ENGINEERED INDUSTRIAL REFRIGERATION SYSTEMS APPLICATION

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Preface:

This manual is intended for someone who has the prior basic education of refrigeration and has the basic knowledge of mechanical engineering such as thermodynamics, heat transfer and etc.

This manual is mostly dealing with application practices for industrial refrigeration application which is for industrial production and hydrocarbon processing.

What is Engineered Refrigeration System?

“Engineered Refrigeration System” means to design a refrigeration system to meet the requirements of a process or for the need of the production for a specific application with a specific design operation. It is absolutely not accommodating the process to suit any fixed designed of standard or packaged refrigeration product. Engineered system is to compose, to design, to structure, to build and to install the refrigeration system precisely to fulfill the refrigeration need of the process or for the production.

System Concept and System Approach:

It is the objective of this manual to emphasize the meaning of “System” concept and “System” approach.

Each of the components in the “refrigeration system” has its assigned duty under the design operating conditions. The “system” operating conditions and the “system” capacity will change if any one of the components which makes up the refrigeration “system” is changed.

Also, it is important that every step taken and the concept used for the refrigeration “system” design should be explainable by the related applicable theory. Always check with P-H Diagram and the associated engineering formula to see if the flow and thermodynamic properties are fundamentally correct. These rules are applicable to both primary refrigerant circuit and the secondary refrigerant circuit if brine is used for the system. Try to exam the function and the character of each component including each device and valve to see how it is fitted into the structure of the “system”.

Refrigeration Industries and Product Segment Classification:

Various industries and customer users require different types of refrigeration technique, equipment, application and system for its process need and for the making of their product. Therefore, base on the user groups, the refrigeration business can be classified into several major segments as the following:
**Industrial Refrigeration** – This is an engineered system either field erected or factory specially designed and assembled systems for the industries such as Chemical, Hydrocarbon Processing, Oil, Petrochemical, Natural Gas Processing and other industries that might require industrial grade refrigeration system.

**Central Systems for Food and Beverage** – This might be a field erected system or factory engineered unit. This application is mainly provided for the installations of cold storage, food processing and freezing.

**Walk-in Cold Storage and Freezers** – This sector of refrigeration application is for smaller scale of food storage. The refrigeration equipment use for this application is mostly factory packaged.

**Transportation Refrigeration** – Packaged product, mainly use for food distribution and transportation. This includes container refrigeration.

**Display Casings** – Packaged refrigeration equipment use for supermarkets.

**Packaged Refrigerator and Freezer** – Units for home, commercial and restaurant users.

The application for other industrial users such as Pharmaceutical, Environmental Simulation, Electronics and etc. are usually covered by the refrigeration equipment and system suppliers of Industrial Refrigeration and/or Food and Beverage.

This manual is mainly to be used for the applications for Industrial Refrigeration and for large central commercial refrigeration.

**Basic Refrigeration System Review:**

A refrigeration system, no matter simple or complex, it is a combination of four principal components, the Compressor, Condenser, Refrigerant Metering Device and the Evaporator as shown in Fig. 1-1. In a central refrigeration system, for various application requirements, each component might have different design as the following:

- **Compressor** can be either Reciprocating or Centrifugal or Screw.
- **Condenser** can be either water cooled shell-and-tube design or air cooled finned-coil or evaporative wetted pipe or finned-coil.
- **Flow Metering Device** can be either Expansion Valve or High Pressure Float Valve or Low Pressure Float Valve.
- **Evaporator** can be either air finned-coil such as Unit Cooler, Product Cooler or horizontal shell-and-tube heat exchanger or vertical special column.
Evaporator:
Water or Brine
Or Special column
Air – Finned Coil
Shell-and-Tube

Metering Device:
Expansion Valve.
Low Pressure Float.
High Pressure Float.

Condenser:
Air Cooled Condenser.
Shell-and-Tube.
Evaporative Condenser.

Compressor:
Screw Compressor.
Reciprocating Compressor.
Centrifugal Compressor.
Rotary Compressor.

Notes: No Loses.
Cycle is the same for all refrigerants.

FIG. 1-1 Simple Refrigeration Cycle
For some system, it requires other components such as Economizer, Intercooler, Suction Trap, High Pressure Receiver, Low Pressure Receiver, Oil Receiver or Pump Recirculation Receiver. Besides of the components, refrigerant piping, oil system, control system and power system are required for a central system. Defrosting system and other things might also be considered for some application.

**Various Refrigeration Systems Description:**

**Single Stage Refrigeration System:**

- Reciprocating Compressor.
- Screw Compressor.
- Single Stage Centrifugal Compressor.

**Compound Refrigeration System:**

- Internally Compound Compressor.
- Reciprocating to Reciprocating Compressor.
- Rotary to Reciprocating Compressor.
- Screw to Reciprocating Compressor.
- Single Stage Centrifugal to Reciprocating Compressor.
- Screw to Screw Compressor.
- Single Stage Centrifugal to Screw Compressor.
- Single Stage to Multistage Centrifugal Compressor.
- Screw to Multistage Compressor.
- Multistage Centrifugal to Multistage Centrifugal Compressor.

**Cascade Refrigeration System:**

- Reciprocating to Reciprocating Compressor.
- Screw to Reciprocating Compressor.
- Screw to Screw Compressor.
- Multistage Centrifugal to Screw Compressor.
- Multistage Centrifugal to Multistage Centrifugal Compressor.

System description also defines refrigerant feed and the type of Evaporator and the arrangement as the following:

- Open System – Refrigerant flow is never returned back to the system.
- Closed System – Refrigerant flow is continuously recirculated,
- Direct Expansion System.
- Flooded System.
- Liquid Recirculation System.
- Brine or Water Chilling System.
- Thermosyphone Evaporator System.

System description is also can be identified by the method of heat rejection:
A refrigeration system can easily be described by highlighting the type of system, type of evaporator, refrigerant feed and method of heat rejection. For example: the system is “screw to screw compound brine chilling system with water cooled condenser”.

Air Cooled System.
Water Cooled System.
Evaporative Condensing System.
Chapter – 2  P-H Diagram Refrigeration Cycle Analysis & Refrigerant Flow Diagram

Industrial refrigeration system design starts from P-H Diagram Refrigeration Cycle Analysis and Refrigerant Flow Diagram:

(1)  P-H Diagram Refrigeration Cycle Analysis.

P-H Diagram is to analysis the feasibility of the refrigeration cycle, to calculate the thermodynamic properties of the refrigeration system. Use the P-H Diagram analysis, all the refrigerant flow rates and operating conditions at the design point for the system can be clearly determined.

(2)  Refrigerant Flow Diagram.

Refrigerant Flow Diagram is show the equipment used for the system, to determine the refrigerant piping between the components; also to determine the pipe sizes, insulation requirement, to determine the pressure drops, suction superheat and etc. The Refrigerant Flow Diagram can be very simple and it also can be expanded to P&ID (Process and Instrumentation Diagram) if required.

The Refrigerant Flow Diagram is to be read in conjunction with the P-H Diagram to get the entire picture of the system.

P-H Diagram Analysis.

FIG. 2-1 shows a typical P-H (Pressure-Enthalpy) diagram for R-22. FIG. 2-2 shows the idea refrigeration cycle imposed on the P-H diagram.

FIG. 2-3 is the image of the refrigeration cycle taken out from FIG. 2-2, but only showing the related data for this idea refrigeration cycle. The vertical Pressure Axis and the horizontal Enthalpy Axis are omitted. The pressures and enthalpy data related to the cycle are shown on the P-H diagram.

Evaporator - Line A-B-C is the evaporative temperature line. The Enthalpy point “B” to Enthalpy point “C” represents the NRE for the system.

Compressor - The Compression line is C-D, it is a line of constant Entropy. The Head of Compression is \( H_D - H_C \) (Btu/Lb.). Compression Head (or Adiabatic Head) is also expressed in Feet, that is \( (H_D - H_C) \times 778 \). For actual compression, the compression no longer follows adiabatic line and it follows polytropic function as indicated by the line C-D’ in FIG. 2-3.

Condensing - The Condenser (Heat rejection) line is D-E or D’-E for actual case. Total heat rejection from the condenser is the sum of heat absorbed from the
FIG. 2-1  Pressure-Enthalpy Diagram for R-22
FIG. 2-2  Refrigeration Cycle on Actual P-H Diagram
The P-H Diagram only shows the related data for the refrigeration system.

FIG. 2-3  A Simplified P-H Diagram
evaporator plus the power input into the system.

Expansion - The line representing the expansion is E-B.

The operating conditions for the refrigeration system (at no loses) read from the P-H Diagram FIG. 2-3 are as the following:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensing Temperature (CT)</td>
<td>110°F</td>
</tr>
<tr>
<td>Evaporative Temperature (ET)</td>
<td>10°F</td>
</tr>
<tr>
<td>Condensing Pressure</td>
<td>241.04 Psia</td>
</tr>
<tr>
<td>Evaporative Pressure</td>
<td>47.46 Psia</td>
</tr>
<tr>
<td>Compressor Suction Temperature</td>
<td>10°F</td>
</tr>
<tr>
<td>Compressor Suction Pressure</td>
<td>47.46 Psia</td>
</tr>
<tr>
<td>Compressor Discharge Pressure</td>
<td>241.04 Psia</td>
</tr>
<tr>
<td>Compressor Suction Enthalpy</td>
<td>105.44 Btu/Lb.</td>
</tr>
<tr>
<td>Compressor Discharge Enthalpy</td>
<td>123.25 Btu/Lb.</td>
</tr>
<tr>
<td>Refrigerant Liquid Enthalpy</td>
<td>42.45 Btu/Lb.</td>
</tr>
<tr>
<td>Compressor Suction Entropy</td>
<td>0.226</td>
</tr>
</tbody>
</table>

For more accurate calculation, it is suggested that all the data are to be taken out from the Refrigerant Property Table or from a Computer Software.

**Expansion (Throttling Process) and Flash Gas:**

From FIG. 2-3, when the refrigerant liquid leaves the condenser at point “E”; it is a high pressure and high temperature liquid. Through the expansion valve, the liquid is throttling down to point “B”. Under this constant enthalpy process, portion of the liquid is evaporated to cool the liquid down to the temperature at the evaporative pressure line A-C. The percent of the Flash Gas is calculated as the following:

\[
\text{Percent of Flash Gas} = \frac{H_B - H_A}{H_C - H_A} \times 100
\]

**Net Refrigerant Effect:**

The remainder amount of the liquid excluding the Flash Gas in the evaporator is the useful effective liquid for refrigeration; the enthalpies difference between C-B is the “Net Refrigerating Effect” (NRE) as shown in the FIG. 2-3.

Therefore: \[\text{NRE} = \text{Enthalpy at point “C”} - \text{Enthalpy at point “B”} \]
\[= H_C - H_B \quad \text{(Btu/Lb.)}\]

The enthalpy of \(H_B\) is the same as the enthalpy of \(H_E\) because adiabatic process of Throttling,

Therefore: \[\text{NRE} = H_C - H_E \quad \text{(Btu/Lb.)}\]
Refrigerant Flow Determination:

Refrigeration Ton (TR) is the unit of heat absorbed from evaporator, which is to absorb 12,000 Btu/Hr (or 200 Btu/Min) from the product or the process. The amount of refrigerant flow required to absorb one TR through the evaporator is determined as the following:

\[
\text{Flow} = \frac{200}{\text{NRE}} \times \text{TR}
\]

Where:
- **Flow**: Refrigerant Flow, Lbs/Min.
- **200**: Heat removal, Btu/Min.
- **NRE**: Net Refrigeration Effect, Btu/Lb.
- **TR**: Tons of Refrigeration.

\[
\text{NRE} = H_C - H_E, \text{ Therefore:}
\]

\[
\text{Flow} = \frac{200}{H_C - H_E} \times \text{TR}
\]

Suction Gas Superheat:

In the real world, the suction gas at the inlet of the compressor is always superheated. The possible sources of superheat are the heat gain from inside of the heat exchanger, suction piping, suction and liquid heat exchanger if used, and compressor casing. Superheat increases the Entropy of suction gas and increases the compression head for the compressor. The suction superheat is represented by the horizontal constant pressure line on the P-H diagram in FIG. 2-4.

Usually, the suction superheat is not considered as part of the NRE, unless the heat exchanger is purposely designed for it as shown in the case (C) of FIG. 2-4.

Too much suction superheat is not desirable for the refrigeration system due to following reasons:

1. Higher suction entropy, higher power consumption for the compressor.
2. Increases the suction gas volume, increases the compressor size.
3. Higher compressor discharge temperature.

Some of the systems, the suction superheat is extremely high due to the special design of the process heat exchanger. If it is the case, it might be justify by quenching the suction gas with refrigerant liquid.
(A) Suction Superheat at constant pressure

(B) Superheat is not part of NRE

(C) Superheat is part of the NRE

FIG. 2-4 Suction Superheat
**Suction Pressure Drop:**

Suction pressure drop is represented by the vertical line as shown in the P-H diagram of FIG. 2-5. Suction pressure drop must be allowed for compressor selection.

Pressure drop from the evaporator to the compressor suction exists because of the following:

(A) The suction pipe line pressure drops including valves, fittings and suction trap.
(B) Suction valve, strainer and check valve for the compressor.
(C) Suction gas acceleration.
(D) Evaporator pressure drop.

Too much suction pressure drop allowance would increase the size of the compressor and increase the design HP. But, too small suction pressure drop would increase the size of the suction piping, fittings and insulation. Therefore, the design allowance for the suction pressure drop should be economically balanced out with the cost for the piping, valves sizes and insulation.

It is important that the suction pressure drop allowance should be expressed in terms of pressure in Psi, not the equivalent of saturated temperature of °F or °C. The value of a \(\Delta P\) for 2~3°F is much smaller at lower ET than at higher ET.

**Suction Penalties:**

Suction penalty is the combination of suction superheat and suction pressure drop as shown in FIG. 2-6. Therefore, the suction penalty is not a horizontal, nor a vertical line. The correct expression shall be a slope as shown in the P-H Diagram.

**Discharge Penalty:**

Compressor discharge pressure is to be higher than the condensing pressure to overcome the pressure resistances in the discharge line. This pressure difference (\(\Delta P\)) is shown in the P-H Diagram of FIG. 2-7.

Too much discharge pressure drop allowed would increase the power consumption of the compressor. But, too small pressure drop allowance might increase the size of discharge piping and fittings. In any case, the discharge penalty should be allowed for the compressor selection for the proper function of the refrigeration system. For screw compressor application, the \(\Delta P\) is usually larger than the reciprocating and centrifugal compressors. The \(\Delta P\) for air cooled or evaporative condenser is larger than the water cooled condenser.

**Effect of Compressor Suction and Discharge Penalties:**

FIG. 2-8 shows the P-H Diagram for the refrigeration system with the suction and discharge penalties. The compression head which is with penalties (shown in dotted line) is larger than the one without penalties.
**FIG. 2-7** Discharge Penalty

**FIG. 2-8** Effect of Suction and Discharge Penalties
FIG. 2-9 shows the Typical P-H Diagram for a Screw System with liquid subcooling economizer; the diagram shows the suction and the discharge penalties which are usually encountered by screw compressor in a refrigeration system.

Both discharge and suction penalties must be included for system design. It exists in the real world. The refrigeration capacity of the system would be under design as much as 10 to 15%, if the compressor is only rated for saturated CT and saturated ET without penalties.

Liquid Subcooling:

FIG. 2-10 (A) shows the Liquid Subcooling on the P-H diagram. If a smaller degree of liquid Subcooling is required by the application, it could be obtained by using cooling water. However, for larger degree of subcooling, it is necessary to include a liquid subcooler.

Liquid Subcooling is one the methods to increase the NRE as shown in the FIG. 2-10 (B). Also, it is used to ensure the liquid supplied to the evaporator is free of flash gas as required by the application, particularly where the liquid line of the installation is with vertical lift.

The relationships Between CT, ET and NRE:

The P-H Diagram of FIG. 2-11 shows the cases of:
1.0 NRE decreases and compression head increases when the CT is higher.
2.0 The NRE increases and compression head decreases when the CT is lower.

The P-H Diagram of FIG. 2-12 shows the cases of:
1.0 NRE increases and compression head decreases when the ET is higher.
2.0 The NRE getting smaller and compression head increase at lower ET.

The NRE is changing all the times during system operation even at the full load condition.

Flash Intercooling (Auto-Refrigeration):

The flash intercooling is also a “throttling” process. A saturated liquid is flashed from a higher pressure at point ‘E’ (FIG. 2-13) through a constant enthalpy process, down to a lower saturated temperature point ‘B’. Portion of the liquid is evaporated to cool the remainder liquid; this flash intercooling function is also called auto-refrigeration. The percent of the Flash Gas is calculated as the following:

\[
\text{Percent of Flash Gas} = \frac{H_B - H_A}{H_C - H_A} \times 100
\]
Take an example for the case of flash intercooling shown in the P-H diagram of FIG. 2-13, the refrigerant is R-22, the enthalpy for each point is shown, if a flow of 200
EFFECT OF LIQUID SUBCOOLING

(B) Effect of Liquid Subcooling

NET REFRIGERANT EFFECT CHANGES:

\[ NRE = 105.44 - 42.45 = 62.99 \text{ BTU/LB OF R-22} \]
\[ NRE' = 105.44 - 18.61 = 86.83 \text{ BTU/LB OF R-22} \]

FIG. 2-10 Liquid Subcooling
FIG. 2-11 Effect of NRE with Condensing Temperature
FIG. 2-12 Effect of NRE with Evaporative Temperature
Flash Intercooling (Auto-Refrigeration):

Lbs/Min liquid at 105°F is throttling through a float valve down to a pressure of 10°F saturated level; the gas and liquid flow at the 10°F interstage level are as the following:

Flash Gas Flow = \frac{40.85 - 13.10}{105.44 - 13.10} \times 200 = 60.10 \text{ Lbs/Min}

Liquid Flow at 10°F = 200 - 60.10 = 139.9 \text{ Lbs/Min}

Power Consumption Calculation:

For power consumption calculation, the enthalpy point for compressor suction and discharge should include the penalties. Therefore, the compression head shall be $H_2'$ and $H_1'$ instead of $H_2$ and $H_1$ as shown in FIG. 2-8. The compression head shall be in Ft. instead of Btu/Lb for power consumption calculation:

$$\text{Head} = (H_2' - H_1') \times 778 = \text{Ft.}$$
Flow x Head

Compressor HP = \frac{33000 \times \text{Eff}}{\text{Flow} \times \text{Head}}

Where:
- Flow: Refrigerant Flow, Lbs/Min.
- Head: Compression Head, Ft.
- Eff: Compressor Overall Efficiency

Refrigerant Flow Diagram

The main purpose of having Refrigerant Flow Diagram is to show:

(a) Refrigerant flows and the operating conditions for the system.
(b) The piping connections between main components for the refrigeration system.
(c) Type of compressor, condenser, evaporator and etc. are used for the system.
(d) Valves and other accessories for the system.
(e) Other important associate components such as suction trap, oil separator and etc. for the proper function of the system.

The Refrigerant Flow Diagram can be very simple just to show the main piping arrangement between the major components or could be complicated as P&ID.

FIG. 2-14 shows a simple Refrigerant Flow Diagram and the correspondent P-H Diagram for the system. The P-H Diagram is showing part of the operating conditions for the refrigeration cycle.

The Refrigerant Flow Diagram shown in FIG. 2-14 is a simple diagram and yet it conveys the messages of the following:

A. It is a brine chilling refrigeration system.
B. The compressor is a screw and it is completed with oil separator.
C. Compressor is with suction stop valve, strainer and check valve.
D. Discharge check and stop valve are used.
E. It is with a water cooled, horizontal shell-and-tube condenser.
F. The condenser is with a Marine Water Box.
G. DX type brine chiller is used with expansion valve, solenoid valve, strainer and check valve.
H. High pressure receiver with full length sight glass.
I. Receiver equalizer.
J. System drier-filter and liquid sightglass.
K. Charging, purge valves and pressure relief valves.
FIG. 2-14  P-H Diagram & Refrigerant Flow Diagram
DX Brine Chilling System
FIG. 2-15  Refrigerant Flow Diagram-Screw System
Summary of P-H Analysis and Refrigerant Flow Diagram

The FIG. 2-15 is the typical Refrigerant Flow Diagram. FIG. 2-16 is the typical P-H Diagram for a refrigeration system. The Refrigerant Flow Diagram shows that the refrigeration system is a screw brine chilling with water cooled condenser, liquid subcooling economizer and the evaporator is a shell-and-tube flooded design with surge drum; water cooled oil cooling system is also shown for the screw compressor.

For practical use purpose, the P-H Diagram and the Refrigerant Flow Diagram shown in FIG. 2-15 and FIG. 2-16 are well enough for system design application.

FIG. 2-16  P-H Diagram- Screw System
Chapter – 3  Components for Refrigeration Cycle Efficiency Improvement

When the evaporative temperature is lower in a single compression refrigeration system, the NRE is smaller, the system efficiency is getting to be less desirable; the power consumption is higher and the compressor size might be also larger. The way to improve the refrigeration cycle efficiency is to use an intercooler or a liquid subcooler to increase the value of NRE and to use a desuperheater to reduce the discharge temperature and to help the compression head of the compressor.

Increasing the NRE means reduces the refrigerant amount flow to the evaporator to produce the required tons of refrigeration; this results in smaller compressor motor and may be even smaller compressor. Because of the smaller amount of refrigerant is circulated, smaller refrigerant piping, valves, fittings and insulation are to be used for the refrigeration system.

Intercooler, liquid subcooler or desuperheater is basically to be used with a refrigeration system that is having multistage compression with exception of the screw compressor because screw compressor has a special feature would allow a side connection to the compressor casing even it is used for single compression. The intercooler used for screw compressor is called economizer; more details are outlined in Chapter 5.

The component of various intercooler, liquid subcooler and desuperheater for the refrigeration cycle efficiency improvement are illustrated as the following:

**Flash Intercooler:**

The FIG. 3-1 is the Flash Intercooler. From the diagram [1] of FIG. 3-1, the liquid either high side or from higher stage at point “A” is throttling through an expansion device of High Pressure Float Valve or Liquid Level Control Valve. The pressure is dropped down to a lower temperature level of “B” Flash gas and the liquid are disengaged inside the intercooler. The flash gas returns to the high stage compressor suction “D”. The refrigerant liquid is supplied to the evaporator at point “C”.

The equipment and the component for this flash intercooler arrangement is shown in Diagram [2] of FIG. 3-1. The equipment is a combination Intercooler and Receiver with a high pressure float valve. The H.P. Float in this arrangement only provides throttling function to dump all the liquid from high side to the Intercooler/Receiver; it has no liquid level control capability for the Intercooler/Receiver.

Diagram [3] of FIG. 3-1 is a vertical type Flash Intercooler with a liquid level control valve. The valve is with liquid level controller to control the refrigerant liquid level inside of the intercooler. The result of this arrangement on the P-H diagram is the same as the equipment listed under [2].
Diagram [4] shows a horizontal intercooler. It has the same functions as of the vertical arrangement as shown in Diagram [3].
The desuperheating capability of these flash intercoolers is limited. The suction gas temperature to the high stage compressor is the mixture of the discharge superheated gas from low stage compressor and the saturated flash gas from the intercooler at the intermediate temperature level.

**Liquid Subcooler:**

FIG. 3-2 illustrates two different types of liquid subcooler. The refrigerant liquid is from either high side or from higher stage at point “A”. A portion of the liquid expands through an expansion device such as DX valve or liquid level control valve. The liquid is throttling down to an intermediate temperature of B-D; this is to subcool the main stream refrigeration from “A” to “C”. The evaporated gas at the intermediate temperature is leaving the intercooler, mixes with the discharge gas from the low stage compressor, then return to the suction of the high stage compressor.

Diagram [2] of FIG. 3-2 is the arrangement of using a DX type shell-and-tube liquid subcooler with a DX valve and Diagram [3] is a flooded type subcooler with liquid level control valve.

**Desuperheater:**

Desuperheater is sometimes being used for special application to quench the high suction superheated gas even for single stage compression application. FIG. 3-3 shows the desuperheater that only provides the desuperheating function without intercooling. Therefore, it does not help to improve the NRE in a multistage system in this case.

The desuperheater shown is a vertical design. It can be horizontal construction. Portion of the liquid from condenser or from high pressure receiver at point “A” expands through a liquid level control valve down to an intermediate temperature of B-D to desuperheat the low stage discharge gas from “E” to “D”. The mixture of the discharge gas from low stage compressor and the desuperheating gas goes to the suction of the high stage compressor.

Desuperheating function can also be accomplished by using DX valve or even less expensive arrangement of liquid injection.

**Flash Intercooler & Desuperheater:**

The intercoolers shown in FIG. 3-4 are with advantages of both flash intercooling and desuperheating. Liquid either from high side or from higher stage at point “A” is throttling through the Low Pressure Float Valve (or Liquid Level Control Valve), the pressure is dropped down to the lower level of “B”, flash gas and the liquid are disengaged inside the intercooler. The refrigerant liquid is supplied to the evaporator at point “C”. Portion of the liquid in the intercooler is to cool the discharge gas from low stage compressor from “E” to “D”. The mixture of flash gas and the desuperheating gas returns to the high stage compressor suction at point as shown.

Diagram [2] is a vertical design and Diagram [3] is the horizontal arrangement. The functions of the both intercooler are the same.
FIG 3-2  Liquid Subcooler
FIG. 3-3  Desuperheater
Fig. 3-4  Flash Intercooler & Desuperheater
**Liquid Subcooler & Desuperheater:**

The intercooler shown in FIG. 3-5 is similar to the vertical intercooler as shown in the diagram [2] of FIG. 3-4 except this intercooler is with a coil type heat exchanger inside of the intercooler to provide subcooled liquid at point “C” for the evaporator.

Actually, the liquid flow A-B and B-D in the P-H Diagram has two streams of refrigerant flow; one represents the refrigerant need to cool the refrigerant flow from “A” to “C” and another one represents the refrigerant flow need to desuperheat the discharge gas from low stage compressor from point “E” to “D”.


[1] COMBINATION LIQUID SUBCOOLING AND DESUPERHEATING INTERCOOLER.

[2] COMPOUND SYSTEM, LIQUID SUBCOOLING AND INTERSTAGE INTERCOOLING

FIG. 3-5  Liquid Subcooler & Desuperheater
Chapter – 4 Compressors Overview & System
Annual Power Consumption

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 a refrigeration system is being designed. Also, the power consumption and economic balance are to be taken into the consideration when selecting the compressor and the major components of the system.

The industrial refrigeration system is designed to control the process temperature or to keep the evaporative temperature to ensure the quality of the product being made or to meet the process requirement. In order to achieve this sole purpose, the compressor for refrigeration application shall have the following features and capabilities:

1. The compressor is to be equipped with automatic control devices for partial load operation and should be able to maintain the design temperature in the evaporator under any load variation.
2. The compressor should have wide range of part load capability.
3. The lubrication system for the compressor is not to be open to outside atmospheric to avoid contamination.
4. Oil used should be refrigeration oil.
5. Seal gas used should be enclosed and shall be the same as the refrigerant being used for the system.

Compressor Characteristics:

FIG. 4-1 shows the compressor group available for industrial refrigeration application. The Internally Compound reciprocating compressor is used mostly for packaged unit; Rotary compressor was used for booster duty and it is now not commonly used; Ejector type compression device and Axial Flow compressors are also not commonly used by industrial refrigeration equipment manufacturers. Therefore, these compressors are not included in this manual.

FIG. 4-2 shows the characteristics of the compressors commonly used for industrial refrigeration application. The horizontal axis represents the flow or capacity and the vertical axis represents the compression head.

From the FIG. 4-2, the reciprocating compressor performance curve is almost vertical; that means the capacity variation for a reciprocating compressor is very limited, however, the compressor can handle wide range of compression head; Because of this character, reciprocating compressor is referred to as a variable head and constant volume machine.
Refrigeration Compressors

- Positive Displacement
  - Reciprocating
    - 1. Single stage
    - 2. Internally compound
  - Rotary comp.
    - 1. Single rotor (rotary)
    - 2. Twin-rotor (Screw)
    - 3. Single rotor (screw)
- Aerodynamic Type
  - Ejector type
    - 1. Single stage
  - Centrifugal
    - 2. Multistage
  - Axial flow

FIG. 4-1 Various Industrial Refrigeration Compressors
In the same diagram, the performance curve for centrifugal compressor is almost flat, that means the centrifugal compressor can handle wide range of capacity variation, but, the head change capability is limited; Because of these characters, the centrifugal compressor is referred to as a variable volume and constant head machine. The screw compressor curve lies between the reciprocating and centrifugal compressors; therefore, the screw compressor is a variable head and also it is a variable volume machine. The screw compressor has the advantages of both centrifugal and reciprocating compressors; best of all, it does not have the disadvantages of both centrifugal and reciprocating machine.
From the Table 4.1, it is very obvious that the screw compressor has all the advantages as compared to other types of compressors for refrigeration application. That is the reason why screw compressors are so popular in the refrigeration industries.

The size of screw compressor is getting smaller; the screw compressors took over the capacity application range which used to be covered by the reciprocating compressor. The size of screw compressor is also getting bigger, particularly the twin-rotor screw, the screw compressors penetrated the areas where the applications are traditionally covered by centrifugal machines.

FIG. 4-3 shows a typical refrigeration system curve. This system curve might change due to load requirement of the process. The industrial refrigeration system operates 365 days per year unless it is scheduled for service and maintenance. The fundamental and primary duty for the compressor for the application is to be able to perform and to fulfill the requirement of evaporative temperature at the full load and to meet the partial load operation.

The installations for industrial refrigeration are usually with open drive. Sometimes, a special driver such turbine or engine is used instead of motor. Therefore, this manual inclines to cover larger size open type compressors instead of hermetic machines.

**System Annual Power Consumption:**

Power consumption and energy conservation are the most important considerations for industrial refrigeration systems design these days. Therefore, when designing the industrial refrigeration system, the power consumption, particularly the annual power consumption should be carefully exam. FIG. 4-4 shows the relationships between the major components of the refrigeration system and also the formula to estimate the annual energy use for the system. This formula can also be used as the justification and comparison for the selection between different compressors or systems.
FIG. 4-3  Typical Refrigeration System Curve
**POWER CONSUMPTION**

\[ \text{BHP} = \frac{\text{FLOW} \times \text{HEAD}}{33000 \times \text{Eff}} \]

\[ \text{BHP} = \text{BHP}_{(\text{PARTIAL LOAD})} \times \frac{(\text{FLOW})\% \times (\text{HEAD})\%}{(\text{Eff})\%} \]

**ANNUAL POWER CONSUMPTION**

\[ \text{ANNUAL POWER CONSUMPTION} = (\text{KW})_{(\text{DESIGN})} \times \sum_{0}^{\text{HOURS}} \left( \frac{\text{HD}\% \times \text{TR}\%}{\text{C}\%} \right) \]

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

**FIG. 4-4  Power Consumption Formula**
The power consumption calculation for the compressor and refrigerant flow for the load (TR) are as the following (See Chapter-2):

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

\[
\text{Flow} = \frac{200}{\text{NRE}} \times \text{TR}
\]

\[
(\text{Compressor HP})_{\text{Design}} = \frac{\text{Flow} \times \text{Head}}{33000 \times \text{Eff}}
\]

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

From the FIG. 4-4, the annual power consumption is greatly depending on the power consumption of the refrigeration system at the design point (BHP)\text{design} or (KW)\text{design}; The annual power consumption also depends on the conditions as how the system is behaved during the partial load operation; It is important that the refrigeration system is to be designed and to be operated in such way that it is able to take the advantages of reduce head (HD%) and reduce capacity (TR%); also the compressor partial load efficiency (\eta %) is another important factor as shown in the formula.

The compressors for industrial refrigeration are practically to be operated year round. Typical operation time is about 6000 to 8000 hours per year. FIG. 4-5 shows the annual Kw-Hr power consumption evaluation for the refrigeration system with operating hours of 8000 per year; the friction HP loses and oil pump are added for this case.
\[
(Kw)_{\text{Design}} = \left[ \frac{(\text{Flow})_{\text{design}} \times (\text{Head})_{\text{design}}}{33000 \times (\gamma)_{\text{design}}} \right] \times 0.7457
\]

ANNUAL POWER CONSUMPTION = \left( \text{Design Kw} \right) \times \sum_{0}^{8000} \left( \frac{\text{HD}\% \times \text{TR}\%}{\gamma} \right)

+ (Kw)_{\text{HP}} \times 8000

+ (Kw)_{\text{oil}} \times 8000

FIG. 4-5  Annual Power Consumption Evaluation  
For 8000 Hours/Year Operation
Chapter – 5  Reciprocating Compressors

The constructions of reciprocating compressor can be classified in accordance with the Cylinder Arrangements such as Horizontal, Vertical, VSA, Radial, V, W and V/W. Most compressors that are with Horizontal, Vertical, VSA or Radial cylinder arrangement are the low speed large cylinder design; the size of these compressors is huge, the compressor operating speed is between 100 RPM to 500 RPM. Those compressors were mostly used in the old days and there are no longer available in the market. The modern reciprocating compressors are designed for high speed operation; the operating speed is from 1,000 RPM to a top speed of 3,600 RPM. Most modern reciprocating compressors are designed for operating speed about 1,000 to 1,800 RPM.

The maintenance and service are very costly for the Horizontal, Vertical or VSA old style reciprocating compressor; most spare parts are not available from stock and it has to be specially made. These old installation can easily be retrofitted by using a new high speed reciprocating or screw compressor.

The general classification for the modern design reciprocating compressors is shown in FIG. 5-1. Open type compressors are usually used for industrial refrigeration application. Therefore, the application information listed in this manual are mostly for the open type reciprocating compressor. The typical construction for the open reciprocating compressor is shown in FIG. 5-2.

Typical capacity ratings are shown in FIG. 5-3 for R-22; FIG. 5-4 is the typical ratings for R-717. Those ratings are for single stage compression for 8, 12 and 16-cylinder compressors respectively.

The general limitations for the reciprocating compressors are as the following:

**General Limitations (Single Stage):**

<table>
<thead>
<tr>
<th>Limitation</th>
<th>Halogens</th>
<th>Ammonia</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Compression Ratio:</td>
<td>14</td>
<td>9.5</td>
</tr>
<tr>
<td>Maximum Oil Temperature:</td>
<td>160°F</td>
<td></td>
</tr>
<tr>
<td>Maximum Discharge Temperature:</td>
<td>275°F</td>
<td>360°F</td>
</tr>
<tr>
<td>Maximum Pressure Differential:</td>
<td>275 Psi</td>
<td></td>
</tr>
<tr>
<td>Maximum Condensing Temperature:</td>
<td>135°F</td>
<td>120°F</td>
</tr>
<tr>
<td>Maximum HP for Belt Driven:</td>
<td>125 HP</td>
<td></td>
</tr>
</tbody>
</table>

The actual limitations recommended by the compressor manufacturer are always to be observed.
FIG. 5-1 Reciprocating Compressor Classification
FIG. 5-2  Typical Reciprocating Compressor Construction

(1) Compressor Housing
(2) External Discharge Manifold
(3) Water Cooled Head
(4) Stop Valves
(5) Suction Strainer
(6) Lubrication Line
(7) Valves Plates
(8) Piston
(9) Connecting Rod
(10) Cylinder Sleeves
(11) Crankshaft
(12) Bearing
(13) Shaft Seal
### Refrigerant -22, 1170 RPM, Direct Drive, 60 Hz.

<table>
<thead>
<tr>
<th>Unit Model</th>
<th>SF4</th>
<th>S124</th>
<th>S164</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sat. DiSH. Temp. F</td>
<td>TONS</td>
<td>BHP</td>
<td>MBH HR</td>
</tr>
<tr>
<td>80</td>
<td>10</td>
<td>16.3</td>
<td>44.0</td>
</tr>
<tr>
<td></td>
<td>20</td>
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<td>62.9</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>43.2</td>
<td>57.8</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>56.0</td>
<td>79.0</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>71.1</td>
<td>85.1</td>
</tr>
<tr>
<td>90</td>
<td>20</td>
<td>46.3</td>
<td>89.3</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>109.6</td>
<td>93.1</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>133.5</td>
<td>95.9</td>
</tr>
<tr>
<td></td>
<td>50</td>
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</tr>
<tr>
<td>100</td>
<td>10</td>
<td>15.0</td>
<td>58.7</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>21.2</td>
<td>53.6</td>
</tr>
<tr>
<td></td>
<td>30</td>
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<td>64.8</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>43.3</td>
<td>71.8</td>
</tr>
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<td>50</td>
<td>52.4</td>
<td>84.0</td>
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<tr>
<td></td>
<td>10</td>
<td>66.5</td>
<td>91.5</td>
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<tr>
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<td>20</td>
<td>7.0</td>
<td>106.8</td>
</tr>
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<td></td>
<td>30</td>
<td>98.2</td>
<td>112.2</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>120.5</td>
<td>117.2</td>
</tr>
<tr>
<td></td>
<td>50</td>
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<td>131.2</td>
</tr>
<tr>
<td>120</td>
<td>10</td>
<td>15.0</td>
<td>58.7</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>21.2</td>
<td>53.6</td>
</tr>
<tr>
<td></td>
<td>30</td>
<td>29.8</td>
<td>64.8</td>
</tr>
<tr>
<td></td>
<td>40</td>
<td>43.3</td>
<td>71.8</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>52.4</td>
<td>84.0</td>
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<tr>
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<td>10</td>
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<td>112.2</td>
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<tr>
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<td>40</td>
<td>120.5</td>
<td>117.2</td>
</tr>
<tr>
<td></td>
<td>50</td>
<td>148.0</td>
<td>131.2</td>
</tr>
</tbody>
</table>

*Operation below 20 F Suction Temperature requires auxiliary oil cooler. For 50 Hz applications (1000 RPM) multiply above TONS by 0.83 and BHP by 0.86.*

**FIG. 5-3 Typical Capacity Rating for R-22**
## REFRIGERANT—717, 1170 RPM, DIRECT DRIVE, 60 Hz.

<table>
<thead>
<tr>
<th>Unit Model</th>
<th>W84</th>
<th>W124</th>
<th>W164</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sat. Disch. Temp. F</td>
<td>TONS</td>
<td>BHP</td>
<td>MBH HR</td>
</tr>
<tr>
<td>80</td>
<td>-15</td>
<td>35.2</td>
<td>64.8</td>
</tr>
<tr>
<td>0</td>
<td>41.5</td>
<td>69.3</td>
<td>674</td>
</tr>
<tr>
<td>10</td>
<td>56.1</td>
<td>77.4</td>
<td>870</td>
</tr>
<tr>
<td>20</td>
<td>73.7</td>
<td>84.2</td>
<td>1099</td>
</tr>
<tr>
<td>30</td>
<td>94.7</td>
<td>88.6</td>
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</tr>
<tr>
<td>40</td>
<td>119.5</td>
<td>92.7</td>
<td>1609</td>
</tr>
<tr>
<td>50</td>
<td>148.5</td>
<td>95.1</td>
<td>2024</td>
</tr>
<tr>
<td>60</td>
<td>182.5</td>
<td>96.6</td>
<td>2435</td>
</tr>
</tbody>
</table>

| 90 | -15 | 32.2 | 57.3 | 557 | 48.3 | 101.0 | 836 | 64.4 | 134.1 | 1114 |
| 0 | 38.4 | 72.6 | 645 | 57.6 | 109.0 | 968 | 76.8 | 144.8 | 1290 |
| 10 | 52.7 | 82.7 | 843 | 79.1 | 124.1 | 1265 | 105.4 | 165.0 | 1685 |
| 20 | 68.8 | 91.4 | 1070 | 104.8 | 137.1 | 1506 | 139.7 | 182.3 | 2140 |
| 30 | 90.4 | 98.7 | 1335 | 135.5 | 148.1 | 2003 | 180.7 | 196.9 | 2670 |
| 40 | 114.7 | 104.6 | 1642 | 172.0 | 156.8 | 2463 | 229.4 | 208.6 | 3293 |
| 50 | 143.0 | 107.6 | 1999 | 214.5 | 161.4 | 2984 | 286.0 | 214.7 | 3978 |
| 60 | 176.1 | 110.3 | 2394 | 264.2 | 165.5 | 3592 | 352.3 | 220.1 | 4788 |

| 96.25 | -10 | 36.4 | 74.3 | 625 | 54.5 | 111.5 | 937 | 72.7 | 148.1 | 1249 |
| 0 | 50.4 | 85.6 | 823 | 75.6 | 128.4 | 1234 | 100.9 | 170.7 | 1645 |
| 10 | 67.4 | 95.6 | 1051 | 101.1 | 143.4 | 1577 | 134.7 | 190.7 | 2102 |
| 20 | 87.5 | 104.2 | 1316 | 131.3 | 156.3 | 1974 | 175.1 | 207.9 | 2631 |
| 30 | 111.5 | 111.2 | 1621 | 167.2 | 166.8 | 2430 | 222.9 | 222.0 | 3240 |
| 40 | 139.5 | 115.9 | 1969 | 209.3 | 173.9 | 2954 | 279.0 | 221.3 | 3937 |
| 50 | 172.1 | 119.0 | 2368 | 258.2 | 178.4 | 3552 | 344.2 | 237.5 | 4735 |
| 100 | -10 | 41.7 | 81.7 | 707 | 62.6 | 121.9 | 1061 | 83.5 | 162.0 | 1414 |
| 0 | 49.1 | 87.1 | 810 | 73.6 | 130.7 | 1218 | 98.1 | 173.8 | 1620 |
| 10 | 65.9 | 96.0 | 1040 | 98.8 | 141.0 | 1560 | 131.8 | 195.6 | 2079 |
| 20 | 85.8 | 107.4 | 1303 | 128.7 | 161.1 | 1955 | 171.7 | 214.2 | 2605 |
| 30 | 109.5 | 115.3 | 1607 | 164.2 | 172.9 | 2411 | 219.0 | 230 | 3213 |
| 40 | 137.4 | 121.1 | 1957 | 206.1 | 181.6 | 2925 | 274.8 | 241.6 | 3912 |
| 50 | 169.7 | 124.4 | 2352 | 254.5 | 186.6 | 3528 | 339.3 | 248.3 | 4703 |

| 110 | 5 | 53.1 | 97.4 | 884 | 79.6 | 146.1 | 1327 | 106.1 | 194.2 | 1768 |
| 10 | 61.6 | 103.9 | 1003 | 92.4 | 155.8 | 1505 | 123.2 | 207.2 | 2060 |
| 20 | 81.1 | 115.5 | 1267 | 121.7 | 173.2 | 1901 | 162.3 | 230.4 | 2534 |
| 30 | 104.3 | 125.6 | 1570 | 156.4 | 188.2 | 2356 | 208.5 | 250.5 | 3140 |
| 40 | 131.4 | 133.9 | 1917 | 197.1 | 200.8 | 2876 | 262.7 | 267.2 | 3833 |
| 50 | 153.0 | 139.6 | 2312 | 244.6 | 209.4 | 3468 | 326.3 | 278.7 | 4622 |

| 120 | 15 | 60.3 | 115.9 | 1091 | 99.5 | 173.8 | 1636 | 132.7 | 231.2 | 2180 |
| 20 | 76.3 | 122.8 | 1228 | 114.5 | 184.2 | 1842 | 152.6 | 245.1 | 2455 |
| 30 | 98.7 | 135.2 | 1529 | 148.1 | 202.8 | 2293 | 197.5 | 268.8 | 3057 |
| 40 | 125.7 | 145.9 | 1874 | 187.9 | 218.9 | 2811 | 250.6 | 291.3 | 3747 |
| 50 | 156.1 | 154.7 | 2267 | 234.2 | 223.2 | 3401 | 312.2 | 309.0 | 4533 |

For 50 Hz applications (1000 RPM) multiply above TONS by 0.83 and BHP by 0.86.

**FIG. 5-4  Typical Capacity Rating for R-717**

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FIG. 5-5 shows the typical Overall Efficiency and Volumetric Efficiency Curve for Halocarbon refrigerant.

FIG. 5-5  Typical Efficiency Curves – Halocarbon
FIG. 5-6 shows a typical oil Cooling and Jacket Cooling for the compressor. Some cases, the water in the oil cooler will not drain completely; therefore, if the installation is located outdoor and exposed to ambient that is below freezing, the oil cooler must be protected against freeze up.

Maximum HP limit for belt drives are to be given by the manufacturer. Usually, maximum driving motor is not to over 125 HP for belt drive application. The horsepower required for belt drive is about 3% higher than the direct drive or correct in accordance with the recommendation of the manufacturer. Maximum operating speed is given by the compressor maker also. Some reciprocating compressor is having maximum speed of 1200 Rpm and some are having 1800 Rpm.

