

# Carbon Foot Print and Energy Consumption Considerations for Industrial Refrigeration System Design

- 節能減碳思維與工業冷凍系統設計 -

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What is a Carbon Footprint? The term of “Carbon Footprint” is the total sum of CO<sub>2</sub> (Carbon Dioxide) emissions due to the consumption of fossil fuels or electricity by the activities for the period of a year. If a refrigeration compressor is driven by a diesel engine or gas engine or gas turbine, the energy source is from fossil fuel; if the compressor is driven by an electric motor, the energy source is electricity; it also emits certain amount of CO<sub>2</sub>, depending upon types of the power generating plant.

The approximate Carbon Footprint for various types of electric power generation plants are shown as the following:

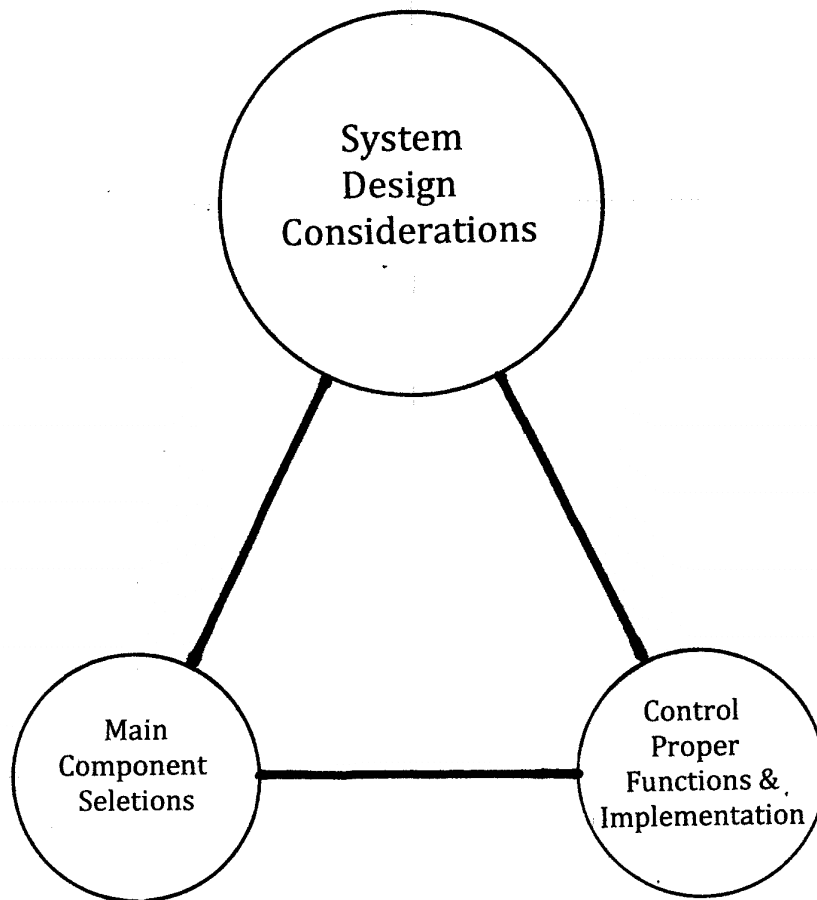
Coal Burning Power Generating Plant:	1,000	g(CO <sub>2</sub> -eq)/Kw-h
Gasification Power Generating Plant:	800	g(CO <sub>2</sub> -eq)/Kw-h
Oil Fired Power Generating Plant:	650	g(CO <sub>2</sub> -eq)/Kw-h
Gas Fired Power Generating Plant:	500	g(CO <sub>2</sub> -eq)/Kw-h
Solar Cell Power Generating Plant:	35~58	g(CO <sub>2</sub> -eq)/Kw-h
Hydroelectric Power Generating Plant:	5~30	g(CO <sub>2</sub> -eq)/Kw-h
Wind Power Generating Plant:	5.25	g(CO <sub>2</sub> -eq)/Kw-h
Nuclear Power Generating Plant:	5	g(CO <sub>2</sub> -eq)/Kw-h

The power generated by so called green energy sources such as solar, wind power and geothermal are actually very small percentage of the total electricity generated for each country; furthermore, the cost of per Kw-h generated by clean energy

source is still commercially unjustifiable except the nuclear power generating plant. The nuclear power was considered as the good clean energy until the unfortunate Fukushima incident in Japan. Unless a technology break-through is obtained for “green” energy power generation, most energy sources for electricity generation today will be still mainly depending upon fossil fuel in the near future. The carbon footprint of any fossil fuel power plant is very high; therefore, the trend of lowering power consumption is emphasized by all the industrial constructions; of course it includes the design, application and fabrication of industrial refrigeration systems.

The Carbon Foot Print of a refrigeration system is actually direct related to the annual energy consumption of the system. In order to lower the Carbon Footprint of a refrigeration system, the design concept must be based on that the system is to be the best efficiency at the design point and also it shall be optimum efficiency for partial load operation.

The refrigeration system design thinking for low energy consumption is generally based on the approaches as the following:



**Figure 1 - Energy Efficient System Design Approaches**

## **Low Carbon Footprint Vs Payback Justification:**

One very important factor when designing a good energy efficient refrigeration system is the economic justification. The initial cost of a lower annual energy consumption system for the sake of low carbon footprint is always costing more. In the practical world, it is not feasible to design a very low carbon footprint refrigeration system, because, the initial cost will be prohibitive which is not justifiable or the payback will not be within the acceptable range by most investors, except by the government for demonstration purpose.

## **System Design Considerations:**

The design approach for low carbon footprint refrigeration system is basically based on the three components as shown in Figure 1:

- System design.
- Main component selections.
- Control and implementation for system proper function.

Those three components cover the system design approaches mainly in the following areas:

### **What system is to be used?**

Single Compression?  
Compound system?  
Cascade system?  
Air cooled ?  
Water cooled?  
Evaporative condenser?

### **What type of compressor is to be used for the system?**

Reciprocating compressor?  
Cylinder unloading?  
Variable speed?  
  
Screw compressor?  
Sliding vane?  
Variable internal volume ratio ( $V_i$ )?  
Variable speed?

Economizing?

Centrifugal compressor?

Single stage or multiple stages with intercooling?

Inlet guide vane control?

Variable speed control?

Combination of pre-rotation vane and variable speed control?

## **Condenser selection:**

The design outdoor temperatures and type of the condenser used determine the design condensing temperature for the refrigeration system. The condensing temperature can be reasonably estimated by using the rule of thumb as the following:

Assuming design outdoor temperature:    95°FDB            80°FWB

Air cooled condenser:             $CT = 95^{\circ}\text{F} + 15^{\circ}\text{F} = 110^{\circ}\text{F}$

Water cooled condenser:         $CT = (80^{\circ}\text{F} + 10^{\circ}\text{F}) + 5^{\circ}\text{F} = 95^{\circ}\text{F}$

Evaporative condenser:         $CT = 80^{\circ}\text{F} + 5^{\circ}\text{F} = 85^{\circ}\text{F}$

Using relatively low design condensing temperature results in lower compression head, therefore, lowering the power consumption at design point for the compressor.

The system with air-cooled condenser or evaporative condenser is always more expensive than the system using water cooled condenser.

A simple step to reduce the power consumption for water-cooled condenser system is to increase the pass arrangement of the condenser. The heat transfer efficiency for the condenser is increased, if the pass arrangement is increased to 3-P or 4-P instead of 2-P. 1-P condenser should not be used unless it is for special design. The water pressure drop for the condenser might be increased in this case; but the energy saving benefit is well worth to consider.

## **Apply system cycle efficiency improvement technique:**

Use intercooling, economizing, liquid subcooling and desuperheating wherever are applicable for the system. These are best tools to improve the system efficiency, to reduce the energy consumption and to reduce the carbon footprint for the refrigeration system.

Economizer of either flash intercooling or liquid subcooling always help improving system cycle efficiency in most cases for screw compressor even single compression.

Intercooler, liquid subcooling and desuperheating improve system efficiency for all compound or multistage centrifugal compressor systems.

### Proper Controls and implementation for system:

Proper automatic control is a must to implement the system function at full load and for partial load operation. The control must be able to take advantages of both reduce head and reduce flow during partial load operation of the system.

In order to keep the refrigeration system function properly, the control shall cover the head control, hot gas bypass and liquid injection if it is required for the system.

### System Overview for Annual Power Consumption

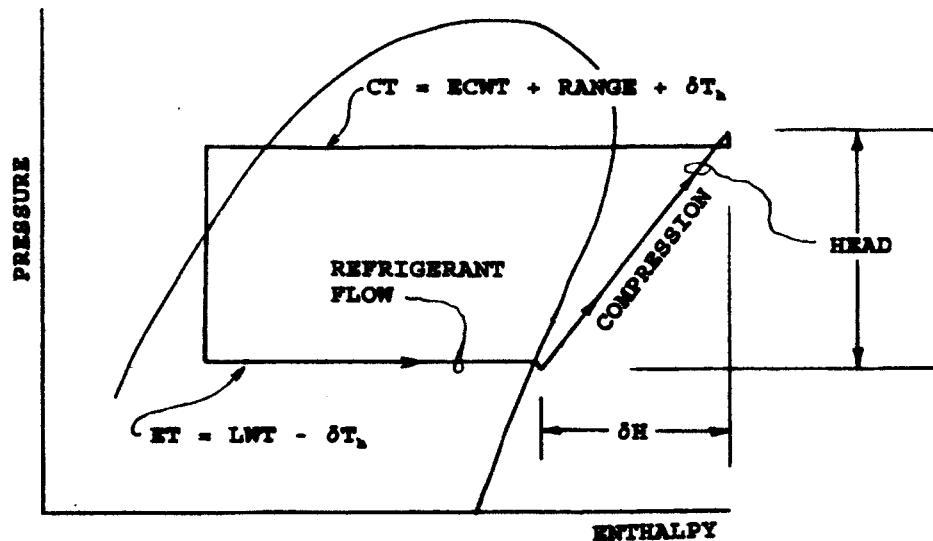


Figure 2 - P-H Diagram Refrigeration Cycle for Energy Efficient

From Figure 2, refrigerant flow for a refrigeration system is determined by Net Refrigerant Effect (NRE); which it is determined by the condensing temperature

(CT) and evaporative temperature (ET) of the refrigeration cycle; the compression ratio influences the head which is also decided by the CT and ET.

For a system using water-cooled condenser the CT is determined by the leaving water temperature plus the small difference; for a shell-and-tube evaporator, the ET is determined by the leaving chilled water temperature minus the small difference. The small difference is the temperature difference required to overcome the heat transfer for the heat exchanger. The small difference is depending on the size and the construction of the heat exchanger.

By selecting different configuration of the condenser or evaporator, the design CT or ET can be changed; when the CT is higher or the ET is lowered, the NRE is smaller, the system efficiency becomes less desirable; the power consumption is higher and the compressor size might be larger; when the CT is lower or the ET is higher, the NRE becomes larger, the system efficiency is more desirable; the power consumption is reduced and the compressor size might be also smaller.

If a screw compressor is used for the system, a flash type or liquid subcooling type economizer can be equipped for the compressor to increase the NRE to reduce the energy consumption, possibly to reduce the size of the compressor.

Increasing the NRE means reduces the refrigerant flow needed for the evaporator to produce same TR for the application; this results in smaller size of compressor and driving motor, furthermore, smaller refrigerant piping, valves, fittings and insulation for the refrigeration system.

