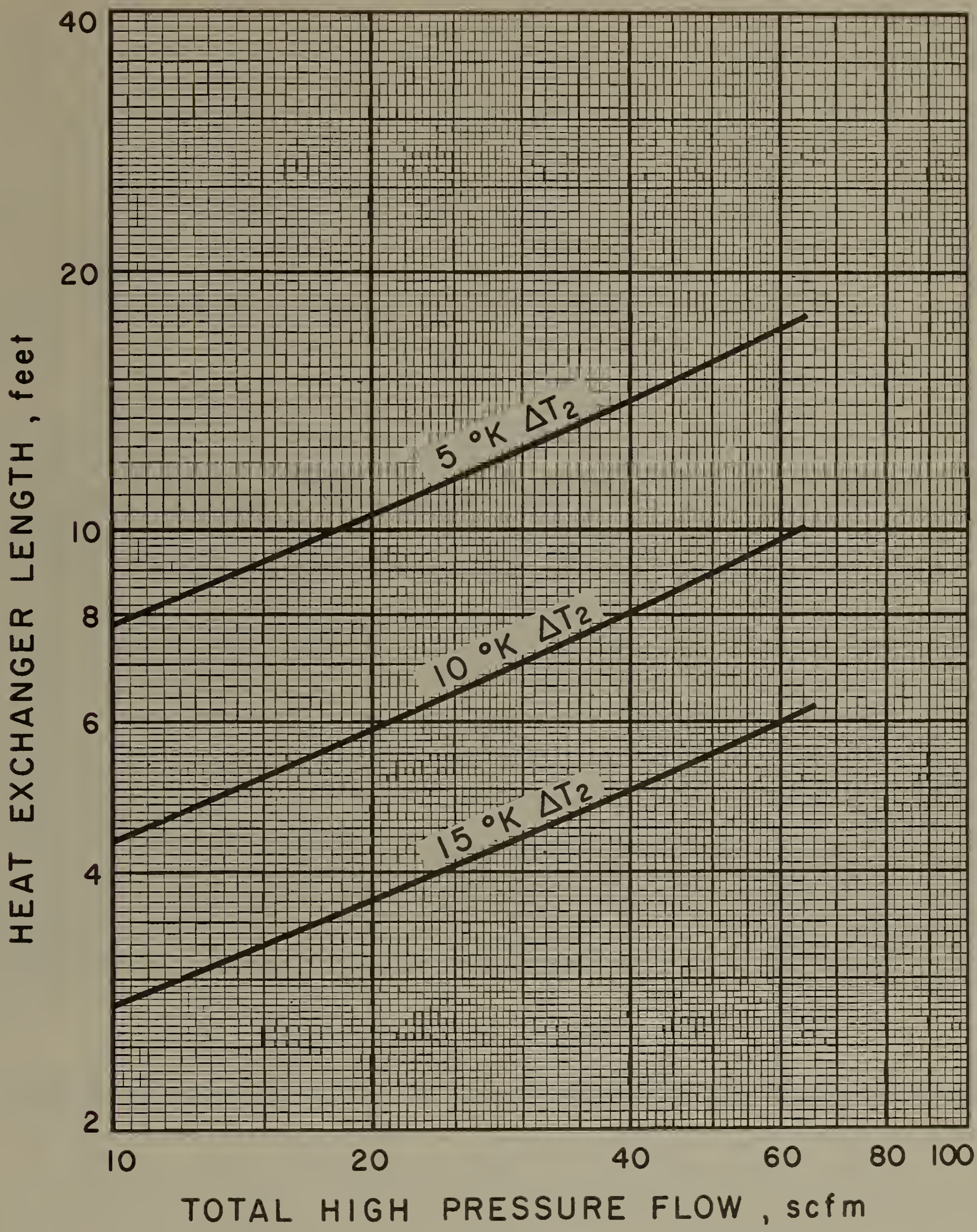


Figure 15. $H_2-H_2-N_2$ Heat Exchanger Length for 78°K Precooled Refrigerator

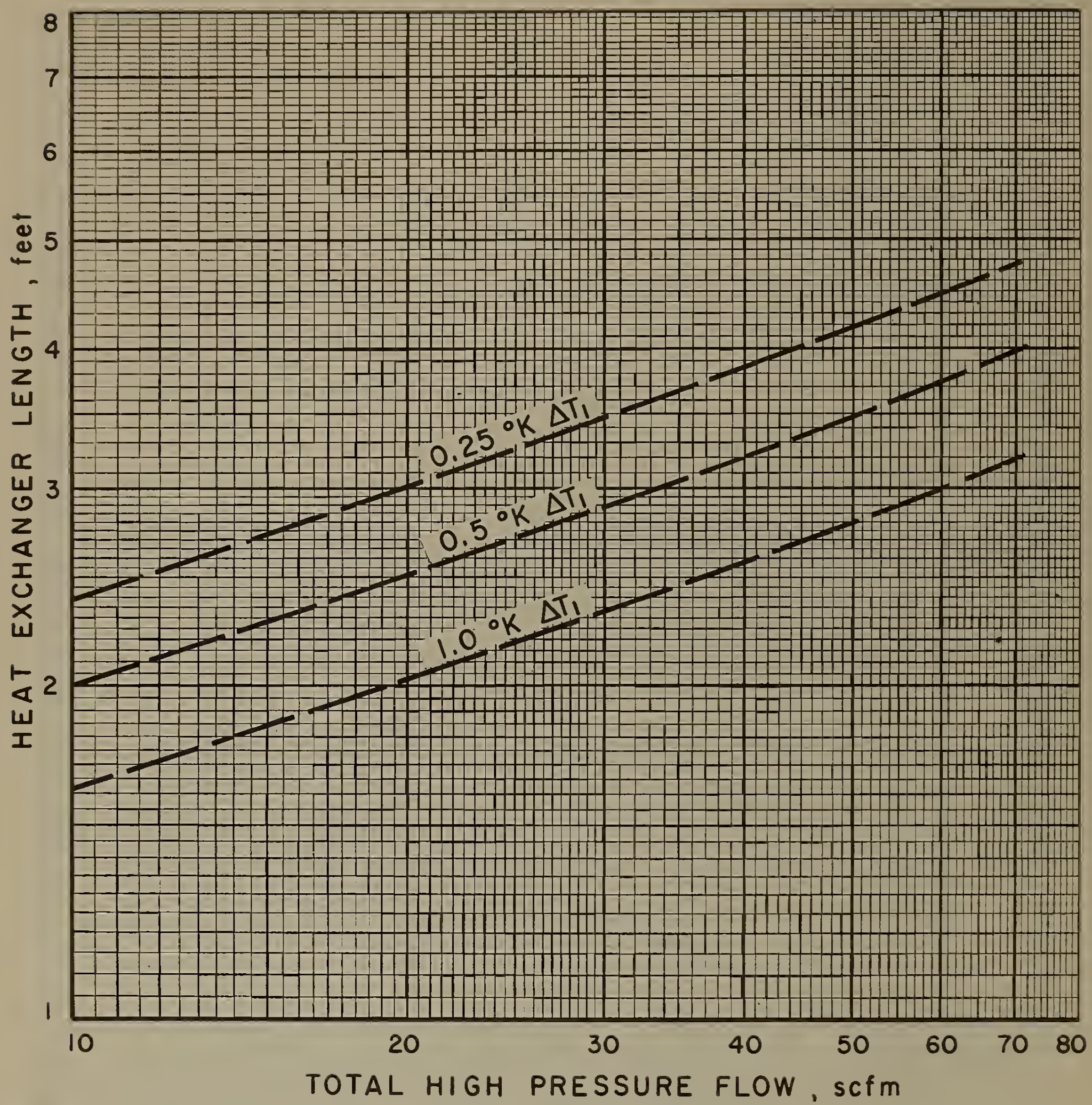


Figure 16. Final Heat Exchanger Length for 65°K Precooled Refrigerator

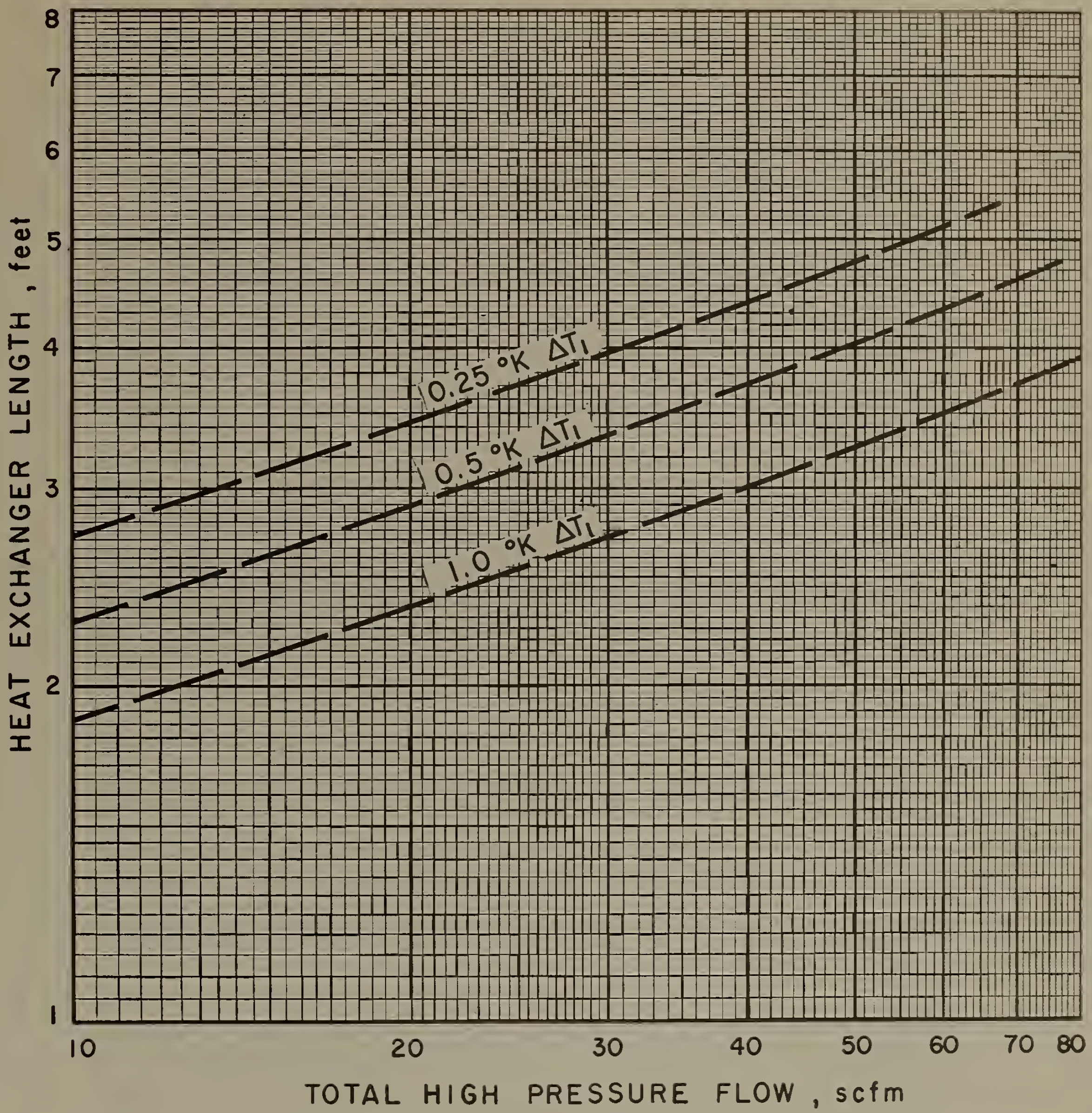


Figure 17. Final Heat Exchanger Length for 78° K Precooled Refrigerator

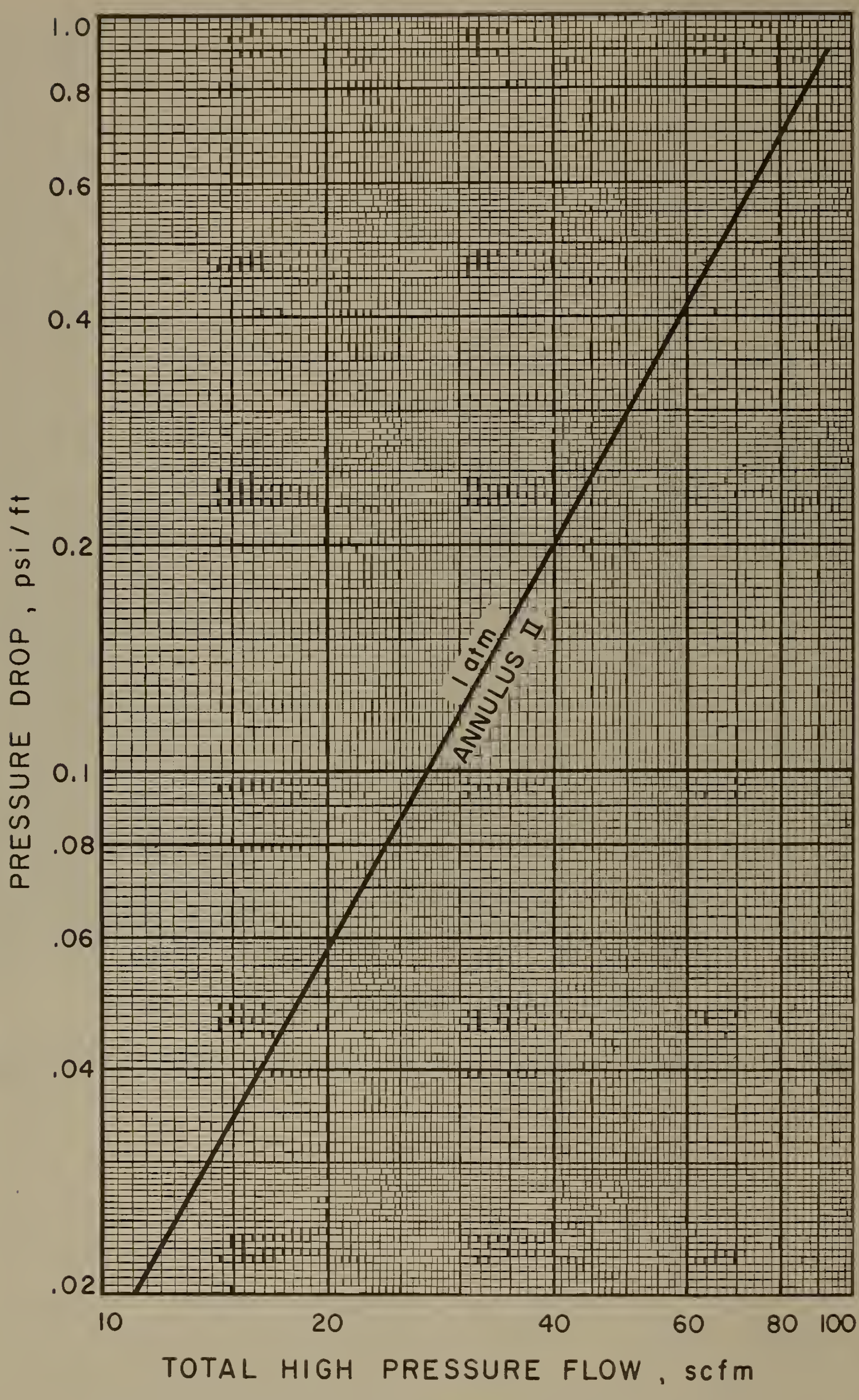


Figure 18. Final Heat Exchanger Pressure Drop

