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A STUDY OF COMPONENT PARTS FOR COMBAT TANK VENTILATION, COOLING, AND HEATING

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FOREWORD

The analytical study of heating, cooling, and ventilation components reported here was completed as one phase of a project whose overall object is to determine the feasibility of increasing combat tank crew comfort and efficiency by conditioning the crew compartment atmosphere.

ABSTRACT

A discussion is made of the various components that might be employed in the design of a system for heating, cooling, and ventilating a combat tank. These components are viewed with consideration to space requirements, power requirements, and other factors that are peculiar to the component under discussion. Charts are presented for certain points of comparison, and conclusions are listed at the end of the discussion. Recommendations are purposely avoided, since it is felt that these can be presented with greater logic in the final report of the overall project.

OBJECTIVE

The objective of this project was to determine the feasibility of ventilating and heating, or ventilating and cooling, the crew compartment of a medium combat tank in environments characterized by extreme temperatures. The upper limit of environmental temperature anticipated by the project is 120°F and the lower limit is -30°F. The possibility of controlling temperatures in the main engine compartment was also to be studied, and an estimate was to be made of the practicability of attaining all the above objectives through the design of a universal package system which could be fitted to all tanks without the necessity for redesign.

I. INTRODUCTION

This report is concerned with the results of a study made to determine the practicability of using commonly known methods for heating and cooling the compartment air in a combat tank. The study was one phase of a project whose overall objective is to determine the feasibility of cooling, heating, and ventilating a combat tank under extreme ambient temperature conditions. The most desirable arrangement is to have a "package" item of equipment that could be installed by an ordnance organization. This package would be suitable, on one hand, for maintaining a supply of cooled air where there are extreme hot temperatures and, on the other hand, for providing heated air in cold ambient temperatures. It would also be desirable that the system be capable of maintaining temperatures conducive to ready starting in the engine compartment after extended shutdown periods in cold climates.

As a result of a literature search it has been found that at least two experimental projects were carried out with the objective of obtaining possible methods of cooling tank crew compartments in extreme hot environments. Both of these projects were sponsored by foreign governments. The German Government, during World War II, engaged in a large-scale project to obtain increased crew efficiency in North African operations by increasing crew comfort. Unfortunately, it has been possible to obtain only a very small amount of information concerning the work, due to the loss of the pertinent reports at the close of the war. The other experimental project of large proportions was also carried out during World War II, under the sponsorship of the Australian Government. Reports concerning this work have been located and were reprinted at the University of Michigan with the author's permission[11,12].

Information on work that has been done with aircraft, buses, railroad cars, and seacraft is available in fairly large amounts, and much of this information might be interpreted in terms of the combat tank cooling and heating problem.

In this phase of the present project, it is desired to determine the feasibility of using certain types of components in the design of a package system capable of attaining the overall objectives listed above.

II. PLAN OF INVESTIGATION

At the start of the project it was tentatively planned to make a comparative study of stock items of various manufacturers. It soon became apparent, however, that the combat tank presented a unique enough problem that few presently manufactured components could be incorporated in the type of system desired. Due to space limitations and rigid configuration requirements, together with a need for minimizing power requirements, practically every piece of equipment to be used will be subject to special design. It seems, then, that the best manner in which the project can serve is to make preliminary determination as to what types of design are even worthwhile considering. It is not expected that the discussion will prove that certain designs are feasible. That must be done by experienced industrial design engineers who specialize in such problems. On the other hand, it is hoped that the discussion can show what direction the design attempts should not take, or at least what penalties to expect in taking such a direction.

It should be noted that, in general, the computations on which the report is based are idealized in nature; that is, efficiency factors have not been applied. This is usually done where it is only desired to show that the results to be obtained from a given design are definitely unattractive. In other cases, estimation of efficiencies have not been attempted due to the fact that in designing for a given objective only experimentation will tell what efficiency a unit will possess. Attempts have been made to consider each component in the light of its most important requirements. In addition, discussion is made of the effects of "external factors", or "other considerations", on the practicability of using a component. These include such conditions as type of operation expected, dust, vibration, lack of maintenance, and other typical conditions that the system will encounter.

Conclusions have been listed at the end of the discussion, but recommendations are reserved for the final report on the overall project where they can be given on the basis of all phases considered together.

III. DISCUSSION OF COMPONENTS

A. REFRIGERANT FLUIDS

The bases for comparison used here are the flammability, toxicity, corrosion, volume displacement of vapor, operating pressures, and power requirements for compression. In order to compare such characteristics as operating pressure and volume displacment, specifications of an evaporation temperature of 40°F and a condensation temperature of 120°F were made.

1. Flammability and Toxicity.—Of the characteristics mentioned above, these two must be cause for rejection from further consideration even if a refrigerant has either of them in only a small degree. This is readily appreciated when one considers the confined space in which a tank crew operates and the fact that crew members are "pinned down" to that space when in danger of being taken under enemy attack. Leakage of toxic or flammable gases under such conditions is intolerable. Table I[1] shows figures relative to the flammability and toxicity of refrigerants that may be used to some extent in present day systems. In view of their pressure and temperature characteristics several of the fluids listed would obviously not warrant consideration even if flammability and toxicity were not important. They are included in the list to show that it is difficult to find any refrigerants that are neither flammable nor toxic.

In making a decision as to whether a fluid should be considered safe for use, column 5 of Table I was referred to and those refrigerants which were not shown as being at least practically nonflammable were considered unsafe. Columns 2, 3, and 4 were then inspected for an indication of toxicity. Of the nonflammable group, those showing either a large "percentage by volume" (column 3) or a large ratio of column 4 to column 3 were considered safe. The latter basis for acceptability was made because of the large mass amount of refrigerant leakage necessary to bring the volume concentration to a dangerously toxic point. Freon-21 would seem perhaps to be on the border line of safety, especially since the "duration of exposure" (column 2) is comparatively low. Toxicity figures were not shown in the reference table for Freon-22. It is noted that carbon dioxide meets the safety requirements, but it will not be considered further because it is so obviously unsuitable for the refrigeration conditions desired in combat tank cooling.

With these points in mind, then, the only refrigerants of those listed that will be compared further on the basis of other characteristics are all of the Freons. Air, because of the possibility of its use in an "open" air cycle, will also be included in the comparison of the remaining refrigerant characteristics.

2. Corrosion.—The pure Freons do not cause any appreciable corrosive attack upon ordinary materials of construction, with the exception of magnesium. In the presence of moisture these refrigerants may form halogen acids which will attack any of the materials of construction ordinarily used. The latter danger is not considered to be great if reasonable

efforts are made to keep the refrigerants as free as possible of moisture. The small corrosive effects due to air oxidation are, of course, well known and these are aggravated by the presence of moisture. Such effects are not great enough to cause serious trouble.

3. Volume Displacement of Vapors.—Table II, in column 2, shows the compressor displacement at inlet conditions for the various refrigerants operating between evaporation and condensation temperatures of 40° and 120°F, respectively. This characteristic, of course, will give an indication of the comparative size of compressor needed for different refrigerants at given temperature specifications. Assuming that a positive displacement compressor is used, the desirable refrigerant would have as small a vapor volume per ton of refrigeration as possible. In the case of dynamic compression the opposite is true since large volumes are needed to enable large enough passages to keep friction losses low in the compressor.

This characteristic will also enable a determination as to whether dynamic (centrifugal) compression can be used. This point is discussed further in the section concerning compressors.

4. Operating Pressures.—Columns 3 and 4 of Table II show the evaporator and condenser pressures for the refrigerants operating between 40° and 120°F, respectively. The air cycle, not being a phase-change cycle, is given in terms of pressure ratio. The power requirement per ton of refrigeration is not determined solely by the pressure ratios, since the specific latent heat of phase change is different in the various fluids. Thus, compression of fewer necessary pounds of fluid may offset the greater horsepower requirement per pound, based on pressure ratio, in producing a ton of refrigeration.

Aside from the compression ratios, the absolute values of the operating pressures would bear some study as such. If the high side pressure is considerably above atmospheric pressure, the equipment must be designed to withstand such pressures, resulting in greater cost and increased space requirement. Also the greater the difference between the system pressure and atmospheric pressure, the greater are the chances for leakage into or out of the system, depending on which pressure is greater.

5. Power Requirements for Compression.—Column 5 of Table II shows the theoretical horsepower requirement for compressing enough refrigerant fluid to produce one ton of refrigeration between the temperature limits of 40° and 120°F. No effort has been made to apply any efficiencies here, since the main purpose of the table is to compare the requirements of one fluid with those of the others. It can be noted that, on this basis, there is no great argument for using one Freon as opposed to another. Any slight difference in theoretical power may be made up by the efficiencies accomplished in the actual system.

Comparison of requirements for the air cycle as opposed to the Freons is another story. If the air cycle is used, justification for its use on other grounds must be great in order to offset the much greater power needed for compression. Indeed, unless most of the work put into compression can be recovered during the cooling expansion, it is possible, or even probable, that no justification can be made for the use of the air cycle.

6. External Factors Affecting the Refrigerants.—It is this category that provides the main basis for considering the use of the air cycle. Since the ultimate objective of the system is to condition the compartment air, the most inviting prospect is to use the air itself as the working fluid if this can be done. A limitless supply of the refrigerant is always abundantly at hand. Considerations of flammability and toxicity are eliminated. Corrosion problems are of negligible importance. The system can be designed with one heat exchanger as opposed to the requirement for at least two in the vapor compression cycles. The importance of maintaining a leakproof system is not as great as for a Freon system. To the credit of the Freons in the consideration of external effects is the fact that, being a closed system, the internal passages and the compressor itself are not subject to the clogging and erosive effects of dust particles. In tank operations, air and dust go hand-in-hand and filtering out the dust is a big problem. Clogging of heat exchangers and air passages will possibly be encountered in any cooling system. In an air cycle system the problem is aggravated by the use of dust-laden air as the working fluid.

In summarizing, a balance must be obtained between the requirements of logistics, maintenance, space, and power before it will be possible to use an air cycle as opposed to a vapor-compression cycle, or the converse. In order to assist in obtaining such a balance, the other components of a system must be studied.

