Chapter – 25 Case of Screw Refrigeration System Design

This shall be a simplified version of the example to illustrate the thinking process how to design a refrigeration system.

Most refrigeration system design starts from very limited information. Discussions and clarifications are very often required between the entity who issues the inquiry and the party who is performing the refrigeration system design. The details of this case example are the result from such communications and clarifications.

The refrigeration system is assumed to be used for a chemical plant. The process used by this chemical plant requires three evaporators at different temperatures. Cooling tower water is used for condenser and compressor oil cooling. Water temperature available is 88°F. Refrigerant preferred is ammonia (R-717); power supply is 6000-3-50 for main motor, 415-3-50 for smaller motors and 220-1-50 for control circuit. The hazardous location classification for the refrigeration system is Ex e, IIA, Zone 2 (or Class I, Group D, Division II).

The refrigeration system is to be designed to handle multiple evaporative loads. All the evaporators are part of the process components of the chemical plant and are remote mounted; the No. 3 evaporator is closer to the engine room. The process loads and the evaporative temperatures requirements are as the following:

Evaporator No. 1:

The process refrigeration load required is 260 TR; evaporative temperature is -40°F.

Evaporator No. 2:

The process refrigeration load required is 30 TR; evaporative temperature is -35°F.

Evaporator No. 3:

This evaporator is a brine cooler which is to cool 300 GPM 30% by wt. Ethylene Glycol brine from 37°F to 30°F. Evaporative temperature required for this heat exchanger is 20°F

The plant operator and the engineers of the user wish to have one central refrigeration system for this installation.

Because no heat load is given for the #3 evaporator, therefore, the refrigeration load for

No. 3 heat exchanger is calculated as the following:

Brine flow: 300 GPM, 30% by wt. Ethylene Glycol brine from 37°F to 30°F

$$\text{Average brine temperature} = \frac{37 + 30}{2} = 33.5°F$$

From ethylene glycol brine property data for
30% by wt. Ethylene Glycol brine at an average temperature of 33.5°F.

Specific Gravity = 1.050
Specific Heat = 0.8712
Brine freezing point: 7.16°F

Brine flow formula is:

$$TR = \frac{GPM}{24} \times S.G. \times C_p \times (T_2 - T_1)$$

Therefore, the refrigeration load for the No.3 heat exchanger is:

$$TR = \frac{300}{24} \times 1.050 \times 0.8712 \times (37 - 30)$$

$$= 80 \text{ TR}$$

For this system, reciprocating compressor is too small and the centrifugal compressor is way too big for the application, particularly the refrigerant is ammonia. Therefore, screw compressor is to be used.

There are three temperature levels, most logical arrangement shall be to use a compound system; the intermediate temperature of the system is to be fixed in such temperature level to accommodate the refrigeration load from evaporator No. 3. Furthermore, the compound system also tends to have better power consumption which might be a concern by the user. The evaporative temperatures of -40°F and -35°F are very close; it can be combined and handled by the low stage (booster) compressor; the 20°F evaporative load is to be connect to intermediate temperature.

No economizing is used. Flash type intermediate intercooler with subcool cooling coil for liquid subcooling is used for various considerations as the following:

(a) The No.1 and No.2 heat exchangers are remotely mounted; subcooled liquid is desirable to avoid flash gas in the liquid which is supplied to these evaporators.

(b) The refrigerant is ammonia, the flow is smaller; therefore, coil type heat exchanger within the flash intercooler instead of separate shell-and-tube liquid subcooler is used, it is less expensive to construct.

(c) Flash intercooling is to deperheat the low stage discharge hot gas to a saturated condition before it enters to the high stage compressor.

The intermediate temperature of the system is to be designed for 15°F to accommodate the #3 heat load at ET of 20°F; this would also allow enough pressure difference for the returning suction gas from No.3 load even with evaporative temperature fluctuation.

Cooling water available is 88°F entering to water cooled condenser. Water cooled condenser is usually designed for water temperature range of 10°F; the leaving cooling water temperature is therefore 98°F. If the heat transfer small temperature difference for allowance for the condenser is 7°F, the design condensing temperature shall be (98 + 7) = 105°F.

P-H Diagram Analysis:

Use the P-H diagram analysis, this compound system now can be started with three base lines as shown in Figure 25-1:

The line (E)-(F) is the lowest evaporative temperature of -40°F.
The line (C)-(D) is the intermediate temperature of 15°F.
The line (A)-(B) is the condensing temperature of 105°F.

Normally, a 10 to 15°F approach between the intermediate temperature and the liquid leaving the intercooler should be allowed. A 15°F approach is used for this case. Therefore, the subcooled liquid leaving the intercooler coil is 15°F + 15°F = 30°F; that is the point (A) as shown in the Figure 25-1.

The liquid for intermediate liquid subcooler is to be from the condenser/receiver at point (G); the throttling line of (G)-(C) is established as shown in Figure 25-2. The subcooled liquid at 30°F which is from point (A) is throttling to the -40°F evaporator, therefore, the throttling line (A)-(E) is also drawn on the same P-H diagram.

The line (E)-(F) is the evaporator No.1 at evaporative temperature of -40°F.
The line (C)-(D) represents the intercooling load at the intermediate temperature of 15°F.

The evaporative temperature of No. 2 load is -35°F which is very close to the lowest ET of -40°F. The line (J)-(K) is Evaporators No.2 is at ET of -35°F. The saturated suction gas of 80 TR load at -35°F from point (K) is to be combined with the -40°F suction through a back pressure regulating valve to regulate the pressure from 12.05 Psia to 10.4 Psia. Both return to the booster compressor. suction at point (L) as shown in Figure 25-4.