The motor should be sized for pull down application for refrigeration application. Pull down HP power is the average of horsepower of standby suction condition and the design suction condition.

The high speed reciprocating compressors are designed for dry compression and the tolerance of liquid carry over in the suction line is very limited; even some liquid particles in the compressor suction may cause faster wear of the cylinder walls and pistons. Therefore, the refrigeration system design using reciprocating compressor must take proper care that the vapor flow to the compressor suction is superheated; Also, no liquid oil should be allowed to slug back to the compressor suction.

Capacity control for reciprocating compressor is achieved by controlling the suction pressure using a throttling valve at the suction or by hot gas bypass or by cylinder unloading. The suction throttling valve is not used by refrigeration application; Cylinder Unloading method is commonly used by Refrigeration Industries.

Most common sizes of reciprocating compressor are 4, 6, 8, 12 and 16 cylinders. The cylinder unload is controlled by solenoid valve. The capacity steps and percent of part loads are as the following:

<table>
<thead>
<tr>
<th>Number of Compressor Cylinder</th>
<th>Minimum Percent of Full Load Capacity</th>
<th>Capacity Steps</th>
</tr>
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<tbody>
<tr>
<td></td>
<td>% Full Load Capacity</td>
<td>% Full Load Capacity</td>
</tr>
<tr>
<td>4</td>
<td>50</td>
<td>100</td>
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<tr>
<td>6</td>
<td>33-1/3</td>
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</tr>
<tr>
<td>8</td>
<td>25</td>
<td>100</td>
</tr>
<tr>
<td>12</td>
<td>33-1/3</td>
<td>100</td>
</tr>
<tr>
<td>16</td>
<td>25</td>
<td>100</td>
</tr>
</tbody>
</table>

FIG. 5-7 shows the oil piping and oil receiver connections for uneven sizes compressor in parallel operation. It is important to note that the oil receiver should be in line with the inlets of the crankcase oil float valves of the both compressors.

FIG. 5-8 shows the oil and piping arrangement for the parallel reciprocating compressors for oil rich liquid return from the system. Also, high pressure refrigerant liquid is used for the oil cooling for the compressor instead of water.
FIG. 5-6 Oil & Jacket Cooling
Reciprocating Compressor

FIG. 5-7 Piping & Oil Connections for Uneven Units
Reciprocating Compressors Parallel Operation
FIG. 5-8 Oil & Piping for Parallel Operation
Refrigerant Cooled Oil Cooling
Reciprocating Compressors
Internally compound reciprocating compressor is a compressor that certain numbers of cylinder are used as the low stage and certain numbers of cylinder are used as the high stage within the same compressor casing; the discharge gas from the low stage cylinders is mixed with intercooling gas then internally discharged to the high stage cylinders. FIG. 5-9 shows the typical internally compound reciprocating compressor.

FIG. 5-10 shows the typical capacity ratings for R-22 internally compound reciprocating compressors; FIG. 5-11 shows a typical factory supplied intercooling arrangement kit for the internally compound reciprocating compressor.
## Typical Capacity Ratings, R-22:

<table>
<thead>
<tr>
<th>Unit Model</th>
<th>C83</th>
<th>C123</th>
<th>C163</th>
</tr>
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<tbody>
<tr>
<td>Sat. Disch.</td>
<td>Saturated</td>
<td>TONS</td>
<td>BHP</td>
</tr>
<tr>
<td>Temp. °F</td>
<td>Suction</td>
<td>°F</td>
<td>°F</td>
</tr>
<tr>
<td>90</td>
<td>-80</td>
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<td>11.7</td>
<td>45.5</td>
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<td>15.8</td>
<td>52.5</td>
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<td>-30</td>
<td>20.7</td>
<td>59.0</td>
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<td>20.0</td>
<td>62.0</td>
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<tr>
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<td>-20</td>
<td>25.7</td>
<td>68.5</td>
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<tr>
<td>120</td>
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<td>3.05</td>
<td>28.5</td>
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<td>5.0</td>
<td>36.0</td>
</tr>
<tr>
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<td>7.5</td>
<td>43.0</td>
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<tr>
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<td>-50</td>
<td>10.6</td>
<td>50.0</td>
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<td>-40</td>
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<td>65.0</td>
</tr>
<tr>
<td></td>
<td>-20</td>
<td>25.0</td>
<td>72.5</td>
</tr>
</tbody>
</table>

Superheat: 25°F from -20°F to -70°F STP.
For STP lower than -70°F,
consult table on page 6.

Subcooling: 10°F above interstage saturation temp.

---

**FIG. 5-10 Typical Capacity Ratings – R22**

Internally Compound Reciprocating Compressor
FIG. 5-11 Typical Intercooling Arrangement
Internally Compound Reciprocating Compressor
Chapter – 6 Screw Compressors

Screw compressors are the most widely used compressors for industrial refrigeration application in the world because of the favorable features of variable head and variable volume characters. Screw compressor practically replaces almost all the applications that used to the areas of reciprocating compressor in industrial refrigeration in recent decade.

Screw compressor was invented long time ago. It was based on dry compression; that is the compressor is with a set of timing gears to ensure that no contacts between the twin rotors of the compressor. The earlier design screws were mostly used and still being used for air and gas compression in oil refinery, petrochemical and gas processing industries. It was not used by refrigeration industries due to various difficulties including high cost.

Oil Injection (Oil Flooded) for screw compressor was invented in 1950’s. It eliminated the timing gears, increasing the compression ratio and reducing the discharge temperature for the compressor. But, it was still not good enough for refrigeration application, because of oil carry over problem and lack of capacity control capability for partial load operation.

The adaptation of the screw compressor for refrigeration duty was made possible in 1970’s after the new inventions plus the further improvements made to the screw compressor. Those major invention and improvement were: The hydraulically operated capacity control; economizer cycle improves compressor efficiency to a point comparable to two-stage; improvements on compressor discharge oil filter, variable Vi, better control by using microprocessor panel and etc.

There are two types of open oil flooded screw compressor available for industrial refrigeration application do-day; one is the Twin-Rotor screw and the other is the Single-Rotor screw. Twin-rotor screw is the most widely used compressor for industrial refrigeration installations worldwide, particularly larger size installations.

FIG. 6.1 shows the typical construction of the twin-rotor screw compressor; FIG. 6.2 shows the typical Asymmetric Twin-Rotor Profile; some manufacturers use other rotor profile such as 4+6 “A” or 5+7 “D” rotor profile or others. FIG. 6.3 is a typical construction of Single-Rotor screw compressor.

The compressor shown in FIG. 6.1 is a bare screw compressor; it can not be used for refrigeration. The screw compressor only can become useful if it is equipped with other necessary subsystems becoming a “compressor unit” as shown in FIG. 6.4. A typical “compressor unit” consists of other subsystems as the following:

Screw Compressor.
Driving Motor & Power Supply System.
Lubrication System.
Oil Reservoir & Oil Separation System.
Hydraulic System.
FIG. 6.1  Typical Twin-Rotor Screw Compressor Construction
FIG. 6.2  Typical Asymmetric Twin-Rotor Profile

FIG. 6.3  Typical Single-Rotor Screw Construction
FIG. 6.4 Typical Screw Compressor Unit Arrangement
Oil Cooling & Filtering System.
Suction and Discharge Valves and Strainer.
Microprocessor Control Panel and Control System.

One of the terms that manufacturers commonly refer to is “Standard” such as “Standard Unit”. “Standard Unit” is the scope of supply for the unit as defined by the manufacturer. A Standard Unit for a screw compressor usually includes the components shown in FIG. 6.1, except the main driver. The standard steel base for the driver is designed to accept NEMA Standards motor and the unit is with single oil filter, single oil cooler and NEMA-1 control panel. Extra cost might be imposed by the maker for the items that is outside the scope of supply of the “Standard Unit”. Therefore, it is recommended to check what exactly the scope of supply is with each supplier.

The screw compressor used for industrial refrigeration is equipped with Hydraulic Operator and Slide Vane capacity control mechanism; the slide vane is capable of controlling the screw compressor for partial load operation down to 10% without surge. FIG. 6.5 is a typical partial load performance curve chart for a screw compressor.

The general compression ratio limit for screw compressor could be as high as 25:1. However, the compressor efficiency is lower when the compression ratio is higher.

Other technical, application details and special feature for the screw compressor are as the following:

**Economizer:**

One of the special features of screw compressor is to allow a side load connection to the compressor casing; this feature makes economizer cycle possible for the refrigeration system to improve the system efficiency. The capacity of the screw can be increased by using the economizer and yet the driving power increase is relatively small.

There are two types of economizer for the screw compressor; one is Flash Economizer and the other is the Liquid Subcooling Economizer. The economizer connection to the compressor casing is located by the compressor maker for the optimum pressure for the economizer. The benefit of economizing is better when the compression ratio of the compressor is higher.

**Flash Type Economizer:**

FIG. 6.6 shows the flash type economizer using a vertical type flash intercooler which is controlled by a liquid level valve. The flash gas returns to the side connection of the compressor. The FIG. 6.7 is also a flash intercooler, but it is with a high pressure float valve, the intercooling vessel serves as the combination of economizer and receiver. The P-H diagram is the same for both flash economizing and is shown in FIG. 6.8. At the top of the FIG. 6.8 shown the compression cycle without economizer; where the Refrigeration Effect $\delta H$ is small. The Refrigeration Effect $\delta H$ is greatly increased when economizer is used.
FIG. 6.5  Typical Partial Load Curves
Fixed Vi Screw Compressor
FIG. 6.6 Screw Compressor with Flash Economizer

FIG. 6.7 Screw Compressor with Economizer/Receiver
FIG. 6.8  P-H Diagram for Screw with Flash Economizer
Liquid Subcooling Type Economizer:

FIG. 6.9 shows the Shell-and-Coil type economizer in a vertical type intercooler which is controlled by a liquid level valve. The flash gas is returned to the side connection of the compressor. The FIG. 6.10 shown is a DX type liquid subcooling economizer. The P-H diagram for these two liquid subcooling (FIG. 6.9 and FIG. 6.10) is the same and it is shown in FIG. 6.11.

For the comparison, the P-H diagram shown at the top of the FIG. 6.11 is the compression cycle without economizer; the Refrigeration Effect $\delta H$ is small. The Refrigeration Effect $\delta H$ is greatly increased when liquid subcooling economizer is used.

Oil Pump:

The purpose of using oil pump is to supply oil injection for the rotors and shaft seal, to cool the compressor, to lubricate the bearings and for the hydraulic operator.

There are three types of compressor design when comes to the use of oil pump:
1) Some compressor design needs full time oil pump for the compressor operation.
2) Some compressor design only requires the oil pump to be operated for the start-up and shut-down. This type of oil pump is the auxiliary oil pump. The oil pump is not in operation after the compressor starts up.
3) Some compressor design can start up the compressor without the auxiliary oil pump and is able to utilize the system pressure differential from the oil reservoir to other parts of the compressor for the positive oil flow; no oil pump at all.

Oil pump is usually required if the system pressure differential is too small or when the compressor is used for booster duty or low stage application.

Oil Separator:

Oil injects into the screw compressor to lubricate the rotating parts and to cool the compressor. The oil absorbs a lot of heat; the oil vapor is mixed with the refrigerant gas when leave the compressor. The oil is to be removed as much as possible through an oil separator-filter before it goes to condenser and evaporator.

The oil separator is a standard component for the screw compressor for industrial refrigeration application. FIG. 6.12 is a typical three-stage oil separator structure for the screw compressor. The oil separator also serves as the oil reservoir. Most the oil separators of modern design screw compressors are equipped with Coalescent Filters inside of the oil separator for better oil separation efficiency. The oil carry over rate could be 3 PPM to 10 PPM depending on the operating conditions if Coalescent Filter is used.

Most Standard Units supplied by the manufacturers are with a horizontal oil separator as shown in FIG. 6.13. Vertical oil separator with floor mounted compressor might be required as special arrangement if the driving motor is too heavy or a special driver such gas engine, steam turbine or gas turbine is being used.
FIG. 6.9  Screw Compressor with Shell-and-Coil
Liquid Subcooling Economizer

FIG. 6.10  Screw Compressor with DX Liquid
Subcooling Economizer
R-22 LIQUID SUBCOOLING ECONOMIZER

LIQUID TO SYSTEM:
225.45 PSIA 11.65°F
11.65°F 225.45 PSIA

105°F 227.95 PSIA

3.65°F 41.71 PSIA

ET -40°F

200 TR 15.22 PSIA

-20°F 13.22 PSIA

FIG. 6.11 P-H Diagram for Screw with Liquid Subcooling Economizer
FIG. 6.12  Oil Separator – Screw Compressor Unit

FIG. 6.13  Screw Unit with Horizontal Oil Separator
**Oil Cooling & Oil Cooler:**

The oil temperature in the oil separator is relatively high and therefore, it is to be cooled down by oil cooler (except the liquid injection oil cooling) to an acceptable temperature level before it is re-injected back to the compressor. There are several types of oil cooling methods available:

**Water Cooled Oil Cooling:**

Oil is cooled by water through a shell-and-tube heat exchanger as shown in FIG. 6.14. FIG. 6.15 shows the relative position between the oil cooler and the oil circuit of the screw compressor.

The major disadvantage of using water cooled oil cooler is that maintenance and service are required for the heat exchanger because of water problem. The big advantage is that a portion of the total heat rejection from the compressor is removed by water from the oil cooler to the cooling tower. The condenser heat rejection load is therefore smaller (see Heat Rejection for Condenser Selection).

**Liquid Injection Oil Cooling:**

Liquid injection oil cooling is that a small amount of refrigerant liquid from high pressure receiver is injected into the screw compressor just before the discharge port of the compressor. The liquid refrigerant evaporated and the mixture of the oil and the refrigerant vapor is cooled down to an acceptable temperature. This type of arrangement is shown in FIG. 6.16. The amount of liquid to be injected is control by a thermostatic expansion valve. The FIG. 6.17 shows the typical arrangement of a screw compressor with liquid injection oil cooling.

The advantage of using liquid injection oil cooling is low cost and very little maintenance is required. The disadvantages are capacity and power consumption penalties. It is generally about 7 to 9% in capacity reduction and also resulting in high power consumption rate. Sometimes, the liquid injection method might not be feasible for low head or high suction temperature applications.

**Thermosyphon Oil Cooling:**

Thermosyphone oil cooling is the method using refrigerant to cool the oil, but without penalizing the capacity or power consumption of the compressor.

FIG. 6.18 is the typical thermosyphone oil cooler arrangement. The thermosyphone oil cooler is a shell-and-tube heat exchanger. The oil flow through the shell side and refrigerant is through the tube side. The refrigerant liquid is supplied to the thermosyphone oil cooler from the thermosyphone receiver by gravity force. Portion of the liquid is vaporized to cool the oil in the heat exchanger; the bubbling mixture of liquid/vapor is circulated back to the thermosyphone receiver. The oil temperature returning to the compressor is regulated by a three-way thermostatic valve as indicated.
FIG. 6.14 Typical Water Cooled Oil Cooler

FIG. 6.15 Screw Compressor with Water Cooled Oil Cooler
FIG. 6.16 Liquid Injection Oil Cooling

FIG. 6.17 Screw Compressor with Liquid Injection Oil Cooling
FIG. 6.18 Typical Thermosyphon Oil Cooler Arrangement
Dual Oil Coolers and Dual Oil Filters:

A lot of oil is circulated for the screw compressor. Oil flow is considered as the blood line for the screw compressor operation. For industrial process refrigeration application, the compressor is required to operate year round without being shut down. If it is the case, some of the users request that compressor unit is to be fitted with dual oil filters with a change over valve; this arrangement allows the switching of the oil flow to the stand-by filter while the other filter is being service and cleaned. Dual oil coolers with a change over valve also might be requested by the user for the same reason if water cooled oil cooler is used.

Capacity Control Slide Valve:

Slide Valve (Slide Vane) is the capacity control device for the screw compressor. This slide valve is able to control the capacity range of the screw compressor from 100% down to 10% without surge. The layout and construction of the Slide Vane for the screw are shown in FIG. 6.19. The operational functions of the sliding vane with corresponding P-V diagram explanations are shown in FIG. 6.20. Case-(1) of FIG. 6.20 is when the slide vane fully closed for full load operation; When the load reduces say to 80% as shown in Case-(2), the slide vane moves to the right by the controller to allow 20% of the gas bypass back to the suction; The slide valve is moving further to the right in Case-(3) to allow more gas to be bypassed to the suction if the load reduces further. The slide valve moves to the left to closed the bypass opening when the refrigeration load is increased.

Internal Volume Ratio Vi:

The characteristic of a screw compressor is determined by the "Internal Volume Ratio" (Vi) of the compressor and the Vi is related to the length of the slide vane. The theory to explain all of this character is as the following:

When the refrigerant gas is compressed by the screw compressor, the gas pressure is increased while its volume is reduced. The internal compression ratio which occurs in the compressor prior to discharge and the discharge pressure of the compressor are governed by the location of the discharge port. For a fixed Vi design compressor, the length of the sliding vane is fixed and the discharge port is tailored at the time when the compressor is manufactured.

The fixed (or the built-in) pressure ratio of the screw compressor is presented by the formulas as the following:

\[ P_i = \frac{P_d}{P_s} \quad \text{(1)} \]

\( P_i \) = Built-in pressure ratio.

\( P_d \) = Pressure of the gas just before discharge.

\( P_s \) = Pressure of the gas just before compression.
Screw Compressor Slide Valve

FIG. 6.19  Capacity Control Slide Valve
Screw Compressor
The term of "Built-in Volume Ratio \(V_i\)" is more frequently used than the built-in pressure ratio for the screw compressor. The \(V_i\) is defined as the following:

\[
V_i = \frac{V_d}{V_s} \quad \text{----------------------------------} \quad (2)
\]

\(V_i\) = Built-in volume ratio.
\(V_d\) = Volume of the gas just before discharge.
\(V_s\) = Volume of the gas just before compression.

From the Gas Law of thermodynamic, the built-in volume ratio is related to the built-in pressure ratio as the following:

\[
P \times V^k = \text{Constant}
\]

\[
P_2 \times (V_2)^k = P_1 \times (V_1)^k \quad \text{----------------------------------} \quad (3)
\]

\(k = \) Gas constant for the refrigerant being compressed.

or

\[
P_d \times (V_d)^k = P_s \times (V_s)^k \quad \text{----------------------------------} \quad (4)
\]

Therefore:

\[
\frac{P_2}{P_1} = \frac{(V_1)^k}{(V_2)^k} \quad \text{----------------------------------} \quad (5)
\]

\[
\frac{P_d}{P_s} = \frac{(V_s)^k}{(V_d)^k} \quad \text{----------------------------------} \quad (6)
\]

\[
\frac{P_d}{P_s} = \left(\frac{V_s}{V_d}\right)^k \quad \text{----------------------------------} \quad (7)
\]

\[
\frac{P_d}{P_s} = V_i^k \quad \text{----------------------------------} \quad (8)
\]

\[
P_d = P_s \times (V_i)^k \quad \text{----------------------------------} \quad (9)
\]
The selection of the size (length) of slide vane is in accordance with the design compression ratio for the compressor. See FIG. 6.21, a shorter slide vane is selected if the design compression ratio is small (low Vi), otherwise, a longer slide vane will be selected if the compression head is high (high Vi). The Vi is fixed once the length of the slide vane is selected for the compressor. The slide vane is selected at the time when the compressor is made in accordance with the design compression ratio.

The Vi becomes a constant once the slide vane length is given, this is referred to as a fixed Vi. From the formula (9), the characteristics of the screw compressor with this fixed Vi are as the following:

(a) The discharge pressure of the compressor is always constant if the suction pressure is controlled at the set point for the refrigeration system.

(b) The internal compression ratio of the screw compressor is always constant no matter how the external system pressure ratio changes. The internal pressure ratio of the compressor will not be able to match the external pressure ratio of the refrigeration system.

(c) The screw compressor internal discharge pressure is not changed in despite of the pressure change in the condenser.

Both the adiabatic efficiency and the volumetric efficiency are fixed when the compressor is given a fixed Vi. A screw maker usually provides several sizes of slide vane available for the selection. For example, the sizes of slide vane to represent Vi of 2.33, 3.0, 3.6 and 4.6 are shown in FIG. 6.22. As indicated earlier, a low compression ratio uses a shorter sliding vane which is having a lower Vi and high compression ratio is logically to choose a longer sliding vane which is having a high Vi for better power consumption. Once a screw compressor is supplied with a specific built-in Vi, this Vi slide vane cannot be altered because it is determined by the geometrical shapes of the axial and radial outlet ports.

Once the compressor is supplied with a fixed Vi, for example Vi = 3.0, the compressor characteristic curve is fixed and the efficiency curve is following the curve "B" of the FIG. 6.22. The peak optimum efficiency of a screw compressor having Vi of 3.0 is at the compression ratio of 3.2. The compressor can be operated at any other compression ratio. However, the compressor shall operate at lower efficiency with higher power consumption.

**Variable Vi Theory & Benefits:**

As shown in (A) of FIG. 6.23, a fixed Vi screw compressor, the motion of the slid valve is stopped when it contacts with the rotor housing. In a variableVi compressor, the slide vane is replaced with a second movable slide stop as shown in the (B) of FIG. 6.23. By moving the slides back and forth, the radial discharge port can be relocated during the operation to match compressor internal compression ratio to the external system pressure ratio to obtain optimum discharge pressure and maintaining the maximum efficiency.
FIG. 6.21  Internal Volume Ratio (Vi) & Slide Vane
FIG. 6.22 Efficiency Curves for Fixed Vi Screw Compressor
(A) View from above  
Fixed Vi Slide Vane

(B) View from above  
Variable Vi Slide Vane

(C) Variable Vi Mechanism  
Full Load Positions

(D) Variable Vi Mechanism  
Partial Load Positions

FIG. 6.23 Fixed Vi and Variable Vi Slide Vanes
A hydraulic controllers “A” and “B” are used to control the slide valve and slide stop separately as shown in FIG. 6.23(C). This allows the compressor Vi to be adjusted during operation and matching the compression discharge pressure precisely as required for the refrigeration system. The stepless control is automatically controlled by the microprocessor control panel for all the operational range and conditions. Thus, the maximum peak volumetric and adiabatic efficiencies of the compressor are maintained over all the operational ranges. Therefore, the efficiency curve of a variable Vi compressor is the line of (X)-(Y), which is at the top of the peak efficiencies of all fixed Vi curves as shown in FIG. 6.24. The power consumption of a variable Vi compressor is the best at all the partial load conditions. A variable Vi compressor is able to take the advantages of both reduce capacity and reduce head during any partial load operation; also the percent of partial load efficiency is actually increased. There, the annual power consumption evaluation is improved in accordance with the formula shown in FIG. 4-4.

**Heat Rejection for Condenser Selection:**

Normally, the heat rejection from a compressor to condenser is the sum of heat absorbed from the evaporator plus the heat input from driver. But, the amount of heat rejection to condenser from a screw compressor, it depends on what type of oil cooling system is used. For liquid injection and thermosyphon oil cooling system, the heat rejection to condenser from the screw compressor shall be the same as for reciprocating or centrifugal compressors. But, if the oil cooling method is water cooled oil cooler, the heat rejection to the condenser for the condenser selection shall be as the following:

\[
\text{Heat rejection for condenser selection, Btu/Hr:} = TR \times 12000 + BHP \times 2545 - \text{[Heat removal by water cooled oil cooler]}
\]

Where

- TR: Tons of Refrigeration
- BHP: Horse power input to the compressor

**Control Panel:**

Most standard unit is with a computer type microprocessor control panel with display and keyboard. It provides automatic control for the continuous operation of the unit or even the refrigeration system; to control slide vane to maintain suction pressure or temperature at a set point; to control the variable Vi for the maximum efficiency. The control panel also can provide self-diagnoses and self-check constantly and continuously against the pre-set safety operation set points of the unit. The control panel can communicate with a building automation system. Most panels are with NEMA-1 enclosure as standard from most makers in the world. But, it can be specially modified to suit any electrical code requirement.
FIG. 6.24 Efficiency Comparison & Improvement
Fixed Vi vs Variable Vi
Screw Compressor
Chapter – 7  Centrifugal Compressors

General speaking, multistage centrifugal compressors are used for large industrial refrigeration installations which are beyond the capability of screw. But, the size of screw compressor is getting bigger; therefore, the capacity of screw overlaps the range of centrifugal compressor. However, centrifugal compressor is still being considered for large refrigeration load application, or where the application requires several refrigeration loads at different temperature levels.

Centrifugal is a high flow, variable volume and constant head machine. The characters of centrifugal compressor as compared to reciprocating and screw compressor are as the following:

<table>
<thead>
<tr>
<th></th>
<th>Centrifugal</th>
<th>Screw</th>
<th>Reciprocating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Classification</td>
<td>Aerodynamic</td>
<td>Positive Displacement</td>
<td>Positive Displacement</td>
</tr>
<tr>
<td>Comp. Head</td>
<td>Constant</td>
<td>Variable</td>
<td>Variable</td>
</tr>
<tr>
<td>Volume</td>
<td>Variable</td>
<td>Variable</td>
<td>Constant</td>
</tr>
<tr>
<td>Flow</td>
<td>High</td>
<td>Medium</td>
<td>Low</td>
</tr>
<tr>
<td>Motion</td>
<td>Rotating</td>
<td>Rotating</td>
<td>Reciprocating</td>
</tr>
<tr>
<td>Capacity Control</td>
<td>Continuously</td>
<td>Continuously</td>
<td>Step Control</td>
</tr>
</tbody>
</table>

Single stage centrifugal compressor is mainly used for booster duty in the compound low temperature refrigeration system. It is also used for water chiller or light brine chiller (see Chapter 31) for process cooling application for some petrochemical industries and also for other industries such as electronics and etc.

**Single Stage and Multistage Centrifugal Compressors:**

Single stage centrifugal compressor refers to as one impeller in one compressor casing; multistage centrifugal compressor means having 2 to 8 impellers in one compressor casing. Figure 7-1 (A) is a typical single stage centrifugal compressor; Figure 7-1 (B) is a typical motor driven single stage centrifugal compressor with internal gear and the Figure 7-1 (C) is a typical multistage centrifugal machine without the driver nor steel base.

Figure 7-2 is a typical cutaway for a 3-stage centrifugal compressor and Figure 7-3 is a typical cutaway for a 5-stage centrifugal compressor.

The compression ratio for single stage centrifugal compressor is relatively small. Therefore, the single stage centrifugal is used only as booster duty in low temperature refrigeration systems. The general compression ratio limit for a multistage centrifugal compressor is about 20:1. The practical minimum suction temperature for a multistage centrifugal is about -50°F for R-22, R-290, R-1270 application. The efficiency of a compressor is getting lower when the compression ratio is getting larger.
(A) Single stage centrifugal compressor, no external gear.

(B) Single stage centrifugal compressor drive line with internal gear.

(C) Multistage centrifugal compressor

Figure 7-1 Industrial Refrigeration Centrifugal Compressors
Performance & Exclusive Characteristics of Centrifugal Compressor:

The typical compressor performance curve at the design speed is shown in Figure 7-4. The curve shows that the centrifugal compressor has limited head capability, but, with variable volume characteristic.

The point “D” is the design point of 100% capacity and 100% design head for the compressor at the given speed. The curve shows that the compressor discharge pressure is balanced with relatively large changes in volume flow. For optimum compressor size selection, the compressor should be selected to have the design point close to the right hand side of the curve, such as point “D”, but not too far to the right, that is too close to the stonewall and unstable area if it is beyond the point “C”. The curve between “A” and “D” is the normal operating range. If the design point is located too far to the left of the curve, that means the compressor is too big for the job, otherwise, the compressor is too small if the operating point if further to the right of point “D”.

Centrifugal compressor is with aerodynamic design impeller, unlike the reciprocating and screw compressor, therefore, the centrifugal compressor has its own exclusive characteristics as the following:

Surge: A surge condition for a centrifugal compressor is when the volume flow is too small, cannot satisfy the space gap of the impeller, the system discharge pressure is greater the developed static pressure of the impeller. This causes a back flow of the gas from the system into the impeller until the discharge pressure is achieved. After the back flow slug has been discharged, the compressor is once again faced the problem of insufficient gas flow and the back flow reoccurs. Surge will cause the compressor vibration and damage the compressor.

Surge Point: Point “A” of Figure 7-4 is end of the performance curve. It is the minimum suction flow of an impeller. The compressor operation becomes unstable if the flow is below this minimum flow. This unstable operation manifests itself in the forms of pressure and flow pulsation or oscillation.

Stonewall: Point “B” of Figure 7-4. Stonewall is the opposite of surge for a centrifugal compressor. Stonewall is the choke condition of the compressor. It is the gas flow more than the impeller can handle. The stonewall condition is the gas flow reaches to the point that no flow is possible.

By theory, the compressor will surge if the centrifugal compressor is operated outside of the performance curve. If the compressor is to be operated outside of the curve, the compressor must be equipped with an automatic head control mechanism or partial load control device as required by refrigeration application.

Partial Load Capability:

The partial load control mechanisms available for centrifugal compressor are variable speed control, inlet guide vane, hot gas bypass, or combination of these methods.
Figure 7-3 5-Stage Multistage Centrifugal Compressor
Figure 7-4 Typical Centrifugal Compressor Performance Curve

Figure 7-5 Typical Variable Speed Control
Centrifugal Compressor Performance Curves
Other head control apparatus is the suction damper control. This suction damper valve is to generate artificial head for the satisfaction of the compressor operation. This method is basically used with the compressors that are primarily designed for gas compression per API-617 and is used by oil refinery and gas processing industries for gas compression. The suction damper control is rarely used by refrigeration industries.

**Variable Speed Control:**

If the compressor drive train (compressor, driver and gear if required) is designed for 85% to 110% speed as shown in Figure 7-5, by varying the compressor speed, a family of curves become to a map, the centrifugal compressor is now can be operated any partial load point on the right hand side of the surge line. The speed control is actually shifting the compressor curve until it balances with the new head and volume requirement of the compressor at the partial load conditions. The compressor can even handle the head higher than the design head by increase the speed of the compressor. However, the compressor will surge if the flow is less than the surge point, unless it is equipped with hot gas bypass. The surge line is formed by connecting all the surge points.

From the Figure 7-5, the refrigeration system line is D-F if the refrigeration is operated under constant head mode. The minimum partial load is about 73% at the surge point of “F”. If the system is operating under fallen head mode, then the system line is D-E, the minimum partial load without surge is about 67% at the surge point of “E”. This percent of minimum partial load that the system can be operated without surge is called the Turn-Down Ratio of the refrigeration system.

The drivers for variable speed control are steam turbine, gas turbine, inverter motor and gas engine. The drivers for variable speed control are expensive, particularly the gas turbine. The automatic control is also expensive and complicated.

**Inlet Guide Vane (Pre-Rotation Vane):**

The Inlet Guide Vane is also called Pre-Rotation Vane (PRV). This capacity control is the far most important invention for centrifugal compressor for refrigeration application. Best of all, it is for constant speed motor drive, also the inlet guide vane has the advantage of saving energy during reduce head partial load operation.

The inlet guide vane is located at the suction inlet of the impeller as shown in Figure 7-6. The inlet guide vane operation is fully automatic controlled by the control panel of the compressor unit.

The principle and the accomplishment of inlet guide vane are as the following:

(a) By varying the position of the vanes of the inlet guide vanes, it changes the entrance angles or angle of approach of the inlet refrigerant gas into the impeller wheel.
During partial load operation, the inlet guide vane is closing, the angle of suction gas entering into the impeller is changed; this changes the gas velocity, specific volume and the pressure of the refrigerant gas.
(c) This changes the performance characteristic of the centrifugal impeller without changing the compressor speed.

(d) This is also equivalently having many new compressors for all different operating conditions just by turning the angle of the inlet guide vanes.

The Figure 7-7 is the typical performance curve for the centrifugal compressor with inlet guide vane control with constant speed drive. The performance line of D-A-G-H is the maximum head capability of the compressor. The inlet guide vane allows the centrifugal compressor to operate any points and any conditions below envelop of D-A-G-H without surge. As compared to Figure 7-6, the surge limits are extended and the partial load capability of the compressor is greatly increased without change of compressor speed.

Theoretically, the inlet guide vane could allow the compressor to operate down to 10% part load capacity without surge. But, for refrigeration application, the turn down ratio of a centrifugal compressor depends on the performance curve of the compressor, the “refrigeration system” operating line and the Mach number of the compressor impeller.

Figure 7-8 shows the typical approximate inlet guide vane openings at constant speed. The compressor is operating with the vane fully open on the line of B-D-A. The inlet guide vane is adjusting and changing its position for the compressor to operate all other areas away from full load. The compressor will exceed its head capability if it is operating above the performance curve of D-A; Hot gas bypass is required if the operating conditions of the compressor is located in the area of the left hand side of the surge limit of A-G-H.

Inlet guide vane can be combined with variable speed control and hot gas bypass arrangement.

**Side Load Capability:**

Multistage centrifugal compressor is equivalent of having many compressors in compound compression. Each impeller represents one compressor. Therefore, a flow of refrigerant gas can be introduced into an inlet connection between the impellers of the multistage centrifugal compressor. This flow of refrigerant vapor can be of either from intercool (economizer) or from the suction of refrigeration load at a different temperature ET, as long as the evaporative pressure of this gas flow is acceptable to the interstage pressure between the impellers. This side connection load is referred to as side load for the multistage centrifugal compressor.

Figure 7-9 illustrates a 4-Stage multistage centrifugal compressor system with three evaporative loads at different temperature levels, plus two stages of economizing. Figure 7-10 is the P-H diagram analysis for the 4-stage centrifugal refrigeration system shown in Figure 7-9; Line A-J is the adiabatic compression line; the compressor suction is at point “A”, the gas is compressed by two impellers to point “B”. A-B is a straight line because no intercooling.
Figure 7-7 Inlet Guide Vane Control – Constant Speed Drive

Figure 7-8 Inlet Guide Vane Position – Constant Speed Drive
Figure 7-9 4-Stage Multistage Centrifugal System with Multiple Temperature Level Loads
Point "C" is the mixture temperatures of the gas from points "B" and "D". The gas from point "D" is the combination of the gas of User Load #2 at ET=38°F and the flash gas from the economizer at 38°F; The location of point “C” depends on the size of the User Load #2. The mixture gas from point “C” which is the suction of 3rd stage impeller is compressed to point “E”. The gas is now mixed with the flows from point “G” and entering to the suction of the 4th stage impeller at point “F”. The flow from point “G” is the combination of the gas of User Load #3 at ET=74°F and the flash gas
from the economizer at 74°F. Compressor discharge is at point “H”. The intention of this illustration is only to show the possible side load connections for the multistage centrifugal compressor, it must be checked to see if the side load pressure is high enough to be accepted by the interstage of the compressor.

**Multistage Centrifugal Compressor Size and Stage:**

Multistage centrifugal compressor size is determined by the actual suction CFM flow; the number stages required is determined by the overall compression head, Mach number, tip speed of the impeller and suction acoustic velocity.

The number of stages or impellers required can be very roughly determined by the following formula:

\[
N = \frac{48 \times H_{ad}}{[V_a]^2}
\]

- \(N\) = Number of Stage (Impeller)
- \(H_{ad}\) = Overall Adiabatic Head, ft.
- \(V_a\) = Suction Acoustic Velocity, Ft./Sec.

This formula is only for budget estimate and it is only good for R-22, R-134a, R-290 or R-1270 with reasonable suction superheat. Entrance losses are to be added in the actual case; Compressor tip speed is not to exceed 850 ft/sec., Mach number not over 1.25 and the side load connection should allow 3 to 5 Psi pressure difference between the inlet and the impeller. The compressor selection must be checked and verified by the compressor manufacturer.

**Internal Float Flash Intercooling Economizer:**

Centrifugal compressor is a high volume flow machine. Some maker provides specially designed internal float flash economizer for the centrifugal refrigeration system. The construction of this type of economizer is shown in Figure 7-11.

**Oil System:**

The amount of oil flow for centrifugal compressor is relatively smaller as compared to screw compressor. The Figure 7-12 is the simplified diagram of the oil lubrication system for a centrifugal compressor. The oil cooler can be water cooled or refrigerant cooled or even air cooled. Some industrial users might request for dual oil coolers with change over three-way valve, if water cooled heat exchanger is used. Dual oil filters with change over 3-way valve might be required for continuous operation of the compressor.
Figure 7-11  Internal Float Flash Intercooling Economizer
If the main oil pump is a shaft mounted pump, auxiliary oil should be provided for start-up to establish the oil pressure before the compressor is brought up to full speed. If the main pump is not shaft mounted, a automatic stand-by oil pump might be considered for continuous service of the compressor.

The lubrication system for a refrigeration centrifugal compressor should be exclusively for the compressor and pressurized with the refrigeration system. The lubrication oil for the compressor is special refrigeration oil and suggest not be contaminated with other equipment or other undesirable environment.
API-617 and 614 Specifications:

Some oil refinery and gas processing industries might call for the centrifugal compressor construction to be in accordance with API-617 Specifications. The API-617 specification was written by American Petroleum Institute. The API-617 compressor is basically and primarily designed and constructed for gas compression and gas transition duty; it is not purposely designed for refrigeration application. The API-617 compressor is not equipped with inlet guide vane control; suction damper must be used for head control if the driver is electric motor. API-617 compressor must use a separate lubrication console with separate seal gas in accordance with API-614 Specification.

Driver, Speed Increaser & Drive Line:

The driver for centrifugal compressor may be electric motor, gas engine, steam turbine or even gas turbine. The compressor operating speed is not the same as the driver speed. If the compressor operating speed is much higher than the driver speed, a speed increaser or an external gear should be used to step up the speed. If the driver speed is higher than the compressor speed, a gear also required to step down the speed. Also, a gear might need just to change the direction of rotation for the compressor.

The train of compressor plus gear and the driver is referred to as the drive line or drive train. The drive train must be checked with torsional analysis to avoid critical speed problem.

Control Panel:

The compressor unit shall include the automatic control panel. The panel shall include all the safeties, alarms and shut-downs function and partial load controls similar to screw compressor unit. Electrical compliances shall be in accordance with the codes and regulations listed in Chapter 24.
Chapter – 8  Low Temperature Refrigeration Systems

It is the general guide line that for industrial refrigeration application, when the system compression ratio exceeds 10 to 15, a compound system might start to look attractive, particularly from the view points of power consumption, annual operating expenses and the pay back; A cascade system might be considered if the evaporative temperature is below -60°F or -80°F. The temperature application range might be overlap widely between various systems. But, at the end, the most effective tool for the comparison is still the system economics.

If the project is to handle a large refrigeration load and the process requires several side loads at different temperature levels, a multistage centrifugal compressor might be the candidate for the project. A multistage compressor can handle the ET down to about -40°F; A compound multistage centrifugal system can handle the ET down to about -100°F and a cascade multistage centrifugal compressor system can handle the ET down to -250°F.

**Compound System:**

A compound system is to use two or more compressors, reciprocating, screw or centrifugal, in series using single refrigerant.

A system with multiple compressors in series without any intercooling and desuperheating is considered same as a single compression and it is not considered as a compound system. Therefore, a compound system must include the necessary required components, such as intercooler, economizer or subcooler to improve the system efficiency as described in Chapter 3. The purpose of using a compound system is as the following:

- Increase the refrigeration effect.
- Reduce discharge temperature.
- Reduce the equipment size.
- Reduce the system power consumption.
- Reduce the annual operation expenses.

Most compound systems are 2-stage compression system. A 2-stage system is actually a combination of two single systems with an intermediate flash intercooler or liquid subcooling type intercooler.

The idea intermediate pressure (Pi) of a 2-stage compression is derived from the formula as the following:

\[ \text{Pi} = \sqrt{P_1 \times P_2} \]

The saturated temperature at the intermediate pressure (Pi) is actually the intermediate temperature of the 2-stage compound system.
Figure 8-1 illustrates how a 2-stage compound with flash intercooling refrigeration system is formed:

Diagram-(1) of the Figure 8-1 is the P-H diagram showing CT, ET and the Pi as indicated in the diagram. The diagram shows that the refrigeration effect $\delta H$ is small if the system were on a simple single compression cycle.

Diagram-(2) of the Figure 8-1 is the P-H diagram for the low stage circuit of the supposed to be compound refrigeration system. The refrigerant liquid to evaporator is now from the intermediate temperature; the intermediate temperature is considered as the condensing temperature for the low stage compressor. As indicated in the diagram, it is evident that by using flash intercooler for the compound system, the refrigeration effect $\delta H'$ is now increased and became much larger as compared to the $\delta H$ shown in Diagram-(1).

Diagram-(3) of the Figure 8-1 is the P-H diagram for the high stage circuit of the supposed to be compound refrigeration system. The intermediate temperature is considered the evaporative temperature for the high stage compressor.

Putting the Diagram-(2) and Diagram-(3) together, it shall be the P-H diagram for this 2-stage screw compound system with flash intercooler as shown in the Diagram-(4) of the Figure 8-1.

The Refrigerant Flow Diagram for this 2-Stage compound system with the flash intercooler is shown in Figure 8-2. The flash intercooler is also providing the desuperheating function for the system.

If the 2-stage compound system is with a shell-and-tube type liquid subcooling intermediate intercooler instead of flash type, the system is shown in Figure 8-3. This type of intercooler provides high pressure subcooled liquid to the evaporator.

The Pi is usually obtained by the idea intermediate pressure formula of square root of $(P_1 \times P_2$). However, it shall be always a good practice to recheck the actual compressor performances to see if the idea Pi yields the optimum efficiencies of both high and low side compressors. Sometimes, the system efficiency can be improved a lot by readjusting the intermediate pressure (Pi) for the actual compressor.

Besides of the intermediate intercooler between the high and low stage, additional intercooler and/or economizer can be used where ever is feasible for the high side and/or the low side compressor to improve the system efficiency.

There is no restriction on what type of compressor to be used for the compound system. However, it is a general practice to use low or medium flow with variable head compressor (reciprocating or screw) as the high stage; use medium or high flow with constant head compressor (screw or centrifugal) as the low stage.

**Multistage Centrifugal System:**

Refrigeration system utilizes one multistage centrifugal compressor, no matter how
Figure 8-1 Structuring a 2-stage Compound System
Figure 8-2 Refrigerant Flow Diagram
2-stage Screw Compound System
With Flash Intercooler
Figure 8-3 Refrigerant Flow Diagram
2-Stage Screw Compound System
With Liquid Subcooling Intercooler
many stages are involved and how complicated the system is, the system is classified as multistage centrifugal system, not compound system, unless the system is with two or more multistage centrifugal compressors.

Figure 8-4 is the refrigerant flow diagram for a refrigeration system with 6-stage centrifugal compressor. The system uses two shell-and-tube type liquid subcoolers to increase the refrigerant effect. The suction gas from the 52°F subcooler is connected to the suction of the 2nd stage impeller of the compressor and the suction gas from the 3°F subcooler is connected to the 4th stage impeller of the compressor. The suction gas from the 35.1°F evaporator is returned to the 3rd stage impeller of the compressor as indicated. The refrigerant use in this case is Propylene (R-1270) which is a commonly used refrigerant for refrigeration systems for hydrocarbon processing industries.

Figure 8-5 is the P-H diagram analysis for this 6-stage centrifugal refrigeration system as shown in Figure 8-4.

Note: The minimum suction gas density for centrifugal compressor when it is used in a compound system is not to be less than 0.01 Lbs/Ft³. The compressor selection might not be feasible if the suction gas density is below this level.

**Booster Compound System:**

When the low stage compressor of a compound system is designed to handle a smaller head as compared to the high stage compressor; this low stage compressor is called the booster compressor. The reasons that might prompt the use of booster compound are as the following:

(I) To improve the high stage compressor efficiency.

(II) To handle evaporative loads at different temperature levels; the intermediate pressure can be set equal to the side load ET and logically it can be handled by the suction of the high stage compressor.

(III) System economic justifications and other considerations.

Figure 8-6 is a typical booster compound refrigeration system using single stage centrifugal compressor as the booster and a screw compressor as the high stage. The advantage of using this screw and centrifugal combination is that constant head machine (centrifugal) for the booster and variable head machine (screw) for the high stage.

The Figure 8-7 is the P-H diagram analysis for this refrigeration system as shown in Figure 8-6.

