# Chapter – 26 Case of Structuring Refrigeration System with Multistage Centrifugal Compressor


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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

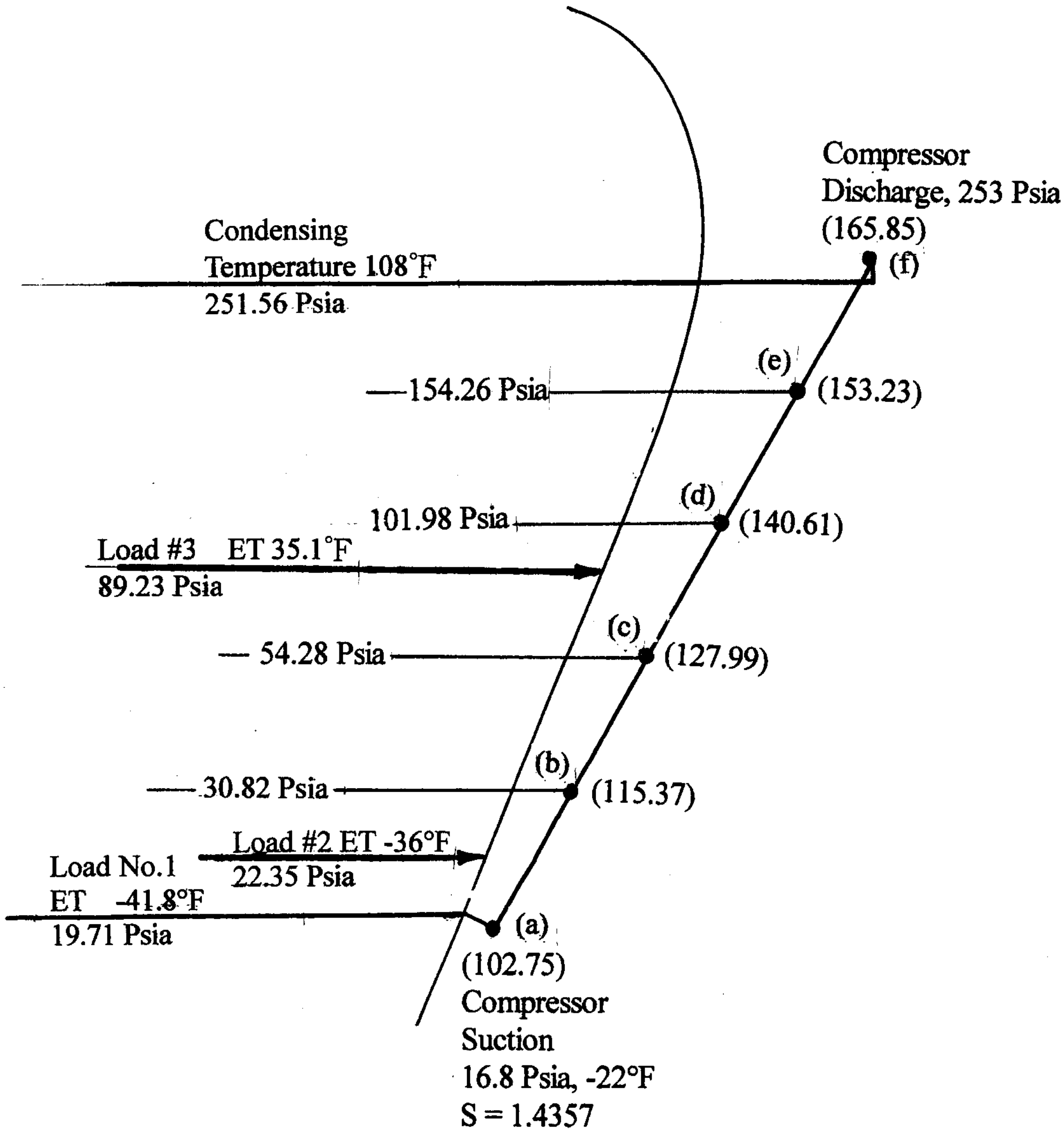


**Figure 26-1 Interstage Pressure**
**5-Stage Multistage Centrifugal**

<u>Guesstimate number stages required for the centrifugal compressor:</u>

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 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:

$$
\begin{aligned}
P &= 16.8 \text{ Psia,} \\
t &= -22°\text{F} \\
H_1 &= 102.75 \\
V_g &= 6.411 \text{ Ft}^3\text{/Lb} \\
V_a &= 761.39 \text{ Ft/Sec.} \\
S &= 1.4357
\end{aligned}
$$

