Case – 11 Brine and Chilled Water Compound System

Case Background:

This is a refrigeration system design to handle two process loads simultaneously; one process load is brine chilling and the other is to produce chilled water. The outline basic requirements of the refrigeration system are as the following:

1.0 Brine Chilling - To cool 1,938 gpm, 35% by weight of Ethanol brine from 15°F to 7°F leaving. Fouling Factor 0.0005.

2.0 Process Cooling - To cool 720 gpm of water from 55°F to 45°F leaving. Fouling Factor is 0.0005.

3.0 Refrigerant shall be R-134a.

4.0 Condenser cooling water is to be 90°F in and 100°F out. Fouling Factor is 0.0005.

5.0 The power consumption of the system is considered important and therefore, options of various systems are to be considered.

6.0 One system for the two refrigeration loads is to be used instead of two separate systems.

7.0 Power supply for the main driver motor is 6000-3-50. Other power supply is 415-3-50 and 230-1-50.

For a better comparison of various proposals, the ET and CT for the system are to be designed in accordance with the following:

The Evaporative temperature for the Brine Chilling is to be –2°F.
The Evaporative temperature for water chilling is to be 41°F.
The Condensing temperature for the condenser is to be 106°F.

The Ethanol brine property is shown in the section of Related Technical Data and Engineering Information for the Base.

Related Technical Data and Engineering Information for the Case:

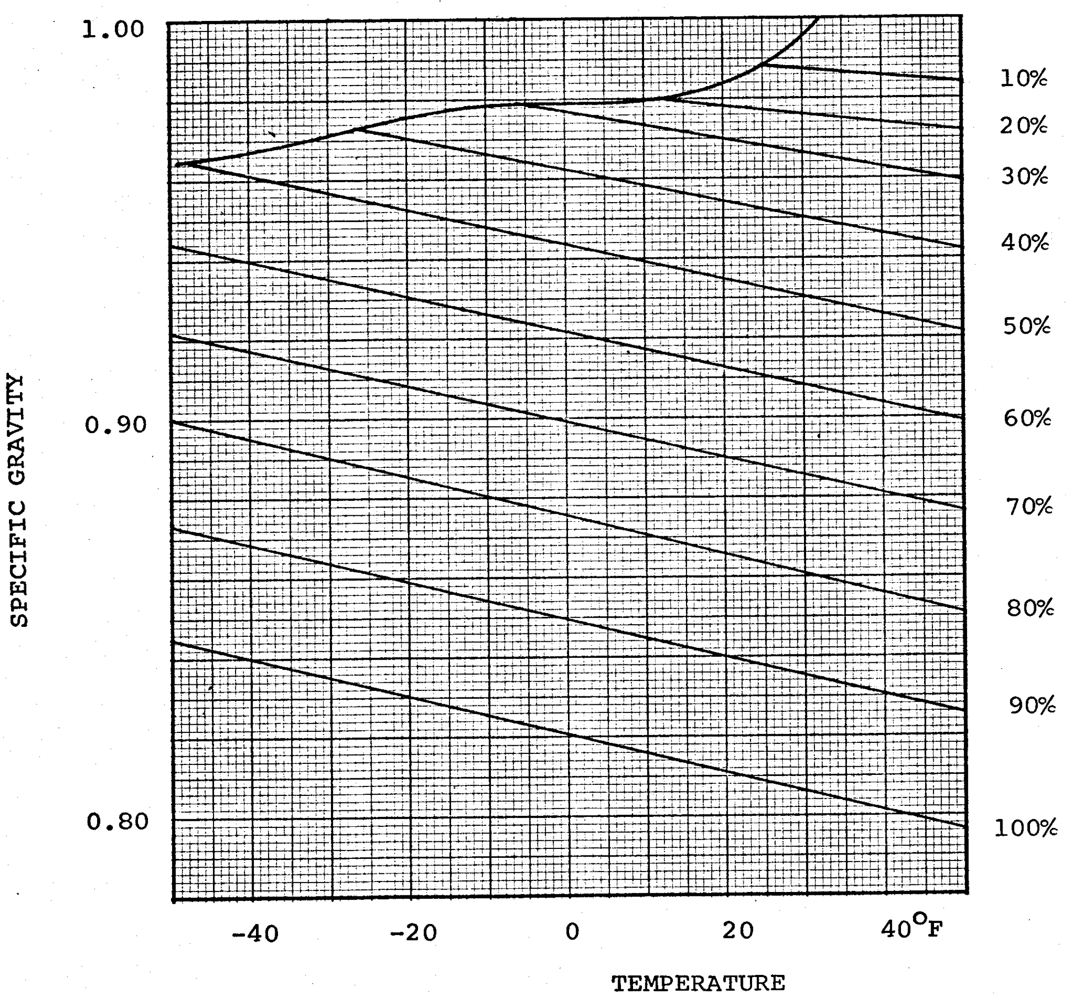

Figure 11-1 Specific Gravity – Ethanol Brine

NOTES:

The brine data of Figure 11-1 and Figure 11-2 are for preliminary design. It is suggested to use the brine data from the evaporator manufacturer's brine data for the selection of the heat exchangers.

