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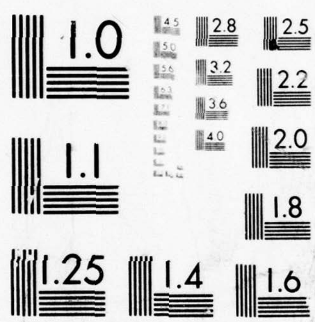

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CONCEPTUAL DESIGNS FOR COMBINED ENVIRONMENTS RELIABILITY TEST FACILITY

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DAYTON T. BROWN, INC.
ENGINEERING AND TEST DIVISION
CHURCH STREET, BOHEMIA, LONG ISLAND
NEW YORK, NEW YORK 11716

JUNE 1977

TECHNICAL REPORT AFFDL-TR-77-3
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AIR FORCE FLIGHT DYNAMICS LABORATORY
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This technical report has been reviewed and is approved for publication.

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Various conceptual designs for CERT facilities were formulated in terms of technical requirements. Two conceptual designs were selected by the Air Force for an economic analysis which included tabulating initial costs, operating costs and maintenance costs.

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PREFACE

Initial data obtained by the Air Force indicates that a new reliability testing method (CERT Concept) provides correlation between field failure rates and failure rates produced in the laboratory. The CERT Concept utilizes controlled interaction of random vibration, chamber air temperature, humidity and altitude, cooling air flow rate, temperature and humidity. This report deals with the task of extending the design of the air Force's chamber plus developing two additional concepts using state of the art techniques, to meet CERT requirements. In using state of the art techniques, only existing commercially available components were to be used and no new components would be designed. The total project was divided into five distinct tasks. Task I began with a meeting between representatives of Dayton T. Brown, Inc. and the Air Force at Wright Patterson Air Force Base. Task I consisted of a source search for information on environmental simulation technology. The information obtained and Dayton T. Brown, Inc. Testing Laboratory's experience in this field provided the basis for the remainder of the project. Task II consisted of a technical development of the Air Force design and two additional conceptual designs. Task III consisted of an economic analysis of two conceptual designs selected from the three in Task II by the Air Force. Task IV consisted of a cost reduction analysis. The final phase, Task V, consisted of a facility scan for existing test chambers which could be modified to CERT requirements.

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NOMENCLATURE

F - Venturi Meter (Flow)

T - Thermocouples (Temperature)

H - Humidity Meter (Specific Humidity)

P - Pressure Transducer (Vacuum)

<u>Valves</u>

⊗ - Manually Adjusted (H_2O)

△○ - Solenoid - Actuated ON-OFF Valve (LN_2)

SC - Speed Control for Blower

EC - Electric Controller (Heaters)

PC - Electro-Mechanical (Air Pressure)

HC - Electro-Mechanical (Humidity)

RC - Solenoid Valve (Refrigerant)

FC - Electro-Mechanical Damper

VC - Vibration Control

SECTION I

INTRODUCTION

TASK - II

1.1 Purpose of the CERT Concept

Aircraft utilization of avionics equipment has increased at an exponential rate over the past ten years. As a result, the need for reliability of the equipment has become a key consideration in the effectiveness of the aircraft. Estimated values of reliability are usually derived from other similar equipment or analytically from that of key components. This method has proven to produce reliability rates that are usually 3-8 times better than those really experienced in the field. The most effective way to predict reliability should be through testing. However, the reliability of avionic equipment cannot be predicted accurately with the currently available test methods. This discrepancy is understandable in view of the lack of correlation between actual flight environment and the laboratory conditions conventionally applied to the equipment. Combined environments reliability testing (CERT) attempts to duplicate the actual flight environment (both cooling air flow and bay conditions) in the testing chamber. In order to establish a realistic test program, studies were made of avionics environmental conditions during aircraft flight missions in tropic and arctic environments. As a result, avionics bay temperature, altitude, humidity, random vibration and cooling air rate, temperature and humidity parameters were established. From this the Air Force established specifications for facility performance, as follows:

Parameter	Range
Facility size	house 3 separate test items individually 3x3x3 feet, 250 lbs. each, 5 kW heat load each
Vibration	vibration as shown in figure 8. Random 0 - 0.4 $\frac{G^2}{Hz}$ over 5 - 2000 Hz. Controls per MIL-STD-810C, Method 514.2
Chamber temperature	-65 to 300°F @ $\pm$ 40°F/min. -65 to 200°F @ $\pm$ 120°F/min. for a $\pm$ 40°F step and $\pm$ 40°F/min. between -65 to 200°F
Cooling air flow	each equipment 0 to 10 pounds air/min. @ $\pm$ 10 pounds/min./min.
Altitude	0 - 70,000 feet @ $\pm$ 40,000 ft./min.
Electrical	supply 15 kW of power to equipment with parameters specified in MIL-STD-704A
Humidity	0.0005 to 0.04 $\frac{\text{pounds } H_2O}{\text{pound air}}$ @ $\pm$ 0.04 pound H_2O/pound air/min.

In an effort to evaluate the CERT concept as a viable method for reliability prediction, the Air Force Flight Dynamic Laboratory modified an existing facility to approach CERT requirements and used it to perform reliability testing. The results indicated good correlation between field failure modes, rates, and test data.

1.2 Program Description

As a result of the data obtained from the CERT evaluation, Dayton T. Brown, Inc. was contracted to perform the following evaluation. Since the Air Force's CERT facility does not meet present CERT requirements, Dayton T. Brown, Inc. was to extend the facility by modification of components to meet these requirements. Two additional concepts using existing state of the art methods and commercially available equipment were also to be developed to meet CERT requirements. This report deals with the engineering evaluation performed and the resulting concepts and extension that resulted.

The Air Force was to select two concepts from the three discussed and a resulting cost analysis would be performed by Dayton T. Brown, Inc. In addition, Dayton T. Brown, Inc. would identify areas where small reductions in facility performance requirements would have significant impact on costs and identify the cost savings. The total investigation should then allow the most cost effective engineering concept to be selected for a CERT facility.

1.3 Statement of Design Objectives

In order to organize the approach to the design of the CERT facility, certain design objectives were formulated. One of the initial prime objectives was to provide three conceptual designs with an approach to air delivery and conditioning that was uniquely its own. Each concept could be independently analyzed and evaluated relative to the other conceptual designs. It was realized, however, that the extension of the existing chamber and the formulation of two additional concepts, had design areas with similar problems that would require solutions utilizing a similar approach. For example, the introduction of humidification into the chamber by the environmental cooling air and the requirement for independent humidity schedules of the cooling air and chamber environment made any attempt at static conditioning of the chamber unfeasible. Although the vacuum pump during operation would remove a certain amount of injected moisture from the chamber, it could not be depended upon to do so due to possible intermittent operation. Furthermore handling moisture in a vacuum pump has adverse effects upon its altitude capabilities. Due to these conditions, it was realized that introduction of ventilation with the proper temperature and humidity schedules would be an approach common to all three of the conceptual designs. Once this decision had been made, alternative methods of supplying and circulating this ventilation air in the chamber were investigated.

In a similar fashion it was realized that any of the three conceptual approaches required two basic conditioning loops, i.e., the cooling air and the chamber air. Regardless of the approach to the conditioning of the two independent temperatures, pressure and humidity schedules, the required air quantity would be delivered in ducting of some type and subjected to convective heat transfer with any material through which or over which, it passed. A common design objective was that material selection would attempt to minimize the convective heat transfer phenomena by utilizing light gauge material with low specific heat capacity.

In the analysis of the cold conditioning of the cooling air and chamber air, a specific design objective was the investigation into the utilization of mechanical refrigeration. This was due to the realization of the high costs of liquid nitrogen.

Finally, it was desired that for any type of equipment which was selected, the design would be optimized in view of performance and system costs. This led to an extensive analysis for the selection of the chamber closed loop fan system and helped specifically in the selection of a refrigerant for the cascade mechanical refrigeration system.

1.4 Approach to Organizing and Solving the Problem

In the early stages of the development of three conceptual designs for the CERT chamber, it was decided to evaluate the performance requirements at the maximum and minimum points of each independent variable. Calculations for environmental air were done for airflows of 10.0 lbs./min. and 0.5 lb./min. and temperatures of -65°F and 200°F. Chamber air calculations were done for sea level altitude, for 70,000 feet, and for temperature conditions of -65°F and 300°F. The specific humidity values for environmental air and chamber air were also selected at their high and low points, i.e., 0.04 and 0.0005 pound of water per pound of air. At any given design point therefore, the combination of the above variables which defined the "worst case" was selected. The result was a design range defined in terms of relevant variables for each piece of equipment.

It was also realized in the early stages of development, that any particular approach should utilize state of the art techniques. This was specifically mentioned in section F, paragraph 2.1, in the statement of work. The final selection therefore, of a particular conceptual design could conceivably lead to a final design which would incorporate currently available equipment in its systems.

Another aspect of the design approach utilized, was the isolation of steady state design conditions versus transient conditions. The design of the mechanical refrigeration systems was sized, as much as possible, for the steady state loads. This reduced the capacity modulation required for the system. Details of this design approach are discussed in the relevant sections of this report. The thermal analysis did not consider heat losses through the chamber wall and duct insulation. In view of the high change rates of air temperature, these losses were not considered significant to the overall analysis. Heat losses in the refrigeration system piping should be minimized by the application of adequate insulation.

Due to state of the art limitations on equipment selection, the method of vibration and electrical generation are handled in a similar fashion for all three of the conceptual designs. An electro-magnetic vibration system capable of handling the performance specifications was selected and recommended for use in each concept. A similar approach was used for the electrical characteristics generation. The control system which utilizes a computer system was conceived as a general solution to test programming.

The input functions to the computer such as temperature, pressure, and humidity, are sensed at different locations for each system. The response of the computer, however, would be similar in all three cases.

As discussed earlier, the introduction of humidity level control in the chamber rendered any attempt at static conditioning of the chamber unfeasible. Several methods which were evaluated are discussed. However, one basic approach has been selected as being feasible for all three conceptual designs. As will be seen in the calculations for chamber air conditioning, it became obvious that the design parameters eliminated the possibility of using one chamber conditioning loop for the supply of different compartments in one single chamber. Consequently, the design concerned itself with the selection of components for separate chambers with their own individual conditioning systems.

Finally, it was felt that the scope of this report should attempt to deal with the existing Air Force CERT chamber not only as a piece of equipment to be modified, but as a workable conceptual approach to the attainment of the design goals. Although the modifications to the existing chamber are specifically dealt with in this report, the concept itself has been extended so that a fair comparison could be made with the additional concepts generated.

SECTION II

COOLING AIR CONCEPTS

2.1 Open Loop Compressor (Extension of AFFDL Design)

The existing Air Force design for cooling air consists of an open loop compressor system. A 150 cfm, 100 psig reciprocating oil compressor supplies a receiver through various filters and traps to a purge air regenerative desiccant dryer. At that point an air flow valve used to control mass flow, feeds parallel hot and cold heat exchanger systems with a common mixing point to obtain the desired air temperature. Humidity is added, by means of steam injection and the air is then fed into the chamber.

A desiccant regenerative dryer is used to obtain the -65°F dew point necessary to assure that icing does not occur on the heat exchanger coils. Since the air is always initially "dry", it also allows precise control of humidity by injecting only the amount needed at any particular time. The dryer can be made to operate at any pressure, but the higher the pressure, the more efficient the drying operation and the smaller the dryer. The increased cost of a compressor producing 100 psig vice 5 psig, is outweighed by the dryer cost to produce a -65°F dew point at 5 psig. Therefore, a 100 psig compressor which is a commercially available standard, is a good choice for the system. The traps and filters serve to control "slugs" of water which the dryer cannot handle and to reduce contamination in the system.

Since the system is high pressure, supply lines are of small copper pipe and the heat exchangers are of the shell and tube type. The cooling was provided by means of LN_2 and the heating by means of steam. The performance, as compared to the requirement of this system as given by the Air Force, was as follows:

Variable	Performance	Requirement
Temperature	-90 to 200°F @ 10°F/min. for 255°FΔ	-65 to 200°F @ 40°F/min. for 265°FΔ
Air Flow	0 to 5.73 lbs./min. @ 120 lbs./min./min.	0 to 10 lbs./min. ± at flow > 0 ±10 lbs./min./min.
Humidity	-90 to 140°F dew point	0.0005 to 0.04 lb. H_2O/lb. air @ ± 0.04 lb. H_2O/lb. air/min.

Apparently, the main problem areas of the supply system were temperature change rates and maximum air flow capabilities.

The modifications to the open loop compressor system, as shown in figure 1, make this system a viable concept to meet CERT requirements. The existing compressor is a 150 scfm, 100 psi oil reciprocating compressor and the dryer is a purge air regenerative desiccant type. The dryer requires 33 percent of the total air flow for regeneration. With this system, a maximum of 5.73 lbs./min. of air was able to be supplied. Consider the following calculations:

Existing Facility

150 cfm   P = 14.7 psia

T = 80° = 540°R

$$\text{Density } d = \frac{P}{RT} = \frac{14.7(144)}{(53.3)(540)} = 0.0735 \text{ lbs./ft.}^3$$

At 150 cfm:

$$\text{Mass Flow} = dV = (0.0735 \frac{\text{lb.}}{\text{ft.}^3}) (150 \text{ cfm}) = 11.025 \frac{\text{lbs.}}{\text{min.}}$$

The dryer requires 33 percent of the total air flow 0.33(11.025) = 3.638 lbs./min. This leaves 11.025 lbs./min. - 3.638 lbs./min. = 7.387 lbs./min. Since 5.73 lbs./min. was actually supplied, the rest or 7.387 - 5.73 = 1.657 lbs./min. must have been used in pneumatic controls and losses.

Considering this, if we kept the same dryer, we would need the following size compressor:

Actual Flow Required = 10 lbs./min.

Line Losses = 1.657 lbs./min.

Total Initial Flow = 11.657 lbs./min.

$11.657 + 0.33(X) = X$ Where X = Total mass flow required

X = 17.39 lbs./min.

$$\text{Therefore, total air flow} = \frac{17.39 \text{ lbs./min}}{0.0735 \text{ lbs./ft.}^3} = 236.6 \text{ cfm}$$

With a purge air dryer we would need a 236.6 cfm compressor to supply the 10 lbs./min. air flow. Without the purge air dryer (using another type, to be discussed later), a compressor of 159 cfm would be needed.

$$\frac{11.657 \text{ lbs./min.}}{0.0735 \text{ lbs./ft.}^3} = 159 \text{ cfm}$$

In order to select a compressor size and type we must first look at the types of dryers available to see if an alternate to the purge air type exists which will determine the required air flow needed.

There are three different varieties of dryers available: refrigerated, deliquescent and desiccant. Refrigerated uses mechanical refrigeration to condense moisture out on cooling coils, but is limited to a 35°F dew point and therefore, is not suitable. Deliquescent uses absorbent chemicals which dissolve during drying and must be periodically replaced. However, there is a chemical mist formed in the drying process which carries over to downstream air lines as a highly corrosive salt and therefore we have eliminated it as a consideration. The regenerative types are the only ones that can meet the -65°F dew point requirement and still not contaminate the air. Among the regenerative varieties, there are three types to consider: the air purge, heat regenerative and heat pump. Each of these varieties uses twin towers of drying desiccant which must be periodically regenerated. The air purge uses already processed air from the main flow to alternately dry the desiccant beds. This requires anywhere from 20 - 35 percent of the total air flow for regeneration which increases the size of the compressor. For this reason and the fact that better alternate methods do exist, we eliminated the use of a purge air dryer. The heat regenerative type uses either electric heaters or steam heat exchangers for main regeneration. Without an external blower, it would still require 7 - 10 percent of the processed air to aid in regeneration. Its main disadvantage is that it is expensive to operate. In the heat pump, moist air is first passed through a refrigerant to an air chiller where 75 percent of the moisture is removed by condensation. The air is then passed through one of two dessicant beds where the remaining moisture is removed by absorption. While one desiccant bed is drying air, the other bed is being reactivated by a portion of the dried processed air (4 percent) as well as heat supplied by warm refrigerant gases passed through heat exchangers in the desiccant beds. There are no electric heaters and the overall process is so efficient, that the only energy needed is power to run a refrigeration compressor. Considering this, the heat pump was selected to replace the air purge drier in the open loop compressor system. The air flow requirement for a compressor would then be 159 cfm plus 4 percent or 165.4 cfm. Therefore, we would need a 165 cfm compressor to supply the 10 lbs./min. air requirement and to maintain the drying cycle in the desiccant beds. To supply the three individual chambers, we would require a 496 cfm compressor.

The main types of compressors available for this type of application are reciprocating and screw. Each can be either oil flooded or oil free. Both types require filters (if oil flooded), water separators, traps and after coolers. However, reciprocating compressors require special foundations, require more maintenance (since there are many more moving parts), are noisier and require more space. A screw compressor is more economical to run. Simple on/off regulation automatically matches air demand precisely from 100 percent down to 0 percent without waste or complication. Reciprocating types can be started and stopped repeatedly, but it imposes high loading on the bearings. Usually they are run at reduced outputs to avoid this and as such are inefficient, requiring more electricity to run. The disadvantages to oil flooded types in this application are obvious. Since the desiccant would be damaged by oil, filters would be required to maintain "oil free" air. If the filters malfunction, the desiccant bed plus all lines will be damaged and will require cleaning or replacement. Oil flooded types require more maintenance due to oil changes needed. For the foregoing reasons, it was felt that the optimum way was to select an oil free screw compressor. Oil free screw compressors are usually very expensive since the "dry" types require close tolerance machining for sealing. However, there is one type of oil free screw compressor which is not expensive. This is the water injected screw compressor. Water injected into the compression chamber provides a near perfect seal between the rotors and also acts as an effective coolant. The result is a very simple design which compresses the air in one stage with discharge air temperature in the range 100 to 140°F vice, approximately 300°F for all other oil free compressors. The water sealed rotors are isolated from the gears and bearings in a separate stator compartment so that contamination by oil is impossible. Therefore, all the advantages of oil free screw compressors (simplicity, low maintenance, cheaper operating costs, no installation costs, low noise, oil free air) are maintained in addition to being priced competitively with oil free reciprocating compressors. Our choice for design would then be a 500 cfm oil free, water injected, screw compressor.

The open loop compressor system has the advantage of providing high pressure air which can be dehumidified efficiently in a desiccant type of dehumidifier. The air leaving the compressor can be cooled to within 15°F of the temperature entering the compressor. Due to the pressurization of the air, it must be dehumidified to point where a minimum amount of moisture will condense and consequently freeze on the -65°F coil.

Although the specification requires a minimum specific humidity ratio of 0.0005 lb. water vapor per lb. of dry air, the dehumidifier should be selected to attain a specific humidity condition which approaches the saturated specific humidity condition at 100 psig and -65°F. This can be calculated in the following manner.

$$H_{sat} = 0.622 \frac{(Pwv)}{(P_T - Pwv)}$$

Where: H_{sat} = Specific humidity (lb. water vapor/lb. dry air)

Pwv = Partial pressure of water vapor (psia)

P_T = Total pressure of air and water vapor (psia)

Therefore since: P_T = 100 psig = 114.7 psia

$Pwv = 35.01 \times 10^{-5}$ psia (Pwv @ -65°F)

$$H_{sat} = 0.622 \frac{(35.01 \times 10^{-5})}{(114.7 - 35.01 \times 10^{-5})} = 1.898 \times 10^{-6} \text{ lb.wv/lb.da}$$

As can be seen the saturated specific humidity condition at -65°F and 100 psig is considerably smaller than the 0.0005 lb.wv/lb.da requested by the specification. Any specific humidity condition higher than the saturated specific humidity condition would result in de-humidification on the low temperature coil and consequent frosting of the coil.

Conditioning

The specification for cooling air reads "The rate of change of cooling air temperature shall be no less than ± 120°F/minute for a ± 40°F temperature step and no less than ± 40°F/minute between -65° to 200°F. In other words, this requires a total system response of 120°F/minute for a change in air temperature of 40°F. For a full scale change of temperature of 265°F (from -65 to 200°F) the total system must respond at 40°F/minute. The principal effects of the thermal mass capacity of the system upon airflow temperature response can be obtained by assuming that the system is adiabatically insulated from ambients and that only the system and airflow exchange heat.

Heat transfer balance can then be made between the system and the airflow as follows.

$$M_{air} \; C_{P_{air}} \; \Delta T_{air} = M_S \; C_{P_s} \; \Delta T_s$$

Where:

M_{air} = mass flow of air (lb./min.)

$C_{P_{air}}$ = Specific heat of air (0.24 BTU/lb./°F)

ΔT_{air} = Change in temperature of air (°F)

M_s = Total weight of delivery system (lb.)

C_{P_s} = Specific heat capacity of system (BTU/lb./°F)

ΔT_s = Temperature response rate of the system (°F/min.)

Since the system can be comprised of different materials with various specific heat capacities the equation becomes

$$M_{air}\ C_{P_{air}}\ \Delta T_{air} = \Sigma(M_s\ C_{P_s})\ \Delta T$$

Where $\Sigma(M_s\ C_{P_s})$ = Sum of thermal mass capacities (BTU/°F).

Considering the various rates of airflow and temperature response rates results in the following calculations.

Condition 1

M_{air} = 10 lbs./min.

$C_{P_{air}}$ = 0.24 BTU/lb./°F

ΔT_{air} = 40°F

ΔT_s = 120°F/min.

Therefore since $M_{air}\ C_{P_{air}}\ \Delta T_{air} = \Sigma(MC_P)_s\ \Delta T_s$

(10 lbs./min.) (0.24 BTU/lb./°F)(40°F) = $\Sigma(MC_P)_s$(120°F/min.)

$$\Sigma(MC_P)_s = 0.8 \text{ BTU/°F}$$

Condition 2

M_{air} = 10 lbs./min.

ΔT_{air} = 265°F

ΔT_s = 40°F/min.

Therefore: (10 lbs./min.) (0.24 BTU/lb./°F)(265°F) = $\Sigma(MC_P)_s$ (40°F/min.)

$$\Sigma(MC_P)_s = 15.9 \text{ BTU/°F}$$

Condition 3

M_{air} = 0.5 lb./min.

ΔT_{air} = 40°F

ΔT_s = 120°F/min.

Therefore: $(0.5 \text{ lb./min.})(0.24 \text{ BTU/lb./°F})(40\text{°F}) = \Sigma(MC_p)_s(120\text{°F/min.})$

$$\Sigma(MC_p)_s = 0.04 \text{ BTU/°F}$$

Condition 4

$M_{air} = 0.5 \text{lb./min.}$

$\Delta T_{air} = 265\text{°F}$

$\Delta T_S = 40\text{°F/min.}$

Therefore:

$$(0.5 \text{ lb./min.})(0.24 \text{ BTU/lb./°F})(265\text{°F}) = \Sigma(MC_p)_s(40\text{°F/min.})$$

$$\Sigma(MC_p)_s = 0.795 \text{ BTU/°F}$$

The thermal mass capacity represents a design limitation on the system. In order for the air flow temperature change to be obtained, the system response rate must be obtained. Similarly, in order for the system response rate to be obtained, the thermal mass capacity (BTU/°F) must not be exceeded. For the above specifications the lowest thermal mass capacity is required at the low airflow condition of 0.5 lbs./min. The thermal heat capacity at condition 3 is 0.04 BTU/°F. Considering the present AFFDL design, the length of copper pipe between the heat exchanger and the chamber is approximately 4 feet. The thermal capacity of the system can be calculated by using the following quantities:

$W_c = 0.75 \text{ lb./ft.}$

$L = 4 \text{ Ft.}$

$C_p = 0.09 \text{ BTU/lb./°F}$

The thermal capacity is therefore:

$$MC_p = W_c \overset{L}{R} C_p = (0.75 \text{ lb./ft.})(4 \text{ ft.})(0.09 \text{ BTU/lb./°F}) = 0.27 \text{ BTU/°F}$$

As can be seen the thermal capacity of the system exceeds the 0.04 BTU/°F which is required for the specification. If the length of copper could be reduced to 3 ft. and the copper pipe replaced by copper tubing the following thermal capacity would result:

$W_c = 0.145 \text{ lb./ft.}$

$L = 3 \text{ ft.}$

$C_p = 0.09 \text{ BTU/lb./°F}$

$MC_p = W_c L C_p = (0.145 \text{ lb./ft.})\ (3 \text{ ft.})\ (0.09 \text{ BTU/lb./°F})$

$MC_p = 0.039 \text{ BTU/°F}$

This represents a reduction of the thermal capacity of the system by a factor of 7.

If this thermal capacity could be achieved in the system design, no additional duct conditioning would be required.

In order to select the heat exchanger, it is necessary to define the maximum values of heat transfer which are required. The temperature of the air entering the heat exchanger is equal to the air temperature leaving the de-humidifier and is therefore a function of the type of drier selected. For the heat regenerative type of drier the leaving air temperature is approximately 100°F and this figure was used in our calculations. The following quantities were used in order to calculate the tonnage of refrigeration required:

M_{air} = 10 lb./min.

C_p = 0.24 BTU/lb./°F

T_{in} = 100°F

T_{out} = -65°F

Therefore:

$$Q = M_{air}\ C_p\ \Delta T$$

$$= (10\ \text{lb./min.})(60\ \text{min./H})(0.24\ \text{BTU/lb./°F})(165°\text{F})$$

$$Q = 23{,}760\ \text{BTU/H}$$

The heat exchanger operation is defined by the following equation:

$$Q = UA\ \Delta T_{LMTD}$$

Q = BTU/H

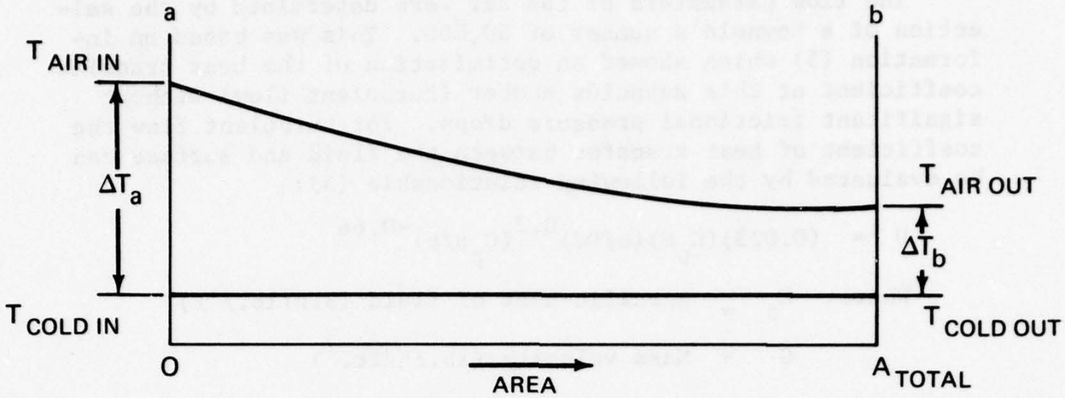
U = Overall heat transfer coefficient BTU/H/ft.^{2}/°F

A = Total effective area (ft.^{2})

ΔT_{LMTD} = Log mean temperature differential

$$= \frac{\Delta Ta - \Delta Tb}{LN\ \frac{\Delta Ta}{\Delta Tb}}$$

We assumed that the refrigerant in the heat exchanger was operating at constant temperature (i.e. - evaporating) and therefore can be represented by the following graph:



Where: $T_{air\ in} = 100°F$

$T_{air\ out} = -65°F$

$T_{cold} = -320°F$ (nitrogen)

Therefore: $\Delta T_{LMTD} = \dfrac{\Delta Ta - \Delta Tb}{LN\,\dfrac{\Delta Ta}{\Delta Tb}} =$

$$\frac{[100-(-320)] - [-65-(320)]}{LN\,\dfrac{[100-(-320)]}{[-65-(-320)]}}$$

$$\Delta T_{LMTD} = \frac{165}{LN\ (1.65)} = \frac{165}{0.5}$$

$$\Delta T_{LMTD} = 330°F$$

The overall heat transfer coefficient can be evaluated by the following equation:

$$\frac{1}{U} = \frac{1}{U_i} + \frac{1}{U_w} + \frac{1}{U_o}$$

Where: U_i = Heat transfer coefficient between the wall and the inside fluid.

U_o = Heat transfer coefficient between the wall and the outside fluid.

U_w = Heat transfer coefficient through the tube wall (thermal conductivity/wall thickness).

The flow parameters of the air were determined by the selection of a Reynold's number of 50,000. This was based on information (5) which showed an optimization of the heat transfer coefficient at this Reynolds number (turbulent flow) without significant frictional pressure drops. For turbulent flow the coefficient of heat transfer between the fluid and surface can be evaluated by the following relationship (5):

$$U = (0.023)(C_p G)(\mu/DG)^{0.2}(C_p \mu/k)^{-0.66}$$

Where: C_p = Specific heat of fluid (BTU/lb./°F)

G = Mass velocity (lb./H/ft.^{2})

μ = Fluid viscosity (lb./ft./H)

D = Diameter (ft.)

k = Thermal conductivity (BTU/H/ft./°F)

The following values were used for the calculation of the overall heat transfer coefficient. The thermal conductivity of brass tubing (11) is 29.4 BTU/H/ft./°F at 100°K. The values for air were taken at 17.5°F which is the medium temperature for the above conditions. The values of nitrogen were selected for saturated liquid at 80°K.

Air	Nitrogen
C_p = 0.24 BTU/lb./°F	C_p = 0.4928 BTU/lb./°F
μ = 0.0428 lb./ft./H	μ = 0.355 lb./ft./H
k = 0.0140 BTU/H/ft./°F	k = 0.077 BTU/H/ft./°F

The density of air was calculated to be:

$$d = P/RT = (114.7)(144)/(53.3)(477.5)$$

$$d = 0.6489 \text{ lb./ft.}^3$$

Since the Reynolds number (Re) is equal to 50,000 the following equation can be made (5).