B. COMPRESSORS

Compressors are classified in two broad categories, "positive displacement" and "dynamic". The compressors that were studied in this project are reciprocating and rotary, both positive-displacement types, and the centrifugal, which is a dynamic type. Axial compressors are omitted because they are primarily machines of much higher capacity than the tank problem calls for. Comparatively, reciprocating compressors are considered small-capacity machines, rotaries are small to medium capacity, and centrifugals are considered medium- to large-capacity machines.

The most important single characteristic to be compared in this problem is the space required. Since centrifugals are considered medium-capacity machines, it was necessary to determine whether they are at all practical for compression of the Freons and of air. The initial limiting factors in the use of centrifugals are the rotative speeds that can be tolerated and the width of the impeller passages needed. Pressure rise in such compressors is a function of the gas velocities which in turn are functions of the impeller tip speed. If the capacity is to be small, then the impellers must be of small diameter and the rotative speed must be accordingly increased in order to provide the proper tip speeds. These speeds are limited in magnitude by the strength of the construction materials used in the compressor. Further, if the capacity is to be small, then the impeller passages must be made narrower along their axial dimension. As these passages are made smaller, friction losses increase and the efficiency of the machine may be too low to allow its further consideration.

1. Practicability of Centrifugal Compression.—Theoretical calculations were carried out in order to determine the practicability of utilizing centrifugal compression for the various fluids under consideration. These calculations assumed that none of the fluid was liquefied during compression, an assumption that might be slightly erroneous in some cases (for example, Freon-113). The amount of error involved, however, would not effect the answer that is being sought. The calculations were carried out on an arbitrary basis of three tons of refrigeration required. A sample calculation of the type carried out is shown in Appendix I. The results are shown in Table III and it is seen that, for the tonnage specified, impeller passages are far too narrow to permit efficient centrifugal compression. Furthermore, inspection of column 2 of Table II shows that in order to obtain what might be termed reasonable efficiencies, the amount of vapor volume required would place refrigeration tonnage at a very high value. This, of course, would result in power requirements much in excess of that which could be spared in a combat tank.

Of the refrigerants shown, air seems to hold out the greatest hope for the practical use of centrifugal compression. With this thought in mind, calculations were carried out to determine the point at which centrifugal compression of air might become feasible. A description of these calculations is shown in Appendix II. In order to determine this feasibility, use was made of an efficiency which is defined as the power theoretically required to produce the necessary tonnage of refrigeration, divided by this theoretical power plus the power required to compensate the friction losses in the impeller passages. Two points should be noted: (1) the theoretically required horsepower to produce the necessary tonnage assumes feedback of the work of expansion to the compressor, and (2) friction losses in the diffusers, return passages, and those due to entrance and construction effects have not been accounted for. Other inefficiencies due to machine friction have also been ignored. It would seem that, aside from impeller losses, reciprocating compressors would have losses that are analogous to those of centrifugal compressors. The results of the calculations of the type described in Appendix II are shown in Figure 1 where the efficiency described above is plotted against the axial width of the impeller passage. In reading these curves it is important to note the assumptions made at the beginning of the sample calculations of Appendix II. Twelve inches was chosen as the impeller diameter because it is believed that this will represent a compressor of the largest dimensions that could be tolerated in the combat tank. Single-stage compression was chosen simply to apply the worst conditions as far as the number of stages is concerned. Actually the assumption of a single stage is an unrealistic one, since the pressure rise required probably could not be attained in a single stage in present design practice.

The AiResearch Manufacturing Company has succeeded in designing and developing a two-stage gas-turbine-driven compressor with a through-flow rate capacity of 2.5 lb/sec at a head of 40,000 ft-lb/lb.(9) The adiabatic efficiency of the unit is stated as being about 80 percent. Maximum envelope dimensions are 16 inches in diameter and 13.5 inches in length. It is seen that the power requirement for this unit would be far too great, and the through-flow rate also is much greater than desired in the combat tank. However, the objectives of the designers were specifically those obtained. It is possible that such designers might also succeed in developing a compressor tailored to the requirements for refrigeration in a tank crew compartment.

On the basis of the results shown in Table III and Figure 1, it seems that the use of centrifugal compression for the vapor compression cycle is not practicable and its use for the air cycle is a marginal possibility. The latter point could only be settled by attempts to design and develop a centrifugal compressor having the desirable characteristics.

2. Space Requirements for Compressors.—It was shown in the previous paragraphs that centrifugal compression for the vapor compression cycle is not feasible. It was also stated that the use of a centrifugal compressor for the air cycle would involve special design and development. If such a development were successful in attaining its objective, there is no way of knowing positively what the dimensions of such a compressor would be. As a hint of the order of magnitude of these dimensions, reference was made above to the gas-turbine-driven compressor developed by the AiResearch Manufacturing Company. Based on the maximum envelope dimensions given there, the space requirement for that particular compressor by itself would be about 1.6 cubic feet. This is as far as the present discussion can go in estimating the space requirement for a centrifugal compressor. It is even possible that a design to answer the tank problem might require additional stages and greater space.

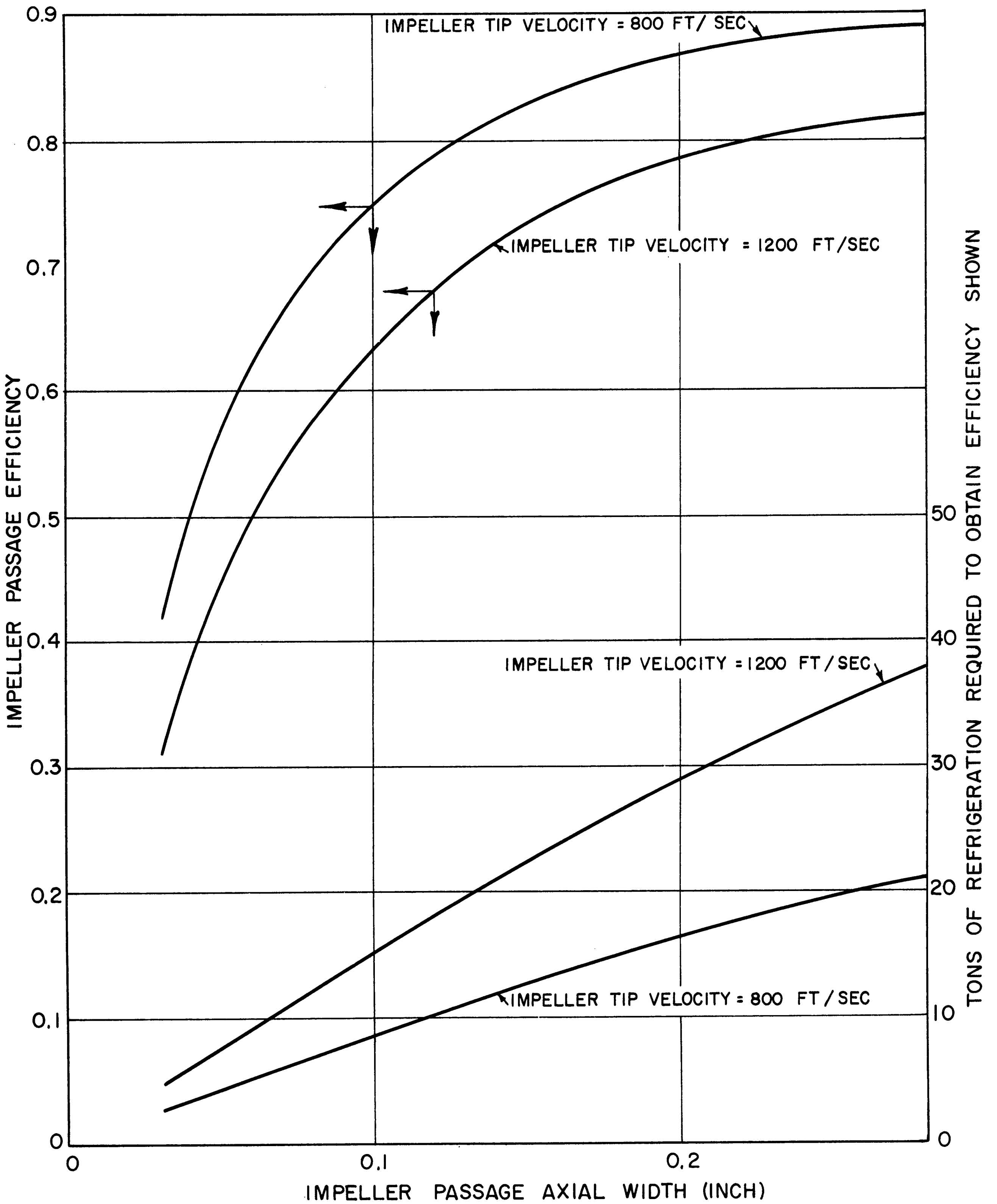

Figure 1. Impeller passage efficiency vs impeller passage axial width (assuming single-stage compression).

Table IV presents figures for space requirements in single-stage rotary and reciprocating compressors. These figures are based on listings in the catalogs of various manufacturers and are approximate. Column 1 for rotary compressors shows the volume required, based on the maximum envelope dimensions of the compressors only. Naturally, duct lines to the inlet nozzle and from the exit nozzle will increase the requirement for space immediately adjacent to the compressor. Column 2 presents the approximate volume of the compressor rotor housing, inlet and exit nozzles, lubricator, and power coupling. Note that allowance has not been made for mounting brackets or, again, for the duct lines leading to and from the compressor. It is believed that in a special design for solving the tank problem, the space requirement for a given rotary compressor would lie somewhere between the limits of columns 1 and 2. Column 3 for reciprocating compressors shows the volume required on the basis of maximum envelope dimensions of the compressor only.

3. Power Requirements for Compressors.—As pointed out in the section on refrigerant fluids, the theoretical power requirements for the various Freons are not greatly different from each other; but the air cycle requires considerably greater power for compression than do the Freons. The theoretical requirements are shown in graphical form in Figures 2 and 3, while the actual power required for reciprocal compression of air is shown in Figure 4. In Figure 2 the requirements of all Freons, except Freon-21, will follow loci almost identical with Freon-22. Also on Figure 2 it will be noted that the power requirement for air with a compression ratio of 4 is less than at a pressure ratio of 3. This seeming anomaly results from the fact that, theoretically at least, a greater amount of work is available for feedback at the higher ratio. Due to inefficiencies, however, realization of this situation is questionable.