## **Outlines of System Design Consideration for Energy Conservation (Lower carbon footprint):**

1.0 System is to be designed to have lower condensing temperature and higher evaporative temperature.

- Better heat transfer in both condenser and evaporator.
- Higher tube velocity to improve heat transfer.
- Use 3-P or 4-P for shell-and-tube heat exchanger instead of 2-P.
- No cooling water regulating valve unless it is required by the system operation or for minimum CT operation.

2.0 Better power consumption overall.

- Use economizer, liquid subcooling, desuperheater or intercooler wherever it is possible.
- Better efficiency of the compressor.

- Better partial load performance of the system.
- Better capacity and head control of the compressor.
- Better efficiency of the main motor.
- Smaller or elimination of oil pump during operation.

Compressor is the heart of the refrigeration system. The character of the compressor greatly influences the performance and the power consumption of the refrigeration system. Therefore, it is important to realize the characteristics of the compressor when making selection of compressor for the system.

- 1.0 Reciprocating compressor – It is a constant volume, variable head machine.  
Step or variable speed capacity control.
- 2.0 Screw compressor – It is a variable volume and variable head machine.  
Sliding vane, variable  $V_i$  and Variable speed control.
- 3.0 Centrifugal compressor – It is a constant head, variable volume machine.  
Variable PRV and variable speed control.

### System Annual Power Consumption:

When designing the industrial refrigeration system, the power consumption, particularly the annual power consumption for carbon footprint consideration should be carefully exam.

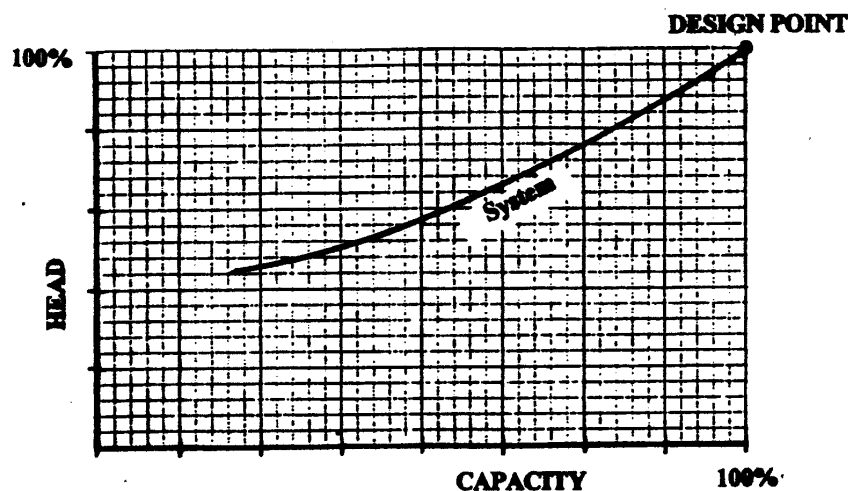


Figure 3 - Typical Partial Load Curve for a Refrigeration System

FIG. 2 shows the relationships between the major components of the refrigeration system. Fig. 3 shows the typical partial load curve; it reflects reduce capacity and reduce head during partial load operation of the refrigeration system.

The power consumption calculation for the compressor and refrigerant flow for the load (TR) at the design point under the design conditions are as the following:

$$\text{Compression Head} = (H_2' - H_1') \times 778 = (\Delta H) \times 778 = \text{Ft.}$$

$$\text{Refrigerant Flow} = \frac{200}{\text{NRE}} \times \text{TR}$$

$$(\text{Compressor HP})_{\text{Design}} = \frac{\text{Flow} \times \text{Head}}{33000 \times \eta}$$

Where:      Flow: Refrigerant Flow, Lbs/Min at design point.  
                  Head: Compression Head, Ft. at design point.  
                   $\eta$  : Compressor Overall Efficiency at design

The power consumption of the compressor unit with bearing friction loss and the oil pump needed for lubrication is as the following:

$$(\text{KW})_{\text{Design}} = \left[ \frac{(\text{Flow})_{\text{design}} \times (\text{Head})_{\text{design}}}{33000 \times (\eta)_{\text{design}}} \right] \times 0.7457$$

$$+ (\text{KW})_{\text{FHP}} + (\text{KW})_{\text{OIL}}$$

Where:  $(\text{KW})_{\text{FHP}}$  Power consumption of Friction bearing loss

$(\text{KW})_{\text{OIL}}$  Power consumption of oil pump



If the overall efficiency of the compressor includes the friction loss and the oil pump design is only for start-up and is not operating during normal operation, then the annual energy consumption estimation of a compressor unit is shown as the following:

$$\text{ANNUAL POWER CONSUMPTION} = (\text{KW})_{\text{DESIGN}} \times \sum_0^{\text{HOURS}} \left[ \frac{\text{HD}\% \times \text{TR}\%}{\eta\%} \right]$$

Where: BHP = Brake HP of compressor shaft input  
Flow = Lbs/Min of refrigerant flow.  
Head = Compression head, ft.  
 $E_R$  = Compressor overall efficiency.  
(FLOW)% = Percent of flow at partial load  
(HEAD)% = Percent of head at partial load.  
( $E_R$ )% = Percent of  $E_R$  at partial load.  
KW = Power consumption at design point.  
HD% = (HEAD)%  
TR% = (FLOW)%  
 $\eta$ % = ( $E_R$ )%

The annual power consumption formula shown above and the Fig. 3 illustrate a very important message, that is:

- (1) The compressor selected should have good efficiency at design.
- (2) The compressor construction should have features for partial load operation.
- (3) The control system designed should be able to take the advantages of both reduce head (HD%) and reduce capacity (TR%) during the partial load operation.

The annual power consumption of the compressor unit is greatly depending on the power consumption of the refrigeration system at the design point (BHP)<sub>design</sub> or (KW)<sub>design</sub>; The annual power consumption also depends greatly on the conditions as how the system is behaved during the partial load operation; the compressor partial load efficiency (  $\eta$  %) is another important factor as shown in the formula as all the refrigeration systems are operated at the partial load most the times.

Compressors for industrial refrigeration are required to operate year round, typical about 6000 to 8000 hours per year; 8000 hours per year operation time is usually used for industrial refrigeration evaluation.

## **Partial Load Operation Analysis - Compressor Load Vs System Load**

What constitutes partial load? What is partial load for the system and what is the partial load for the compressor used for the system? Is it the percent partial load of the compressor equals to the percent of the system partial load?

The process heat load is the system load. The refrigeration system design load is based on the process heat load at 100% under the design operating conditions; the compressor is operated at 100% full load under these conditions. However, the compressor might be operated at partial load if the operating conditions are outside of the design conditions even the system is at full load (100% process heat load).

The followings are the two extreme cases to illustrate the partial load condition. Both cases use brine chiller as example:

Case-1 If the ECWT (Entering Cooling Water Temperature) is controlled and it is constant; the fluid outlet temperature is kept constant; will the compression head (compression ratio) of the compressor change when the system is at partial load?

Case-2 shall be if the system load is always full load at 100% and the leaving brine temperature is constant, will the compressor load change, if the ECWT (Entering Cooling Water Temperature) drops when the ambient temperature drops.

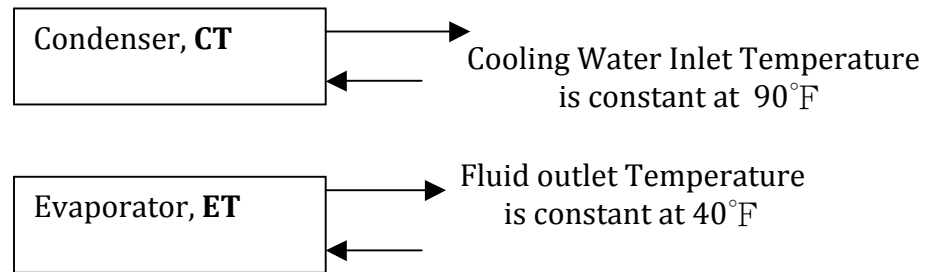
These two cases will help to better understand the meaning of (Head)% and (Flow)% for the compressor in the Annual Energy Consumption Formula listed earlier.

### **Case – 1- ECWT and leaving fluid temperature are constant; what CT, ET and compression ratio will be if system load is at partial load?**

This case is to take a water chiller with water-cooled condenser as an example with the following conditions:

- A. The cooling water flow for condenser is constant.
- B. Entering cooling water temperature is constant.
- C. The chilled water flow in the evaporator is constant.
- D. The leaving chilled water temperature is constant.

The question is asked if the ECWT (entering cooling water temperature) is kept constant at 90°F and the chilled water outlet temperature leaving from evaporator is also being kept at constant at 40°F, will the CT or ET change or will the compression head (ratio) change when the refrigeration system is operated at partial load?



### (1) Will condensing Temperature (CT) change for Case-1?

Figure 4 is the performance curves for the water cooled condenser. The horizontal axis is the  $Ft^2/Ton$ ; the vertical axis represents the  $\delta T$ , which is the Small Difference; the small difference is the overall temperature difference to overcome the heat transfer resistance between the condensing temperature and the condenser leaving cooling water temperature.

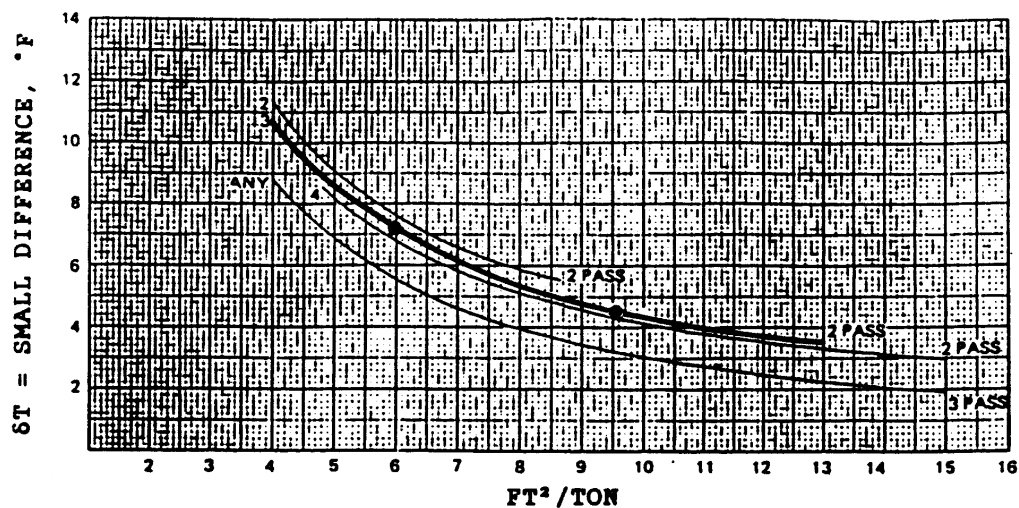


Figure 4 - Performance Curves of Water Cooled Condenser

$$\delta T = CT - T_w$$

$\delta T$  = Small Difference, °F

$CT$  = Condensing Temperature, °F

$T_w$  = Leaving Condenser Water Temperature, °F

Therefore:  $CT = T_w + \delta T$

### **CONDENSING TEMPERATURE AT 100% SYSTEM DESIGN LOAD:**

Assume the condenser is having 1,500 Ft<sup>2</sup> 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 at design; 2-P pass arrangement.