$Re = dvD/\mu$ where d = density
v = velocity
D = Diameter
μ = viscosity

therefore:

$$50{,}000 = (0.6489)(VD)/0.0428$$

$$(vD) = 3297.88 \text{ ft.}^2/\text{H}$$

The mass flow of 10 lbs./min. results in a volumetric flow for the calculated density of 924.64 ft.^{3}/H. The total cross-sectional area was then calculated to be equal to 0.10 ft.^{2}. The mass velocity (G) becomes 6000 lb./H/ft.^{2}. The following equation (5):

$$G = dv$$

can be modified to:

$$DG = dvD = Re(\mu)$$

Substituting this in the equation for the coefficient of heat transfer it becomes:

$$U = (0.023)(C_p G)(\mu/Re\mu)^{0.2}(C_p\mu/k)^{-0.66}$$

Substitution of the air properties in the equation yields:

$$U_o = 4.668 \text{ BTU/H/ft.}^2/°F$$

Nitrogen

If the liquid nitrogen changes from saturated liquid to saturated vapor after expanding from three atmospheres to one atmosphere, the change in enthalpy is approximately 71 BTU/lb. In order to cool at the rate of 23,760 BTU/H, the following mass flow would be required:

$$M_n = (23{,}760 \text{ BTU/H})(1/71 \text{ BTU/lb.}) = 335 \text{ lb./H}$$

The density of the vapor leaving the heat exchanger (6) is 0.3795 lb./ft.^{3} A recommended (6) flow velocity for refrigerants was selected to be 1200 ft./min. This results in a total flow area of 0.0123 ft.^{2}. Brass tubing (3/8") with a wall thickness of 0.0825 in. (0.0069 ft.) and an inner diameter of 0.375 inches has a cross-sectional area of 0.008 ft.^{2} (12). The total number of tubes required would be approximately 16. Assuming the mass flow is evenly distributed, this results in a mass flow per tube of 20.94 lb./H and a mass velocity of 26175 lb./H/ft.^{2}. The determination of the heat transfer coefficient was done by the following equation (5).

$$U = 0.06\ (k_1/D)(d_1/dv)^{0.28}(DGx/\mu_1)^{0.87}\ (C_p\mu_1/k)^{0.4}$$

Where the subscripts l and v refer to the liquid and vapor phases, respectively. The quantity x is the vapor quality and was selected as 50 percent. Substitution yields the following:

U_i = 371.13 BTU/H/ft.^{2}/°F

The thermal conductivity of the tubing is 29.4 BTU/H/ft/°F which results in a U-factor of 4260.87 BTU/H/ft.^{2}/°F.

Substitution of the above values yields:

$$\frac{1}{U} = \frac{1}{U_i} + \frac{1}{U_o} + \frac{1}{U_w}$$

$$\frac{1}{U} = \frac{1}{371.13} + \frac{1}{4.668} + \frac{1}{4260.87}$$

$$\frac{1}{U} = 0.0027 + .214 + 0.0002$$

$$\frac{1}{U} = .2169$$

$$U = 4.610 \text{ BTU/H/ft.}^2/°\text{F}$$

Using this value to determine the required heat transfer area:

$$A = (Q)/(U)(\Delta T_{LMTD})$$

$$A = (23760)/(4.610)(330°)$$

$$A = 15.62 \text{ ft.}^2$$

If refrigerant 503 at -100°F was used the ΔT_{LMTD} would become:

$$\Delta T_{LMTD} = \frac{\Delta Ta - \Delta Tb}{LN\frac{\Delta Ta}{\Delta Tb}}$$

$$= \frac{[100 - (-100)] - [-65 - (-100)]}{LN \frac{[100 - (-100)]}{[-65 - (-100)]}}$$

$$\Delta T_{LMTD} = \frac{[200 - 35]}{LN\frac{200}{35}} = \frac{165}{1.74} = 94.83°\text{F}$$

The following values (6) were used for the determination of the heat transfer coefficient.

R - 503

C_p = 0.195 BTU/lb./°F

μ = 0.61 lb./ft./H

k = 0.0575 BTU/H/ft./°F

d_1 = 89.05 lb./ft.^{3}

d_v = 0.772 lb./ft.^{3}

If the R-503 cycle is defined by figure 10, the enthalpy change is 54 BTU/lb. The mass flow required is:

$$M_{503} = (23,760)/54 = 440 \text{ lb./H}$$

Since the vapor density is 0.772 lb./ft.^{3} and a maximum velocity of 1200 ft./min. was selected, the required cross sectional flow area is .0079 ft.^{2}. If the same tubing used for the nitrogen heat exchanger was selected, the total number of tubes required would be 10. Assuming an even distribution of mass flow this would result in a mass velocity of 55000 lb/H/ft.^{2}. Substituting the above values in the following equation:

$$U = 0.06\,(K_1/D)(d_1/d_v)^{0.28}(DGx/\mu_1)^{0.87}(C_p\mu_1/k)^{0.4}$$

yields:

$$U_i = 303.69 \text{ BTU/H/ft.}^2/°\text{F}$$

The overall heat transfer coefficient of the heat exchanger becomes:

$$\frac{1}{U} = \frac{1}{U_i} + \frac{1}{U_o} + \frac{1}{U_w}$$

$$\frac{1}{U} = \frac{1}{303.69} + \frac{1}{4.668} + \frac{1}{4260.87}$$

$$\frac{1}{U} = .0033 + 0.214 + 0.0002$$

$$\frac{1}{U} = 0.2175$$

$$U = 4.597 \text{ BTU/H/ft.}^2/°\text{F}$$

The required heat transfer area is:

$$A = (Q)/(U)(\Delta T_{LMTD})$$

$$A = (23{,}760)/(4.597)(94.83)$$

$$A = 54.50 \text{ ft.}^2$$

This represents approximately 3.5 times as much heat transfer surface area as the cryogenic heat exchanger. The selection of the heat exchanger tubing configuration must take several factors into consideration. There are detailed discussions in the literature (11, 12, 13) which enumerate the material selection and construction for cryogenic applications. The total cross sectional area required for acceptable airflow parameters (A = 0.10 square feet) applies to any configuration. The majority of the available types are either 1) shell and tube or 2) concentric tube configurations. Scott (11) discusses both types. The final selection would be based on cost, low heat capacity and dimensions of the space available for the exchanger. Based on the nitrogen heat exchanger parameters described (A = 15.62 square feet, 16 tubes of 0.540 O.D. tubing) the total length of the nitrogen heat exchanger would be 6.9 feet. The total length required for an R-503 system would be approximately 38.5 feet. Although this is certainly an unrealistic number for a single heat exchanger, the installation of several heat exchangers could be an advantage in a system with such a wide range of airflows. An R-503 heat exchanger of similar length (8.2 ft.) to the nitrogen heat exchanger was investigated. Calculations were done for a design utilizing 32 brass tubes of 1.062 in. diameter and a wall thickness of .1265 in. This resulted in a total heat transfer surface area required of 69.21 square feet. The resultant low velocity of the refrigerant vapor however, could lead to oil return problems from the heat exchanger to the compressor. Due to the complexity of problems involved in the design of a mechanical refrigeration system in the open loop compressor system, a cryogenic heat exchanger utilizing liquid nitrogen as the refrigerant was used as the basis for the economic analysis.

The major problem with the cryogenic heat exchanger in a system with a wide range of flows is that since the unit must be sized for its maximum load, a low airflow condition causes a temperature control problem. Since liquid nitrogen is operating in the heat exchanger at -320°F, the temperature of the air approaches this temperature at a low flow condition. The current method of controlling the final temperature of the air by allowing air heated by the steam heat exchanger to mix with the small airflow leaving the liquid nitrogen exchanger can be avoided. By utilizing compact electric heat exchangers in series with the cryogenic heat exchanger, the air could be reheated to the desired temperature. These heaters are capable of being in a compressed air system and can be installed in series or parallel to increase capacity. Inner-fin compact heat exchangers have an extended surface on the inside of the heat exchanger tube. Also, they have a very low pressure drop characteristic versus airflow. Capacity modulation is also available due to their electrical operation.

If, during a low airflow condition, the air was leaving the liquid nitrogen heat exchanger at -320°F and a condition into the chamber of 200°F was required, this would result in a re-heat capacity needed of:

$$Q = (0.5\ \text{lb./min.})(0.24\ \text{BTU/lb./°F})(520\text{°F})(60\ \text{min./H})$$

$$= (3744\ \text{BTU/H})(1\ \text{kw}/3413\ \text{BTU/H}) = 1.097\ \text{kw}$$

Since it is required to install a capacity of 4.22 KW for normal heating operation, this represents no additional installed capacity.

With the control system designed it would be possible to eliminate the constant supply of nitrogen through the heat exchanger. If the test conditions were programmed in the computer, a new control point requiring cooling could signal the operation of the nitrogen heat exchanger. The heat exchanger mass therefore could be brought down to operational temperature while maintaining the zero time condition by utilizing the electrical heaters in a reheat capacity.

This can be considered the main advantage of a series system over a parallel system when using electrical heaters. The total heat capacity of the electrical heaters can be attained instantaneously while the cooling heat exchanger requires time to reach full capacity due to its large mass. If the heaters are installed in series with the cooling heat exchanger, they should be located as near as possible to the entry point of the ducting to the chamber. If the heaters are maintaining a zero time condition while the cooling heat exchanger is reaching full capacity, the ducting between the cold-heat exchanger and the electrical heaters will be approaching an equilibrium condition with the air simultaneously.

Humidification

For humidification, all the standard methods, (steam injection, pan humidification, water sprays, air washers and ultrasonic humidification) were considered.

Steam jet humidifiers permit accurate control because of their quick response. Their major disadvantages are: (1) the fact that sensible heat may be introduced into the controlled space, and (2) the possibility of condensation if the jet is directed to a cooler surface or to a restricted area. These advantages are negligible in a large air handling system in which humidity is not added directly into the controlled space.

Pan humidifiers are used where quick response and large capacity are not essential. They are normally not suitable where load changes are rapid. Therefore, this method was eliminated from consideration.

Water sprays offer quick response in a process which is nearly adiabatic. However, because there can be a sensible cooling effect, a spray humidifier is undesirable where precise control of dry bulb temperature is a must. Conditioning the air remote from the chamber in which it is to be used can eliminate this effect.

Air washers are not particularly suitable for situations involving low relative humidities. A provision must be made to prevent icing if the mixed air temperature falls below 32°F and to ensure that droplets of moisture are not carried along with the air stream. Response is fair and the process is adiabatic as long as unheated recirculated water is used in the washer. Since low temperature humidification may exist, this method was eliminated from consideration.

Ultrasonic humidification uses nozzles operated in the same manner as any air atomizing nozzle, but produces a quality of atomization which cannot be achieved with any other type of atomizing device. The nozzles create droplets of low mass, (averaging 10 to 15 microns) low fluid velocity and low impingement characteristics which prevent wetting of duct walls and surrounding areas. The atomizers are activated by pressurized gas, either compressed air or steam. Gas passes through the nozzle's inner bore at high velocities and into a resonator cavity. The result is an intensified sonic energy field between the nozzle exit and the resonator. Any liquid pumped, poured, or sucked into this field is atomized into fine droplets. The typical system consists of a single or ganged nozzle(s), air and water solenoids, a flowmeter and pressure regulating valves.

Of all these methods, the steam jet and ultrasonic humidification lend themselves most readily to proportional control in large systems requiring fast response. Each has a major advantage, the addition of sensible heat from steam when requiring humidity at high temperature and the fact that ultrasonic humidification does not add any heat when requiring humidity while lowering the temmperature. These would work together to aid in temperature change rates and in the most efficient humidification throughout the whole temperature change.

A combination of both systems is ideal, however, ultrasonic humidification cannot be used with the open loop compressor system without the addition of the plenum, since the nozzles will not fit in the small diameter high pressure lines. It will however, be used on both of the other concepts developed. Therefore, for the AFFDL chamber only steam humidification, as is presently used, is recommended.

2.2 Open Loop Blower Cooling Air

The first additional conceptual design for supplying cooling air is the open loop blower system. See figure 2 for a schematic of the system. The system consists of a 15 psig oil free positive displacement axial type blower that feeds, through ducting and filters, a heat regenerative desiccant dryer. The dryer then feeds an insulated conditioning plenum which contains both cooling coils and heaters. After the plenum, humidity is added and ducted into the chamber. Using a high pressure blower rather than a fan or low pressure type allows the use of smaller valving and ducting and still allows the use of a desiccant dryer to obtain -65°F dew points. Although the dryer must be considerably larger in size than a compressor system, additional advantages result. A speed control or bypass system is used to control flow; a receiver and throttling system is not needed. Open element duct heaters can be used which have a quicker response and better total heat transfer capability than a steam heat exchanger. The cooling coil is of the direct expansion type with low pressure drop and provides excellent heat transfer characteristics.

In selecting a blower, many factors were considered. The air flow requirement was still 10 lb./min. for each chamber and the losses were expected to be approximately 10 percent for the ducting system. The heat regenerative dryer required 7 to 10 percent of the total flow for regeneration. Considering these factors, the end blower must be capable of supplying 162 cfm to maintain mass flow. For three systems the total requirement would be 486 cfm. It was still desirable to have oil free air which eliminated single and double stage rotary blowers. A low dew point was also needed and the most effective way to do this is by a desiccant dryer. Therefore, a trade-off was considered. To dry air at only a few inches of water (low pressure blower) would require a very large and costly dryer. Therefore, a blower was selected to give maximum pressure (15 psi) but still stay within a range of pressure that could be used with a ducting system. The blower selected was an axial helical screw type positive displacement blower. It delivers 496 cfm at 15 psi requiring 45 hp. Since the lobes of the rotors do not touch each other, no lubrication in the compression chamber is required. Therefore, we have both oil free air and very low maintenance. Oil is only used for lubrication of bearings and timing gears. All parts are balanced, producing very little vibration and therefore requiring no special installation. Since the unit operates with low friction and at relatively low speed (3500 rpm), power costs are low and wear is minimized.

The same reasons for selecting a regenerative desiccant dryer apply here as they did in the selection of one for the compressor system. However, at the time of this report, a 15 psig heat pump dryer was not commercially available. If a 100 psig dryer was used in this system, it would not provide the low dew point necessary for the system. Therefore, a 15 psig electric heat regenerative type is recommended.

The humidification system used would be the combination steam injection plus ultrasonic humidification as discussed in the open loop compressor system. At maximum air flow (10 lb./min.) and for maximum humidity (.04 lb. H_2O/lb. air), we would need a humidity injection rate of 10 lbs./min.(.04 lb. H_2O/lb. air) = 0.4 lb./min. to maintain humidification. This is not a problem with either steam injection or ultrasonic humidification. Since for ultrasonic the nozzle selection is based on gallons H_2O/ hour, we would select one unit with the following capacity:

$$\frac{0.4 \ \frac{\text{lbs. } H_2O}{\text{min.}} \ (60 \ \frac{\text{min.}}{\text{hr.}})}{8.345 \ \frac{\text{lbs.}}{\text{gal.}}} = 3 \text{ gallons } \frac{H_2O}{\text{hr.}}$$

Cooling and Heating

The determining factor in the selection of the open loop blower is the desiccant dehumidifier. In order for the dehumidifier to operate efficiently, the blower had to be selected to deliver the required quantity of air at a minimum pressure of 15 psig. The density of the air at these conditions is:

$$d = \frac{P}{RT} = \frac{(15 + 14.7)\ (144)}{(53.3\ (460 + 80)} = 0.1485 \text{ lb./ft.}^3$$

At a mass flow of 10 lb./min., this represents a volumetric flow of:

$$\text{CFM} = \frac{10 \text{ lb./min.}}{0.1485 \text{ lb./ft.}^3} = 67.34 \text{ CFM}$$

The thermal capacity of the ducting should be designed to approach the 0.04 BTU/°F previously calculated as required for the specified response. As in the open loop compressor system, if the heaters are installed in series with the cooling heat exchanger, they should be located as near as possible to the entry point of the ducting to the chamber. If the heaters are maintaining a zero time condition while the cooling heat exchanger is reaching full capacity, the ducting between the cold-heat exchanger and the electrical heaters will be approaching an equilibrium condition with the air simultaneously.

The sizing of the mechanical refrigeration system was done for an inlet air temperature of 100°F. Since the maximum airflow is 10 lb./min. and it is required to attain -65°F, the following refrigeration is required in heat exchanger.

M_{air} = 10 lb./min. $\quad T_{in}$ = 100°F

C_p = 0.24 BTU/lb./°F

T_{out} = -65°F

Therefore $Q = MC_p \Delta T$

$= (10 \text{ lb./min.})(0.24 \text{ BTU/lb./°F}(165°\text{F})(60 \text{ min./H})$

$= 23{,}760 \text{ BTU/H}$

A mechanical refrigeration system capable of operating at -65°F with the above capacity would be selected. A typical cascade system uses R-503 in the low stage and R-22 in the high stage. Capacity modulation by means of cylinder unloading and hot gas bypass are recommended for the design.

2.3 Closed Loop Blower

Statement of Operation

The main advantage of the closed loop system is that it imposes no additional air capacity requirement for the vacuum pump. By balanced circulation of the environmental cooling air, the vacuum pump is only required to remove the air inside the chamber in order to attain an altitude condition. In order to understand the design specifications required for the closed loop blower, a brief description of the loop will be given. After the air enters the chamber and passes across the test item, it is removed by the ducting connecting the chamber and the low pressure side of the blower. So that the temperature extremes be avoided in the blower, a conditioning plenum would have to be installed between the chamber and the blower. After being pressurized in the blower, the air would be sensed for its temperature, pressure and specific humidity conditions. The control system would use these conditions in conjunction with a flow measurement device in order to deliver the correct mass flow rate to the chamber. The air is then conditioned (cooled, dehumidified, or heated) to the required design condition specified by the test program. Finally humidification is done prior to entering the chamber. The main disadvantage of the system is the fact that the blower and ducting system must be sized to handle air which is at an altitude density. The lowest density air returning from the chamber could be at a condition of 70,000 feet altitude and a temperature of 300°F. The density, therefore, of the return air would be:

$$d = \frac{P}{RT} = \frac{(.065)(144)}{(53.3)(760)} = 0.0023 \text{ lb./ft.}^3$$

Since the maximum mass flow required is 10 lb./min., the blower must be capable of handling a volumetric flow of:

$$\text{CFM} = \frac{10 \text{ lb./min.}}{0.0023 \text{ lb./ft.}^3} = 4378 \text{ CFM}$$

The system does not include a desicant dryer in the system due to the pressure limitations on the blower selection. This means that any de-humidification will be handled by the cooling coils. In order to avoid inordinate frosting on the -65°F coil, the conditioning plenum prior to the blower should be designed to eliminate as much humidity as possible from the air returning from the chamber. A 35° coil would be capable of removing moisture over a good percentage of the range of the possible inlet conditions.

As can be seen from figure 17, above 50,000 feet the dew-point is below 35°F over the full range of the specific humidity design conditions. For any design point above 50,000 feet requiring cooling air below 35°, frosting will take place on the low temperature refrigeration coil.

The possibility of air returning to the mechanical refrigeration system at 300°F places a sizeable requirement on the conditioning system. In addition to providing de-humidification, a 35° coil prior to the cascade system would reduce the required capacity of the cascade mechanical refrigeration unit. The capacity of the system required to reduce 10 lb./min. of air at an inlet condition of 300°F to a leaving condition of 35° would be:

$$Q = M_{air} \; C_p \; \Delta T$$

Where Q = heat removal rate (BTU/H)

M_{air} = Air Flow (lb./min.)

C_p = Specific Heat of Air (0.24 BTU/lb./°F)

ΔT = Temperature drop (°F)

Q = (10 lb./min.) (0.24 BTU/lb./°F)(265°F)(60 min./H)

Q = 38,160 BTU/H = 3.18 tons.

If further conditioning of the 10 lb./min. airflow was required to -65°F, the required capacity of the cascade system would be:

$$Q = M_{air} \; C_p \; \Delta T$$

Q = (10 lb./min.) (0.24 BTU/lb./°F)(100°F)(60 min./H)

Q = 14,400 BTU/H.

Heating

If the air was returning from the chamber at 10 lb./min. and a temperature of -65°F and cooling air was required at 200°F, the required heating capacity would be:

$$Q = M_{air} C_p \Delta T$$

$$Q = (10 \text{ lb./min.}) (0.24 \text{ BTU/lb./°F})(265\text{°F})(60 \text{ min./H})$$

$$Q = 38{,}160 \text{ BTU/H} = 11.2 \text{ KW.}$$

Duct Design

The duct design is complicated by the wide range of volumetric capacities which it must handle. Attention should also be given to the design limits of the thermal mass capacity as discussed in section 6.1.

If a duct cross section area of 1 ft.^{2} and a minimum volumetric airflow of 150 CFM was selected, the minimum velocity of the ducting system would be 150 ft./min. This minimum volumetric airflow could be accomplished with a bypass loop around the blower. Regardless of the mass flow required in the cooling air delivery system, the blower would operate at a point capable of delivering the equivalent of 150 CFM. If an annular duct system was utilized as a means of accomplishing a bypass, additional benefits could be gained in terms of system response at low airflows. If a minimum of 150 ft./min. were selected for a coil face velocity, the required face area would be 1 ft.^{2}. In order to prevent excessive water carryover from the coil during de-humidification, a velocity of 750 ft./min. should not be exceeded (13). This would require approximately 6 coils (1 ft.^{2}/coil) on each conditioning plenum.

Blower Sizing

The blower must be capable of delivering air between 150 CFM and 4348 CFM. In order to estimate the pressure requirements, the components contributing to the pressure drop in the system should be evaluated. These components are:

1) duct friction
2) cooling coils
3) heaters

Duct Friction

The pressure drop of 150 CFM in a duct of 1 ft.^{2} (D = 1.13 ft.) can be evaluated by the following equation (6)

$$\Delta P = f(L/D) (V/4005)^2$$

Where: ΔP = frictional pressure drop (in. H_2O)

L = duct length (ft.)

D = inside diameter (ft.)

V = velocity (ft./min.)

F = friction coefficient = 0.02

Assuming 10 feet of ducting therefore:

$$\Delta P = (0.02)(10/1.13)(150/4005)^2 = .00025 \text{ in. } H_2O$$

35° Coil

Since the coil operates between an inlet of 300°F and a leaving air temperature of 35°F and the total area per coil is 1 ft.^{2}, the coil rating becomes:

$$BTU/H/ft.^2/°F = (38160)/(1)(265)(6)$$

$$= 24.$$

A Singer-Wilmington coil (2 row-4 fins/in. - J fin-flat) would satisfy this requirement and has an approximate ΔP at 150 ft./min. of 0.012 in H_2O.

Cascade Coil

Assuming a suction temperature of -100°F and an inlet air temperature of approximately 35°F, the coil rating for 6 coils at 1 ft.^{2} per coil is:

$$BTU/H/ft.^2/°F = (14,400)/(1)(135)(6)$$

$$= 17.7.$$

The same configuration coil was satisfactory and would also have a ΔP of 0.012 in. H_2O at 150 ft./min.

Heaters

The pressure drop of finned tube electrical duct heaters at 150 ft./ min. is 0.004 in. H_2O (Electric Heaters, Inc. - Custom Built Duct Heaters - Catalog EHD-72).

The pressure drop of the system is the sum of the individual component drops, therefore:

$$\Delta P_{total} = \Delta P_{duct} + \Delta P_{35° \text{ coil}} + \Delta P_{coil} + \Delta P_{heaters}$$

$$= 0.0025 + 0.0120 + 0.012 + 0.0040$$

$$= 0.02825 \text{ in. } H_2O.$$

The pressure drop at 4348 CFM becomes:

$$\Delta P_2 = \Delta P_1/(CFM_1/CFM_2)^2 \qquad (10)$$

Where: ΔP_1 = pressure drop at 150 CFM

ΔP_2 = pressure drop at 4348 CFM

CFM_1 = 150 CFM

CFM_2 = 4348 CFM

therefore:

$$\Delta P_2 = (0.02825)/(150/4348)^2$$

$$\Delta P_2 = 23.74 \text{ in. } H_2O$$

The blower selected should be capable of supplying 4348 CFM at 23.74 in. H_2O. The input power of the blower motor would be an additional load on the refrigeration equipment and would be included in the final design load estimate.

SECTION III

3.1 Chamber Construction

The chamber constructions for the CERT facility are shown in figures 4 through 6. Figure 4 shows the modifications to the existing AFFDL chamber that, as a bare minimum, are needed to only approach CERT requirements. Additional modifications should be incorporated but would basically change the whole chamber. However, it is considered that to perform a fair evaluation based on CERT requirements, a completely new chamber facility design should be conceived. The chamber should utilize one of the two additional conceptual chamber designs developed (figures 5 and 6). The approach to cooling air using an open loop compressor, as discussed previously, should still be considered as a third approach to meeting that part of the CERT requirements.

For the AFFDL extension (figure 4) the requirement of humidification and dehumidification inside the chamber renders the static conditioning (free convection of cold wall and electric heaters) undesirable. We can put humidity in the chamber by steam or ultrasonic injection and obtain fast response. However, dehumidification, without changing the air inside the chamber continuously is a problem. Two methods are available: (1) having a separate refrigeration system with a cooling coil inside the chamber which will condense water out on the coils to be drained; or (2) a dry nitrogen injection while venting to the atmosphere (no altitude) or venting through the pump. The first is a much more practical approach but requires the addition of a circulating fan. This method would also increase the temperature response time since the coil would be capable of maintaining -65°F temperature. However, the spacing of the coils to prevent frosting is critical. The vacuum pump would also assist dehumidification, while at altitude, since the design is capable of handling the wet air.

The cold wall would be left in the chamber but it should be changed and used in the following way: since the copper cold wall represents a large load to heat up when going from low to high temperature, its construction should be changed. The construction should consist of a light stainless steel panelcoil type heat exchanger. This would reduce the heat capacity and mass and thus, lower the temperature response time. Since we have added mechanical refrigeration with a -65°F coil in the chamber, the new cold wall should only be used when fast rates of change from high to low temperature are needed. It would basically be used to take care of the latent heat load built up in the chamber wall at high temperature. Once low temperature is reached, the LN_2 supply would be shut off and the mechanical refrigeration would be used for steady state response.

The number of heaters used for the present chamber is four. Based on our calculations, we can increase the number of heaters to twelve to increase the heat response necessary for maximum temperature change. It is also noted that the present chamber (4 feet in diameter) is not capable of holding a 3 foot cube test item. The dimensions must be limited to 2 1/2 feet x 2 1/2 feet x 2 1/2 feet. This will still allow flow around the test item however, much turbulence will exist in the chamber.

The first additional concept for chamber conditioning is shown in figure 5. The chamber is approximately 7 feet long (not including end caps), 6 feet in interior diameter, with a 6 inch annulus inside. It can be reduced to 6 feet long if it is used with a compressor system. The exterior of the chamber should be the pressure member. This will allow the interior of the chamber to be a light-weight 316 stainless steel of panel coil type construction. This means that only a small mass of metal would have to be heated or cooled for each temperature cycle. Having the external member as the pressure member allows the insulation to be at altitude conditions. This will also cut down on heat loss through the wall. Precautions must be taken so that humidity does not enter the insulation, which would destroy its insulating qualities. This can be accomplished by less steel interior by welding and allowing pressure to equalize walls through valving and desiccant type filters. A glass insulating material should be used since it has the best insulating characteristics and is capable of withstanding high temperatures.

For the interior of the chamber, the design includes a 6 inch annulus, created by a fiberglass shroud, through which the air returns to the fan. This allows conditioned air to be delivered to the test item and returned separately to the fan to be reconditioned. Turbulence is kept to a minimum. An additional benefit is that the conditioned air comes in contact with a fiberglass shroud which has air of similar temperature on its other side. This creates a gradient of temperature to the inner wall and cuts down on heat transfer from the conditioned flow.

Fiberglass was selected as the shroud material to increase temperature response rate. Assuming a 1/16 inch thick shroud to give a 6 inch annulus, the following calculations are compared:

Fiberglass	Stainless Steel
Cp (specific heat) $\sim$ 0.25 BTU/°F/lb.	Cp = 0.12 BTU/°F/lb.
Specific Volume $\sim 15 \frac{\text{cu. in.}}{\text{lb.}}$	Density = 0.28 lb./cu. in.

For 5 feet diameter annulus and 7 feet long, the volume of the annulus is 0.573 cubic foot.

Fiberglass

$$\text{Mass} = \frac{1}{15}\ \frac{\text{lb.}}{\text{cu. in.}}\ \left(\frac{1728\ \text{cu. in.}}{\text{cu. ft.}}\right)\ (0.573\ \text{cu. ft.})$$

= 66.01 lbs.

Stainless Steel

$$\text{Mass} = 0.28\ \frac{\text{lb.}}{\text{cu. in.}}\ \left(\frac{1728\ \text{cu. in.}}{\text{cu. ft.}}\right)\ (0.573\ \text{cu. ft.})$$

= 277.24 lbs.

For a 40°F change the heat capacity would be:

$$Q = MC_p\Delta T$$

Fiberglass

Q = (66.01 lbs.) (0.25 BTU/lb./°F)(40°F)

= 660.1 BTU

Stainless Steel

Q = (277.24 lbs.)(0.12 BTU/lb./°F)(40°F)

= 1330.8 BTU

As can be seen, the heat capacity of the fiberglass is approximately half of what it would be if 316 stainless steel was used. By decreasing heat capacity, the temperature response is increased. Fiberglass air deflectors are also used at the end of the chamber to cut down on turbulance and reduce heat transfer.

Even though the interior of the chamber is a lightweight stainless steel to reduce heat capacity, it is felt that the interior should be of a panel coil type of construction to allow LN_2 to be circulated within it. The panel coil wall should be used to remove the heat load in the wall to aid in temperature response when going from high to low temperature.

Equipment was placed in the chamber as shown in figure 5. The 35°F coil would do the majority of dehumidification by condensation using the -65°F coil for the rest. At altitude the vacuum pump, since it is placed at the return end of the air flow, will also aid in dehumidification. Humidification is accomplished by means of steam and ultrasonic injection as discussed previously and is placed after the heaters.

The second conceptual design for the CERT facility is shown in figure 6. The concept used in the approach is total mixing of the air in the chamber, which is the more commercially used method, rather than separated flow as in the annulus design. The chamber is still 6 feet in diameter and 7 feet long (not including end caps) if used with a blower system but can be reduced to 6 feet long if a compressor system is used. The chamber would be constructed using the same materials and insulation as the annulus design. The cold wall, refrigeration coils, heaters, fan and humidification system would also be the same as the annulus design. However, some equipment placement was changed to facilitate better operation. Since a total mixing was desired, the fan was moved to utilize it as a blow-through rather than a draw-through. This would allow the 35°F dehumidification coil to be located at the opposite end of the flow to enable more effective dehumidification to take place. The vacuum pump was also moved, again, to accommodate dehumidification at the 35°F coil end. The -65°F coil, heaters and cold wall were left the same.

3.2 Chamber Conditioning

Fan Selection

Regardless of the method of air circulation conditioning inside the chamber, the conditioned air will change in temperature as it travels through the chamber. With the several sources of heat load inside the chamber, it is impossible to maintain a constant temperature throughout the conditioned space. In order to evaluate the quantity of air which should be circulated in the space it was necessary to select a temperature change which would be acceptable. This change in temperature selected was 25°F. The steady state sensible heat loads on the chamber were then calculated for a condition of -65°F inside the chamber at a sea level altitude.