The enthalpy point for the compressor discharge:

$H_f = 165.85$ at discharge pressure of 253.0 Psia and entropy of 1.4357

Therefore, the adiabatic Compression Head $= H_f - H_a = 165.85 - 102.75$

$= 63.10$ Btu/Lb

or $= 63.10 \times 778 = 49{,}092$ Ft.

Use the formula listed in Chapter 7 to guesstimate the number of stages needed for the multistage centrifugal compressor for the application:

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

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

$$N \fallingdotseq \frac{48 \times 49{,}092}{[761.39]^2} = 4.065$$

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:

$$\text{Each impeller is to handle} = \frac{63.10}{5 \text{ Stages}} = 12.62 \text{ Btu/Lb per wheel}$$

Therefore, the enthalpy for the point (b) is 102.75 + 12.62 = 115.37 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 $4^{th}$ stage suction. Therefore, the suction gas of Load #3 has to be returned to the $3^{rd}$ 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:

$$\text{Compression Head} = H_2 - H_1 = 165.85 - 102.75$$

$$= 63.10 \text{ Btu/Lb}$$

$$\text{or} \quad = 63.10 \times 778 = 49{,}092 \text{ Ft.}$$

$$\text{Each impeller is to handle} = \frac{63.10}{6 \text{ Stages}} = 10.517 \text{ Btu/Lb enthalpy per wheel}$$

Or

$$\text{Each impeller is to handle} = \frac{49{,}092}{\quad} = 8{,}182 \text{ ft head per wheel}$$

6-stage

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°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°F; the evaporative pressure is 89.23 Psia. The pressure is high enough to be returned to the $4^{th}$ wheel inlet as shown in Figure 26-3. The suction pressure of the $4^{th}$ 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 $2^{nd}$ stage inlet and the other is to be located at $3^{rd}$ 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 $1^{st}$ intercooler is set for 52°F and the $2^{nd}$ 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 $2^{nd}$ 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.