Figure 3, in addition to showing power requirement versus refrigeration tonnage, also shows theoretical displacement versus refrigeration tonnage. A cross-reading of the plot will, of course, show power requirement versus displacement.

Figure 4 is based on data taken from manufacturers' catalogs, and compression and mechanical inefficiencies are allowed for. A comparison of this chart with that of Figure 3 shows that, in general, the overall efficiencies to be expected are about 80 percent. Approximately the same efficiencies can be expected in reciprocal compression of the Freons.

Efficiencies for rotary compressors will be generally slightly lower than those for reciprocating compressors. The variations, however, will not be great.

Centrigufal compressors used in industry have good adiabatic

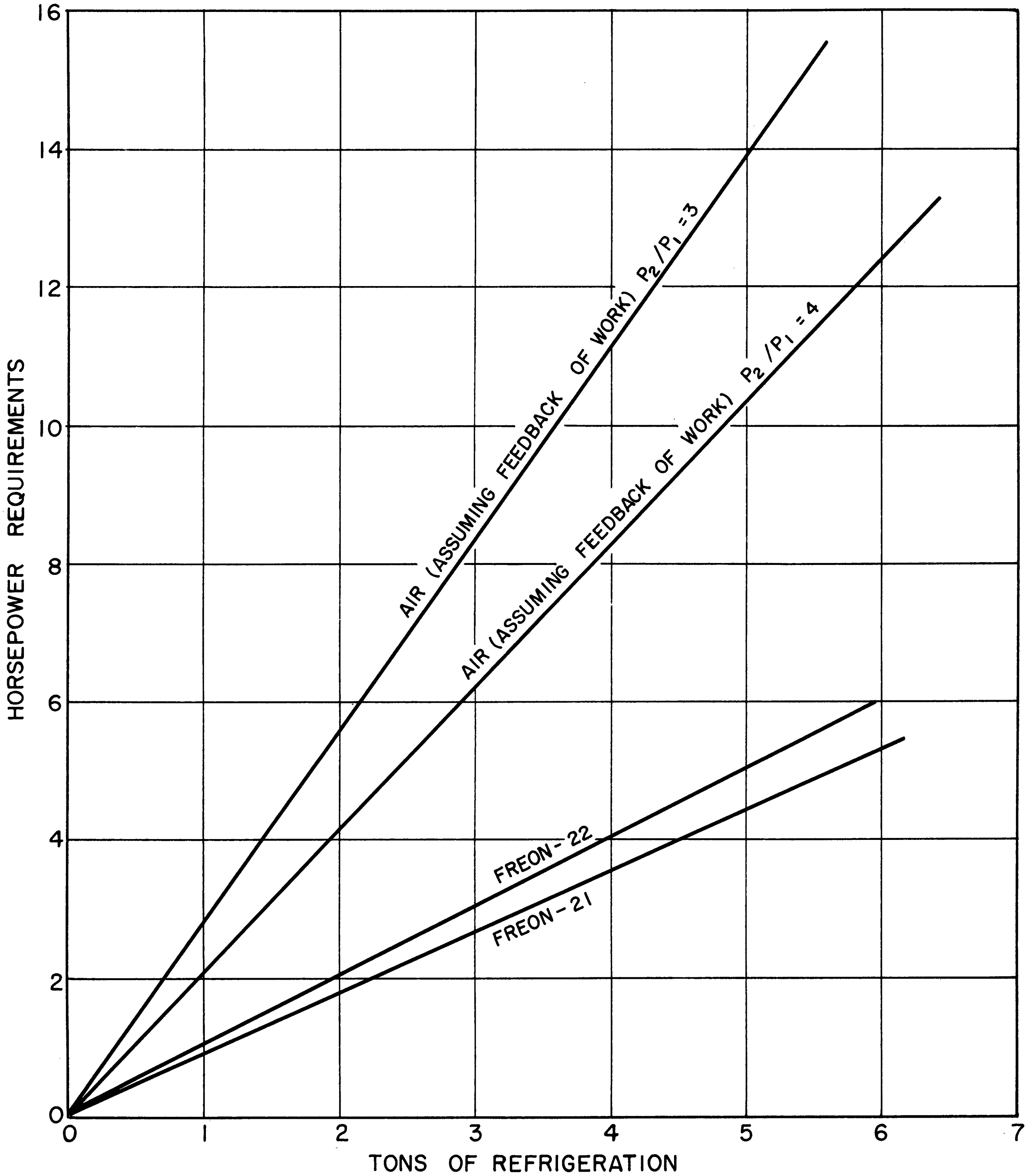

Figure 2. Theoretical horsepower required for refrigeration air (assuming total work feedback); Freon-21 and Freon-22 vapor compression cycles.

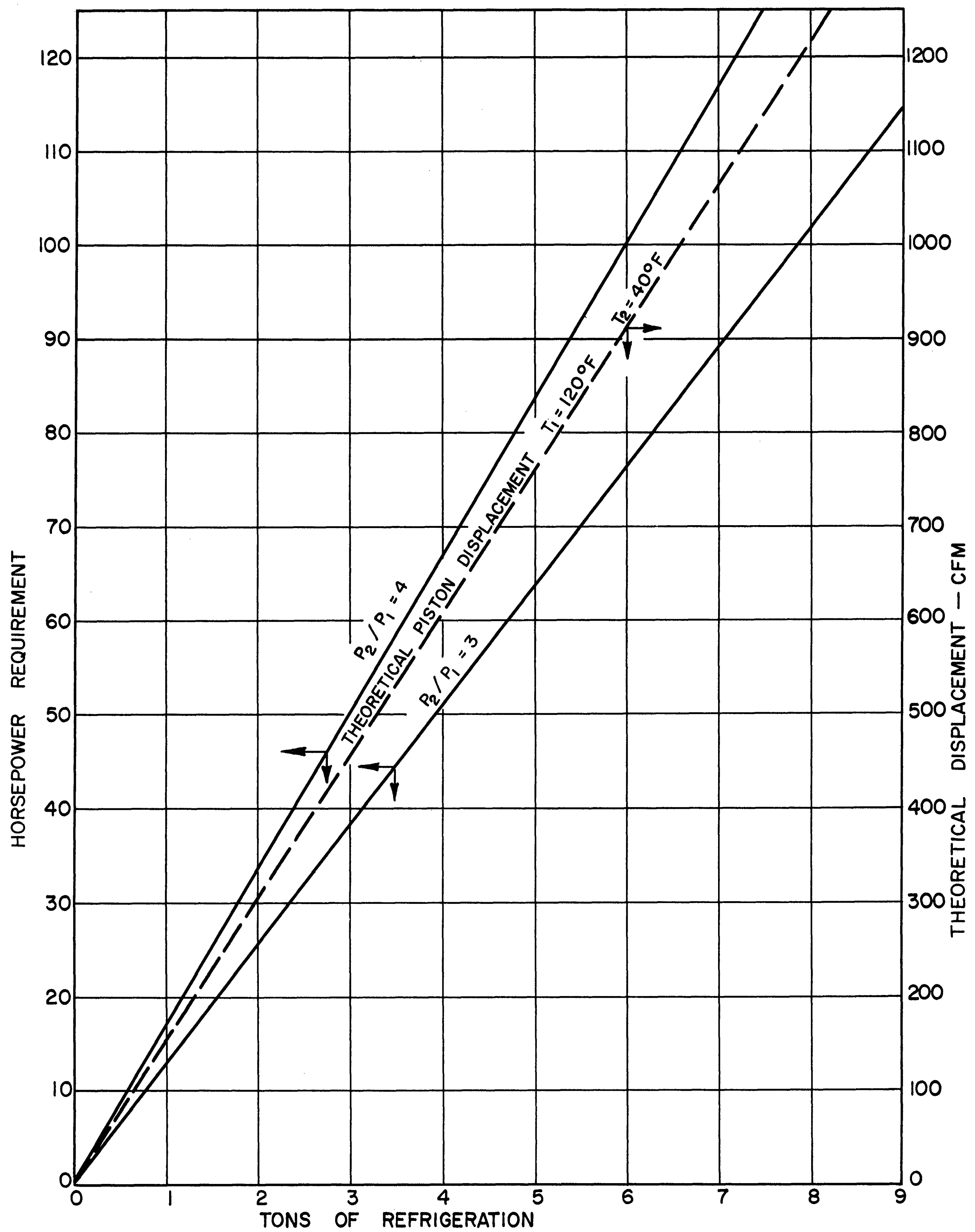

Figure 3. Theoretical horsepower requirement and piston displacement vs tons of refrigeration (assuming no work feedback).

