Chapter – 11 Partial Load Performance Of Refrigeration System

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Hot Gas Bypass & Liquid Injection:

When the compressor is under surge condition, the operation is unstable. Hot gas bypass is the anti-surge method (see Figure 11-1) by introducing the compressor discharge gas back to compressor suction to avoid surge. Liquid injection is to desuperheat the hot gas down to a lower temperature to prevent over heating the compressor. Liquid for the injection is usually taken from higher pressure such as high pressure receiver, through an expansion device, mix with the hot gas and then return to compressor suction.

The preferred location of hot gas connection is at the inlet of evaporator as shown in the Figure 11-2 for DX evaporator or at the up stream of the suction scrubber such as shown in the Figure 11-3 or far away from the compressor suction.

Manual hot gas bypass might be sufficient if the system is only occasionally under surge condition and the artificial load is rather small. However, automatic control of liquid injection is always recommended for desuperheating the hot gas even the hot gas bypass is manually controlled.

Most sliding vane controls for the screw compressors are for partial load operation down to 10% without surge. Therefore, no hot gas bypass is required, unless the screw compressor is required to be operated less than 10% or down to 0% load.

It is recommended to check the turn-down ratio of a centrifugal compressor to see if hot gas bypass is required for the application. Hot gas bypass and liquid injection should be provided if the system is to be operated down to 0% load.

Compressor Load Vs System Load:

What is the definition of partial load? Is it the percent partial load of the compressor always equal to the percent of the system partial load? Actually, most the cases, the compressor might be operating at partial load already while the refrigeration load is at 100% full load, except at the design point. Most cases, the compressor part load is not the same as the system part load.

The followings are the two extreme cases to illustrate the partial load condition: One is if the system load is always full load at 100%, what will be the compressor load if the ECWT (Entering Cooling Water Temperature) is not controlled? Another case shall be if the ECWT is controlled and it is constant, what compressor head will be when the system is operating at partial load? These two cases will also help to better understand the meaning of (Head)% and (Flow)% in the Annual Energy Consumption Formula as shown in FIG. 4-4.