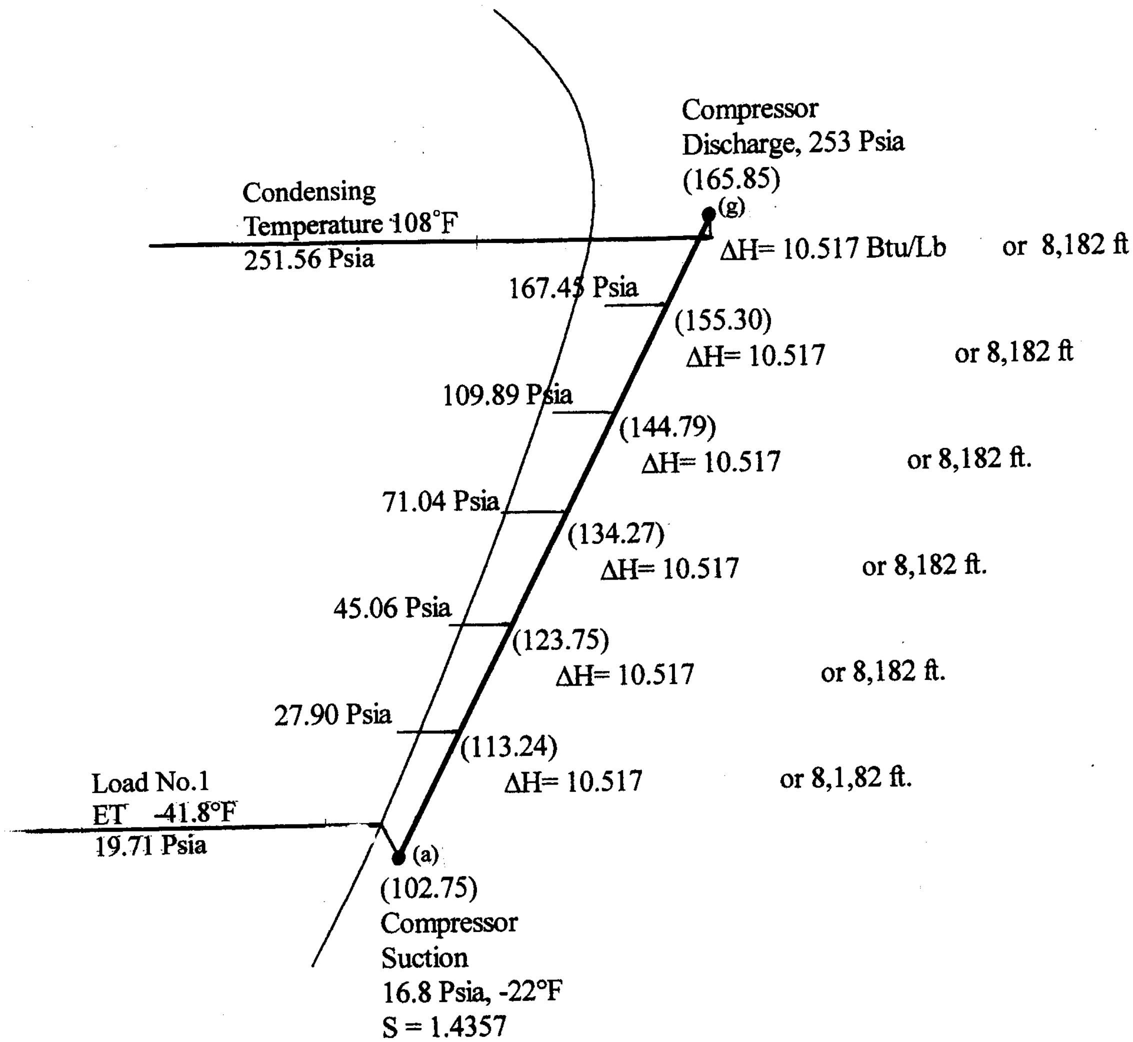


**Figure 26-2** **Interstage Pressure**
**6-Stage Multistage Centrifugal**

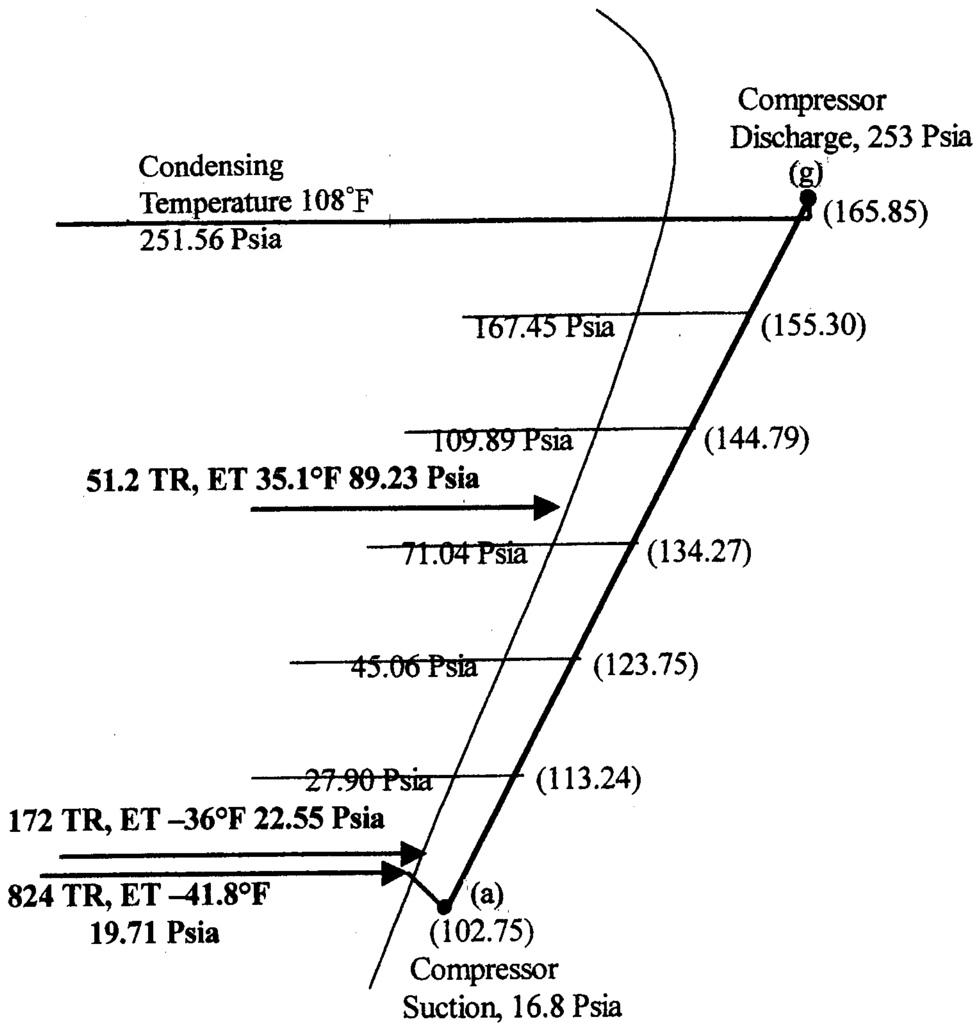


