Case – 12 Multistage Centrifugal Refrigeration System Halocarbon Refrigerant

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Case Background:

This case is to show how to achieve the following tasks:

(1) To determine the number of stages and how to accommodate a side load for multistage centrifugal compressor.

(2) To select a multistage centrifugal compressor to meet the design conditions.

(3) To design a refrigeration system for the centrifugal compressor selected and the operating conditions specified.

The formula for the estimation of the number of stages for centrifugal shown in the book of Chapter 7 of Engineered Industrial Refrigeration Systems Application is a formula for very rough estimate. The selection method show in this case shall provide a more detail information as how to determine the number of stages. It is important to note that the centrifugal compressor selection should be made and confirmed by the compressor manufacturer. The method shown in this case is just for the purpose for preliminary design of the refrigeration system.

The operating conditions for the multistage centrifugal compressor are as the following:

Refrigerant:	R-22
Refrigeration Capacity (1):	606 TR at ET of –22°F
Refrigeration Capacity (2):	216 TR at ET of 14°F
Condensing Temperature:	104°F
Discharge piping pressure drop:	0.64 Psi
Suction piping pressure drop:	0.3 Psi for both evaporators
Suction superheat:	6°F for –22°F evaporator, 4°F for 14°F evaporator.

Related Technical Data and Engineering Information for the Case:

Table 12-1 Compressor Impeller Diameter-Inches and $(Dia.)^2$

CASING SIZE	DIA.	$(DIA.)^2$
26B	12.2	149
26A	14.8	219
38B	18.0	324
38A	21.9	480
55B	26.7	713
55A	31.5	993

Table 12-2 Maximum Allowable HP Per 1,000 RPM

CASING SIZE	SHAFT OR COUPLING	EACH IMPELLER
26	356	91
38	1,360	295
55	2,650	877

Table 12-3 Maximum Allowable Compressor Speed and CFM Flow

CASING SIZE	MAXIMUM	
	RPM	CFM*
26B	15,950	3,690
26A	13,150	5,450
38B	10,800	8,050
38A	8,900	11,900
55B	7,300	17,700
55A	6,180	24,600

*Note:
Maximum CFM may be less than shown depending on head requirements and mol. wt. of gas being pumped.

Table 12-4 Approximate Compressor First Critical Speed - RPM

Comp. Model	Impeller Material: All Aluminum	Impeller Material: All Steel	Comp. Model	Impeller Material: All Aluminum	Impeller Material: All Steel
226B	61,200	48,700	526B	14,400	11,400
226A	55,700	41,500	526A	13,100	9,600
238B	41,000	32,700	538B	9,700	7,600
238A	37,300	27,800	538A	8,800	6,500
255B	28,100	22,400	555B	6,600	5,200
255A	25,600	19,100	555A	6,000	4,400
326B	31,900	25,400	626B	11,100	8,700
326A	29,200	21,500	626A	10,000	7,400
338B	21,400	17,000	638B	7,400	5,800
338A	19,600	14,500	638A	6,700	4,900
355B	14,700	11,700	655B	5,100	4,000
355A	13,400	10,000	655A	4,600	3,400
426B	20,300	16,100	726B	8,900	7,000
426A	18,400	13,600	726A	8,000	5,900
438B	13,600	10,800	738B	6,000	4,700
438A	12,300	9,200	738A	5,400	3,900
455B	9,300	7,400	755B	4,100	3,200
455A	8,400	6,300	755A	3,700	2,700

Note: The first number refers to number of stages.
The operating compressor speed of the compressor shall not exceed 80% of the first critical speed.

Table 12-5 Maximum Temperature Limitation for Impellers

<table>
<tr><th colspan="2">Aluminum</th><th>Stainless Steel</th></tr>
<tr><th>FPS</th><th>°F</th><td rowspan="3">520°F at Any Speed</td></tr>
<tr><td>900</td><td>283</td></tr>
<tr><td>850</td><td>300</td></tr>
<tr><td>800</td><td>317</td><td rowspan="7">** Temp. with "P.R.V. closed," i.e., Design Temp. Rise Times 1.3 Plus Suction Temp.</td></tr>
<tr><td>750</td><td>334</td></tr>
<tr><td>700</td><td>351</td></tr>
<tr><td>650</td><td>368</td></tr>
<tr><td>600</td><td>384</td></tr>
<tr><td>550 & Less</td><td>400</td></tr>
</table>

EFF. MULT. #1		EFF. MULT. #2			
			MACH. NO.		
COMP. SIZE	MULT.	STAGES	UP TO 1.10	1.20	1.30
26	1.00	1	1.0	.98	.96
38	1.01	2	1.0	.98	.96
		3	1.0	.97	.94
55	1.02	4	1.0	.97	.92
		5	1.0	.96	.90

INTERPOLATE VALUES FOR EFFICIENCY MULTIPLIER, ADIABATIC HEAD COEFFICIENT, AND MAXIMUM CAPACITY FACTOR.

ADIABATIC HEAD COEF. μ_A

	MACH. NO.		
STAGES	UP TO 1.10	1.20	1.30
1	.51	.50	.49
2	.50	.49	.48
3	.49	.48	.47
4	.48	.47	.46
5	.47	.46	.45

CAPACITY LIMITS

MACH. NO.	MAX. CAP FACTOR
.6	.200 *
.7	.205 *
.8	.210 *
.9	.215
1.0	.220
1.1	.215
1.2	.210
1.3	.200

* CAN BE EXCEEDED WITH REDUCED μ AND η.