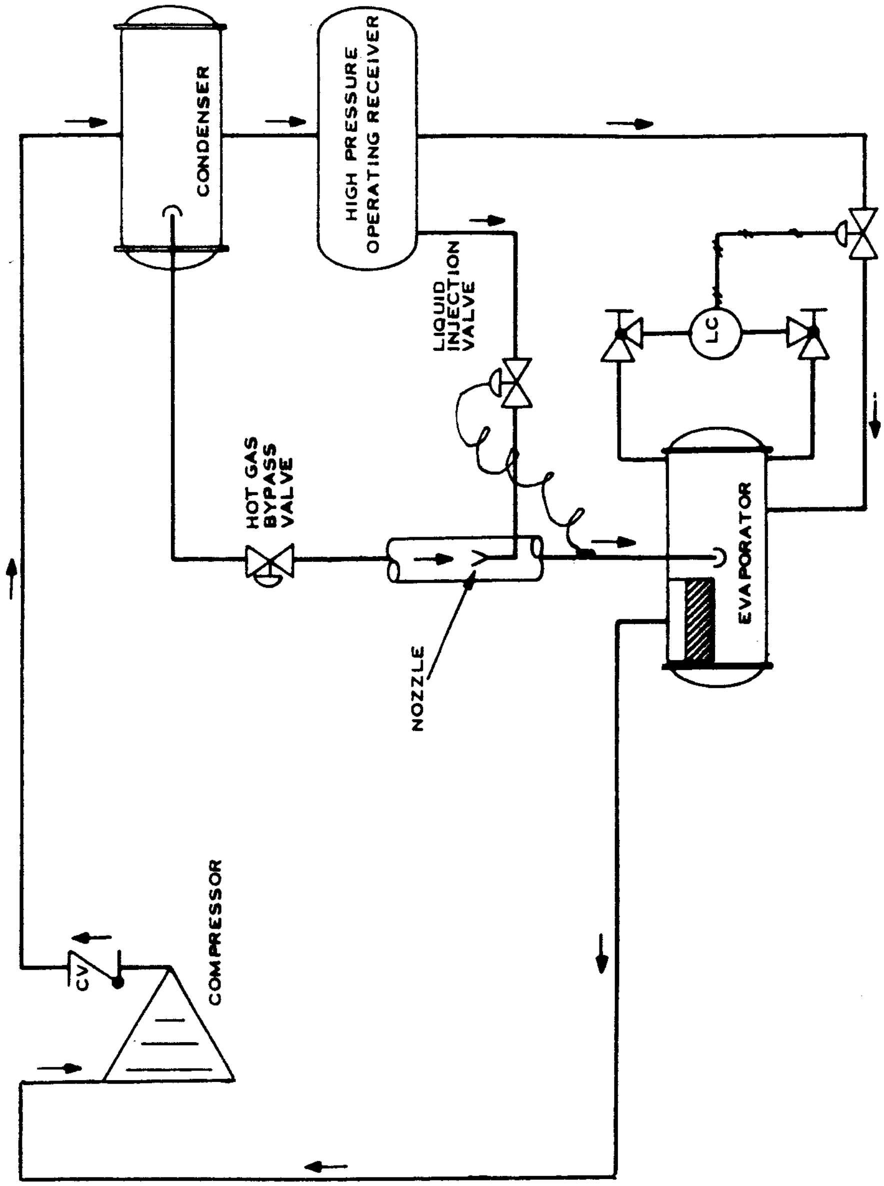


Figure 11-1 Anti-Surge Arrangement
Flooded Evaporator

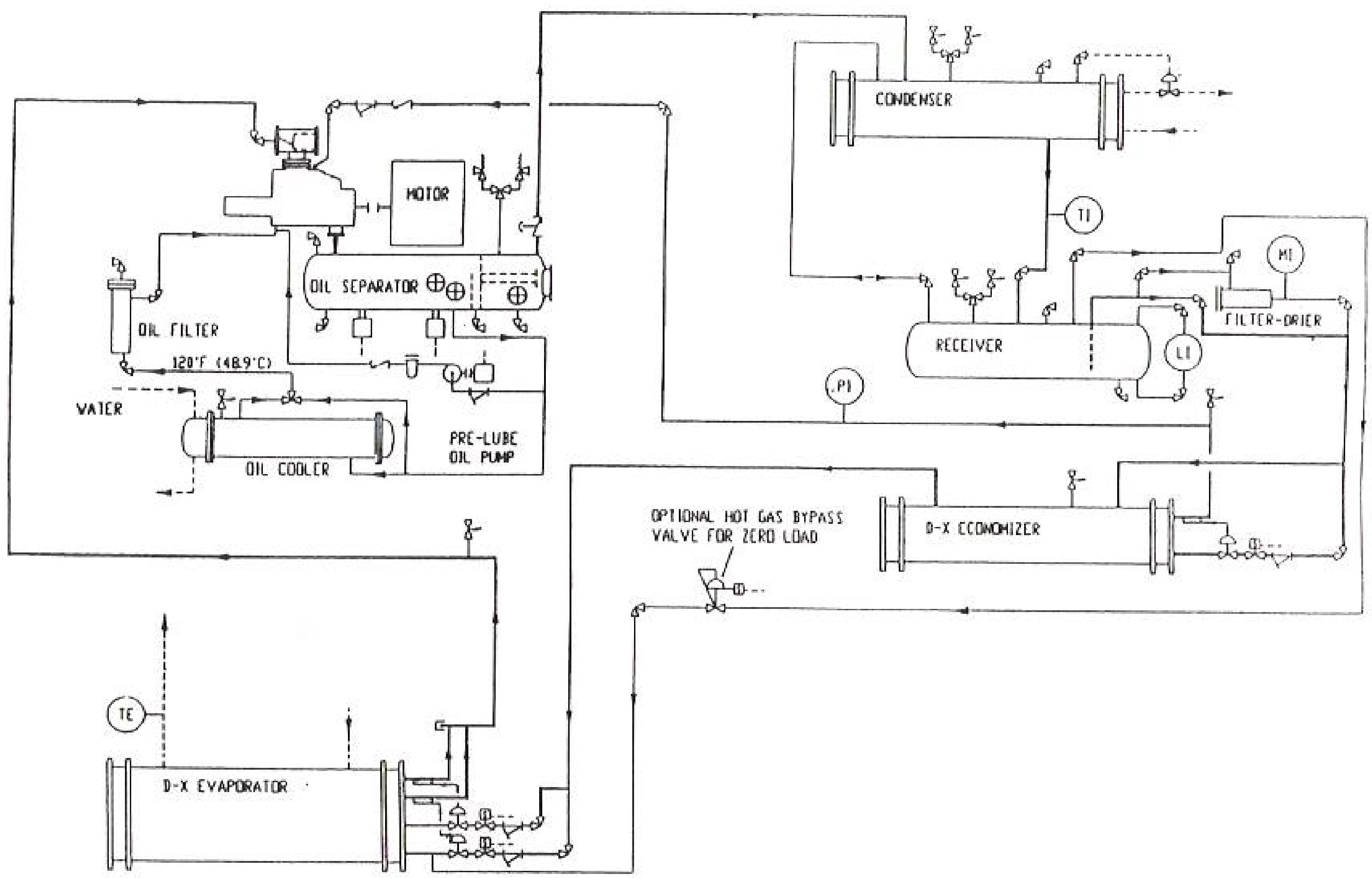