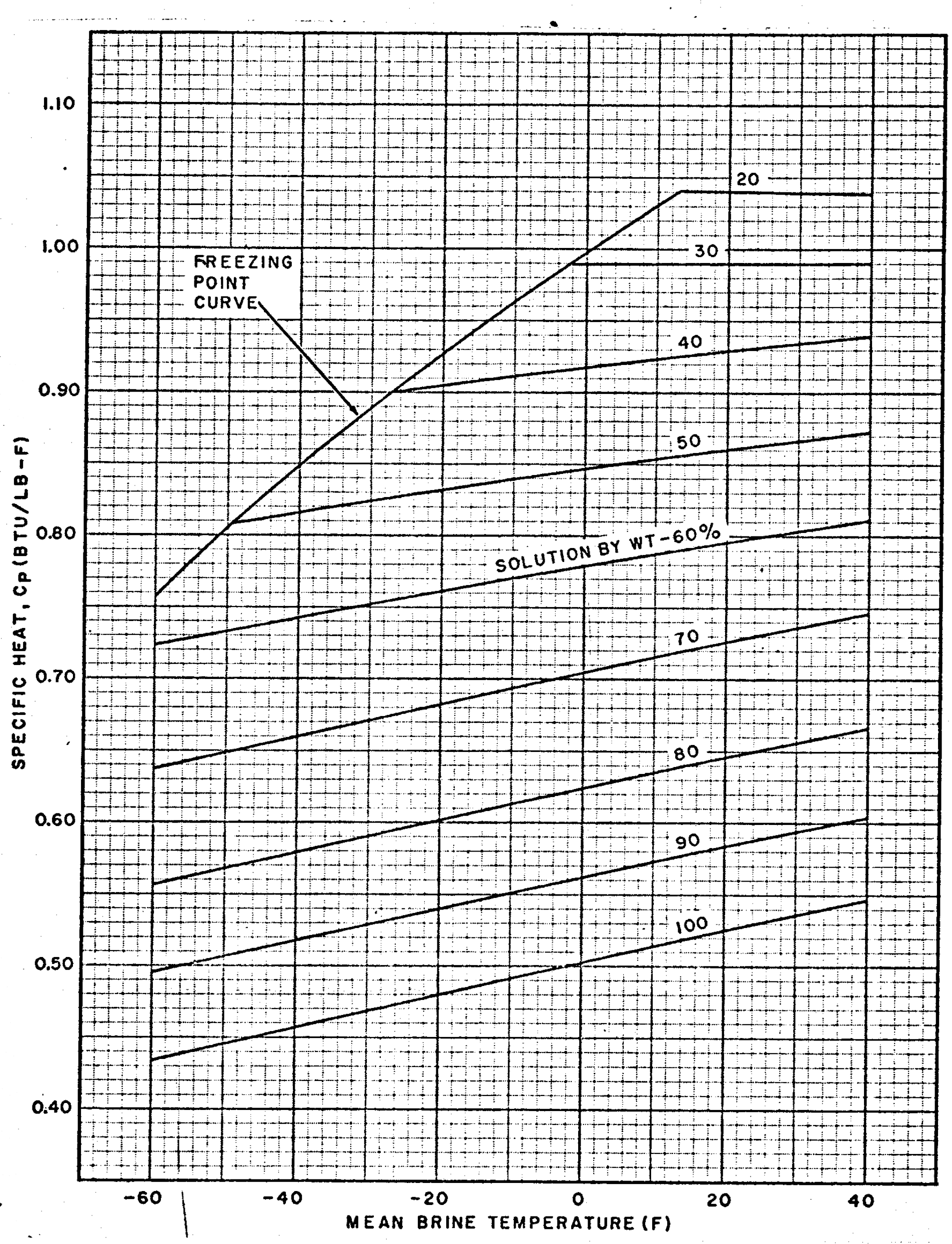

Figure 11-2 Specific Heat – Ethanol Brine

It is suggested to have the following computer programs available from the equipment manufacturer:

(1) Brine Properties Programs.
(2) R-134a Refrigerant Properties Computer Program.
(3) Centrifugal Compressor Selection and Rating Computer Programs.
(4) Screw Compressor Selection and Rating Computer Programs.

If those computer programs are not ready available, ask the manufacturer to make the selections base on the operating conditions as specified.

Case Cogitation:

In order to determine what type of equipment and system are to be used, first is to see what is the refrigeration capacity for the system.

Capacity of Brine Chilling for Process Refrigeration:

Calculating the refrigeration load for the brine chilling:

Brine:	35% by weight of Ethanol
Entering Temperature:	15°F
Leaving Temperature:	7°F
Brine flow:	1,938 gpm
ET :	–2°F

$$\text{Average Brine Temperature: } \frac{15 + 7}{2} = 11°F$$

From computer brine property program or from Figure 11-1 and 11-2:
At brine average temperature 11°F and 35% by weight.

Specific Gravity:	0.964
Specific Heat:	0.9634

Refrigeration Load:

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

$$= \frac{1938}{24} \times 0.964 \times 0.9634 \times (15 - 7)$$

$$= 600 \text{ TR}$$

Therefore: Process refrigeration load is 600 TR

Capacity of Water Chilling for Process Cooling Load:

Chilled water flow:	720 GPM
Entering Temperature:	55°F
Leaving Temperature:	45°F

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

$$= \frac{720}{24} \times 1.00 \quad \times 1.00 \quad \times (54 - 45)$$

$$= 300 \quad TR$$

Therefore, Process cooling load for water chilling is 300 TR

From the load requirements, it is too large for the use of reciprocating compressors. Therefore, centrifugal or screw compressor are to be considered for the application.

Evaporative Temperature specified for brine chilling is –2°F and the process cooling ET is 41°F. Therefore, three lines are already fixed on the P-H diagram as shown in the Figure 11-3:

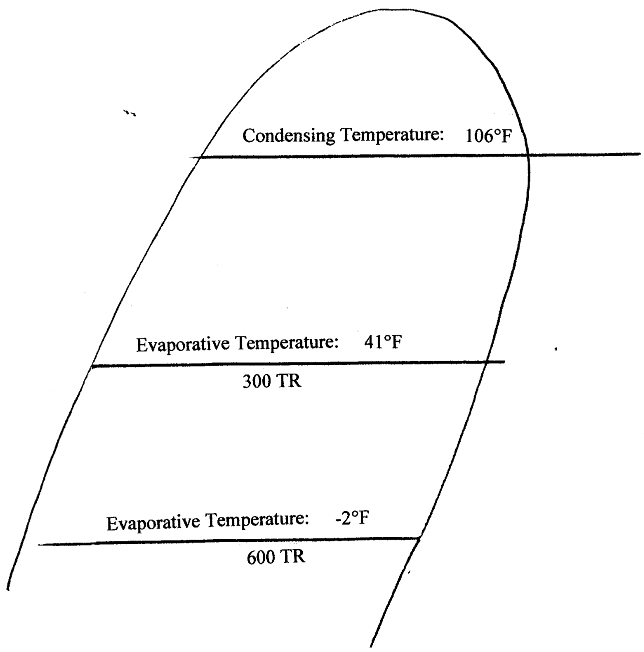

Figure 11-3 Preliminary P-H Diagram

Process heat load is 300 TR at ET of 41°F and 600 TR at ET of -2°F. If a compound (high stage & low stage) system is to be used, it is logical to use the 41°F which is the same as the ET of the water chiller as the intermediate temperature for the compound sytem to simplify the system design; the refrigeration system is a close coupled arrangement within the engine room; therefore, a flash type intercooler can be used; the liquid to brine chiller is supplied from intermediate intercooler at temperature of 41°F. The P-H diagram for the compound system is now being formed as shown in Figure 11-4.

