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INVESTIGATION OF EXTERNAL REFRIGERATION SYSTEMS FOR LONG TERM CRYOGENIC STORAGE

PREPARED FOR

MANNED SPACECRAFT CENTER

UNDER CONTRACT NAS 9-10412

LMSC

SYSTEMS REVIEW REPORT

LOCKHEED MISSILES & SPACE COMPANY

A GROUP DIVISION OF LOCKHEED AIRCRAFT CORPORATION

SPACE SYSTEMS DIVISION · SUNNYVALE, CALIFORNIA

CR-115141

LMSC/A903162

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PREPARED BY

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FOREWORD

This report was prepared by Lockheed Missiles and Space Company (LMSC) for The Manned Spacecraft Center, National Aeronautics and Space Administration, Houston, Texas, under Contract NAS 9-10412. This report covers the portion of the study pertaining to a review of cryogenic refrigeration systems. The study is being conducted under the direction of Thomas L. Davies, Propulsion and Power Division. Major contributors to this report were H. L. Jensen, Project Manager, T. C. Nast, Thermophysics and A. P. M. Glassford, Thermophysics.

SUMMARY

This Systems Review Report is the product of the first part of a study entitled "An Investigation of External Refrigeration Systems for Long-Term Cryogenic Storage Systems" being performed by Lockheed Missiles and Space Co. for The NASA, Manned Spacecraft Center. The objective of the study is to perform refrigeration systems review, technical analyses, and parametric studies that will yield design trade-off information that may be used for external refrigeration system selection and optimization for a variety of space applicable cryogenic tankage.

The refrigeration capacity of interest is between 5 and 100 watts at temperatures from 20°K to 110°K. Application of the refrigeration systems will be for single and double walled cryogenic tanks ranging in size from 20 to 200 cubic feet. The mission durations of interest are from 6 to 24 months and the heat rejection sink temperatures range from 20°K to 110°K.

The goal of the first part of the study was to review the data and material available on existing and prototype cryogenic refrigeration systems that would have applicability to this study and to present this data in a Systems Review Report.

The data contained in this report are to form a part of a refrigeration system Design Reference Manual that will be completed at the end of the study. This Systems Review Report is not intended to be used as a refrigerator selection guide in itself, since developed refrigeration systems for space applications simply are not generally available. Rather, it is intended to provide some guidance to the spacecraft systems designer as to overall state of refrigeration technology so that he may define the shortest, and most appropriate refrigerator development program for his particular task.

This report provides some introductory material in Sections 1 through 4 that is intended to help clarify and identify the various thermodynamic processes and refrigeration cycles that are employed by most of the manufacturers currently

producing cryogenic refrigerators. Section 5 gives data on existing refrigerators (including prototype models) of interest to this study. Section 6 provides considerations of maintenance and lifetime and Section 7 gives a brief overall summary.

CONTENTS

ILLUSTRATIONS

TABLES

1.0 INTRODUCTION

The purpose of this report is to summarize the current available information on small refrigerator systems that are applicable to in-space cryogenic cooling operations. Most of the data is applicable to cooling loads of 1 to 100 watts and cooling temperatures of $20^{o}K$ to $110^{o}K$. To date no closed cycle refrigeration system has been used in space, although a few elementary short life open cycle systems have been flown and a few programs presently underway will place closed cycle refrigerators in space in the near future. Consequently, such systems must be developed and proven for space use, and the content of this report necessarily relates as much to assessing developmental potential based upon current information, as to reporting actual performance. The assessment of the suitability of a refrigerator for a given cooling task in this context depends upon the current performance, development potential and peculiarities of the task.

The current state of refrigerator technology is governed largely by a combination of technical performance limitations and prevailing economic market. For discussion purposes the interaction of these influences may be divided into three categories. In the first would be those purely technical limitations imposed by an incomplete understanding of the basic process, or by fundamental limits of the processes themselves. Refrigerator designs are based on the best available heat transfer, fluid flow and material property data and an upper limit as thermodynamic performance will be set by the adequacy of these data, the methods of handling the data for design purposes, and the techniques for obtaining the performance of actual machines. These limitations are discussed in detail in the following sections describing particular configuration systems. In a second category of influences would be the influence of the market along on refrigerator technology. In this category would be placed the simple observation that one of the reasons that spaceborne refrigerators are not available is that there has been no demand for them. This deficiency can be remedied by support of the refrigerator market by research and development programs such as those funded by the Defense Department. These programs effectively permit systems of limited economic appeal to be developed to their full technical capabilities. This aspect is discussed further in Section 2. In the third category are those limitations placed by an interaction between market requirements, mechanical design and refrigerator performance. For example, the required maintenance intervals of many commercially available refrigerators are currently optimized with respect to cost, and could be extended considerably if the customer were prepared to pay the increased cost of larger maintenance intervals.

From a knowledge of the current state of technology one may select a suitable refrigerator system, define a program leading to the design or development of a satisfactory hardware from existing technology, or define a program of research and development on one or more new types of refrigeration systems. The selection process must be based upon a complete specification of the total spacecraft environment in which the refrigerator must operate. Such a specification must include the following parameters:

- Magnitudes and temperature levels of cooling loads
- Interface requirements between load and refrigerator such as permissible vibration; whether the refrigerator can be integrated with the cooling load or whether the refrigeration must be transferred by heat pipe, convective loop or other thermal link from a remote location; duty cycle; heat flux and temperature level control requirements, etc.
- Nature of spacecraft power supply, particularly as defined in terms of system penalties for both primary thermal power and generated electrical power, since both electrically and thermally powered refrigerators exist.
- Interface requirements between power source and refrigerator. For electrically powered systems this requirement would be quite simple. For thermally powered systems it must be decided whether the source can be integrated with the load or should be located remotely and linked by heat pipe, convective loop or other thermal link.

- Interface requirements between refrigerator and spacecraft heat rejection systems. Heat must be rejected from the refrigerator at temperatures in the general range of earth ambient temperatures. It must be transported from the refrigerator to the radiator by heat pipe, convective loop or other thermal link.

- Maintenance possibilities. Depending upon the particular mission, this will range from zero maintenance to a maximum value considerably less than that permissible for ground and airborne units.

- Required operational lifetime.

It can be seen from the foregoing remarks that this report is not intended to be used as a refrigerator selection guide in itself, since developed refrigeration systems are simply not generally available for space. Rather, it is intended to provide some guidance to the spacecraft systems designer as to the overall state of refrigerator technology so that he may define the shortest and most appropriate refrigerator development program for his particular task. It is possible, of course, that, in certain particular applications existing systems may be suitable, but this situation is not expected to occur often.

Since the primary objective of this report is to present a summary of the current type of refrigeration systems that can be applied to space operations, a major portion of the report is devoted to describing existing (including prototype) cryogenic refrigerators. This information is contained in Section 5 and consists of tables and curves giving coefficient of performance, weight, size, and other performance parameters. However, prior to Section 5 some introductory material on factors influencing refrigerator technology (Section 2), basic refrigeration cycles (Section 3), and refrigerator components (Section 4) is presented in order to provide some continuity and understanding of the different cycles and processes as applied to practical refrigerators.

2.0 FACTORS INFLUENCING REFRIGERATOR TECHNOLOGY

During the nineteenth century, the principal interest in attaining very low temperatures was to attempt to liquify the so-called permanent gases, culminating in the liquification of hydrogen in 1898 and helium in 1908. Early very low temperature refrigeration was achieved by a relatively inefficient cycle involving one or more stages of compression, heat exchange and throttling, known as the Linde or Joule-Thomson cycle.

After this first liquification of helium considerable research was performed on the behavior of helium and other substances at the newly-attainable low temperatures in apparatuses which included integral liquifiers or in the very small numbers of laboratories in the world which possessed a self-contained but inevitably tempermental and inefficient liquifier. These early low temperature experiments gradually revealed the vast possibilities of basic research at very low temperatures and the demand for liquid helium as a basic laboratory utility rapidly increased. In the late 1940's, Arthur D. Little, Inc., began marketing a commercial version of a significantly more efficient and practical helium liquifier using expansion engines and operating on a modified Claude cycle which was developed by Dr. S. C. Collins.

Several hundred units of this refrigerator/liquifier have been built to date and its introduction, in retrospect, marked the beginning of the commercial low temperature refrigerator market. During the 1950's, a small number of other laboratory model refrigerators appeared on the market and several large scale gas liquifiers were built. These all worked on basically similar Claude or Brayton cycles, or the simple Joule-Thomsen cycle. A second most significant event in the development of practical very low temperature refrigerators was the introduction in the late 1940's and early 1950's of a Stirling-cycle-based refrigerator by Phillips Electrical Company. This Stirling cycle refrigerator developed into a highly successful commercial product. The most significant feature of this cycle is the use of regenerative rather than counterflow heat exchangers. Since that time many types of refrigerators have been built which use regenerative heat exchangers which are, in general, derived from this cycle.

From the late 1950's to the present day several new fields of application for refrigerators and liquifiers developed whose influence is almost entirely responsible for the present shape of technology. With the development of the space program, liquid fueled ballistic missiles, and the large scale use of liquified natural gas came a demand for very large scale liquification plants. At the same time there was a growing interest in the field of very low temperature electronics which led to a demand for convenient laboratory-type refrigerators for basic research, and for small flight weight units for airborne detection systems. Although many other particular applications could be noted, it is in these three areas that the greatest amount of design effort, cumulative experience and reliable hardware can be found.

In general, these existing refrigerators are not immediately suitable for space flight use and a substantial number of research and development programs have been pursued by the U. S. Government in order to reduce or eliminate these inadequacies. Some of the programs have been general in nature and have been intended to raise the overall level of technology. Others have aimed at procuring a refrigerator for a particular mission. This research and development activity is sufficiently intense to be considered as part of the current state of technology.

3.0 BASIC REFRIGERATION CYCLES

3.1 General Theory

A refrigerator is a device which absorbs heat at one temperature and rejects it at a higher temperature. In order to perform this operation, an expenditure of mechanical work is required, as shown in Fig. 3-1. According to the second law of thermodynamics, this operation must result in a zero or positive production of entropy. In terms of the quantities shown in Figure 3-1

$$\frac{q_a}{T_a} \geq \frac{q_c}{T_c} \tag{3-1}$$

According to the First Law of Thermodynamics

$$q_a = q_c + W \tag{3-2}$$

Thus,

$$W \geq q_c \left[\frac{T_a - T_c}{T_c}\right] \tag{3-3}$$

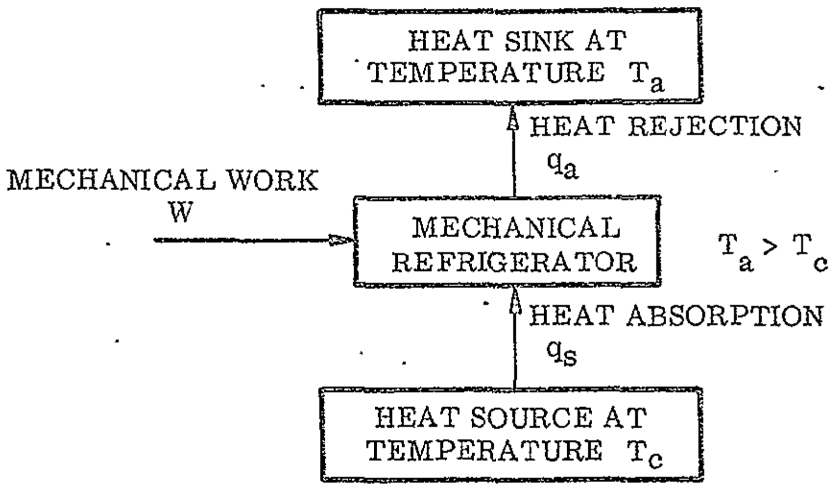


Figure 3-1 Mechanically-Powered Refrigerator Operation

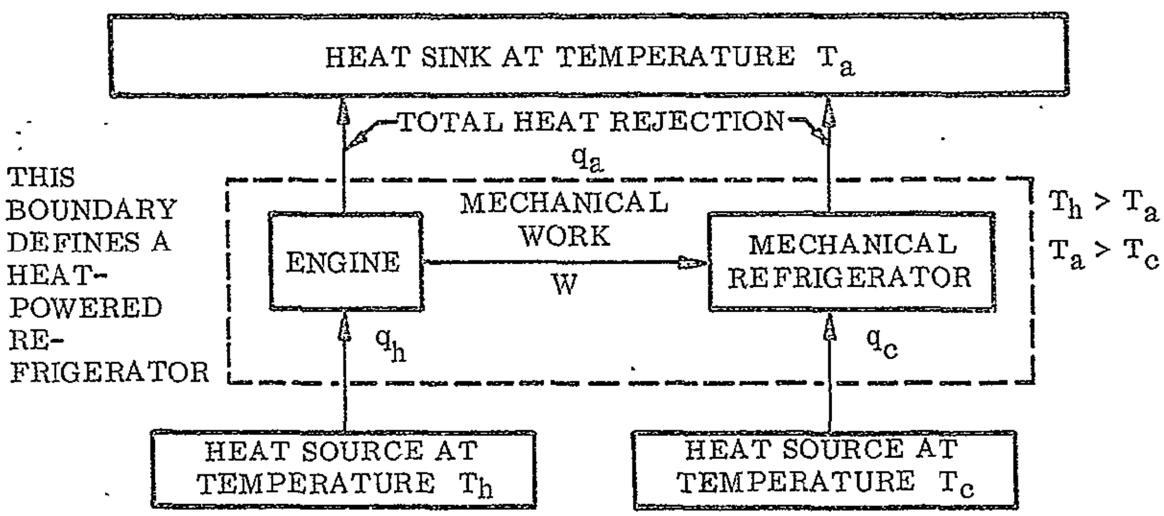


Figure 3-2 Heat-Powered Refrigerator Operation

Eq. 3-1, 3-2 and 3-3 relate to a system of heat source, heat sink, mechanical refrigerator and mechanical work source. It is sometimes desirable to include the work source in the definition of the refrigerator, in which case the situation shown in Fig. 3-2 applies. In this case the work input required by the refrigerator is generated by an engine, which takes heat from a high temperature source, rejects heat to a lower temperature sink and produces work. For the whole system the operation must produce a zero or positive production of entropy. If both engine and refrigerator share a common sink then

$$\frac{q_a}{T_a} \geq \frac{q_h}{T_h} + \frac{q_c}{T_c} \tag{3-4}$$

According to the First Law of Thermodynamics

$$q_a = q_h + q_c \tag{3-5}$$

thus,

$$q_h \geq q_c \cdot \left[\frac{T_h}{T_c} \cdot \frac{T_a - T_c}{T_h - T_a}\right] \tag{3-6}$$

There is no input of mechanical work to the system. Energy is supplied as heat and the whole system may be called a heat powered refrigerator. The performance of a refrigerator is customarily expressed by its "coefficient of performance" "c.o.p."

$$\text{c.o.p.} = \frac{\text{refrigerative effect}}{\text{power supplied}}$$

This function is a satisfactory basis for comparison if all systems are of the type of Fig. 3-1. For most mechanical refrigerators the source of power will be electrical energy converted to mechanical power via an electric motor. The electrical energy will originally have been produced by some process whose operation is completely independent of the refrigerator and its influence may be neglected for comparison purposes. In the case of heat powered refrigerators the coefficient of performance is of less value as a standard. Some heat powered refrigerators operate by electrical resistance heating, while others operate on heat input from a primary source, such as radioisotope or solar collector. If the heat input, q_h, in Eq. 3-6 were to be provided by an electrical heater then based upon electrical power consumption, q_h, would clearly be greater than W in Eq. 3-3 and the refrigerator of Figure 3-1 would be always superior in efficiency. If, however, the means of obtaining heat and electric power are included in the assessment of power required, then a different conclusion may possibly be reached. It was not intended that generalized expressions for the behavior of power sources be included in this report; however, one should consider the power source to obtain the proper perspective on refrigerator systems.

The heat and work interactions implied by the devices shown in Fig. 3-1 and 3-2 are produced by circulating a fluid through the system and causing it to undergo appropriate processes at the heat source and sink. The First Law of Thermodynamics can be written for working fluid in a given process as follows:

$$\begin{bmatrix}\text{heat addition}\\ \text{to the fluid}\end{bmatrix} = \begin{bmatrix}\text{increase in internal energy}\\ \text{of the fluid}\end{bmatrix} + \begin{bmatrix}\text{work performed}\\ \text{by the fluid}\end{bmatrix}$$

Heat may be transferred from the load to the fluid by causing the latter to perform expansion work and replacing this energy with heat from the load either during or after expansion. Heat may be transferred from the fluid to the sink by performing work on the fluid by compressing it and rejecting this energy to the heat sink during or after compression. In either case, the system requires a compressor and sink heat exchanger, and an expander and load heat exchanger. The device of Fig. 3-1 will require a source of mechanical work which may be provided by some type of separate motor. The device of Fig. 3-2 produces the necessary mechanical work by incorporating a heat engine within the system. A heat engine is a reversed refrigerator so this system will require an expander and source heat exchanger and a compressor and heat sink exchanger in addition to the refrigerator components. These components are essential to all refrigerators. Very low temperature refrigerators are distinguished from other refrigerators by the use of a heat exchanger between the load and sink temperatures. Working fluid flowing from the compressor to the expander is pre-cooled by fluid returning to the compressor from the expander. This process permits operation of the refrigerator over a much greater temperature differential than could be obtained by a single expansion.

3.2 Thermodynamic Cycles

It is desirable to keep the values of W and q_h in Eq. 3-3 and 3-6, respectively, as small as possible with respect to q_c. Their values will be a minimum when all cycle processes are reversible, i.e., they produce no overall increase in entropy. Cycles based upon reversible processes can be achieved but are difficult to execute. In fact, practical refrigeration cycles are notable for their very high degree of irreversibility, and successful practical cycles are usually based on expeditious juggling of the many sources of performance loss. This characteristic highly irreversible behavior is traceable to the basic expression for entropy change, dq/T. It is apparent that the entropy changes associated with a given quantity of transferred heat is very much greater at low temperatures than at high temperatures. Much more emphasis must, therefore, be placed upon cold end performance than hot end performance and the resulting practical cycles frequently bear little resemblance to textbook ideal cycles. Nevertheless, it is useful to review the basic ideal cycles in order to obtain a better understanding of their faults and to indicate why the variations shown in the practical cycles of the next section are necessary.

The Carnot Cycle is the best known reversible cycle, shown on the temperature entropy diagram of Fig. 3-3. The compression/cooling and expansion/heating process are accomplished isothermally. The heat transfer processes during these phases are effected over negligibly small temperature differences, resulting in no overall increase in entropy. The fluid is cooled and heated between these temperatures by isentropic expansion and compression, respectively. In practice, the isothermal processes require an infinitely long duration if finite quantities of heat are to be transferred across infinitesimal temperature difference. It is necessary to run practical machines at relatively high speeds and compression is generally accomplished so fast that the process is adiabatic and the working fluid temperature rises. The heat of compression would thus be rejected to the sink after compression and across a finite temperature difference. The same

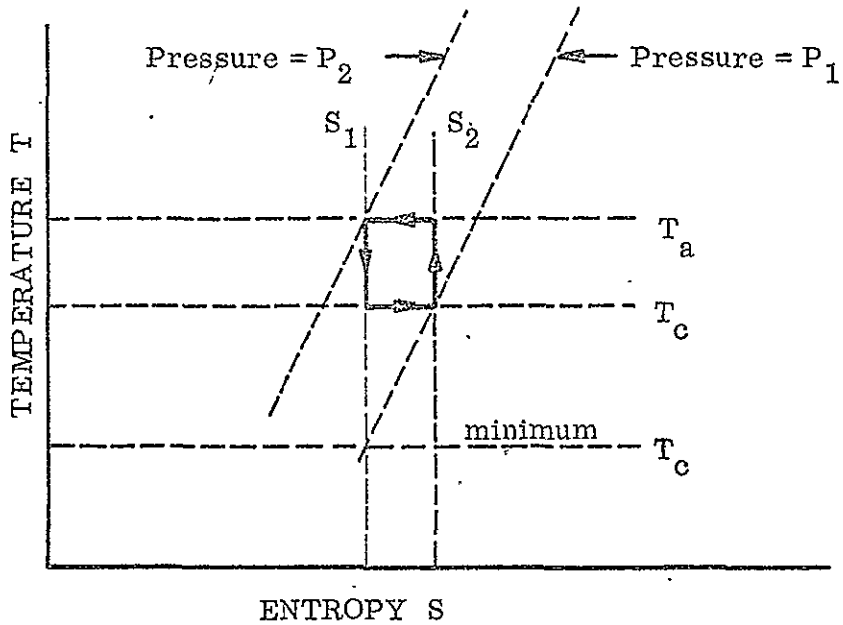


Fig. 3-3 The Carnot RefrigerationCycle

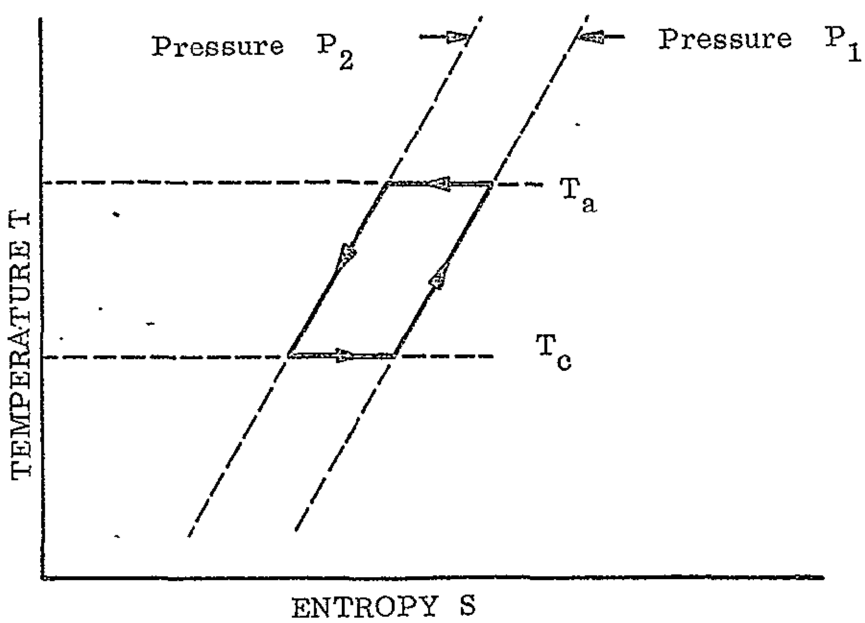


Fig. 3-4 The Stirling Refrigeration Cycle

comment applies to the expander and the heat from the cooling load. Another serious limitation of the Carnot cycle is that the ratio of the minimum sink temperature to the load temperature is governed by the pressure ratio used in the compression and expansion processes.

$$\frac{T_a}{T_c} = \left(\frac{P_2}{P_1}\right)^{\frac{\alpha-1}{\alpha}} \qquad (3\text{-}7)$$

This places a severe restriction upon the practical temperature range.

A second reversible cycle is <u>The Stirling Cycle</u>, shown in Fig. 3-4 on a temperature-entropy diagram. Compression and expansion are performed isothermally, as in the Carnot cycle and the same comments apply. However, heating and cooling is accomplished at constant pressure in a heat exchanger. If the exchanger is 100 percent efficient, which is to say that the temperature difference between the working fluid and exchanger is zero at all points, the cycle is reversible. The Stirling cycle has the desirable quality of being able to span large temperature differences. It would undoubtedly be the most popular refrigeration cycle were it possible to perform reversible heat transfer in all the components. In practical machines the compression and expansion processes are performed so rapidly that they are closer to being adiabatic than isothermal. The Stirling cycle then more closely resembles the non-reversible Brayton cycle, shown in Fig. 3-5. <u>The Brayton cycle</u> is basically a Stirling cycle operated too rapidly. It deserves its own name because recognition of this speed of operation requires separate heat exchangers downstream of the compressor and expanders for transferring the heat that in the Stirling cycle would have been transferred inside these components. It is a reasonable generalization that most refrigeration cycles are modifications or variations of this Brayton cycle. The principal variations are the use of differing types of compressors, heat exchangers and expanders, and the use of varying numbers of these components to exploit the benefits or avoid the problems of non-ideal behavior of the working fluid and the components themselves.

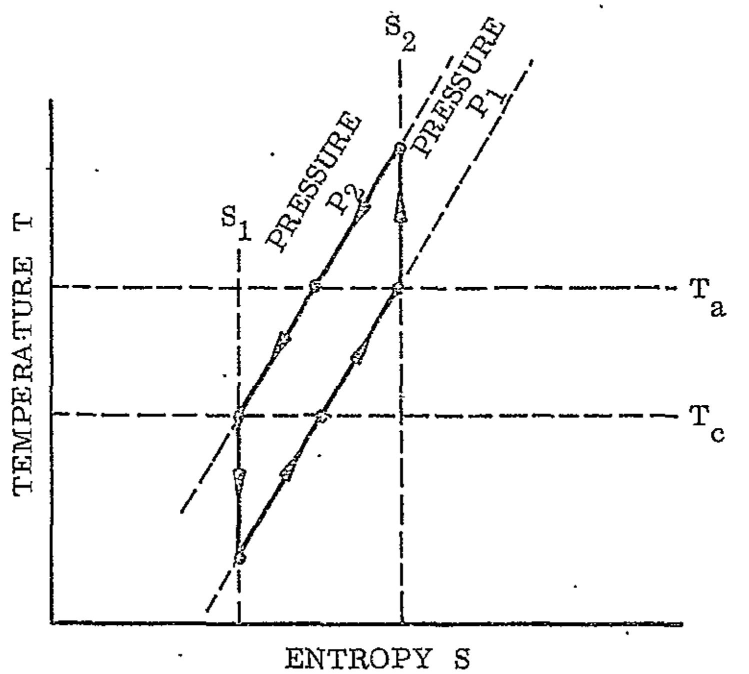


Fig. 3-5 The Brayton Refrigeration Cycle

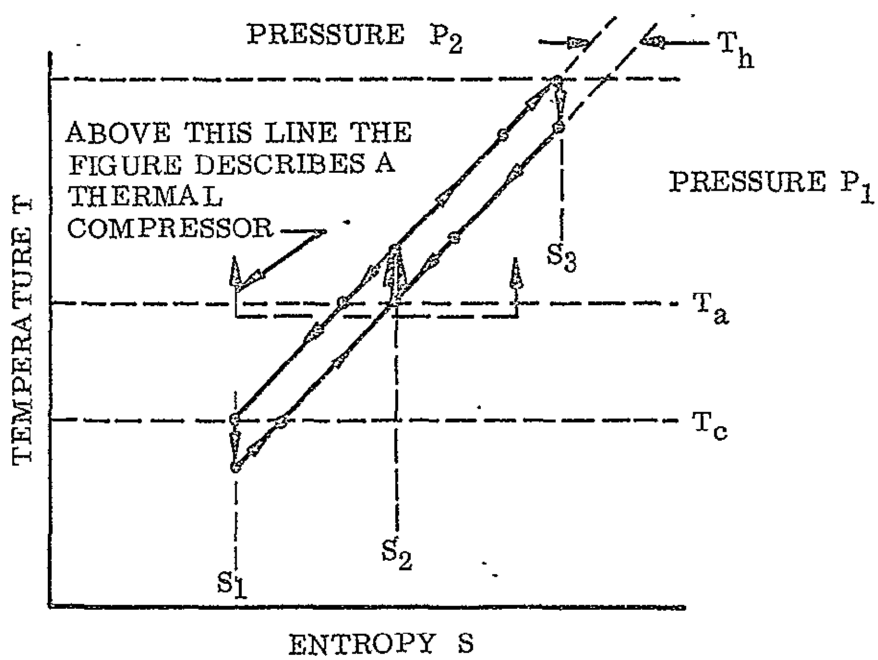


Fig. 3-6 The Heat Powered Brayton Refrigeration Cycle

So far the discussion of cycles has been confined to systems consuming mechanical work. Fig. 3-6 shows the cycle which would be followed by a heat powered system such as that shown in Fig. 3-2. It is composed of a coupled engine and refrigerator which both operate on the Brayton cycle. The compressor handles working fluid for both the refrigerator and the engine. The power produced by the engine is equal to the power required by the refrigerator. This power may be transmitted directly through the fluid or may be transmitted from component to component by mechanical linkages.

4.0 BASIC REFRIGERATOR COMPONENTS (THEORY AND DATA)

The basic components of a refrigerator are compressor, expander or expansion device, and heat exchangers. Some practical refrigerators exist in which some or all of these components are incorporated in the same mechanical device, for example, the Stirling and Vuilleumier refrigerators. Others represent a system comprised of separable components which may be studied and selected independently. In this section, a brief survey is given of the state of development of these three types of components.

4.1 Compressors

General Consideration: The compressor is the component in which most or all of the power required for refrigeration is supplied to the working fluid. It can be seen from Equation 3-3 for very low temperature refrigeration that even in the ideal case the compressive work will be many times the cooling capacity and the compressor is therefore usually the largest and is always the most heavily stressed component in the system. For many terrestrial refrigeration applications the compressor can be located remotely from the heat exchanger, and, has not been subjected to much weight and volume optimization. The most important requirements of a stationary compressor are that it should be reliable and that it should not contaminate the working fluid. To this end, extensive development work has been performed on solid lubricated seals made of carbon or impregnated teflon to eliminate the need for oil.

For terrestrial applications there are two broad types of solid lubricated reciprocating compressors on the market. There are many high capacity process plant compressors which have been designed for high reliability and low overall operating cost, but whose weight and volume have not been optimized. There is also a family of lightweight portable compressors of about 1 horsepower and below, whose operating pressures range up to about 150 psig. These compressors would probably need to be strengthened slightly to take the pressures required by most refrigeration systems, but they do provide an excellent indication of the order of unit weight that can be achieved with moderate design effort.

An alternative approach to providing oil-free compressed gas is to use a conventionally lubricated compressor and apply the design effort to removing the oil from the high pressure discharge stream. If this approach is acceptable, it is possible to use hermetically sealed freon compressors, whose weight is relatively low. The overall operational weight would have to include the weight of the oil separation equipment, however. All other things being equal, the solid lubricated system should show a lighter weight, at the expense of a possibly shorter lifetime.

The possibility of using welded metal bellows to seal the space between piston and cylinder has always been considered an attractive concept but until recently bellows technology has not been able to provide the lifetime necessary to permit design of a competitive unit. At the present time, however, there are several small capacity bellows compressors on the market which have demonstrated impressive lifetimes and low unit weight. This type of compressor is not yet fully proven nor have all its possibilities been adequately explored, and it is thus deserving of greater attention.

For relatively high flow rates, rotary dynamic machines are generally found to be more suitable than reciprocating machines in that the system size and weight can be reduced considerably for a given throughput. Dynamic compressors raise fluid pressure by increasing the kinetic energy of the flow stream and thus converting the kinetic energy to pressure head. This can be achieved by a variety of machine configuations. Generally speaking, rotary dynamic compressors do not provide as much pressure rise per stage as reciprocating positive displacement machines and it is thus found that those rotary machines which provide the highest pressure ratio per stage are the first to become attractive as flow rates are increased and reciprocating compressors become less attractive. With increasing flow rates first the regenerative or drag compressor, then centrifugal compressors, then axial compressors would be considered. The compression efficiency of the regenerative compressor is quite low, that of the centrifugal compressor is somewhat better while the axial flow compressor has the highest efficiency of dynamic compressors. Normally, drag and centrifugal compressors are not attractive by comparison with positive displacement and axial compressors from the point of view of efficiency. They are, however, compatible with gas bearings and retain considerable appeal because of this feature.

Between the fields of application of reciprocating positive displacement and rotary dynamic compressors are rotary positive displacement compressors. This type of compressor can attain compression ratios and efficiencies comparable to those obtained in reciprocating compressors, but at higher flow rates. They are, like the reciprocating compressor, quite heavy, and are not likely candidates for spaceborne refrigeration systems, since at the higher flow rates gas bearing dynamic compressors will be more attractive.

In recent years, more attention has been given to lightweight long-life, contaminant-free compressors suitable for aircraft and spacecraft use. A number of experimental compressors exist in the relatively small size range which eliminate solid seals, bearings or crank shaft. Among these are high speed rotary compressors using gas bearings; a reciprocating compressor operated by linear actuator and metallic spring, and using clearance seals; a reciprocating compressor with linear actuation, gas springs, clearance seals and rotary motion for centralization and valve port alignment; and an electrodynamically operated free piston compressor. These compressors are all in the early development stage.

Work on heat-powered compressors is in a very early stage of development. Such compressors may be defined as a single device which compresses a fluid while accepting heat from a high temperature source and is ejecting it to a lower temperature sink but which does not consume or produce mechanical work. Such systems may be rotary or reciprocating in action. They would effectively constitute the upper portion of Fig. 3-6, in that the thermal compressor constitutes a Brayton cycle engine whose output is exactly sufficient to compress the working fluid of the refrigerator. Also, the working fluid is common to both the engine and the refrigerator.

Performance of Practical Compressors: The minimum amount of power required to compress unit mass of a fluid from a pressure P_1 to a pressure P_2 is that required by the isothermal process and is equal to $RT \ln (P_2/P_1)$. The actual work required is increased by power losses in transmitting energy from the power supply to the working fluid, such as motor losses and friction losses at bearing and seal surfaces. If the compressor has valves then the flow pressure losses through these valves will require the fluid in the compressor to be compressed from a lower to a higher pressure than the corresponding fluid pressures in the outside circuit, and hence increasing the power requirement. During the compression process some heat will be transferred from the fluid to the walls of the compressor, but it will invariably be insufficient to prevent the fluid temperature rising and the process will thus not be reversible and will require more power than isothermal compression. Also during compression there will be turbulence within the compression spaces which will result in further heating of the fluid by internal friction, and hence further deviation from the isothermal process. Finally, not all the fluid compressed will be available in the refrigeration cycle because there will be leakage from high to low pressure sides through clearance spaces, and in the case of gas bearing systems a portion of the flow will be required by the bearings.

In the case of the thermal compresssor the losses associated with the valves, leakage, and non-isothermal compression will still be present. The bearing and seal losses will usually be much less since there will be no moving boundary to the device through which mechanical work must be transmitted. Since the thermal compressor is an integrated heat engine and compressor there will be heat losses associated with the heat engine function as well as the compressor function.

The thermodynamic performance of a compressor takes into account all effects internal to the compressor and is usually expressed by the isentropic efficiency, η_{ad}, or the isothermal efficiency, η_{is}

$$\eta_{ad} = \frac{\text{actual work required for compression}}{\text{isentropic work, } W_{ie}} \qquad (4\text{-}1)$$

$$\eta_{is} = \frac{\text{actual work required for compression}}{\text{isothermal work, } W_{is}} \qquad (4\text{-}2)$$

$$W_{ad} = RT_1 \frac{\gamma}{\gamma-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma}{\gamma-1}} -1\right] \qquad (4\text{-}3)$$

$$W_{is} = RT_1 \ln (P_2/P_1) \qquad (4\text{-}4)$$

The theoretical isentropic work of single stage, compressor is shown on Fig. 4-1. The actual work of compression is obtained using equation 4-1 and a known value for isentropic efficiency η_{is}. The value of η_{is} varies with the type of compressor, fluid and operating conditions, and cannot be presented in a convenient graphical or tabular form but representative values may be quoted for good typical designs. For reciprocating compressors an adiabatic efficiency of 0.85 to 0.90 may be assumed. For centrifugal compressors η_{is} can range from 0.50 and lower for very small units, up to about 0.82 for well designed high capacity machines. Drag or regenerative compressors have η_{is} figures characteristically less than 0.50. In calculating the required motor size allowance must be made for motor efficiency, mechanical efficiency, leakage or loss of compressed fluid, etc.

Having determined the required motor input power Figures 4-2 and 4-3 may be consulted to obtain compressor weight and volume, respectively. These figures were obtained by plotting data for comparatively low weight electrically driven compressors for terrestrial applications. The data are plotted against input power because a substantial portion of the weight is that of the electric motor. The weight of the compression components will also vary with number of compression stages and operating pressure, but the variations due to

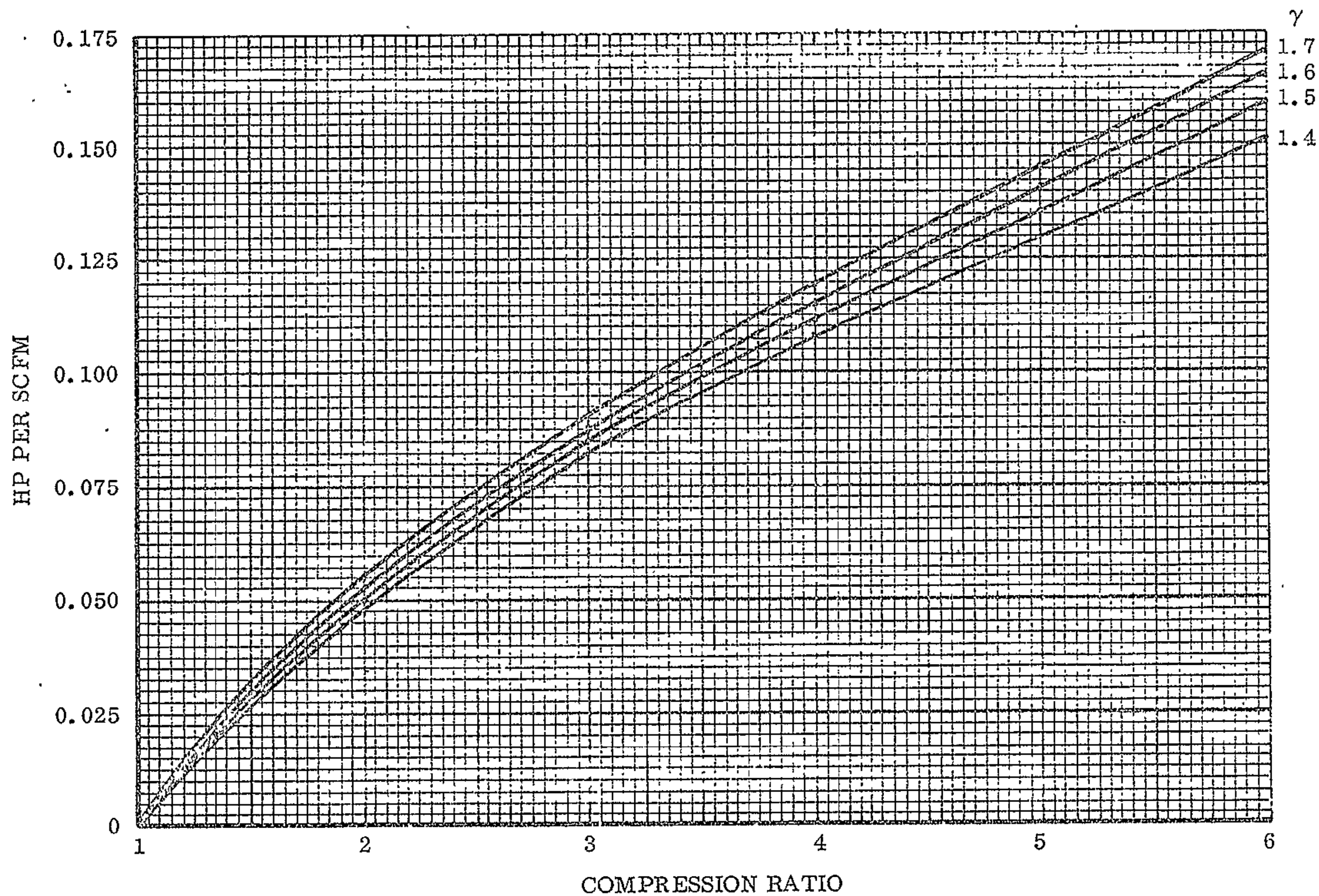


Fig. 4-1 Theoretical Isentropic Work for Single Stage Compressor

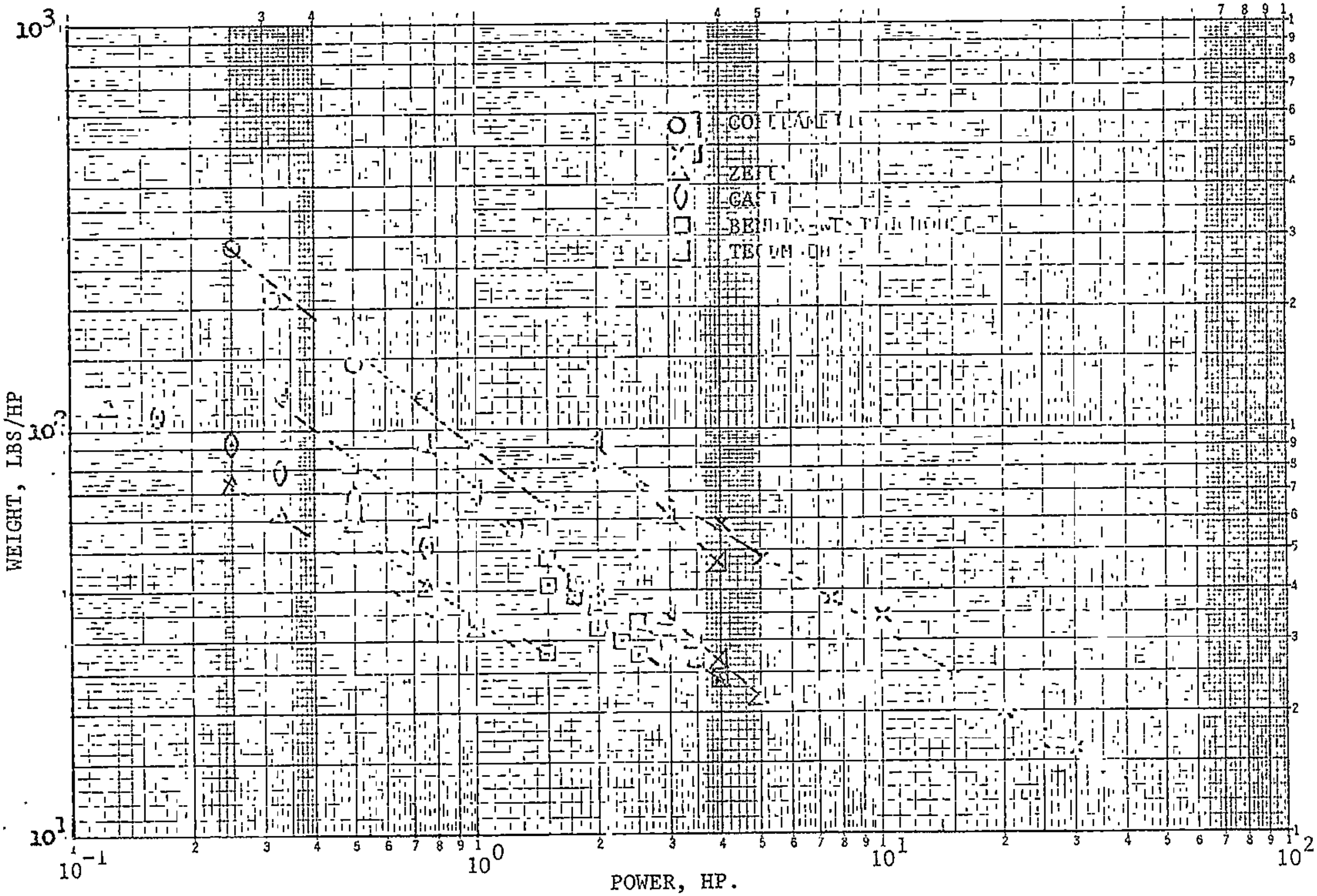


Fig. 4-2 Compressor Weight as a Function of Power Required

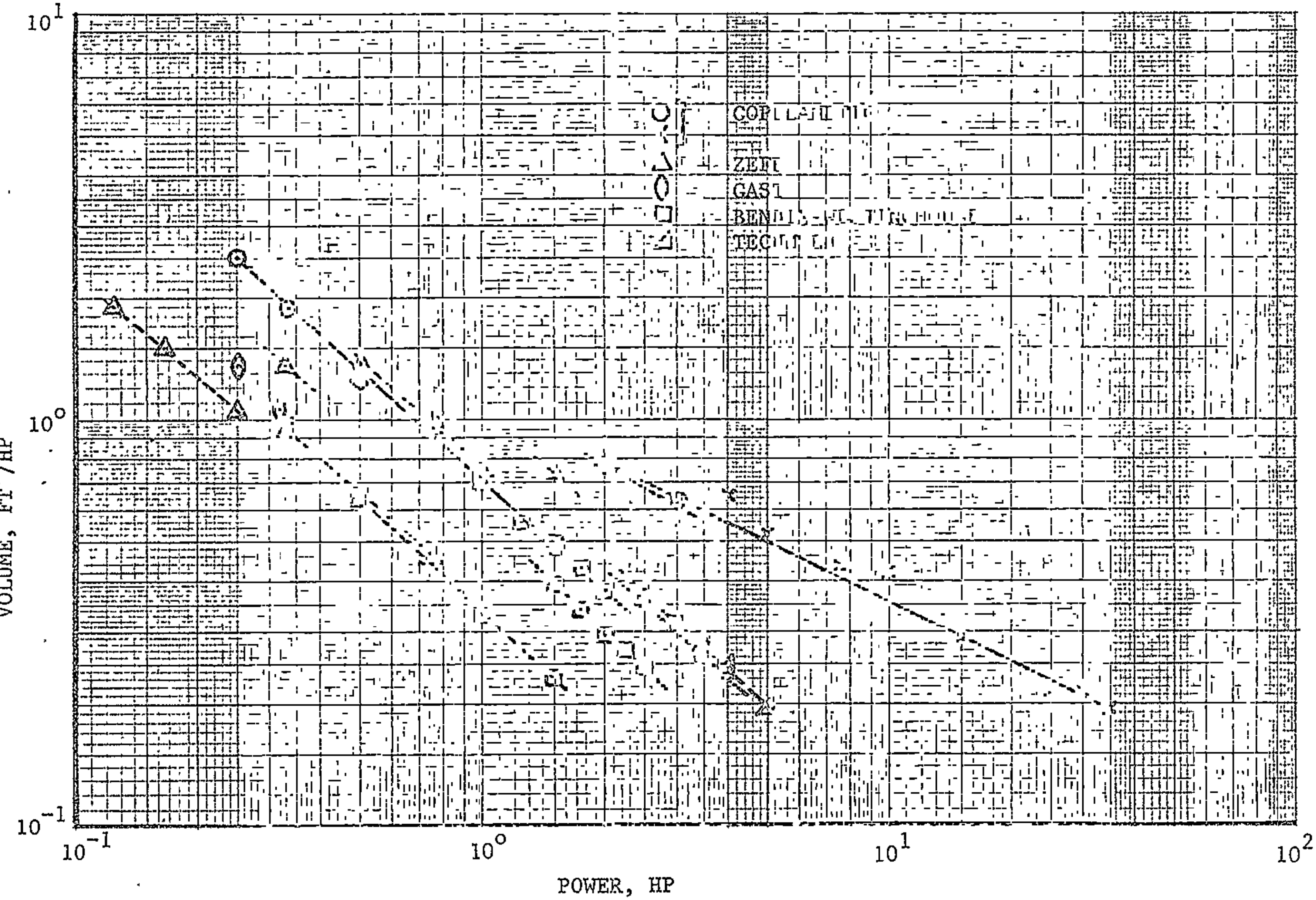


FIG. 4-3 COMPRESSOR VOLUME AS A FUNCTION OF POWER REQUIRED

these effects are less than those between different design concepts. The data shown for Zefex [1] and Gast [2] compressors are typical of commercially available lightweight, oil-free, solid lubricated compressors. Their range of application, as dictated by the market rather than technology, is up to 1 to 2 HP for compression from 14.7 psia to 35-175 psia. They consist of little more than lightweight compression cylinders attached to conventional low cost electric motors. The extent to which their weight could be reduced would be dependent mainly on possible motor lightening and to a much lesser extent on compression piston and cylinder lightening. Data are also shown for Tecumseh[3] and Bendix-Westinghouse[4] hermetically sealed oil-lubricated freon compressors. The weights shown are for the bare compressor, excluding the oil separation components that would be needed for a low temperature refrigeration application. The data are significant in that some attention has been given to reducing the weight of this type of compressor. The electric motor is integrated with the compression cylinders in these designs. They are known to be strong enough to operate as helium compressors at pressure levels up to 350 psig and are offered as part of many split component refrigerators. Their power range is generally up to about 3 HP. Also plotted on Figures 4-2 and 4-3 are data for Copeland hermetically sealed freon compressors[5]. These compressors are not weight optimized and are built in the more conventional manner of detachable motor and compression cylinders. However, Copeland makes a range of similarly designed compressors from ¼HP to 35 HP and a plot of the weight and volume data for these compressors is helpful in showing trends. The Copeland data show two linear relationships reflecting air cooling for lower powers and water cooling for the higher powers.