Figure 11-2 Hot Gas Bypass DX Heat Exchanger

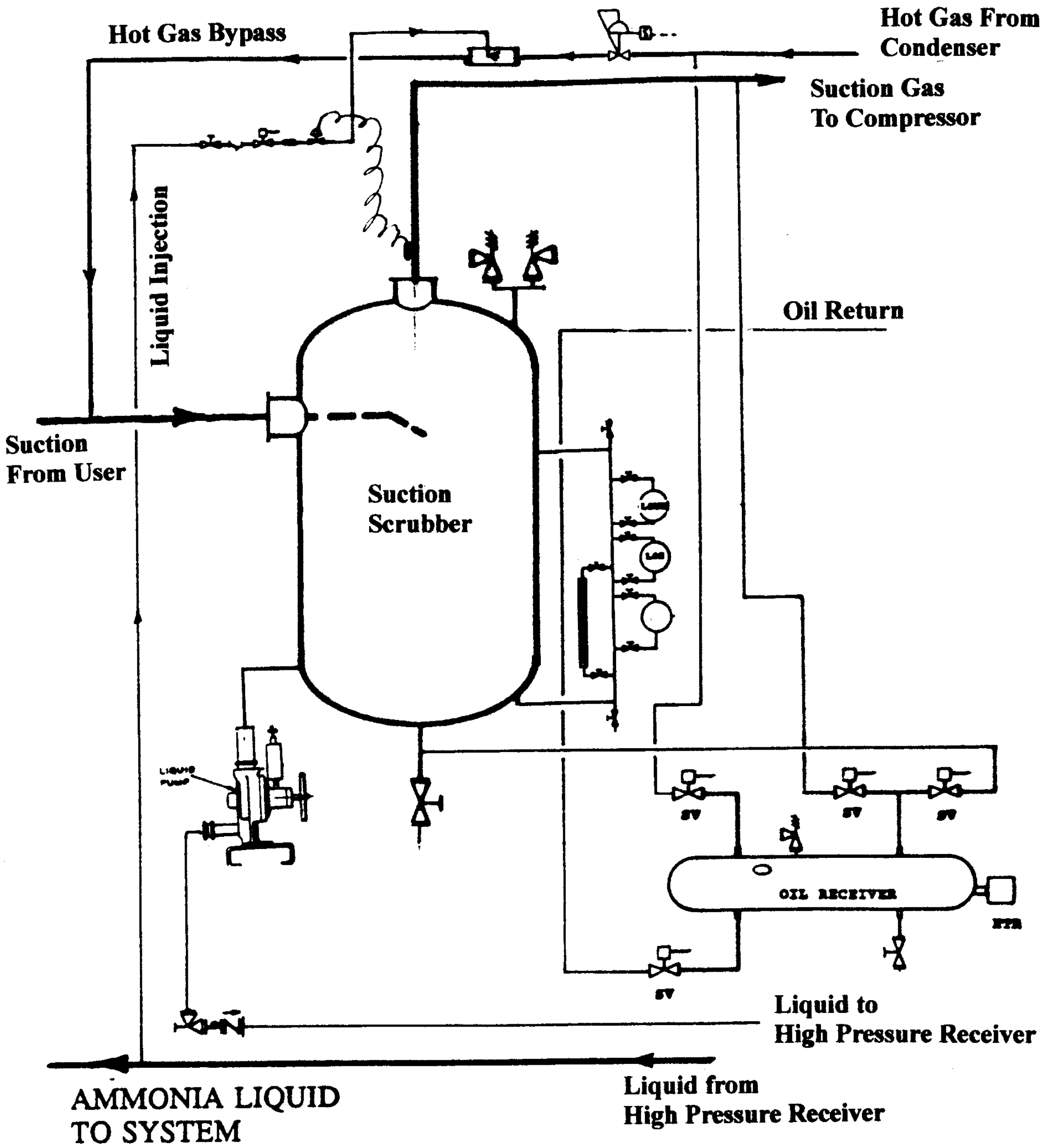
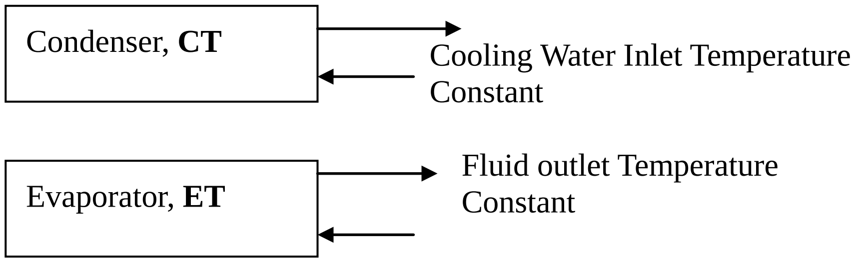


