THE DESIGN OF REFRIGERATION EQUIPMENT

FOR

LOW TEMPERATURE TESTING ROOMS

by

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Thesis submitted to the Graduate Faculty of the

Virginia Polytechnic Institute

in candidacy for the degree of

MASTER OF SCIENCE

in

Mechanical Engineering

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May, 1954

Blacksburg, Virginia

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I

INTRODUCTION

During World War II, with the armed forces waging war on several fronts and with the development of high altitude aircraft, the necessity for a great deal of low temperature testing was brought about. The production testing of various instruments and appliances, such as radios, cameras, clocks, and meters, which would be subjected to low temperatures in arctic climates or at high altitudes, was found necessary. This subject had been given very little consideration prior to the war. When it became necessary to test equipment at a low temperature, it was normally transported to a cold climate and there the tests were conducted. With the increased demand for low temperature testing, the expense and time delay encountered in such movements became prohibitive.

During and since the war many low temperature test chambers have been built. The design problems encountered, such as load determination, defrosting, lubrication, refrigerant selection, proper staging, and control selection, have been similar, but the solutions have been different. Moore, Gellalles, and Young state: "Experience has shown that research requirements change so quickly and so radically that the original design specification cannot be much more than a hopeful guess."(22)

Mr. Coy W. Brown states: "Any systems requiring evaporating refrigerant temperatures below -40° F. should be attempted only by engineers experienced in this presently special field."(5)

Since definite data on equipment for low temperature test rooms seems to be limited, it is believed that from a design of such a machine much valuable information can be gained.

Two low temperature testing rooms have been constructed in the Mechanical Engineering Laboratory at The Virginia Polytechnic Institute. One will operate at temperatures down to 0° F. and the other will operate at temperatures down to -40° F. In addition to being used for low temperature research, it is anticipated that the refrigerating equipment for these rooms will be used by students taking Mechanical Laboratory in routine experiments. The object of this thesis is to design suitable refrigeration equipment for the two rooms. The design will include specifications, bill of material and equipment, detail and assembly drawings, and operating instructions. Available data on the design, control, and operation of low temperature refrigerating equipment will be assembled and correlated. This information, with interpretation thereof, will be used in solving a practical engineering problem which is, at the present, in a special field.

II

REVIEW OF LITERATURE

A - Load Determination

1 - General

Load determination is of extreme importance in low temperature design. Approximate methods must be discarded and the load determined to a reasonable degree of accuracy. A small inaccuracy in load determination for a refrigerating system that operates at a low suction temperature can cause the horsepower and sizing of the compressor to vary a considerable amount. As the suction temperature of a condensing unit decreases, this effect becomes more pronounced. Mr. Aulsebrook has published approximate performance curves of production condensing units used in sub-zero work. The curves have been prepared using both Freon-12 and Freon-22 as the refrigerants.(3) He has plotted curves with British thermal units (BTU) per watt hour as the ordinate and suction temperature as the absissa. The suction temperature ranges from -110° F. to 0° F. From 0° F. to -30° F. the curves have been plotted for single stage compression. From -40° F. to -70° F. the curves have been plotted for two stage compression. From -80° F. to -110° F. the curves have been plotted for three stage compression. With a suction temperature of 0° F., a condensing unit using Freon-12 as a refrigerant has a capacity of five BTU per watt hour. When

the suction temperature is decreased to -50° F., a two stage unit has a capacity of 2.66 BTU per watt hour, and at -110° F. a three stage unit has a capacity of one BTU per watt hour. It is noted from the approximate performance data that an inaccuracy in load determination will vary the horsepower of the condensing unit approximately five times as much when the suction temperature is -110° F. as it will when the suction temperature is 0° F. Even though research requirements change quickly and radically, every effort should be made to accurately determine the load and thus prevent improper sizing of expensive equipment.

The load sources commonly encountered in low-temperature test work may be summarized in the following groups: (1) Wall heat gain or leakage, (2) air changes and (3) product load. (3) This summation is slightly different from that used in general refrigeration work in that it does not include a fourth group called miscellaneous. The miscellaneous load normally consists of the heat dissipated from electrical apparatus such as lights and motors and also the heat dissipated from people working in the refrigerated space. This load is classed as product load when applied to load calculations for low-temperature test cabinets. For a complete load calculation, it is necessary for each of the three load sources to be treated separately to determine the total load.

2 - Wall Heat Gain

The heat entering the cabinets by virtue of conduction through the wall structure is subject to calculation by conventional methods, provided the conductivity factors for all insulating materials are accurately known. Conductivity factors used should be based on experimental data taken at the approximate temperature level to be attained. These data are available from numerous sources and for a special material it can usually be obtained from the manufacturer. Changes of as much as 100 percent have been observed in the conductivity of some materials at extreme temperatures.(3)

In some cases the wall structure of test chambers requires that special allowances be made in the calculation of the conducted load. A careful inspection should be made for structual features such as framing elements, breaker strips, and conductive parts projecting through the insulation. Such features are very commonly found in the frame and the supporting members of the frame for a doorway. In common experience the insulating value of a door is approximately the same as a wall of similar insulation. This is especially true if the door is of the overlap type. The stiffeners for this type door are usually contained in the overlap of the door on the frame, thus presenting to the cold side only the insulated area of the door.(31)

Stoever gives the following equation based on Fourier's law that can be used in calculating the conducted load through the wall structure:(34)

$$\underline{1} \qquad Q = UA\,(\Delta t)$$

where: Q = heat conducted, BTU/hr.

U = coefficient of transmittance, BTU/(hr.) (^{o}F.) (sq. ft.)

A = average area of the section, sq. ft.

Δt = difference between the average outside temperature and the average refrigerator temperature, degrees F.

It will be noted that a decision was made to use the arithmetic mean of the inner and outer areas in calculating the heat conducted due to its general use in industry. Mr. Hechler states: "For such cases as boxwalls and the like, the arithmetic mean of the inner and outer areas is often used. For small rectangular enclosures with relatively thick walls, the arithmetic mean area causes serious error because at the edges and corners the heat flow is complicated, formulas applicable under these conditions are available, giving a smaller effective area than other methods commonly used."(14) In Mr. Hechler's bibliography, he refers to the formulas given by McAdams.(20) McAdams was reviewed and it was found that he began his study of heat transmission with rectangular bodies having walls at least one-half as thick as the shortest inside dimension. On the basis of

this information, the walls of the test chambers were not considered of a relative thickness to warrant the application of shape factors.

Allen, Walker, and James use the following equation for determining the transmission coefficient for a wall consisting of two materials.(2) This formula can be expanded and used for walls consisting of more than two materials.

$$\underline{2} \quad U = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{1}{f_o}}$$

f_i = surface conductance of the inside wall in BTU per hour per square foot per degree difference in temperature between the air and the wall

f_o = surface conductance of the outside wall in BTU per hour per square foot per degree difference in temperature between the air and the wall

x_1 = thickness of the first material in inches

x_2 = thickness of the second material in inches

k_1 = conductivity of the first material in BTU per hour per square foot of area per inch of thickness per degree F. difference in temperature

k_2 = conductivity of the second material in BTU per square foot of area per inch of thickness per degree F. difference in temperature

The quantity U is the over-all transmission coefficient for a wall, and it should not be confused with the internal conductivity of the material. The transmission coefficient U also includes the surface resistances. Since the quantities f_i, f_o, and $\frac{k}{x}$ are the conductivities of the respective elements of a wall, their reciprocals may be considered as the resistances to heat flow. The transmission coefficient of any type of wall can thus be computed if the surface resistances and the conductivities of the various materials that make up the wall are known. With nonhomogeneous materials such as hollow tile it is more convenient to express the conductivity as the total conductance for the stated thickness. The formula which expresses this relationship is:

$$C = \frac{k}{x} \qquad (3)$$

where: C = total conductance of a material in BTU per hour per square foot of area per degree F. difference in temperature

The American Society of Heating and Ventilating Engineers has published values for surface conductance, total conductance, and material conductivity for numerous materials.(13) The Johns-Manville Company has published the conductivity factors for Rock Cork at various mean temperatures.(32)

3 - Air Changes

The next load group to be considered will be the sensible and latent heat load caused by air changes. Each time a door to a test room is opened, a certain amount of cold air spills out and is replaced by the warm outside air. When the doors are closed, test rooms are usually subject to minor air changes due to infiltration of air through small cracks such as those normally found around doors. Test rooms that are to be used for testing internal combustion engines require that a quantity of outside air be supplied for combustion purposes. All of the warm outside air that enters a test room must be reduced in temperature to that of the refrigerated space, thus adding to the refrigeration load.

It is difficult to determine, with any degree of accuracy, the quantity of air that enters a refrigerated space by infiltration and door openings. Mr. Segal has published a table of expected air changes that has proven practical.(26) He has combined the air changes due to door openings and those due to infiltration into one figure. The table has been prepared for room volumes from 200 cubic feet to 100,000 cubic feet. It also includes variations for three types of storage conditions: average, heavy usage, and long storage. This table can be applied in the determination of air changes for a test room if the door opening is similar to that of a cold storage room.

The sensible heat, latent heat, and the moisture to be removed from the outside air that enters the test rooms must be determined. Martin and Willey, in their preparation of a low-temperature psychrometric chart, outlined six heat quantities that must be considered in cooling air from a temperature above 32° F. to a temperature below 32° F.[19] They are:

"1. Sensible heat of the vapor between dry bulb temperature and dew point.

2. Latent heat of the vapor condensed at the dew point.

3. Sensible heat of the condensate between the dew point and 32° F. if the dew point is above 32° F. If the dew point is below 32° F. this value does not appear.

4. Latent heat of fusion of the water frozen.

5. Sensible heat of ice and vapor between 32° F. and the final temperature.

6. Sensible heat of the air."

Martin and Willey assumed that when a mixture of air and water vapor was cooled, all condensation occurred at the dew point. The actual process was known to be progressive but the assumption simplified the calculations.

In the application of the six steps outlined by Martin and Willey, it is first necessary to determine the specific humidity

of the air both before and after it has been cooled. This can be accomplished by the application of Dalton's law of partial pressure to the familiar laws of Boyle and Charles. Theoretically this application is not scientifically correct but it is accurate enough for all practical purposes.(18) The following equation is derived when a mixture of one pound of dry air and a quantity of water vapor occupy the same volume and are at the same temperature.

$$\frac{B - e}{e} = \frac{7,000}{w} \times \frac{53.35}{85.78}$$

$$w = \frac{4353e}{B - e} \qquad (4)$$

w = weight of water vapor, grains per pound of dry air

B = the barometric pressure, inches of Hg.

e = pressure of the water vapor, inches of Hg.

7,000 = grains per pound

53.35 = gas constant (R) for air

85.78 = gas constant (R) for vapor

Thus if the vapor pressure can be determined, the weight of water vapor per pound of dry air can be determined.

When the wet-bulb temperature, dry-bulb temperature, and the barometric pressure of a mixture of dry air and water

vapor are known, the vapor pressure of the water in the mixture can be determined by Carrier's equation.(7)

$$5 \quad e = e' - \frac{(B - e')(t - t')}{2,800 - 1.3t'}$$

where: e' = saturation pressure at the wet-bulb temperature, inches of Hg.

t = dry-bulb temperature of the air, °F.

t' = wet-bulb temperature, °F.

When the barometric pressure, dry-bulb temperature, and dew point temperature of a mixture are known, the partial pressure of the vapor can be determined from the steam tables. This is possible since the dew point of a mixture of air and water vapor is the saturation temperature corresponding to the actual partial pressure of the vapor.

4 - Product Load

The product load for a test room consists of all sensible and latent heat dissipated inside the test room. It includes the heat introduced by all electrical apparatus, the heat generated by test equipment, the heat added by the presence of operators, and any heat added by the movement of a warm product into the cold test room.

The heat introduced by electrical energy must be accounted for at 3415 BTU per kilowatt-hour. This includes the input to the fan motors on forced convection evaporators

provided the motors are mounted inside. Some installations, such as The Canadian Lagoratory for Arctic Testing, have the fan motors mounted outside the test rooms and connected to the fans by shafts.(24) The net losses in heat radiated to the rooms are eliminated, leaving only the heat equivalent of the useful work done as an addition to the refrigerated load. Mr. Aulsebrook states that for small motors the saving will amount to 25 to 40 per cent of the total motor input.(3) This is also substantiated by data published by Mr. Segal.(26)

The heat and moisture losses from the human body have been found to vary with atmospheric conditions, exertion, and clothing.(26) Considerable information on this subject was available for temperatures ranging from 25° F. to 95° F. For temperatures below 25° F., the information was very limited. Segal(26) has published a table giving the average hourly load due to occupancy for room temperatures from 50° F. down to -10° F. He has given the heat equivalent per person in BTU per hour. He did not divide the heat equivalent into sensible and latent heat. He gave the average heat equivalent for occupancy of a room at a temperature of 0° F. as 1,300 BTU per hour per person. In the design of stratochambers, Mr. Bergdoll used 1,380 BTU per hour per person as the heat equivalent for occupancy.(4) This figure was used for a design room temperature of -70° F. and with the occupants wearing heated suits.