Meaningful data on the prototype systems mentioned earlier are difficult to obtain but an indication of the performance of gas bearing compressors can be obtained from the following figures. Maddocks[6] reports a gas-bearing six-stage centrifugal compressor of approximately 163 lbs. and 9600 watts input, giving a weight of 13 lbs./HP at 12.4 HP. Breckenridge[7] reports a gas-bearing two-stage rotary-reciprocating compressor design of about 72 lbs and 746 watts input, giving a weight of 77 lbs/HP at 0.93 HP.

4.2 Expanders

Configurations: An expander is a device in which a fluid may perform work against the environment which, in the context of refrigeration system, means the environment outside the refrigerator. Two principal types of expanders are commonly used - reciprocating position displacement and rotary dynamic. The work produced by these expanders can be transmitted to the environment mechanically or, in the case of the rotary expander, by generating electricity at the low temperature and dissipating it in the environment.

The reciprocating expander is much like an increased reciprocating compressor in operation. Fluid is admitted to an expansion cylinder at high pressure, is expanded against a piston and is then discharged from the cylinder at low pressure. Rotary dynamic expanders, or turbines, can be constructed in a variety of ways. For lower flow rates radial impulse turbines are generally used. For higher flow rates axial impulse and radial reaction turbines may be considered. A survey of individual applications[8] suggests that the range of flow rates and cooling loads covered by this report is best handled by radial impulse machines. The flow rates usually encountered in relatively small capacity turbines are usually so low that even with radial impulse turbines partial admission must be used. In a pure impulse turbine the pressure head of the working fluid is converted entirely to kinetic energy in the inlet nozzles and the turbine wheel operates essentially like a Pelton wheel to remove the kinetic energy. Reaction turbines require a static pressure differential across the wheel since part of the expansion is performed in the wheel passages. As size is reduced it becomes increasingly difficult to maintain this pressure head. The axial turbines, both impulse and reaction, have a generally higher flow capacity than radial turbines and are thus to be found in high flow applications.

Performance of Practical Expanders: Representative unit data for expander weight and volume are difficult to obtain since few have been built by comparison with compressors. However, practical refrigerator systems show values of coefficient of performance of about 1/20 at $100^{\circ}K$ to 1/200 at $20^{\circ}K$ so that the expander power will be between 5 and 0.5% of the compressor power. Thus its unit weight and volume need not be known to as high a degree of accuracy as those for the compressors. In order to obtain a conservative estimate of the order of magnitude of size of expanders it is suggested that the following approximations be made. Compressors and expanders are both power transmission devices and it is proposed that the assumption be made that the system size per unit power transmitted are the same for both types of devices at a given value of transmitted power. Expander weight and volume should thus be estimated from Figures 4-2 and 4-3, in which case the abscissa will refer to expander power rather than compressor power.

It is important to know the isentropic efficiency of the expander in order to perform a cycle analysis because the efficiency of the expander will have a great bearing on the overall system size and weight. The remarks and equations relating to compressor work and efficiency apply to the expansion process. The loss mechanisms are the same, with one important exception. In the compressor, heat transfer from the hot fluid to the compressor walls during compression will tend to increase compression efficiency. In the expander the walls will be warmer than the fluid and heat transfer from them to the expanding fluid will tend to decrease expansion efficiency. Data for isentropic efficiency of reciprocating expansion engines and overall efficiency (including electric generator) for radial impulse turbines are shown in Figures 4-4 and 4-5. The data are to be regarded as showing the rough order of magnitude and the trends, rather than exact information, since each expander is designed for a different application. Efficiency is not a function of pressure ratio or output primarily and these parameters have been selected only for graphic convenience.

4.3 Heat Exchangers

General Consideration: There are three principal types of heat exchangers in a refrigerator system - the compressor after-cooler, the cooling load exchanger and the main heat exchanger in which heat is exchanged between the working fluid streams passing to and from the expander. The performance of these three types of exchangers influences refrigeration systems in differing ways.

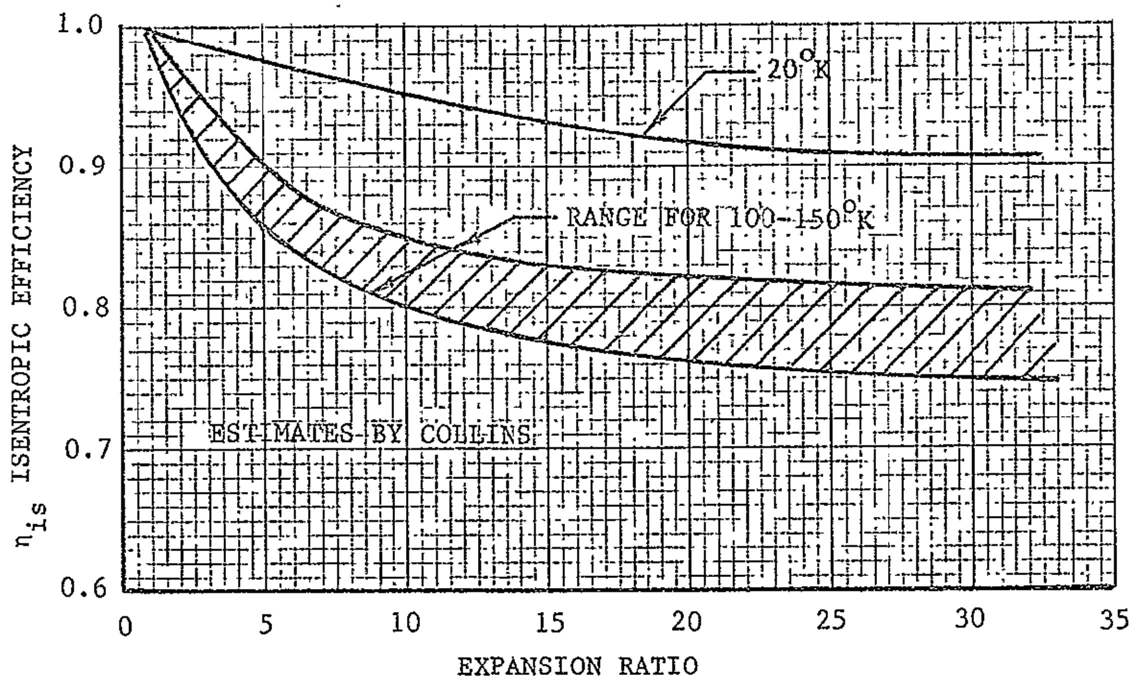


Fig. 4-4 Efficiency of Reciprocating Expansion Engine

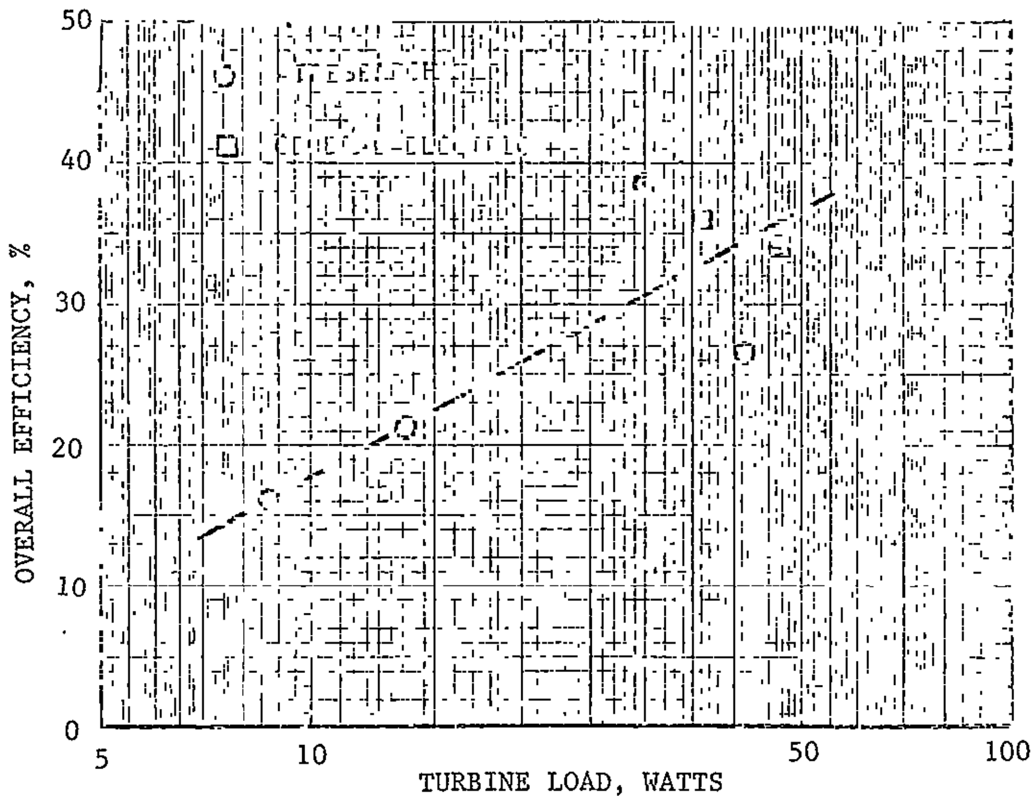


Fig. 4-5 Efficiency of Radial Impulse Turbines

The most important heat exchanger is the main exchanger because it must operate over the relatively large temperature range of load temperature to sink temperature. The two principal configurations used for this exchanger are shown schematically in Figures 4-6 and 4-7. In the counterflow heat exchanger, Figure 4-6, the high pressure and low pressure fluid streams pass continuously through separate flow channels which are in thermal contact. Heat is transferred from the warm high pressure fluid through the channel walls to the cooler low pressure fluid. At each point in the exchanger the properties and parameters of the fluid flows are constant with time. In the regenerative heat exchanger, Figure 4-7, the warmer high pressure stream and the cooler low pressure stream flow alternately in opposite directions through the same flow channel. Initially, the warm high pressure fluid flows through the exchanger giving up its heat to the exchanger walls. After a period of time the high pressure flow is then temporarily discontinued and cold low pressure fluid is passed through the exchanger in the opposite direction, picking up heat from the exchanger walls. After a similar period the low pressure flow is stopped and the high pressure flow resumes. This type of exchanger thus achieves its heat transfer by temporary heat storage on the exchanger walls rather than heat transfer through the walls of the flow channels. The heat transfer area per unit volume can be increased considerably if the transfer area has no structural responsibility such as maintaining separation of the flows. Materials such as fine screen or small spheres can be used to pack the regenerator to provide heat storage capacity. At low temperatures the specific heat of solids falls off very markedly, however, and the thermal storage capacity of practical regenerators becomes very small. Refrigerators using regenerative exchangers currently have a lower limit of operation of about $7^{\circ}K$, with poor performance due to this effect appearing below about $20^{\circ}K$. The intermittent nature of the flow in the regenerator tends to reduce the chances of heat exchanger fouling. A disadvantage of the regenerator is that its operating pressure must vary cyclically and losses are introduced because this process cannot be performed. For this reason, regenerative exchangers are best applied to systems in which the exchanger is in constant communication with the expander and, in some cases, the compressor, so that the cyclic pressure variation is at least accomplished smoothly. Alternatively, the flow reversal would have to be performed by valve

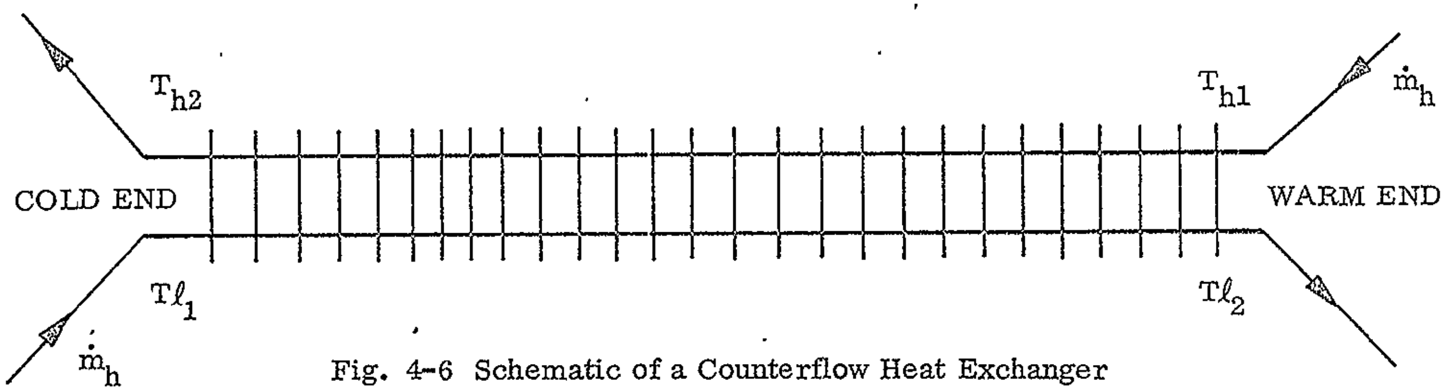
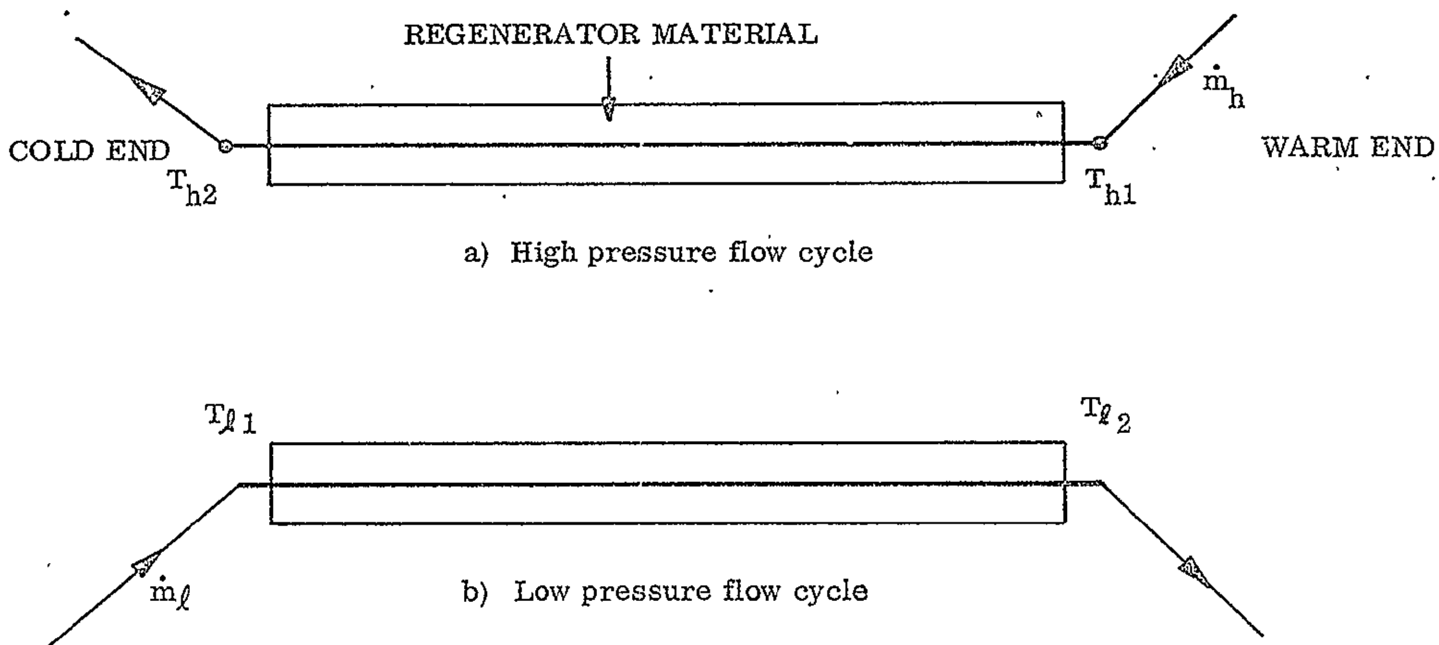


Fig. 4-6 Schematic of a Counterflow Heat Exchanger



a) High pressure flow cycle

b) Low pressure flow cycle

Fig. 4-7 Schematic of a Regenerative Heat Exchanger operation, resulting in large losses due to sudden expansion of either the inflowing high pressure stream or the outflowing regenerator contents.

The <u>counterflow heat exchanger</u> has the disadvantage that fluid containment and heat transfer are accomplished in the same channels. This leads to inefficient use of materials of construction and increased weight per unit heat transfer area. Miniaturization is difficult for this reason. The counterflow exchanger does not, however, depend upon the specific heat of the walls for its operation and thus can operate at all temperatures. It is also considerably easier to design, since its performance is not time dependent.

<u>Heat Exchanger Design</u>: It is impossible to present generalized heat exchanger performance data in the manner that is possible for compressors and expanders. This is because heat exchanger performance is not limited by natural phenomena so much as by system optimization criteria. Heat exchangers can be built to any desired degree of efficiency as long as the weight, volume and pressure drop penalties can be paid. There is a virtual infinity of possible heat transfer surfaces, each with its own heat transfer and pressure drop characteristics and, many materials of construction may be considered. The following sections are thus brief surveys of possible analytical approaches. For actual design data the literature must be consulted, for example Kays and London, "Compact Heat Exchangers" (McGraw Hill, 1954).

The above remarks relate principally to counterflow heat exchangers. Regenerative heat exchangers are somewhat simpler to characterize because the heat transfer surfaces for low temperature exchangers have been confined to screens or spheres. Also, it is noted that regenerative exchangers are usually used in machines using reciprocating expanders and in a broad category of machines the exchangers and expanders comprise an integrated unit. For these systems a rough plot can be made of practical expander-exchanger performance, and this has been done in the appropriate following section.

<u>Counterflow Heat Exchangers</u>: Figure 4-6 shows a schematic of a counterflow heat exchanger. The high and low pressure mass flow rate and entering temperatures are $\dot{m}_h$, $\dot{m}_\ell$, T_h, T_ℓ, respectively. The object of a design analysis is

to determine the fluid exit temperatures T_{h_2} and T_{ℓ_2} in terms of the given entering conditions, the heat exchanger geometry, and the fluid properties. A heat balance for a cross section of the heat exchanger of length dx gives the following equations, ignoring longitudinal conduction:

$$-\dot{m}_h . \frac{dh_n}{dx} . dx = \dot{m}_\ell . \frac{dh_\ell}{dx} . dx \qquad (4\text{-}5)$$

$$\dot{m}_\ell \frac{dh_\ell}{dx} . dx = (US) . (T_h - T_\ell) . dx \qquad (4\text{-}6)$$

Here (US) is the overall heat transfer conductance, given by

$$\frac{1}{US} = \frac{1}{U_\ell S_\ell} + \frac{tw}{k_w S_w} + \frac{1}{U_h S_h} \qquad (4\text{-}7)$$

S_ℓ, S_h and S_W are the areas per unit length for heat transfer. U_ℓ and U_h are the heat transfer coefficients between low pressure and high pressure fluid streams, respectively, and k_w and t_w are the thermal conductivity and wall thickness for the wall separating the two flows.

The heat transfer coefficients can be found from empirical data reported in the literature in the general form

$$N_u = f\ (R_e, P_r) \qquad (4\text{-}8)$$

where

$$N_u = \frac{UD_e}{k} \qquad (4\text{-}9)$$

$$R_e = \frac{\dot{m}_\mu D_e}{A_f\ \mu} \qquad (4\text{-}10)$$

$$P_r = \frac{Cp\mu}{k} \qquad (4\text{-}11)$$

D_e and A_f are the flow channel hydraulic diameter and flow area, respectively. k, Cp and μ are the thermal conductivity, specific heat, and viscosity of the working fluid, respectively.

A typical empirical expression in the form of equation is the following, (appropriate for flow inside round tubes):

$$Nu = 0.023\ R_e^{0.08}\ P_r^{0.4} \tag{4-12}$$

For flow outside round tubes there are many geometric possibilities leading to many heat transfer correlations and the literature should be consulted for particular data. A more commonly occuring situation is flow perpendicular to round tubes, for which the following correlation is prepared:

$$Nu = C\ .\ R_e^{0.8}\ P_r^{0.33} \tag{4-13}$$

C is 0.26 for in-line tubes and 0.33 for staggered tubes. Here R_e is based upon the tube diameter and the minimum flow area past the tubes.

In general, C_p, k and μ and hence U, are temperature dependent, and are thus functions of x. Also, at the temperatures existing in very low temperature refrigerators the working fluids do not behave as ideal gases and thus the expression Cp dT cannot be substituted for dh in Equations 4-5 and 4-6. Equations 4-5 and 4-6 must therefore be integrated by numerical methods since the non-ideal gas behavior and temperature dependence of Cp, k and μ cannot be expressed in simple analytical forms which permit direct integration of the equations.

As a result of a numerical integration the outlet temperatures T_{h_2} and T_{ℓ_2} may be determined.

It is customary to express the performance of a heat exchanger in terms of its effectiveness, ε, defined as

$$\varepsilon = \frac{\text{actual heat transfer}}{\text{maximum theoretical heat transfer}} \tag{4-14}$$

Using this definition and assuming a constant specific heat for the working fluid through the high and low pressure sides of the heat exchanger, it is possible to derive an expression for heat exchanger effectiveness in terms of the geometry, mass flow rates and fluid specific heats from which outlet temperatures can be found for specified inlet temperatures. This approach is of great general utility in exchanger design but is not recommended for the design of exchangers for very low temperature refrigerators, where property changes are too severe to be successfully accounted for by average values. In this case the effectiveness must be found from the temperatures calculated by the numerical solution and therefore ε is a somewhat redundant quantity if the recommended rigorous temperature solution is employed.

For those cases where an approximate analysis of the exchangers is desired, the following widely-employed expressions may be used for counterflow heat exchangers

$$\varepsilon = \frac{C_h \, [T_{h_1} - T_{h_2}]}{C_{min} \, [T_{h_1} - T_h]} \qquad (4\text{-}15)$$

where C_h and C_ℓ are the averaged quantities $(mC_p)_h$ and $(mCp)_\ell$ respectively and C_{min} is the smaller of C_h and C_ℓ.

The effectiveness is expressed analytically by

$$\varepsilon = \frac{1 - \exp\,[-N(1-C_{min}/C_{max})]}{1 - (C_{min}/C_{max})\,\exp\,[-N(1-C_{min}/C_{max})]} \qquad (4\text{-}16)$$

N is the number of transfer units, given by

$$N = \frac{USL}{C_{min}} \qquad (4\text{-}17)$$

L is the heat exchanger length and U is the averaged heat transfer coefficient.

For a given application C_{min} will be specified. The heat transfer area, SL, will depend upon the type of heat exchanger geometry selected. The heat transfer coefficient, U, will be given by an equation of the form of 4-7 whose constant and exponents have been determined experimentally. From these quantities and either question 4-15 or a numerical solution for ε, the relationship between weight, volume and effectiveness can be obtained for a particular geometry and flow rate.

Flow through heat exchanger passages is accompanied by pressure losses which may be calculated using well-known standard analyses.

For an element of heat exchanger passage in either flow path the pressure drop can be expressed as

$$dP = \frac{\dot{m}}{A_f} \frac{d}{dx} \left(\frac{\dot{m}}{\rho A_f}\right) dx + \frac{4\dot{m}^2 \, f \, dx}{2A_f^2 \, \rho \, D_e} \qquad (4\text{-}18)$$

and $$f = 0.00140 + 0.125 \, R_e^{-0.32} \qquad (4\text{-}19)$$

As with the heat transfer calculation, it is recommended that a numerical integration be made along the flow path so as to account for property variations with temperature.

If an approximate expression is required for conditions under which $(\dot{m}/A)$ is constant, the fluid properties are constant, and perfect gas behavior exists, equations 4-18 and 4-19 can be integrated to give

$$P_1^2 - P_2^2 = 2 \, R\bar{T} \left(\frac{\dot{m}}{A_f}\right)^2 \left[\ln \frac{P_1 T_2}{P_2 T_1} + \frac{4 \, \bar{f} L}{2 \, De} \right] \qquad (4\text{-}20)$$

$\bar{f}$ and $\bar{T}$ are mean values of friction factor and temperature respectively.

The preceding remarks apply strictly to load, main and after-cooler heat exchangers. In the case of the load and after-cooler exchangers the assumption of constant fluid properties is more usually acceptable and use of the simple equation (4-16) would therefore be more reasonable.

Regenerative Heat Exchangers: The analysis of a regenerative heat exchanger is basically similar to that of the counterflow exchanger. The differences are that the fluid and surface temperatures vary with time as well as position. The heat balance equations, ignoring longitudinal conduction, for a section of heat exchanger of length dx is thus:

$$M\,dx\,\frac{\partial h_m}{\partial t} = US\,[T_g - T_m]\,dx \qquad (4\text{-}21)$$

$$-\,US\,[T_g - T_m]\,dx = \frac{\partial(\dot{m}_g h_g)dx}{\partial x} + A_f dx\,\frac{\partial(\rho_g u_g)}{\partial t} \qquad (4\text{-}22)$$

The subscripts refer to gas and metal temperatures and M refers to the heat exchanger mass per unit length. These equations are obviously more complex than those governing counterflow exchangers. Unless many simplifying assumptions are made the equations cannot be solved in closed form. In low temperature applications these simplifications, such as assumption of infite specific heat of the matrix, zero heat storage in the fluid and constant fluid properties, are invalid.

Regenerative exchanger efficiency must therefore be found using numerical analysis or by consulting published tables of solutions. The weight and volume of regenerative exchangers will be found from a knowledge of the geometry and the material of construction.

It was noted earlier that regenerative heat exchangers are often used in systems in which the expander and heat exchanger are built as a unit. A system employing such an expander-exchanger combination requires only a compressor to form a complete refrigeration system.

Some weight and volume data for practical exchanger-expanders are presented in Figures 4-8 and 4-9. They refer to several different types of expansion process but a degree of correlation which is satisfactory for the present purposes is apparent. Figures 4-8 and 4-9 show clearly the rapid rise in weight and volume with decreasing operating temperature. Figure 4-10 is a less satisfactory correlation of weight against cooling power for various temperatures.

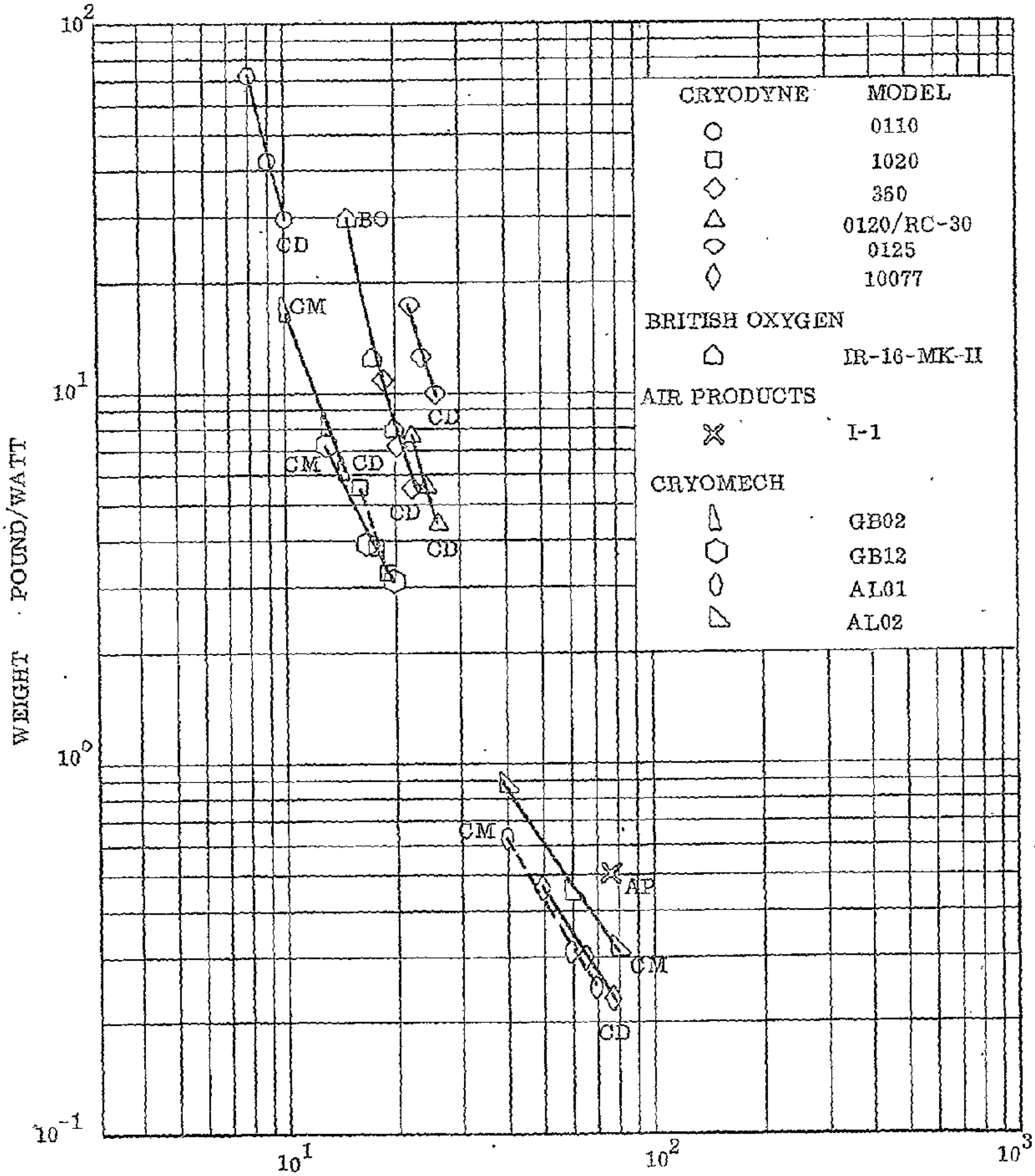


Fig. 4-8 Weight of Exchanger-Expanders Versus Temperature

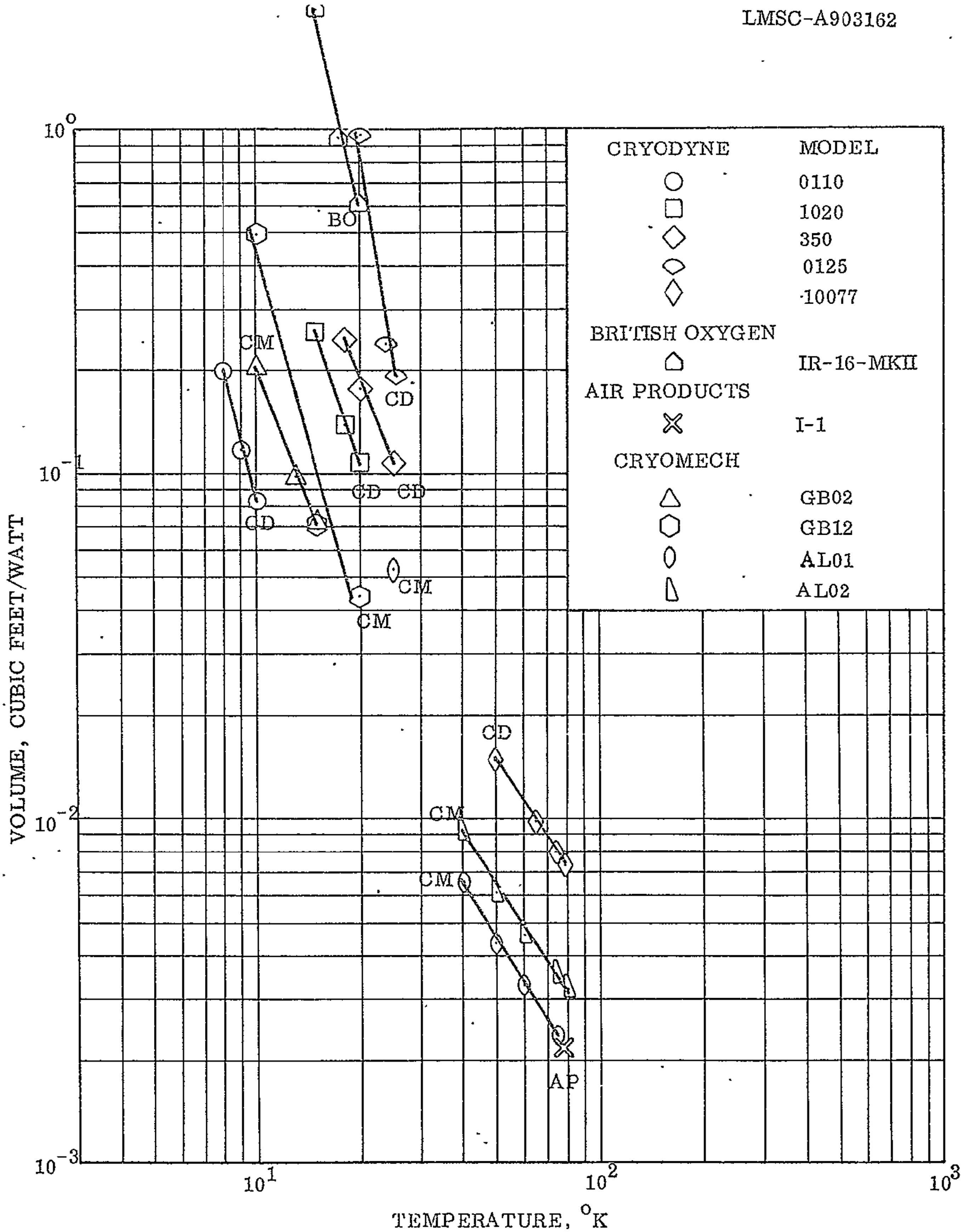


Fig. 4-9 Volume of Exchanger-Expanders Versus Temperature

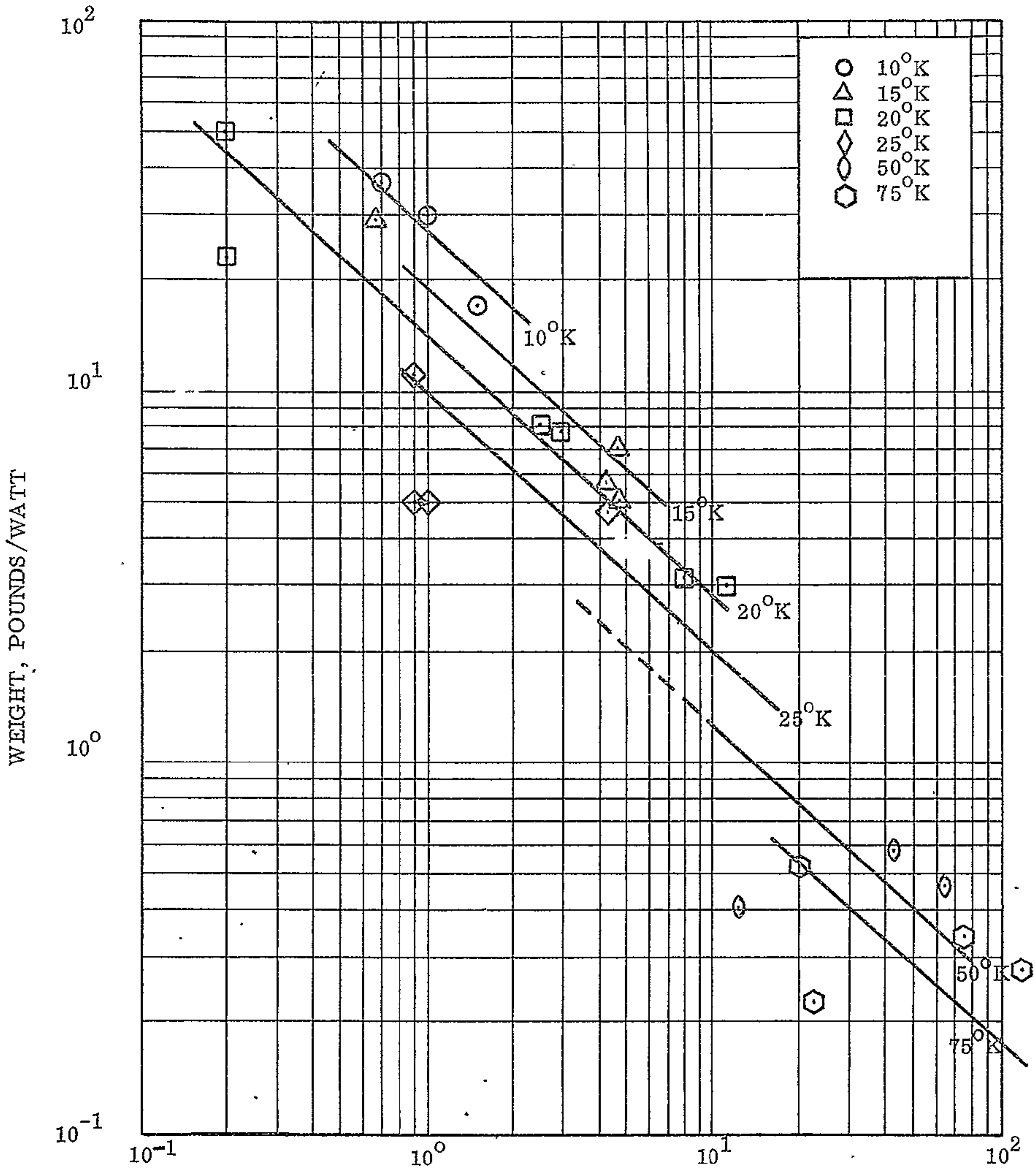


Fig. 4-10 Weight of Exchangers-Expanders Versus Cooling Power

These curves may be used to estimate the weight of split component systems. For a given load temperature the coefficient of performance gives the required compressor power per watt of cooling. The size of the complete system is then obtained by adding the component size from this section and section 4.1.

5.0 EXISTING REFRIGERATOR SYSTEMS

This section describes practical refrigeration cycles which use the thermodynamic principles and the components described in the previous sections. Although most cycles are basically related to the Brayton cycle, their practical execution has led to a wide variety of configurations. Most refrigeration systems can, however, be placed into one of a small number of subgroups whose members are closely related. The cycles selected for discussion were limited to those which it was felt had potential for satisfying the requirements of this study, i.e., potential for long term operation, low weight and volume and high thermal efficiency. The data on operating characteristics of the various units has been obtained from an extensive search of the literature and from contacts and discussions with the companies and agencies engaged in the production and development of the units.

These systems can be divided into two broad groups; one employing counterflow heat exchangers and another employing regenerative heat exchangers (see Fig. 5-1). If counterflow exchangers are used then the working fluid flow rate at any point in the refrigeration system is constant with time. The working fluid flows at constant rate and direction through all the system components. These components can hence be designed for continuous steady state operation at prescribable conditions. This category includes Claude, Joule-Thomson, and orthodox Brayton cycle systems. On the other hand those systems which employ regenerative heat exchangers must make some provision for intermittently reversing the direction of flow and alternately compressing and decompressing the working fluid in the regenerator. This can be performed in a refrigerator in which the cycle processes are executed successively in different regions of the same component. The working fluid is compressed while it occupies the warm end and the regenerator spaces, and is expanded while it occupies the cold end and regenerator spaces.

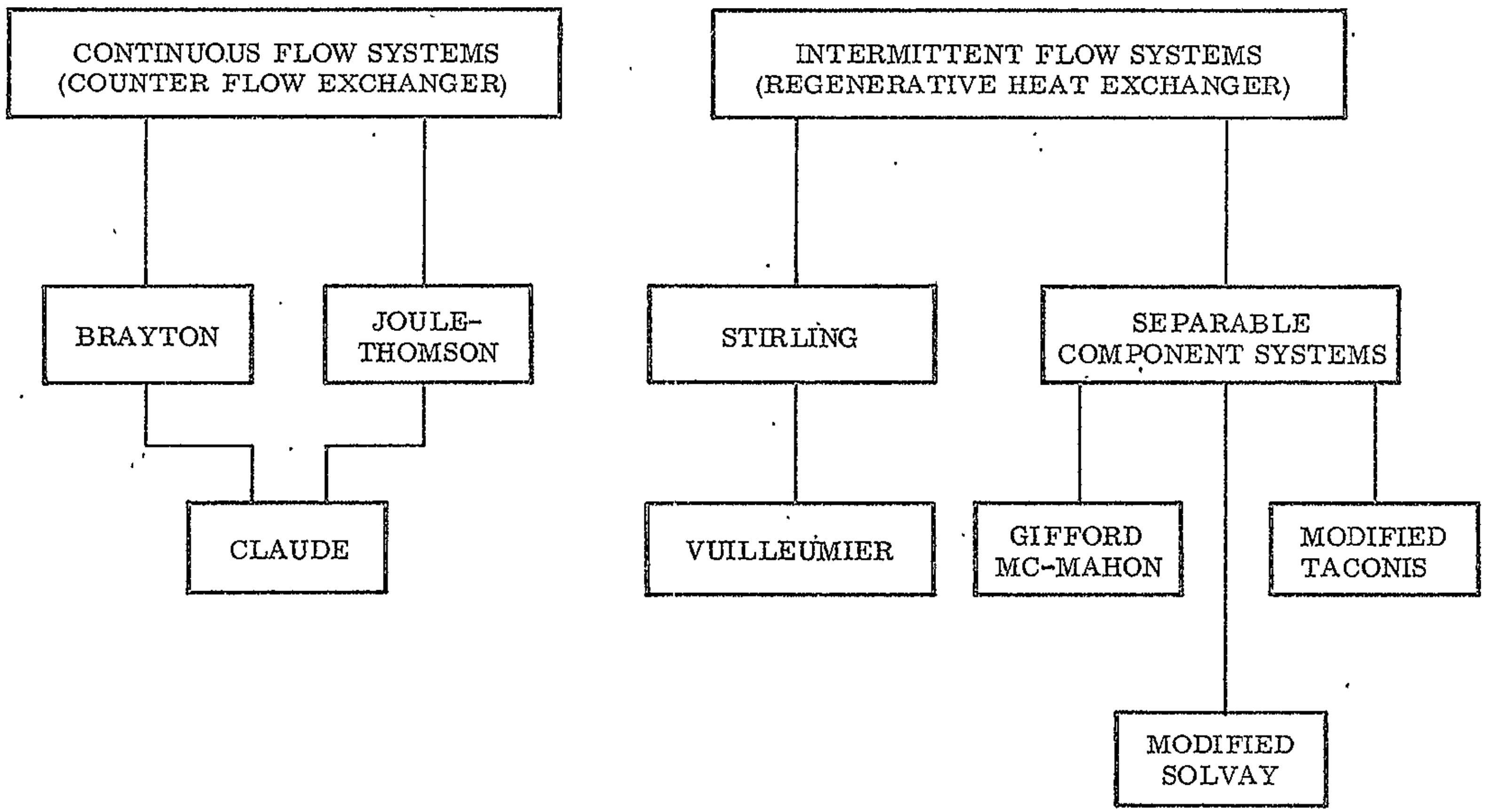


Fig. 5-1 Cycles Included for Analysis

The first heat engine to use regenerators was built by Robert Stirling and an important distinction should be made here between the Stirling cycle and what are commonly known as Stirling engines or refrigerators. The ideal Stirling cycle could be executed with a counterflow heat exchanger just as well as a regenerative heat exchanger. On the other hand, practical so-called Stirling devices actually operate on a Brayton cycle because the working spaces do not operate isothermally and the heats of compression and expansion are transferred in separate heat exchangers. Since the expression "Stirling refrigerator" is in common use it will be used in this report, but the above remarks should be noted. In the Stirling refrigerator compression and expansion is effected mechanically by movement of a single piston. The Vuilleumier refrigerator which also uses regenerative heat exchangers is essentially a heat powered version of the Stirling refrigerator in which the compression and expansion of the working fluid is effected thermally by movement of a part of the fluid between hot and ambient spaces.