**Cascade System:**

As a general guide line, cascade system might be considered for one or more following reasons:
Figure 8-4 R-1270 (Propylene) Refrigerant Flow Diagram
6-Stage Multistage Centrifugal Refrigeration System
R-1270 (PROPYLENE)
P-H DIAGRAM ANALYSIS

Figure 8-5 R-1270 P-H Diagram
For 6-Stage Multistage Centrifugal Refrigeration System
Figure 8-6 Refrigerant Flow Diagram R-22
Centrifugal Booster / Screw Compound System
A. When compound system cannot meet the design evaporative temperature as required.

B. When the suction temperature is too low, the suction pressure cannot have a stable operation of the compressor.

C. When the suction specific volume of the refrigerant gas is too large, the initial investment for the compressor and system are too expensive.
D. Positive pressure requirement; the low side is to be operated above atmospheric pressure.

E. When the overall temperature differential exceeds the practical range of one refrigerant.

F. When the ET is below -80°F.

Cascade system is made up by two entirely separate independent refrigerant circuits as shown in Figure 8-8, each using a refrigerant appropriate for its temperature range. Heat is exchanged through a cascade condenser between the high side and the low side circuits. The high side is the high temperature system and the low side is the low temperature system.

The cascade condenser is a shell-and-tube heat exchanger to tie the low side and high side systems together. The refrigerant gas discharge from the low side compressor is condensed inside the tubes of the cascade condenser. The cascade condenser is also the evaporator for the high temperature system; the high side refrigerant liquid in the shell absorbs the heat from low side, evaporates and returns to the suction of high side compressor.

The evaporative temperature for the high stage for the cascade system depends on the evaporative temperature level of the low stage; it is usually in the range of 0°F to -30°F. The refrigerant use for the high stage circuit may be R-22, R-717, R-290 (Propane) or R-1270 (Propylene). The low temperature cycle usually uses a high pressure and low specific volume refrigerant, such as R-23, R-116, R-508B, R-170 (Ethane) or R-1150 (Ethylene); the sole purpose of using these refrigerants for the low side is to minimize the size and number of compressors.

The specific volume of the low side refrigerants is relatively small at the low temperature. However, the specific volume is getting larger and the pressure is getting higher when the temperature increases to the stand-by ambient temperature. In order to avoid high design working pressure (DWP) for the low side, an expansion tank is to be sized large enough to permit vaporization of all the refrigerant charge of the low side during the system is shut-down and warming up to the ambient temperature. The DWP of the system should be 300 Psig and the expansion tank volume should be sized large enough to limit the system pressure to be below 225 Psig.

Some case, a Hold-down System is provided for the cascade system. The hold-down system is a small refrigeration system to keep the low side cold when the cascade system is shut-sown. For safety consideration, it is suggested that the hold-down system should not replace the need of an expansion tank.
Figure 8-8 Typical Cascade Refrigeration System
Chapter – 9  Open Cycle Refrigeration System

An open cycle refrigeration system is that the system is without a traditional evaporator. The suction gas for the refrigeration system might be from a storage tank or from a pipeline, the refrigeration system is to liquefy the gas and send the liquid to other facility or to a storage tank. The refrigerant of the system is an open flow and is not a closed recirculation circuit. The typical open cycle refrigeration system is that the system for Ammonia storage or LPG storage.

**Ammonia Storage:**

Refrigeration system is needed for ammonia storage for the following reasons:

(1) When the ammonia liquid coming out from the fertilizer plant, the pressure and temperature are relatively high. Usually, the working pressure for the storage tank is constructed for just slightly above the atmospheric pressure. Therefore, the pressure and the temperature of the ammonia liquid from pipe line are to be reduced before it is stored in the storage tank.

(2) The ammonia liquid inside the tank is boiled off due to heat gain through the storage tank. This boiled off gas must be re-condensed to avoid pressure build up in the tank; the re-condensed ammonia liquid is to be returned back to storage tank.

(3) To condense the flash gas from the loading system.

(4) To re-condense the flash gas from the liquid throttling to the storage tank.

An example is shown in Figure 9-1. The Ammonia liquid flow from pipeline is 98.6 °F and 242.22 Psia; the compressor suction flow is the boiled off gas and the flash gas from the storage tank and the flash gas from the loading system.

The refrigeration system shown in Figure 9-2 is designed to handle the ammonia liquid flow from the pipe line and to re-condense the gases from the storage tank and from the loading system. This refrigeration system is a compound system with a 7-stage multistage centrifugal compressor as the low stage and a 7-stage multistage centrifugal compressor as the high stage. The intermediate temperature of this compound system is 27°F.

The Figure 9-3 is the P-H diagram for this compound system. The ammonia liquid from the pipe line is first throttling down to 126.5 Psia pressure level, then drops down to 59 Psia, then 44.12 Psia. The suction flow of the low stage compressor is from the boiled off gas and the flash gases. Side load of low stage compressor is the flash gas from the 16 °F intercooler and it is connected to the inlet of the 6th stage impeller. The side load for the high stage compressor is the flash gas from 69 °F intercooler; this side load is connected to the inlet of the 5th stage impeller of the high
Figure 9-1 Ammonia Storage

- Pipeline: Liquid from pipeline 98.6°F 242.22 Psia
- Refrigeration System: Vapor to Refrigeration system 15.2 Psia -27.4°F
- Liquid to storage tank 16°F 44.12 Psia
- Ammonia Storage Tank: Boil off gas & Flash gas from storage tank 15.2 Psia -27.4°F
- Loading System: Vapor from Loading System -27.4°F
Figure 9-2 Open Cycle Refrigeration System
For Ammonia Storage
Figure 9-3 P-H Diagram for Ammonia Storage
stage compressor. The liquid leaves from the low stage intercooler at 16°F 44.12 Psia is returned to the storage tank through a throttling valve.

The liquid leaves the refrigeration system in this case is saturated liquid at 44.12 Psia. The system should be modified to provide subcooled liquid instead of saturated if the pipe line is with a vertical lift.

Screw compressor instead of centrifugal should be considered if the ammonia flow rates are more suitable for the capability of screw compressors.

LPG Storage:

LPG (Liquefied Propane Gas) storage is another typical application of open cycle refrigeration. When producing LPG, it also produces Butane and Pentane in most cases. Therefore, LPG storage refrigeration system shown in Figure 9-4 also includes the Butane and Pentane storage.

All the storage tanks are designed for working pressure about atmospheric pressure. The refrigeration system is to reduce the pressure inside the storage tank by liquefying the boiled off gases and return the condensed liquid back to the storage tank.

The Figure 9-4 is a refrigeration system to liquefy the propane boiled off gas. It also uses propane as the refrigerant to condense the boiled off gases of Butane and Pentane. The Figure 9-5 is the P-H diagram for this refrigeration system.

This system was designed for a LPG terminal where the ambient temperature is high and water supply is a problem. Therefore, air cooled condenser is used. An 8-stage centrifugal multistage compressor was used to allow different evaporative temperature levels for Butane and Pentane condensing.

The description of the refrigeration system is as the following:

Condensing temperature is 135°F.

Compressor main suction gas is the boiled off gas from the Propane Storage Tank at -40°F and 14.7 Psia.

Use Propane as the refrigerant for two evaporators. One is the evaporator for Pentane condensing. ET is 110°F. Suction from this evaporator is connected to the inlet of 8th stage impeller. Another evaporator is for Butane condensing. ET of 0°F. The suction from this evaporator is connected to the inlet of the 3rd stage impeller.

Three flash intercoolers are used for the compressor: (1) at 83°F 149.89 Psia, flash gas inlet to the 7th stage impeller; (2) at 37°F, 74.73 Psia, flash gas inlet to the 5th stage impeller and (3) at 0°F 38.3 Psia, flash gas inlet to the 3rd stage impeller of the compressor.

A two-stage flash intercooler is used, for the 37°F and 83°F intercooling.
Figure 9-4  Refrigeration System for Propane, Butane and Pentane Storage
There are two streams of side loads flow from 0°F 38.3 Psia to the inlet of 3rd stage impeller of the compressor; one is the flow from the flash intercooler and the other is the vapor flow from the evaporator for the Butane condensing.

Multistage centrifugal compressor is used for this example. The refrigeration system can be designed by using screw compressor.
Chapter – 10  System Balancing

The major components of a refrigeration system are the compressor, condenser and the evaporator. These three components are the structure of the triangle of the refrigeration system. Each component has its performance curve under the design operating conditions and the balance point of these three curves shall be the exact refrigeration capacity of the system at the design conditions. Changing any one of the three components might change the shape of the triangle and therefore, changes the character and the capacity of the system.

Figure 10-1 is the typical performance curves for a compressor. At the constant condensing temperature, say 110˚F, the compressor capacities at various saturated suction temperatures are:

<table>
<thead>
<tr>
<th>Saturated Suction</th>
<th>Compressor Capacity, TR</th>
</tr>
</thead>
<tbody>
<tr>
<td>30 F</td>
<td>185 TR</td>
</tr>
<tr>
<td>20 F</td>
<td>147 TR</td>
</tr>
<tr>
<td>10 F</td>
<td>116 TR</td>
</tr>
<tr>
<td>0 F</td>
<td>90 TR</td>
</tr>
<tr>
<td>-10 F</td>
<td>67 TR</td>
</tr>
<tr>
<td>-20 F</td>
<td>49 TR</td>
</tr>
</tbody>
</table>

From the Table A-1, at constant condensing temperature, the compressor capacity is higher if the suction temperature is higher and the capacity is smaller if the suction temperature is lower.

From Figure 10-1, if the suction temperature is constant, the compressor capacities at various condensing temperatures are as the following:

<table>
<thead>
<tr>
<th>Condensing Temperature</th>
<th>Constant Saturated Suction Temp. 0 F</th>
<th>Constant Saturated Suction Temp. 20 F</th>
</tr>
</thead>
<tbody>
<tr>
<td>80 F</td>
<td>112 TR</td>
<td>178 TR</td>
</tr>
<tr>
<td>90 F</td>
<td>105 TR</td>
<td>168 TR</td>
</tr>
<tr>
<td>100 F</td>
<td>98 TR</td>
<td>158 TR</td>
</tr>
<tr>
<td>110 F</td>
<td>90 TR</td>
<td>148 TR</td>
</tr>
<tr>
<td>120 F</td>
<td>82 TR</td>
<td>138 TR</td>
</tr>
<tr>
<td>130 F</td>
<td>74 TR</td>
<td>127 TR</td>
</tr>
</tbody>
</table>

From Table A-2, if the compressor suction temperature is constant, the compressor capacity is higher if the condensing temperature is lower and the compressor capacity
Figure 10-1   Typical Compressor Performance Curves
is smaller if the condensing temperature is higher. In general speaking, the compressor capacities are higher at various condensing temperatures if the suction temperature of the compressor is higher as shown.

The Figure 10-2 is the characteristic and performance lines representing the major components of compressor, condenser and evaporator. The diagram [A] of the Figure 10-2 is the compressor general curves. The horizontal axis represents the capacity and the vertical axis is the condensing temperature. The performance line at the left is the compressor performance line of a lower constant suction temperature and the right hand side line is the compressor performance of a higher constant suction temperature.

The diagram [B] of Figure 10-2 is the condenser characteristic line at a fixed GPM/TR and fixed cooling water temperatures range, a larger condenser with higher Sq.Ft./TR heat transfer surface will have higher capacity and have lower condensing temperatures; likewise, a smaller condenser will have higher condensing temperature and smaller capacities.

The diagram [C] of Figure 10-2 is the general curves for an evaporator. At the same capacity, the smaller size evaporator will result in lower evaporative temperature while higher ET for the larger evaporator; larger size evaporator produces more TR than the smaller evaporator is having smaller tonnage at the same ET.

By combining all the general characteristics of the major components as described as the above, by plotting the actual performance data of each compressor, condenser and the evaporator, a typical system balancing chart is shown in Figure 10-3; the horizontal axis is the TR of the system; the upper vertical line is the condensing temperature and the lower vertical axis is the evaporative temperature of the system.

The Figure 10-3 shows the system balance chart for the compressor of model R-blv2, R-22 at the compressor speed of 1450 RPM, the condenser is a model HC-xlb6, 2-Pass arrangement, 85 to 95 °F cooling water range at 3 GPM/TR, 0.001 scale factor and the brine cooler is model C-R2B2 at leaving brine temperature of -5 °F, 40% by wt. of Ethanol brine, 0.001 scale factor, 2-pass arrangement.

The compressor model R-blv2 performance lines are shown; the left hand side line (1) is the compressor at constant suction temperature of -20 °F and the right hand side line (2) is the compressor at the constant suction temperature of -10 °F. The condenser performance line is the line (3); the line (3) intersects with the lines (1) and (2), the line between points of (A) and (B) is the combined performance curve of the condenser HC-xlb6 and the compressor R-blv2., representing the compressor and the condenser at the various capacities at the corresponding condensing temperature at the design operating condition for the compressor and the condenser. For example, the compressor and condenser will produce 29 TR at condensing temperature of 101.6 °F.

Locate point (C) by projecting point (A) which is the compressor suction at -20 °F, also locate point (D) by projecting point (B) which is the compressor suction at -10 °F. Connecting point (C) and (D), this line represents the combined compressor and the condenser performance at various points for the compressor and the condenser with
Figure 10-2  Characteristics of Major Components of Refrigeration System
Figure 10-3  System Balancing
suction temperature added.

Now, plotting the brine cooler performance line (4) against the TR and ET on the chart, the intersection point of line (4) and line (C)-(D) is the point (E) which shall the balance point of the compressor, condenser and the brine cooler at the design operating conditions.

The balance point of the system using the specified three components is shown as:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>System Capacity</td>
<td>31.1 TR</td>
</tr>
<tr>
<td>Saturated Suction</td>
<td>-13.8°F</td>
</tr>
<tr>
<td>Condensing Temperature</td>
<td>102.3°F</td>
</tr>
<tr>
<td>Refrigerant</td>
<td>R-22</td>
</tr>
<tr>
<td>Compressor Model</td>
<td>R-blv2</td>
</tr>
<tr>
<td>Compressor Speed</td>
<td>1450 RPM</td>
</tr>
<tr>
<td>Condenser Model</td>
<td>HC-xlb6</td>
</tr>
<tr>
<td>Condenser Pass</td>
<td>2-Pass</td>
</tr>
<tr>
<td>Cooling Water Temperature</td>
<td>85 to 95°F</td>
</tr>
<tr>
<td>Condenser Scale</td>
<td>0.001</td>
</tr>
<tr>
<td>Cooler Model</td>
<td>C-R2B2</td>
</tr>
<tr>
<td>Leaving Brine Temperature</td>
<td>-5°F</td>
</tr>
<tr>
<td>Brine</td>
<td>40% by wt. of Ethanol brine</td>
</tr>
<tr>
<td>Cooler Scale Factor</td>
<td>0.001</td>
</tr>
<tr>
<td>Cooler Pass</td>
<td>2-Pass</td>
</tr>
</tbody>
</table>
Chapter – 11  Partial Load Performance
          Of Refrigeration System

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 11-3  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.

![Diagram](image)

(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 $\delta T$ 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.

\[
\delta T = CT - T_w
\]

$\delta T$ = Small Difference, °F
CT = Condensing Temperature, °F
$T_w$ = Leaving Condenser Water Temperature, °F

Therefore: $CT = \delta T + T_w$

CONDENSING TEMPERATURE AT 100% SYSTEM DESIGN LOAD:

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 $\delta T = $ 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.

CONDENSING TEMPERATURE AT 60% SYSTEM PARTIAL LOAD:

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
\( \text{Ft}^2/\text{Ton} = 1,500/156 = 9.6 \)

From Figure 11-4, the \( \delta T = \text{Small Difference} = 4.5^\circ F \)

Entering cooling water temperature = 90\(^\circ\)F (Constant)

Leaving cooling water temperature is now = 90\(^\circ\)F + 6\(^\circ\)F = 96\(^\circ\)F

Therefore:

\[
\text{CT} = \delta T + T_w = 4.5 + 96 = 100.5^\circ F
\]

The condensing temperature is now 100.5\(^\circ\)F at 60% partial load and it is no longer at 107.2\(^\circ\)F as design. Therefore, the compressor head is changed even the entering cooling water temperature entering to the condenser is kept constant at 90\(^\circ\)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 \( \text{Ft}^2/\text{Ton} \); the vertical axis is the \( \delta T \) 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.

\[
\delta T = T_w - \text{ET}
\]

\[
\delta T = \text{Small Difference, } ^\circ F
\]

\[
\text{ET} = \text{Evaporative Temperature, } ^\circ F
\]

\[
T_w = \text{Leaving Chilled Water Temperature, } ^\circ F
\]

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

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

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

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

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

From Figure 11-5, the \( \delta T = \text{Small Difference} = 7.0^\circ F \)

Therefore:

\[
\text{ET} = T_w - \delta T = 40 - 7.0 = 33^\circ F
\]

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

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

Refrigeration load at 60% load = 200 x 0.6 = 120 TR
\[
\text{Ft}^2/\text{Ton} = 1,500/120 = 12.5
\]
From Figure 11-5 the \( \delta T \) = Small Difference = 6.0°F
Leaving chilled water temperature = 40°F (Constant)

Therefore:

\[
\text{ET} = T_w - \delta T = 40 - 6 = 34°F
\]

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.
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 ft²°F-Hr/Btu
  - Cooler: 0.002 ft²°F-Hr/Btu
- Condensing temperature: 115°F
- Evaporative temperature: 13°F

SUMMER PEAK LOAD OPERATION:

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

OFF-SEASONS AND WINTER OPERATION:

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

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:**

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:

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

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.
Chapter – 12  Heat Exchangers for Industrial Refrigeration Systems

This section deals with the heat exchangers used for industrial refrigeration application, such as air cooled condenser, evaporative condenser, water cooled condenser and shell-and-tube evaporators. Other types of heat exchangers such as pipe-coil, unit coolers, product coolers, plate type heat exchanger, coil-and-drum, pipe-in-pipe, air-to-air, fluid-to-gas or gas-to-fluid or oil coolers are not covered in this section.

**Condensing Ton and Evaporative Ton:**

When dealing with heat transfer, the heat transfer unit is Btu/Hr. Evaporative ton or ton of refrigeration is 12,000 Btu/Hr. However, if the heat transfer ton is referred to the heat rejection from condenser, it is the condensing ton. The condensing ton is not 12,000 Btu/Hr, it is 14,545 Btu/Hr.

\[
\text{Condensing Ton} = 1 \text{ TR} + 1 \text{ HP} = 12,000 + 2,545 = 14,545 \text{ Btu/Hr}
\]

The condensing ton is only used for the term of Sq.Ft/ton for water cooled condenser curves. To avoid any confusion, the condensing ton is not used for heat rejection calculation.

**System Heat Rejection to Condenser:**

\[
\text{Heat rejection} = \text{TR} \times 12000 + \text{BHP} \times 2545
\]

<table>
<thead>
<tr>
<th>Heat rejection:</th>
<th>Btu/Hr</th>
</tr>
</thead>
<tbody>
<tr>
<td>TR:</td>
<td>Tons of Refrigeration</td>
</tr>
<tr>
<td>BHP:</td>
<td>Compressor BHP at design load</td>
</tr>
</tbody>
</table>

For example: 500 TR, power consumption is 625 BHP

\[
\text{Heat rejection} = 500 \times 12,000 + 625 \times 2545 \\
= 6,000,000 + 1,590,625 \\
= 7,590,625 \text{ Btu/Hr}
\]

Note: If screw compressor is used and it is with water cooled oil cooling, the heat rejection to condenser should be:

\[
\text{Heat rejection} = \text{TR} \times 12000 + \text{BHP} \times 2545 - \text{Oil Cooling Heat Removal}
\]
Air Cooled Condensers

Basically, air cooled condensers are mostly used for the installation where water is not available or it is economically justified to use air cooled instead of water cooled for large refrigeration installation. In most cases, use of air cooled condenser could result in higher cost and higher power consumption as compared to water cooled condenser project.

Figure 12-1 shows a typical construction of an air cooled condenser. An air cooled condenser consists of a fan section, a coil section, casing and supporting structure. Figure 12-2 shows the belt driving fan assembly for the air cooled condenser. Figure 12-3 is a smaller size air cooled condenser and the Figure 12-4 is a typical larger size air cooled condenser.

The performance of the air cooled condenser is as the following:

\[ CT = ADB + TD \]

- \( CT \) = Condensing Temperature, \(^{\circ}F\)
- \( ADB \) = Entering Dry Bulb Air Temperature, \(^{\circ}F\)
- \( TD \) = Approach, \(^{\circ}F\)

The approach for the air cooled condenser is the temperature difference between the design condensing temperature and the design outdoor ambient temperature; smaller approach requires a larger size condenser because more heat transfer coil surface is needed; larger TD permits a smaller air cooled condenser because less heat transfer coil surface is required for the heat rejection.

Figure 12-5 is the typical air cooled condenser performance data sheet. The performance sheet shows the condenser has less heat rejection capacity at a smaller TD and higher heat rejection capacity at a higher TD.

What TD should be used depends on the maximum DWP (design working pressure) of the condensing coil and the refrigerant being used. Most installations use a TD between 15\(^{\circ}F\) to 30\(^{\circ}F\). The air cooled condenser is getting expensive if the TD is less than 15\(^{\circ}F\) and the DWP of high side is getting too high if the TD is more than 30\(^{\circ}F\), unless the ambient design dry bulb temperature is low. The power consumption will be higher if the TD is higher for the system; the condenser size is smaller, but the compressor and driving motor is more expensive.

For small and medium size installation, the condenser can have either horizontal fan discharge or vertical discharge. Also, two types of fan draft flow for the condenser: one is Draw Through (Induced Draft) design and the other is Blow Through (Forced Draft) design.

Most large size air cooled condensers are with vertical air discharge. Figure 12-6 shows a typical large air cooled condenser with a forced draft fan; Figure 12-7 is a typical large air cooled condenser with an induced draft fan.
Figure 12-1  Typical Air Cooled Condenser

Figure 12-2  Fan Drive Assembly for Air Cooled Condenser
Figure 12-3  Smaller Size Air Cooled Condenser

Figure 12-4  Larger Size Air Cooled Condenser
### Single Circuit Condensers-Refrigerant-12, -22 and -502

**Condenser Heat Rejection Capacity (BTU/hr)**

<table>
<thead>
<tr>
<th>UNIT SIZE</th>
<th>&quot;TD&quot; (°F) + Condenser Condensing Temp. - Entering Air Dry Bulb Temp.</th>
<th>10</th>
<th>15</th>
<th>20</th>
<th>25</th>
<th>30</th>
<th>35</th>
<th>40</th>
<th>45</th>
<th>50</th>
</tr>
</thead>
<tbody>
<tr>
<td>36</td>
<td>23,350</td>
<td>35,000</td>
<td>46,700</td>
<td>58,250</td>
<td>70,000</td>
<td>81,600</td>
<td>91,400</td>
<td>105,000</td>
<td>118,500</td>
<td></td>
</tr>
<tr>
<td>37</td>
<td>27,350</td>
<td>41,000</td>
<td>54,700</td>
<td>68,250</td>
<td>82,000</td>
<td>95,600</td>
<td>109,400</td>
<td>123,000</td>
<td>136,500</td>
<td></td>
</tr>
<tr>
<td>39</td>
<td>30,350</td>
<td>45,500</td>
<td>60,700</td>
<td>75,750</td>
<td>91,000</td>
<td>106,100</td>
<td>121,400</td>
<td>135,500</td>
<td>149,100</td>
<td></td>
</tr>
<tr>
<td>41</td>
<td>36,000</td>
<td>61,000</td>
<td>72,000</td>
<td>90,000</td>
<td>108,000</td>
<td>126,000</td>
<td>145,000</td>
<td>162,000</td>
<td>180,000</td>
<td></td>
</tr>
<tr>
<td>54</td>
<td>42,000</td>
<td>63,000</td>
<td>81,000</td>
<td>105,000</td>
<td>128,000</td>
<td>147,000</td>
<td>168,000</td>
<td>187,000</td>
<td>210,000</td>
<td></td>
</tr>
<tr>
<td>61</td>
<td>47,000</td>
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<td>94,000</td>
<td>117,500</td>
<td>141,000</td>
<td>164,500</td>
<td>186,000</td>
<td>211,500</td>
<td>235,000</td>
<td></td>
</tr>
<tr>
<td>71</td>
<td>56,000</td>
<td>84,000</td>
<td>112,000</td>
<td>140,000</td>
<td>168,000</td>
<td>198,000</td>
<td>226,000</td>
<td>252,000</td>
<td>280,000</td>
<td></td>
</tr>
<tr>
<td>111</td>
<td>61,600</td>
<td>92,500</td>
<td>123,000</td>
<td>155,000</td>
<td>185,000</td>
<td>216,000</td>
<td>247,000</td>
<td>277,000</td>
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</tr>
<tr>
<td>142</td>
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<td>143,500</td>
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<td>322,500</td>
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<td>209,000</td>
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<td>335,000</td>
<td>372,000</td>
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<td>187</td>
<td>93,500</td>
<td>140,000</td>
<td>187,000</td>
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<td>420,000</td>
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<td>210,000</td>
<td>275,000</td>
<td>330,000</td>
<td>385,000</td>
<td>440,000</td>
<td>495,000</td>
<td>550,000</td>
<td></td>
</tr>
<tr>
<td>242</td>
<td>121,500</td>
<td>182,500</td>
<td>243,000</td>
<td>305,000</td>
<td>365,000</td>
<td>426,000</td>
<td>490,000</td>
<td>548,000</td>
<td>608,000</td>
<td></td>
</tr>
<tr>
<td>289</td>
<td>145,000</td>
<td>217,500</td>
<td>290,000</td>
<td>362,500</td>
<td>435,000</td>
<td>507,000</td>
<td>580,000</td>
<td>653,000</td>
<td>725,000</td>
<td></td>
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<tr>
<td>339</td>
<td>170,000</td>
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<td>855,000</td>
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<td>534,000</td>
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<td>1,190,000</td>
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<td>1,440,000</td>
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<td>1,690,000</td>
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<td>2,535,000</td>
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<td>1,245,000</td>
<td>1,595,000</td>
<td>1,865,000</td>
<td>2,135,000</td>
<td>2,400,000</td>
<td>2,795,000</td>
<td>3,110,000</td>
<td></td>
</tr>
</tbody>
</table>

**Figure 12-5**  Typical Performance Data

R-22 Air Cooled Condenser
Advantages of Induced Draft Fan:

Less Air Recirculation – The air from induced draft unit exhausts about 2.5 times the velocity of the forced draft unit thereby moving the air further away from the unit, minimize the possibility of recirculation of hot air.

Figure 12-6  Forced Draft Fan Design
Larger Size Air Cooled Condenser

Figure 12-7  Induced Draft Fan Design
Larger Size Air Cooled Condenser
Induced draft units are more efficient – The inlet air more uniformly covers the bottom rows of tubes.

Might require less fan horsepower – Induced Draft unit could move more air at same fan horsepower than Forced Draft unit.

Protect bundles from atmospheric corrosion – The exhaust hoods of the Induced Draft unit cover the top of the units; minimize the amount of rain that drops into the fan ring. It might require less maintenance for corrosion protection.

Better operational heat transfer – Hood cover shields the tube from direct rays of the sun, more uniform cooling.

**Minimum data required for air cooled condenser selection and pricing:**

The followings are the minimum data required to make an air cooled condenser selection or to make quotation proposal:

1. Heat rejection, Btu/HR.
2. Refrigerant.
3. Ambient air temperature.
5. Vertical or horizontal arrangement, Blow Through or Draw Through fan.
6. Centrifugal, propeller or vane axial fan.
7. Type of motor, power supply and code requirements.
8. Special coil material requirement, if any.
9. Special supporting structure, if any.
10. Type of head control, if required.

**Head Control Requirement:**

Since air cooled condensers are often required to operate over a wide range of ambient air temperatures and variable loading conditions, therefore, air cooled condenser tends to run a very low head pressure when operating in a low ambient air temperature. It is a good practice to take advantage of the falling condensing temperature to save the power consumption. However, the system must maintain a minimum head pressure in order to keep system pressure differential high enough to allow the controls and expansion devices function properly. Therefore, refrigeration application should have head control provision to allow the system operating pressure to vary with the ambient air temperature within certain limits, but without adversely affecting the system operation.

**Various Head Controls for Air Cooled Condenser:**

A) Fan Cycling Control.

Individual fan of a multiple fans can be cycled to maintain the condensing temperature of the system. This is most commonly used control for air cooled condenser particularly for small size installations.
B) Face Dampers.

This is good for low ambient temperature installation. Air flow reduction through the condenser coil can be accomplished with closing the face damper. Damper motor(s) is controlled by ambient sensing thermostat or condenser head pressure sensing switch.

C) Flooding-Type Condenser Control and Low Ambient Operation.

Head control valve will partially close the liquid to receiver to flood the condenser coil to maintain the preset pressure for the condenser. An automatic hot gas valve provides pressure differential between compressor discharge and receiver of no less than 30 Psi.

This type of condenser pressure control is effective for cold climates even below 0°F.

The receiver must be sized large enough to hold the extra amount of liquid for the coil flooding.

Low Ambient Bypass Timer - A five minute time delay to bypass the low pressure cutout during low ambient startup.

D) Two-Speed Fan Control or Variable Speed Fan Control.

Two speed fan control or variable fan speed control provides more fine tune of head control for the air cooled condenser.

Variable Fan Pitch Head Control for Large Size Air Cooled Condenser:

In additional to the methods of the head controls available for smaller size air cooled condenser, the other head control available for larger size air condenser is the Automatic Variable Pitch Fan Control. This is to control the pitch of the fan to vary the air volume flow to control the temperature or condensing pressure.

System Shut Down During Cold Ambient:

In case the refrigeration system is shut down during cold ambient, protection against refrigerant migrant in the system should be considered. These protections are such as drain off process fluid in the evaporator if the process fluid will freeze in low ambient; or the refrigerant in the system is to be pumped out into storage receiver or the service valves for the condenser are to be shut off.

Piping and Parallel Multiple Unit Application:

Piping hook-up and the recommendations for parallel multiple units operation are mostly the same as for the evaporative condenser.
Evaporative Condensers

Evaporative condenser is the same as air cooled condenser except water spray is added to wet the entire condensing coil. The water circulation rate for the evaporative condenser is about 1 GPM/Ton.

Most evaporative condensers are with vertical discharge. Fan draft flow is Blow Thru or Induce Draft. Figure 12-8 shows the Blow Thru type evaporative condenser. Figure 12-9 shows the Induce Draft type design. Fans available for evaporative condenser are axial, vane axial or centrifugal.

The performance of an evaporative condenser is rated by °F of Approach.

\[ \text{Deg.F Approach} = \text{CT} - \text{FWB} \]

\[ \text{CT} = \text{Condensing Temperature, °F} \]
\[ \text{FWB} = \text{Wetbulb Air Temperature, °F} \]

Therefore:

\[ \text{CT} = \text{°F Approach} + \text{FWB} \]

For example: Wetbulb Air Temperature is 80°F
Approach is 10°F

\[ \text{CT} = 10°F + 80°F = 90°F \]

Figure 12-10 is the typical heat rejection factors for R-22 for various approaches between condensing temperature and air °FWB for a typical evaporative condenser; Figure 12-11 is the typical heat rejection factors for R-717 at various approaches. Those heat rejection factors are to be applied for the determination of the size of the evaporative condenser. Higher the factor, larger size of the evaporative condenser is required. The heat rejection factor will be larger if the approach between the CT and °FWB is smaller.

Generally speaking, the minimum approach is about 10°F for most applications. For special application the approach could be 5~6°F; however, the evaporative condenser would be huge and expensive.

**Head Control Methods for Evaporative Condenser:**

The head controls for evaporative condenser, such as variable speed drive and multiple fan cycling are the same as for the air cooled condenser except the following:

1.0 Automatic Capacity Control Damper control:

(a) Damper installed in the centrifugal fan housings – This is to use the damper to modulate the quantity of air flowing through the unit. When
Figure 12-8  Blow Thru Type Evaporative Condenser
Figure 12-9  Induce Draft Type Evaporative Condenser
<table>
<thead>
<tr>
<th>Condensing Pressure (PSIG)</th>
<th>Cond. Temp. (°F)</th>
<th>Entering Air Wet Bulb Temperature (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>50</td>
<td>55</td>
</tr>
<tr>
<td>R-12</td>
<td></td>
<td>85</td>
</tr>
<tr>
<td>R-22</td>
<td></td>
<td>85</td>
</tr>
<tr>
<td>91.8</td>
<td>155.8</td>
<td>85</td>
</tr>
<tr>
<td>99.8</td>
<td>168.4</td>
<td>90</td>
</tr>
<tr>
<td>108.3</td>
<td>181.8</td>
<td>95</td>
</tr>
<tr>
<td>117.2</td>
<td>195.9</td>
<td>100</td>
</tr>
<tr>
<td>126.6</td>
<td>210.8</td>
<td>105</td>
</tr>
<tr>
<td>136.4</td>
<td>226.4</td>
<td>110</td>
</tr>
<tr>
<td>146.8</td>
<td>242.7</td>
<td>115</td>
</tr>
<tr>
<td>157.7</td>
<td>259.9</td>
<td>120</td>
</tr>
</tbody>
</table>

Figure 12-10  Typical Heat Rejection Factor  
For Evaporative Condenser, R-22

<table>
<thead>
<tr>
<th>Cond. Press. (PSIG)</th>
<th>Cond. Temp. (°F)</th>
<th>Entering Air Wet Bulb Temperature (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>50</td>
<td>55</td>
</tr>
<tr>
<td>151.7</td>
<td>85</td>
<td>1.00</td>
</tr>
<tr>
<td>165.9</td>
<td>90</td>
<td>0.85</td>
</tr>
<tr>
<td>181.1</td>
<td>95</td>
<td>0.73</td>
</tr>
<tr>
<td>195.3</td>
<td>100</td>
<td>0.71</td>
</tr>
<tr>
<td>207.4</td>
<td>105</td>
<td>0.64</td>
</tr>
<tr>
<td>214.2</td>
<td>110</td>
<td>0.57</td>
</tr>
<tr>
<td>232.3</td>
<td>115</td>
<td>0.51</td>
</tr>
<tr>
<td>251.5</td>
<td>120</td>
<td>—</td>
</tr>
<tr>
<td>271.7</td>
<td>—</td>
<td>—</td>
</tr>
</tbody>
</table>

Figure 12-11  Typical Heat Rejection Factor  
For Evaporative Condenser, R-717
the dampers close to their minimum air flow position, an auxiliary
switch will turn off the fan motor.

(b) The discharge hood and damper – The dampers are to be fully open
before the fans are running and closed when the fan is off. The damper
actuator is controlled by a temperature controller.

2.0 Water Pump Variable Speed Control. And Water Pump On-Off

Reduction of air or water flow would greatly reduce the capacity. Therefore,
shutting off the water flow should be done only after other means of control
have been used.

The Head Control Function Sequence for evaporative condenser:

On decreasing ambient temperature in winter time:

A. Fan speed reduction.
B. Fans cycling.
C. Damper closing.
D. Pump cycling.

On condensing pressure increasing:

a. Fan speed increase.
b. Fans cycling.
c. Damper opens.
d. Pump cycle on.

Piping Hook-up Suggestion for Single Evaporative Condenser Application:

For single unit of evaporative condenser with single coil, the drain line to receiver can
be served as internal equalizer; no external equalizing may be required as shown in
Figure 12-12.

However, if liquid drain line is trapped or a surge receiver is used with the
evaporative condenser as shown in Figure 12-13, an equalizer line should be used and
a liquid trap height (h) should be provided. The liquid drain from the evaporative
condenser should be sized for a maximum velocity of 100 Ft/Min. A higher liquid
column should be provided if a maximum velocity of 150 Ft/Min and a valve is used
for the drain line:

<table>
<thead>
<tr>
<th>Maximum Velocity of Drain Line, Ft/Min</th>
<th>Valve Between Evaporative Condenser and Receiver</th>
<th>Liquid Column (h), Inches</th>
</tr>
</thead>
<tbody>
<tr>
<td>100 Ft/Min</td>
<td>None, Angle or Globe</td>
<td>14 Inches</td>
</tr>
<tr>
<td>150 Ft/Min</td>
<td>None</td>
<td>14 Inches</td>
</tr>
<tr>
<td>150 Ft/Min</td>
<td>Angle</td>
<td>16 Inches</td>
</tr>
<tr>
<td>150 Ft/Min</td>
<td>Globe</td>
<td>28 Inches</td>
</tr>
</tbody>
</table>
Figure 12-12  Single Coil Evaporative Condenser
Top Inlet to Receiver

Figure 12-13  Single Evaporative Condenser with
Surge Receiver
Liquid Trap Column for Multiple Parallel Units Operation:

Liquid trap column is important for multiple coils or multiple evaporative condenser application when one or more condenser is to be shut off during operation. Figure 12-14 shows one condenser is operating while the other condenser is idling. The pressure at the outlet of the idle condenser is the same as inlet pressure. The liquid shall back up into and flooding the operating condenser coil if no trap liquid column is provided in this case.

Assuming the refrigerant is Ammonia (R-717), CT is 95°F. The liquid column needed to off set the pressure unbalance under this operating condition is calculated as the following:

Condensing Temperature: 95°F
Condensing Pressure: 195.9 Psia
Liquid Density: 36.68 Lbs/Ft³
Valve P.D.: 0.30 Psi
Pressure after valve: 195.6 Psia
Coil Pressure Drop: 0.8 Psi
Liquid pressure leaving coil: 195.6 – 0.8 = 194.8 Psia
Total pressure differential: 195.9 – 194.8 = 1.1 Psi

\[
\text{Pressure Diff. Psi} = \frac{\text{Liquid Density (Lbs/Ft}^3\text{)} \times \text{Column (Ft)}}{144}
\]

\[
1.1 \text{ Psi} = \frac{36.68 \text{ (Lbs/Ft}^3\text{)} \times \text{Column (Ft)}}{144}
\]

\[
\text{Column (Ft)} = \frac{1.1 \times 144}{36.68} = 4.32 \text{ Ft}
\]

\[
= 52 \text{ Inches liquid column of R-717}
\]

From the above calculation, 52” liquid trap column is required to balance the pressure drop through the condenser coil and the valve for the R-717 at 95°F CT. The liquid column required for the various common refrigerants for each Psi PD are as the following:

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Liquid Trap Column</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-717 Ammonia</td>
<td>48” for each Psi P.D.</td>
</tr>
<tr>
<td>R-290 (Propane)</td>
<td>58” for each Psi P.D.</td>
</tr>
<tr>
<td>R-1270 (Propylene)</td>
<td>57” for each Psi P.D.</td>
</tr>
<tr>
<td>R-22</td>
<td>22” for each Psi P.D.</td>
</tr>
</tbody>
</table>
R-717 Refrigerant
CT = 95°F
195.9 Psia

Figure 12-14  Liquid Trap Column for Multiple Parallel Units
Evaporative Condenser Operation
It is recommended to check with the evaporative condenser manufacturer to obtain the coil pressure drop for the coil or their recommendation as what is height of the liquid column trap should be provided for the parallel application.

**Evaporative Condensers in Parallel Operation:**

For the multiple evaporative condensers system, it becomes necessary to shut off one or more evaporative condenser from a multiple system to maintain sufficient condensing temperature and system pressure difference. Therefore, the piping hook-up and the liquid trap column are important for proper operation of the refrigeration system.

Should the multiple evaporative condensers be used for a constant load or where the local wet bulb air temperature variation is not too much or all the condensers are expected to be operated all time even at partial load operation, liquid trap is not required under these circumstances.

For multiple evaporative condenser units operation, the requisites for the arrangements are as the following:

I) A receiver must be used. The receiver must be sized to have storage capacity to hold the operating charge for all the conditions.

II) An external equalizer line must be provided between the receiver and the gas inlet line to the evaporative condensers.

III) Enough vertical height liquid trap column shall be properly provided.

IV) The vertical liquid drain lines from the evaporative condensers to receiver are to be sized to have maximum velocity not more than 150 Ft/Min; the horizontal drain line is to be sized to have maximum velocity not more than 100 Ft/Min.

V) The equalizer line should be connected to the point where the pressure drop in the gas inlet line to each evaporative condenser is approximately the same.

VI) The shut off valves in the evaporative condenser liquid drain lines should be located at the lowest elevation point as possible to minimize flash gas.

VII) Normally, purge valve is to be located at the highest point of the liquid drain line.

The followings are the suggested piping hook-up for various evaporative condensers connected in parallel operation:

Figure 12-15 is the parallel condenser, each is having multiple coil circuits and with a storage liquid receiver.

Figure 12-16 is the piping hookup for parallel evaporative condensers with a surge receiver.
Figure 12-15
Parallel Units with Multiple Coils
Top Inlet Receiver

Figure 12-16  Parallel Operation with Surge Receiver
Figure 12-17
Parallel Units & Multiple Coils with Surge Receiver

Figure 12-18
Parallel Operation
Condensers and Receiver on Same Floor
Figure 12-19 Parallel Operation
Mixed Sizes Evaporative Condensers

Figure 12-20 Parallel Operation
When Trapping Height not Available
Figure 12-21  Parallel Operation of Evaporative Condenser and Shell-and-Tube Water Cooled Condenser

Figure 12-22  Parallel Operation of Evaporative Condenser and Shell-and-Tube Water Cooled Condenser with Surge Receiver
Figure 12-17 is the piping hookup for parallel operation and each unit is having multiple coils with a surge receiver.

Figure 12-18 is the parallel condensers mounted on the same floor level.

Figure 12-19 is the piping hookup for parallel operation but with mixed sizes of evaporative condensers.

Figure 12-20 is the piping hookup suggestion for parallel evaporative condensers on the same floor when the trap liquid column is not available. In this case, dimension “A” is the elevation difference between evaporative condenser outlet and the top of the receiver; dimension “A” must be at least 20 inches. Dimension “B” should be as small as possible. It is suggested that both evaporative condensers are to be remain operated at the same time.

Figure 12-21 shows an evaporative condenser and a shell-and-tube water cooled condenser are connected in parallel with top inlet storage receiver. H2 in this case is the height between the evaporative condenser and the operating liquid level in the receiver. This height is determined for the pressure drop along the circuit “a-b-c-d” including the pressure drop through the evaporative coil. H3 is the elevation between the bottom connection of the water cooled shell-and-tube condenser and the operating liquid level in the receiver; this height is determined by the pressure drop through the refrigerant circuit of “a-b’-c’-d’”.

Figure 12-22 shows the evaporative condenser and shell-and-tube water cooled condenser are connected in parallel with bottom inlet surge receiver. In this case, H2 is the height between the evaporative condenser and the operating liquid level in the receiver. This height is determined for the pressure drop along the circuit “a-b-c-d” including the pressure drop through the evaporative coil. H3 is the elevation between the bottom connection of the shell-and-tube condenser and the operating liquid level in the receiver; this height is determined by the pressure drop through the refrigerant circuit of “a-b’-c’-d’”. The horizontal liquid drain line from the water cooled condenser to the receiver is to be sized for 100 Ft/Min velocity maximum at the design load.

**Liquid Subcooling:**

Certain degree of liquid subcooling can be obtained by the evaporative condenser. However, the subcooling should be arranged in a separate circuit in the evaporative condenser as shown in the Figure 12-23. The liquid from receiver for the subcooling should be taken out at the downstream of the receiver to minimize the chance that vapor mixes with liquid at this point.

**Winter and Low Ambient Operation:**

For industrial refrigeration installations, the refrigeration system is required to be operated year round. Therefore, it is necessary to take provisions of maintaining a minimum required condensing pressure for the refrigeration system and also to prevent the water being freeze up in the evaporative condenser.
One of the better control methods for winter operation for evaporative condenser is to have automatic damper control of air flow for the evaporative condenser instead of dry coil operation. Dry coil operation needs change over and the method of control is not entirely satisfactory.

In order to prevent the water in the evaporative condenser freeze up during the system during shut down is to have a remote water recirculation system consists of sump tank and a pump to be located indoor as shown in Figure 12-24.

**Data required for evaporative condenser selection and inquiry:**

- Heat rejection, Btu/Hr.
- Condensing temperature.
- Refrigerant.
- Ambient wetbulb air temperature, °F.
- Fan type, vane axial or centrifugal.
- Induced draft or blow thru arrangement.
- Subcooling coil, if required.
- Special coating of additional corrosion protection.
- Head control requirement.
- Power supply, type of motor and electrical code requirement.
Figure 12-23  Evaporative Condenser with Separate Circuit for Liquid Subcooling

Figure 12-24  Evaporative Condenser with Remote Sump Tank and Pump
Shell-and-Tube Heat Exchangers

Various heat exchangers for refrigeration application under the category of shell-and-tube are as the following:

(I) Water Cooled Condenser. (Figure 12-33)

(II) Evaporators:
- Flooded Type: (Figure 12-39)
- Half Bundle type. (Figure 12-39)
- Full Bundle with Surge Drum. (Figure 12-40)
- Dry Expansion (DX). (Figure 12-42)
- Thermosyphone. (Figure 12-43)
- Spray Evaporator. (Figure 12-44)
- Overfeed Evaporator. (Figure 12-45)
- Kettle Type Evaporator. (Figure 12-46)

Most the shell-and-tube heat exchangers for refrigeration application are horizontal with fixed tube sheet design that includes both condenser and evaporators. The efficiency of a vertical heat exchanger is not as good as the horizontal design; floating tube sheets design heat exchanger is not warranted for refrigeration system. The heat exchangers with U-tubes and/or floating tube sheet design are mainly used for evaporator duty and mostly are provided by the hydrocarbon processing industries users, not by refrigeration system providers.