A product placed in a test room at a higher temperature than the room will lose heat until its temperature is equal to that of the room. The heat introduced in this manner can be determined from the weight and specific heat of the product, provided there is not a change in state. If there is a change of state, it is necessary to know its state upon entering the test room, its final state, its weight, specific heat above and below freezing, its freezing temperature, and latent heat.(26)

When a test is designed to start with the test room and the apparatus at room temperature, a great source of the load may be the sensible heat from the cabinet walls, insulation, evaporator, and the product. Mr. Aulsebrook states: "For practical purposes it is safe to assume that the liner, evaporator and approximately one-half of the total insulation weight must be cooled through the entire range."(3) Tests of this type are more often encountered in the design of stratochambers.

B - Refrigerant

Mr. Aulsebrook states: "It is theoretically possible to use almost any refrigerant for low-temperature work, but, practically, the field narrows down to Freon-12, Freon-22 and methyl chloride."(3) He further states that since methyl chloride is to some extent inflammable and under certain circumstances toxic, it has found little acceptance.

In the review of literature conducted by the author, the complete absence of a low-temperature test installation utilizing methyl chloride as the refrigerant was noted, but several of the installations reviewed employed ammonia. Mr. Orr and Mr. Henshaw state that ammonia was used as the refrigerant in the Low Temperature Laboratory located in Ottawa, Canada.(24) Moore, Gellalles, and Young state that ammonia was used as the refrigerant in the Engine Research Building of the N. A. C. A. Lewis Laboratory because it was thought to meet more fully the specified flexibility requirements and offer an installation lower in initial and operating cost.(22) Soling and Brickford tell us that ammonia has been successfully employed for the production of refrigeration temperatures as low as -90° F. They also tell us that most manufacturers employ low-speed compressors for ammonia duty as compared with the commonly used high-speed Freon compressors. The highly efficient, compact, and inexpensive low-side equipment that can be employed with the Freon refrigerants is not generally available for use with ammonia.(29)

Freon-12 is satisfactory to use in the production of evaporator temperatures down to -125° F. This temperature corresponds to an evaporator pressure of approximately 0.5 pounds per square inch absolute. Applying the same low pressure limit to Freon-22, it can be used in the production of

evaporator temperatures down to -137° F. When using reciprocating compressors, an evaporator pressure of 0.5 pounds per square inch absolute is too low for practical operation. The practical low limits in evaporator temperatures when using reciprocating compressors are -115° F. for Freon-12 and -125° F. for Freon-22.(29)

All authorities seem to agree on the physical characteristics of the following refrigerants:

Methyl chloride is to some extent inflammable and under certain circumstances toxic. It has a suction temperature higher than Freon-12 or Freon-22 at a comparative low pressure. Therefore it has no particular advantage over the Freons.

Ammonia freezes at -108° F. It is explosive and toxic, but the conditions that must be satisfied for combustion are such that with ordinary care the vapor is not dangerous. Since ammonia is a strong irritant, any inhalation which is tolerable does not seem to have any after effects. In the presence of moisture it is corrosive to copper-bearing metals.

Freon-12 has a low condensing pressure in comparison with ammonia, thus permitting ordinary lightweight equipment construction. It is non-corrosive to all metals used in refrigeration apparatus. It is not inflammable, explosive, toxic or an irritant.

Freon-22 has generally the same physical characteristics as Freon-12 except that it has a lower boiling point than Freon-12

at low pressures and a higher condensing pressure at normal condensing temperatures.

Carbon dioxide freezes at -69.9° F. It is odorless, noninflammable and is considered nontoxic. It has a very high condensing pressure at normal condensing temperatures.

Propane is inflammable, explosive and toxic. It has suction and condensing temperatures a few degrees below Freon-22 at comparative pressures.

Ethane and Ethylene are inflammable, explosive, and toxic. They have very low boiling points but their condensing pressures are almost as high as carbon dioxide.

C - Evaporators

Mr. Aulsebrook states: "Evaporators are of the forced convection and plate types. In applications in which frequent heating and cooling are involved, the forced convection evaporator has a distinct advantage. The plate coil has found more general acceptance in those applications in which the cabinet is to be chilled to a predetermined low temperature and held there for long periods. It is easily defrosted while cold, maintains a uniform temperature on relatively long cycles, and when used as part of the cabinet wall occupies a minimum of space. The heat absorption per degree temperature difference on either of these two types of evaporators remains substantially the same as has been obtained on similar evaporators in the higher temperature

brackets, but, because of the rapid increase in size and cost of condensing units as evaporator temperatures go down, every effort should be made to improve the evaporator performance in order to reduce to the lowest practical level the temperature difference between refrigerant and space. That is, one must incorporate the largest amount of surface consistent with available space, and design the refrigerant passages, air flow, etc., to utilize this material at its highest efficiency. The pressure drop must be held to a minimum because it handicaps the performance of the condensing unit, yet reasonable velocities are necessary to obtain efficient transfer. Because of these divergent factors, a 'tailor made' evaporator based on considerable trial-and-error experimenting is usually necessary to obtain the best results."(3)

Frost and Lopiccolo have written a very comprehensive article on the "Design of Low Temperature Test Cabinets." In their discussion of evaporators they state: "Finned coil blast type evaporators are most common in the low temperature field.

"Dry expansion evaporators feed refrigerant in at the top of a coil from where it flows down hill to the suction manifold. Distributors are necessary on the valve, and the valve requires sensitive regulation. However oil problems are eliminated.

"Flooded coil systems eliminate expansion valve troubles since uniform distribution can be obtained by the hydrostatic principle. There is the ever present possibility of oil becoming trapped in the evaporator unless overcome by design.

"The use of cold plates has been carefully considered and although certain features are advantageous, some objections bear further study. Cold plate shelves carry away heat by conduction. Should a piece of equipment require no loss of heat to surrounding objects, other than all to the air (this being a duplication of actual field performance), then cold plates supporting such objects would give false test data."(12)

Mr. Soling states that for test rooms having considerable mass in which a slight variation of temperature of the air off the coil is not critical, thermal valves and direct expansion coils can be used satisfactorily with Freon-12 and Freon-22 for room temperatures as low as -70° F. He states that at temperatures below -70° F., it is not possible to allow the necessary refrigerant pressure drop through the direct expansion coils. It is therefore essential on such applications to use a pumped recirculation system for the refrigerant.(29)

D - Defrosting

Finned coil evaporators can be kept clear of frost accumulation by several different methods. The most common

methods used with low-temperature equipment are electric defrost, hot-gas defrost, and the glycol spray system.

The electric defrost employs the use of electric heating coils which are inserted in copper tubing and spaced between and in contact with each row of finned tubing. The heaters can be operated by hand or made automatic by the use of a timing device.(23)

The hot-gas defrost employs the use of hot gas from the discharge side of a compressor for defrosting the evaporator. A simple system can be operated with manual control valves. An automatic system can be controlled by a timing device and automatic valves.(24)

When either the hot gas or the electric heater method of defrosting is used, the evaporator ceases to operate for cooling purposes during the defrost period. This is a definite disadvantage to most test installations. The disadvantage can be eliminated by installing two evaporators in the test room and defrosting them alternately.

The system that seems to have found more general acceptance is the glycol spray. During the operating cycle of the evaporator, the tubes are sprayed with glycol. The solution absorbs the condensate as it collects on the tube surface. In reality, a glycol spray system prevents the formation of frost

instead of removing it after it accumulates. The glycol system is made to operate continuously by circulating part of the solution to a regenerating unit located outside the test room. The moisture is boiled off and the glycol is returned to the sump at the evaporator for recirculation. The regenerating unit maintains the glycol in constant concentration. This being a continuous process, inoperative test periods are eliminated.(22) (17)

E - Staging

Mr. Sulsbrook states: "Physically, there are two methods of carrying out this operation in common use, the cascade and multiple-compression systems. In a cascade system, the first or low-pressure compressor discharges into a heat exchanger from which the second compressor takes its refrigerant to be compressed into another heat exchanger, which in turn may be taken up by a third compressor. This system offers the advantage that refrigerant circuits are comparatively simple, each assembly having its own refrigerant, valves, and control. When desirable, two or perhaps three different refrigerants may be used. The disadvantages lie in the fact that there must be a definite temperature difference in each heat exchanger to transfer the heat from one stage to the next. The second disadvantage is in the fact that, in the lowest stage especially, the total pressure available from the condenser may be so low as to make it difficult to control expansion valves or other

pressure-restricting devices with any degree of accuracy. For example, in a three-stage system there might be the circumstance of a suction pressure of, say, 28 inches vacuum and a head pressure of 6 inches vacuum; the total differential available will not give satisfactory refrigerant flow through standard expansion valves.

"By the multiple-compression method the first compressor discharges into the second and the second into the third, thus maintaining the lowest possible compression ratio per stage. Here the condensing pressure is high enough to actuate ordinary expansion valves. The chief disadvantage is that the refrigerant fed to the second and successive stages is highly superheated, causing compressor overheating. Moreover, the refrigerant fed to the low temperature evaporator comes from a comparatively warm condenser, and thus generates a very high percentage of flash gas at the valve."(3) As a means of overcoming both of these disadvantages, intercooling equipment must be used.(29)

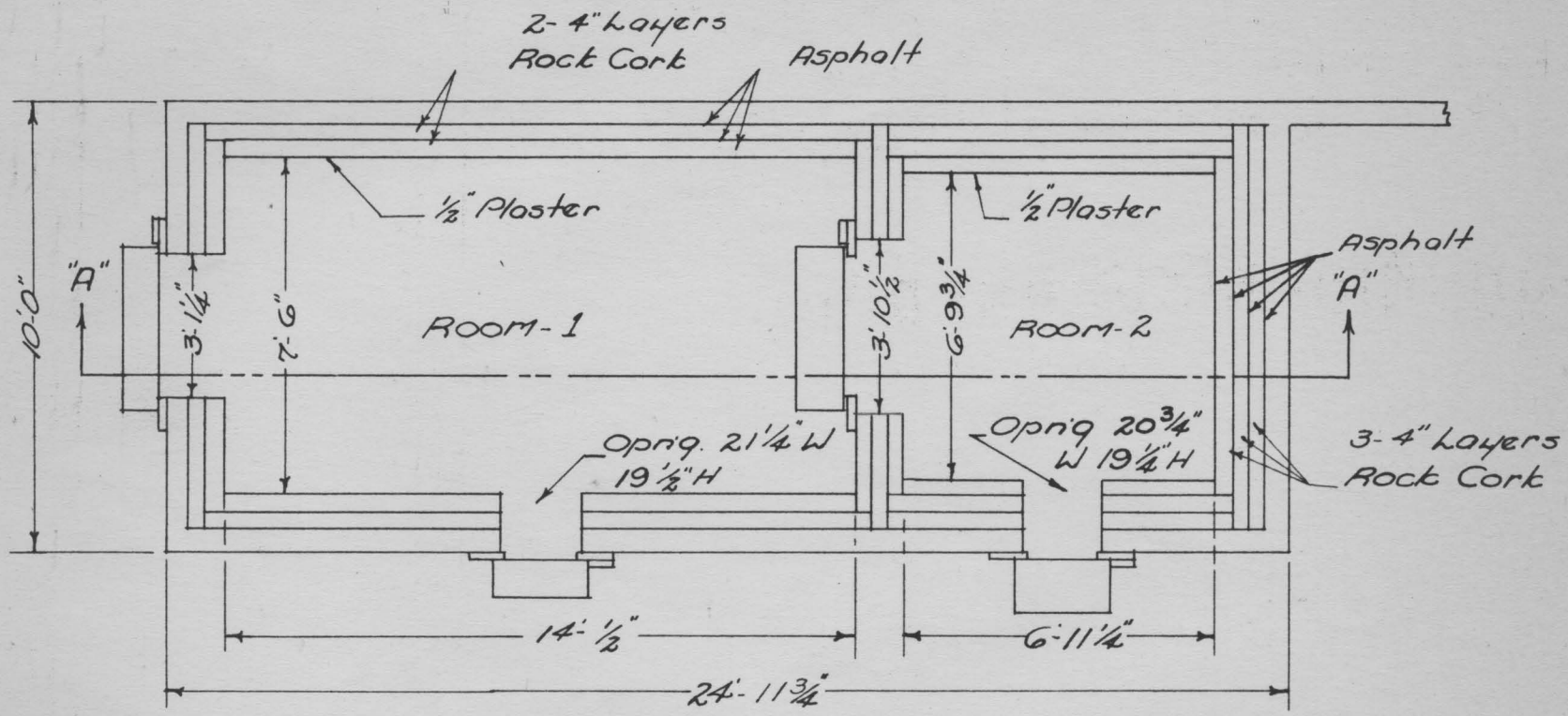
Mr. Aulsebrook further states: "Using Freon-12, it is possible to attain fairly satisfactory efficiencies on two-stage units at evaporator temperatures as low as -75° F., at which point the compression ratio per stage is approximately 8 to 1. It is advisable to resort to the third stage if the temperatures must be carried lower. By substituting Freon-22, the absolute pressure in the evaporator will be increased by about 2 lb. per sq. in."(3)

Generally, staging limits are based on the manufacturer's recommended limits of compression ratio and discharge temperature for a compressor. Mr. Miller states that the manufacturer's maximum recommended compression ratios usually are from 9 to 10 and the maximum recommended discharge temperatures are approximately 300° F.(21) It must be noted that the selection of intermediate stage pressures on the basis of manufacturer's limits does not necessarily provide the most efficient installation.