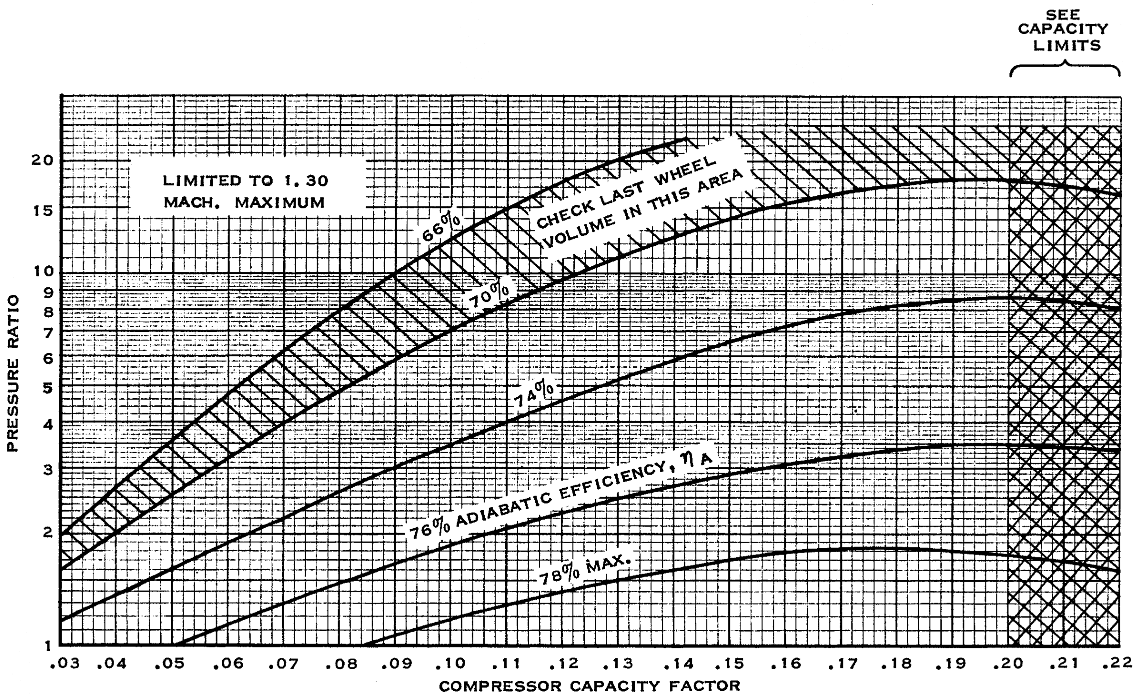


Figure 12-1 Adiabatic Efficiency Map for Multistage Centrifugal Compressor for Halocarbon Refrigerant

Note: Compressor efficiency and part load performance can be improved by changing the impeller profile design. Ask the compressor manufacturer for a better energy consumption selection for energy conservation application.

