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HELIUM REFRIGERATION AND LIQUEFACTION USING A LIQUID HYDROGEN REFRIGERATOR FOR PRECOOLING

by

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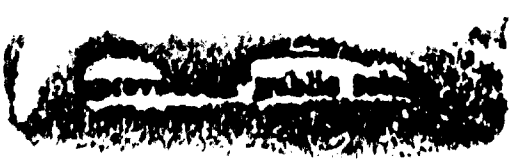

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Helium Refrigeration and Liquefaction Using a Liquid Hydrogen Refrigerator for Precooling

by

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ABSTRACT

Consideration is given to the use of a hydrogen refrigerator to assist in the production of temperatures below those obtained with hydrogen. A hydrogen refrigerator is used to maintain a precooling evaporator at or near 21°K in a helium gas circuit. The helium circuit may be arranged to produce liquid for external use or to produce refrigeration between 21°K and 4.2°K in a closed system.

Charts have been developed that show the requirements of the composite helium - hydrogen system and the effect of heat exchanger performance. The relative quantities of refrigeration (hydrogen and helium) at various temperature levels have been determined.

INTRODUCTION

At the present time, a number of hydrogen refrigerators and liquefiers are being used to maintain experimental apparatus at low temperatures. For example, several 21-27°K hydrogen refrigerators are used for continuous refrigeration of liquid hydrogen bubble chambers[1,2]. Refrigeration at this temperature level is also being planned for experimental apparatus associated with nuclear reactors and electromagnets. Certain applications may require temperature below those obtainable with hydrogen.

Hydrogen refrigerators can be used for the production of lower temperatures by using a hydrogen evaporator to precool a stream of helium gas in an additional circuit. In some instances, it may be feasible to convert a hydrogen liquefier to perform the function of a refrigerator for this precooling.

This paper discusses a relatively simple method of using a hydrogen refrigerator to assist in the production of temperatures from 21°K to liquid helium temperature. With proper considerations it should be possible to operate the hydrogen precooled helium system at any temperature within these limits. Consideration in the text is also given to the production of liquid helium for external use. Cycle information is presented with text only sufficient to define the particular conditions or assumptions upon which the calculation is based.

GENERAL DISCUSSION

A schematic arrangement of the composite helium-hydrogen low temperature apparatus is shown in Figure 1. Symbols indicated on the schematic are used throughout the text. The automatic control of such a unit has been discussed elsewhere[2] and will not be considered here. Both the helium and the hydrogen circuits use liquid nitrogen precooling at 65°K, although 78°K is considered in a few instances. The liquid nitrogen required as precoolant is given in liters per hour since, for most applications, it will be supplied as liquid from an external source. It is further assumed that the nitrogen will be supplied to the unit as saturated liquid at 2 atmospheres.

The hydrogen gas is further cooled, after liquid nitrogen precooling, in a final heat exchanger and expanded at the hydrogen throttle valve to produce liquid in the evaporator. The evaporator can be a reservoir as shown, or a bundle of heat exchanger tubes. Hydrogen evaporator temperatures from 21°K (1.23 atm) to 15°K (0.13 atm) are studied.

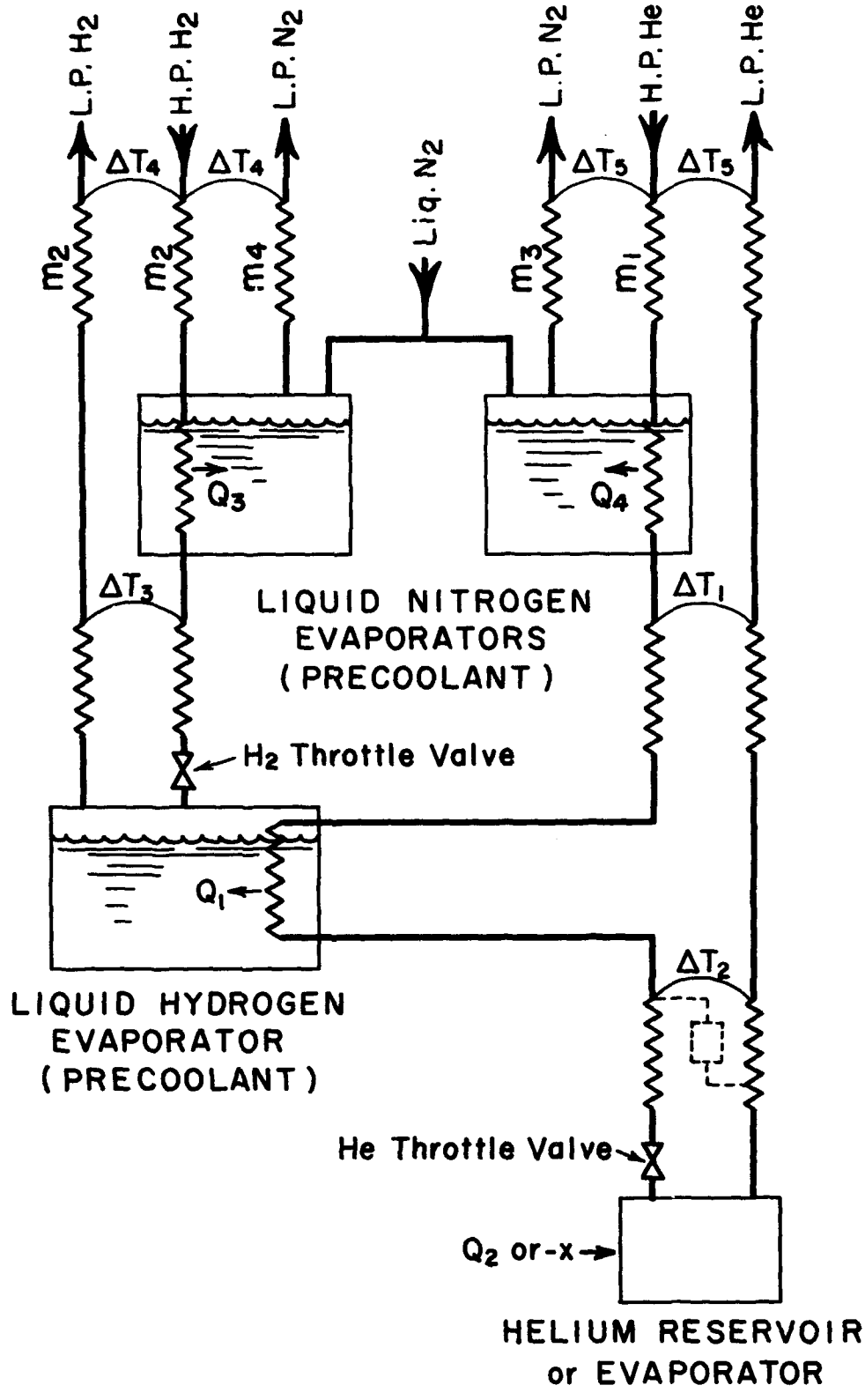

FIGURE I. SCHEMATIC ARRANGEMENT OF LOW TEMPERATURE CYCLE

The helium gas, after liquid nitrogen precooling, is passed through a counterflow heat exchanger and further cooled in the liquid hydrogen evaporator. The temperature of the gas is still further reduced in a final heat exchanger and expanded at the helium throttle valve where liquid can be formed. Liquid formed in the expansion process can be evaporated in the reservoir, as in a refrigerator, or can be extracted from the reservoir, as in a helium liquefier. If refrigeration is required at temperatures above liquid helium, the final heat exchanger is partially by-passed. The quantity of by-passed gas determines the operating temperature of the refrigerator.

If a hydrogen refrigerator is available, the method described above is a very simple way of producing temperatures lower than that obtainable with hydrogen. An alternate method for the final stage cooling of the helium gas stream would utilize an expansion engine as indicated by the dotted lines of Figure 1. The expander inlet temperature could range from the temperature of the hydrogen evaporator to some lower temperature. This method requires the use of an expansion engine operating at low temperatures, but it substantially increases the thermodynamic efficiency of the cycle. Additional improvement in the thermodynamic efficiency of the cycle can be obtained by replacing the helium throttle valve with an expansion engine. Use of such an expander is limited to temperatures above that of liquid helium if condensation in the expander is to be avoided. The performance of this alternate cycle must be computed for the refrigeration temperature desired. These more extensive calculations have not been considered in the present paper.

Small capacity refrigeration systems do not necessarily require a high thermodynamic efficiency. Often the simplest cycle is desirable for reliability and ease of operation. For any cycle, however, it is desirable to produce refrigeration with a minimum expenditure of energy, since the cost of gas compression equipment materially affects the total capital investment. Compression equipment may account for 20-40 percent of the total system cost.

Sufficient information is presented later to determine the performance of a practical helium refrigeration or liquefaction system based on the cycle indicated by the solid lines in Figure 1. A convenient reference is the theoretical performance based on a heat exchanger efficiency of 100 percent. For the final heat exchangers in the cycle under consideration, this corresponds to $\Delta T_3 = \Delta T_2 = 0$. The design parameters considered are:

1) heat exchanger efficiency (presented as functions of ΔT_1 through ΔT_5)

2) liquid nitrogen consumption

3) mass flow

4) power required for compression

The power required reflects only that necessary for compression of the gas in each circuit. A 65 percent isothermal compression efficiency is used for the calculations. With the aid of the design charts presented, it is possible to determine the quantity of refrigeration (Q_1) required at liquid hydrogen temperature for a given quantity of refrigeration (Q_2) at lower temperatures or for a given quantity of helium liquefied (x).

All helium calculations are based on the thermodynamic properties presented by Mann and Stewart[3], Keesom[4] and Zelmanov[5]. The hydrogen calculations are based on the properties given by Woolley, Scott and Brickwedde[6]. The results of the calculations have been smoothed where necessary since they require greater accuracy than obtainable from the above properties.

HYDROGEN REFRIGERATOR

Hydrogen refrigerators usually operate from 21°K to 27°K. For efficient operation of the helium refrigerator or liquefier, it is desirable to maintain the hydrogen evaporator as low in temperature as possible. In fact, it would be desirable, on this basis, to operate the evaporator at a reduced pressure. Practical considerations usually limit the precooling temperature to about 15°K (0.132 atm), slightly above the triple point of hydrogen. Operating the hydrogen evaporator at a reduced pressure results in a substantial increase in the work of hydrogen compression although there is a considerable reduction in the work of compression for the total cycle in Figure 1. Figure 2 shows the work of hydrogen compression (kw) required per unit (kw) of refrigeration at the evaporator temperature for theoretical performance (100 percent efficiency) of the final heat exchanger. Minimum work occurs for an inlet high pressure near 120 atmospheres and this pressure has been used throughout the remainder of the calculations. The power required to compress 1 scfm (1 atm, 20°C) of hydrogen gas from 1 atmosphere to 120 atmospheres is equal to 0.37 kw. Compression from 0.132 atm requires 0.52 kw per scfm (1 scfm= 0.04 gm/sec).

The refrigeration available as a function of heat exchanger performance is shown in Figure 3. It is observed that the quantity of refrigeration is reduced because of finite temperature differences, by 5 percent per degree K for 65°K precooling and by 6.7 percent per degree K for 78°K precooling.

The liquid nitrogen required in the precooling evaporator is affected by both ΔT_3 and ΔT_4. A lower limit is imposed on the value of ΔT_4 because of the variation in the specific heat of hydrogen. This limit can be seen in a cooling curve analysis[2]. The quantity of nitrogen is shown in Figure 4 in units of liters per hour per watt of refrigeration (Q_1) at the hydrogen evaporator. Both Figure 3 and 4 are independent of the hydrogen evaporator temperature.

Since it is thermodynamically advantageous to reduce the temperature (and pressure) of a hydrogen evaporator, it is of interest to determine the effect of a reduced pressure on the performance of a hydrogen refrigerator not specifically designed for such operation. The reduction in temperature (and pressure) will require that the heat exchangers operate under conditions other than the original design. The important design parameters influenced by reduced pressure operation are the heat exchanger efficiency and the pressure drop across the heat exchangers. The heat exchanger efficiency is affected by both the quantity of heat transferred and the heat transfer coefficients within the heat exchanger.