Regenerative exchangers are also used by another subgroup of refrigerators known variously under such names as modified Solvay, modified Taconis, or Gifford-McMahan refrigerators. These refrigerators are essentially Stirling refrigerators in which compression and expansion of the working fluid is affected by successively operating inlet and exhaust valves to admit and release high pressure gas. The presence of valves permits the use of a separate compressor, which could be of any configuration.

Systems using regenerative exchangers cannot be analyzed assuming steady state conditions since the pressure, fluid content and temperature of all components varies cyclically with time as well as position, and complex numerical methods are needed. The relationship among the cycles which will be described in this section is shown in Fig. 5-1.

In the following subsections single stage compression, expansion and heat exchange are assumed in all cases for the sake of clarity. It is noted, however, that the work of compression can always be reduced by multistaging in those systems which use separate compressors. The efficiency of expansion can similarly be improved by multistage expansion. The efficiency of heat exchange may be improved by what might be referred to as multistage heat exchange by splitting the exchanger into sections to reduce the temperature range. Between heat exchange stages the temperatures of the two streams are brought together by supplying refrigeration at this point by means of an intermediate expansion engine.

5.1 The Stirling Cycle Refrigerator

It was noted earlier that these refrigerators which are commonly called Stirling refrigerators, do not in practice operate on the ideal Stirling cycle. Due to the speed of operation heat cannot be transferred to and from the working spaces fast enough to permit isothermal compression and expansion. As a result, these processes are carried out under conditions closer to adiabatic and the necessary heat transfer is effected in separate heat exchangers. This operation is more characteristic of the Brayton cycle. The truly characteristic feature of the practical so-called Stirling refrigerator and its derivatives is the use of regenerative heat exchangers.

In the refrigeration application heat exchangers are used to exchange heat between high and low pressure gas streams meaning that the single flow passage in the regenerator must be alternately pressurized and depressurized. This could be achieved by using a continuously operating compressor and expander, ballast tanks or dual regenerators, and reversing valves between compressor and regenerator and expander and compressor. Such a system would incur substantial losses due to irreversible sudden compression and expansion when the valves were switched and due to the pressure drop through the valves. The practical Stirling refrigerator avoids these losses because the regenerator is in communication with the expander and compressor at all times, resulting in smooth and therefore less irreversible pressure cycling in the regenerator and elimination of flow losses through the valves.

5.1.1 Operation

The operation of a Stirling refrigerator is shown in Figure 5-2.

In position 1 the working fluid occupies the ambient space, after-cooler and regenerator. From 1 to 2 the fluid is compressed by inward motion of the

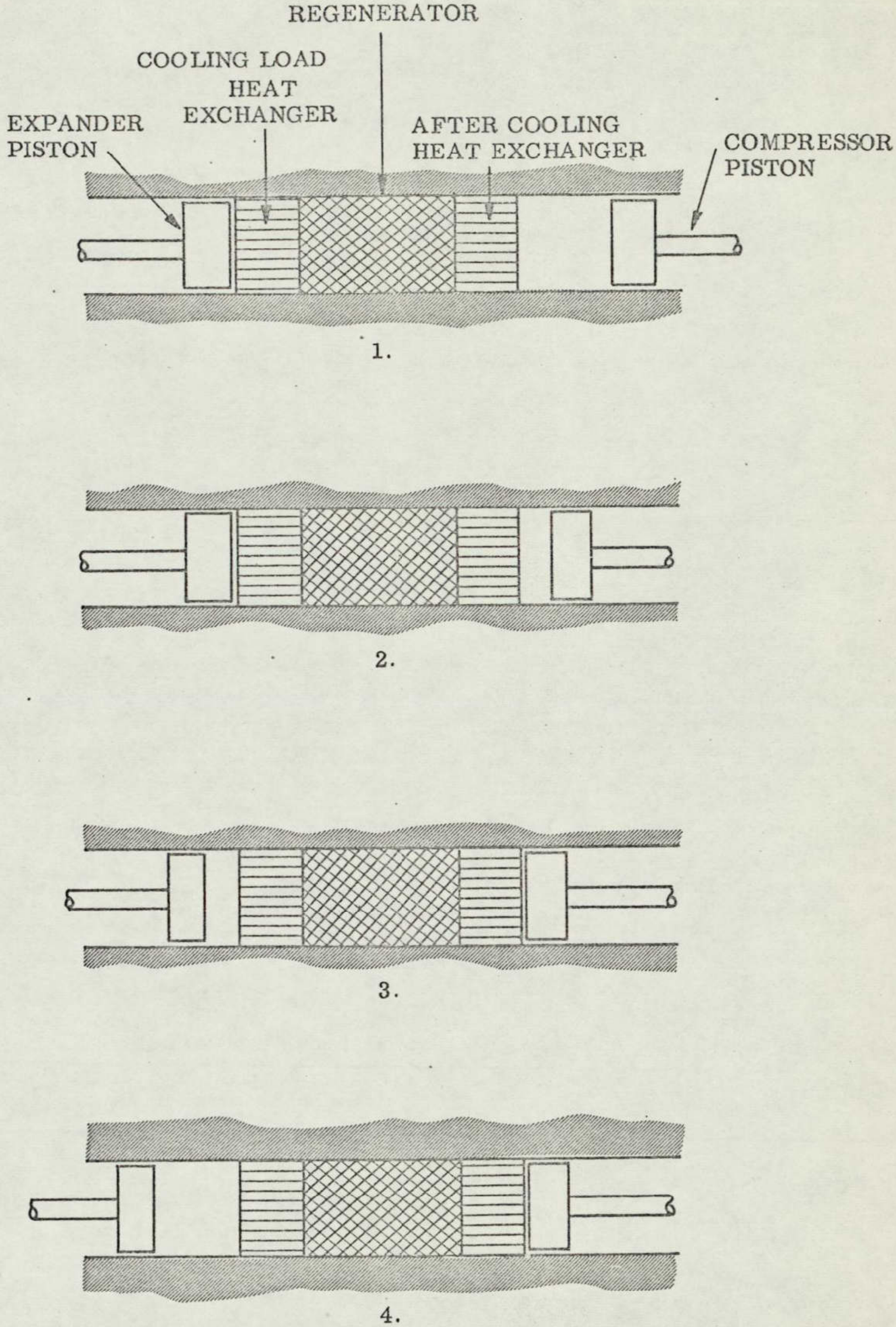


Fig. 5-2 The Practical Stirling Refrigerator

compression piston. From 2 to 3 the compressed fluid is transferred from the ambient end to the cold end at constant overall volume by equal increment of both pistons. During this transfer heat of compression is rejected to the after cooler and the temperature is reduced to the cold end temperature in the regenerator. With the fluid now occupying the cold space, load heat exchangers, and regenerator the fluid is expanded by outward movement of the expander piston, 3 to 4 . The fluid is returned from the cold end to the ambient end at constant volume by equal increment of both pistons. During this transfer the lost energy of expansion is replaced in the load exchanger and the temperature is raised to the ambient temperature in the regenerator.

This cycle can equally well be executed using just one piston to perform both expansion and compression processes, and using a passive displacer to move the fluid from one space to another. This configuration of refrigerator is shown in Figure 5-3.

In practice, it is not practical to move either the two pistons or the piston and displacer in the intermittent manner shown. It is customery to drive both components from the same crank shaft for practical convenience. Both components are thus continually in motion but the cycle can be satisfactorily executed by phasing the piston or displacer motions such that compression occurs with most of the fluid in the warm space and expansion occurs with most of the fluid in the cold space.

Because of the cyclic operation of the practical Stirling refrigerator and the fact that working fluid will be distributed through several temperature regimes during compression and expansion, it is impossible to show the steady state cycle processes on a temperature entropy diagram in the conventional way. It is consequently very difficult to perform a reliable thermal analyses of this type of system without resort to quite complex digital and/or analog computational techniques. For approximate engineering analysis purposes simplified representation of the processes can be made which provide a more accessible, if less reliable, method.

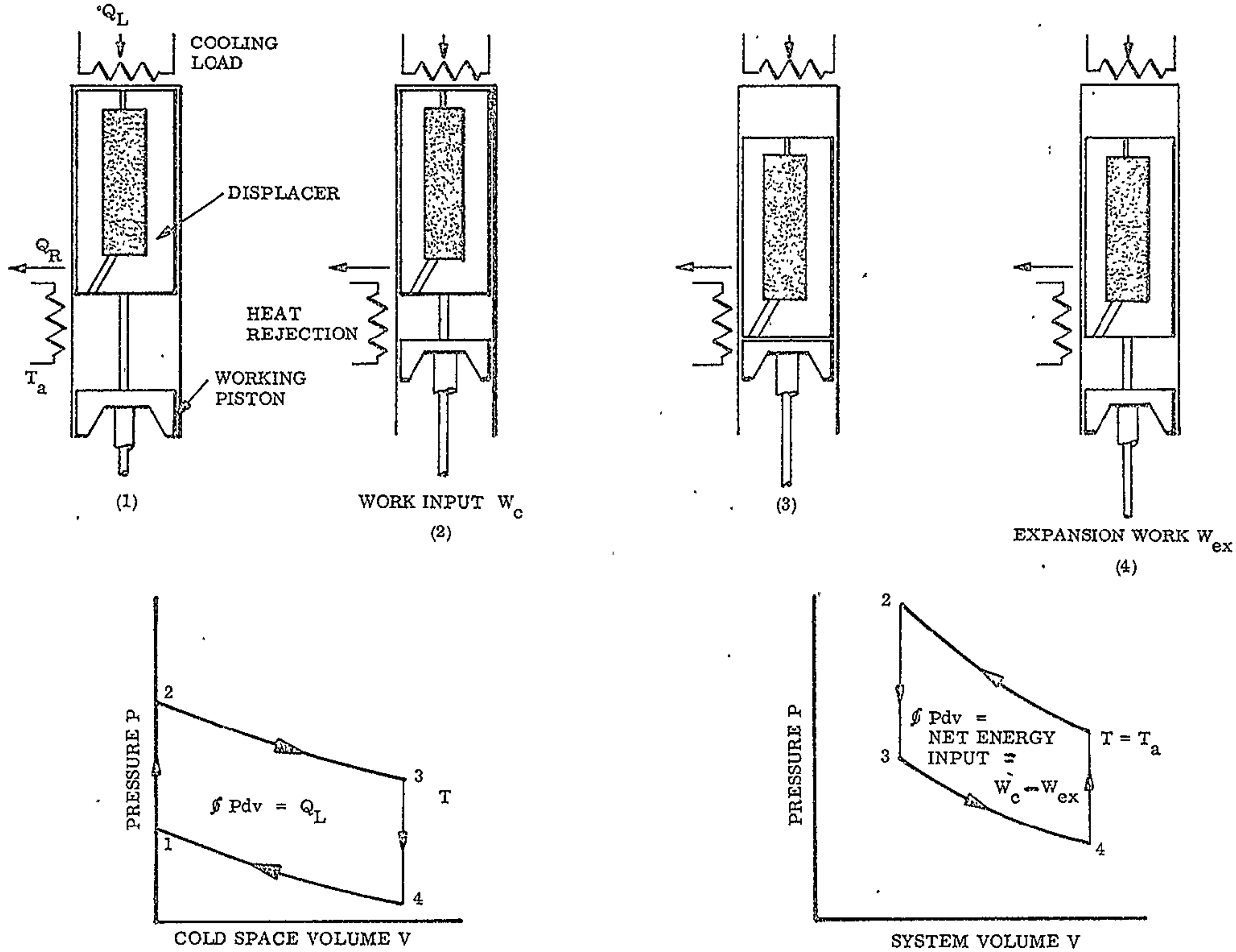


Fig. 5-3 Stirling Cycle

The detailed analysis of a practical Stirling refrigerator requires that the processes occurring in the various locations of the machine be characterized by a set of partial differential equations. There are five major locations. The compression space, after-cooler, regenerator, load exchanger, and expansion space. Within these regions there will be property variations which can be allowed for by averaging or by subdivision of the region. Equations are written to characterize the mass and heat flow rates into these regions and subdivisions in terms of pressure and temperature differentials, fluid properties and system geometry. The equations are then solved for the boundary conditions of total mass conservation, given load and after-cooler temperature and given compressor and expander displacement, speed and phase relationship. Such a system of equations can only be solved by numerical techniques on a digital computer, or by analog methods. It has been found that solution by these methods is quite difficult because of the length of time required both to write a program and then to solve the problem on the computer. As time progresses it is likely that improved numerical techniques will be found to reduce the amount of machine time, but currently this exact analysis approach is often rejected in favor of the more convenient approximate approach.

In the approximate analysis the performance of the refrigerator is assessed by writing the cooling capacity of the machine, q_c, as

$$q_c = \oint_c Pdv - \Sigma \text{ losses}$$

Here $\oint_c Pdv$ is the gross work performed by the cold end of the refrigerator, equal to the gross cooling capacity. The losses due to the various undersirable mechanisms which introduce heat from the environment to the cold end and reduce the effective cooling capacity are considered to be analytically separable effects. The work input to the ambient end, W, is

$$W = \oint_a Pdv$$

The terms $\oint_c Pdv$ and $\oint_a Pdv$ can be evaluated by writing mass and energy conservation equations for the working spaces. In the simplest case these equations can be solved by assuming isothermal conditions in the spaces and modifying the answer by means of an empirical constant to allow for non-isothermal behavior. This so-called Schmidt analysis permits analytic expressions to be written for the integrals. The integrals can also be evaluated for adiabatic conditions using a relatively simple numerical solution as shown by Rios[14]. Rios' method can be modified quite simply to include the effect of heat transfer with the walls of the working spaces to obtain a quite accurate evaluation of $\oint_c$ Pdv and $\oint_a$ Pdv. The separable loss mechanisms are generally accepted to be as follows:

a) Heat conduction from ambient to cold end by conduction through the structural members.

b) Heat transfer by radiation and convection from the ambient environment.

c) Heat transfer from the ambient end to the cold end due to the relative motion of the displacer or expander and cylinder walls.

d) Heat flux into the cold space due to regenerator inefficiency.

e) Heat loss due to cyclical pressurization and depressurization of the clearance spaces.

f) Non-ideal heat transfer in the load and after-cooler exchangers.

These losses can be assessed by performing simple individual engineering analyses of the mechanisms, assuming that they are decoupled from each other. Generalized expression for these loss mechanisms cannot be written because the most appropriate analytical model may be different from one case to another.

The analysis based upon decoupled loss mechanisms is a great deal easier to use than the complex analysis but it cannot allow for the strong interaction that may occur between the loss mechanisms, particularly a), c), e) and heat transfer within the working spaces. This method will continue to be used until the complex analysis can be made more available and less costly in computer time.

5.1.2 Companies Engaged in Production and Development of Stirling Refrigerators

The following companies are presently engaged in production and development of Stirling cycle refrigeration systems:

U.S. Philips Corporation
Malaker Corporation
Hughes Aircraft Corporation

U. S. Phillips: Phillips was the pioneer in development of the Stirling refrigerator, having built their first machine in 1954 for the purpose of air liquification. Initial research on the Stirling cycle as a refrigeration device was initiated in 1945 by Phillips, and in 1950 the first drops of liquid air were obtained from a Stirling-cycle refrigerator. Additional history on the development of the Stirling refrigerator at Phillips is given in Refs. 9 to 12.

Presently, Phillips produces a variety of Stirling cycle machines for laboratory and industrial use as well as miniature units for aircraft use. The miniature flight units designated "Cryogem" include models 42100 and 42151 which are two stage units which provide cooling in the range of 1 watt at $25^{o}K$ and 2 watts at $30^{o}K$. These units are intended primarily for aircraft usage in cooling infra-red detectors and as such provide lower refrigeration capability than required for this study.

Of particular interest for long-term cryogenic storage requirements is Phillips Model A-20 "cryogenerator". This unit appears to be the only one which provides $20^{o}K$ refrigeration at levels near 100 watts, other available production units generally providing 1-10 watts of refrigeration at that temperature, followed by very large, heavy industrial units. The unit is a two-stage machine based on the Phillips-Stirling cycle and provides refrigeration at two temperature levels, one over the $60\text{-}90^{o}K$ range and the second stage between 15 and $30^{o}K$. This unit, then has the potential of cooling two different cryogens at the two temperature levels. Conceivably all two-stage Stirling cycles have this capability, however, development along these lines has not been pursued in many units. Various arrangements of the A-20 cryogenerator have been selected by Phillips for various usages including gas liquification and cooling of experiments.

Malaker: Malaker laboratories has been engaged in the development of Stirling cycle refrigeration since the 1950's. The majority of their research and development has gone into the Stirling cycle and has been concentrated on small units. Malaker has produced units with very high thermal efficiencies, and is actively engaged in additional development and modification of their units. Some of their earlier work under contract to WPAFB is reported in Ref. 17. Recent work of interest here has been devoted to the modification of one of their units to make it adaptable to space operation. (Designated Model SS-I). The primary modification consists of providing an all welded case around the units to allow larger temperature excursion during operation without freezing up the existing O-ring seals and leaking the working gas. A Malaker unit has been successfully tested in vacuum in the laboratory. Vacuum operation would not appear to present a problem for the Stirling units since they are hermetically sealed as are the Phillips machines. Malaker has concentrated on miniature units for laboratory use, for aircraft support and various field uses. They have not built large industrial units such as Phillips. Units are available for cooling down to approximately 15°K in two stages and to near 60°K in a single stage. Production units are available and fall within the requirement of this study; 10-100 watts at 20 to 110°K.

Hughes Aircraft Corporation: Hughes Aircraft Corporation produces Stirling cycle machines for various uses. The majority of their units manufactured to date are for IR cooling on aircraft. Refrigeration units are not commercially available from HAC but essentially provide a support function for in-house activities.

One of the prototype models[13] provides 15W @80°K and therefore falls within the study range, while the other units provide only a few watts. HAC is extensively engaged in research and development on the Vuilleumier cycle unit and these activities are discussed in another section (5.2.2).

5.1.3 Analysis of Stirling Cycle Data

Data has been assembled on the characteristics of the following Stirling refrigerators:

Malaker Corporation
Mark VII-C
XIV-A
VII-R
XX

U. S. Phillips Corporation
A-20

N. V. Phillips Gloeilampenfabrieken
X-20 (prototype)

Hughes Aircraft Company
Hughes prototype

These four companies make many additional units as well; however, they fall outside the range of the cooling requirements for this study, and were not included in the data correlations, since sufficient data was available on units of the required capacity (5-100 watts).

The parameters of the various units are tabulated in Table 5-1 for the seven units. The tabulation also includes the calculations which were made to provide the basis of the curves which are plotted.

Figures 5-4 through 5-8 present the cooling capacity as a function of the temperature for the individual units. Also included is the power input vs. temperature where available. The Phillips A-20 unit provides cooling at two temperatures corresponding to the first and second stage of the machine, and the performance of both stages is included. This feature is an attractive consideration for application to cooling both the fuel and oxidizer of a vehicle system using cryogenic propellants. The other Stirling cycle units are not arranged so that net cooling is available at the 1st stage, although this might be accomplished with a substantial redesign of the systems. Other cycles presented in this study also provide cooling at two stages.

Table 5-1

EXISTING STIRLING CYCLE
COOLING AT 20 TO 110°K AT

Manufacturer		Malaker Corp.	Malaker Corp.	Malaker Corp.
Trade Name		Cryomite	Cryomite	Cryomite
Model		Mark VII-C	Mark ViV-A	Mark VII-R
I.D. Number		1	2	3
Refrigeration Range		17.5 - 80°K	44 - 100°K	40 - 125°K
Cycle		Stirling	Stirling	Stirling
Working Fluid		Helium	Helium	Helium
High Pressure		NI	NI	NI
Low Pressure		17 atm. Fill	NI	NI
Minimum Temp.		17.5°K	44°K	40°K
Cool-Down Time		8 min.	7 min.	3.8 min.
Expander RPM		NI	NI	NI
Volts - Phase - Frequency		208 - 3/4 - 400/60	208 - 3 - 400	208 - 3 - 400
Cooling Means		Air or Water	Air	Air
Ambient Temp. Reqmts.				
Required Attitude		Any	Any	Any
Cryostat Dimensions		4.8" D x 11.5" L	2.9" D x 13" L	6 1/2" D x 23 1/2" L
Compressor Dimensions				
System Volume		209 in.^{3}	86 in.^{3}	781 in.^{3}
System Weight		15.5 lb	5.5 lb	40 lb
MTBF		40,000 hr	40,000 hr	40,000 hr
Maintenance Interval		1,000 hr	1,000 hr	1,000 hr
System Cost		$5,195	$9,000	$17,500
20°K	Refrigeration	1 watt	0	0
	Power Input	700 w		
	COP	.00143		
	% Carnot Eff.	2.0%		
	lb/watt	15.5		
	in.^{3}/watt	209		
77°K	Refrigeration	15 w	2.8 w	60 w
	Power Input	640 w	108 w	1,220 w
	COP	.0234	.0259	.0492
	% Carnot	6.8%	7.5%	14.3%
	lb/watt	1.03	1.97	0.667
	in.^{3}/watt	13.9	30.8	13.0
110°K	Refrigeration	23 w	5 w	90 w
	Power Input	560	96 w	1,220 w
	COP	.0411	.0522	.0738
	% Carnot	6.5%	8.25%	11.7%
	lb/watt	0.674	1.10	0.445
	in.^{3}/watt	9.1	17.2	8.7

FOLDOUT FRAME 1

5-1

CLE REFRIGERATORS
K AT 5 TO 100 WATTS

	Malker Corp	Phillips Laboratory	Phillips Laboratory	Hughes Aircraft		
	Cryomite	None	Prototype	Prototype		
	Mark XX	A-20	X-20			
	4	5	6	7	8	9
	40 - 120°K	12 - 300°K	12 - 300°K	45°K up		
	Stirling	Stirling	Stirling	Stirling		
	Helium	Helium	Helium	Helium		
	NI	427 psia	NI	NI		
	NI		NI	NI		
	40°K	12°K	12°K	45°K		
	.4 min.	40 min.	15 min.	3 min.		
	NI	1450 - 1750	1750	NI		
	208 - 3 - 400	400 - 3 - 50/60	2000 VA - 3 - 50/60	115 - 3 - 400		
	Air or Liquid	198 gal/hr. H_2O	Air or Liquid	Air or Liquid		
				-55 +071°C		
	Any		Any (g = 0)	Any		
2" L	19"x 18" x 15 1/2"	43.5"x37.4"x19.7"	4" D x 7.5" L	8" x 6" x 6"		
			18.5" x 13.8" x 13"			
	1500 in.^{3}					
	65 lb	660 lb	112 lb	10 lb		
	40,000 hr		NI	NI		
	1,000 hr	500 hr	4,000 hr	500 hr		
	$24,000		NI	NI		
	0	100 w	10 w			
		8,300 w	1,750 w			
		.0121	.00572			
		17%	8%			
		6.60	11.2			
	110 w		36 w	14 w		
	1,990		1,750 w	500 w		
	.0553		.0206	.0280		
	.6%		6%	8.1%		
	0.591		3.12	0.715		
	13.6					
	164 w		NI			
	1,860 w					
	.0883					
	14%					
	0.396					
	9.15					

FOLDOUT FRAME 2

(2)

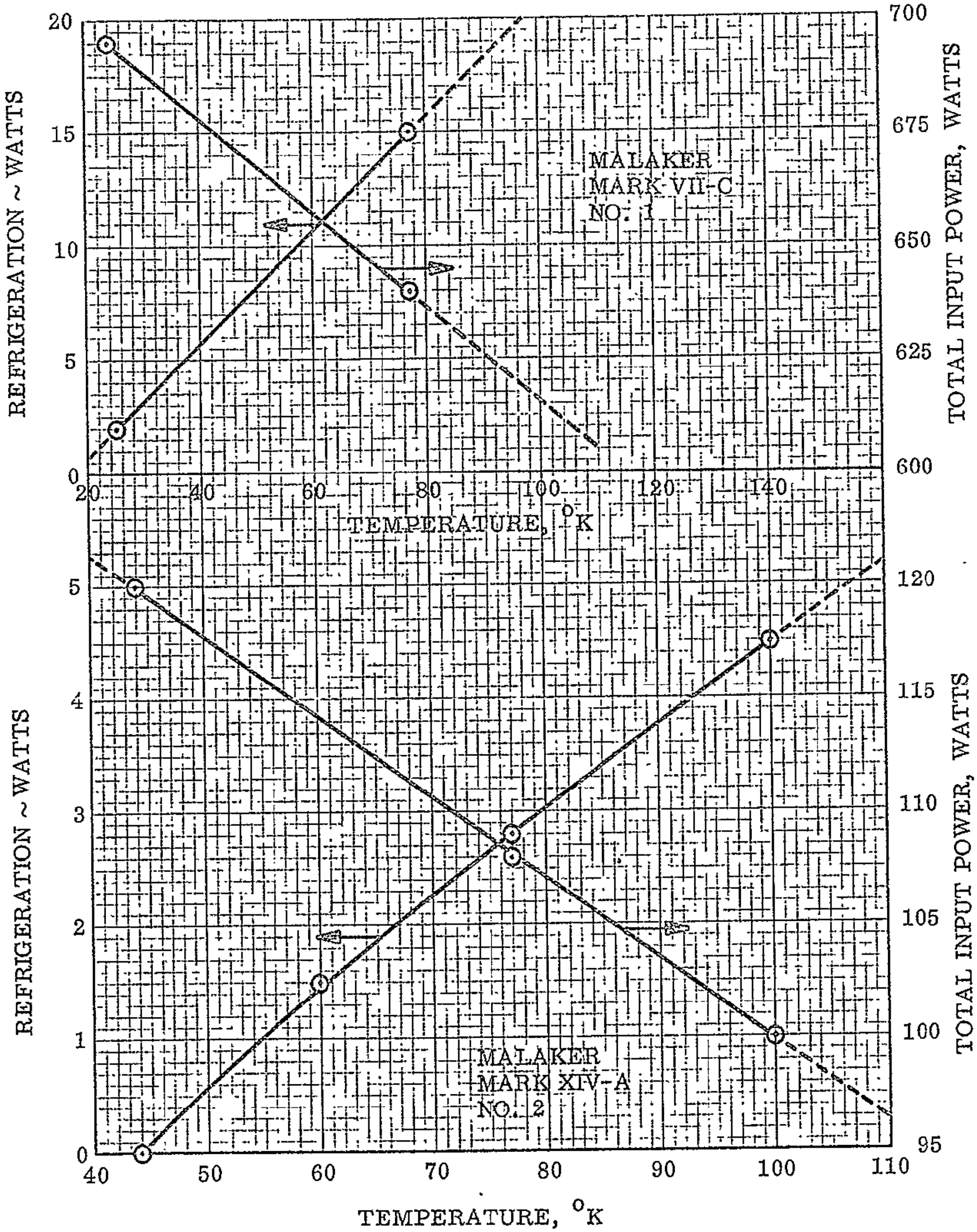


Fig. 5-4 Refrigeration Versus Temperature (Stirling Cycle)

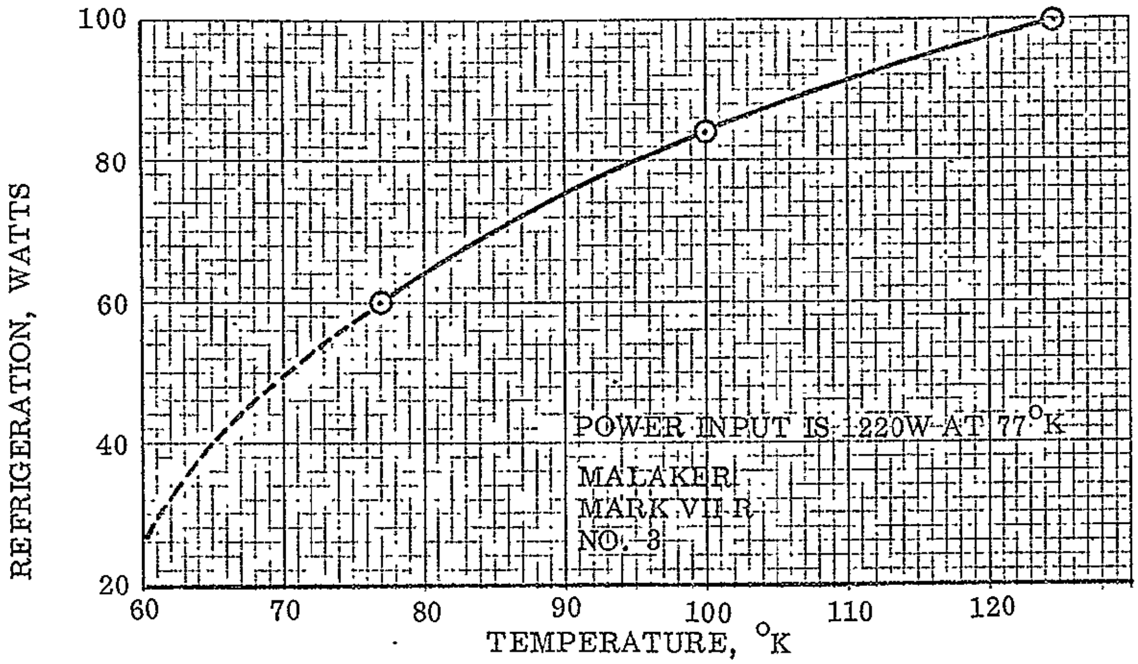


Fig. 5-5 Refrigeration Versus Temperature (Stirling Cycle)

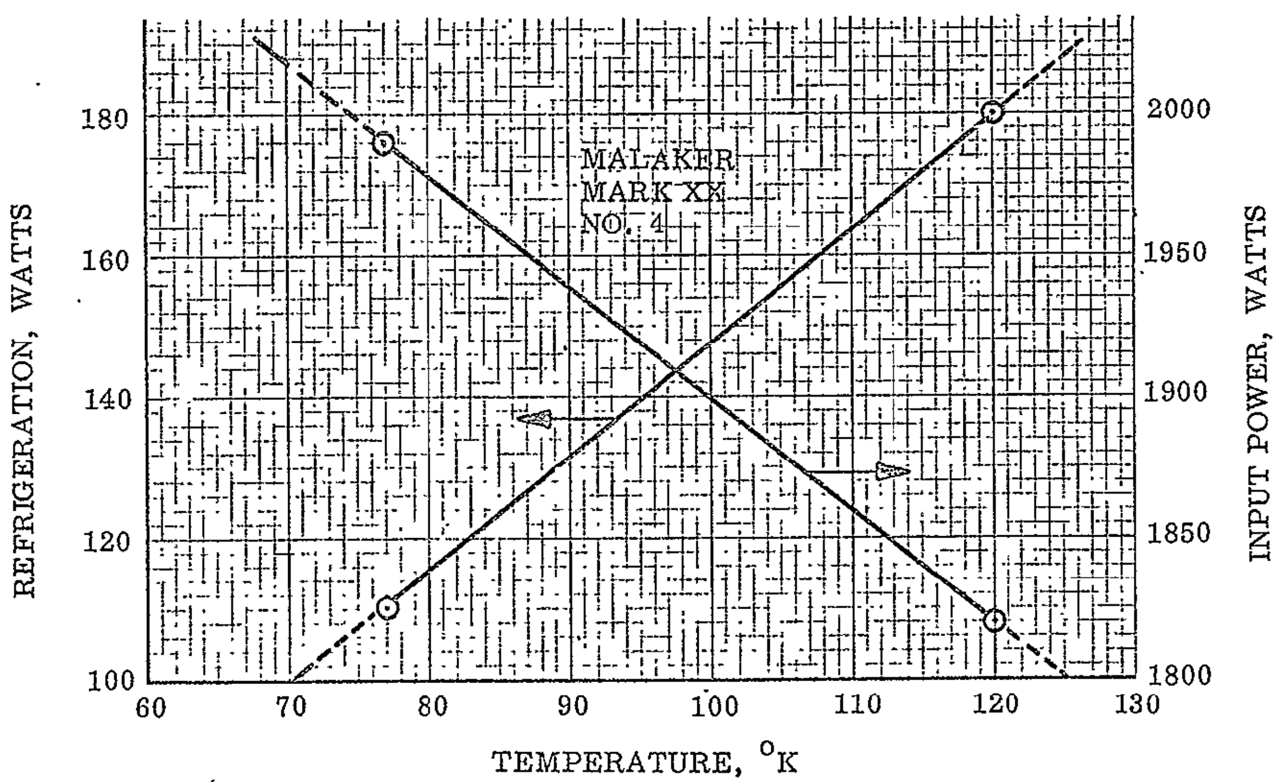


Fig. 5-6 Refrigeration Versus Temperature (Stirling Cycle)

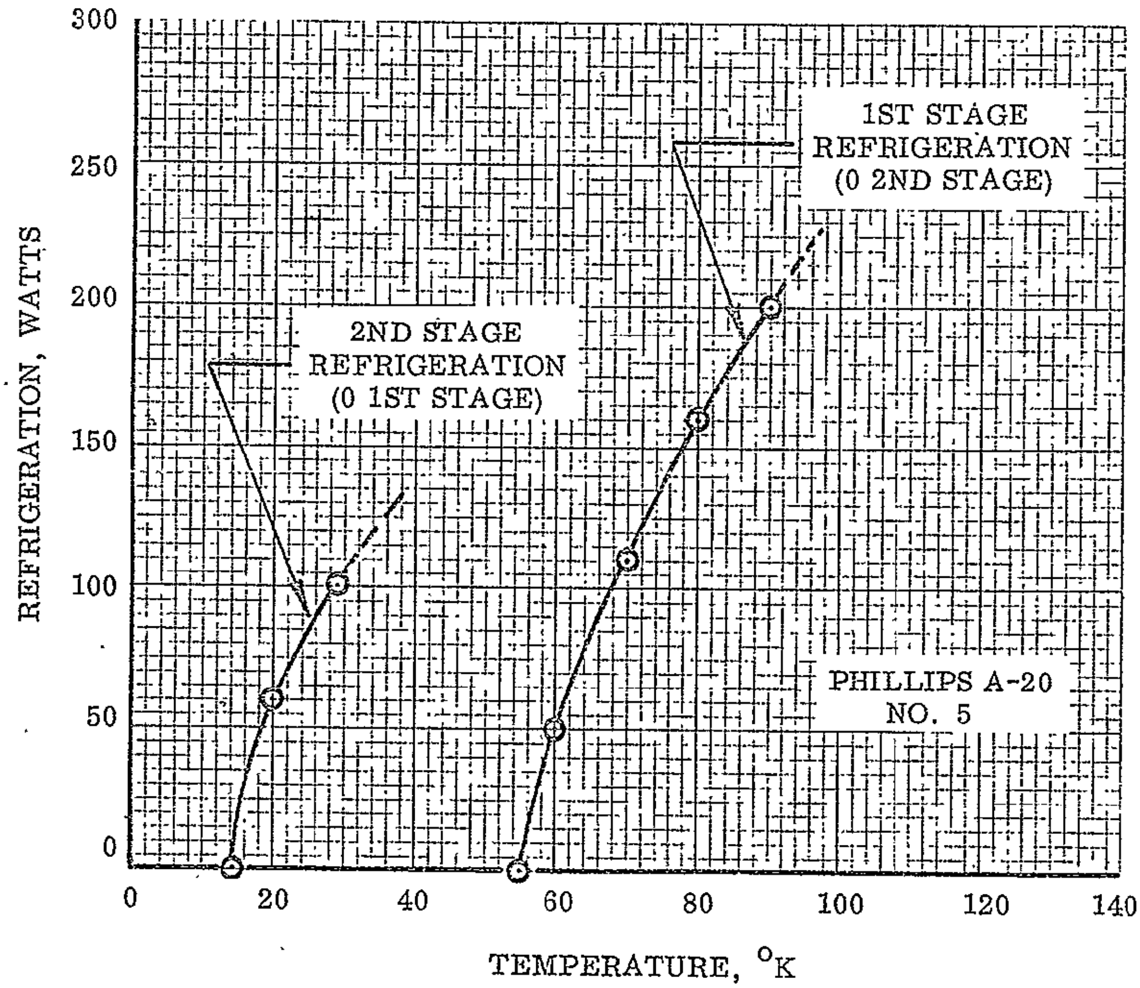


Fig. 5-7 Refrigeration vs. Temperature (Stirling Cycle)

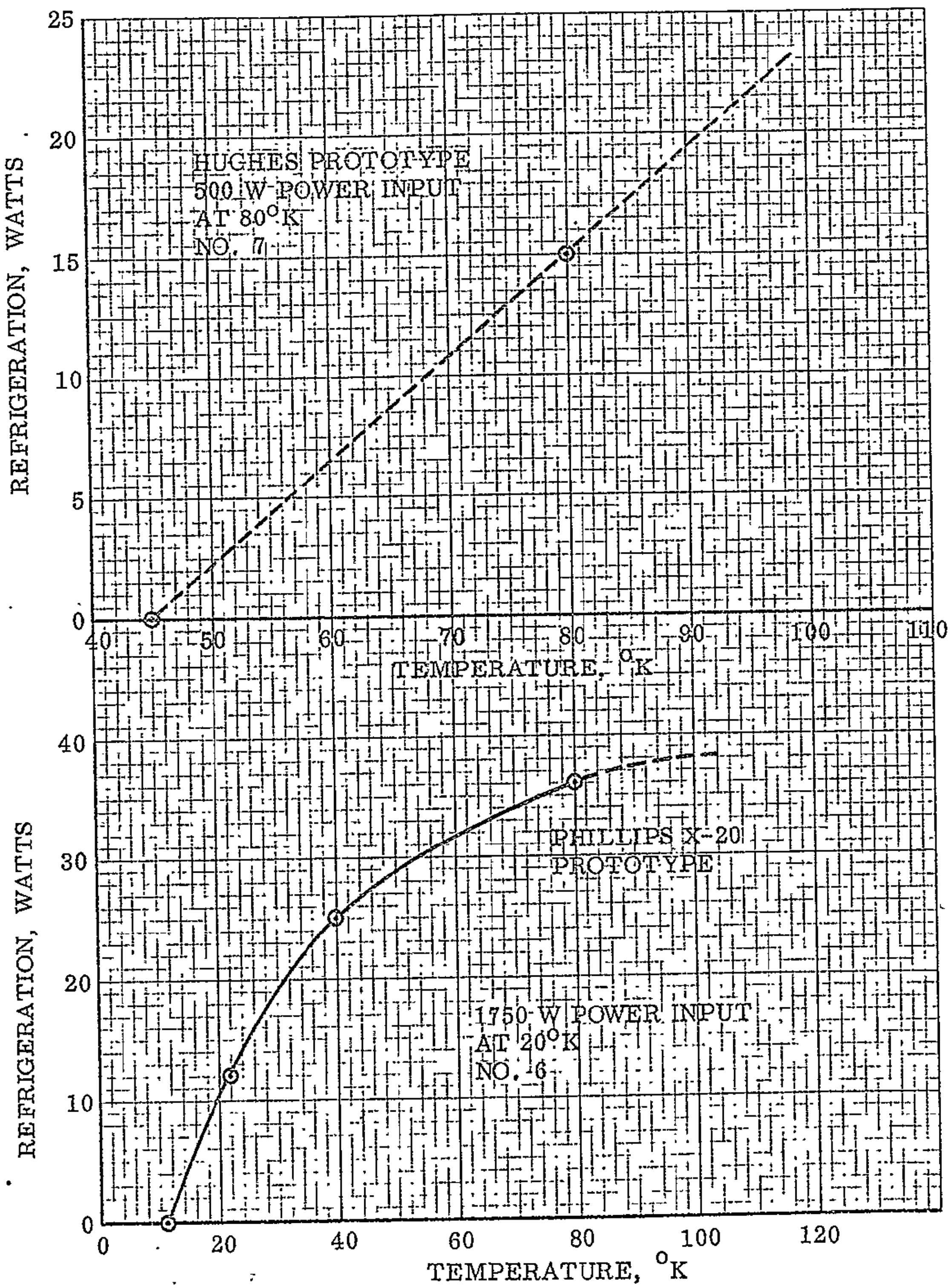


Fig. 5-8 Refrigeration Versus Temperature (Stirling Cycle)

Figure 5-9 presents the coefficient of performance (C.O.P.) of the units as a function of net refrigeration provided. The data is presented at three temperatures of interest: 20°K (LH_2), 77°K (LN_2) and 110°K. The 20°K and 110°K temperatures represent the lower and upper limits of the study and the 77°K point is a convenient intermediate temperature at which operating data is commonly reported. The data at the three temperatures was curve fit as shown on Figure 5-9. In general, the curve fitting was done so that the curve represented maximum performance (highest value of C.O.P.). A notable exception is the Malaker Mark XIV - (I.D. No. 2) which falls considerably above the curve fit at 110°K. A curve fit through this point would give twice the C.O.P. of the curve fit.

Figure 5-10 presents the C.O.P. as a function of temperature. The Carnot efficiency is also shown for comparison, and represents the theoretical performance of the Stirling cycle. Curve fits are also shown for 100 watt and 5 watt capacity, and were obtained from the previous figure. The figures, as expected, show a pronounced effect of both temperature and capacity. It is interesting to note that the slope of the 5 and 100 watt curves are parallel to the slope of the Carnot C.O.P. vs. temperature, although they were curve fit independently of the Carnot curve.

Figure 5-11 shows the percent Carnot efficiency which the units have achieved as a function of refrigeration capacity. The units provided from 6 to a maximum of 20% of the theoretical efficiency, and this is the best performance of any of the cycles as will be seen in later sections.

Figure 5-12 presents the specific weight of the systems as a function of cooling capacity. The data are fitted with lines at 20°K, 77°K and 110°K as for the C.O.P. data.

Figure 5-13 shows the data points plotted as specific weight vs. temperature, with line at 5 watt and 100 watt capacity. The capacity lines were obtained by cross-plotting the result of the previous plot. The results show the expected strong dependence on both capacity and temperature.