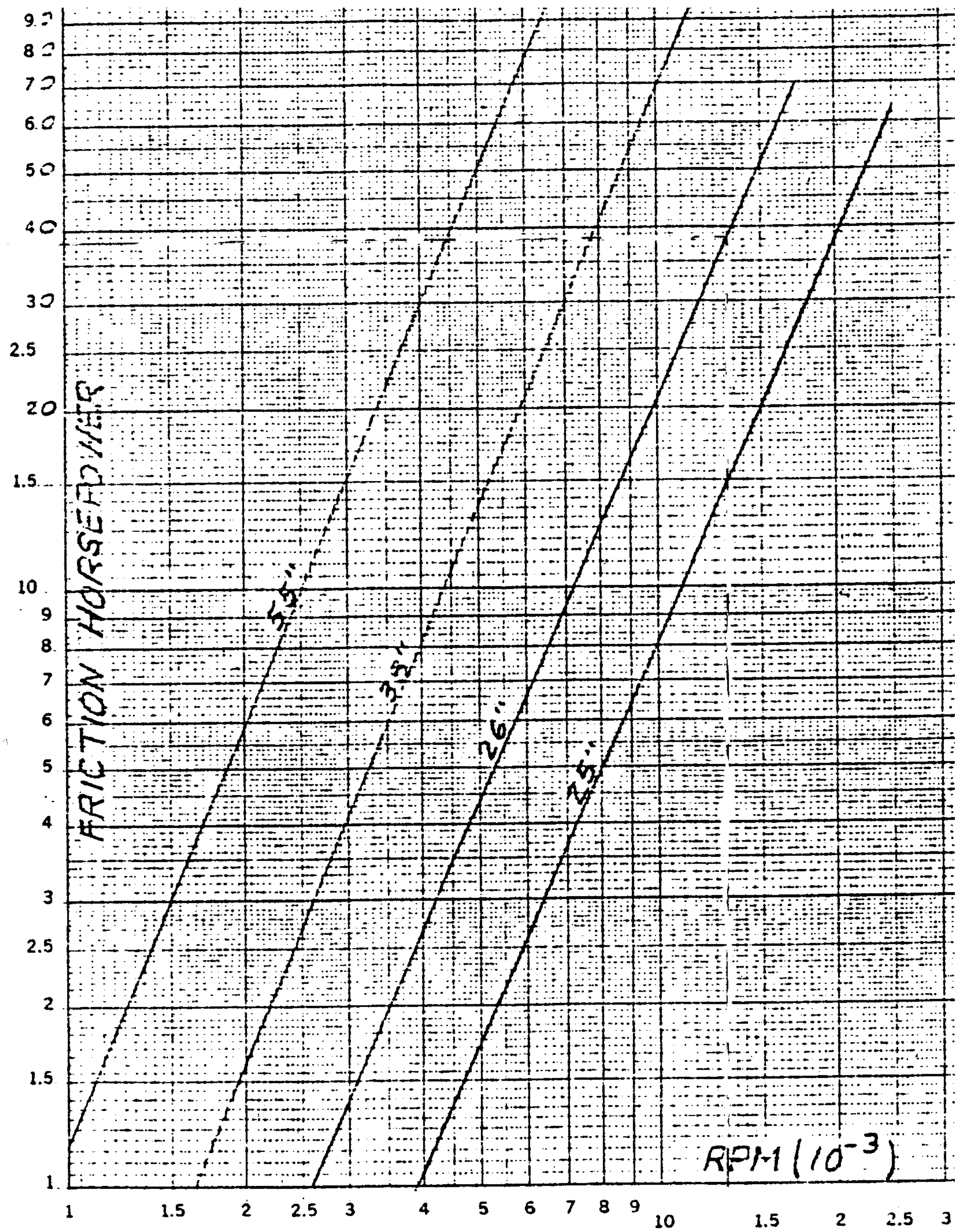


Figure 12-2 Friction HP for Multistage Centrifugal Compressor

Cogitation

Recap the operating conditions specified for the compressor:

Refrigerant:	R-22
Refrigeration Capacity (1):	606 TR at ET of –22°F
Refrigeration Capacity (2):	216 TR at ET of 14°F
Condensing Temperature:	104°F
Discharge piping pressure drop:	0.64 Psi
Suction piping pressure drop:	0.3 Psi for both evaporators
Suction superheat:	6°F for –22°F evaporator, 4°F for 14°F evaporator.

Evaporative Pressure at –22°F = 23.7 Psia
Condensing Pressure at 104°F = 222.36 Psia

Assuming compressor suction entrance loss = 0.9 Psi
Assuming compressor discharge nozzles loss = 8.0 Psi

Compressor Suction conditions:

Suction Pressure: 23.7 – 0.3 – 0.9 = 22.5 Psia
Suction temperature: -22 + 6 = -16°F

Compressor discharge pressure:

Discharge pressure: 222.36 + 0.64 + 8.0 = 231.0 Psia

Notes:

Centrifugal compressor is determined by both the compression head and the actual suction flow of the refrigerant (capacity).

(a) Head determines the requirement of how many stages to be used.

(b) Refrigerant Flow (Capacity) determines the compressor size.

Step No. 1 Determine how many stages are needed for the application base on the lowest compressor suction conditions and the highest compressor discharge pressure:

From computer refrigerant property program, all the data at suction conditions can be obtained as the following:

Suction conditions:	P	=	Pressure	=	22.5 Psia,
	t	=	Temperature	=	-16°F
	H_1	=	Enthalpy	=	103.26 Btu/Lb
	V_g	=	Suction Specific Volume	=	2.3365 Ft^3/Lb
	V_a	=	Accoustic Velocity	=	536.39 Ft/Sec.
	S	=	Entropy	=	0.2372

Discharge conditions:

231 Psia

$H_2 = 129.80$

Compression Head (H_{ad}) $= H_2 - H_1 = 129.80 - 103.26$

$= 26.54$ Btu/Lb

or $H_{ad} = 26.54 \times 778 = 20{,}648$ Ft.

Maximum Mach number for the application is 1.3

$$M_o = \frac{T_s}{V_a}$$

M_o : Mach number

T_s : Impeller Tip Speed, Ft/Sec

V_a : Accoustic Velocity. Ft/Sec.

<u>Reasonable</u> Max. impeller Tip Speed = $M_o \times V_a = \underline{1.25} \times 536.39 = 670$ Ft/Sec.

Tip Speed Formula:

$$T_s = \sqrt{\frac{32.2 \times H_{ad}}{N \times \mu_{ad}}}$$

Assume $\mu_{ad} = 0.51$

$$670 = \sqrt{\frac{32.2 \times 20{,}648}{N \times 0.51}}$$

From the above formula:

Number of stages, N = 2.904 Say 3-stage T_s = 659.2 Ft/Sec.
M_o = 1.229

It is preferred that the Mach number to be around 1.1 unless it is under tight cost consideration.

Therefore, try to use a 4-stage compressor instead of 3-stage

$$T_s = \sqrt{\frac{32.2 \times 20{,}648}{4 \times 0.51}}$$

$$= 570.9 \text{ Ft/Sec}$$

$$M_o = \frac{570.9}{536.39} = 1.064$$

From the Adiabatic Head Coefficient Table, Figure 12-1, the Maximum value of μ_{ad} for Mach number up to 1.10 is 0.48, therefore the tip speed should be corrected as the following:

$$T_s = \sqrt{\frac{32.2 \times 20{,}648}{4 \times 0.48}}$$

$$= 588.46 \text{ Ft/Sec}$$

$$M_o = \frac{588.46}{536.39} = 1.098 \qquad \text{Ok!}$$

4-stage compressor is used.

<u>Step No. 2</u> Determine the inter-stage pressures of the compressor.

Compression Head = $H_2 - H_1$ = 129.80 - 103.26
= 26.54 Btu/Lb

$$\text{Each impeller is to handle} = \frac{26.54}{4 \text{ Stages}} = 6.635 \text{ Btu/Lb}$$

Standard arrangement for the compressor is all the wheels are having the same diameter; each impeller handles equal head, not equal compression ratio as shown in Figure 12-3.