Small Difference and LMTD:

The heat transfer formula for condenser or evaporator is as the following:

\[ Q = U \times A_o \times \text{LMTD} \]

- \( Q \) = Heat transferred, Btu/Hr.
- \( U \) = Overall heat transfer coefficient, Btu/HR-\( \text{Fr}^2 \cdot \text{F} \).
- \( A_o \) = Total effective outside tube surface, \( \text{Fr}^2 \).

\[ \text{LMTD} = \frac{L - S}{\log_e \left\{ \frac{L}{S} \right\}} \]

The heat transmission and the thermal resistances through the tube are shown in Figure 12-25. Figure 12-26 shows the heat transmission resistances and the temperature gradients for the tube wall and each layer of film through the tube section from \( t_1 \) to \( t_7 \).

Small difference value is the difference between \( CT \) and the leaving cooling water or the
standard tube layout

triangular tube pitch

TUBE SECTION

OUTSIDE TUBE

INSIDE TUBE

Figure 12-25  Heat Transfer & Thermal Resistances
\[ U = \frac{1}{\left(\frac{1}{h_i} R + r_{fi} + r_m + r_s + r_{fo} + \frac{1}{h_o}\right)} \]

Figure 12-26  Heat Transmission
difference between the leaving fluid temperature and the ET as shown in Figure 12-27. Some of the condenser or cooler performance curves are expressed in small difference instead of LMTD.

**Shell Diameter and Tubes Count:**

Figure 12-28 shows the optimum total possible tubes insert for a given shell diameter, the general heat exchanger data and the total external surface for the heat exchanger for given size of the heat exchanger. Figure 12-29 is the table for various shell diameter and number of tube counts for various pass arrangement for the heat exchanger. All the data are based on 3/4” OD tubes on 15/16” Triangular Pitch for both condenser and evaporator. For example: For a shell diameter of 24”OD, 2-Pass arrangement, full bundle flooded evaporator, maximum tubes count is 388.

Figure 12-30 shows the physical data for various commonly used 3/4” OD finned tubes. Figure 12-31 is the commonly used tube material and the corresponding thermal conductivity, Btu/ft²°F·hr/ft at 60°F. Figure 12-32 is the physical data for various commonly used bare tubes.

**Code and Standard Compliance:**

The manufacturing and the construction of shell-and-tube heat exchanger is to conform with ASME code Section VIII for Unfired Pressure Vessels, see Chapter 13 for further details.

The water heads for the shell-and-tube heat exchanger under ASME code can be either water channel or regular water box or marine water box.

The definition of “marine” water box is that the water box for the heat exchanger shall be constructed in such way without disturbing the water piping connections while cleaning tubes for the heat exchanger.
The MTD for differences off the chart may be found by the formula:

$$MTD = \frac{L - S}{\log_e \left(\frac{L}{S}\right)}$$

Where $L =$ Large Difference
$S =$ Small Difference

Figure 12-27 LMTD Chart
### Table: Condenser Dimensions and Weights

<table>
<thead>
<tr>
<th>Condenser Size</th>
<th>Nominal Shell OD</th>
<th>No. of Tubes</th>
<th>Area, Sq. Ft.</th>
<th>K-22, Lbs.</th>
<th>Weight, Lbs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>13-X-96</td>
<td>14&quot;</td>
<td>120</td>
<td>475</td>
<td>576</td>
<td>302</td>
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<tr>
<td>13-X-120</td>
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<td>120</td>
<td>597</td>
<td>726</td>
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<td>13-X-184</td>
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<td>719</td>
<td>872</td>
<td>458</td>
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<td>13-X-168</td>
<td>14&quot;</td>
<td>120</td>
<td>841</td>
<td>1020</td>
<td>535</td>
</tr>
<tr>
<td>13-X-192</td>
<td>14&quot;</td>
<td>120</td>
<td>963</td>
<td>1168</td>
<td>613</td>
</tr>
<tr>
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<td>1085</td>
<td>1316</td>
<td>690</td>
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<td>1207</td>
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<td>768</td>
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<tr>
<td>15-X-120</td>
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<td>164</td>
<td>813</td>
<td>986</td>
<td>480</td>
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<td>979</td>
<td>1188</td>
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<td>15-X-168</td>
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<td>1146</td>
<td>1390</td>
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<td>16&quot;</td>
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<td>1313</td>
<td>1592</td>
<td>776</td>
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<td>15-X-216</td>
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<td>164</td>
<td>1479</td>
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<td>1996</td>
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<td>20&quot;</td>
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<td>1593</td>
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<td>2595</td>
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<td>4399</td>
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<td>4038</td>
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<td>460</td>
<td>3196</td>
<td>3875</td>
<td>1826</td>
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<tr>
<td>25-X-192</td>
<td>26&quot;</td>
<td>460</td>
<td>3663</td>
<td>4442</td>
<td>2093</td>
</tr>
<tr>
<td>25-X-216</td>
<td>26&quot;</td>
<td>460</td>
<td>4131</td>
<td>5009</td>
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<td>5576</td>
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<td>600</td>
<td>4234</td>
<td>5013</td>
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</tr>
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<td>4743</td>
<td>5752</td>
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<td>6592</td>
<td>7968</td>
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<td>29-X-144</td>
<td>30&quot;</td>
<td>750</td>
<td>5167</td>
<td>6266</td>
<td>2162</td>
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<tr>
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<td>5792</td>
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<td>8114</td>
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<td>7433</td>
<td>9038</td>
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<td>5856</td>
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<td>850</td>
<td>6730</td>
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<td>850</td>
<td>8447</td>
<td>10242</td>
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</tr>
</tbody>
</table>

**Notes:** All based on 3/4" OD Tubes.

**Weight Based on steel tubes, 0.049 wall at root of fins.**

---

**Figure 12-28 Shell Size & Tube Count**

---

**NTL – Normal Tube Length, Inches**

**Finned Tube**

**Nominal Shell Diameter**

---

168
Shell Diameter, Pass and Maximum Tube Counts of 3/4”OD Tubes

<table>
<thead>
<tr>
<th>Shell Diameter, Inches OD / ID</th>
<th>Number of Passes of the Shell</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1-Pass</td>
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<tr>
<td>8.625” / 7.981”</td>
<td>49</td>
</tr>
<tr>
<td>10.75” / 10.02”</td>
<td>77</td>
</tr>
<tr>
<td>12.75” / 12.00”</td>
<td>111</td>
</tr>
<tr>
<td>14.00” / 13.25”</td>
<td>129</td>
</tr>
<tr>
<td>16.00” / 15.25”</td>
<td>177</td>
</tr>
<tr>
<td>18.00” / 17.25”</td>
<td>227</td>
</tr>
<tr>
<td>20.00” / 19.25”</td>
<td>273</td>
</tr>
<tr>
<td>22.00” / 21.25”</td>
<td>339</td>
</tr>
<tr>
<td>24.00” / 23.25”</td>
<td>411</td>
</tr>
<tr>
<td>26” OD</td>
<td>475</td>
</tr>
<tr>
<td>28” OD</td>
<td>559</td>
</tr>
<tr>
<td>30” OD</td>
<td>649</td>
</tr>
<tr>
<td>32” OD</td>
<td>749</td>
</tr>
<tr>
<td>34” OD</td>
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</tr>
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<td>66” OD</td>
<td>3657</td>
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<tr>
<td>72” OD</td>
<td>4301</td>
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Figure 12-29 Shell Diameter, Pass and Maximum Tube Count
<table>
<thead>
<tr>
<th>BWG</th>
<th>Nominal Wall at Finned Section</th>
<th>Fins per Inch</th>
<th>External Surface Sq.Ft. per Lin. Ft.</th>
<th>Internal Diameter, Inch</th>
<th>Surface Area Ratio Outside to Inside</th>
<th>Flow Area Square In.</th>
<th>Weight Per Ft. Length (Copper) Lbs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>0.035”</td>
<td>19</td>
<td>0.499</td>
<td>0.555</td>
<td>3.43</td>
<td>0.242</td>
<td>0.401</td>
</tr>
<tr>
<td>19</td>
<td>0.042”</td>
<td>19</td>
<td>0.499</td>
<td>0.541</td>
<td>3.53</td>
<td>0.229</td>
<td>0.448</td>
</tr>
<tr>
<td>18</td>
<td>0.049”</td>
<td>19</td>
<td>0.499</td>
<td>0.527</td>
<td>3.63</td>
<td>0.218</td>
<td>0.493</td>
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<tr>
<td>16</td>
<td>0.065”</td>
<td>19</td>
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<td>0.495</td>
<td>3.86</td>
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<tr>
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<td>0.085”</td>
<td>19</td>
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<td>0.459</td>
<td>4.16</td>
<td>0.162</td>
<td>0.697</td>
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<td>0.438</td>
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<td>3.39</td>
<td>0.192</td>
<td>0.665</td>
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<td>16</td>
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<td>0.459</td>
<td>3.65</td>
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<td>0.035”</td>
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<td>0.630</td>
<td>0.555</td>
<td>4.34</td>
<td>0.242</td>
<td>0.405</td>
</tr>
<tr>
<td>19</td>
<td>0.042”</td>
<td>26</td>
<td>0.630</td>
<td>0.541</td>
<td>4.46</td>
<td>0.229</td>
<td>0.452</td>
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<tr>
<td>18</td>
<td>0.049”</td>
<td>26</td>
<td>0.630</td>
<td>0.527</td>
<td>4.58</td>
<td>0.218</td>
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<tr>
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<td>0.065”</td>
<td>26</td>
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<td>0.495</td>
<td>4.87</td>
<td>0.192</td>
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<tr>
<td>14</td>
<td>0.083</td>
<td>26</td>
<td>0.630</td>
<td>0.459</td>
<td>5.25</td>
<td>0.162</td>
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Figure 12-30 Physical Data for Commonly Used 3/4”OD Tubes

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<thead>
<tr>
<th>Tube Material</th>
<th>Conductivity</th>
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<tbody>
<tr>
<td>Copper</td>
<td>196</td>
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<tr>
<td>90/10 Copper Nickel</td>
<td>27</td>
</tr>
<tr>
<td>70/30 Copper Nickel</td>
<td>17</td>
</tr>
<tr>
<td>85% Red Brass</td>
<td>92</td>
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<tr>
<td>Admiralty, Type B</td>
<td>64</td>
</tr>
<tr>
<td>Aluminum Brass, Type B</td>
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</tr>
<tr>
<td>Arsenical Copper</td>
<td>196</td>
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<tr>
<td>3003 Aluminum</td>
<td>92</td>
</tr>
<tr>
<td>*Aluminum Bronze, 5%</td>
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<tr>
<td>Admiralty, Types C and D</td>
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</tr>
<tr>
<td>Low Carbon Steel (Seamless or Welded)</td>
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</tr>
<tr>
<td>Steel, ASTM A334</td>
<td>26</td>
</tr>
<tr>
<td>*Monel</td>
<td>15</td>
</tr>
<tr>
<td>*SS Steels, 304, 304L, 316, 316L or 321</td>
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</tbody>
</table>

Note: *Available 16 FPI only

Figure 12-31 Commonly Used Tube Materials and Conductivity
Physical Data for Commonly Used Bare Tubes

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<tr>
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<th></th>
<th></th>
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<td></td>
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<tr>
<td>3/4”</td>
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<td>20*</td>
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<td>0.1964</td>
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<td>0.1964</td>
<td>0.584</td>
<td>0.1530</td>
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<td>0.546</td>
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<td>1.18</td>
<td>0.882</td>
<td>1.33</td>
<td>14.75</td>
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Note No.1:
*0.035” wall bare tube is too thin to be recommended for rolled tube joints.

Note No.2:
**1-1/4”OD, 13 BWG, Black steel tubes, electric resistance welded; mostly used for ammonia horizontal coolers and condensers. Weight per feet length is 1.172 lbs.

Figure 12-32 Physical Data for Commonly Used Bare Tubes
Water Cooled Condenser

Figure 12-33 shows the typical horizontal design shell-and-tube water cooled condenser.

Cooling tower water is used for most shell-and-tube condenser installations. Besides of cooling tower, the cooling medium for condenser can be river water, well water or sea water. Water softening is commonly used these days for cooling tower to minimize the fouling build-up in the tubes.

The most commonly used tube material for the condenser is copper. The condenser tube material should be changed to other corrosion resistance material if sea water is used. Some cases, the river or well water is also salty and corrosive, special tube material such as 90/10 Cupro Nickel or 70/30 Curpo Nickel should be used instead of copper for these application. Titanium tubes and tube sheet design condenser is the best for corrosion resistance, however it is the most expensive.

Cooling tower water is usually designed to have leaving water temperature 7 to 10°F above the design outdoor ambient wet bulb air temperature. The water range for condenser is usually about 10 to 15°F.

Water Cooled Condenser Performance:

\[ \delta T = CT - T_w \]

\( \delta T \) = Small Difference, °F

\( CT \) = Condensing Temperature, °F

\( T_w \) = Leaving Condenser Water Temperature, °F

Therefore:

\[ CT = \delta T + T_w \]

Figure 12-34 is a typical water cooled condenser performance curve. Figure 12-35 is a diagram to show the impact of higher fouling factor for condenser rather than the normal allowance of 0.0005 Ft²·Hr·°F/Btu. Figure 12-36 is the typical curve to show the impact of using other material for the condenser instead of copper. Those correction factors are in approximate LMTD to be added to the overall LMTD of the heat transfer for the condenser.

Cooling fluid flow calculation:

\[ \text{GPM} = \frac{\text{Btu/Hr.}}{499.8 \times \text{S.G.} \times \text{Cp} \times (T_2 - T_1)} \]

Btu/Hr = Heat rejection

GPM = Cooling fluid flow, Gal/Min
Figure 12-33  Typical Shell-and-Tube Water Cooled Condenser
Condoner Performance Without Subcooler:
R-500 0.0003 Fouling Resistance
Based on 90 Deg F Mean Water Temperature
Correction Factor to be Added to L-T-D-
Value from Curve .02x(90-Mean Water
Temperature in Condenser)
26 Fins Per Inch, 15 FT. Copper Tubes

Figure 12-34 Typical Shell-and-Tube Water Cooled
Condenser Performance
Figure 12-35  Typical Fouling Penalties
For Water Cooled Condenser
Figure 12-36  Typical Special Material Penalties
For Water Cooled Condenser
S.G. = Specific Gravity of fluid at average temperature
C_p = Specific Heat of fluid at average temperature
T_2 - T_1 = Range, fluid temperature difference, °F

If the cooling fluid is water, the formula becomes:

\[
\text{GPM} = \frac{\text{Btu/Hr.}}{499.8 \times (T_2 - T_1)}
\]

Data required for water cooled condenser selection and pricing:

- Heat Load, Btu/Hr.
- Cooling water in temperature, °F.
- Condensing temperature, °F.
- Refrigerant.
- Shell side DWP and Tube side DWP.
- Overall length or NTL limitation, if any.
- Water pressure drop limitation, if any.
- Special tube material, Gauge, FPI, Tube OD requirements, if any.
- Fouling factor requirement.
- Pass arrangement.
- Cooling water GPM limitation, if any.
Evaporator

All the heat exchangers are to be selected by the heat exchanger manufacturer because the type of heat exchanger, NTL length, shell diameter, tube type, tube size, tube material and heat transfer coefficients vary widely depending on the brine or product and temperatures handled.

Evaporator Performance:

\[ \delta T = T_w - ET \]

\[ \delta T = \text{Small Difference, } ^\circ\text{F} \]
\[ ET = \text{Evaporative Temperature, } ^\circ\text{F} \]
\[ T_w = \text{Leaving Chilled Water Temperature, } ^\circ\text{F} \]

Therefore: \( ET = T_w - \delta T \)

Figure 12-37 shows the impact on various fouling factor for evaporator. The Figure 12-38 is the impact on evaporator if other tube material is used instead of copper tubes. These penalties for higher fouling factor or for special tube material are to be included in the value of \( \delta T \).

Calculation for brine cooling is more complicated than for water because of the specific gravity, viscosity and thermal conductivity and etc. vary widely. Viscosity of the brine impacts greatly on the size of the heat exchanger. More tube heat transfer surface \( \text{Ft}^2/\text{TR} \) is needed for the application if the brine is having higher viscosity. As a rough guide and a rule of thumb, if a flooded evaporator is with 20 BWG 19 FPI 3/4” tubes, the relationships between \( \text{Ft}^2/\text{TR} \) and the viscosity of the brine are as the following:

<table>
<thead>
<tr>
<th>( \text{Ft}^2/\text{TR} )</th>
<th>Brine Viscosity</th>
</tr>
</thead>
<tbody>
<tr>
<td>8 Square Ft. per TR</td>
<td>2 to 3 CP</td>
</tr>
<tr>
<td>10 Square Ft. per TR</td>
<td>3 to 5 CP</td>
</tr>
<tr>
<td>14 Square Ft. per TR</td>
<td>5 to 8 CP</td>
</tr>
<tr>
<td>18 Square Ft. per TR</td>
<td>9 to 12 CP</td>
</tr>
</tbody>
</table>

A different type of evaporator should be considered instead of flooded evaporator if the brine viscosity is above 12 CP.

Fluid or Brine flow for evaporator:

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

OR

\[ \text{Btu/Hr.} = \text{GPM} \times 499.8 \times \text{S.G.} \times \text{Cp} \times (T_2 - T_1) \]
Figure 12-37  Penalties for Fouling Factor for Evaporator
TR = Tons of refrigeration
GPM = Fluid flow, Gal/Min
S.G. = Specific Gravity of fluid at average temperature
Cp = Specific Heat of fluid at average temperature
T₂ - T₁ = Range, Temperature Difference, °F

Submerge Penalty:

For low temperature application, the density of the refrigerant liquid inside the evaporator increases rapidly as the evaporative temperature getting lower, the static head is increasingly greater accordingly. The evaporative temperature is to be lowered to offset this submerge penalty. The larger the shell diameter and lower the ET, more submerge penalty should be allowed. The degree of submergence effect also greatly depends on what refrigerant is used.

Data Required for Evaporator Selection and Pricing:

Refrigeration Load, TR or Btu/Hr.
In and out temperatures, °F.
Refrigerant.
Shell side DWP and Tube side DWP.
Evaporative Temperature, °F.
Pass arrangement.
Fouling factor.
Pressure drop allowed, if any.
Special tube material, if any.
Special requirements for tube OD, FPI, NTL, if any.

Additional information required if the cooling fluid is common brine instead of water:
   Name of the brine, WT% brine concentration, if specified.

Additional information required if the medium is special fluid or special brine:
   Name of the brine for fluid.
   Specific heat at the average temperature.
   Specific gravity at the average temperature.
   Thermal Conductivity at the average temperature.
   Viscosity at four temperature points:
      At the brine average temperature.
      At the leaving brine temperature.
      At 10°F below brine leaving temperature.
      At 15°F below brine leaving temperature.

Additional information required if the application is for gas condensation:
   Name of the gas.
   Mole fraction of each components of the gas mixture.

Half Bundle Flooded Evaporator:

Figure 12-39 is the half bundle or partial bundle evaporator design. This heat exchanger is one cylindrical shell. Lower half of the bundle is for tubes insert and the integral upper
Figure 12-39  Shell-and-Tube Flooded Evaporator Half Bundle Design
half space is for gas/liquid separation. The advantage of using this half bundle design is for installation where ceiling height is limited. The disadvantages are bigger shell diameter is required; might require higher submerging penalty if the ET is low and operating refrigerant charge is higher.

**Full Bundle Flooded Evaporator:**

Figure 12-40 is the full bundle shell-and-tube flooded evaporator. This heat exchanger consists of two shells. The lower shell is the evaporator which contains full bundle of tubes insert. The upper shell is the surge drum or the accumulator which provides the space for gas and liquid separation. These two shells are connected by the risers as shown.

**Dry Expansion Evaporator:**

Figure 12-41 is the DX evaporator. It is a low cost heat exchanger and it simplifies the refrigeration system design. DX system is basically used for small capacity installation.

**Thermosyphon Evaporator:**

Figure 12-42 is a typical thermosyphon evaporator. The thermosyphon evaporator is a shell-and-tube heat exchanger coupled with a surge drum or accumulator. The brine or fluid flows through the shell side and the refrigerant is through the tube side. This heat exchanger is designed to cool high viscous fluid or brine. The refrigerant liquid is supplied to the thermosyphon evaporator from the surge drum on top of the evaporator by gravity force. Portion of the liquid is vaporized to cool the fluid in the heat exchanger; the bubbling mixture of liquid/vapor is circulated back to the surge drum. The refrigerant liquid from high side receiver is supplied to the surge drum through an expansion device such as liquid level control valve. Gas is returned to compressor suction from the surge drum. The surge drum must be located at a certain height above the evaporator in order to generate the thermosyphone effect.

**Spray Evaporator:**

Figure 12-43 shows the spray type shell-and-tube evaporator. This type of evaporator is used to minimize the refrigerant charge and also is used for very low temperature refrigeration to eliminate the submerge penalty in the evaporator. Liquid is pump recirculated. The liquid refrigerant charge is limited. The shell should be sized to provide gas/liquid separation or moisture eliminator should be provided to prevent liquid carry over back to compressor suction.

**Overfeed Evaporator:**

Figure 12-44 is the liquid overfeed type evaporator. The refrigerant liquid is forced feed through the tube. This heat exchanger is designed to cool high viscous product or brine because the product or brine is through the shell, not through the tubes. The application of this evaporator is the same as the thermosyphon evaporator except that the liquid is forced fed by a pump instead of by gravity fed, therefore, no height limitation is needed between the separation compartment and the evaporator.
Figure 12-40  Shell-and-Tube Flooded Evaporator Full Bundle Design
Figure 12-41  Shell-and-Tube Dry Expansion Evaporator

Figure 12-42  Shell-and-Tube Thermosyphon Evaporator
Figure 12-43  Shell-and-Tube
Spray Evaporator

Figure 12-44  Shell-and-Tube
Overfeed Evaporator

Figure 12-45  Shell-and-Tube
Kettle Type Evaporator
Kettle Type Evaporator:

Figure 12-45 is the Kettle type evaporator. This type of evaporator is usually used by oil refinery, petrochemical and hydrocarbon processing industries. It only used for very special application and it usually provided by the user.
Chapter – 13  Major Mechanical Codes for Refrigeration Systems

Pressure Vessel Code ASME:

All the pressure vessels including the heat exchangers are to be designed in accordance with the ASME Pressure Vessel Code Section VIII. The ASME code applies for all the pressure vessels over 6” in internal diameter and the design working pressure over 15 Psig. However, the code does not apply to a shell or portion thereof contains water or certain common brine and the volume is less than 120 gallons.

The tube side of the flooded cooler is for the cooling medium or brine which the pressure is the hydrostatic pressure plus pumping head. In this case, no impact tested welds or nickel steel is required providing that the parts subject to this pressure are designed to 2.5 times the maximum pressure to be encountered in the service. For example: If the tube side of a vessel working pressure is 60 Psig, the channel heads and water box is to be designed for 60 x 2.5 = 150 Psig. It is suggested to check with the manufacturer for each case for these requirements.

All the pressure vessels for refrigeration applications are to be designed, constructed, inspected and stamped in accordance with the ASME code where it applies.

Alternation or modification for pressure vessels that have been inspected and stamped by ASME is not allowed and is in violation of the code unless proper arrangement is made from code authorities is obtained.

ASME Code Requirement for Low Temperature Application:

Special steel and impact test requirement for low temperature is because the metal becomes brittle at the low temperature. The normal guide line of ASME for low temperature application is -20°F. Under ASME code, carbon steel can be used for operating temperature above -20°F, A-516 Carbon Steel normalized may be used for temperature between -20°F to -50°F; 3-1/2% Nickel steel is for temperature between -50°F to -150°F and 304 Stainless steel may be used for temperature between -150°F to -325°F. Also impact testing of welds and materials are required for temperature below -20°F.

However, the character of refrigerant is that temperature is a function of the pressure; the pressure is low when the temperature is low and the operating pressure is higher only when the operating temperature is higher. Therefore, under ASME section UCS-66(c)(2), the ASME code permits within practical pressure limits, carbon steel is allowed for low temperature application. That means no impact test and no nickel steel or other special steel is required for operating temperature below -20°F if the operating pressure below -20°F never exceeds 0.4 times of the DWP at -20°F.
An example is for a refrigeration system using R-717 (Ammonia) as the refrigerant. The low side DWP is 225 Psig, the pressure limit allowing using carbon steel for low side evaporator under UCS-66(c)(2) is 225 x 0.4 = 90 Psig. The ET for the evaporator is -40°F which the operating pressure is 8.7” Hg. 8.7” Hg is below 0 Psig and is below than 90 Psig allowed under ASME UCS-66(c)(2). Therefore, impact tested welds or special shell material will not be required for code conformance. However, carbon steel should not be used below -150°F; nor 3-1/2% nickel steel below -200°F.

The case of R-1270 (Propylene) refrigerant: The low side DWP is 225 Psig, the pressure limit allowing using carbon steel for low side evaporator under UCS-66(c)(2) is 225 x 0.4 = 90 Psig. The ET for the evaporator is -40°F which the operating pressure is 5.91 Psig. The operating pressure of 5.91 Psig is below than 90 Psig allowed under ASME UCS-66(c)(2); therefore, impact tested welds and special shell material are not required for code conformance to ASME.

No impact tested welds or nickel steel is required for tubes side of brine cooler provided that parts subjected to the pressure are designed to 2.5 times the maximum pressure of the service encountered.

Some users might insist to use low temperature material such as A516 or Nickel Steel for the fabrication of the heat exchanger, the cost extra for the using of these material is significant.

**TEMA Code for Shell-and-Tube Heat Exchanger:**

TEMA (Tubular Exchanger Manufacturers Association) Standard might be sometimes specified for the heat exchanger by the users of hydrocarbon processing or chemical industries.

The TEMA specification is not a replacement of ASME, but it is an additional to ASME. If TEMA is required, the inspection of TEMA is actually to be carried out by ASME inspector and the certification is also to be done by ASME.

Figure 13-1 shows the TEMA heat exchanger nomenclature for the constructions of shell-and-tube heat exchangers. The first column of the chart is the front end stationary head types; the second column is the shell types and the third column is for the rear end head types. Selecting one description from each column, it shall be the basic construction requirements of the heat exchanger design. For example, the heat exchanger of AEL designates: “A”- represents the front end head with removal cover; “E”- represents one pass shell and “L”-Represents the rear end channel fixed tube sheet stationary head with removal cover.

The TEMA specification covers three classifications as the following:

- **TEMA-C**: For the general moderate requirements of commercial general process application.

- **TEMA-B**: For the service requirement of chemical process application.
Figure 13-1   TEMA Heat Exchanger Nomenclature
TEMA-R: For the general severe service requirements of petroleum and related processing application.

If the heat exchanger is to be designed and constructed in accordance with the TEMA specification, the class of TEMA design such as either “C” or “B” or “R” must be indicated.

The major differences of the heat exchanger design between TEMA and ASME are the front and rear end head type construction, hydrostatic test, corrosion allowance minimum tube pitch, minimum tube supports, minimum shell thickness and etc. The heat exchanger of TEMA design is larger than ASME for the same heat load requirement.

The major differences between TEMA-C, TEMA-B and TEMA-R are that the shell thickness of TEMA-B is less severe than TEMA-R; TEMA-C is less severe than TEMA-B and etc.

Direct expansion shell-and-tube heat exchanger is not under the classification of TEMA. Therefore, exception must be taken to the head design.

**Mechanical Refrigeration Safety Code:**

Safety Code for Mechanical Refrigeration ANSI/ASHRAE 15 is the primary safety reference for mechanical refrigeration systems. It covers the safety requirements for the design, construction, installation, testing and inspection.

ANSI/IIAR-2 governs the standards for the equipment, design and installation of ammonia mechanical refrigeration systems.

**Refrigeration Piping Codes:**

Pressure piping code of ASME/ANSI Standard B31.5 covers the design requirements of the refrigerant piping, valves and fittings for the refrigeration system.

B31.5 effects shell connections in some area where no cast iron, wrought iron or carbon steel be used for pipe fittings below -100°F; also a 2% increase over the actual design working pressure should be added for each degree below 0°F.

B31.3 is the Pressure Piping, Petroleum Refinery Piping Code. This code is compatible with B31.5 the design and material requirements except for operating temperatures below -20°F, where the B31.1 becomes involved with requirements for impact tested materials.
Chapter – 14 Refrigerants and Refrigerant Selection

Refrigerants for Industrial Refrigeration

What constitutes a good refrigerant? The desirable features of a refrigerant are as the following:

-- Non-Corrosive.
-- Acceptable vapor pressure characteristics.
-- Favorable flow rates of Lbs/Min and CFM.
-- Favorable Refrigerant Effect.
-- Non-Toxicity.
-- Non-Flammability.
-- Good Film Conductivity of Heat Transfer.
-- Good Thermal Stability.
-- No O-Zone Problem.
-- No GWP (Global Warming Potential) Problem.
-- Low Cost for the Refrigerant & Relatively Low Cost for the Equipment Used.

There are about 115 refrigerant listed in ASHRAE Standard 34, none of the refrigerants fulfills all the desirable criteria of a good refrigerant. Therefore, the refrigerant selection is a matter of balancing between advantages and disadvantages for the particular installation.

The ASHRAE Standard 34 also defines the classifications of safety for the refrigerants:

<table>
<thead>
<tr>
<th>Refrigerant Safety Classification</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Toxicity</strong></td>
</tr>
<tr>
<td>Group A</td>
</tr>
<tr>
<td>Group B</td>
</tr>
<tr>
<td><strong>Flammability</strong></td>
</tr>
<tr>
<td>Class 1</td>
</tr>
<tr>
<td>Class 2</td>
</tr>
<tr>
<td>Class 3</td>
</tr>
</tbody>
</table>

If the Refrigerant is classified under group A1; that means the refrigerant is a lower toxicity with no flame propagation.

R-718 (Water) is nearly a good refrigerant; unfortunately, it cannot be used for any low temperature application, it is mainly used for absorption system for water chilling duty. Furthermore, the absorption system is more expensive than rotating equipment system using other type of refrigerant for the same temperature level application. The
disadvantage of water use as refrigerant out weights the advantage, therefore, the R-718 (water) is not considered as a good refrigerant.

R-11, R-12, R-114 and R-13 are classified under safety group of A1, but are being phased out because of O-Zone and GWP problems.

R-123 was supposedly the replacement for R-11. Unfortunately, it is under B1 safety classification because the toxicity is high; the Allowable Exposure Limit (AEL) and Threshold Limit Value (TLV) is 10 PPM. R-123 refrigerant is basically used for centrifugal system because the volume flow is high and it is mostly used for water chilling duty.

At the present time, the common refrigerants used for industrial refrigeration are as the following:

Table 14.2 Common Refrigerants

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Boiling Point @ Atm Pres.</th>
<th>Refrigeration System</th>
<th>Safety Classification</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-134a, CH₂FCF₃</td>
<td>-15.0°F</td>
<td>High stage or Compound</td>
<td>A1</td>
</tr>
<tr>
<td>R-717 (Ammonia), NH₃</td>
<td>-28.0°F</td>
<td>High stage or Compound</td>
<td>B2</td>
</tr>
<tr>
<td>R-22 ,CHClF₂</td>
<td>-41.4°F</td>
<td>High stage or Compound</td>
<td>A1</td>
</tr>
<tr>
<td>R-290 (Propane), C₃H₈</td>
<td>-43.7°F</td>
<td>High stage or Compound</td>
<td>A3</td>
</tr>
<tr>
<td>R-1270 (Propylene) , C₃H₆</td>
<td>-53.9°F</td>
<td>High stage or Compound</td>
<td>A3</td>
</tr>
<tr>
<td>R-170 (Ethane), C₂H₆</td>
<td>-127.9°F</td>
<td>Cascade low stage</td>
<td>A3</td>
</tr>
<tr>
<td>R-23, CHF₃</td>
<td>-115.7°F</td>
<td>Cascade low stage</td>
<td>A1</td>
</tr>
<tr>
<td>R-1150 (Ethylene), C₂H₄</td>
<td>-154.7°F</td>
<td>Cascade low stage</td>
<td>A3</td>
</tr>
<tr>
<td>R-50 (Methane), CH₄</td>
<td>-258.7°F</td>
<td>Cascade low stage</td>
<td>A3</td>
</tr>
</tbody>
</table>

The R-134a is the replacement for R-12. Therefore, the boiling point, the working pressure and the flow CFM for R-134a are very similar to R-12.

The safety classification of both R-134a and R-22 are A1. The GWP value for R-22 is 0.35 as compared to 0.26 for R-134a. The DWP for air cooled application might be lower if R-134a refrigerant is used instead of R-22; however, the capacity of a compressor rated on R-134a is about 30% to 40% less than R-22 for the same operating conditions.

R-717, ammonia is a very popular refrigerant because this refrigerant provides all the desirable features except the toxicity and the flammability ratings. The safety classification for ammonia is B2. The explosion range for ammonia is 16% to 25% mixed with air; the ignition source temperature is 1,203 F or higher.
The flammability class is “2” for ammonia, the electrical equipment using R-717 as refrigerant should be designed and constructed in accordance with hazardous location application if it is strictly in confirmation with the code requirements. However, many users in food processing industries waive this requirement; might be because of the narrow explosion range of ammonia and air mix and high ignition temperature.

Both R-290 (Propane) and R-1270 (Propylene) refrigerants are widely used for refrigeration systems, particularly for oil, petrochemical, hydrocarbon processing and chemical industries, it is because the electrical equipment used for those industries is required to confirm with hazardous regulations regardless what refrigerant is used.

Both R-290 and R-1270 do not have EPA, O-Zone or GWP problems; furthermore, it is a low cost refrigerant and it is convenient for those industries to handle this type of refrigerants. The character comparison between R-290 and R-1270 is very similar to the comparison between R-134a and R-22; a compressor rated on R-290 refrigerant will produce smaller capacity as compared to R-1270 for the same duty.

The above capacity comparisons are based on screw compressors. The centrifugal compressor is a high volume flow machine; therefore, it might be justifiable to select R-134a over R-22 or R-290 over R-1270 if a centrifugal compressor is used. However, it might be not be feasible to use R-717 refrigerant for centrifugal compressor, unless the refrigeration capacity is very large.

The costs for R-1270, R-290 and R-717 are the cheapest amount all the commonly used refrigerants.

R-170 (Ethane), R-23, R-1150 (Ethylene) and R-50 (Methane) are the refrigerant commonly used for the low stage of a cascade refrigeration system. R-23 is one of the group A1 refrigerants and it is used for non-hazardous installation. R-50 (Methane), R-170 (Ethane) and R-1150 (Ethylene) are group A3 refrigerants; therefore, all the electrical items including motors for the refrigeration system are to be designed and constructed in accordance with regulations of the NEC code for hazardous application.
Dual Refrigerants System of R22 and Ammonia (R717):

The comparisons between R-22 and R-717 are generally outlined as the following:

1.0 Heat transfer and thermodynamic properties of R-717 are better than R-22. Piping and valve sizes of R-717 system are smaller than R-22.

2.0 R-717 is one of the refrigerants under safety group B2; R-22 is A1 safety refrigerant. Electrical equipment for R-717 system is more expensive than R-22 if code requirement is not waived by the user.

3.0 R-22 has higher TR than R-717 based on same screw compressor size and same operating conditions.

4.0 R-717 has slightly advantage on BHP/TR at lower compression ratio as compared to R-22 for screw compressor; R-22 has better BHP/TR over R-717 at higher compression ratio.

5.0 The environmental effects of the R-22 refrigerant are:
   - Atmospheric Lifetime: 18 years
   - Ozone Depletion Potential (ODP): 5%
   - Global Warming Potential (GWP): 40%.
   - Availability of R-22 in the future: Restricted

6.0 On the other hand, the environmental effects of R-717 are very favorable:
   - Atmospheric Lifetime: <2 weeks
   - Ozone Depletion Potential (ODP): 0%
   - Global Warming Potential (GWP): 0%.
   - Availability of R-717 in the future: No Restriction

The economical impact between R-22 and R-717 is hinged on the code requirement for the electrical items. If the electrical system for ammonia is to be designed and constructed in accordance with the NEC code requirement, then, the R-717 system might be more expensive than the R-22 system. However, if the user waives the NEC code compliance such as in food processing industries and is willingly to accept the toxicity classification of ammonia, then the ammonia might be better over R-22. However, the conclusion is not absolute; it is recommended that the analysis should be based on initial investment and annual energy consumption of the system.

Some of the application, the user prefers to use A1 group refrigerant such as R-22 for the installation. However, by knowing the availability of R-22 may be in jeopardy in the future, a provision is made that the refrigeration system is to be designed for the dual use of R-22 and R-717. That means the system is designed for the use of R-22 for now; in case the availability of the R-22 refrigerant is restricted in the future, the refrigeration system is to be converted to R-717 at that time.

The refrigerant flows and the general data for system using R-22 and R-717 are listed in Table 14.3. This comparisons are based on same size of screw compressor and under the same evaporative temperature of 10°F and same condensing temperature of 104°F.
Table 14.3 Comparisons R-717 and R-22

<table>
<thead>
<tr>
<th>Description</th>
<th>Ammonia (R-717)</th>
<th>R-22</th>
</tr>
</thead>
<tbody>
<tr>
<td>Same size compressor, ET/CT</td>
<td>ET=10 F</td>
<td>ET=10 F</td>
</tr>
<tr>
<td></td>
<td>CT=104 F</td>
<td>CT=104 F</td>
</tr>
<tr>
<td>Net Refrigeration Effect (NRE)</td>
<td>455.46 Btu/Lb</td>
<td>64.91 Btu/Lb</td>
</tr>
<tr>
<td>Capacity</td>
<td>209.3 TR</td>
<td>205.2 TR</td>
</tr>
<tr>
<td>Power consumption</td>
<td>363.5 BHP</td>
<td>364.2 BHP</td>
</tr>
<tr>
<td>BHP/TR</td>
<td>1.7367</td>
<td>1.7744</td>
</tr>
<tr>
<td>Evaporative pressure</td>
<td>38.5 Psia</td>
<td>47.5 Psia</td>
</tr>
<tr>
<td>Condensing pressure</td>
<td>225.6 Psia</td>
<td>222.4 Psia</td>
</tr>
<tr>
<td>Discharge temperature</td>
<td>197.5 F</td>
<td>168.0 F</td>
</tr>
<tr>
<td>Compressor Vi required</td>
<td>3.6</td>
<td>3.6</td>
</tr>
<tr>
<td>Refrigerant flow</td>
<td>92.0 Lbs/Min</td>
<td>632.3 Lbs/Min</td>
</tr>
<tr>
<td>Suction Specific Volume</td>
<td>7.832 Ft³/Lb</td>
<td>1.206 Ft³/Lb</td>
</tr>
<tr>
<td>Suction volume flow</td>
<td>720.5 ACFM</td>
<td>762.6 ACFM</td>
</tr>
<tr>
<td>Oil cooler heat rejection</td>
<td>488,200 Btu/HR</td>
<td>206,200 Btu/HR</td>
</tr>
<tr>
<td>Condenser heat rejection</td>
<td>2,978,000 Btu/HR</td>
<td>3,235,000 Btu/HR</td>
</tr>
</tbody>
</table>

The general guidelines and the major considerations for the refrigeration system design for the dual refrigerants of R-22 and R-717 application are as the following:

**Compressor:**

- R-22 has higher volume flow than R-717.
- R-22 has higher power consumption than R-717.
- R-717 has higher discharge pressure than R-22.
- R-717 has higher condensing pressure than R-22.
- R-717 has higher adiabatic efficiency than R-22.
- The compressor is to be based on R-22. But, all the parts are to be for R-717.

**Condenser and Cooler:**

- Cooler heat transfer: R-717 has about 17% higher U-value than R-22.
- Condenser heat transfer: R-717 has about 7% higher U-value than R-22.
- The surfaces are to be based on R-22. But, materials are to be good for R-717.

**Valves and Piping:**

- Valve sizes for Ammonia are smaller than R-22.
- Piping sizes for Ammonia are smaller than R-22.
- All the piping and valves are to be selected for R-22.
- All the piping and valves materials are to be good for R-717.
Automatic control valves are to be for R-22; but, are changeable for R-717 in future.

**Materials:**

- All materials are to be non-ferrous.
- Carbon steel tubes are to be used for all heat exchanger.
- No brass materials are to be in the system.
- All the materials are to be good for both R-717 and R-22

**Electrical Requirements and Considerations:**

- All the electrical are to be designed and constructed in accordance with R-717 requirement. All the electrical items, motors and control panel are to be designed and constructed for Class I, Division II in accordance with the NEC Code.

**System Conversion:**

- The system is to be designed and selected for R-22 for initial operation. A complete kit is to be provided by the manufacturer for future conversion to R-717. Also, the system manufacturer shall submit a list of conversion procedure and instruction such as drain, purge, evacuation and part changes for the R-717 conversion.

**Highlights of Safety and Code Required Procedures for the Installation:**

- If the system is to be fully complied with the code requirements, all the electrical items, motors and control panel are to be designed and constructed in accordance with the NEC code for Class 1, Division 2 hazardous location. The engine room ventilation rate is to be in accordance with the requirement of NFPA. ASHRAE 90.1 Standard also requires that no ammonia machinery or piping be located in the public hallway or public assembly areas.

- Some users in food processing industries would accept the IIAR (International Institute of Ammonia Refrigeration) Ammonia Machinery Room Ventilation regulation as the substitute for NEC code. That means to use ventilation method to reduce the machine room to a non-hazardous location.
Pure Hydrocarbon and Mixed Hydrocarbons as Refrigerant

All the refrigerants R-50, R-170, R-290, R-600, R-1150 and R-1270 are the hydrocarbons and it is actually one of the components of the natural gas. The natural gas consists of the hydrocarbons of methane, ethane, propane, propylene, butane, pentane and etc. The specific gravity of the natural gas might vary from 0.6 to 1.3.

R-50 (Methane CH\textsubscript{4}), R-1150 (Ethylene C\textsubscript{2}H\textsubscript{4}) and R-170 (Ethane C\textsubscript{2}H\textsubscript{6}) are used as the refrigerant for the low stage in a cascade refrigeration system. The refrigerants R-1270 (Propylene C\textsubscript{3}H\textsubscript{6}) and R-290 (Propane C\textsubscript{3}H\textsubscript{8}) are used as the refrigerant for the high stage. The R-600 (Butane C\textsubscript{4}H\textsubscript{10}) is mostly used as the refrigerant for heat pump application or used for water chilling duty.

Figure 14-1 shows the P-H diagram for R-22, the bubble point temperature and the dew point temperature are the same under the same pressure. Figure 14-2 is the P-H diagram for 100% pure Propylene (R-1270) and the Figure 14-3 is the P-H diagram for 100% pure Propane (R-290); it also shows the saturation temperatures are constant at constant pressure. However, if the hydrocarbon refrigerant is not pure and it is a mixture with other hydrocarbon, the P-H diagram for the mixture is shown in Figure 14-4, the saturated bubble point and dew point temperatures are not the same at the same pressure, or the saturated temperature changes in accordance with the equilibrium condition even under the same pressure.

Figure 14-5 is the P-H diagram for a Mole percent mixture of 95.39% of Propane, 3.26% of Ethane and 1.32% of Butane and other components of hydrocarbons. The Propane is not a pure substance, the bubble point and dew point temperatures are not the same even it is under saturated condition under same pressure. In this case, at a constant pressure of 14.9 Psia, the bubble point is -50 F and the dew point is -39 F.

Figure 14-6 is the P-H diagram for a Mole percent mixture of 80.7% of Propane, 19.2% of Ethane and 0.1% of Butane. The split between the bubble point temperature and the dew point temperature are much greater. For comparison, the saturated temperature would be -42.17 F at 14.9 Psia if it is a pure 100% Propane.