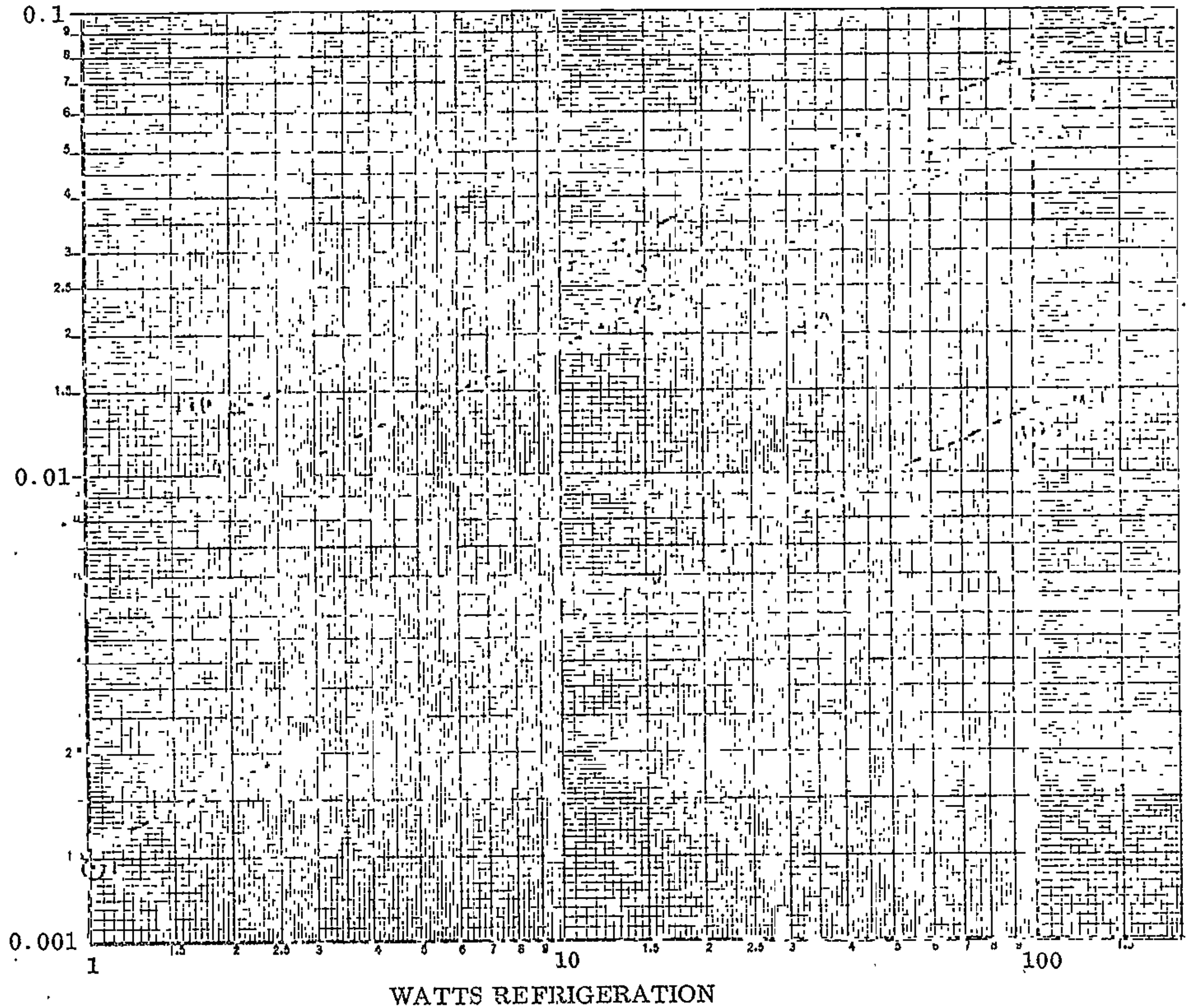


Fig. 5-9 Coefficient of Performance Versus Cooling Capacity (Stirling Cycle)

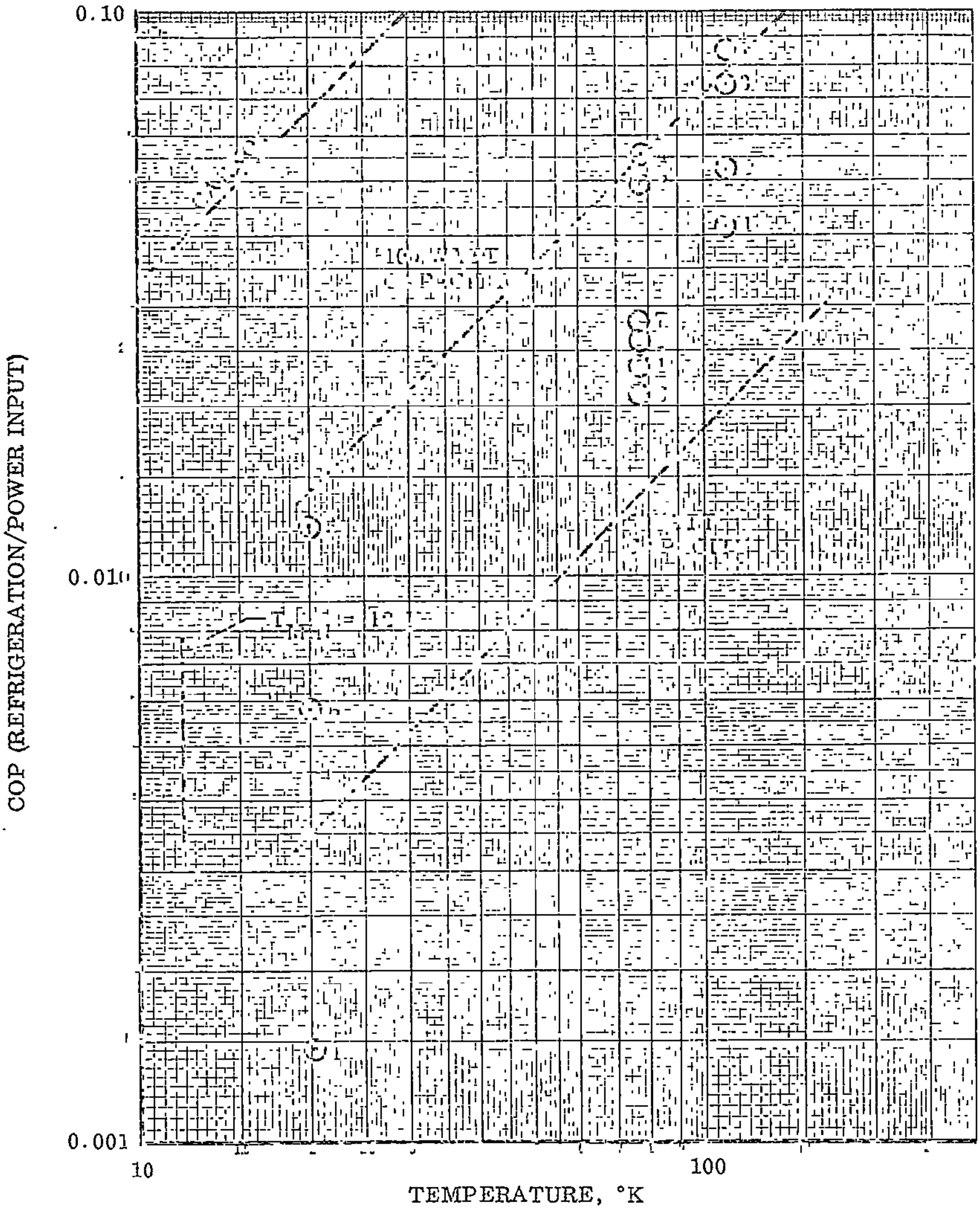


Fig. 5-10 Coefficient of Performance Versus Temperature (Stirling Cycle)

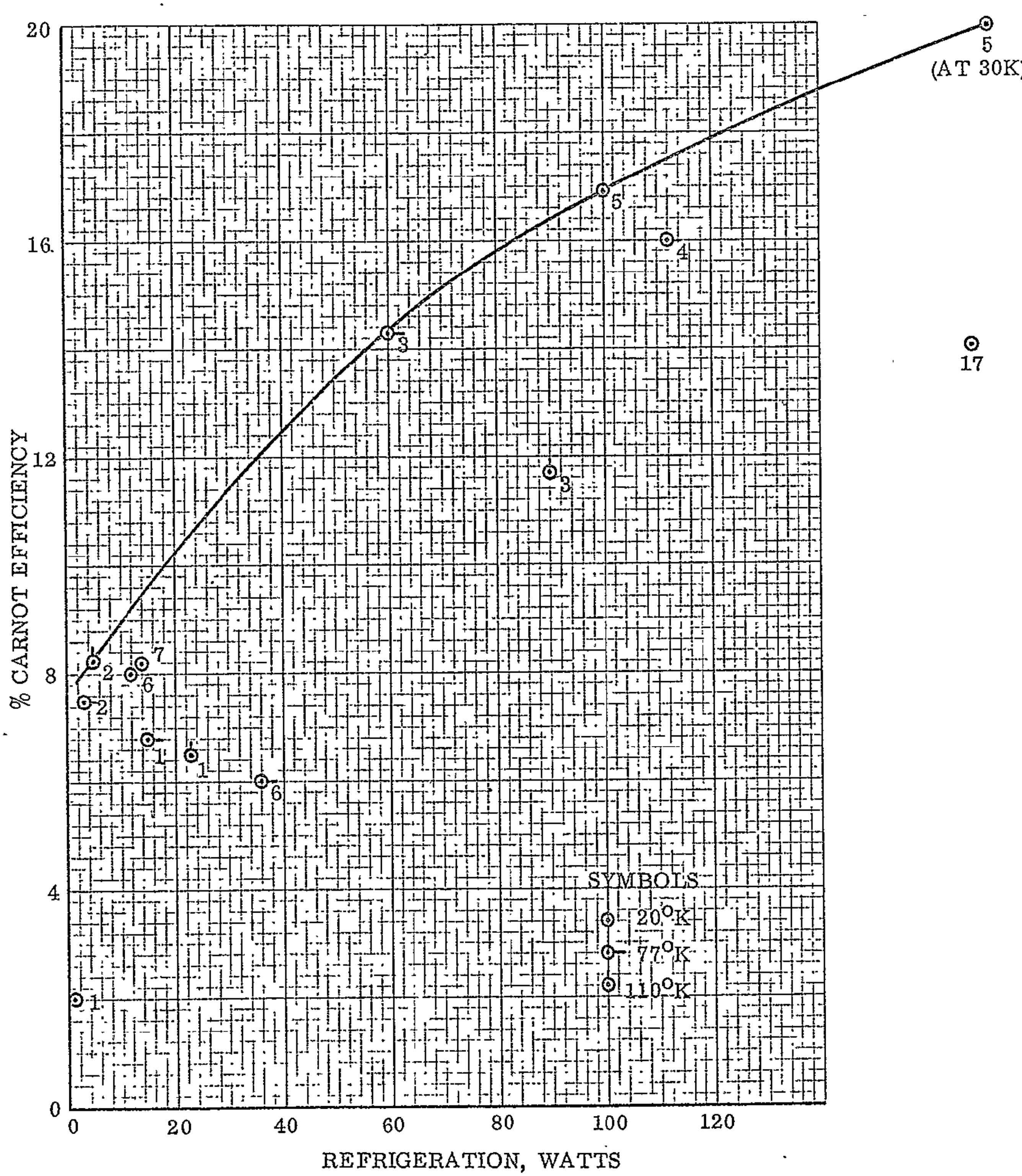


Fig. 5-11 Percent Carnot Efficiency of Stirling Cycle Refrigerators

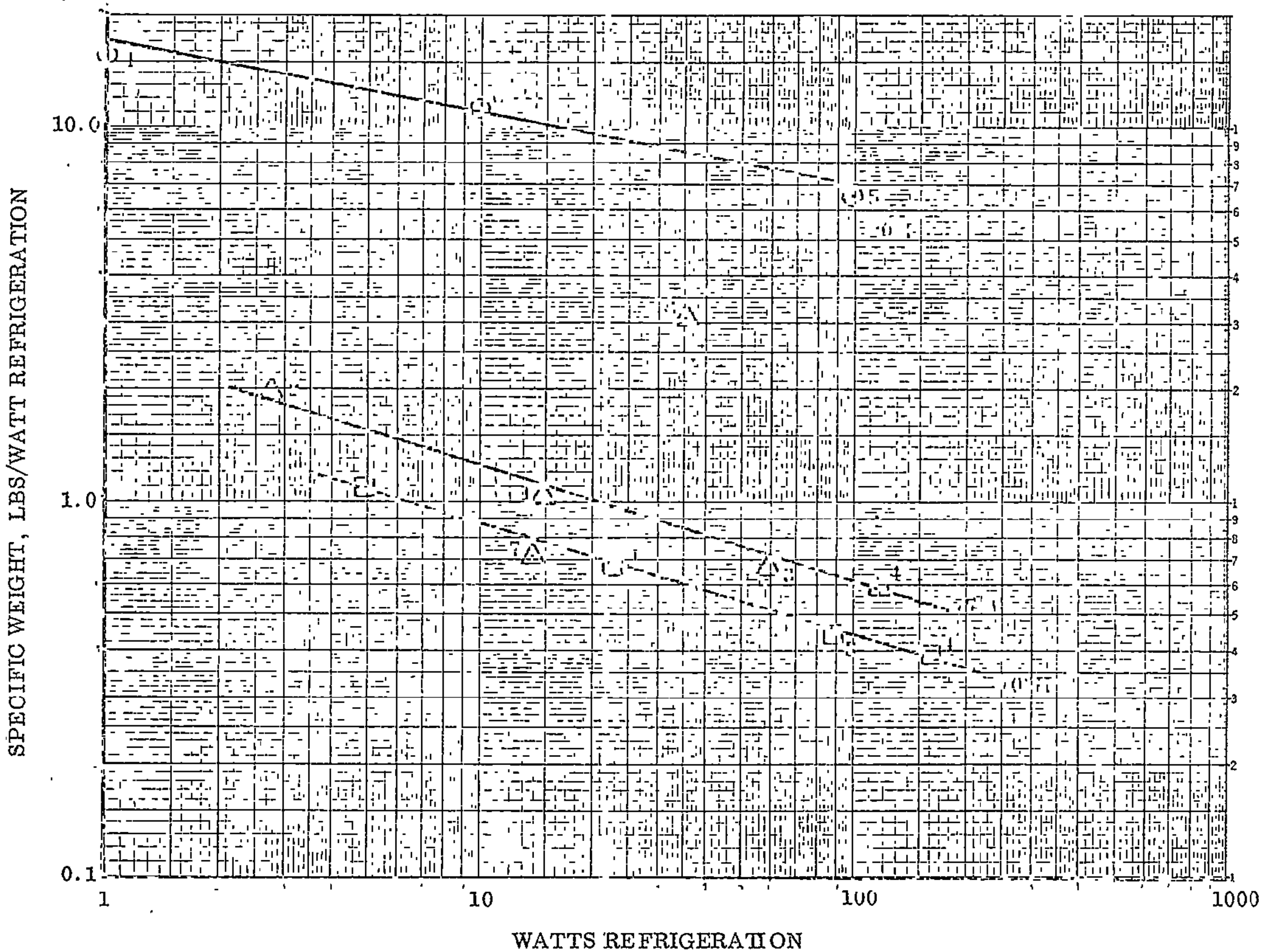


Fig. 5-12 Specific Weight Versus Cooling Capacity (Stirling Cycle)

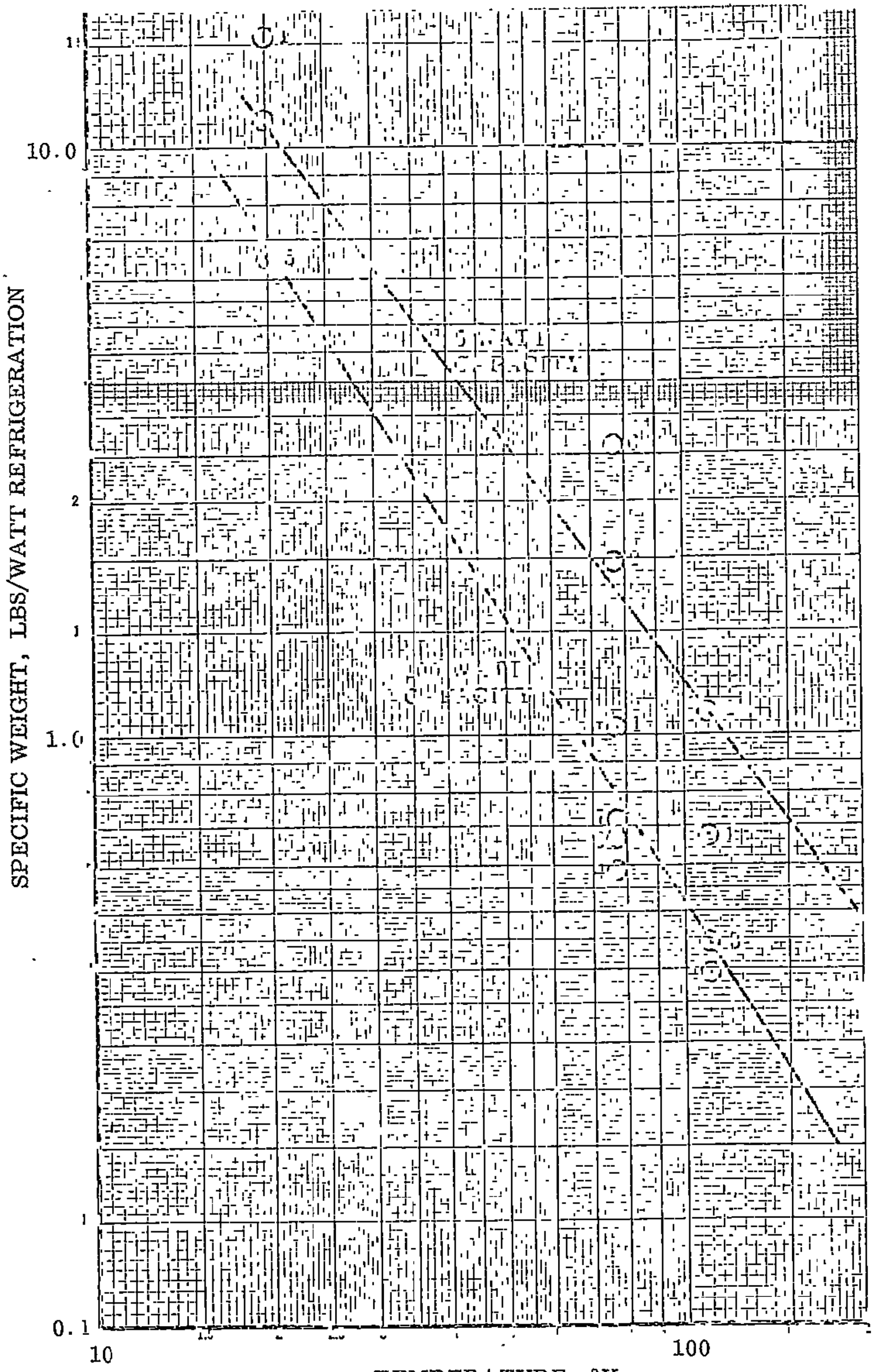


Fig. 5-13 Specific Weight Versus Temperature (Stirling Cycle)

Figure 5-14 shows the specific volume of the various units vs. cooling capacity at the three temperatures selected.

5.2 The Vuilleumier Refrigerator

5.2.1 Operation

The Vuilleumier refrigerator is in essence a practical Stirling refrigerator in which compresssion and expansion of the working fluid is effected thermally instead of mechanically. This modification is best illustrated in connection with the Stirling refrigerator configuration of Figure 5-3. The working piston is removed and is replaced by a thermal compressor/expander consisting of a hot space, ambient space regenerative heat exchanger and displacer. The cycle of operations, shown in Figure 5-15 closely parallels that of the Stirling refrigerator. In position 1, the fluid is all in the ambient space. From 1 to 2, the compressor displacer moves from hot to ambient end, causing fluid to move from ambient to hot spaces as constant volume, resulting in an increase in system pressure and hence compression of the fluid remaining in the ambient space. From 2 to 3 the expander displacer is moved to displace this remaining fluid to the cold end. From 3 to 4 the pressure is reduced by displacing fluid from the hot space back to the ambient space, thereby expanding the fluid in the cold space. From 1 to 2 the cold gas is returned to the ambient space by movement of the expander displacer. The heat interactions in the exchangers are similar to those in the Stirling refrigerator. In addition to the load and after-cooler heat exchangers, however, there is also a power heat exchanger required at the hot end through which the energy required to compress the fluid is supplied. As noted in Section 3, this energy will be higher than the actual work of compression since the device is in essence a combined engine and compressor, and thus the supplied energy must also include the necessary rejection heat besides the compressive work.

In practice the intermittent movement of the displacers is achieved by driving both of them from the same crankshaft but displaced in phase such that during compression most of the fluid is in the ambient space, and during expansion most of the fluid is in the cold space.

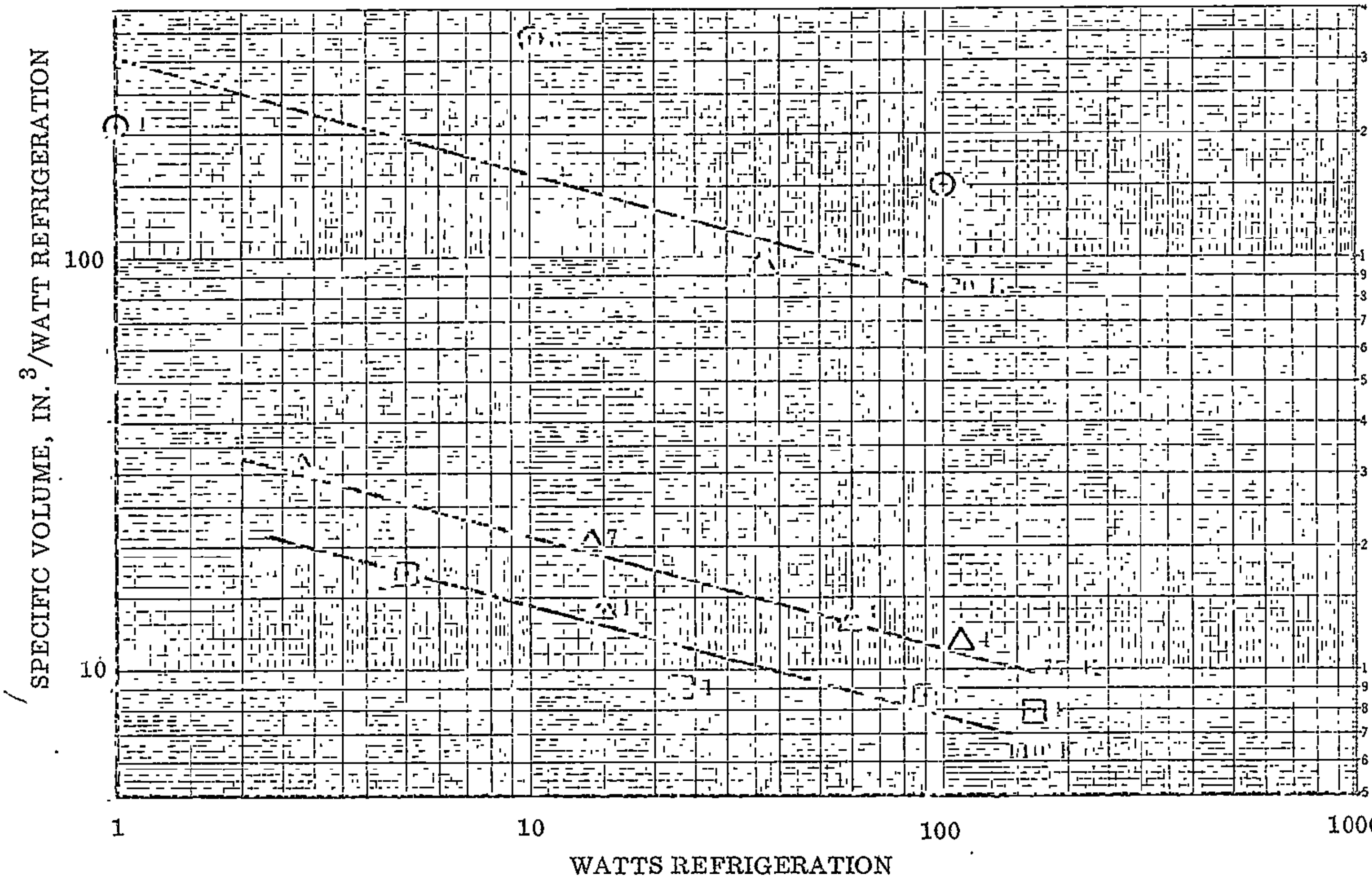


Fig. 5-14 Specific Volume Verses Refrigeration Load (Stirling Cycle)

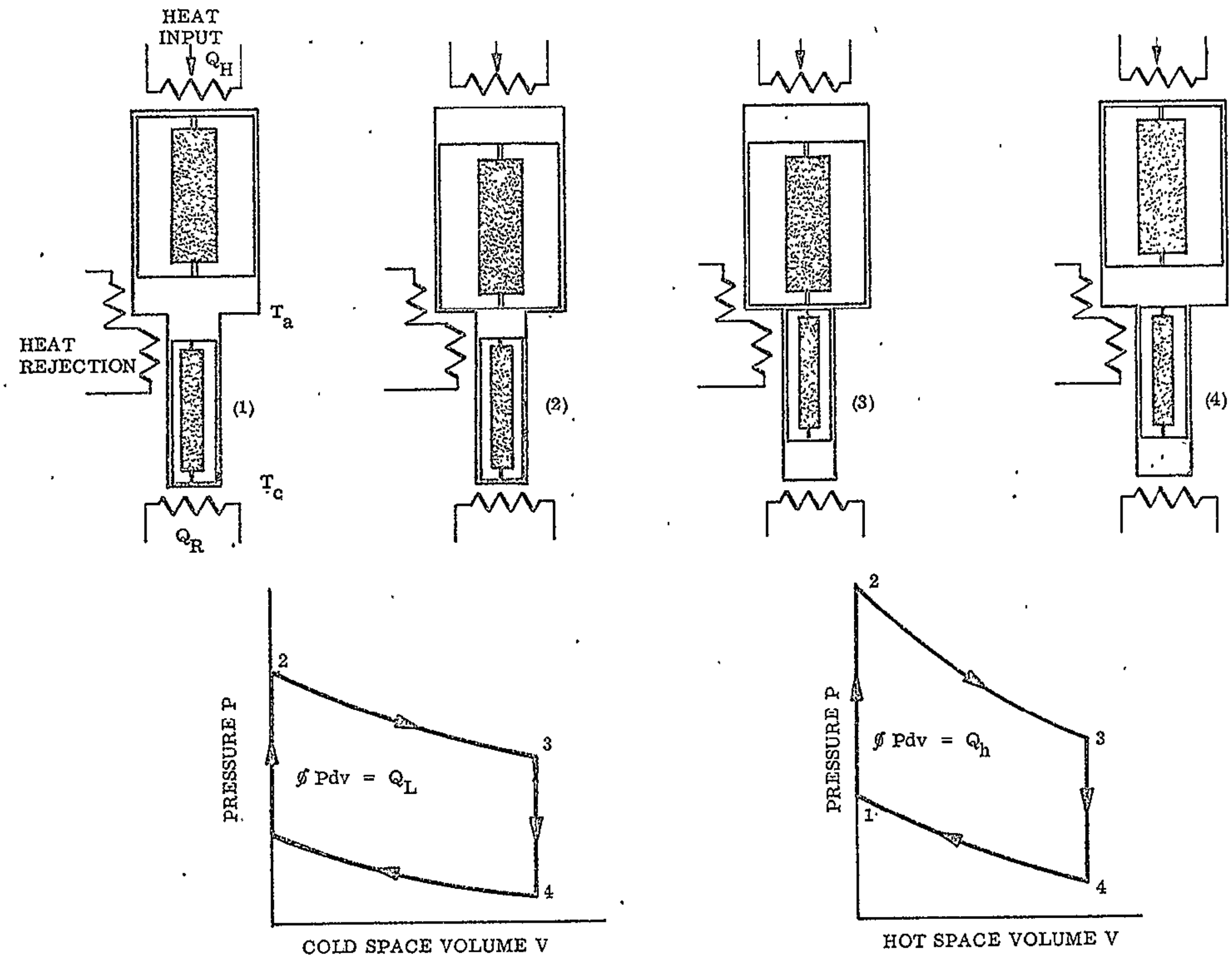


Fig. 5-15 Vuilleumier Cycle

Thermal analysis of the VM refrigerator is very similar to that of the Stirling refrigerator and all comments made in Section 5-1 on this subject apply. Two analytical approaches are possible as in the Stirling case. The VM refrigerator is a constant volume device whereas the Stirling refrigerator changes overall volume at a prescribed rate. Otherwise the same general approach is employed.

In the case of the approximate analysis exactly the same approach is followed in analyzing the cold end, in that the effective cooling capacity is equal to the gross work performed at the cold end less the heat leakage into the cold end. In the VM refrigerator power is supplied as thermal rather than mechanical energy. This required thermal energy, q_h, is given by

$$q_h = \oint_h Pdv + \Sigma\text{losses}$$

Here, $\oint_h Pdv$ is the gross work performed at the hot end and the losses represent the additional heat which has to be supplied to make good the heat lost by various mechanisms to the environment. The loss mechanisms are qualitatively identical to those occuring in the cold end and listed in Section 5-1.

Strictly speaking a practical Stirling refrigerator using separate compression, heat exchanger and expansion components could be successfully operated, but dual drive mechanisms would be needed, the movement of the compressor piston and displacer or expander piston would have to be accurately synchronized, and the separation of the components would introduce undesirable void volume. The requirement that the compressor and expander motors be synchronized in effect requires that system to be designed and operated as a single unit even though the components are in fact separately located.

5.2.2 Companies Engaged in the Development of Vuilleumier Refrigerators

At the present time no companies are known to be engaged in the development of Vuilleumier (VM) refrigerators in the range required for this study (5-100 watts). Several companies are engaged in the development of miniature units in the range of 0.1 to 2 watts, however, and several prototype units have been built and tested. Many of these units are being developed with the express goal of long term orbital operation. The VM cycle has certain advantages over other cycles considered which make it sufficiently interesting to explore for this application, even though units do not exist in the 5-100 watt range. The following list summarizes the development efforts known to be currently underway:

Hughes Aircraft Company: Hughes has been actively engaged in the development of the VM cycle for at least 3 years. Their activities include the fabrication and testing of five prototype units. HAC is preparing to put one of their VM prototype units in space for extended orbital use in the near future. Some of the prototype models developed were under contract to Air Force Flight Dynamics Laboratory, WPAFB, during the period March 1967 to April 1968 (Contract F33615-67-C-1532)[19].

Phillips Laboratories: Phillips Laboratories has built two small prototype units recently and has successfully tested these. The units have been built with the application of military infrared systems in mind, utilizing their quiet operation to advantage.[18]

Garrett Air Research: Garrett Air Research presently is engaged in a contract (NAS 5-21096) with Goddard to develop and test a VM cryogenic refrigerator for approximately 5W cooling at $75^{\circ}K$ for space-flight usage. The lifetime goal of the system is 2 yr. to 5 yr. Work on this contract was recently initiated.

Submarine Systems (Division of Sterling Electronics): Submarine Systems has recently entered the VM development area. The principle man responsible for the refrigeration work is Kenneth Cowans, formerly with Hughes Aircraft Company. Work on VM units includes programs under contract to WPAFB for development of a space flight unit and Ft. Belvoir in the area of night vision, Contracts F 33615-70-C-1130 and DAAK 02-70-C-0436, respectively. These contracts call for development, fabrication and testing of VM units.

5.2.3 Analysis of Data on Units

Data on prototype units from Hughes and Phillips were the only information found on this recently developed unit. The data for five Hughes prototype units and two Phillips prototype units is presented in Table 5-2.

The coefficient of performance of these units is plotted as a function of refrigeration capacity in Figure 5-16. As shown, the cooling capacity only goes to two watts, substantially below the requirements of this study. Figure 5-17 presents the C.O.P. as a function of the refrigeration temperature, and Figure 5-18 presents the % Carnot efficiency, as a function of refrigeration capacity, showing efficiencies in the range of 1 to 4% for the small prototype units.

Figures 5-19 through 5-21 present the data on specific weight and specific volume for the VM system. Curve fits are made at 77°K only where four data points are available on the prototypes.

It should be anticipated that since the units are prototype and the development history is quite limited that substantial improvements may be forthcoming in the performance of units based on this cycle. The comparison between the performance of this cycle and others is discussed in Section 7.

5.3 The Joule-Thomson Refrigerator

5.3.1 Operation

A practical Joule-Thomson refrigerator is shown in Figure 5-21a. The cycle is identical to the Brayton cycle of Figure 5-41 except for one important modification. The expansion process, 4 to 5, is accomplished by isenthalpic expansion through a throttling valve rather than by expansion in a work producing device. Since no expansion work is produced, no heat addition is required in the load exchanger to replenish the lost internal energy of the working fluid. The cycle produces refrigeration by virtue of a useful side effect of non-ideal behavior at the sink temperature. The cooling effect is given by

$$q_c = \dot{m}\,[h_5 - h_5]$$
$$= \dot{m}\,[h_5 - h_4]$$

A heat balance on the main heat exchanger yields

$$h_1 - h_6 = h_3 - h_4$$

Table 5-2

EXISTING VUILLEUMIER PROTOTYPE RE

Manufacture		Hughes Aircraft	Hughes Aircraft	Hughes Aircraft	Hughes Aircraft	Hughes Airc
Trade Name						Prototype
Model		Prototype	Prototype	Prototype	Prototype	X447550-10
I.D. Number		11	12	13	14	15
Refrigeration Range		≈ 77°K	≈77°K	15 K to 75°K	25 - 75°K	30 - 75°K
Cycle		Vuilleumier	Vuilleumier	Vuilleumier	Vuilleumier	Vuilleumier
Working Fluid		Helium	Helium	Helium	Helium	Helium
High Pressure			600 psi			400 psi
Low Pressure						
Minimum Temp					16°K	
Cool-Down Time		30 min.	10 min.	30 min.	30 min.	30 min.
Expander RPM			600			240
Volts - Phase - Frequency		28 VDC	28 VDC	28 VDC	115 - 3 - 400	28 VDC
Cooling Means		Air	Air	Liquid	Liquid	Liquid
Ambient Temp. Reqmts.			-55°C to 71°C			
Required Attitude		Any	Any	Any	Any	Any
Cryostat Dimensions		7.15 x 7.15 x 8	6.5 x 5.7 x 5.1	10.5 x 13.6 x 7.8	7.5 x 9.5 x 10"	10.5 x 13.6
Compressor Dimensions						
System Volume		410 in.^{3}	190 in.^{3}	1,110 in.^{3}	712 in.^{3}	1,110 in.^{3}
System Weight			5.75 lb			181 lb
MTBF			5,000 hr goal			
Maintenance Interval		3,000 hr goal	1,000 hr	10,000 hr goal	1,000 hr	10,000 hr gc
Near 20°K	Refrigeration			0.15 w at 15°K	2 w at 25°K	0.5 w at 30
	Power Input			370 w	1,200 w	550 w
	COP			.000405	.00167	.00091
	% Carnot			0.77%	1.84	0.82%
	lb/watt					36
	in.^{3}/watt			16,700	365	2,220 (1st Stage)
77°K	Refrigeration	0.6 w	1.5 w			6 w
	Power Input	60 w	200 w			500 w
	COP	.01	.0075			
	% Carnot	2.9%	2.2%			
	lb/watt		3.83			
	in.^{3}/watt	683	127			

(1) Same Units at two different operating conditions
(2) Based on 350°K Ambient

FOLDOUT FRAME 1

5-2
REFRIGERATORS (SMALL UNITS)

Aircraft	Phillips Laboratory	Phillips Laboratory	Phillips Laboratory		
pe					
)-100	Prototype	Prototype[1]	Prototype[1]		
	16	17	18		
K	77 - 200°K	≈77°K	≈77°K		
nier	Vuilleumier	Vuilleumier	Vuilleumier		
	Helium	Helium	Helium		
	38 atm	30 atm	40 atm		
	28 atm				
	70°K				
	9 min.				
	750	600	600		
	Air	Air	Air		
	Any	Any	Any		
.3.6 x 7.8"	12 x 8 x 6	16.5 x 7.1 x 7.1"	16.5 x 7.1 x 7.1		
1.^{3}	580 in.^{3}	820 in.^{3}	820 in.^{3}		
	81 lb	15 lb	15 lb		
ır goal					
: 30°K					
ge)					
	0.5 w	1 w	2 w		
	70 w	120 w	191 w		
	.00715	.00833	.0105		
	2.07%	2.9%[2]	3.7%[2]		
	16	15	7.5		
	1,160	820	410		

FOLDOUT FRAME 2

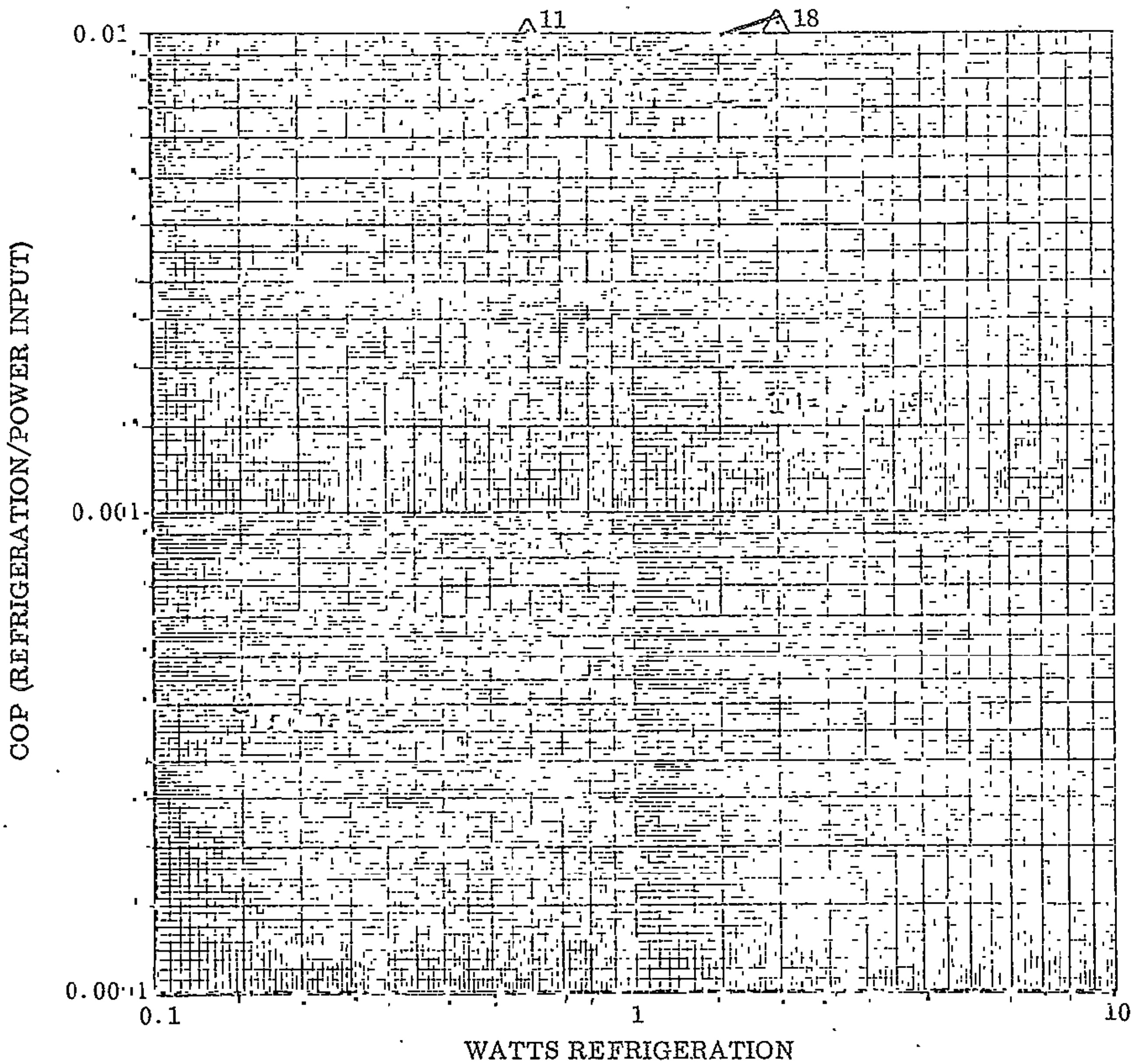


Fig. 5-16 Coefficient of Performance Verses Cooling Capacity (Vuilleumier Cycle)

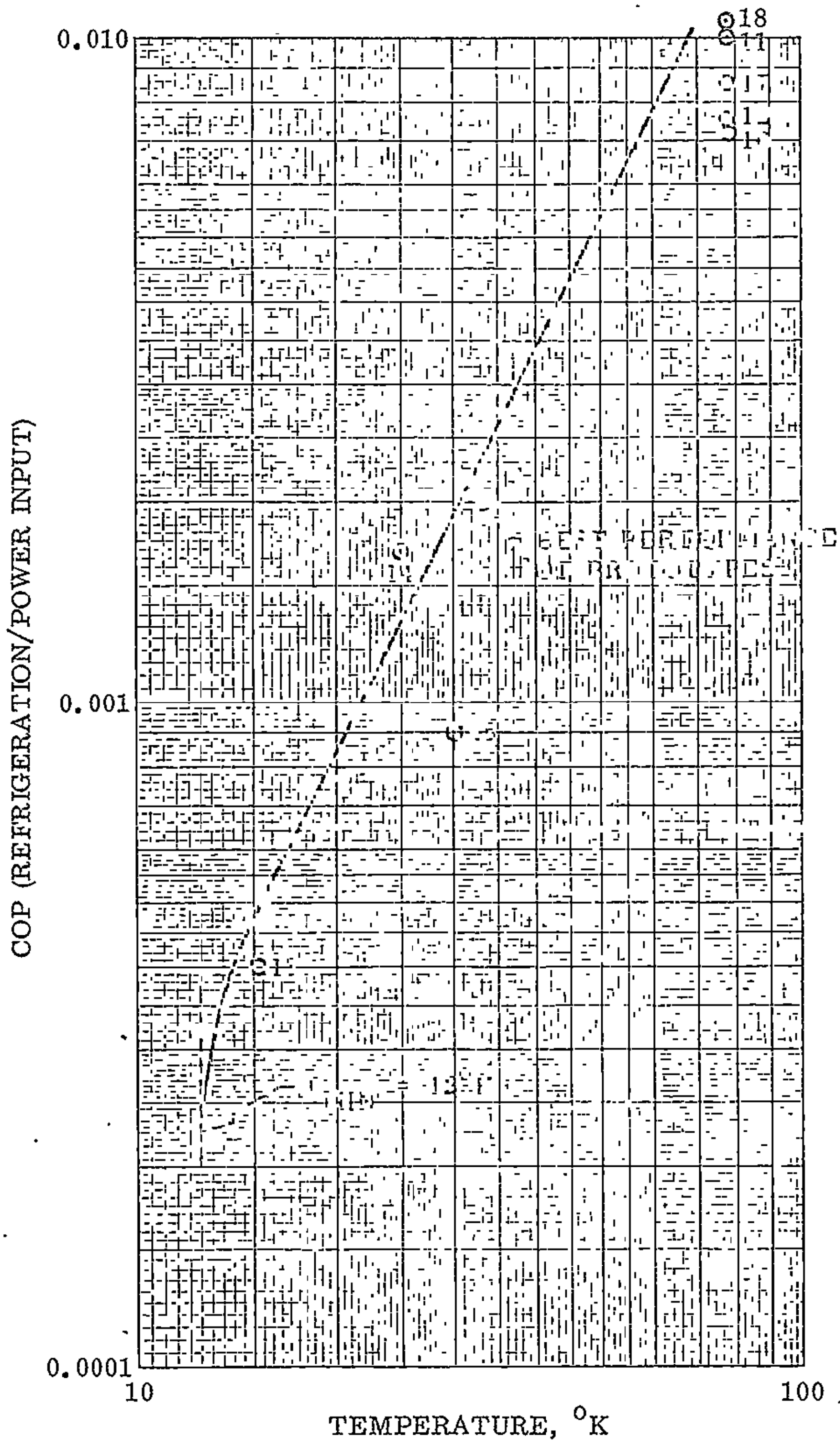


Fig. 5-17 Coefficient of Performance of Prototype Vuilleumier Refrigerators (Small Units)