By considering the basic equations of heat transfer, it can be shown that the heat transfer coefficients will remain approximately the same for a given flow rate regardless of pressure. The change in efficiency of a heat exchanger will therefore be determined by the additional heat removed and its effect on the log mean temperature difference. However, for the case considered this effect has been calculated and found to be insignificant. Thus, neither the change in the heat transfer coefficients nor the quantity of heat transferred has an appreciable effect on the heat exchanger efficiency.

The pressure drop across a heat exchanger, for a given flow rate, is proportional to the specific volume of the process fluid. For a given heat exchanger, the pressure drop in a 15°K hydrogen refrigerator will be approximately 10 times the pressure drop in a 21°K refrigerator. Therefore, the lowest attainable operating temperature of the evaporator will usually be dictated by the heat exchanger pressure drop. However, the actual value must be determined for each heat exchanger design and cannot be computed on a generalized basis.

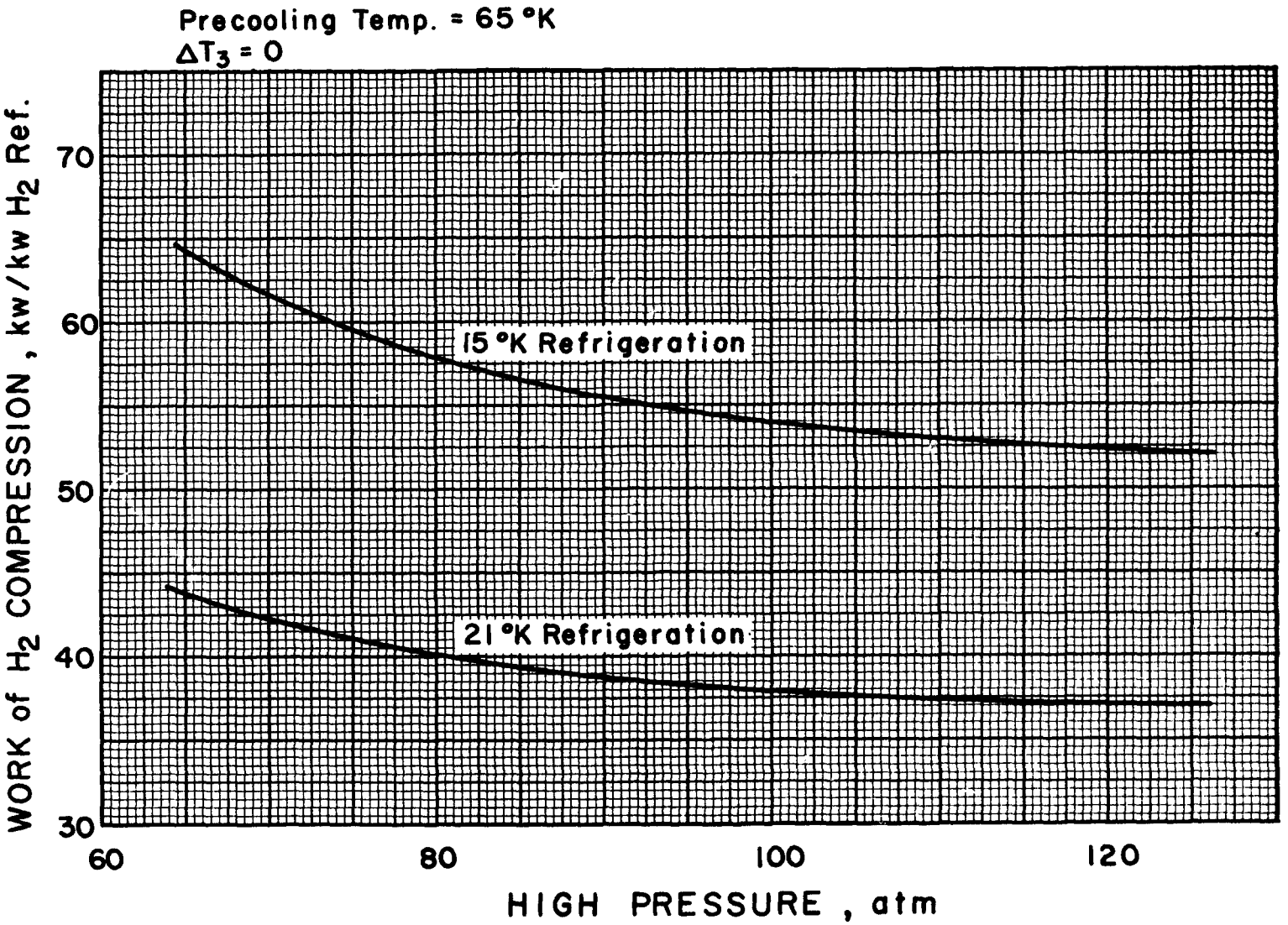

FIGURE 2. HYDROGEN REFRIGERATOR - WORK OF HYDROGEN COMPRESSION

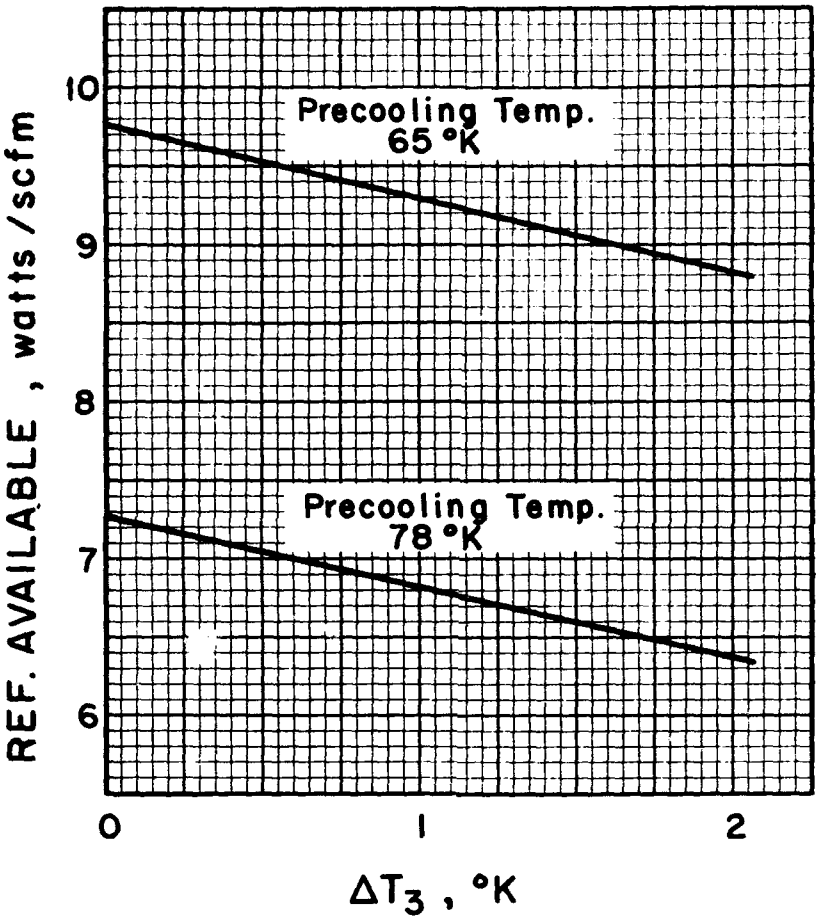

FIGURE 3. HYDROGEN REFRIGERATOR REFRIGERATION AVAILABLE

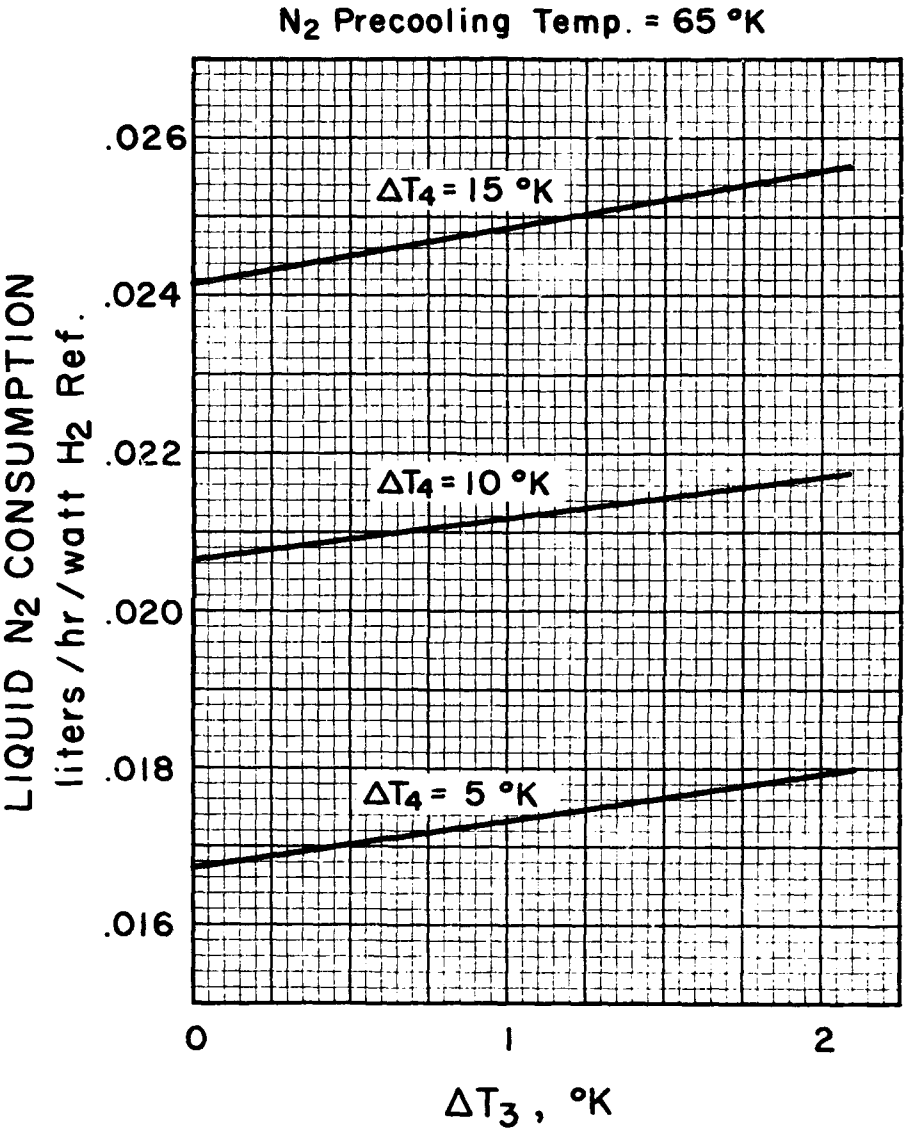

FIGURE 4. HYDROGEN REFRIGERATOR LIQUID NITROGEN CONSUMPTION

HELIUM REFRIGERATOR

The performance of a helium refrigerator with liquid hydrogen precooling has been calculated on the basis of the assumptions set forth in the General Discussion. The design charts presented (Figures 5 through 12) are applicable to refrigeration temperatures from the temperature of the liquid hydrogen evaporator to the temperature of liquid helium.

Figure 5 shows the work of compression (kw) required per unit (kw) of refrigeration at the helium evaporator temperature for theoretical performance of the final heat exchanger, $\Delta T_2 = 0$. Minimum work occurs for an inlet high pressure near 30 atmospheres and this pressure has been used throughout the remainder of the calculations. The compression of 1 scfm (1 atm, 20°C) of helium gas from 1 to 30 atmospheres requires 0.274 kw (1 scfm = 0.0835 gm/sec).