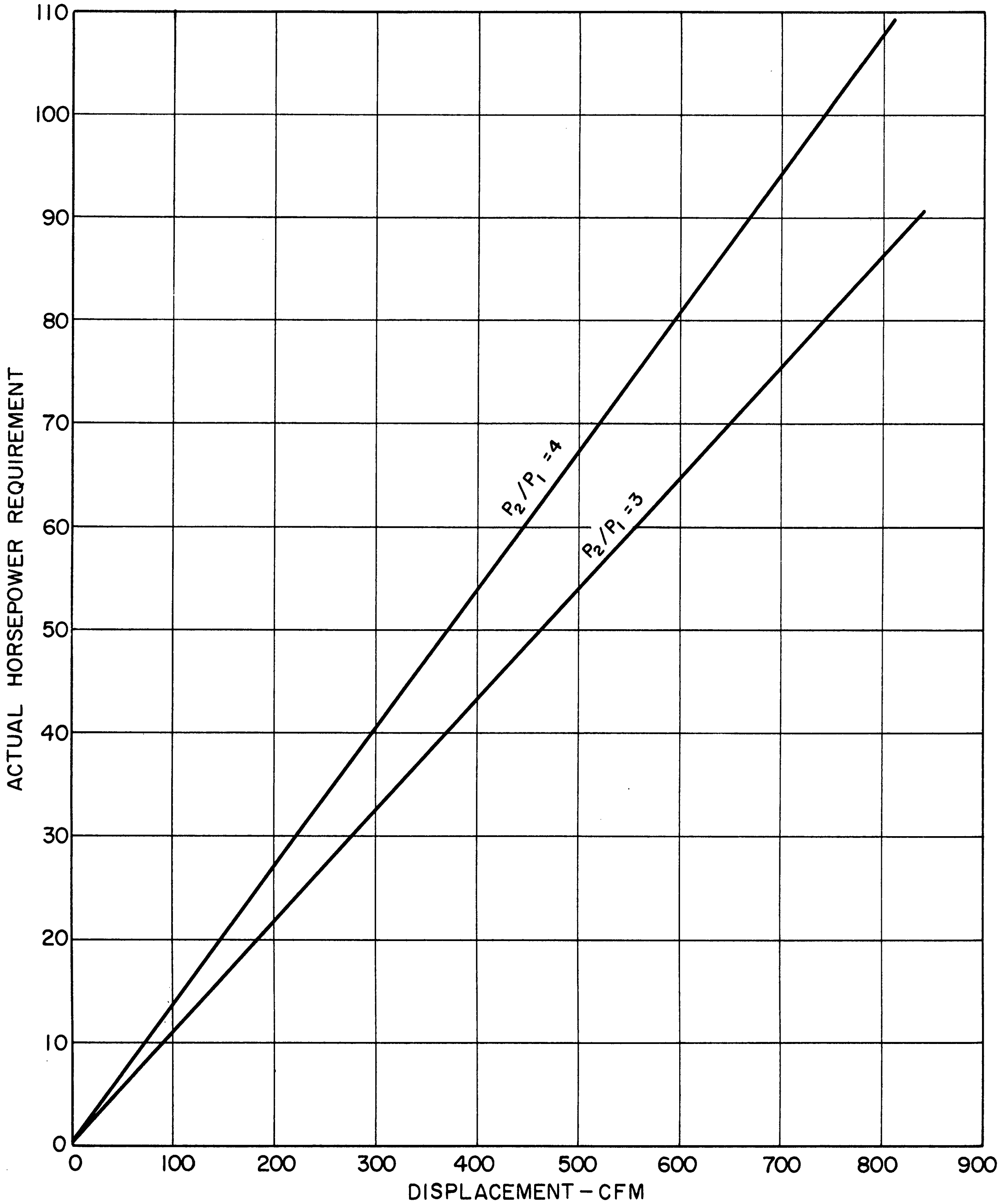

Figure 4. Actual horsepower requirement vs displacement for reciprocating compressors (assuming no work feedback); P_1 = 14.7 psia; (Reference 10).

efficiency ratings, as high as 80 to 90 percent. Such compressors, though, have much higher capacity ratings than the tank problem is concerned with. The adiabatic efficiency of the AiResearch gas-turbine-driven compressor referred to earlier is about 80 percent, but even this comparatively small machine has a capacity rating much in excess of that desired for a combat tank. It would seem certain that any design for a compressor having capacities suitable for a tank will have efficiencies below those mentioned above.

4. Other Considerations.—Reciprocating compressors have been the most commonly used in the past, but rotary compressors have experienced increased favor in recent years. If an effort is to be made to utiliize system components as presently manufactured, it seems that reciprocating compressors would bear first inspection, with rotary compressors being worthy of consideration also. If modification of design is to be made, based substantially on existing designs, the converse order of consideration seems logical. If a completely new design is to be contemplated, then the advantages of centrifugal compression in an air cycle would make this method worthy of at least a preliminary design study.

Rotary compressors require precise machining which would make initial cost comparatively high. Because of close tolerances that are built in, lubrication and maintenance are larger problems than in the reciprocating machines. Under the conditions to be experienced in a combat tank, the required operating periods for the compressor are long; in fact, it might be running constantly for a whole day at a time. Reciprocating compressors would probably be more adaptable to this type of service. Aside from the advantages of small size, centrifugal compressors have no contact moving parts other than in the shaft. Lubricating problems are thus minimized and mechanical maintenance would not be as pressing a problem. In addition, over long periods the efficiency of the maching should remain essentially constant due to the absence of rubbing parts.

It is believed that any of the types of compressors discussed can be made sturdy enough to withstand the shock and vibration inherent in tank operation. In certain atmospheres, especially in desert operations, dust intake becomes a major problem with the air cycle, since the air to be compressed and subsequently rejected to the compartment is drawn from the surrounding atmosphere. This problem is exhibited in dust contamination of the compartment air, but similar difficulties would also be experienced with a vapor-compression system. Unique to the air cycle, though, is the possible resulting difficulty of erosion of certain parts of the compressor. This holds true no matter what type of compression is used, and it particularly militates against use of positive-displacement compression in an "open" air cycle where the troubles result from abrasion rather than impingement.

C. HEAT EXCHANGERS

In these components the main points to be considered are the space required to accommodate the necessary heat-transfer area, and the pressure drops across the exchangers. This section will be concerned with a short preliminary study of compact heat-exchange surfaces of the type developed largely for the aircraft industry, but which is also used to an increasing extent for many other purposes. Calculated, too, were estimates of tubing needed for smooth, tubular heat exchange.

1. Space Requirements.—It is most difficult in any heat exchange between two fluids separated by a solid wall to predict the overall heat-transfer coefficient. With present-day designs it can be estimated, as a first approximation, that it is possible to design air-to-air heat exchangers to have an overall heat-transfer coefficient of up to 10 Btu/(lb-ft^2-°F). About the same, or perhaps slightly higher, values are possible between air and condensing or vaporizing fluids. This study was made, then, on the basis of assumed overall coefficient values of two, six, and ten for air-to-air exchange, and of four and ten for air-condensable vapor exchange.

The study of the compact heat-exchange surface is based on a report by Kays, London, and Johnson,[8] and is considerably simplified for the purposes of this project. The actual design of such heat exchangers is even much more involved than the presentation of the authors shows, as they point out. The project reported by the authors made a study of 34 different core surface configurations, which were classified in three groups: tubular, plate-fin, and finned-flat tubes. Depending on the particular core geometry, these exchangers had heat-transfer areas on a given side of up to about 600 sq ft/cu ft of core volume. Appendix III shows a sample calculation of the type used in the present project to study the compact heat exchanger. In the calculations a heat-transfer-area to core-volume ratio of 250 sq ft/cu ft was used in all cases. The results are shown in Figures 5, 6, 7, 8, and 9.

2. Pressure Drops Across Exchangers.—For the size of the heat exchangers necessary in the tank cooling problem, it should be possible to keep the pressure drop within a maximum of two or three percent of the entering pressure of the gases involved. On this basis, the horsepower needed to overcome friction losses in the exchanger should not exceed a value of about one for the tank problem. The higher requirements will be experienced in the denser core structures and will increase with increasing flowrates.

3. Other Considerations.—As in the case of compressors the external factors of chief concern in the use of the exchangers is dust.

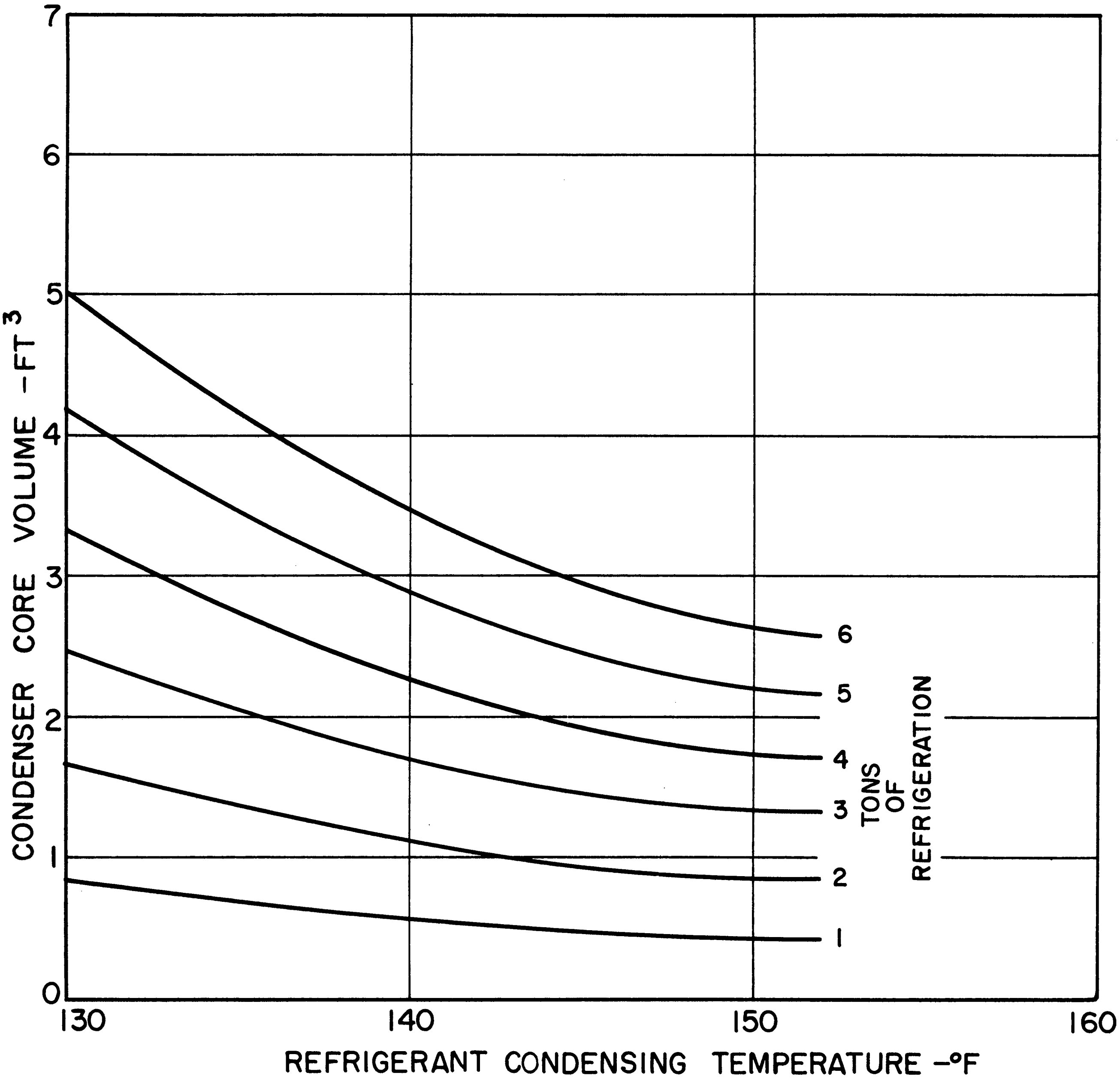

Figure 5. Condenser core volume estimates vs refrigerant condensing temperature; U = 10 Btu/(hr-ft^2-°F).