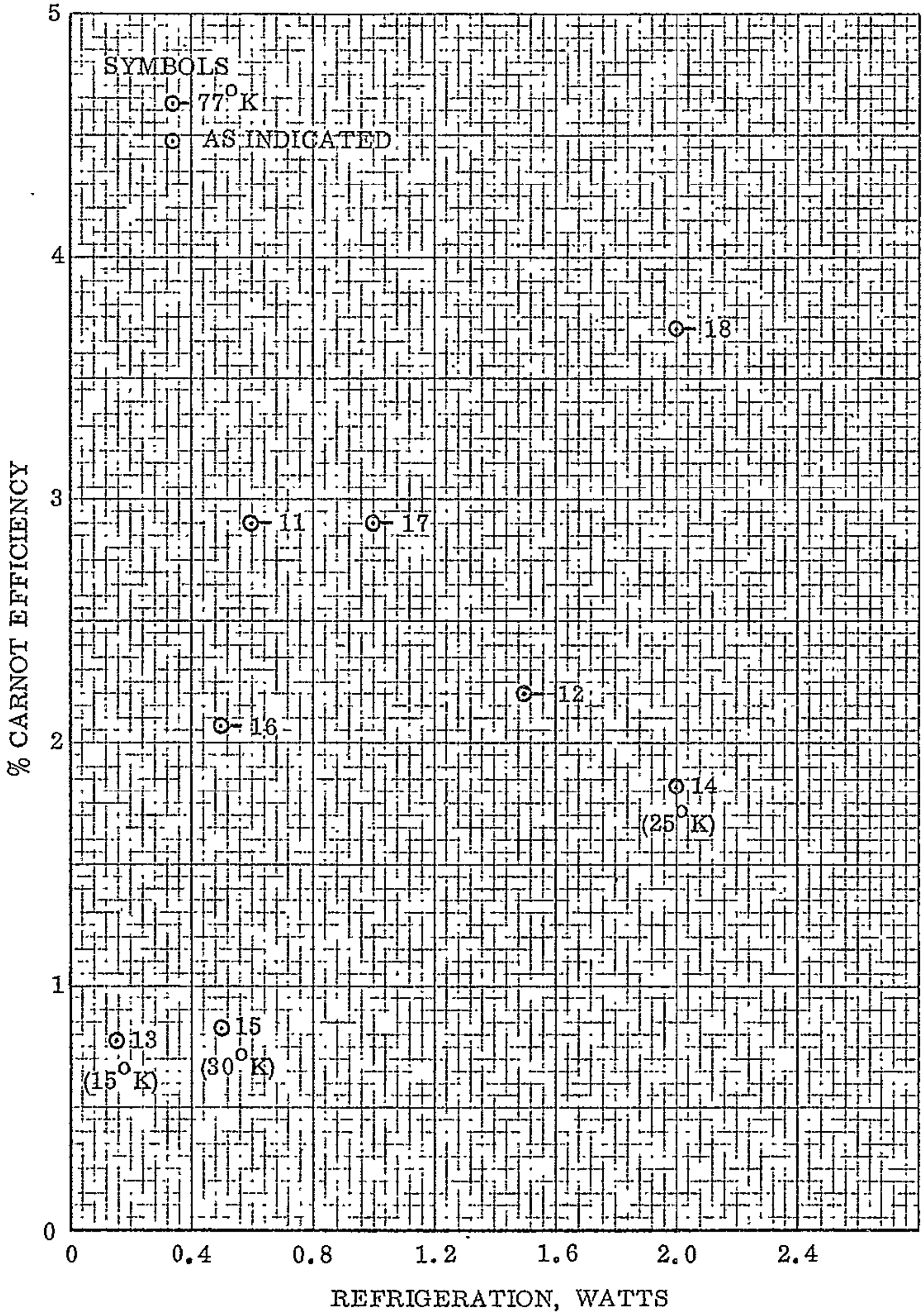


Fig. 5-18 Percent Carnot Efficiency of Vuilleumier Prototype Refrigerators

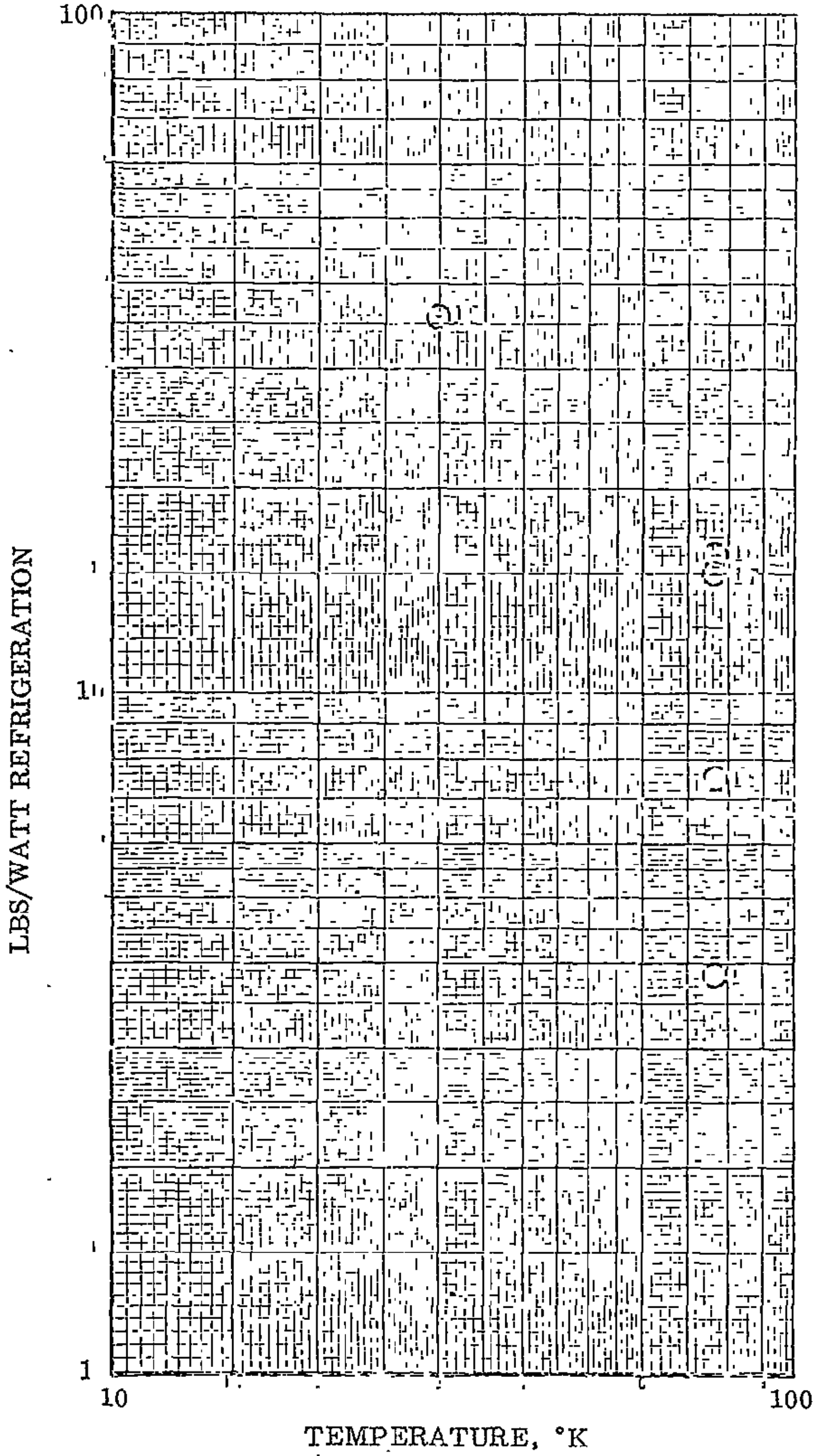


Fig. 5-19 Specific Weight Versus Temperature (Vuilleumier Cycle)

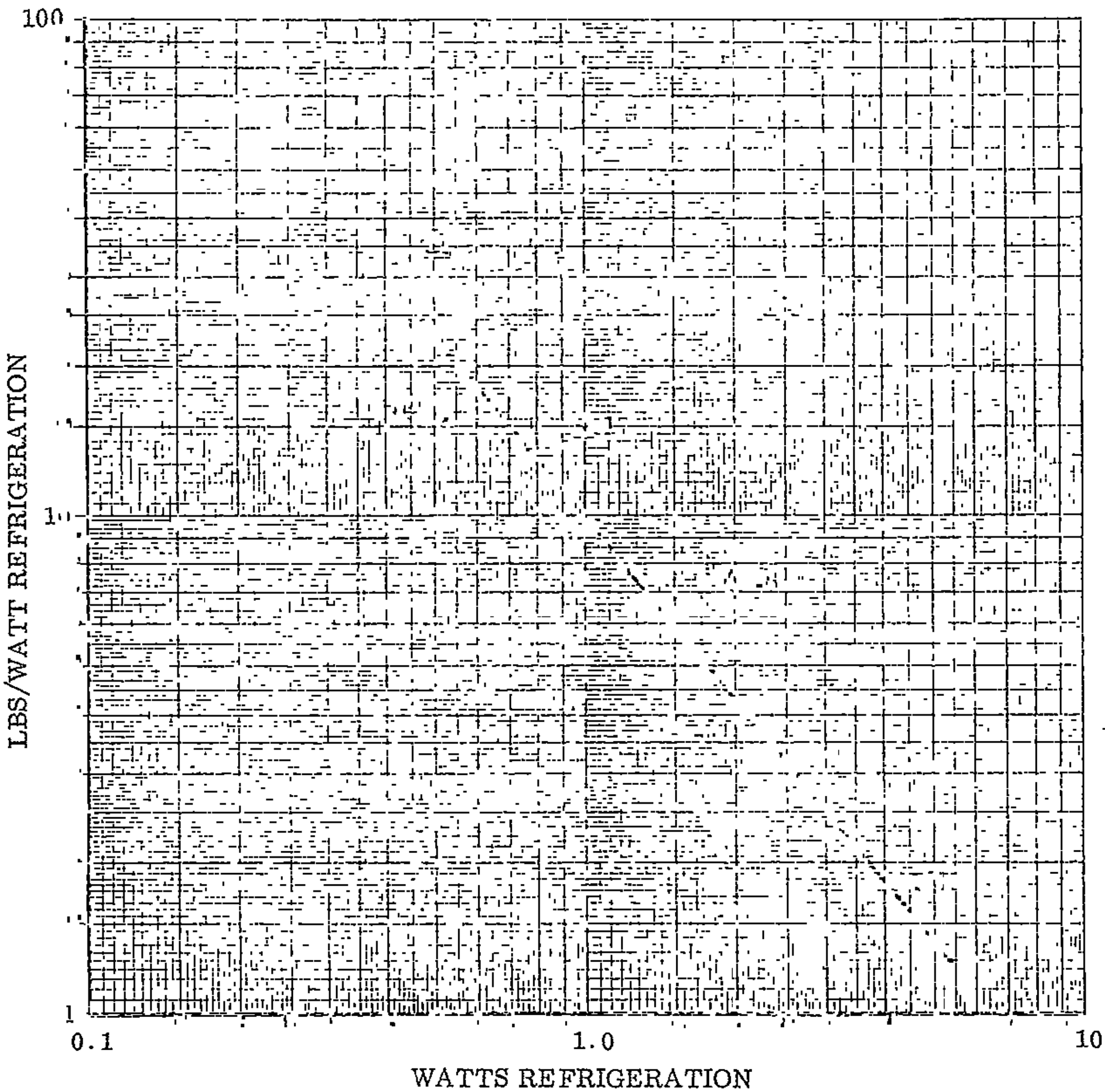


Fig. 5-20 Specific Weight Versus Refrigeration Capacity (Vuilleumier Cycle)

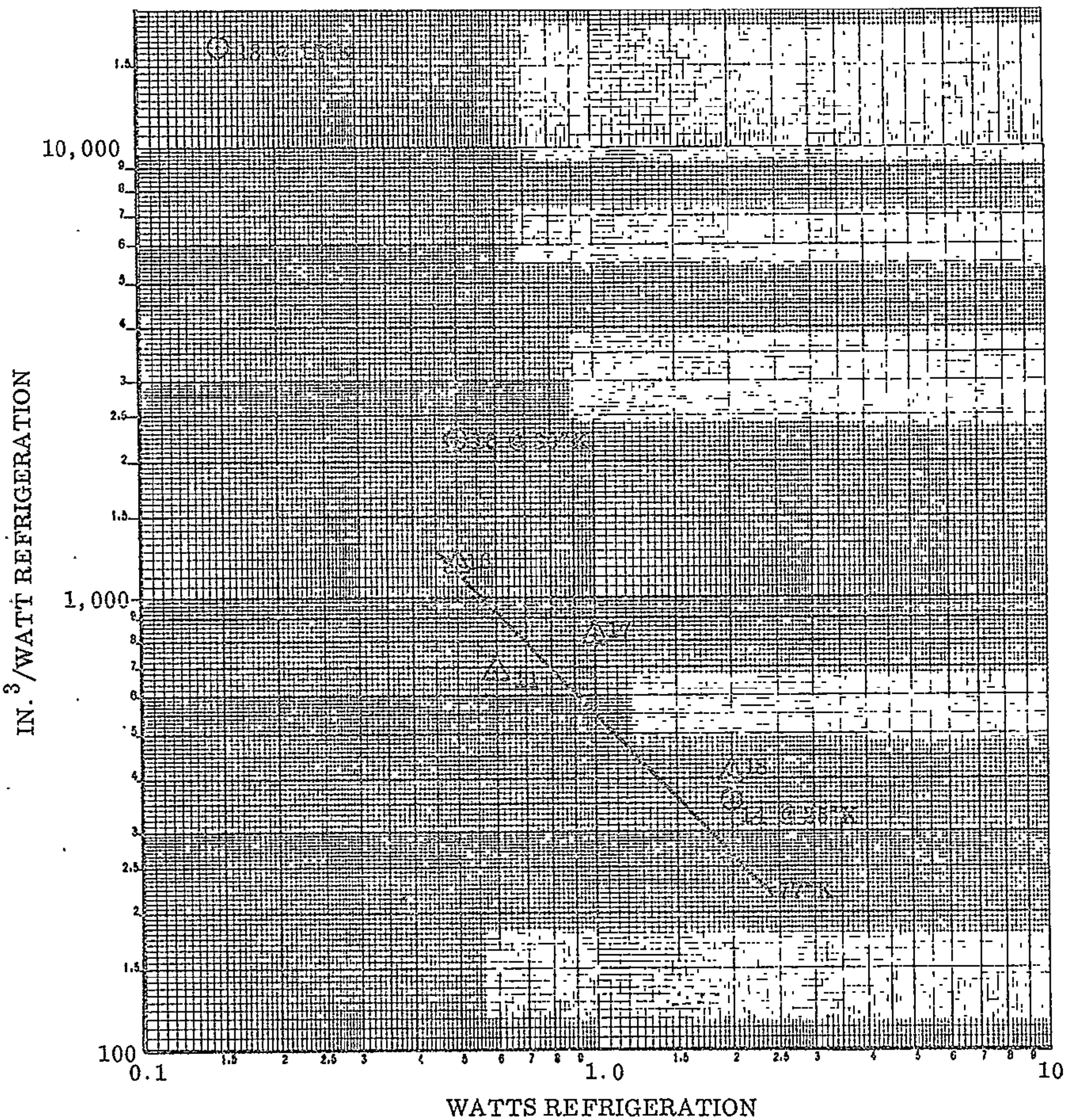


Fig. 5-21 Specific Volume Versus Refrigeration Capacity (Vuilleumier Cycle)

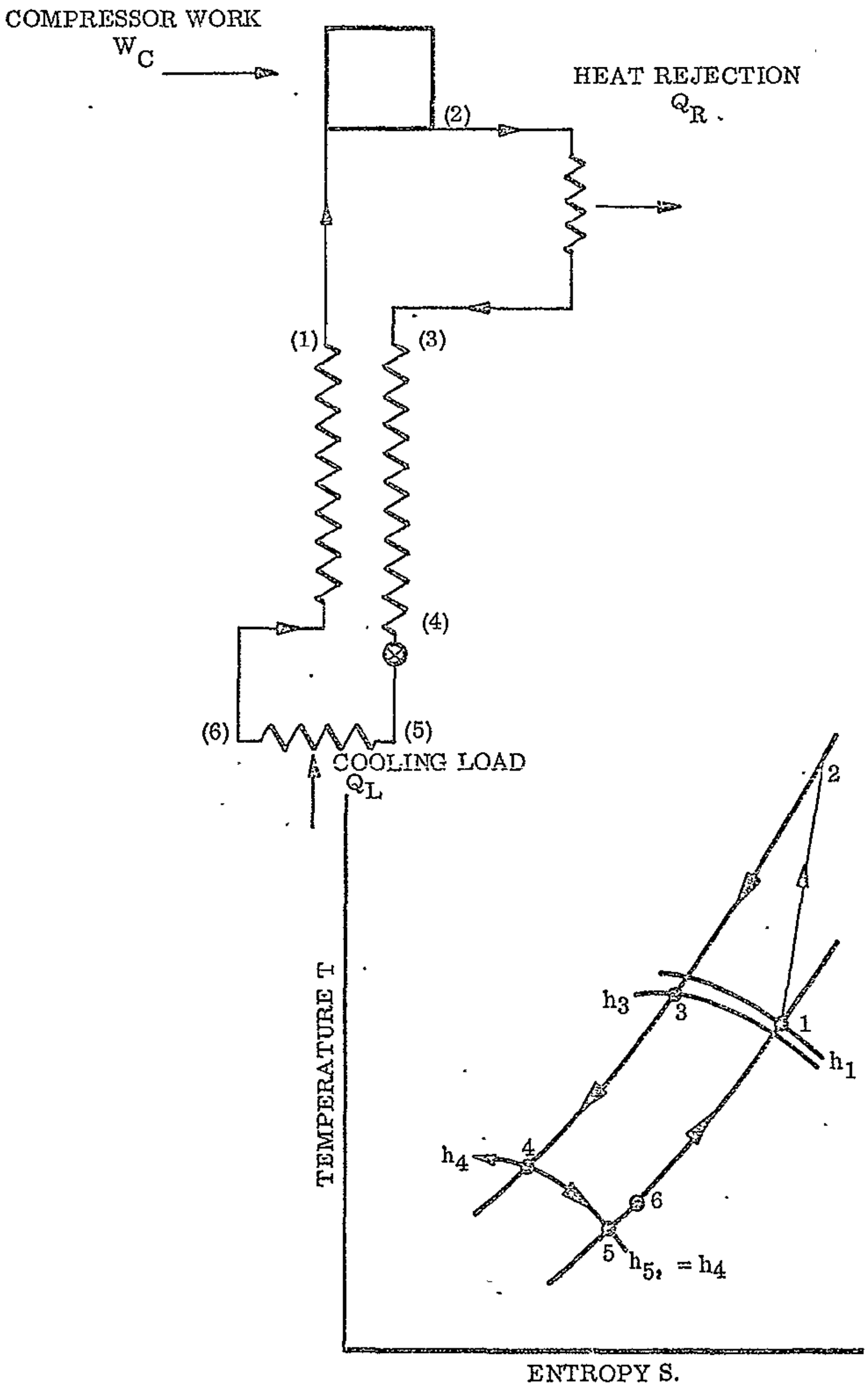


Fig. 5-21a Joule-Thomson Cycle

Hence

$$q_c = \dot{m}\,[h_1 - h_3]$$

For q_c to be positive $(\partial h/\partial p)_{T_3}$ must be negative.

With a 100 percent efficient main heat exchanger

$T_1 = T_3$ and the maximum value of q_c is given by

$$q_c(\max) = \dot{m}\,[h(P_1T_3) - h(P_2,T_3)]$$

With less than 100 percent efficiency, q_c is given by

$$q_c = \dot{m}\,[h(P_1,T_1) - h(P_2,T_3)]$$

As T_1 is reduced due to less efficient heat exchange, $h\ (P_1T_2)$ will decrease and q_c will eventually become zero. The performance of a Joule-Thomson system is therefore limited by the sign and magnitude of $(\partial h/\partial P)_{T_3}$, and by the ability of the main heat exchanger to permit utilization of this effect.

For the present application only nitrogen has both a negative value of $(\partial h/\partial P)_{T_3}$ at normal ambient temperatures and is still a vapor in the temperature range of interest. Those fluids which condense at temperatures lower than nitrogen – neon, hydrogen and helium – can be used for very low temperature Joule-Thomson refrigerators if T_3 is reduced to a point where $(\partial h/\partial P)_{T_3}$ is negative. This can be done by precooling the fluid using another type of refrigeration system. Thus, Joule-Thomson systems can be used in double or triple cascade to obtain cooling in the range of liquid hydrogen or liquid helium temperatures, as shown in Figure 5-22.

The cooling capacity, q_c, of a single stage Joule-Thomson system is given by equation

$$q_c = \dot{m}\,(h_1 - h_3)$$

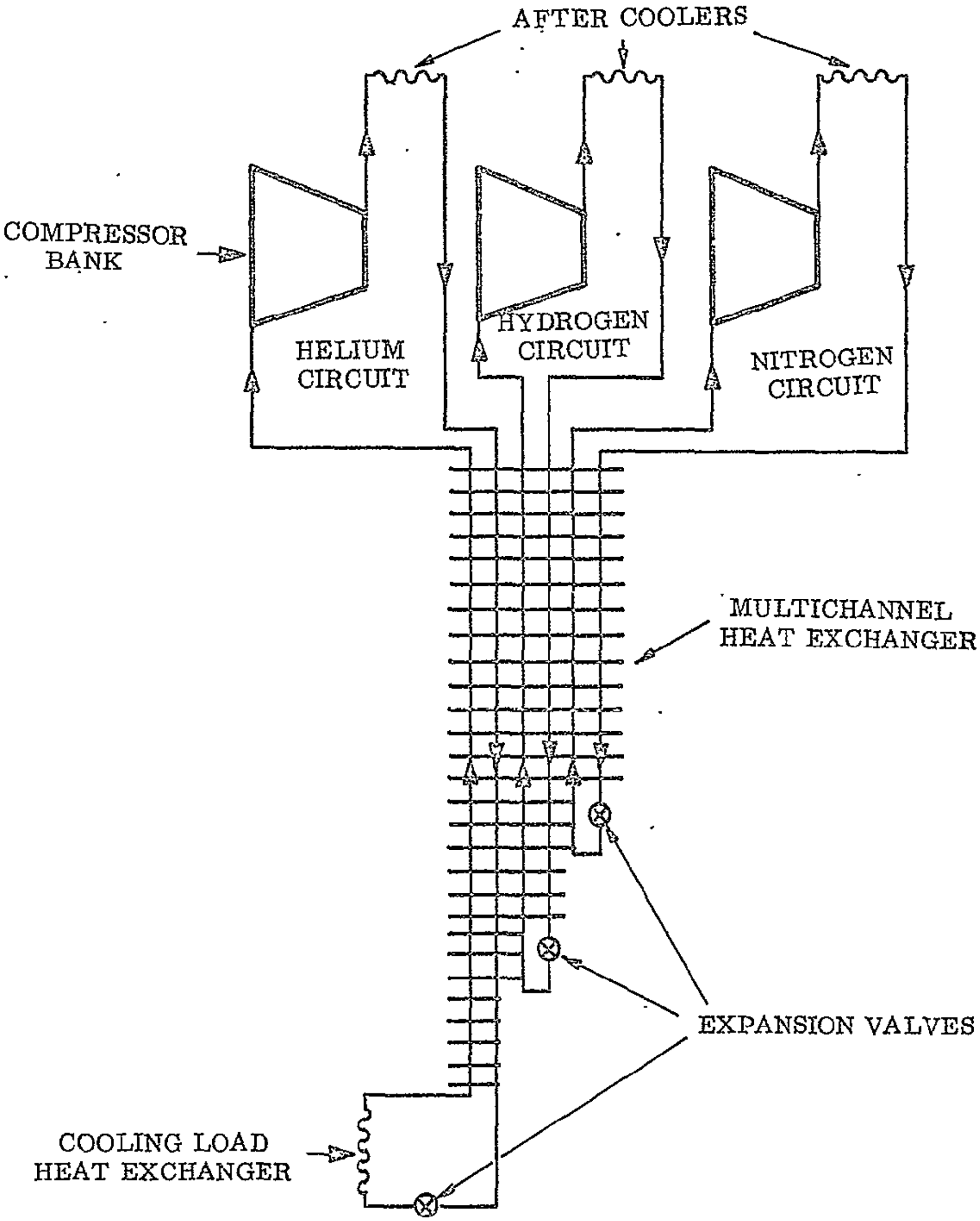


Fig. 5-22 A Three-Fluid Cascaded Joule-Thomson Refrigerator

the power required, W, is given by

$$W = \dot{m}\,[h_2 - h_3]$$

The significant parameters for assessing the performance of a Joule-Thomson refrigerator are the compression ratio, the gas properties at the warm end and the heat exchanger effectiveness. Dean and Mann[20] present values of W/q_c as a function of these quantities for Joule-Thomson refrigerators using nitrogen, hydrogen and helium as working fluid. Available data on closed cycle systems is sparse and generally limited to a few watts of cooling. The parameters of larger J-T units, in the range of this study, may be obtained by combining the data from this study with the physical dimensions, capacity and efficiency of practical compressors, and heat exchangers, from Section 4.

5.3.2 Manufacturers of Joule-Thomson Refrigerators

The following companies are presently engaged in production and development of J-T refrigeration systems.

Air Products and Chemicals, Inc.
Garrett Corporation
Santa Barbara Research
Hymatic Engineering Company

The majority of J-T units produced by these companies are open cycle systems, in which the working fluid is supplied by a high pressure stored gas source. In this usage only relatively short cooling periods are available. J-T units have been used in space in various applications, but their principal advantage is in providing short term cooling (i.e., minutes or hours) after extended durations in space. For example, a two-stage J-T unit was supplied by Air Products for cooling of an infra-red system on the Mariner missions.

Data on existing closed cycle J-T units is quite sparse and there appears to be no production models available which meet the refrigeration requirements of this study. The performance of the closed cycle J-T system is keyed to the compressor, and data in Section 4 can be utilized to make fairly close estimates of the performance which would be expected from closed cycle J-T systems in the range of 5-100 watts.

Garrett AiResearch Manufacturing: Garrett Corporation makes several closed cycle units primarily for use in aircraft. These units use N_2 as the working fluid and are therefore limited to a lower temperature of $75^{o}K$. These units provide cooling up to 5 watts.

Air Products and Chemicals: Air Products and Chemicals produces a wide variety of open cycle J-T units. In addition they produce two closed cycle J-T units, one a single stage unit to provide 2W at $77^{o}K$ and a second stage unit which supplies 0.35W at $23^{o}K$. Air Products has recently developed a modified Solvay unit for commercial and military application, and they feel that the potential of this unit for long term closed cycle application is superior to the J-T unit.

Santa Barbara Research: Santa Barbara Research has produced a variety of open cycle J-T refrigerator and also has produced a single, closed cycle J-T unit for aircraft use. It is believed that they are not active in the development of closed cycle J-T systems at this time.

Hymatic Engineering Co., Ltd: Hymatic Engineering Company specializes in open cycle type J-T coolers for infrared detectors. Hymatic has been engaged to a small degree in closed cycle J-T systems.

Hughes Aircraft Company: Hughes has produced several units in the past for cooling systems on aircraft, but is not now engaged in development of J-T units, and does not produce a unit which is generally available to industry.

5.3.3 Discussion of Data on J-T Units

Characteristics of Existing Joule-Thomson Refrigerators: Data has been assembled on the characteristics of closed-cycle J-T systems made by four companies. None of these units produces the degree of refrigeration required in this study, however, the data is useful in assessing the relative performance of units. Data from the following units has been assembled.

<u>Garrett AiResearch</u>

Model 133488
144406

<u>Air Products</u>

Model J-80-1000
J-30-3500

<u>Santa Barbara Research</u>

Prototype unit

<u>Hughes Aircraft Company</u>

Prototype Unit

These data are not complete and primarily cover the near 77°K temperature range where a single stage unit with N_2 as the working fluid can be used. One exception to this is the two stage unit produced by Air Products which provides cooling to approximately 23°K. The largest cooling capacity of the units are 12 watts.

The data on the closed cycle systems is tabulated in Table 5-3. The C.O.P. of existing closed cycle units is presented in Figs. 5-23 and 5-24. The ideal performance of a J-T unit operating at near optimum pressure (2400 psi) is also shown for comparison at 75 to 95°K. The system which provides the highest thermodynamic performance (No. 26) delivers about 25% of the ideal performance of the J-T cycle.

A curve fit of the data points was made only at 77°K, and is shown on Fig. 5-24.

Although the maximum theoretical performance of the J-T system is not given by the Carnot efficiency, the % Carnot efficiency vs. refrigeration capacity is shown in Fig. 5-25 so that it may be compared with the other cycles on the same basis. The plot shows that the J-T systems considered produce approximately 2% of the Carnot efficiency. A considerable improvement should be experienced

CLOSED CYCLE [illegible]

Manufacturer	Garrett [illegible]
Trade Name	None
Model	133[illegible]
I.D. Number	21
Refrigeration Range	$\approx 77^{\circ}K$
Cycle	J-T
Working Fluid	N_2
High Pressure	155 atm
Low Pressure	1 atm
Minimum Temperature	$75^{\circ}K$
Cool-Down Time	12 min.
Compressor RPM	
Volts - Phase - Frequency	115/[illegible]
Cooling Means	Air
Ambient Temp. Reqmts.	$-40^{\circ}C$ to [illegible]
Required Attitude	Any
Cryostat Dimensions	
Compressor Dimensions	6.5" [illegible] (3-stage)
Cryostat Wt.	
Compressor Wt.	
System Wt.	22.5 lb
Compressor Volume	
Cryostat Volume	
System Volume	$1.5\ ft^3$
MTBF	1,000 hr [illegible]
Maintenance Interval	300 and [illegible]
System Cost	$9,[illegible]
$23^{\circ}K$ Refrigeration	
$23^{\circ}K$ Power Input	
$23^{\circ}K$ COP	
$23^{\circ}K$ % Carnot	
$23^{\circ}K$ lb/watt	
$23^{\circ}K$ $in.^3$/watt	
$77^{\circ}K$ Refrigeration	5 w
$77^{\circ}K$ Power Input	650 w
$77^{\circ}K$ COP	.0077
$77^{\circ}K$ % Carnot	2.[illegible]
$77^{\circ}K$ lb/watt	4.5
$77^{\circ}K$ $in.^3$/watt	518

FOLDOUT FRAME 1

le 5-3

ON REFRIGERATORS (SMALL UNITS)

rrett AiResearch	Air Products	Air Products	Santa Barbara Research Center	Hughes Aircraft		
ne	None	None	None	Prototype		
4406	J-80-1000	J-30-3500				
	23	24	25	26		
77°K	≈ 77°K	23°K	≈79°K	77°K		
r	J-T	J-T	J-T	J-T		
	N_2	N_2 and He	N_2	N_2		
6 atm						
tm						
°K	75°K		75°K	75°K		
5 m	5 min.		5 min.	5 min.		
	3,850	3,850				
5/208 - 3 - 400						
r	Air	Air	Air			
0°C to 71°C						
y						
	5" x 8" x 12" (2-stage)		7" D x 12.5" L			
.5 lb	18 lb		16 lb	40 lb		
45 ft^3						
000 hr est						
0 hr	500 hr	500 hr	500 hr			
,000	$9,000		$10,000			
		0.35 w (23°K)				
		1,050 w				
		0.000333				
		0.4%				
v	2 w		2 w	12 w		
0 w	600 w		326 w	750 w		
067	.00333		.00614	0.016		
0%	0.9%		1.8%			
5	9		8			
9						

FOLDOUT FRAME 2

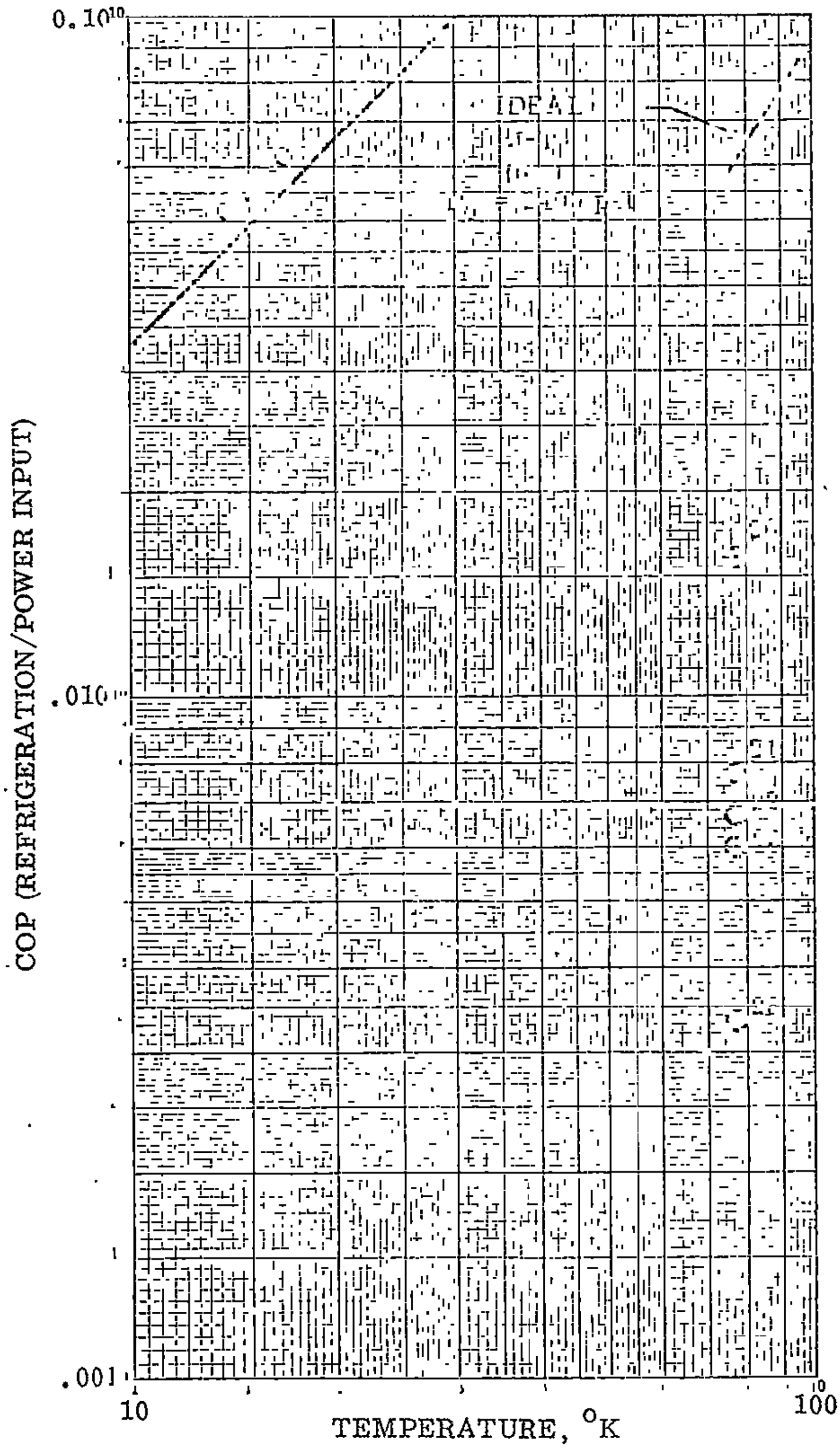


Fig. 5-23 C.O.P. Versus Temperature for Joule-Thomson Closed Cycle Systems

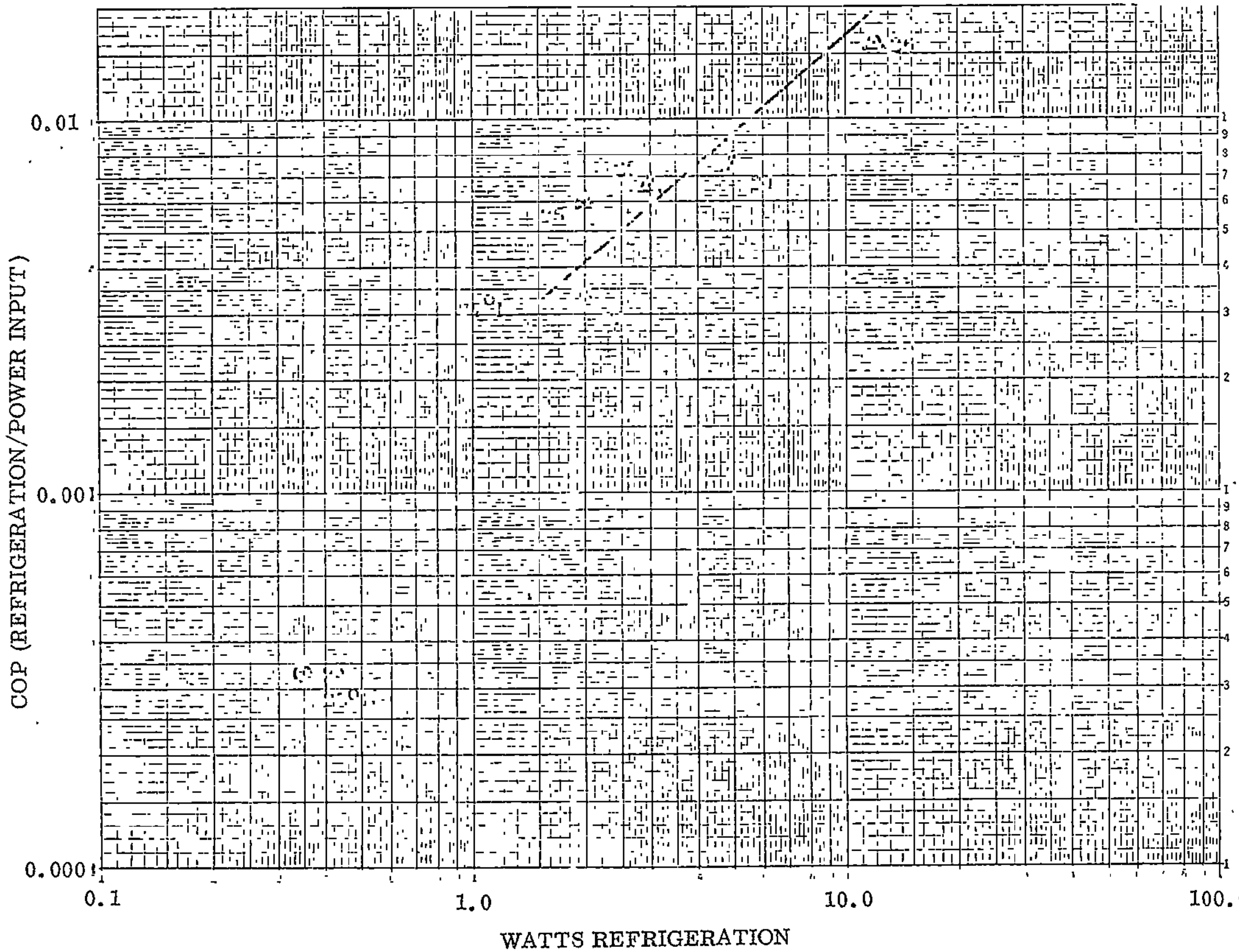


Fig. 5-24 C.O.P. Versus Capacity for Joule-Thomson Closed Cycle Systems

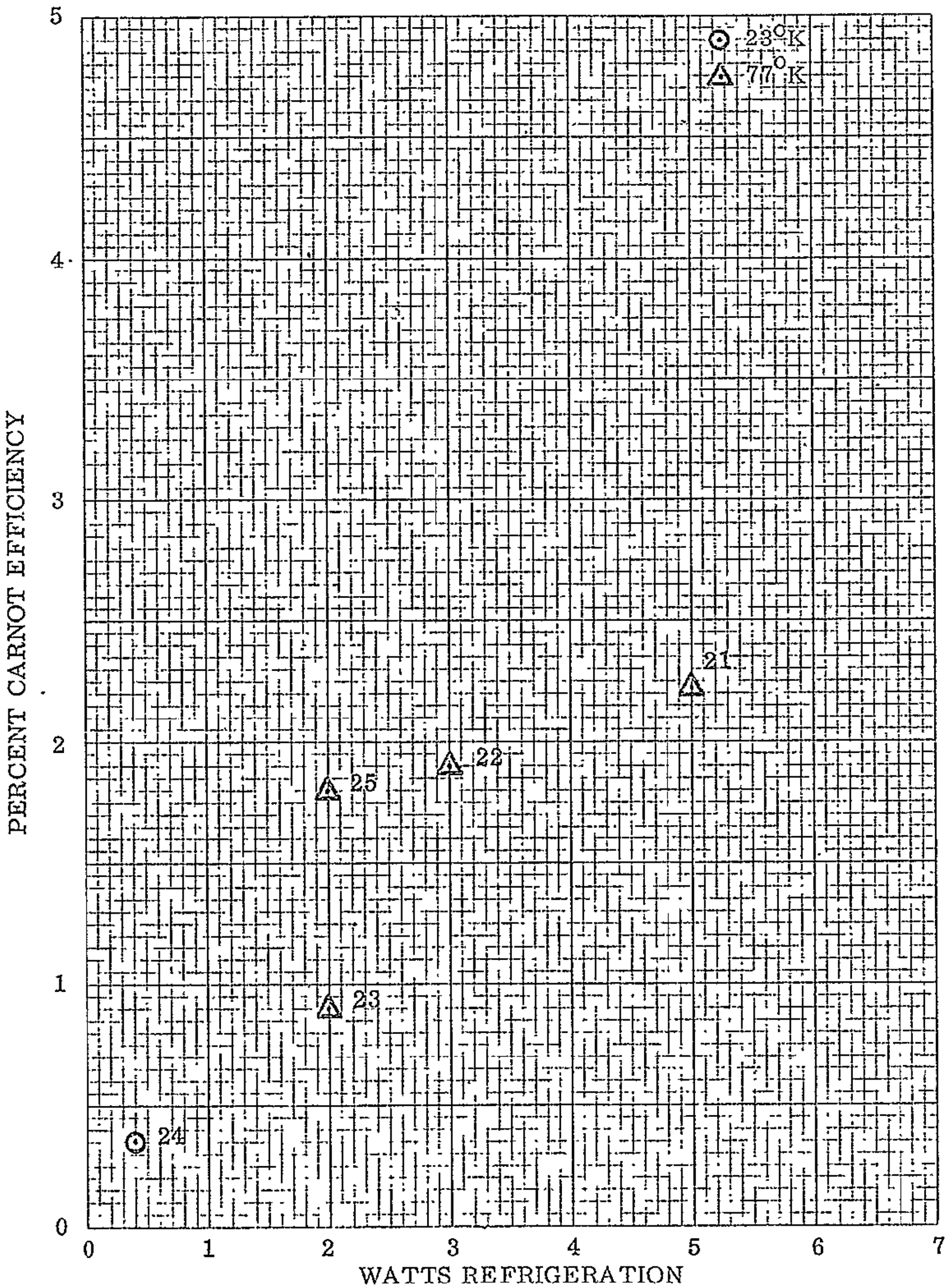


Fig. 5-25 Percent Carnot Efficiency for Joule Thomson Systems

for units of greater capacity, however, data was not available for large closed cycle units.

Figure 5-26 shows the specific weight of the units as a function of refrigeration capacity at 77°K only. Figure 5-27 presents the specific volume of two units for which system volume was available. The available data does not permit curve fitting; the trend from the two points is obviously wrong, since it indicates the specific volume of the systems would increase with capacity.

5.4 Independent Component Regenerator Refrigerators (Gifford-McMahan, Solvay, and Taconis)

The practical Stirling and VM refrigerators achieve compression, expansion and heat transfer processes in a single mechanical unit however refrigerators can be built which use regenerative exchangers in which the compression, expansion and heat exchange components are completely separate, if switching valves and surge volumes are used to isolate the time-dependent operation of the exchanger from the operation of the compressor and expander. Valving introduces irreversibility which cause more harm to system efficiency if they occur at the cold end than at the ambient end. It is therefore possible to conceive a refrigeration system in which the main regenerative heat exchanger, load exchanger and expander operate as one unit and the compressor as another. Such a system has many practical advantages in that separation of components is achieved but no low temperature valving is required. Valves and surge tanks are used only at the ambient end. Such an arrangement permits many areas of design freedom compared to the Stirling or VM refrigerator. The compressor and heat exchanger-expander interface requirements are confined to working fluid flow rates and pressures. The type of compressor used to supply the working fluid at these rates and pressure can be selected optimally from all possible types - dynamic, positive displacement or thermal.

The exchanger-expander unit will be very similar to the cold end of a Stirling or a VM refrigerator. With separation, however, there is greater freedom of choice of displacer drive and means of extracting expansion work. By changing the valve timing the shape of the Pv diagram can be influenced to some degree.