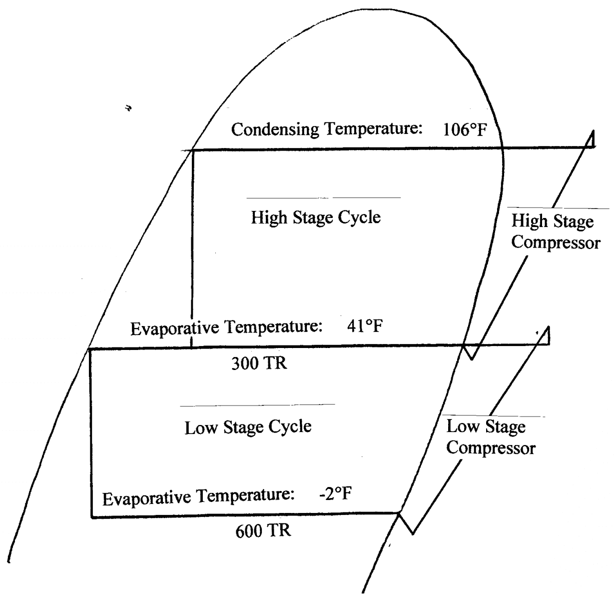

Figure 11-4 P-H Diagram for the Compound System

The corresponding refrigerant flow diagram is shown in Figure 11-5. Half Bundle Flooded heat exchangers are used for the Water Chiller and the Brine Chiller.

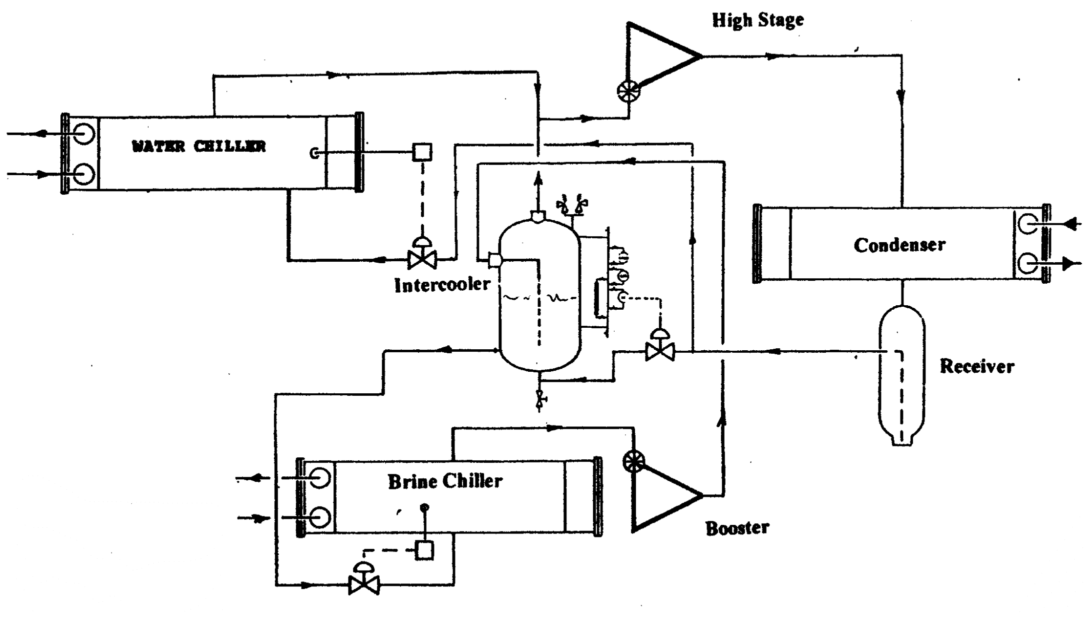

Figure 11-5 Refrigerant Flow Diagram for the Compound System

The basic system is now fixed, but there still several options are available for the evaluation for the compound system by using different combination of the centrifugal and screw compressors for the high stage and low stage for the system; the possible combination options are listed as the following:

	Low Stage Compressor	High Stage Compressor
Option-1	Centrifugal	Centrifugal
Option-2	Screw Booster	Screw W/Economizer
Option-3	Centrifugal	Screw W/Economizer
Option-4	Screw Booster	Screw No Economizer
Option-5	Centrifugal	Screw No Economizer

NOTES:
If a screw is used for booster duty, it is without economizer due to low pressure differential application; centrifugal compressors used are single stage or two stages design, but, no economizer is used because of low head.

General Considerations for Compressor Selection:

(a) Use manufacturer's computer selection program for screw and centrifugal compressor selection.

(b) The external suction pressure drop is 0.48 Psia.
The external suction superheat is 5°F.
The external discharge pressure loss is 0.75 Psia.

(c) For centrifugal compressor: The compressor suction entrance loss is 0.2 psi and discharge nozzle loss is 0.4 psi.

(d) For screw compressors: Standard suction and discharge valves. Water cooled oil cooler. Pre-lube oil pump for high stage and demand oil pump for the booster. No economizing for low stage. Liquid subcooling economizer for the high stage, pressure drop 4.5 psi and approach is 10°F. No economizing for booster.

(e) If the screw compressor rating program does not include the pressure drops for the discharge oil separator, valves, strainer and check valve, the pressure drops are to be included in the external pressure drop in addition to the external piping loses.