Test Item

The "worst case" would be a situation in which -65°F was required and the test item was imposing its maximum heat load. Therefore:

$$Q_{TI} = (5kW)\ (3413\ BTU/H/kW) = 17065\ BTU/H$$

Cooling Air

If the cooling air was at full flow (10 lb./min.) and at its highest design temperature (200°F), it would be imposing its maximum load upon the chamber conditioning system. Therefore:

$$Q_{CA} = MCp\Delta T = (10 \text{ lb./min.})(0.24 \text{ BTU/lb./°F})(265°F)(60 \text{ min./H})$$

$$= 38{,}160 \text{ BTU/H}$$

Blower Load

A motor size of 5 hp was assumed to be required to run the fan at the above conditions. This assumption was checked and a discussion of the horsepower required follows. The blower load is a result of the energy added to the circulating air by the blower. The motor was assumed to be located outside of the air stream and therefore, the motor efficiency was not included in the cooling load estimate. Therefore:

$$Q_{blower} = (5 \text{ hp})(2545 \text{ BTU/H/hp}) = 12725 \text{ BTU/H}$$

The total of the steady state loads inside the chamber is therefore:

$$Q_{total} = Q_{TI} + Q_{CA} + Q_B$$

$$= 17065 + 38160 + 12725 = 67950 \text{ BTU/H}$$

Assuming a 25°F rise in the chamber, the CFM required is:

$$CFM = \frac{Q \text{ total}}{(d)(Cp)(\Delta T)} = \frac{67950}{(0.100)(0.24)(60)(25)} = 1888 \text{ CFM}$$

If the same temperature condition existed at an altitude of 70,000 feet ($P = 0.65$ psia) the density would be:

$$d = \frac{P}{RT} = \frac{(0.65)(144)}{(53.3)(395)} = 0.0044 \text{ lb./ft.}^3$$

In a later analysis of fan performance, the motor sizing of the fan at altitude will be discussed. It will be assumed at this point, that the fan requires 15 hp at 70,000 feet in order to provide the CFM required for cooling. The blower load then becomes at the altitude condition.

$$Q_B = 15 \text{ hp } (2545 \text{ BTU/H/hp})$$

$$= 38{,}175 \text{ BTU/H}$$

The total of the steady state loads inside the chamber is therefore:

$$Q_{total} = Q_{TI} + Q_{CA} + Q_B$$

$$= 17{,}065 + 38{,}160 + 38{,}175$$

$$= 93{,}400 \text{ BTU/H}$$

Assuming a 25°F rise in the chamber, the CFM required is:

$$CFM = \frac{Q\ total}{(d)(Cp)(\Delta T)} = \frac{93{,}400}{(0.0044)(0.24)(25)(60)}$$

$$CFM \cong 58{,}965$$

On the first analysis, it becomes obvious that the fan selection should be limited to providing the air for one chamber. However, even the restrictions on selecting a fan for 58,965 CFM are numerous, and the chance of finding one with all of the capabilities required are next to impossible.

In order to define the static pressure requirements of the fan at these operating conditions, it became necessary to estimate the system resistance of the fan at the sea level and altitude conditions. The calculations of system resistance were done for sea level altitude and then corrected for the density of air at 70,000 feet. For the chamber there are five items which contribute to the static pressure drop in the system:

1) wall friction
2) cascade refrigeration coil
3) 35°F coil
4) heaters
5) turning elbows at end of chamber

<u>Wall Friction</u>

The area of the annulus is 8.64 $ft.^2$. This is equivalent to a diameter of 39.8 inches. The pressure drop of 1888 CFM on a duct of 39.8 inches diameter can be calculated by the following equations (6).

$$\Delta P = f(L/D)\ (V/4005)^2$$

Where ΔP = frictional pressure loss (in. H_2O)

L = duct length (feet)

D = inside diameter (feet)

V = velocity (ft./min.)

F = friction coefficient

The Reynolds number for the above conditions was calculated to be:

$$Re = \frac{dvD}{\mu} = \frac{(0.75\ lb./ft.^3)(13110.0\ ft./H)(3.32\ ft.)/.04}{lb./ft./H}$$

$$= 8.2 \times 10^4$$

therefore f = .02 [for e/D = .0005 (6)]

Substitution yields

$$\Delta P = (.02)(7/3.32)(2.85/4005)^2$$

$$\Delta P = (.02)(2.11)(2.98 \times 10^{-5})$$

$$\Delta P = .13 \times 10^{-5} \text{ in. } H_2O$$

Cascade Refrigeration Coil

The performance characteristics of the coil were selected based upon Figure 12F for a 75 CFM compressor. The coil was selected for a leaving air temperature of -65°F at a refrigerant temperature of -100°F. Since the room air rises 25°F during its circulation, the difference in temperature between the entering air temperature and the refrigerant temperature is 60°F. At this point of operation, the coil and compressor combination must have a rating of 16 tons (192,000 BTU/H). Assuming a total face area of 12 $ft.^2$, the coil rating becomes 266 BTU/H/$ft.^2$/°F. A coil was selected (Singer-Wilmington - J fin (flat) 8 row with 4 fins/inch) and the pressure drop of this coil at 1888 CFM is 0.0575 in. H_2O.

35° Coil

The recommendation for the 35° coil was made in order to minimize the amount of frosting on the cascade (-100°F) coil. If the cooling air was entering the chamber at 200°F and .04 lb. water vapor per lb. of air, the enthalpy condition could be determined by a calculation of the partial pressure of the water vapor:

$$Pwv = \frac{H(P_T)}{.6218 + H} \qquad (6)$$

Where P_T = total pressure = 50 psia

H = .04 lb. water vapor per lb. of air

Pwv = Water vapor pressure (psia)

$$Pwv = \frac{(.04)\ (50\ psia)}{(.6218 + .04)} = 3.022\ psia$$

From the steam tables (Keenan, Keyes, Hill and Moore) the enthalpy at this condition is approximately 1149 BTU/lb. If the air was flowing at 10 lb. per min. this would represent a water vapor flow rate of 0.4 lb./min. If the chamber was initially at a density of 0.075 lb. per $ft.^3$, the quantity of air inside the chamber (197 $ft.^3$) would be:

$$\text{lb. of air} = (197\ ft.^3)\ (.075\ lb./ft.^3)$$

$$= 14.775\ lb.$$

Since the humidity must change at the rate of 0.04 lb. water vapor per lb. of air per minute, the removal rate of water vapor must be 0.591 lb. water vapor per minute. Therefore, the de-humidification coil must be capable of removing a total of 0.991 lb. water vapor per minute. If the chamber condition was -65° F and saturated, the enthalpy condition would be 1032.36 BTU/lb. (6). The total latent heat capacity of the coil would be:

$$Q_{latent} = M_{wv}\ (\Delta h)$$

$$= (0.991)(1149 - 1032.36)$$

$$= (0.991)(116.64) = 115.59\ BTU/min.$$

$$\cong 6935\ BTU/H.$$

Since the standard refrigeration coil (6) is designed for 60% sensible cooling, a coil with a total cooling capacity of 17,338 BTU/H would be selected. If a total face area of 5 $ft.^2$ were selected, this would represent a coil with a rating of 3468 BTU/H/$ft.^2$. The pressure drop of a coil with this rating (Singer-Wilmington - J fin (flat) - 1 row with 12 fins per inch) would be 0.035 in. H_2O at 1888 CFM.

Heaters

If the heat capacity was designed to equal the refrigeration capacity (16 tons) the total KW input required would be approximately 56 KW. Assuming that a minimum velocity of 500 ft. per min. could be attained across the heaters, a pressure drop of 0.024 in. H_2O would result for finned tube type duct heaters. (Electric Heaters Inc. - Custom Built Duct Heaters - Catalog EHD-72).

Turning Elbows

The pressure drop of the elbows can be calculated from the equation (6)

$$\Delta P = C\ (v/4005)^2$$

where: ΔP = pressure drop (in. H_2O)

C = coefficient approximately .9

V = velocity (ft./min.)

therefore for four elbows (considering two 90° elbows at each end of the chamber).

$$\Delta P = (4)(0.9)(2.9/4005)^2 = 0.011 \text{ in. } H_2O$$

The total system resistance is therefore:

$$\Delta P_{total} = \Delta P_{wall} + \Delta P_{coil} + \Delta P_{35° \; coil} + \Delta P_{heaters} + \Delta P_{elbows}$$

$$\Delta P = 0.0000013 + 0.0575 + 0.035 + 0.024 + 0.011$$
$$\Delta P = 0.128 \text{ in. } H_2O$$

Since all fan curves are rated at standard density, however, this static pressure drop must be corrected.

Figure 13 is a representation of a typical fan performance curve. Indicated on the sketch are curves for system resistance and also fan performance at certain densities and speeds. For any given system the fan will operate at the intersection of the fan performance curve and the system resistance curve. The above pressure drop calculations were done for sea level altitude and a temperature of -65°F. This results in a non-standard density of 0.100 lb./ft.^{3}.

This point of operation is shown by point A on figure 13 at which the fan would deliver 1888 CFM at a static pressure of .128 in. H_2O. In order to make proper fan selection, this point must be located on the standard density curve. This can be done by the following relationship: (10)

$$CFM_B = CFM_A \; (dA/dB)^{0.5}$$

therefore: $1888 \; (.100/.075)^{0.5} = 2180$ CFM.

Point B on figure 13 is the point at which the fan should be selected from the fan catalog as the operating condition at sea level altitude and temperature of -65°F.

A previous calculation showed that 58,965 CFM was required at an altitude density of 0.0044 lbs./ft.^{3}. This is shown as point C on figure 13. In order for the fan to be selected for this upper operating limit, it is again necessary to make a conversion to standard density. This can be done by using the following relationship. (10)

$$(S.P)_D/(S.P)_B = [(CFM)_D/(CFM)_B]^2$$

therefore: $(S.P)_D = (S.P)_B\ [(CFM)_D/(CFM)_B]^2$

$$= (0.128)[58965/2180] = 93.65 \text{ in. } H_2O.$$

The fan to be selected therefore must have the capacity to deliver 58,965 CFM at a static pressure of 93.65 in. H_2O. It is not available. Even if a fan of this type were available, the horsepower of the fan would be an inordinate load upon the air conditioning.

At this point of the design it was realized that a method of fan optimization was required in order to select a fan. The first step in the procedure was a calculation of the total steady state cooling loads in the chamber for different fan horsepower inputs. There was also an assumption that the temperature difference between the chamber and the cooling air was not at the maximum. Therefore:

$Q_{test\ item}$ = 17,065 BTU/H

Q_{CA} = (10)(0.24)(165)(60) = 23,760 BTU/H

Q_{5hp} = 12,725 BTU/H

Q_{10hp} = 25,450 BTU/H

Q_{15hp} = 38,175 BTU/H

Q_{20hp} = 50,900 BTU/h

Q_{25hp} = 63,625 BTU/H

The ratio of the heat load of the motor versus the total heat load was then plotted on ~~graph (15)~~. figure 15.

In addition, a graph (figure 14) was made of the system resistance curve for standard density and theoretical values of CFM versus static pressures were plotted on the same graph for different horsepower inputs between 5 and 25 hp. The calculation assumed a fan static efficiency of 80 percent and used the following equation (reference 10):

$$CFM \times SP = (SE)\ (6356)\ (BHP)$$

From this graph a determination was made of the fan performance for various horsepower inputs. The intersection of the respective curves represented the operating point of the system.

Finally the ratio of the operating point CFM to the horsepower input was plotted on graph 15 also. The intersection of the two curves on graph 15 represented the optimization point for the system. It represented the delivery of the maximum CFM for the smallest motor load to total load ratio. As can be seen, the horsepower determined was 15 hp. The system would operate at a static pressure of 5 inches and the fan could theoretically deliver 14,800 CFM at this condition. Since this represented a reduction on the mass flow capability of the system, one would expect a higher temperature rise in the chamber. This is calculated for the conditions at altitude (70,000 ft.) and a temperature of -65°F (395°R).

$$\Delta T = \frac{Q\ \text{total}}{MC_p} = \frac{79{,}000}{(0.0044)(14{,}800)(0.24)(60)}$$

$$\Delta T_{rise} = 84.25°F$$

In other words, at these conditions the air leaving the coil would increase 84.25°F during its circulation. This can certainly be viewed as a limitation on the system. The type of fan anticipated for use in this system would be an axial type capable of operating in the range of temperatures (-65 to 300°F) specified.

In order to evaluate the point at which the fan begins to lose its effectiveness, it is necessary to calculate a density and convert it to an altitude pressure at -65°F. Therefore:

since $Q = MC_p\ \Delta T$

$d = 79{,}000/(14{,}800)(0.24)(25)(60)$

$d = .015\ \text{lb./ft.}^3$

and

$P = d\ R\ T = (0.015)(53.3)(395) = 316\ \text{PSF}$

This is equivalent to an altitude of approximately 44,500 feet. The effectiveness therefore with this loading condition begins to deteriorate at this altitude and the temperature rise of the air through the chamber begins to increase.

In view of the problems encountered with the fan during operation at altitude, an alternative method of providing ventilation air to the chamber was investigated. If a vacuum pump was selected which was capable of evacuating 10,000 CFM at altitude, this would represent a mass flow of 33 lb./min. since:

$$\dot{M} = d(CFM) = \frac{(0.65)(144)}{(53.3)(530)} (10{,}000) = 33 \text{ lb./min.}$$

Instead of drawing this air from an atmospheric condition, an antechamber could be used to store conditioned air. The antechamber would consist of a receiver which would store compressed air which had been dehumidified and conditioned to 70°F. In order to supply a minimum mass flow to the receiver of 33 lb. /min., the compressor would have to be rated for a volumetric flow of:

$$CFM = \frac{33 \text{ lb./min.}}{0.075 \text{ lb./ft.}^3} = 440 \text{ CFM.}$$

The conditioned air in the receiver therefore would be delivered from the receiver through a valve in the end of the chamber and across the coils for further conditioning.

The amount of air drawn across the coils would then be evacuated by the vacuum pump.

The number of restrictions upon this type of system caused us to reject it as a workable conceptual design. This system would have to be used in a closed-loop cooling air concept since it would no longer have the capacity to evacuate the mass flow of air imposed by an open-loop cooling air conceptual design. Secondly, if the receiver was designed to hold 64 ft.^{3} (36 inch diameter x 96 inch long) of air at a pressure of 100 psi, the elapsed time before full evacuation of the receiver would be:

$$\text{Time} = \frac{(d)\ V}{\dot{M}} = \frac{(100)(144)}{(53.3)(530)} (64) \frac{1}{(33)} = 1 \text{ minute}$$

This is certainly not long enough to provide conditioning in the chamber.

Also the vacuum pump booster system would not be operating at a condition above a certain pressure and therefore the system could only be used at an altitude condition.

Cooling

The conditioning of the chamber utilizes mechanical refrigeration with expansion coils located in the air stream inside the chamber. As has been shown, the coils were sized for the full cross-sectional area of the chamber. The pressure drop across the coils affects the system resistance and this system resistance changes in proportion to the ratios of the sea level density and altitude density. In order to pre-cool the air prior to the low temperature cascade mechanical refrigeration coil, a standard mechanical refrigeration coil operating at a suction temperature of 35°F has been designed into the system. This will serve the dual purpose of pre-cooling the chamber air prior to the low-temperature coil and dehumidifying prior to the low-temperature coil. The suction temperature of the coil has been selected to minimize the affects of frosting on the coil during a dehumidification process. Although the cooling effects of a coil actually increase up to a certain level of frosting, (due to the increased surface area available for heat transfer) this soon reaches a maximum and its cooling effects decrease due to the increased pressure drop in the system. In view of the fact that both of these coils would be custom-made for the chamber it is felt that the design should attempt to include extended fins on the front rows of both the low-temperature and 35°F coils and eliminate them on the final rows of the coil. The benefits therefore, of extended surface heat transfer could be gained while eliminating the deleterious effects of frosting between fins. In effect, this type of construction would aid in the optimization of the coil resistance by eliminating fins on the final row where most of the dehumidification process is taking place.

As a means of evaluating the performance of the coils during dehumidification, curves were plotted of dew point temperature versus specific humidity for various pressures. See figure (17). The equation utilized to plot the curves was derived from Dalton's Law of Partial Pressures and the Ideal Law for water vapor and air.

$$Pwv = \frac{H(P_T)}{0.6218+H}$$

Where Pwv = Water Vapor Pressure (PSIA)

P_T = Total Pressure (PSIA)

H = Specific Humidity. (lb. Water Vapor/lb. Dry Air)

Values over the full range of specific humidities specified for the chamber were used in the equation. The water vapor pressure was then converted to a dew point temperature by referring to the steam tables. As can be seen on the graph, the entire range of specified humidities falls below a 35°F dew point, once the pressure-altitude point is above 50,000 feet. This means that any dehumidification above 50,000 feet would be handled entirely by the low-temperature coil. Therefore, above 50,000 feet, any dehumidification process should be followed by a cycle in which defrosting of the low-temperature coil is accomplished. Values of pressure above atmospheric were also plotted in order to aid in the determination of dew points at elevated pressures.

As mentioned, the low-temperature coil located in the chamber is part of a cascade mechanical refrigeration system. A cascade mechanical refrigeration system is one in which one refrigerant circuit is used to condense the refrigerant of a second operating at a lower temperature. This permits operation at ordinary pressures and achieves the inherent advantages of a single stage system. The selection of the low-temperature refrigerant with the proper pressure characteristics at low-temperature enables the compressor to be selected with reasonable compression ratios.

The calculations for the heat transfer capacities of the refrigerants were made under certain assumptions in order to simplify the analysis:

(1) a negligible amount of sub-cooling along the saturated liquid line in the condenser,

(2) the process through the expansion valve is along a line of constant enthalpy.

(3) the vapor leaving the evaporator is at a condition on the saturated vapor line.

(4) compression is isentropic.

Referring to the pressure-enthalpy diagrams for the two refrigerants considered for the low-temperature coil (R-13 and R-503) the following points have been indicated on the diagrams in order to aid in the understanding of the analysis:

Point 1 - Entering Evaporator
Point 2 - Entering Compressor
Point 3 - Entering Condensor
Point 4 - Entering Evaporator

Both the refrigerants were evaluated at a -100°F suction temperature and a -30°F condensing temperature. The following calculations can be done using the chart:

R-13

Section Temperature = -100°F
Evaporating Pressure = 7.6 PSIG
Condensing Pressure = 90.9 PSIG

$$\text{Compression Ratio} = \frac{90.9 + 14.7}{7.6 + 14.7} = 4.8$$

Refrigeration Effect = 46.4 BTU/lb.

$$\text{Mass Flow Per Ton} = \frac{12000 \text{ BTU/lb./Ton}}{60 \frac{\text{min.}}{\text{H}} (\Delta H)} = \frac{200}{46.4} = 4.31 \text{ lb./min./Ton}$$

Specific Volume @ -100°F = 1.56 Ft.^3/lb.

$$\text{Compressor } \frac{\text{CFM}}{\text{Ton}} = (M) \times (V) = (4.31)(1.56) = 6.74 \frac{\text{CFM}}{\text{Ton}}$$

$$\text{Compressor H.P. Per Ton} = \frac{(M)\ (\Delta H)\ \text{COMP}}{42.4 \frac{\text{BTU/min.}}{\text{HP}}} =$$

$$\frac{(4.31)(11.12)}{42.4} = 1.13 \frac{\text{HP}}{\text{Ton}}$$

R-503

Suction Temperature = -100°F
Evaporating Pressure = 16.3 PSIG
Condensing Pressure = 135.5 PSIG

$$\text{Compression Ratio} = \frac{135.5 + 14.7}{16.3 + 14.7} = 4.8$$

Refrigeration Effect = 54.0 BTU/lb.

$$\text{Mass Flow Per Ton} = \frac{200}{54.0} = 3.70 \text{ lb./min./Ton}$$

Specific Volume @ -100°F = 1.30 Ft.^3/lb.
Compressor CFM Per Ton = (3.20)(1.30) = 4.8 CFM/Ton

$$\text{Compressor HP Per Ton} = \frac{(3.70)(72-56)}{42.4} = 1.2 \frac{\text{HP}}{\text{Ton}}$$

As can be seen, the refrigeration effect of R-503 is better than R-13 with the same compression ratio and horsepower requirements. R-503 can also run at -120°F suction temperature and still run at positive pressure while R-13 is limited to -100°F suction temperatures. Due to these qualities, R-503 should be considered for the working fluid on the low-temperature side of all the cascade mechanical refrigeration systems.

The evaporator coil is cooled not by contact with the air but by actual contact with the refrigerant charge inside the coil. The air itself cannot be cooled unless the temperature of the coils has been decreased sufficiently. It is possible to eliminate the time required to cool the coils and refrigerant charge by precooling, which would therefore not be part of the required pulldown time. The control system could begin the pulldown of the coil to the anticipated condition while maintaining the zero time condition with reheating. At the preset time, the reheat cycle would stop and the coils would begin their chamber pulldown having already reached their steady state new load temperature.

There are some particular problems peculiar to the actual design of mechanical refrigeration systems which become more acute in the low temperature multi-stage systems than in the average high temperature units. One is the ability to return oil from the evaporator to the compressor. In general, the direct expansion type of evaporator is likely to give the least oil problems since there is a continuous flow from one end to the other and all of the fluid, refrigerant and oil, which comes in one end should go out the other end.

Unless the flow rate is extremely low, the concentration of oil sufficiently high, or the system is at a low enough temperature to result in separation of oil in a form which adheres to the walls of the evaporator, there is little possibility of the oil remaining behind. Particular care, however, should be given to this problem in the actual design of the system.

Also, since a compressor can handle a greatly increased mass flow of refrigerant at the higher absolute pressures corresponding to the starting conditions than it can handle at the low absolute pressures in the final operating conditions, some degree of capacity control must be included. The means of controlling compressor capacity and evaporator capacity is about the same with multi-stage systems as with single stage systems. One of the most effective and simplest methods of taking care of this is to feed discharge gas into the liquid line between the expansion valve and the distributor of the direct expansion evaporator; the flow of the discharge gas can be controlled by evaporator pressure, using a constant pressure expansion valve or any other way which is appropriate. This supplies an artificial load to the evaporator equal to the difference between the lowest capacity at which the system will operate satisfactorily and the lowest actual load which will exist when operation is required. This type of control is generally referred to as hot gas bypass.

Transient Loads

As mentioned previously, it was desired to isolate the loads inside the chamber which could be considered as steady state and those which could be considered transient. In the case of the chamber conditioning the steady state loads have already been discussed. These steady state loads, namely the test item, cooling air, and blower load total approximately 93,400 BTU/H. These items will continue to load the air conditioning system before, during and after the pulldown period for any two chamber conditions.

Certain other items however exert a load on the system due only to the difference in its temperature before and after the start of the pulldown period. The heat transfer between the objects and the air being conditioned changes over a period of time due to the gradual elimination of the temperature difference between the mass of the item and the air temperature. If these items could be conditioned by means other than the mechanical refrigeration unit, a savings would likely result in the design due to a reduction in the overall capacity and capacity modulation features of the mechanical refrigeration system. The refrigeration coils themselves are usually one of the transient loads in the system, however this transient load can be eliminated by anticipation of the pulldown period and a time allowance using reheat for coil pulldown.

The additional transient loads in the chamber are:

1) chamber walls & mass insulation
2) fiberglass shroud
3) test item mass
4) fan casing

The design of the walls of the chamber must attempt to balance the advantages of insulation in reducing the heat transfer through the chamber walls and the disadvantages of the load imposed on the air conditioning system by the mass of insulation used. Studies (3) have shown that as little as 1/4 inch to 1/2 inch inch of wall thickness has changed temperature during a fast pulldown period similar to one that would exist in the CERT chamber.

Pownall and Soling (7) did studies on a chamber wall with large temperature changes and mass type insulation and developed methods of predicting transient loads for certain pulldown times and ranges. Graph 16 is an extrapolation of their curves for our particular application. As can be seen, the plot is for tons of refrigeration required per 1000 square feet of surface area versus pulldown time in minutes. This was done for several pulldown ranges. In the case of the CERT chamber the pulldown time for the maximum change is:

$$\frac{\text{Pulldown Range}}{\text{Pulldown Rate}} = \frac{300 - (-65)}{40°\text{F/min.}} = 9.125 \text{ minutes}$$

From the graph therefore, the load for this time period would be 34 tons/1000 $ft.^2$. For a chamber surface area of 175 $ft.^2$, this would represent a load of:

$$Q = \frac{(34 \text{ tons})}{1000 \text{ ft.}^2} (175 \text{ ft.}^2) = 6 \text{ tons} = 72{,}000 \text{ BTU/H}$$

Similar calculation for the fibergalss shroud yields a total load of approximately 3 tons.

The test item mass imposes an additional load upon the chamber conditioning beyond its input of 5 kW. The above reference (7) also performed heat studies of the heat load imposed by objects directly in the air stream as our test item is. As a result of the study, a chart was developed from which one could determine a heat load factor for uninsulated objects requiring certain pulldown times. The heat load factor was determined for a certain pulldown times. For the test item in the test chamber therefore:

Weight = 250 lb.
Size = 3 feet x 3 feet x 3 feet
Specific Heat = 0.12 BTU/lb/°F (mild steel)

consequently:

$$F = \frac{\text{sq. ft. surface area}}{(\text{WT})\ (\text{SP.HT})} = \frac{4 \times (3 \times 3)}{(250)(0.12)} = 1.2$$

The heat load factor was determined (7) to be equal to 0.3.

The total heat load of the item for the pulldown time is therefore:

$$\text{Heat Load} = \frac{(\text{Heat Load Factor})\ (\text{WT})\ (\text{SP.HT})\ (\Delta T)}{200\ (\text{Pulldown Time})}$$

For a pulldown range of 300 to -65°F the ΔT = 365°F and the time is 9.125 minutes therefore:

$$\text{Heat Load} = \frac{(0.3)(250 \times 0.12)(365)}{(200)(9.125)} = 1.8 \text{ tons}$$

Pulldown Time

In order to evaluate the operation of the mechanical refrigeration unit, a study was done on its operating temperature at the last stages of pulldown. Holladay (reference 3) provides a method of analysis which is suitable for our purposes. The first step in the analysis is to construct R-503 compressor curves for various volumetric capacities. Values of refrigeration effect (change in enthalpy) were determined by referring to the pressure - enthalpy diagram for R-503 between suction temperatures of -100°F and -30°F. Corresponding values of refrigerant properties and capacities were then evaluated at those conditions. The following chart was then constructed:

Temp °F	RE-ΔH BTU/lb	M=200/ΔH lb/min/ton	V CF/lb	MxV CFM/ton	Compressor Capacity (Tons) 50 CFM	60 CFM	75 CFM	100 CFM
-100	54	3.70	1.30	4.80	10.42	12.50	15.62	20.80
-90	54	3.70	1.01	4.25	11.76	14.12	17.64	23.53
-80	55	3.60	0.80	2.88	17.36	20.83	26.04	32.72
-70	55	3.60	0.64	2.23	22.42	26.91	33.63	44.84
-60	56	3.57	0.52	1.96	25.51	30.61	38.26	51.02
-50	57	3.51	0.42	1.47	34.02	40.82	51.02	68.03
-40	58	3.45	0.35	1.21	41.32	49.58	61.98	82.64
-30	60	3.33	0.29	0.96	52.08	62.50	78.12	104.16

Graphs 12 (A, B, C, D) were constructed for values of compressor capacity (tons) versus refrigerant temperatures. Evaporator curves were drawn on the same graph. The intersection of the evaporator curves and compressor curves is the operating point of the system. The evaporator selected was designed to deliver air at -65°F with a refrigerant temperature of -100°F. From this curve, the compressor capacity at various air temperatures could be determined.

The specification of a 40°F/minute chamber pulldown rate results in a pulldown time of 9.125 minutes for a chamber pulldown from 300°F to -65°F. This was used in the calculation of the total BTU levels resulting from the following load inputs.

Loads	Calculation		Total BTU
Test Item	(5KW)(3413 BTU/H/KW)(1H/60min.)(9.125min.)	=	2595 BTU
Cooling Air	(10lb./min)(9.125min.)(0.24)(200 + 65)	=	5804
Blower	(15HP)(2545BTU/H/HP)(1H/60min.)(9.125min.)	=	5806
Room Air	(198 ft.^{3})(0.044 lb./ft.^{3})(0.24)(365°F)	=	76
Fiberglass	(0.8)(72.34)(0.25)(365)	=	5281
Fan Casing	(0.8)(185)(0.11)(365)	=	5942
Test Item Mass	(0.3)(250)(0.12)(365)	=	3285
			28,789 BTU

By comparing this total BTU to the total temperature change inside the chamber, we obtain a "thermal rating". This represents the average BTU per degree over the pulldown range.

therefore: $\frac{28,789 \text{ BTU}}{365°\text{F}} = 78.8 \text{ BTU/°F}$

A comparison of the compressor capacity and total capacity required for air temperature reduction can then be made. For the analysis, the pulldown range selected was the most difficult range, i.e. the lowest temperature portion of the pulldown. This was a condition from -15°F to -65°F which represents a 50°F pulldown. The following chart can then be constructed for a 75 CFM compressor.

Air Temp.	ΔT	Comp. Capacity	BTU = (BTU/°F)(ΔT)	Time
-15				
-30	15°	23 = 276,000	78.8(15) = 1182	0.257
-40	10°	20 = 240,000	78.8(10) = 788	0.197
-50	10°	18 = 216,000	78.8(10) = 788	0.219
-60	10°	16 = 192,000	78.8(10) = 788	0.246
-65	5°	15 = 186,000	78.8(5) = 394	0.127
				1.046

This represents a pulldown rate of:

50°F/1.046 = 47.8 °F/Min.

Performing a similar analysis for a 60 CFM compressor yields the following:

Air Temp.	ΔT	Comp. Capacity	BTU = (BTU/PF)(ΔT)	Time
-15				
-30	15°	18 = 216,000	1182	0.328
-40	10°	15 = 180,000	788	0.263
-50	10°	14 = 168,000	788	0.281
-60	10°	13 = 156,000	788	0.303
-65	5°	12.5 = 150,000	394	0.158
				1.333 min.

This results in a pulldown rate of 37.51°F/minute.

The specification of 40°F/minute lies between these two capacities. In the design and selection of the cascade system, the volumetric efficiency of 75 percent could be expected. The actual efficiencies of the compressor at the operational compression ratio and suction temperature must be obtained from the compressor manufacturer in order to make the final design. The volumetric capacities used for these calculations represent actual compressor capacities.

The selection of total kW input of the heaters should approach the total cooling capacity of the refrigeration unit. This ensures full reheat capacity during a dehumidification process. Wherever possible, radiant heaters should be located in order to reduce the transient load of the chamber walls during a period of transition from low temperature to high temperature. Open element electrical resistance heaters may be used if condensation on the heaters at low temperature can be avoided. The pressure drop calculations for the fan selection were based upon electrical heaters of the metal enclosed tubular type. Arcing between terminals under vacuum conditions may be avoided by locating the terminals outside of the vacuum space through a vacuum tight fitting. Electrical heaters have advantages over other types for several reasons. They have a low mass which benefits the pulldown time for the chamber when it is operating on a cooling cycle. Secondly, due to their electrical feature, modulation of capacity is easily accomplished. They have a very quick response time to input. Finally, they present minimal resistance to airflow.