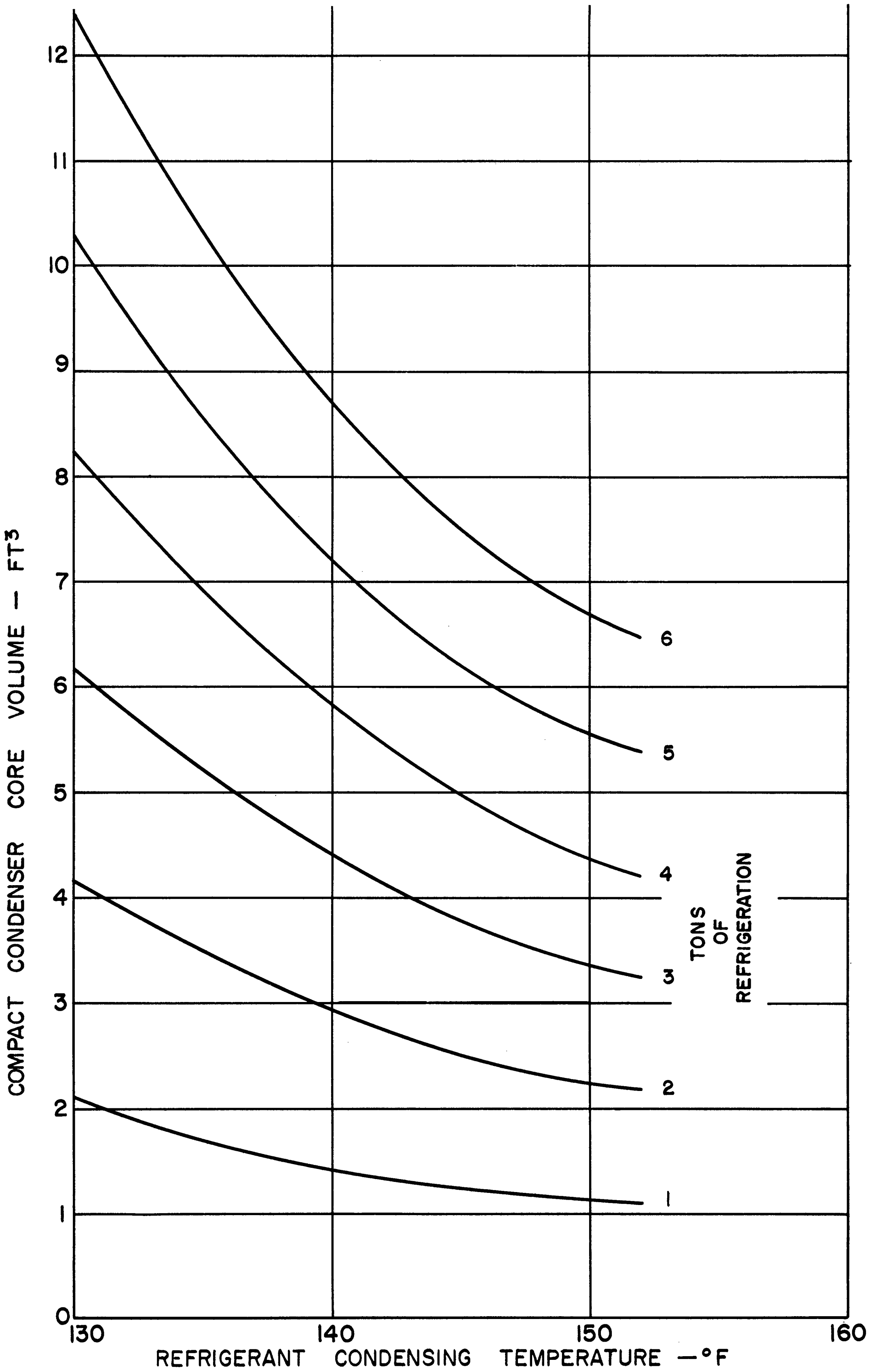

Figure 6. Condenser core volume vs refrigerant condensing temperature; U = 4 Btu/(hr-ft^2-°F).

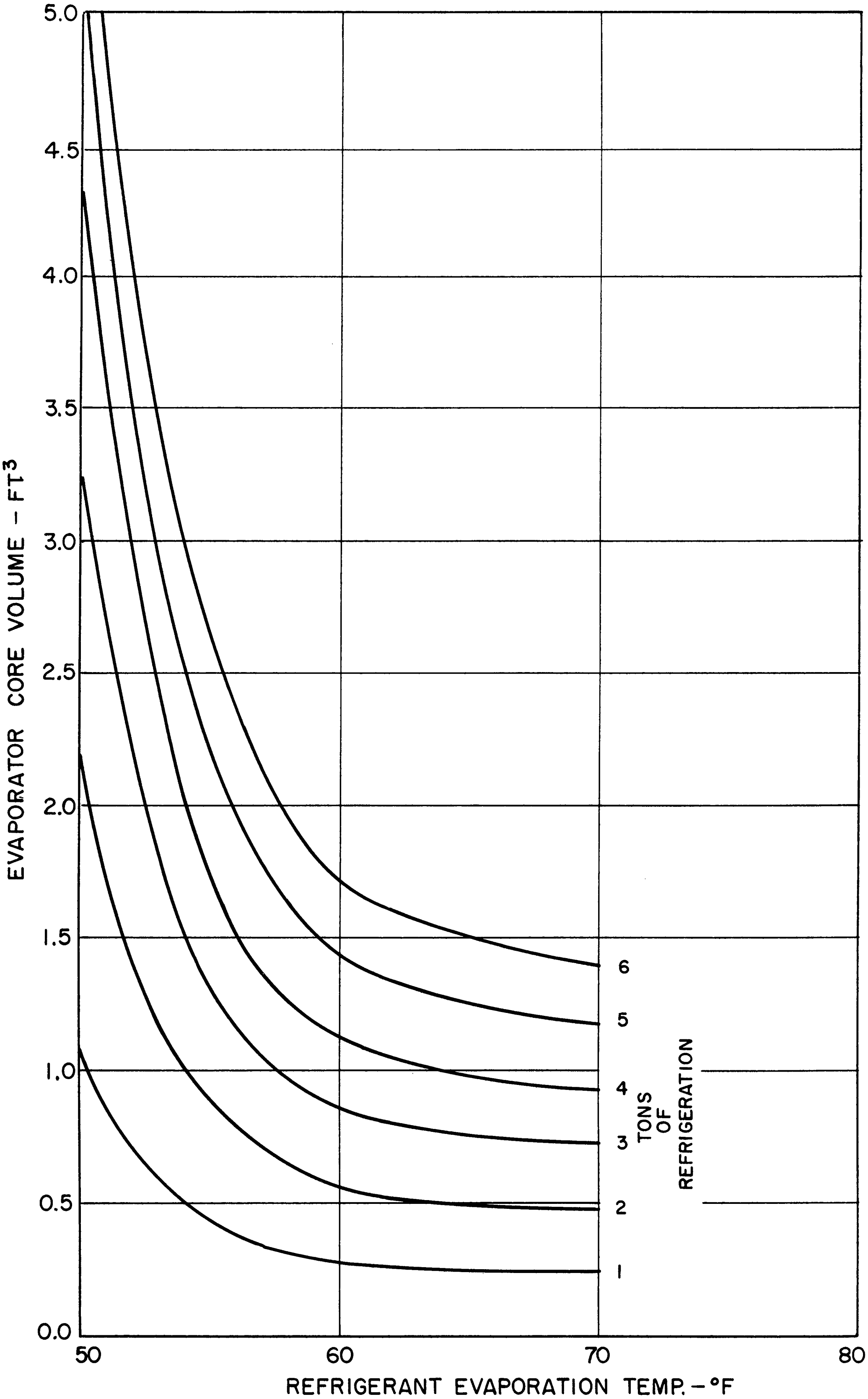

Figure 7. Evaporator core volume vs refrigerant evaporation temperature; U = 10 Btu/(hr-ft^2-°F).