The System Calculation and Compressor Selection for System Option-1:

This compound system is to use centrifugal compressors for both high stage and low stage:

Booster (Low Stage) Centrifugal Compressor Calculation:

Operating conditions given:

Brine Chilling::	600 TR.
Refrigerant:	R-134a
Evaporative Temperature:	-2°F
Evaporative Pressure:	20.18 Psia
Intermediate Temperature:	41°F
Intermediate Pressure:	50.75 Psia
Suction piping loss:	0.48 Psi
Centrifugal Entrance Loss:	0.2 Psi
Suction line superheat:	5°F
Discharge Piping Loss:	0.75 Psi
Compressor Discharge Nozzle PD:	0.2 Psi
Input speed:	2,950 RPM At 50 Hz

The low stage compressor cycle can be treated as a simple refrigeration cycle and the intermediate temperature is to be considered as the condensing temperature for the low stage as shown in Figure 11-6:

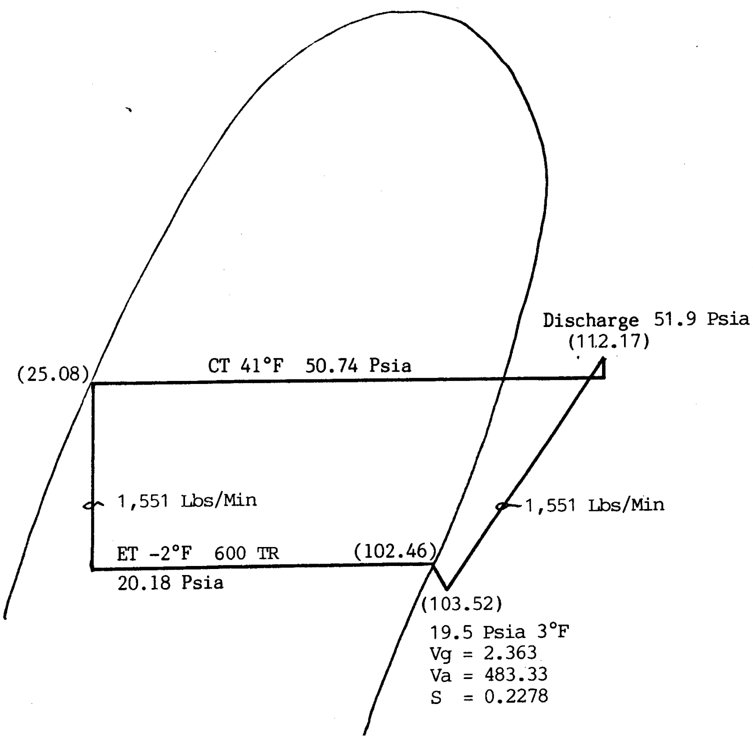

Figure 11-6 Low Stage Cycle

The enthalpy and property data shown above for the R-134a are from P-H diagram, refrigerant tables or from computer program..

Discharge Pressure = 50.75 + 0.75 + 0.4
= 51.9 Psia

Suction Pressure = 20.18 – 0.48 – 0.2
= 19.5 Psia

Suction Temperature = -2 + 5
= 3°F

Compressor
Suction Conditions: 19.5 Psia, 3°F

Refrigeration Load : 600 TR

Low stage compressor suction refrigerant flow:

$$\text{Flow, Lbs/Min} = \frac{200}{\text{Refrigeration Effect}} \times \text{TR}$$

$$= \frac{200}{102.46 - 25.08} \times 600$$

$$= 1551 \text{ Lbs/Min.}$$

$$\text{ACFM} = (\text{Lbs/Min}) \times V_g$$

$$= 1551 \times 2.363$$

$$= 3665 \text{ ACFM}$$

Low stage centrifugal compressor selected from the manufacturer is model KA-65
SHP = 432 HP

Heat Load generated from low stage compressor:

$$= 600 \text{ TR} + \frac{432 \times 2545}{12000}$$

$$= 691.6 \text{ TR}$$

Total Heat Load for the High Stage Compressor:

= Heat load from low stage + Process Cooling Load

= 691.6 TR + 300 TR

= 991.6 TR

High Stage Compressor Calculation:

Operating conditions for High Stage K-Centrifugal :

Capacity:	991.6 TR.
Refrigerant:	R-134a
Evaporative Temperature:	41°F
Evaporative Pressure:	50.75 Psia
Condensing Temperature:	106°F
Condensing Pressure:	152.03 Psia
Suction piping loss:	0.48 Psi
Suction Entrance PD:	0.2 Psi
Suction line superheat:	5°F
Discharge Piping Loss:	0.75 Psi
Compressor Discharge PD:	0.4 Psi
Input speed:	2,950 RPM At 50 Hz

Again, the high stage compressor cycle can be treated as a simple refrigeration cycle and in this case, the evaporative temperature is also the intermediate temperature for the high stage. The cycle is as shown in Figure 11-7.

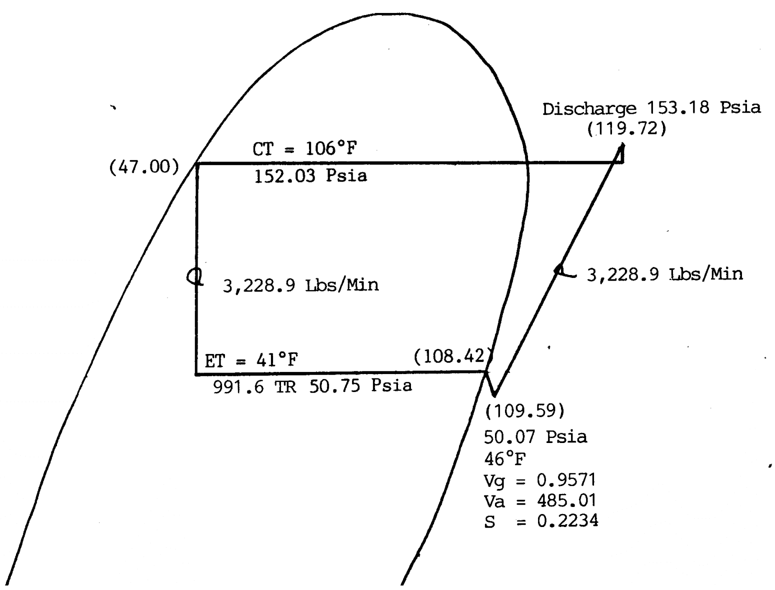

Figure 11-7 P-H Diagram for the High Stage Compression

Compressor calculation:

Discharge Pressure = 152.03 + 0.75 + 0.4
= 153.18 Psia

Suction Pressure = 50.75 – 0.48 – 0.2
= 50.07 Psia

Suction Temperature = 41 + 5
= 46°F

Compressor
Suction Conditions: 50.07 Psia, 46°F

Refrigeration Load : 991.6 TR

Thermodynamic data are shown in the P-H diagram.

$$\text{Flow, Lbs/Min} = \frac{200}{\text{Refrigeration Effect}} \times \text{TR}$$

$$= \frac{200}{108.42 - 47.00} \times 991.6$$

$$= 3228.9 \quad \text{Lbs/Min.}$$

ACFM = (Lbs/Min) x V_g

= 3228.9 x 0.9571 = 3090.4 ACFM

The high stage compressor selected by the manufacturer is KA-65
SHP = 990.2 HP

System Heat Rejection:

= 991.6 x 12000 + 990.2 x 2545

= 14,419,259 Btu/Hr.