**Figure 26-3 Side Load Connections**

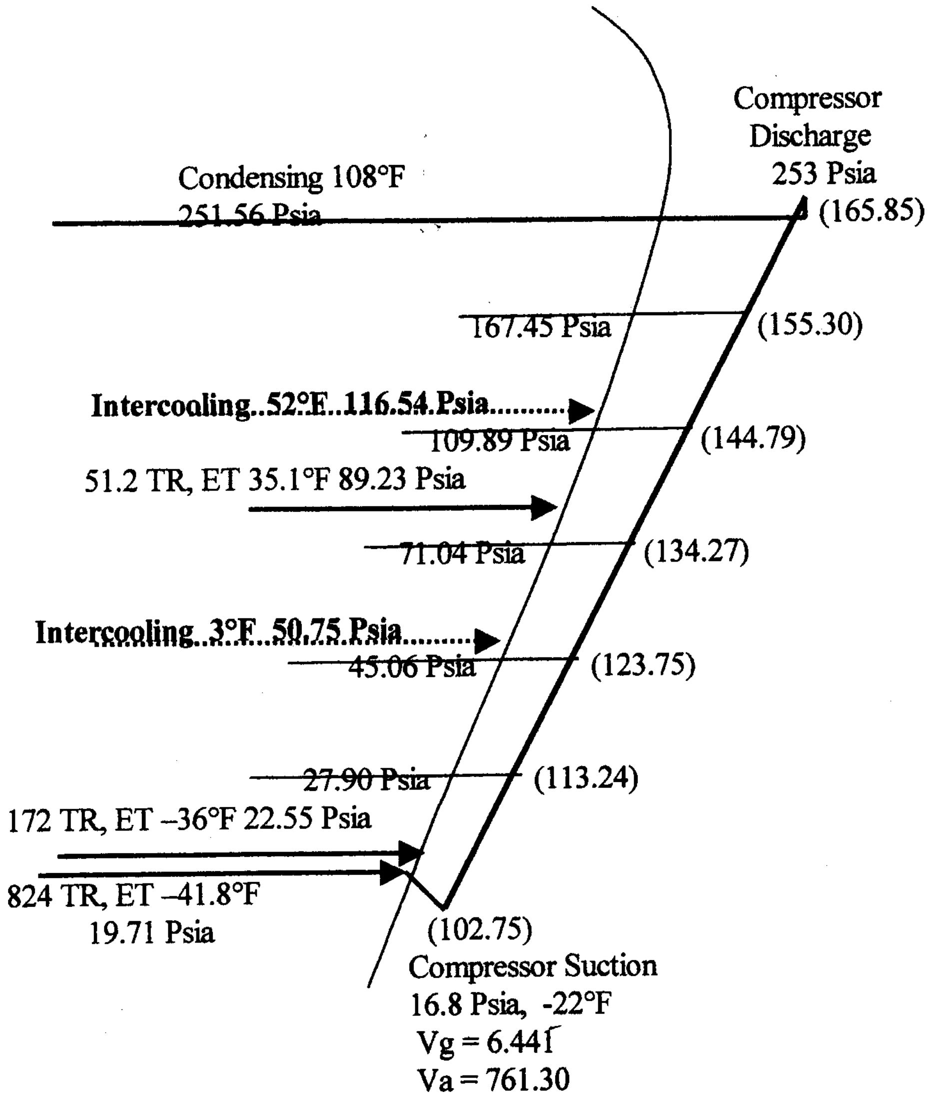


Figure 26-4 Intercooler-Economizer Locations.