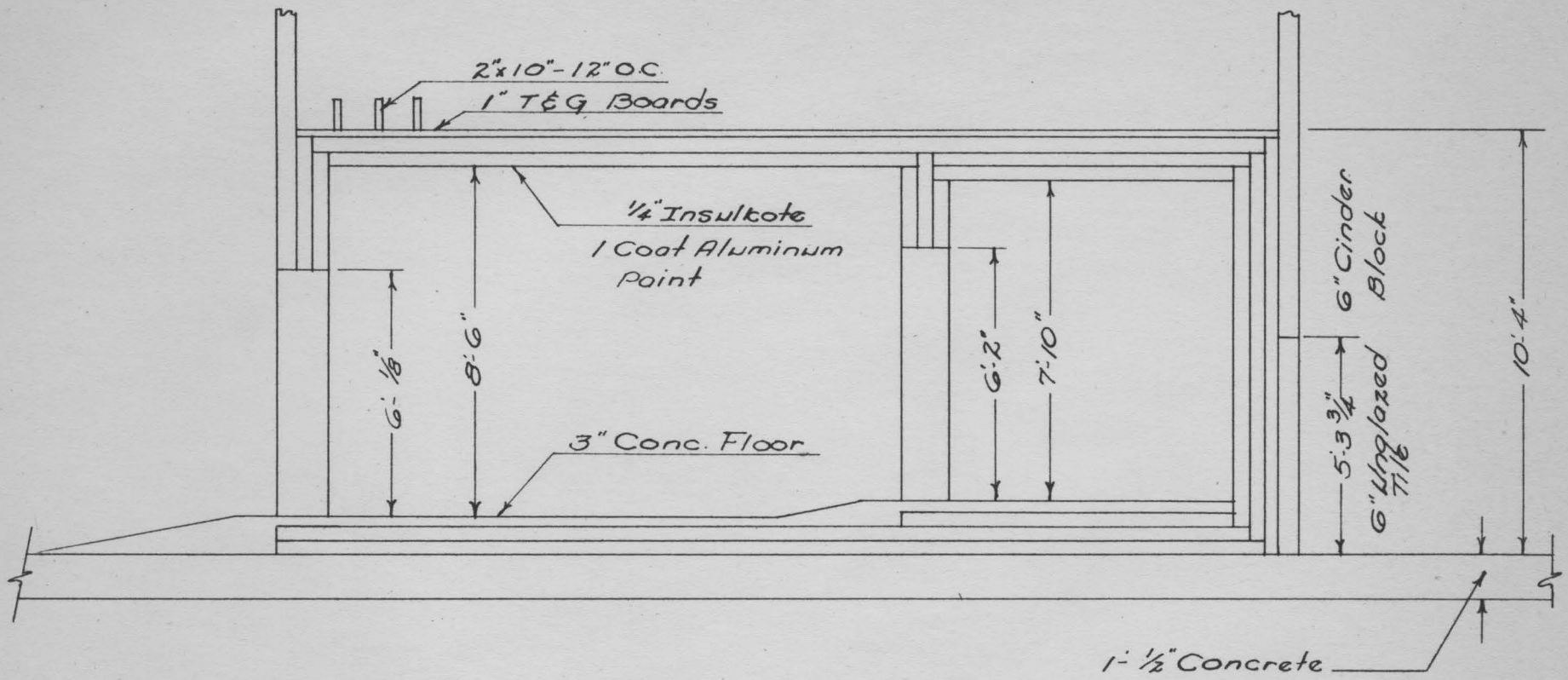
III

LOAD CALCULATIONS

A - Test Room Plans



VIRGINIA POLYTECHNIC INSTITUTE MECHANICAL ENGINEERING DEPARTMENT	
LOW TEMPERATURE TESTING ROOMS	
Drawn by: E.L.B.	Date: March 12, 1954
Checked by:	Scale: ¼" = 1'-0"
Approved by:	Drawing No. 1



VIRGINIA POLYTECHNIC INSTITUTE
MECHANICAL ENGINEERING DEPARTMENT

SECTION A-A

Drawn by E.L.B.	Date: March 12, 1954
Checked by:	Scale: 1/4" = 1'-0"
Approved by:	Drawing No. 2

B - Wall Heat Gain

With a test being conducted in Room number 2, the design temperatures for the installations were given as follows:

Room number 1:

Dry-bulb temperature = 0° F.

Room number 2:

Dry-bulb temperature = -40° F.

With a test being conducted in Room number 1, the design temperatures for the installations were given as follows:

Room number 1:

Dry-bulb temperature = 0° F.

Room number 2:

Dry-bulb temperature = 0° F.

Air surrounding test rooms:

Dry-bulb temperature = 100° F.

Wet-bulb temperature = 80° F.

The values for surface conductance, total conductance and material conductivity, with the exception of the values of material conductivity for Rock Cork, will be taken from those published by the American Society of Heating and Ventilating Engineers.(13) The conductivity factors for Rock Cork were obtained by a special letter.(32) Since it is anticipated that forced convection type evaporators will be used in the installation, values for the surface conductance

will be chosen with an air velocity of 5 miles per hour (m. p. h.) inside the test rooms and 0 miles per hour outside the test rooms.

Sample calculations for heat gain through floor of Room number 2:

Equation 2 $$U = \frac{1}{\frac{1}{f_i} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{1}{f_o}}$$

f_i for air velocity of 5 m. p. h. = 3.5

f_o for air velocity of 0 m. p. h. = 2

k_1 for concrete = 12

k_2 for Rock Cork = .305

k_3 for concrete = 12

x_1 = 3

x_2 = 12

x_3 = 12.5

Substituting values in Equation 2:

$$U = \frac{1}{\frac{1}{3.5} + \frac{3}{12} + \frac{12}{.305} + \frac{12.5}{12} + \frac{1}{2}}$$

$$U = .0241$$

Equation 1 $$Q = UA\,(\Delta t)$$

$U = .0241$ $\Delta t = 140$

$A = 68.89$

$$Q = .0241 \times 68.89 \times 140$$

$$Q = 232.43 \text{ BTU/hr.}$$

Table 1 - Wall Heat Gain for Room No. 1
with Test Being Conducted in Room No. 2

Section	Average Area Sq. Ft.	U $\frac{\text{BTU}}{(\text{hr})(^{\circ}\text{F})(\text{sq.ft.})}$	Δt °F.	Q BTU/hr
Floor	131.92	.0364	100	480.19
Tile Wall	173.37	.0360	100	624.13
Cinder Block Wall	171.79	.0365	100	627.03
Ceiling	131.92	.0364	100	480.19
Partition Wall	34.66	.0230	-40	-31.89
Partition Door	22.35	.0222	-40	-19.85
Part. door frame 1 3/4" pine	1.16	.4138	-40	-19.20
Part. door frame 3 3/4" pine	.38	.2264	-40	- 3.44
Outside Door	18.66	.0359	100	66.99
Inspection Door	2.88	.0359	100	10.34
Total				2214.49

Table 2 - Wall Heat Gain for Room No. 2
with Test Being Conducted in Room No. 2

Section	Average Area Sq. Ft.	U $\frac{\text{BTU}}{(\text{hr})(^{\circ}\text{F.})(\text{sq.ft.})}$	Δt $^{\circ}$F.	Q BTU/hr
Floor	68.89	.0241	140	232.43
Tile Wall	114.33	.0239	140	382.55
Cinder Block Wall	109.13	.0242	140	369.73
Ceiling	68.89	.0241	140	232.43
Partition Wall	34.66	.0230	40	31.89
Partition Door	22.35	.0222	40	19.85
Part. door frame 1 3/4" pine	1.16	.4138	40	19.20
Part. door frame 3 3/4" pine	.38	.2264	40	3.44
Inspection Door	2.77	.0239	140	9.27
Total				1300.79

Table 3 - Wall Heat Gain for Room No. 1
with Test Being Conducted in Room No. 1

Section	Average Area Sq. Ft.	U $\frac{\text{BTU}}{(\text{hr})(^{\circ}\text{F.})(\text{sq.ft.})}$	Δt $^{\circ}$F.	Q BTU/hr
Floor	131.92	.0364	100	480.19
Tile Wall	173.37	.0360	100	624.13
Cinder Block Wall	171.79	.0365	100	627.03
Ceiling	131.92	.0364	100	480.19
Outside Door	18.66	.0359	100	66.99
Inspection Door	2.88	.0359	100	10.34
Total				2288.87

Table 4 - Wall Heat Gain for Room No. 2
with Test Being Conducted in Room No. 1

Section	Average Area Sq. Ft.	U $\frac{\text{BTU}}{(\text{hr})(^{\circ}\text{F.})(\text{sq.ft.})}$	Δt $^{\circ}$F.	Q BTU/hr.
Floor	68.89	.0241	100	166.02
Tile Wall	114.33	.0239	100	273.25
Cinder Block Wall	109.13	.0242	100	264.10
Ceiling	68.89	.0241	100	166.02
Inspection Door	2.77	.0239	100	6.62
Total				876.01

C - Air Changes

1 - General

The design conditions are as follows:

Barometric pressure = 28.1 inches Hg.

Air surrounding test rooms:

Dry-bulb temperature = 100° F.

Wet-bulb temperature = 80° F.

Room number 1

Dry-bulb temperature = 0° F.

Assumed dew point = -10° F.

Room number 2

Dry-bulb temperature = -40° F.

Assumed dew point = -50° F.

The moisture condensed and the latent and sensible heat removed per pound (lb.) of dry air from a mixture of air and water vapor at 100° F. wet-bulb temperature, when it is cooled to -40° F. dry-bulb temperature and -50° F. dew point temperature, can be calculated as follows:

The weight of water vapor in the 100° F. dry-bulb mixture can be calculated by use of equations 4 and 5.

$$5 \quad e = e' - \frac{(B - e')(t - t')}{2{,}800 - 1.3t'}$$

$e' = 1.0321$ inches Hg.[6]

$$e = 1.0321 - \frac{(28.1 - 1.0321)(100 - 80)}{2{,}800 - 1.3(80)}$$

$$e = 1.0120 \text{ inches Hg.}$$

From the steam tables the dew point was found to be 79.4° F.

$$\underline{4} \quad w = \frac{4353e}{B - e}$$

$$w = \frac{4353 \times 1.0120}{28.1 - 1.0120}$$

$$w = 162.62 \text{ grains/lb. of dry air}$$

$$w = \frac{162.62}{7000}$$

$$w = .023231 \text{ lbs./lb. of dry air}$$

At -40° F. dry-bulb temperature and a dew point temperature of -50° F., the water vapor pressure (e) was found to be .0031126 inches Hg. (35)

The water vapor content per pound of dry air was calculated to be .000069 lbs.

The weight of water vapor condensed and frozen when the original mixture is cooled to -40° F. dry-bulb temperature and a dew point of -50° F.

$$= .023231 - .000069$$

$$= .023162 \text{ lbs./lb. of dry air}$$

All condensation will be assumed to occur at the dew point.

The sensible heat of the vapor between 100° F. and 79.4° F. based on a specific heat of 0.45

= 0.45 x 20.6 x .023231

= .2154 BTU/lb. dry air

The latent heat of the vapor condensed at the dew point

= .023162 x 1049

= 24.2969 BTU/lb. dry air

The sensible heat of the condensate between 79.4° F. and 32° F.