From the above cases, if the hydrocarbon such as R-1270, R-290, R-600 and etc. is to be used as the refrigerant, the purity of the refrigerant is to be in accordance with the ASHRAE Standard 34. Otherwise, the heat exchanger selection shall follow the equilibrium calculation if the refrigerant is a hydrocarbons mixture.
Figure 14-1  P-H Diagram R-22

Figure 14-2  P-H Diagram, Pure Propylene (R-1270)
Figure 14-3  P-H Diagram, Pure Propane (R290)

Figure 14-4  P-H Diagram, Mixed Hydrocarbons
Figure 14-5  P-H Diagram, Hydrocarbon Mixture

Figure 14-6  P-H Diagram, Mix Hydrocarbons
Chapter – 15    Refrigerant Liquid Feed Circuits

The way of supplying refrigerant liquid to the evaporator depends on the type of evaporator is used. The most common use evaporators for industrial refrigeration application are DX, Flooded, Liquid Recirculation and Thermosyphone.

Direct Expansion (DX):

Direct Expansion is also called Dry Expansion. The special character of a DX circuit is that the refrigerant is always through the tubes instead of shell; the liquid refrigerant is controlled by a thermostatic expansion valve which is to throttle just enough liquid to match the refrigeration load; the liquid is evaporated completely before leaving the heat exchanger and the thermostatic expansion valve also controls to provide just enough liquid refrigerant flow to provide 10°F to 15°F superheat at the outlet of the heat exchanger as shown in Figure 15-1.

Figure 15-2 and Figure 15-3 are typical DX shell-and-tube heat exchanger for either process fluid or brine or water cooling. The refrigerant is through the tubes and the fluid is through the shell is guided by vertical baffles. One of the advantage of the DX cooler has no submerge penalty for low temperature application because the refrigerant is through the tubes.

The advantages of using a DX heat exchanger are low cost, operation charge is minimized and oil return problem is simplified. However, the size of DX valve is limited; therefore, the DX evaporator is mostly used for smaller capacity system. Also, the DX system is usually used for higher evaporative temperature application.

The cooler may be with single refrigerant circuit or multi-circuit with each circuit fed by a DX valve or by a pilot operated valve as shown in Figure 15-4. A pilot operated valve is usually used for a larger capacity.

Sometimes, a pipe-in-pipe heat exchanger is used in the suction line to ensure the suction gas is superheated as shown in Figure 15-4.

Flooded:

Refrigerant liquid feed to a flooded type evaporator is through a throttling valve with liquid level controller to control the liquid level inside the shell; the refrigerant liquid level inside the shell is automatically maintained just enough to flood all the tubes no matter how the heat load fluctuates.

Figure 15-5 shows a sketch of refrigerant feed diagram for a half bundle shell-and-tube heat exchanger. The evaporator is with a single liquid inlet and multiple outlet connections arrangement. The sketch only shows an idea of liquid valve with a liquid level controller; no specific type of liquid feed valve and what type of liquid level controller is shown.

Figure 15-6 shows a half bundle shell-and-tube evaporator with a pilot operated low
Figure 15-1  Direct Expansion and Coil Heat Exchanger

Figure 15-2  Direct Expansion Shell-and-tube Heat Exchanger
Figure 15-3  DX Shell-and-tube Heat Exchanger

Figure 15-4  Direct Expansion Cooler with Pilot Operated DX Control Valve
Figure 15-5  Refrigerant Feed for Half Bundle Shell-and-Tube Heat Exchanger

Figure 15-6  Pilot Control Liquid Valve for Half Bundle Shell-and-Tube Heat Exchanger
pressure float valve. The liquid level is controlled by the pilot float valve.

In Figure 15-7, the refrigerant level is controlled by a modulating liquid level controller which is to regulate the pneumatic power to open or close the liquid throttling valve for the half bundle design evaporator. Figure 15-8 shows the same application except that the heat exchanger is a full bundle design. The liquid and gas separation is taken place in the upper shell.

Figure 15-9 shows the refrigerant feed for coil type flooded heat exchanger. The refrigerant from high pressure receiver or from intercooler is throttling through a low pressure float valve to the surge drum. The low pressure float valve also serves as the liquid level controller to maintain the liquid level inside the surge drum. If the design ET is 20 °F, the liquid in the surge drum is 20 °F. The coil is flooded with 20 °F refrigerant liquid. Evaporated gas in the coil flows back to the surge drum; the flash gas combines with the evaporated gas return to compressor suction.

Figure 15-10 is a typical refrigerant feed arrangement for a product cooler with flooded coil. The refrigerant valve is a low cost solenoid expansion valve which is controlled by a simple liquid level float switch. Unit cooler or product cooler air coil type heat exchangers are mostly used for food process industries.

**Liquid Recirculation:**

Liquid recirculation system is to pump the refrigerant liquid to the evaporator. This system is mostly used for food process industries to increase the heat transfer efficiency for the evaporators. This system is also used for those industrial applications to ensure no flash gas is in the liquid line for a high vertical column or for remote evaporators.

Figure 15-11 shows the sketch for the liquid recirculation set-up. The liquid from high pressure receiver or from intermediate intercooler, is throttling into a horizontal design pump receiver through a liquid valve which is controlled by a liquid level controller to maintain the liquid level in the pump receiver. The temperature in the pump receiver is controlled at the evaporative temperature as design. For example, the ET is -15 °F as shown in the diagram of Figure 15-11. The liquid from the pump receiver is overfed to the evaporator by the liquid pump and the amount to be fed to the evaporator(s) is about 2 to 4 time more than what is needed for the heat load depending on what type of refrigerant is used. The overfed rate is about 4:1 for ammonia and 3:1 for R-22. If the recirculation rate is 3:1; that means only 1/3 of liquid refrigerant is evaporated for the heat load; 2/3 of liquid and 1/3 of the vapor are returned to the pump receiver; the liquid and gas are separated in the pump receiver. The temperature is still at the saturated temperature of -15 °F. The evaporated gas in the evaporator combining with the flash gas from the liquid throttling to pump receiver are returned to the compressor suction. The gas leaving the pump receiver is still at the saturated temperature of -15 °F as shown.

For all the liquid recirculation systems, the refrigerant flow leaving the evaporator back to the pump receiver is a two-phase flow of liquid and vapor. The flow pressure drop of a two-phase flow is much greater than a single phase flow of either 100%
Figure 15-7  Modulating Refrigerant Feed Valve for Half Bundle Shell-and-Tube Heat Exchanger

Figure 15-8  Modulating Refrigerant Feed Valve for Full Bundle Shell-and-Tube Heat Exchanger
Figure 15-9  Flooded Evaporator  
Coil Type Heat Exchanger

Figure 15-10  Flooded Evaporator  
Product Cooler
Figure 15-11  Liquid Recirculation
liquid or 100% gas.

Only single refrigerant pump is shown in the sketch. However, to ensure continuous operation of the system, most liquid recirculation systems are equipped with dual liquid pumps; one is the stand-by for service and repair.

Figure 15-12 is a diagram of liquid recirculation circuit for a vertical pump receiver. The liquid level control is by a low pressure float type liquid level control valve. The refrigerant from high stage is throttling through this float valve. Then; combining with the return liquid/gas from evaporator, flows back to the vertical pump receiver. The liquid and gas are separated in the pump receiver; the saturated vapor is returned to compressor suction. Three liquid level switches are shown for the safety operation of the pump receiver; one is for high liquid level cutout; one is for high liquid level alarm and one is the float switch for low liquid level alarm.

The charts of Figure 15-13 and Figure 15-14 are the rates of evaporation for R-22 and R-717 respectively; it can be used for the rough approximate estimates to obtain the rate of evaporation for either R-22 or R-717.

For example: A 65 TR R-22 system, ET is 25°F; from Figure 15-13, the rate of evaporation is 0.205 GPM/TR; the amount of R-22 liquid evaporated is 65 x 0.205 = 13.325 GPM; the normal liquid recirculation rate recommended for R-22 is 3:1; therefore, the total liquid recirculation is 13.324 x 3 = 39.975 GPM or a 40 GPM pump is to be selected for this liquid recirculation system.

Same rules apply for the system if the refrigerant is ammonia instead of R-22, except that the liquid recirculation rate recommended for R-717 is 4:1.

Use P-H diagram to calculation the refrigerant flow if accurate rate of recirculation is desired.

**Thermosyphone:**

The typical liquid feed circuit for the Thermosypone evaporator is shown in Figure 15-15. The liquid is throttling through the liquid valve to the inlet of the surge drum which is on top of the heat exchanger; the pneumatic operator of the valve is controlled by the liquid level controller to control the liquid level in the surge drum. Usually, a pneumatic air supply approximately 20 psig is required for this modulating proportional liquid level control valve operation.

The refrigerant is circulated through the tubes by gravity thermal siphoning. Refrigerant liquid enters to the heat exchanger from the bottom side of the front channel box of the heat exchanger; the liquid and evaporated vapor mix returns to the surge accumulator from the top of the rear channel box of the heat exchanger. The liquid and gas are separated in the surge accumulator. The evaporated gas and the throttling flash gas are returned to compressor suction.
Figure 15-12  Liquid Recirculation System with Vertical Pump Receiver
Figure 15-13  Rate of Evaporation for R-22
Figure 15.14  Rate of Evaporation for R-717
Figure 15-15  Refrigerant Feed
Thermosympone Evaporator
Chapter – 16 Refrigerant Flow Control Valves

Refrigerant flow control valves are used to ensure the refrigeration system is performed in accordance with the conditions as designed, particularly during partial load operation. All the valves have some undesirable side effect such as line pressure drop, except it is required by design such as throttling valve. Therefore, when making valve selection, it is important to select a valve that is producing less pressure drop. All the valves must be checked if it is designed and constructed for refrigeration duty.

Every valve used for the refrigeration system must serve the purpose and the duty as designated, regardless if it is manual valve or automatic control valve. Manual stop valves are for the purpose of providing the conveniences of services and maintenances. Manual valves also include the check valves and hand expansion.

Typical hand expansion valve is shown in Figure 16-1. Manual expansion valve or needle valve is used as a throttling orifice in application wherever is required.

All the valves used for system performance control or for system safety should be fully automatic. The common use automatic valves and its function are described as the following:

**Solenoid Valve:**

Solenoid valve is an electrical control on-off valve. It is for the automatic control of refrigerant flow for either vapor or liquid line; the valve can be either normally open or normally closed. The electrical power supply can be either 220 or 110 or even 24 volts.

**Thermostatic Expansion Valve:**

Thermostatic expansion valve is also referred to as direct expansion (DX) or dry expansion valve. The DX valve is used as the automatic control throttling valve for DX evaporator. DX valve has two important elements as shown in Figure 16-2; one is the external bulb which is to allow enough refrigerant flow for the refrigeration capacity requirement and the other element is the external equalization line which is to restrict the refrigerant flow just enough to provide a 10 to 15°F superheat at the outlet of the heat exchanger.

**Low Pressure Float Valve:**

Low pressure float valve is a throttling expansion device. It provides a seal between the high side and the low side; it controls the refrigerant flow from the high side to the low side intercooler or flooded type heat exchanger, and to maintain a desired liquid level in that vessel.
Figure 16-1  Typical Manual Expansion Valve

Figure 16-2  Thermostatic Expansion Valve
Figure 16-3 is the cross section of the low pressure float valve. The assembly consists of the float ball, the body and the valve.

Figure 16-3  Cross Section of Low Pressure Float Valve

Figure 16-4  Low Pressure Float Valve Connections
Figure 16-4 shows the typical connections for the low pressure float valve. Liquid inlet is from high pressure receiver; liquid out is to evaporator or to low side. Gas balance line connects to the upper gas portion of the vessel and the liquid balance line connects to lower liquid portion of the vessel. The float ball inside of the float valve chamber controls the liquid level inside the vessel. The valve is open when the float ball is below the liquid line to allow more liquid flow into the vessel and the valve is close when the float ball is above the liquid level line.

Figure 16-5 is the typical application of the low pressure float valve for an intermediate flash intercooler; Figure 16-6 shows the typical connections of the low pressure float valve for an intercooler with a subcool coil. Figure 16-7 is a typical used of low pressure float valve for a half bundle design shell-and-tube evaporator.

Figure 16-8 shows another design of the low pressure float valve and Figure 16-9 is a pilot operated float liquid level control valve.

Low pressure float valve is generally used for larger capacity system. A high side received is needed to hold the refrigerant variation wherever a low pressure float valve is used.

**Proportional Level Control Valve:**

The function of a proportional level control valve is similar to the low pressure float valve, but it provides more accurate and positive control of the liquid level. The valve assembly consists of liquid level controller, control transmitter, valve positioner and the liquid control valve. This liquid level valve assembly requires pneumatic instrument air supply for the operation.

Diagram [B] of Figure 16-10 is the cut away of the level control caged displacer assembly of the Diagram [A]; Diagram [C] is the liquid level control valve with the valve operator, the “positioner”. The controller senses the liquid level in the vessel, it sends the output signal from the transmitter to the valve positioner to open or close the valve.

Figure 16-11 is a typical piping hookup for a flash type intercooler using a modulating level control valve. Figure 16-12 is a typical piping arrangement of the modulating valve for a half bundle shell-and-tube evaporator and the Figure 16-13 is the suggested piping hookup for a full bundle shell-and-tube flooded evaporator.

This proportional level control valve is an industrial grade product and is commonly used for heavy duty industrial refrigeration application. It is the more expensive than any other previous low pressure float valves.

**High Pressure Float Valve:**

High pressure float valve (Figure 16-14) is also referred to as liquid drain valve. The high pressure float valve is a seal between the high side and the low side. It is an expansion device to drain all the liquid from high side to an intercooler or a heat exchanger.
Figure 16-5  Low Pressure Float Valve for Intermediate Flash Intercooler

Figure 16-6  Low Pressure Float Valve for Intercooler with Subcool Coil

Figure 16-7  Low Pressure Float Valve with Shell-and-Tube Evaporator
Figure 16-8  Float Type Liquid Level Valve
Figure 16-10  Modulating Proportional Liquid Level Control Valve
Figure 16-11  Modulating Proportional Level Control Valve & Flash Type Intercooler

Figure 16-12  Typical Piping Hookup of Modulating Liquid Level Control Valve for Heat Exchanger
The operation of the high pressure float valve is similar to low pressure float valve except that it has no liquid level control function; therefore, the low side vessel must be large enough to take care of the fluctuation of the liquid refrigerant and must be able to hold the entire critical charge (balance charge) of the refrigerant.

Figure 16-15 shows a typical connection for the high pressure float valve. The Figure 16-16 is the cross section diagram of another design of the high pressure float.

**Liquid Level Float Switch:**

The general construction of a refrigerant float switches are shown in Figure 16-17. Diagram [A] is a hermetically sealed float switch and Diagram [B] is mercury float switch. Typical use of liquid level switch is for high liquid level alarm and cutout, low liquid level alarm and cutout as shown in Figures 16-5 and 16-6.

Liquid level switch can be also used as the liquid level control for flooded evaporator such as shown in Figure 16-18, wherein, the liquid level switch controls the liquid inlet solenoid valve; the high pressure liquid is throttling through the hand expansion valve. This type of arrangement is the least expansive way of providing the liquid level control as compared to other liquid level control arrangement.

The liquid level float switch can be either normally open or normally closed. SPST, SPDT or DPDT.

**Pressure Regulators:**

Pressure regulators include the valve that controls inlet pressure, outlet pressure and differential pressure.

Inlet pressure control valve is shown in Diagram [A] of Figure 16-19 and is to control the upstream pressure of the valve; Outlet Pressure control valve is shown in Diagram [B] of Figure 16-19 and it is to control the down stream pressure of the valve. Outlet pressure control valve is also referred to as the back pressure regulating valve which is to throttle the suction pressure from a higher pressure level to a lower pressure.

**Pressure Relief Valve:**

Pressure relief valve is a safety valve; it is to be installed for pressure vessel as determined by the Safety Code. The cross section of the relief valve is shown in Diagram [A] of Figure 16-20. Diagram [B] is the single relief valve and the Diagram [C] is a dual pressure relief valves with a change over valve. The function of the changer over valve is to open one of the valves while closes the other relief valve for service.
Figure 16-14  Cross Section of High Pressure Float Valve

Figure 16-15  High Pressure Float Valve Connections
Figure 16-16  Liquid Drain Valve  
(High Pressure Float)

[A] Hermetically Sealed Type  
[B] Mercury Switch Type

Figure 16-17  Refrigerant Float Switches
Figure 16-18  Liquid Level Control with Level Switch

Figure 16-19  Pressure Regulators
[A] Cross Section of Pressure Relief Valve

[B] Single Relief Valve

[C] Dual Pressure Relief Valve with Change Over Valve

Figure 16-20 Pressure Relief Valves
Chapter – 17  Liquid Carry Over and Gas/Liquid Disengagement

Refrigeration compressors are designed and constructed for gas compression duty only. It is detrimental to the compressor if refrigerant liquid is slug over to the compressor suction. Therefore, it is vital important that the refrigeration system is to be designed in such way to prevent the liquid carry over from the low side.

Liquid slug over to compressor suction might be due to poor design of the evaporator; or malfunction of control; or liquid trapped in the suction line.

Liquid carry over is not allowed for centrifugal and reciprocating compressors. Screw compressor is able to take very small flow of the liquid, as long as only if the flow is small and the flow is steady such as the oil/refrigerant liquid from oil return. To ensure the safe operation of the system, it is preferred that no liquid is presented in the suction even it is screw compressor.

The most effective way to prevent the liquid slug over to the compressor suction is to place a suction trap right before the compressor suction as shown in Figure 17-1.

The Suction Trap is also referred to as:
- Suction Drum.
- Suction Surge Drum.
- Suction Scrubber.
- Knockout Drum.
- Dead-end Trap.
- Suction Accumulator.

The suction scrubber, if used, it should be placed as close as possible to the compressor suction, to strip the liquid and oil out from the suction stream before it is entering into the compressor. Therefore, suction trap provided shall be large enough to accumulate the excess liquid from the low side.

Usually a suction trap should be installed if the evaporator is remote mounted or if no assurance that the evaporator is properly designed. No suction trap is required if the evaporator and the compressor are close coupled within the same engine room and the evaporator design is properly designed. Other conditions or applications that might require the use of suction trap are as the following:

- Multiple evaporators are used for the installation.
- Rapid changing refrigeration load condition.
- High range of cooling fluid.
- For central collection of liquid and oil.
- Quench or to desuperheating the high superheated suction gas.

The size of a suction trap is expressed in diameter by height of the vessel. It can be
Figure 17-1  Suction Trap (Knockout Drum) Arrangement
either vertical or horizontal design. In any cases, it must be sized large enough to provide the separation space for gas and liquid disengagement in the trap. The criteria of sizing the suction trap are as the following:

1.0 The cross section area of the suction trap should be large enough that the vertical refrigeration vapor travel velocity does not exceed the maximum allowable gravity separation velocity for the refrigerant used.

2.0 The height between the liquid level and the suction nozzle in the suction trap shall provide enough distance for vertical separation.

3.0 It shall be large enough to hold the accumulated liquid during operation.

The Figure 17-2 is the maximum allowable gravity separation vapor velocity for various commonly used refrigerants to prevent liquid carryover.

The separation velocity and travel distance are also applicable for the design of intermediate intercooler and flooded evaporator; enough separation space is to be provided to prevent liquid carryover from the intercooler or from the evaporator.

Some suction traps are equipped with moisture eliminator or demister to increase the separation efficiency. The suction trap size could be smaller if effective moisture eliminator is used.

Figure 17-3 is the typical suction trap for gravity separation. Normally, the suction connection is at the top of the suction trap. The alternate suction connection can be at the side of the suction trap if the engine room ceiling height is limited.

Figure 17-4 is the typical intermediate intercooler with a subcooling coil. The intercooler is with a demister (moisture eliminator); the inlet gas connection for the low stage compressor is at the upper portion of the intercooler, the perforated pipe is extended below the liquid level in the intercooler with a perforated plate to hold down the boiling and foaming of the refrigerant.

The common used accessories for the suction trap are as the following:

Pressure safety relief valve.
High liquid level alarm switch.
High liquid level cutout switch.
Liquid transfer switches.
Liquid level sight glass and indicator.
Liquid subcooling coil.
Vaporize heater.
Liquid drain valve or liquid transfer arrangement.
Oil drain valve.
Figure 17-2  Maximum Gas & Liquid Separation Velocity
Chapter – 18  Refrigerant Piping Systems

Normally, the refrigerant flows in the refrigerant piping system are single phase flow of either vapor or liquid. It is a requirement to design the refrigerant piping system for single phase flow, unless it is purposely designed for two-phase flow such as the return flow of a liquid recirculation system.

Refrigerant Piping for Single Phase Flow of Gas or Liquid:

Pressure drops in refrigerant piping system must be optimized, particularly the pressure drop in suction line of the refrigeration system; it must be carefully exam and considered, because it impacts greatly the selection of the compressor which adversely reducing the capacity of the compressor and also increases the power consumption for the system, especially for low temperature application.

Pressure drop allowance for the suction line is the balance of economic considerations of the size of suction line, power consumption, compressor size, suction line insulation and installation costs. As a general guide, suction line size should generally be selected for a pressure drop of 1 to 3 psi per 100 feet of pipe for evaporative temperature above 20 °F; 0.2 to 2 psi per 100 feet of pipe for evaporative temperature between 20 °F to -60 °F.

Increase discharge line pressure drop would increase the compression ratio for the compressor and reduces the volumetric efficiency of the compressor. This reduces capacity of the compressor, also increase power consumption of the system. Discharge line is generally sized to have 2 to 4 psi per 100 feet of pipe run.

Pressure drop can be critical as well in liquid lines since it is the most common cause of generating flash gas; flashing of liquid in the liquid line might cause loss of system capacity.

Piping line from condenser to receiver is generally sized to have maximum liquid velocity of 300 fpm for a free draining installation and a liquid velocity not to exceed 150 fpm for trapped liquid line.

For refrigerant piping from receiver to evaporator or intercooler, the maximum liquid velocity should be limited to 300 fpm and the line pressure drop is suggested to be less than 2 psi per 100 feet pipe run. When the liquid flow is against a riser column or when the pressure drop may generate flash gas, liquid subcooling is suggested to eliminate the flashing problem. Liquid pipe line should be insulated if subcooling method is used.

The liquid line between the expansion or throttling device and the evaporator should be short; common practice is to use same size pipe as the expansion outlet or one size larger than normal liquid line. Two-phase flow calculation should be considered if the
throttling device and the evaporator are not close, because this line is carrying both liquid and gas.

The minimum size of liquid or gas connection to auxiliary devices such as liquid float valve, surge drum, etc. should not be smaller than the connection of the float valve.

Table 18-1 lists the dimensions and the physical data for the copper or seamless steel tubing; Table 18-2 lists same information for the carbon steel piping. These tables also provide the tube or the pipe volume information for the estimation of refrigerant charge for the piping system.

Table 18-3 is for the quick selection of suction line sizes at various suction temperatures, pressure drops and at 105°F condensing temperature for R-22. Interpolation is allowed to determine the line capacity, however, the interpolation should be based between saturated suction temperatures at a fixed pressure drop. No interpolation is allowed between pressure drop columns. For other condensing temperature, use refrigerant flow rate and the pressure drop charts for the estimate. Use P-H diagram method to calculate the refrigerant flow if more accurate flow rate is desirable.

Table 18-4 is the quick selection chart to determine the capacities of discharge and liquid lines at fixed pressure drop or velocity as shown in the table for R-22. This chart is based on 105°F CT and 40°F ET. Use refrigerant flow rate and the pressure drop charts for CT and ET temperatures other than 105°F and 40°F.

Table 18-5 is the approximate K-Factors for valves and fittings for halocarbon refrigeration application.

Table 18-6 lists the equivalent lengths for the valves and fittings for halocarbon refrigerant.

Figure 18-1 is the quick reference chart for the approximate flow rate per TR for various ET and CT for R-22; the flow rate is Lbs. of R-22 per minute per TR. Again, use P-H diagram method to calculate the refrigerant flow if more accurate flow rate is desirable.

Figure 18-2 is the chart to determine the psi pressure drop per 100 ft. for vapor flow of R-22 for carbon steel piping. (Figure 18-5 is for copper tubing).

Figure 18-3 is to determine the velocity for the R-22 vapor flow for steel piping. (Figure 18-6 is for copper tubing).

Figure 18-4 is to determine the velocity of liquid flow of R-22 for steel piping. (Figure 18-7 is for copper tubing).

Figure 18-8 is the chart to convert the vertical R-22 liquid column to pressure drop between receiver to evaporator.

Table 18-7 shows the maximum TR capacities for various pipe sizes under various conditions for R-717.
<table>
<thead>
<tr>
<th>Size</th>
<th>Nominal Pipe Size (inches)</th>
<th>External Dia. (inches)</th>
<th>Internal Dia. (inches)</th>
<th>Thickness of Metal (inches)</th>
<th>Transverse Area (Square inches)</th>
<th>Lin. Ft. of Pipe Per Square Foot of External Surface</th>
<th>Length of Pipe in Ft. Containing 1-Cu. Ft. of Space</th>
<th>Length of Pipe in Ft. Occupying 1-Cu. Ft. of Space</th>
<th>Weight Per Foot Pounds</th>
</tr>
</thead>
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<td>.375</td>
<td>K</td>
<td>0.311</td>
<td>.033</td>
<td>.110</td>
<td>0.076</td>
<td>10.45</td>
<td>12.29</td>
<td>1195.0</td>
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Table 18-1 Dimensions and Physical Data
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<th>Length of Pipe Occupying Cu. Ft. of Space per Inch</th>
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Table 18-2  Dimensions and Physical Data
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NOTES:
1. Based on fluid flow at 105°F saturated condensing temperature
2. "IPS" data based on Schedule 40 steel piping  "OD" data based on Type L copper tubing

Table 18-3   R-22, Suction Line Capacities, TR
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<td>316</td>
<td>459</td>
</tr>
</tbody>
</table>

NOTES: * Based on fluid flow at 105°F saturated condensing temperature and 40°F saturated evaporating temperature
** "IPS" data based on Schedule 40 steel piping except that liquid lines 1½" and smaller are Schedule 80
*** "OD" data based on Type L copper tubing

Table 18-4  R-22, Discharge and Liquid Line Capacities, TR
### Ferrous Valves and Fittings

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
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<td>1/2</td>
<td>1.4</td>
<td>1.7</td>
<td>1.1</td>
<td>0.9</td>
<td>0.9</td>
<td>0.9</td>
</tr>
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<td>3/4</td>
<td>2.9</td>
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<td>1.3</td>
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<td>0.32</td>
<td>0.9</td>
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<td>0.32</td>
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<td>0.32</td>
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<td>0.32</td>
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<td>0.3</td>
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<td>0.9</td>
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<td>0.3</td>
<td>0.32</td>
<td>0.9</td>
</tr>
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<td>1.4</td>
<td>0.3</td>
<td>0.32</td>
<td>0.9</td>
</tr>
<tr>
<td>8</td>
<td>12.7</td>
<td>12.7</td>
<td>1.4</td>
<td>0.3</td>
<td>0.32</td>
<td>0.9</td>
</tr>
<tr>
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<td>13.9</td>
<td>13.9</td>
<td>1.4</td>
<td>0.3</td>
<td>0.32</td>
<td>0.9</td>
</tr>
<tr>
<td>12</td>
<td>15.2</td>
<td>15.2</td>
<td>1.4</td>
<td>0.3</td>
<td>0.32</td>
<td>0.9</td>
</tr>
</tbody>
</table>

### Non-Ferrous Valves and Fittings

<table>
<thead>
<tr>
<th>LINE SIZE (Inches)</th>
<th>GLOBE VALVE Flare or Sweat</th>
<th>ANGLE VALVE Flare or Sweat</th>
<th>SHORT-RADIUS ELL Flare or Sweat</th>
<th>LONG-RADIUS ELL Flare or Sweat</th>
<th>TEE, LINE-FLOW Flare or Sweat</th>
<th>TEE, BRANCH-FLOW Flare or Sweat</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/2</td>
<td>5.0</td>
<td>5.0</td>
<td>1.4</td>
<td>1.7</td>
<td>0.9</td>
<td>3.5</td>
</tr>
<tr>
<td>3/4</td>
<td>7.3</td>
<td>7.3</td>
<td>1.4</td>
<td>1.7</td>
<td>0.9</td>
<td>3.5</td>
</tr>
<tr>
<td>1</td>
<td>9.6</td>
<td>9.6</td>
<td>1.4</td>
<td>1.7</td>
<td>0.9</td>
<td>3.5</td>
</tr>
<tr>
<td>1 1/4</td>
<td>11.9</td>
<td>11.9</td>
<td>1.4</td>
<td>1.7</td>
<td>0.9</td>
<td>3.5</td>
</tr>
<tr>
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<td>14.2</td>
<td>14.2</td>
<td>1.4</td>
<td>1.7</td>
<td>0.9</td>
<td>3.5</td>
</tr>
</tbody>
</table>

**NOTES:**
- *K* = 2gh/p
- Based on Schedule 40 pipe
- Based on Type L copper tubing
- For screwed valves and fittings, use ferrous *K*-Factors
- For OD sizes above 2 1/4", use welded ferrous *K*-Factors

Table 18-5 K-Factors (Velocity Heads)
Hallocon Alves and Fittings
### Ferrous Valves and Fittings

**Line Size (Inches)**

<table>
<thead>
<tr>
<th>Valve Type</th>
<th>Angle Valves</th>
<th>Short-Radius Ell</th>
<th>Long-Radius Ell</th>
<th>Tee, Line-Flow</th>
<th>Tee, Branch-Flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/4</td>
<td>16</td>
<td>10</td>
<td>9</td>
<td>4.1</td>
<td>-</td>
</tr>
<tr>
<td>3/8</td>
<td>22</td>
<td>12</td>
<td>12</td>
<td>4.7</td>
<td>-</td>
</tr>
<tr>
<td>1/2</td>
<td>33</td>
<td>14</td>
<td>14</td>
<td>5.3</td>
<td>1.6</td>
</tr>
<tr>
<td>1</td>
<td>60</td>
<td>20</td>
<td>20</td>
<td>7.1</td>
<td>2.2</td>
</tr>
<tr>
<td>1 1/2</td>
<td>76</td>
<td>22</td>
<td>22</td>
<td>7.9</td>
<td>2.6</td>
</tr>
<tr>
<td>2</td>
<td>90</td>
<td>26</td>
<td>26</td>
<td>9.1</td>
<td>3.2</td>
</tr>
<tr>
<td>2 1/4</td>
<td>100</td>
<td>30</td>
<td>28</td>
<td>9.8</td>
<td>3.8</td>
</tr>
<tr>
<td>3</td>
<td>122</td>
<td>34</td>
<td>34</td>
<td>12.3</td>
<td>4.9</td>
</tr>
<tr>
<td>4</td>
<td>162</td>
<td>40</td>
<td>40</td>
<td>16.2</td>
<td>6.2</td>
</tr>
<tr>
<td>5</td>
<td>210</td>
<td>43</td>
<td>43</td>
<td>21.0</td>
<td>8.1</td>
</tr>
<tr>
<td>6</td>
<td>250</td>
<td>76</td>
<td>76</td>
<td>25.0</td>
<td>9.5</td>
</tr>
</tbody>
</table>

| **Non-Ferrous Valves and Fittings** |

<table>
<thead>
<tr>
<th>Valve Type</th>
<th>Angle Valves</th>
<th>Short-Radius Ell</th>
<th>Long-Radius Ell</th>
<th>Tee, Line-Flow</th>
<th>Tee, Branch-Flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/4</td>
<td>16</td>
<td>10</td>
<td>9</td>
<td>4.1</td>
<td>-</td>
</tr>
<tr>
<td>3/8</td>
<td>22</td>
<td>12</td>
<td>12</td>
<td>4.7</td>
<td>-</td>
</tr>
<tr>
<td>1/2</td>
<td>33</td>
<td>14</td>
<td>14</td>
<td>5.3</td>
<td>1.6</td>
</tr>
<tr>
<td>1</td>
<td>60</td>
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<td>20</td>
<td>7.1</td>
<td>2.2</td>
</tr>
<tr>
<td>1 1/2</td>
<td>76</td>
<td>22</td>
<td>22</td>
<td>7.9</td>
<td>2.6</td>
</tr>
<tr>
<td>2</td>
<td>90</td>
<td>26</td>
<td>26</td>
<td>9.1</td>
<td>3.2</td>
</tr>
<tr>
<td>2 1/4</td>
<td>100</td>
<td>30</td>
<td>28</td>
<td>9.8</td>
<td>3.8</td>
</tr>
<tr>
<td>3</td>
<td>122</td>
<td>34</td>
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<td>162</td>
<td>40</td>
<td>40</td>
<td>16.2</td>
<td>6.2</td>
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<tr>
<td>5</td>
<td>210</td>
<td>43</td>
<td>43</td>
<td>21.0</td>
<td>8.1</td>
</tr>
<tr>
<td>6</td>
<td>250</td>
<td>76</td>
<td>76</td>
<td>25.0</td>
<td>9.5</td>
</tr>
</tbody>
</table>

**NOTES:**

1. *L* = K(D/F)
2. Friction factors (f) determined at "practical" Reynolds Numbers based on 40°F suction lines having pressure drop of 1.8 psi/100 ft.
3. Based on Schedule 40 pipe
4. Flare, sweat, flanged, etc., and based on Type L copper tubing

Table 18-6  Equivalent Lengths
Halocarbon Valves and Fittings

242
Figure 18-1    R-22, Approximate Flow Rate per TR
Figure 18-2   R-22, Vapor Pressure Drop in Steel Piping
Figure 18-4  R-22, Liquid Velocity and Pressure Drop
Steel Piping
Figure 18-5  R-22, Vapor Pressure Drop in Copper Tubing
Figure 18-7  R-22, Liquid Velocity and Pressure Drop
Copper Tubing

NOTE: CURVE BASED ON LIQUID AT 90°F
AND CAN BE USED FOR LIQUID
FROM 70°F TO 110°F. FOR OTHER
TEMPERATURES, USE BASIC EQUA-
TION. CURVE DOES NOT ALLOW
FOR LIQUID FLASHING.
Figure 18-8  R-22 Liquid – Pressure and Elevation Equivalent
### Maximum Tons Refrigeration

<table>
<thead>
<tr>
<th>Nominal Pipe Size</th>
<th>Suction Line</th>
<th></th>
<th></th>
<th>Liquid Line</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inches</td>
<td>Suction Pressure—psig and</td>
<td>Discharge Line</td>
<td>Condenser to Receiver</td>
<td>Receiver to System</td>
</tr>
<tr>
<td></td>
<td>Corresponding Temperature—°F</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>5 psig (—17.2°)</td>
<td>20 psig (5.5°)</td>
<td>45 psig (30°)</td>
<td></td>
</tr>
<tr>
<td>1/8</td>
<td>—</td>
<td>—</td>
<td>—</td>
<td>—</td>
</tr>
<tr>
<td>3/8</td>
<td>0.6</td>
<td>1.1</td>
<td>2.0</td>
<td>2.5</td>
</tr>
<tr>
<td>1/2</td>
<td>1.2</td>
<td>2.2</td>
<td>4.1</td>
<td>6.0</td>
</tr>
<tr>
<td>1/4</td>
<td>2.2</td>
<td>4.0</td>
<td>7.5</td>
<td>8.0</td>
</tr>
<tr>
<td>1/2</td>
<td>3.4</td>
<td>6.0</td>
<td>15.0</td>
<td>15.0</td>
</tr>
<tr>
<td>3/4</td>
<td>6.4</td>
<td>11.8</td>
<td>21.6</td>
<td>24.0</td>
</tr>
<tr>
<td>13/4</td>
<td>12.1</td>
<td>22.2</td>
<td>42.0</td>
<td>26.0</td>
</tr>
<tr>
<td>1</td>
<td>19.1</td>
<td>35.5</td>
<td>65.0</td>
<td>29.0</td>
</tr>
<tr>
<td>2</td>
<td>31.5</td>
<td>59.0</td>
<td>108.0</td>
<td>40.0</td>
</tr>
<tr>
<td>3</td>
<td>64.0</td>
<td>118.0</td>
<td>240.0</td>
<td>74.0</td>
</tr>
<tr>
<td>4</td>
<td>172.0</td>
<td>306.0</td>
<td>600.0</td>
<td>192.0</td>
</tr>
<tr>
<td>8</td>
<td>362.0</td>
<td>650.0</td>
<td>1200.0</td>
<td>420.0</td>
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<td>10</td>
<td>640.0</td>
<td>1180.0</td>
<td>2160.0</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>940.0</td>
<td>1850.0</td>
<td></td>
<td></td>
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</table>

Friction Loss Basis
in psig Per 100 Ft
Eqiv. Length
0.25 0.50 1.00 1.00 — —

Table 18-7  R-717 Pipe Line Maximum TR

### Equivalent Feet of Pipe for Valves and Fittings

<table>
<thead>
<tr>
<th>Line Size, Inches</th>
<th>IPS 1/8</th>
<th>IPS 1/4</th>
<th>IPS 3/8</th>
<th>IPS 1/2</th>
<th>IPS 3/4</th>
<th>IPS 1</th>
<th>IPS 1 1/4</th>
<th>IPS 1 1/2</th>
<th>IPS 2 1/4</th>
<th>IPS 2 1/2</th>
<th>IPS 3</th>
<th>IPS 4</th>
<th>IPS 6</th>
<th>IPS 8</th>
<th>IPS 10</th>
<th>IPS 12</th>
</tr>
</thead>
<tbody>
<tr>
<td>Globe Valve (Open)</td>
<td>14</td>
<td>16</td>
<td>22</td>
<td>28</td>
<td>36</td>
<td>42</td>
<td>57</td>
<td>69</td>
<td>83</td>
<td>118</td>
<td>168</td>
<td>225</td>
<td>280</td>
<td>335</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Angle Valve (Open)</td>
<td>7</td>
<td>9</td>
<td>12</td>
<td>15</td>
<td>18</td>
<td>21</td>
<td>28</td>
<td>34</td>
<td>42</td>
<td>57</td>
<td>83</td>
<td>117</td>
<td>140</td>
<td>165</td>
<td></td>
<td></td>
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<tr>
<td>Standard Elbow</td>
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<td>2</td>
<td>3</td>
<td>4</td>
<td>5</td>
<td>6</td>
<td>8</td>
<td>10</td>
<td>12</td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Standard Tree</td>
<td>3</td>
<td>4</td>
<td>5</td>
<td>6</td>
<td>8</td>
<td>9</td>
<td>12</td>
<td>14</td>
<td>17</td>
<td>22</td>
<td>34</td>
<td>44</td>
<td>56</td>
<td>65</td>
<td></td>
<td></td>
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<tr>
<td>(Through Side Outlet)</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>2</td>
<td>2</td>
<td>3</td>
<td>4</td>
<td>2</td>
<td>2</td>
<td>3</td>
<td>3</td>
<td>4</td>
<td>5</td>
<td>6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(Through Straight Flow)</td>
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<td>1</td>
<td>1</td>
<td>2</td>
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<td>3</td>
<td>4</td>
<td>5</td>
<td>6</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

IPS—Iron pipe size.
Values shown are average values based on standard-weight pipe.
Note—Right-angle, forged fittings give a high pressure drop and their use in suction lines should, in general, be discouraged.

Table 18-8  Equivalent Lengths
Ammonia (R-717) Valves and Fittings

251
Table 18-8 is the equivalent pipe lengths for the valves and fittings for R-717.

Table 18-9 is the equivalent pipe lengths for welded ells and return bends for R-717 piping.

Table 18-10 is the maximum refrigerant flow capacities in Lbs. per Minute for piping from condenser to receiver for R-717.

Figure 18-9 is the quick reference chart for the approximate flow rate per TR for various ET and CT for R-717; the flow rate is Lbs. per minute per TR. Use P-H diagram method to calculate the refrigerant flow if more accurate flow rate is desirable.

Figure 18-10 is the chart to determine the psi pressure drop per 100 ft. for vapor flow of R-717 refrigerant; this chart is for carbon steel pipe and it is for discharge gas line and for suction piping for temperature between 40°F to -40°F.

Figure 18-11 is the same as Figure 18-10 except it is for suction piping for R-717 and for temperature between -40°F to -100°F.

Figure 18-12 is to determine the pressure drop psi per 100 ft. for R-717 liquid flow.

Figure 18-13 is the chart to estimate the vertical R-717 liquid static column to pressure drop between receiver and evaporator.

General Guides for Pipe Size Selection and Estimation:

1.0 Design and develop the refrigerant flow diagram for the refrigeration system.
2.0 Sketch a proper P-H diagram for the refrigeration system.
3.0 Show all the design operating conditions on the P-H diagram including all the pressures, temperatures, enthalpy points, CT, ET, TR, intermediate temperature if any.
4.0 Indicate all the design pressure drops and superheated allowance.
5.0 Calculate the refrigerant flows for each section.
6.0 Estimate the piping run between each component and each piping section; estimate the number of elbows, tees, valves, etc. convert all the valves and fittings into equivalent length to obtain the total length of the pipe run for each section.
7.0 Select and adjust the pipe size to match the pressure drop allowed.
8.0 Pressure drop calculated must be less than the pressure drop allowed in the design, particularly the suction piping pressure drop. If it is higher than the designed, either to change the pipe to a larger size pipe to reduce the pressure drop or changing the compressor selection to accept the larger pressure drop.
9.0 Check vertical static column to see if liquid subcooling is needed for the system.
10.0 Check to avoid liquid traps in the piping line to protect the compressor.

Refrigerant Piping System for Two-Phase Gas & Liquid Flow:

In refrigerant piping system, only three areas involve with two-phase flow.
Figure 18-9  R-717 Approximate Flow Rate per TR
### Equivalent Feet of Pipe for Welded Elks and Return Bends

<table>
<thead>
<tr>
<th>Radius of Bend</th>
<th>Designation of Bend</th>
<th>Fitting</th>
<th>1 1/2&quot; IPS</th>
<th>1 1/4&quot; IPS</th>
<th>2 IPS</th>
<th>2 1/4&quot; IPS</th>
<th>3&quot; IPS</th>
<th>4&quot; IPS</th>
<th>6&quot; IPS</th>
<th>8&quot; IPS</th>
<th>10&quot; IPS</th>
<th>12&quot; IPS</th>
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</thead>
<tbody>
<tr>
<td>1D</td>
<td>1R</td>
<td>45° Ell</td>
<td>1.3</td>
<td>1.6</td>
<td>1.9</td>
<td>2.6</td>
<td>3.2</td>
<td>3.9</td>
<td>5.8</td>
<td>7.7</td>
<td>9.6</td>
<td>11.5</td>
</tr>
<tr>
<td></td>
<td>90° Ell</td>
<td>2</td>
<td>2.5</td>
<td>3.0</td>
<td>4.0</td>
<td>5.0</td>
<td>6.0</td>
<td>9.0</td>
<td>12.0</td>
<td>15.0</td>
<td>18.0</td>
<td></td>
</tr>
<tr>
<td></td>
<td>180° Return Bend</td>
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<td>4.0</td>
<td>5.4</td>
<td>6.8</td>
<td>8.1</td>
<td>12.1</td>
<td>16.1</td>
<td>20.0</td>
<td>24.2</td>
<td></td>
</tr>
<tr>
<td>1 1/2D</td>
<td>1 1/2R</td>
<td>45° Ell</td>
<td>1.0</td>
<td>1.3</td>
<td>1.6</td>
<td>1.9</td>
<td>2.2</td>
<td>2.8</td>
<td>3.9</td>
<td>5.1</td>
<td>6.4</td>
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<td></td>
<td>90° Ell</td>
<td>1.5</td>
<td>2.0</td>
<td>2.5</td>
<td>3.0</td>
<td>3.5</td>
<td>4.5</td>
<td>6.0</td>
<td>8.0</td>
<td>10.0</td>
<td>12.0</td>
<td></td>
</tr>
<tr>
<td></td>
<td>180° Return Bend</td>
<td>2.0</td>
<td>2.7</td>
<td>3.4</td>
<td>4.0</td>
<td>4.7</td>
<td>6.0</td>
<td>8.1</td>
<td>10.7</td>
<td>13.4</td>
<td>16.1</td>
<td></td>
</tr>
</tbody>
</table>

R—Radius, D—Nominal Pipe Diameter.