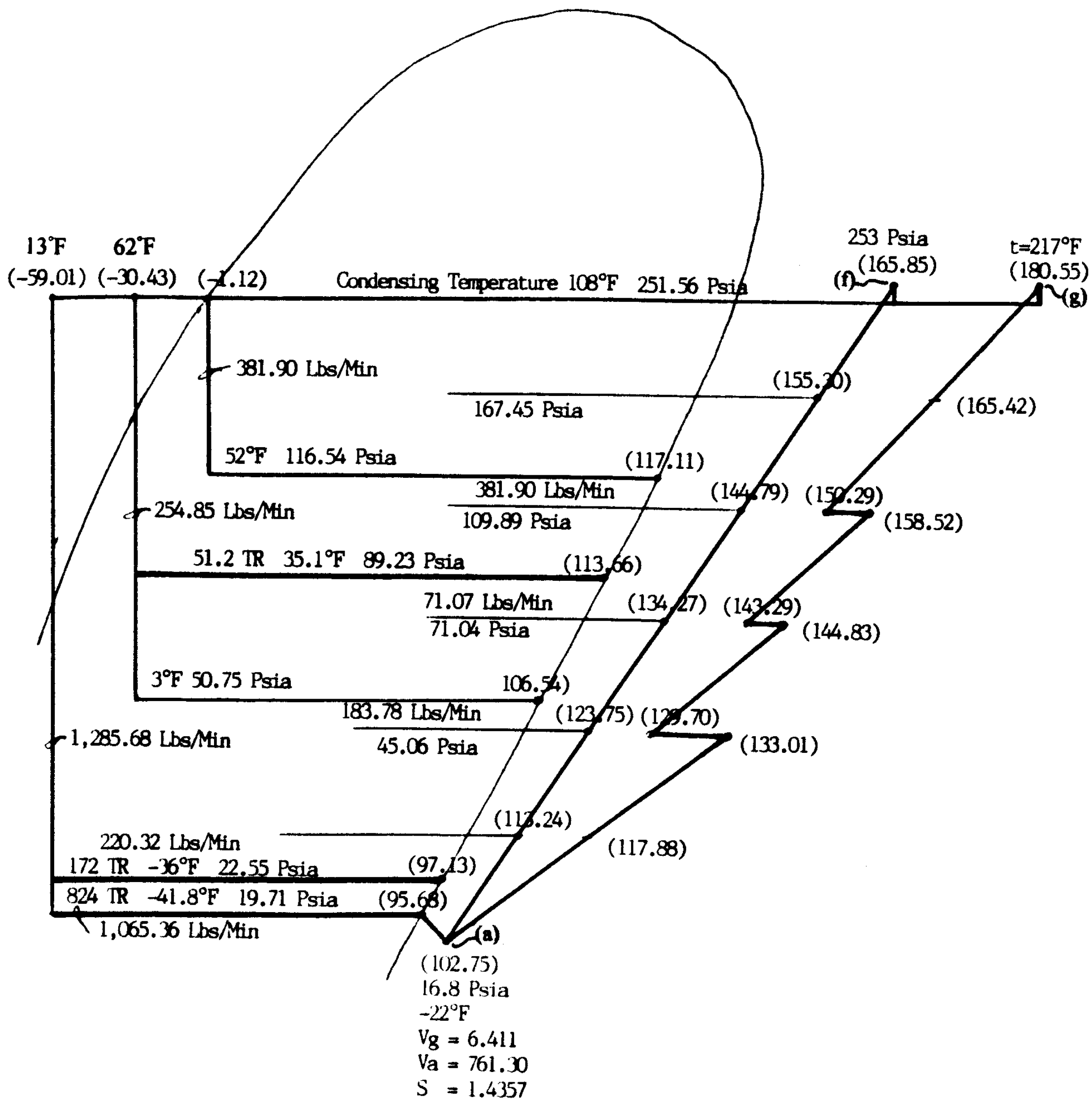


**Figure 26-5** System P-H Diagram for 6-Stage Centrifugal Refrigeration System

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.