There are several safety precautions which must be taken in order to ensure proper operation of the heater. Since with most coils there are minimal airflow requirements across the elements, care must be taken in sizing of the individual heaters. There should also be a safety control which eliminates operation of the heater if the chamber air fan fails to operate.

SECTION IV

ALTITUDE GENERATION

The selection of the proper capacity of a vacuum pump for a given installation is straightforward in principle but depends on many operating factors contained in the setup. The first two factors to be considered are:

(a) the operating pressure required, and

(b) the pumping down time desired.

The first factor determines the type of pump required (single stage, compound, mechanical booster, diffusion) as well as the minimum pumping speed (volumetric flow rate) required to maintain the operating pressure. The second factor determines the minimum pumping speed required to pump down in the desired time. The first depends on the rate of flow and evolution of gas in the system and the second depends primarily upon the volume to be evacuated. The last factor to be considered is the type of gas to be evacuated. Large quantities of condensable vapor are more easily handled by certain types of pumps.

Due to the pressure (0.65 psi) and the volume of the gas to be evacuated, a mechanical booster pumping system was selected. This system combines a positive displacement, lobe-type rotary booster pump with a rotary piston high vacuum second stage pump. The system operates automatically from atmospheric pressure to its blank-off pressure. Each pump is equipped with its own motor and the booster is automatically controlled by a self-contained pressure switch. When started, the second stage pump roughs out the system to the cut-in pressure of the booster pump. At this point the booster pump automatically cuts in, reducing the prime utility of the second stage pump to that of a backing pump for the booster operation. When started, the booster pump quickly reaches maximum pumping speed. If the pressure should rise above the set cut-in pressure, the booster automatically stops operation and the second stage pump resumes with its roughing function until the cut-in pressure is once again reached. A temperature switch protects the booster pump in the event of excessive temperature.

This system has many advantages to it. It can operate at maximum efficiency over a wider range of low pressures. This advantage is in contrast to conventional mechanical pumps which gradually decrease pumping speed as the pressure drops. Therefore, conventional systems have a relatively low volumetric efficiency in the low range. Another advantage is that the booster pump is free of sealing oil. This makes the system cleaner in operation. It provides a far cleaner pumping action than that of either a conventional oil-sealed mechanical high vacuum pump or that of an oil ejector pump. The system is completely automatic, requires minimum floor space, and has a low dollar per cfm cost.

Due to the high humidity conditions involved, a liquid ring vacuum pump was selected as the second stage pump. The liquid ring pump is a non-pulsating, vibrationless machine that removes gases by means of rotating impeller blades entering and leaving a ring of liquid, usually water. Therefore, it is especially suited for application requiring clean, oil-free vacuum pumping and for processes that liberate water vapor. When the vapors entrained in the air flow contact the liquid, some condensing action takes place. Thus, it functions as a direct contact condenser to increase the pumping speed for vapor without appreciably impairing its pumping speed for air. This characteristic offers functional as well as economic advantages in contrast to oil sealed pumps which may have to depend on water cooled condensers or refrigerated traps to maintain efficiency. The pumping mechanism of the liquid ring vacuum pump is also insensitive to contamination by liquids and vapors if they are compatible with the sealing fluid. Liquid slugs do no mechanical harm to the pump.

Even through the mechanical booster system was selected, different capacities are needed for the different concepts developed. For the open loop compressor system (AFFDL extension), utilizing a 4 foot diameter by 5 foot long chamber, the following applies:

<u>Open Loop Compressor</u>

Volume Chamber - 62.8 cubic feet

$$S_1 = 2.3 \frac{V}{t} \log \frac{P_1}{P_2}$$

V = volume

t = time to pump down

t for a $\pm$ 40,000 ft./min. rate and 70,000 ft. max. is 1.75 minutes

P_1 = initial pressure

P_2 = final pressure

S_1 = capacity without forced air flow

$$S_1 = \frac{2.3(62.8)}{1.75} \log \frac{760}{33}$$

S_1 = 112.77 CFM

With a forced airflow of 10 lb./min., referring to the Wright Patterson Air Force Base report (1), the relationship for the additional vacuum pump capacity is:

$$Vp = \frac{V_F P_1}{P_2}$$

$$Vp = \frac{\text{[illegible]}0(760)}{0.075(33)}$$

$$Vp = 3070.63 \text{ CFM}$$

P_1 = initial pressure

V_F = forced air flow

P_2 = final pressure

Vp = additional pump capacity with forced air flow

Therefore the total capacity should be:

$$112.77 + 3070.63 = 3183.4 \text{ CFM}$$

For the two additional conceptual designs we will consider a 6 foot diameter by 7 foot long chamber.

Open Loop Blower

Without forced air flow:

$$S_1 = 2.3 \frac{V}{t} \log \frac{P_1}{P_2}$$

Volume = 197.82 cubic feet

$$S_1 = \frac{2.3(197.82)}{1.75} \log \frac{760}{33}$$

$$S_1 = 354.11 \text{ cfm}$$

With forced air flow, capacity is 354.11 + 3070.63 = 3424.74 cfm

Closed Loop Blower

Including 10 square feet of ducting, the capacity becomes:

$$S_1 = \frac{2.3(207.82)}{1.75} \log \frac{760}{33}$$

$$S_1 = 372.01 \text{ cfm}$$

Since this is a totally closed system this is the actual capacity needed.

One additional aspect must be discussed concerning the required capacity of the vacuum pumping system. Since the ultrasonic humidification operates using air pressure, it is injecting additional air flow into the chamber which must be removed. A selection of nozzles was based on the following information:

Cooling Air

10 lbs./min. maximum air flow

$0.04 \frac{\text{lbs. } H_2O}{\text{lb. air}}$ maximum

$$\frac{10 \frac{\text{lbs.}}{\text{min.}} \left(0.04 \frac{\text{lbs. } H_2O}{\text{lb. air}}\right)}{8.345 \frac{\text{lbs.}}{\text{gal.}}} \times 60 \frac{\text{min}}{\text{hr.}} = 2.88 \frac{\text{gal. } H_2O}{\text{hr.}}$$

Chamber Air

$150 \frac{\text{lbs.}}{\text{min.}}$ maximum air flow

(supplied from fan data)

$0.04 \frac{\text{lbs. } H_2O}{\text{lb. air}}$ maximum humidity

$$\frac{150 \frac{\text{lbs.}}{\text{min.}} \left(0.04 \frac{\text{lb. } H_2O}{\text{lb. air}}\right)}{8.345 \frac{\text{lbs.}}{\text{gal.}}} \times 60 = 43.1 \frac{\text{gal.}}{\text{hr.}}$$

This is the amount of water added during maximum humidification in each case. A selection of nozzles was made to deliver this water. These require the following air flow, air pressure and liquid pressure to operate:

Cooling Air

#052 nozzle
3.5 psi liquid pressure
11 psi air pressure
1.1 scfm air volume added

Chamber

#281 nozzle
4 psi liquid pressure
12 psi air pressure
25.5 scfm air volume added

Mass flow added is the following:

$$1.1 \text{ scfm } (0.075 \frac{\text{lbs.}}{\text{cf}}) = 0.0825 \frac{\text{lbs.}}{\text{min.}} \quad 24.4 \text{ scfm } (0.075) = 1.91 \frac{\text{lbs.}}{\text{min.}}$$

For these two air flows added, we need an additional capacity of the vacuum pump of:

Total air flow = 26.6 scfm

$$Vp = \frac{V_F P_1}{P_2}$$

$$Vp = \frac{26.6 \text{ scfm } (760)}{33}$$

Vp = 612.61 cfm additional

This would be added to the values calculated for both closed and open loop blower systems. Since in an open loop compressor system we are only using ultrasonic humidification in the chamber, the additional capacity becomes:

$$Vp = \frac{25.5\ (760)}{33} = 587.27 \text{ cfm}$$

SECTION V

VIBRATION

The requirements given for the CERT facility lead to only one possible method for obtaining them; the use of an electro-mechanical vibrator, specifically a Ling system, Model D300.

Since the shaker is cooled by a completely enclosed tubular system carrying distilled water, the need for outside atmospheric pressure air for cooling is not needed. The inside of the shaker can be evacuated and can operate through the chamber wall without the atmospheric pressure loading problem. There is no tendency for the chamber pressure to "suck" the armature out of the shaker body.

If the shaker was placed inside the chamber, the chamber would have to be large enough to house the shaker and strong enough to support the shaker weight. This would make the chamber extremely large and heavy. Furthermore, the large load of the shaker would have to be overcome by additional capacity of the system. Therefore, it is obvious that the shaker should be in a piggy back arrangement with the chamber.

In the piggy back arrangement, (see figure 19) the top surface of the shaker, which is machined, forms a part of the chamber wall. A pressure seal gasket (o-ring plus gasket) is placed between the shaker body machined surface and the chamber. The plane of the shaker table is inside the chamber. Most of the shaker body is outside. The inside of the shaker is sealed from the chamber by a low pressure diaphragm seal.

The top thermal accessories include a laminated fiberglass thermal barrier which covers the armature table. It is held in place by the table inserts and perhaps additional screws threaded into the armature top depending on the size. Silicone rubber shims are used to help prevent the thermal barrier from vibrating. The normal stainless steel load mounting standoffs on the armature are replaced by higher units which are designed to clamp the thermal barrier and elevate the interface of the fixture/payload above the armature table to provide additional isolation against thermal conduction.

The control system selected for vibration is the Ling, Model 61000 Digital Vibration Control and Analysis System. It gives the response of an analog system with the convenience of digital computer control. The operator specifies test parameters through a single conversational language and hence set-up time is greatly reduced. Test results can be recorded and repeated with near perfect accuracy.

The system itself consists of the following components:

(56K)
1) Two Bay Cabinet
2) Vibration Control Panel
3) Model 1400 Analog Input System

4) Model 1410 Analog Output System
5) Model 1540 Real Time Clock
6) Model 1753 Teletype System
7) Model LDP-1801 Computer
8) Model 1802 Expansion Chassis
9) Model 1811 Memory (32K)
10) Model 1820 Priority Interrupt System
11) Random Control and Analysis Software Package

The following options are also recommended and included in the price of the control system in the cost analysis:

(3K) 12) Sine Control and Analysis (which includes Model 1411 Programmable Sine Generator)
(4K) 13) Model 1752 High Speed Reader and Printer
(2K) 14) Model 1530 Two digital to analog converters under computer control for X-Y Plotter and Oscilloscope

The approximate prices are shown in the left hand column.

The system is capable of single channel random control and analysis to 25 KHz and single channel sine control. With the memory supplied the system is capable of storing up to ten individual programs and although it is a single channel system it can accept up to 8 channels with the addition of a Model 1409 input channel for each additional channel desired. The system is modularly upgradeable with control and analysis being performed by software. This means easy expansion and modification.

The heart of the system is the Model 1801 Digital Computer. It is a 16K word, 1 msec core memory expandable to 32K words. Its unique features are automatic bootstrapping, memory parody, hardware multiply and divide (faster then software), vector interrupt and power failure auto restart. Data is transferred up to a 10 KHz rate over the programmed Input/Output bus. This would be used with all the standard control functions plus the high speed paper tape reader and punch and the display for the oscilloscope and X-Y plotter.

The shaker itself along with its control system is easily capable of meeting all sine and random requirements for this weight test item. No other additional equipment is needed.

SCETION VI

ELECTRICAL GENERATION

6.1 Introduction

The nominal power requirements for units under test in the CERT facility are specified as 15 KVA of 3-phase, 4-wire, 400 Hz and 28 VDC at 100 amperes. The control of the quality of this power shall be in accordance with the requirements of specification MIL-STD-704A. The total power available during the AC test simulations shall be a maximum of 15 KVA at 120 VAC line-neutral and the total current available during the DC test simulations shall be a maximum of 100 amperes.

6.2 Description of Power Sources

(1) 3-phase AC at 15 KVA

Alternating current power sources capable of simulating the characteristic variations in voltage or frequency or phase angle as required by MIL-STD-704A must be electronic types in order to meet the response demands of test parameters. In addition, the power source must be programmable in all parameters of the test requirement.

The specific system contemplated for the CERT facility shall be capable of manual or computer control. All MIL-STD-704A AC voltage and frequency tests shall be under direct computer program control. Dynamic responses of the system are such that there are no interactions between the parameters of voltage, frequency and phase angle when any one function is varied during a MIL-STD-704A test program.

During manual control of the AC power source, the operator has total and complete control over each phase amplitude, phase angle and overall frequency via thumbwheel switches. If a single phase source of power is required, electrical rewiring at the output terminals will make available a power level of 5 KVA. If the AC power source was left connected as a three phase system, and only single phase power was required, then only 3 KVA could be supplied while maintaining a load regulation of $\pm$ 1 percent.

(2) DC Power at 100 Amperes

The direct current power source is capable of simulating the characteristic variations in voltage and ripple content as required by MIL-STD-704A while under direct computer program control. If there is a requirement, manual adjustment of voltage and/or current limits is possible, without the aid of the computer. The no load to full load or full load to no load transient response of the system is better than 200 microseconds.

(3) ± 600 VDC Spike

This MIL-STD-704A test shall not be under computer control because of its simplicity in application. The test technique shall employ a capacitive discharge coupled with wave shaping circuitry. The required waveform shall be established with a 50 ohm resistive load prior to connection to the unit under test DC power lines. The power source shall then be charged to 600 VDC and the proper polarity, and then discharged (injected) onto the DC power line.

SECTION VII

CONTROL SYSTEM

7.1 Operation

Referring to the processor block diagram (figure 7), a single minicomputer with a nonvolatile memory will control all three test chambers and the MIL-STD-704A electrical tests. The real time clock is a programmable unit that will interrupt the processor at a fixed frequency. All processor inputs and outputs for control functions will be in response to the real time clock. The teletype will be used for initial program development and debugging. The final operating program will be stored on a diskette. The teletype will also be used to initiate any test.

The operator will have the facility to select the method of data entry by setting a toggle switch on the processor control panel. Data may be entered into the program via the teletype paper tape reader or a diskette, with a diskette being the preferred input medium. Each diskette may store many test runs. To execute any test the operator need only type in on the teletype the track name of the appropriate test.

The diskette system to be used shall be a dual system. One diskette will be used exclusively for program storage and bootstrap loading. The second diskette will have a number of files of formatted data corresponding to each test to be run. The operating program for environmental testing will be structured to accept correctly formatted input data from the diskette. The formatted data will include temperature, humidity, air flow, and altitude all as a function of time. Only one general purpose environmental test program will be required for this type of data input. The header sector at the beginning of each track will identify the type of test to be run, either MIL-STD-704A or environmental and the chamber under test (1, 2, or 3). The dual diskette system will also incorporate a write protect feature and automatic positioning at power on, to facilitate bootstrap loading of the program.

After the automatic program load, the processor will set the initial parameters as determined by the data input, and check that all parameters have stabilized by monitoring the transducer inputs before commencing the test. All input parameters will be switched at the designated time in response to the processor's real time clock. A single processor shall control up to three chambers simultaneously.

The processor will output six parameters for each chamber under test once per second to a remote LED display panel. The same parameters will be printed on a line printer to provide a permanent hardcopy record of the test. The operator will have the ability to delete the hardcopy output by setting a toggle switch on the processor control panel. The processor output to the remote LED displays and line printer, will indicate the cooling air mass flow, temperature, humidity, and the chamber pressure, temperature, and humidity. Six LED displays will thus be provided for each of the three test chambers.

All external hardware that interfaces to the processor's bus is designed to be modular in nature. Each environmental chamber that is controlled by the processor will have a dedicated analog multiplexer and 12 bit A/D converter. This approach will allow simultaneous control of up to three chambers through foreground/background computer processing.

7.2 Processor Software Functions

The software functions performed by the processor will include sampling all transducer inputs and correlating them with the real time clock to perform real time switching of the appropriate controls. The program will be structured to incorporate anticipatory changes in the system due to the ramping of one or more control functions. For example, if the data input calls for ramping temperature as a function of time, the program will automatically compensate the humidity controls to maintain the desired level.

Another typical software function is the computation of mass flow. The transducer inputs corresponding to upstream temperature and pressure and the downstream pressure will be substituted into an orifice equation to compute mass flow. Thus mass flow is a computed value and is not directly measured by a mass flow meter. This approach will yield both better accuracy and resolution.

The system software will also compensate for all transducer offsets. When the program is initialized, the computer will scan all transducer inputs and form a table in memory of the required error correction for each transducer. All transducer inputs during system operation will then be added/subtracted to the appropriate offset to obtain a true reading. For any transducer which is out of specification, a fault message will be printed on the line printer for operator action.

7.3 Diskette Input Data Format

The data input from the diskette will be structed in matrix form. The data matrix will describe temperature, humidity, air flow, and altitude as a function of time. This format of data entry will facilitate a universal environmental program where all parameters that change are entered via the diskette. Only one program is thus required for environmental testing by utilizing this approach.

The MIL-STD-704A diskette data input will be similarly structured. The parameters of interest will be voltage and current as a function of time expressed in matrix form.

7.4 MIL-STD-704A Electrical Control

The selected processor can control one MIL-STD-704A electrical power test station. The programmable AC oscillator and the programmable DC source are controlled and addressed by the processor in a manner similar to memory addressing. The processor will initiate the test as specified from the diskette data input. The AC and DC sources will be digitally addressed and controlled as a function of time for the duration of the test.

7.5 Digital Computer Control Versus Analog Recording Control

The use of a digital computer for process controlling offers distinct advantages over analog control techniques. This section will briefly outline a few of those advantages.

(a) Speed - Digital data from a diskette can be entered into memory at a transfer rate in excess of 75,000 words per second. This is a significant improvement in the system throughput rate.

(b) Accuracy - A 16 bit minicomputer can resolve data to one bit (2^{16}) or one part in 65,536. This is a considerable improvement over most analog processing techniques.

(c) Data Formatting - If the diskette data input is structured in matrix form as previously described, the data can be edited or changed one element at a time without altering any other data. This is a considerable advantage when updating or generating new test runs.

(d) Data which is recorded on magnetic tape and played back to control equipment is subject to tape synchronization errors and the possible loss of data due to a bad section of tape. This can cause inoperation of the system with unpredictable results on the process under control. Data which is recorded digitally is not subject to this type of error. There are no digitizing or quantization errors which are inherent in analog recordings.

7.6 Equipment Configuration

Six identical cabinets will be required to mount the hardware shown in the processor block diagram. Each cabinet will be a standard 6 foot tall cabinet for mounting 19 inch RETMA racks on EIA and WE standards.

One cabinet will be required to house the processor and the related peripherals and interface hardware. Two cabinets will be used to mount the equipment used for the DC testing in MIL-STD-704A. Three cabinets are necessary for the MIL-STD-704A AC testing.

7.7 Control

Sensors

The requirements for the location of the sensors located in the conceptual designs discussed depend upon the particular design being reviewed. As an example in the open loop system the flowmeter can be located prior to the conditioning equipment. Since the flowmeter is located after the de-humidifier the effects of water vapor flow will have little effect upon the mass flow calculation. In the closed loop system however, the measurement device will possibly be affected by the water vapor flow. Regardless of the measurement system the measured mass flow is a combination of dry air and water vapor:

$$M_{total} = M_{DA} + M_{water\ vapor}$$

The mass flow of the water vapor is the product of the air flow and the specific humidity, therefore:

$$M_{total} = M_{DA} + M_{DA}\,(H) = M_{DA}\,(1 + H)$$

The actual mass flow of dry air is therefore the measured mass flow divided by the correction factor of 1 + H:

$$M_{DA} = \frac{M_{total}}{1 + H}$$

For a condition of 10 lb./min. dry air flow and a specific humidity of 0.04 lb. water vapor per lb. of dry air this becomes:

$$M_{total} = 10(1 + 0.04) = 10(1.04) = 10.4 \text{ lb./min.}$$

for this reason a dew point sensor was located in the air stream before the flowmeter in the closed loop system. The correction factor could be programmed into the computer for automatic data reduction. In addition to the problem of specific humidity correction the necessary corrections for temperature and pressure effects must also be included in the mass flow measurement systems. This is due to the fact that most measurement systems are for volumetric flow rates that require correction for densities. This poses no serious problem to the open loop systems, however, the wide temperature and pressure variations of the closed loop systems require careful considerations of the problems inherent to its design.

In view of the size limitations in terms of available flowmeters of the turbine type, we anticipate the need for orifice or venturi type flowmeters to be used in the closed loop blower and open loop blower systems. The advantage of a computer type control system is that of pre-programmed equations for the flowmeter selected and inputs to the computer as a result of temperature, pressure and humidity sensing at the desired location.

A copper constantan type thermocouple is capable of operating in the range of -300°F to 500°F and therefore would be an adequate type to use in the facility. There are numerous temperature controllers available which could handle the fast change rates of which the chamber must be capable. One particular type which we recommend is the trendtrak programmer. A black-ink record of the program desired is drawn on a special disk which is then mounted on the program unit. A photo-electric detector continuously senses the drawn curve. As the disk turns the photo-electric detector "tracks" the ink-line, changing the position of the servo-driven shaft and consequently the set point of your controller and the input to your process.

The measurement of specific humidity is also complex since there are no devices available which measure it directly. The condensation type dew point hygrometer is one of the most accurate, reliable and wide range sensors available for humidity measurements. With proper attention to the condensate surface temperature measurement, system errors can be reduced to less then $\pm$ 0.5°F. The conversion to specific humidity must be made by use of the equation:

$$H = 0.62198 \frac{Pwv}{P - Pwv}$$

The total pressure of the system could be monitored and the values of the water vapor pressure could be obtained by preprogramming water vapor pressures for the dew point temperature range anticipated in the system. The computer would automatically reduce the input data by solution of the equation with required input.

SECTION VIII

CONCLUSIONS

The advantages and disadvantages of each particular part of the concepts are listed below:

Cooling Air

	Advantages	Disadvantages
Open Loop Compressor	1) small desiccant drier 2) Little frosting on low temperature coil 3) high pressure allows use of extremely small lines, flow sensors and valving 4) air compressor and drier can be located in a room some distance from chamber setup	1) larger vacuum pump required to remove cooling air injected
Open Loop Blower	1) lower compression 2) mechanical refrigeration 3) little frosting on low temperature coil 4) allows use of conditioning plenum for heat exchanger	1) large high cost desiccant dryer 2) larger vacuum pump required to remove cooling air injected
Closed Loop Blower	1) small vacuum pump 2) allows use of conditioning plenum for heat exchangers	1) large blower required 2) cryogenic cooling required 3) desiccant dryer not feasible 4) entire system requires design for vacuum condition 5) frosting of low temperature coil

Chamber Air

	Advantages	Disadvantages
Annulus Design	1) less heat loss of conditioned air through annulus 2) less turbulence	1) large ΔT through length of chamber 2) shroud adds additional heat load 3) radiant heaters cannot be used for latent heat load on cold wall

	Advantages	Disadvantages
Open Air Design	1) smaller ΔT in chamber due to mixing 2) allow addition of heaters onto cold wall to remove latent heat load of cold wall	1) larger heat loss of conditioned air 2) larger turbulence

From the above parts a system for CERT requirements may be selected.

SECTION IX

ECONOMIC ANALYSIS

TASK III

As a result of a review of the technical analysis the Air Force chose two concepts for cost analysis. These were the open loop blower cooling air (figure 2) with AFFDL extension design chamber (figure 4) and open loop compressor (figure 1) with annulus chamber design (figure 5). The cost analysis include obtaining or developing costs for initial costs, maintenance costs, costs and the sum of all three for 1000, 2000, 4000 and 6000 hours of operation per year.

Two additional pieces of equipment were added to the AFFDL extension chamber as a result of discussions with the Air Force. A small fan would be used in chamber and would be sized only for the CFM capacity to maintain 200 FPM in chamber and not a constant mass flow required for cooling capacity as in the annulus design chamber. For a four foot diameter chamber this amounts to approximately a 2500 cfm fan. Secondly, a standard refrigeration system (35°F coil) would be included in the extension to perform the major amount of dehumidification and to avoid substantial frosting of the -65°F coil.

In order to define the scope and method of financial analysis the following assumptions for initial, operating and maintenance costs were made.

Initial costs were based on quotes received from suppliers or estimates based on in-house price information.

The operating costs were based on energy and material consumption (electrical, water and liquid nitrogen). For the various hours of operation during the year operating conditions were defined as follows:

1) 50% operation at maximum heating (cooling air)
2) 50% operation at maximum cooling (cooling air)
3) 50% operation at maximum humidity
4) 100% operation at full cooling air flow
5) 50% chamber operation at maximum heating
6) 50% chamber operation at maximum cooling
7) 100% operation at maximum altitude
8) 100% operation with vibration operating at maximum power
9) 100% operation with computer operating at maximum power

The following definitions apply to the above conditions:

<u>Cooling Air Maximum Cooling</u>

$M =$ 10 lb./min.

$\Delta T = [100-(-65)] = 165°F$

Cooling Air Maximum Heating

$M = 10$ lb./min.
$\Delta T = [200-(-65)] = 265°F$

Chamber Air Maximum Cooling

Cooling Air	=	10 lb./min. @ 200°F
Test Item	=	5Kw maximum load
Fan	=	Maximum HP

Chamber Air Maximum Heating

Cooling Air	=	10 lb./min. @ -65°F
Test Item	=	Not operating
Fan	=	Minimum HP
Chamber	=	Pulldown from maximum cooling

Energy Costs

Electricity = \$0.026/Kwh
LN_2 = \$0.32/gallon
Water = \$0.422/1000 gallon
Air = \$0.1/1000 cubic feet
Steam = \$0.004/pound

The maintenance costs of all of the equipment except the computer and vibration systems were determined as follows. The failure rates of the various pieces of equipment were taken from the government industry data exchange program (GIDEP) "Summary of Failure Rate Data" (volume 1, revised August 1975). A cost factor multiplier was determined. The equipment with the highest cost was assigned a cost factor of 0.1 and the item with the lowest cost was assigned a value of 1.0. The remaining cost factors were then the intermediate points determined by applying the equation of a straight line graph that connects the two extreme points. The maintenance cost was calculated by the following equation:

$$\text{Maintenance Cost} = (\text{Hours Operation}) \left(\frac{\text{Failure Rate}}{\text{Hours}}\right) (\text{Cost Factor}) (\text{Cost})$$

This assumes that the higher cost items will have a larger maintenance cost. The vibration maintenance costs were determined from systems in actual use at Dayton T. Brown, Inc. The computer system maintenance cost is a standard maintenance contract cost from a computer company.

Tables I and II list the various factors (power requirements, water consumption etc.) used for calculation of operating costs. Tables II and IV list the maintenance factors used. Table V and VI list the initial costs, VII and VIII the operating costs, IX and X the maintenance costs and Table XI list the sum of all the costs for each facility.