As previously indicated, the quantity of helium refrigeration (Q_2) available is substantially influenced by the hydrogen precooling temperature. Theoretical refrigeration as a function of the precooling temperature is shown in Figure 6. It can be seen that an increase in refrigeration of 80 percent is achieved by reducing the precooling temperature from 21°K to 15°K when the return helium stream is at a pressure of one atmosphere. An increase in the low pressure of the system will decrease the available refrigeration but will also decrease the power of compression. A low pressure of 5 atmospheres results in a decrease in the work of compression of 47 percent (30 atmospheres high pressure) while the theoretical refrigeration available is decreased by 20-25 percent. Thus, operation at a slightly elevated low pressure may be beneficial if refrigeration temperatures above liquid helium are desired. If refrigeration is required near the normal boiling point of helium (4.2°K), it is important to keep this pressure as low as practicable.

Applying the First Law of Thermodynamics to the process shown in the schematic of Figure 1 shows that ΔT_2 affects the final quantity of refrigeration (Q_2) and that ΔT_1 affects the relationship of Q_2 to the quantity of refrigeration required in the hydrogen evaporator (Q_1). Figures 7 and 8 indicate the ratio of refrigeration, Q_2/Q_1, as a function of both ΔT_1 and ΔT_2 for 21°K and 15°K precooling, respectively. The temperature difference at the warm end of the final helium heat exchanger (ΔT_2) has a profound influence on the refrigeration available - considerably greater than the influence of ΔT_1. In fact, when $\Delta T_2 = 1.9$°K with 21°K precooling, no refrigeration is available below the hydrogen evaporator temperature. The point of zero refrigeration with 15°K precooling occurs when $\Delta T_2 = 3.1$°K.

The basic assumption of theoretical refrigeration, $\Delta T_2 = 0$, requires further consideration for the helium refrigeration cycle. By a cooling curve analysis[2], for the conditions under consideration, a violation of the Second Law of Thermodynamics is observed. To avoid this difficulty, a modification to the schematic shown in Figure 1 must be made for the portion of the helium circuit below the hydrogen evaporator.[7] This modification is shown in Figure 1a below.

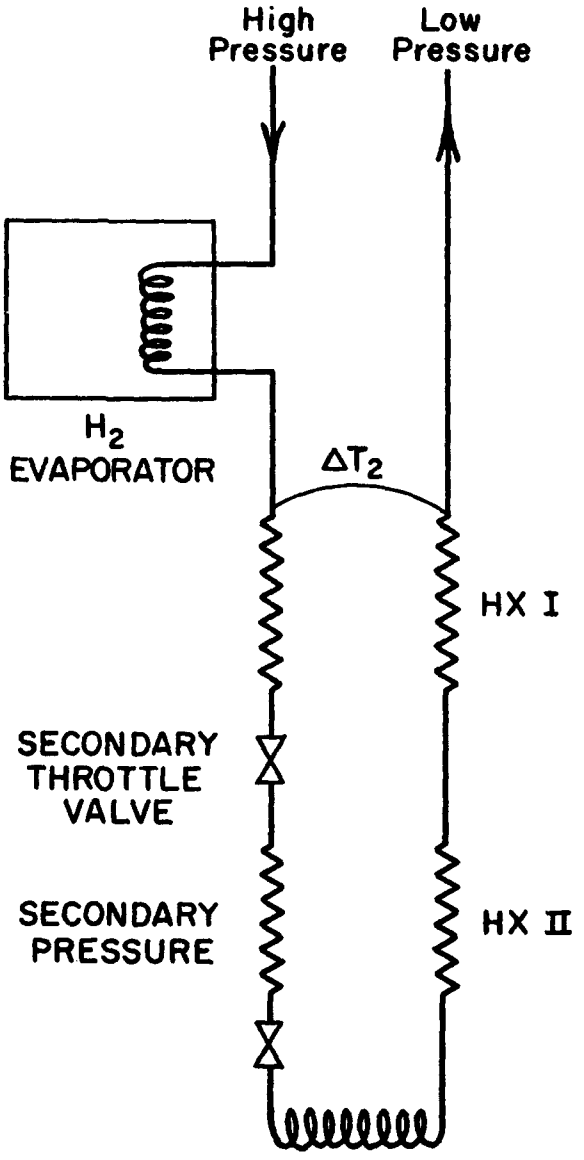

Figure 1a Modified Schematic for Helium Refrigerator

The final helium heat exchanger of Figure 1 is subdivided into two heat exchangers by the addition of a secondary throttle valve. The inlet temperature of this valve and the resulting down stream pressure is selected to prevent a violation of the Second Law of Thermodynamics. A cooling curve for this arrangement is shown in Figure 9. For a 30 atmosphere high pressure and the precooling temperatures under consideration, a secondary pressure of 5-10 atmospheres is sufficient.

To be consistent with the Second Law of Thermodynamics, if provisions for the two throttle valves are not made, the minimum values of ΔT_2 are $0.5^{\circ}K$ for $21^{\circ}K$ precooling and $1.7^{\circ}K$ for $15^{\circ}K$ precooling. These temperature differences are sufficient to provide a $\Delta T = 0$ at the cold end of the final heat exchanger. The effect of this ΔT_2 can be seen from Figures 7 and 8.

Additional design parameters are given in Figures 10 and 11. From these figures, it is possible to determine the helium flow rate and the consumption of liquid nitrogen for a helium refrigerator. The liquid nitrogen consumption shown in Figure 11 is independent of the hydrogen evaporator temperature.

The effect of eliminating the nitrogen evaporator in the helium circuit is shown in Figure 12 for a $21^{\circ}K$ hydrogen evaporator. The ratio of the quantity of helium refrigeration to the quantity of hydrogen refrigeration is greatly reduced, conclusively indicating the desirability of this precooling in the helium circuit.

As indicated in the General Discussion, the installation of an expansion engine across the final helium heat exchanger, as shown by the dotted line in Figure 1, adds substantially to the thermodynamic efficiency of the cycle. Preliminary calculations indicate that the addition of an expansion engine having a $21^{\circ}K$ inlet temperature and an 85 percent isentropic expansion efficiency will decrease the work of compression by at least a factor of 2 and will increase the available refrigeration (Q_2) by a considerable factor. The optimum high pressure, based on work of compression, is reduced to about 15 atmospheres.