Water temperature range is 10°F at design conditions, 3.0 gpm/Tr

$T_w$  = Leaving Condenser Water Temperature = 90 + 10 = 100°F

Ft<sup>2</sup>/Ton = 1,500/250 = 6.0

From Figure 4, the  $\delta T$  = Small Difference = 7.2°F

Therefore:

$CT = T_w + \delta T = 100 + 7.2$

= 107.2°F

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

### **CONDENSING TEMPERATURE AT 60% SYSTEM PARTIAL LOAD:**

Heat rejection at 60% load = 250 x 0.6 = 156 Condensing-ton  
(Part load efficiency of the compressor is not considered)

Ft<sup>2</sup>/Ton = 1,500/156 = 9.6

From Figure 4, the  $\delta T$  = Small Difference = 4.5°F

Entering cooling water temperature = 90°F (Constant)

Water temperature range is now 6°F instead of 10°F.

Leaving cooling water temperature is now = 90°F + 6°F = 96°F

Therefore:

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

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, mainly because the CT is lower at partial load operation.

## **(2) Will evaporative temperature (ET) change for Case-1?**

Figure 5 is the performance curves for the water cooler. The fluid is water for this illustration; the result is the same if brine was used. The horizontal axis is the Ft<sup>2</sup>/Ton; the vertical axis is the δT, which is the Small Difference; small difference 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<sub>w</sub> = Leaving Chilled Water Temperature, °F

$$\text{Therefore: } ET = T_w - \delta T$$

### **EVAPORATIVE TEMPERATURE AT 100% SYSTEM DESIGN LOAD:**

Assume the evaporator cooler is having 1,900 Ft<sup>2</sup> total 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. The design flow is 2.4 gpm/Tr at design.

$$T_w = \text{Chilled Water Leaving Evaporator Cooler} = 40^\circ\text{F (Constant)}$$

$$\text{Ft}^2/\text{Ton} = 1,900/200 = 9.5$$

From Figure 5, the  $\delta T = \text{Small Difference} = 7.0^\circ\text{F}$   
 Therefore:

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

The design evaporative temperature (ET) is  $33^\circ\text{F}$  at 100% refrigeration load.

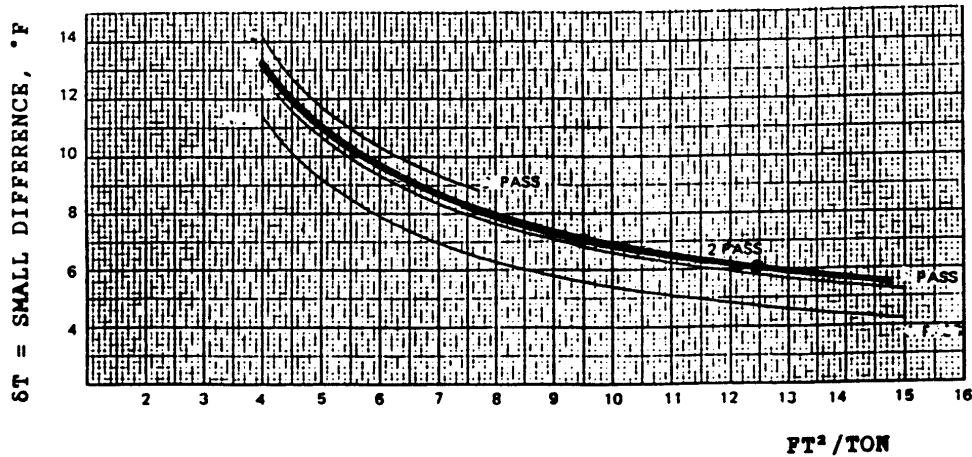


Figure 5 - Performance Curves of Shell-and-Tube Evaporator

**EVAPORATIVE TEMPERATURE AT 60% SYSTEM PARTIAL LOAD:**

System refrigeration load is reduced to 60% load =  $200 \times 0.6 = 120 \text{ TR}$

$$Ft^2/\text{Ton} = 1,500/120 = 12.5$$

From Figure 5 the  $\delta T = \text{Small Difference} = 6.0^\circ\text{F}$

Leaving chilled water temperature =  $40^\circ\text{F}$  (Constant)

Therefore:

$$\begin{aligned} ET &= T_w - \delta T = 40 - 6 \\ &= 34^\circ\text{F} \end{aligned}$$

The evaporative temperature is now 34°F at this 60% partial load and it is no longer at 33°F as design even the leaving chilled water temperature is kept at 40°F. From this example, the fact shall be:

- (a) The ET is always higher than the design ET during the partial operation.
- (b) The compression head of the compressor is smaller because of higher ET.

### (3) Would Compressor Load Change for Case-1?

Figure 6 is a typical performance curves for the compressor. The horizontal axis represents the evaporative temperature (ET) or saturated suction temperature. The vertical axis is the compressor capacity in tons of refrigeration.

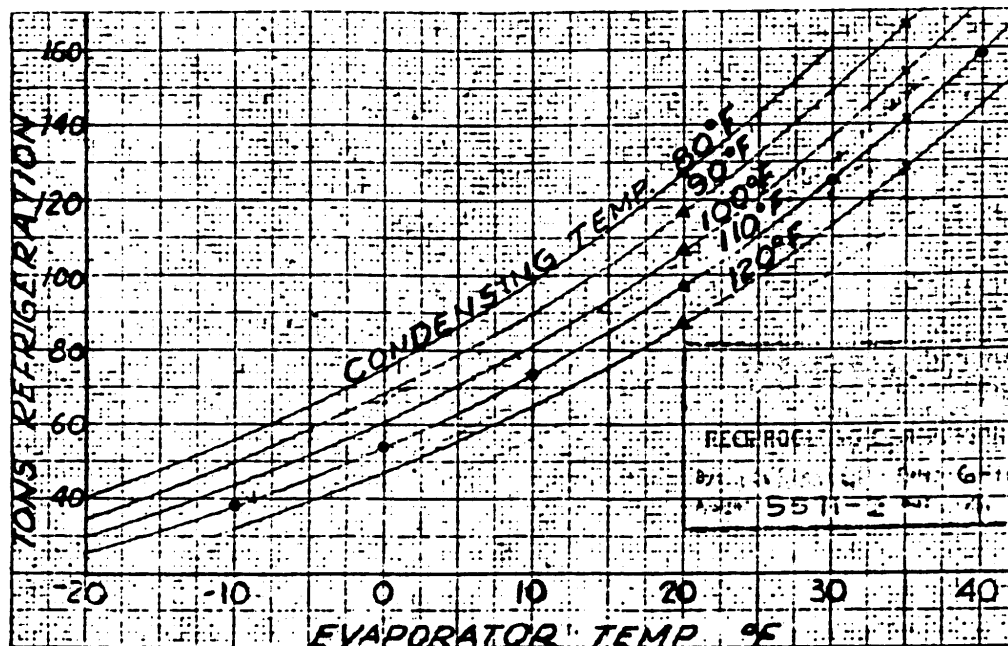


Figure 6 - Typical Performance Curves of a Compressor

From this compressor performance curve shown above, at the constant condensing temperature (CT), the compressor will produce higher TR if the evaporative temperature is higher; the following example listed the TR at various ET at the constant condensing temperature of 110°F:

Constant Condensing Temperature at 110°F	
Evaporative Temperature	Tons Refrigeration
-10°F	30 TR
0°F	54 TR
10°F	74 TR
20°F	97 TR

At the constant evaporative temperature, the same compressor will have higher TR when the condensing temperature is lowered; the following example shows the TR at various CT at constant ET of 20°F:

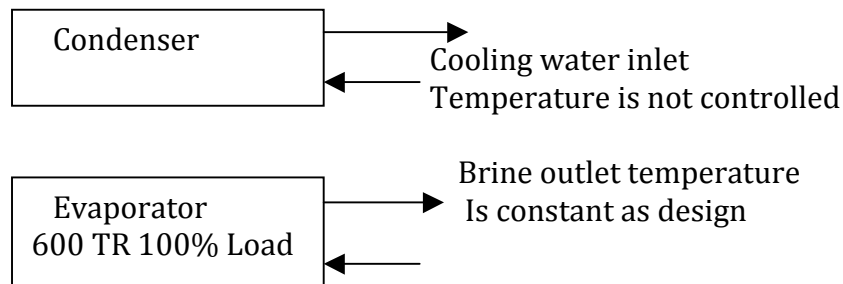
Constant Evaporative Temperature at 20°F	
Condensing Temperature	Tons Refrigeration
120°F	87 TR
110°F	98 TR
100°F	106 TR
90°F	117 TR
80°F	126 TR

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 lowered than the design CT and/or if the evaporative temperature is higher than the design ET. In other word, the compressor is actually running at partial load when the CT is lower and/or ET is higher than the design point. Furthermore, the compressor is actually operating less than 60% of designed load even the system load is at 60% part load.

## **Case – 2 - System load is constant at 100% full load; leaving brine temperature is constant, what the compressor load drops if ECWT drop?**

This Case-2 is to exam what compressor load will be if the system load is always running at 100% full load and the brine leaving temperature from the evaporator is kept constant; however, the entering cooling water temperature to condenser is not controlled, that means the cooling water temperature drops along with the ambient temperature.





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

If the system load is always full load; therefore, the ET in the evaporator is constant in order to produce the constant leaving brine temperature. However, the condensing temperature in the condenser is going to drop when the entering cooling water temperature drops in accordance with the ambient temperature. See the analysis below for design load and off-season operation of the system.

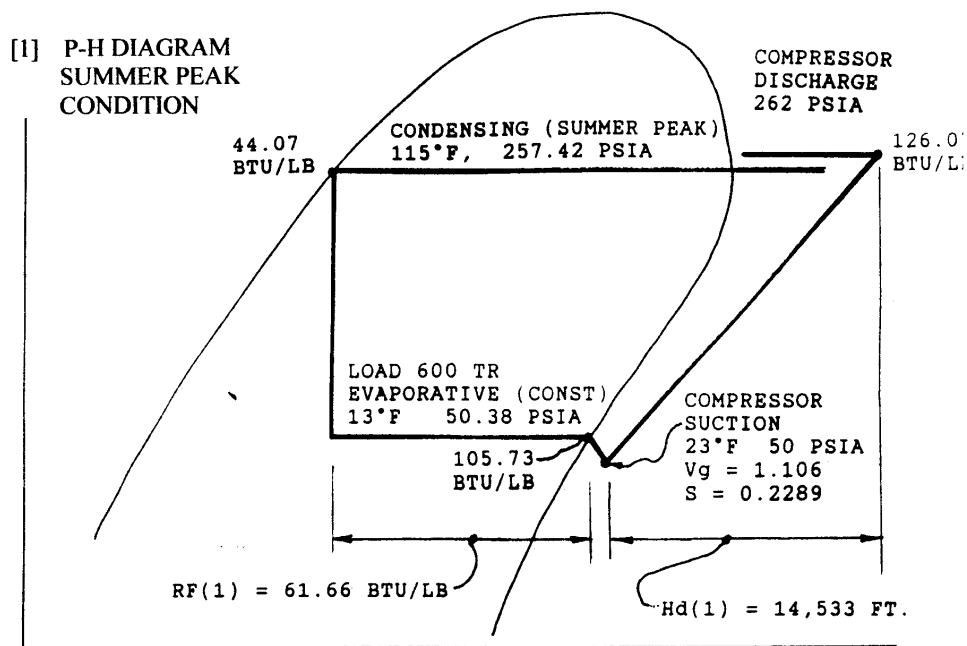
The refrigeration system in this case is an Ethylene Glycol brine chiller, the design Conditions are:

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 <sup>2</sup> -°F-Hr/Btu
Cooler:	0.002 Ft <sup>2</sup> -°F-Hr/Btu
Condensing temperature:	115°F
Evaporative temperature:	13°F

#### **SUMMER PEAK LOAD OPERATION AT DESIGN:**

Figure 7 is the P-H diagram analysis for the refrigeration system during the summer peak design conditions:

Refrigeration Load:	600 TR (Constant)
Brine leaving temperature:	23°F (constant)
Evaporative temperature:	13°F
Condensing temperature:	115°F



**Figure 7 - Compressor Flow and Head Variation**

The compression head, NRE, refrigerant flow and the ACFM are calculated as the following:

Compressor head,  $HD(1) = (126.07 - 107.39) \times 778 = 14,533 \text{ ft.}$

Refrigerant Effect,  $RE(1) = 105.73 - 44.07 = 61.66 \text{ Btu/Lb}$

Refrigerant Flow,  $FLOW(1) = 600 \times 200 / 61.66 = 1,946.16 \text{ Lbs/Min}$

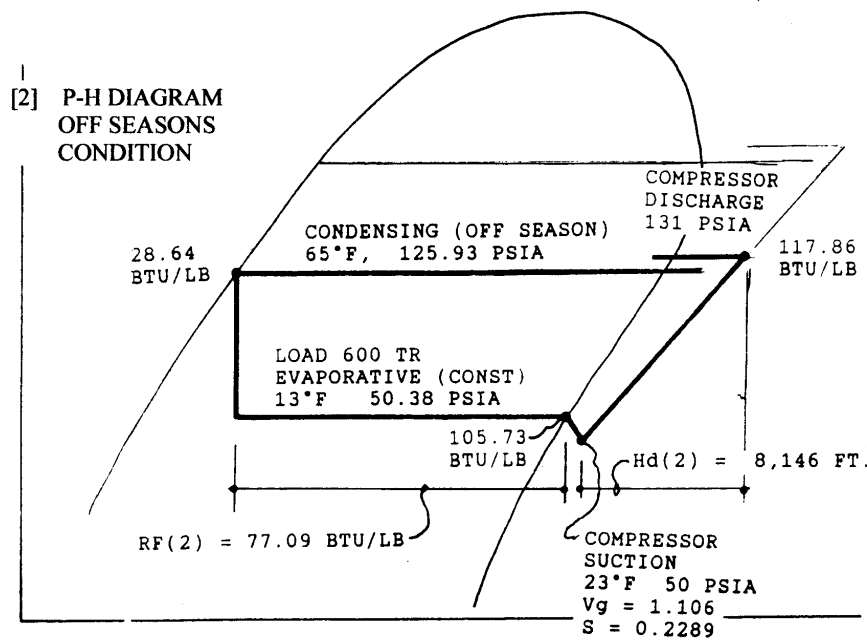
Compressor Suction ACFM,  $CFM(1) = 1,946.16 \times 1.106 = 2,152.44 \text{ CFM}$

### **OFF-SEASONS AND WINTER OPERATION:**

Figure 8 is the P-H diagram analysis for the same refrigeration system during off-seasons and winter minimum head operation. The refrigeration system load is still at 100% at 600 TR, but the condensing temperature is dropping to 65°F due to the fact the condenser cooling water temperature is lowered during off seasons. The condensing temperature is not to be lowered below 65°F to

prevent the system pressure difference being too low for the proper function of the refrigeration system operation.

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



**Figure 8 - Compressor Head and Flow Variations – Off Season Operation**

The compression head, NRE, refrigerant flow and the ACFM for this off-season lowest CT operation are calculated as the following:

Compressor head,  $HD(2) = (117.86 - 107.39) \times 778 = 8,146 \text{ ft.}$

Refrigerant Effect,  $RE(2) = 105.73 - 28.64 = 77.09 \text{ Btu/Lb}$

Refrigerant Flow,  $FLOW(2) = 600 \times 200 / 77.09 = 1,556.62 \text{ Lbs/Min}$

Compressor Suction ACFM,  $CFM(2) = 1,556.62 \times 1.106 = 1,721.62 \text{ CFM}$

### **COMPRESSOR LOAD COMPARISONS AT POINT (1) AND POINT (2):**

This case is to show two extreme operating conditions between the design point and lowest CT during off-season operation; this is to show that even the system refrigeration load is 100%, the “compressor” load and “compression” head will change during the operation.

The followings are the summary of these two conditions:

	Summer Peak	Off season
System Capacity	600 TR	600 TR
Leaving Brine Temp.	23°F	23°F
Evaporative Temp.	13°F	13°F
Condensing Temp.	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

At the system load of 100% at the design point, the compressor flow is 1,946.16 Lbs/Min, the compressor head is 14,533 ft. But, even the load is still 100% at 600 TR, the refrigerant flow to compressor is reduced by 20% to 1,556.62 Lbs/Min and the compressor compression head is reduced about 44% to 1,556.62 ft.

CONCLUSION - From the above two cases, it is obvious that:

(A) At the system full load, the compressor might be running at partial load except at design peak conditions.

(B) The condensing temperature might be lower than design even the system load is at full load.

(C) The evaporative temperature might be higher than design if the system is at partial load.

The above two cases also verify the fact that in order to obtain low annual energy consumption, the industrial refrigeration system designed and the compressor selected should include the features that it is able to take advantage of both reduce flow and reduce head for the compressor during partial load operation.

## **Compressor Performance and Energy Saving Feature**

Energy saving feature for compressor is that the compressor design and construction are able to provide full energy saving capabilities when the system is operated at partial load conditions.

### **Energy Saving Features for Centrifugal Compressor**

Most standard centrifugal compressors are designed for constant speed drive with automatic control of inlet guide vane or Pre-rotation Vane (PRV). PRV is a low cost and yet the effective tool for energy saving operation for centrifugal compressor besides of variable speed control for partial load operation.

Some centrifugal compressor is with suction damper control instead of PRV. Centrifugal compressor without PRV is basically designed for variable speed drive. Sometimes, a suction damper control is provided for the compressor for a constant speed drive application if the compressor design is without PRV; the annual energy consumption is higher by using suction damper instead of PRV.

The best energy saving feature of centrifugal compressor is the combination of automatic variable speed and Pre-Rotation Vane (PRV) control. The typical performance map for the centrifugal compressor with variable speed and PRV control is show in Figure 9.

In the Figure 9, the curve of A-C-B is the compressor design speed at 100%. The point "A" is the design point; Line A-B represents the typical system line at reduce head; the line A-C line represents the system operated at constant head of the compressor.

Line D-E is the surge line of the compressor at various speeds. Area H-D-E-K is for variable speed control with the PRV fully open. In the case if the operating point is located inside the area of D-F-G-E, the compressor shall be speed control with the adjustment of the pre-rotation vane. Some partial load operation might need to use hot gas bypass and liquid injection.

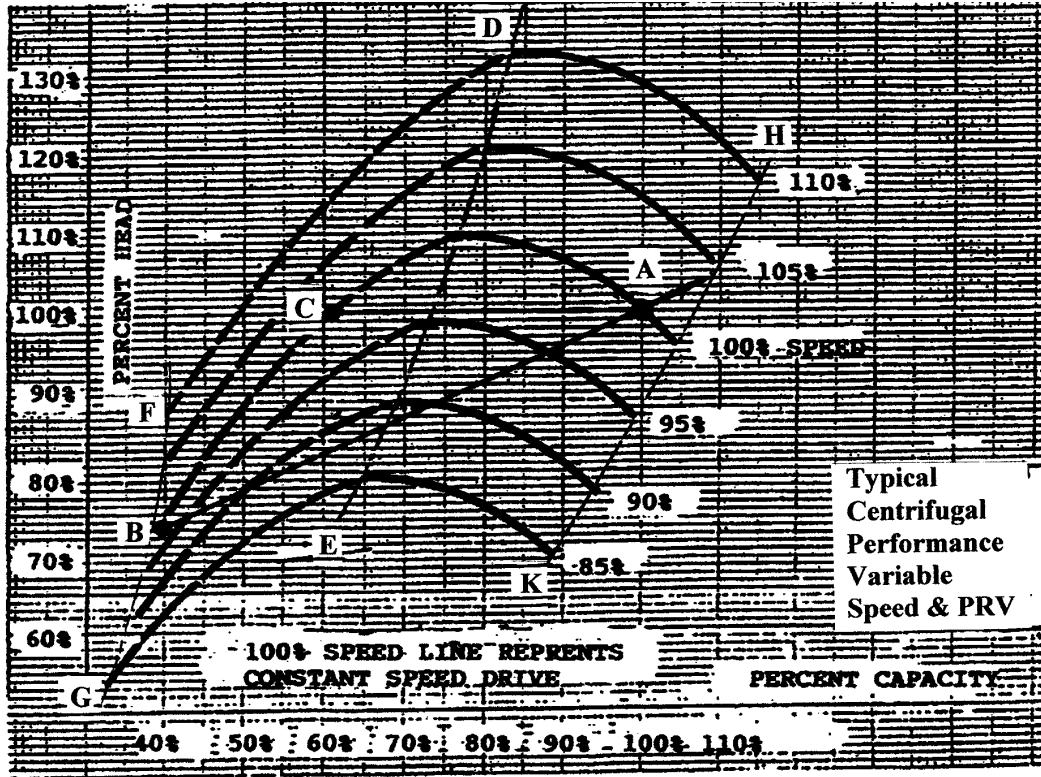


Figure 9 – Typical Performance Map for Centrifugal Compressor with Speed and PRV Control

The variable speed energy saving feature for centrifugal compressor is based on the Centrifugal Affinity Laws as shown in the following formula:

Flow Rate: 
$$Q_2 \approx Q_1 \times \left[ \frac{N_2}{N_1} \right]$$

Head: 
$$H_2 \approx H_1 \times \left[ \frac{N_2}{N_1} \right]^2$$

HP: 
$$HP_2 \approx HP_1 \times \left[ \frac{N_2}{N_1} \right]^3$$

Where:      Q:      Flow rate  
                  N:      Compressor impeller speed  
                  H:      Compression head  
                  HP:     Horsepower

Assuming the percent of efficiency is not changed during the partial load operation, the percent of HP reduction in accordance with the Affinity Law is as the following:

RPM% Change	HP% Reduction
100%	100%
95%	85.7%
90%	72.9%
85%	61.4%
80%	51.2%
75%	42.2%
70%	34.2%

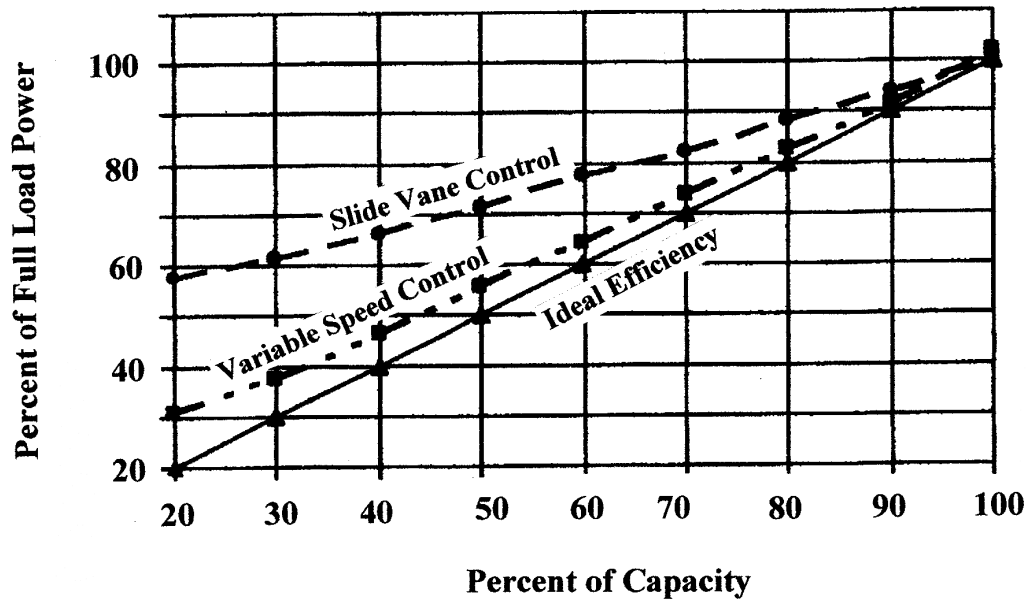
The apparatus of variable speed control for centrifugal compressor is more expensive than pre-rotation vane (PRV) control and it might have maximum HP driving motor size limitation and might be the power supply voltage is also restricted.