Tables are arabic numerals

TABLE 1

CERT Facility - Operating Parameters/Open Loop Compressor with Annulus Chamber

Item	Electricity	Water(GPH)	Air	LN2	Steam
Compressor	36.3 HP	600	-	-	-
Dryer (Heat Pump)	1 Kw	20	5.3 scfm	-	-
Vacuum Pump	50 HP	660	-	-	-
Chamber Fan	15 HP	-	-	-	-
Vibration (incl. control)	40 Kw	540	-	-	-
Electrical Generation	20 Kw	-	-	-	-
Humidification					
Cooling Air (Steam)	-	-	-	-	24 lbs./hr
Chamber Air Ultrasonic	-	435	244 scfm		-
Steam	-	-	-	-	360 lbs./hr
Heaters					
Cooling Air	11 Kw	-	-	-	-
Chamber Air	50 Kw	-	-	-	-
Computer System	3 Kw	-	-	-	-
Cold Wall*	Assume 30 temperature changes @ $135 per change (Typical Reliability Test)				
Cryogenic Cooling Air	-	-	-	385 lbs/hr	-
Refrigeration System					
Chamber Cascade	68 HP	-	-	-	-
Chamber Standard	3 HP	-	-	-	-

TABLE 2

CERT Facility - Operating Parameters/Open Loop Blower with AFFDL Chamber

Item	Electricity	Water(GPH)	Air	LN_2	Steam
Blower	16 HP	300	-	-	-
Dryer (Heat Type)	1.8 Kw	-	9 scfm	-	-
Vacuum Pump	50 HP	660	-	-	-
Chamber Fan	2 HP	-	-	-	-
Vibration (incl. control)	40 Kw	540	-	-	-
Electrical Generation	20 Kw	-	-	-	-
Humidification					
Cooling Air					
Ultrasonic	-	3	1.1 scfm	-	-
Steam	-	-	-	-	24 lbs./hr
Chamber Air					
Ultrasonic	-	55	31.2 scfm	-	-
Steam	-	-	-	-	459 lbs./hr
Heaters					
Cooling Air	11 Kw	-	-	-	-
Chamber Air	20 Kw	-	-	-	-
Computer System	3 Kw	-	-	-	-
Cold Wall* Assume 30 temperature changes @ $135 per change.					
Refrigeration System					
Cooling Air	12 HP	-	-	-	-
Chamber Cascade	28.5 HP	-	-	-	-
Chamber Standard	3 HP	-	-	-	-

*Average amount of changes for a reliability test

TABLE 3

CERT Facility - Maintenance Factors

Open Loop Compressor with Annulus Chamber

Item	Cost Factor	Maintenance Factor ($x10^{-6}$)
Thermocouple	1	62.992
Separator and Trap	0.000625	22.61
LN_2 Solenoids	0.999375	24.848
Water Hx	0.999375	12.42
Lytron Hx	0.999	12.42
Steam Pressure Regulator	0.999881	47.488
Humidity Control Valve	0.9975	87.082
Flow Control Valve	0.9968	29.767
Pressure Transducer	0.9956	63.327
Cryogenic Hx	0.9944	12.42
Pre-Filter	0.9931	15.05
After Filter	0.9931	15.05
Altitude Control Valve	0.9881	29.767
Standard Refrigeration	0.9819	37.093
Heaters	0.9756	2.151
Fan	0.9731	6.23
Cold Wall	0.9693	1.242
Humidity	0.9656	13.617
Dryer	0.9681	21.364
Cascade Refrig. (Chamber)	0.5631	37.093
Compressor	0.9188	37.093
Vacuum Pump	0.5006	25.641

Item	Cost Factor	Maintenance Factor ($x10^{-6}$)
Vibration	Not Used, cost is based on actual DTB numbers	
Computer System	Not Used, cost is for a 1 year maintenance contract	
Electrical Generation	0.200625	126.5
Ultrasonic Humidification	0.9068	19.313

TABLE 4

CERT Facility - Maintenance Factors

Open Loop Blower with AFFDL Chamber

Item	Cost Factor	Maintenance Factor
Thermocouple	1	62.992
Separator and Trap	.999625	22.61
LN_2 Solenoids	.999375	24.848
Water Hx	.999375	12.42
Steam Pressure Regulator	.99881	12.42
Fan	.99812	6.23
Humidity Control Valve	.9975	87.082
Flow Control Valve	.9968	29.767
Pressure Transducer	.9956	63.327
Pre-Filter	.9931	15.05
After-Filter	.9931	15.05
Alt. Control Valve	.9881	29.767
Heaters	.9844	2.151
Standard Refrigeration	.9819	37.093
Humidity Meter	.9656	13.617
Blower	.8944	37.093
Cascade Refrig. (Cool Air)	.8756	37.093
Cascade Refrig. (Chamber Air)	.5631	37.093
Cold Wall	.9693	1.242
Dryer	.7006	13.499
Vacuum Pump	.5006	25.641
Vibration	Not Used	

Item	Cost Factor	Maintenance Factor
Computer System	Not Used	
Ultr. Humidity System	.9068	19.312
Electrical Generation	.200625	126.5

TABLE 5

CERT Facility - Initial Costs

Open Loop Compressor with Annulus Chamber

Item-Cooling Air	Cost	Item Chamber	Cost
Compressor and Aftercooler	6,543	Vacuum Pump	40,000
Receiver	433	Water Hx	1,300
Pre-Filter	600	Altitude Control Valve	1,000
After Filter	600	Vibration	60,000
		Digital Vibration Control	65,000
Dryer	2,600	Chamber Const. (Structural only)	50,000
Flow Control Valve	300	Fan	2,200
Cryogenic Hx	500	Cascade Refrig.	35,000
Lytron Hx (3)	390	Standard Refrig.	1,500
Steam Humidification (In plant)	-	Electric Heaters	2,000
Separator and Trap	80		
Steam Press. Reg.	145	Steam Humidification (In plant)	-
Humidity Control	250		
		Separator and Trap	80
		Steam Press. Reg.	145
LN_2 Solenoids (2)	200	Humidity Control	250
Humidity Meter	2,800	Pressure Transducer	400
Thermocouples (4)	200	Humidity Meter	2,800
Flow Meter	1,200	Thermocouple (3)	150
		Ultrasonic Humidification	7,500
		LN_2 Solenoids (2)	200
		Computer System (Total)	57,000
		Electrical Generation	64,000
Total	16,841	Total	389,325
Subtotal (Chamber plus Cooling Air)	406,166		
		Contingency for installation 10%	40,600
		Total	446,766

TABLE 6

CERT Facility - Initial Costs

Open Loop Blower with AFFDL Chamber

Item-Cooling Air	Cost	Item-Chamber	Cost
Blower	2,833	Vacuum Pump	40,000
Aftercooler	500	Water Hx	100
Pre-Filter	600	Altitude Control Valve	1,000
After Filter	600	Vibration	60,000
		Digital Vibration Control	65,000
Dryer	8,000	Chamber Const. (Structural only)	50,000
Flow Control Valve	300	Fan	200
Steam Humidification (In plant)	-	Cascade Refrig.	35,000
Separator and Trap	80		
Steam Press. Reg.	145	Standard Refrig.	1,500
Humidity Valve	250		
		Electric Heaters	800
Ultrasonic Humidification	500		
		Ultrasonic Humidification	7,500
Electric Duct Heaters	440		
		Steam Humidification (In plant)	-
Insulated Conditioning	800		
		Separator and Trap	80
		Steam Press. Reg.	145
Thermocouples (4)	200	Humidity Valve	250
Flow Meter	1,200	Press. Transducer	400
Humidity Meter	2,800	Humidity Meter	2,800
		Thermocouples (3)	150
Cascade Refrigeration	10,000		
		LN_2 Solenoids (2)	200
		Computer (Total)	57,000
		Elect. Generation	64,000
Total	29,248	Total	386,725
Total (Chamber plus Cooling Air)	415,973	Contingency for Installation 10%	41,500
		Total	457,473

TABLE 7

CERT Facility - Operating Costs

Open Loop Compressor with Annulus Chamber

Item	Type Cost	1000 hr.	2000 hr.	4000 hr.	6000 hr.
Compressor	Electricity	704	1409	2817	4226
	Water	253	507	1013	1520
Dryer	Electricity	26	52	104	156
	Air	32	64	128	192
	Water	8	16	32	48
Vacuum Pump	Electricity	969	1938	3876	5814
	Water	279	558	1116	1674
Chamber Fan	Electricity	291	582	1164	1746
Vibration	Electricity	1040	2080	4160	6240
	Water	228	456	912	1368
Elect. Generation	Electricity	520	1040	2080	3120
Humidification					
Cooling Air	Steam	48	96	192	288
Chamber Air	Ultrasonic	41	82	164	246
	Steam	360	720	1440	2160
Heaters					
Cooling Air	Electricity	143	286	572	858
Chamber Air	Electricity	650	1300	2600	3900
Computer System	Electricity	78	156	312	468

TABLE 7
(Continued)

Item	Type Cost	1000 hr.	2000 hr.	4000 hr.	6000 hr.
Cold Wall	LN_2	4050	8100	16,200	24,300
Cryogenic Cooling Air	LN_2	9129	18,258	36,516	54,774
Refrigeration System					
Cascade Chamber	Electricity	659	1318	2637	3956
Standard Chamber	Electricity	29	58	116	174
Personnel	One Technician @ $6.00/hr.	6000	12,000	24,000	36,000
	Total	25,537	51,074	102,148	153,222

TABLE 8

CERT Facility - Operating Costs

Open Loop Blower with AFFDL Chamber

Item	Type Cost	1000 hr.	2000 hr.	4000 hr.	6000 hr.
Blower	Electricity	310	621	1241	1862
	Water	127	253	507	760
Dryer	Electricity	48	95	191	286
	Air	56	112	221	336
Vacuum Pump	Electricity	969	1938	3876	5814
	Water	279	558	1116	1674
Chamber Fan	Electricity	39	78	156	234
Vibration	Electricity	1040	2080	4160	6240
	Water	228	456	912	1368
Elect. Generation	Electricity	520	1040	2080	3120
Humidification					
Cooling Air	Ultrasonic	53	106	211	316
	Steam	24	48	96	144
Chamber Air	Ultrasonic	2	4	8	12
	Steam	459	918	1836	2754
Heaters					
Cooling Air	Electricity	143	286	572	858
Chamber Air	Electricity	260	520	1040	156
Computer System	Electricity	78	156	312	468

TABLE 8
(Continued)

Item	Type Cost	100 hr.	2000 hr.	4000 hr.	6000 hr.
Cold Wall	LN_2	4050	8100	16,200	24,300
Refrigeration System					
Cooling Air	Electricity	117	233	465	698
Cascade Chamber	Electricity	277	553	1105	1658
Standard Chamber	Electricity	29	58	116	174
Personnel	One Technician @ $6.00/hr.	6000	12,000	24,000	36,000
	Total	15,108	30,216	60,432	90,648

TABLE 9

CERT Facility - Maintenance Costs

Open Loop Compressor With Annulus Chamber

Item (Number)	1000 hr.	2000 hr.	4000 hr.	6000 hr.
Thermocouple (7)	3.15(22.05)	6.30(44.10)	12.60(88.20)	18.90(13.23)
Separator and Trap (2)	1.80(3.60)	3.60(7.20)	7.20(14.40)	10.85(21.60)
LN_2 Solenoids (4)	2.48(9.92)	4.97(19.84)	9.93(39.68)	14.90(59.52)
Water Hx (1)	1.24	2.48	4.96	7.40
Lytron Hx (3)	1.61(4.83)	3.23(9.66)	6.45(19.32)	9.68(28.98)
Steam Press. Reg. (2)	6.87(13.74)	13.75(27.48)	27.51(54.96)	41.26(82.44)
Humidity Control Valve (1)	21.72(43.44)	43.43(86.86)	86.86(173.76)	130.29(260.64)
Flow Control Valve (1)	8.90	17.80	35.60	53.41
Pressure Transducer (1)	25.22	50.44	100.88	151.31
Cryogenic Hx (1)	6.17	12.35	24.70	37.05
Pre-Filter (1)	8.96	17.94	35.87	53.81
After Filter (1)	8.96	17.94	35.87	53.81
Alt. Control Valve (1)	29.71	59.42	118.94	178.26
Standard Refrigeration (1)	27.31	54.63	109.26	163.89
Heaters	2.09	4.19	8.38	12.57

TABLE 9

(Continued)

Item (Number)	1000 hr.	2000 hr.	4000 hr.	6000 hr.
Fan (1)	13.33	26.67	53.34	80.02
Cold Wall (1)	3.00	6.00	12.00	18.00
Humidity Meter (1)	36.81	73.63	147.26	220.89
Dryer (1)	53.77	107.54	215.09	322.64
Cascade Chamber Refrig. (1)	731.05	1462.09	2924.18	4386.28
Compressor (1)	222.99	445.98	891.97	1337.95
Vacuum Pump (1)	513.43	1026.87	2053.74	3080.61
Vibration	5000	10,000	20,000	30,000
Computer System	3984.00	3984.00	3984.00	3984.00
Elect. Generation	1685	3370	6941	10,112
Ultrasonic Humidification	131.36	262.72	525.40	799.11
Total	12,590.88	25,181.76	50,363.52	75,545.28

TABLE 10

CERT Facility - Maintenance Costs

Open Loop Blower with AFFDL Chamber

Item (Number)	1000 hr.	2000 hr.	4000 hr.	6000 hr.
Thermocouples (7)	3.15(22.05)	6.30(44.10)	12.60(88.20)	18.90(132.30)
Separator and Trap (2)	1.80(3.60)	3.60(7.20)	7.20(14.40)	10.85(12.60)
LN_2 Solenoids (4)	2.48(9.92)	4.97(19.84)	9.93(39.68)	14.90(59.52)
Water Hx (1)	1.24	2.48	4.96	7.40
Steam Press. Reg. (1)	6.87(13.74)	13.75(27.48)	27.51(54.96)	41.26(82.52)
Fan (1)	1.24	2.48	4.97	7.46
Humidity Control Valve (1)	21.72(43.44)	43.43(86.86)	86.86(173.76)	130.29(260.64)
Flow Control Valve (1)	8.90	17.80	35.60	53.41
Pressure Transducer (1)	25.22	50.44	100.88	151.31
Pre-Filter (1)	8.96	17.94	35.87	53.81
After Filter (1)	8.96	17.94	35.87	53.81
Alt. Control Valve (1)	29.71	59.42	118.84	178.26
Heaters (1)	1.38	2.75	5.50	8.25
Standard Refrigeration (1)	27.31	54.63	109.26	163.89
Humidity Meter (1)	36.81	73.63	147.26	220.89

TABLE 10
(Continued)

Item (Number)	1000 hr.	2000 hr.	4000 hr.	6000 hr.
Blower	101.42	202.85	405.71	608.57
Cascade Refrig. (Cool Air)	324.78	649.57	1299.14	1948.72
Cascade Refrig. (Chamber Air)	731.05	1462.09	2924.18	4386.28
Cold Wall	3.00	6.00	12.00	18.00
Dryer	97.26	194.52	389.03	583.55
Vacuum Pump	513.43	1026.87	2053.74	3080.61
Vibration	5000	10,000	20,000	30,000
Computer System	3984.00	3984.00	3984.00	3984.00
Ultra. Humidification (2)	131.36	262.72	525.40	788.11
Elect. Generation	1685	3370	6941	10,112
Total	12,813.78	25,627.56	51,255.12	76,882.68

TABLE 11

CERT Facility - Initial + Operating + Maintenance Costs

Item	1000 hr.	2000 hr.	4000 hr.	6000 hr.
Open Loop Compressor with Annulus Chamber	484,893	523,021	599,277	675,533
Open Loop Blower with AFFDL Chamber	485,394	513,316	569,160	625,003

ECONOMIC ANALYSIS - (Continued)

As can be seen from tables V and VI the initial costs for both facilities are fairly close (i.e. 446,766 versus 457,473). Likewise the maintenance costs (IX and X) are almost the same. However the operating costs for the open loop compressor system are twice that of the open loop blower system for either 1000, 2000, 4000 or 6000 hours the difference increasing from approximately $10,000 at 1000 hours to $60,000 at 6000 hours. This makes the total facility cost substantially more for the open loop compressor system at the end of 6000 hours. This difference in cost basically stems from the high cost of liquid nitrogen for the cryogenic cooling air system. The initial cost of the mechanical refrigeration system for the open loop blower is $10,000. However the operating cost of the cryogenic cooling is $9,129 for 1000 hours and $54,774 for 6,000 hours. Therefore the refrigeration system cost would be paid for after approximately 1000 hours of operation. A mechanical refrigeration system for the compressor will be discussed in the cost reduction section.

SECTION X

COST REDUCTION

TASK - IV

There are several areas in which a reduction in the cost of the facility can be attained. After several meetings and telephone conversations with the Air Force, the following were selected for investigation.

(1) Brine system for cold wall.
(2) Cascade refrigeration system for open loop compressor.
(3) Reduction in chamber temperature change rates.
(4) Reduction in altitude capability.
(5) Water tower installation for cooling water requirements.

The following analysis will discuss each cost reduction area individually.

(1) Secondary Refrigerant System for Cold Wall

In order to eliminate the high operating costs of using liquid nitrogen on the chamber cold wall, an investigation was made into the costs of a secondary refrigerant system for heat removal. The panel coil has a total length of 7 feet and a diameter of 6 feet. The cross-sectional area of the channel is 0.93 in^2. This results in a total panel weight of 744 pounds.

The cost of the refrigeration system was estimated at $35,000. The cost of installation would be approximately $3000.00. This results in a total initial cost of $38,000. The operating costs over 6000 hours of operation were calculated to be $3,900.00. Maintenance costs for 6000 hours of operation were calculated to be $4,400.00. Since the operational costs of using liquid nitrogen in the cold wall were calculated to be $4,050.00 per 1000 hours of operation, the initial cost investment would be realized after approximately 9000 hours of operation. On this basis we would recommend it as a cost reduction.

(2) Cascade System for Open Loop Compressor

The cost analysis in task III used a liquid nitrogen heat exchanger for the cooling air. The total capacity of the heat exchanger was determined by the maximum capacity required by the specifications. This resulted in the following capacity.

$$Q = MC_p\Delta T = (10 \text{ lb./min.})(60 \text{ min./H})(0.24 \text{ BTU/lb./°F})(165\text{°F})$$
$$= 23,760 \text{ BTU/H} \cong 2 \text{ tons}$$

The initial cost of a two ton mechanical refrigeration system of this type would be approximately $10,000.00. Operating costs for 6000 hours of operation were calculated to be $700.00. Maintenance costs for 600 hours of operation were estimated to be $2,000.00. The following is a listing of the initial, operating and maintenance costs for the two kinds of systems.

	Initial Cost	Operational	Maintenance	Total
Cascade	10,000	700	2,000	12,700
LN_2	500	54,774	100	55,374

The cost advantages of the cascade system are obvious and we recommend it as a cost reduction possibility.

(3) Reduction in Chamber Temperature Change Rates

The relaxing of the 40°F/min. temperature change rate could result in a reduction of the refrigeration capacity of the chamber cascade mechanical refrigeration unit. In order to determine the cost savings of this type of reduction, a refrigeration analysis was done for several compressor capacities. This kind of analysis is described in detail in section III of the report.

The operating and maintenance costs were determined by using the assumptions and methods used in the economic analysis (section IX) of this report. The following table was organized for 6000 hours of operation.

Change Rate °F/Min	Capacity Tons	Initial Costs $	Power HP	Operating Costs $	Maintenance Costs $	Total Costs $
37	12.5	28,000	60	3,315.00	3,500.00	34,185.00
28	8.3	19,000	40	2,150.00	2,380.00	23,530.00
22	6.25	14,000	30	1,700.00	1,750.00	17,450.00

This can be compared to the total costs for 6000 hours of operation of the proposed system of $43,342 (I.C. = $35,000, O.C. = $3,956.00 and M.C. = $4,386.00).

(4) Reduction in Altitude Capability

The altitude requirement inside the chamber imposes the most problems in operation on the following two pieces of equipment:

1. Vacuum Pump
2. Chamber Fan

As can be seen from the discussion in section IV of this report, more than 95 percent of the vacuum pump capacity is required merely to remove the 10 lb. per minute flow while maintaining an altitude of 70,000 ft. This required a vacuum system with the capacity to remove approximately 3800 CFM at altitude. If the altitude requirement was only 60,000 ft., the total CFM capacity of the pump would be approximately 2400 CFM. Initial cost on a pump with this capacity would be approximately $30,000. Operating costs would be $5,600.00 for 6000 hours of operation and maintenance costs would be $2,300.00. The total costs for 6000 hours of operation would be $37,900.00. This compares to total costs for 6000 hours of operation of the pump required for an altitude of 70,000 ft. of $50,568.00.

(5) Water Tower Installation for Cooling Water Requirements

Water consumption rates for the items requiring cooling water were totaled and a total refrigeration capacity was determined by allowing a 20°F rise on the water temperature. The total capacity was determined to be 420,588 BTUH with a circulation capacity of 42 GPM. Total initial costs for supply and installation of a complete water cooling tower were estimated to be $5,100.00. The tower selected has a 480,000 BTUH capacity at 70° water temperature. The unit weighs approximately 3500 lbs. and has the dimensions of 6 feet x 6 feet x 6 feet. This compares with total operating costs for 6000 hours of operation of the open loop compressor system of approximately $7,800.00. The cooling water requirements of the compressor, dryer, vacuum pump and the vibration system were considered in the estimate.

SECTION XI

FACILITY SCAN

TASK - V

The purpose of the scan was to find three commercially available chambers that could be modified to CERT requirements and estimate the cost savings of modification rather than building one from scratch. Two standard chambers closest to CERT requirements are the Tenney agree chamber (temperature - humidity - vibration) and the special design Thermotron (temperature-altitude-humidity-vibration) chamber. Neither contain cooling air and can only change temperature at the rate of 10°F per minute. The altitude capability of the chamber could not support cooling air and maintain altitude. Basically a complete modification to either chamber would be required to meet CERT requirements. All the systems including control would have to be replaced. There would be no cost savings by modification since the CERT requirements are so unique and do not exist in any form on the commercial market.

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CONCEPTUAL DESIGNS FOR COMBINED ENVIRONMENTS RELIABILITY TEST F--ETC(U)
JUN 77 K TRELEWICZ, J HERLIHY F33615-76-C-7026

UNCLASSIFIED DTB26R76-1109 AFFDL-TR-77-3 NL

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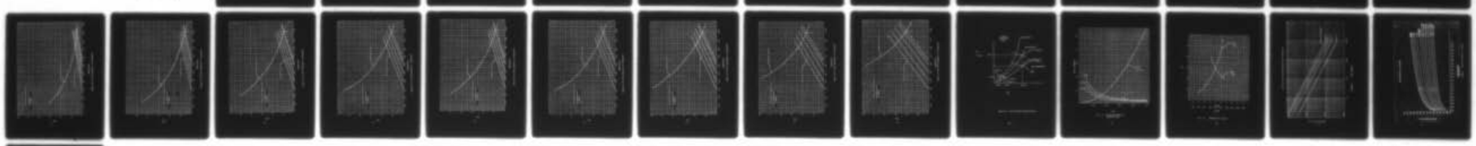

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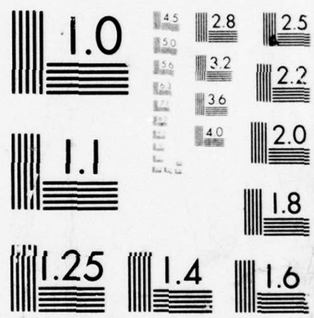

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

REFERENCES

1. Arnold H. Mayer and David K. Prather, "An Experimental Dynamic Combined Environments Facility for Reliability Testing of Airborne Equipment," ~~AFFDLT2, September, 1975.~~ AFFDL-TR-76-108, October, 1976.

2. "System Practices for Multi-stage Applications," 1973, Ashrae Systems Handbook, Chapter 29.

3. W. L. Holladay, "Low Temperature Test Chamber Design, Refrigerating Engineering," July 1950, Page 656.

4. H. Edmund Karig, "Psychrometric Charts for High Altitude Calculations," Journal of the A.S.R.E., November, 1946.

5. F. Kreith, <u>Principles of Heat Transfer</u>, International Textbook Company, Second Edition, July, 1968.

6. <u>Handbook of Fundamentals</u>, ASHRAE, 1972.

7. H. B. Pownall and S. P. Soling, "Heat Flow Rates with Rapid Temperature Changes in Enclosed Spaces," Journal of the A.S.R.E., July, 1944.

8. T. L. Pondek and W. Elmendor, "A Versatile Environmental Cooling Air Facility," Proceedings - I.E.S.

9. D. K. Prather and D. Earls, "Combined Environment Reliability Test (CERT) for Avionics Subsystems," The Journal of Environmental Sciences," March/April, 1976.

10. <u>Fans and Their Application in Air Conditioning</u>, the Trane Company, 1970.

11. R. Scott, <u>Cryogenic Engineering</u>, D. Van Nostrand Company, Inc. 1959.

12. T. Baumeister, <u>Mechanical Engineers' Handbook</u>, McGraw-Hill, 1958.

13. <u>Equipment Guide and Data Book</u>, ASHRAE, 1969.

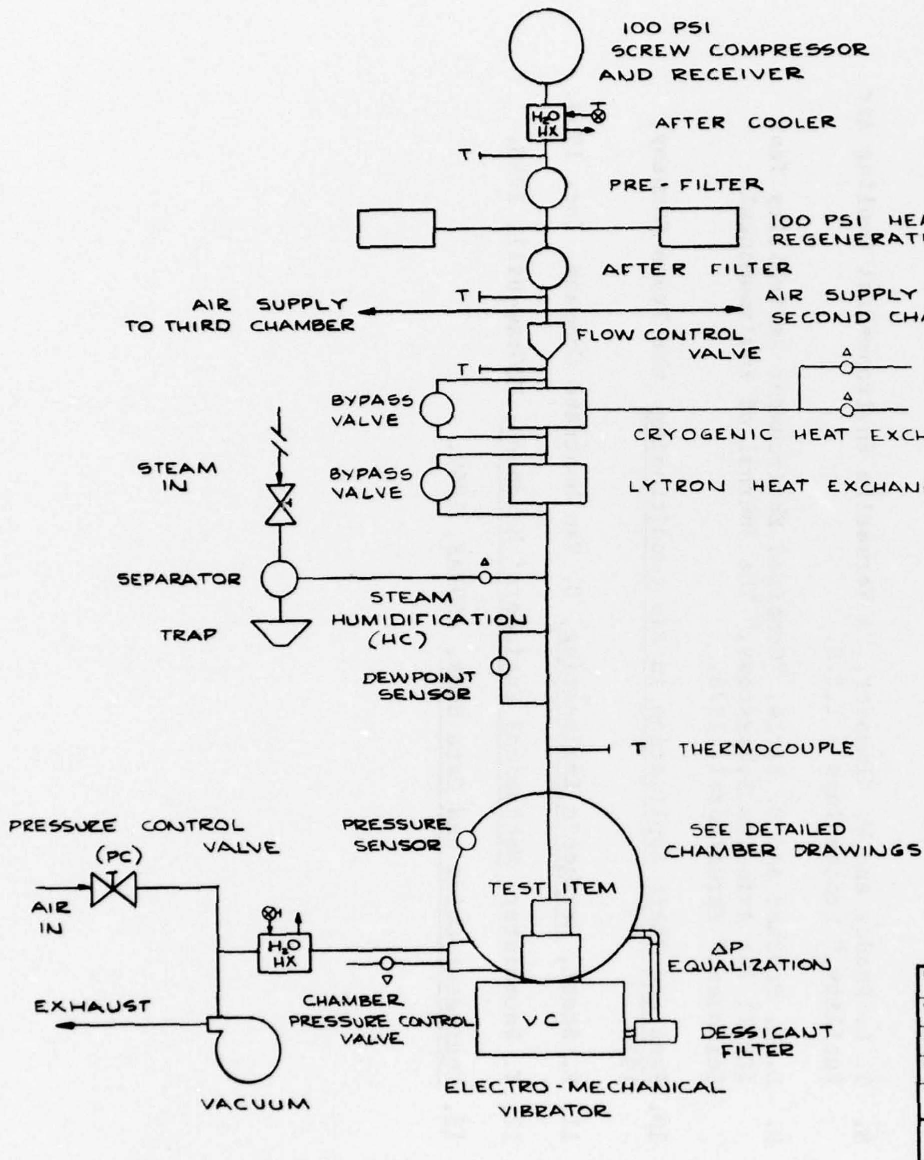

FIGURE NO. 1

SCHEMATIC

- OPEN LOOP COMPRESSOR SYSTEM -

DESIGNER J. HERLIHY K. TRELEWICZ		DAYTON T. BROWN INC. BOHEMIA, NEW YORK		
CHECKER		COMBINED ENVIRONMENTS RELIABILITY TEST FACILITY		
ENG K. Trelewicz K. TRELEWICZ	7-26-76			
APPROVAL				
CUST. APPROVAL				
UNLESS OTHERWISE SPECIFIED ALL DIMENSIONS ARE IN INCHES TOLERANCE FRACTIONS ± 1/32 DEC X ± .02 XX ± .01 XXX ± .005		SIZE C	CODE IDENT 96362	DWG. NO. L8675
		SCALE: NONE		SH 1 OF 6

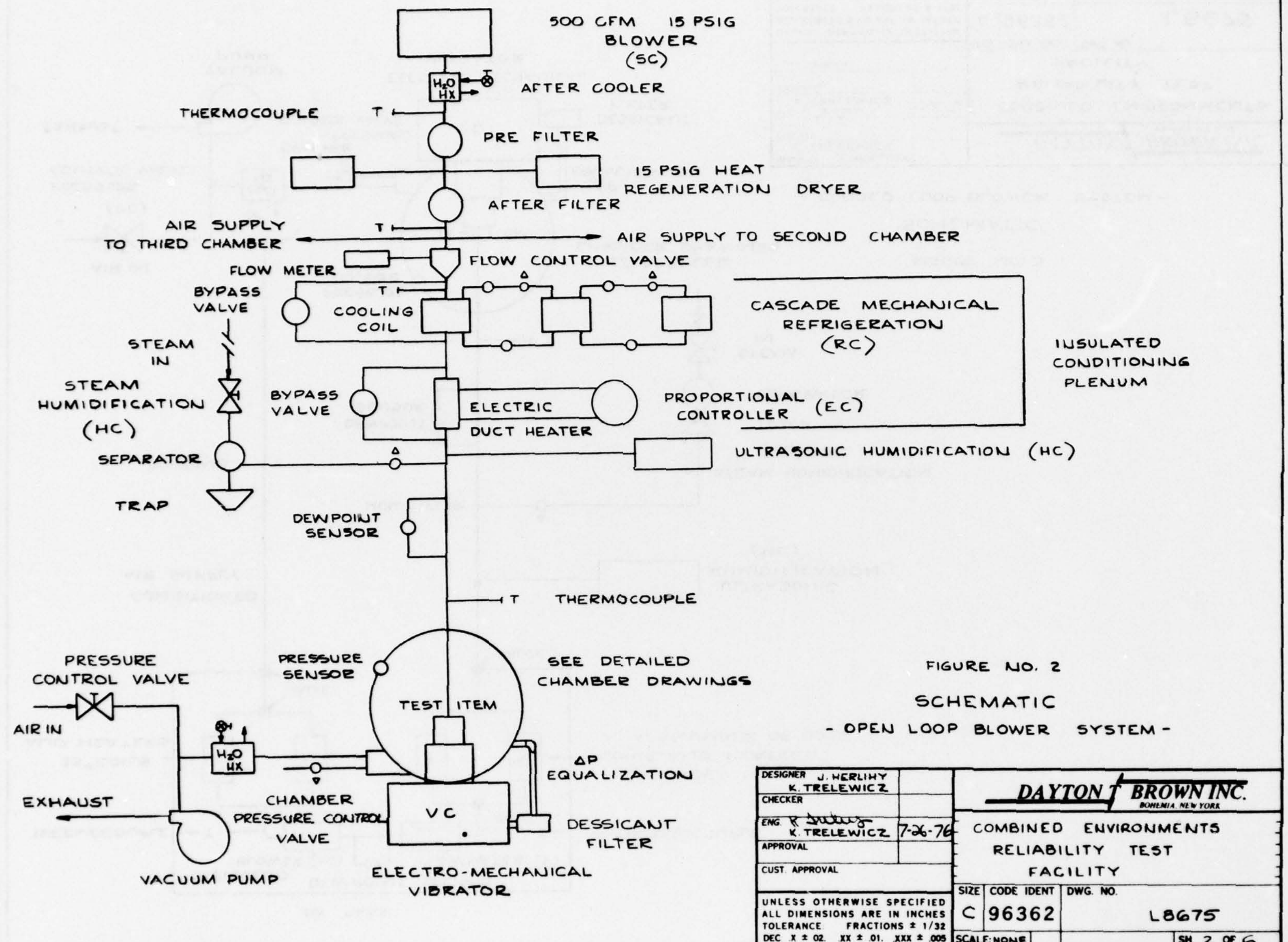

FIGURE NO. 2

SCHEMATIC

- OPEN LOOP BLOWER SYSTEM -

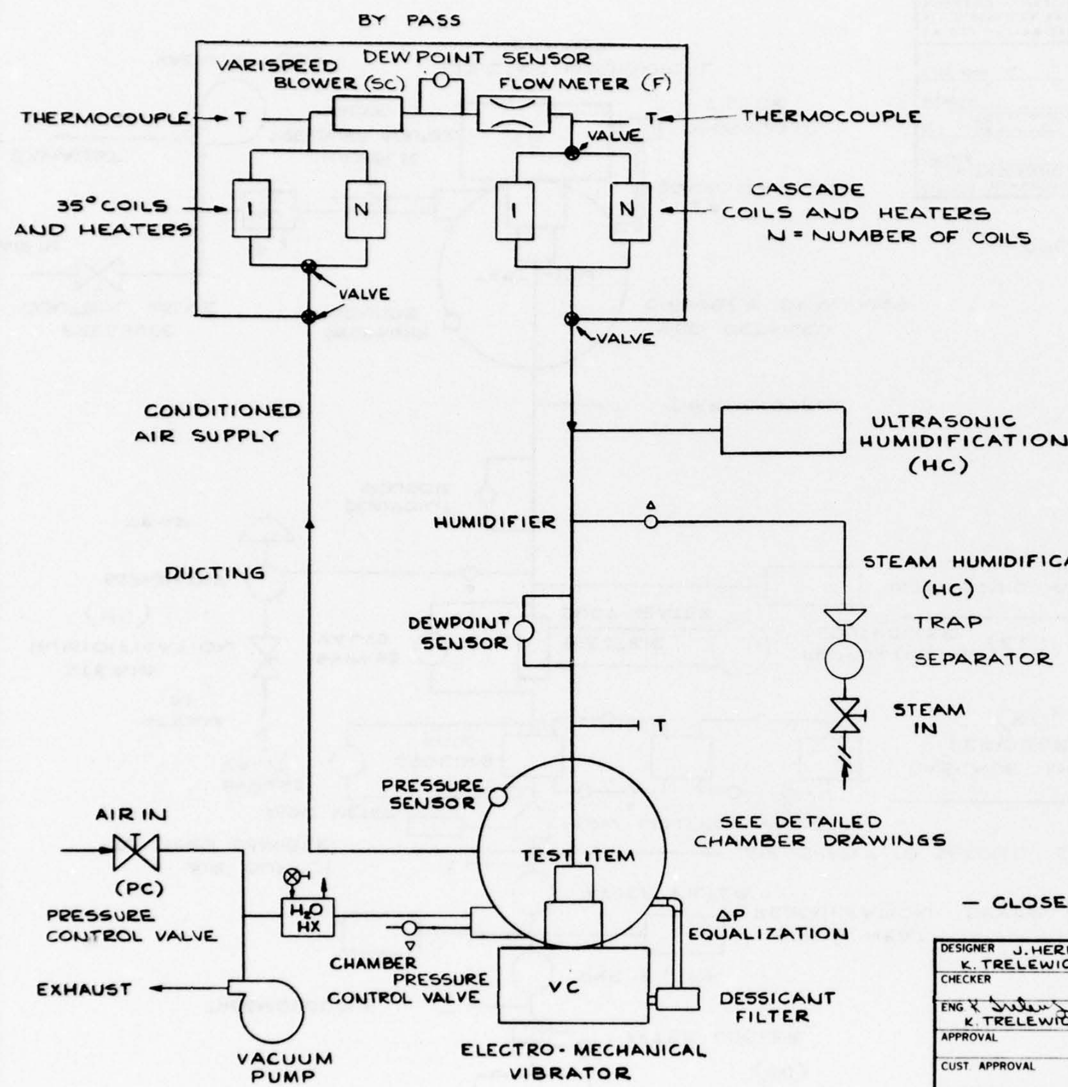

FIGURE NO. 3

SCHEMATIC

— CLOSED LOOP BLOWER SYSTEM —

DESIGNER J. HERLIHY K. TRELEWICZ		DAYTON T. BROWN INC. BOHEMIA NEW YORK
CHECKER		COMBINED ENVIRONMENTS RELIABILITY TEST FACILITY
ENG K. Trelewicz K. TRELEWICZ	7-26-76	
APPROVAL		
CUST APPROVAL		SIZE C · CODE IDENT 96362 · DWG. NO. L8675
UNLESS OTHERWISE SPECIFIED ALL DIMENSIONS ARE IN INCHES TOLERANCE FRACTIONS ± 1/32 DEC X ± .02 XX ± .01 XXX ± .005		SCALE: NONE · SH 3 OF 6