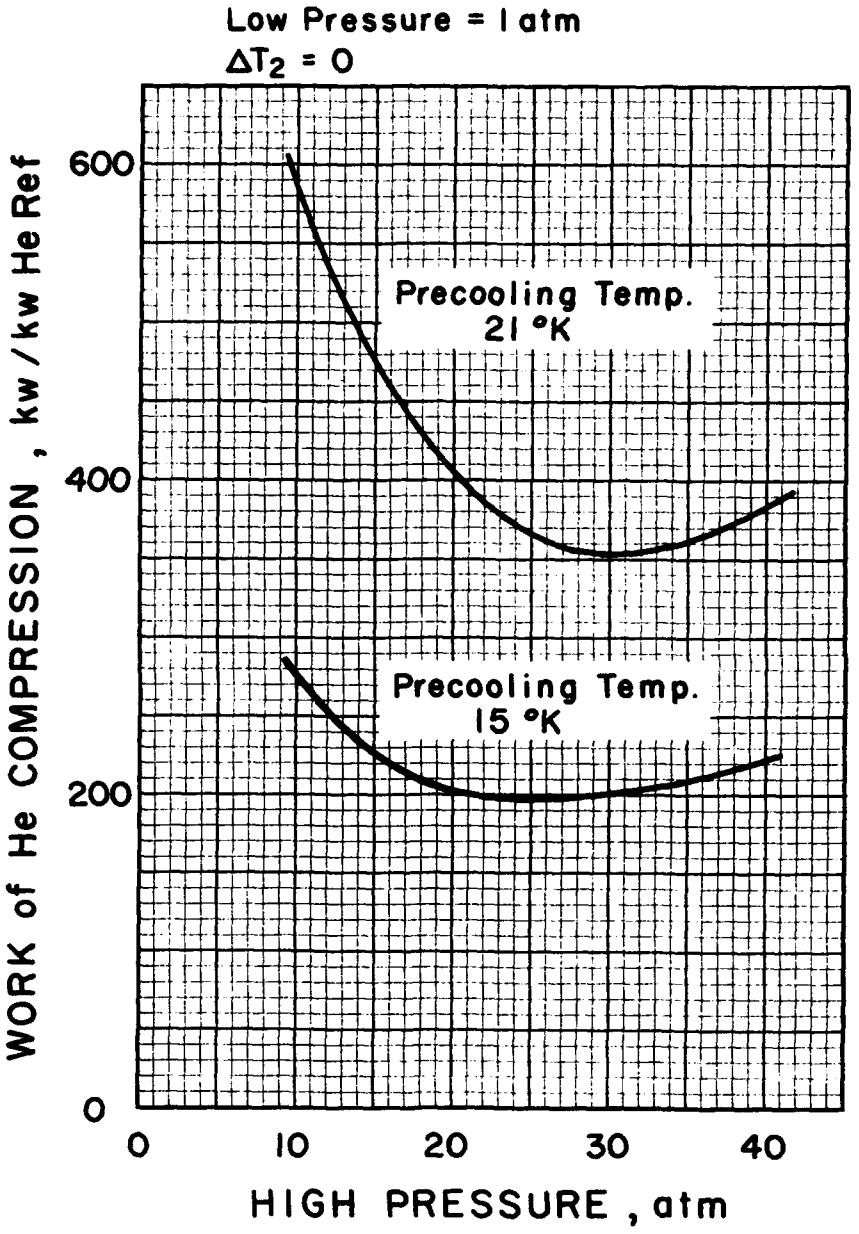

FIGURE 5. HELIUM REFRIGERATOR
WORK OF HELIUM COMPRESSION

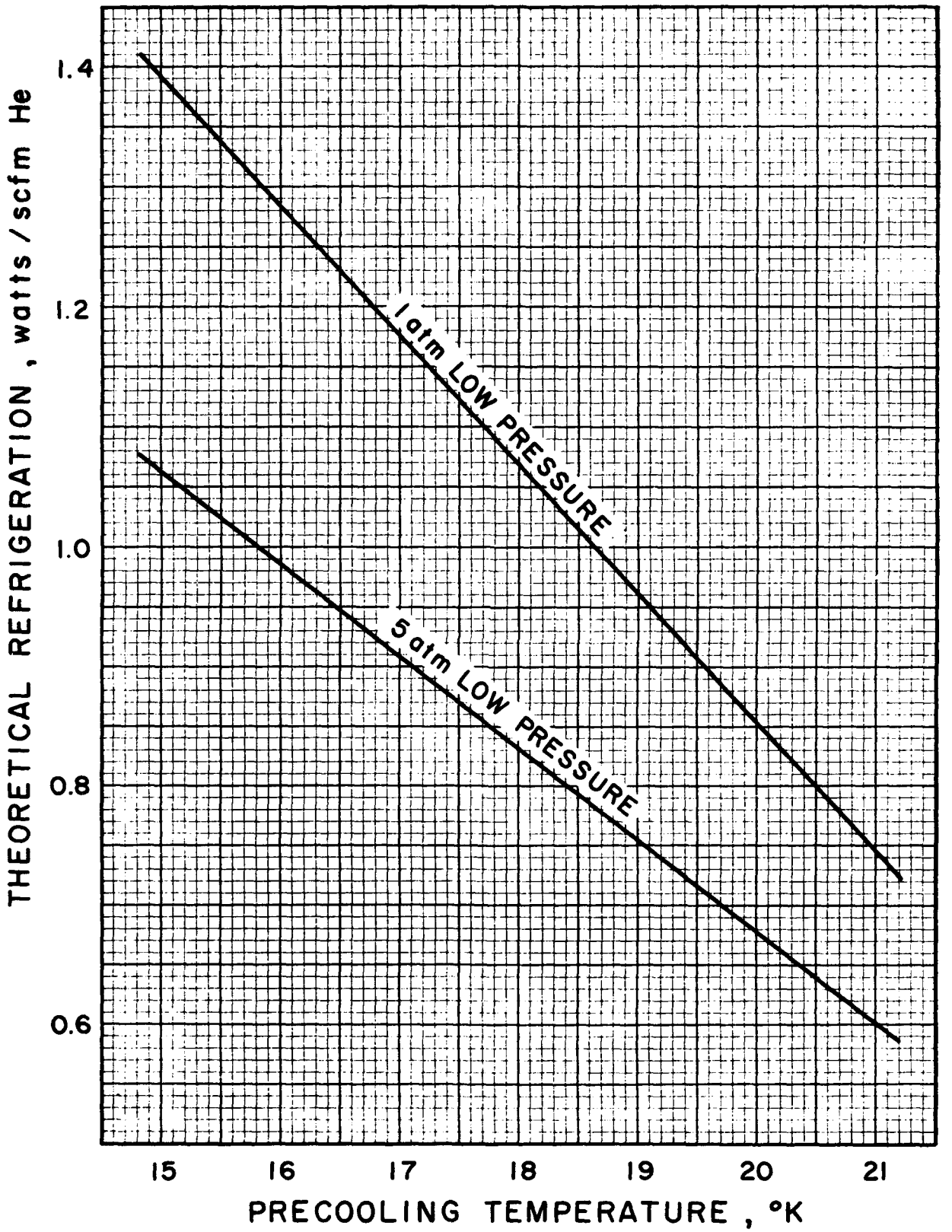

FIGURE 6. HELIUM REFRIGERATOR - THEORETICAL REFRIGERATION

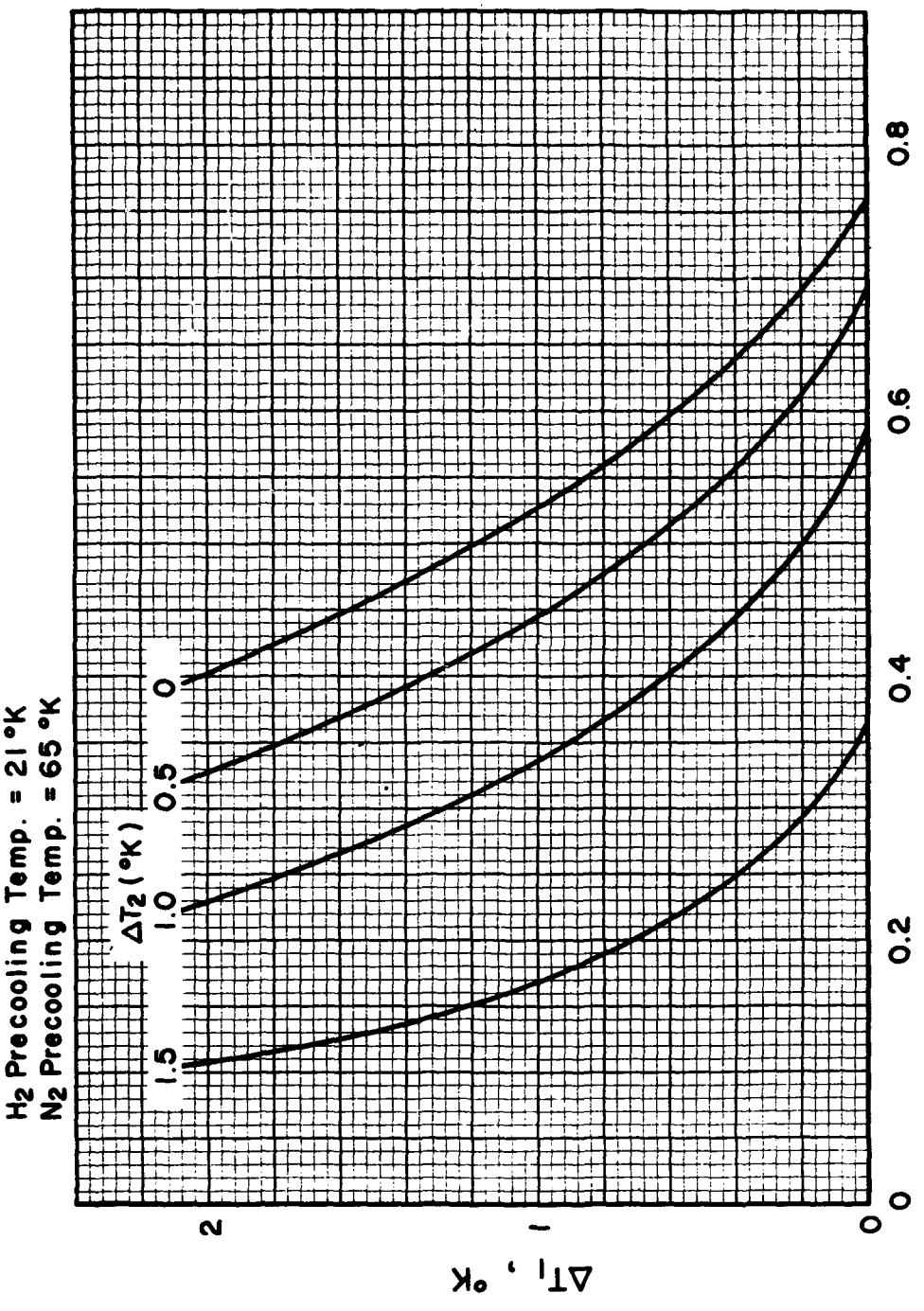

FIGURE 7. HELIUM REFRIGERATOR - EFFECT OF HEAT EXCHANGER PERFORMANCE ($21^{\circ}K$ PRECOOLING).

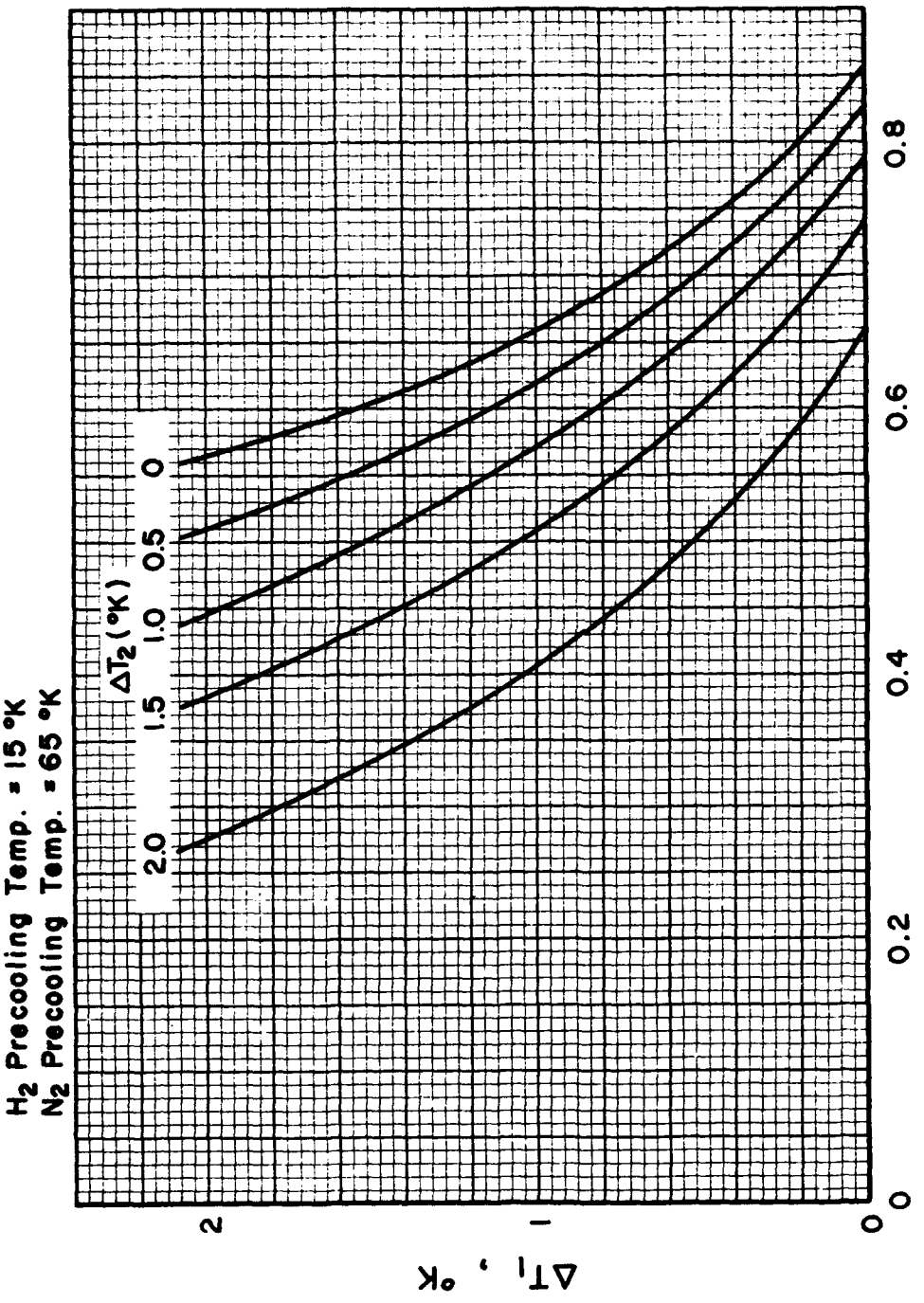

FIGURE 8. HELIUM REFRIGERATOR - EFFECT OF HEAT EXCHANGER PERFORMANCE (15°K PRECOOLING).