= .023162 x 47.4

= 1.0979 BTU/lb. dry air

The latent heat of fusion of water frozen

= .023162 x 144

= 3.3353 BTU/lb. dry air

The sensible heat of ice and vapor between 32° F. and -40° F. based on 0.5 specific heat for each

= 0.5 x 72 x .023231

= .8363 BTU/lb. dry air

The sensible heat of the air

= 0.24 x 140 x 1

= 33.6000 BTU/lb. dry air

A summation of the sensible and latent heat removed gives the following totals:

Total sensible heat removed

= 35.7496 BTU/lb. dry air

Total latent heat removed

= 27.6322 BTU/lb. dry air

The moisture condensed and the latent and sensible heat removed per pound of dry air from a mixture of air and water vapor at 100° F. dry-bulb temperature and 80° F. wet-bulb temperature, when it is cooled to 0° F. dry-bulb temperature and -10° F. dew point temperature, was calculated to be:

Water vapor condensed and frozen

= .022620 lbs./lb. dry air

Total sensible heat removed

= 25.6593 BTU/lb. dry air

Total latent heat removed

= 26.9857 BTU/lb. dry air

The moisture condensed and the latent and sensible heat removed per pound of dry air from a mixture of air and water vapor at 0° F. dry-bulb temperature and -10° F. dew point temperature, when it is cooled to -40° F. dry-bulb temperature and -50° F. dew point temperature, was calculated to be:

Water vapor condensed and frozen

= .000542 lbs./lb. dry air

Total sensible heat removed

= 10.0903 BTU/lb. dry air

Total latent heat removed

= .6465 BTU/lb. dry air

2 - Air Change Load Due to the Test Engine

The design conditions were that a five horsepower test engine would be tested in each of the test rooms. The tests would not be simultaneous. The fuel consumption would be .5 lb. per horsepower hour. The air fuel ratio would be 25 lbs. of air per lb. of fuel.

The hourly fuel consumption rate

= .5 x 5 = 2.5 lb./hr.

The hourly air consumption rate

= 2.5 x 25 = 62.5 lbs./hr.

The sensible and latent heat and the moisture to be removed from the outside air to be supplied the test engine when it is operated in Room number 1 will be:

Sensible heat removed

= 62.5 x 25.6593

= 1603.71 BTU/hr.

Latent heat removed

= 62.5 x 26.9857

= 1686.61 BTU/hr.

Moisture removed

= 62.5 x .022620

= 1.4138 lb./hr.

The sensible and latent heat and the moisture to be removed from the outside air that is supplied the test engine when it is operated in Room number 2 will be:

Sensible heat removed

$$= 62.5 \times 35.7496$$

$$= 2,234.35 \text{ BTU/hr.}$$

Latent heat removed

$$= 62.5 \times 27.6322$$

$$= 1727.01 \text{ BTU/hr.}$$

Moisture removed

$$= 62.5 \times .023162$$

$$= 1.4476 \text{ lbs/hr.}$$

3 - Air Change Load Due to Infiltration and Door Opening

The design conditions for the door openings of the test rooms were given to be the same as that of long storage rooms.

Air changes per 24 hours due to infiltration and door openings for long storage rooms were taken from Mr. Segal's published table[26] and converted to cubic feet per hour as follows:

For Room number 1:

$$\frac{18.811 \times .6}{24} \times 895.11$$

$$= 421.06 \text{ cubic ft./hr.}$$

For Room number 2:

$$\frac{30.989 \times .6 \times 370.22}{24} = 286.81 \text{ cubic ft./hr.}$$

When the test is conducted in Room number 1, the only load due to infiltration and door openings will be the 421.06 cubic ft./hr. of 100° F. air that will enter the outside door. When the test is conducted in Room number 2, there will be a different condition. It is necessary to pass through Room number 1 in order to enter Room number 2. Under this condition, Room number 1 will be subjected to the same load of 100° F. air as when the test was being conducted in it. There will also be an interchange of 286.81 cubic ft./hr. of air between Rooms 1 and 2. The 286.81 cubic ft./hr. of 0° F. air will add an additional load to Room number 2, but the same quantity of -40° F. air will tend to decrease the total load on Room number 1.

The weight of dry air in one cubic foot of mixture can be calculated from the equation:

$$PV = WRT$$

where: P = absolute pressure of the gas

V = its volume

W = its weight

T = its absolute temperature

R = a constant for the particular gas

Transposing the equation:

$$W = \frac{PV}{RT}$$

The weight of dry air in one cubic foot of the 100° F. mixture can be calculated as follows:

1 inch mercury = 0.492 lbs./sq. inch

e = 1.0120 inches Hg.

B = 28.1 inches Hg.

W = weight of dry air in lbs./cu. ft. of mixture

The pressure exerted by the dry air in the mixture is:

28.1 - 1.0120 = 27.088 inches Hg.

$$W = \frac{144 \times 0.492 \times 27.088 \times 1}{53.35(459.7 + 100)}$$

$$W = .06427 \frac{\text{lb. dry air}}{\text{cu. ft. mixture}}$$

The weight of dry air in one cubic foot of 0° F. mixture that moves from Room number 1 to Room number 2 must also be determined.

At a dry-bulb temperature of 0° F. and a dew point temperature of -10° F., the water vapor pressure (e) was found to be .027680 inches Hg.(35)

Pressure of dry air:

= 28.1 - .02768

= 28.07232

$$W = \frac{144 \times 0.492 \times 28.07232 \times 1}{53.35(459.7 + 0)}$$

$$W = .08110 \ \frac{\text{lb. dry air}}{\text{cu. ft. mixture}}$$

When a test is being conducted in Room number 1, the heat and moisture load will be as follows:

Total weight of dry air entering room

$$= 421.06 \times .06427$$

$$= 27.062 \ \frac{\text{lb. dry air}}{\text{hr.}}$$

Sensible heat

$$= 27.062 \times 25.6593$$

$$= 694.39 \text{ BTU/hr.}$$

Latent heat

$$= 27.062 \times 26.9857$$

$$= 730.29 \text{ BTU/hr.}$$

Moisture

$$= 27.062 \times .022620$$

$$= .6121 \text{ lbs./hr.}$$

When a test is being conducted in Room number 2, the load due to outside air entering Room number 1 by infiltration and door openings will be as follows:

Sensible heat

$$= 694.39 \text{ BTU/hr.}$$

Latent heat

$$= 730.29 \text{ BTU/hr.}$$

Moisture

= .6121 lbs./hr.

The load of Room number 2 is increased by the amount of heat that is removed in cooling 286.81 cubic feet of air per hour from 0° F. dry-bulb temperature and -10° F. dew point temperature to -40° F. dry-bulb temperature and a dew point of -50° F. The weight of air that infiltrates from Room number 1 to Room number 2 per hour will be:

$$286.81 \times .08110 = 23.260 \ \frac{\text{lb. dry air}}{\text{hr.}}$$

Sensible heat

= 23.260 x 10.0903

= 234.70 BTU/hr.

Latent heat

= 23.260 x .6465

= 15.04 BTU/hr.

Moisture removed

= 23.260 x .000542

= .0126 lb./hr.

The air that moves from Room number 1 to Room number 2 is displaced in Room number 1 by colder air from Room number 2. This tends to decrease the net air change load on Room number 1 by the following amounts, and the net load on Room number 1 becomes as follows:

Sensible heat

= 694.39 - 234.70

= 459.69 BTU/hr.

Latent heat

= 730.29 - 15.04

= 715.25 BTU/hr.

Moisture

= .6121 - .0126

= .5995 lb./hr.

D - Product Load

The test engine will burn 2.5 lbs./hr. of fuel that has a high heat value of 19,500 BTU/lb. A design factor of 40 per cent of the fuel input was given as the quantity of heat that the engine would radiate to the test room. One person will be present in the test room during a test. There will be 100 watts of lighting for each test room.

Sensible heat radiated to the test room by the engine

= 2.5 x 19,500 x .40

= 19,500 BTU/hr.

Sensible heat gain due to lights

= 100 x 3.415

= 341.5 BTU/hr.

The load due to occupancy will be taken as 1300 BTU/hr.[26] for Room number 1 and 1380 BTU/hr.[4] for Room

number 2. The occupancy load was given by the reference in total heat only, therefore it will be entered in the load summation as sensible heat.

E - Tabulation of Load

Table 5 - Load with test conducted in Room No. 1
and Room No. 2 maintained at 0° F. with no internal load

Room Number 1

Source of Load	Sensible heat BTU/hr.	Latent heat BTU/hr.	Moisture lb./hr.
Wall heat gain	2,288.87		
Air for combustion in test engine	1,603.71	1,686.61	1.4138
Heat radiated to room by test engine	19,500.00		
Lights	341.5		
Occupant	1,300		
Infiltration & door opening	694.39	730.29	.6121
Total	25,728.47	2,416.90	2.0259
Total Heat	28,145.37 BTU/hr.		—

Room Number 2

Source of Load	Sensible heat BTU/hr.	Latent heat BTU/hr.	Moisture lb./hr.
Wall heat gain	876.01		
Total Heat	876.01 BTU/hr.		—

Table 6 - Load with test conducted in Room No. 2 and Room No. 1 maintained at 0° F. with no internal load

Room Number 2

Source of Load	Sensible heat BTU/hr.	Latent heat BTU/hr.	Moisture lb./hr.
Wall heat gain	1,300.79		
Air for combustion in test engine	2,234.35	1,727.01	1.4476
Heat radiated to room by test engine	19,500.00		
Lights	341.5		
Occupant	1,380.0		
Infiltration & door opening	234.70	15.04	.0126
Total	24,991.34	1,742.05	1.4602
Total Heat	26,733.39 BTU/hr.		—

Room Number 1

Source of Load	Sensible heat BTU/hr.	Latent heat BTU/hr.	Moisture lb./hr.
Wall heat gain	2,214.49		
Infiltration & door opening	459.69	715.25	.5995
Total	2,674.18	715.25	.5995
Total Heat	3,389.43 BTU/hr.		—

IV

SELECTION OF EQUIPMENT

A - General

Since the primary idea in the design of this equipment is its installation in the Mechanical Laboratory, standard equipment will be used whenever possible. The manufacturer's recommendations in the equipment selection will be followed.

It was given as a design condition that both test rooms could be maintained at 0° F. with only the high stage compressor in operation. This situation dictates that the intermediate pressure be approximately the same as that of the evaporator in Room number 1. As a design condition, the temperature of water available for condensing purposes was given to be 82° F. Freon-12 will be used as the refrigerant.

The air supplied the test rooms for combustion will be precooled in order to reduce the formation of frost on the evaporator. The precooler will be designed to reduce the entering air to a dry-bulb and a dew point temperature of -5° F. This will reduce the calculated test load of each room by 3,161.98 BTU/hr. total heat. The moisture reduction will be 1.4031 lbs/hr.

Ten per cent will be added to the heat loads to allow for heat gains in piping and as a safety factor in the selection of equipment. This was recommended by the Frick Company.(28)

B - Evaporator Selection

Due to Mr. Aulsebrook's statement that a "tailor-made" evaporator based on considerable trial-and-error experimenting is usually necessary to obtain the best results, a number of manufacturers were contacted for their recommendations.(3) The Frick Company was found to have supplied the equipment for a number of such test installations. They recommended that two Frick Air Cooling Units Model FED 120 be used in Room number 1 and two units Model FED 180 be used in Room number 2. They stated that the units were available in stock sizes and would meet the necessary requirements. They recommended that the evaporator temperature be -15° F. for Room number 1 and -55° F. for Room number 2. They stated that the units would operate 12 hours before their efficiency would become sufficiently impaired as to require defrosting.(28) The only information available for publication on the units was included in the specification sheet.(11) The units are electric defrost, forced convection, and have a finned coil. The ratings for a 15° F. temperature difference are 18,000 BTU/hr. for the FED 120 operating at -15° F. refrigerant temperature and 18,900 BTU/hr. for the FED 180 operating at a -55° F. refrigerant temperature. The motor heat for the FED 120 was given as 18,300 BTU/24 hrs. and for the FED 180 it was given as 34,000 BTU/24 hrs.(11)

Checking the published specifications with the loads, we find the following:

Room number 1

Maximum calculated total heat load	28,145.37 BTU/hr.
Load added by fan motors	1,525.00 BTU/hr.
Total	29,670.37 BTU/hr.
Minus total heat removed by precooler	3,161.98 BTU/hr.
Total	26,508.39 BTU/hr.
10% for safety factor & pipe gain	2,650.84 BTU/hr.
Total heat to be removed by evaporators	29,159.23 BTU/hr.

Two FED 120 units would have a maximum rating of 36,000 BTU/hr.

Room number 2

Maximum calculated total heat load	26,733.39 BTU/hr.
Load added by fan motors	2,833.33 BTU/hr.
Total	29,566.72 BTU/hr.
Minus total heat removed by precooler	3,161.98 BTU/hr.
Total	26,404.74 BTU/hr.
10% for safety factor & pipe gain	2,640.47 BTU/hr.
Total heat to be removed by evaporators	29,045.21 BTU/hr.