The liquid supplies to No. 3 evaporator can be either from the subcooled liquid at 30°F point (A) or from receiver at 105°F at point (G). For this project, because the No. 3 evaporator is located not far away from the engine room, the decision is made that the liquid for No. 3 evaporator is to be from receiver at point (G); the suction gas from No.3 load is to be returned to the suction of the high stage compressor at point (N). Line

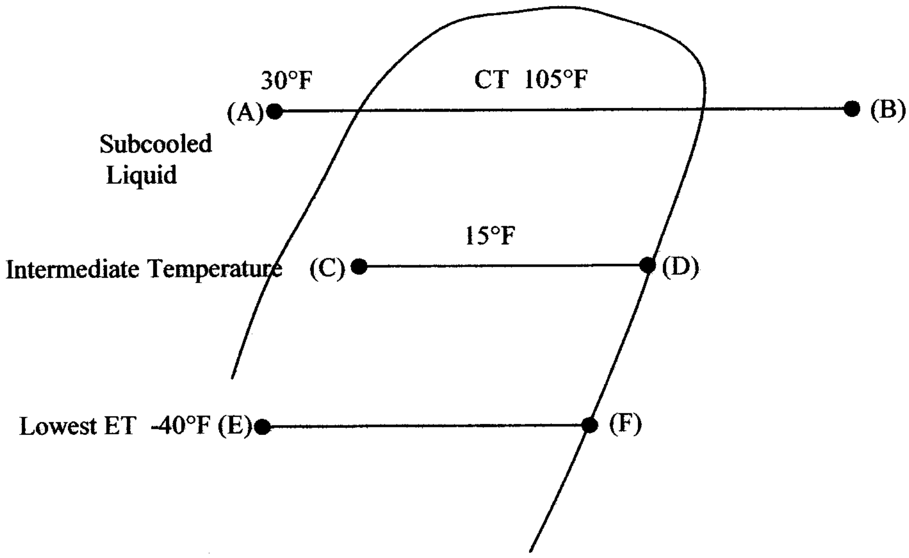

Figure 25-1 Determine CT, ET & Intermediate Temperatures on PH Diagram

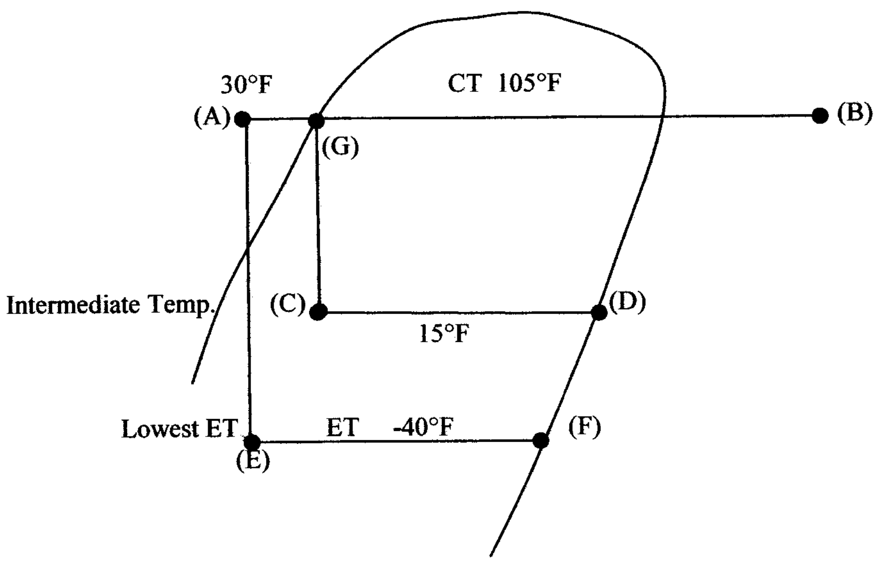

Figure 25-2 Liquid Subcooling & Liquid Feed Lines

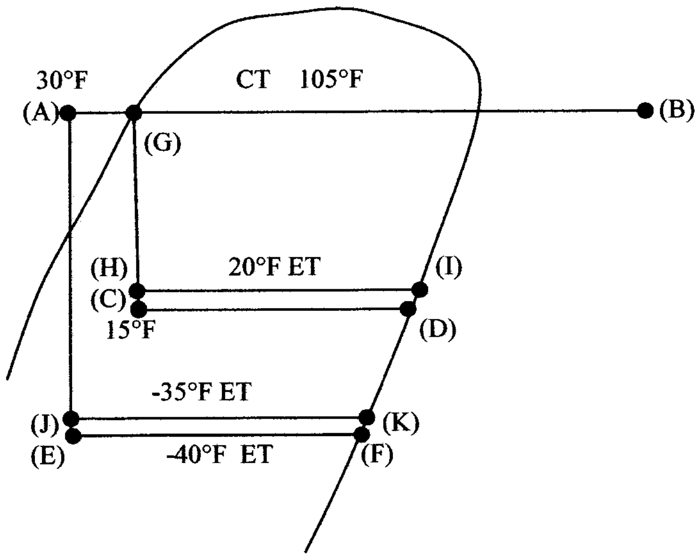

Figure 25-3 No. 2 & No. 3 Refrigeration Loads

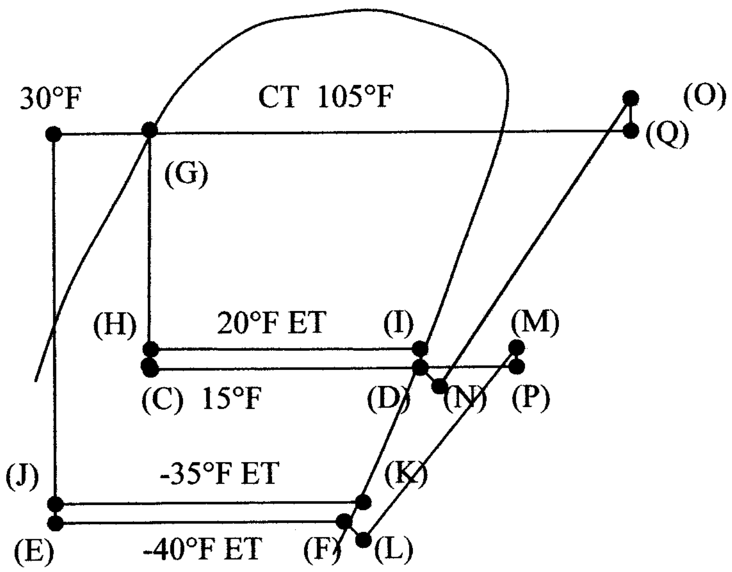

Figure 25-4 High Stage & Booster Compressor Lines Pressure Penalties

between points (H) to (I) represents the evaporative load No.3 as shown in Figure 25-4.

The saturated pressure corresponding to respective evaporative temperature for various main temperature lines on the P-H diagram are as the following:

Load No. 1:	ET -40°F	(10.4 Pisa)
Load No. 2:	ET -35°F	(12.05 Psia)
Load No. 3:	ET 20°F	(48.2 Psia)
Condensing Temperature:	105°F	(228.9 Psia)
Intermediate temperature:	15°F	(43.1 Psia)

For proper equipment selection and the valid piping system design, the suction pressure drop and suction superheat to determine the operating conditions for the compressor must be known. If it is not known, assumption should be made and then the result must be verified. Reasonable assumption of the line pressure drops and superheat are listed below:

Booster Compressor Suction Conditions:	
Suction external pressure drop:	0.5 Psia
Suction line superheat:	10°F
Booster Compressor Discharge External Pressure Drop:	0.5 Psia
High Stage Suction Conditions:	
Suction external piping pressure drop:	0.5 Psia
Suction superheat:	5°F
High Stage Discharge External Pressure Drop:	0.8 Psia

Line (D)-(N) shown in Figure 25-4 is the combination of suction pressure drop 0.5 psi and suction superheat 5°F for the high stage compressor suction; line (O)-(Q) represents the internal pressure of the compressor unit including the oil separator and the external discharge line pressure drop of 0.8 psi for the high stage compressor. Likewise, the line (F)-(L) shown in Figure 25-4 is the combination of the suction pressure drop of 0.5 psi and the suction superheat 10°F for the low stage (booster) compressor suction; the line (M)-(P) is the internal pressure drop plus the external pressure drop of 0.5 psi for the booster compressor discharge on the P-H diagram.