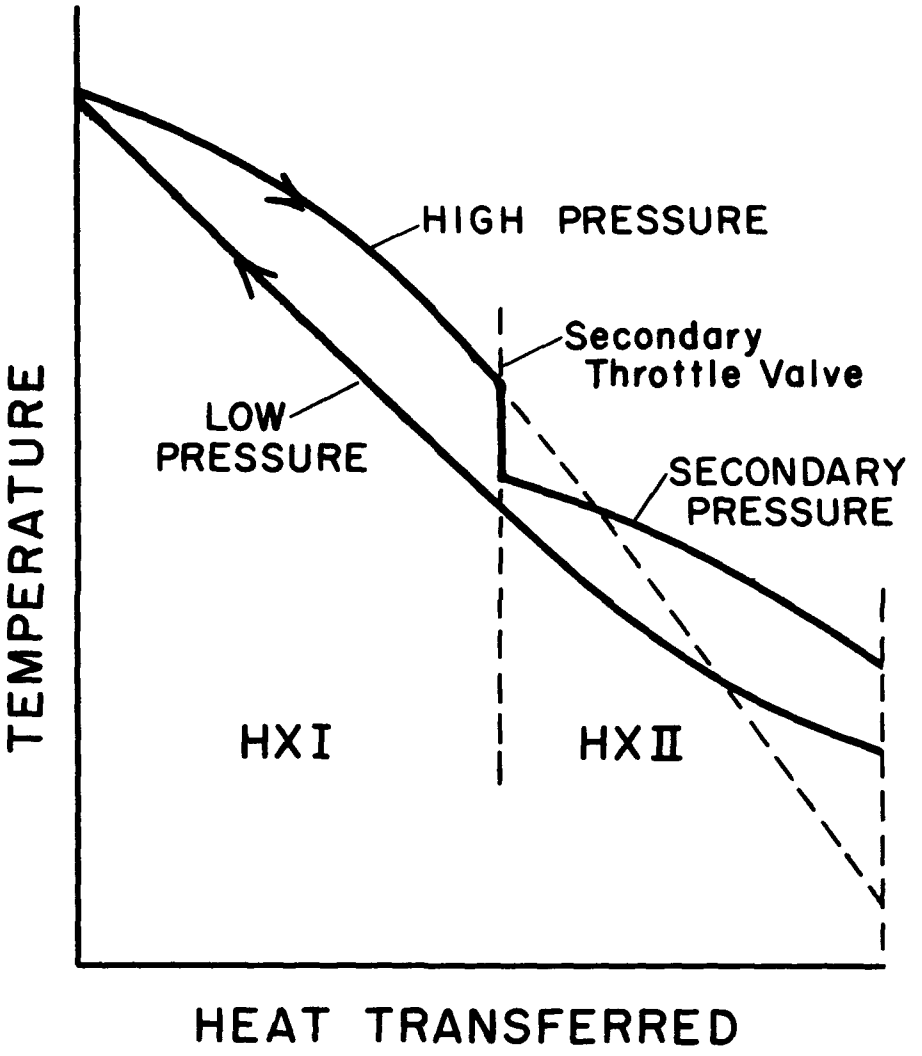

FIGURE 9. HELIUM REFRIGERATOR - COOLING CURVE FOR MODIFIED CYCLE

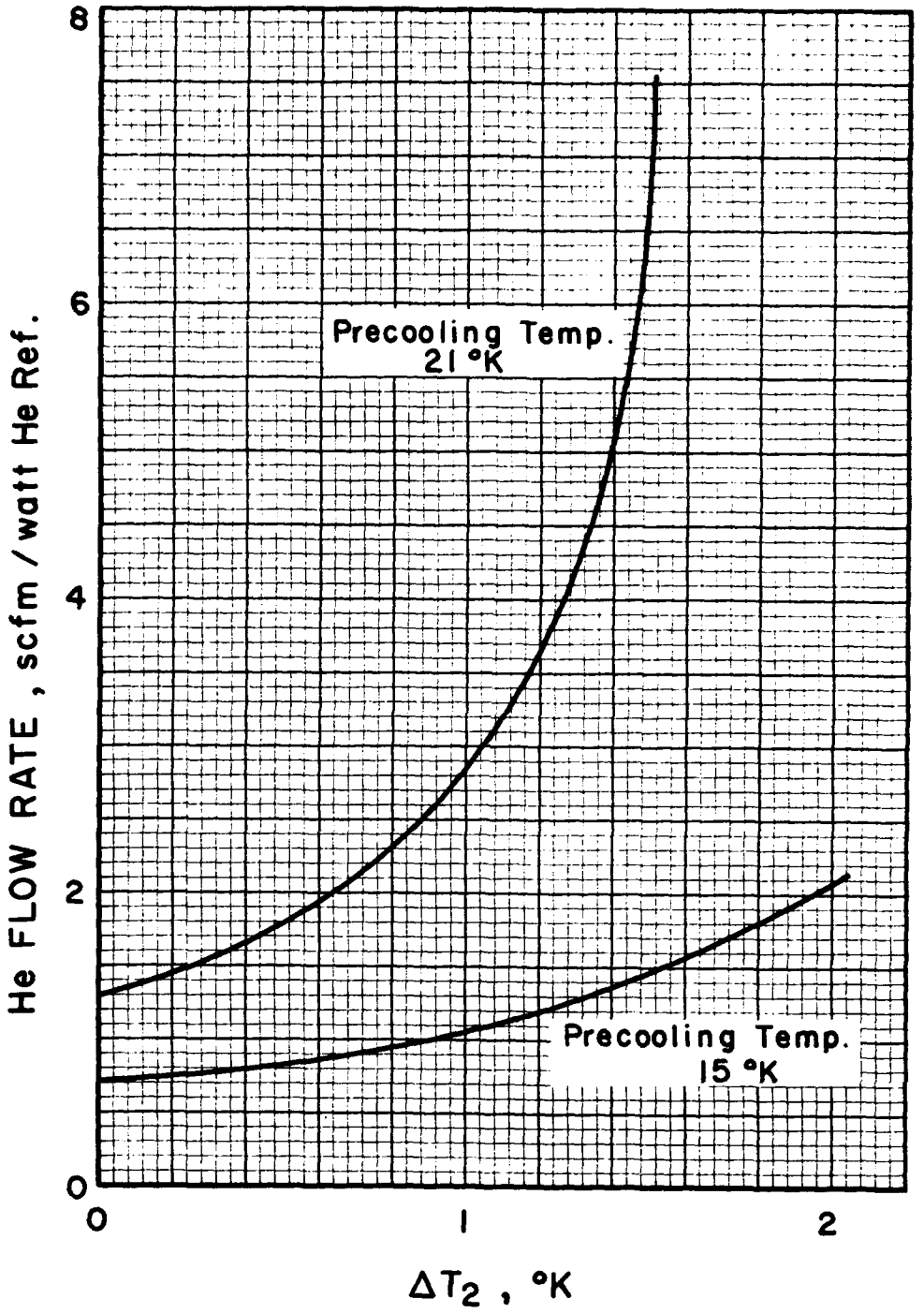

FIGURE 10. HELIUM REFRIGERATOR - HELIUM FLOW RATE

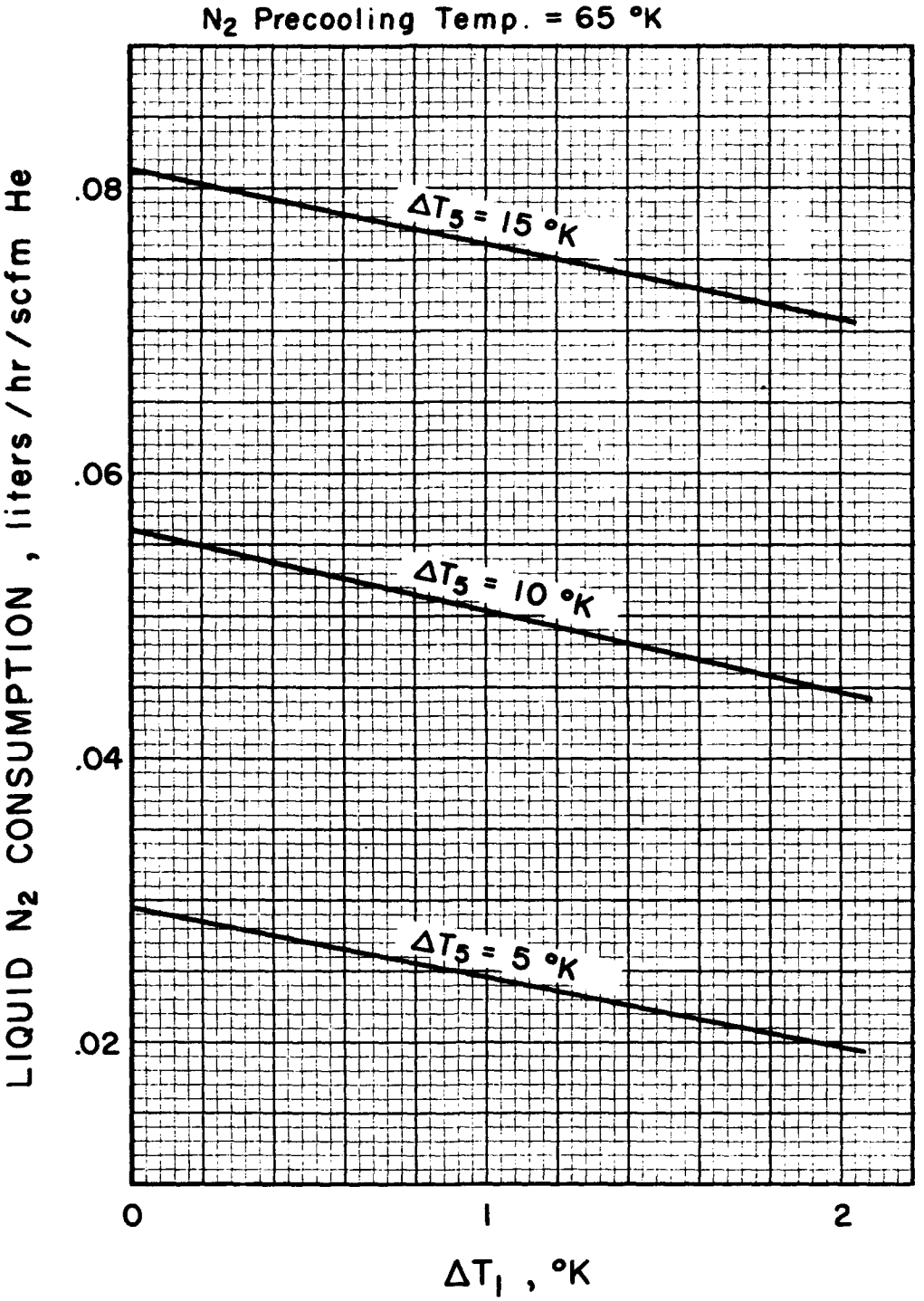

FIGURE II. HELIUM REFRIGERATOR - LIQUID NITROGEN CONSUMPTION

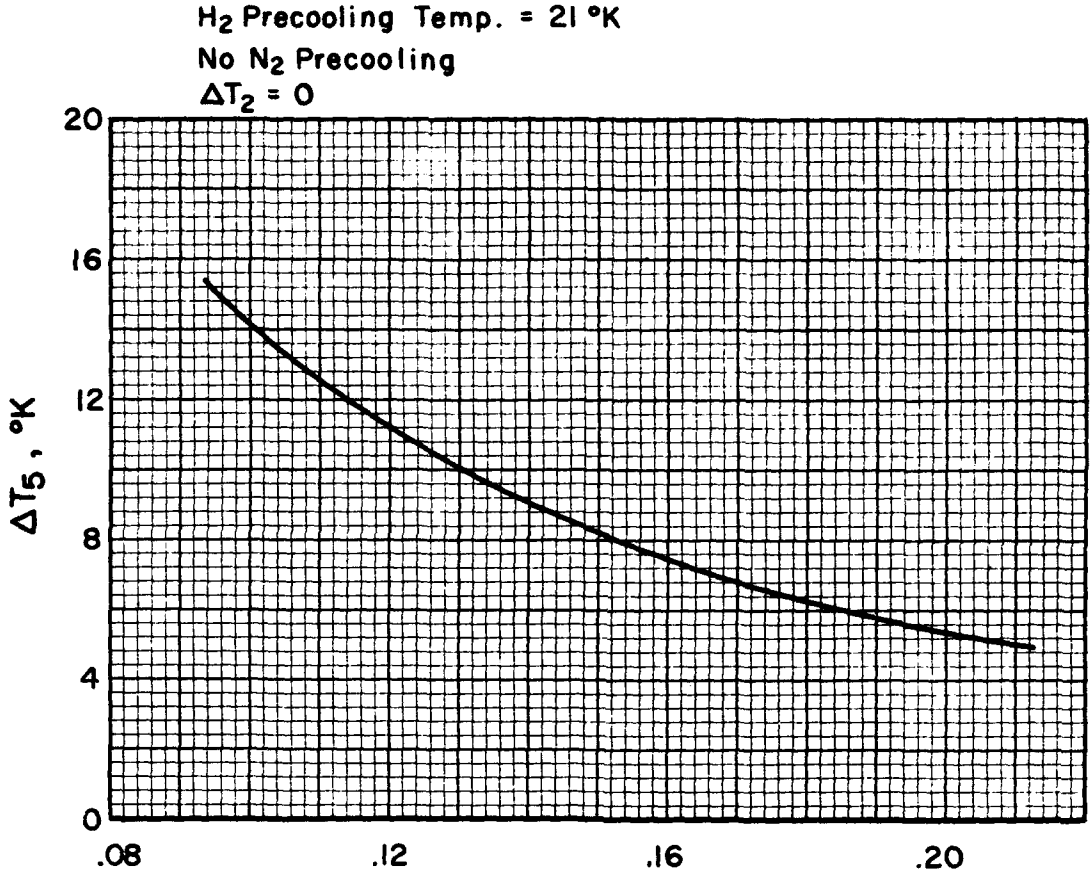

FIGURE 12. HELIUM REFRIGERATOR - EFFECT OF HEAT EXCHANGER PERFORMANCE (NO NITROGEN PRECOOLING).