Two FED 180 units would have a maximum rating of 37,800 BTU/hr.

According to the published ratings of the evaporators, they seem large for the installation. It must be noted at this time that the published ratings do not account for the frost accumulation, dew point temperature, and the low barometric pressure. These factors were considered by the Frick Company Engineers in their selection.(28)

The specified units are selected for the installation.

C - Pipe Sizing and Friction Loss

The Carrier Corporation recommends that an equivalent pressure drop of 2° F. be used as a design factor in the suction lines, liquid lines, and hot gas lines to allow for friction.(27)(6) The 2° F. equivalent pressure drop will be used in the selection of equipment and the pipes will be sized on the final plans by using the pipe sizing charts published by Carrier.(6)

D - Properties of the Refrigerant at Various Points in the System

The properties will be determined with both compressors in operation. All points referred to in the system will be those indicated on Fig. 1, page 55. The generally accepted practice in the industry is to refer to pressures in terms of temperature. When this is done, the pressure will be that of saturation for the temperature stated.(6) Properties of Freon-12 will be taken from tables published by The American Society of Refrigerating Engineers.(8)

The units of the properties will be as follows:

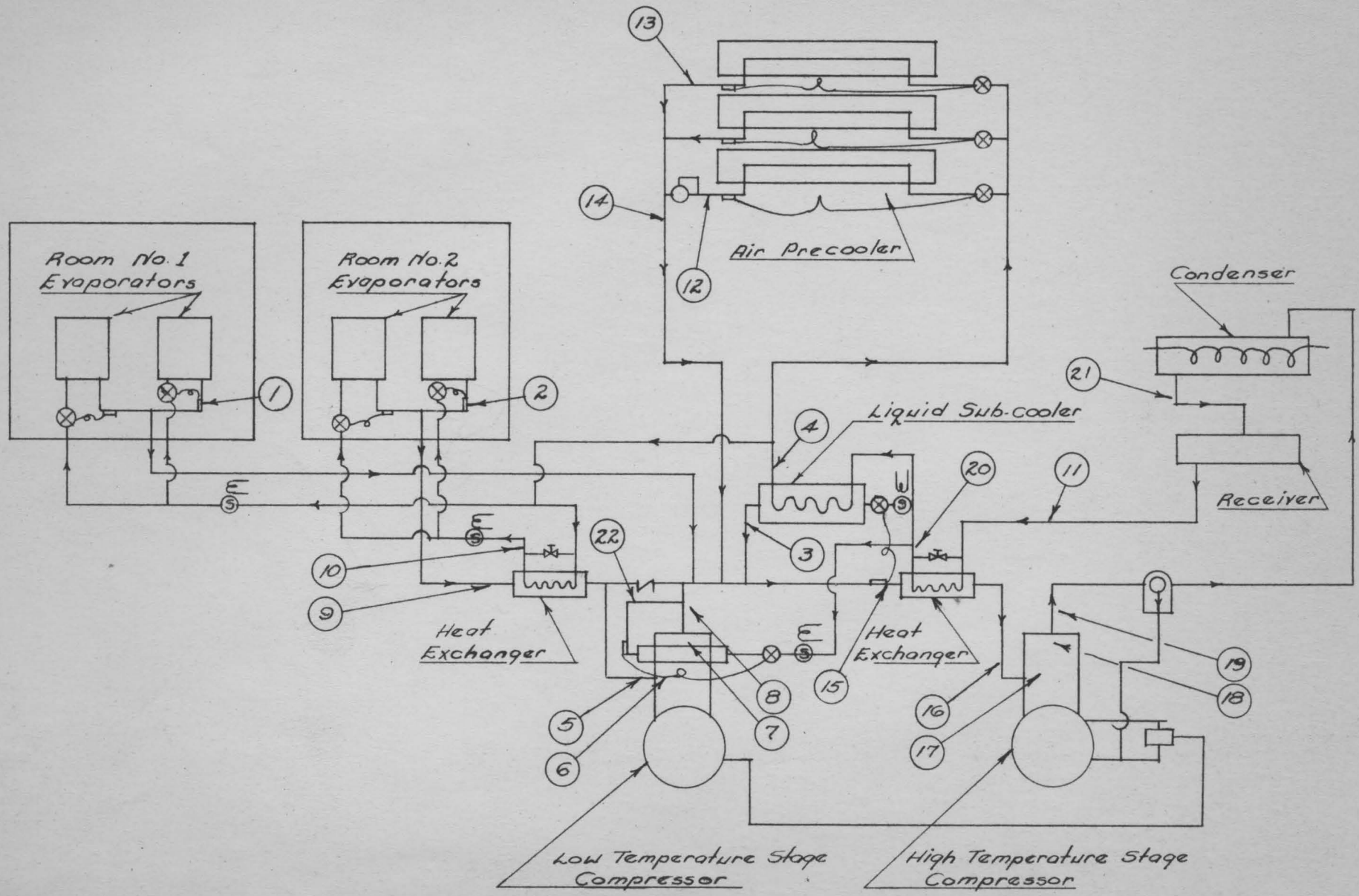
Temperature = (°F.)

Pressure (P) = pounds per square inch absolute (psia)

Enthalpy (h) from -40° F. = BTU/lb.

Entropy (s) from -40° F. = BTU/lb.°F.

Specific volume (v) = cubic feet per pound



-FIGURE 1-

SCHEMATIC LAYOUT OF EQUIPMENT

Point 1:

Pressure = -15° F. = 17.16 psia

Temperature = -5° F.

An allowance of 10° F. superheat will be made for valve operation.

h = 77.92 BTU/lb.

Point 2:

Pressure = -55° F. = 6.20 psia

Temperature = -45° F.

h = 73.01 BTU/lb.

Point 3:

There will be a mixture of liquid and saturated vapor leaving the liquid sub-cooler at this point.

Pressure = -15° F. = 17.16 psia

Temperature = -15° F.

Point 4:

Assume a 10° F. temperature difference between liquid temperature leaving the sub-cooler and the -15° F. intermediate temperature

Temperature = -5° F.

h = 7.20 BTU/lb.

Point 5:

Pressure = -57° F. = 5.868 psia

Temperature = -12° F.

h = 77.38 BTU/lb.

The Frick Company recommends that the vapor be superheated 45° F. at this point to eliminate the entrance of liquid into the compressor.[28] The heat exchanger will be designed to superheat the vapor entering the compressor to -12° F.

Point 6 (entering suction valve):

There is a .5 lb/sq. in. pressure drop through the suction stop valve of the Frick booster compressor and the temperature of the vapor is increased in the valve and manifold to the point where it is superheated 75° F.[28]

Pressure = 5.368 psia

Temperature = 15° F.

v = 8.064 cubic feet per pound

h = 81.06 BTU/lb.

s = .2013 BTU/lb.°F.

Point 7 (leaving discharge valve):

There is a .5 lb/sq. in. pressure drop through the discharge stop valve.

Pressure = 16.89 psia

Compression will be assumed to take place at constant entropy. Therefore

s = .2013 BTU/lb.°F.

h = 90.28 BTU/lb.

Temperature = 82.69° F.

Point 8:

Pressure = -17° F. = 16.39 psia

h = 90.28

Point 9:

h at point 9 = h at point 2

h = 73.01

Approximate Temperature = -45° F.

Approximate Pressure = 6.20 psia

Point 10:

The heat gained by the vapor will be removed from the liquid.

$$h_5 - h_9 = h_4 - h_{10}$$

$$77.38 - 73.01 = 7.20 - h_{10}$$

h_{10} = 2.83 BTU/lb.

Temperature = -26.23°F.

Point 11:

Temperature = 100° F.

h = 31.16 BTU/lb.

Point 12:

Pressure = 30° F. = 43.16 psia

Allow 10° F. superheat for valve operation.

Temperature = 40° F.

h = 83.06 BTU/lb.

Point 13:

Pressure = -15° F. = 17.16 psia

Allow 10° F. superheat for valve operation.

Temperature = -5° F.

h = 77.92 BTU/lb.

Point 14:

For each pound of refrigerant circulated through the air precooler, 65.4% will pass point 12 and 34.6% will pass point 13. The relative quantities were determined from the refrigeration effect of the refrigerant through each section of the precooler and the quantity of heat to be removed.

h = 81.28 BTU/lb.

Pressure = -15° F. = 17.16 psia

Point 15:

Pressure = -17° F. = 16.39 psia

Allow 10° F. superheat for valve operation.

Temperature = -7° F.

h = 77.59 BTU/lb.

Point 16:

Pressure = -17° F. = 16.39 psia

The recommended superheat at this point is 30° F. for the Frick compressor.(28)

Temperature = 13° F.

h = 80.32 BTU/lb.

Point 17 (entering suction valve):

The pressure loss through the suction stop valve of the Frick compressor

= 1 lb/sq. in.[28]

The superheat of the vapor entering the suction valve will be increased to 50° F. by the suction stop valve and the manifold.[28]

Pressure = 15.39 psia

Temperature = 30.3° F.

v = 3.111 cubic feet per pound

h = 82.77 BTU/lb.

s = .18846 BTU/lb.°F.

Point 18 (leaving discharge valve):

The pressure loss through the discharge stop valve

= 1 lb/sq. in.[28]

Pressure = 145.8 psia

Compression will be assumed to take place at constant entropy.

s = .18846 BTU/lb.°F.

Temperature = 189.66°F.

h = 103.18 BTU/lb.

Point 19:

h = 103.18 BTU/lb.

Pressure = 107° F. = 144.8 psia

Point 20:

$h_{16} - h_{15} = h_{11} - h_{20}$

$80.32 - 77.59 = 31.16 - h_{20}$

$h_{20} = 28.43$ BTU/lb.

Temperature = 88.9° F.

Point 21:

Pressure = 140.9 psia

Temperature = 105° F.

h = 32.40 BTU/lb.

Point 22:

Pressure = -17° F. = 16.39 psia

h = 77.59 BTU/lb.

E - Compressor and Motor Selection for Low Temperature Stage

It is obvious that the low temperature stage compressor will receive its greatest balanced load when a test is being conducted in Room number 2. The selection of the low temperature stage equipment will be based on this load condition.

The refrigeration effect of the Freon supplied Room number 2

$= h_2 - h_{10}$

$= 73.01 - 2.83$

$= 70.18$ BTU/lb.

The total maximum heat to be removed from Room number 2 was previously calculated to be 29,045.21 BTU/hr.

The weight of refrigerant required

$$= \frac{29,045.21}{70.18}$$

$$= 413.87 \text{ lb/hr.}$$

The volume of vapor to be compressed in cubic feet per minute (cfm)

$$= \frac{413.87 \times v_6}{60}$$

$$= \frac{413.87 \times 8.064}{60}$$

$$= 55.62 \text{ cfm}$$

Frick booster compressors have an actual volumetric efficiency (e_a) of 81.68 per cent at the desired operating conditions.[28]

Displacement required by Frick compressor

$$= \frac{55.62}{.8168}$$

$$= 68.1 \text{ cfm}$$

The Frick Booster Compressor Model PF 303 operating at a standard speed of 620 revolutions per minute (rpm) has a displacement of 68.70 cfm. This unit will be selected for the installation.

The actual weight of refrigerant circulated through the evaporators in Room number 2 will be:

$$\frac{68.7}{68.1} \times 413.87 = 417.59 \text{ lb/hr.}$$

The clearance of the compressor Model PF 303 is 3 per cent.[28]

The volumetric efficiency based on clearance (e_c) can be calculated from the following equation by Faires:[9]

$$e_c = 1 + c - c\left(\frac{P_7}{P_6}\right)^{\frac{1}{n}}$$

P_7 = discharge pressure (psia)

P_6 = suction pressure (psia)

c = per cent clearance

n = compression exponent

= 1.138 for Freon-12[25]

$$e_c = 1 + .03 - .03\left(\frac{16.89}{5.368}\right)^{\frac{1}{1.138}}$$

$$= .948$$

The indicated horsepower (ihp) of the compressor can be calculated from the following equation derived by Faires:[9]

$$ihp = \frac{144}{33{,}000}\left(\frac{n}{n-1}\right) P_6 V_6 \left[\left(\frac{P_7}{P_6}\right)^{\frac{n-1}{n}} - 1\right] e_c$$

where: 144 = Factor converting lb/sq. in. to lb/sq. ft.