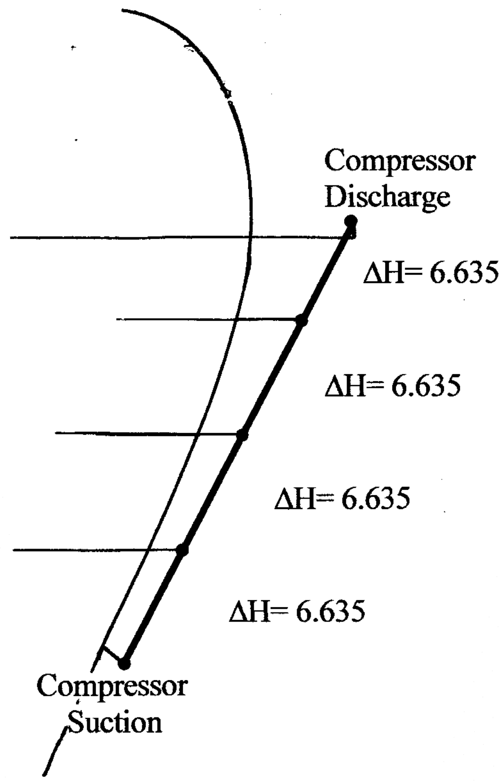


Figure 12-3 Enthalpy Difference Between Impellers

At compressor suction the suction pressure is 22.5 Psia, the suction temperature is -16°F, from refrigerant R-22 property, the enthalpy at suction condition H_1 is 103.26 Btu/Lb and the entropy is 0.2372.

See Figure 12-4, the ΔH for each stage is 6.635, therefore, the enthalpy for the first impeller discharge is 103.26 + 6.635 = 109.90; the enthalpy value for each impeller discharge can be obtained by using the same method as the following:

Enthalpy for 1st stage impeller:	109.90
Enthalpy for 2nd stage impeller:	116.53
Enthalpy for 3rd stage impeller:	123.17

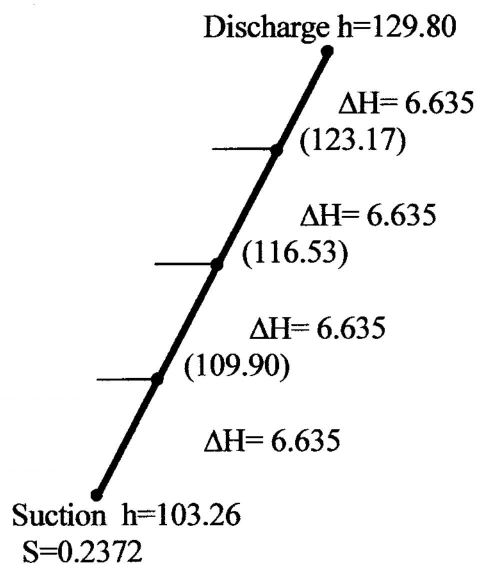


Figure 12-4 The Enthaply Points Between Impeller Discharge

See Figure 12-5, under adiabatic compression function, the S = 0.2372 is constant, the interstage pressure between stage can be obtain by using various enthalpy points at constant entropy of 0.2372.

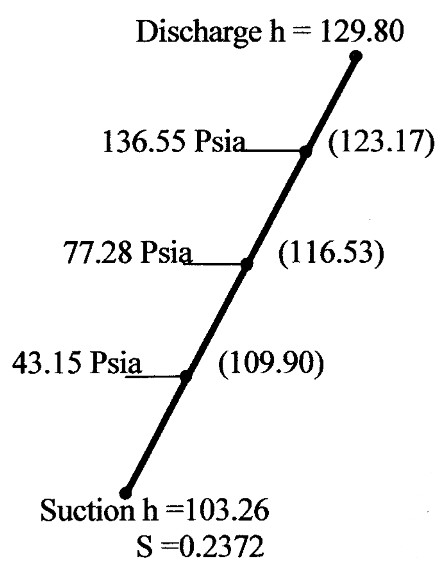


Figure 12-5 Interstage Pressure Between Stages

Suction pressure for 2nd stage impeller:	43.15 Psia
Suction pressure for 3rd stage impeller:	77.28 Psia
Suction pressure for 4th stage impeller:	136.55 Psia

<u>Step No. 3</u> – To determine where the side load connection should be located for the ET of 14°F No.2 refrigeration load.

The evaporative pressure for the 14°F refrigeration load is 51.37 Psia. Therefore, side load connection for the No. 2 refrigeration should be at the second stage inlet where the impeller inlet pressure is 43.15 Psia.

The total pressure difference for piping loss and the nozzle connection pressure drop is 51.37 – 43.15 = 8.22 Psi.

Pressure drop allocations:

External piping PD =	0.3 Psi
Control Valves =	4 Psi
PD available for Side Connection =	3.92 Psi

<u>Step No. 4</u> - To format the P-H diagram in accordance with the design concept and select the number of stage intercooling for the compressor:

Assume the system is a close couple refrigeration system. The evaporators are close to each other in the same engine room. Therefore, flash intercoolers are to be used.

4-Stage compressor, maximum 3 intercooling are allowed. 2 stages flash intercooling are used for the system for this case instead of 3-stage intercooling because the suction of the side load 216 TR at ET of 14°F already connected to the 2nd stage inlet.

Figure 12-6 is the P-H diagram which reflects all the design approaches with 2-stage flash intercooling.

The evaporative temperature for intercooler is fixed to allow approximately 5 Psi pressure difference between the saturated evaporative pressure of the intercooler and the interstage suction pressure of the impeller; this 5 Psi pressure difference allowed is for external piping and control valve pressure drops for the intercooler.