## Energy Saving Features for Screw Compressor

The best energy saving feature of screw compressor was used to be the automatic variable internal volume ratio (Vi) with the slide vane mechanism before the available of variable speed drive. With the implementation of variable speed control, the energy saving and partial load efficiency of the screw compressor is now further improved.

Figure 10 shows the typical partial load curves for a screw compressor, these curves are based on the screw compressor for ammonia refrigeration with system design compression ratio of 10.42. From the Figure 10, it shows is the partial load efficiency curve of the variable speed control is much better than the partial load efficiency curve of the compressor equipped with slide vane control; the efficiency

line of variable speed control is closer to the ideal part load efficiency curve as compared to that is having slide vane control.



**Figure 10 – Typical Partial Load Curves for Screw Compressor  
R-717 Compression Ratio 10.42**

The speed control apparatus must be linked together with the automatic variable internal ratio ( $V_i$ ) mechanism for the compressor in order to achieve the objective of better energy saving during the partial load operation; The function of variable speed control is to take advantage of reduced flow at the partial load and the variable internal volume ratio ( $V_i$ ) is to take advantage of the reduced head at the partial load of the compressor; The sliding vane for the compressor is to be at fully opened position during the speed control operation.

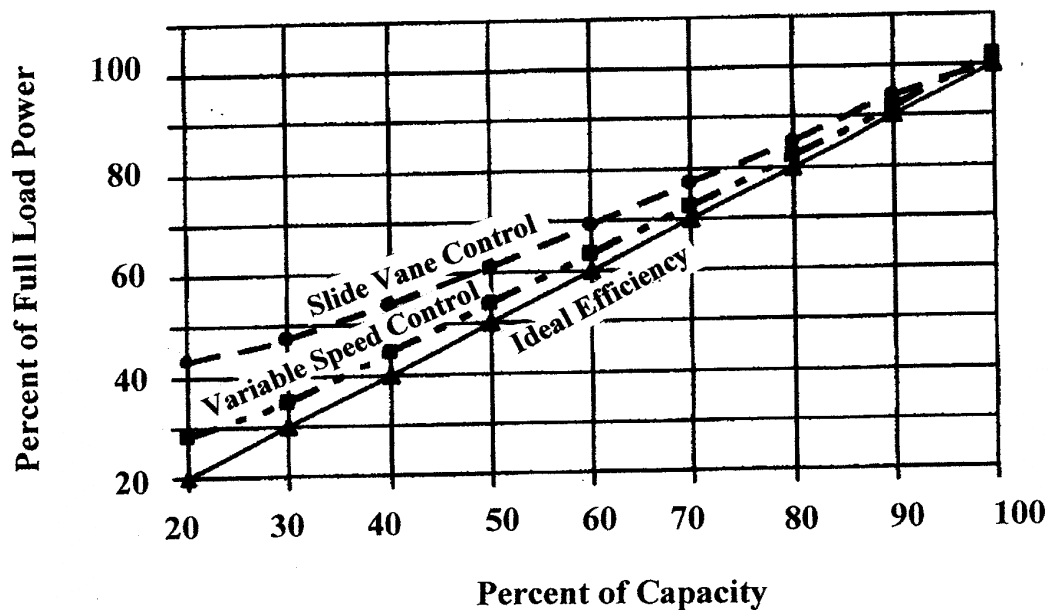
Some screw compressors have minimum operation speed restriction; the sliding valve is to be activated instead of variable speed control for conditions below the minimum speed set for the compressor.

Variable speed control provides even better partial load efficiency if the compressor is for higher compression ratio application. Figure 11 shows a typical partial load curves based on compressor for ammonia refrigerant with design compression ratio of 4.06; As compared to Figure 10, the energy saving of variable speed control compared to sliding vane for this compressor is not as good as the one working at higher compression ratio as shown in Figure 10.



The variable speed option is expensive for the compressor unit; it might be more justifiable to switch the system to compound system instead of single compression if the system design compression ratio is getting too high.

The HP of the driving motor size and the power supply voltage for variable speed control might be restricted.



**Figure 11 – Typical Partial Load Curves for Screw Compressor  
R-717 Compression Ratio 4.06**

## **Energy Saving Features for Reciprocating Compressor**

Reciprocating compressors use for industrial refrigeration system are mostly open drive. The compressors are with cylinder unloading for partial load operation as standard. Some manufacturer offers variable speed drive for energy saving application. Most reciprocating compressors equipped with variable speed control are for partial load operation from 100% down to 10%. Hot gas bypass and liquid injection is to be incorporated with the system if the system is required to be operated below 10%.

Most reciprocating compressor are replaced by screw compressor for most industrial applications. It is suggested to consider screw instead of reciprocating compressor for pay back justification, particularly for larger capacity with variable speed drive.

## Oil Migration in Refrigeration System Increases Carbon Footprint

Another factor that influences greatly the annual energy consumption and carbon footprint of a refrigeration system is the lubrication oil accumulation in the low side.

Lubrication oil in refrigeration system is not avoidable. The oil accumulated over the time is harmful to heat transfer of the evaporator and might reduce the capacity of the refrigeration system.

The heat transfer efficiency is reduced if the oil film accumulated on the heat transfer surface is increased. Figure 12 is the typical diagram showing the relationship between the oil film and the heat transfer coefficient at various oil film accumulations on the tubes.

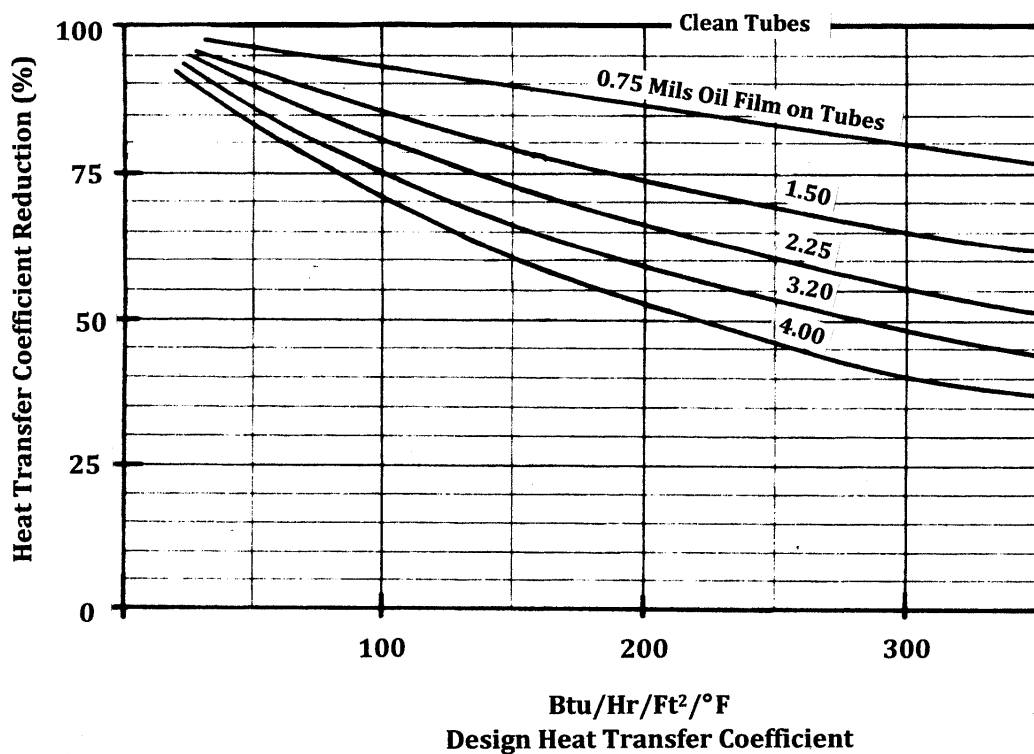


Figure 12 - Influence of Oil Film

From Figure 12, if the heat transfer coefficient of clean tubes of the heat exchanger is 200 Btu/Hr/Ft²/°F at design, the heat transfer coefficient will decrease to 154 Btu/Hr/Ft²/°F or 77% reduction, if the tubes surface is accumulated with a 1.5 Mils oil film thickness after certain period of operation; the heat transfer coefficient is

further reduced to 105 Btu/Hr/Ft<sup>2</sup>/°F or 52.5% reduction, if the oil accumulation in the heat exchanger is increased and the oil film thickness on the tubes is further increased to 4.0 Mils over the times.

The refrigeration capacity of the system will greatly reduce if excess oil is remained in the evaporator as the heat transfer coefficient is reduced; therefore, it is detrimental to refrigeration system performance if the oil is continuously accumulated and is not removed from the low side, particularly from the evaporator. In view of this, the oil in refrigeration system should be managed in such way that:

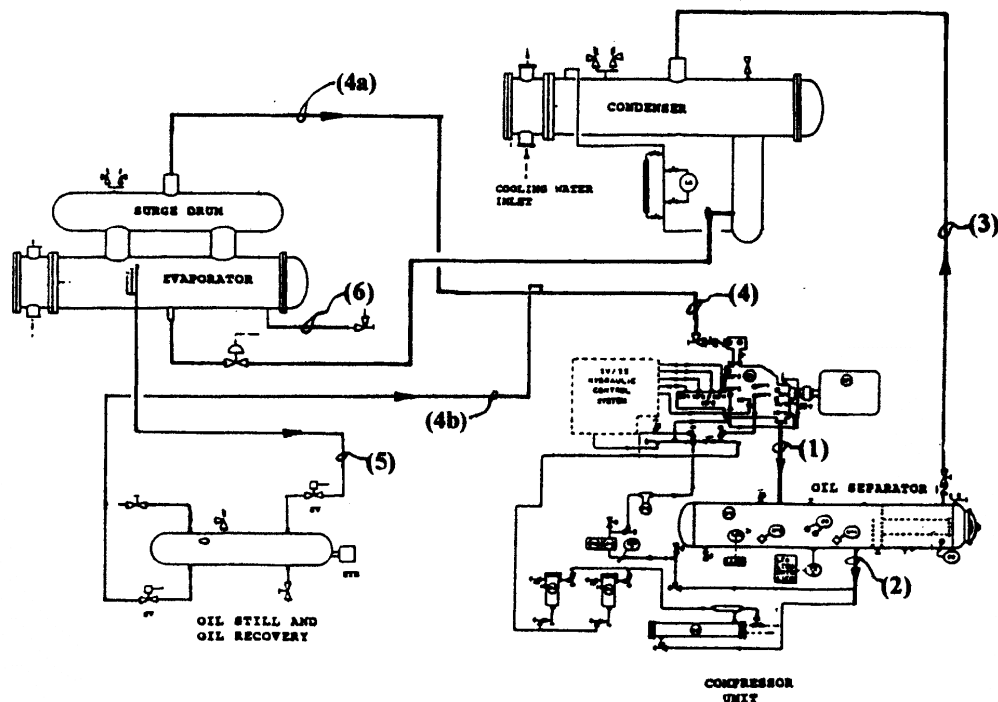
- (A) The oil carryover from compressor discharge is minimized.
- (B) The oil is recovered or removed from the evaporator.
- (C) The oil is returned to compressor or oil reservoir.