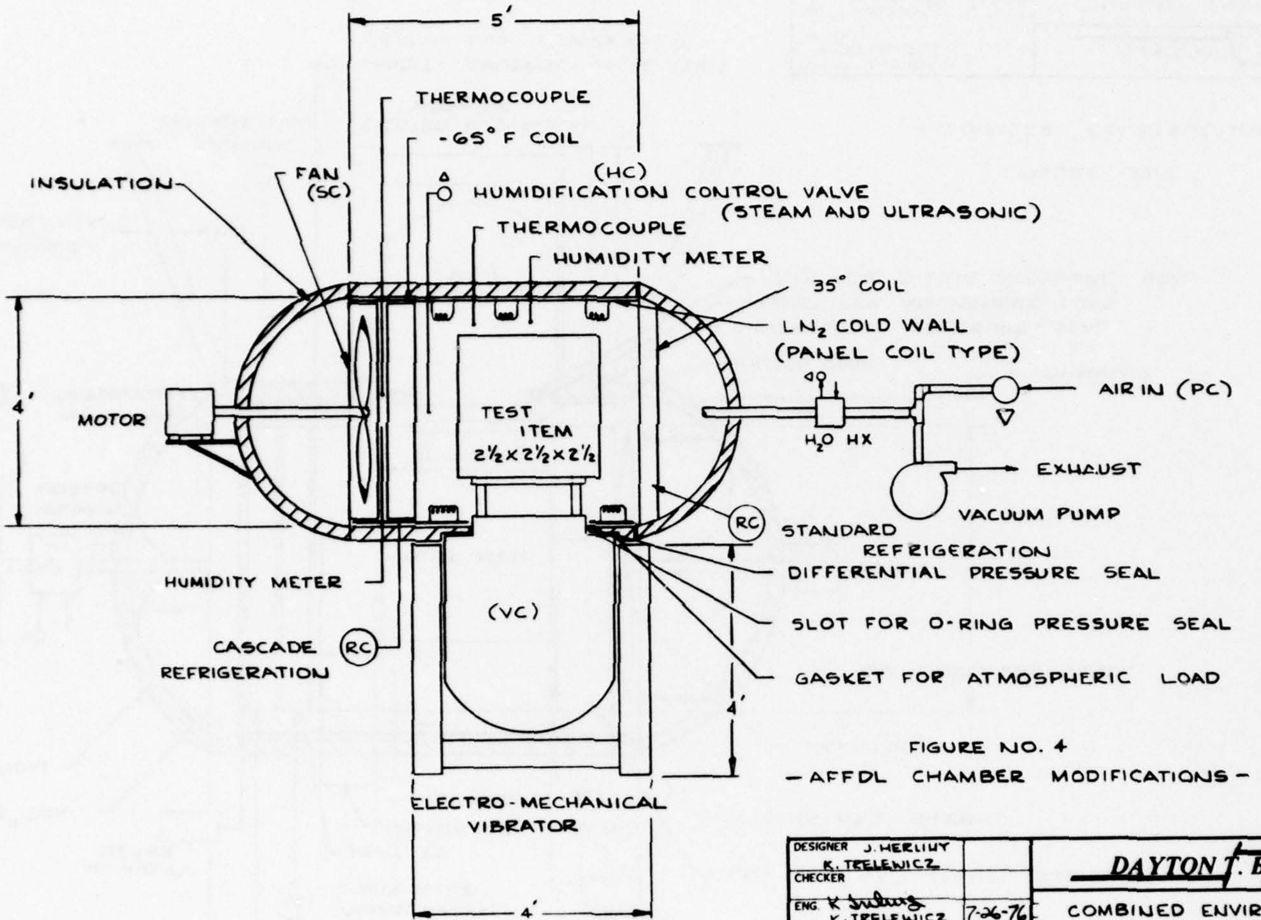

FIGURE NO. 4

– AFFDL CHAMBER MODIFICATIONS –

DESIGNER J. HERLIHY K. TRELEWICZ		DAYTON T. BROWN INC. BOHEMIA, NEW YORK		
CHECKER		COMBINED ENVIRONMENTS RELIABILITY TEST FACILITY		
ENG. K Trelewicz K. TRELEWICZ	7-26-76			
APPROVAL				
CUST. APPROVAL				
UNLESS OTHERWISE SPECIFIED ALL DIMENSIONS ARE IN INCHES TOLERANCE: FRACTIONS ± 1/32 DEC X ± .02 XX ± .01 XXX ± .005	SIZE C	CODE IDENT 96362	DWG. NO. L8675	
	SCALE: NONE			SH 4 OF 6

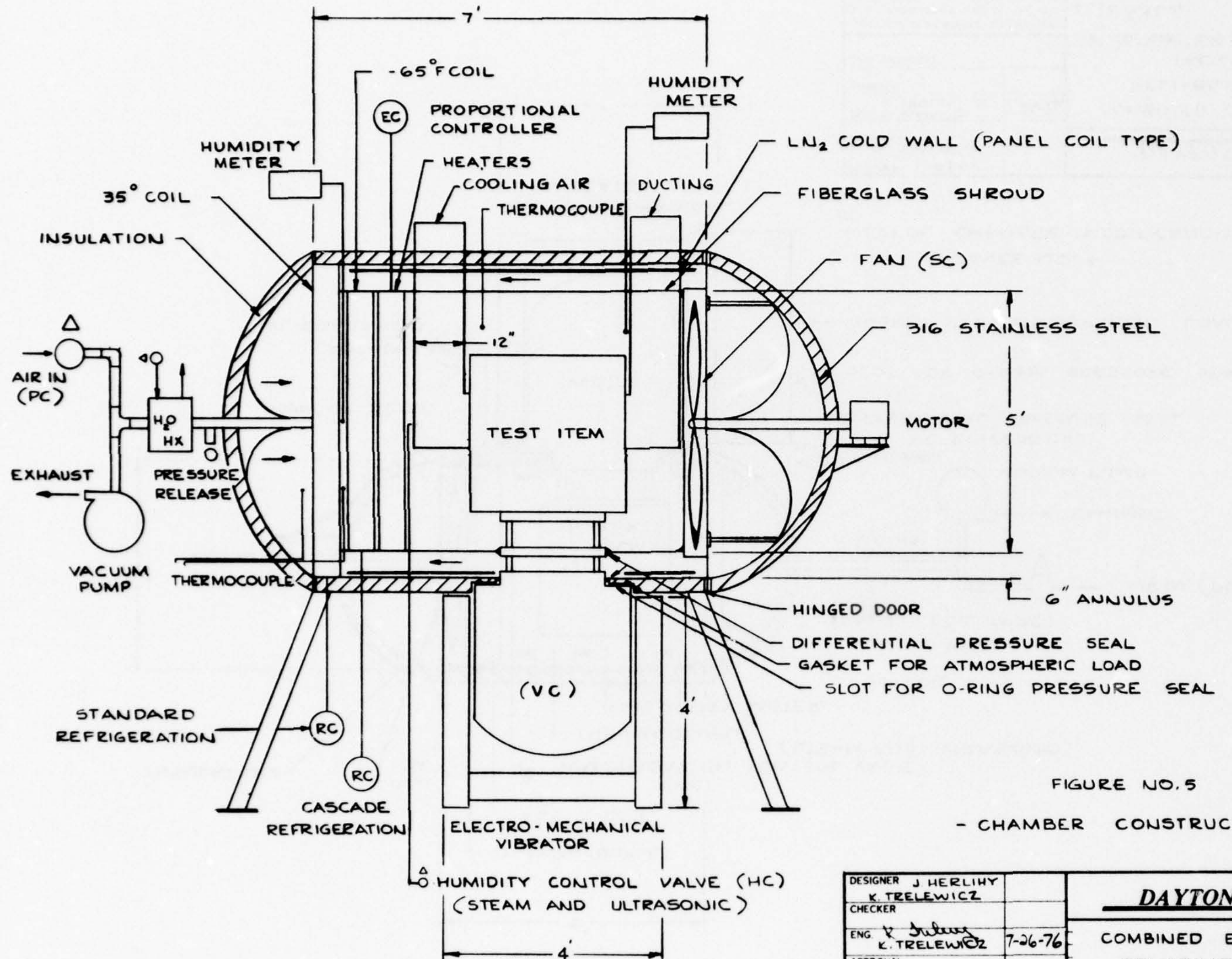

FIGURE NO. 5

- CHAMBER CONSTRUCTION -

DESIGNER J HERLIHY K. TRELEWICZ		DAYTON T. BROWN INC. BOHEMIA, NEW YORK	
CHECKER			
ENG K. Trelewicz K. TRELEWICZ	7-26-76	COMBINED ENVIRONMENTS RELIABILITY TEST FACILITY	
APPROVAL			
CUST. APPROVAL			
UNLESS OTHERWISE SPECIFIED ALL DIMENSIONS ARE IN INCHES TOLERANCE FRACTIONS ± 1/32 DEC X ± 02 XX ± 01 XXX ± 005		SIZE C CODE IDENT 96362	DWG. NO. L8675
		SCALE: NONE	SH 5 OF 6

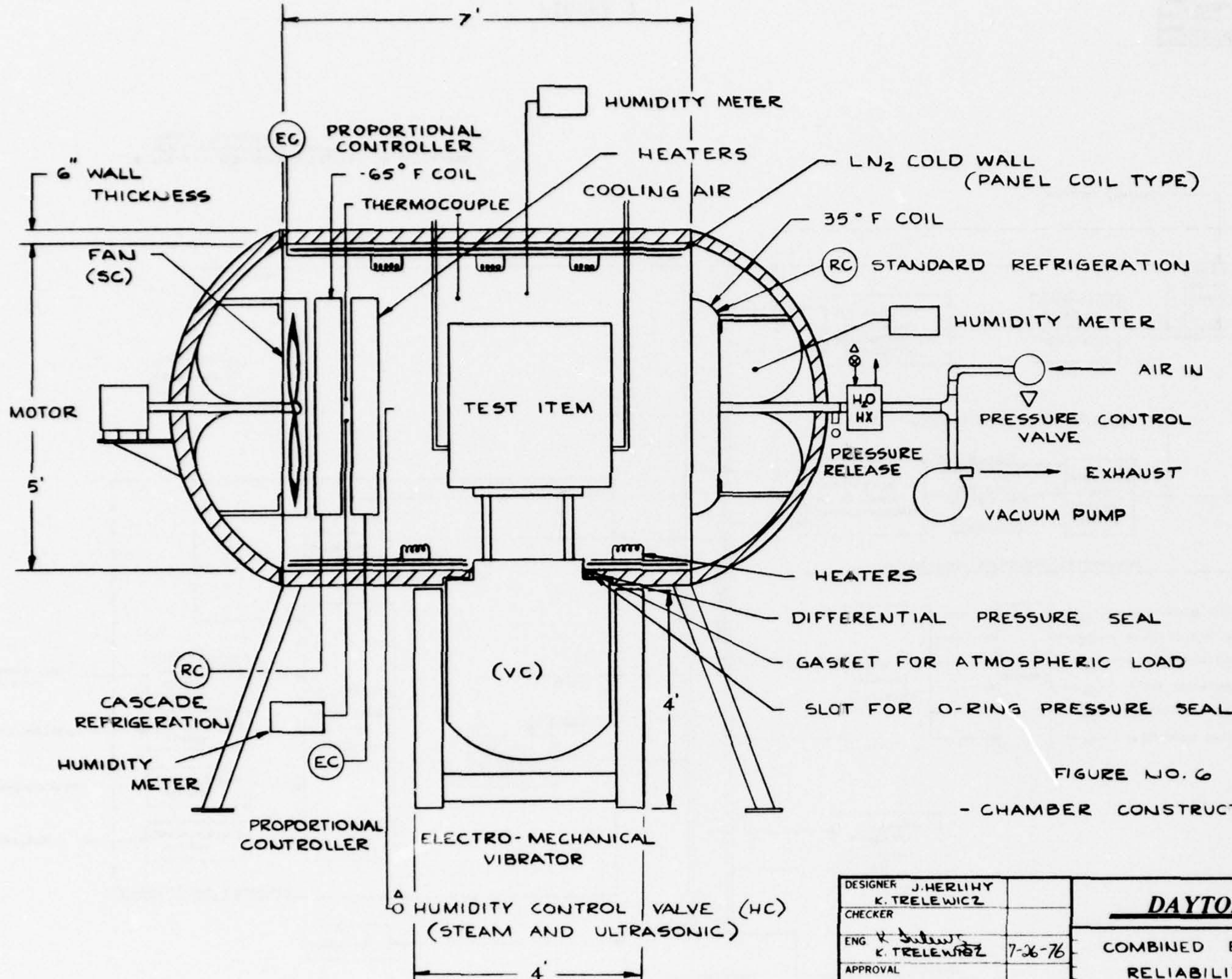

FIGURE NO. 6

- CHAMBER CONSTRUCTION -

DESIGNER J. HERLIHY / K. TRELEWICZ		DAYTON T. BROWN INC. BOHEMIA NEW YORK
CHECKER		COMBINED ENVIRONMENTS RELIABILITY TEST FACILITY
ENG K. Trelewicz / K. TRELEWICZ	7-26-76	
APPROVAL		
CUST APPROVAL		SIZE C · CODE IDENT 96362 · DWG NO. L8675
UNLESS OTHERWISE SPECIFIED ALL DIMENSIONS ARE IN INCHES TOLERANCE FRACTIONS ± 1/32 DEC X ± .02 XX ± .01 XXX ± .005		SCALE: NONE · SH 6 OF 6

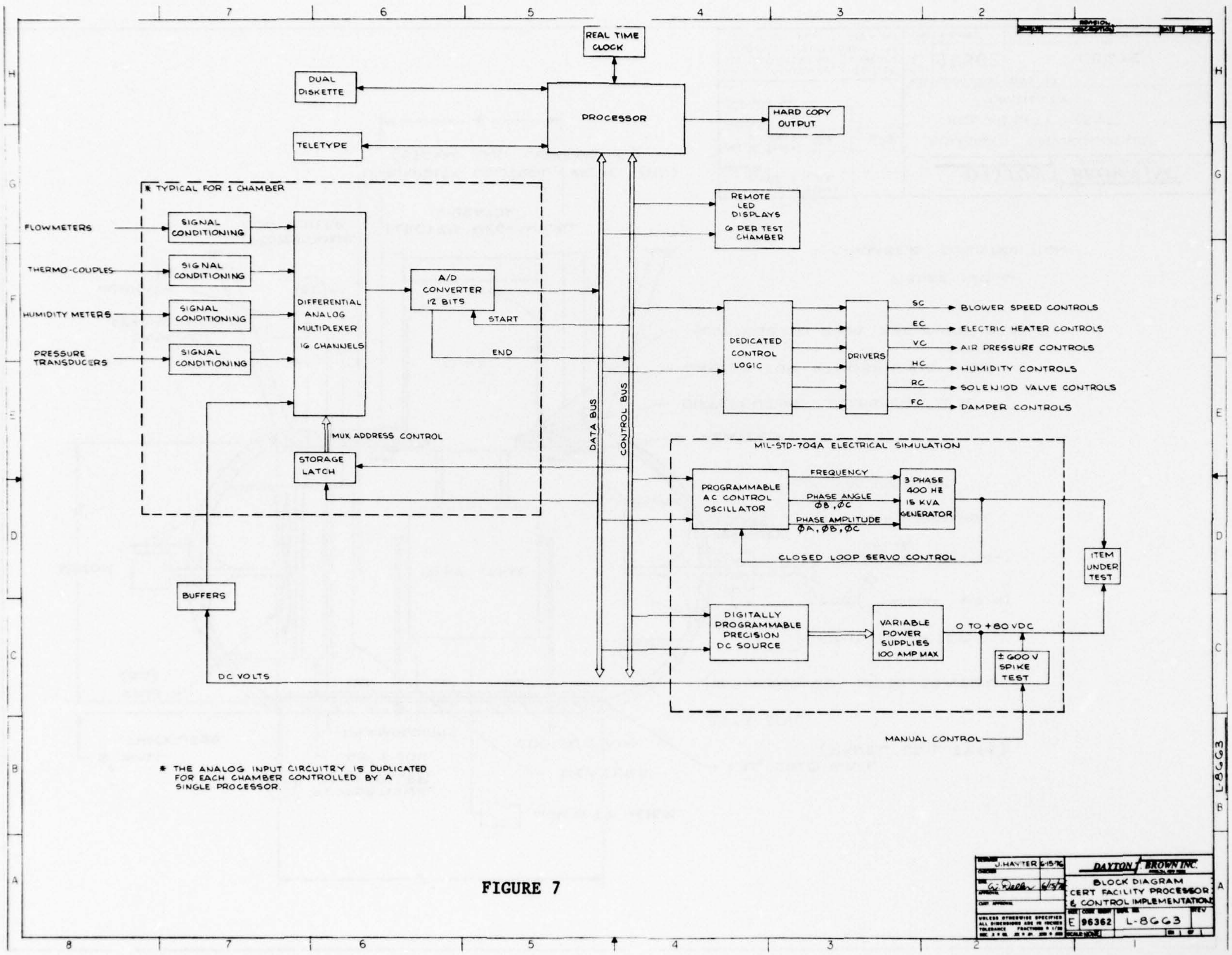
REAL TIME CLOCK
PROCESSOR
DUAL DISKETTE
TELETYPE
HARD COPY OUTPUT
* TYPICAL FOR 1 CHAMBER
FLOWMETERS
THERMO-COUPLES
HUMIDITY METERS
PRESSURE TRANSDUCERS
SIGNAL CONDITIONING
SIGNAL CONDITIONING
SIGNAL CONDITIONING
SIGNAL CONDITIONING
DIFFERENTIAL ANALOG MULTIPLEXER 16 CHANNELS
A/D CONVERTER 12 BITS
START
END
MUX ADDRESS CONTROL
STORAGE LATCH
BUFFERS
DC VOLTS
DATA BUS
CONTROL BUS
REMOTE LED DISPLAYS 6 PER TEST CHAMBER
DEDICATED CONTROL LOGIC
DRIVERS
SC
EC
VC
HC
RC
FC
BLOWER SPEED CONTROLS
ELECTRIC HEATER CONTROLS
AIR PRESSURE CONTROLS
HUMIDITY CONTROLS
SOLENIOD VALVE CONTROLS
DAMPER CONTROLS
MIL-STD-704A ELECTRICAL SIMULATION
PROGRAMMABLE AC CONTROL OSCILLATOR
FREQUENCY
PHASE ANGLE ØB, ØC
PHASE AMPLITUDE ØA, ØB, ØC
3 PHASE 400 HZ 15 KVA GENERATOR
CLOSED LOOP SERVO CONTROL
ITEM UNDER TEST
DIGITALLY PROGRAMMABLE PRECISION DC SOURCE
VARIABLE POWER SUPPLIES 100 AMP MAX
0 TO +80 VDC
± 600 V SPIKE TEST
MANUAL CONTROL
* THE ANALOG INPUT CIRCUITRY IS DUPLICATED FOR EACH CHAMBER CONTROLLED BY A SINGLE PROCESSOR.
J. HAYTER 6-15-76
DAYTON T. BROWN INC.
BLOCK DIAGRAM
CERT FACILITY PROCESSOR & CONTROL IMPLEMENTATION
E 96362 L-8663