Refrigerant Flow Diagram:

Refrigerant flow diagram starts from the major components needed for the system. The refrigerant flow diagram is structured, composed and laying out exactly by following with the system design concepts and thermodynamic bases which were used to develop the P-H diagram for the system.

The major components for this compound system indicated in the P-H diagram of Figure 25-4 are water cooled condenser, high stage compressor, booster compressor and intermediate coil type intercooler; those components are shown in Figure 25-5 plus high pressure receiver and suction trap. Three suction lines from three evaporators are also shown.

Some of the components such as receiver, suction trap, valves and etc. are the hidden and it can not be shown in the P-H diagram. But, it is important for the good application practice for most refrigeration system design; it is to be shown in refrigerant flow diagram.

Connecting the components in accordance with the design logics established earlier in the P-H diagram, the system is now being and is shown in Figure 25-6, this is the refrigerant flow diagram for the system and it is formed in accordance with the design concepts as shown in the P-H diagram of Figure 25-4. P-H diagram is to show if the system is designed to meet with thermodynamic feasibilities and the refrigerant flow diagram is to show a mechanical working system reflecting the thermodynamic theory compliance to the P-H analysis.

PERFORMANCE REQUIRED FOR BOOSTER (LOW STAGE) COMPRESSOR:

Assign the enthalpy for points of point (A), (F) and (K). Point (A) is the subcooled liquid at 30°F, the enthalpy is 75.7 Btu/Lb; the saturated vapor enthalpy at -40°F for point (F) is 577.6 Btu/Lb and the saturated vapor enthalpy at -35°F for point (K) is 599.5 Btu/Lb as shown in Figure 25-7.

Calculating the refrigerant flow for the low stage compressor:

Flow (1) = Refrigerant flow required for the No.1 load at ET of -40°F:

$$= \frac{200}{(H_F - H_A)} \times TR$$

$$= \frac{200}{597.6 - 75.7} \times 260$$

$$= 99.64 \quad \text{Lbs/Min}$$

Flow (2) = Refrigerant flow required for the No.2 load at ET of -35°F:

$$= \frac{200}{(H_K - H_A)} \times TR$$

$$= \frac{200}{599.5 - 75.7} \times 30$$

$$= 11.45 \quad \text{Lbs/Min.}$$

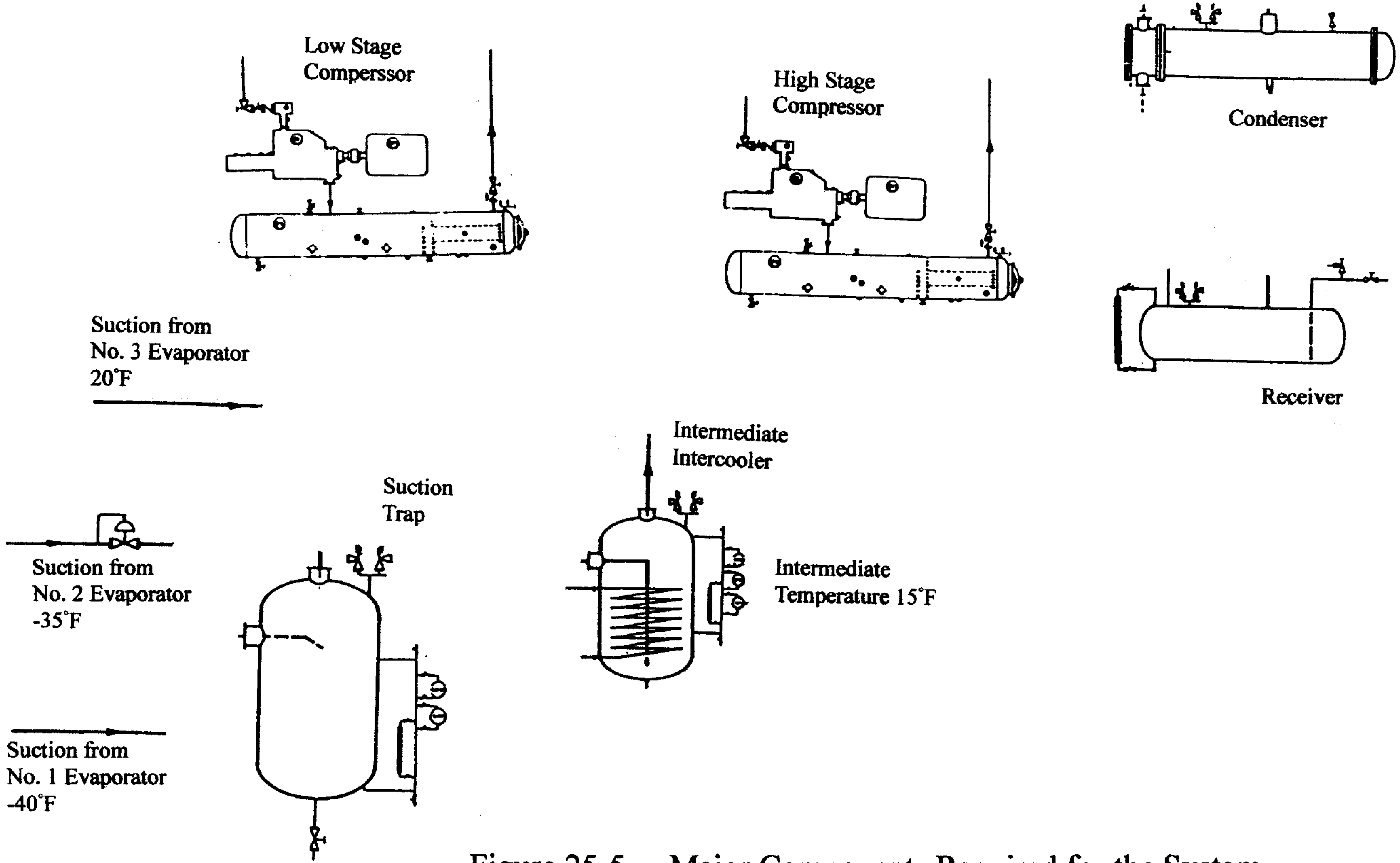

Figure 25-5 Major Components Required for the System

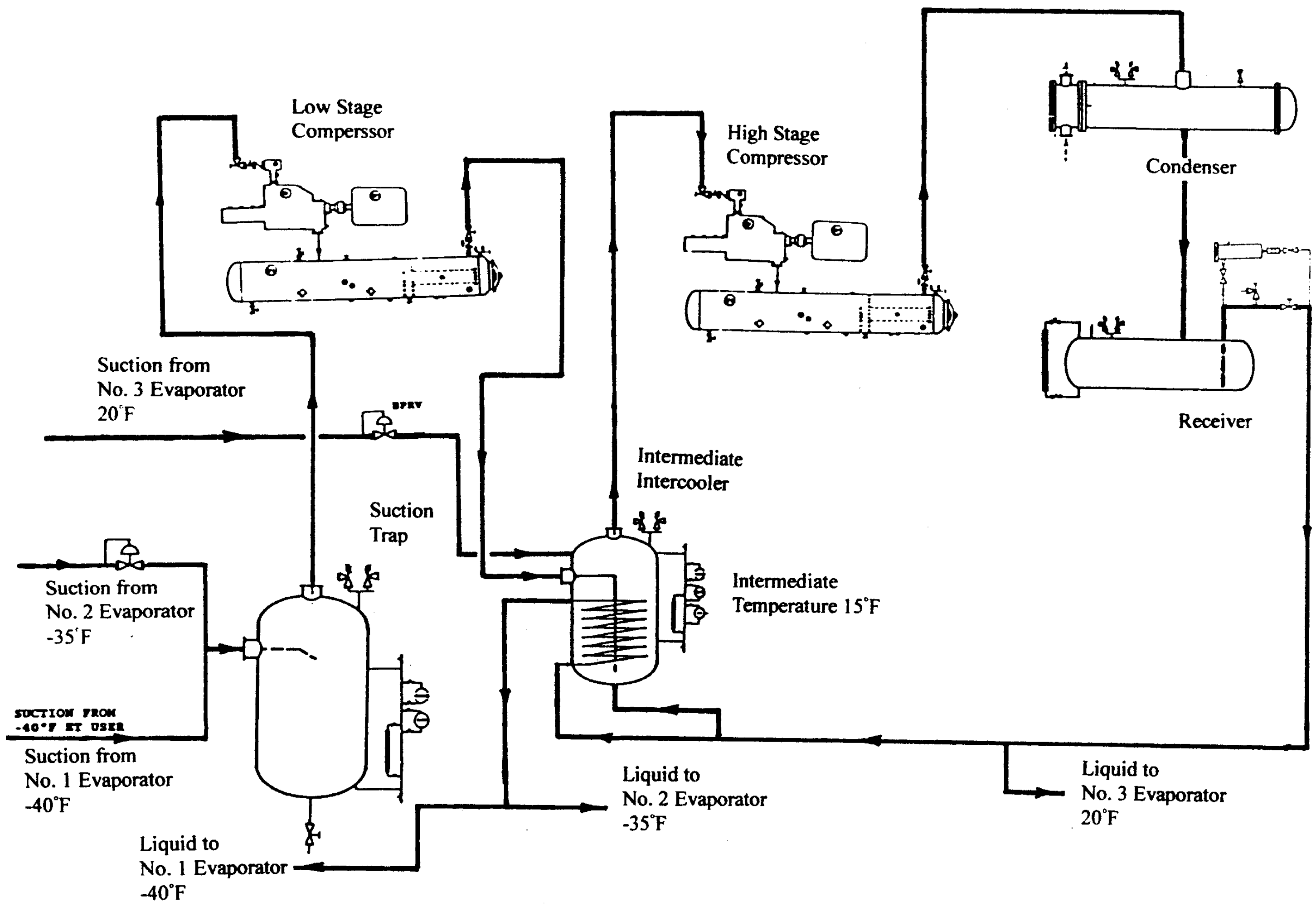

Figure 25-6 Formulating the Refrigeration System

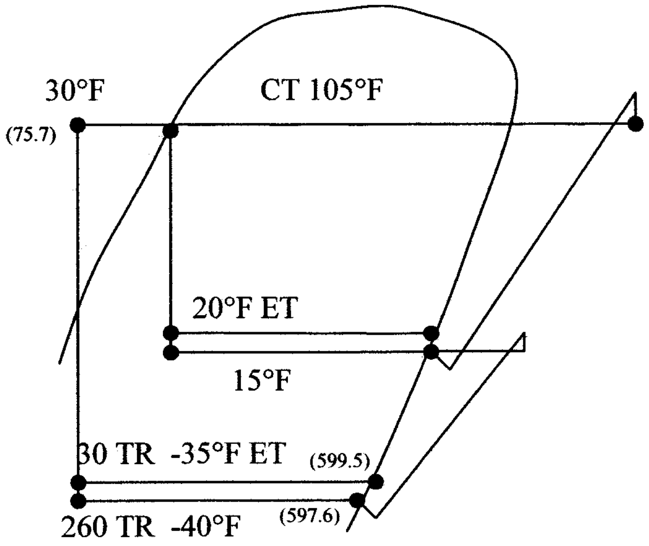

Figure 25-7 Enthalpy Points for No. 1 & No. 2 Loads

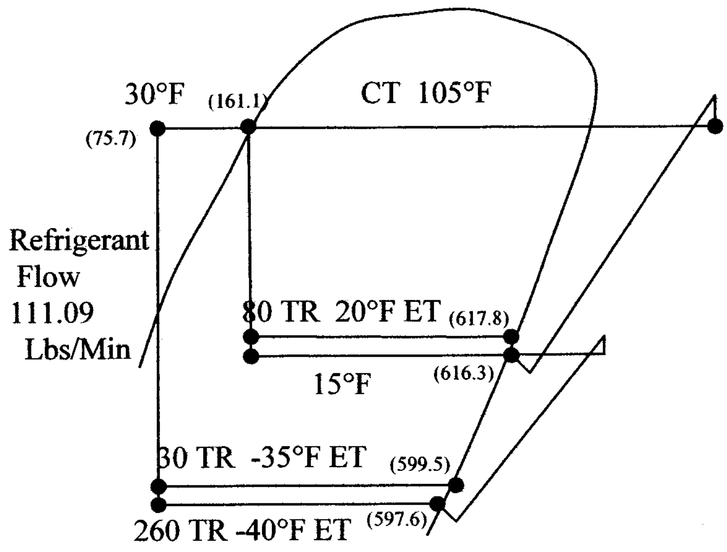

Figure 25-8 Enthalpy Points for No. 3 Load and Intercooler

Total refrigerant flow for the booster compressor:

= Flow (1) + Flow (2)
= 99.64 + 11.45
= 111.09 Lbs/Min

Operating conditions for the low stage (booster) compressor:

Refrigerant:	R-717
Refrigerant flow:	111.0 Lbs/Min.
Valves:	Standard
Intermediate temperature:	15°F
Compressor suction pressure:	10.35 Psi
Compressor suction temperature:	-30°F
Discharge external pressure drop:	0.5 Psi
Liquid temperature to evaporator:	30°F
Compressor speed:	2,950 RPM
Oil cooling:	Water cooled

The compressor can be selected by using the compressor manufacturer Computer Selection Program with the above operating conditions, or ask compressor maker to make compressor selection. The compressor selected is as the following:

Compressor Model:	RB-676B
Power Consumption:	363.6 BHP
Equivalent TR:	290 TR
Oil Cooler Heat Rejection:	295,400 Btu/Hr.