### Table 18-9 Equivalent Lengths
Ammonia (R-717) Welded Elks and Return Bends

### Condenser to Receiver Liquid Lines
Capacities in Total Pounds of Ammonia per Minute

<table>
<thead>
<tr>
<th>Pipe Size</th>
<th>100 ft/min Velocity</th>
<th>200 ft/min Velocity</th>
</tr>
</thead>
<tbody>
<tr>
<td>3/4&quot;</td>
<td>3.55</td>
<td>7.1</td>
</tr>
<tr>
<td>5/8&quot;</td>
<td>5.90</td>
<td>11.8</td>
</tr>
<tr>
<td>1&quot;</td>
<td>10.9</td>
<td>21.8</td>
</tr>
<tr>
<td>1 1/4&quot;</td>
<td>18.2</td>
<td>36.4</td>
</tr>
<tr>
<td>1 1/2&quot;</td>
<td>32.5</td>
<td>65.0</td>
</tr>
<tr>
<td>2&quot;</td>
<td>44.6</td>
<td>89.2</td>
</tr>
<tr>
<td>2 1/4&quot;</td>
<td>55.0</td>
<td>107.0</td>
</tr>
<tr>
<td>3&quot;</td>
<td>125.0</td>
<td>242.0</td>
</tr>
<tr>
<td>4&quot;</td>
<td>186.5</td>
<td>373.0</td>
</tr>
<tr>
<td>6&quot;</td>
<td>322</td>
<td>644</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pipe Size</th>
<th>100 ft/min Velocity</th>
<th>200 ft/min Velocity</th>
</tr>
</thead>
<tbody>
<tr>
<td>3/4&quot;</td>
<td>3.55</td>
<td>7.1</td>
</tr>
<tr>
<td>5/8&quot;</td>
<td>5.90</td>
<td>11.8</td>
</tr>
<tr>
<td>1&quot;</td>
<td>10.9</td>
<td>21.8</td>
</tr>
<tr>
<td>1 1/4&quot;</td>
<td>18.2</td>
<td>36.4</td>
</tr>
<tr>
<td>1 1/2&quot;</td>
<td>32.5</td>
<td>65.0</td>
</tr>
<tr>
<td>2&quot;</td>
<td>44.6</td>
<td>89.2</td>
</tr>
<tr>
<td>2 1/4&quot;</td>
<td>55.0</td>
<td>107.0</td>
</tr>
<tr>
<td>3&quot;</td>
<td>125.0</td>
<td>242.0</td>
</tr>
<tr>
<td>4&quot;</td>
<td>186.5</td>
<td>373.0</td>
</tr>
<tr>
<td>6&quot;</td>
<td>322</td>
<td>644</td>
</tr>
</tbody>
</table>

Table 18-10 Ammonia Flow Capacities, Lbs/Min Condenser to Receiver
PERFORMANCE DATA

FRICION LOSS, PSI PER 100 FT. OF EQUIVALENT LENGTH

FRICION IN GAS MAINS
AMMONIA

BASED ON
DISCHARGE GAS TEMP. = 100° F.

NOTES:
1. EXAMPLE ABCD FOR DISCHARGE LINES.
2. EXAMPLE EFGH FOR SUCTION LINES.
3. FOR COMPUTING FRICION LOSS BEYOND LIMITS OF GRAPH USE FOLLOWING RELATION:
SHIPS: FRICION LOSS OF THE REFRIGERANT FLOW.

VARGES:
A. DIRECTLY AS THE SQUARE OF THE WEIGHT.
B. INVERSELY AS THE DENUTY.
C. INVERSELY AS THE FIFTH POWER OF THE INTERNAL PIPE DIAMETER.

FRIGERANT FLOW RATE, POUNDS PER MINUTE

Figure 18-10  R-717 Vapor Pressure Drop in Steel Pipe
Discharge and Suction Line ET 40°F to -40°F
Figure 18-11  R-717 Vapor Pressure Drop in Steel Pipe
Discharge & Suction Line, ET -40°F to -100°F
Figure 18-12  R-717 Liquid Pressure Drop in Steel Pipe
(A) One is inside the DX evaporator; the DX evaporator is designed for this application and it is not part of the piping system.

(B) The second area is the piping between the outlet of the expansion device and the inlet of the evaporator, this section of the piping is very close; the usual rule of thumb is to use the same size of the outlet connection of the expansion device.

(C) The third area is the return piping of a liquid recirculation system.

The pressure drop is considerably higher than a single phase vapor flow if the flow is a two-phase flow of gas and liquid combination. Figure 18-14 is the pressure drop multiplier factor for the two-phase flow for R-22 based on vapor pressure drop Lbs/Min flow of same pipe size. Figure 18-15 is the pressure drop multiplier factor for the two-phase flow for R-717 based on vapor pressure drop Lbs/Min flow of same pipe size. Both Figure 18-14 and Figure 18-15 are the simplified version of the pressure drop multiplier curves. It is suggested to use industrial acceptable software if more accurate estimate is required.
Figure 18-14  Flow Pressure Drop Multiplier for R-22

Figure 18-15 Flow Pressure Drop Multiplier for R-717
Chapter – 19  Brines and Brine Piping System

Brine is the common name referring to all the secondary refrigerants or heat transfer fluids other than water. Water is the least expensive and it has best heat transfer properties among all the secondary refrigerants. But, its freezing point is 32 °F and therefore, cannot be used for low temperature refrigeration application.

It is the preference to use direct refrigerant to cool the product or process in industrial refrigeration application; heat transfer is penalized and power consumption is increased if secondary refrigerant is used instead of direct refrigerant. However, in some cases, use brine (secondary refrigerant) instead of direct refrigerant is more desirable and justifiable because of one or the combination of the following reasons:

1.0 Cost considerations; sometimes, the construction of brine piping system is less expensive than direct refrigerant piping due to the requirements of codes and regulations, particularly if the installation is with remote evaporator (more than 100 feet) or if the evaporator is located at high elevation. Cost comparison should be carefully examined to see if use of brine is justified; the brine system involves additional costs of brine cooler, brine pump, pumping HP, compressor power consumption, brine and refrigerant charges, maintenance and etc.

2.0 In the event of leakage, brine is more acceptable than refrigerant.

3.0 For multiple evaporator application, individual temperature control for each evaporator or space may be simpler, more accurate or more efficient with brine instead of refrigerant.

4.0 Some application requires sharp but short load peaks application, brine storage tank is able to provide the flywheel effect to average out the refrigeration load requirement.

5.0 The need of defrosting can be eliminated if brine is used for spray cooler application.

Inhibitor should be considered whenever brine is used to reduce the corrosion. The corrosion might be from the brine or might be caused by oxygen and/or carbon dioxide contaminations.

Corrosion inhibition is to form a surface of barrier that protects the metal from attack, some by reaction of the inhibitor and some by absorption. Environment Stabilizers and adjusters are also being used for corrosion inhibition; this is stabilizing or altering the overall environment to maintain in alkaline condition. It is best to use combination of the two types of additives, inhibitors and stabilizers. Do not use oxidants inhibitors such as sodium chromate as inhibitor for Ethylene Glycol. Both Methylene Chloride and Trichlorethylene do not require additives.
Figure 19-1 lists the common brines and the practical working brine temperature for the brine. The figure also shows the recommended tube material for the heat exchanger. The temperature limits shown are the approximate brine leaving temperature limit for flooded shell-and-tube heat exchanger. The temperature limitation is that heat exchanger selection would become unpractical and unviable if the leaving brine temperature is lower the temperature indicated. However, if the heat exchanger is DX, the leaving brine temperature limit for propylene glycol can be extended down to \(-25^\circ F\) and ethylene glycol down to \(-30^\circ F\) respectively at the proper brine concentration.

There are basically five groups of common brines:

(A) Salt brines such as calcium chloride and sodium chloride.

(B) Glycol brines such as propylene glycol and ethylene glycol.

(C) Alcohol brines such as methanol and ethanol.

(D) Chlorinated Hydrocarbon brines such as Methylene Chloride and Trichlorethylene.

(E) Floriated hydrocarbon brine such as R-11.

Sodium chloride and calcium chloride brine were very popular because low cost. Due to the corrosive nature of the brine, maintenance and initial cost are to be balanced and justified if sodium chloride or calcium chloride is to be selected. Other commonly used brines for refrigeration application are propylene glycol, ethylene glycol and methanol. Propylene glycol is the preferred brine for food, beverage and daily product application.

Ethylene Glycol is none corrosive brine. But, it becomes an acid end product when it is oxidized by air. Therefore, an inhibitor should be used for Ethylene Glycol brine. A ready mixed and specially engineered heat transfer fluids of propylene glycol and ethylene glycol are available in the market from suppliers such as Dow Chemical.

Ethanol and methanol are the derivative from alcohol and it is a flammable liquid. Ethanol is more expensive than methanol and the heat transfer properties are not as good as methanol brine. Therefore, methanol is better choice over ethanol.

Trichlorethylene and Methyl Chloride are halocarbons. It is only used as secondary refrigerant for very special low temperature application that well below \(-40^\circ F\).

R-11 is a very good secondary refrigerant; it can be used for very low temperature brine application down to \(-150^\circ F\). The only problem is that it is restricted due to CFC O-zone problem.

The safety and toxicity concerns for various common secondary refrigerants are listed as the following:
Figure 19-1  Common Brine Working Temperatures

<table>
<thead>
<tr>
<th>BRINE TYPE</th>
<th>EXCELLENT</th>
<th>GOOD</th>
</tr>
</thead>
<tbody>
<tr>
<td>SODIUM CHLORIDE</td>
<td>Cu, Ni.</td>
<td>Copper</td>
</tr>
<tr>
<td>PROPYLENE GLYCOL</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>ETHYLENE GLYCOL</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>CALCIUM CHLORIDE</td>
<td>Cu, Ni.</td>
<td>Steel *</td>
</tr>
<tr>
<td>METHANOL</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>ETHANOL</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>TRICHLOROETHYLENE</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>METHYLENE CHLORIDE</td>
<td>Copper</td>
<td>Copper</td>
</tr>
</tbody>
</table>
Table 19-01  **Brine Safety Ratings**

<table>
<thead>
<tr>
<th>Brine</th>
<th>Safety Rating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sodium Chloride</td>
<td>Non-flammable</td>
</tr>
<tr>
<td>Calcium Chloride</td>
<td>Non-flammable</td>
</tr>
<tr>
<td>Propylene Glycol</td>
<td>Flammable, moderate fire hazard. Flash point 210°F to 225°F undiluted.</td>
</tr>
<tr>
<td>Ethylene Glycol</td>
<td>Flammable, moderate fire hazard. Flash point 232°F to 240°F undiluted.</td>
</tr>
<tr>
<td>Methanol water</td>
<td>Flammable, fire hazard. Flash point 54°F to 60°F undiluted. Flash point 75°F 30% solution.</td>
</tr>
<tr>
<td>Ethanol</td>
<td>Flammable, fire hazard. Flash point 55°F undiluted.</td>
</tr>
<tr>
<td>Trichloroethylene</td>
<td>Non-flammable at ordinary ambient</td>
</tr>
<tr>
<td>Methylene Chloride</td>
<td>Non-flammable at ordinary ambient</td>
</tr>
<tr>
<td>R-11</td>
<td>Non-flammable</td>
</tr>
</tbody>
</table>

Table 19-02  **Brine Toxicity Data**

<table>
<thead>
<tr>
<th>Brine</th>
<th>Safety Rating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sodium Chloride</td>
<td>Non-toxic. Suitable for direct contact with food</td>
</tr>
<tr>
<td>Calcium Chloride</td>
<td>Non-toxic.</td>
</tr>
<tr>
<td>Propylene Glycol</td>
<td>Non-toxic. Suitable for food processing</td>
</tr>
<tr>
<td>Ethylene Glycol</td>
<td>Toxic.</td>
</tr>
<tr>
<td>Methanol water</td>
<td>Toxic.</td>
</tr>
<tr>
<td>Ethanol</td>
<td>Non-toxic</td>
</tr>
<tr>
<td>Trichloroethylene</td>
<td>Toxic. Threshold limit is about 100 ppm</td>
</tr>
<tr>
<td>Methylene Chloride</td>
<td>Slight toxicity, but generally considered Non-toxic, but threshold limit is about 500 ppm</td>
</tr>
<tr>
<td>R-11</td>
<td>Non-toxic, but threshold limit is about 1000 ppm</td>
</tr>
</tbody>
</table>

The data and the transportation properties of the brine required for heat transfer calculation or heat exchanger selection are as the following:

1. Brine solution concentration, percent by weight.
2. Brine inlet temperature and outlet temperature.
3. Specific heat.
4. Specific gravity.
5. Viscosity.
6. Thermal conductivity.

The transportation properties for the three typical brines of propylene glycol, ethylene glycol and methanol are shown in the figures from 19-2 to 19-16:

**For Propylene Glycol Brine:**
For Ethylene Glycol Brine:

For Methanol Brine:

Most heat exchanger selection software programs include the transportation properties of the commonly used brine. Refer to brine software program for more accurate brine data and all other brines.

Sometime, the refrigeration system is required to be designed to cool the process fluid. The process fluid is not common brine and therefore, the specific heat, specific gravity, thermal conductivity and viscosity information are to be furnished by the user in order to make heat exchanger selection.

Brine concentration is usually expressed by percent by weight, not percent by volume. Percent by Weight and Percent by Volume conversion curve is shown in the Figure 19-2 for propylene glycol, Figure 19-7 for ethylene glycol and Figure 19-12 for methanol.

The viscosity of a brine effects heat transfer greatly. The viscosity is higher when the brine temperature is lower. Also, the viscosity is higher when the brine concentration is higher except the ethanol and methanol (see viscosity curve in Figure 19-16 for methanol. Ethanol has similar characteristic as methanol).

In heat exchanger selection and heat transfer calculation, the Reynolds number is a very important indicator if the heat exchanger for brine cooling is viable.

**REYNOLDS NUMBER (Re):**

\[
\text{Re} = \frac{M \times \text{TUBE ID}}{\mu \times 2.42}
\]
Figure 19-2  Propylene Glycol – Freeze Point
Figure 19-3  Propylene Glycol – Specific Gravity
Figure 19-4  Propylene Glycol – Thermal Conductivity
Figure 19-5  Propylene Glycol – Specific Heat

Figure 19-6  Propylene Glycol – Viscosity
Figure 19-7  Ethylene Glycol – Freeze Point
Figure 19-8  Ethylene Glycol – Specific Gravity
Figure 19-9  Ethylene Glycol – Thermal Conductivity

Figure 19-10  Ethylene Glycol – Specific Heat
Figure 19-11 Ethylene Glycol – Viscosity
Figure 19-12  Methanol – Freeze Point
Figure 19-13 Methanol – Specific Gravity
Figure 19-14  Methanol – Thermal Conductivity
Figure 19-15  Methanol – Specific Heat
Figure 19-16   Methanol – Viscosity
\[
M = \frac{\text{GPM} \times \text{S.G.} \times 500 \times \text{NO.PASS}}{\text{NO.TUBES} \times \text{TUBE FLOW AREA}}
\]

\[
Re = \frac{V \times 92,880 \times \text{S.G.} \times \text{TUBE ID}}{\mu}
\]

Where:
- \(\text{GPM}\) = Gallons per minute brine flow
- \(\text{S.G.}\) = Specific gravity of the brine
- \(\text{NO.PASS}\) = Pass arrangement of the heat exchanger
- \(\text{NO.TUBES}\) = Number of tubes in the heat exchanger.
- \(\text{TUBE AREA}\) = Square feet area of tube surface.
- \(\text{ID}\) = Tube ID, Feet
- \(V\) = Brine velocity, ft/sec.
- \(\text{TUBE ID}\) = Inside diameter of the tube, Ft.
- \(\mu\) = Viscosity, Centipoise

From the above formula, it is obvious that the viscosity of the fluid is the most important factor to determine the Reynolds number; the Reynolds number is larger if the viscosity is lower. For a viable heat exchanger selection, the \(Re\) value must be greater than 2,100 for turbulent flow. If the \(Re\) number is below 2,100, it represents the laminar flow and the heat exchanger selection is not viable.

Viscosity will greatly impact on the size of the heat exchanger. Higher viscosity of the brine requires bigger heat exchanger and larger heat transfer surface. From experience, in case of R-22 application, the general guide for the tube external surface requirements is as the following:

<table>
<thead>
<tr>
<th>For R-22</th>
<th>Viscosity</th>
<th>Sq.Ft./TR Tube Surface</th>
</tr>
</thead>
<tbody>
<tr>
<td>2 to 3 CP</td>
<td>8</td>
<td>Sq.Ft./TR</td>
</tr>
<tr>
<td>3 to 5 CP</td>
<td>10</td>
<td>Sq.Ft./TR</td>
</tr>
<tr>
<td>5 to 8 CP</td>
<td>14</td>
<td>Sq.Ft./TR</td>
</tr>
<tr>
<td>12 CP</td>
<td>18</td>
<td>Sq.Ft./TR</td>
</tr>
</tbody>
</table>

**Brine Flow Calculation:**

Brine flow is determined by:
(1) Heat Load.
(2) Brine Transportation Properties:
   Specific Gravity
   Specific Heat
(3) Brine Temperature Difference or Range

**Formula for Brine Flow & Refrigeration Capacity:**

\[
\text{TR} = \frac{\text{GPM} \times \text{S.G.} \times \text{C}_p \times \varnothing \text{T}}{24}
\]

\[
\text{Btu/Min} = \text{GPM} \times 8.33 \times \text{S.G.} \times \text{C}_p \times \varnothing \text{T}
\]

\[
\text{Btu/Hr} = 499.8 \times \text{GPM} \times \text{S.G.} \times \text{C}_p \times \varnothing \text{T}
\]

- **TR** = Tons of Refrigeration
- **GPM** = Fluid Flow Rate, Gallons per Minute
- **S.G.** = Specific Gravity of the Fluid
- **C}_p = Specific Heat of the Fluid, BTU/LB-°F
- **\varnothing T** = Temperature Range of the Fluid, °F, \((T_2 - T_1)\)

Or

\[
\text{GPM} = \frac{\text{TR} \times 24}{\text{S.G.} \times \text{C}_p \times \varnothing \text{T}}
\]

\[
\text{GPM} = \frac{\text{Btu/Min}}{8.33 \times \text{S.G.} \times \text{C}_p \times \varnothing \text{T}}
\]

\[
\text{GPM} = \frac{\text{Btu/Hr}}{499.8 \times \text{S.G.} \times \text{C}_p \times \varnothing \text{T}}
\]

**Example for Brine Flow Calculation:**

Capacity: 220 TR
Brine range: 27.5°F to 21°F
Fluid: 35% By weight ethylene glycol brine
Brine average temperature \(= \frac{27.5 + 21}{2} = 24.3^\circ F\)

Brine properties at the mean temperature of 24.3°F:
- Specific Gravity: 1.056
- Specific Heat: 0.845 BTU/LB-°F

\[
TR = \frac{GPM \times S.G. \times C_p \times \Delta T}{24}
\]

\[
GPM \times 1.056 \times 0.845 \times (27.5 - 21) = \frac{220}{24}
\]

Brine Flow = 910 GPM

**Formula for Brine Pumping HP:**

\[
BHP = \frac{W \times H}{33,000 \times \text{EFFICIENCY}}
\]

- \(BHP\) = Brake HP.
- \(W\) = Weight of liquid. Lbs/Min.
- \(H\) = Total head in feet (Corrected for brine flow)

Heat Exchanger Tube Material
for Common Refrigerants
and Secondary Refrigerants

<table>
<thead>
<tr>
<th></th>
<th>Excellent</th>
<th>Good</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sodium Chloride</td>
<td>Cu. Ni</td>
<td>Copper</td>
</tr>
<tr>
<td>Propylene Glycol</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>Ethylene Glycol</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>Calcium Chloride</td>
<td>Cu. Ni.</td>
<td>Steel*</td>
</tr>
<tr>
<td>Methanol</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>Trichlorethylene</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>Methyl Chloride</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>Sodium Nitrate</td>
<td>S.S.</td>
<td>Copper*</td>
</tr>
<tr>
<td>Calcium Nitrate</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>Acetone</td>
<td>Copper</td>
<td>Copper</td>
</tr>
<tr>
<td>Component</td>
<td>Material</td>
<td></td>
</tr>
<tr>
<td>-----------------</td>
<td>------------------</td>
<td></td>
</tr>
<tr>
<td>Ethanol</td>
<td>Copper</td>
<td></td>
</tr>
<tr>
<td>Water</td>
<td>Copper</td>
<td></td>
</tr>
<tr>
<td>Sea Water</td>
<td>Cu. Ni.</td>
<td></td>
</tr>
<tr>
<td>Halocarbons</td>
<td>Copper</td>
<td></td>
</tr>
<tr>
<td>Hydrocarbons</td>
<td>Copper</td>
<td></td>
</tr>
<tr>
<td>Ammonia</td>
<td>Steel</td>
<td></td>
</tr>
</tbody>
</table>

**Notes:**
*Must be inhibited.*
Ethylene Glycol and Sodium Chloride brine are also recommended to be inhibited.

**Brine Concentration Determination:**

In general speaking, the brine concentration should be optimized such that the freezing point of the brine should be more than 5°F below the lowest possible evaporative temperature for the refrigeration system. For example: In the case of ethylene glycol brine system, the lowest evaporative temperature is 14°F, the freezing temperature of the brine should be at least 14°F - 5°F = 9°F, therefore, the concentration of the ethylene glycol brine should be 24% by weight. General speaking, too much of conservative by increase the percent of brine concentration has no advantage to the operation and it is harmful to energy consumption, because the heat transfer is penalized and the power consumption is increased due to higher viscosity.

**Brine Piping:**

The brine piping system construction shall generally follow the water piping system practices.

Figure 19-17 is the equivalent length of valves and fittings for the carbon steel piping in addition to the regular piping run. Figure 19-18 is the pressure drop through pipe for water flow.

Figure 19-19 is the pressure drop correction factor for brine flow. The flow must be turbulent flow; that is the Turbulent Flow Factor ($\beta$) must be greater 0.1.
<table>
<thead>
<tr>
<th>Type of Fitting</th>
<th>3/4</th>
<th>1</th>
<th>1 1/4</th>
<th>1 1/2</th>
<th>2</th>
<th>2 1/4</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gate Valve—Open</td>
<td>3</td>
<td>5</td>
<td>6</td>
<td>.8</td>
<td>1.2</td>
<td>1.4</td>
<td>1.7</td>
<td>2.0</td>
<td>2.5</td>
<td>2.7</td>
<td>3.0</td>
<td>3.5</td>
<td>4.0</td>
<td>4.5</td>
<td>5.0</td>
<td>6.0</td>
</tr>
<tr>
<td>Globe Valve—Open</td>
<td>16</td>
<td>21</td>
<td>26</td>
<td>.35</td>
<td>.43</td>
<td>.54</td>
<td>.65</td>
<td>.80</td>
<td>.95</td>
<td>1.10</td>
<td>1.30</td>
<td>1.40</td>
<td>1.60</td>
<td>1.80</td>
<td>2.10</td>
<td>2.50</td>
</tr>
<tr>
<td>Angle Valve—Open</td>
<td>8</td>
<td>11</td>
<td>14</td>
<td>.18</td>
<td>.20</td>
<td>.25</td>
<td>.31</td>
<td>.40</td>
<td>.45</td>
<td>.51</td>
<td>.60</td>
<td>.70</td>
<td>.80</td>
<td>.91</td>
<td>1.10</td>
<td>1.25</td>
</tr>
<tr>
<td>Standard 45 deg. Elbow</td>
<td>.8</td>
<td>1.0</td>
<td>1.3</td>
<td>.16</td>
<td>.20</td>
<td>.25</td>
<td>.31</td>
<td>.38</td>
<td>.45</td>
<td>.50</td>
<td>.58</td>
<td>6</td>
<td>8</td>
<td>8.5</td>
<td>10</td>
<td>11</td>
</tr>
<tr>
<td>Standard 90 deg. Elbow</td>
<td>1.5</td>
<td>2.0</td>
<td>2.5</td>
<td>.35</td>
<td>.45</td>
<td>.50</td>
<td>.57</td>
<td>.65</td>
<td>.80</td>
<td>1.10</td>
<td>1.30</td>
<td>1.40</td>
<td>1.60</td>
<td>1.80</td>
<td>2.10</td>
<td>2.50</td>
</tr>
<tr>
<td>Medium Sweep 90 deg. Elbow</td>
<td>1.4</td>
<td>1.8</td>
<td>2.3</td>
<td>.30</td>
<td>.35</td>
<td>.45</td>
<td>.52</td>
<td>.68</td>
<td>.8</td>
<td>9</td>
<td>10</td>
<td>11</td>
<td>14</td>
<td>15</td>
<td>17</td>
<td>19</td>
</tr>
<tr>
<td>Long Sweep 90 deg. Elbow</td>
<td>1.0</td>
<td>1.5</td>
<td>2.0</td>
<td>.25</td>
<td>.30</td>
<td>.35</td>
<td>.45</td>
<td>.56</td>
<td>.67</td>
<td>.8</td>
<td>9</td>
<td>10</td>
<td>12</td>
<td>14</td>
<td>16</td>
<td>18</td>
</tr>
<tr>
<td>Square Elbow 90 deg.</td>
<td>3.0</td>
<td>4.5</td>
<td>5.5</td>
<td>.75</td>
<td>.9</td>
<td>1.2</td>
<td>1.5</td>
<td>2.2</td>
<td>2.4</td>
<td>2.7</td>
<td>3.0</td>
<td>3.3</td>
<td>3.8</td>
<td>4.3</td>
<td>4.9</td>
<td>5.6</td>
</tr>
<tr>
<td>Class Return Bend</td>
<td>3.5</td>
<td>5</td>
<td>6</td>
<td>.8</td>
<td>10</td>
<td>12</td>
<td>15</td>
<td>18</td>
<td>20</td>
<td>24</td>
<td>26</td>
<td>30</td>
<td>35</td>
<td>40</td>
<td>45</td>
<td>54</td>
</tr>
<tr>
<td>Stand Tee—Full Size Branch*</td>
<td>3.0</td>
<td>4.5</td>
<td>5.5</td>
<td>.75</td>
<td>.9</td>
<td>1.2</td>
<td>1.5</td>
<td>2.2</td>
<td>2.4</td>
<td>2.7</td>
<td>3.0</td>
<td>3.3</td>
<td>3.8</td>
<td>4.3</td>
<td>4.9</td>
<td>5.6</td>
</tr>
<tr>
<td>Stand Tee—Through Run</td>
<td>1.0</td>
<td>1.5</td>
<td>2.0</td>
<td>.25</td>
<td>.30</td>
<td>.35</td>
<td>.45</td>
<td>.56</td>
<td>.67</td>
<td>.8</td>
<td>9</td>
<td>10</td>
<td>12</td>
<td>14</td>
<td>16</td>
<td>18</td>
</tr>
<tr>
<td>Sudden Enlargement from d to D**</td>
<td>1.5</td>
<td>2.0</td>
<td>2.5</td>
<td>.35</td>
<td>.45</td>
<td>.50</td>
<td>.65</td>
<td>.80</td>
<td>.91</td>
<td>1.10</td>
<td>1.30</td>
<td>1.40</td>
<td>1.60</td>
<td>1.80</td>
<td>2.10</td>
<td>2.50</td>
</tr>
<tr>
<td>d/D = 1/4</td>
<td>1.0</td>
<td>1.5</td>
<td>2.0</td>
<td>.25</td>
<td>.30</td>
<td>.35</td>
<td>.45</td>
<td>.56</td>
<td>.67</td>
<td>.8</td>
<td>9</td>
<td>10</td>
<td>12</td>
<td>14</td>
<td>16</td>
<td>18</td>
</tr>
<tr>
<td>d/D = 1/2</td>
<td>.8</td>
<td>1.3</td>
<td>1.6</td>
<td>.22</td>
<td>.26</td>
<td>.33</td>
<td>.38</td>
<td>.49</td>
<td>.56</td>
<td>6.4</td>
<td>8.1</td>
<td>10</td>
<td>11</td>
<td>13</td>
<td>15</td>
<td>17</td>
</tr>
<tr>
<td>Sudden Contraction from D to d**</td>
<td>.8</td>
<td>1.3</td>
<td>1.6</td>
<td>.22</td>
<td>.26</td>
<td>.33</td>
<td>.38</td>
<td>.49</td>
<td>.56</td>
<td>6.4</td>
<td>8.1</td>
<td>10</td>
<td>11</td>
<td>13</td>
<td>15</td>
<td>17</td>
</tr>
<tr>
<td>d/D = 1/4</td>
<td>.6</td>
<td>1.0</td>
<td>1.3</td>
<td>.22</td>
<td>.26</td>
<td>.33</td>
<td>.38</td>
<td>.49</td>
<td>.56</td>
<td>6.4</td>
<td>8.1</td>
<td>10</td>
<td>11</td>
<td>13</td>
<td>15</td>
<td>17</td>
</tr>
<tr>
<td>d/D = 1/2</td>
<td>.6</td>
<td>1.0</td>
<td>1.3</td>
<td>.22</td>
<td>.26</td>
<td>.33</td>
<td>.38</td>
<td>.49</td>
<td>.56</td>
<td>6.4</td>
<td>8.1</td>
<td>10</td>
<td>11</td>
<td>13</td>
<td>15</td>
<td>17</td>
</tr>
<tr>
<td>Ordinary Pipe Entrance with upstream end of pipe flush with inside of tank.</td>
<td></td>
<td>.9</td>
<td>1.5</td>
<td>1.8</td>
<td>2.4</td>
<td>3.0</td>
<td>3.6</td>
<td>4.5</td>
<td>5.1</td>
<td>6.0</td>
<td>6.6</td>
<td>7.5</td>
<td>9.0</td>
<td>11</td>
<td>12</td>
<td>14</td>
</tr>
<tr>
<td>Entrance with pipe projecting into tank beyond inside face (Borda Entrance)</td>
<td>1.5</td>
<td>2.0</td>
<td>2.5</td>
<td>.35</td>
<td>.40</td>
<td>.50</td>
<td>.60</td>
<td>.70</td>
<td>.8</td>
<td>9.0</td>
<td>10</td>
<td>12</td>
<td>13</td>
<td>15</td>
<td>17</td>
<td>19</td>
</tr>
</tbody>
</table>

*Pressure drop through side outlet, or from side outlet through run.

**Equivalent feet of the smaller diameter pipe, "d."

Figure 19-17  Water Main – Equivalent Length Valves & Fittings
Figure 19-18  Water Resistance Pressure Drop through Pipe
KINEMATIC VISCOSITY (Y) *

*Kinematic Viscosity (Y) is Obtained From Formula $Y = \frac{U}{S}$

Where U is Absolute Viscosity in Centipoise & S is Specific Gravity

Figure 19-19  Pressure Drop Correction Factor for Brine Flow
\[
\beta = \frac{D \times V \times S}{\mu} > 0.1
\]

Where:

- \(\beta\) = Turbulent Factor
- \(D\) = Diameter of the pipe, inches.
- \(V\) = Velocity brine flow, Ft/Sec.
- \(S\) = Specific gravity.
- \(\mu\) = Absolute viscosity, centipoises.
Refrigeration compressors need oil for lubrication to reduce friction and to reduce wear. Oil flow rate required for various compressors varies. Centrifugal compressor requires minimal amount of oil flow, reciprocating compressor requires more oil circulation and the amount of oil injection and oil recirculation for screw compressor is a lot. Oil carryover in refrigeration system is not avoidable except the oil free compressor; however, all the oil free compressors are not designed for refrigeration duty. Most refrigeration systems of modern design are with screw compressors, therefore, the refrigeration system is to be designed with features to handle and to manage the oil problems.

The Roles of Lubricant for Refrigeration Compressors:

- To prevent or minimize wear of moving parts.
- To provide mechanical cooling for the compressor.
- To reduce gas leakage, to improve compression ratio and efficiency.
- To reduce friction driving horsepower.
- Protecting metal from rust and corrosion.
- Washing away dirt and wear particles.

The Basic Terms for Compressor Oil:

Oil characteristics are defined by viscosity, solubility, pour point, floc point, flash point, stability, foaming, dielectric and cleanliness; the details are as the following:

Viscosity:

Viscosity of the oil is the degree to which oil resists internal flow measured by SUS (Saybolt Universal Seconds), usually at one of three temperatures of 0°F, 100°F or 210°F. SUS is measured in Saybolt cup. It is the time in seconds for the oil to flow out from the cup at the temperature specified. If the oil takes 100 seconds 100°F, the oil is 100 SUS 100°F. Viscosity of oil must be high enough to provide a hydrodynamic film for compressor and yet thin enough to flow through the refrigeration system at low temperature application. SUS is the unit which is merely a measurement of viscosity, it has nothing to do with the quality of the oil.

Solubility (Miscibility):

Solubility or miscibility of the oil is the ability of the oil to mix and dissolve with the refrigerant. The oil return rate is good if the solubility of the oil with the refrigerant is good. The solubility is depending upon the oil characteristic, temperature and pressure.
Pour Point:
Pour point is that the lowest temperature which the oil still can flow by gravity. Pour Point is measured by putting oil in a beaker and measure the lowest temperature that it pours.

Floc Point:
Floc point is the temperature when refrigerant mixed with oil shows flocculent and waxy material. Floc Point also called cloud point. The Floc Point must be below the lowest system temperature of the system. Refrigeration oil must be exceptionally free of wax.

Flash Point:
Flash point is the temperature the oil is breaking down. The flash point reflects the volatility of the oil.

Stability:
Stability of the oil is the quality of oil to minimize corrosion, gum, varnish, tar and sludge, protection against oxidation and thermal instability, particularly the quality of minimizing chemical reaction to metal due to moisture in the oil. Stability is a vital characteristic of good lubricant, which must resist the deterioration over prolonged period of service to minimize compressor wear and maintenance problem. Also, the oil should not easily carbonize when working at high temperatures or around hot spots such as oil heater.

Defoaming Characteristics:
Oil having low foaming character is preferred in refrigeration system. Most refrigeration oils contain with foam inhibitor ingredient.

Dielectric Strength:
Water content in oil is measured by the electric current that can pass through the oil. Refrigeration oil should have extremely good dielectric strength, particularly in hermetic compressor system. Refrigeration oil should be kept water free.

Cleanness:
Oil should be free of contamination.

Requirements of Compressor Oil:
Refrigeration system is a clean, sealed and enclosed environment. The lubrication oil selected should be compatible with the refrigerant and the compressor; the compressor oil shall be with the preferred features as listed below:
1.0 Lubricity at all operating temperature and pressure of the refrigerant.
2.0 Chemically stable at high and low temperatures.
3.0 Thermally resistant to high temperature breakdown.
4.0 High flash points.
5.0 Dirt and moisture free.
6.0 Low pour points to resist congealing in condensers and evaporators.
7.0 Exceptionally wax free.
8.0 Formulated to have proper viscosities for specific applications.

**Influence of Oil Migration in Refrigeration System**

Lubrication oil in refrigeration system is not avoidable. But, the oil is harmful to heat transfer in refrigeration system. The heat transfer efficiency is reduced if the oil film accumulated on the heat transfer surface is increased. Figure 20-1 is the typical diagram showing the relationship between the oil film and the heat transfer coefficient at various oil film accumulations on the tubes. For example: if a clean tubes heat exchanger is having heat transfer coefficient of 162 Btu/Hr/Ft²/°F, the heat transfer coefficient is reduced to 126.36 162 Btu/Hr/Ft²/°F or 78% if the tubes surface is with 1.5 Mils oil film thickness; the heat transfer coefficient is further reduced to only 58% or to 93.96 Btu/Hr/Ft²/°F at 4.0 Mils oil film accumulation on the tubes. The refrigeration capacity of the refrigeration system is greatly reduced if excess oil is remained in the evaporator; therefore, it is detrimental to refrigeration system if the oil is not removed from the evaporator. In view of this, therefore, the oil in refrigeration system should be managed in such way the oil carryover from compressor discharge is minimized and the oil is recovered and is returned to compressor or oil reservoir from the evaporator.

The Figure 20-2 is a typical lubrication oil migration diagram for a screw refrigeration system; the oil migration rates are 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).
Figure 20-2  Lubrication Oil Migration
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.

In this oil migration diagram of Figure 20-2, the oil should be separated as much as possible from the compressor discharge and then the oil is recovered from the evaporator. A good automatic oil management system is that the oil flows of (4) and (5) should be equal to the oil flow of (3) at normal operation of the refrigeration system.

### Wire Mesh or Demister Type Discharge Oil Separator

Wire mesh or demister type discharge oil separator is also referred to as Knitted wire fabric type discharge oil separator. The velocity is critical when this type of separator is used. Too high a velocity tends to apply too much force to the droplets on the wires, re-entraining some and carrying them through. Too low a velocity permits some of the droplets to follow the gas stream without impingement.

Maximum allowable vapor velocity through the element $V$, ft/sec:

$$ V = 0.35 \times \sqrt{\frac{\delta_L - \delta_V}{\delta_V}} $$

- $V$ = Velocity, Ft/Sec.
- $\delta_L$ = Density of the entrained liquid. Lbs/Ft$^3$
- $\delta_V$ = Density of the gas. Lbs/Ft$^3$

The minimum velocity for maintained separator efficiency of approximately 25% of the design value may be used.

The separation efficiency depends to some extent on the thickness of pad or pads and also on the vessel arrangement, but it is usually in the region of 30 to 50 ppm. Some application might be 15 to 25 ppm.

### Oil Vapor Carry Over by Compression

Assuming an oil separator is 100% efficiency to eliminate all the mists and droplets from the discharge from compressor, but, oil carryover still exists in the form of oil
vapor. Oil vapor is a gas and it carries over through the mechanical filter. The amount of oil vapor in the discharge line of the compressor 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 as the following:

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\(^3\)
- \( T \) = Temperature, \((460 + ^\circ\text{F})\)

\[ P_o \times V = W_o \times R_o \times T \]
\[ P_g \times V = W_g \times R_g \times T \]

- \( P_o \) = Oil vapor pressure, Psia
- \( W_o \) = Weight, Oil, Lbs
- \( R_o \) = Gas Constant of Oil
- \( P_g \) = Refrigerant vapor pressure, Psia
- \( W_g \) = Weight, Refrigerant, Lbs
- \( R_g \) = Gas Constant of Refrigerant
- \( V \) = Volume, ft\(^3\)
- \( T \) = Temperature, \((460 + ^\circ\text{F})\)

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

\[ R = \frac{1545}{MW} \]

- \( R \) = Gas Constant
- \( MW \) = Molecular Weight

\[ R_o = \frac{1545}{MW_o} \]
\[
R_g = \frac{1545}{MW_g}
\]

\[
MW_g = \text{Molecular Weight of the Refrigerant Vapor}
\]

\[
MW_o = \text{Molecular Weight of the Oil Vapor}
\]

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

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

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

Oil Vapor Carry Over In PPM
\[
= \frac{P_o}{P_g} \times \frac{MW_o}{MW_g} \times 10^6
\]

Gas pressure is the system pressure, therefore:

\[
P_g \cong P_s
\]

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:

Oil Vapor Carry Over In PPM
\[
= \frac{VP_o}{Ps} \times \frac{14.7}{760} \times \frac{MW_o}{MW_g} \times 10^6
\]

\[
VP_o = \text{Oil vapor pressure, mm Hg}
\]
Figure 20-3 shows the relationship of oil vapor carryover rate at various condensing temperature, evaporative temperature and percent of refrigeration load at 50 Hz or 60 Hz compressor speed.

**Refrigerant Gas Molecular Weight:**

The gas molecular weight for various commonly use refrigerant as listed in Table 20.1.

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Molecular Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ammonia, R-717</td>
<td>17.03</td>
</tr>
<tr>
<td>R-12</td>
<td>120.93</td>
</tr>
<tr>
<td>R-13</td>
<td>104.47</td>
</tr>
<tr>
<td>R-13B1</td>
<td>148.93</td>
</tr>
<tr>
<td>R-22</td>
<td>86.48</td>
</tr>
<tr>
<td>R-134a</td>
<td>102.03</td>
</tr>
<tr>
<td>R-290</td>
<td>44.10</td>
</tr>
<tr>
<td>R-1270</td>
<td>42.09</td>
</tr>
<tr>
<td>R-503</td>
<td>87.50</td>
</tr>
</tbody>
</table>

**Molecular Weight of Common Use Refrigeration Oil:**

The molecular weights of various refrigeration oil is listed in Table 20.2

<table>
<thead>
<tr>
<th>Refrigeration Oil</th>
<th>Molecular Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capella D</td>
<td>300 – 310</td>
</tr>
<tr>
<td>Suniso 4G</td>
<td>261 – 265</td>
</tr>
<tr>
<td>Capella D</td>
<td>240 – 243</td>
</tr>
</tbody>
</table>

**Vapor Pressure of Various Oils:**

The oil vapor pressures for various refrigeration oil is shown in Table 20.3,

<table>
<thead>
<tr>
<th>Refrigeration Oil</th>
<th>100°F</th>
<th>150°F</th>
<th>200°F</th>
<th>250°F</th>
<th>300°F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capella D</td>
<td>0.0001</td>
<td>0.0013</td>
<td>0.013</td>
<td>0.085</td>
<td></td>
</tr>
<tr>
<td>Terristic 52</td>
<td>0.000011</td>
<td>0.00014</td>
<td>0.0012</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Zerice S-41</td>
<td>0.00065</td>
<td>0.0074</td>
<td>0.055</td>
<td>0.31</td>
<td>1.3</td>
</tr>
<tr>
<td>Suniso 3GS</td>
<td>0.0003</td>
<td>0.004</td>
<td>0.031</td>
<td>0.2</td>
<td>0.9</td>
</tr>
<tr>
<td>Suniso 5G</td>
<td>0.000075</td>
<td>0.0011</td>
<td>0.0099</td>
<td>0.061</td>
<td>0.3</td>
</tr>
</tbody>
</table>
Figure 20-3  Oil Vapor Carryover – Screw Compressor
## Theoretical Oil Vapor Carryover Rates

Oil vapor carryover rate varies at various discharge temperature even with same condensing temperature based on the oil separator is with 100% separation efficiency, all droplets and mists are removed. The oil vapor carryover rate is lower if the discharge temperature is lower.

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 for these refrigerants at various discharge temperatures in Table 20.4.

### Table 20.4 Oil Vapor Carryover Rates

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

## Amount of Vaporized Oil Carryover to Evaporator

The example is made 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. Assuming the oil separator is with 100% separation efficiency, all droplets and mists are removed. The oil vapor carryover rates which are for 30 days operation for this refrigeration system are shown in Table 20.5.

### Table 20.5 Approximate Oil Carryover Rates

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Refrigerant Flow, Lbs/Hr</th>
<th>Oil Vapor Carryover, Lbs/Hr</th>
<th>If for 30 days, 24 hours per day. Oil Vapor carryover, Lbs</th>
<th>Approximate Oil Carryover in Gal of Oil at Sp.Gr 0.93</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-22</td>
<td>9,066.87</td>
<td>0.04633</td>
<td>33.36</td>
<td>3.55</td>
</tr>
<tr>
<td>R-717</td>
<td>1,305.20</td>
<td>0.01779</td>
<td>12.81</td>
<td>2.58</td>
</tr>
</tbody>
</table>
A lot of oil will be accumulated in the evaporator after long period of operation. The refrigeration capacity of the evaporator will be reduced if the oil is not removed from low side.

**General Classification of Oils**

There are three groups of compressor lubricant, i.e. mineral oils, synthetic oils and semisynthetic oils. Many selections are available, but, it shall be always follow the recommendation of the compressor maker as which type of oil is recommended. Some of the oil is miscible in the refrigerant and some is immiscible with the lubricant. The oil recovery rate is better if the oil use is miscible with oil.

Synthetic oil is specially designed for refrigeration use and it has many advantages as the following:

- Extending oil change period.
- Reduce oil carry over.
- Reduce oil consumption.
- Lower operating oil temperature.
- Higher flash point.
- Higher film strength.
- Reduce compressor wear and longer compressor operating life.
- Non-foaming character.
- Low toxicity.
- Good for wide operating temperature range.

The disadvantage of Synthetic oil is the higher initial cost and lesser compatibility with seal materials.

Oil in refrigeration system should be checked and analysis periodically for water content, viscosity, alkalinity/acidic level and chemical element content.

**Discharge Oil Separator, Oil Still and Oil Return**

The main purpose of using compressor discharge oil separator is to eliminate most of the oil mists and droplets from the discharge line before it is carried over to the refrigeration system. The amount of oil carryover from centrifugal compressor discharge is smallest and therefore, discharge oil separator is not usually used for centrifugal systems. Discharge oil separator is commonly used for reciprocating and discharge oil separator is part of the screw compressor standard accessories.