FIGURE 7

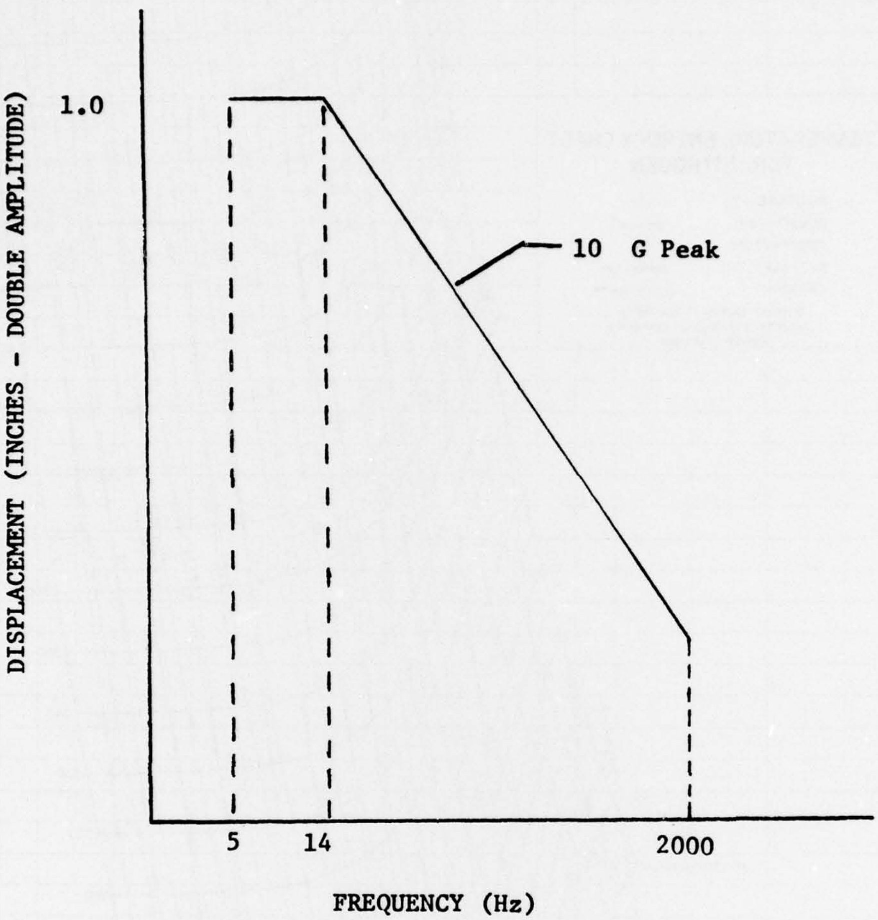

FIGURE 8 SINUSOIDAL VIBRATION CURVES

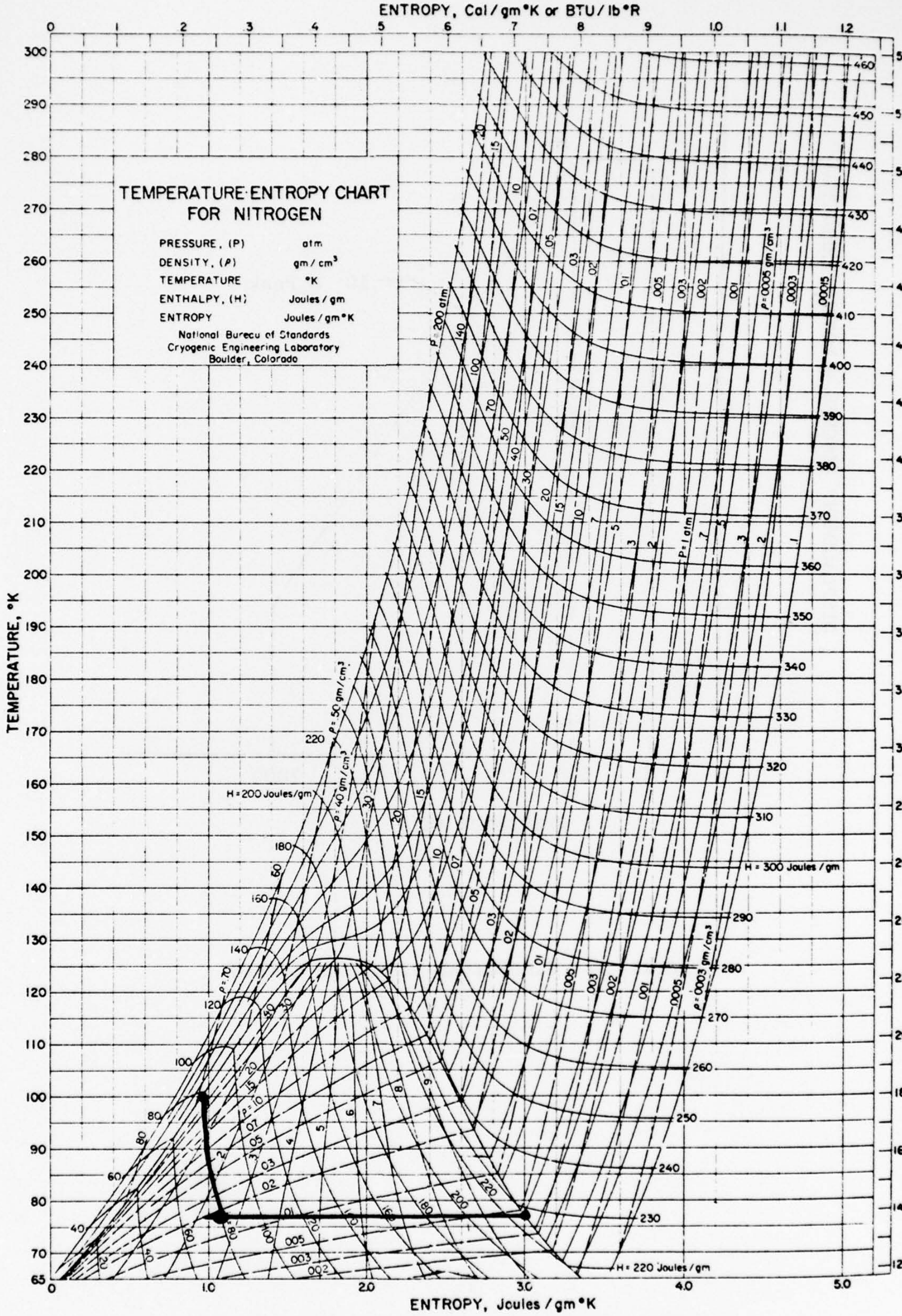

Fig. 9 Temperature-Entropy Diagram for Refrigerant 728

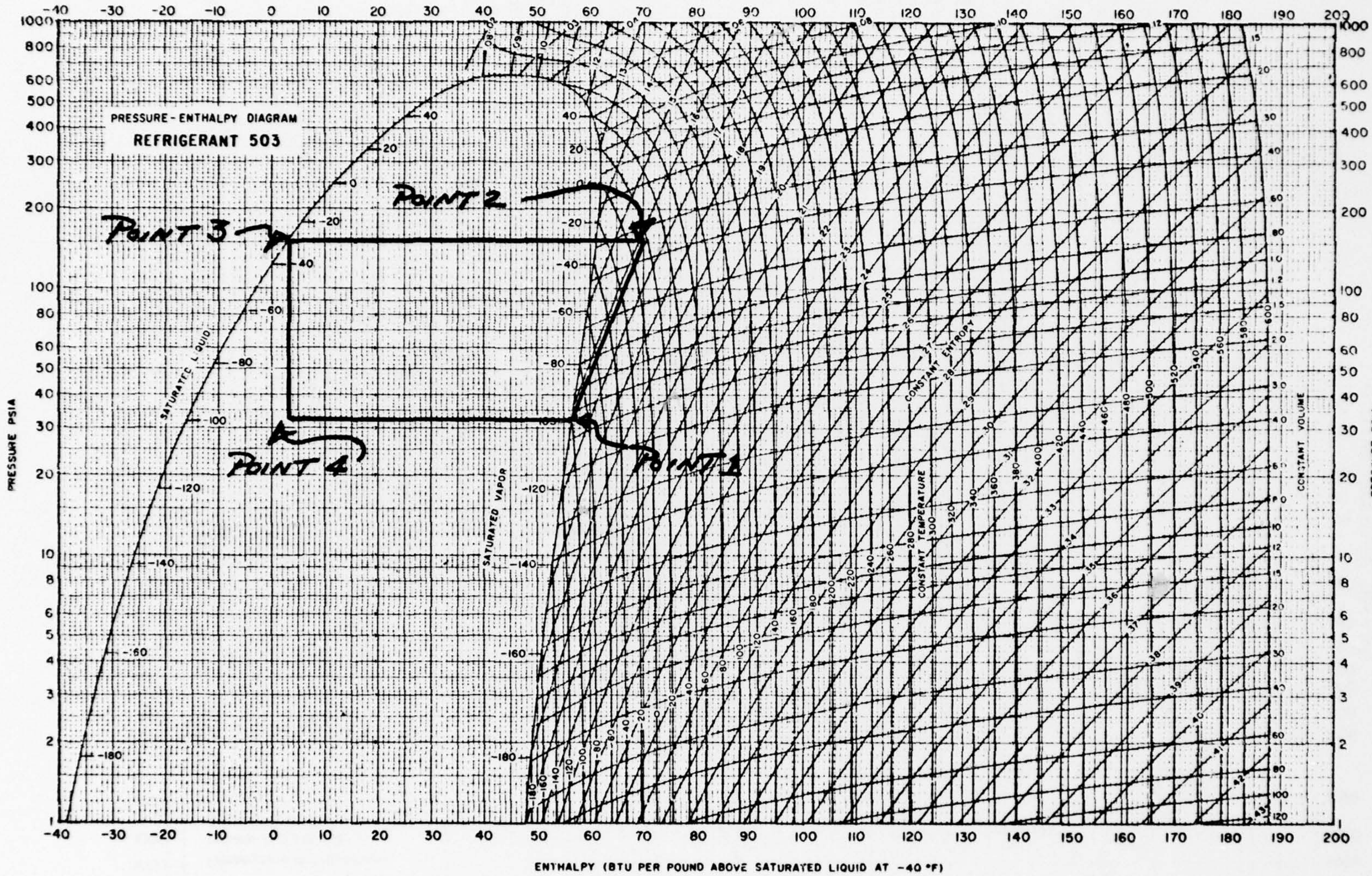

Fig. 10 Pressure-Enthalpy Diagram for Refrigerant 503

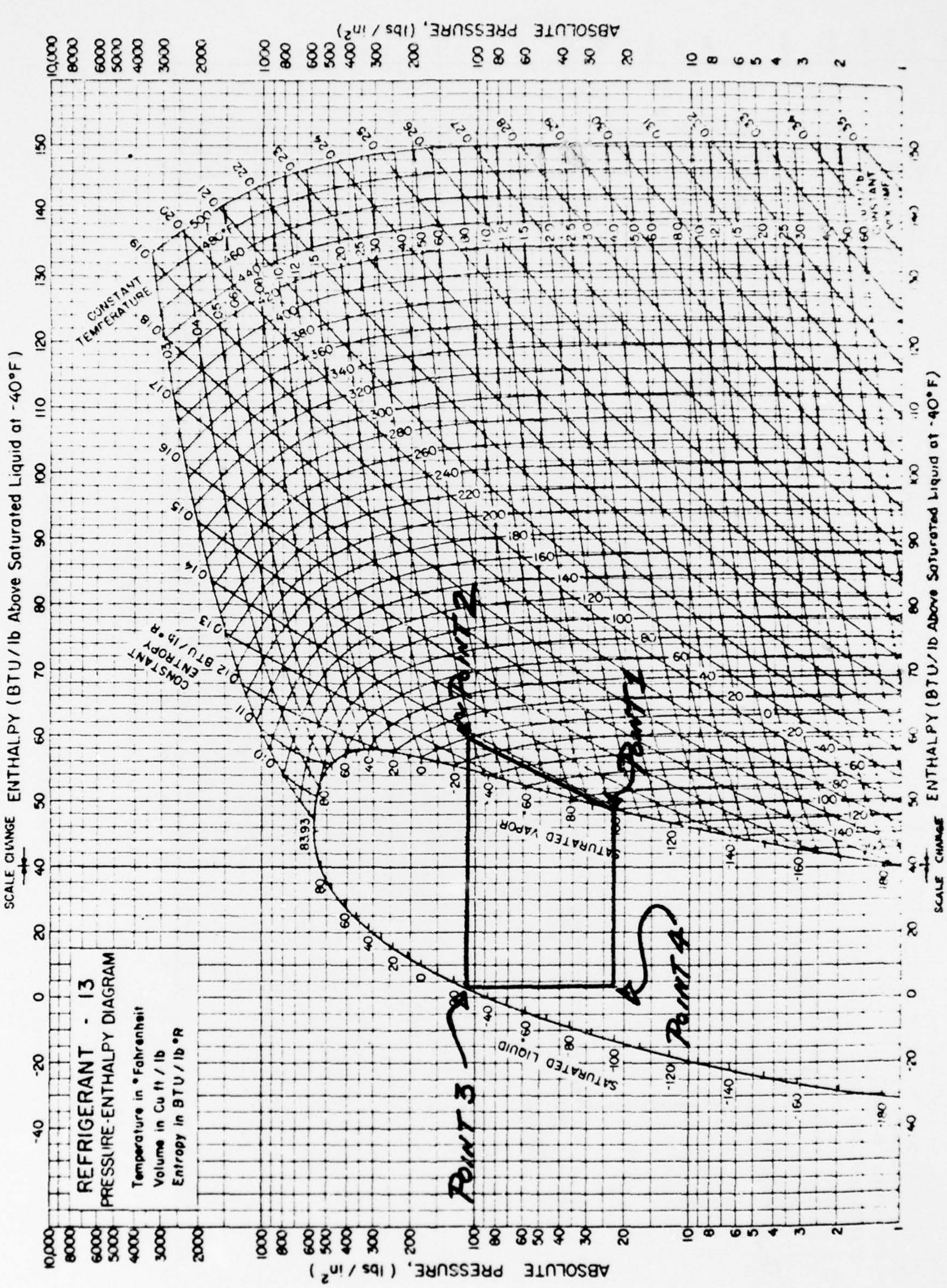

Fig. 11 Pressure-Enthalpy Diagram for Refrigerant 13

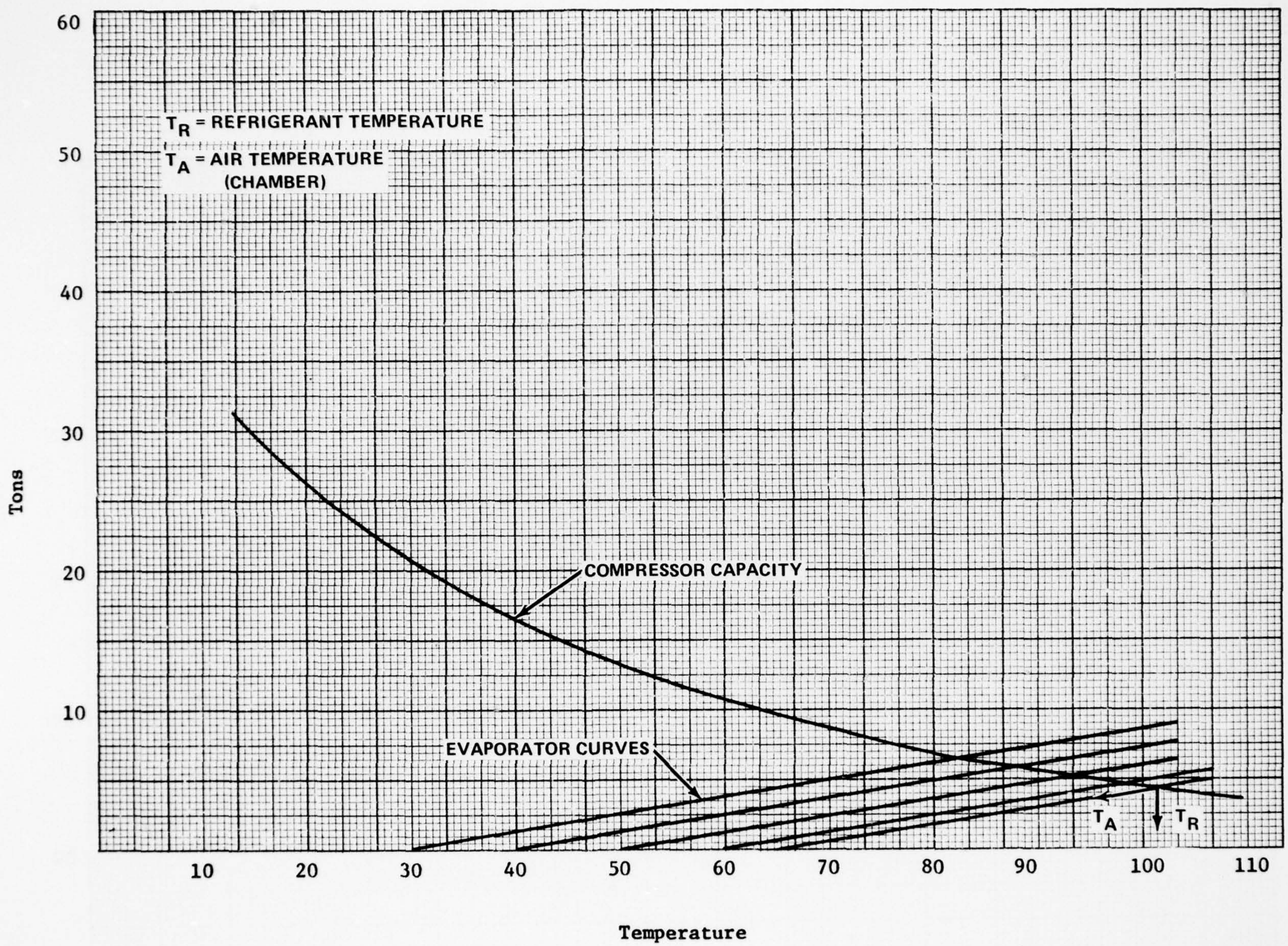

Figure 12A 20 CFM Compressor vs Evaporator

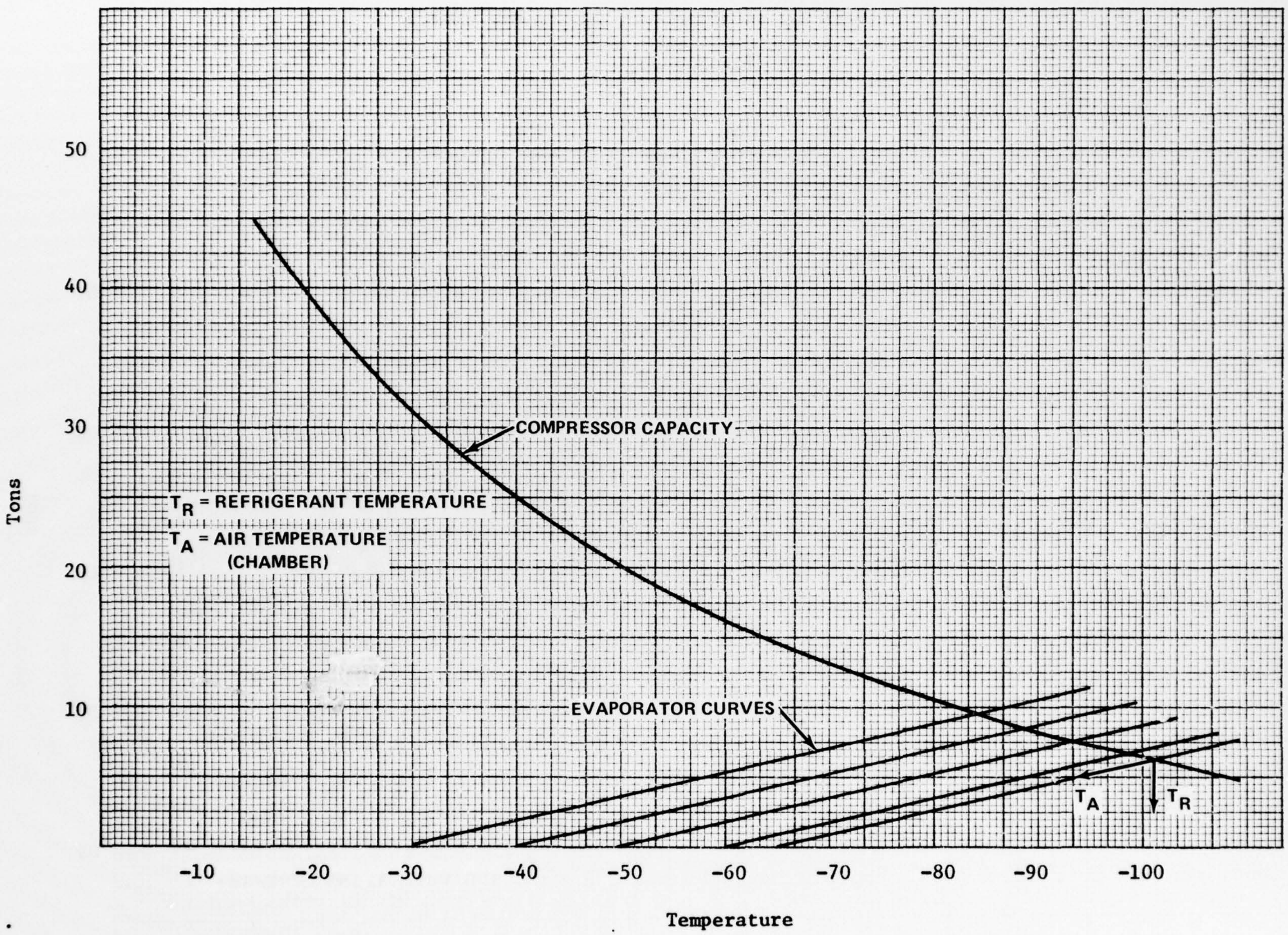

Figure 12B 30 CFM Compressor vs Evaporator

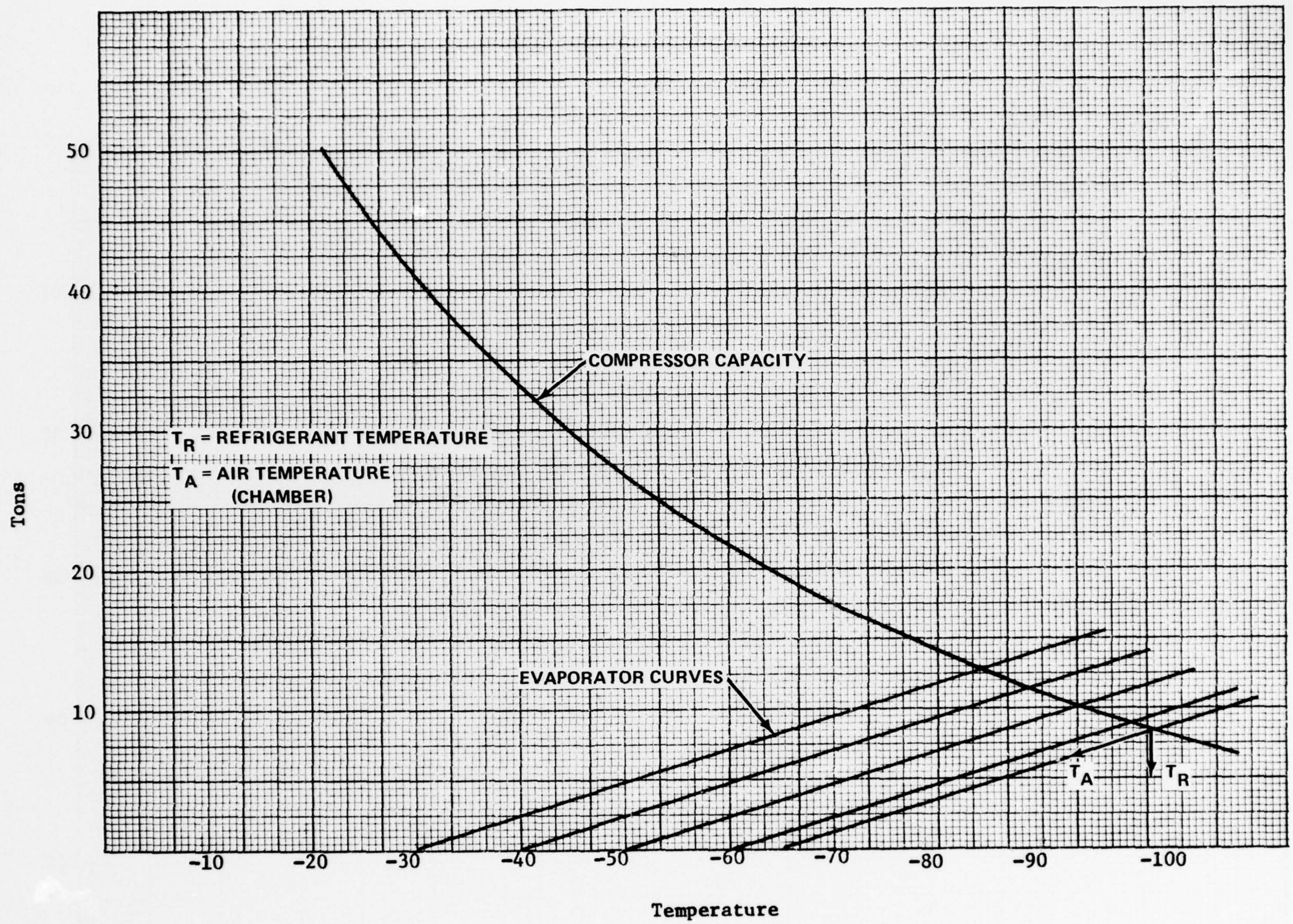

Figure 12C 40 CFM Compressor vs Evaporator

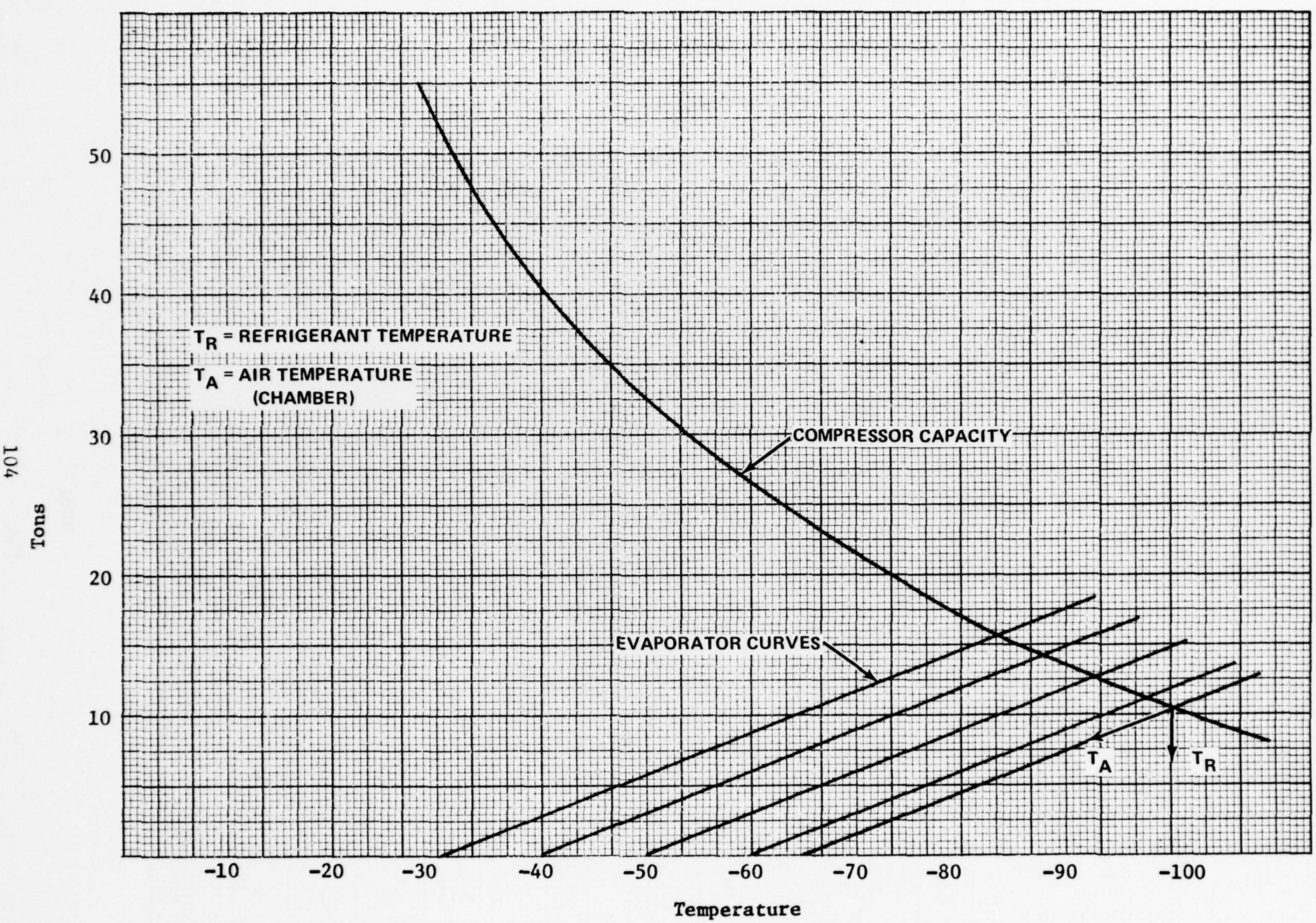

Figure 12D 50 CFM Compressor vs Evaporator

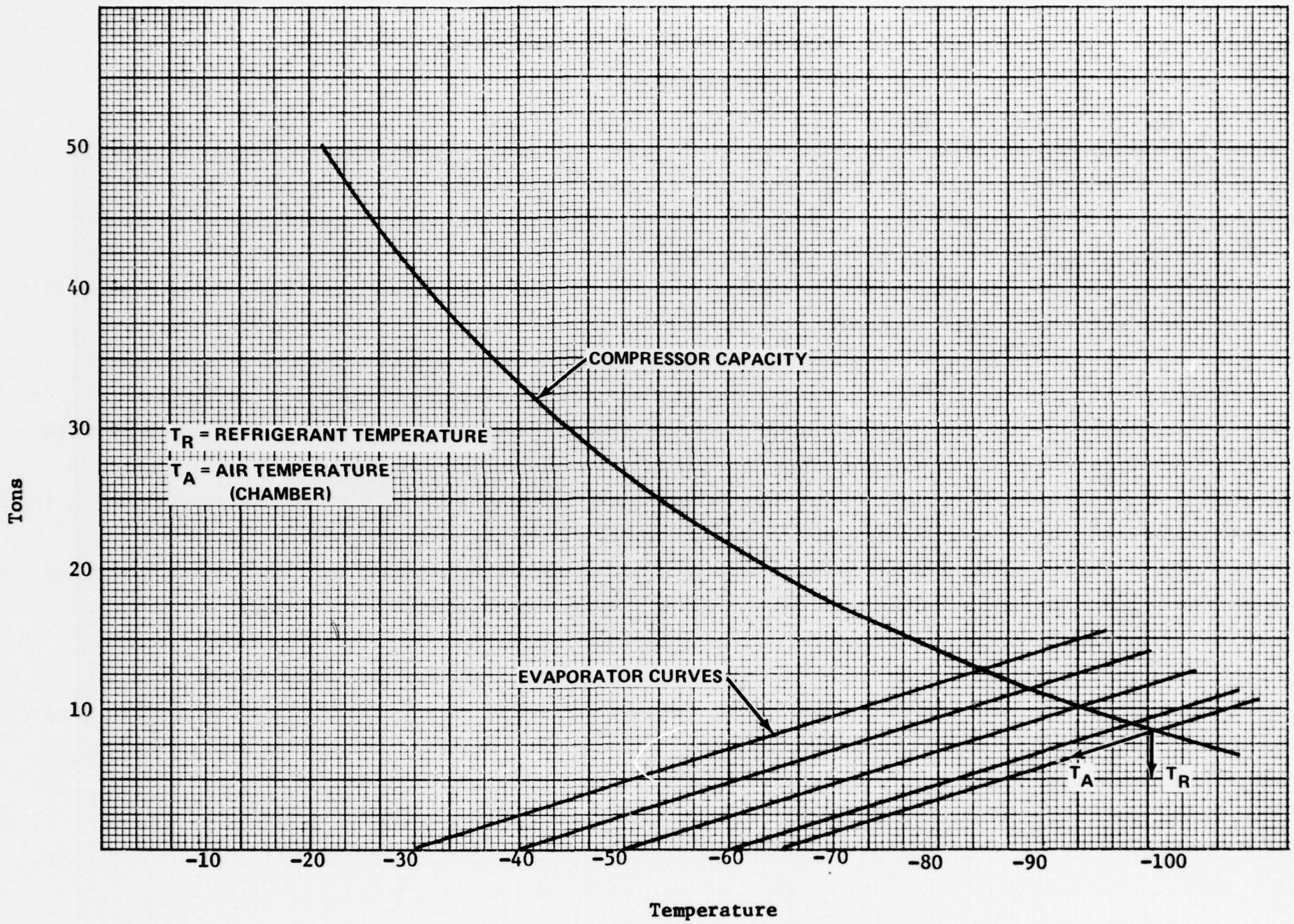

Figure 12C 40 CFM Compressor vs Evaporator

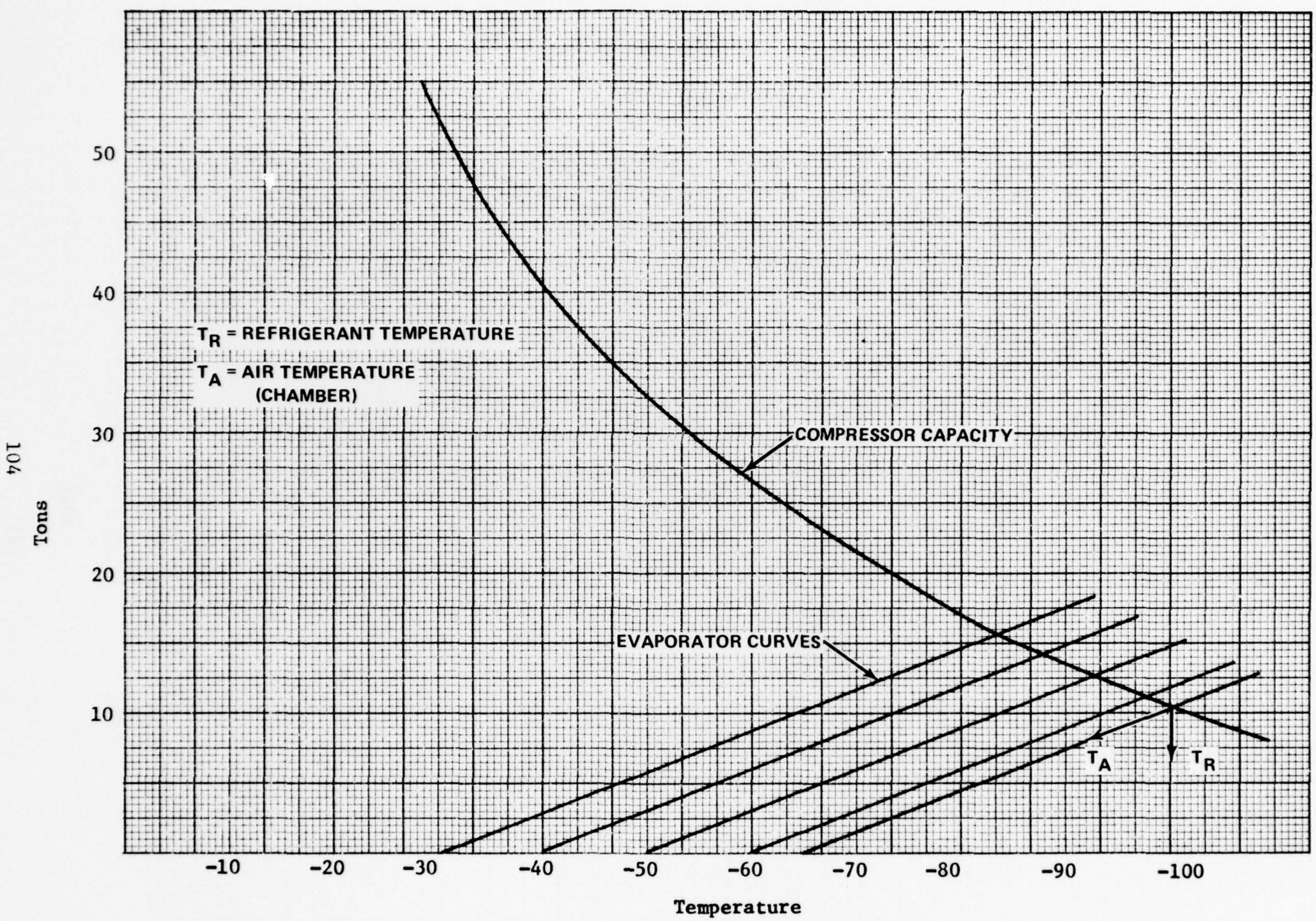