= 14,420 MBH

Cooling Water Required:

$$GPM = \frac{Btu/Hr}{499.8 \times \Delta T}$$

$$= \frac{14{,}419{,}259}{499.8 \times (100 - 90)}$$

$$= 2885 \text{ GPM} \qquad 90°F \text{ to } 100°F$$

(A) Centrifugal Compressor Compound System:

(1)	High stage compressor BHP:	990.2 BHP
(2)	Low stage compressor BHP:	432 BHP
(3)	Total System Power Consumption:	1,422 BHP
(4)	Total heat rejection:	14,420 MBH
(5)	Cooling Water Flow:	2,885 GPM

P-H Diagram for the compound system is shown in Figure 11-8.

Refrigerant Flow Diagram is shown in Figure 11-9.

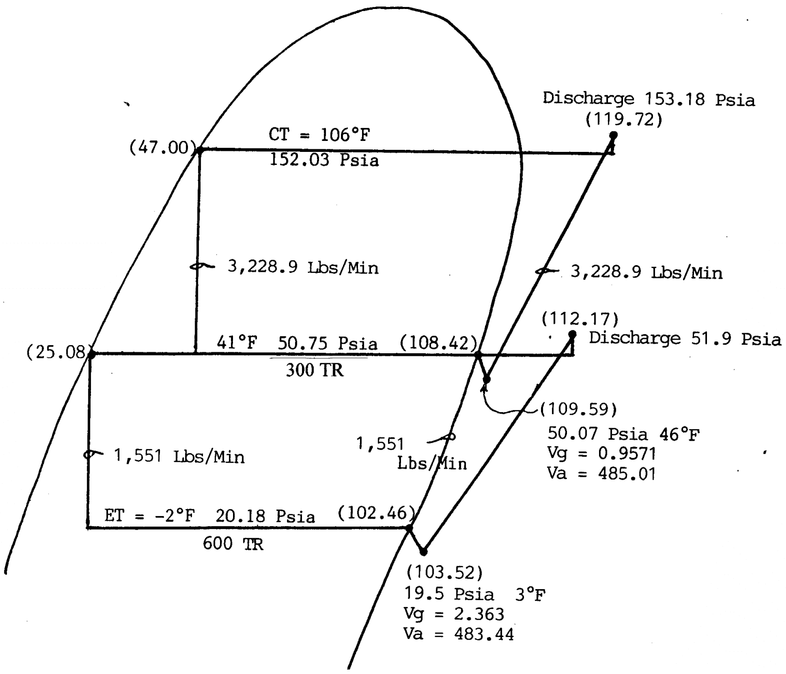

Figure 11-8 P-H Diagram for Compound System Option-1

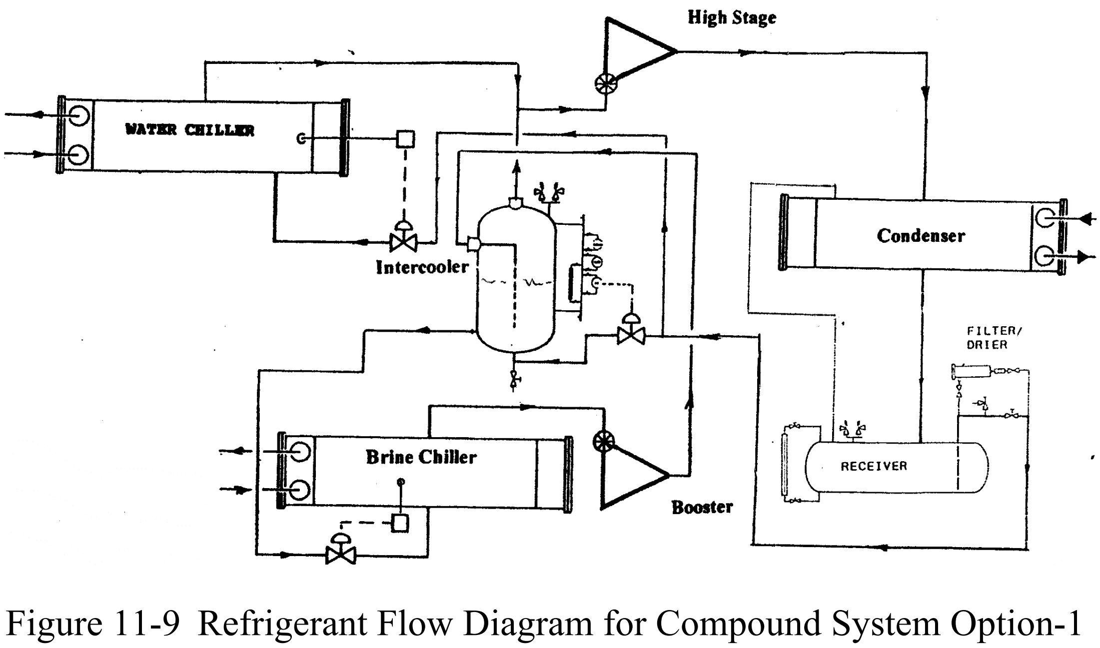

Figure 11-9 Refrigerant Flow Diagram for Compound System Option-1

The System Calculation and Compressor Selection for System Option-2:

This compound system is to use screw compressors for both high stage and low stage. The high stage screw compressor is with economizer, but the low stage (Booster) compressor is without economizer.

Compressor used for this Option-2:

Low Stage (Booster): Screw Compressor without Economizer
High Stage: Screw Compressor with Economizer

The calculation and selection procedures are almost the same as for the System Option-1, except that the oil cooling for the screw compressor is water cooled, portion of the heat for oil cooling is to be deducted from the total heat rejection to high side or to the condenser.