HELIUM LIQUEFIER

Essentially the same principles apply to the helium liquefier that apply to the helium refrigerator discussed in the previous section. However, for the pressures and temperatures considered, there is no danger of violations of the Second Law of Thermodynamics. Although the work of helium compression in a helium liquefier differs from that of a refrigerator, the level of optimum high pressure remains the same. This is shown in Figure 13 where the work of helium compression per unit of helium liquefied is presented as a function of the high pressure. The liquid produced is calculated assuming theoretical performance of the final heat exchanger, $\Delta T_2 = 0$, and a low pressure in the cycle of one atmosphere. The remainder of the calculations are based on the optimum high pressure of 30 atmospheres. The influence of the heat exchanger temperature difference (ΔT_2) on the quantity of liquid produced (fraction liquefied) is shown in Figure 14. If liquid helium at one atmosphere is desired as an end product, increasing the low pressure will both reduce the fraction liquefied and cause flashing losses when the liquid is transferred from the liquefier to an external storage container. It is particularly important therefore, to keep the low pressure as near one atmosphere as possible. The pressure drop on the low pressure side of the helium system must be carefully evaluated.

The ratio of helium liquefied to the quantity of hydrogen refrigeration required is shown in Figures 15 and 16. The effect of heat exchanger performance can clearly be seen from these figures. With $21^{\circ}K$ precooling, no liquefaction occurs when $\Delta T_2 = 1.9^{\circ}K$. With $15^{\circ}K$ precooling, no liquefaction occurs when $\Delta T_2 = 3.1^{\circ}K$. The required helium flow rate and liquid nitrogen consumption are shown in Figures 17 through 20.

The effect of eliminating the liquid nitrogen evaporator in the helium circuit is shown in Figure 21. The ratio of helium liquefied to the quantity of hydrogen refrigeration required is greatly reduced, again indicating the desirability of this precooling in the helium circuit.

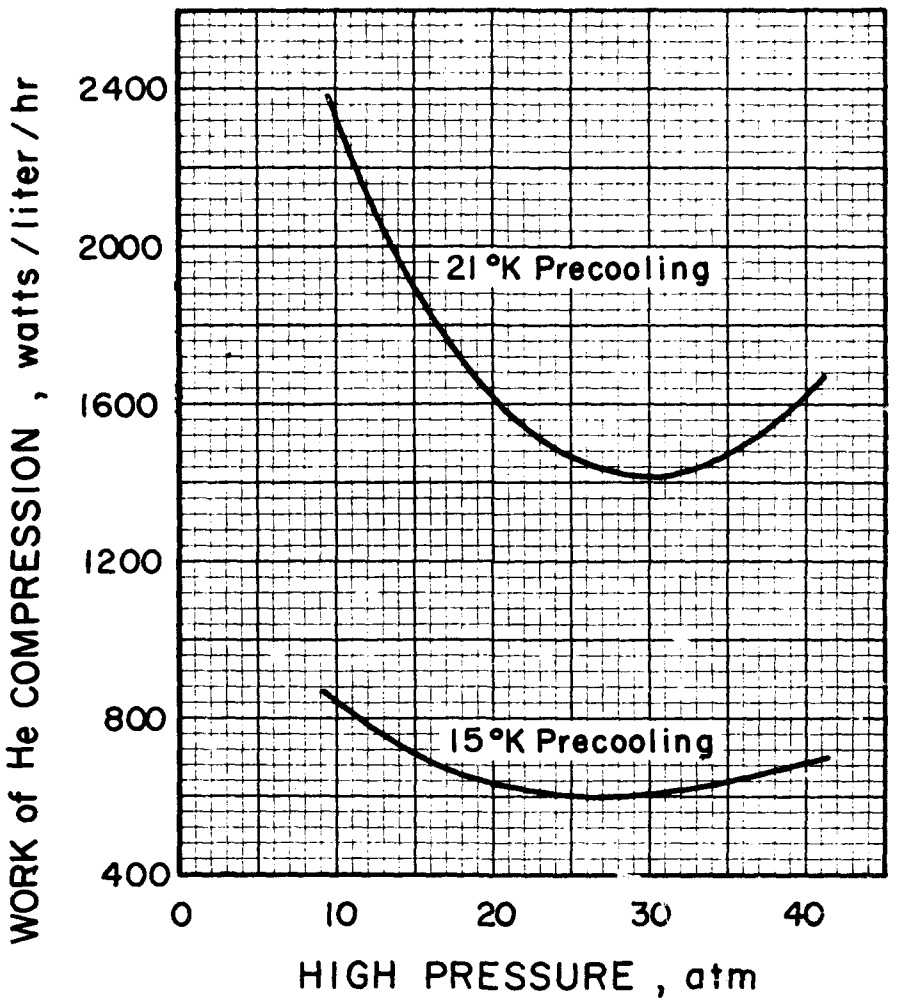

FIGURE 13. HELIUM LIQUEFIER - WORK OF HELIUM COMPRESSION.

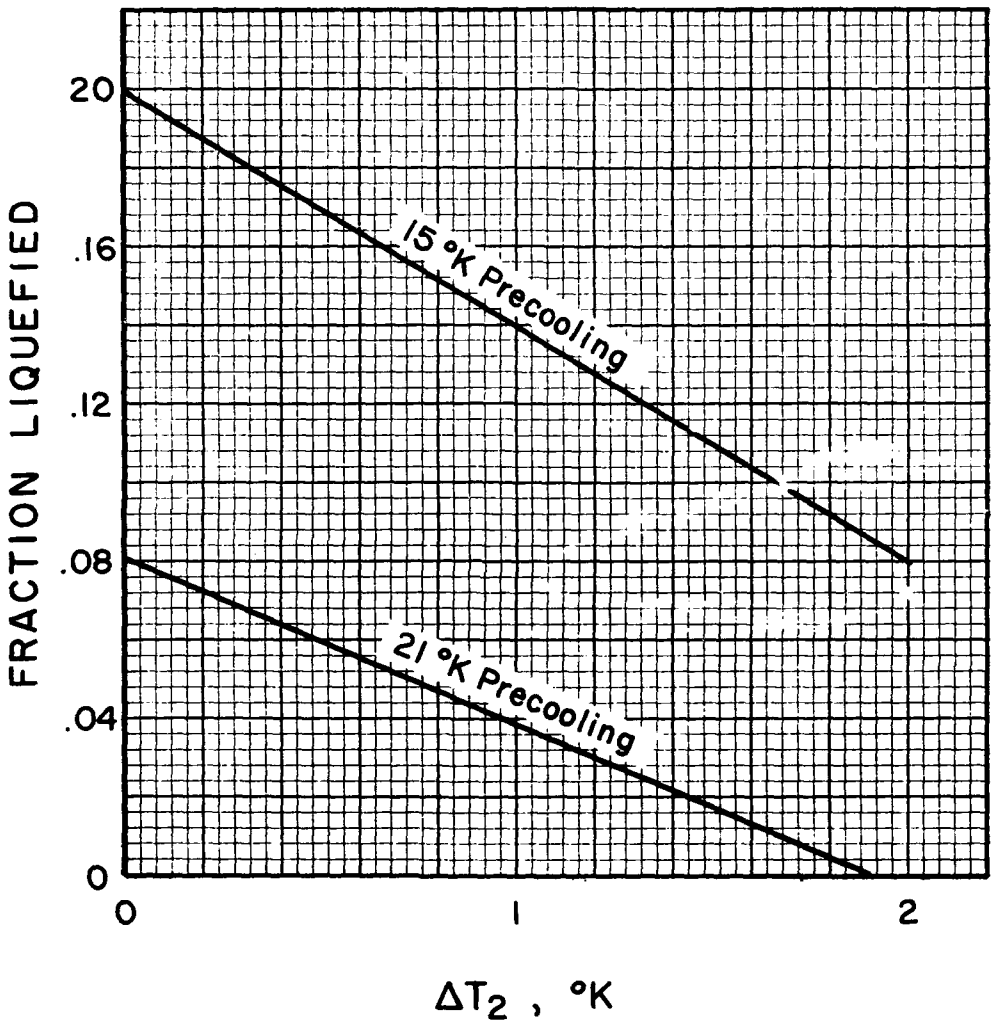

FIGURE 14. HELIUM LIQUEFIER - FRACTION LIQUEFIED.

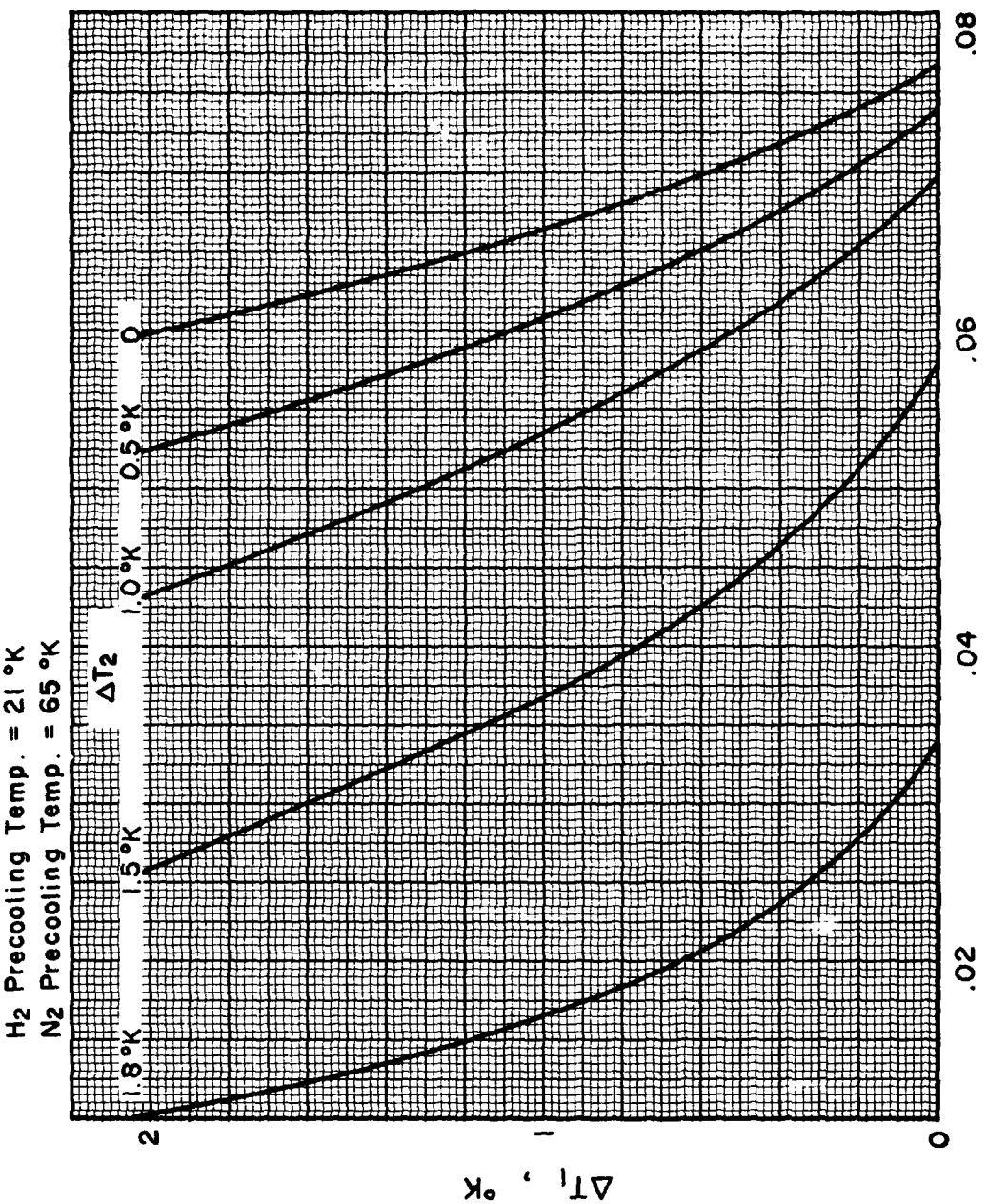

FIGURE 15. HELIUM LIQUEFIER - EFFECT OF HEAT EXCHANGER PERFORMANCE (21° K PRECOOLING).