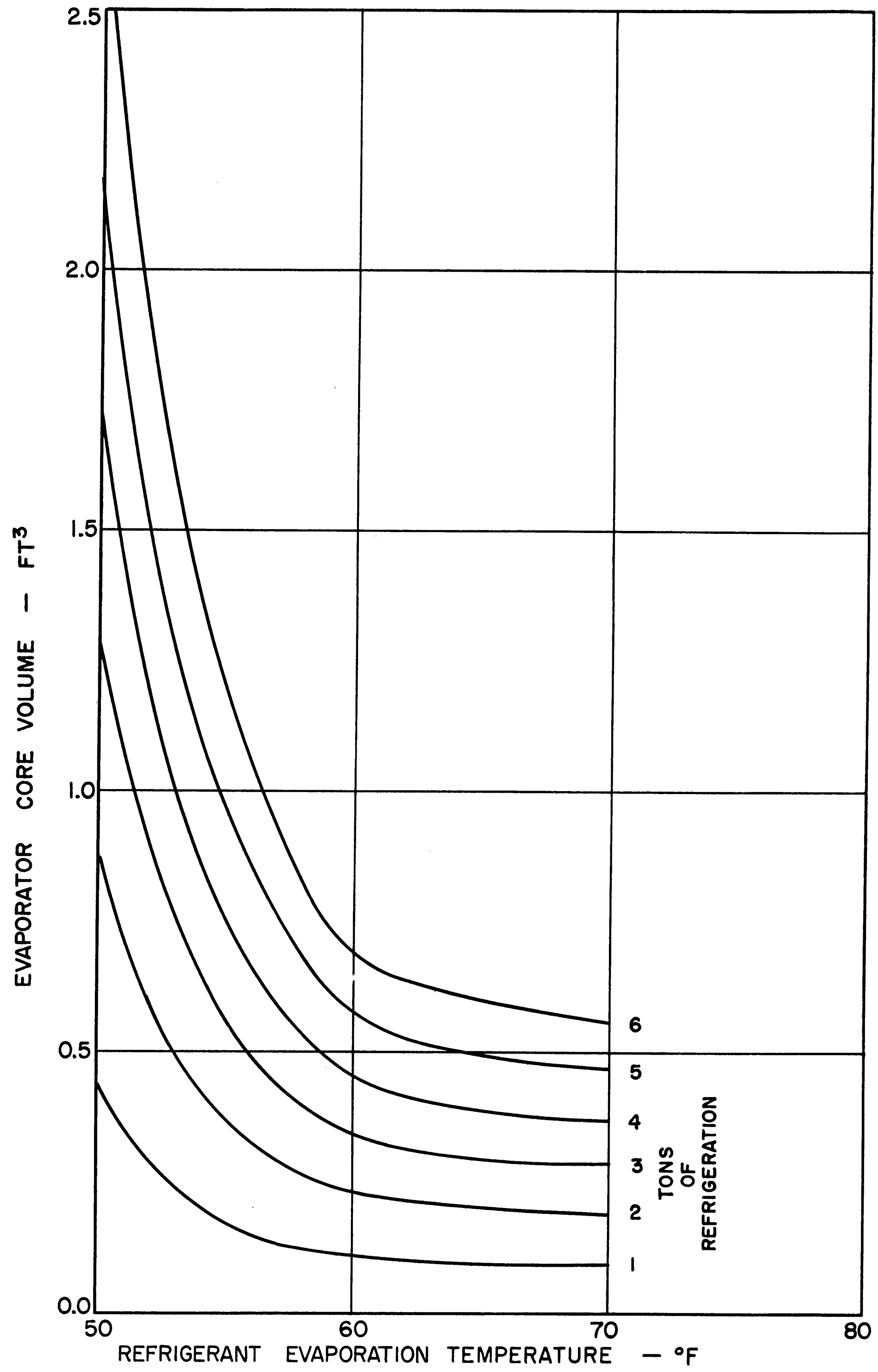

Figure 8. Evaporator core volume vs refrigerant evaporation temperature; $U = 4$ Btu/(hr-ft^2-°F).

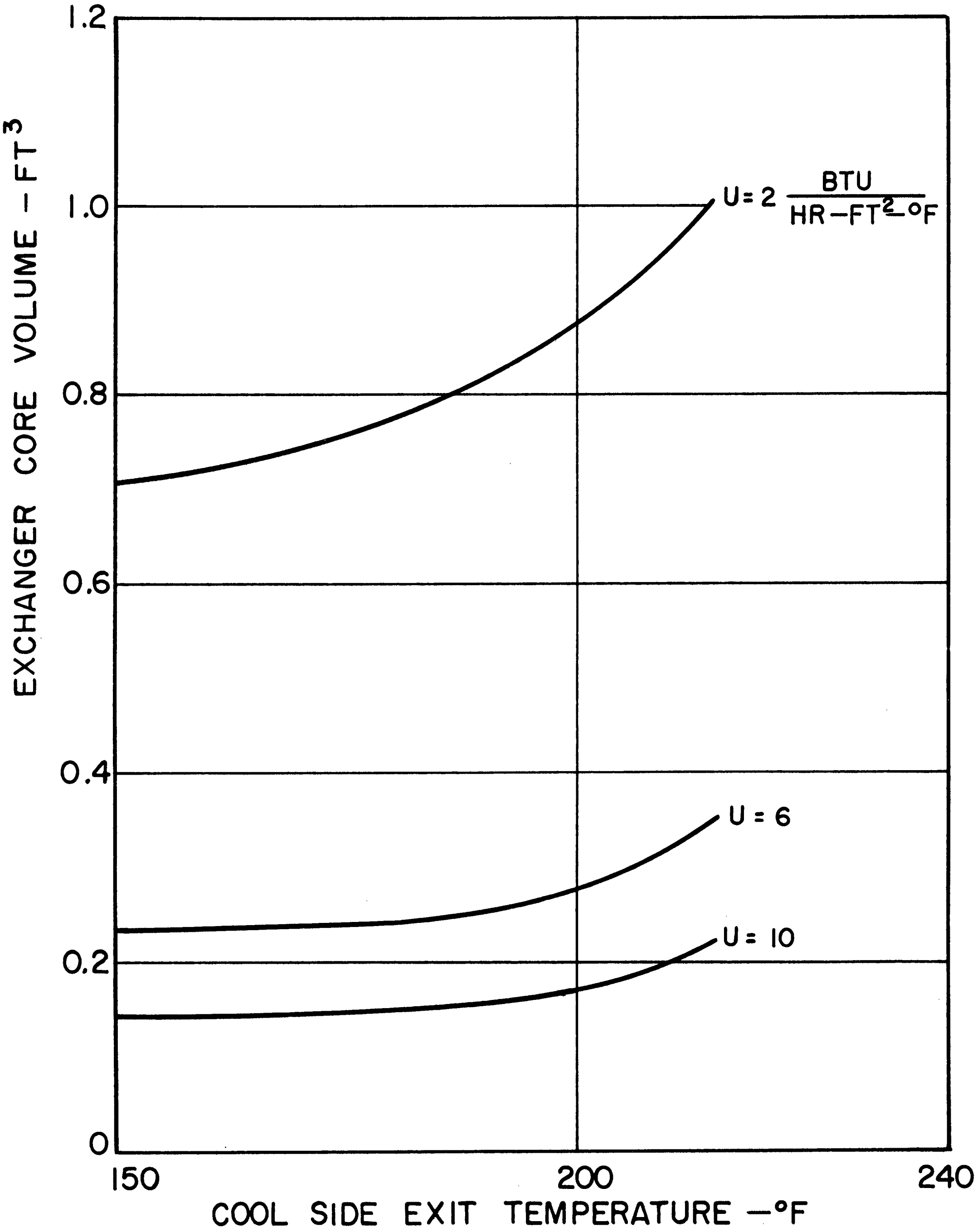

Figure 9. Air-cycle exchanger core volume vs cool side exit temperature; pressure ratio of compressor = 3; compartment air rate = 350 cfm; cool side entrance temperature = 120°F.

Enough dust may accumulate to choke the passageways and cause greater pressure loss than was designed for through the exchanger. More serious than this, though, from the point of view of the system as a whole is that it might not take much dust accumulation on the surfaces of the exchanger to appreciably change the film heat-transfer coefficients and, thus, the overall coefficients.

The tank operates under conditions so that the only cooling medium readily available for the condensers, or for the aftercooler in air compression, is the ambient air. In the case of exchangers on the interior of the tank it might be difficult to get the full benefit of the cooling system due to the presence of heated air leaving the cooling side of the exchanger. The suggestion follows then that it might be better to place all heat-exchange surfaces on the exterior of the tank, this seeming the most logical place if ambient air is the cooling medium. Two immediate difficulties arise, however, as a result of this solution. First, unless the exchanger is shielded from the direct impingement of solar radiation, any cooling effect of the ambient air is largely or completely nullified. Secondly, an exterior exchanger would be quite vulnerable to damage or complete destruction by enemy rifle fire or shrapnel, or by impact of tree branches, stumps, and other obstructions.

Another problem would be to obtain heat transfer by greater forced convection than is caused by the normal movements of the tank. The mounting of suitable fans on the outside of the vehicle for this purpose might prove to be impractical.

D. DUCTING

Here again the important factors to be considered are space requirements and power requirements to overcome friction losses in the ducts.

1. Space Requirements.—Configuration, as opposed to volume, becomes more important as far as space is concerned. On the other hand, there is tolerance for a greater choice of configurations than in any other component; by this is meant not only the transverse geometry of the ducts themselves, but also the variety of paths they follow throughout the vehicle. The greatest single problem in designing the ducts is in the accomplishment of cross-over between the two parts of the tank that are movable with respect to each other, the turret, and the hull. In this, the most feasible method thus far suggested is to bring the duct up through the center of the turret floor. In any event, it would seem that a properly designed duct system can have the major effect in reducing the heating or cooling requirements in the tank, and thus in keeping space and power requirements for other components to a minimum. The report on the phase of this project

concerned with ventilation flow patterns points out the inadequacy of the ventilating and heating system as presently used in the M-47 medium tank. This inadequacy could be remedied simply by a redesigned duct system.

2. Power Requirements.—The power requirement for overcoming friction losses in the ducts would be comparatively small relative to that required for compression. Friction losses per 100 feet in round ducts for air flow up to 1000 cu ft/min attain values up to 2 inches of water for velocities up to 1000 feet per minute and for duct diameters from 22 inches down to 1-1/2 inches. Fairly similar losses are experienced in rectangular-shaped ducts. Depending on total length of ducts and numbers of elbows, changes in cross-sectional area, and other constrictions, as well as on volume and velocity flowrates, the power required for this portion of a system will be at some value up to about one horsepower. Thus, in the overall system, power requirements for the duct losses are not a major item of consideration.

3. Other Considerations.—The use of ducts for routing heated or cooled air to specific points in the tank creates the need for insulation of these ducts. Unless such insulation is used, the transfer of heat across the duct walls probably would result in some cancellation of the beneficial effects desired from the system. It is possible that the use of sufficient insulation to reduce heat transfer to a reasonable level will result in difficulties in fitting the ducts into proper spaces.

As stated above, there is a problem in connecting ducts between the hull and the turret because of the need for movement of one portion of the tank relative to the other. If a system of flexible hose is planned for use in this connection, or for that matter if flexible hose is used anywhere, there is the possible event of the hose pinching or crimping. The results of this mishap may be twofold: the supply of conditioned air might be cut off from some portion of the vehicle, and the fault might jam some vital moving part of the tank at a critical moment.

E. HEATING METHODS

There are three general methods that might possibly be used in heating the crew compartment of the tank. Combustion-type heaters utilizing fuel directly in the production of heat may be used, as they are being used in the M-47 medium tank and other vehicles; waste heat from the main engine may be carried through the crew compartment before being rejected to the surroundings; or, in the case of centrifugal compression, the heat generated by the compressor may be fed directly into the crew compartment instead of being carried through an exchanger to the surroundings. The combustion heater has been studied quite thoroughly and is in practical use by the army; a discussion of the combustion heater is made in the report on the heat-transfer-characteristics phase of this project. A complete study of a method of utilizing engine-waste heat has been made by the Remsel Corporation, Chicago, Illinois, and a report on that project is in the files at the Detroit Arsenal. This section, therefore, will be concerned only with a short discussion of the practicability of using compression heat from a centrifugal compressor.