Figure 11-3 Hot Gas Bypass & Liquid Injection

Compressor Head Variation at Variable Load but Constant ECWT

This case is to exam if the ECWT (entering cooling water temperature) is constant and the evaporator fluid outlet temperature is constant, will the CT or ET change or would compression head change when the refrigeration system is in partial load.



(1) Would Condensing Temperature (CT) change?

Figure 11-4 is the typical characteristics and performance curves for the water cooled condenser. The horizontal axis is the Ft^2/Ton; the vertical axis represents the δT which is the Small Difference; it is the overall temperature difference to overcome the heat transfer resistance between the condensing temperature and the condenser leaving cooling water temperature.

$$\delta T = CT - T_w$$

δT = Small Difference, °F
CT = Condensing Temperature, °F
T_w = Leaving Condenser Water Temperature, °F

Therefore: $CT = \delta T + T_w$

<u>CONDENSING TEMPERATURE AT 100% SYSTEM DESIGN LOAD</u>:

Assume the condenser is having 1,500 Ft^2 heat transfer surface area. Heat rejection is 250 Condensing-Ton; cooling water flow is constant, 90°F entering to the condenser and 100°F leaving, 2-P condenser arrangement.

T_w = Leaving Condenser Water Temperature = 100°F
Ft^2/Ton = 1,500/250 = 6.0
From Figure 11-4, the δT = Small Difference = 7.2°F
Therefore:

$$CT = \delta T + T_w = 7.2 + 100 = 107.2°F$$

The condensing temperature is 107.2°F at the design point 100% load.

<u>CONDENSING TEMPERATURE AT 60% SYSTEM PARTIAL LOAD</u>:

Heat rejection at 60% load = 250 x 0.6 = 156 Condensing-ton

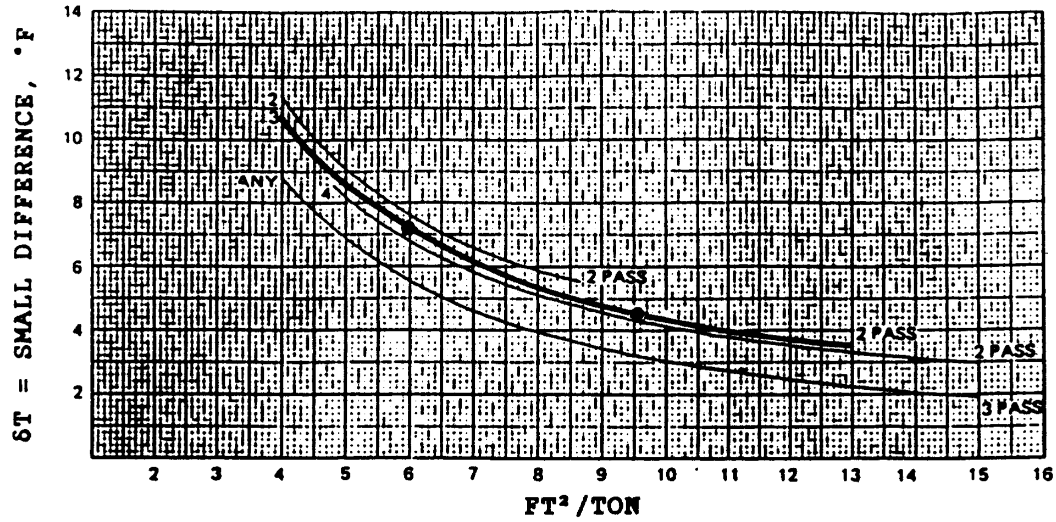


Figure 11-4 Typical Characteristics
Water Cooled Condenser

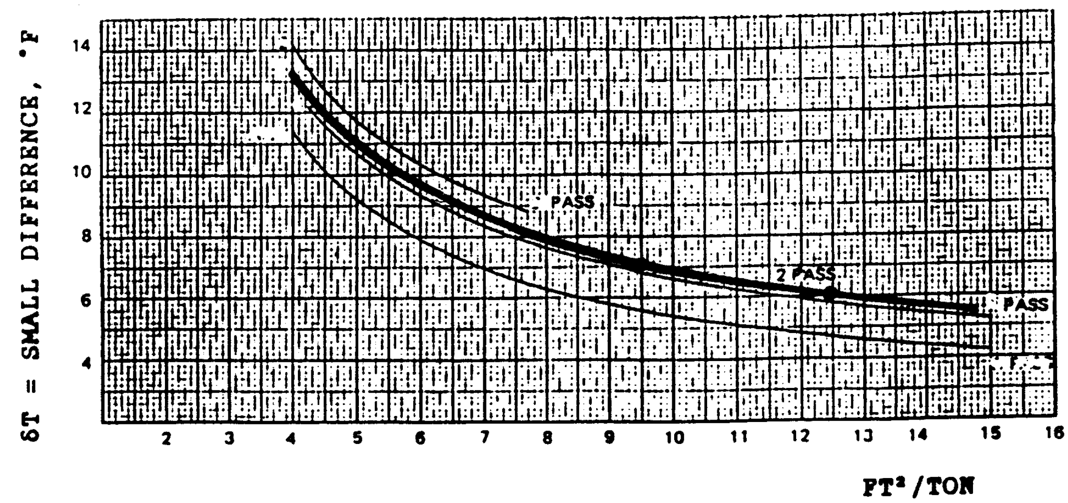


Figure 11-5 Typical Characteristics
Shell-and-Tube Evaporator

$Ft^2/Ton = 1,500/156 = 9.6$
From Figure 11-4, the δT = Small Difference = 4.5°F
Entering cooling water temperature = 90°F (Constant)
Leaving cooling water temperature is now = 90°F + 6°F = 96°F
Therefore:

$$\begin{aligned} CT &= \delta T + T_w = 4.5 + 96 \\ &= 100.5°F \end{aligned}$$

The condensing temperature is now 100.5°F at 60% partial load and it is no longer at 107.2°F as design. Therefore, the compressor head is changed even the entering cooling water temperature entering to the condenser is kept constant at 90°F.

(2) Would Evaporative Temperature (ET) change?

Figure 11-5 is the typical characteristics and performance curves for the water cooler. Water fluid is used for the illustration; the result is the same for brine. The horizontal axis is the Ft^2/Ton; the vertical axis is the δT which is the Small Difference; which is the overall temperature difference to overcome the heat transfer resistance between the leaving chilled water temperature and the evaporative temperature.

$$\delta T = T_w - ET$$

δT = Small Difference, °F
ET = Evaporative Temperature, °F
T_w = Leaving Chilled Water Temperature, °F

Therefore: $ET = T_w - \delta T$

EVAPORATIVE TEMPERATURE AT 100% SYSTEM DESIGN LOAD:

Assume the evaporator cooler is having 1,900 Ft^2 heat transfer surface area. The refrigeration load is 200 TR; leaving chilled water temperature from the evaporator is 40°F. 2-P cooler arrangement is used.

T_w = Chilled Water Leaving Evaporator Cooler = 40°F (Constant)
Ft^2/Ton = 1,900/200 = 9.5
From Figure 11-5, the δT = Small Difference = 7.0°F
Therefore:

$$\begin{aligned} ET &= T_w - \delta T = 40 - 7.0 \\ &= 33\ °F \end{aligned}$$

The design evaporative temperature (ET) is 33°F at 100% refrigeration load.

EVAPORATIVE TEMPERATURE AT 60% SYSTEM PARTIAL LOAD:

Refrigeration load at 60% load = 200 x 0.6 = 120 TR

Ft^2/Ton = 1,500/120 = 12.5
From Figure 11-5 the δT = Small Difference = 6.0°F
Leaving chilled water temperature = 40°F (Constant)
Therefore:

$$ET = T_w - \delta T = 40 - 6 = 34°F$$

The evaporative temperature is now 34°F at 60% partial load and it is no longer at 33°F as design. Therefore, the compressor head is smaller. The ET is always higher than the design during the partial operation.