## **Oil Migration in Refrigeration System**

The Figure 13 is a typical lubrication oil migration diagram for a screw refrigeration system; the various rates of oil migration are defined as the following:

- 1.0 Oil Pumping Rate: This is the total flow of oil discharge from compressor together with refrigerant gas. The oil carry over from compressor discharge exists in two forms, one is the aerosols; the other form is the oil vapor. Aerosols are fine mists or droplets that can be separated and removed by mechanical filter; oil vapor is in gas form and cannot be removed by mechanical separation.
- 2.0 Oil Separation Rate: This is the amount of oil separated by an oil separator in discharge line and returned to the compressor for compressor cooling and lubrication. Most oil mists are separated by the oil separator, particularly if coalesce filter is used.
- 3.0 Oil Carryover Rate: The amount of oil that goes to condenser. Most of the oil is in the form of vapor after the oil separator. The flow of oil shall be equal to (1) if no oil separator is used.
- 4.0 Oil Return Rate: The oil returns from system to the compressor suction. This flow includes the oil return with the suction refrigerant gas (4a), if the oil is miscible with oil, plus the amount of oil return from oil still (4b).

- 5.0 Oil Recovery Rate: The amount of oil which is recovered from the evaporator by means of an oil still/recovery unit and returned to oil receiver. Some system, the oil is recovered from intercooler if compound system is used.
- 6.0 Oil Draw-off Rate: The amount of oil is drawn off from evaporator and discarded. Some ammonia system, the oil is drawn off from the condenser.
- 7.0 Oil Make-up Rate: Amount of oil is added to the oil reservoir in the oil separator.



**Figure 13 – Oil Migration in Refrigeration System**

First of all, the oil should be filtered and separated out as much as possible from the compressor discharge by using a good efficiency oil separator; the oil carry over rate should be kept as small as possible for the system; secondly, the oil should be recovered from the evaporator and returning to the oil receiver periodically. It is suggested that a good automatic oil management system such as oil still and oil return unit is to be provided for all the evaporators in the refrigeration system, that the oil return rate is equal to the oil carry over rate during entire normal operation period of the refrigeration system.

The initial cost of the oil still and oil return unit for a refrigeration system is relatively small and it is well justifiable by the reasons of annual energy saving and the performance assurance for the refrigeration system.

## Oil Carry Over by Compression

There is a total misconception that oil from the compressor discharge can be completely eliminated by a good oil filter-separator that no oil would be carried over to the evaporator. This is only true if the oil separator is a combination of mechanical and chemical eliminator. A mechanical-chemical type oil filter is expensive; it is usually not used for refrigeration system, unless it is required by special purpose.

The best mechanical oil separator used today even with 100% separation efficiency only can eliminate all the mists and droplets from discharge gas of the compressor, oil carryover still exists in the form of oil vapor when leaving from the oil separator.

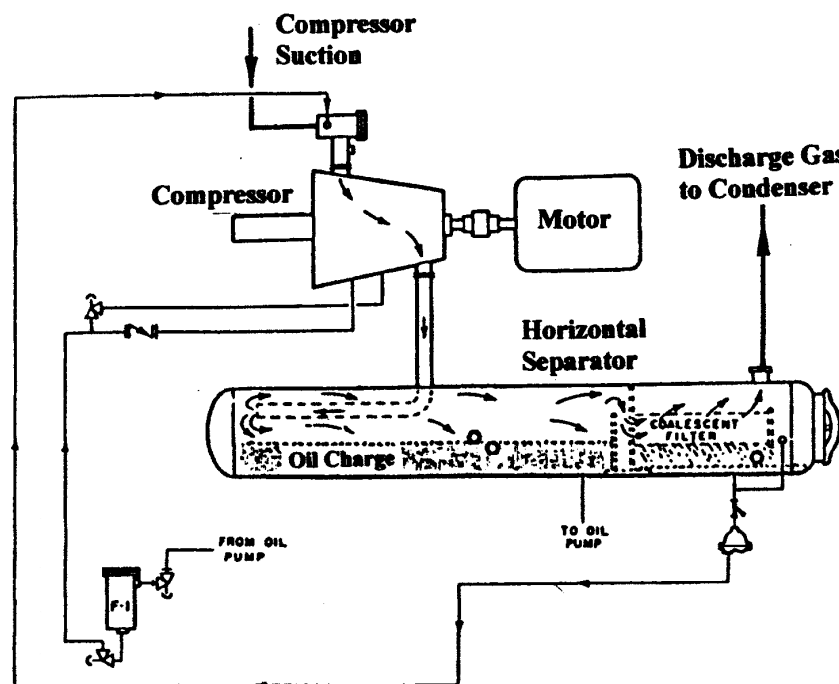


Figure 14 - Discharge Oil Separator and Filter

## Oil Vapor Carry Over By Compression

Assuming the oil separator used for the compressor unit is able to eliminate all the mists and droplets of the oil; but the oil in form of gas/vapor is still going through the mechanical oil filter/separator (see Figure 14) to the condenser and then to the evaporator. The amount of oil vapor carry over in the outlet of the separator depends on the type or brand of oil, type of refrigerant, discharge pressure and discharge temperature. The oil carryover in vapor form can be calculated by the following method:

Gas Law:

$$P \times V = W \times R \times T$$

- P = Gas pressure, Psia
- W = Weight of Gas, Lbs
- R = Gas Constant
- V = Volume, ft<sup>3</sup>
- T = Temperature, (460 + °F)

$$P_o \times V = W_o \times R_o \times T$$

$$P_g \times V = W_g \times R_g \times T$$

- P<sub>o</sub> = Oil vapor pressure, Psia
- W<sub>o</sub> = Weight, Oil, Lbs
- R<sub>o</sub> = Gas Constant of Oil
- P<sub>g</sub> = Refrigerant vapor pressure, Psia
- W<sub>g</sub> = Weight, Refrigerant, Lbs
- R<sub>g</sub> = Gas Constant of Refrigerant
- V = Volume, ft<sup>3</sup>
- T = Temperature, (460 + °F)

$$\frac{P_o \times V}{P_g \times V} = \frac{W_o \times R_o \times T}{W_g \times R_g \times T}$$

$$R = \frac{1545}{MW}$$

R = Gas Constant  
 MW = Molecular Weight

$$R_o = \frac{1545}{MW_o}$$

$$R_g = \frac{1545}{MW_g}$$

MW<sub>g</sub> = Molecular Weight of the Refrigerant Vapor  
 MW<sub>o</sub> = Molecular Weight of the Oil Vapor

$$\frac{P_o \times V}{P_g \times V} = \frac{W_o \times R_o \times T}{W_g \times R_g \times T} = \frac{W_o \times MW_g \times T}{W_g \times MW_o \times T}$$

$$\frac{P_o}{P_g} = \frac{W_o}{W_g} \times \frac{MW_g}{MW_o}$$

$$\frac{W_o}{W_g} = \frac{P_o}{P_g} \times \frac{MW_o}{MW_g}$$

$$\begin{aligned} \text{Oil Vapor Carry Over} \\ \text{In PPM} &= \frac{P_o}{P_g} \times \frac{MW_o}{MW_g} \times 10^6 \end{aligned}$$

Gas pressure is the system pressure, therefore:

$$P_g \cong P_s$$

$$\text{Oil Vapor Carry Over In PPM} = \frac{P_o}{P_s} \times \frac{MW_o}{MW_g} \times 10^6$$

If the oil vapor pressure is in mm Hg instead of Psia, the formula is as the following:

$$\text{Oil Vapor Carry Over In PPM} = \frac{VP_o}{P_s} \times \frac{14.7}{760} \times \frac{MW_o}{MW_g} \times 10^6$$

$VP_o$  = Oil vapor pressure, mm Hg

The Figure 15 shows the relationships of oil vapor (gas) carry over between refrigeration load, compressor speed, evaporative temperature and condensing temperature of the refrigeration system.

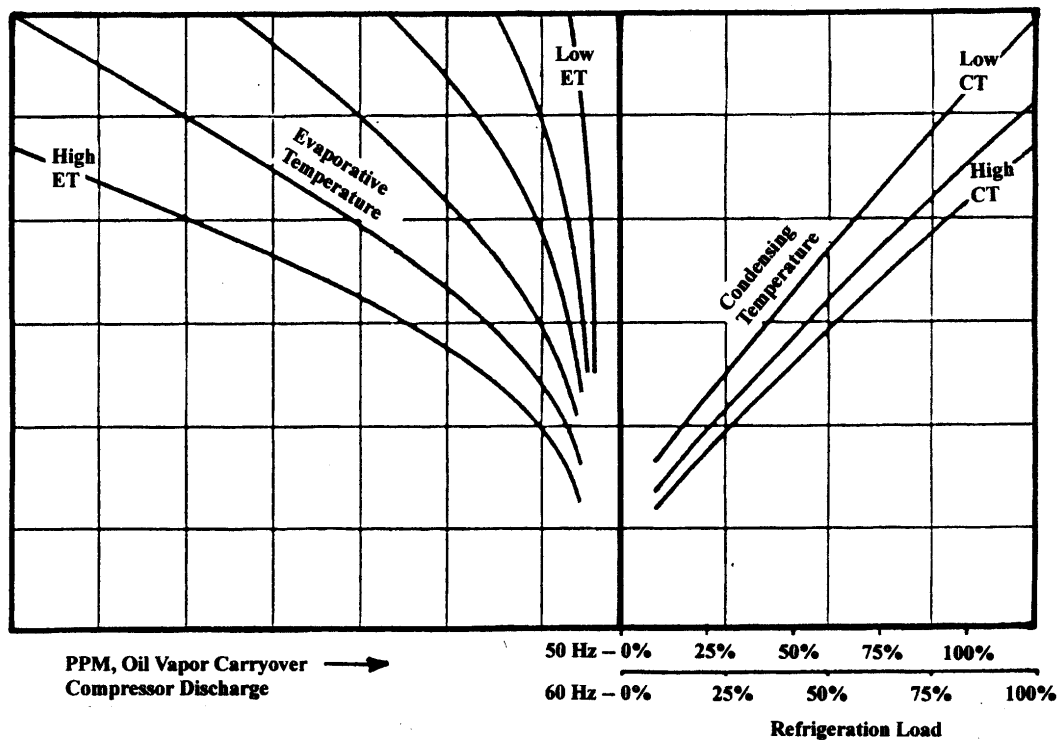


Figure 15 – Oil Carry Over Rate



From the above ideal gas formula and the chart, oil vapor carryover rate varies at various discharge temperatures even with same condensing temperature and the oil separator is having 100% separation efficiency, that all droplets and mists are removed. The oil vapor carryover rate is lower if the discharge temperature is lower.