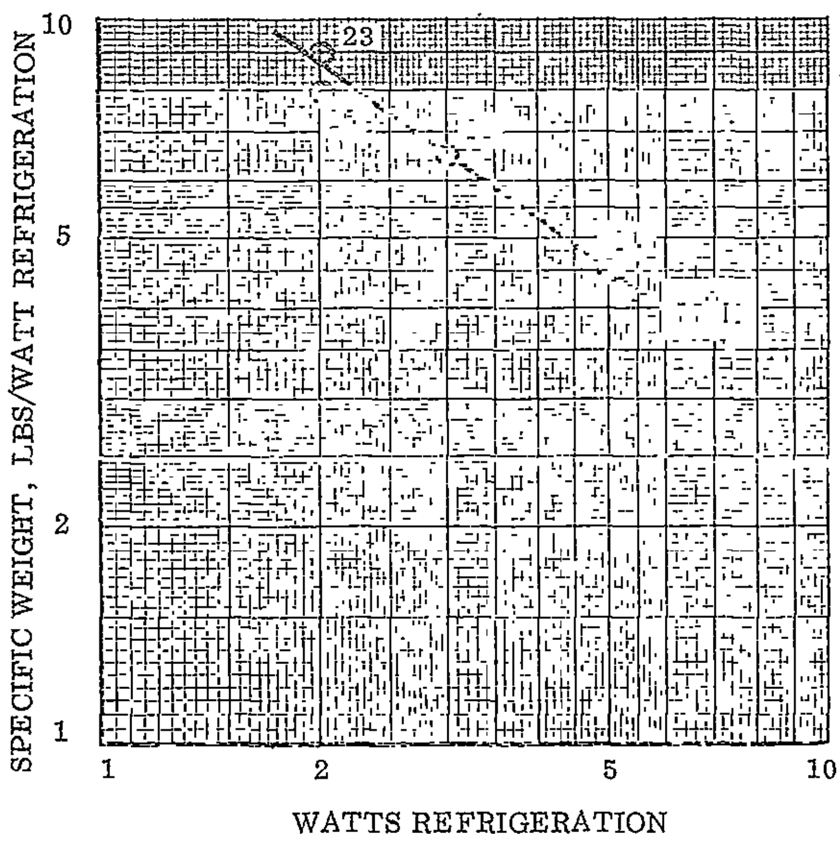


Fig. 5-26 Specific Weight vs. Capacity for Closed Cycle Joule-Thomson Units

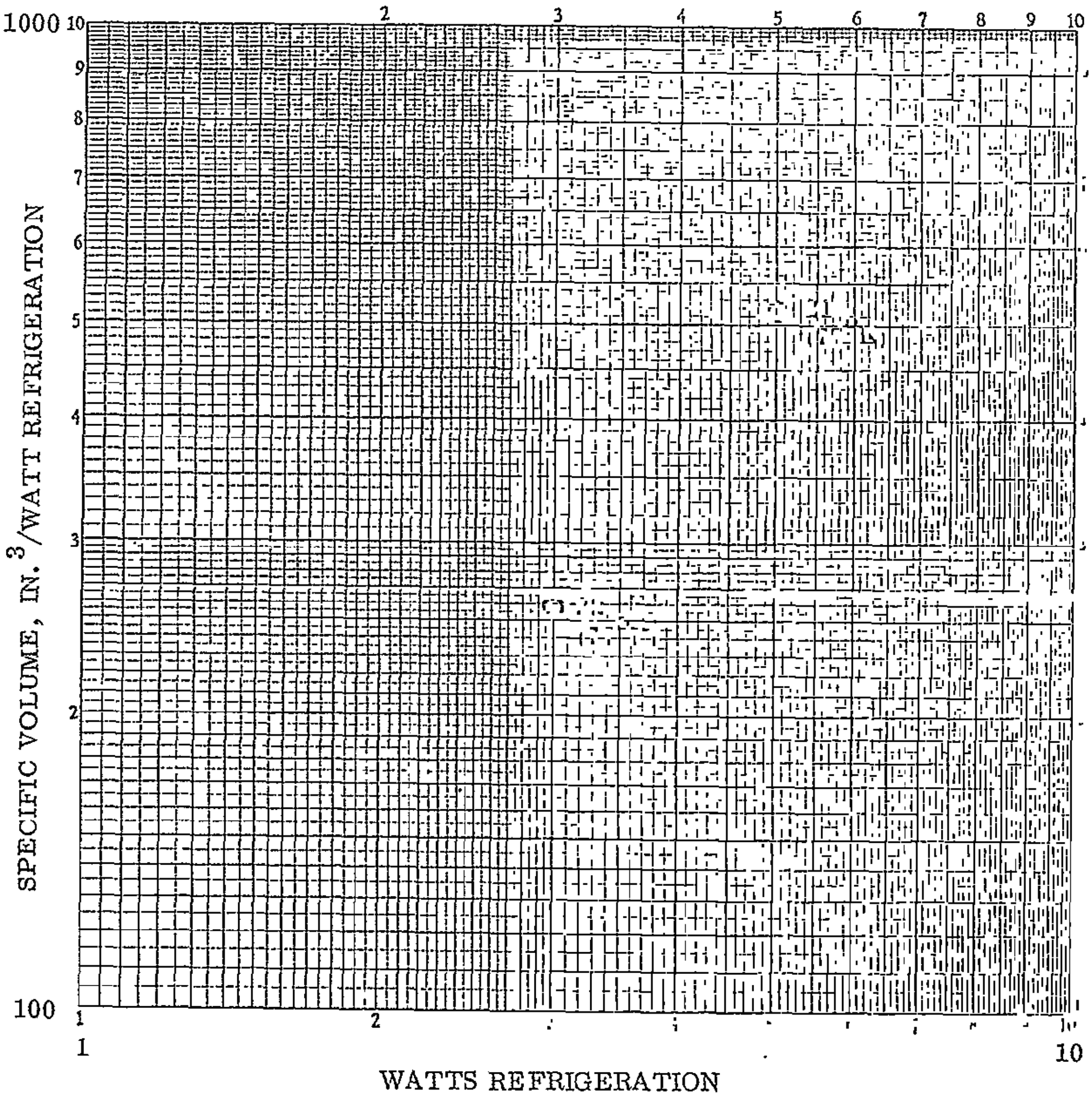


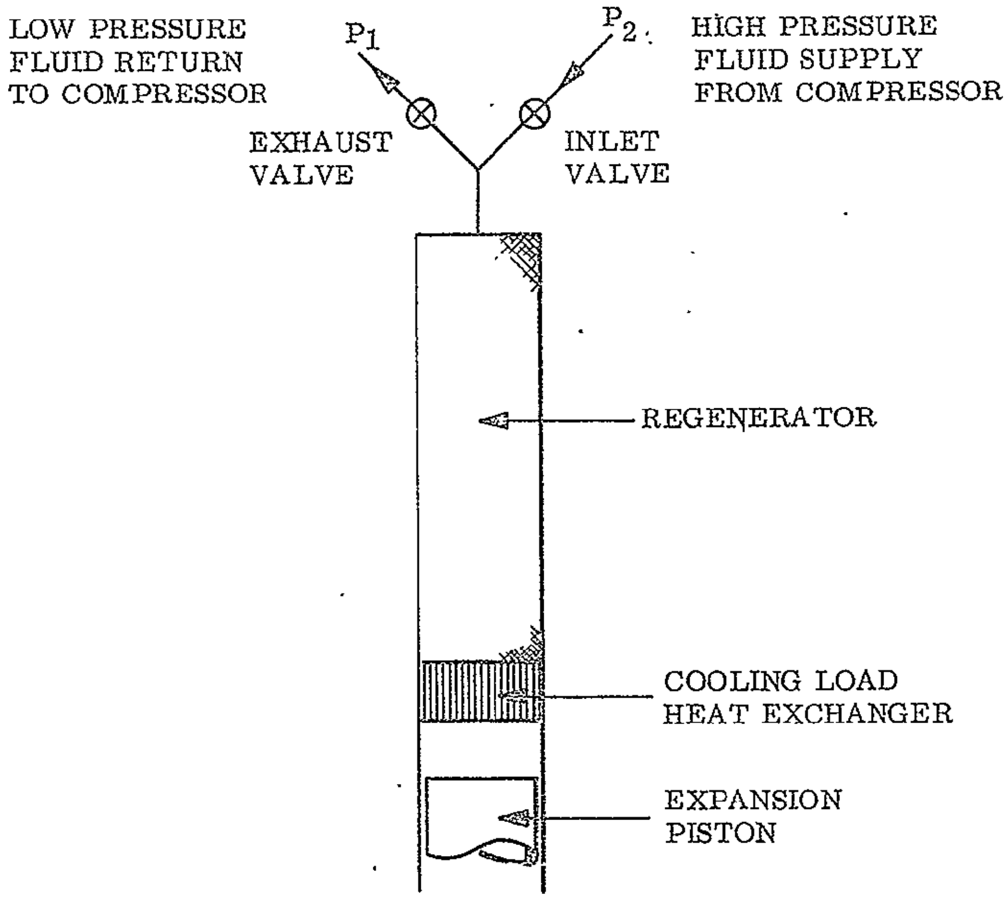
Fig. 5-27 Specific Volume of Closed Cycle Joule-Thomson Units Vs Capacity

In recent years, this split component system has gained a great deal of popularity. By separating the expander from the compressor, it is possible to construct a system consisting of a simple lightweight, compact cooling unit which can be more easily integrated with the load, and a compressor which can be located remotely and connected to the expander with long flexible lines carrying the high and low pressure working fluid. Because of this remote location the compressor design can be optimized for convenience and reliability rather than compactness. Because of the use of valves the fluid flow is unidirectional in the lines, oil separators and filters can be inserted in the lines, permitting the use of reliable and proven oil-lubricated compressors instead of solid lubricated compressors.

It was noted above that this type of refrigerator permits many design variations to be considered within the same basic concept. Because of this characteristic and the commercial attractiveness of the system there are many varities of split component refrigerators on the market. These systems are basically the same in that they nearly all use modified hermetically sealed freon compressors, so that the system variations are confined to the method of operating the exchanger expander unit. However, for reasons of commercial advertising and patent justification plus a certain amount of pedantry, a profusion of names has been applied to the individual expander modifications. They include, Taconis, Solvay, and Gifford-McMahon, with and without the adjective "modified". There seems to be some justification for crediting Taconis with first appreciating the full possibilities of this type of exchanger-expander, although many persons have proposed devices using regenerators all the way back to Robert Stirling and possibly earlier. It is outside the scope of this report to establish the correct name for this family of devices and they will be referred to simply as exchanger-expanders. However, there is a tendency to refer to different expansion processes incorrectly, as different thermodynamic cycles. In this report the distinction between cycles and processes will be kept.

5.4.1 Operation

There are two major techniques for operating expander displacers. One technique is exemplified by the basic Solvay process. In Figure 5-28 the expander



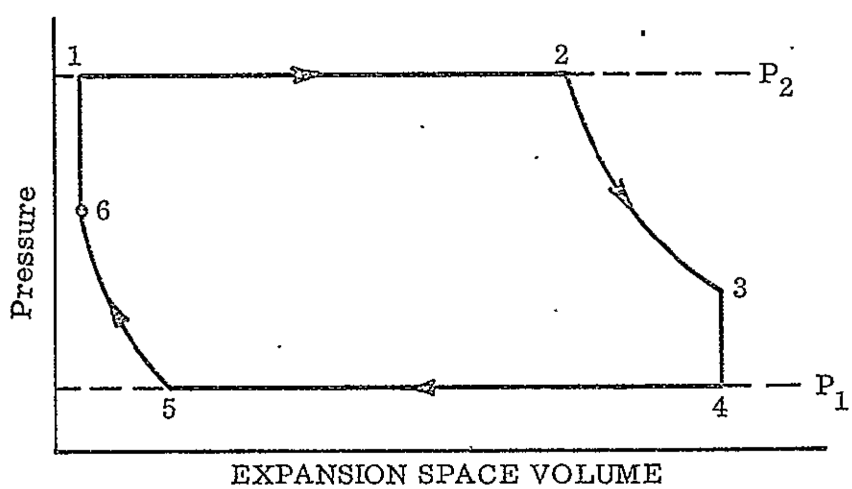


Fig. 5-28 The Solvay Expansion Process

consists of an expansion piston connected to the working fluid and the inlet and exhaust valves through a regenerator. In position 1 the inlet valve is open and the exhaust closed. The regenerator and other void volumes are filled to the higher pressure. From 1 to 2, the piston moves outward and working fluid enters the cylinder after first being cooled in the regenerator. At point 2, the inlet valve is closed and the fluid pressure falls until the piston reaches its outermost position. At position 3, the exhaust valve is opened and the fluid in the system expands irreversibly to the valve at 4. From 4 to 5, the piston moves inward, expelling the cold working fluid from the system after first being warmed in the regenerator. At 5, the exhaust valve is closed and the piston continues to move until it reaches the innermost position at 6. At position 6, the inlet valve is opened and the fluid in the system is compressed irreversibly from 6 to 1. The valve timing points 2 and 5 can be selected such that compression and expansion are reversible, i.e., position 3 and 4, and 6 and 1 are identical. Alternately, the valve timing can be chosen so that 2 and 3, and 5 and 6 are identical maximizing the area of the Pv diagram.

In the solvay process the work of expansion can be extracted mechanically by connecting the piston to a crank mechanism. Alternately, the opposite end of the expansion piston can be operated as a compression piston which dissipates the expansion work either quasi-reversibly as work of compression, or irreversibly in the form of heat by causing fluid to pass through a throttle valve and heat exchanger.

The other significant expansion technique is exemplified by the basic <u>Taconis process</u> (Fig. 5-29). The system consists of a cylinder containing a movable displacer. Working fluid can be introduced or rejected from the system via inlet and exhaust valves which communicate directly with the ambient temperature end of the cylinder, and with the cold end through a regenerative heat exchanger. In position 1, the inlet valve is open, the exhaust valve is closed, and the displacer is at the cold end. The ambient space and the regenerator contain high pressure working fluid. From 1 to 2, the displacer is moved from the cold end to the ambient end and the cold space fills with high

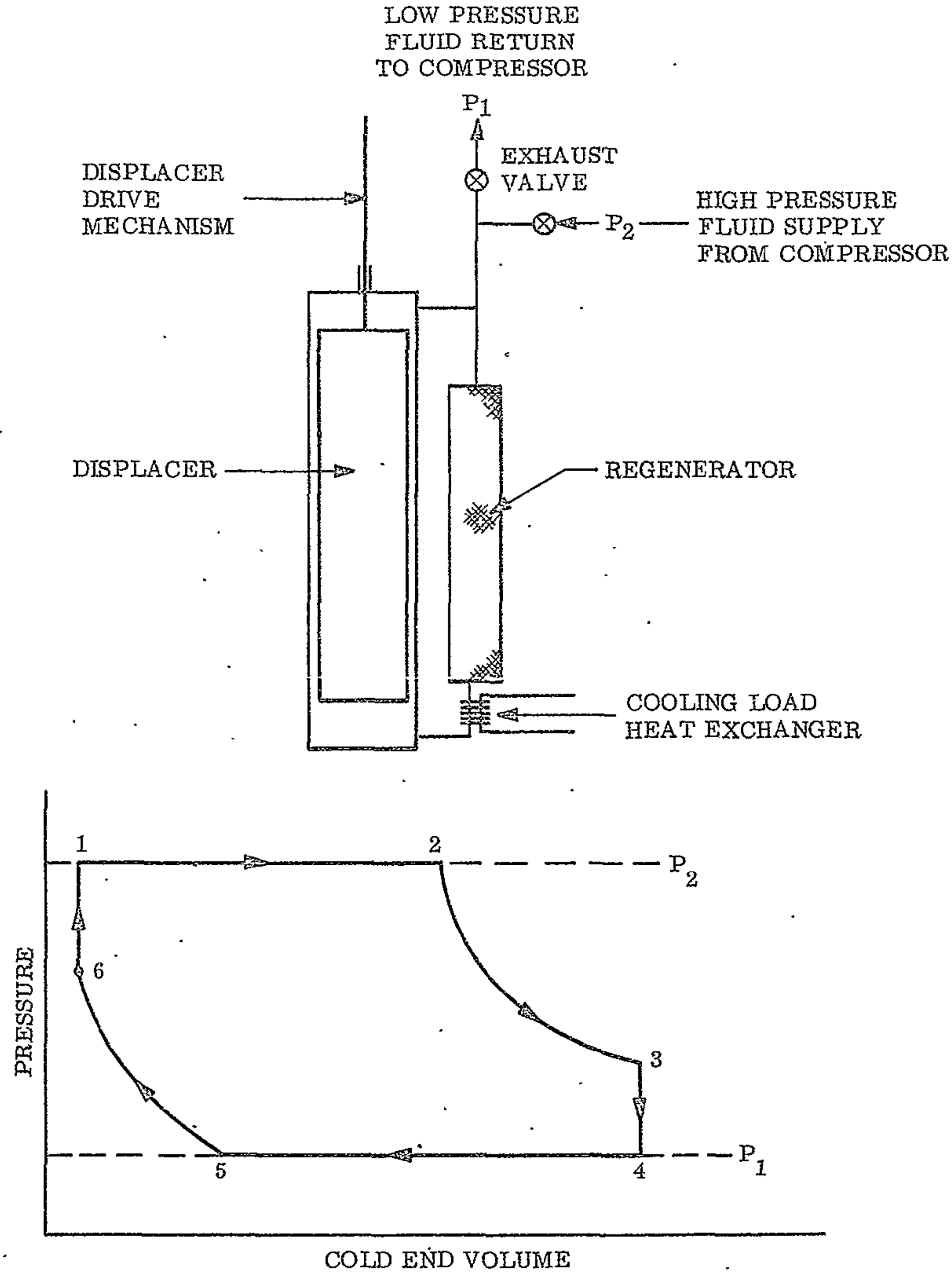


Fig. 5-29 The Taconis Expansion Process

pressure fluid. At point 2, the inlet valve closes and the displacer continues moving until it reaches the ambient end at 3. The pressure at 3 is lower than at 2 by virtue of the cooling which occurs when fluid is transferred from ambient to cold spaces. At 3, the exhaust valve is opened and the fluid expands irreversibly to 4. At point 4, the displacer is moved back towards the cold end, expelling low pressure fluid, until the exhaust valve is closed at 5. From 5 to 6, the fluid is compressed by displacement from cold to ambient spaces. At 6, the displacer is at the cold end and the inlet valve is opened, compressing the gas in the ambient space irreversibly to point 1. As in the case of the Solvay cycle the valve timing points 2 and 5 can be varied to maximize either cycle efficiency or unit performance.

In the Taconis cycle the work of expansion is extracted from the system by a somewhat devious route. When the inlet valve is opened the working fluid performs work as it flows into the expander to compress the fluid in the ambient space. When this fluid is displaced to the cold end, the heat of compression is deposited in the regenerator. During the exhaust phase this heat is picked up by the exhausting fluid and removed from the system.

The variations upon the Solvay and Taconis process usually involve valve timing, method of operation of displacer or piston and geometric configuration.

The thermal analysis of the Taconis process is very similar to that of the Stirling and VM refrigerator in that remarks relating to the complex and simplified analysis approaches outlined in Section 3 apply.

In the case of the Taconis expander the boundary conditions differ in that the system is open and the condition of constant mass is replaced by the valve flow rate equations and the specified inlet and exhaust fluid pressures. The work required by the overall cycle comprising compressor and exchanger-expander is equal to the work needed to compress the fluid consumed by the expander.

In the case of the Solvay process, the analyses is somewhat simpler because one is concerned with events in only one working space, but this space is nevertheless connected to a regenerator, whose behavior is time-dependent.

Though the complexity is diminished, the analysis of the Solvay process at very low temperatures should still preferably be made by the complex method. The boundary conditions for the complex method for the Solvay process analysis will be the motion of the working piston, as governed by the crank mechanism or the dynamics of the free piston-compressor arrangement, and the inlet and exhaust valve timings and fluid pressures.

5.4.2 Manufacturers of Systems

The following companies are engaged in the manufacture and development of systems which incorporate separate expanders and compressors in conjunction with regenerators.

Cryogenic Technology, Inc.
Cryomech Inc.
Air Products and Chemicals
British Oxygen Co., Ltd. .
The Welch Scientific Company

Cryogenic Technology, Inc: Cryogenic Technology, Inc., evolved from Arthur D. Little, Inc., originally. In 1967, an outgrowth of the activity of ADL was established as 500 Incorporated, a subsidiary of ADL, whose purpose was to produce and market low-temperature equipment. The company name was later changed to Cryogenic Technology, Inc., and established and operated as a separate company. CTI did the pioneering work in the development of units of this type. Their units have been used since 1959 to provide cooling for amplifiers. Several hundred units of this type have been placed in use and CTI has the largest backlog of experience in this area. CTI produces a standard line of approximately 15 units based on the Gifford-McMahan cycle, which provide cooling from approximately $10^{o}K$ to $150^{o}K$ at capacities of from 1 to 100 watts. As with the other manufacturers they utilize freon compressors which have been developed for aircraft service. Many of their units see service in aircraft, and these are lightweight, compact units. Extensive data on the system performance is available from CTI, primarily due to their long operating history and this data is readily available.

Cryomech, Inc: Cryomech, Inc. is a small company which was started in 1964 by Professor W. E. Gifford. Dr. Gifford is a well known, recognized authority in the field of cryogenic refrigeration and did much of the early development of the system which bears his name. Cryomech has a standard line of seven Gifford-McMahon refrigerators which provide cooling from 7.5°K (minimum temperature) to 200°K at rates from approximately ½ to 100 watts. The system uses a standard oil lubricated compressor (Tecumseh Model AJT15).

Air Products and Chemicals: Air Products who specialize in Joule-Thomson systems, has recently introduced a new cryogenic refrigerator system based on the modified Solvay cycle. Air Products manufacturers two units at present, and state that they will soon have a militarized version available using a dry lube compressor. Their two units provide cooling of 1 to 30 watts over a temperature range from 12°K to 200°K. They have recently introduced a third small-scale unit.

British Oxygen Co., Ltd.: British Oxygen manufacturers a Taconis cycle unit which has a capacity of 1.5 watts at 16°K. In general most of the company effort is concerned with large capacity systems at a temperature near 4°K.

The Welch Scientific Company: Welch Scientific Company makes a Gifford-McMahon cycle unit which uses compressed air as the working fluid. The unit is not applicable to requirements here since it is open cycle and does not provide low enough temperature using air as the working fluid.

5.4.3 Discussion of Data on Units

The parameters and operating data for cryogenic refrigerators based on the Gifford-McMahon, modified Solvay and modified Taconis cycles are tabulated in Table 5-4. The units selected for tabulation and analysis were limited primarily to those which provide cooling in the range of 5 to 100 watts. A substantial number of other smaller units are available.

Figures 5-30 through 5-34 present net refrigeration capacity vs. temperature for the individual units. Most of the units provide additional cooling on the first stage at approximately 77°K and this feature is particularly attractive

Table 5

EXISTING GIFFORD-McI
COOLING AT 20-11

Manufacturer	Cryomech, Inc.	Cryomech, Inc.	Cryomech, Inc.	Cryomech, Inc.	CTI
Trade Name	None	None	None	None	Cryodyne
Model	AL01	AL02	GB02	GB12	350
I.D. Number	30	31	32	33	34
Refrigeration Range	32 - 80°K	23 - 80°K	7.5 to 20°K	9 to 25°K	15 to 77°K
Cycle	G-M	G-M	G-M	G-M	G-M
Working Fluid	He	He	He	He	He
High Pressure	24 atm	24 atm	24 atm	24 atm	185 psi
Low Pressure	10 atm	10 atm	10 atm	10 atm	65 psi
Minimum Temp	23°K	23°K	7.5°K	9°K	15°K
Cool-Down Time	12 min	25 min	25 min	35 min	45 min
Expander RPM	144	144	144	144	72
Volts-Phase-Frequency	110/220 - 1 - 5/60	220 - 1 - 50/60	220 - 1 - 50/60	220 - 1 - 50/60	200/300 - 1 - 50/6(
Cooling Means	Air	Air	Air	Air	Air
Ambient Temp Req	NI	NI	NI	NI	-25°F to +125°F
Required Attitude	Cryostat any	Cryostat any	Cryostat any	Cryostat any	Cryostat any
Cryostat Dim. (in.)	2.5 x 2.5 x 14.5	5 x 5 x 18	5 x 5 x 21	5 x 5 x 24	19 x 5 x 9
Compressor Dim. (in.)		29 W x 19 x 27 H	29 x 19 x 27	29 x 19 x 27	28 x 17 x 16
System Volume (in.^{3})		15,350	15,420	15,500	8,460
Compressor Wt	125 lb	175 lb	175 lb	175 lb	175 lb
Cryostat Wt	5 lb	25 lb	25 lb	25 lb	22 lb
System Wt	130 lb	300 lb	200 lb	200 lb	229 lb(2)
MTBF	5000	5000	5000	5000	10,000
Maintenance Interval(1)	3000/5000	3000/1500	3000/1500	3000/1500	3000/3000/6000
System Cost	$8,610	$10,290	$13,200	$13,200	$13,000
20°K Refrigeration			5.5 W	8 W	3W (2nd Stage)
20°K Power Input			3000 W	3000 W	2100 W
20°K COP			.00184	.00267	.00143
20°K % Carnot			2.57%	3.7%	2%
20°K Lb/Watt			36.4	25	76
20°K In.^{3}/Watt			2800	1940	2820
77°K Refrigeration	18 W	75 W			5 W (1st Stage)
77°K Power Input	900 W	3000 W			2100 W
77°K COP	.020	.025			
77°K % Carnot	5.8%	7.2%			
77°K Lb/Watt	7.22	2.67			
77°K In.^{3}/Watt					
100°K Refrigeration	29	89 W			
100°K Power Input	100	3000 W			
100°K COP	.0322	.0297			
100°K % Carnot	5.08%	4.7%			
100°K Lb/Watt	4.5	2.2			
100°K In.^{3}/Watt		172			

(1) Cryostat/Compressor oil filter/compressor
(2) Total weight excludes instrument panel

FOLDOUT FRAME 1

: 5-4

cMAHON REFRIGERATORS
$10^{o}K$ AT 5-100 WATTS

	CTI	CTI	CTI	Air Products		
	Cryodyne	Cryodyne	Cryodyne	Display		
	355	1020	10077	CS-102		
	35	36	38	38		
	15 to 77°K	13 to 77°K	30 to 77°K	30 to 200°K		
	G-M	G-M	G-M	Mod. Solray		
	He	He	He	He		
	275 psi	275 psi	300 psir	320 psig		
	75 psi	75 psi	100 psi	115 psig		
	15°K	13°K	25°K	30°K		
	[illegible]min	50 min	30 min	20 min		
	[illegible]	82	82	144		
'60	208/440 - 3 - 50/60	208/400 - 3 - 40/60	208/440 - 3 - 50/60	208/440 - 3 - 60		
	Air	Air	Air	Air		
	-25°F to 125°F	-25°F to 125°F	-25°F to +125°F	40 - 110°F		
	Styostat any	Cryostat any	Cryostat any	Cryostat any		
	18 x 10 x 6	20 x 13 x 8	16.5 x 13 x 8	4 D x 19 L		
	41 x 27 x 26	41 x 27 x 26	41 x 27 x 26	22 x 17 x 15		
	29,900	30,880	30,500	5,600		
	425 lb	425 lb	425 lb	150 lb		
	22 lb	33 lb	30 lb	10.6 lb		
	468lb(2)	488 lb(2)	480 lb(2)	161 lb		
	10,000	13,000	14,000	3000 - 5000 est		
	3000/3000/6000	3000/3000/6000	3000/3000/6000	3000/6000		
	$16,000	$17,000	$18,000	$7,000		
	5 W (2nd Stage)	11 W (2nd Stage)				
	5600 W	5600 W				
	.00089	.00197				
	1.2%	2.7%				
	93	44				
	5980	2810				
	5 W (1st Stage)	10 W (1st Stage)	100 W	17 W		
	5600 W	5600 W	5600 W	1700 W		
			.0179	.01		
			5.2%	2.9%		
			4.8	9.5		
			205	330		
				22 W		
				1700 W		
				.013		
				2%		
				7.3		
				255		

FOLDOUT FRAME 2

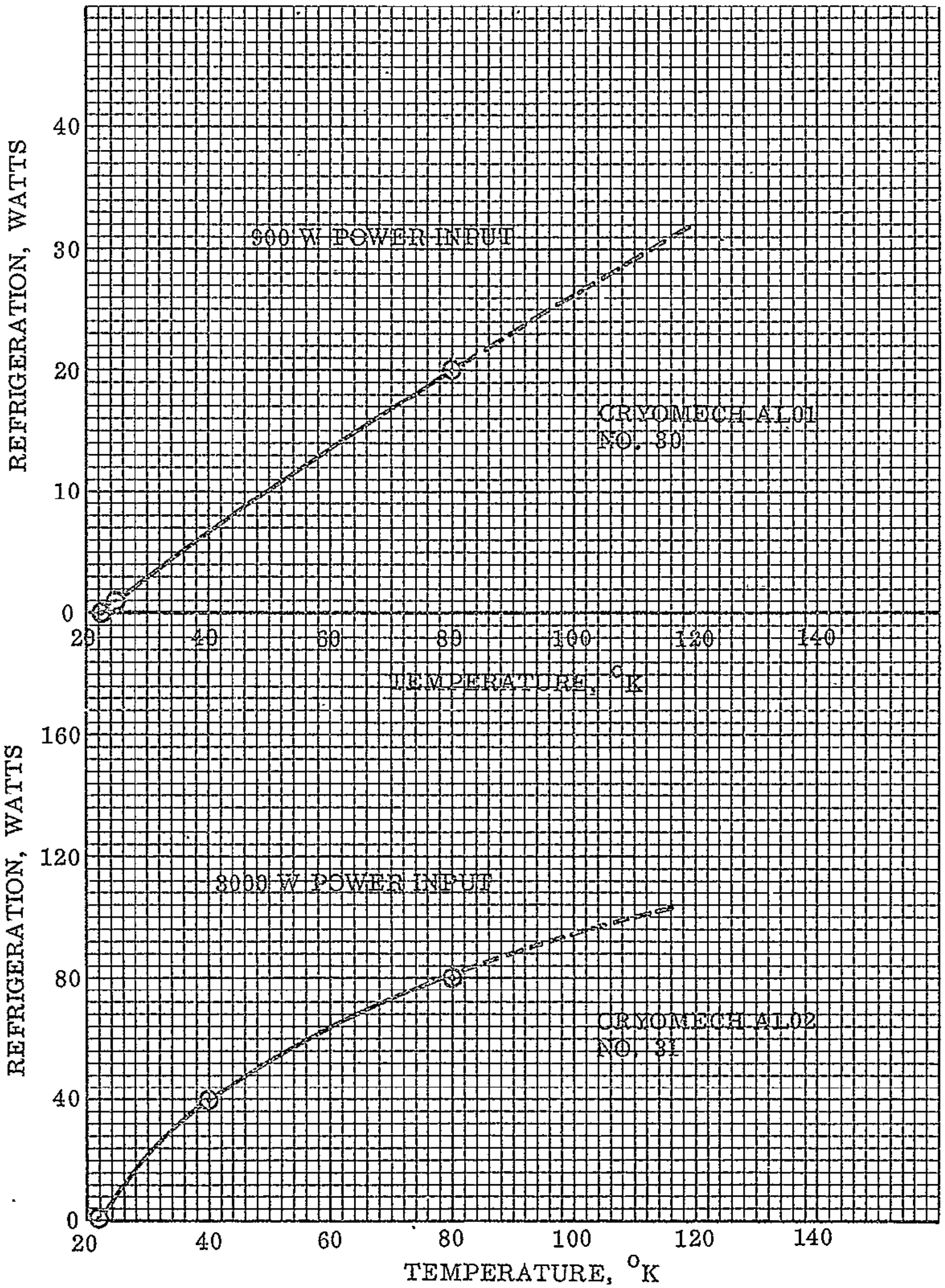
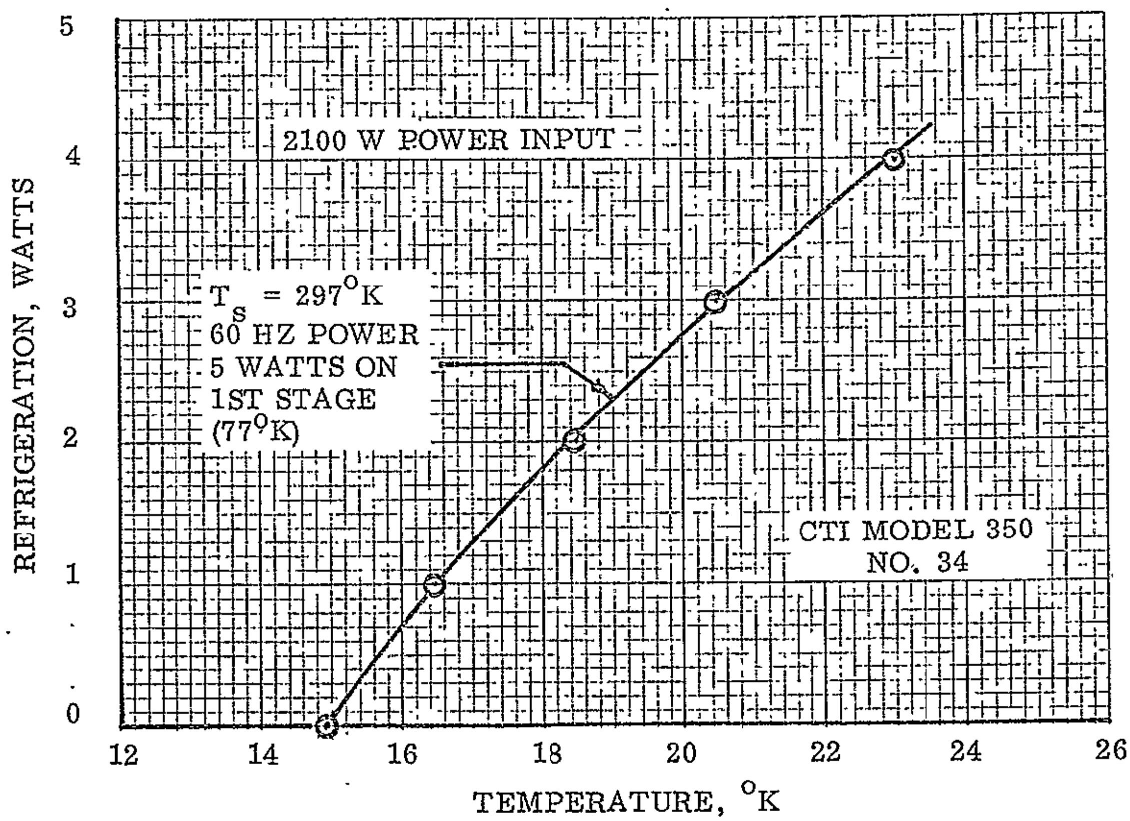


Fig. 5-30 Refrigeration vs. Temperature (Gifford-McMahon Cycle)



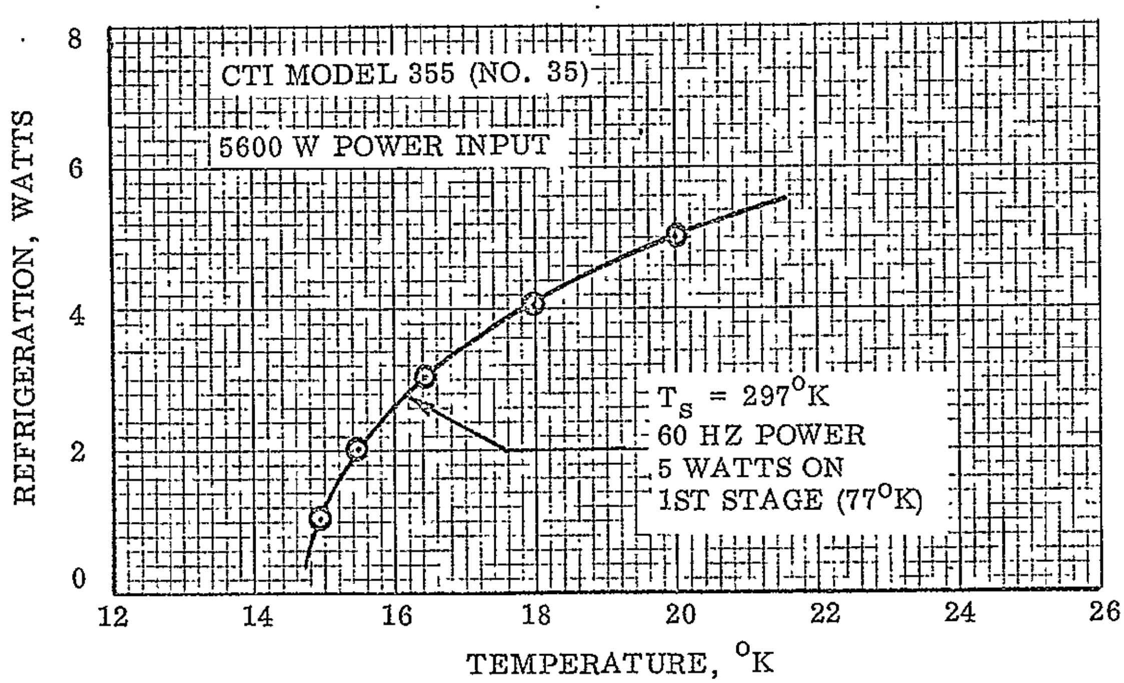
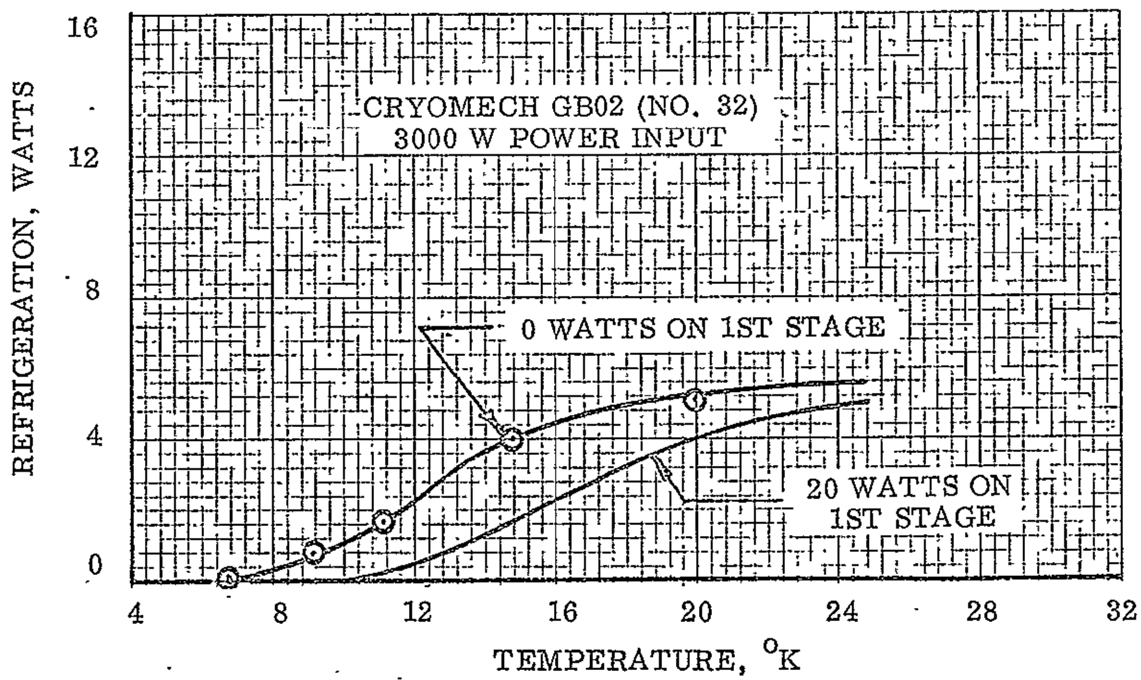


Fig. 5-31 Refrigeration vs. Temperature (Gifford-McMahon Units)



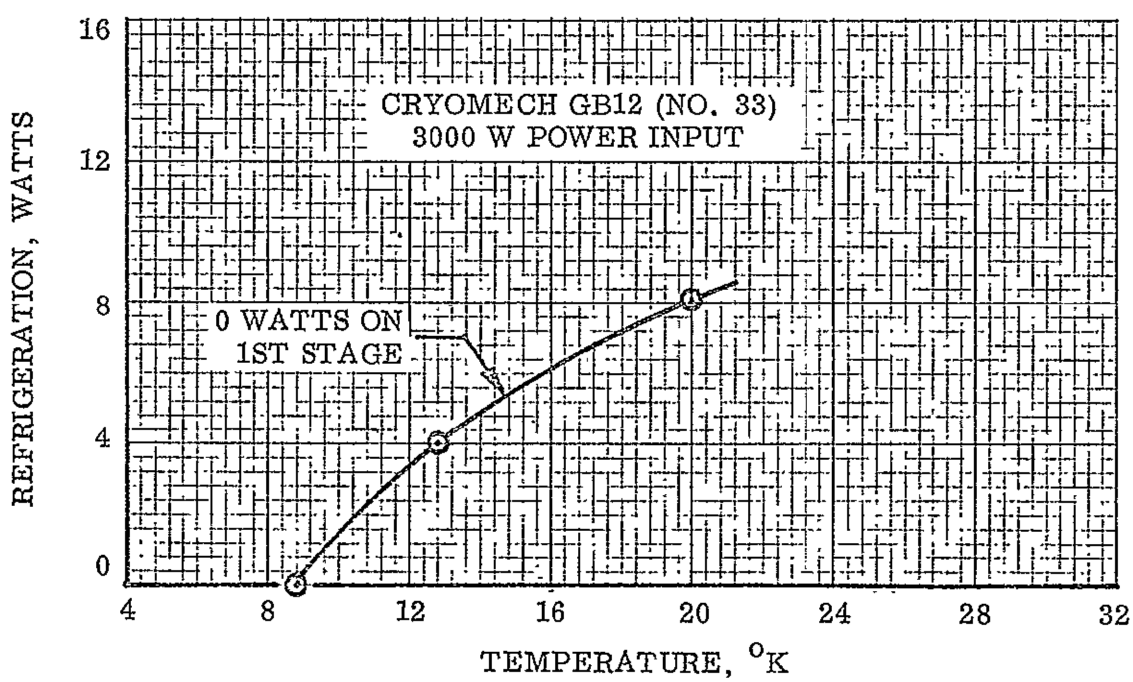
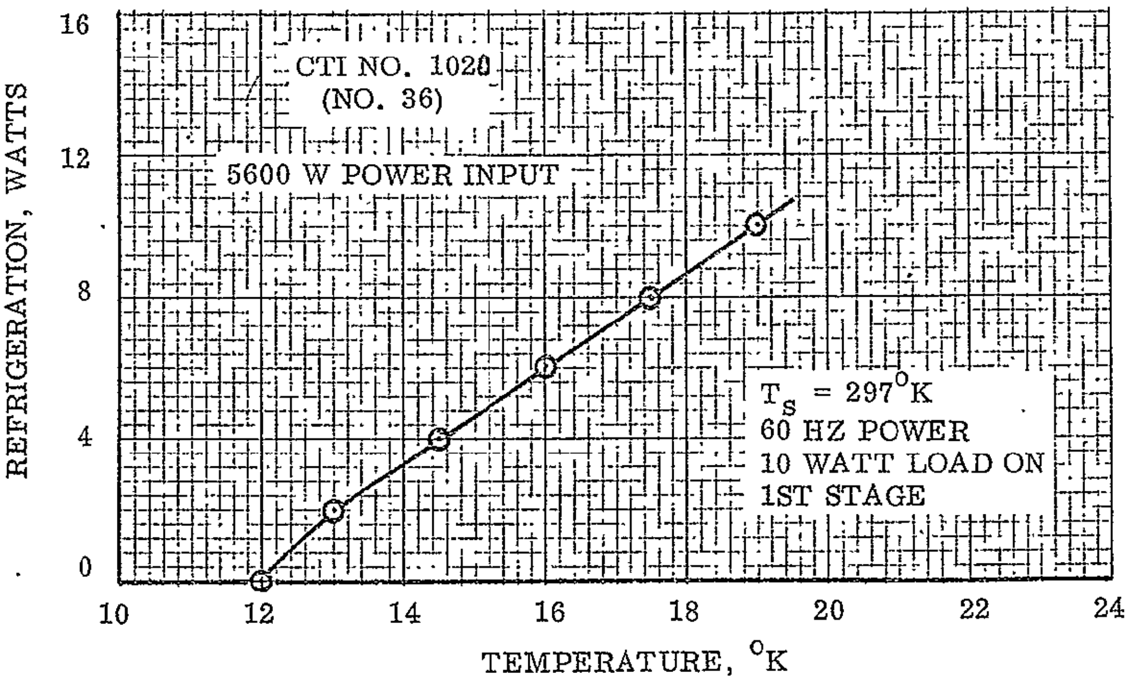


Fig. 5-32 Refrigeration vs. Temperature (Gifford McMahon Units)



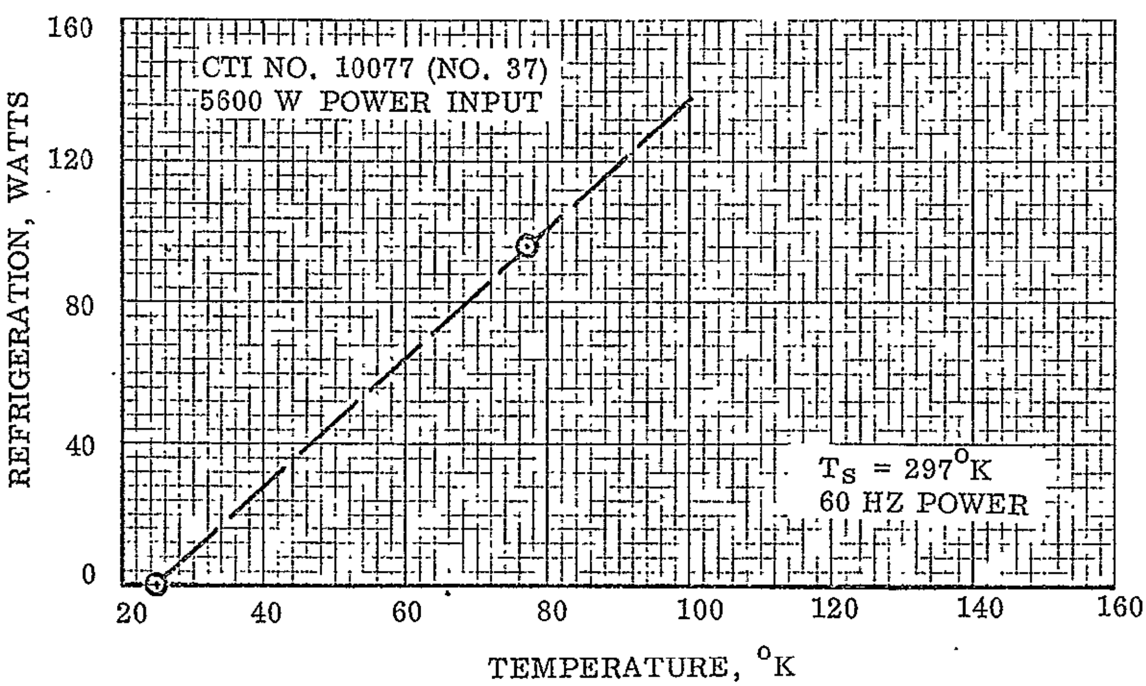
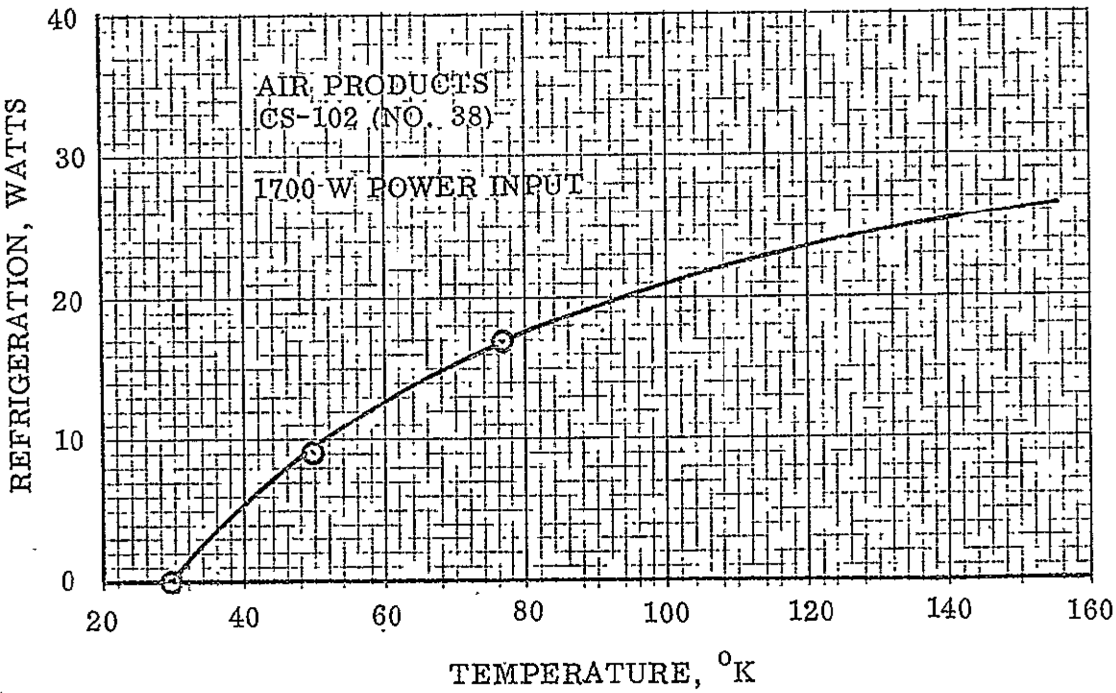


Fig. 5-33 Refrigeration vs. Temperature (Gifford-McMahon Units)



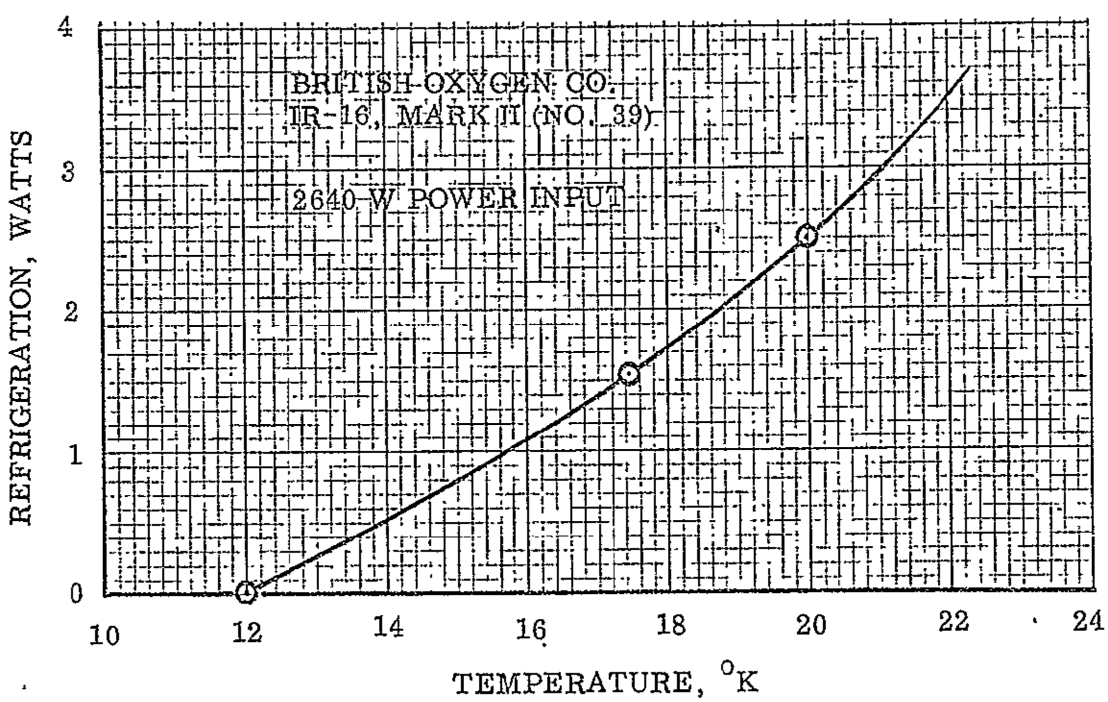


Fig. 5-34 Refrigeration vs. Temperature (Gifford-McMahon Units)

for application to vehicle systems which utilize fuel and oxidizer that require cooling at two temperatures. Data on first stage refrigeration are presented where available. In most cases the first stage cooling exceeds the second stage (lower temperature) cooling. Most manufacturers do not include data on the coupling between first and second stage cooling rates. Second stage cooling rates are reduced as heat loads are introduced on the first stage. This effect is shown in Fig. 5-32 which presents data for Cryomech GB02.