The P-H diagram for this Option-2 is as the Figure 11-10:

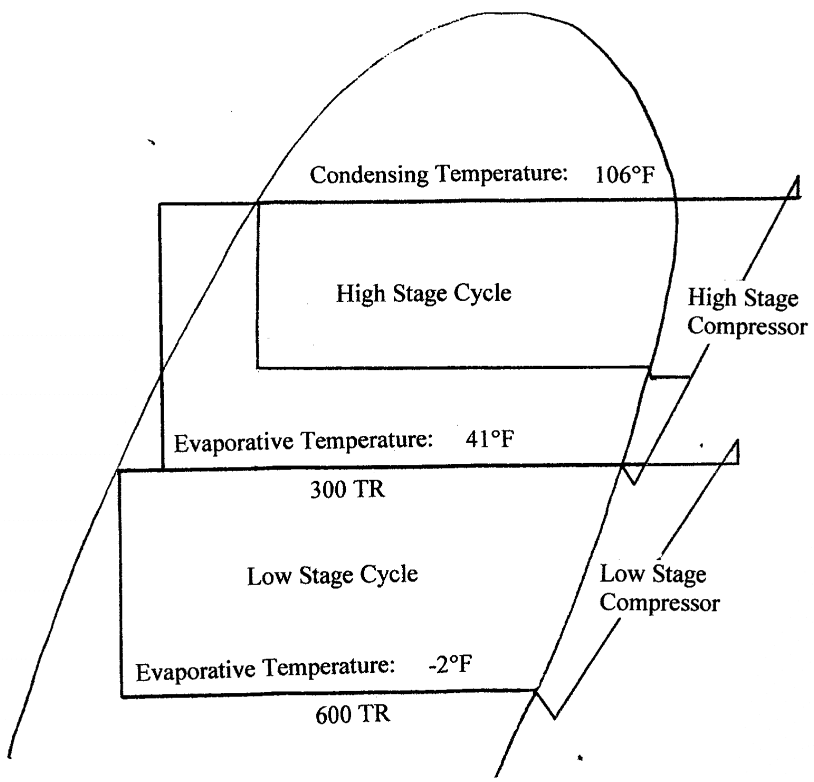

Figure 11-10 P-H Diagram for Compound System Option-2

The refrigerant flow diagram for the Compound System Option-2 is as shown in Figure 11-11:

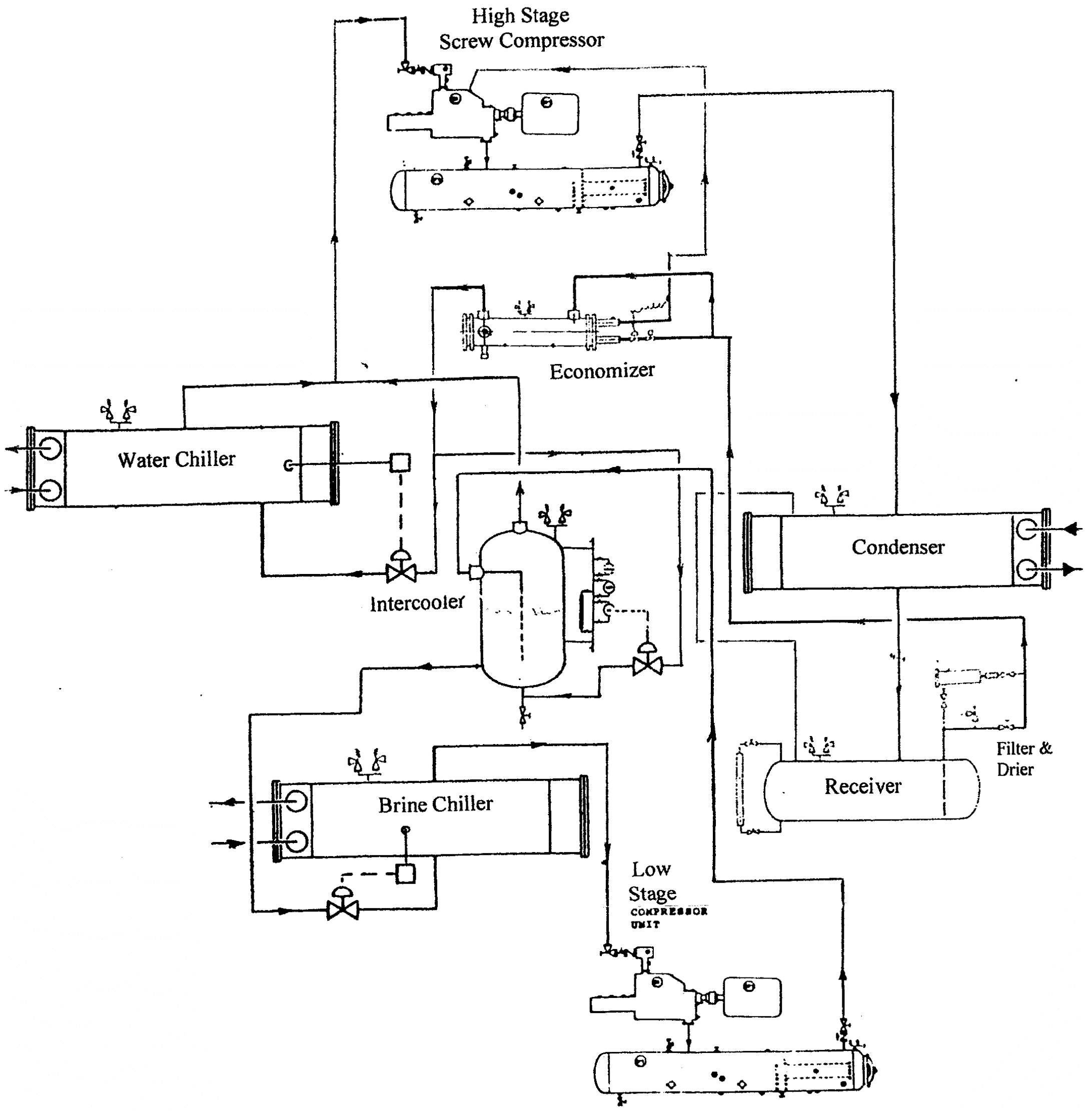

Figure 11-11 Refrigerant Flow Diagram for Compound System Option-2

The System Calculation and Compressor Selection for System Option-3:

This compound system is to use screw compressor for high stage and centrifugal compressor for the low stage (Booster). The high stage screw compressor is with economizer.