PERFORMANCE REQUIREMENT FOR HIGH STAGE COMPRESSOR:

Calculate the refrigerant for the high stage compressor:

The liquid refrigerant for No.3 load is from receiver. Assign the enthalpy for points of (I) (D) and (G) as indicated in Figure 25-8. The enthalpy for points (C), (H) and (G) are the same.

The total refrigerant gases flow to the suction of the high stage compressor are as the following:

(1) Refrigerant flow from booster compressor = 111.09 Lbs/Min

(2) Refrigerant flow from the 80 TR, No. 3 load at ET of 20°F:

$$= \frac{200}{617.8 - 161.1} \times 80$$

= 35.03 Lbs/Min

(3) Refrigerant flow for the liquid subcooling:

Liquid to be subcooled = 111.09 Lbs/Min.

Heat load required for liquid subcooling:

$$= \frac{111.09 \text{ x } (161.1 - 75.7)}{200}$$

= 47.4 TR

Refrigerant required for liquid subcooling in the intercooler:

$$\text{Refrigerant flow} = \frac{200}{616.3 - 161.1} \text{ x } 47.4$$

= 20.8 Lbs/Min

(4) The motor heat input to the booster compressor & heat removal from oil cooling for the booster compressor:

Booster Compressor Motor BHP = 363.6 BHP

Motor heat input = 363.6 x 2545
= 925,362 Btu/Hr

Water oil cooling heat removal = 295,400 Btu/hr.

Net heat input to the high stage compressor:

= Motor heat input – Oil cooling heat removal
= 925,362 – 295,400
= 629,962 Btu/Hr.

or 52.5 TR

Refrigerant flow required for the net heat input:

$$\text{Refrigerant flow} = \frac{200}{616.3 - 161.1} \text{ x } 52.5$$

= 23.07 Lbs/Min

Total suction refrigerant gas flow to the high stage compressor:

Total flow to the high stage:

= 111.09 + 35.03 + 20.8 + 23.07

= 189.99 Lbs/Min

The operating conditions for the high stage compressor are as the following:

Refrigerant:	R-717
Refrigerant flow:	189.99 Lbs/Min.
Valves:	Standard
Suction pressure:	42.6 Psi
Suction temperature:	20°F
Condensing temperature:	105°F
Discharge ext. pressure drop:	0.8 Psi
Compressor speed:	2,950 RPM
Oil cooling:	Water cooled

The compressor can be selected by using the compressor manufacturer Computer Selection Program with the above operating conditions, or ask compressor maker to make compressor selection. The compressor selected is as the following:

Compressor Model:	RB-316H
Power Consumption:	641.9 BHP
Equivalent TR:	434.2 TR
Oil Cooler Heat Rejection:	903,300 Btu/Hr.

PERFORMANCE REQUIREMENTS FOR WATER COOLED CONDENSER:

Oil Cooling Heat Rejection for the high stage compressor: 903,300 Btu/Hr.

Condenser Heat Rejection:

= 434.2 TR x 12000 + 641.9 BHP x 2545 – 903,300
= 5,940,736 Btu/Hr.

Cooling water flow calculation:

Cooling water entering temperature is 88°F.
Assuming 10°F range.
Leaving cooling water temperature is 98°F.

$$\text{GPM} = \frac{\text{Btu/Hr.}}{499.8 \quad \text{x} \quad (T_2 - T_1)}$$

Btu/Hr = 5,940,736 Btu/Hr

$T_2 - T_1 = 10°F$

Therefore, the cooling water requirement is:

$$GPM = \frac{5,940,736}{499.8 \times 10}$$

$$= 1,189 \text{ GPM}$$

Operating conditions for the condenser:

Heat Load:	5,940,736 Btu/Hr
Refrigerant:	R-717
Cooling water in temperature:	88°F
Cooling water leaving temperature:	98°F
Condensing temperature:	105°F.
Shell side DWP:	300 Psig
Tube side DWP:	150 Psig
Shell material:	Carbon steel
Tube material:	Carbon steel
Tube:	¾" 14 BWG Bare tubes
Fouling factor:	0.001
Pass arrangement:	2-P
Overall length or NTL limitation:	26'-0"
Water pressure drop limitation:	15 Psi

External Piping Pressure Drop Available for the #3 Evaporator at normal operation:

Evaporative pressure for the #3 Evaporator = 48.2 Psia
Intermediate intercooler pressure = 43.1 Psia
Pressure drop available for the #3 Evap.
= 48.2 – 43.1
= 5.1 Psi
Use a back pressure regulating valve for the suction line of #3 load is suggested.

Suction Traps:

Suction trap (scrubber) is required for remote evaporators. Therefore, it is suggested to use a suction trap for the evaporative loads No.1 and No.2 as shown in Figure 25-6. A back pressure regulator is used for the suction of the refrigeration load No.2.

The suction line of the No.3 load is connected to the intermediate intercooler. The intercooler is used as the suction trap for the No.3 refrigeration load suction instead of using a separate suction trap as shown in Figure 25-6.

Performance requirements for Intermediate Intercooler:

The intercooler coil is to cool 111.09 Lbs/Min ammonia liquid from 105°F to 30°F with an evaporative temperature of 15°F. The liquid flow to intercooler 43.87 Lbs/Min, this liquid is for the liquid subcooling and also to desuperheat 111.09 Lbs/Min discharge gas from the booster compressor. The cross section area of the intercooler shall be sized to handle total vapor flow of (35.03 + 43.87 + 111.09) = 189.99 Lbs/Min of saturated ammonia vapor.