**Oil Separator for Reciprocating Compressors**

Most discharge oil separators designed for reciprocating compressor are wire mesh or demister type because of initial cost concern. The efficiency of wire mesh or demister oil separator is usually can be as good as 99.99%; the oil carryover is about 30 to 400 PPM by weight. Figure 20-4 shows a typical horizontal design of the mesh type oil separator; the Figure 20-5 is the typical arrangement of a vertical mesh type oil
separator. Both figures show that the separated oil drains to an oil float valve from the oil separator and then returned to oil receiver of back to the crankcase of the compressor.
Figure 20-6 is the typical piping arrangement for equal size reciprocating compressor in parallel operation with a common oil separator. Figure 20-7 is the typical piping arrangement for mix sizes of parallel reciprocating compressors with a common oil separator. In either case, the oil inlet connection to the crankcase oil float vale of the both compressors should be same level to the bottom of the common oil receiver.

**Oil Separator for Screw Compressors**

Screw compressor circulates lots of oil during operation; an oil separator must be used for the screw compressor, otherwise, the compressor is unsuitable for refrigeration application. This is the reason why an oil separator is part of a standard equipment of a screw compressor unit (See Figure 6-4). Almost all the oil separators for screw compressor are designed with multiple stages of oil separation and are equipped with coalescent filters; some designs can eliminate 100% of all mists and droplets of the oil, the remaining oil carryover from the discharge line in this case is the oil vapor.

Figure 20-8 shows the screw compressor with a horizontal oil separator. Most screw compressor units, the compressor is mounted on top of the horizontal oil separator, except that the motor is too heavy or over 1,000 HP. Figure 20-9 shows a typical screw compressor with vertical oil separator; the screw compressor-motor drive line is floor mounted on a steel base in this case.

**Oil Still and Oil Recovery**

If the oil is miscible with refrigerant, oil is returned with refrigerant to compressor suction automatically, no forced oil return system is required. However, if the oil is not miscible with the refrigerant, oil still with forced oil return as shown in Figure 20-10 should be provided for the refrigeration system.

The oil still and oil recovery system consists of an oil receiver with heater and operating valves. This set up provides oil return function 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 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 flow of oil in the suction. For reciprocating compressor or large amount of oil flow, it is suggested that the oil is to be returned to oil reservoir instead of compressor suction.

Figure 20-11 is a typical oil still and oil receiver with an oil pump; the oil is returned to oil reservoir by the oil pump.
Figure 20-6  Oil Separation and Oil Return
Equal Size Reciprocating Compressors

Figure 20-7  Oil Separation and Oil Return
Mix Size Reciprocating Compressors
Figure 20-8  Horizontal Oil Separator – Screw Compressor

Figure 20-9  Vertical Oil Separator – Screw Compressor
Figure 20-11  Oil Still with Pump Oil Return

Figure 20-12  Evaporator with Possible Oil Return Connections
Figure 20-12 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 than 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.

**Oil Return by Ejector (Eductor)**

The principal of operation of the ejector is to use high pressure gas to induce the oil and liquid sample from the evaporator, returning the oil/liquid mixture back to compressor suction. The cost of this type of oil return is the least expensive and it is only about 20% of the normal oil still and oil recovery system. However, the refrigeration capacity penalty is about 4% for this type of oil return because the hot gas is bypassed back to the suction.
Purge

Purge Unit is to remove none condensable gases and moisture from the refrigeration system. Non-condensable gases are mainly air and moisture. Moisture mixes with refrigerant might cause harmful acids and is harmful to the performance of the refrigeration system. Non condensable gases in the system are accumulated in the high side such in condenser; cause the condensing temperature to rise which increases the power consumption. Therefore, the main purpose of use purge is to improve the system efficiency and to reduce the power consumption.

Where to use purge unit:

1.0 Purge unit is used for low pressure refrigerant refrigeration system when the refrigerant pressure is below atmospheric pressure under normal temperature.

2.0 Use for refrigeration system that the suction pressure is below atmospheric pressure.

3.0 Very often, refrigeration system is improperly evacuated or purging after a portion of the system is opened for service or maintenance, particularly large refrigeration system. Purge unit is a very effective tool to restore the optimum efficiency of the refrigeration system.

In any case, the purge is a relatively inexpensive item as compared to total capital spending and yet is one of the best investments to ensure the peak efficiency of the refrigeration system.

Connecting point for purge:

For air cooled condenser or evaporative condenser system – The purge connection should be on top of the receiver if the receiver is located in a cool place. If the receiver is located in a warm space, the purge connection should be located at the top of liquid outlet of the air cooled or evaporative condenser.

For water cooled condenser system – The purge connection should be on top of the water cooled condenser if no receiver. Otherwise, the purge connection should be on top of the other side of the receiver away from the liquid inlet.

Methods of purge:

The least expensive way of purging is just use a manual purge valve. A lot of refrigerant is wasted if the manual purging is done directly to atmosphere without a refrigerated purge, because non-condensable gases are mixed with refrigerant. Therefore, manual purge is not economical and also might not be acceptable to environment concern.
Thermal drum refrigerated purge is the most popular purge arrangement being used for refrigeration application today. Figure 21-1 is a typical single drum purge arrangement and Figure 21-2 is a typical double drum thermal purge unit. Compressor type purge was used in the old days and is rarely used in today’s application.

The double drum purge unit is used for those cases that a greater degree of refrigerant recovery in the purging operation is desired and it is often used for low temperature refrigeration systems.

The working theory of thermal drum purge is to introduce high pressure liquid from receiver and evaporating in the coil inside of the drum. The mixture of refrigerant and non-condensable gases enters to the drum; the refrigerant vapor is condensed and returned to the system. The build-up of non-condensable gas pressure in the drum will activate a pressure regulating valve and 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 bled of manually.

**Refrigerant Transfer**

The transfer unit is used to transfer the refrigerant from one part to another part of the refrigeration system for the conveniences of service and maintenance. Transfer unit is mostly provided for centrifugal refrigeration system because centrifugal compressor cannot be used for refrigerant evacuation. Transfer unit is rarely used for reciprocating or screw compressor system because both reciprocating and screw compressor are positive displacement machine, it can be used to pump the refrigerant down to receiver without the transfer unit.

The typical flow diagram for refrigerant transfer unit is shown in Figure 21-3. The scope of a transfer unit includes a small compressor of 3 HP, 5 HP, 7-1/2 HP or 10 HP, water cooled or air cooled condenser and service valves as shown in Figure 21-3. Connection (A) is usually connected to top side of the condenser or evaporator; Connection (B) and Connection (C) are usually connected to a receiver.

Some case the transfer unit is included as part of the centrifugal system which is with storage receiver to provide the service convenience.

**Pumpout System**

If the transfer unit is not part of the system and it is constructed with a receiver as a portable unit, this unit is a pumpout system and it is shown in Figure 21-4. The receiver in this pumpout unit is referred to as pumpout receiver. Figure 21-4 shows a typical pumpout system connected to a critical charged centrifugal refrigeration system. Critical charge system is a refrigeration system without a storage receiver. For off season shutdown or for system service and maintenance, the refrigerant is pumped out to the pumpout receiver.

The operation of the pumpout system is described as the following:
Figure 21-1  Single Drum Thermal Purge Unit
Figure 21-3  Typical Transfer Unit with Air Cooled Condenser
Figure 21-4  Typical Pumpout System Connection
1.0 **Liquid drain from evaporator to pumpout receiver** – Open V1, V2, V3, V4 and V8, close other valves. The pumpout receiver needs to be located below the evaporator for this operation.

2.0 **Transfer liquid from evaporator to pumpout receiver** – Open V1, V6, V3, and V4, close other valves, run the compressor of the transfer unit.

3.0 **Evacuate refrigerant gas from system to pumpout receiver** – Open V4, V8, V7, V9, and V5, close other valves, run the transfer unit.

4.0 **Transfer liquid from pumpout receiver back to the system** – Open V1, V4, V8, V7 and V2, close other valves, run the compressor of the transfer unit.

The process of evacuating refrigerant from the system to a pumpout receiver is referred to as pumpout cycle.

**Pumpdown Cycle**

If the refrigeration system is equipped with storage receiver as part of the system and the compressor is a reciprocating or screw, the refrigerant in the system can be pumped down to the receiver for system service. The process and operation of pumping down the refrigerant to storage receiver is referred to as pumpdown cycle. The pumpdown cycle can be designed for automatic operation with automatic valves and low pressure cutout.

**Liquid Transfer from Suction Trap (Scrubber)**

Suction trap (scrubber) is to prevent any liquid slug over from the evaporator. When the refrigerant liquid in the suction trap (scrubber) is accumulated over the setting of high level alarm, the liquid must be removed from the suction trap for continuous safe operation of the refrigeration system. The suction trap must be provided with some means of facility for liquid removal; otherwise, the system will face a great deal of operational difficulty in case the trap is full of liquid.

There are several ways to remove the liquid from the suction trap without shutdown the refrigeration system, the followings are some of the possible arrangements:

(a) Figure 21-5 shows a method of liquid removal by draining. The liquid drains to the external drum first and then close the drain and equalizing valves, use high pressure gas to push the liquid back to intermediate receiver or intercooler through a pressure regulating valve. The liquid can be gravity drain back to system receiver if the external drum is high enough to allow gravity drain passage.

(b) Figure 21-6 shows the methods by evaporating the liquid inside of the suction trap. One is to use electric heater and one is to use steam or hot water through the heat exchanger coil; the liquid is evaporated and return to compressor suction. It is important to drain the water completely out from the coil after the operation if steam or hot water is used, otherwise the water inside of coil will freeze and it might create a great deal of damage.
Figure 21-5  Liquid Removal by Draining

Figure 21-6  Liquid Removal by Evaporating
Figure 21-7  Liquid Removal by Discharge Hot Gas

Figure 21-8  Liquid Removal by Liquid Pump
(c) Figure 21-7 shows the method to remove the liquid by using compressor discharge hot gas. The hot gas is condensed in the coil, liquid in the suction trap is evaporated and return to compressor suction. The condensed liquid is collected in the liquid receiver then returns to intermediate intercooler, or drains to high pressure receiver if pitch slope allows.

(d) Figure 21-8 is the method use refrigerant liquid pump to return the liquid back to the system.

All the above methods can be designed and constructed for automatic instead of manually operated.
Chapter – 22  Electrical Codes for Refrigeration Equipment and System

The major electrical codes which are applicable to refrigeration system are NEC and IEC for equipment and NEMA and IP for enclosures. All the electrical including motors, enclosures of starters, control panels, and control devices; electrical wirings and conduits use for refrigeration system are under the jurisdiction of these codes and all are to be designed and constructed accordingly with the code requirements. The codes cover all hazardous and non-hazardous, indoor and outdoor locations application.

**NEC Code**

NEC (National Electric Code) is the highest code for electrical standard in USA. If the refrigeration system is for hazardous location, the system must be designed and constructed in accordance with the Article 500 of the NEC code.

Article 500 of the NEC Code defines the hazardous environment and degree of protection requirements by three categorizations as the following:

“Class” – This category is to define the atmospheres that the refrigeration equipment is to be located.

“Group” – This category is to define the hazardous characteristics of the atmosphere.

“Division” – This category is to define the degree of the hazardous concentration and degree of protection.

When a refrigeration system is to be designed and constructed in accordance with NEC code for hazardous application, all three categories of “Class”, “Group” and “Division” are to be indicated. An example of the description of the NEC for a hazardous application is shown in Figure 22-1. The full description for this example defines the type of atmosphere where the equipment is located; type of gas or vapor that the equipment is exposed to; type of protection is required and the maximum surface temperature for the equipment.

Further details of the definition for “Class”, “Group” and “Division” are outlined as the following:

“Class” is the hazardous atmosphere where the equipment is to be located:

Class I: Atmosphere is with gas or vapor.

Class II: Atmosphere is with combustible dusts.

Class III: Atmosphere is with flammable fibers or flyings.
Description for NEC Code
For Hazardous Application

Atmospheres:

Class I : For gas or vapor
Class II : For combustible dusts
Class III : For flammable fibers or flyings

Gas Group:

Group A: Acetylene
Group B: Hydrogen
Group C: Ethylene
Group D: Methane, Propane

Class I, Group D, Division II, T3

Area & Risk:

Division I: Hazard likely during normal operation
Division II: Hazard during abnormal operation

Surface Temperature Limit

Figure 22-1 Expression for NEC for Hazardous Application
“Group” is to identify various hazardous chemical substances in the atmosphere:

Group A: Acetylene.

Group B: Hydrogen or equivalent.

Group C: Ethyl ether, ethylene, cyclo-propane and etc.

Group D: Gasoline, hexane, naphtha, benzene, butane, propane, alcohol, acetone, benzyl, lacquer solvent, natural gas and etc.

Group E: Dusts of aluminum, magnesium and etc.

Group F: Dusts of carbon black, charcoal, coal or coke and etc.

Group G: Containing flour, starch or grain dust.

“Division” is to identify the probability existing of the hazardous chemical concentration and degree of protection required for the equipment.

Division I: Location in which the hazardous concentrations in the air exists continuously, intermittently or periodically under normal operating conditions or during maintenance.

Division II: Location in which the hazardous concentrations are only under abnormal or unusual conditions (breaking of a pipe, for instance).

If the industrial refrigeration system is used for hydrocarbon processing, the atmosphere shall be gas and vapor and the gases exposed are hydrocarbon gases; therefore, the installation area is “Class I” and the chemical exposure is “Group D”. The degree of protection of either “Division I” or “Division II” shall be determined by the process engineer of the user.

Beside of Class, Group and Division, NEC also defines the maximum surface operating temperature. The temperature identification numbers list by NEC 500-2(b) are as the following:

<table>
<thead>
<tr>
<th>Maximum Temperature</th>
<th>Identification Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>Degree C</td>
<td>Degree F</td>
</tr>
<tr>
<td>450</td>
<td>842</td>
</tr>
<tr>
<td>300</td>
<td>572</td>
</tr>
<tr>
<td>280</td>
<td>536</td>
</tr>
<tr>
<td>260</td>
<td>500</td>
</tr>
<tr>
<td>230</td>
<td>446</td>
</tr>
<tr>
<td>215</td>
<td>419</td>
</tr>
<tr>
<td>200</td>
<td>392</td>
</tr>
<tr>
<td>180</td>
<td>356</td>
</tr>
</tbody>
</table>
IEC Code

IEC (International Electric Code) is used by European countries. Some countries in Asia Pacific are having electric code closely resembling to IEC code. Similar to NEC code, the IEC code also have hazardous identification in three categories. An example of the description for hazardous application expression for the IEC code is shown in Figure 22-2.

The symbols denote the type of protection or enclosure is (Ex)e, (Ex)d, (Ex)f and (Ex)i as indicated. The degree of protection is outlined as the following:

(Ex)e – This refers to Increased Safety Design. This type of enclosure is used for all items of electrical equipment which do not produce sparks in normal operation.

(Ex)d – This refers to Flameproof Enclosure. All the parts of the item of electrical equipment where igniting arcs or sparks may be produced are housed in a flameproof enclosure. The sealing faces, cable entries, shaft glands, etc., are made with comparatively large gap length and limited gap clearances to prevent the transmission of flames or particles which might ignite the surrounding explosive atmosphere. During operation, explosive gas-air mixtures penetrate only seldom into the interior of the enclosure. Should an internal explosion occur, however, it is prevented from spreading to the mixture in the ambient atmosphere.

(Ex)p – This refers to Pressurized Enclosure. Those parts of the item of electrical equipment which may become a source of explosion, or the unit itself, are artificially ventilated with fresh air or and inert gas to produce a pressure above atmospheric inside the enclosure, thus preventing any explosive mixtures contained in the surrounding atmosphere from reaching the parts which may become the source of an explosion.

(Ex)i – This refers to Intrinsically Safe Construction. Generation of electric arcs (and of unduly high temperatures) which may ignite an explosive mixture is prevented by appropriately limiting the current and voltage as a function of inductance, capacitance and resistance. The power rating of the circuit is smaller than the minimum ignition power of 20 μWs.

IEC code lists the classification of hazardous areas into three zones according to the degree of hazard as the following:

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>165</td>
<td>329</td>
<td>T3B</td>
</tr>
<tr>
<td>160</td>
<td>320</td>
<td>T3C</td>
</tr>
<tr>
<td>135</td>
<td>275</td>
<td>T4</td>
</tr>
<tr>
<td>120</td>
<td>248</td>
<td>T4A</td>
</tr>
<tr>
<td>100</td>
<td>212</td>
<td>T5</td>
</tr>
<tr>
<td>85</td>
<td>185</td>
<td>T6</td>
</tr>
</tbody>
</table>
Description for IEC Code
For Hazardous Application

**Enclosure**
Degree of Protection:

- "e" = Increased Safety Design
- "d" = Flame Proof
- "p" = Pressurized
- "i" = Intrinsically Safe

**Gas Group:**
- IIA = Methane and Propane
- IIB = Ethylene
- IIC = Hydrogen, Acetylene

**Ex e IIA T4 Zone 0**

**Area Risk Classification:**
- Zone 0 = Continuous Hazard
- Zone 1 = Likely in Normal Operation
- Zone 2 = Hazard Under Abnormal

**Temperature Class:**
- Maximum Temperature Limit
  - T1 = 450°C
  - T2 = 300°C
  - T3 = 200°C
  - T4 = 135°C
  - T5 = 100°C
  - T6 = 85°C

Figure 22-2  Expression for IEC for Hazardous Application
Zone 0 – That is a zone in which an explosive gas and air mixture is continuously present or present for long periods.

Zone 1 – That is a zone in which an explosive gas and air mixture is likely to occur in normal operation.

Zone 2 – That is a zone in which an explosive gas and air mixture is not likely to occur, and if it occurs it will only exist for a short time.

Temperature Classification under IEC

There are six temperature classes for the grouping of gases relative to ignition temperature and for designing the maximum surface temperature for electrical equipment for use in hazardous area under IEC as shown in Table 22.2.

<table>
<thead>
<tr>
<th>Temperature Class</th>
<th>Maximum Surface Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>T1</td>
<td>450 °C</td>
</tr>
<tr>
<td>T2</td>
<td>300 °C</td>
</tr>
<tr>
<td>T3</td>
<td>200 °C</td>
</tr>
<tr>
<td>T4</td>
<td>135 °C</td>
</tr>
<tr>
<td>T5</td>
<td>100 °C</td>
</tr>
<tr>
<td>T6</td>
<td>85 °C</td>
</tr>
</tbody>
</table>

Classification Equivalent between IEC and NEC

The grouping of gases under IEC is similar but not identical to NEC. Although there are exceptions, the following comparisons are generally applied:

<table>
<thead>
<tr>
<th>IEC Code</th>
<th>NEC Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>Group IIA</td>
<td>Group D</td>
</tr>
<tr>
<td>Group IIB</td>
<td>Group C</td>
</tr>
<tr>
<td>Group IIC</td>
<td>Group B</td>
</tr>
</tbody>
</table>

The area classification, enclosure and degree of protection for IEC and NEC equivalent are as the following:

<table>
<thead>
<tr>
<th>IEC Code</th>
<th>NEC Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>Group II, Zone 0</td>
<td>Class I, Division 1</td>
</tr>
<tr>
<td>Group II, Zone 1</td>
<td>Class I, Division 1</td>
</tr>
<tr>
<td>Group II, Zone 2</td>
<td>Class I, Division 2</td>
</tr>
</tbody>
</table>
Electrical Codes for Enclosures & Control Devices

NEMA Code

All enclosures of electrical apparatus should be designed and constructed in accordance with the NEC or NEMA (National Electrical Manufacturers Association) whichever is applicable or specified for the area of applications either hazardous or non-hazardous as defined by NEMA or NEC. Various enclosures which are classified by NEMA are outlined as the following:

NEMA-1 General Purpose
A general purpose enclosure is intended primarily to prevent accidental contact with the enclosed apparatus. It is suitable for general purpose applications indoors where it is not exposed to unusual service conditions.

NEMA-2 Drip-Tight
A drip-tight enclosure is intended primarily to prevent accidental contact with the enclosed apparatus and, in addition, is so constructed as to exclude falling moistures or dirt.

NEMA-3 Weatherproof (Weather-Resistant)
A weatherproof enclosure is intended to provide suitable protection against specified weather hazards. It is suitable for use outdoors.

NEMA-3R Rain-Tight
A rain-tight enclosure is intended primarily to meet the requirements for rain-tight apparatus. It is suitable for general applications outdoors where sleet-proof construction is not required.

NEMA-4 Water-Tight
A watertight enclosure is designed to exclude water applied in the form of hose stream. It is suitable for application where the apparatus may be subjected to a stream of water during cleaning operations.

NEMA-5 Dust-Tight
A dust-tight enclosure is so constructed as to exclude dust.

NEMA-6 Submersible
A submersible enclosure is intended to permit the enclosed apparatus to operate successfully when submerged in water under specified conditions of pressure and time.

NEMA-7 (A, B, C or D) Hazardous Location – Class I – Air Break
The enclosures are designed to meet the application requirements of Group A, B, C or D of the NEC code for Class I hazardous location which may be in effect from time to time that the circuit interruption occurs in air.

NEMA-8 (A, B, C, or D) Hazardous Location – Class I – Oil Immersed
The enclosures are designed to meet the application requirements of the NEC code for Class I hazardous locations which may be in effect from time to time that the apparatus is immersed in coil.

**NEMA-9**  
(E, F or G) Hazardous Location – Class II  
These enclosures are designed to meet the application requirements of the NEC code for Class II hazardous locations.

**NEMA-10**  
Bureau of Mines – Explosion Proof  
This enclosure is designed to meet the explosion proof requirements of the U.S. Bureau of Mines. It is suitable for use in gassy coal mines.

**NEMA-11**  
Acid or Fume-Resistant – Oil Immersed  
This enclosure provides for the immersion of the apparatus in oil such that it is suitable for application where the equipment is subject to acid or other corrosive fumes.

**NEMA-12**  
Industrial Use  
An industrial use enclosure is designed for use in those industries where it is desired to exclude such materials as dust, line, fibers and flyings, oil seepage or coolant seepage.

**NEMA-13**  
Dust-Proof  
A dust-proof enclosure is intended primarily to prevent accidental contact with the enclosed apparatus, and in addition, is so constructed that dust which may enter will not interfere with the operation of the apparatus. The construction of the enclosure can be defined only in relation to the apparatus and to the amount and kind of dust present.

**IP (Ingress Protection) Code**

IP code is described in IEC and is generally used in Europe and some Asia Pacific countries. IP code is to define the degrees of protection of enclosures.

The expression of IP code is for example IP 21. The first characteristic numeral defines Protection against Contact and Ingress of Foreign Bodies; the second characteristic numeral defines Protection against Ingress of Liquid. The classifications for the degrees of protection of enclosures are as shown in Table 22.5.

<table>
<thead>
<tr>
<th>First Characteristic Numeral</th>
<th>Second Characteristic Numeral</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>IP 00</td>
</tr>
<tr>
<td>1</td>
<td>IP 10</td>
</tr>
<tr>
<td>2</td>
<td>IP 20</td>
</tr>
<tr>
<td>3</td>
<td>IP 30</td>
</tr>
<tr>
<td>4</td>
<td>IP 40</td>
</tr>
<tr>
<td>5</td>
<td>IP 50</td>
</tr>
<tr>
<td>6</td>
<td>IP 60</td>
</tr>
</tbody>
</table>
The brief definition for the First Digit Numeral (Protection against Contact and Ingress of Foreign Bodies) is as the following:

( 0 ) No protection of equipment against ingress of solid foreign bodies.
( 1 ) Protection against ingress of large solid foreign bodies.
( 2 ) Protection against ingress of medium size solid foreign bodies.
( 3 ) Protection against ingress of small solid foreign bodies greater in thickness than 2.5 mm.
( 4 ) Protection against ingress of small solid foreign bodies greater in thickness than 1 mm.
( 5 ) Protection against the ingress of dust in an amount sufficient to interfere with satisfactory operation of the equipment enclosed.
( 6 ) Complete protection against ingress of dust.

The brief definition for the Second Digit Numeral (Protection against Ingress of Liquid) is as the following:

[ 0 ] No protection of equipment against ingress of solid foreign bodies.
[ 1 ] Protection against drops of condensed water.
[ 2 ] Protection against drops of liquid falling at any angle up to 15° from the vertical.
[ 3 ] Protection against drops of rain falling at any angle up to 60° from the vertical.
[ 4 ] Protection against splashing. Liquid splashed from any direction shall have no harmful effect.
[ 5 ] Protection against water projected by a nozzle from any direction.
[ 6 ] Protection against conditions of ship’s decks.
[ 8 ] Protection against indefinite immersion in water.

Both NEMA and IP codes are also applicable for the design and construction of enclosures for electric motors and starters, see Chapter 23 for details.
Purged and Pressurized Enclosures

NFPA-496 is a standard for purged and pressurized enclosures for electrical equipment if the layout of the electrical installation of the plant is so ingeniously arranged. NFPA-496 is not for the purpose of replace NEC NFPA-70, it is to provide information for the design of purged enclosure for the purpose of eliminating or reducing within the enclosure a Class I hazardous location classification as defined in Article 500 of National Electrical Code. By this means, equipment which is not otherwise acceptable for hazardous location may be utilized in accordance with the NEC. The enclosure with type “Z” purge meets Division II classification and the type “X” purging is to reduce the classification of Division I to non-hazardous.

Intrinsically Safe Devices

Intrinsically safe devices are allowed to be used for hazardous location for which it is approved under NEC 500-1. By definition, intrinsically safe equipment or device and its wiring are incapable, under normal or reasonably abnormal operating conditions, of igniting a specified hazardous material, or any in a group of hazardous materials. Normal operating conditions assume maximum supply voltage and environmental factors within the rated extremes given for the equipment. Abnormal operating conditions assume any two independent electrical faults occurring in combination. In other words, intrinsically safe equipment is incapable of releasing sufficient electrical or thermal energy under normal or abnormal conditions to cause ignition of a specific hazardous atmospheric mixture in its most ignited concentration. Intrinsically safe electrical equipment and wiring may be installed in any hazardous location of any Group classification for which it is accepted without requiring NEMA-7 enclosure.
Chapter – 23  Motor and Starter

Electric Motors

All the motors are to be designed and constructed in accordance with NEMA code. Electric motors being used to-day are mostly squirrel cage induction motor. Wound rotor, Synchronous or D-C motors are rarely involved with industrial refrigeration application. In accordance with the NEMA classification, there are two types of motor construction, one is “Open” design and the other is the Totally-Enclosed machine. An open motor is having ventilating openings which permit a passage of external cooling air over and around the windings of the motor. A totally-enclosed motor is the one that is so enclosed as to prevent the free exchange of air between the inside and the outside of the case but not necessarily as air-tight.

Open Motors

**Drip-Proof Motor** – A drip-proof machine is an open motor in which the ventilating openings are so constructed that drops of liquid or solid particles falling on the machine at any angle not greater than 15° from the vertical cannot enter the machine either directly or by striking and running along a horizontal or inwardly inclined surface of the machine.

**Splash-Proof Motor** – A splash-proof machine is an open machine in which the ventilating openings are so constructed that drops of liquid or solid particles falling on the machine or coming towards it in a straight line at any angle not greater than 100° from the vertical cannot enter the machine either directly or be striking and running along a surface of the machine.

**Semi-Guarded Motor** – A semi-guarded machine is an open machine in which part of the ventilating openings in the machine, usually in the top half, are guarded as in the case of a “guarded motor” but the others are left open.

**Guarded Motor** – A guarded machine is an open machine in which all openings giving direct access to live or rotating parts (except smooth shafts) are limited in size by the design of the structural parts or by screens, grills, expanded metal, etc., to prevent accidental contact with such parts. Such openings shall not permit the passage of a cylindrical rod 1/4” in diameter, except that, where the distance from the guard to the live or rotating parts is more than 4”, they shall not permit the passage of a cylindrical rod ¾” in diameter.

**Drip-Proof Fully Guarded Motor** – A drip-proof fully guarded machine is a drip-proof machine whose ventilating openings are guarded.

**Open Externally-Ventilated Motor** – A open externally-ventilated machine is one which is ventilated by means of a separate motor-driven blower mounted on the machine enclosure.
**Open Pipe-Ventilated Motor** – An open pipe-ventilated machine is an open machine except that openings for the admission of the ventilating air are so arranged that inlet ducts or pipes can be connected to them. This air may be circulated by means integral with the machine or by means external to and not a part of the machine. In the latter case, this machine is sometimes known as separately or forced-ventilated machine.

**Weather-Protected Type-I Motor (WP-I)** – A weather-protected Type I machine is an open machine with its ventilating passages so constructed as to minimize the entrance of rain, snow and air-borne particles to the electric parts and having its ventilated openings so constructed as to prevent the passage of a cylindrical rod \( \frac{3}{4} \)" in diameter.

**Weather-Protected Type-II Motor (WP-II)** – A weather-protected Type II machine shall have, in addition to the enclosure defined for a weather-protected Type I machine, its ventilating passages at both intake and discharge so arranged that high-velocity air and air-borne particles blown into the machine by storms or high winds can be discharged without entering the internal ventilating passages leading directly to the electric parts of the machine itself. The normal path of the ventilating air which enters the electric parts of the machine shall be so arranged by baffling or separate housings as to provide at least three abrupt changes in direction, none of which shall be less than 90°. In addition, an area of low velocity not exceeding 600 feet per minute shall be provided in the intake air path to minimize the possibility of moisture or dirt being carried into the electric parts of the machine.

**Totally-Enclosed Motors**

**Totally Enclosed Non-ventilated Motor** – A totally-enclosed non-ventilated machine is a totally-enclosed machine which is not equipped for cooling by means external to the enclosing parts.

**Totally-Enclosed Fan-Cooled Motor** – A totally-enclosed fan-cooled machine is a totally-enclosed machine equipped for exterior cooling by means of a fan or fans integral with the machine but external to the enclosing parts.

**Explosion-Proof Motor** – An explosion-Proof machine is a totally-enclosed machine whose enclosure is designed and constructed to withstand an explosion of a specified gas or vapor which may occur within it and to prevent the ignition of the specified gas or vapor surrounding the machine by sparks, flashes or explosions of the specified gas or vapor which may occur within the machine casing.

**Dust-Ignition-Proof Motor** – A dust-ignition-proof machine is a totally-enclosed machine whose enclosure is designed and constructed in a manner which will exclude ignitable amounts of dust or amount which might affect performance or rating, and which will not permit arcs, sparks, or heat otherwise generated or liberated inside of the enclosure to cause ignition of exterior accumulations or atmospheric suspensions of a specific dust on or in the vicinity of the enclosure.

**Water-Proof Motor** – A water-proof machine is a totally-enclosed machine so constructed that it will exclude water applied in the form of a stream from a hose,
except that leakage may occur around the shaft provided it is prevented from entering the oil reservoir and provision is made for automatically draining the machine. The means for automatic draining may be a check valve or a tapped hole at the lowest part of the frame which will serve for application of a drain pipe.

**Totally-Enclosed Pipe-Ventilated Motor** – A totally-enclosed pipe-ventilated machine is a totally-enclosed machine except for openings so arranged that inlet and outlet ducts or pipes may be connected to them for the admission and discharge of the ventilating air. This air may be circulated by means integral with the machine or by means external to and not a part of the machine. In the latter case, these machines shall be known as separately or forced-ventilated machines.

**Totally-Enclosed Water-Cooled Motor** – A totally-enclosed water-cooled machine is a totally-enclosed machine which is cooled by circulating water, the waater or water conductors coming in direct contact with the machine parts.

**Totally-Enclosed Water-Air-Cooled Motor** – A totally-enclosed water-air-cooled machine is a totally-enclosed machine which is cooled by circulating air which, in turn, is cooled by circulating water. It is provided with a water-cooled heat exchanger for cooling the ventilating air and a fan or fans, integral with the rotor shaft or separate, for circulating the ventilating air.

**Totally-Enclosed Air-To-Air-Cooled Motor** – A totally-enclosed air-to-air-cooled machine is a totally-enclosed machine which is cooled by circulating the internal air through a heat exchanger which, in turn, is cooled by circulating external air. It is provided with an air-to-air heat exchanger for cooling the ventilating air and a fan or fans, integral with the rotor shaft or separate, for circulating the internal air and a separate fan for circulating the external air.

**Totally-Enclosed Fan-Cooled Guarded Motor** – A totally-enclosed, fan cooled guarded machine is a totally-enclosed, fan-cooled machine in which all openings giving direct access to the fan are limited in size by the design of the structural parts or by screens grills, expanded metal, etc., to prevent accidental contact with the fan. Such openings shall not permit the passage of a cylindrical rod ½” in diameter except that, where the distance from the guard to the fan is more than 4”, they shall not permit the passage of a cylindrical rod ¾” in diameter.

**Most Commonly used Motors for Industrial Refrigeration**

With all the motors listed above, only few are commonly used for industrial refrigeration. The most frequently used motors are:

Open Motors:
- Open Drip-Proof (ODP) motor.
- Weather-Protected type I (WP-I) motor.
- Weather-Protected type II (WP-II) motor.

Totally Enclosed Motors:
- Totally-Enclosed Fan-Cooled (TEFC) motor.
Totally-Enclosed Force Ventilated (TEFV) motor.
Totally-Enclosed Water-Air-Cooled (TEWAC) motor.
Totally-Enclosed Air-To-Air Cooled (TEAAC) motor.

Explosion Proof (Division I) Motors:
Totally-Enclosed Fan Cooled motor.

All other types of motor are occasionally used for special application to meet the specific job requirement.

Motor General Selection Guide for Non-Hazardous Location:

The following Table 23.1 shall serve as a general guide in selecting the motor to meet the non-hazardous service conditions at various locations as encountered.

<table>
<thead>
<tr>
<th>Installation Environment</th>
<th>Motor Type</th>
<th>Protective Option</th>
</tr>
</thead>
<tbody>
<tr>
<td>Indoor Installation</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Normal service, dry and clean</td>
<td>ODP</td>
<td>Bearing &amp; winding RTD</td>
</tr>
<tr>
<td>Dry, dusty</td>
<td>ODP</td>
<td>Space heater</td>
</tr>
<tr>
<td>Humid, some moisture</td>
<td>ODP</td>
<td>Space heater and bearing RTD</td>
</tr>
<tr>
<td>High moisture, contaminants</td>
<td>ODP or WP-1</td>
<td></td>
</tr>
<tr>
<td></td>
<td>or TEFC</td>
<td></td>
</tr>
</tbody>
</table>

| Outdoor Installation             |              |                                          |
| Normal service, dry and clean    | ODP          | Space heater                             |
| Occasional moisture and light dust| WP-1 or TEFC| Space heater                            |
| Moisture, wind, rain, dust       | WP-1 or WP-2 Or TEFC | Space heater, lightning arrestors, surge capacitors |
| Excessive dust, corrosives and contaminants | WP-II or TEFC | Space heater, bearing & winding RTD |

Motor General Selection Guide for Hazardous Location:

The following is the general guide in selecting the motor for hazardous environment application:

(1) Class I, Division 2 refers to locations where the atmosphere may become hazardous only under abnormal or unusual conditions. In general, motors in standard enclosures can be installed in Division II location if the motor has no normally sparking parts. Thus, open or standard totally enclosed squirrel-cage
motors are acceptable for NEC Class I, Division II application. However, beside the hazardous consideration, the motor enclosure selection shall also meet the NEMA standard for moisture, dust, and rain exposure particularly for outdoor installation.

(2) **Class I, Division I** motors must be totally-enclosed double shell construction with special seals, fits, breathers, drains and conduit boxes plus three windings mounted with thermostats to detect over temperature in accordance with NEC, NFPA 70 code. Therefore, the motors which might be qualified for Division I atmospheric environment, it must be a totally enclosed construction such as TEFC, TEAAC, TECWAC or Totally Enclosed Inert Gas Filled Water Cooled design with modification for explosion application.

**NEMA Compared IP (IEC) for Motor Enclosure**

The following Table 23.2 lists the frequently specified types of enclosure and the equivalent protection in accordance with IEC IP code.

<table>
<thead>
<tr>
<th>NEMA Enclosure</th>
<th>Enclosure per IEC (IP)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Drip-Proof Motor</td>
<td>IP 12</td>
</tr>
<tr>
<td>Splash-Proof Motor</td>
<td>IP 13</td>
</tr>
<tr>
<td>Semi-guarded Motor</td>
<td>No equivalent Item</td>
</tr>
<tr>
<td>Guarded Motor</td>
<td>IP 22</td>
</tr>
<tr>
<td>Open Pipe-Ventilated Motor</td>
<td>IP 23 or IP 44</td>
</tr>
<tr>
<td>Weather-Protected Motor</td>
<td>IPW 23</td>
</tr>
<tr>
<td>Weather-Protected Motor</td>
<td>IPW 24</td>
</tr>
<tr>
<td>Totally-Enclosed Non-Ventilated Motor</td>
<td>IP 44 or higher without fan</td>
</tr>
<tr>
<td>Totally-Enclosed Fan Cooled Motor</td>
<td>IP 44 or higher fan cooled</td>
</tr>
<tr>
<td>Explosion Proof Motor</td>
<td>Ex d IIA, IIB or IIC</td>
</tr>
<tr>
<td>Dust-Ignition-Proof Motor</td>
<td>Ex e, IP 44 or higher</td>
</tr>
<tr>
<td>Water-Proof Motor</td>
<td>IP 45 or higher</td>
</tr>
</tbody>
</table>

**Motor Options, Modifications and Accessories:**

Special testing and special modifications are available from the manufacturer for custom-built motor at a price addition. The common modifications are as the following:

**Service Factor** – Service factor is one of the options available from the motor manufacturers in USA. When a service factor which usually 1.15 is called for; it means that the motor shall have the capability of 15% continuous overload ability.

**Non-Standard Insulation** – Motor insulation is other than standard. Or use type “F” insulation rated for “B” temperature rise.
**Special Voltage** – Motors are rated at plus and minus 10% of the nominal name plate voltage with rated frequency applied.

**Special Frequency** – Standard frequency in USA is 60 Hz. Other frequencies are non-standard including 50 Hz and 25 Hz.

**Efficiency** – Guaranteed of efficiency or efficiency higher than normal can be obtained by price addition, the motor frame size may be increased by several size larger than normal.

**Low Starting current** – Low starting current motor might have lower breakdown and starting torque.

**Enclosure** – All enclosures other than open-drip proof (ODP).

**Space Heater** – Space heater is to prevent undue collection of condensation when the motor is not in operation.

**Lightning Arrestor or Surge Protection** – For motor having voltage of 2300V or higher. Surge capacitor must be used when lightning arrester is specified.

** Forced Lubrication Bearings** – Forced feed lubrication is required for certain type of motor especially large HP motor. Lubrication oil can be supplied from the external gear if an external gear is employed and extra amount lubrication system consists of oil pump, oil reservoir, oil cooler, oil filter, piping and control should be furnished for the motor.

**Hazardous Application** – Motor to be modified to meet NEC or IEC codes.

**Dual Voltage or Dual Frequency** – Can be designed for either dual voltages or dual frequencies of both 60 Hz and 50 Hz.

**Other Common Modifications:**
- Vibration probes and monitory system.
- Winding temperature detecting equipment.
- Bearing temperature sensing and indicating equipment.
- Low voltage starting.
- Power factor correction.
- Corrosion resistance hardware.
- Altitude above 3300 feet.
- Special bearing.
- Special starting torque.

**Motor Specifications:**

The minimum information required for motor selection are as the following:

- HP size for the motor.
- Voltage, phase and frequency of the power supply.
- Control voltage.
Starters

All the starters are to be designed and constructed in accordance with NEMA standard. The starter used for refrigeration systems are mainly electro-mechanical starter. Solid state starter is a reduced voltage starter with solid state control. The solid state starter operates quietly and smoothly, eliminates the light flicker which is the biggest disadvantage of an electro-mechanical starter. The following Table 23.3 lists all types of the electro-mechanical starters, types of switching transition, approximate percent of locked rotor ampere (LRA) for each step, and approximate percent of locked rotor torque (LRT) for each step.

Table 23.3 Starter Type - LRA & LRT

<table>
<thead>
<tr>
<th>Starter Type</th>
<th>Switching Transition</th>
<th>% LRA</th>
<th>% LRT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Across the Line</td>
<td></td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Part Winding, 2 Steps</td>
<td>Closed</td>
<td>65%</td>
<td>48%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Part Winding, 3 Steps</td>
<td>Closed</td>
<td>45%</td>
<td>24%</td>
</tr>
<tr>
<td>with Resistance</td>
<td></td>
<td>65%</td>
<td>48%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Part Winding, 4 Steps</td>
<td>Closed</td>
<td>51%</td>
<td>22%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>74%</td>
<td>60%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>87%</td>
<td>81%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Star Delta Open</td>
<td>Open</td>
<td>33%</td>
<td>33%</td>
</tr>
<tr>
<td>Transition</td>
<td></td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Star Delta Closed</td>
<td>Closed</td>
<td>33%</td>
<td>33%</td>
</tr>
<tr>
<td>Transition</td>
<td></td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Auto Transformer Open</td>
<td>Open</td>
<td>50% Tap</td>
<td>25%</td>
</tr>
<tr>
<td>Transition</td>
<td></td>
<td>27%</td>
<td>42%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>65% Tap</td>
<td>42%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>45%</td>
<td>64%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>80% Tap</td>
<td>64%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>66%</td>
<td>100%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Auto Transformer Closed</td>
<td>Closed</td>
<td>50% Tap</td>
<td>25%</td>
</tr>
<tr>
<td>Transition</td>
<td></td>
<td>27%</td>
<td>42%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>65% Tap</td>
<td>42%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>45%</td>
<td>64%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>80% Tap</td>
<td>64%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>66%</td>
<td>100%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Primary Resistance 2</td>
<td>Closed</td>
<td>65%</td>
<td>43%</td>
</tr>
<tr>
<td>Point</td>
<td></td>
<td>100%</td>
<td>100%</td>
</tr>
</tbody>
</table>
Part Winding (P-W) starters are mainly used for smaller HP motor. Other reduce voltage starters such as star-delta are mainly used for voltage below 600V. Auto-transformer starter is more expensive if used for 600V range as compared to star-delta. Medium and high voltage such as 4160 V, 6000 V or 10000 V mostly use across-the-line (direct-on-line), auto-transformer or primary Resistance type starter.

**Starter Specifications**

The minimum information for starter selection are as the following:

- Size of starter, FLA & LRA.
- Voltage, phase and frequency of the power supply.
- Type of starter.
- Type of enclosure.
- Motor data: HP; Maximum allowable stall time; Acceleration time, S.F.
- Modifications and Accessories.
- Codes compliance, NEMA or IP.

**Starting Torque Requirement and Starting Acceleration**

Starting torque is a very important consideration for selecting the driver for the compressor in industrial refrigeration application. Figure 23-1 is the typical screw compressor torque requirement and the typical motor starting torque capabilities. A motor with across-the-line (ACL) starter or auto-transformer starter with 80% tap might be able to start the screw compressor at 185 Psi pressure differential; however, the motor might have difficulty to start the same compressor if the starter is with 65% tap and is not enough starting torque to start a low stage compressor at 30 Psi pressure differential; a motor with Star-Delta starter might not provide enough torque to start the compressors under normal conditions for refrigeration application.

All drivers including motors should be checked for starting torque and starting acceleration time to see if the driver is with enough torque to start the compressor at the specified conditions within the acceleration time allowed for the motor. Figure 23-2 shows the pull-up torque of the motor (B) is below the compressor torque curve and therefore, the motor is not providing enough torque to pull the compressor up to the design speed.

Figure 23-3 is the typical speed torque curves for a single stage centrifugal compressor. The starting torque requirement of the motor depends on the starting gas density ratio for the compressor. Figure 23-4 is the typical speed torque curves for a multistage centrifugal compressor with various density ratios.
Figure 23-1  Typical Screw Compressor Starting Torque Requirement & Motor Starting Torque
Motor "A"
Adequate starting torque to start the compressor

Motor "B"
Not enough starting torque to start the compressor

Figure 23-2 Typical Starting Torque Requirement for Motor

Figure 23-3 Typical Starting Torque for Single Stage Centrifugal Compressor
Figure 23-4  Typical Starting Torque for Multistage Centrifugal Compressor
Special Driver

Drivers other than motors available for screw or centrifugal compressor drive are gasoline engine, gas engine, diesel engine, steam turbine and gas turbine as special application. Motor is the most commonly use driver for compressor because it is simple and least expensive as compare to other type of drivers.