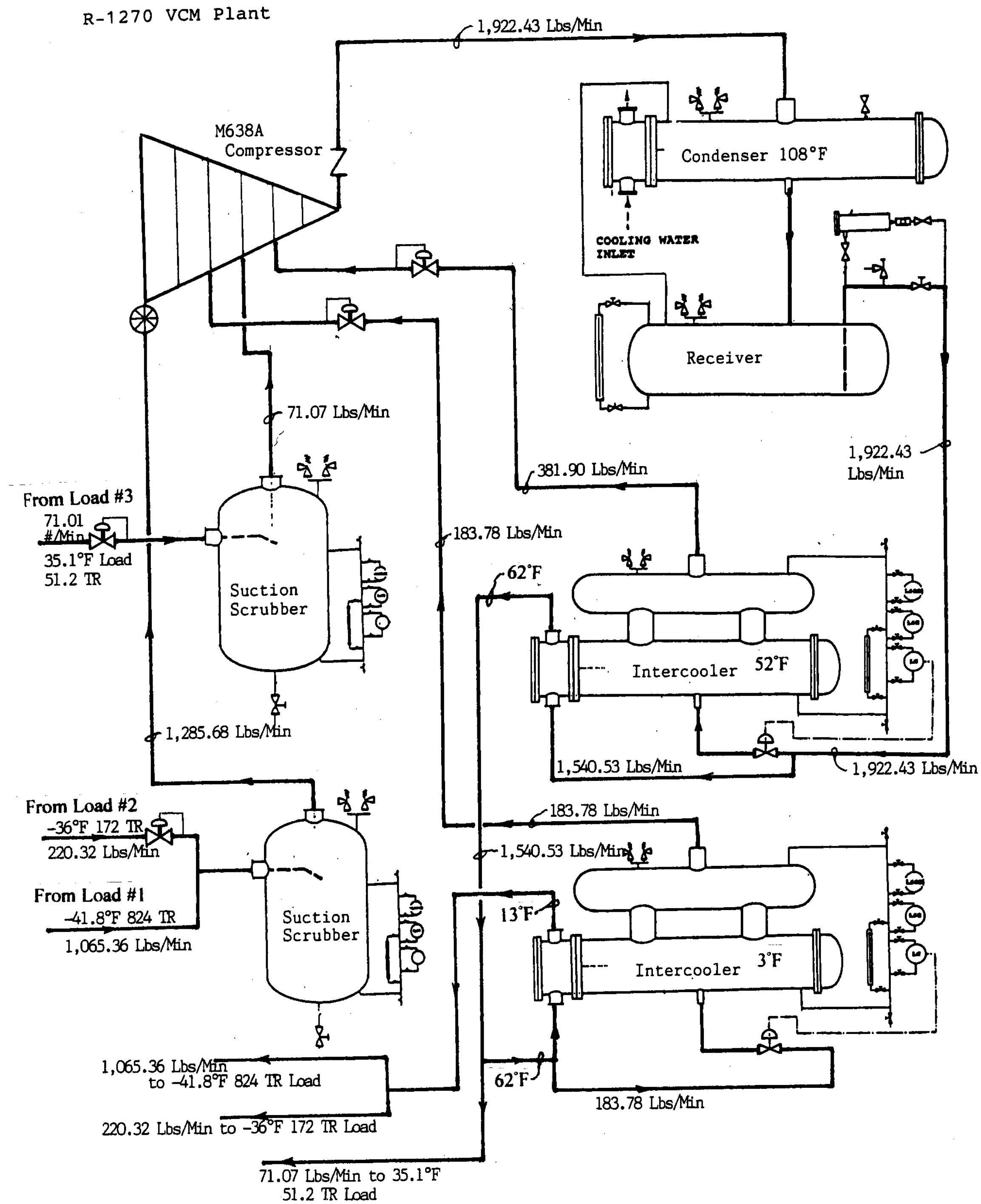


**Figure 26-6 System Refrigerant Flow Diagram 6-Stage Centrifugal**