V_6 = Gross compression displacement (cfm)

33,000 = Foot pounds per minute per 1 horsepower

$$ihp = \frac{144}{33{,}000}\left(\frac{1.138}{1.138 - 1}\right)5.368 \times 68.70\left[\left(\frac{16.89}{5.368}\right)^{\frac{1.138 - 1}{1.138}} - 1\right].948$$

$$= 1.860$$

An allowance of reserve power must be made for operation periods at higher suction pressures which result during pull-down periods. For booster compressors where the ihp is below 5 horsepower, multiply ihp by 3 to determine motor size.(28)

$$\text{Motor size} = 3 \times 1.860 = 5.580$$

A standard size motor of $7\frac{1}{2}$ horsepower will be selected.

F - Compressor and Motor Selection for High Temperature Stage

There are two possible balanced load conditions under which the high temperature stage equipment must operate. One load condition exists with a test being conducted in Room number 1 at 0° F. and Room number 2 being maintained at 0° F. with no internal load. The second load condition exists with a test being conducted in Room number 2 at -40° F. and Room number 1 being maintained at 0° F. with no internal load. The selection of equipment for this stage must be based on the load condition that requires the greatest weight of refrigerant to be compressed.

The weight of refrigerant to be compressed when a test is being conducted in Room number 2 will be calculated first.

Weight of refrigerant from Room number 1:

Calculated total heat load Room 1	3,389.43 BTU/hr.
Load added by fan motors	1,525.00 BTU/hr.
Total	4,914.43 BTU/hr.
10% for safety factor & pipe gain	491.44 BTU/hr.
Total	5,405.87 BTU/hr.

Refrigeration effect

$$= h_1 - h_4$$

$$= 77.92 - 7.20$$

$$= 70.72 \text{ BTU/lb.}$$

Weight of refrigerant required

$$= \frac{5,405.87}{70.72}$$

$$= 76.44 \text{ lb/hr.}$$

Weight of refrigerant from air precooler:

Total heat removed from air by precooler	3,161.98 BTU/hr.
10% safety factor & line loss	316.20 BTU/hr.
Total	3,478.18 BTU/hr.

Refrigeration effect

$$= h_{14} - h_4$$

$$= 81.28 - 7.20$$

$$= 74.08 \text{ BTU/lb.}$$

Weight of refrigerant required

$$= \frac{3,478.18}{74.08}$$

$$= 46.95 \text{ lb/hr.}$$

Weight of refrigerant used for cooling jacket of low temperature stage compressor:

The Frick Company allows 5 per cent of tne compressor load for jacket cooling.(28)

Heat to be removed from compressor jacket

$$= .05 \times 29,045.21$$

$$= 1,452.26 \text{ BTU/hr.}$$

Refrigeration effect

$$= h_{22} - h_{20}$$

$$= 77.59 - 28.43$$

$$= 49.16 \text{ BTU/lb.}$$

Weight of refrigerant required

$$= \frac{1,452.26}{49.16}$$

$$= 29.54 \text{ lb./hr.}$$

Weight of refrigerant from low temperature stage compressor:

This weight has been previously calculated to be 417.59 lb./hr.

Weight of refrigerant required for subcooling liquid and desuperheating the vapor between stages:

Liquid to Room number 1	76.44 lb/hr.
Liquid to Room number 2	417.59 lb/hr.
Liquid to air precooler	46.95 lb/hr.
Total weight of liquid to be subcooled	540.98 lb/hr.

Heat removed from liquid

$$= 540.98 (h_{20} - h_4)$$

$$= 540.98 (28.43 - 7.20)$$

$$= 11,468.78 \text{ BTU/hr.}$$

Refrigeration effect required for desuperheating the vapor from low temperature stage

$$= 417.59 (h_g - h_{15})$$

$$= 417.59 (90.28 - 77.59)$$

$$= 5,299.22 \text{ BTU/hr.}$$

Refrigeration effect required for desuperheating the vapor from Room number 1

$$= 76.44 (h_1 - h_{15})$$

$$= 76.44 (77.92 - 77.59)$$

$$= 25.23 \text{ BTU/hr.}$$

Refrigeration effect required for desuperheating the vapor from the air precooler

$$= 46.95 (h_{14} - h_{15})$$

$$= 46.95 (81.28 - 77.59)$$

$$= 173.25 \text{ BTU/hr.}$$

Heat removed from liquid	11,468.78 BTU/hr.
Desuperheating vapor from low temp. stage	5,299.22 BTU/hr.
Desuperheating vapor from Room 1	25.23 BTU/hr.
Desuperheating vapor from air precooler ..	173.25 BTU/hr.
Total refrigeration effect required	16,966.48 BTU/hr.

Refrigeration effect

$$= h_{15} - h_{20}$$

$$= 77.59 - 28.43$$

$$= 49.16 \text{ BTU/lb.}$$

Total weight of refrigerant required for subcooling liquid and desuperheating the vapor between the stages

$$= \frac{16,966.48}{49.16} = 345.13 \text{ lb/hr.}$$

Summarizing, we get the total refrigerant weight passing through the compressor.

Refrigerant from low temperature stage	417.59 lb/hr.
Refrigerant from Room number 1	76.44 lb/hr.
Refrigerant from air precooler	46.95 lb/hr.
Refrigerant from jacket of low temp. compressor	29.54 lb/hr.
Refrigerant from liquid sub-cooler	345.13 lb/hr.
Total	915.65 lb/hr.

The weight of refrigerant that will be compressed when a test is being conducted in Room number 1 will now be calculated.

Weight of refrigerant from Room number 1:

The total heat to be removed from Room 1 was previously calculated to be 29,159.23 BTU/hr.

Refrigeration effect

$$= h_1 - h_4$$

$$= 77.92 - 7.20$$

$$= 70.72 \text{ BTU/lb.}$$

Weight of refrigerant required

$$= \frac{29,159.23}{70.72}$$

$$= 412.32 \text{ lb/hr.}$$

Weight of refrigerant from Room number 2:

Calculated total heat Room 2	876.01 BTU/hr.
Load added by fan motors	2,833.33 BTU/hr.
Total	3,709.34 BTU/hr.
10% safety factor & pipe loss	370.93 BTU/hr.
Total	4,080.27 BTU/hr.

With the low temperature stage compressor not operating the refrigeration effect for Room number 2 becomes equal to that of Room number 1 or 70.72 BTU/lb.

Weight of refrigerant required

$$= \frac{4,080.27}{70.72}$$

$$= 57.70 \text{ lb/hr.}$$

The weight of refrigerant required for air precooler will remain the same as previously calculated or

$$= 46.95 \text{ lb/hr.}$$

The weight of refrigerant required for subcooling the liquid and desuperheating the vapor:

Liquid to Room number 1	412.32 lb/hr.
Liquid to Room number 2	57.70 lb/hr.
Liquid to air precooler	46.95 lb/hr.
Total weight of liquid to be subcooled	516.97 lb/hr.

Heat removed from liquid

$$= 516.97 (28.43 - 7.20)$$

$$= 10,975.27 \text{ BTU/hr.}$$

Total refrigeration effect required for desuperheating vapor was calculated to be

$$= 328.36 \text{ BTU/hr.}$$

The weight of refrigerant required for subcooling the liquid and desuperheating the vapor

$$= \frac{11,303.63}{49.16}$$

$$= 229.94 \text{ lb/hr.}$$

The total weight of refrigerant circulated to the high temperature stage compressor with a test being conducted in Room number 1 was found by summary to be 746.91 lbs/hr.

The greater compressor displacement will be required with a test being conducted in Room number 2. The equipment selection will be based on this load condition.

The volume of vapor to be compressed

$$= \frac{915.65 \times v_{17}}{60}$$

$$= \frac{915.65 \times 3.111}{60}$$

$$= 47.48 \text{ cfm}$$

The Frick "Eclipse" Compressor Model PF 303 has a displacement of 94.20 cfm when operated at 850 revolutions per minute, and an actual volumetric efficiency of 53.5 per cent at the desired operating conditions.(28)

Required compressor displacement

$$\frac{47.48}{.535} = 88.75 \text{ cfm}$$

The Frick "Eclipse" Compressor Model PF 303 will be selected for the installation and operated at the standard speed of 850 rpm.

Actual capacity of compressor

$$= \frac{94.20}{88.75} \times 746.91$$

$$= 792.47 \text{ lb/hr.}$$

$$e_c = 1 + c - c\left(\frac{P_{18}}{P_{17}}\right)^{\frac{1}{n}}$$

$$= 1 + .03 - .03\left(\frac{145.8}{15.39}\right)^{\frac{1}{1.138}}$$

$$= .8137$$

$$ihp = \frac{144}{33,000}\left(\frac{n}{n-1}\right) P_{17}V_{17}\left[\left(\frac{P_{18}}{P_{17}}\right)^{\frac{n-1}{n}} - 1\right] e_c$$

$$= \frac{144}{33,000}\left(\frac{1.138}{1.138-1}\right) 15.39 \times 94.20\left[\left(\frac{145.8}{15.39}\right)^{\frac{1.138-1}{1.138}} - 1\right] .8137$$

$$= 13.303$$

An allowance of reserve power must be made for operation periods at high suction pressures which result during pull-down periods. The general practice is to drive the high-stage compressor with a motor that will carry the compressor at the highest expected evaporator temperature. This is generally the air conditioning rating of the unit.(30) The Frick Company stated that the compressor required a 25 horsepower motor when it was operated in air conditioning service.(28)

A 25 horsepower motor will be selected for the installation.

G - Condenser Selection

The Carrier Corporation has published rating curves for their standard water cooled condensers.(10) The ratings are based on heat rejection in BTU per hour per °F. temperature difference between entering water temperature and condensing temperature for various fouling factors. Recommended fouling factors are given for various water classifications.(10)

Condenser heat rejection load

= (weight of refrigerant to condenser lb/hr.) $(h_{19} - h_{21})$

= 792.47 (103.18 - 32.40)

= 56,091.03 BTU/hr.

The condensing temperature will be 105° F.

The temperature of the available water is 82° F.

Condenser heat rejection capacity required in BTU/hr/°F. temperature difference

$$= \frac{56,091.03}{105 - 82}$$

= 2,438.7 BTU/°F. temperature difference

The recommended fouling factor to be used for cooling tower water with treated make-up is 0.001.(10) From the capacity curves of the Carrier 8 Pass Condenser Number 5F30, the quantity of condenser water required was found to be 6.5 gallons per minute and the pressure drop through the condenser was found to be 1.5 pounds per square inch.

Soling and Rescorla state that it is necessary to oversize the condenser to some extent for low temperature installations that require a period of operation for pull-down.(30) Checking the operation of condenser number 5F30, we find, at 50 per cent over balanced load, that 11 gallons of condenser water per minute will be required and that the pressure drop through the condenser will be 3.5 pounds per square inch. The next smaller size condenser, the Carrier 6 Pass Condenser Number 5F20, when operated at 50 per cent over balanced load, requires 19.5 gallons per minute of water and the pressure drop would be 7 pounds per square inch.

The Carrier water cooled 8 Pass Condenser Number 5F30 will be selected for the installation. By using condenser number 5F30, adequate overload protection will be provided, there will be a decrease in the cost of the water pump and piping, and the cost of pump operation will be decreased. The 5F30 condenser is a shell and coil type. It has 3/4 inch finned tubes, with 15.4 square feet of water side surface and 49.9 square feet of refrigerant side surface.

H - Receiver Selection

The receiver must provide a liquid seal to prevent refrigerant vapor from entering the liquid lines that supply refrigerant to the evaporators. It is also desirable for it to provide sufficient storage space to house the entire charge

during a shut-down period and to store an excess amount of refrigerant over and above the actual amount required for operation.(33) Acme Industries Inc. state that a receiver should be sized to allow for a 20 to 25 per cent gas expansion space as a safety factor.(16)

The operating charge for various pieces of equipment to be used in the installation will be determined from the manufacturers' specifications. The charge for the piping will be determined from the size and length of pipe to be installed. The receiver will be sized to have an excess storage capacity of 25 per cent of the operating charge when it is 80 per cent full of liquid refrigerant.

The operating charge for the refrigerant piping cannot be determined until the actual size and length of pipe is known. The receiver size will be determined when the final plans are drawn and a standard size Acme receiver will be selected. The selected size will be stated in the specifications.

I - Heat Exchanger and Liquid Sub-cooler Selection

The heat exchangers manufactured by Acme Industries Inc. are rated on the basis of per cent efficiency at various rates of flow. If it were possible to build a heat exchanger that would be 100 per cent efficient, the suction gas would be returned to the compressor at the same temperature as that of the entering liquid. The formula for heat exchanger selection

is given as follows:[1]

$$\%\text{Efficiency} = \frac{\text{Leaving gas temp.} - \text{Entering gas temp.}}{\text{Entering liquid temp.} - \text{Entering gas temp.}} \times 100$$

The gas temperatures, liquid temperature, and the weight of refrigerant circulated per hour have been previously determined for both the low temperature stage and the high temperature stage of the system. By substituting the proper values in the above formula, we find the desired efficiency for the low temperature stage heat exchanger to be as follows:

$$\% \text{ Efficiency} = \frac{(-12) - (-45)}{(-5) - (-45)} \times 100$$

$$= 82.5\%$$

The weight of refrigerant circulated through the heat exchanger per hour has been previously calculated to be 417.59 pounds per hour.

$$\text{Weight of refrigerant circulated} = \frac{417.59}{60}$$

$$= 6.96 \text{ lbs/minute}$$

Acme heat exchanger model H-642 has an operating efficiency of 81 per cent when 6 pounds per minute of refrigerant are circulated and the efficiency decreases as the rate of flow increases.[1] It is obvious that this model is too small. The next larger size, Acme model H-840, will be selected for the low temperature stage.