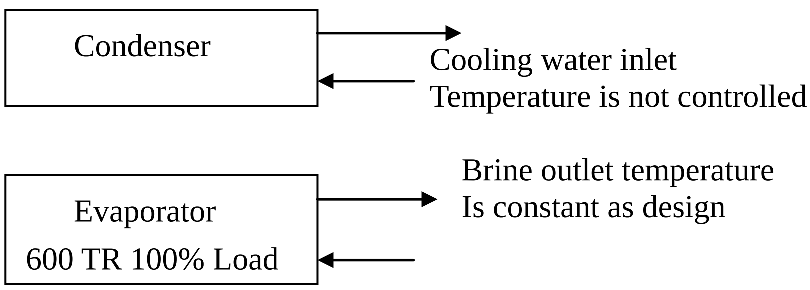
Note: Water cooler is used to simplify the illustration. The contrast will be even greater if brine is used.

(3) Would Compressor Load Change?

Figure 11-6 is a typical performance curves for a compressor. The horizontal axis represents the evaporative temperature (ET) or saturated suction temperature. The vertical axis is the compressor capacity in tons of refrigeration. From the compressor performance curve, at the constant condensing temperature (CT), the same compressor will produce higher TR when the evaporative temperature is higher. At the constant evaporative temperature, the same compressor will have higher TR when the condensing temperature is decreased. Therefore, it is very obvious that during the operation, the same compressor is able to produce higher TR than the design TR if the condensing temperature is lower than the design CT and/or the evaporative temperature is higher than the design ET. Therefore, the compressor is actually running less than 60% partial load while the system load is at 60% partial load.

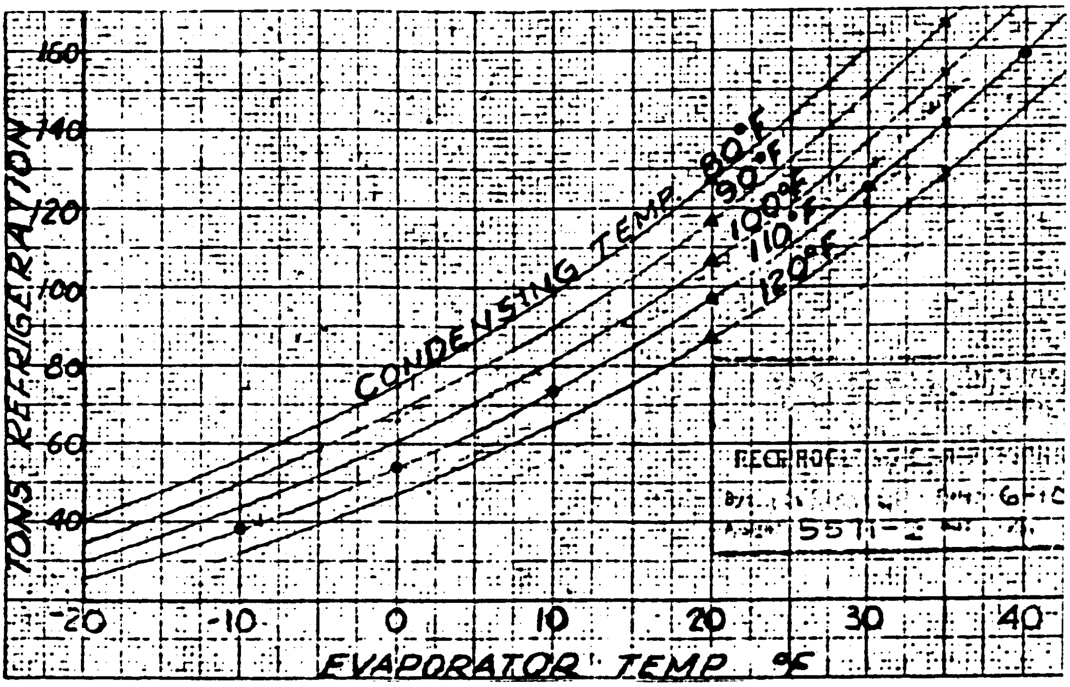
Compressor Partial Load at 100% Full Load with Variable ECWT

This is the case is to exam what compressor loading will be if the system load is always 100% full load while the brine outlet temperature from the evaporator is constant; but, the entering cooling water temperature to condenser is not controlled.