All the enthalpy points, temperatures and pressures shown on the Figure 12-6 are obtained from the R-22 refrigerant property software.

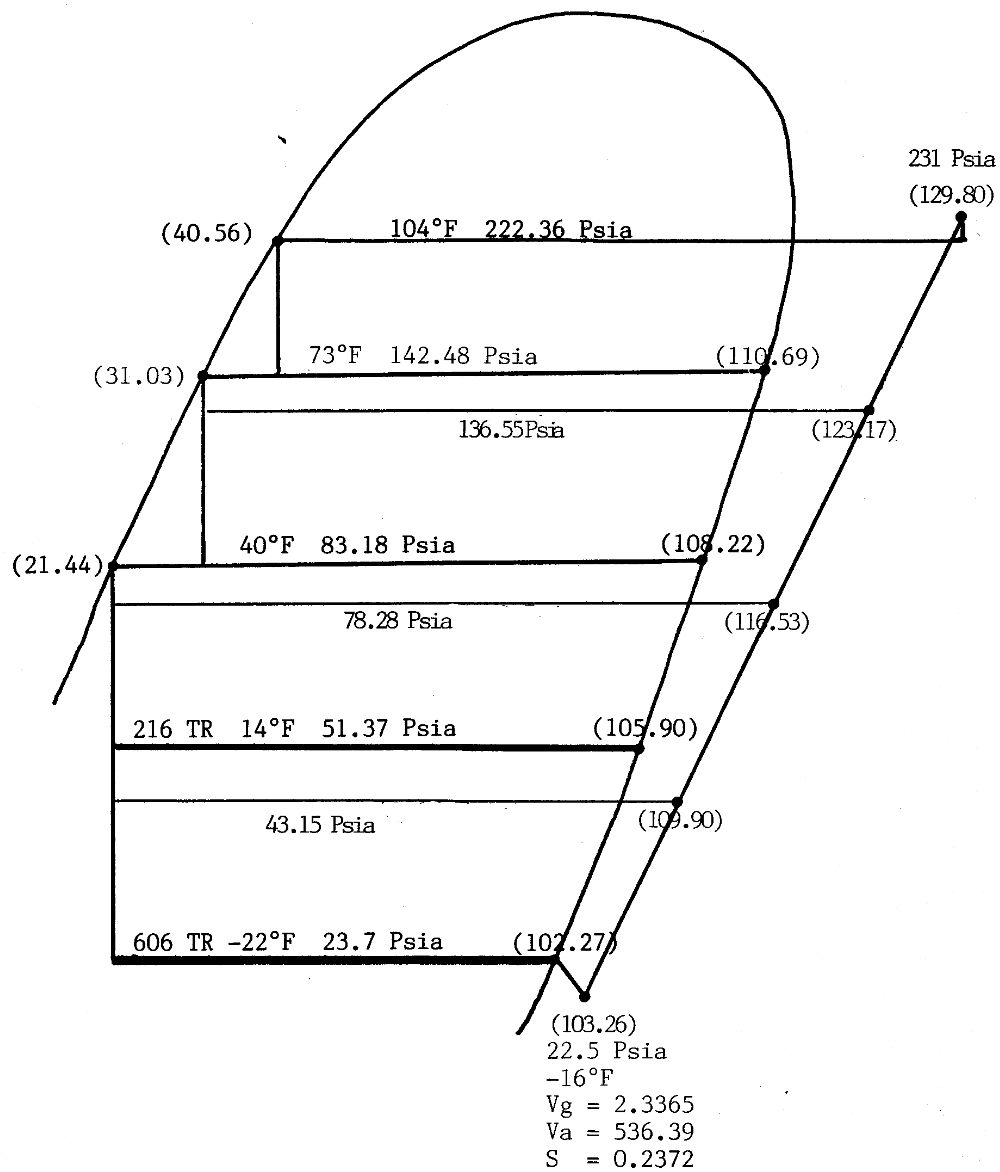


Figure 12-6 Constructing P-H Diagram for the System

The refrigerant flow diagram of Figure 12-7 is composed in accordance with the functions as defined in the P-H diagram of Figure 12-6.

High pressure float valve with combination economizer-receiver is used to reduce the initial cost for the 2nd stage intercooling.

Refrigerant flows are calculated and based on the P-H diagram of Figure 12-6.