The coefficient of performance of the various machines are presented in Figs. 5-35 and 5-36 vs. refrigeration and temperature. The data are curve fit using the technique as described for the Stirling cycle. The highest values for C.O.P. are shown by the Cryomech Units. The minimum temperature achieved by the various units is approximately $7^{o}K$. The machines which provide the data on C.O.P. represent a substantial amount of experience, and it is not expected that large improvements in performance will be forthcoming in the near future, although some improvements will almost certainly evolve.

Figures 5-37 and 5-38 present the system weight per refrigeration (specific weight) vs. temperature and cooling capacity. The data clearly shows the effect of both cooling capacity and temperature level. As in the other cycles the curve fits were made through the "best" points (i.e., highest C.O.P. and minimum specific weight).

Unlike the C.O.P. data it is expected that substantial weight reduction could be made in the systems by selecting compressor units which are optimized for minimum weight. Section 4.1 discusses the various compressors available and indicates the relative weight gains which appear obtainable in the compressor unit. As previously discussed in Section 4, the compressor is a large fraction of the total system weight.

Figure 5-39 presents the specific volume of the systems (In^3/Watt) and the curve fits at the three temperatures. It is expected that large reductions in the specific volume of the units would be achieved by optimization of the compressor for space use.

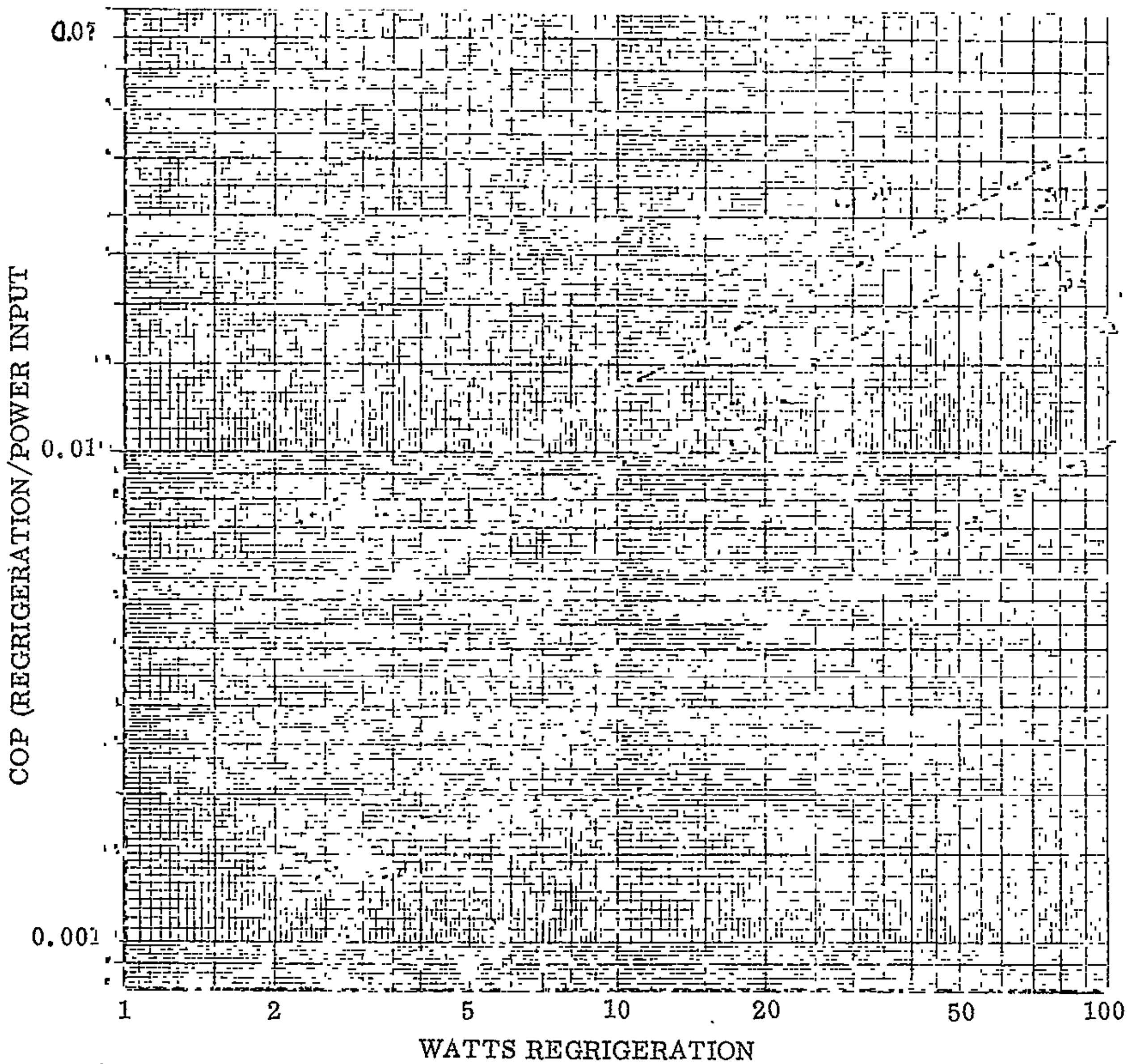


Fig. 5-35 C.O.P. Versus Refrigeration Capacity For Gifford-McMahon Units

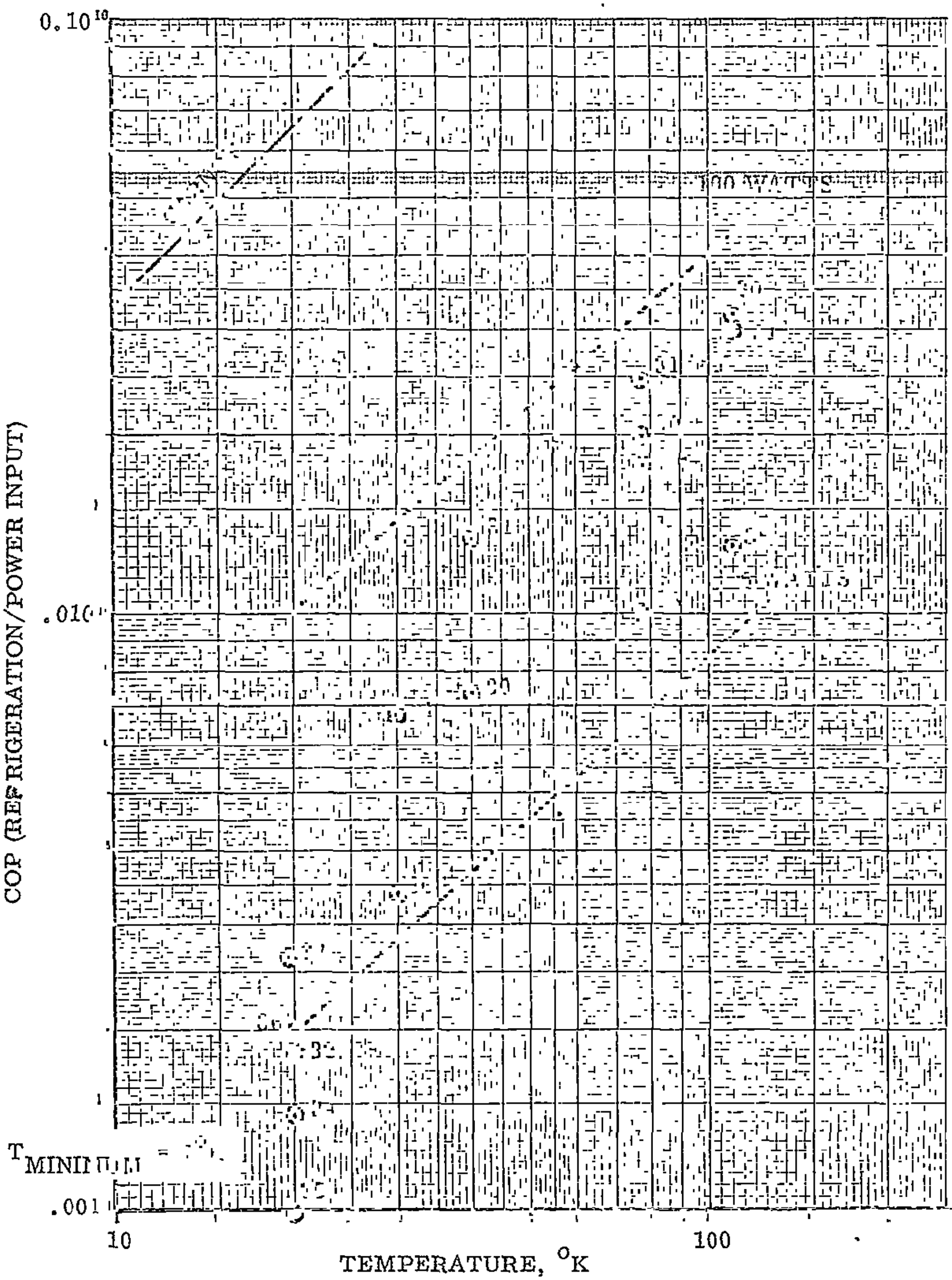


Fig. 5-36 Coefficient of Performance of Gifford-McMahon, Taconis, and Solvay Refrigerators

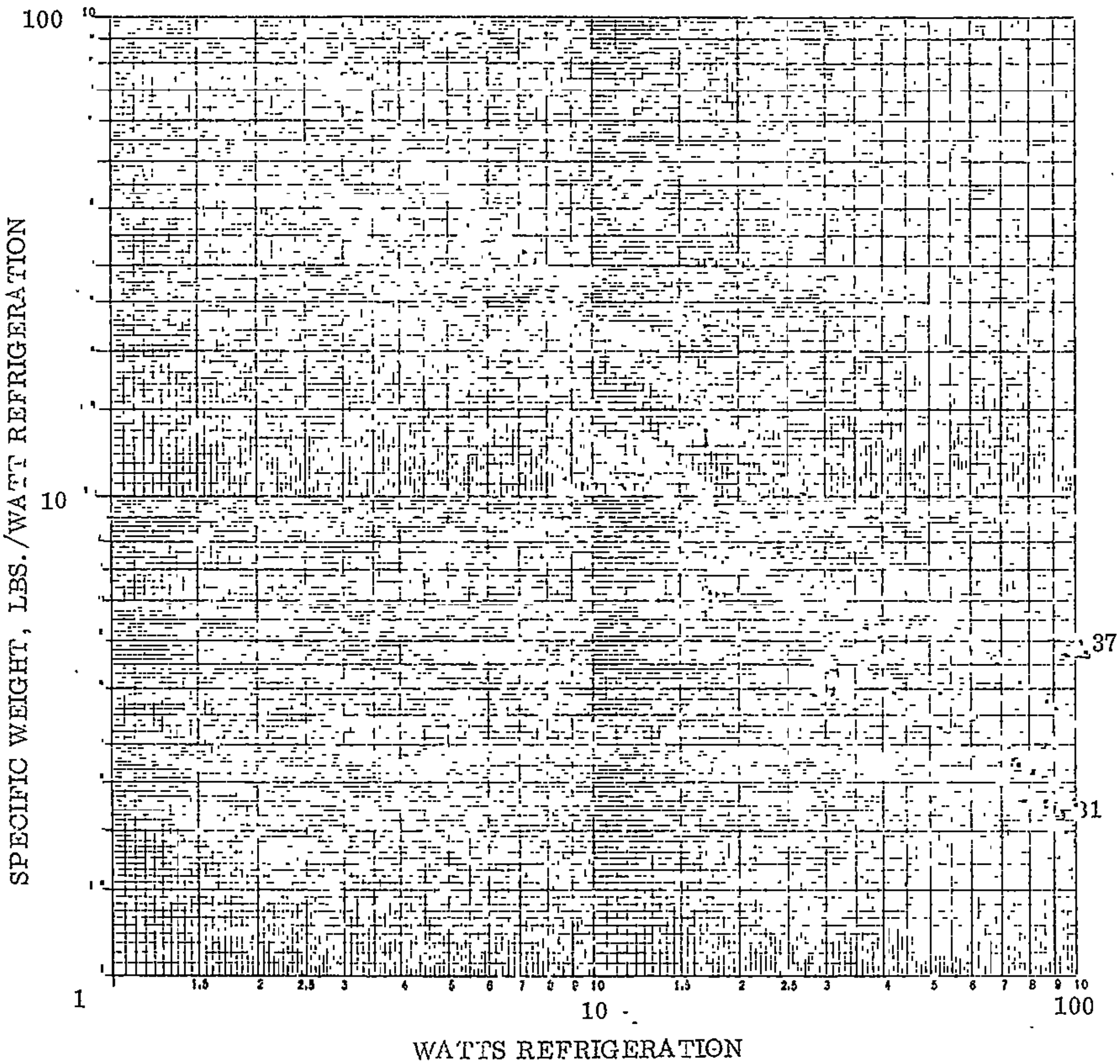


Fig. 5-37 Specific Weight Vs Watts for Gifford-McMahon Units

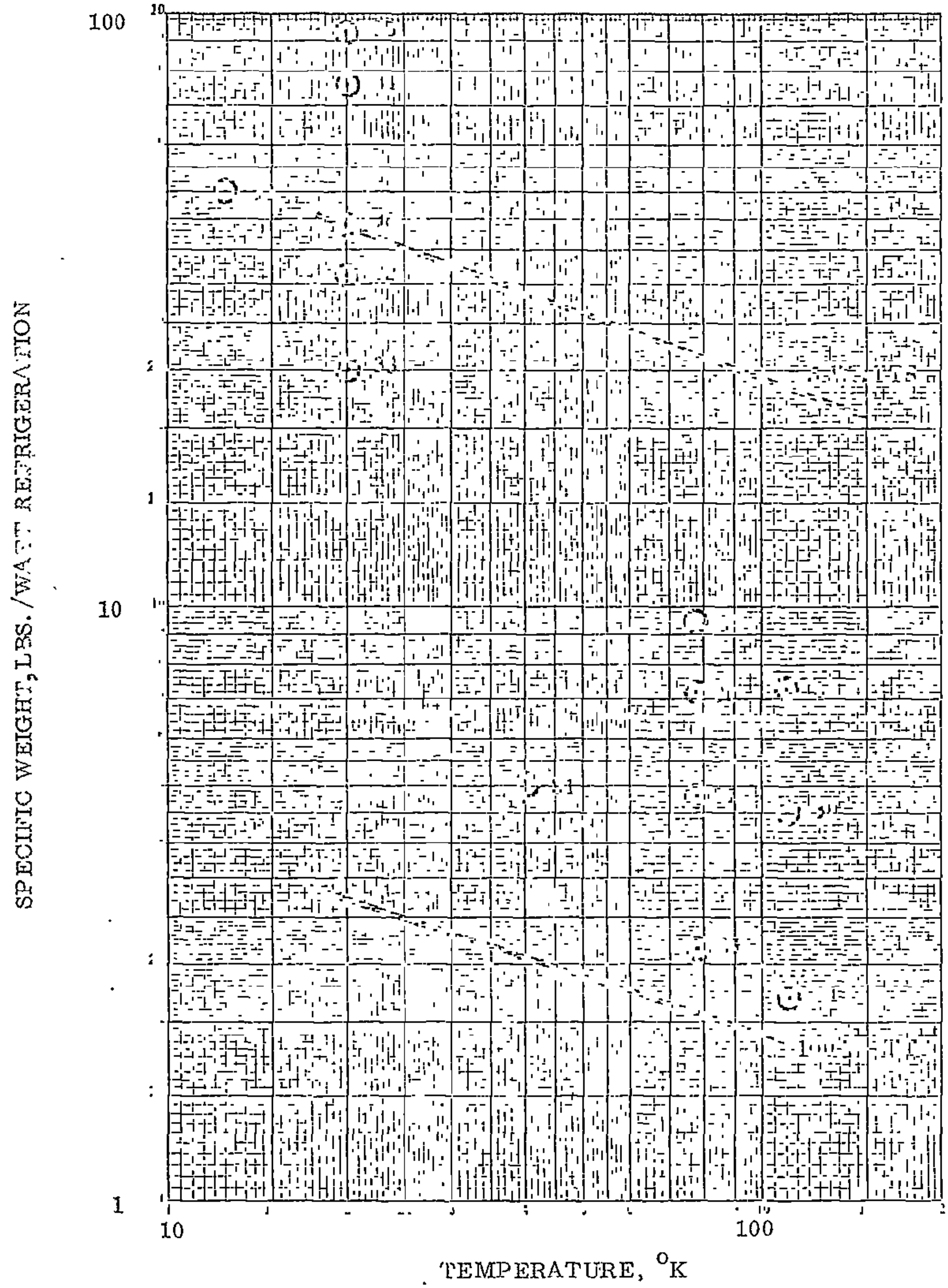


Fig. 5-38 Specific Weight Versus Temperature of Gifford-McMahon, Solvay, and Taconis Cycle Refrigerators

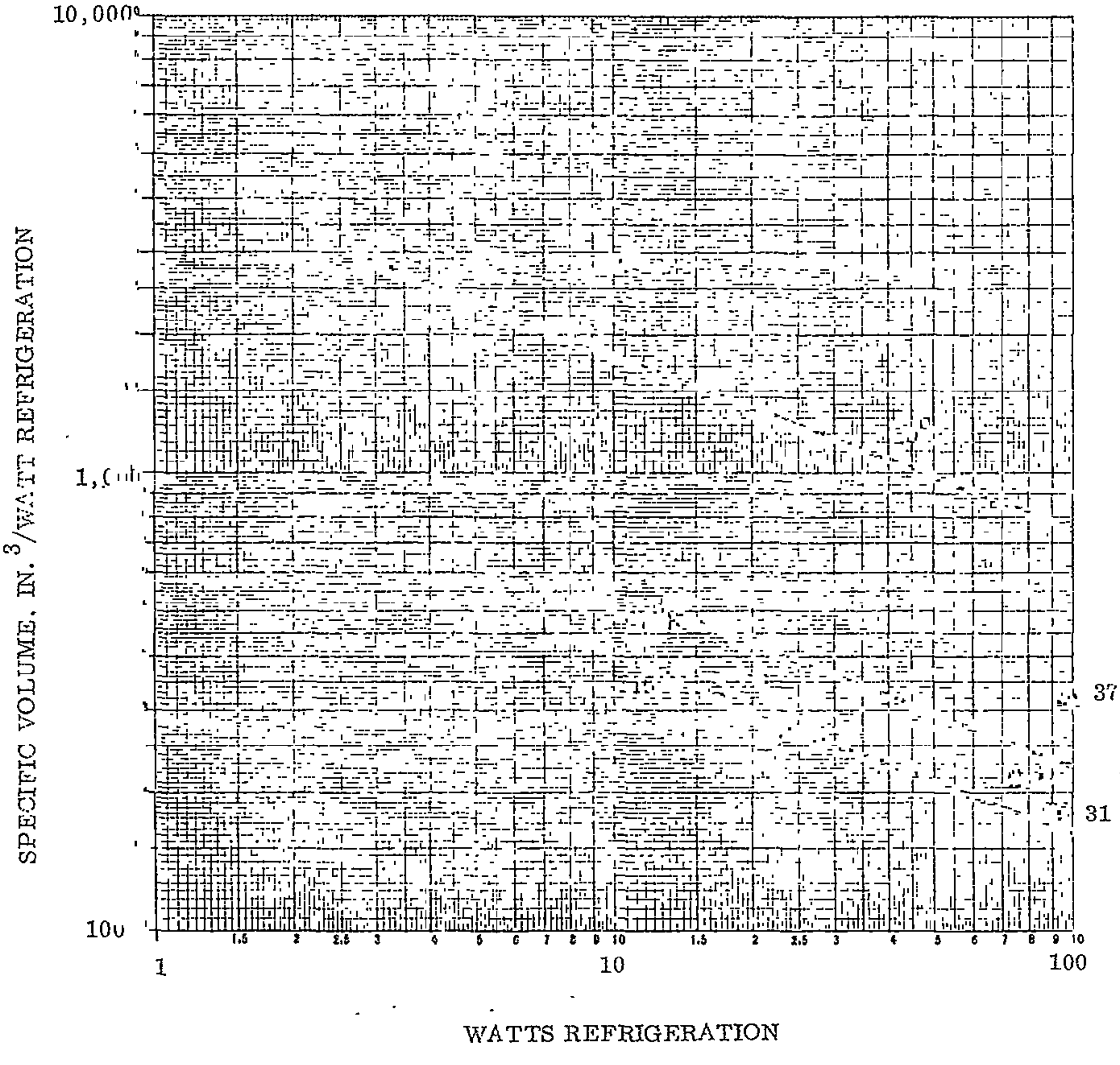


Fig. 5-39 Specific Volume Versus Capacity of Gifford-McMahon Systems

A further comment which should be noted is that none of the units would be capable of space usage since their oil separation systems would not operate in zero gravity conditions.

The percent Carnot efficiency is presented in Figure 5-40 and shows that the units provide from 1 to 7% Carnot efficiency, which is substantially below that achieved with the most efficient cycle consideredhere, the Stirling cycle, which provides from 6 to 20% in the same operating range.

In spite of this shortcoming the G-M and similar cycles provide a high degree of flexibility due to the separable components and more importantly <u>currently provide the longest unattended lifetime</u> for the refrigerators which fall within this study.

5.5 Brayton/Claude Cycles

5.5.1 Operation

<u>Brayton Refrigerator</u>: A practical Brayton cycle refrigerator is shown in Figure 5-41. Gas is compressed with some increase in entropy from 1 to 2. The heat of compression is rejected to the ambient temperature heat sink in an after-cooler from 2 to 3. The high pressure fluid is cooled from 3 to 4 in the main heat exchanger. The pressure at 4 is slightly less than at 2 due to the flow losses in the two heat exchangers. The fluid is expanded from 4 to 5 with some entropy increase, and is then warmed to 6 by passage through the load heat exchanger. The fluid is warmed from 6 to 1 in the main heat exchanger as it returns to the inlet side of the compressor. The pressure at 4 is slightly higher than at 1 because of pressure losses in the heat exchangers.

Analysis of the cycle is performed by selecting high and low fluid pressures, load and sink temperatures, and either choosing mass flow rates and component dimensions from which efficiencies can be determined (as described in Section 4), or assuming efficiencies from which required component dimensions may be found in a separate calculation (as described in Section 3). The analysis begins by assuming a value for T, and hence,

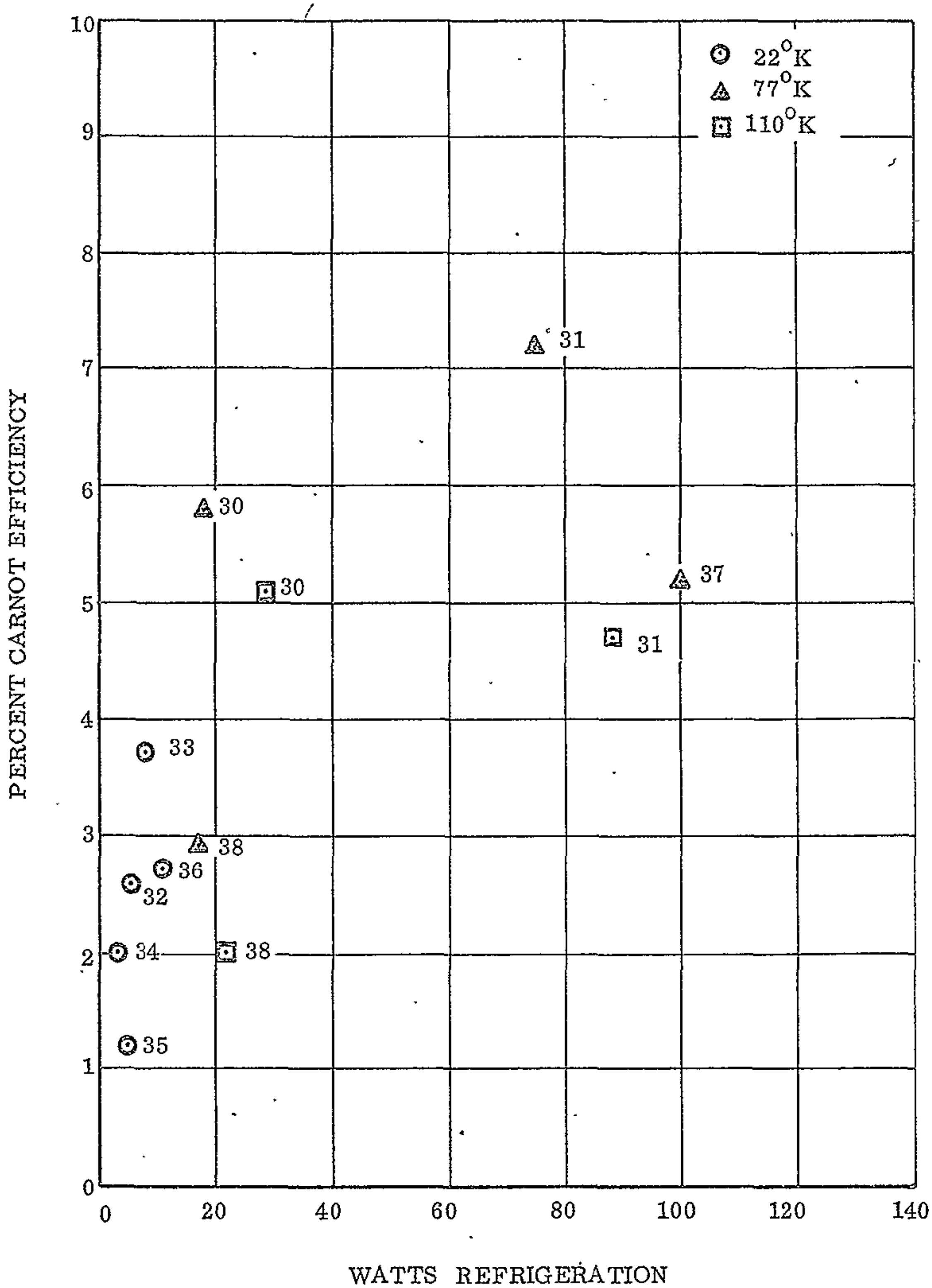


Fig. 5-40 Percent Carnot Efficiency of Gifford McMahon Systems

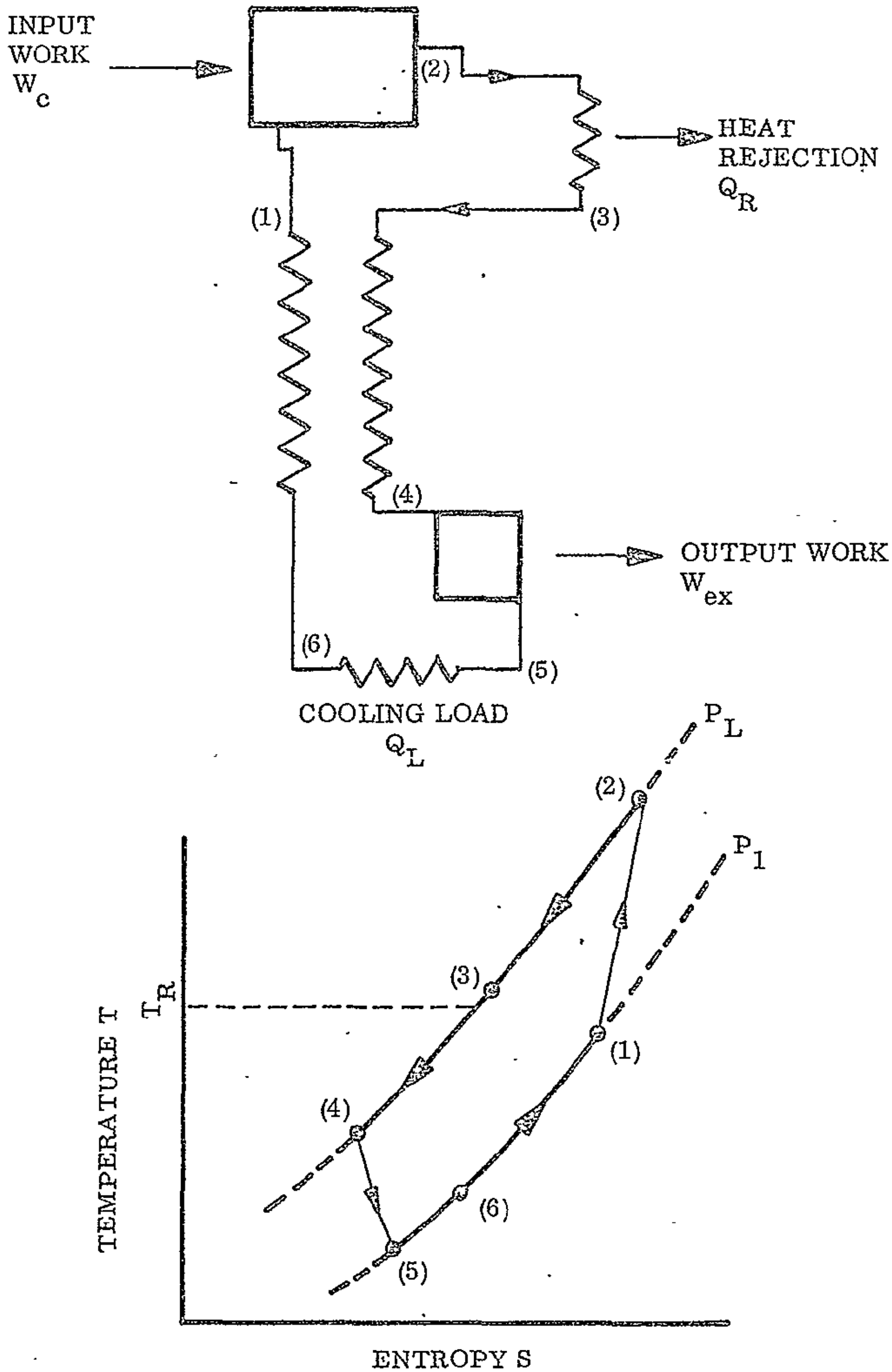


Fig. 5-41 The Reverse Brayton Cycle

h_2 is found from the assumed or calculated compressor isentropic efficiency,

$$\eta_{isc} = \frac{h\ (P_2,S_1) - h_1}{h_2 - h_1}$$

T_2 is found from h_2 and P_c.

$$T_2 = T\ (P_2,h_2)$$

P_3 is found from the assumed or calculated after-cooler loss coefficient K_a

$$P_3 = P_2 - \frac{P_2 + P_3}{2} K_a$$

h_3 is found from the assumed or calculated after cooler effectiveness ε_a, T_2, and the sink temperature T_a

$$\varepsilon_a = \frac{h_2 - h_3}{h_2 - h\ (P_3,T_a)}$$

T_3 is found from the fluid equation of state

$$T_3 = T\ (P_3,\ H_3)$$

h_4 is found from the assumed or calculated main heat exchanger effectiveness, ε_m

$$\varepsilon_m = \frac{h_3 - h_4}{h_3 - h_6}$$

P_4 is found from the assumed or calculated main heat exchanger high pressure side pressure loss coefficient, K_{mh}

$$P_4 = P_3 - \frac{(P_3 + P_4)}{2} K_{mh}$$

T_4 is found from the fluid equation of state

$$T_4 = T(H_4, P_4)$$

P_o is found from the assumed or calculated load heat exchanger pressure loss coefficient, K_L

$$P_o = P_s - \left[\frac{P_o + P_s}{2}\right] K_L$$

h_5 is found from the assumed or calculated expander isentropic efficiency, η_{isc}.

$$\eta_{isc} = \frac{h_4 - h_5}{h_4 - h(P_5, S_4)}$$

T_5 is found from the fluid equation of state

$$T_5 = T(P_5, h_s)$$

h_6 is found from the assumed or calculated load heat exchanger efficiency, ε_e

$$\varepsilon_e = \frac{h_6 - h_5}{h(P_o, T_e) - h_5}$$

T_6 is found from the fluid equation of state

$$T_6 = T(P_o, h_6)$$

P_1 is found from the assumed or calculated main heat exchanger low pressure side loss coefficient, K_{me}

$$P_1 = P_6 - \left[\frac{P_1 + P_6}{2}\right] K_{me}$$

h_1 is found from the assumed or calculated main heat exchanger effectiveness ε_m.

$$\varepsilon_m = \frac{h_1 - h_6}{h_3 - h_6}$$

T_1 is found from the fluid equation of state

$$T_1 = T\ (P_1,\ h_1)$$

The calculated values of P_1 and T_1 will not, in general, agree with the assumed values. Adjustments are made in the assumed expander pressure ratio and the cycle is recalculated using the new T_1. The process is repeated until a consistent set of figures is obtained. If component efficiencies rather than dimensions were assumed then the component sizes and flow rates required to provide this performance must then be determined.

The cooling capacity of the refrigerator, q, is the heat absorbed by the load heat exchanger.

$$q_c = \dot{m}\ [h_o - h_s]$$

The power required by the refrigerator, W, is the work of compression

$$W = \dot{m}\ [h_2 - h_1]$$

It is apparent that the analysis of continuous flow Brayton Cycle refrigerators is relatively straightforward. Performance data can be prepared quite readily as a function of component efficiencies and the results of two extensive parametric studies are reported in the literature [(21)(22)]. Muhlenhaupt and Strobridge present calculated values of W/q_c for a wide range of expander and exchanger efficiencies and helium, hydrogen and nitrogen working fluids. A total of 66 charts are presented for W/q_c as a function of P_2, each presented for a range of three other cycle parameters. Wilson and D'Arbeloff [(23)] present a similar range of calculated performance data with component efficiencies as parameters. These data are calculated for helium, hydrogen, and neon as working fluids. The effects of multiple stage compression, multiple stage expansions and the use of intermediate temperature expansion stages to improve apparent effectiveness are shown.

The dimensions and performance characteristics of a practical Brayton cycle refrigerators may thus be determined by combining the parametric system performance data of these two studies with the physical dimensions, capacity, and efficiency of practical compressors, heat exchangers and expanders.

<u>The Claude Refrigeration Cycle</u>: As the operating temperature of the Brayton refrigerator is lowered point 5 (Fig. 5-41) will enter the two-phase region of the working fluid, and the fluid will leave the expander as a two-phase mixture. Up to the present time, it has not been considered good engineering practice to permit expanders to operate in the two phase region because of possible mechanical damage to the expander. For refrigeration at temperatures within the two-phase region of the working fluid it has therefore become accepted practice to perform the expansion process isenthalpically through a throttle valve as in the Joule-Thomson cycle, rather than in an expansion engine.

As explained in Section 5.3, this process will not produce net refrigeration unless the value of $(\partial h/\partial P)_T$ is negative at the effective sink temperature. For helium, hydrogen and neon this means that the effective sink temperature must be reduced by use of an auxiliary heat exchanger. The Claude cycle is effectively a Joule-Thomson cycle in which the effective sink temperature is lowered by a Brayton cycle refrigerator. It is designed so that the two systems share the same working fluids.

Figure 5-42 shows a practical Claude cycle. The cycle closely resembles the Brayton and Joule-Thomson cycles, Figures 5-41 and 5-21, and the cycle description is similar. The difference is that upon reaching point 4 the flow divides and a portion of the flow is expanded as in a Brayton refrigerator and is returned to the compressor via a combined load and precooling exchanger, and the main exchangers. The remaining portion of the stream at 4 is passed through the other side of the load/precooling refrigerator and is cooled

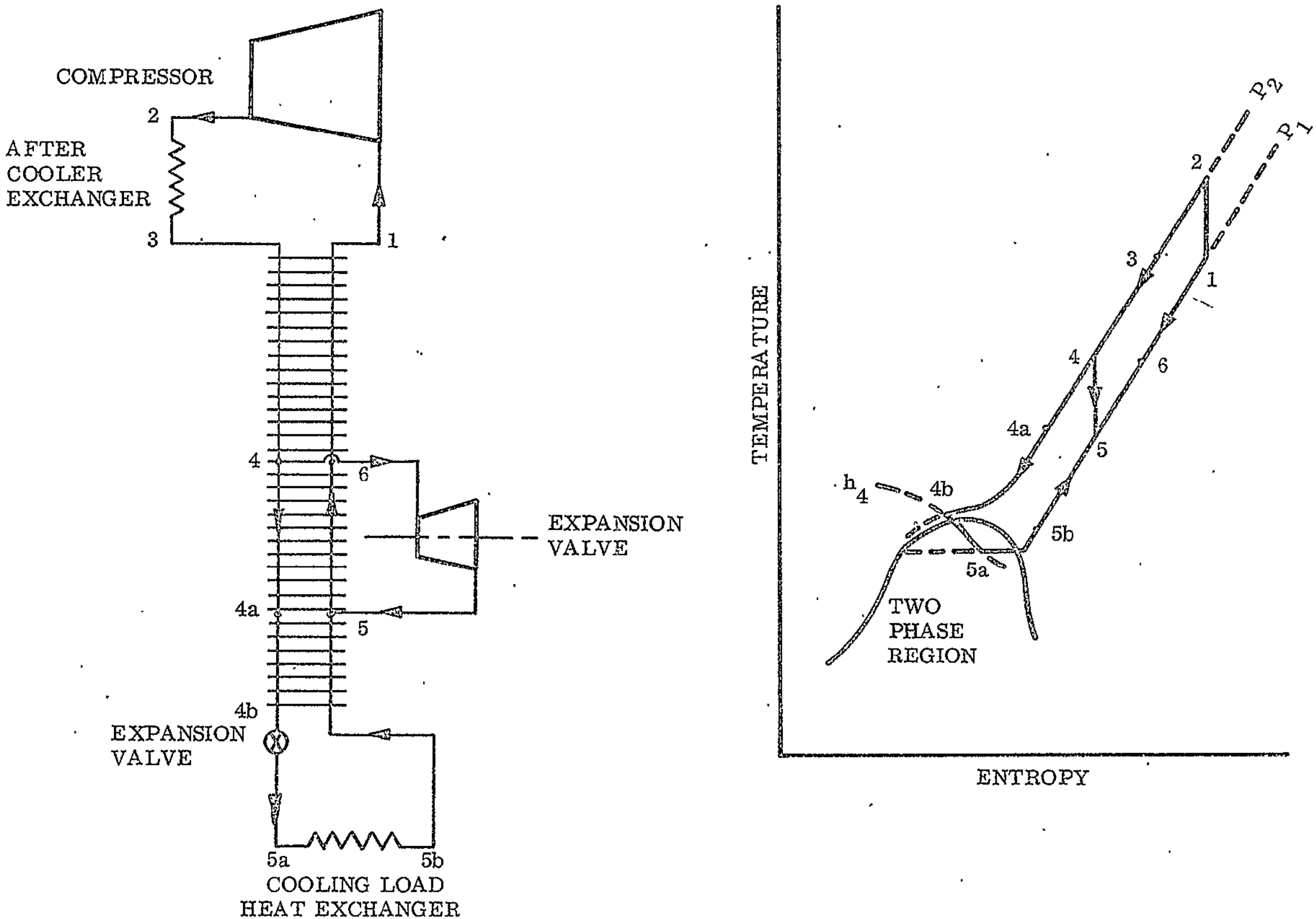


Fig. 5-42 The Claude Refrigeration Cycle

to T_{4a}. The system $4-4_a-4_b=5_a-5_b-5-6$ is a standard Joule-Thomson refrigerator and functions like the system 3-4-5-6-1 in Figure 5-21. The fluid is cooled in the main Joule-Thomson heat exchanger, 4a to 4b, after which it is expanded isenthalpically to 5a. From 5a to 5b, the fluid is warmed in the Joule-Thomson load heat exchanger and then is reheated in the main exchanger to point 5 where the two flows are united and pass back to the compressor via the load/precooling exchanger and main exchanger.

It can be seen that the Claude cycle is essentially more complex and less efficient than the Brayton cycle inasmuch as a Joule-Thomson refrigerator has been added and that because of this the cooling effect at the load is produced by isenthalpic expansion, which produces a greater increase in entropy than even the most inefficient expander. These implications are considered desirable if a two-phase mixture in the expander is to be avoided. Recently a reciprocating expander has been operated successfully in the two-phase region[24]. If it proves possible to develop expander technology to the point where such expanders become generally available, the technical advantage of the Claude cycle would be eliminated.

The Claude cycle can be analyzed in a manner analogous to the Brayton cycle and Joule-Thomson cycle analyses. For the temperature range of interest to this program, $20^oK - 100^oK$, the possible working fluids are helium, neon, hydrogen and nitrogen. Because of the low condensation temperature of helium, there is no necessity for adding a Joule-Thomson stage to a helium Brayton refrigerator. Neon or hydrogen Claude systems would be appropriate for the $20^oK - 30^oK$ range. A Claude system using nitrogen could be used in the 75^oK to 85^oK range, where its efficiency would be higher than that of a single stage Joule-Thomson system.

An extensive parametric study of a helium Claude cycle refrigerator for use in the temperature range of 42^oK is presented by Muhlenhaupt and Strobridge[22]. The temperature range of this refrigerator is below the range of current interest and although the technique of analysis would be applicable to hydrogen, neon or nitrogen system, the data are of no direct interest. However, the data do show that the efficiency passes through a maximum with increasing magnitude at high pressure, and that the optimum pressure is higher than that found for Brayton cycle systems [22].

5.5.2 Companies Engaged in Brayton/Claude Cycle Refrigerator Development

The following companies have been engaged in research and development activities associated with a miniature Brayton cycle refrigerator:

- General Electric Company, Schenectady, New York
- AiResearch Manufacturing Company (A Division of the Garrett Corporation) Los Angeles, California
- Arthur D. Little, Inc., Cambridge, Massachusetts
- Hymatic Engineering Co., Ltd.