Compressor used for this Option-3:

Low Stage (Booster):	**Centrifugal Compressor**
High Stage:	**Screw Compressor with Economizer**

The calculation and selection procedures are almost the same as for the System Option-1, except that the oil cooling for the high stage screw compressor is water cooled, portion of the heat for oil cooling is to be deducted from the total heat rejection to the condenser.

The P-H diagram for this Option-3 is same as the Figure 11-10. The refrigerant flow diagram is shown in Figure 11-12.

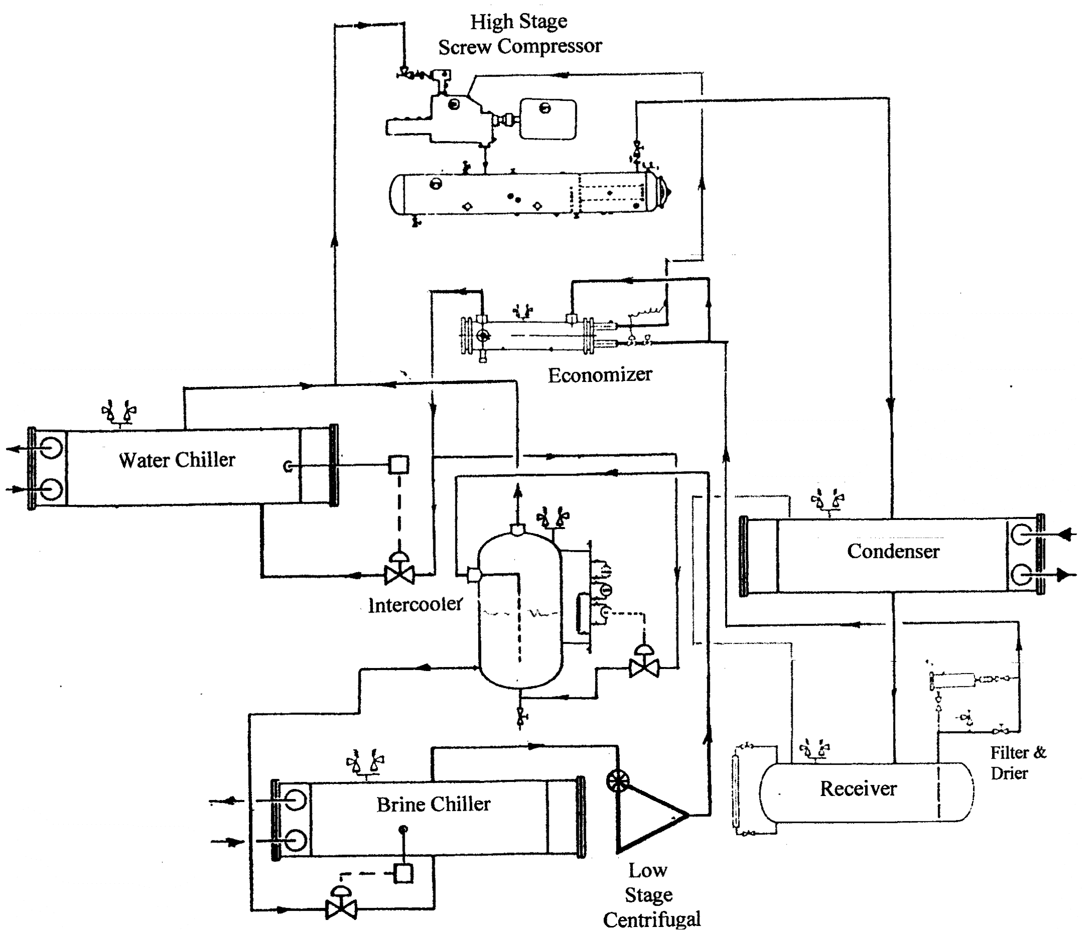

Figure 11-12 Refrigerant Flow Diagram for Compound System Option-3

The System Calculation and Compressor Selection for System Option-4:

This compound system is to use screw compressors for both high stage and for the low stage (Booster). Both are without the economizer.

Compressor used for this Option-4:

Low Stage (Booster): Screw Compressor without Economizer
High Stage: Screw Compressor without Economizer

The calculation and selection procedures are almost similar to the System Option-1, except that the oil cooling for the screw compressors are water cooled, portion of the heat for oil cooling is to be deducted from the total heat rejection to the high side or to the condenser.

The P-H diagram for this Option-4 is same as the Figure 11-4. The refrigerant flow diagram is shown in Figure 11-13.

