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.
(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
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.
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 gain 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.

Figure 7-6 Inlet Guide Vane for Centrifugal Compressor
This changes the performance characteristic of the centrifugal impeller without changing the compressor speed.

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.
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:

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

- \(N\) = 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, an 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.