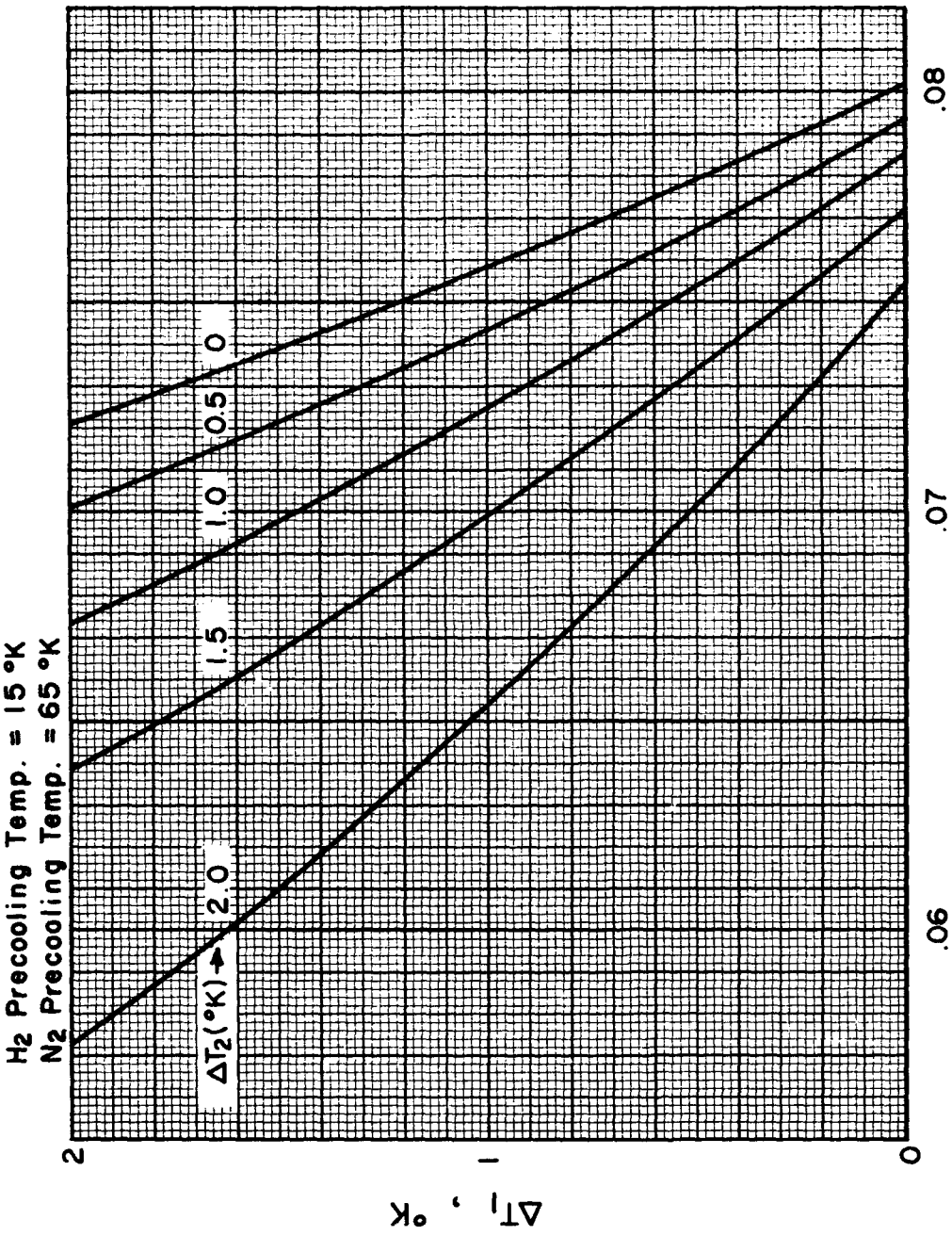

FIGURE 16. HELIUM LIQUEFIER - EFFECT HEAT EXCHANGER PERFORMANCE (15° K PRECOOLING).

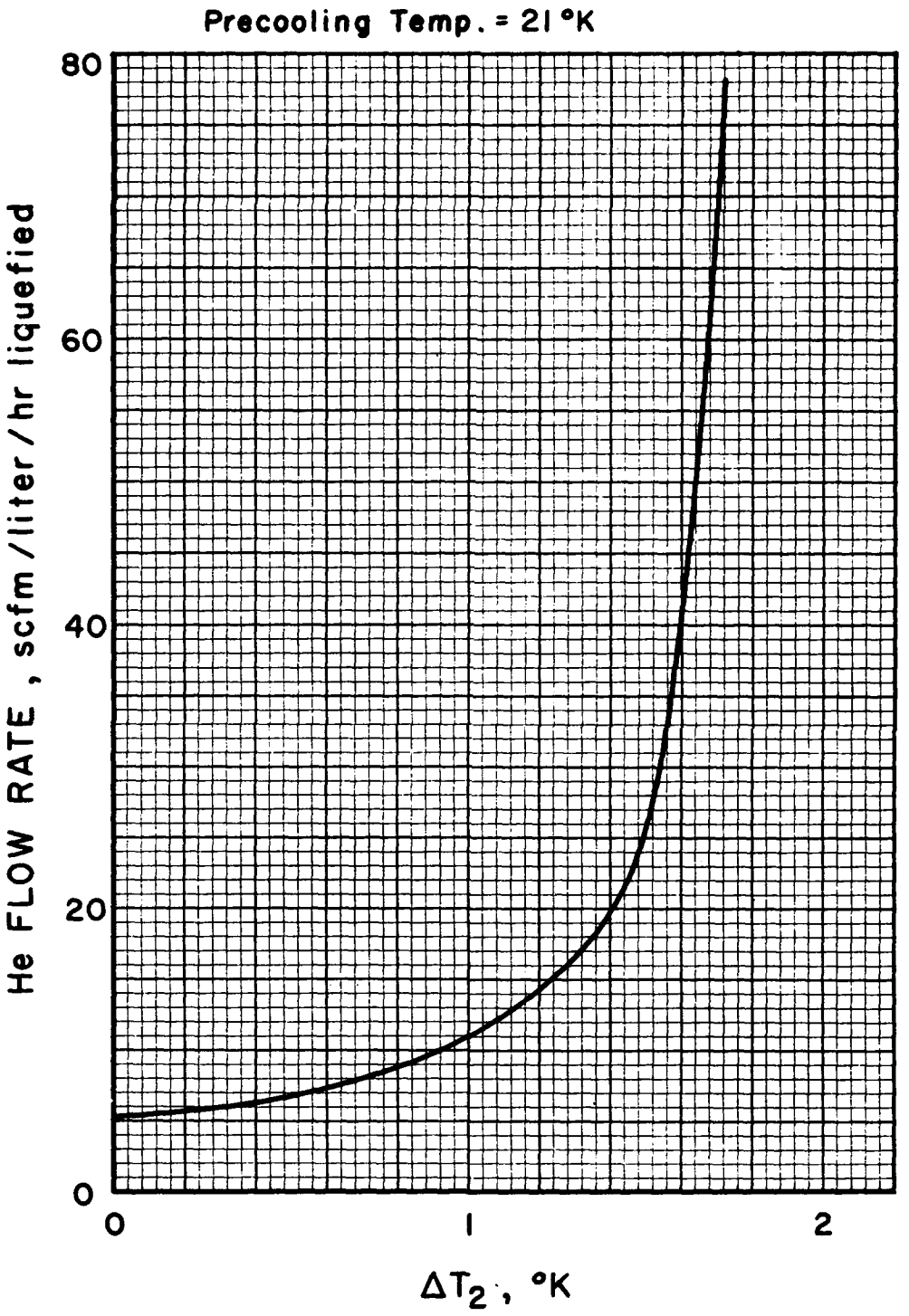

FIGURE 17 HELIUM LIQUEFIER - FLOW RATE (21° K PRECOOLING).

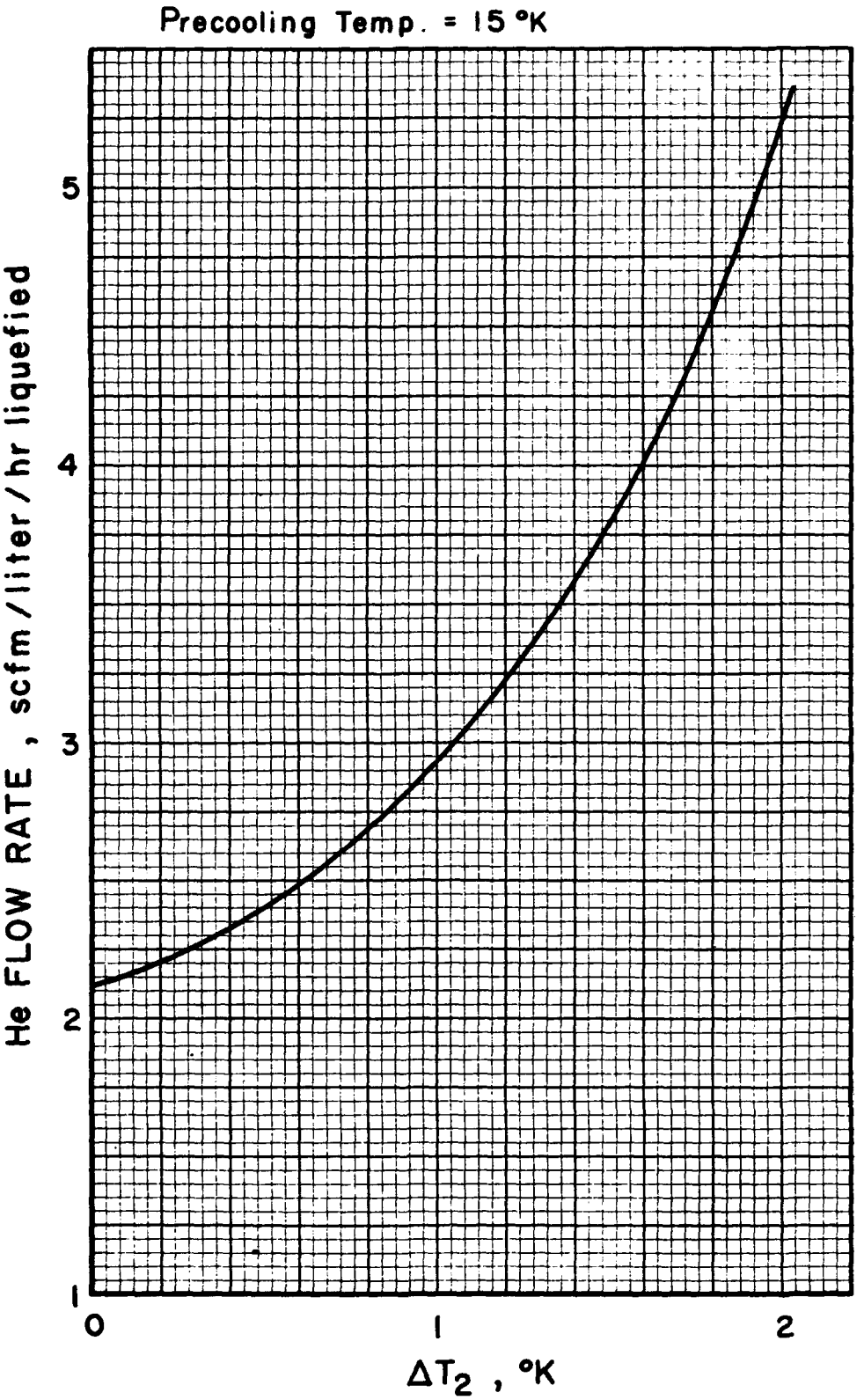

FIGURE 18. HELIUM LIQUEFIER - FLOW RATE (15° K PRECOOLING).