The rate of refrigerant flow and the desired efficiency were determined for the high temperature stage and an Acme model H-615 was selected for the installation.

The Frick Company recommends the use of an Acme heat exchanger for the liquid sub-cooler. From experience they recommended that the size of the heat exchanger be based on a heat transfer coefficient of U = 25 BTU/hr./°F./ft.^{2} (28)

The temperature of the liquid in has been previously determined to be 88.9° F.

The temperature of the liquid out has been previously determined to be -5° F.

The temperature of the vapor in the unit has been previously determined to be -15° F.

The log mean temperature difference was calculated to be 40.1° F.

The heat to be removed from the liquid in the cooler has been previously calculated to be 11,468.78 BTU/hr.

$$Q = UA\ (LMTD)$$

$$11{,}468.78 = 25A\ (40.1)$$

$$A = \frac{11{,}468.78}{25 \times 40.1}$$

$$= 11.45\ ft.^2$$

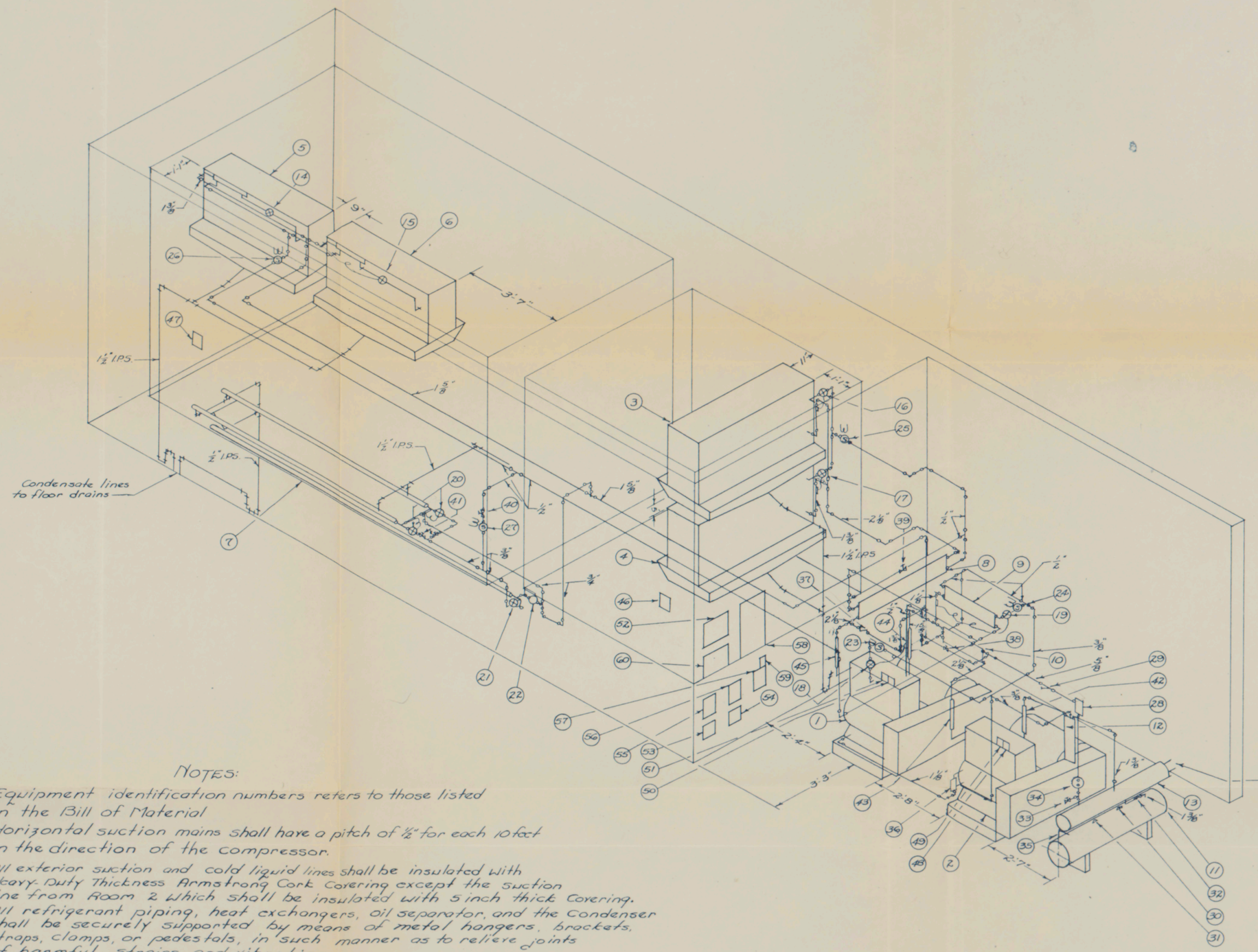
The Acme heat exchanger model H-624 has 13 square feet of surface and it will be selected for the installation.

J - Air Precooler Selection

The Brown Fintube Company manufactures heat transfer equipment for special application. Representatives of the company were given the desired operating conditions for the two stage air precooler, and they submitted a proposed design that would satisfy the requirements. A detailed plan is included as Drawing Number 5. This unit will be selected for the installation.

V

EQUIPMENT PLANS



NOTES:

Equipment identification numbers refers to those listed in the Bill of Material

Horizontal suction mains shall have a pitch of ½" for each 10 foot in the direction of the compressor.

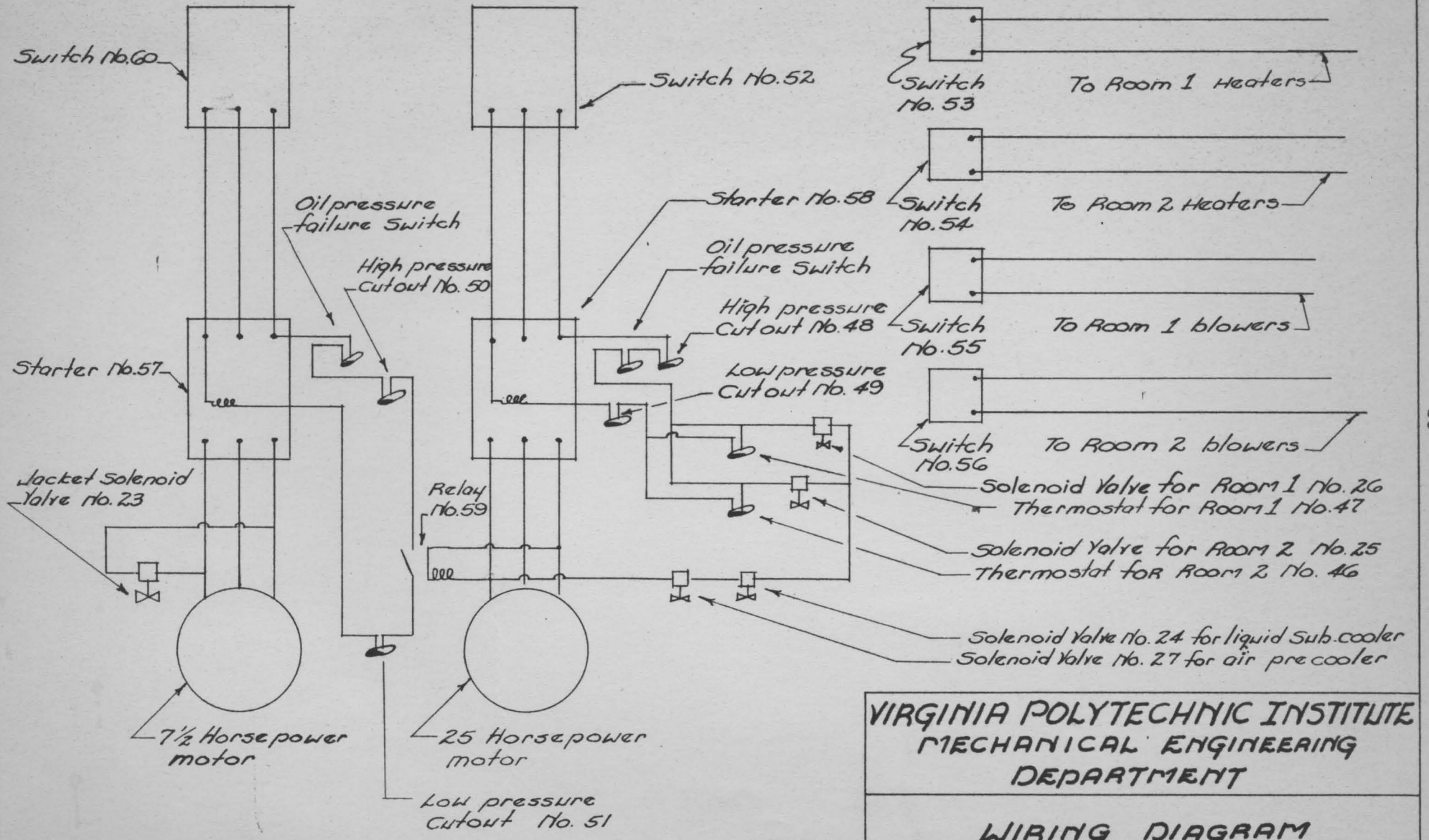
All exterior suction and cold liquid lines shall be insulated with Heavy-Duty Thickness Armstrong Cork Covering except the suction line from Room 2 which shall be insulated with 5 inch thick Covering.

All refrigerant piping, heat exchongers, oil separator, and the Condenser shall be securely supported by means of metal hangers, brackets, straps, clamps, or pedestals, in such manner as to relieve joints of harmful strains and vibration.

Refrigerant and oil return pipe sizes are O.D. and are for Type L copper tubing.

Connect Condensate lines from evaporators 3 and 4 to floor drain in the equipment room.

VIRGINIA POLYTECHNIC INSTITUTE MECHANICAL ENGINEERING DEPARTMENT	
EQUIPMENT PLANS	
Drawn by: E.L.B.	Date: May 9, 1954
Checked by:	Scale: ⅜" = 1'-0"
Approved by:	Drawing No. 3



Equipment Identification numbers refer to those listed in the Bill of Material

VIRGINIA POLYTECHNIC INSTITUTE
MECHANICAL ENGINEERING DEPARTMENT

WIRING DIAGRAM

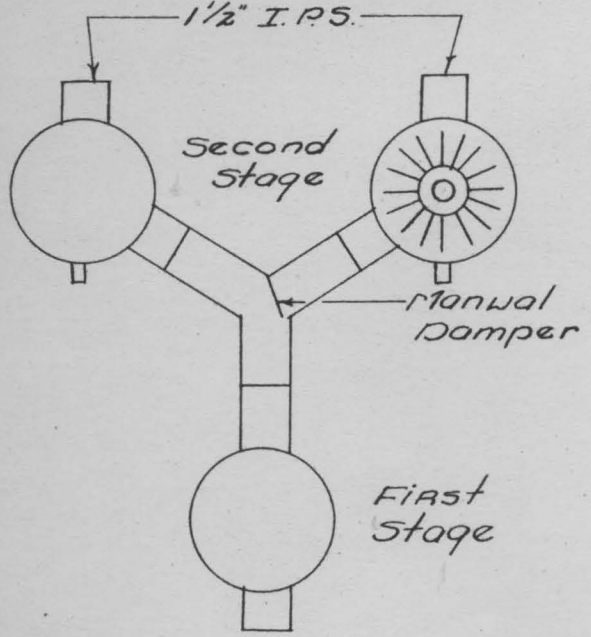
Drawn by: E.L.B.	Date: May 12, 1954
Checked by:	Scale: None
Approved by:	Drawing No.4