In addition to these companies, whose activities have been directed toward miniature units for aircraft and space usage, the following companies have built large capacity industrial units, using the Claude and Brayton cycle:

- National Bureau of Standards
- L'Air Liquide
- British Oxygen Cryoproducts
- CVI Corporation
- Sulzer Brothers Limited

Garrett AiResearch: Garrett AiResearch began activities on the design and development of a miniature non-reciprocating closed-cycle cryogenic cooler in 1962 under contract to WPAFB (AF 04(695)-313), and AF 04(695)-146. The objective of this contract was to develop a refrigerator with a cooling capacity of approximately 2 watts at 77°K which would be suitable for use with space-based infrared sensor devices. The work conducted on this system resulted in the fabrication of a non-reciprocating system based on the Brayton cycle and using nitrogen as the working fluid. The system parameters for this unit are shown in Table 5-5 (No. 40).

Long term unattended operation, one of the goals of this effort was not demonstrated, the actual testing of the unit being quite limited in duration. The work on this unit terminated in March 1967. A second development contract carried on at Garrett was begun in February 1964 and ended in December 1965. This effort was funded by U. S. Army Satellite Communications Agency under contract DA-36-039-AMC-30725. The program objective was the development of a helium refrigerator system suitable for cooling low-noise amplifiers at 4.2°K. The work accomplished under this contract consisted of component feasibility studies. A working model was not built under this contract. (Reference 25)

A third effort by Garrett was funded by WPAFB during the period April 1964 to October 1968. The objective was to develop a turbomachinery-type closed-cycle refrigeration system to provide 1W of cooling at 3.6°K, for a continuous operating period of 10,000 hr minimum in a space component environment. Fabrication and testing was conducted during this contract; however, a complete prototype unit was not fabricated. (Reference 24). The work was done under Contract AF33(615)-1015.

Arthur D. Little, Inc: Arthur D. Little, Inc., began their activities on Brayton cycle systems in mid-1962. Their activities, like Garrett's, have been funded by WPAFB and the objectives of their initial work was the development of a refrigerator to provide 2 watts of cooling at 77°K in a space environment. Work on this area was completed in May 1967 and resulted in the fabrication and testing of a development model. In addition to this a lightweight model was constructed but not tested. Further information on this development effort is contained in Reference 27.

Additional work was performed by A. D. Little under contract to WPAFB from July 1966 to October 1968, on Brayton cycle refrigeration [28]. The design objectives of this study were to provide 1 watt of refrigeration at 3.6°K. The system was to operate in space and had an operating lifetime goal of 10,000 hours. The major components of the refrigerator system were built and tested; however, a complete working model of the system was not built.

General Electric: General Electric has investigated small cryogenic refrigeration units based on the Brayton cycle for a period of approximately 5 years. These activities have been conducted using in-house funds and more recently under contract to WPAFB. Like Garrett and A. D. Little, two cryogenic refrigeration systems have been investigated. The first unit designed and built was an 80°K refrigerator. This was a prototype unit which was tested. Test data on this unit is not available. As a result of this testing a compact 77°K unit was designed.

In addition, a second unit for cooling at 4.4°K was designed and various components were tested. General Electric has published various papers on their development effort and presented operating data on components. Extensive testing of a complete refrigeration system was not accomplished in these programs.

Hymatic Engineering: Hymatic Engineering has developed a prototype Brayton cycle unit which produces 0.3w at 28°K. Additional information on their activities is not available at this time. The available parameters of their prototype model are listed in Table 5-5 along with the other Brayton cycle units.

Other companies produce large Brayton and Claude cycle units for much higher refrigeration rates and at generally lower temperatures (≈4°K), these companies were included in the list and limited data on the parameters of these units can be found in Reference 29. Additional information on available miniature gas bearing cryogenic turbines with low to moderate flowrate is presented in Reference 30 which include the results of a survey of these units.

5.5.3 Discussion of Data for Brayton/Claude Systems

Data has been assembled on the following prototype units which operate on the Brayton cycle:

Garrett AiResearch Company
Prototype Unit (No. 40)

Hymatic Engineering
Prototype Unit (No. 42)

A. D. Little Company
Prototype Unit (No. 44)

Table 5-5

PROTOTYPE BRAYTON CY

	Manufacturer	Garrett AiResearch	Hymatic
	Trade Name	None	None
	Model	Prototype	Prototype
	I. D. Number	40	42
	Refrigeration Range	$\approx 80^{o}K$	19 - $28^{o}K$
	Cycle	Brayton	Brayton
	Working Fluid	N_2	He
	High Pressure	0.72 atm.	20 - 30 atm
	Low Pressure	0.30 atm.	1 atm.
	Minimum Temperature	$\approx 75^{o}K$	$19^{o}K$
	Cool-Down Time	≈6 hr	30 min
	Expander RPM		1500
	Volts-Phase-Frequency	115 - 1 - 60	
	Cooling Means	Water	
	Ambient Temp Reqmt	NI	$-40^{o}C$ to $70^{o}C$
	Required Attitude	NI	NI
	Cryostat Dim. (in.)	8.5 x 10.6 x 28.3	6 x 4 x 15
	Compressor Dim. (in.)		NI
	System Volume		
	Compressor Wt	15 lb	NI
	Cryostat Wt		20 lb
	System Wt	151 lb	NI
Min. Temp	Refrigeration		0.3 W at $28^{o}K$
	Power Input		
	COP		
	% Carnot		
	Lb/Watt		
	$In.^3$/Watt		
As Indicated	Refrigeration	2 W at $80^{o}K$	
	Power Input	375 W	
	COP	.00533	
	% Carnot		
	Lb/Watt		
	$In.^3$/Watt		

(1) Excluding vacuum case.

5-5
CYCLE REFRIGERATORS

	A.D. Little		
	None		
	44		
	$3.6^{o}K$		
	Brayton		
	He		
m			
	Radiative		
$0^{o}C$	NI		
	NI		
	8 D x 60 L		
	5.5 D x 52 L		
	72 lb		
	52 lb		
	124 lb		
$8^{o}K$	1 W at $3.6^{o}K$		
	1310 W		
	.000763		

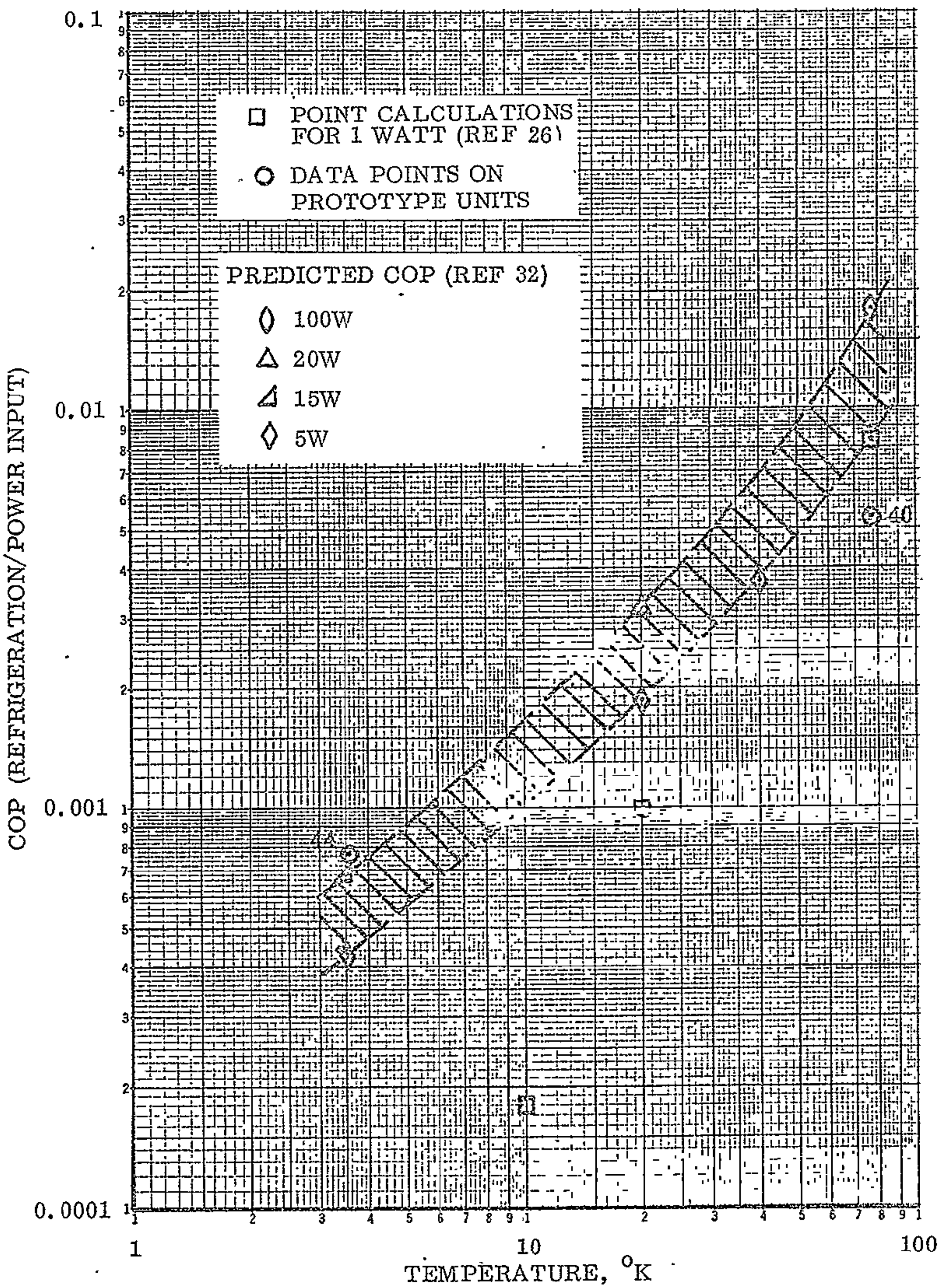


Fig. 5-43 Thermodynamic Performance of Brayton/Claude Cycle for Prototype Units and Prediction

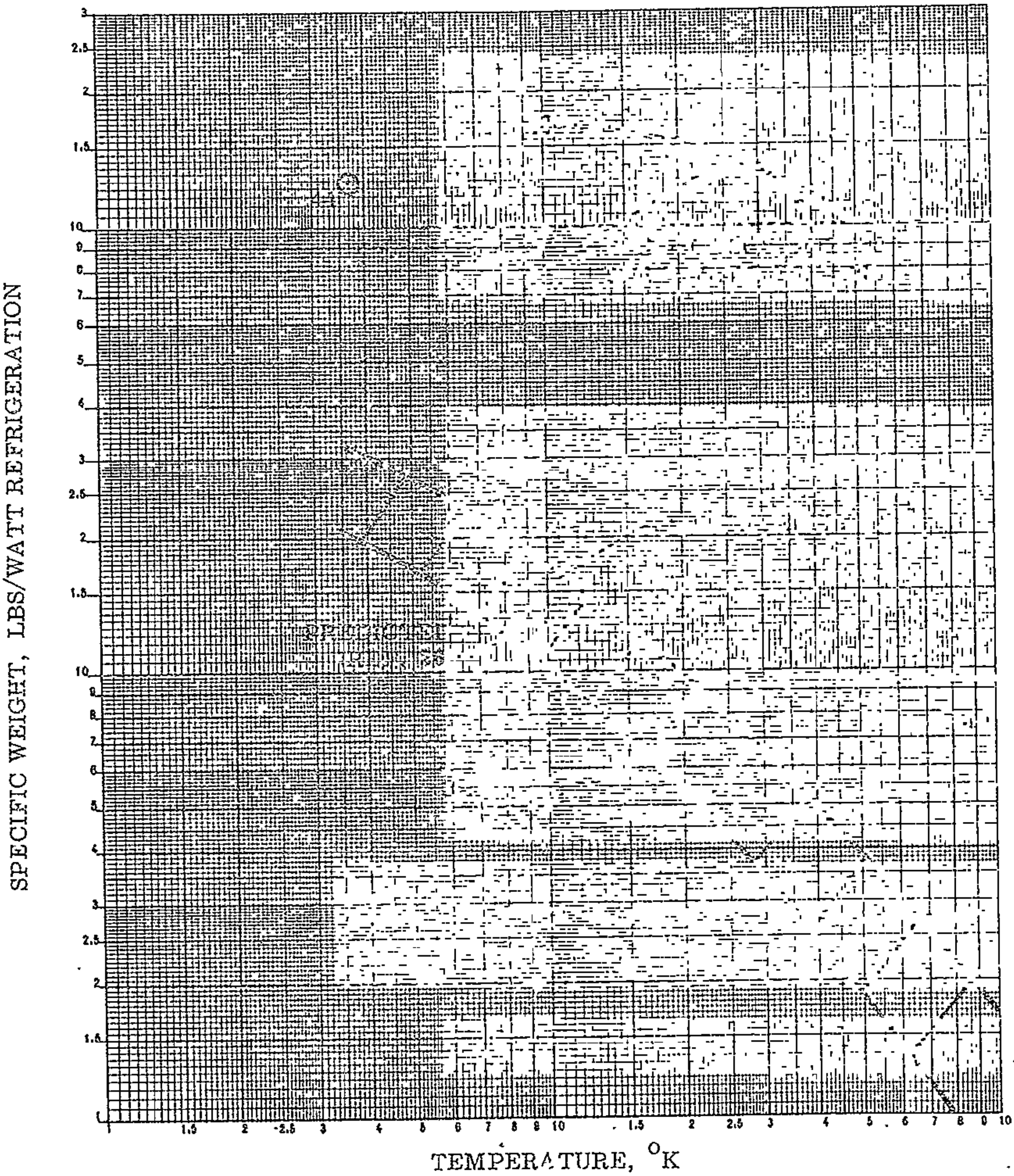


Fig. 5-44 Specific Weight of Brayton/Claude Cycle Refrigerators for Prototype Units and Prediction

All of these units are prototype units and a complete list of operating characteristics was not available, but available data is indicated in Table 5-5.

Figures 5-43 and 5-44 present the C.O.P. and specific weight data as a function of refrigeration temperature. Since experimental data on these units is severely limited, predicted performance data are also shown from two sources (26)(32)

6.0 LIFETIME AND MAINTENANCE CONSIDERATIONS

It is especially important in space application to distinguish between the stated absolute lifetime of a practical device and its maintenance interval. If the refrigerator must operate in a completely unattended fashion, then the maintenance interval becomes synonymous with the lifetime and a long life system that requires relatively frequent but minor attention might have no advantage over a system which might require no maintenance but has a short lifetime. At the present time refrigerators of the cooling capacity and load temperatures of concern to this report are unable to operate for periods much longer than 3000 hours without some kind of attention. Most systems show some kind of deterioration after a much shorter period than this, requiring anything from minor component replacement to complete overhaul. Figure 6-1 shows a summary of maintenance intervals claimed by various manufacturers, taken from data presented in Section 5.

It is clear that for long term space missions an improvement must be made in refrigerator technology. A result of this situation is that several development programs have been initiated by the U. S. Air Force, U. S. Army, and NASA to explore promising techniques for extending component lifetimes. It is obvious, therefore, that no data exist upon the reliability of long life spaceborne refrigerators, per se. The lifetime potential of these developmental refrigerators must be extrapolated from early test results, engineering background information and experience with ground based units. Reliability data for ground based units is again difficult to obtain since on the one hand, the only large scale user is the Defense Department and on the other hand the refrigerator manufacturers are not enthusiastic to divulge proprietary information.

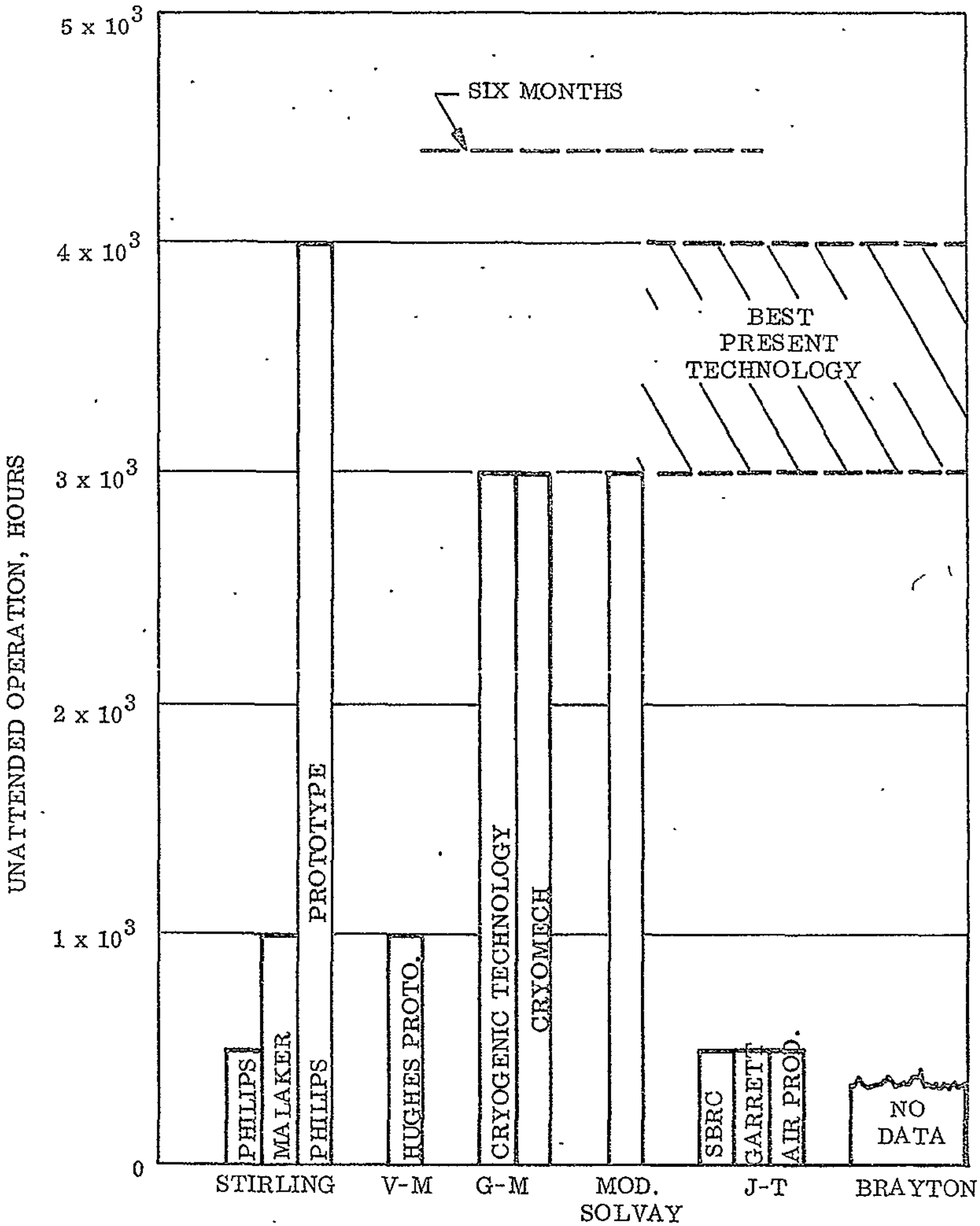


Fig. 6-1 Maintenance Intervals of Existing Refrigeration Systems

Several manufacturers claim to be able to build modified overdesigned versions of existing short life refrigerators which would operate unattended in space for as long as one year, but this has yet to be proved and the necessary weight penalties may turn out to be excessive.

At this stage of the study it has been concluded that a definitive analysis of refrigerator lifetimes and failure modes cannot be made without further investigative work. The kind of data needed for generalized lifetime prediction will have to be generated by a program of detailed component analysis rather than by questioning users or manufacturers. In this report it is possible only to define the scope of such a program.

6.1 Sources of Failure or Performance Degradation

The performance of a refrigerator can gradually deteriorate due to loss of working fluid, contamination of working fluid, internal seal wear leading to increased thermodynamic losses, and mechanical wear in the drive mechanism and other external components. These effects tend to reduce the cooling capacity and/or increase the power requirement. The performance of a refrigerator can also deteriorate abruptly by sudden failure of a component, but it is quite impossible to comment further on such a possibility at this stage.

The detailed manner in which the noted effects will cause the performance to deteriorate are as follows:

6.1.1 Effect of Working Fluid Contamination

The working fluid can become contaminated in two ways. Particular matter can enter the fluid as a result of wear or other source of breakdown of the construction materials. Considerable vapors can enter the fluid by desorption from the construction materials or as an original impurity. In those systems which use oil lubricated components, the fluid can become contaminated with oil mist due to inadequate filtration. If the compressor operates at too high a temperature the oil may effect the working of the refrigerator as follows:

- Condensing vapors or particulare matter may plug the heat exchanger passages or low temperature throttle valves thereby raising the flow resistance and reducing the cooling capacity.

- Condensing vapors or particulate matter may plug the heat exchanger thereby lowering the heat exchanger efficiency and reducing cooling capacity.

- Condensing vapors or particulate matter may enter low temperature clearance spaces, causing increased friction, increased wear rates or seizure; resulting in reduced cooling capacity and increased power consumption.

- Particulate matter may cause abrasive wear in moving clearance spaces such as in compression cylinders, gas bearings, valve mechanisms, etc., causing increased power consumption.

- Particulate matter may cause excessive abrasive wear in moving seals, causing increased input power consumption and/or fluid leakage and loss of cooling power.

- Particulate matter may prevent proper closing of compressor check valves, causing loss of cooling capacity and possible compressor overheating.

6.1.2 Effect of Loss of Working Fluid

A loss of working fluid will have only one major effect. The output of the machine will be reduced due to the reduced circulation rate and pressure levels.

6.1.3 Effect of Internal Seal Wear

The wear of internal seals will permit working fluid to bypass process components. This will result in a loss of cooling capacity due to the following effects:

- Leakage of fluid past the compressor piston or impeller from high to low pressure regions will lower the delivery flow rate.

- Leakage of fluid from high to low pressure or sides of expansion pressure valves will lower the available expansion pressure ratio and cooling capacity.

- o Leakage of fluid past displacer seals will cause inefficient heat transfer and lower cooling capacity.

6.1.4 Effect of Mechanical Wear Outside the Working Spaces

This includes normal bearing wear, seal wear between working and drive mechanism spaces, valve mechanism wear, electric motor wear, etc. These wear sources will have the following effects.

- o Increased power loss in bearings due to surface deterioration or increased play, vibration, etc., resulting in an increased input power requirement.

- o Fluid leakage from working to drive mechanism spaces, resulting in loss of working fluid, decreased pressure ratio or lower supply or working fluid to the expander.

- o Leakage of fluid from the high to low pressure sides of the valves.

- o Increased play in the bearings can magnify stress levels and lead to abrupt mechanical failure.

6.2 Possible Techniques for Extending Reliability

Based upon knowledge of the main causes of failure several areas of technology should be investigated. The lifetime of a single unattended unit can be extended by the following methods:

- o Selection of materials of construction that can be thoroughly cleaned, baked out and which produce no outgassing.

- o Selection of materials which will produce a minimum of abrasive wear particles.

- o Development of continuous working fluid filtration techniques.

- o Hermetically seal the apparatus by welding joints to minimize loss of fluid. Allow for make up of working fluid from a reservoir, if necessary.

- o Development of systems using only two lightly loaded dynamic seals, such as constant volume heat-powered systems.

- o Improve seal materials and design techniques and maintain very high tolerances on machining and assembly to minimize static loading.

- o Develop gas bearing systems.

- o Design wear compensation into these parts which will lose material with usage.

The reliability of a system can be extended by the following methods.

- o Use of refrigerators serially to obtain extended refrigeration.
- o Making provision for inflight maintenance such as by periodic purging, warm-up, serial use of oil filters, etc.

7.0 SUMMARY

This report summarizes the results of an extensive review of the state of the art of cryogenic refrigeration systems and components which have the potential for development for space usage. In conducting the survey and preparing the data the studies were generally limited to the following parameters and cycles:

Cycles	Parameters
Brayton	
Joule-Thomson	Refrigeration temperature $20^{\circ}K$ to $110^{\circ}K$
Stirling	Refrigeration capacity 5 watts to 100 watts
Vuilleumier	
Gifford-McMahon	
Modified Taconis	
Modified Solvay	

In cases where insufficient data were available for prototype units of interest, data were obtained outside the range of these parameters, primarily at lower refrigeration capacity which are required for infrared detector cooling.

The data contained within this report are to form a part of a refrigeration system handbook which is to be used as a guide and source of data for refrigeration systems analysis as applied to cryogenic space storage.

The data accumulated in this study have been tabulated and graphed, and where the data permits a curve fit has been made which allows limited extrapolation to various conditions, and which forms a basis for system performance studies. The primary performance parameters which were graphed included the following:

Specific Weight (System weight per watt of refrigeration)
Specific Volume (System weight per watt of refrigeration)
Coefficient of Performance (Watts of refrigeration per watts of power input)
System Unattended Operating Lifetime
Additional parameters were also tabulated and discussed in the text.

Figures 7-1 through 7-5 present these parameters as a function of temperature for the various cycles considered. The comparative performance parameters are presented at refrigeration levels of both 5 watts and 100 watts where possible. The performance curves at 5 watts are well supported by data on operating units, however, at 100 watts an extrapolation from smaller units is generally required. For the Vuilleumier and Brayton cycle data are available on prototype units only and extensive data on the Joule-Thomson cycle is only available at 77°K for closed cycle systems. It is felt that the data accumulated and reduced presents the best available basis for performance trade-off studies of refrigeration systems in this range. The data on the Stirling cycle units is extensive and essentially represents a baseline to which the other cycles and systems can be compared. Although the Stirling cycle shows the best performance in terms of efficiency, size and weight, the other cycles have advantages in other areas, such as lifetime and ease of integration, which require careful consideration.

None of the units investigated presently has an unattended operating lifetime as great as six months although substantial efforts to achieve longer lifetimes have been made in various Government sponsored studies. However, long lifetimes have not been demonstrated. Detailed data on failure modes of the systems is

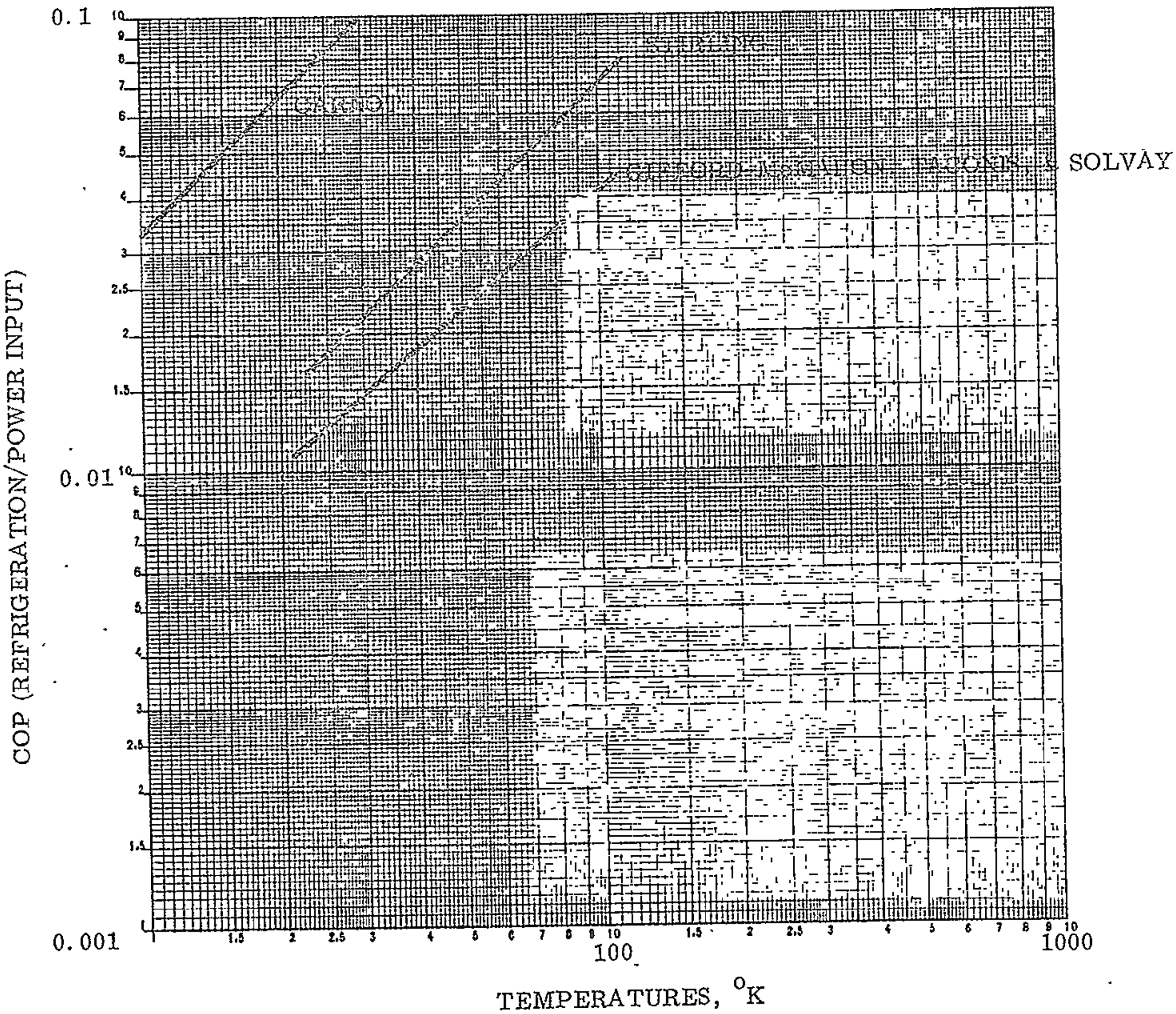


Fig. 7-1 COP of Various Cycles at 100 Watt Cooling Capacity Versus Temperature

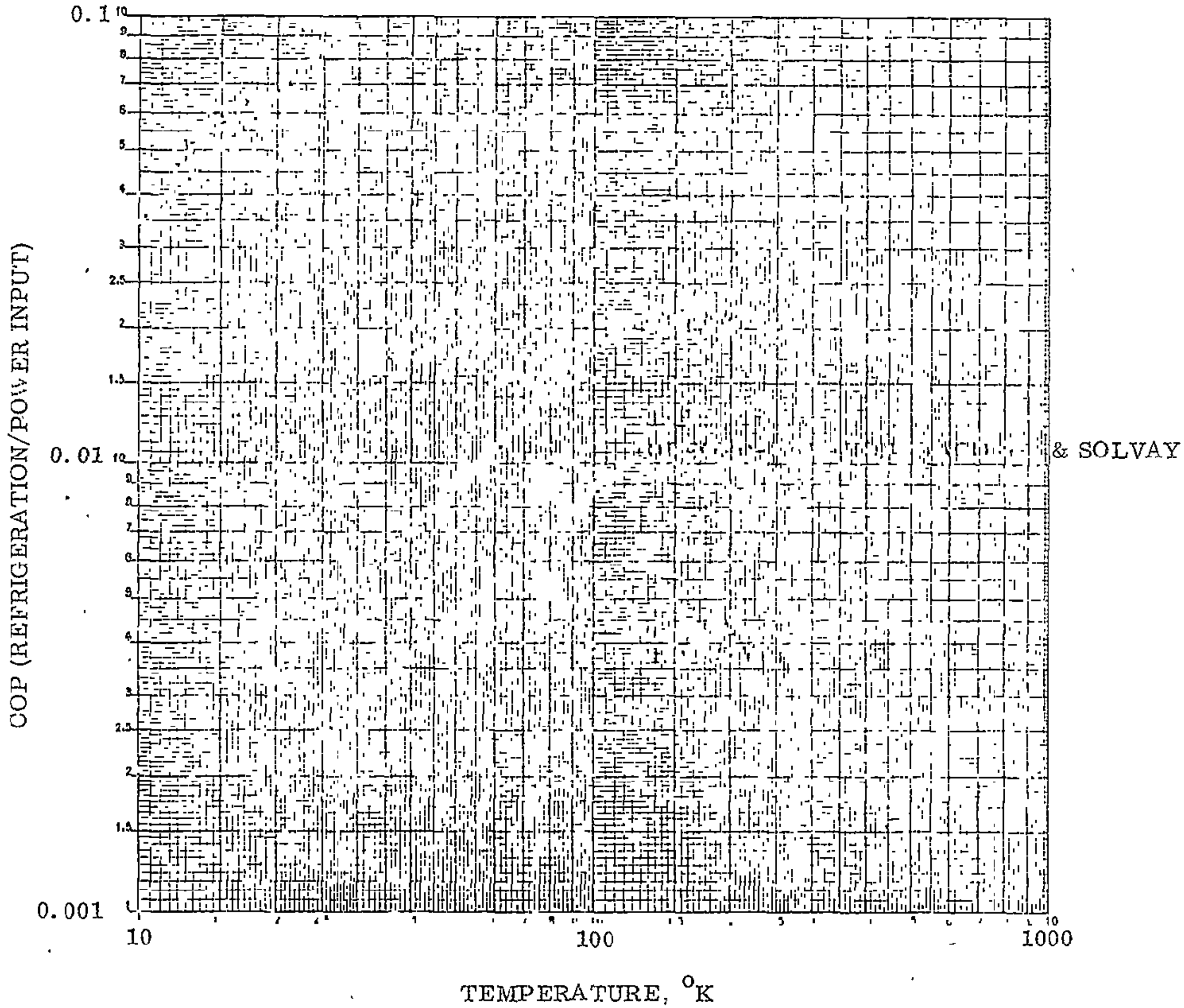


Fig. 7-2 COP of Various Cycles at 5 Watt Cooling Capacity Versus Temperature

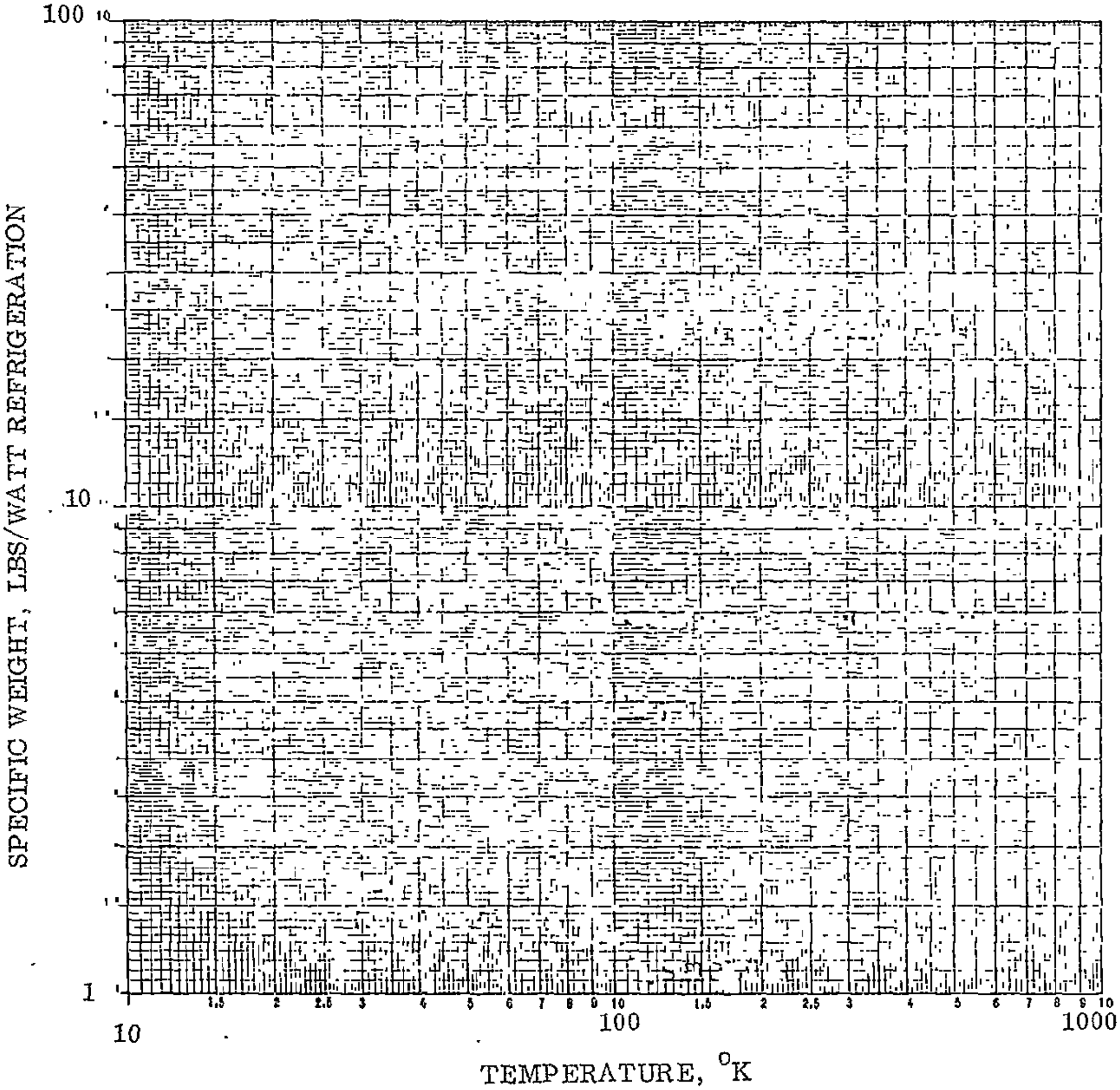


Fig. 7-3 Specific Weight of Various Cycles at 5 Watt Capacity Versus Temperature

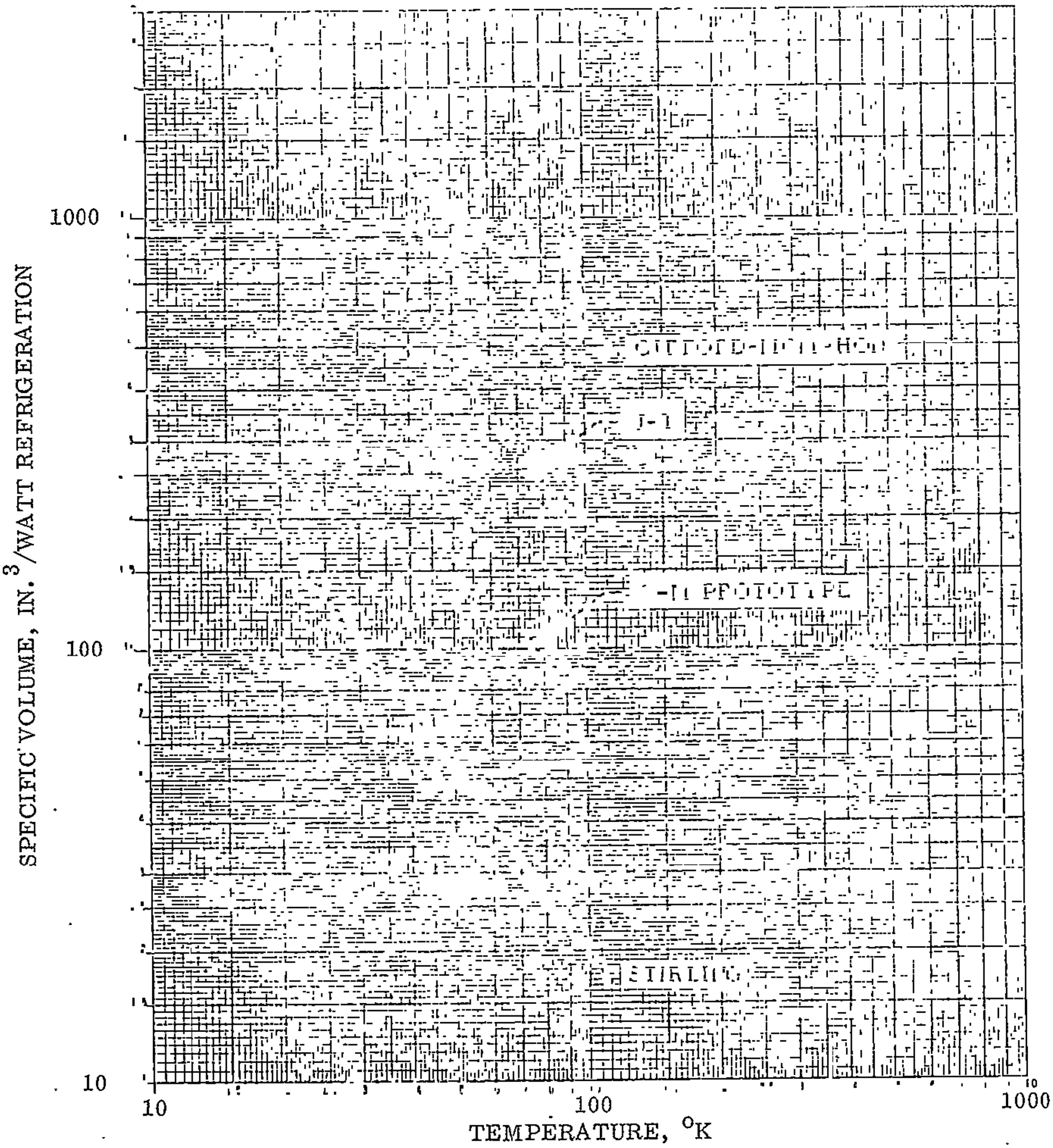


Fig. 7-4 Specific Volume of Various Cycles at 5 Watt Capacity Versus Temperature

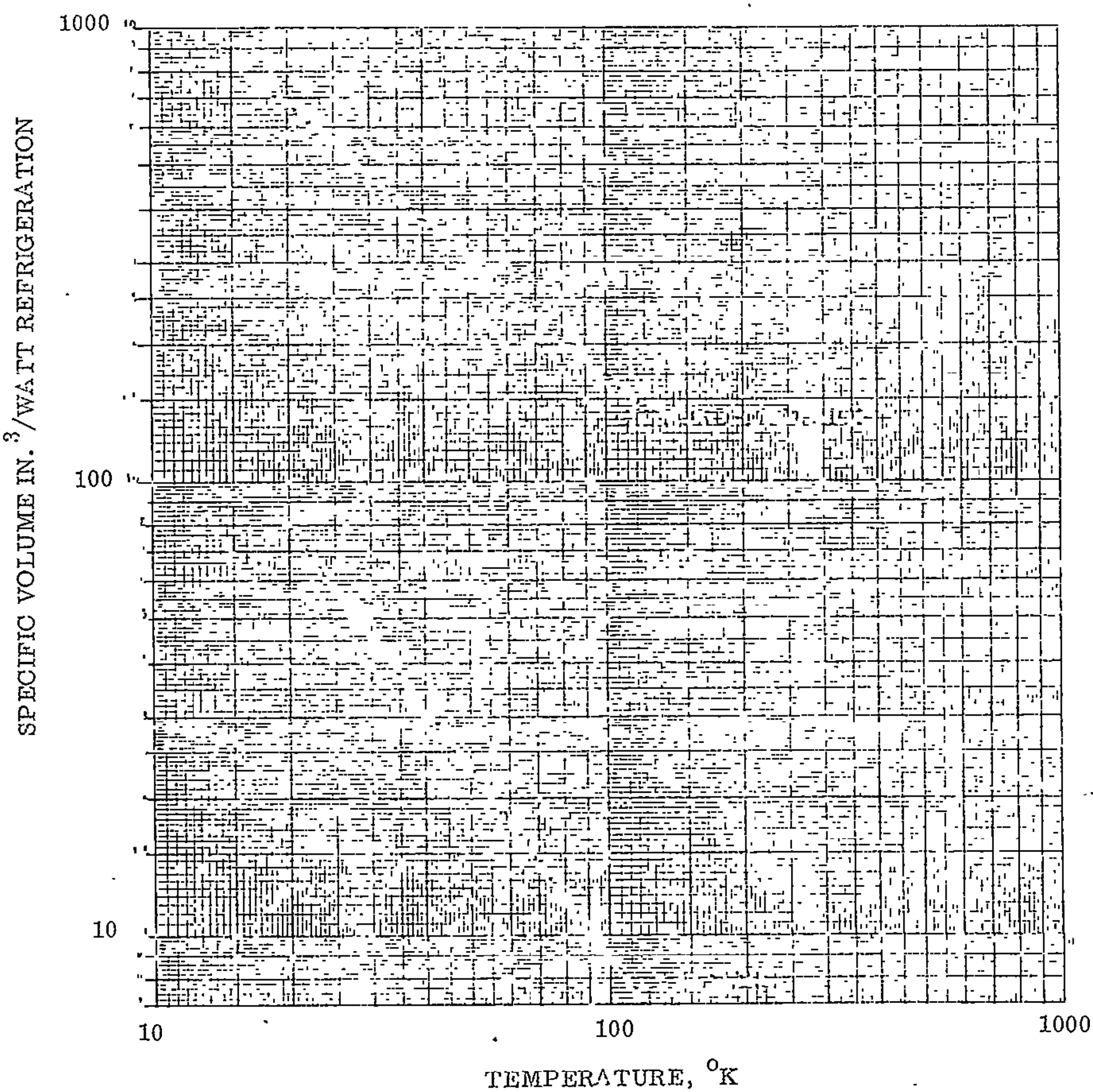


Fig. 7-5 Specific Volume of Various Cycles at 100 Watt Capacity Versus Temperature

quite scarce and undoubtedly this is due in part to the proprietary nature of the machines. The area of lifetime for the refrigeration units is a key item in their development and much additional work will be required to devise ways in which operating lifetime can be extended.

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From the beginning of the program to the present a comprehensive survey of refrigerator development has been made. This has included establishing communication with the following specialist refrigerator manufacturers.

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