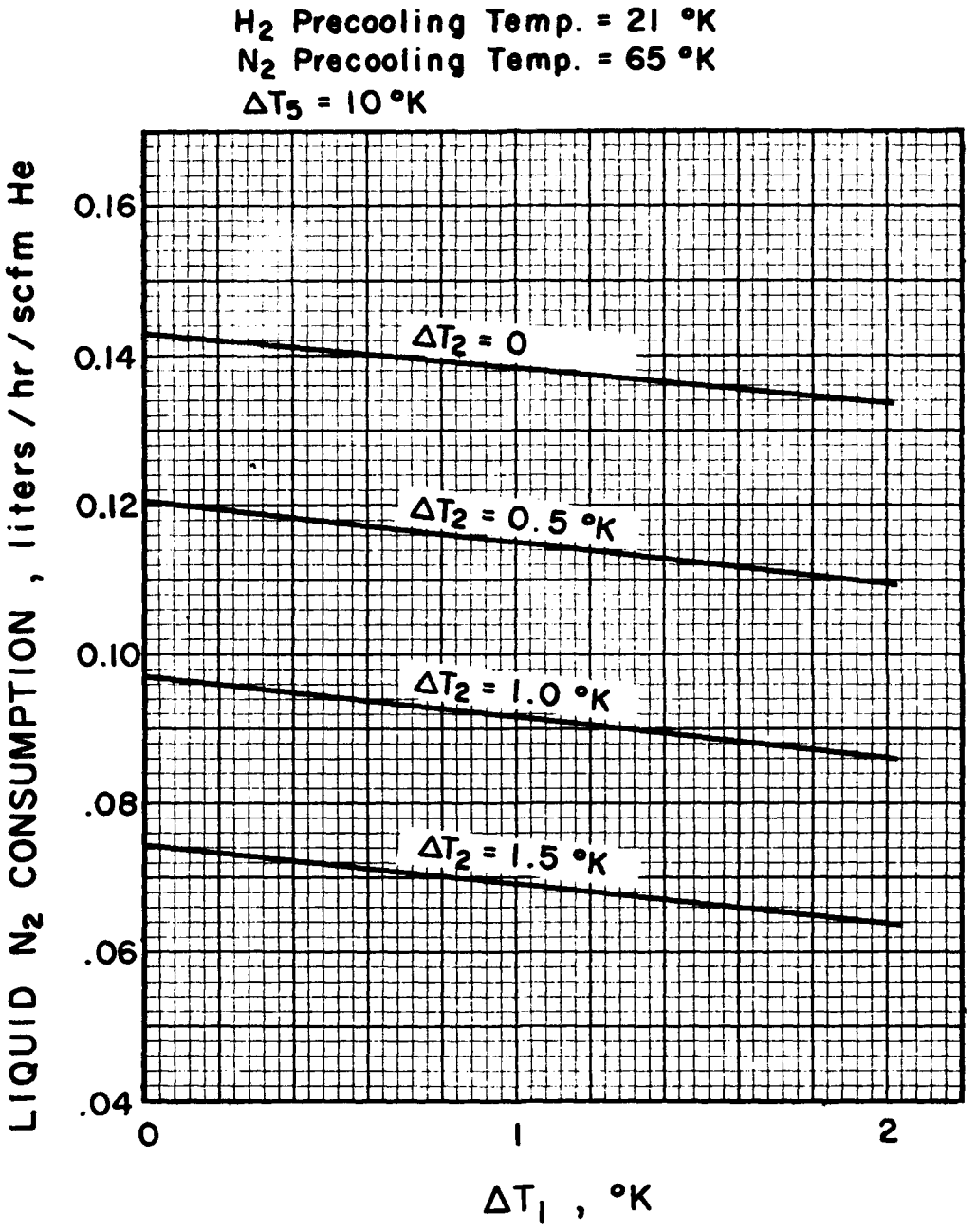

FIGURE 19. HELIUM LIQUEFIER - LIQUID NITROGEN CONSUMPTION (21° K PRECOOLING).

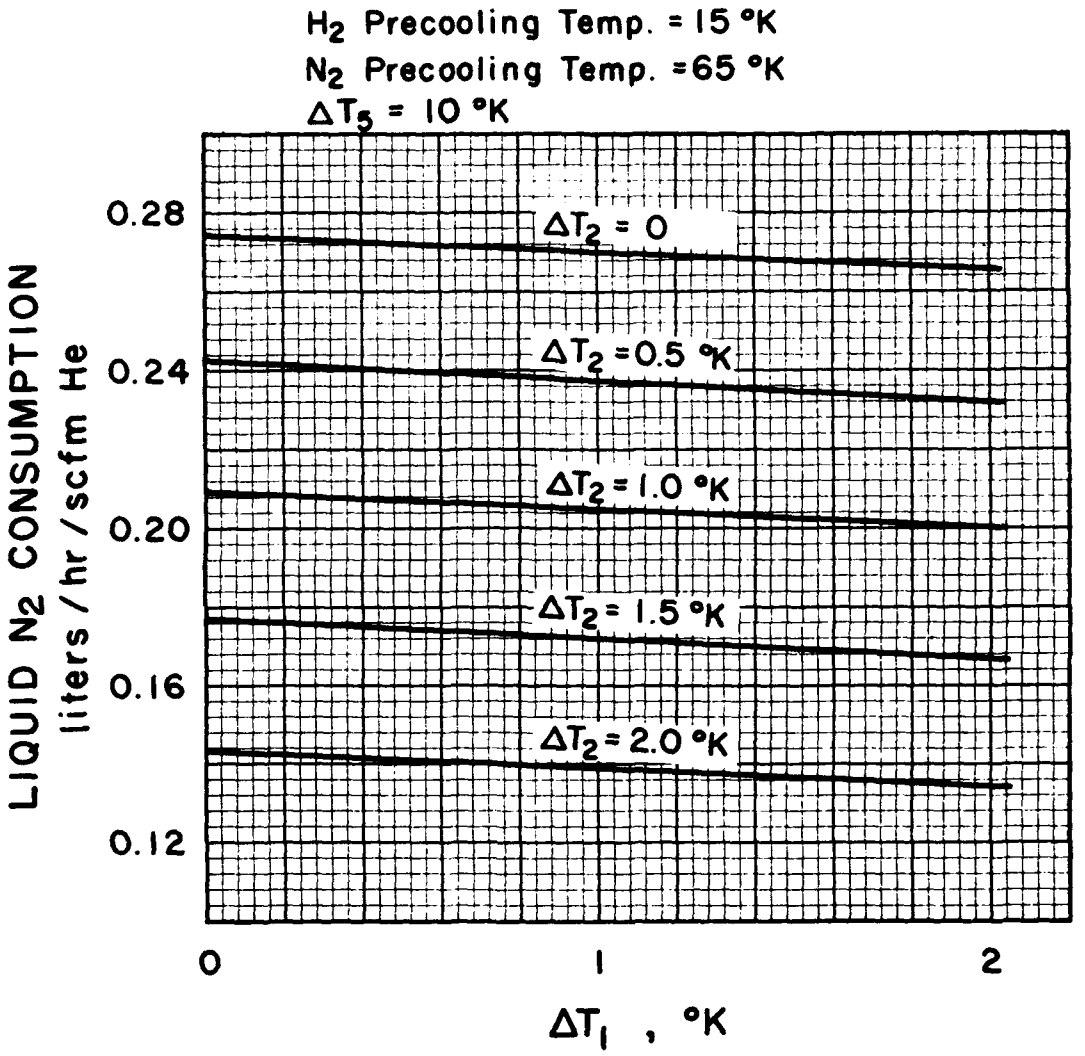

FIGURE 20. HELIUM LIQUEFIER - LIQUID NITROGEN CONSUMPTION (15° K PRECOOLING).

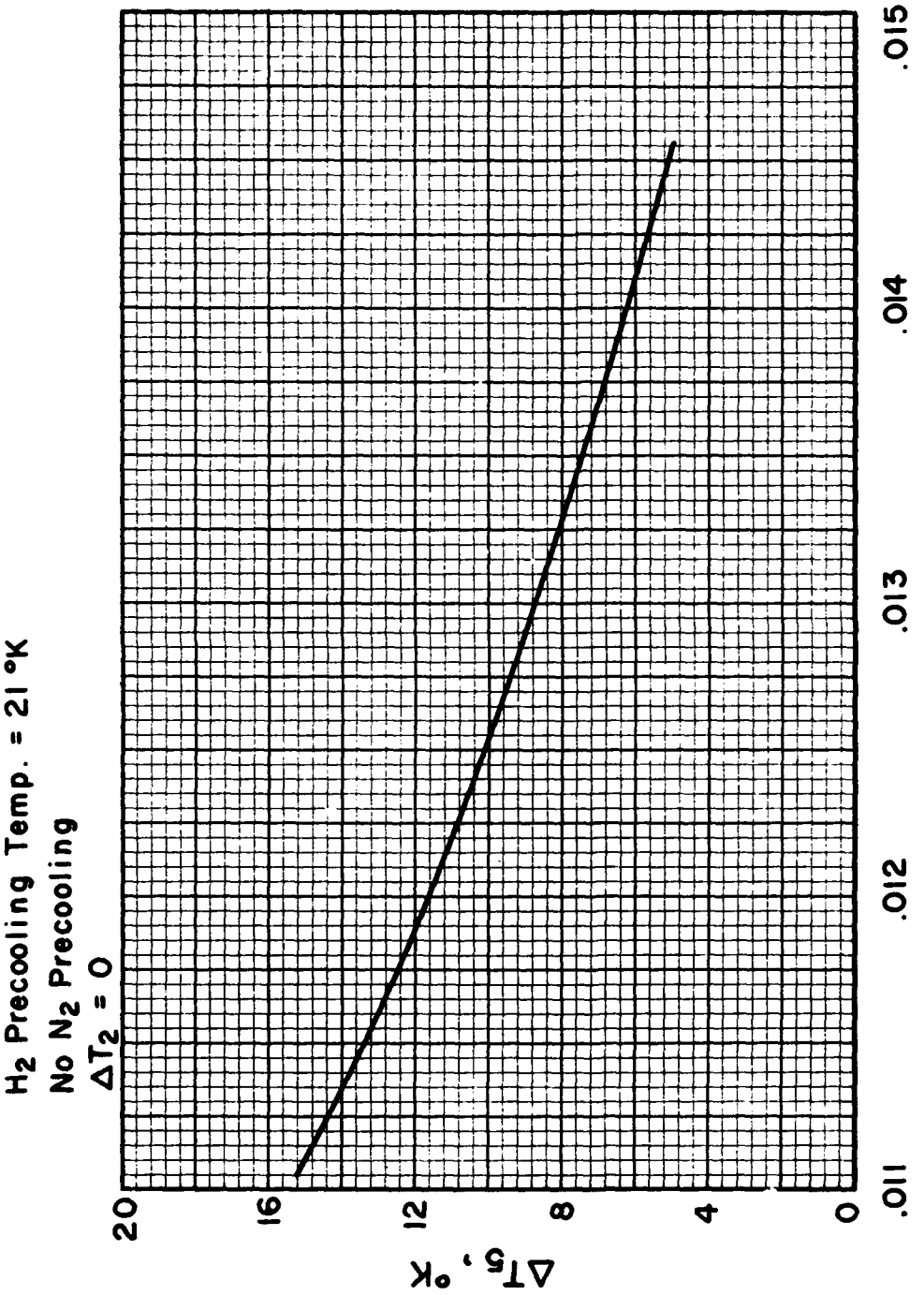

FIGURE 21. HELIUM LIQUEFIER - EFFECT OF HEAT EXCHANGER PERFORMANCE (NO NITROGEN PRECOOLING)

BIBLIOGRAPHY

1. 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).

2. D. B. Chelton and J. W. Dean, "Design and Construction of a Liquid Hydrogen Temperature Refrigeration System", National Bureau of Standards Technical Note No. 38 (1960).

3. D. B. Mann and R. B. Stewart, "Thermodynamic Properties of Helium at Low Temperatures and High Pressures", National Bureau of Standards Technical Note No. 8 (1959).

4. W. H. Keesom, A. Bijl and L. A. J Monte, "Le Diagramme log T, Sde l'helium, Leiden Communication, Supplement No. 108c.

5. J. Zelmanov, "The *Entropy* Diagram for Helium at Low Temperatures", Journal of Physics (USSR), Volume 8, Number 3, (1944) pp. 135-141.

6. 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.

7. I. L. Zelmanov, Comptes rendus (Doklady) Acad. Sci. URSS, 22, 25 (1939).