Intercooler is usually equipment with liquid level sight glass and liquid level alarm and cutout switches.

Calculate Booster Compressor Suction Trap Size for Liquid/Gas Separation:

Suction refrigerant flow = 111.09 Lbs./Min. R-717
Refrigerant temperature: -30°F
Maximum vapor/liquid separation velocity: 143 Ft/Min.
Specific Volume at –30°F Vg = 18.97 Ft^3/Lb
CFM flow = 111.09 x 18.97 = 2,107.4 CFM

Cross Section Area of the Suction Trap:

$$A = \frac{1}{4} \times \pi \times D^2$$

A = Cross Section Area of the Suction Trap.
D = Diameter of the suction trap

Maximum Allowable Vapor Velocity, FPM = 143 Ft/Min:

$$FPM = \frac{CFM}{A}$$

$$FPM = \frac{CFM}{\frac{1}{4} \times \pi \times D^2}$$

CFM = Refrigerant Vapor Flow = 2,107.4

$$D \quad (\text{Inches}) = \sqrt{\frac{576 \times CFM}{\pi \times FPM}}$$

FPM = 143 Ft/Min.

$$D \text{ (Inches)} = \sqrt{\frac{576 \times 2107.4}{\pi \times 143}}$$

= 51.98 Inches
52”

The size of the suction trap might be 52” x 8’-0”

The suction trap is usually equipped with liquid level sight glass, liquid level alarm and cutout switches. Also liquid transfer accessories are to be provided as desired.

The high pressure receiver size is to be determined if it is for storage receiver or just for operating receiver.

The electrical requirements are:

Electrical Classification:	Class I, Group D, Division II or Ex e, IIA, Zone 2
Power supply:	Compressors Oil heater, 220-1-50. Compressor Oil pump, 415-3-50. Control panel, 220-1-50
Compressor driving motors:	WP-II enclosure, 6000-3-50. with space heater 220-1-50.

Final P-H Diagram and Refrigerant Flow Diagram:

The final P-H diagram and the system refrigerant flow diagram for this refrigeration system are shown in Figure 25-9 and Figure 25-10 respectively.

Discussion:

In order to minimize the field construction, therefore, the refrigeration unit is to be a custom-built and skid-mounted unit and shall be fabricated by the equipment manufacturer.

The pressure drop values are assumed. Piping sizes and distances are to be checked and verified to see if the pressure drops allowed are enough. If not, the piping size is to be increased or the screw compressors are to be reselected.

The control voltage for 50 Hz power supply is 220-1-50. Most control voltage is 110V or 24V. Most control panel is design for 110~220V power supply. If the control voltage is 110V, a 3~5 KVA control transformer should be provided in the starter for controls.