For industrial application, the refrigeration system is required to be operated year round; day and night; spring, summer, fall and winter.

TYPICAL COMPRESSOR CHARACTERISTICS



Constant Condensing Temperature 110°F

Evaporative Temperature	Tons Refrigeration
-10°F	38 TR
0°F	54 TR
10°F	74 TR
20°F	97 TR

Constant Evaporative Temperature 20°F

Condensing Temperature	Tons Refrigeration
120°F	87 TR
110°F	98 TR
100°F	106 TR
90°F	117 TR
80°F	126 TR

Figure 11-6 Typical Characteristics of Compressor

The refrigeration system in this case is an Ethylene Glycol brine chiller.

Design Conditions are:

Refrigerant:		R-22
Refrigeration Load:		600 TR
Brine leaving temperature:		23°F
Brine return temperature:		33.8°F
Cooling water entering temperature:		91.4°F
Fouling factors:	Condenser:	0.002 Ft^2-°F-Hr/Btu
	Cooler:	0.002 Ft^2-°F-Hr/Btu
Condensing temperature:		115°F
Evaporative temperature:		13°F

<u>SUMMER PEAK LOAD OPERATION</u>:

Diagram [1] of Figure 11-7 is the P-H diagram analysis for the refrigeration system during the summer peak conditions:

Refrigeration Load:	600 TR
Brine leaving temperature:	23°F
Evaporative temperature:	13°F
Condensing temperature:	115°F

Compressor head, HD(1) = (126.07 – 107.39) x 778 = 14,533 ft.

Refrigerant Effect, RE(1) = 105.73 – 44.07 = 61.66 Btu/Lb

Refrigerant Flow, FLOW(1) = 600 x 200 / 61.66 = 1,946.16 Lbs/Min

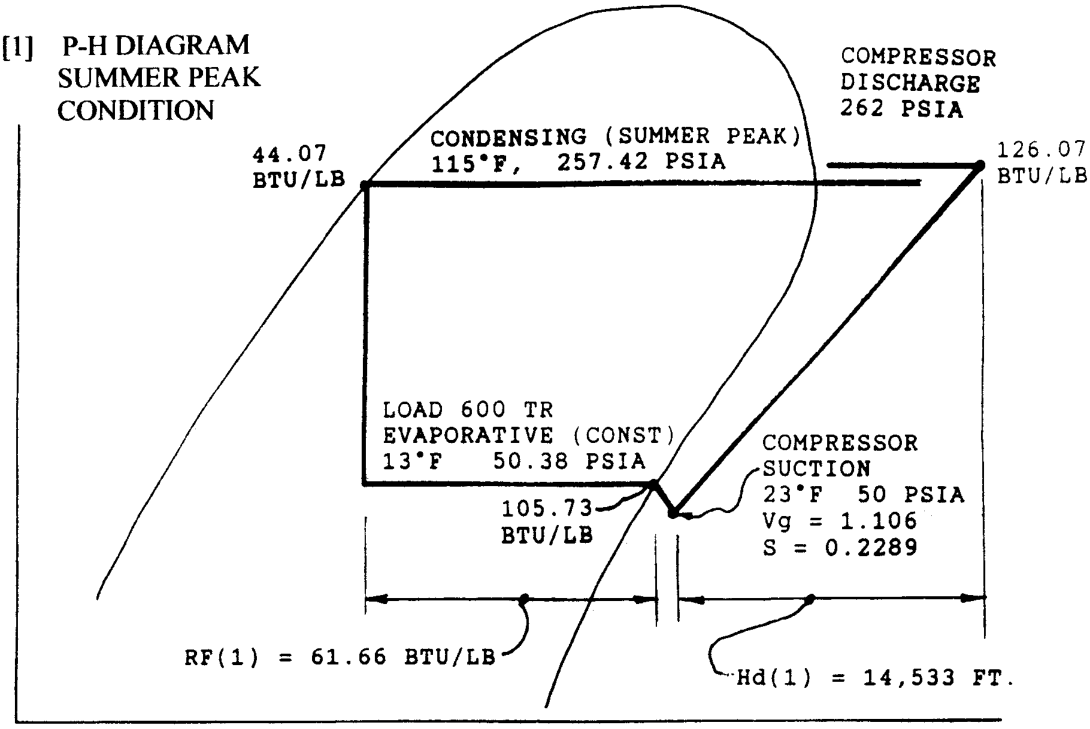
Compressor Suction ACFM, CFM(1) = 1,946.16 x 1.106 = 2,152.44 CFM

<u>OFF-SEASONS AND WINTER OPERATION</u>:

Diagram [2] of Figure 11-7 is the P-H diagram analysis for the same refrigeration system during off-seasons and winter operation. The refrigeration system load is still at 100% 600 TR, but the condensing temperature is lowered to 65°F due to the fact the condenser cooling water temperature is lower during off seasons. The condensing temperature is not to be lower than 65°F to prevent the system pressure difference being too low for the proper function of the refrigeration system operation.