## Theoretical Oil Vapor Carryover Rates

Ammonia (R-717), propylene (R-1270) and R-22 are the most commonly used refrigerant for industrial process refrigeration application. The oil vapor carryover amount is shown below for these refrigerants at various discharge temperatures:

Refrigerant	Ammonia (R-717)	Propylene (R-1270)	R-22
Molecular weight, gas	17.03	42.09	86.68
Molecular weight, oil	305	305	305
System pressure, 105°F CT	228.9 Psia	242.41 Psia	225.45 Psia
At discharge temperature 250°F	128.6 PPM	49.15 PPM	25.72 PPM
At discharge temperature 200°F	19.6 PPM	7.52 PPM	3.03 PPM
At discharge temperature 150°F	2.0 PPM	0.75 PPM	0.39 PPM
At discharge temperature 100°F	0.2 PPM	0.06 PPM	0.03 PPM

## Amount of Vaporized Oil Carryover to Evaporator

The oil carry over rates are calculated for R-22 and R-717, based on a single stage screw compressor flooded refrigeration system, 100°F condensing temperature, 10°F evaporative temperature; the refrigeration full load is 50 TR at design operating conditions. Assuming the oil separator is with 100% separation efficiency, all droplets and mists are removed. The oil vapor carryover rates and amount for 30 days operation for this refrigeration system are shown in the following table:

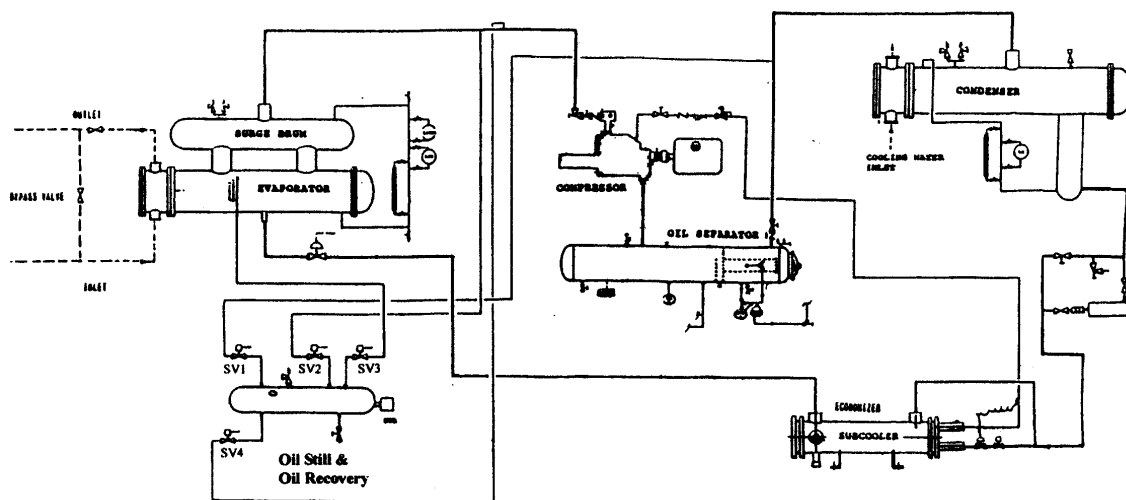
Refrigerant	Refrigerant Flow, Lbs/Hr	Oil Vapor Carryover, Lbs/Hr	If for 30 days, 24 hours per day. Oil Vapor carryover, Lbs	Approximate Oil Carryover in Gal of Oil at Sp.Gr 0.93
R-22	9,066.87	0.04633	33.36	3.55
R-717	1,305.20	0.01779	12.81	2.58

## Oil Still and Oil Recovery Unit for Energy Saving

As explained previously, a large amount of oil will be accumulated in the evaporator after a certain period of operation. The refrigeration capacity of the evaporator will be reduced if the oil is not removed from low side, because the system will be forced to operate at a lower ET and possibly at a higher CT to maintain the refrigeration load if the compressor and motor provided for the system are large enough; the system will result in higher energy consumption and higher carbon footprint under this undesirable situation.

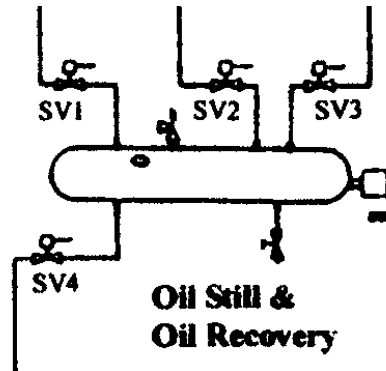
Oil-free type compressor is not feasible for use in refrigeration system; furthermore, the energy consumption and the initial investment is too high.

If the oil used is completely miscible with refrigerant, oil is automatically returned with refrigerant to compressor suction; in this case, no separate oil return unit is required. However, if the oil is not miscible with the refrigerant, oil still with a forced oil return unit should be provided for the system; this oil still and oil recovery unit is shown in Figure 16.



**Figure 16 - Refrigeration System with Automatic Oil Still and Oil Return Unit**

The oil recovery unit consists of an oil still/receiver with heater and operating valves is shown in Figure 17. The suggested piping connection for this oil recovery unit is shown in Figure 16.



**Figure 17 - Oil Still and Oil Recovery System**

The operation sequence of the oil still and oil recovery is as the following:

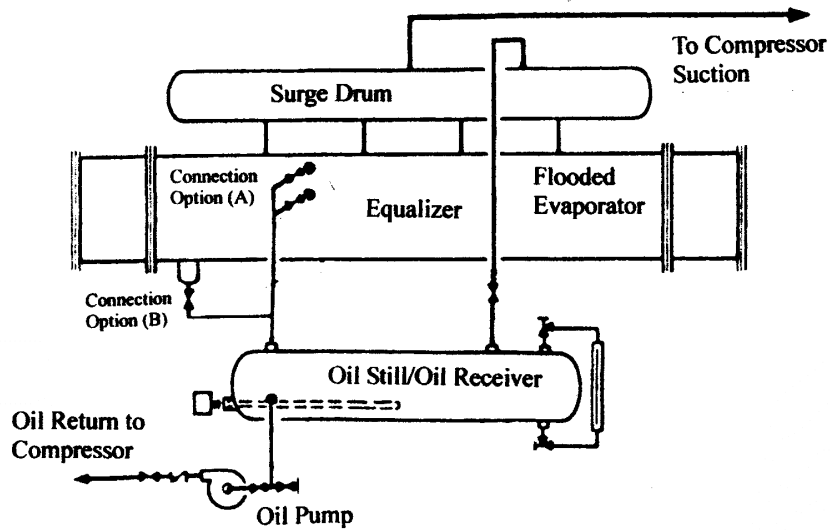
Refrigerant liquid and oil sample flow through SV3 to the oil receiver. Equalizing line SV2 is open; SV1 and SV2 are closed. The refrigerant liquid in the oil receiver is boiled off by the oil heater, the gas is returned to compressor suction via SV2. Oil return function is achieved by closing the SV2 and SV3; open SV1 and SV4; high pressure gas enters into the oil receiver through SV1, presses the oil in the oil receiver and oil is returned to compressor suction through SV4.

This oil return system can be automatically operated by a timer or by level switches.

Screw compressor is able to take a steady small amount of oil flow in the suction. For reciprocating compressor or large amount of oil flow, it is suggested that the oil is to be returned to crankcase for reciprocating compressor or to oil separator/reservoir for screw compressor instead of suction of the compressor.

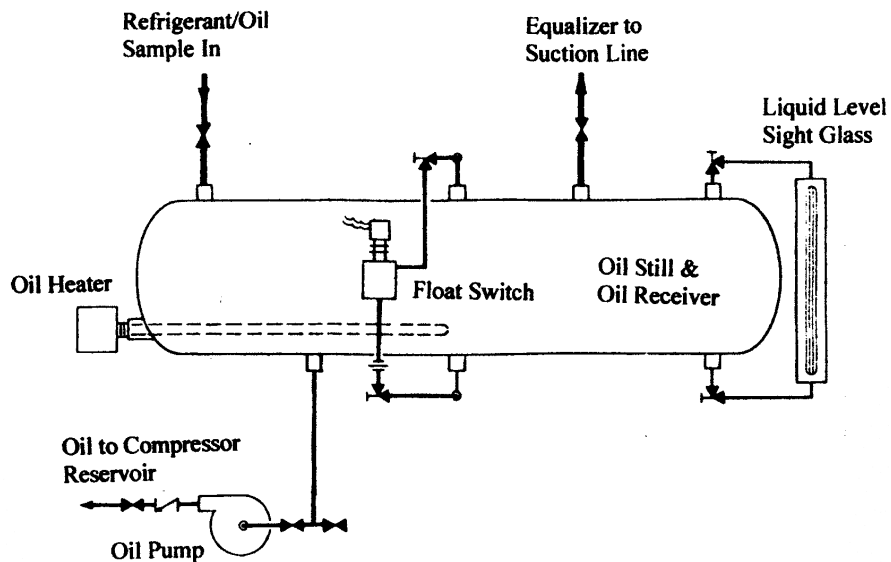
Figure 18 is a typical oil still and oil receiver with an oil pump; the oil is force returned to oil reservoir by the oil pump.

In Figure 18, it shows two possible oil tapping connections for oil return; the Connection Option (A) is for oil gravity which is lighter than the refrigerant, the oil is floating at the top layer of the liquid level in the evaporator, so the oil/liquid sample is taken out from the upper layer of the liquid level in the evaporator; connection Option (B) is for oil gravity which is heavier then the refrigerant, the oil is accumulated at the bottom of the evaporator in this case, so the oil/liquid sample is taken out from the lower point of the evaporator.