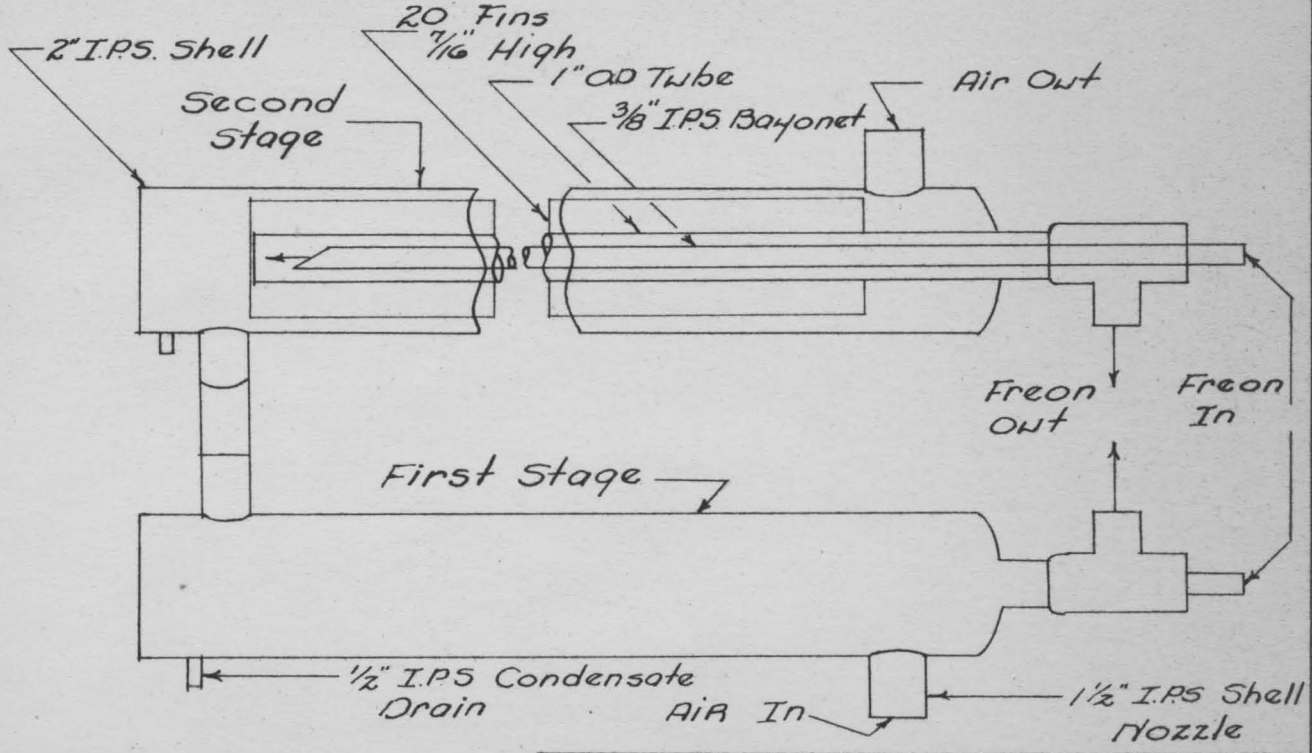
FIRST STAGE

Fin. Length = 11'-0"
Overall Length = 12'-6"
Surface 18.9 Sq. Ft.

SECOND STAGE

Fin. Length = 7'-0"
Overall Length = 8'-6"
Surface Per Section 12 Sq. Ft.
Total Surface 24 Sq. Ft.





BROWN FINTUBE COMPANY	
Air Precooler	
Drawing by:	Date: March 11, 1954
Checked by:	Scale: None
Approved by:	Drawing No. 5

VI

BILL OF MATERIAL

Equipment identification numbers refer to those indicated on Drawing Number 3 and Drawing Number 4.

1. Frick booster compressor model PF 303 with structural steel base; unloader; 7½ horsepower, 220 volt, 3 phase, 60 cycle motor; and a 620 rpm drive.
2. Frick compressor model PF 303 with structural steel base; unloader; 25 horsepower, 220 volt, 3 phase, 60 cycle motor; and a 850 rpm drive.
3. Frick air cooling unit model FED 180.
4. Frick air cooling unit model FED 180.
5. Frick air cooling unit model FED 120.
6. Frick air cooling unit model FED 120.
7. Brown Fintube air precooler.
8. Acme heat exchanger model H-840.
9. Acme heat exchanger model H-624.
10. Acme heat exchanger model H-615.
11. Acme liquid receiver size 10 3/4 inches by 48 inches with saddle brackets.
12. Acme oil separator model F-305 with 1 3/8 inches O. D. gas connections.
13. Carrier condenser number 5F30.

14. Alco expansion valve type TCL100FZ with strainer.
15. Alco expansion valve type TCL100FZ with strainer.
16. Alco expansion valve type ZZ100FX with strainer.
17. Alco expansion valve type ZZ100FX with strainer.
18. Alco expansion valve type TCL50FZ with strainer.
19. Alco expansion valve type TCL200FZ with strainer.
20. Two Alco expansion valves type TCL50FZ with strainers.
21. Alco expansion valve type TCL50FG with strainer.
22. Alco suction line regulator type 760 Evapotrol.
23. Alco solenoid valve number S155.
24. Alco solenoid valve number S120.
25. Alco solenoid valve number S120.
26. Alco solenoid valve number S120.
27. Alco solenoid valve number S120.
28. Henry drier number 7570-B.
29. Henry "Y" strainer number 895E.
30. Henry relief valve number 527.
31. Kerotest packless angle valve number 3971S.
32. Henry packless angle shut-off valve number 6421.
33. Henry packless charging valve number 6233.
34. Henry liquid indicator number LI-94.
35. Acme gauge glass assembly.
36. Aminco float level trap.
37. Henry check valve number 2051.

38. Henry throttle valve number 6295.

39. Henry throttle valve number 6294.

40. Henry line shut-off valve number 6264.

41. Two Henry line shut-off valves number 6263.

42. Techniflex vibration eliminator number F-138.

43. Techniflex vibration eliminator number F-218.

44. Techniflex vibration eliminator number F-158.

45. Techniflex vibration eliminator number F-218.

46. Minneapolis Honeywell temperature controller type L426A range -50° F. to +10° F.

47. Minneapolis Honeywell temperature controller type L426A range -20° F. to +50° F.

48. Minneapolis Honeywell pressure controller type L414B.

49. Minneapolis Honeywell pressure controller type L414A.

50. Minneapolis Honeywell pressure controller type L614A.

51. Minneapolis Honeywell pressure controller type L414A.

52. Cutler-Hammer standard duty safety switch number 4131H344.

53. Cutler-Hammer standard duty safety switch number 4131H342.

54. Cutler-Hammer standard duty safety switch number 4131H342.

55. Cutler-Hammer standard duty safety switch number 4131H341.

56. Cutler-Hammer standard duty safety switch number 4131H341.

57. Cutler-Hammer across-the-line starter number 9586H2492.

58. Cutler-Hammer reduced voltage starter number 9605H792.

59. Square D magnetic relay type JNG-1.

60. Cutler-Hammer standard duty safety switch number 4131H342.

61. Type L Copper Tubing:

(Sizes listed in inches O. D.)

18 ft. 3/8 in.

81 ft. 1/2 in.

14 ft. 5/8 in.

12 ft. 3/4 in.

8 ft. 1 1/8 in.

17 ft. 1 3/8 in.

47 ft. 1 5/8 in.

12 ft. 2 1/8 in.

62. Wrought Copper 90° elbows, solder connection:

(Sizes listed in inches O. D.)

9 - 3/8 in.

23 - 1/2 in.

5 - 5/8 in.

6 - 3/4 in.

4 - 1 1/8 in.

10 - 1 3/8 in.

10 - 1 5/8 in.

7 - 2 1/8 in.

63. Wrought Copper tees, solder connection:

(Sizes listed as inches O. D.)

1 - 3/8" x 3/8" x 3/8"

1 - 3/8" x 3/8" x 1/2"

1 - 1/2" x 3/8" x 1/2"

6 - 1/2" x 1/2" x 1/2"

1 - 1/2" x 1/2" x 5/8"

3 - 5/8" x 5/8" x 5/8"

2 - 3/4" x 3/4" x 3/4"

1 - 1 3/8" x 1 3/8" x 1 5/8"

1 - 1 5/8" x 1 5/8" x 3/4"

1 - 1 5/8" x 1 5/8" x 1 3/8"

1 - 1 5/8" x 1 5/8" x 1/2"

2 - 2 1/8" x 1 5/8" x 5/8"

1 - 2 1/8" x 2 1/8" x 1 1/8"

1 - 2 1/8" x 1 1/8" x 1 3/8"

64. Steel pipe and fittings:

43 ft. $1\frac{1}{2}$ in. pipe

14 ft. $\frac{1}{2}$ in. pipe

4 - $1\frac{1}{2}$ in. 45° elbows

14 - $1\frac{1}{2}$ in. 90° elbows

4 - $1\frac{1}{2}$ in. tees

7 - $\frac{1}{2}$ in. 90° elbows

2 - $\frac{1}{2}$ in. tees

65. Armstrong Cork covering, 5 inches thick:

4 ft. for 2 1/8 in. O. D. pipe

4 covers for 2 1/8 in. O. D. elbows

Lagging for Acme model H840 heat exchanger

66. Armstrong Cork covering heavy duty thickness:

Lagging for Acme heat exchanger model H-624

Lagging for Acme heat exchanger model H-615

2 covers for Alco expansion valves type TCL50FZ

1 cover for Alco solenoid valve number S120

1 cover for Alco expansion valve type TCL50FG.

1 cover for Henry check valve number 2051

2 covers for $\frac{1}{2}$ in. globe valves

2 covers for 3/8 in. globe valves

1 cover for $1\frac{1}{2}$ in. IPS tee

1 cover for $1\frac{1}{2}$ in. IPS 90° elbow

2.5 feet for $1\frac{1}{2}$ in. IPS pipe

67. Cork covering for the following tubing and fittings will be for solder connections and tne sizes will be listed as O. D.:

Tubing

5 ft. - 3/8 in.

55 ft. - 1/2 in.

12 ft. - 3/4 in.

1 ft. - 1 1/8 in.

3 ft. - 1 3/8 in.

31 ft. - 1 5/8 in.

2 ft. - 2 1/8 in.

Wrought copper 90° elbows

3 - 3/8 in.

12 - 1/2 in.

6 - 3/4 in.

1 - 1 1/8 in.

2 - 1 3/8 in.

4 - 1 5/8 in.

2 - 2 1/8 in.

Wrought Copper tees

1 - 3/8" x 3/8" x 3/8"

1 - 3/8" x 3/8" x 1/2"

4 - 1/2" x 1/2" x 1/2"

2 - 3/4" x 3/4" x 3/4"

1 - 1 5/8" x 1 5/8" x 3/4"

1 - 1 5/8" x 1 5/8" x 1/2"

1 - 1 5/8" x 1 5/8" x 1 3/8"

1 - 2 1/8" x 1 5/8" x 1 5/8"

1 - 2 1/8" x 2 1/8" x 1 1/8"

68. 100 lbs. Freon-12.

VII

OPERATING INSTRUCTIONS

The refrigeration equipment has been designed to operate automatically by the use of pressure and temperature controls with three exceptions: the control of the condenser water, the defrosting of the evaporators, and the defrosting of the second stage of the air precooler.

The control of the condenser water can be made automatic by the installation of a pressure controlled automatic water regulating valve. The valve should be installed at the water inlet to the condenser and the pilot tube connected to the condenser hot gas inlet. If the automatic valve is not used, the flow of water must be regulated by the use of a hand valve.

The evaporators must be defrosted after every 12 hours of operation. This can be accomplished by turning off the compressors and the circulation fans and turning on the electric defrost heaters located in each evaporator. When the coils are clear of frost, the heaters are turned off and the circulation fans and the compressors are restarted. If automatic evaporator defrosting is desired, it can be accomplished by the installation of a time switch with the necessary relays.

The second stage of the air precooler must be defrosted periodically while a test is being conducted in either test room.

The second stage is divided into two sections with the refrigerant flow to each being controlled by a hand valve. This makes it possible to defrost one section while the other is being operated. Since an operator will be present when a test is being conducted, it will be more practical to defrost the precooler by hand control.

All pressure controls should be adjusted to maintain the refrigerant properties outlined in Section IV of this thesis. As an aid to the adjustment, liquid by-pass lines with throttle valves have been specified for each heat exchanger. Manufacturers' recommendations as to equipment maintenance, adjustment, and operation should, of course, be followed.

VIII

ACKNOWLEDGMENTS

The author wishes to express his gratitude to the graduate committee comprised of Professor J. B. Jones, Professor J. P. Mahaney, Associate Professor J. L. Jones, and Associate Professor H. S. Miles. Associate Professor J. L. Jones served as chairman of the committee. The guidance, helpful suggestions, and friendly criticism contributed by all members of the graduate committee made the writing of this thesis possible. Thanks are also extended to Associate Professor C. E. Trent and Associate Professor H. L. Wood for their helpful suggestions.

The help and advice of Mr. T. F. Sloan, Engineer, Frick Company, was found invaluable.

IX

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