Refrigeration Load:	600 TR (No change)
Brine leaving temperature:	23°F (No Change)
Evaporative temperature:	13°F (No change)
Condensing temperature:	65°F

Compressor head, HD(2) = (117.86 – 107.39) x 778 = 8,146 ft.



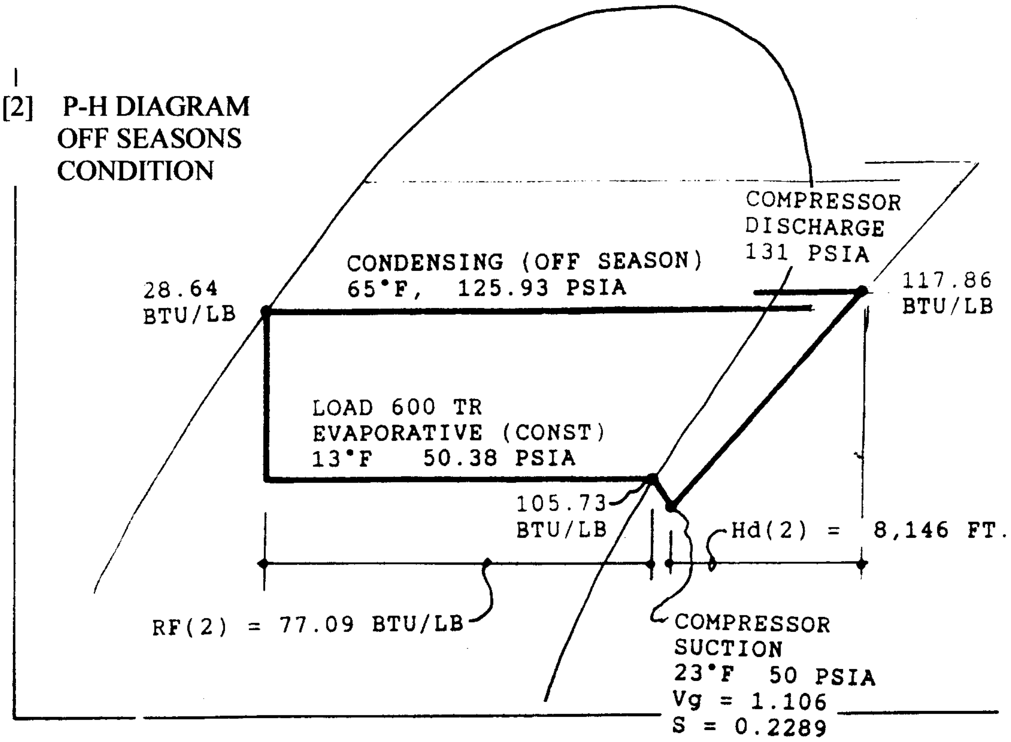


Figure 11-7 Compressor Head and Flow Variations

Refrigerant Effect, RE(2) = 105.73 – 28.64 = 77.09 Btu/Lb

Refrigerant Flow, FLOW(2) = 600 x 200 / 77.09 = 1,556.62 Lbs/Min

Compressor Suction ACFM, CFM(2) = 1,556.62 x 1.106 = 1,721.62 CFM

COMPRESSOR LOAD COMPARISONS:

This case is a hypothetical extreme case; it is just to show that even the refrigeration load is 100%, the compressor load and compression head will change during the operation.

The followings are the summary of this case:

	Summer Peak	Off season
System Capacity	600 TR	600 TR
Evaporative Temperature	13°F	13°F
Condensing Temperature	115°F	65°F
Compression Head	14,533 Ft.	8,146 Ft.
Refrigerant Flow	1,946.16 Lbs/Min	1,556.62 Lbs/Min
Compressor Suction	2,142.44 CFM	1,721.62 CFM

From the above case study, whenever a refrigeration system is designed, it is important to let the cooling water temperature to fall along with the ambient temperature. This feature will help to save annual energy consumption tremendously.