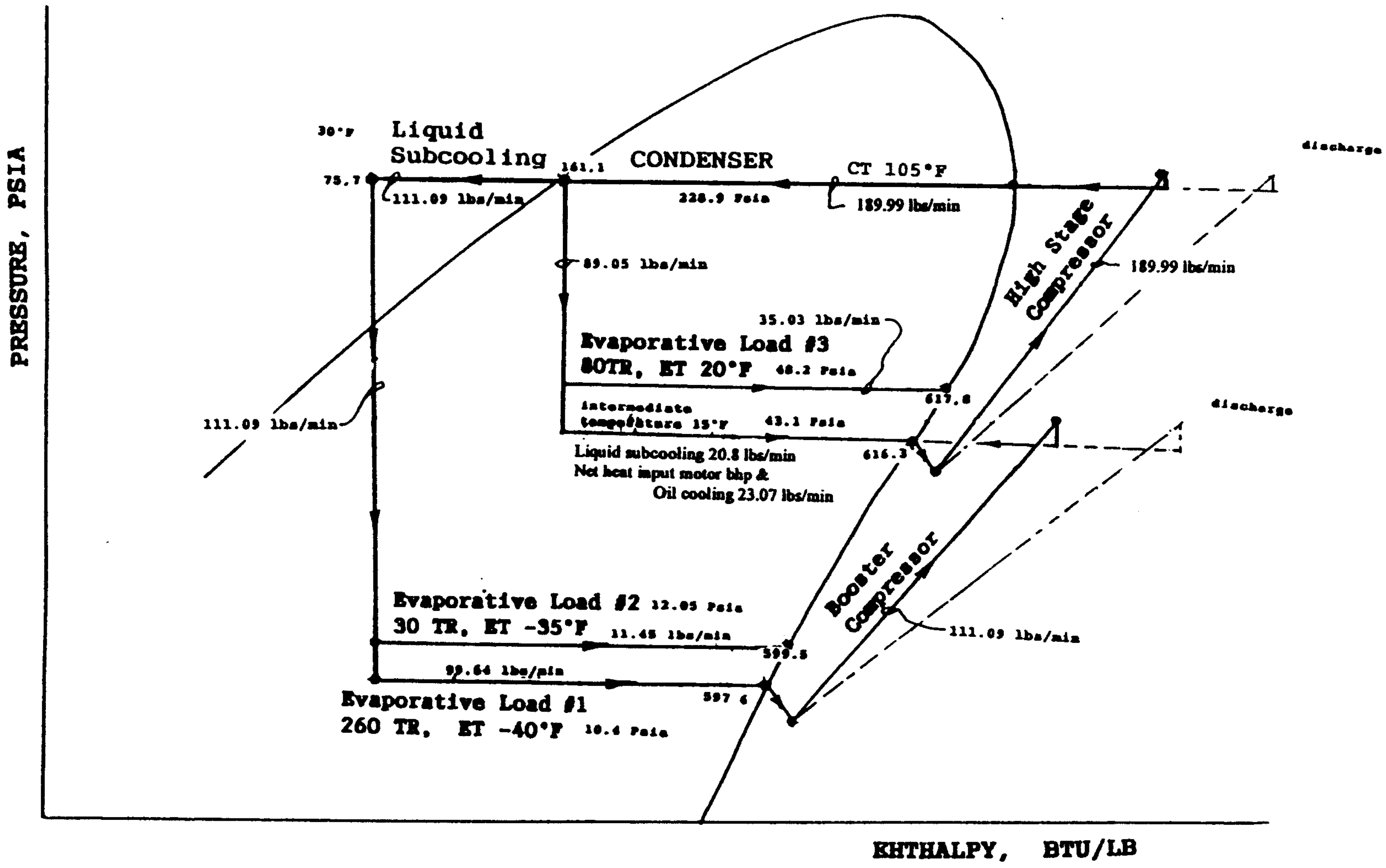

Figure 25-9 P-H Diagram for the Compound System

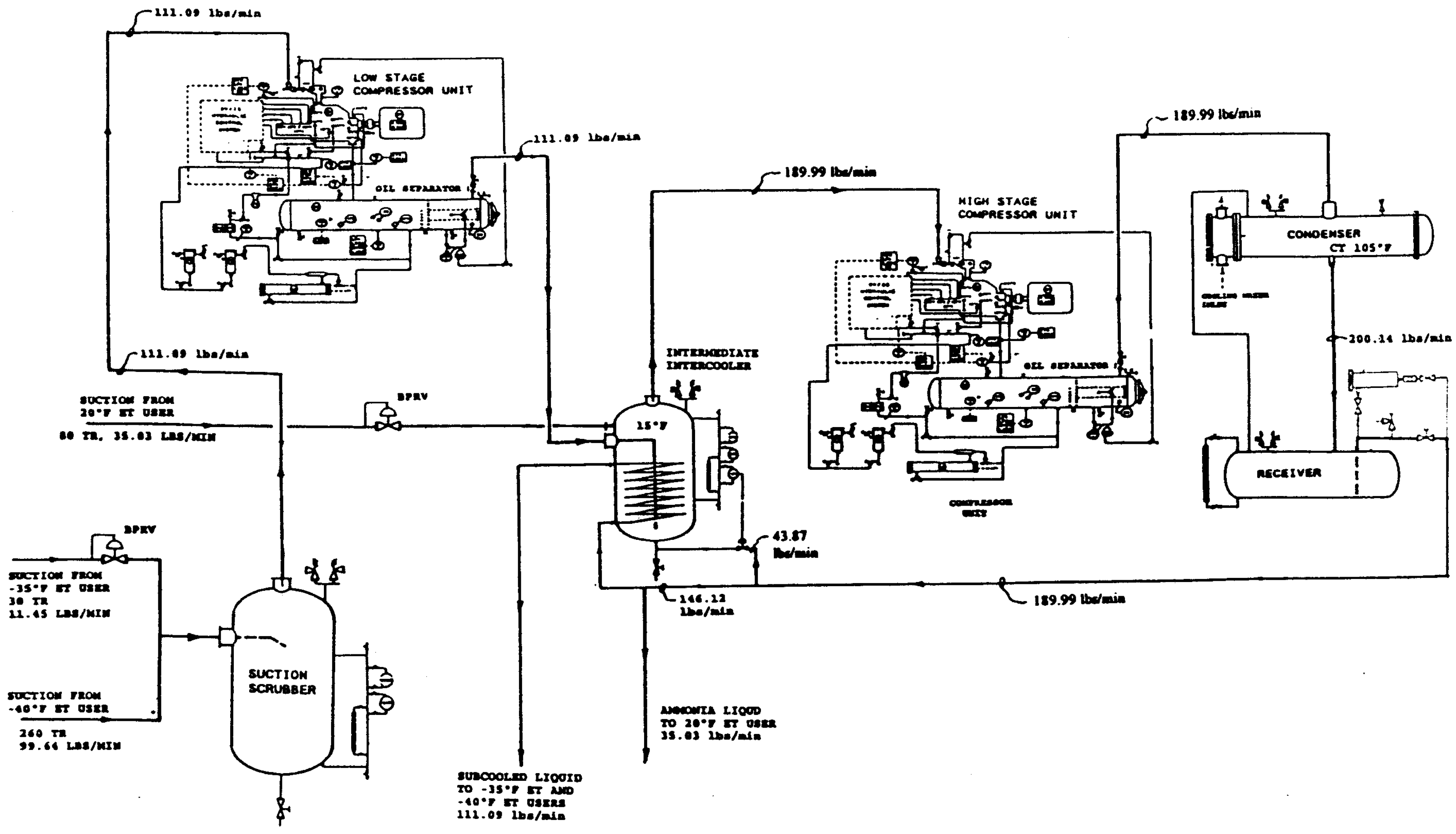

Figure 25-10 Refrigerant Flow Diagram
Compound System

The design procedures are the same if the system is used for 60 Hz application instead of 50 Hz power supply. The power supply for motor for 60 Hz might be 4160-3-60; for oil heater and space heater for motor might be 230-3-60.

The control panel is to be NEMA-4 with purge for Class I, Group D, Division II or Ex e, IIA, Zone 2.

All the equipment and controls are to be designed for outdoor installation if the system is to be located outdoor.