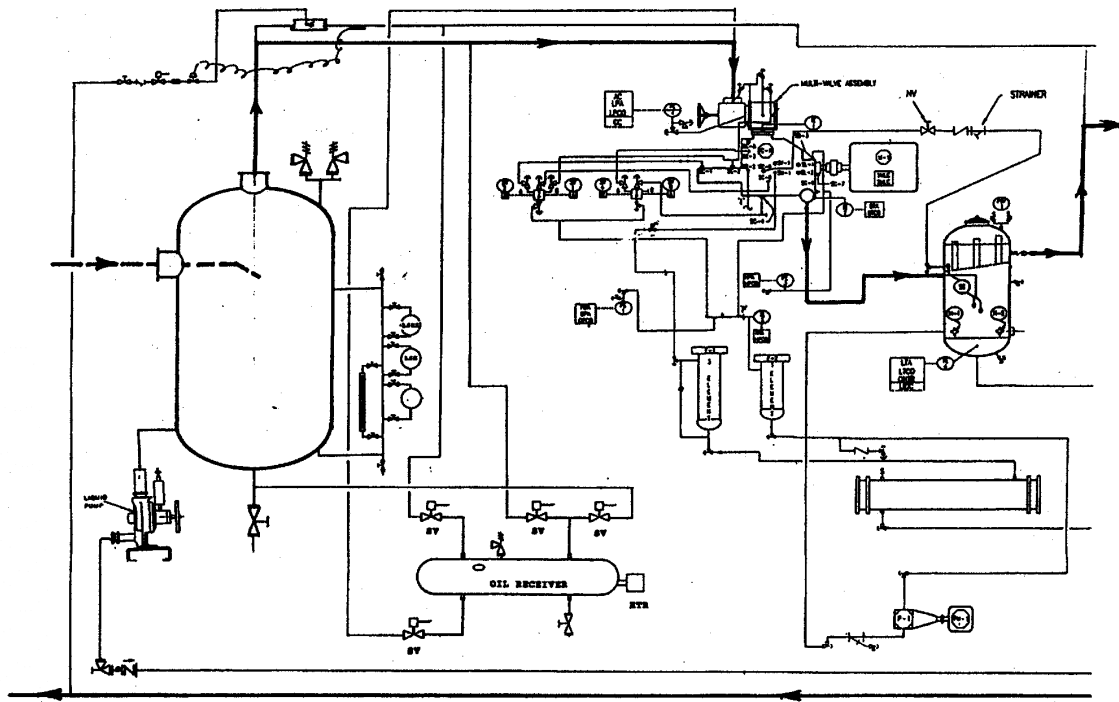
**Figure 18 - Evaporator with Possible Oil Return Connections**

This arrangement can be designed for automatic oil level control and automatic oil return. The oil level in the still and oil receiver is controlled by a level control and flow switch as shown in Figure 19



**Figure 19 - Oil Still with Pump Oil Return**

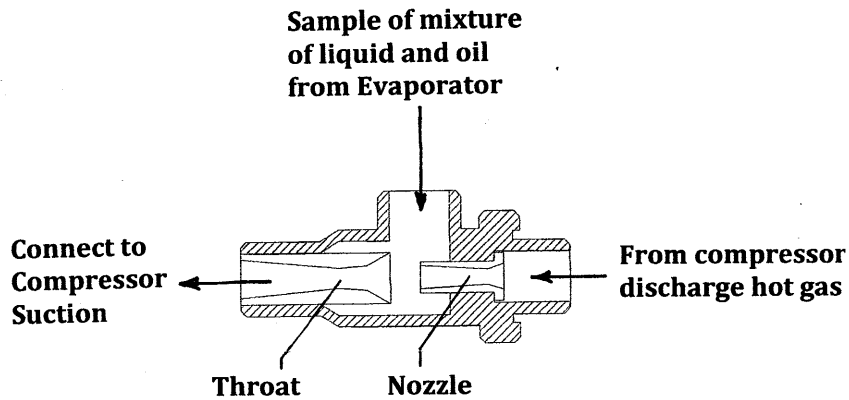
The Figure 20 shown the oil still and oil recovery unit, which is connected to the compressor suction scrubber. Hot gas bypass and liquid injection arrangement is also shown in the same diagram. A liquid pump is also provided for returning the excess liquid refrigerant to the high side of the refrigeration system.



**Figure 20 - Oil Still and Oil Return for Suction Scrubber**

### **Oil Return by Ejector (Eductor)**

The figure 21 shows the simple and low cost device for oil return. The principal of operation of an ejector is to use compressor hot discharge gas to induce the oil and liquid sample from the evaporator, returning the oil/liquid mixture back to compressor suction. The ejector is a simple and the least expensive way for oil return as compared to other oil recovery systems. The use of ejector type oil return will penalize the refrigeration capacity about 4~5%, because the hot gas is bypassed back to the compressor suction.



**Figure 21 Ejector for Oil Return**

## **Removal of None Condensable Gases From System for Energy Efficient Operation and System Performance Assurance**

Non-condensable gases accumulated inside the refrigeration system are mainly air and moisture. Excess none condensable gas accumulated in the condenser causes the condensing temperature to rise and increase the power consumption of the compressor and increases the carbon footprint of the refrigeration system. Furthermore, moisture mixes with refrigerant might cause harmful acids and is also harmful to the performance of the refrigeration system.

None condensable gases in the system are mostly accumulated in the high side such in the condenser and also possibly in the high-pressure receiver. The easiest way to remove the non-condensable is to provide the refrigeration system with an automatic Purge Unit.

The price paid for providing the purge unit is relatively small and yet it improves the system efficiency; reduces the power consumption of the compressor and maintain the performance of the system.

Of course, the least expensive way of purging is just use a manual purge valve. However, a lot of refrigerant is wasted, because non-condensable gases are mixed with refrigerant. Refrigerant is expensive. Therefore, manual purge might not be economical and also it might not be acceptable to environment concern.

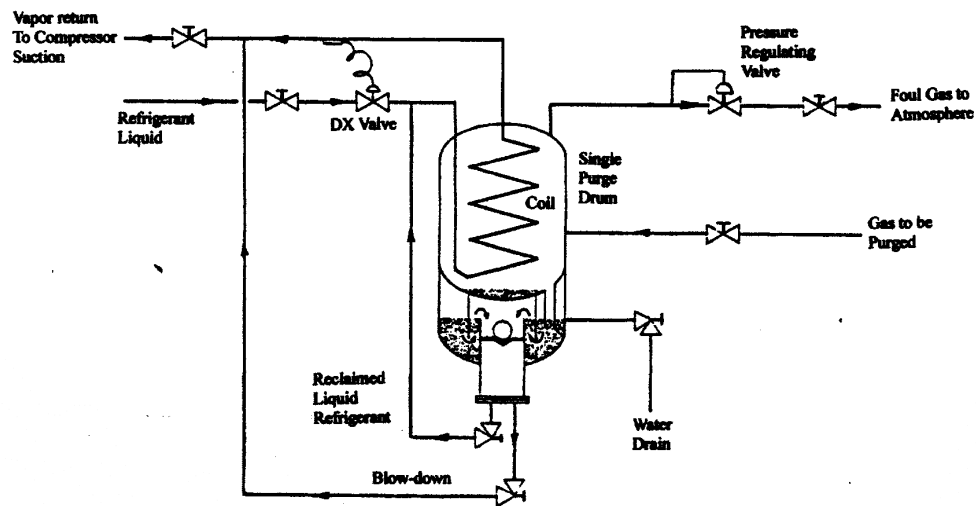
## Purge Drum Method:

Thermal drum refrigerated purge is the most popular purge arrangement being used for refrigeration application these days; it is low cost and it is effective.

Figure 21 is a typical single drum purge arrangement.

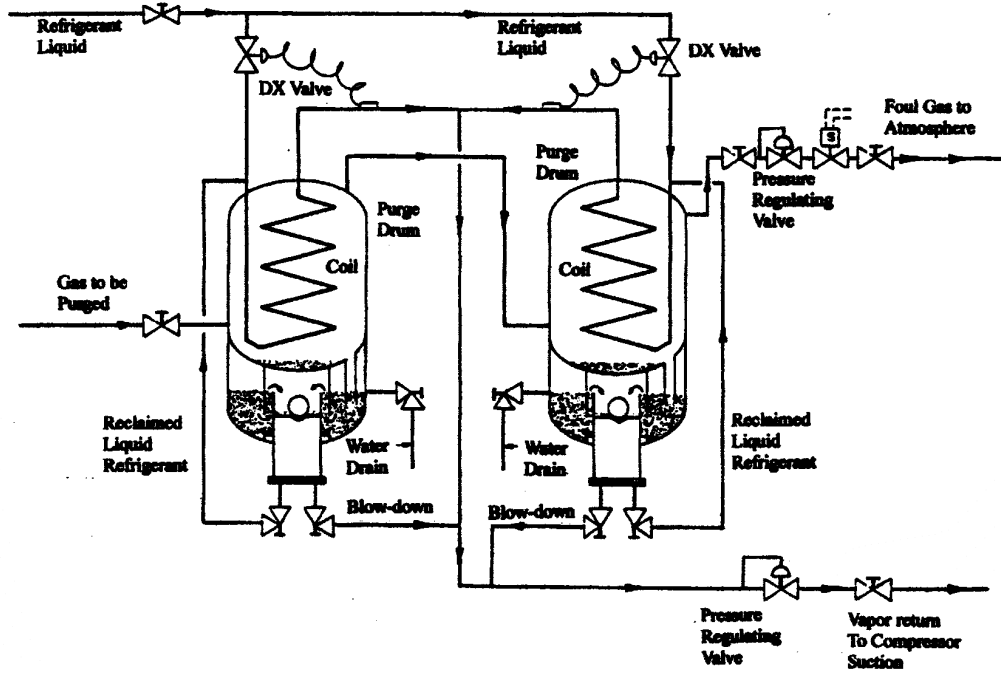
The working theory of thermal drum purge is as the following:

The liquid from the high side liquid receiver; through an expansion valve, the liquid evaporates in the coil inside of the drum. The sample of refrigerant and non-condensable gases mixture from condenser or high pressure receiver enters to the drum consisting of the cooling coil. The refrigerant vapor in the drum is condensed and returned to the system. The build-up of non-condensable gas pressure in the drum will increase and activate a pressure regulating valve and it is released to atmosphere. Any water moisture, which is condensed along with the refrigerant vapor will float on the surface of the refrigerant liquid in the bottom of the drum and is to be manually bled. The refrigerant liquid condensed in the drum is returned to the low side of the system.



**Figure 21 – Single Drum Purge Unit**

Figure 22 is a typical double drum thermal purge unit. The theory for the double drum purge works the same as the single drum purge, except it is 2-stage.



**Figure 22 – Double Drum Purge Unit**

Usually, the single drum purge is used for system is having higher evaporative temperature or the evaporative pressure is higher than atmospheric pressure. The double drum purge is used for lower evaporative temperature, particularly for evaporative pressure below atmospheric pressure.

Compressor type purge was used in the old days and is rarely used for in today's industrial refrigeration application.

## **Recap of Low Energy Consumption and Low Carbon Footprint Refrigeration System Design**

From the above case studies, whenever a refrigeration system is designed for low annual energy consumption and low carbon footprint consideration, it is important to keep in mind the following factors:

- (1) The construction and the design of the compressor should have best efficiency at the design point.
- (2) Each of the main components should be selected for low head operation.



- (3) The system should be designed to produce highest Net Refrigeration Effect (NRE) whenever it is economically justifiable and acceptable.
- (4) The compressor construction and design features should be able to take advantages of reduced head and reduced refrigerant flow during partial load operation.
- (5) The compressor selected should have good partial load efficiency.
- (6) The cooling water, if a water cooled condenser is used, the water temperature for the system should be allowed to fall with the ambient temperature; That means do not use water regulating valve to control the inlet water temperature, unless it is used to keep the minimum head required for the proper operation of the system.
- (7) The refrigeration system should be designed with automatic oil still and oil recovery system for automatic oil return.
- (8) An automatic purge unit should be provided to remove the non-condensable gases from the refrigeration system.

## Notes and acknowledgements

This article is the summary of the application information outlined in the book of “Engineered Industrial Refrigeration Systems Application” by Thomas T.S. Wan. This book is available on-line at <http://tswan888.wordpress.com>.

Some of the data and efficiency curves used in the illustration are from Frick manual.



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