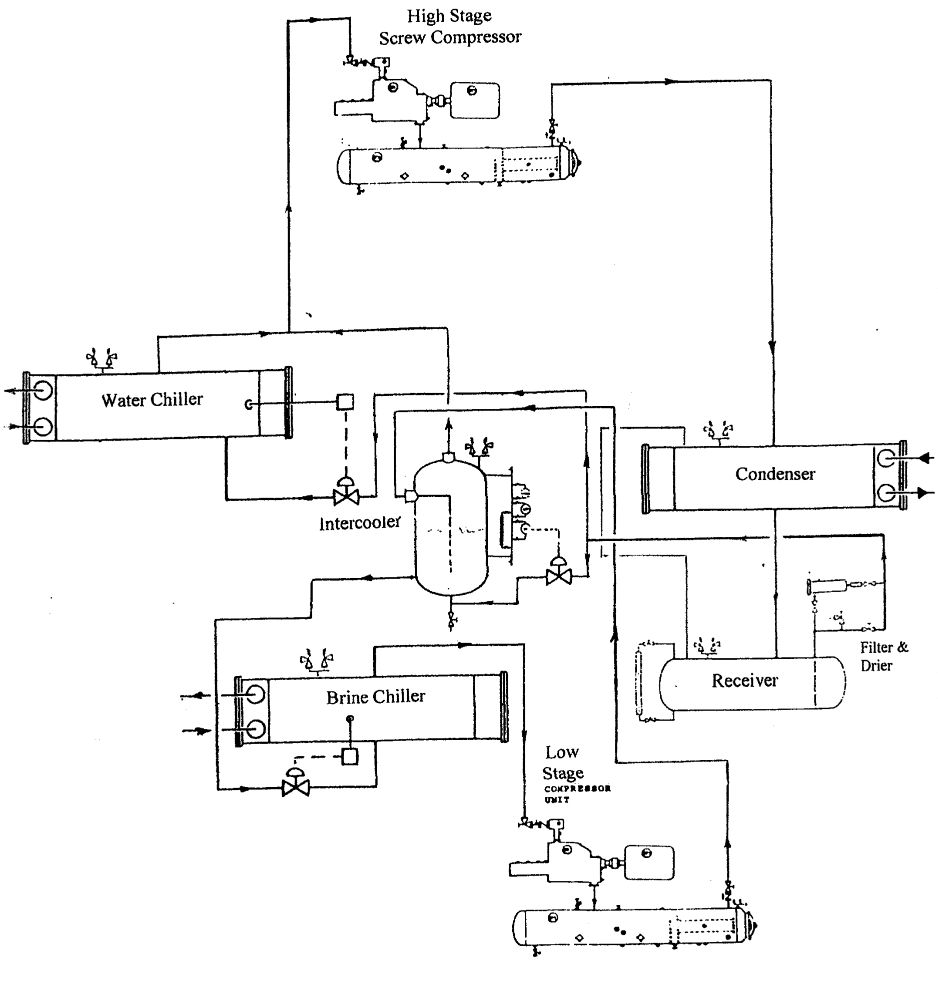

Figure 11-13 Refrigerant Flow Diagram for Compound System Option-4

The System Calculation and Compressor Selection for System Option-5:

This compound system is to use screw compressor for the high stage and the centrifugal compressor for the low stage (Booster).

Compressor used for this Option-5:

Low Stage (Booster): **Centrifugal Compressor**
High Stage: **Screw Compressor without Economizer**

The calculation and selection procedures are almost similar to the System Option-1, except that the oil cooling for the screw compressors are water cooled, portion of the heat for oil cooling is to be deducted from the total heat rejection to the condenser.

The P-H diagram for this Option-5 is same as the Figure 11-4. The refrigerant flow diagram is shown in Figure 11-14.

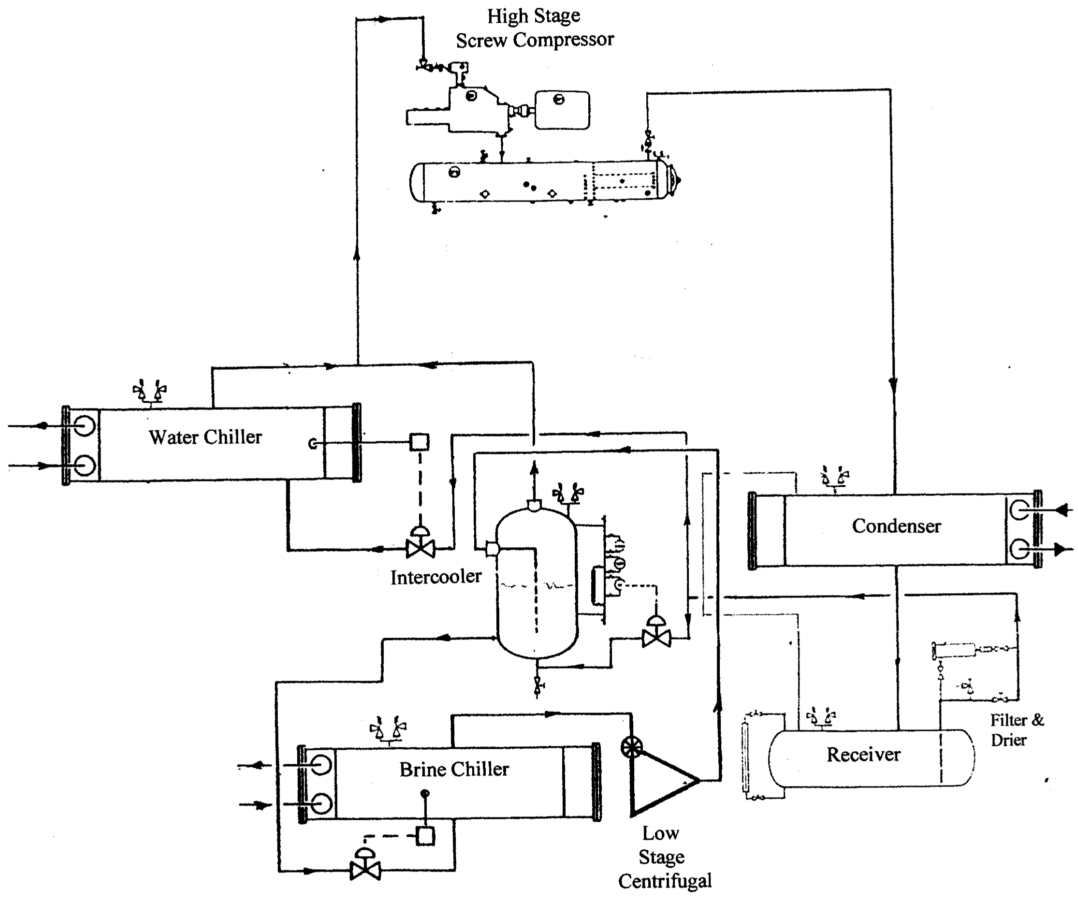

Figure 11-14 Refrigerant Flow Diagram for Compound System Option-5

Conclusion:

System Power Consumption Comparisons					
Option No.	System Description		Compressor Model	BHP	Total BHP
No. 1	Centrifugal Low stage to Centrifugal High Stage	Low Stage	KA-65	432	1,422
		High Stage	KA-65	990	
No. 2	Screw Booster to Screw High Stage	Low Stage	RB-1080	704	1,797
		High Stage	RB-856 with economizer	1093	
No. 3	Centrifugal Low Stage to Screw High Stage	Low Stage	KA-65	432	1,423
		High Stage	RB-676 with economizer	991	
No. 4	Screw Booster to Screw High Stage	Low Stage	RB-1080	704	1,874
		High Stage	RB-856	1170	
No. 5	Centrifugal Low Stage to Screw High Stage	Low Stage	KA-65	432	1,541
		High Stage	RB-856	1109	

For large capacity low temperature compound application, it is better to use a booster compressor which is designed for high volume flow to improve operation efficiency to reduce the power consumption.

Systems of Option No. 1 and Option No. 3 have best power consumption; therefore, system costs should be studied to make comparison. It is believe that the System Option No. 1 should be used because the compressors for the high stage and the low stage are the same model. It is more convenient for service, maintenance and spare parts stocking program if Option No. 1 is used.