Figure 12D 50 CFM Compressor vs Evaporator

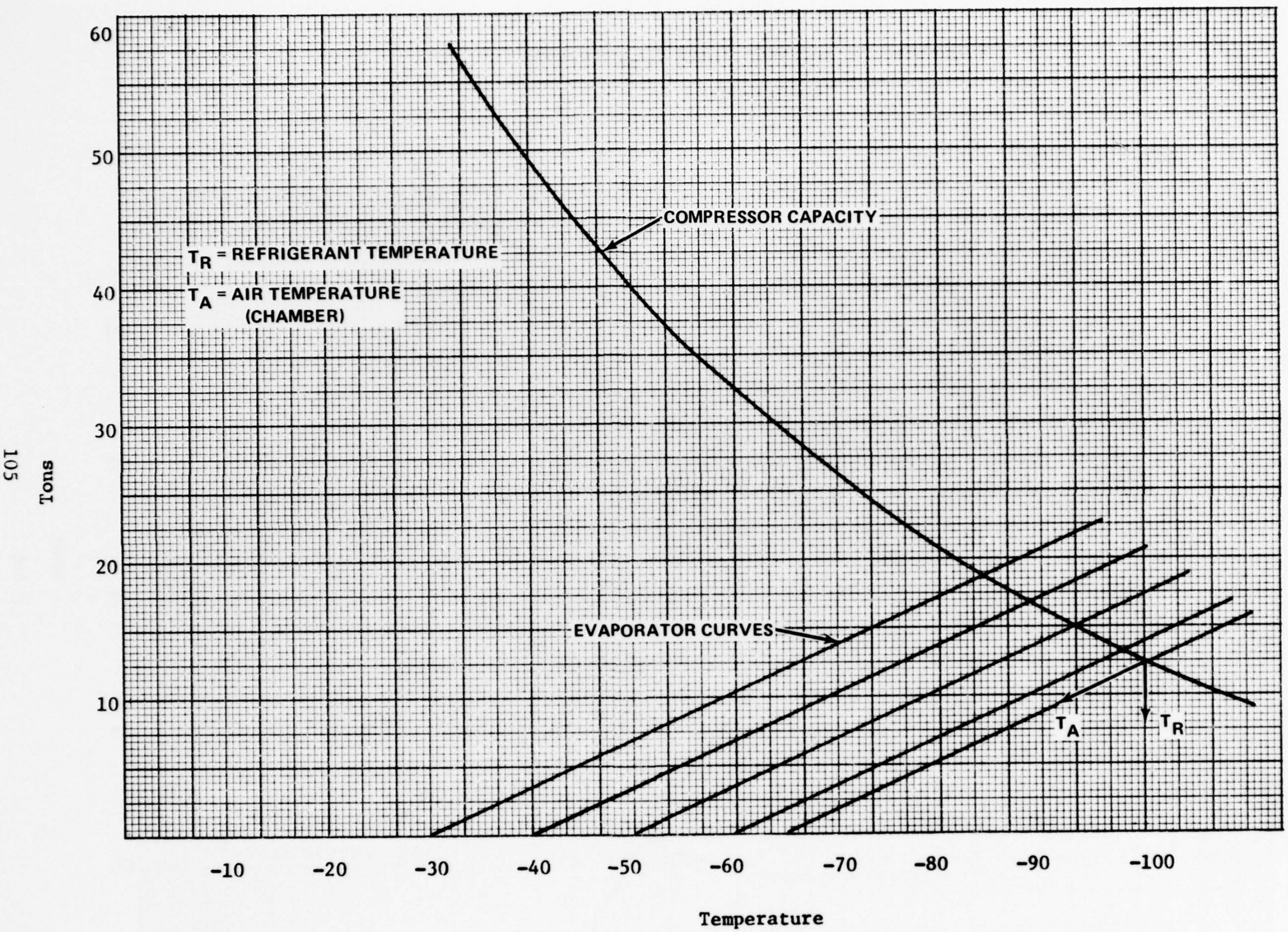

Figure 12E 60 CFM Compressor vs Evaporator

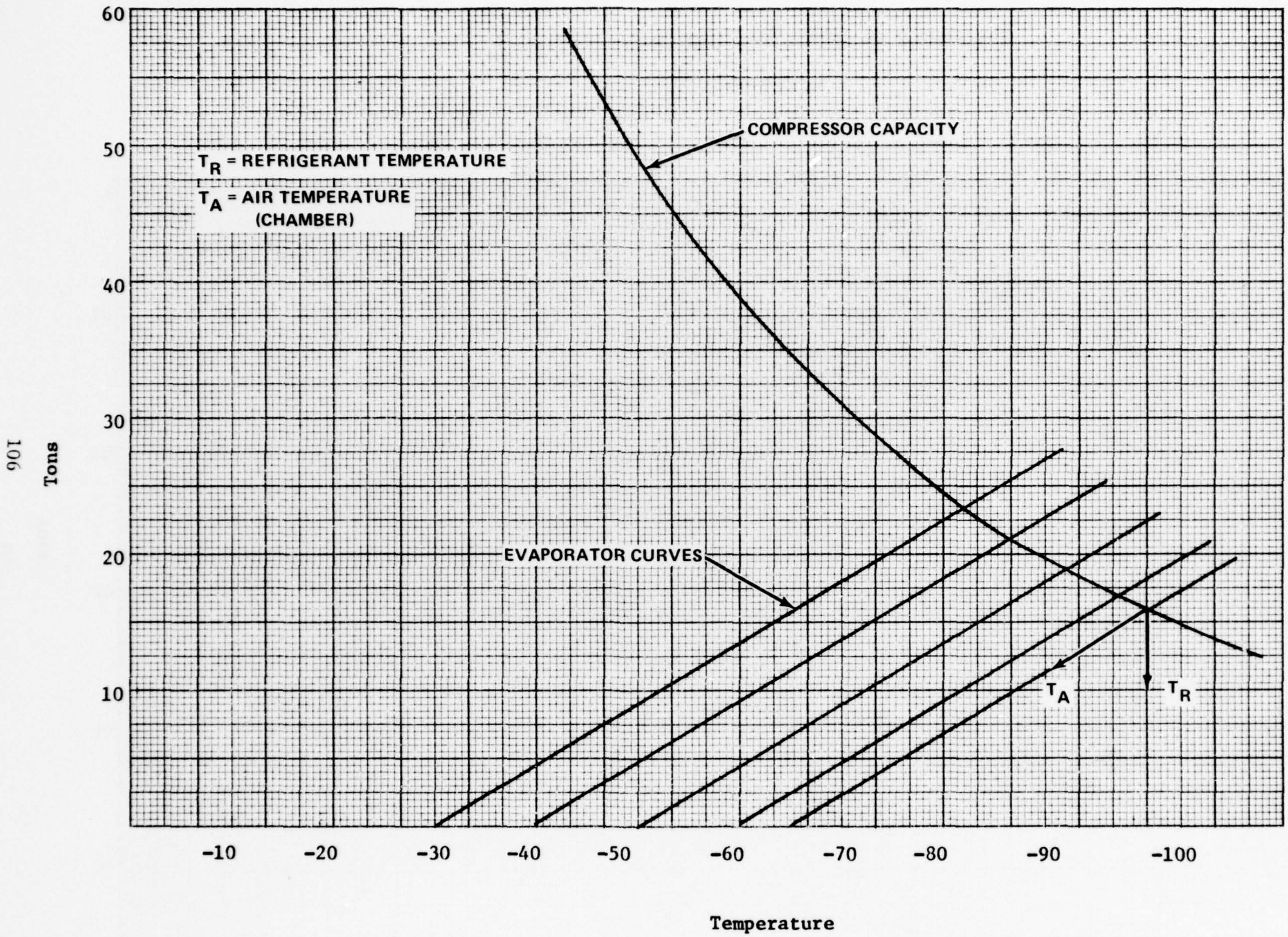

Figure 12F 75 CFM Compressor vs Evaporator

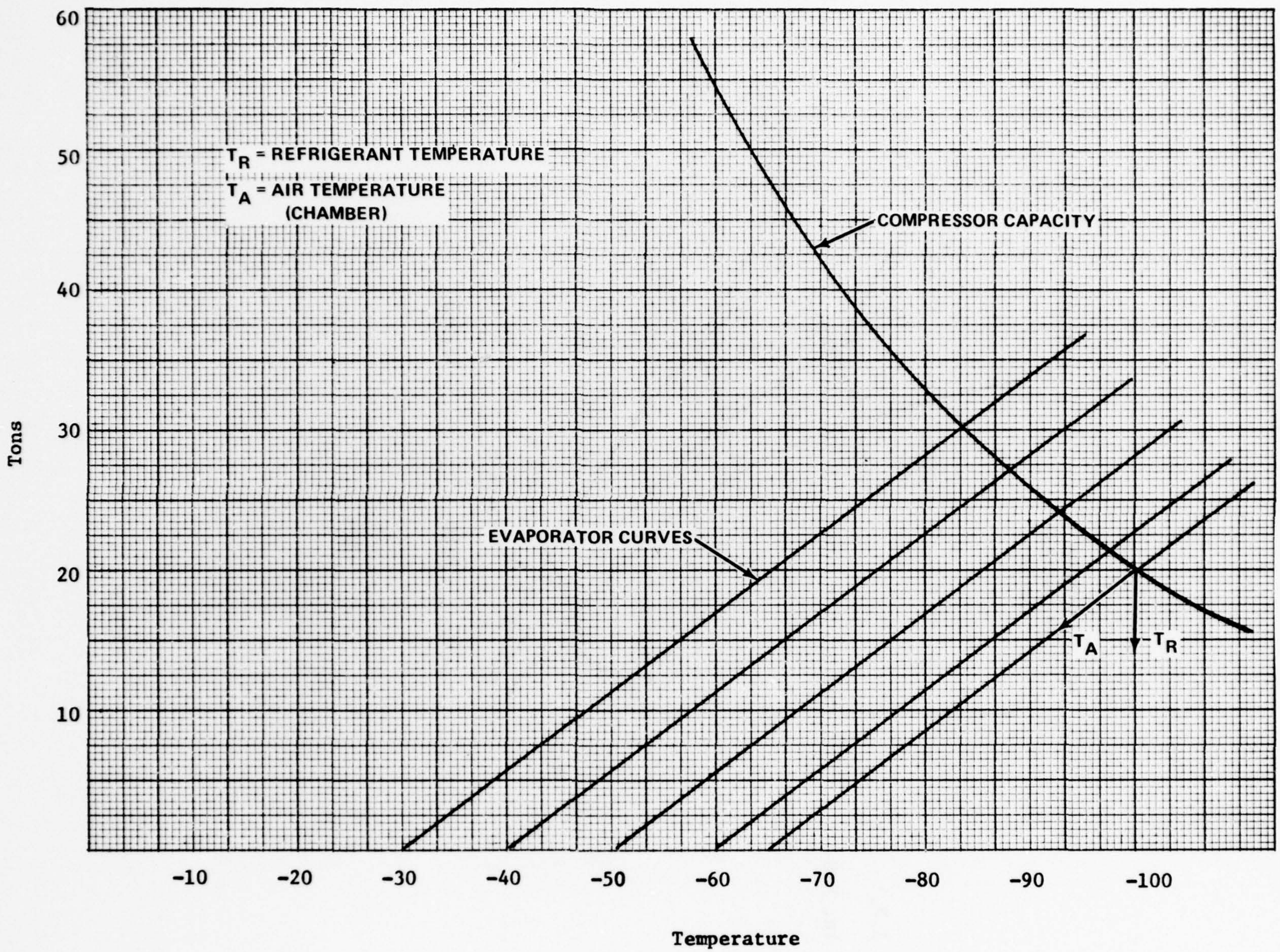

Figure 12G 100 CFM Compressor vs Evaporator

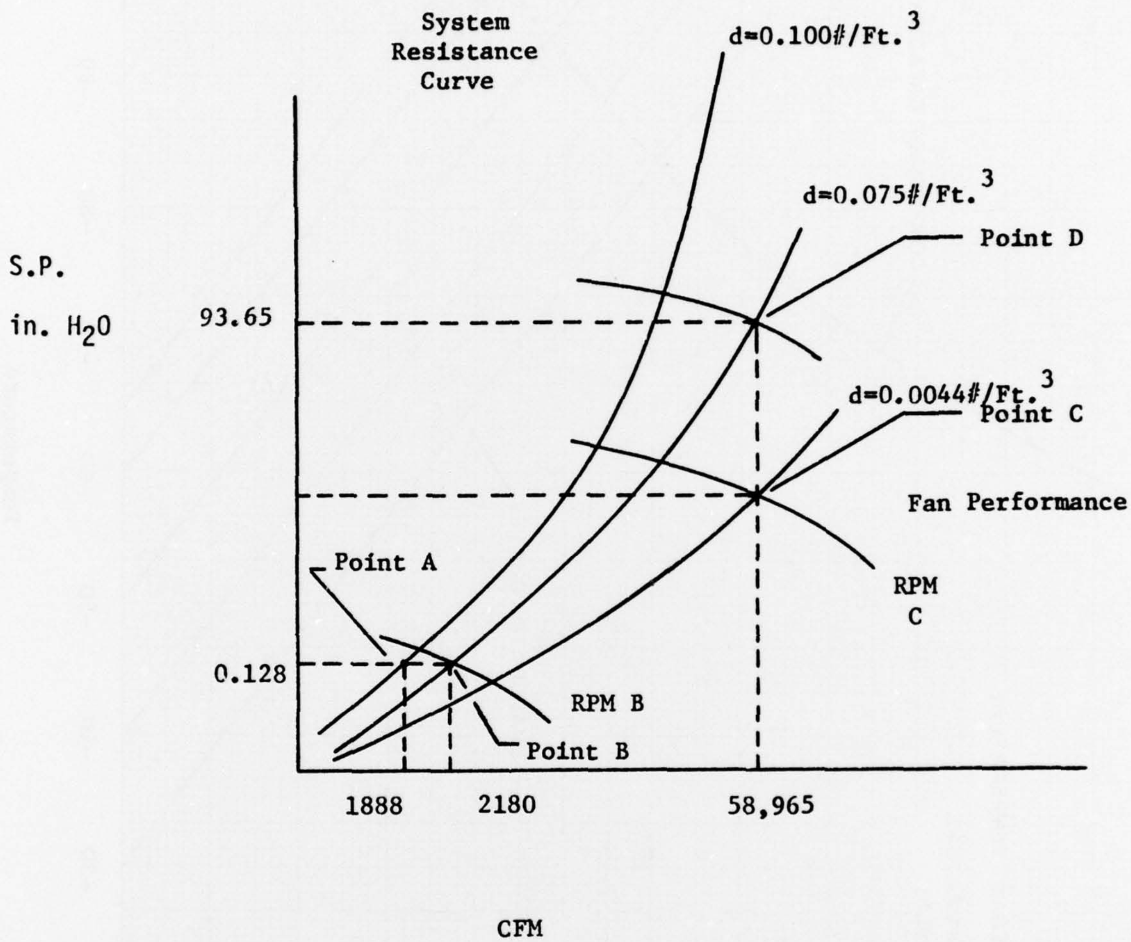

Figure 13. Fan Operating Characteristics

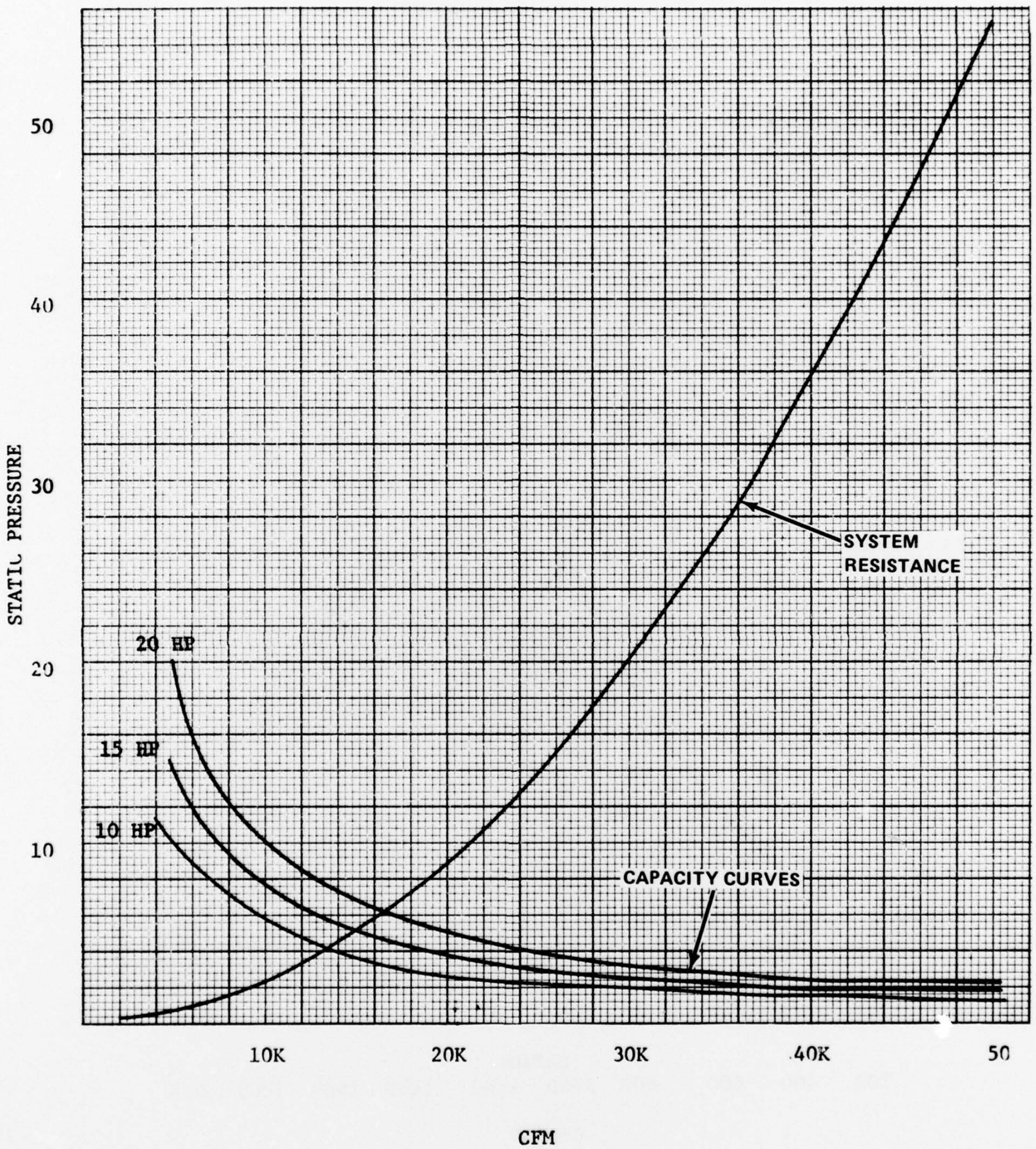

FIG. 14 SYSTEM RESISTANCE VS. VOLUMETRIC FLOW

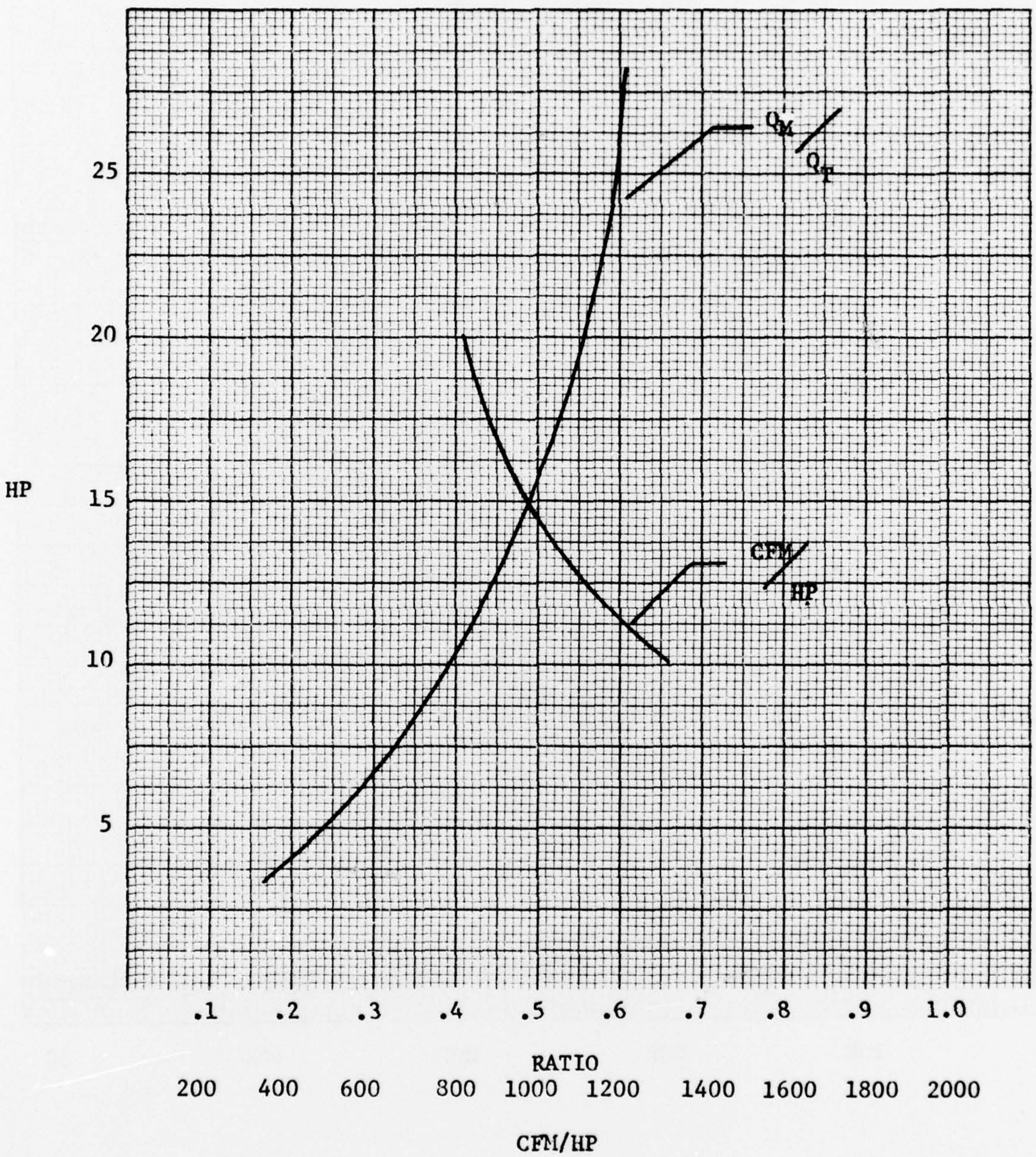

FIG. 15 HORSEPOWER VS. RATIO

HEAT LOAD FOR A GIVEN SPACE-MASS TYPE INSULATION

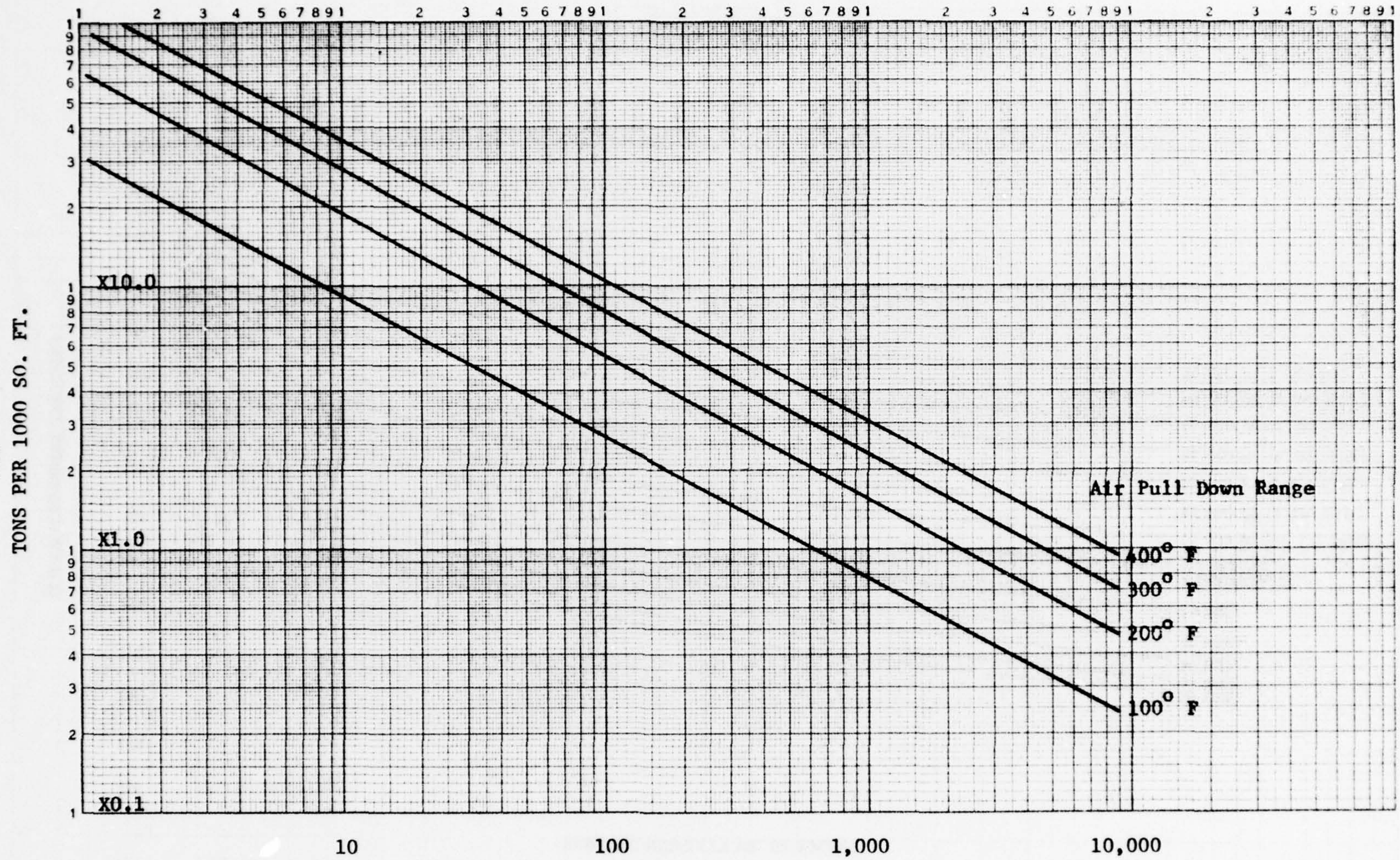

FIGURE 16 TIME (MINUTES)

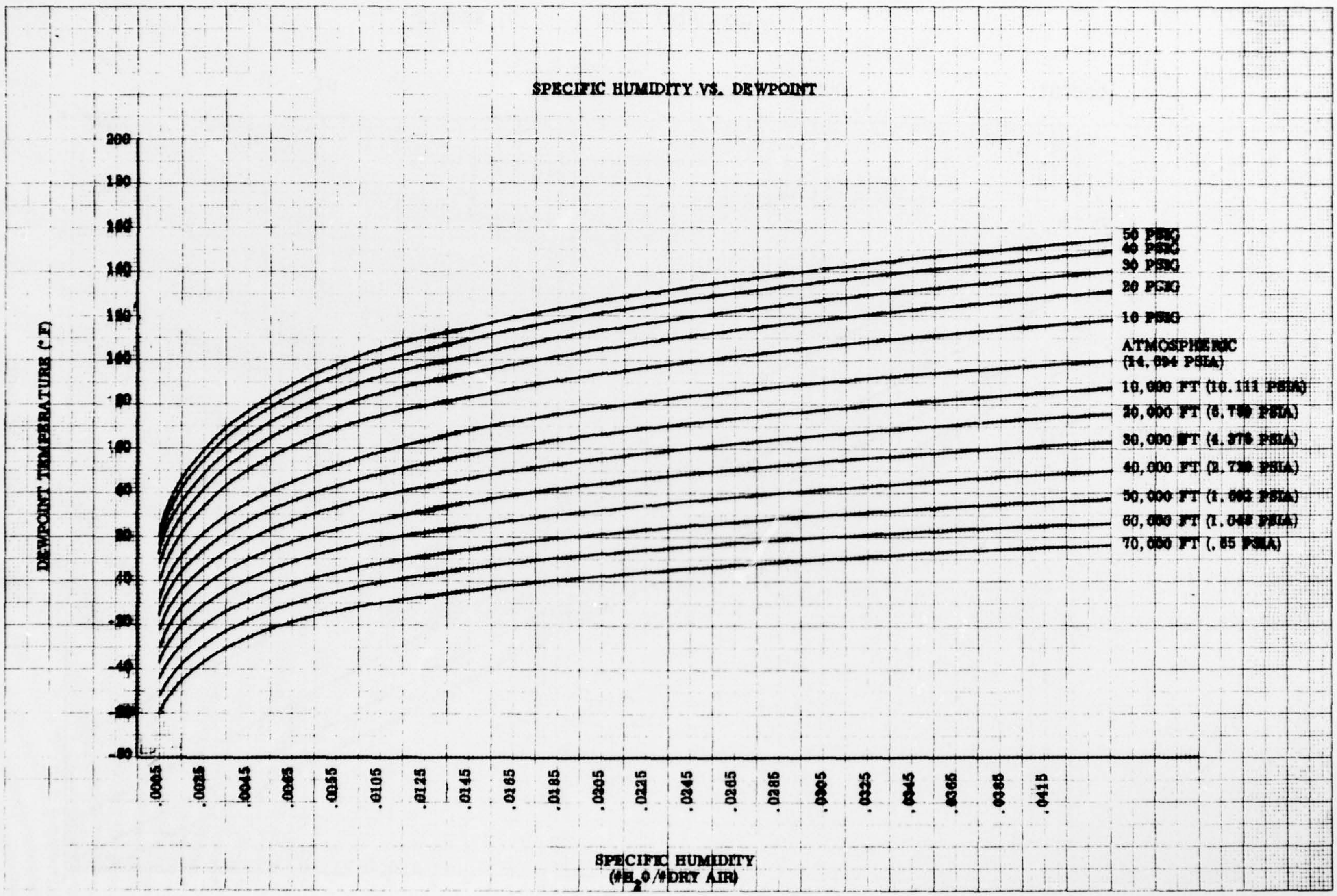

FIGURE 17 SPECIFIC HUMIDITY VS. DEWPONT

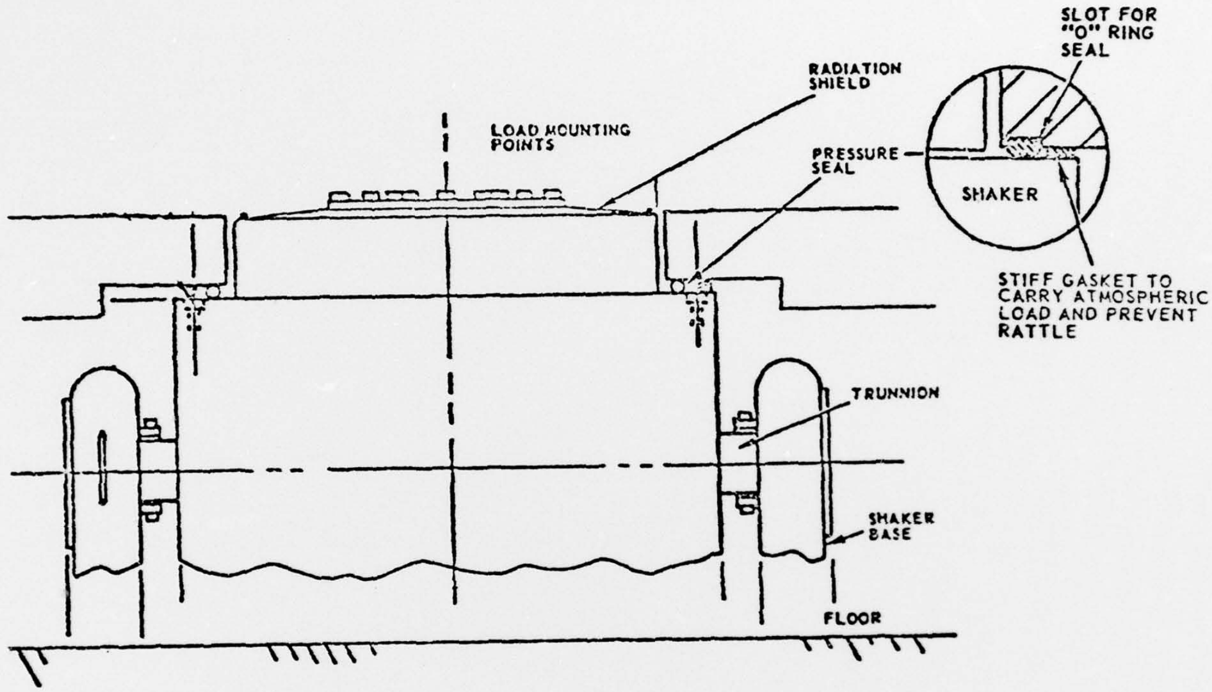

Figure 18 Piggyback Chamber - Shaker Outline

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