The temperature rise of a gas going through a centrifugal compressor is completely independent of the pressure rise across the compressor. It depends only on the velocity of the gas leaving the impeller tips. Some of the work done on the gas by the impeller will result in the creation of a pressure rise and some of it will be used to overcome losses in the machine. The temperature rise, however, is not altered in any way by the mode of distribution of the imposed work between pressure rise and friction losses. If there were no pressure rise at all, it would simply be that losses were extremely high, and the temperature would still be the same as though there were an appreciable pressure rise and few losses.

The work done by an impeller on a gas passing through it is theoretically wv^2/g, where w is the mass rate of flow, v is the tip tangential velocity of the impeller, and g is the gravitational constant. The actual amount of work absorbed by the gas is, however, somewhat lower than this theoretical amount. The theoretical value assumes that the gas leaving the impeller is at the same linear velocity as the tangential velocity of the impeller tips, but this is not true for a practical machine. Some slippage occurs between the impeller surface and the air, and the velocity of the air leaving the impeller is actually lower than the tip velocity.

It is shown in textbooks[3] dealing with centrifugal compression that the total temperature increase across an impeller is constant for a given tip speed and is equal to v^2/JgC_p, where v is again the tip tangential velocity of the impeller, J is the mechanical equivalent of heat, g is the gravitational constant, and C_p is the specific heat for the gas and is assumed constant over the range of compression. If the equation is expressed in English units and the gas is air, having a constant value of C_p equal to 0.24 Btu/(lb-°F) over the range of the pressure ratio, then

$$\Delta T = T_2 - T_1 = \frac{v^2}{6040} \quad \text{(in degrees Fahrenheit).}$$

Thus, in the ideal case, if the tip speed of the impeller were 1200 ft/sec (a reasonable figure) and if the temperature of the air going into the compressor were -30°F, the theoretical temperature of the air leaving the would be

$$T_2 = \frac{v^2}{6040} + T_1$$

$$= \frac{\overline{1200}^2}{6040} + (460-30)$$

$$= 668° \text{ Rankine}$$

$$= 208° \text{ Fahrenheit} \quad .$$

The actual velocity of the air leaving the impeller would be at some figure less than the tangential velocity, v, of the formula above, and the actual temperature of the air would also be at some lower figure. The point of the discussion, though, is to show that a considerable rise in air temperature can be attained by use of the centrifugal compressor. If, then, a centrifugal compressor of small enough capacity for use in a medium-tank air-cycle refrigerator can be developed, here within the same system is also the possibility of having a heater simply by cutting the exchanger and work-loaded expansion turbine out of the system.

F. INSULATION

The advantages to be gained through the use of some insulation on the walls of the tank are discussed in the report dealing with the heat-transfer characteristics of the M-47 medium tank. The discussion there, however, does not go into the practicability of applying such insulation. It is the purpose here, then, to discuss briefly the mechanical feasibility of applying insulating material as desired.

Ideally, any material to be considered, in addition to its insulation qualities, should possess pliability, ruggedness against abrasion and crushing, nonflammability, imperviousness to moisture, and ease of application. The usual materials have, of course, the good insulating qualities, but unfortunately none that has been studied could be considered practical for use; they all failed by a wide margin to meet one or more of the qualities listed above. Each of the qualities listed is important enough that an extremely poor characteristic in a material is considered grounds for rejection from further thought.

Just prior to completion of this report, however, a new development was disclosed concerning a sprayable insulation[13]. The product, a resinous liquid, is manufactured by the Insul-Mastic Corporation of America under the trade name "Poly-Cell". It is stated that the liquid is sprayed on the surface to be insulated and at first resembles a coat of varnish. After a few minutes it foams and swells to a thick, airy cushion and after foaming has ceased (about 15 minutes) a normal looking, semirigid insulation is left. The company claims that the insulation can be applied to any desired thickness in one coat. The K factor is given as "24 per inch of thickness", density is slightly more than 2 lb/cu ft, and known temperature limits are -40° to 225°F. The article referenced does not explain what is meant by "K factor", but it is assumed that the term applied to conductivity. On this basis the conductivity, expressed on the usual basis of a foot thickness, is 2 Btu/(hr-ft^2-°F/ft). Further claims are that the substance contains no volatiles and is nonflammable; it can be colored to almost any desired color; it cuts application costs since no hand work is needed in fitting and fastening; and it forms a natural vapor or weather barrier because of its structural strength and absence of joints to expand and contract.

If all claims enumerated are well-founded, this is a material in which a certain amount of insulation value is sacrificed in favor of obtaining other characteristics that are extremely favorable for use in combat tanks. It would seem reasonably worthwhile to investigate the possibilities of this material, in view of the advantages to be gained by insulation.

IV. CONCLUSIONS

1. Based on safety requirements most of the refrigerants commonly used for vapor compression cycles, except the various Freons, should be eliminated from consideration in a combat tank cooling systems.

2. Based on the additional factors of cubic displacement, operating pressures, and power requirements, Freon-21 seems to be the best refrigerant of those studied for use in a vapor-compression cycle.

3. The "open" air cycle presents enough advantages so that, in spite of the large amount of power needed for its use, it is worthy of at least a preliminary design and development study.

4. For the needs of combat tank cooling, centrifugal compression is not practical for use in a vapor-compression cycle. Reciprocating compression is considered better for use than rotary compression because of lesser lubrication and maintenance problems.

5. The air cycle will be worth consideration only if centrifugal compression can be used. The design and development of a centrifugal compressor of small enough capacity for use in the tank will be, at best, difficult to attain.

6. With respect to heat exchangers, space requirements and pressure drops will present relatively minor problems. In the interests of conserving interior space and obtaining better heat transfer, it would seem preferable to mount the exchangers on the exterior side of the tank. An intriguing possibility is that of mounting the exchangers between the hull and tracks of the tank and of obtaining power directly from the track suspension system for forced convection on the cold side of the exchanger when the tank is in motion.

7. Dust accumulation presents the largest single "external" problem in the consideration of the overall system. It can cause abrasion, erosion, clogging (causing decreased efficiency), and lower rates of heat transfer; and, of course, more directly it will cause increased perspiring and breathing discomfort of crew members.

8. Careful planning of the duct system can be the biggest single factor in keeping both refrigeration and heating requirements to a minimum. The chief problems in planning the duct system lie in achieving the crossover of ducts between the hull and turret, and in maintaining nearly equalized velocities at all duct exits despite varying distances and paths from the cooling or heating source to the distribution point.

9. There seem to be three feasible methods of heating: utilization of combustion heaters (of the type now used or some variation), waste heat from the engine, and compression heat if centrifugal compression of air is used. Of these, only the combustion-type heater can be used to activate the vehicle in extremely cold weather after prolonged shutdown of <u>all components</u> of the vehicle. For conditions where only the main engine has been shut down, sufficient heat for starting can be maintained by keeping either a combustion heater or centrifugal compressor running. The utilization of waste heat is, of course, only practicable when the main engine is running.

10. The best heating system for a combat tank would make use of combustion or compression heat, augmented by the use of waste heat from the main engine.

11. Insulation on the walls of the tank would greatly assist in solving the crew comfort problem. However, the practicability of installing and maintaining such insulation is questionable. The newly introduced sprayable insulation presents a possibility that insulation can be used.

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APPENDIX I

POSSIBLE USE OF CENTRIFUGAL VAPOR COMPRESSION

The basis used is the production of three tons of refrigeration. It is assumed that the impeller diameter should be no greater than 12 inches in order to conform to space needs for the unit. Single-stage compression is also assumed. In general the head normally handled by centrifugal compressors lies in the range from 2500 to 75,000 feet of fluid. Rotative speeds in present designs may be as high as 30,000 rpm and tip tangential speeds up to 1300 to 1600 ft/sec are permissible. There are several other considerations in centrifugal design, but the one on which the possibility of vapor compression is most likely to fail is in the need for passages (as in the impeller) that are of such size as to preclude excessive aerodynamic losses.

The following calculations deal with theoretical possibilities for Freon-113 ($C_2Cl_3F_3$—molecular weight = 187.4).

Assume: refrigerator temperature = 40°F, pressure = 2.66 psia,
condenser temperature = 120°F, pressure = 15.4 psia.

1. Head Required

$$H = \frac{Z\, R_g\, T_1 \left[\left(P_2/P_1\right)^{\frac{k-1}{k}} - 1\right]}{\left(\frac{k-1}{k}\right)}$$

where R_g = gas constant for Freon-113 = 8.25
T_1 = suction temperature = 40°F = 500°R
P_2/P_1 = compression ratio = 15.4/2.66 = 5.8
k = ratio of specific heats = 1.07
Z = compressibility factor $\cong$ 0.97.

$$\therefore H = \frac{[0.97][8.25][500]\left[(5.8)^{0.065} - 1\right]}{0.065}$$

$$= 7450 \text{ ft-lb/lb.}$$

2. Impeller Rotative Speed

$$N = \frac{1300}{D}\sqrt{\frac{H}{\mu}}$$

where N = rotative speed in rpm
D = impeller diameter = 12 inches
H = head = 7450 feet
μ = pressure coefficient = 0.55 (assumed).

$$\therefore N = \frac{1300}{12}\sqrt{\frac{7450}{0.55}}$$

$$= 12600 \text{ rpm.}$$

3. Impeller Tip Tangential Speed

$$v = \frac{\pi DN}{720}$$

$$= \frac{(3.14)(12)(12600)}{720}$$

$$= 660 \text{ ft/sec.}$$

4. Impeller Exit Axial Length

The circumferential length of the impeller wheel has been fixed by the diameter of 12 inches. Of the total circumferential length, a certain amount has to be subtracted for the area that is taken up by the cross section of the impeller vanes. The axial length of the impeller passage is dependent on the volumetric flow to be designed through the passages when other dimensions are fixed. For estimating the volumetric flow at the impeller exits it is required that the pressure and temperature of the gas at the exits be estimated.