All compressors are basically designed for motor drive as standard. The base of the drive-line for the compressor is changed if it is changed to special driver. Cost of engine or turbine is more expensive than motor; and the cost penalty for the special system design is very heavy; therefore, the case should be evaluated carefully to see if overall benefit is justified including the return on investment. Furthermore, the drive train should be checked for torsional vibration, particularly if external gear is used between the driver and the compressor.
Chapter – 24  Compressor and System Controls

The modern compressor unit is equipped with a control panel as a standard. This control panel is a microprocessor type panel. It might actually a computer with display and keyboard. This type of control panel can provide all the major control and monitoring functions for the compressor or refrigeration system. The major control functions are listed as the following:

(1) To provide automatic control for continuous safe operation of the compressor or the refrigeration system.

(2) To maintain suction pressure or leaving process fluid temperature automatically at a predetermined set point.

(3) To control and adjust automatically the capacity controller such as sliding vane, or internal volume ratio in case of screw compressor, or inlet guide vane if centrifugal compressor, to meet the process load demand at any given time and any partial load operating conditions that are within the design scope.

(4) The control panel shall be able to do self-diagnoses and self-check constantly and continuously against the pre-set safety operation set points of the compressor unit. Also display the status of all the operating conditions at any given time.

(5) The control panel should be able to perform pre-alarm and display on a first out bases. The unit will be still in operation after the pre-alarm until the cutout preset point is reached.

(6) The display monitor of the control panel should be able to display the information of cutout and the safety functions of the compressor or the refrigeration unit. Shut-down records and freezes operating conditions at each time of shut-down are to be on first out bases.

(7) The control panel shall be pre-programmable and for new data entering; it shall be able to be programmed for automatic start-up and shutdown on time schedules. The control panel shall be able to communicate with any major building or facility automation systems.

(8) The control panel shall have other features such as:

- Security identification.
- Real-time clock to report time, day, date and year.
- Motor overload protection.
- Emergency stop.
- Battery back-up.

The control panel should have display for the individual safety function, pre-alarm and shutdown as shown in Table 24.1 as the following:
Table 24.1 Safety display, pre-alarm and shut-down

<table>
<thead>
<tr>
<th></th>
<th>Display</th>
<th>Pre-Alarm</th>
<th>Shut-Down</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low suction pressure</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>High discharge pressure</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>High discharge temperature</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>Low oil pressure</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>High oil pressure</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>High oil temperature</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>Low oil temperature</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>High oil filter differential</td>
<td>x</td>
<td>x</td>
<td></td>
</tr>
<tr>
<td>Oil heater</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>High motor current</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>Freezing temperature</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>Annunciators</td>
<td>x</td>
<td>x</td>
<td>x</td>
</tr>
</tbody>
</table>

Control panel should be able to include other control functions such as hot gas bypass and liquid injection; oil still and oil return control and even automatic pump-down if so desired.
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:
\[
\text{GPM} \times \text{S.G.} \times \text{C}_p \times (T_2 - T_1)
\]
\[
\frac{24}{24}
\]

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

\[
\frac{300}{24} \times 1.050 \times 0.8712 \times (37 - 30)
\]
\[
= 80 \text{ TR}
\]

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 subcooling 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 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 Psia) |

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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
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:

\[
\text{Flow (1)} = 200 \times \frac{\text{TR} (H_F - H_A)}{597.6 - 75.7}
\]

\[
= 200 \times \frac{260}{597.6 - 75.7}
\]

\[
= 99.64 \text{ Lbs/Min}
\]

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

\[
\text{Flow (2)} = 200 \times \frac{\text{TR} (H_K - H_A)}{599.5 - 75.7}
\]

\[
= 200 \times \frac{30}{599.5 - 75.7}
\]

\[
= 11.45 \text{ Lbs/Min}
\]

Total refrigerant flow for the booster compressor:

\[
= \text{Flow (1)} + \text{Flow (2)}
\]

\[
= 99.64 + 11.45
\]

\[
= 111.09 \text{ Lbs/Min}
\]
Figure 25-7  Enthalpy Points for No. 1 & No. 2 Loads

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

\[
\frac{200}{617.8 - 161.1} \times 80 = 35.03 \text{ Lbs/Min}
\]
3. Refrigerant flow for the liquid subcooling:
Liquid to be subcooled = 111.09 Lbs/Min.

Heat load required for liquid subcooling:

\[
\frac{111.09 \times (161.1 - 75.7)}{200} \]

= 47.4 TR

Refrigerant required for liquid subcooling in the intercooler:

\[
\text{Refrigerant flow} = \frac{200}{616.3 - 161.1} \times 47.4
\]

= 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 \times 2545

= 925,362 Btu/Hr

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

Net heat input to the high stage compressor:

\[
= \text{Motor heat input} - \text{Oil cooling heat removal}
\]

= 925,362 – 295,400

= 629,962 Btu/Hr.

or 52.5 TR

Refrigerant flow required for the net heat input:

\[
\text{Refrigerant flow} = \frac{200}{616.3 - 161.1} \times 52.5
\]

= 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.

\[
\text{GPM} = \frac{\text{Btu/Hr}}{499.8 \times (T_2 - T_1)}
\]

\[
\text{Btu/Hr} = 5,940,736 \text{ Btu/Hr} \\
T_2 - T_1 = 10°F
\]

Therefore, the cooling water requirement is:
5,940,736
\[ \text{GPM} = \frac{5,940,736}{499.8 \times 10} \]
\[ = 1,189 \text{ GPM} \]

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.
  \[ = 48.2 - 43.1 \]
  \[ = 5.1 \text{ Psi} \]
- 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 \(V_g = 18.97\) Ft\(^3\)/Lb  
CFM flow = 111.09 x 18.97 = 2,107.4 CFM

Cross Section Area of the Suction Trap:

\[
A = \frac{1}{4} \pi D^2
\]

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

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

\[
\text{FPM} = \frac{\text{CFM}}{A}
\]

\[
\text{FPM} = \frac{\text{CFM}}{\frac{1}{4} \pi D^2}
\]

\(\text{CFM} = \) Refrigerant Vapor Flow = 2,107.4

\[
D \text{ (Inches)} = \sqrt{\frac{576 \times \text{CFM}}{\pi \times \text{FPM}}}
\]
FPM = 143 Ft/Min.

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

= 51.98 Inches
Say: 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.
Chapter – 26  Case of Structuring Refrigeration System with Multistage Centrifugal Compressor

This case of illustration is to show how to connect the refrigeration side and how to place the intercooler-economizer for the refrigeration system which is with a multistage centrifugal compressor. The refrigeration system is with three users (three evaporative loads), the refrigerant is propylene (R-1270). The refrigeration system is for a VCM plant in a petrochemical complex.

The design conditions are:

| Load (User) #1 :  | 824 TR, ET at -41.8°F. |
| Load (User) #2:  | 172 TR, ET at -36°F.   |
| Load (User) #3:  | 51.2 TR, ET at 35.1°F. |

The design condensing temperature is 108°F. Water cooled condenser is used.

External suction piping pressure drop and entrance loss are 2.91 Psi.
Suction line superheat is 19.8°F.

The external discharge piping pressure drop and discharge nozzle loss are 1.44 Psi.

The total compression head for the compressor of the refrigeration system is to base on the lowest ET of -41.8°F and the condensing temperature of 108°F plus suction and discharge penalties.

For R-1270 Propylene refrigerant:

- Evaporative Pressure at -41.8°F = 19.71 Psia
- Condensing Pressure at 108°F = 251.56 Psia

Compressor Suction and Discharge Operating Conditions are:

- Suction Pressure: 19.71 - 2.91 = 16.8 Psia
- Suction temperature: -41.8 + 19.8 = -22°F
- Discharge pressure: 251.56 + 1.44 = 253.0 Psia

Guesstimate number stages required for the centrifugal compressor:

The first step is to determine how many stages are needed for the multistage centrifugal compressor for this application base on the design operating conditions listed. From Figure 26-1 shows the compression line on the partial P-H diagram, the adiabatic
Figure 26-1  Interstage Pressure
5-Stage Multistage Centrifugal
compression line is the line (a)-(f). The design suction conditions are the points (a) and the design discharge point is (f) as shown.

All the thermodynamic properties and data can be obtained from computer refrigerant property program or from the refrigerant property table for the refrigerant R-1270 (propylene) through the following process:

Compressor suction conditions:

\[
\begin{align*}
P &= 16.8 \text{ Psia,} \\
t &= -22^\circ \text{F} \\
H_1 &= 102.75 \\
V_g &= 6.411 \text{ Ft}^3/\text{Lb} \\
V_a &= 761.39 \text{ Ft/Sec.} \\
S &= 1.4357
\end{align*}
\]

The enthalpy point for the compressor discharge:

\[
H_f = 165.85 \text{ at discharge pressure of 253.0 Psia and entropy of 1.4357}
\]

Therefore, the adiabatic Compression Head \( = H_f - H_a = 165.85 - 102.75 \)

\[
= 63.10 \text{ Btu/Lb}
\]

or \( = 63.10 \times 778 = 49,092 \text{ Ft.} \)

Use the formula listed in Chapter 7 to guesstimate the number of stages needed for the multistage centrifugal compressor for the application:

\[
N \approx \frac{48 \times H_{ad}}{[V_a]^2}
\]

\[
N = \text{Number of Stage (Impeller)} \\
H_{ad} = \text{Overall Adiabatic Head, ft.} \\
V_a = \text{Suction Acoustic Velocity, Ft./Sec.}
\]

\[
N \approx \frac{48 \times 49,092}{[761.39]^2} = 4.065
\]

Therefore, a 5-stage compressor is needed for the application.

The theory of multistage centrifugal compression stimulates that total compression head is shared equally by the impellers if the compressor impellers are having same diameter.
This is the Hypothesis of equal head for wheel analysis for multistage centrifugal machine.

Therefore, the enthalpy difference for each impeller is as the following:

\[
\frac{63.10}{5 \text{ Stages}} = 12.62 \text{ Btu/Lb per wheel}
\]

Therefore, the enthalpy for the point (b) is \(102.75 + 12.62 = 115.37 \text{ Btu/Lb.}\)

The interstage pressure is 30.82 Psia at enthalpy of 115.37 and the entropy of 1.4357.

Following the same procedure, enthalpy and pressure for other points of the interstage of each impeller can be obtained and are shown in Figure 26-1.

The evaporative load #2 is to be returned to compressor main suction, because the pressure of load #2 is very close to the main suction pressure.

The evaporative pressure of Load #3 at ET of 51.2°F is 89.23 Psia; this pressure is just below the pressure of the 4th stage suction. Therefore, the suction gas of Load #3 has to be returned to the 3rd stage inlet at 54.28 Psia. The throttling from 89.23 Psia to 54.28 Psia might be too wasteful from the stand point of energy consumption. This problem can be corrected by the method of wheel trimming, however, wheel trimming manufacturing process is very expensive. Another option is to change the compressor to 6-stage instead of 5-stage. The cost addition for having a 6-stage machine is reasonable.

**Determination of interstage pressures for the 6-stage multistage compressor:**

Compression Head = \(H_2 - H_1 = 165.85 - 102.75\)

\[= 63.10 \text{ Btu/Lb}\]

or \[= 63.10 \times 778 = 49,092 \text{ Ft.}\]

Each impeller is to handle = \(\frac{63.10}{6 \text{ Stages}} = 10.517 \text{ Btu/Lb enthalpy per wheel}\)

Or \[
\frac{49,092}{6\text{-stage}} = 8,182 \text{ ft head per wheel}
\]

Base on equal head hypothesis, the enthalpy and pressure can be obtained for the interstage between in impellers. These values are shown in Figure 26-2.
Side load connection and intercooling location:

The ET for the user refrigeration load #2 is \(-36^\circ\)F; the evaporative pressure is 22.55 Psia which very close to the main suction. Therefore, the gas is to be returned to the compressor suction as shown in the partial P-H diagram of Figure 26-3.

The ET for refrigeration load #3 is \(35.1^\circ\)F; the evaporative pressure is 89.23 Psia. The pressure is high enough to be returned to the 4\textsuperscript{th} wheel inlet as shown in Figure 26-3. The suction pressure of the 4\textsuperscript{th} wheel is 71.04 Psia.

The throttling pressure differential between the evaporative temperature of load #3 and the interstage pressure is 18.19 Psi which is half the throttling pressure difference as compared to the 5-stage machine. In view of this, it is decided that 6-stage compressor is to be used instead of 5-stage. Furthermore, the Mach number of the compressor is lower, this helps the partial load performance of the compressor.

Two intercoolers (economizers) are used in this case. One is to be located at the 2\textsuperscript{nd} stage inlet and the other is to be located at 3\textsuperscript{rd} stage inlet. These are indicated in the P-H diagram as shown in the Figure 26-4. The evaporative temperatures of the intercooling are based on the rule of thumb that a \(\Delta P\) of 5 Psi difference between the evaporative pressure of the intercooler and the compressor interstage pressure; this pressure difference should be enough for the control valve, the piping loss and the nozzle pressure drop. Therefore, the ET for the 1\textsuperscript{st} intercooler is set for 52°F and the 2\textsuperscript{nd} state is to be 3°F.

Shell-and-tube liquid subcooling type economizer is used, because the evaporators are remote mount and are far away from the compressor. The basic rule for the intercooler heat exchanger design is to set about 10°F approach between the leaving subcooled liquid temperature and the ET of the intercooler. Therefore, the liquid temperature leaving first stage intercooler is to be 62°F and the liquid leaving the 2\textsuperscript{nd} stage intercooler is to be 13°F respectively.

System P-H Diagram and Refrigerant Flow Diagram:

Figure 26-5 is the P-H diagram analysis for this refrigeration system with two liquid subcoolers (economizers). The line (a)-(f) is the adiabatic compression line; the zigzag line of (a)-(g) is the actual polytropic compression line with efficiency included.
Compressor Discharge, 253 Psia  
(165.85)  
ΔH = 10.517 Btu/Lb  
or 8,182 ft

(155.30)  
ΔH = 10.517  
or 8,182 ft

(144.79)  
ΔH = 10.517  
or 8,182 ft

(134.27)  
ΔH = 10.517  
or 8,182 ft

(123.75)  
ΔH = 10.517  
or 8,182 ft

(113.24)  
ΔH = 10.517  
or 8,182 ft

Condensing Temperature 108°F  
251.56 Psia

167.45 Psia  

109.89 Psia

71.04 Psia  

45.06 Psia

27.90 Psia

19.71 Psia

Load No. 1  
ET = -41.8°F

(102.75)  
Compressor Suction
16.8 Psia, -22°F  
S = 1.4357

Figure 26-2  Interstage Pressure  
6-Stage Multistage Centrifugal
Figure 26-3 Side Load Connections
Figure 26-4  Intercooler-Economizer Locations.
Figure 26-5 System P-H Diagram for 6-Stage Centrifugal Refrigeration System
The refrigerant flow diagram for the system is shown in Figure 26-6. A main suction trap (scrubber) is provided for suction gas returning from the #1 load at ET of -41.8°F. The suction gas flow from load #2 is returned to the main suction trap through a back pressure regulating valve. A separate suction trap is used for the suction gas from the #3 load at ET of 35.1°F; a back pressure regulating valve is used to control the suction pressure for the suction gas from load #3.

The subcooled R-1270 liquid for the user load #3 is from the first stage liquid subcooler; the subcooled liquid for load #1 and load #2 is from the second stage liquid subcooler.
Figure 26-6  System Refrigerant Flow Diagram
6-Stage Centrifugal
The air side evaporators for cold storage and frozen food are unit coolers and product coolers. Bare pipe coil was used in the old days and is rarely being used for modern cold storage installation because of poor efficiency. The construction of unit cooler or product cooler is basically consists of fan assembly, coil assembly, drain pan, casing and defrosting accessories, if any.

The coil is usually rated for either ammonia or halocarbon if coil is constructed with steel or aluminum; however, the coil is only good for halocarbon refrigerant if it is constructed with copper.

Brine capacity rating is available from the cooler manufacturer; information needed for brine selection shall be type of brine, percent by wt. concentration, in and out temperatures, flow rate in GPM, room design temperature and cooling capacity.

**Unit Cooler:**

Unit cooler is generally referred to the units which are with smaller coil; it is used for smaller cold storage room application. Most unit Coolers are Ceiling Mounted design as shown in Figure 27-1. Most ceiling mounted units are designed for rear air inlet, front air discharge. Figure 27-2 shows the typical unit cooler with multiple fans. Figure 27-3 shows the typical capacity ratings and physical data of the unit coolers. Most unit coolers are with propeller fans; units are available for air, water or hot gas defrosting application. Units are mostly with ¾” OD coil; coil material is aluminum, copper or steel. Fins are 3 or 4 F.P.I.

**Product Cooler:**

Product cooler generally refers to the units with larger coil for larger capacity; for large cold storage, blast freezing or wind tunnel application. Product coolers have ceiling or floor mounted models. Figure 27-4 is a typical Ceiling Mounting Unit with horizontal rear air inlet, horizontal front air discharge arrangement; Figure 27-5 is a typical Floor Mounting Unit with front horizontal air inlet, horizontal front air discharge. Other discharge and inlet orientations are available for ceiling or floor mounting product units.

Figure 27-6 is the table for typical capacities and physical data for the product coolers. The units are with ¾" OD, 4 FPI steel coils. Other options for product coolers are such 1" OD coils; aluminum or copper coil material.

Most product coolers are with centrifugal or vane axial fan instead of propeller fans.

**Performance of Unit Cooler and Product Cooler:**

Capacity ratings of unit cooler and product coolers are similar, both are rated on
Figure 27-1  Typical Unit Cooler

Figure 27-2  Typical Multiple Fans Unit Cooler
### HT & Lt Models: ½"OD Aluminum Tubes, Ammonia Ratings

<table>
<thead>
<tr>
<th>MODEL NO.</th>
<th>CAPACITY BTUH /T.D.</th>
<th>WET COIL</th>
<th>FROSTED COIL</th>
<th>FAN &amp; MOTOR DATA</th>
<th>COIL DATA</th>
<th>SURF.- VOLL. INT.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>DX REC./FL.</td>
<td>DX REC./FL.</td>
<td>QTY./HP CFM FPM</td>
<td>FPI</td>
<td>FT.3 FT.3</td>
</tr>
<tr>
<td>HT254</td>
<td>3523</td>
<td>4098</td>
<td>3171</td>
<td>3688</td>
<td>2-1/3</td>
<td>7140 643</td>
</tr>
<tr>
<td>HT264</td>
<td>3962</td>
<td>4607</td>
<td>3565</td>
<td>4146</td>
<td>2-1/3</td>
<td>6920 623</td>
</tr>
<tr>
<td>HT354</td>
<td>5425</td>
<td>6309</td>
<td>4805</td>
<td>5678</td>
<td>3-1/3</td>
<td>10710 612</td>
</tr>
<tr>
<td>HT364</td>
<td>6102</td>
<td>7095</td>
<td>5492</td>
<td>6366</td>
<td>3-1/3</td>
<td>10380 594</td>
</tr>
<tr>
<td>HT454</td>
<td>7095</td>
<td>8250</td>
<td>6386</td>
<td>7425</td>
<td>4-1/3</td>
<td>14280 635</td>
</tr>
<tr>
<td>HT464</td>
<td>7961</td>
<td>9257</td>
<td>7165</td>
<td>8331</td>
<td>4-1/3</td>
<td>13840 615</td>
</tr>
<tr>
<td>HT554</td>
<td>8993</td>
<td>10456</td>
<td>8093</td>
<td>9410</td>
<td>5-1/3</td>
<td>17850 618</td>
</tr>
<tr>
<td>HT564</td>
<td>10134</td>
<td>11784</td>
<td>9121</td>
<td>10606</td>
<td>5-1/3</td>
<td>17300 599</td>
</tr>
<tr>
<td>HT654</td>
<td>10772</td>
<td>12472</td>
<td>9654</td>
<td>11225</td>
<td>6-1/3</td>
<td>21420 607</td>
</tr>
<tr>
<td>HT664</td>
<td>12194</td>
<td>14179</td>
<td>10974</td>
<td>12781</td>
<td>6-1/3</td>
<td>20760 588</td>
</tr>
<tr>
<td>HT754</td>
<td>12773</td>
<td>14853</td>
<td>11612</td>
<td>13503</td>
<td>7-1/3</td>
<td>24990 621</td>
</tr>
<tr>
<td>HT764</td>
<td>14239</td>
<td>16558</td>
<td>12945</td>
<td>15053</td>
<td>7-1/3</td>
<td>24220 602</td>
</tr>
<tr>
<td>HT854</td>
<td>14419</td>
<td>16768</td>
<td>13109</td>
<td>15244</td>
<td>8-1/3</td>
<td>28560 613</td>
</tr>
<tr>
<td>HT864</td>
<td>16266</td>
<td>18915</td>
<td>14788</td>
<td>17196</td>
<td>8-1/3</td>
<td>27680 594</td>
</tr>
</tbody>
</table>

| LT264     | -                   | -         | 3696         | 4298             | 2-1/3     | 7300 676 | 4 828  .93 |
| LT263     | -                   | -         | 3931         | 4572             | 2-1/3     | 9400 848 | 3 641  .93 |
| LT264     | -                   | -         | 4192         | 4876             | 2-1/2     | 9200 830 | 4 828  .93 |
| LT364     | -                   | -         | 5679         | 6603             | 3-1/3     | 10950 626 | 4 1308 1.46 |
| LT363     | -                   | -         | 5993         | 6969             | 3-1/3     | 14100 806 | 3 1013 1.46 |
| LT364     | -                   | -         | 6457         | 7508             | 3-1/2     | 13800 789 | 4 1308 1.38 |
| LT464     | -                   | -         | 7450         | 8663             | 4-1/3     | 14600 649 | 4 1679 1.88 |
| LT463     | -                   | -         | 7861         | 9140             | 4-1/3     | 18800 837 | 3 1300 1.38 |
| LT464     | -                   | -         | 8456         | 9833             | 4-1/3     | 18400 819 | 4 1679 1.88 |
| LT564     | -                   | -         | 9431         | 10967            | 5-1/3     | 18250 631 | 4 2159 2.41 |
| LT563     | -                   | -         | 10052        | 11689            | 5-1/2     | 23500 814 | 3 1672 2.41 |
| LT564     | -                   | -         | 10707        | 12451            | 5-1/2     | 23000 796 | 4 2159 1.88 |
| LT664     | -                   | -         | 11421        | 13280            | 6-1/3     | 21900 620 | 4 2638 2.95 |
| LT663     | -                   | -         | 12052        | 14013            | 6-1/2     | 28200 799 | 3 2043 2.95 |
| LT664     | -                   | -         | 12953        | 15061            | 6-1/2     | 27600 782 | 4 2638 2.95 |
| LT764     | -                   | -         | 13135        | 15274            | 7-1/3     | 25550 635 | 4 3009 3.36 |
| LT763     | -                   | -         | 14011        | 16293            | 7-1/2     | 32900 817 | 3 2330 3.36 |
| LT764     | -                   | -         | 15039        | 17487            | 7-1/2     | 32200 800 | 4 3009 3.36 |
| LT864     | -                   | -         | 15096        | 17554            | 8-1/3     | 29200 627 | 4 3484 3.36 |
| LT863     | -                   | -         | 16112        | 18736            | 8-1/2     | 37600 807 | 3 2698 3.36 |
| LT864     | -                   | -         | 17216        | 20018            | 8-1/2     | 36800 790 | 4 3484 3.36 |

Figure 27-3  Typical Capacity Ratings & Physical Data
For Unit Cooler
Figure 27-4  Typical Horizontal Ceiling Mounting Product Cooler

Figure 27-5  Typical Vertical Floor Mounting Product Cooler
CF Model: ¾" OD Steel Coil, Ammonia Ratings

<table>
<thead>
<tr>
<th>MODEL NUMBER</th>
<th>ROW COIL CFM @ 600 FPM</th>
<th>NO. OF FANS</th>
<th>FAN SIZE INCHES</th>
<th>FAN MOTOR HP</th>
<th>CAPACITIES BTU/HR - °F TD</th>
<th>TOTAL COIL SURFACE (SQ FT)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>DX</td>
<td>3 FINS/INCH</td>
<td>4 FINS/INCH</td>
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<tr>
<td>CF 134</td>
<td>4</td>
<td>7,500</td>
<td>2</td>
<td>12</td>
<td>2</td>
<td>4,773</td>
</tr>
<tr>
<td>CF 136</td>
<td>4</td>
<td>7,500</td>
<td>2</td>
<td>12</td>
<td>2</td>
<td>5,020</td>
</tr>
<tr>
<td>CF 138</td>
<td>4</td>
<td>7,500</td>
<td>2</td>
<td>12</td>
<td>3</td>
<td>4,128</td>
</tr>
<tr>
<td>CF 194</td>
<td>4</td>
<td>11,250</td>
<td>2</td>
<td>17</td>
<td>2</td>
<td>3,711</td>
</tr>
<tr>
<td>CF 196</td>
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<td>11,250</td>
<td>2</td>
<td>17</td>
<td>2</td>
<td>5,128</td>
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<tr>
<td>CF 198</td>
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<td>11,250</td>
<td>2</td>
<td>17</td>
<td>3</td>
<td>6,192</td>
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<tr>
<td>CF 234</td>
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<td>2</td>
<td>18</td>
<td>3</td>
<td>4,451</td>
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<td>3</td>
<td>6,153</td>
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<tr>
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<td>13,600</td>
<td>2</td>
<td>18</td>
<td>5</td>
<td>7,430</td>
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<tr>
<td>CF 306</td>
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<td>2</td>
<td>21</td>
<td>5</td>
<td>8,204</td>
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<tr>
<td>CF 308</td>
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<td>18,000</td>
<td>2</td>
<td>21</td>
<td>5</td>
<td>9,307</td>
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<tr>
<td>CF 3010</td>
<td>4</td>
<td>18,000</td>
<td>2</td>
<td>19</td>
<td>7/1/2</td>
<td>11,739</td>
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<tr>
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<td>3</td>
<td>18</td>
<td>5</td>
<td>10,258</td>
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<tr>
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<td>3</td>
<td>18</td>
<td>7/1/2</td>
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<tr>
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<td>3</td>
<td>21</td>
<td>5</td>
<td>11,989</td>
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<tr>
<td>CF 448</td>
<td>8</td>
<td>26,250</td>
<td>3</td>
<td>21</td>
<td>7/1/2</td>
<td>14,448</td>
</tr>
<tr>
<td>CF 4410</td>
<td>8</td>
<td>26,250</td>
<td>3</td>
<td>19</td>
<td>10</td>
<td>17,120</td>
</tr>
<tr>
<td>CF 536</td>
<td>8</td>
<td>31,500</td>
<td>3</td>
<td>21</td>
<td>7/1/2</td>
<td>14,358</td>
</tr>
<tr>
<td>CF 538</td>
<td>8</td>
<td>31,500</td>
<td>3</td>
<td>21</td>
<td>7/1/2</td>
<td>17,338</td>
</tr>
<tr>
<td>CF 5310</td>
<td>8</td>
<td>31,500</td>
<td>3</td>
<td>21</td>
<td>10</td>
<td>20,544</td>
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</table>

FINNED COIL CAPACITY CORRECTION FACTORS

<table>
<thead>
<tr>
<th>FIN SPACING COIL CONDITION &amp; TYPE</th>
<th>FIN FACTOR</th>
<th>MULTIPLY FACTORS BY:</th>
</tr>
</thead>
<tbody>
<tr>
<td>2/3 FPI</td>
<td>.88</td>
<td>92</td>
</tr>
<tr>
<td>2/4 FPI</td>
<td>.85</td>
<td>91</td>
</tr>
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</table>

LOW SUCTION TEMPERATURE CAPACITY CORRECTION FACTORS

<table>
<thead>
<tr>
<th>T.D.</th>
<th>SUCTION TEMPERATURE °F</th>
</tr>
</thead>
<tbody>
<tr>
<td>-10</td>
<td>1.00</td>
</tr>
<tr>
<td>-20</td>
<td>1.00</td>
</tr>
<tr>
<td>-30</td>
<td>0.95</td>
</tr>
<tr>
<td>-40</td>
<td>0.90</td>
</tr>
<tr>
<td>-50</td>
<td>0.85</td>
</tr>
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</table>

Figure 27-6 Typical Capacity Ratings & Physical Data For Product Cooler
Btu/Hr per F of TD. TD is the temperature difference between the cold room design temperature and the coil design evaporative temperature.

An example for a product cooler selection is shown as the following:

- Cooling capacity: 123,000 Btu/hr.
- Evaporative temperature: 12°F.
- Room temperature: 20°F.
- Refrigerant: Ammonia.
- Refrigerant feed: Liquid recirculation.
- Coil fins spacing: 4 FPI.

Design TD = °F Diff = Room Design Temp. – ET (Evaporative Temperature)

\[
\text{Room Design Temperature} = 20°F \\
\text{ET} = 12°F \\
TD = 20 - 12 = 8°F
\]

\[
\frac{123,000 \text{ Btu/Hr}}{8} = 15,375 \text{ Btu/Hr per } °F
\]

See Figure 27-6, the typical capacity rating for the product cooler, the product cooler model CF-388, 8-row deep coil, 4 FPI for ammonia liquid recirculation is rated at 15,750 Btu/hr./°F TD.

The capacity of model CF-388 is 15,750 x 8 = 126,000 Btu/Hr, which is larger than the capacity specified. Therefore, the product cooler model CF-388 is selected.

From the typical ratings for unit coolers and product coolers shown in Figures 27-2 and 27-6, the capacity for a unit with DX refrigerant feed is lower than flooded or liquid recirculation refrigerant feed, because the heat transfer of a DX coil is less efficient than the coil for flooded or liquid recirculation.

Coil Fins Spacing:

The coil fins spacing for cooler is usually 3 or 4 FPI. A 6 FPI coil only can be used for room temperature above 50 °F with ET not lower than 40 °F. The coil can be constructed for vari-fin for low temperature application. Vari-fin is to use different fin spacing for the coil, that is to use less fins per inch for entering air side of the coil where heavier frost build-up is likely. For example, an 8-row 4 FPI coil to use 2 FPI on air entering two rows and remaining 6 rows to be 4 FPI. Some units are even available with 1.5 FPI vari-fin spacing option. The capacity of the vari-fin coil should
be corrected in accordance with the manufacturer’s recommendation. Other typical capacity corrections for vari-fin are shown in Table 27.1.

Table 27.1 Capacity Correction Factor for Vari-fin

<table>
<thead>
<tr>
<th>FPI of 2 Rows of Entering Air</th>
<th>FPI for Remaining Rows</th>
<th>Capacity Correction Factors</th>
<th>Multiply to Rating</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Coi Rows Deep</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>6-Row</td>
<td>8 Rows</td>
</tr>
<tr>
<td>1.5 FPI</td>
<td>3 FPI</td>
<td>0.88</td>
<td>0.90</td>
</tr>
<tr>
<td>2 FPI</td>
<td>4 FPI</td>
<td>0.88</td>
<td>0.90</td>
</tr>
<tr>
<td>3 FPI</td>
<td>4 FPI</td>
<td>0.92</td>
<td>0.93</td>
</tr>
</tbody>
</table>

Cooler Capacity Correction for Low Temperature Application:

The coil capacity should be also corrected in accordance with manufacturer’s recommendation for low temperature application. It is suggested not to use DX feed for evaporative temperature below 0°F. The capacity of the cooler should be corrected if the CFM flow is changed because of 60 Hz for 50 Hz application, if applicable.

Wind Tunnel and Blast Freezers:

Product cooler might not meet the requirements of wind tunnel or blast freezer or for IQF and Spiral Freezers, because the fan and coil for product cooler are mostly fix designed for general cold storage application, therefore, for wind tunnel or blast freezing application, the fan, CFM flow, external static pressure for the fan and coil capacity are to be specially selected for specific installation.

The Relationship between Room Relative Humidity & Design TD for Cold Storage:

The refrigeration capacity for a unit cooler or product cooler is higher at larger TD. Therefore, the cooler size is smaller if the TD is larger for same refrigeration capacity. On the other hand, smaller TD requires large size cooler. The cost for smaller size cooler is cheaper; but the refrigeration equipment could be more expensive and the power consumption will be higher, because larger TD requires lower ET for the same room design temperature. Therefore, the initial cost of the cooler should be balanced out with the refrigeration equipment and the power consumption if no restriction on TD.

The TD should be carefully considered, if the room relative humidity is important for the cold storage room, because the humidity in the room is closely related to the TD. Table 27.2 is the approximate relationship between the room humidity and the design TD:

Table 27.2 Design TD vs Room Humidity

<table>
<thead>
<tr>
<th>Room Humidity, %RH</th>
<th>Design TD</th>
</tr>
</thead>
</table>
### Defrosting Systems:

All the coolers used for cold storage have frost problem, because most cold storage room temperatures are below freezing. Frost accumulated on the cooler coil is harmful to the refrigeration, because the frost build-up decreases the heat transfer efficiency and dampers the air flow through the coil. Therefore, the ice built-up on the coil must be defrosted periodically, if the cooler is operated in a cold room which the room design temperature is below 34°F. The defrosting function is preferred to be fully automatic.

Several defrosting methods which are commonly used for the ice removal are as the following:

(a) Air Defrost.
(b) Electric Defrost.
(c) Water Defrost.
(d) Hot Gas Defrost.
(e) Combination of Water and Hot Gas Defrost.

The followings are the general considerations for defrosting arrangement:

1) The method of defrosting under consideration should be compatible with the refrigeration system being designed; the initial cost, power consumption, operating and maintenance costs should also be optimized and evaluated.

2) The defrosting period should be as short as possible to minimize the temperature rise in the refrigerated space no matter what type of defrosting method is used.

3) The defrost system should be simple to operate.

4) It is very important that the drain line should be pitched from the cooler unit; the pitch slope should have 3 to 4 inches per foot. A trap should be provided and located outside from the refrigerated space.

5) Defrost all coils in a room at one time, wherever is possible.

6) When all coils in one room are not defrosted at one time, the drain from the coil or group of coils being defrosted should be brought out of the room as soon as possible and trapped before becoming common with other drains from the same room.

7) The drain line inside of the refrigerated space should be wrapped with a heat tape.
The refrigeration system design should provide maximum protection against liquid carryover to the compressor suction; particularly defrost system for DX and flooded coils.

**Air Defrost System:**

Air defrost may be used for cold storage rooms with design room temperatures not lower than 40°F.

The operation principle is when air defrosting cycle starts; the liquid and suction line solenoid valves are closed. The evaporator fans continue to run. The room air of 40°F or higher flows over the coil melting the frost on the coil; the defrosted water drops into the drain pan of the cooler. It is suggested to use a defrost relief regulating valve with the suction line solenoid valve and a trap at the suction before the compressor suction if refrigerant feed is DX.

**Electric Defrost System:**

Electric defrost is to place an electric heater network in the coil evaporator assembly. The electric heater is to heat up the entire coil to melt the frosted ice on the coil. Electric heaters are provided for the drain pans and drain lines. Drain pan and drain line are heated during defrost. Electric defrost system requires a longer defrost time and higher energy consumption than the hot gas or water defrost method.

Figure 27-7 shows the typical piping arrangement for direct expansion cooler with electric defrost arrangement. During the cooling cycle, the fans are on; the suction solenoid valve (SSV) and the liquid solenoid valve (LSV) are open. When the unit is activated for defrost cycle; the unit defrost door is closed; the liquid solenoid valve (LSV) and suction solenoid valve (SSV) are closed; the electric heaters are energized.

If the refrigerant pressure built-up in the coil is relieved through the defrost relief regulator (DRR). When the time of defrosting is over, the heater is de-energized, solenoid valves are opened. The fan is to be turned on until all the water drained out from the cooler. A suction accumulator is recommended to prevent liquid lug over to compressor suction.

**Water Defrost System:**

Figure 27-8 is the typical water defrosting piping system for a direct expansion cooler. Water defrost method is an effective way of defrosting. Water defrost unit is equipped with water spray heads above the coil. The water spray over the coil melts the ice off the coil. The water spray rate is approximately about 3 gpm per square foot of coil face area for a 5 to 15 minutes period.

During the cooling cycle, the fans are on; the suction solenoid valve (SSV) and the liquid solenoid valve (LSV) are open; water solenoid valve (WSV) is closed. When the unit is activated for defrost cycle; the fans are off; the liquid solenoid valve (LSV) and suction solenoid valve (SSV) are closed; the water solenoid valve (WSV) is energized, the water spray over the coil melts the ice on the coil. The refrigerant
pressure in the coil is relieved through the defrost relief regulator (DRR). When the time of defrosting is over, the water solenoid valve (WSV) is de-energized. The fan is turned on after all the water drained out from the cooler. A suction accumulator is recommended to prevent liquid lug over to compressor suction.

Figure 27-7  Typical Electric Defrost DX Piping Diagram

Figure 27-8  Typical Water Defrost DX Piping Diagram
Water defrost is effective and it is economically to install. But, water defrosting system has some disadvantages where they apply:

A. Water temperature must be maintained above 60°F.

B. Sand, scale and other impurities in the water sometimes cause the water solenoid valves to stick open, causing the room to be flooded.

C. If unit is used for room below freezing, the cooler unit must be perfectly level. If it is not, the water distribution header within the unit might not completely drain.

**Air and Water Defrosting for Liquid Recirculation System:**

Figure 27-9 shows the typical air defrost for liquid recirculation system. No separate suction trap is necessary; because liquid recirculation receiver serves as the suction accumulator (See Figure 27-11).

Figure 27-10 shows the typical water defrost for liquid recirculation system. No separate suction trap is necessary; because liquid recirculation receiver serves as the suction accumulator (See Figure 27-11).

**Hot Gas Defrosting Systems:**

Hot gas defrost is to apply the hot gas from the compressor discharge directly to the evaporator coil and the drain pan. Heat for defrosting is mainly from the latent heat of the compressed vapor, the vapor is condensed in the coil. This condensed liquid must be returned to somewhere, either to an oversized surge drum for flooded cooler; a suction trap for DX system or back to liquid recirculation receiver.

Liquid recirculation system is best suitable for hot gas defrosting application. Figure 27-11 shows a typical liquid recirculation piping with hot gas defrost. The vapor is supplied to coil and drain pan, heat of the gas melts the frost on the coil, the vapor is condensed in the coil; the liquid flows through suction pressure regulating valve and back to the liquid recirculation receiver via the suction piping line.

If the refrigerant feed is direct expansion instead of liquid recirculation, a suction trap before the compressor suction is to prevent liquid slug back to compressor. If the refrigerant feed is flooded, the surge drum should be sized large enough to prevent the liquid slug over during hot gas defrost cycle.

**Hot Gas Defrosting for Flooded Cooler:**

Figure 27-12 shows a typical flooded evaporator using hot gas defrosting. In this case, the surge drum of the flooded cooler is used to accumulate the condensed liquid, providing that the surge drum is sized large enough for this purpose. The sequence of operation is as the following:
Figure 27-9  Typical Air Defrost
Liquid Recirculation Piping Diagram

Figure 27-10  Typical Water Defrost
Liquid Recirculation Piping Diagram
Figure 27-12 Typical Hot Gas Defrost
Flooded Cooler Piping Diagram

Figure 27-13 Typical Hot Gas Defrost
Liquid Recirculation Piping Diagram
REFRIGERATION CYCLE – The fans are on, the liquid from receiver flows into the surge drum through the liquid line valve (LLV) which is controlled by the liquid level controller (LC); the pilot solenoid valve (PSV), hot gas solenoid valve (HGS) and the defrost relief regulator (DRR) are closed. The liquid line check valve (LLCV) and gas line check valve (GLCV) are open. Suction gas returns to compressor suction through the back pressure regulator (BPR).

DEFROST CYCLE – The fans are off, the liquid line valve (LLV) is closed; the pilot solenoid valve (PSV), hot gas solenoid valve (HGS) and the defrost relief regulator (DRR) are open. The liquid line check valve (LLCV) and gas line check valve (GLCV) are closed. The hot gas flows through the drain pan coil and to the coil of the evaporator to defrost the ice on the coil. The hot gas is condensed inside of the coil, the liquid condensate flows to surge drum through defrost relief regulator (DRR). The defrost relief valve is to be set to 70 psig back pressure in case of ammonia on the coil during defrost cycle; the hot gas pressure for the defrosting should be at least 110 psig.

**Hot Gas Defrosting for Liquid Recirculation Evaporator:**

Figure 27-13 shows a typical piping system for hot gas defrosting liquid recirculation evaporator.

REFRIGERATION CYCLE – Fans are on, liquid line solenoid valve (LLSV) and suction shut valve (SSV) are open, defrost regulating valve (DRV) and hot gas solenoid valve (HGSV) are closed.

DEFROSTING CYCLE – Fans are off, liquid line solenoid valve (LLSV) and suction shut valve (SSV) are closed, defrost regulating valve (DRV) and hot gas solenoid valve (HGSV) are open. The hot gas flows through the drain pan coil and to the coil of the evaporator to defrost the ice on the coil. The hot gas is condensed inside of the coil, the liquid condensate flows through suction line back to liquid recirculation receiver through defrost relief valve (DRV).

**Top Feed or Bottom Feed for Liquid Recirculation Coil:**

Figure 27-14 shows a product cooler with liquid recirculation coil. The coil has two headers, inlet and outlet. The inlet header feeds number of tubes; the tubes are routed back and forth through the length of the evaporator. Each tube passage, from liquid header to the suction heater; this tube passage is called circuit. If the refrigerant liquid is unevenly distributed the circuits, due to pressure differences, the liquid usually “starves” the top circuits of the coil. There, orifices should be inserted into the tube openings in the liquid header to resolve this problem as shown in Figure 27-15.

Figure 27-15 also shows “Top Feed” and “Bottom Feed” arrangements for the liquid recirculation coil. There is no definite conclusion as which one method is better than the other.
Figure 27-14  Typical Liquid Recirculation Unit Cooler

Figure 27-15  Top Feed Vs Bottom Feed
Liquid Recirculation Coil
Data Required for Product Cooler or Unit Cooler Selection:

- Refrigeration capacity, Btu/Hr.
- Room design temperature.
- Evaporative temperature.
- Refrigerant.
- DX, Flooded, Liquid Recirculation or Brine.
- Coil material.
- Coil FPI. 3, 4, or 6 FPI.
- Vari-fin application.
- Type of fan and motor, power supply.
- Type of defrosting.
- Other options.

Liquid Spray No-Frost Cooler:

The liquid spray no-frost cooler is to use a brine solution such as propylene glycol or ethylene glycol, the brine is spray continuously over the evaporator coil to keep the coil from frosting. But, this type of spray cooler is not widely used for modern cold storage facilities except that the brine solution is Lithium Chloride (LiCL) or Tri-Ethylene Glycol (TEG) for hygienic applications.

The liquid spray no-frost cooler system consists of two sections as shown in Figure 27-16. The No-Frost evaporator section is to be located inside the cold room or outside of the cold room connected with supply and return air ducts. The Solution Concentrator or Regenerator section is to be located outdoor.

The no-frost cooler section consists of the fan assembly, the refrigeration coil, pump and spray assembly, eliminator assembly and the casing with solution sump. No-frost liquid or brine solution is pumped and sprayed over the evaporator coil; the cooler fan circulates the air from the refrigerated space through the coil, the eliminator and back to the cold storage area. The no-frost liquid solution washes the frost on the coil surface continuously. The no-frost liquid solution then becomes diluted and is pumped to the solution concentrator which is located at the outdoor.

The Solution Concentrator (or regenerator) section consists of fan assembly, the heating coil, pump and spray assembly, eliminator assembly and the casing. The diluted solution from the no-frost cooler is pumped to the solution concentrator through a heat recovery heat exchanger. The diluted solution is sprayed in the regeneration section where excessive moisture is evaporated from the liquid mixture by circulating outside air and also with supplement heat from a heating coil, if needed. The enriched and regenerated solution is then circulated through the heat recovery heat exchanger, back to the no-frost cooler.

Hygienic No-Frost Coolers:

The operation theory for the hygienic no-frost cooler is same as shown in Figure 27-16. Hygienic no-frost cooler is used for those installations where sterilization is required.
Figure 27-16  No-Frost Cooler System
The hygienic type no-frost cooler removes airborne mold spores, bacteria, particles, germs and micro-organism. Therefore, the hygienic no-frost cooler is mostly used for food processing, breweries and hospital rooms, etc. Also, it might be used for area that great amount of washing such as aging and fermenting tank in brewery or area produces extremely high latent loads; it also use for meat packing, food processing plants, food distribution centers, frozen food warehouse, chemical plants and laboratories.

There are basically two types of liquid absorbents used for no-frost hygienic application, Lithium Chloride (LiCL) and Tri-Ethylene Glycol (TEG). Lithium Chloride (LiCL) is a corrosive and toxic material, therefore, the eliminator efficiency must be very good for this type of solution. On the other hand, Tri-Ethylene Glycol is a non-toxic and non-corrosive material; it meets the approval of USDA and CFIA for use with food grade application.
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