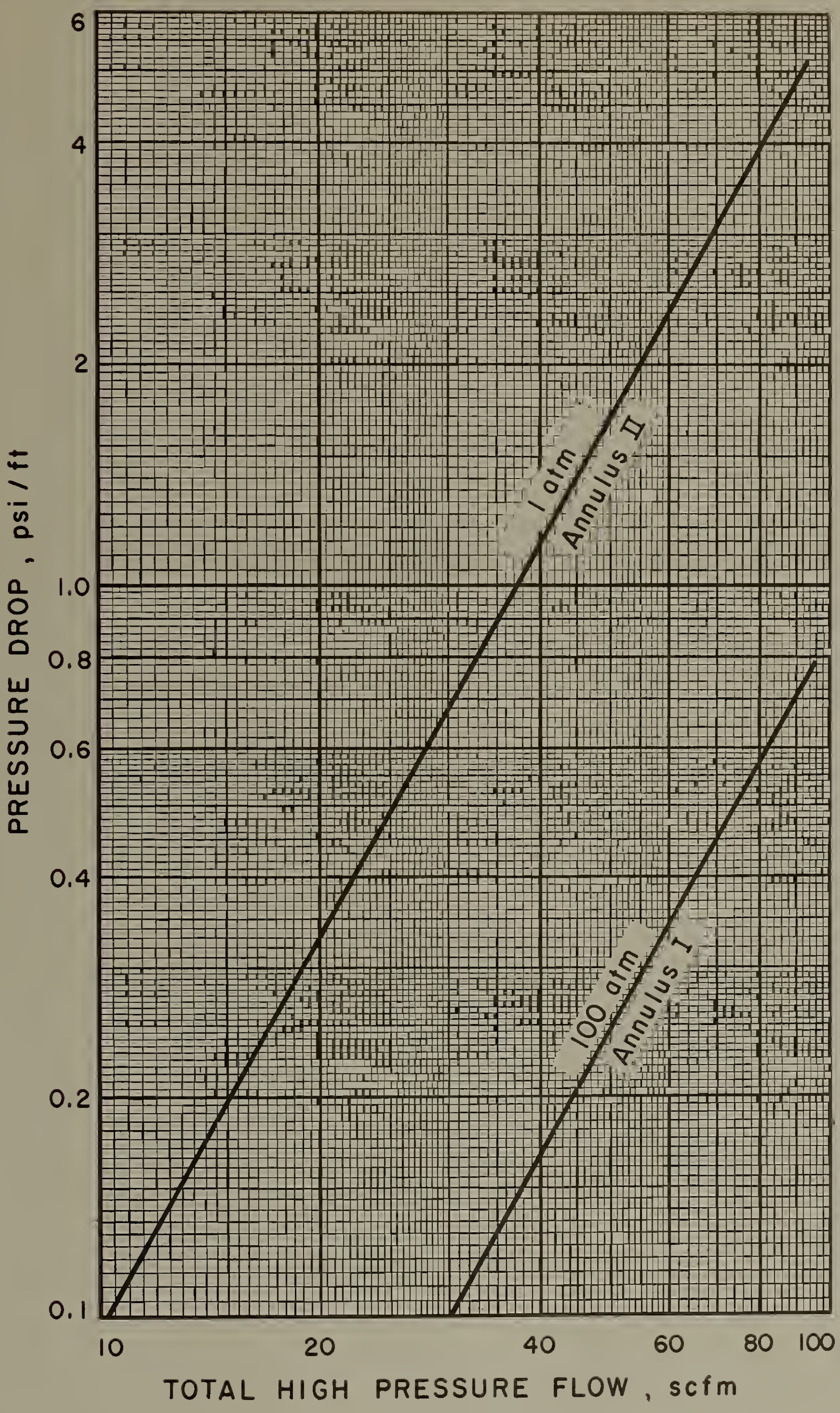


Figure 19. $H_2-H_2-N_2$ Heat Exchanger H_2 Pressure Drop

U. S. DEPARTMENT OF COMMERCE
Frederick H. Mueller, Secretary

NATIONAL BUREAU OF STANDARDS
A. V. Astin, Director

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Organic and Fibrous Materials. Rubber. Textiles. Paper. Leather. Testing and Specifications. Polymer Structure. Plastics. Dental Research.

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Mineral Products. Engineering Ceramics. Glass. Refractories. Enameled Metals. Constitution and Microstructure.

Building Technology. Structural Engineering. Fire Protection. Air Conditioning, Heating, and Refrigeration. Floor, Roof, and Wall Coverings. Codes and Safety Standards. Heat Transfer. Concreting Materials.

Applied Mathematics. Numerical Analysis. Computation. Statistical Engineering. Mathematical Physics.

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Radio Propagation Engineering. Data Reduction Instrumentation. Modulation Research. Radio Noise. Tropospheric Measurements. Tropospheric Analysis. Propagation Obstacles Engineering. Radio-Meteorology. Lower Atmosphere Physics.

Radio Standards. High Frequency Electrical Standards. Radio Broadcast Service. High Frequency Impedance Standards. Electronic Calibration Center. Microwave Physics. Microwave Circuit Standards.

Radio Communication and Systems. Low Frequency and Very Low Frequency Research. High Frequency and Very High Frequency Research. Ultra High Frequency and Super High Frequency Research. Modulation Research. Antenna Research. Navigation Systems. Systems Analysis. Field Operations.

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