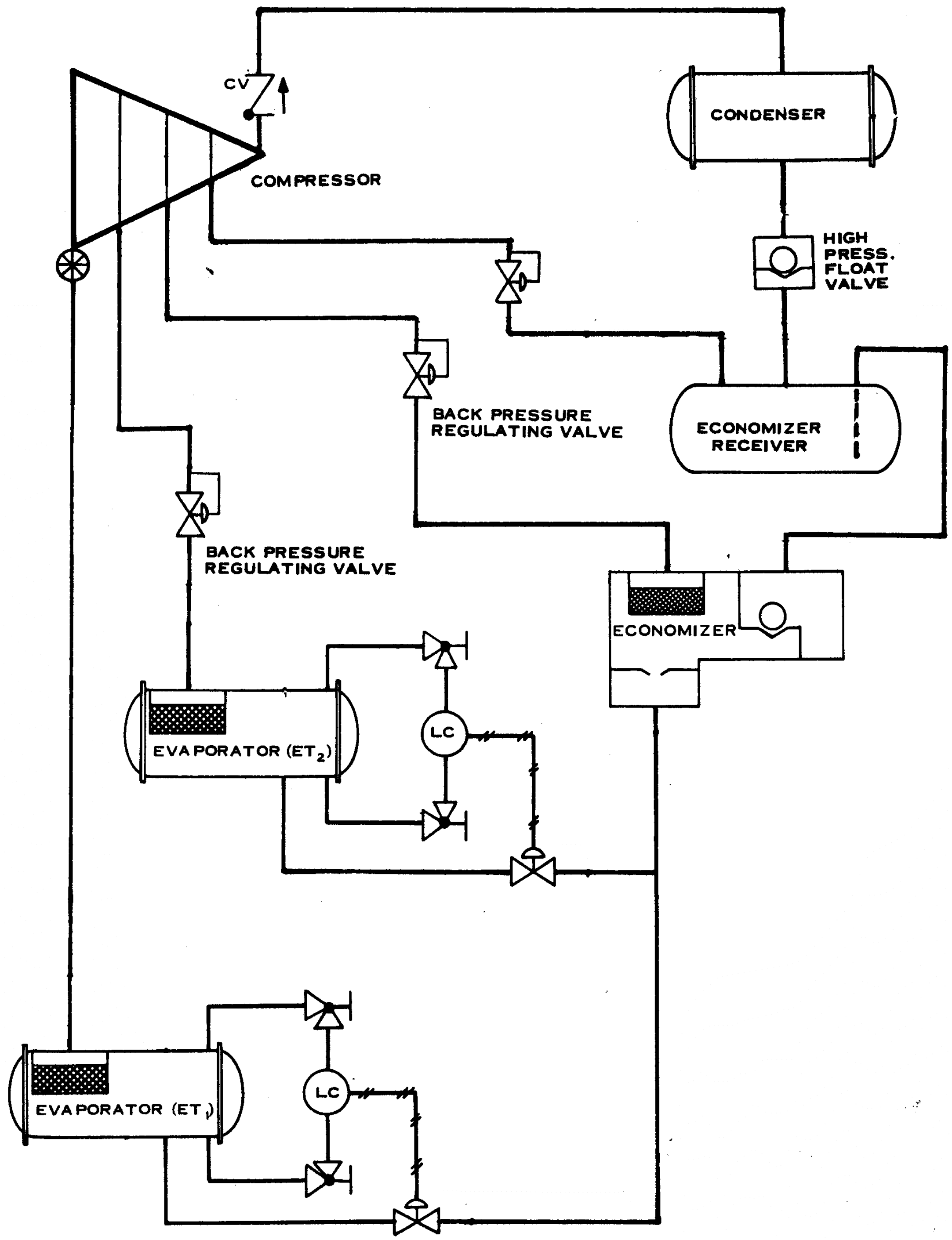


Figure 12-7 Composed Refrigerant Flow Diagram for the System

Step No. 5 – Calculate the refrigerant flows of the system.

Compressor main suction flow, Refrigeration Load No.1:

$$\frac{200}{102.27 - 21.44} \times 606 = 1{,}499.44 \text{ Lbs/Min.}$$

Refrigerant flow for Load No.2:

$$\frac{200}{105.90 - 21.44} \times 216 = 511.48 \text{ Lbs./Min.}$$

Total refrigerant flow for Loads No.1 and No.2:

$$= 1{,}499.44 + 511.48 = 2{,}010.92 \text{ Lbs./Min.}$$

Flash gas to the compressor 2nd stage inlet:

$$= 2{,}010.92 \times \left[\frac{108.22 - 21.44}{108.22 - 31.03} - 1 \right]$$

$$= 249.83 \text{ Lbs./Min.}$$

Liquid leaving 1st stage economizer:

$$= 2{,}010.92 + 249.83$$
$$= 2{,}260.75 \text{ Lbs./Min.}$$

Flash gas to the compressor 1st impeller inlet:

$$= 2{,}260.75 \times \left[\frac{110.69 - 31.03}{110.69 - 40.56} - 1 \right]$$

$$= 307.21 \text{ Lbs./Min.}$$

Step No. 6 – Determine Compressor suction ACFM:

Compressor suction ACFM flow:

= Suction flow (Lbs./Min.) x V_g

= 1,499.44 x 2.3365 = 3,503.44 CFM

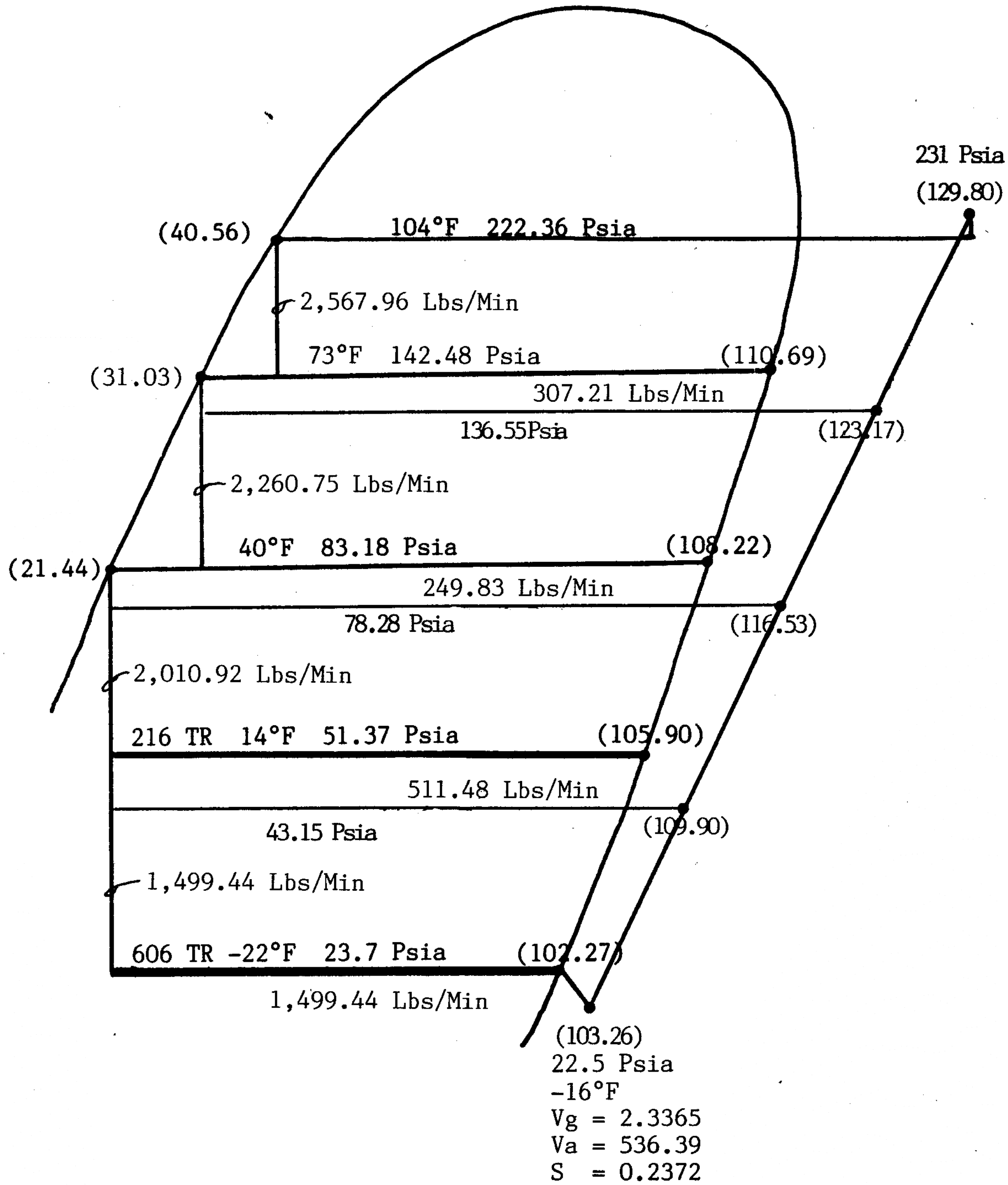


Figure 12-8 System P-H Diagram

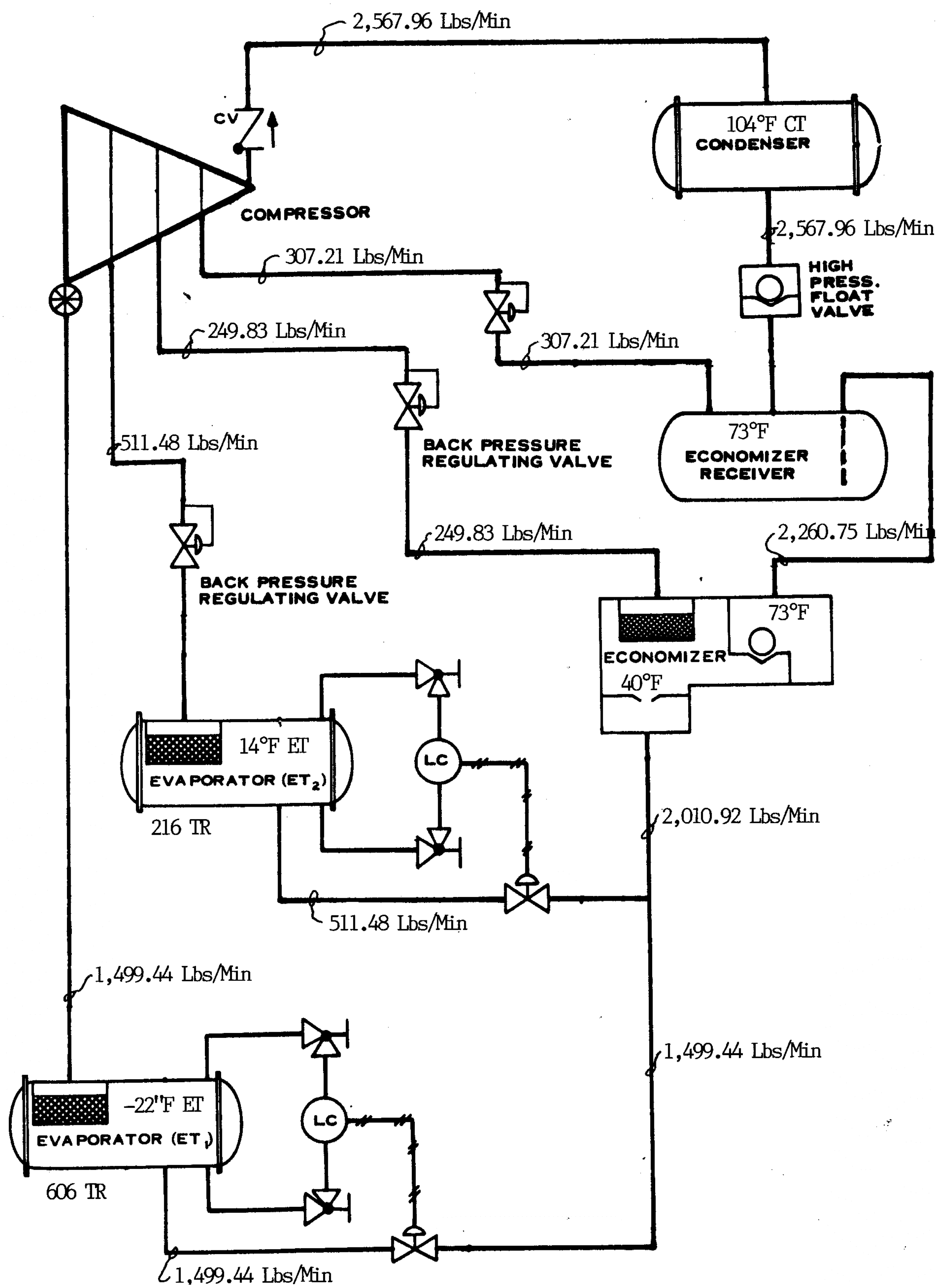


Figure 12-9 System Refrigerant Flow Diagram

Step No. 7 – Compressor Calculation:

Capacity Factor = Q/ND^3

$$Q/ND^3 = \frac{7.54 \text{ x CFM}}{T_s \text{ x } D^2}$$

Try M426A compressor. (4-stage, 26A casing)

From the compressor data sheet Table 12-1.

$D^2 = 219$

$T_s = 588.46$ ft/sec.
CFM = 3,503.44

$$Q/ND^3 = \frac{7.54 \text{ x } 3{,}503.44}{588.46 \text{ x } 219} = 0.205 \qquad \text{Ok.}$$

Therefore, M426A compressor is selected .

Notes:

(a) The compressor size is determined by the suction ACFM and the tip speed of the compressor.

(b) Overall compression ratio of the machine must be used for determining the efficiency of the compressor. Wheel by wheel analysis method is not allowed.

(c) The capacity limitation must be check if the capacity factor Q/ND^3 is over 0.20.

(d) The efficiency of the compressor is generally fixed by the size of the first stage impeller.

Step No. 8 – Determine the compressor efficiency:

Compression Ratio:

$$CR = \frac{P_2}{P_1} = \frac{231.0}{22.5} = 10.27$$

This is Halocarbon application, therefore, Adiabatic Rating curve should be used.

From the compressor performance map Figure 12-1,

Capacity Factor $Q/ND^3 = 0.205$ & $CR = 10.27$ $\eta_{ad} = 73.0\%$

Check capacity limitation, maximum Q/ND^3 allowed is 0.215 for Mach number 1.1, therefore, the selection of M425A is Ok.

The efficiency multipliers:	Compressor size:	1.0
	Mach Number:	1.0

Corrected $\eta_{ad} = 73.0\% \times 1.0 \times 1.0 = 73.0\%$

Step No. 9 - Horse Power calculation:

$$GHP = \frac{W \times H_{ad}}{33000 \times \eta_{ad}}$$