The full pressure rise for the compressor is not realized at the impeller exits, since the velocity head has not yet been converted to pressure head until after this point. It is usually estimated that at these points about one half the expected static pressure rise has taken place, and also that about one half the expected temperature rise has occurred.

Then

$$P_e = P_1 + \frac{P_2 - P_1}{2}$$

where P_e = pressure at impeller exit
P_1 = suction pressure = 2.66 psia
P_2 = discharge pressure = 15.4 psia.

$$\therefore P_e = 2.66 + \frac{15.4 - 2.66}{2} = 9.03 \text{ psia.}$$

$$T_e = \frac{1}{2}\left[\frac{v^2}{6040}\right] + T_1$$

where T_e = temperature at impeller exit
T_1 = suction temperature = 500°R
v = tip velocity = 660 ft/sec.

$$\therefore T_e = \frac{1}{2}\left[\frac{\overline{660}^2}{6040}\right] + 500 = 536°\text{R} = 76°\text{F} .$$

With the pressure and temperature at the impeller outlet fixed, it is now possible to determine the volumetric flow at that point, by using the relation:

$$Q_e = \frac{w\ R_g\ T_e}{144\ P_e}$$

where w = 11.73 lb/min (for 3 tons of refrigeration)
R_g = 8.25
T_e = 536°R
P_e = 9.03 psia.

$$\therefore Q_e = \frac{(11.73)(8.25)(536)}{(144)(9.03)}$$

$$= 40 \text{ ft}^3\text{/min}$$

$$= 0.67 \text{ ft}^3\text{/sec.}$$

The relation for finding the impeller exit area is

$$A_e = [\pi D - nt]\, b$$

where $A_e = \frac{Q_e}{v_e}$ = impeller exit area (sq in.)
D = impeller diameter = 12 inches
n = number of vanes = 20 (assumed)
t = vane thickness = 0.125 inch (assumed)
v_e = 660 ft/sec
Q_e = 0.67 ft^3/sec
b = axial length of impeller exit (inches)

$$\therefore b = \frac{[144][0.67]}{[(3.14)(12) - (20)(0.125)][660]}$$

$$= 0.004 \text{ inch.}$$

Based on the assumed specifications the impeller passages at the exit would each have an area of about 1-3/4 inches along the circumference by about 0.004 inch axially. A reasonable radial length for the impeller passages might be 3 to 4 inches. It can be shown that impeller passages of these dimensions would result in aerodynamic losses so that the compressor efficiencies would be ridiculously low.

APPENDIX II

CALCULATIONS FOR IMPELLER AERODYNAMIC LOSSES

The impeller wheel diameter was assumed at 12 inches and the impeller inlet vane tip diameter at 5 inches. This makes a radial passage length of 3.5 inches, or 0.292 foot. The number of vanes and their thicknesses were taken to be such that the impeller circumferential free exit length was 35 inches, and it was assumed that there are 20 vanes and passageways. Each passage has, then, a circumferential exit length of 1.75 inches and a circumferential inlet length of 0.73 inch. This makes an average circumferential length for the impeller passage equal to 1.24 inches, or 0.103 foot. For purposes of the calculations the parameters that were varied were (1) impeller axial width, (2) average air velocity through the impeller passage, and (3) average circumferential passage length.

Numerical values for viscosity and density were obtained from the

Chemical Engineers Handbook[2], and the friction factor chart as shown in Figure 125, page 140, of Unit Operations[5] was used.

The Reynold's number was determined by the relation

$$R_{e(avg)} = \left[\frac{4\ R_H\ v\ \rho}{\mu}\right]_{avg}$$

where R_e = Reynold's number
R_H = hydraulic radius
v = arithmetic average gas velocity through the passage
ρ = arithmetic average gas density through the passage
μ = arithmetic average gas viscosity through the passage.

Assuming arithmetic averages in the above equation is not absolutely correct, since it implies linear velocity distribution through the passage. The latter condition does not hold true. On the other hand, they are far easier to use than some other mean values and the errors introduced in the final answers are not of great significance, in view of all the other assumptions made.

Having found the Reynold's number, a value for friction factor was found from the chart of Figure 125[5] with the assumption of smooth passage surfaces.

The required quantities were now known so that the work expended in overcoming aerodynamic friction could be found with the relationship

$$lw_f = \frac{f\ v_{avg}^2\ L}{8\ g_e\ R_H}$$

where lw_f = friction-loss
f = friction factor
L = radial length of impeller passage
g_e = gravitational constant
v_{avg} and R_H are as defined above.

To find the power loss, the mass flow associated with the impeller dimensions chosen must be known. This was found by methods similar to those shown in Appendix I, above. Power loss was then found by using the relation

$$HP_f = \frac{(lw_f)\ W}{33000}$$

where W = mass flow of gas.

Additional assumptions that were made during the course of the calculations were those concerning impeller entrance velocity, pressure, and temperature.

APPENDIX III

CORE-SIZE ESTIMATION FOR COMPACT HEAT EXCHANGERS

Calculations for a rough estimation of core sizes for compact exchangers were based on the report by Kays, London, and Johnson.[8] The method used in this project was greatly simplified from the methods reported. This simplification was done by assuming various values of U, the overall heat-transfer coefficient, whereas in their method the authors calculated approximate values of this parameter. In addition to the assumed values of the overall coefficient, the entering and leaving temperatures of both streams in the exchanger were specified; and it was further specified that the exchanger would be of the cross-flow, unmixed fluid type having a heat-exchange-area to core-volume density of 250 sq ft/cu ft on both sides.

By means of an energy balance, assuming an adiabatic exchanger, the coolant flow requirement was determined for the conditions imposed. From this the capacity rates, defined as the hourly mass flowrate multiplied by the specific heat, were found for each stream, and with a knowledge of these an exchanger effectiveness could be determined. Knowing the exchanger effectiveness and the capacity-rate ratios between the two streams it was possible to obtain values for the "number of exchanger heat-transfer units" (NTU) from charts in the reference report. These charts consist of plots of heat-exchanger effectiveness versus NTU for lines of constant capacity-rate ratios, a separate chart being used for the various possible flow arrangements, such as concurrent and countercurrent flow.

The "number of exchanger heat-transfer units" (NTU) is defined by

$$NTU = AU/C_{min}$$

where U = overall heat transfer coefficient

A = same heat-transfer area as is used in the definition of U

C_{min} = the capacity rate of the hot or cold streams, whichever is lower.

With values of NTU, C_{min}, and U fixed, the equation was solved for the required heat-transfer area, and this, together with a specified ratio of transfer area to core volume, enabled an estimation of the required exchanger core volume.

TABLE 1

FLAMMABILITY AND TOXICITY OF REFRIGERANT FLUIDS
(Reference 1, page 205)

Refrigerant	Toxicity—Lethal or Serious Injury (refrigerant in air)			Flammable or Explosive Limits
	Duration of Exposure, hr	Percentage by Volume	Pounds per 1000 Cu Ft	Percentage by Volume
Propane	2	37.5-51.7	42.8-58.5	2.3-7.3
Freon-22				Nonflammable
Ammonia	1/2	0.5-0.6	0.221-0.256	16.0-25.0
Freon-12	2	28.5-30.4	89.6-95.7	Nonflammable
Methyl chloride	2	2-2.5	2.62-3.28	8.1-17.2
Isobutane				1.8-8.4
Sulfur dioxide	1/12	0.7	1.165	Nonflammable
Butane	2	37.5-51.7		1.6-6.5
Freon-114	2	20.1-21.5	90.5-96.8	Nonflammable
Freon-21	1/2	10.2	27.1	Nonflammable
Freon-11	2	10	35.7	Nonflammable
Methyl formate	1	2-2.5	3.12-3.9	4.5-20.0
Methylene chloride	1/2	5.1-5.3	11.25-11.7	Nonflammable
Freon-113	1	4.8-5.2	23.3-25.2	Nonflammable
Carbon dioxide	1/2 to 1	29-30	33.2-34.3	Nonflammable

TABLE 2

OPERATING CHARACTERISTICS OF REFRIGERANTS BETWEEN THE TEMPERATURE LIMITS FROM 120°F TO 40°F
(Carnot horsepower requirement is 0.75.)

Refrigerant	Vapor Volume per Ton of Refrigeration, cfm	Evaporator Pressure, psia	Condenser Pressure, psia	Horsepower per Ton of Refrigeration
Air (assuming feedback of power to compressor)	152.4 152.4	(Noncondensing) (Noncondensing)	Pressure ratio = 4 Pressure ratio = 3	2.05 2.76
Air (assuming no feedback of power)	152.4 152.4	(Noncondensing) (Noncondensing)	Pressure ratio = 4 Pressure ratio = 3	16.61 12.65
Freon-11	17.1	7.0	33.2	0.94
Freon-12	3.4	51.7	171.8	0.99
Freon-21	9.75	12.3	55.8	0.88
Freon-22	2.15	83.7	277.3	1.00
Freon-113	41.7	2.66	15.4	0.95
Freon-114	9.95	15.2	63.4	0.96

TABLE 3

IMPELLER PASSAGE AXIAL WIDTH FOR THEORETICAL COMPRESSION OF ENOUGH REFRIGERANT TO PRODUCE THREE TONS OF REFRIGERATION, WITH CONDENSER TEMPERATURE AT 120°F AND EVAPORATOR TEMPERATURE AT 40°F

Refrigerant	Impeller Passage Axial Width, in.
Freon-11	0.0018
Freon-12	0.0006
Freon-21	0.0009
Freon-22	0.0003
Freon-113	0.0041
Freon-114	0.0014

TABLE 4

SPACE REQUIREMENTS FOR SINGLE-STAGE ROTARY AND RECIPROCATING COMPRESSORS

Actual Free Air Delivery, cfm	Rotary Compressors (based on envelope dimensions), cu ft	Rotary Compressors (theoretical smallest size), cu ft	Reciprocating Compressors (based on envelope dimensions), cu ft
3			1.46
16			2.29
31	5.00	3.05	4.01
43	5.00	3.05	6.01
74	7.34	4.23	
109	8.84	4.41	
127	8.84	4.41	
152	14.7	5.46	
194	14.7	5.46	
228	14.7	5.46	