$$= \frac{(1499.44 \times 20648) + (511.48 \times 15486) + (249.83 \times 10324) + (307.21 \times 5162)}{33000 \times 0.73}$$

$$= 1{,}786.9 \text{ HP}$$

Step No. 10 - Determine the Compressor Speed:

$$Rpm = \frac{229 \times T_s}{D}$$

$T_s = 588.46$ ft/sec.

26A compressor, D = 14.8” (Table 12-1)

$$\text{Rpm} = \frac{229 \times 588.46}{14.8} = 9{,}105 \text{ rpm}$$

<u>Step No. 11</u> – Friction HP, using Tilting Pad thrust bearing:

FHP = 17.0 HP for 26A compressor at 9,105 Rpm speed.

<u>Step No. 12</u> – Compressor Shaft horsepower:

$$\begin{aligned} \text{SHP} &= (\text{GHP} + \text{FHP}) \times 1.03 \\ &= (1{,}786.9 + 17.0) \times 1.03 \\ &= 1{,}858 \text{ HP} \end{aligned}$$

Notes:

(a) 3% safety factor for proposal use only. The final SHP is to be verified and confirmed by the Turbo Design group.

(b) Addition factor is to be applied if no (-) tolerance is allowed.

<u>Step No. 13</u> – Motor driving HP:

M compressor always requires external speed increaser (external gear). Gear loss is about 3% of the driving HP. Therefore, the motor driving HP is 1,858 x 1.03 = 1,913 HP

<u>Step No. 14</u> – Coupling Size:

At 9,105 Rpm and 1,858 BHP
Coupling size required is 1-7/8” ϕ

<u>Step No. 15</u> – Compressor Casing:

Compressor maximum discharge pressure is 209 Psig. Therefore, Cast Iron casing is satisfactory with 300 Psig DWP.

Step No. 16 – Coupling Maximum HP:

$$\text{Coupling Max. HP} = 356 \times \frac{9{,}105}{1{,}000} = 3{,}241 \text{ HP} \quad \text{Ok.}$$

Step No. 17 – Impeller Fastening:

$$\text{Impeller Fastening HP} = 120 \times \frac{9{,}105}{1{,}000} = 1{,}092.6 \text{ HP}$$

The 4th impeller carry highest BHP:

$$= \frac{2{,}567.96 \times 5{,}162}{33{,}000 \times 0.73} \; 550.26 \text{ BHP} < 1{,}092.6 \text{ HP} \quad \text{Ok.}$$

Step No. 18 – Complete the P-H Diagram.

See the attached P-H with all the points calculated.

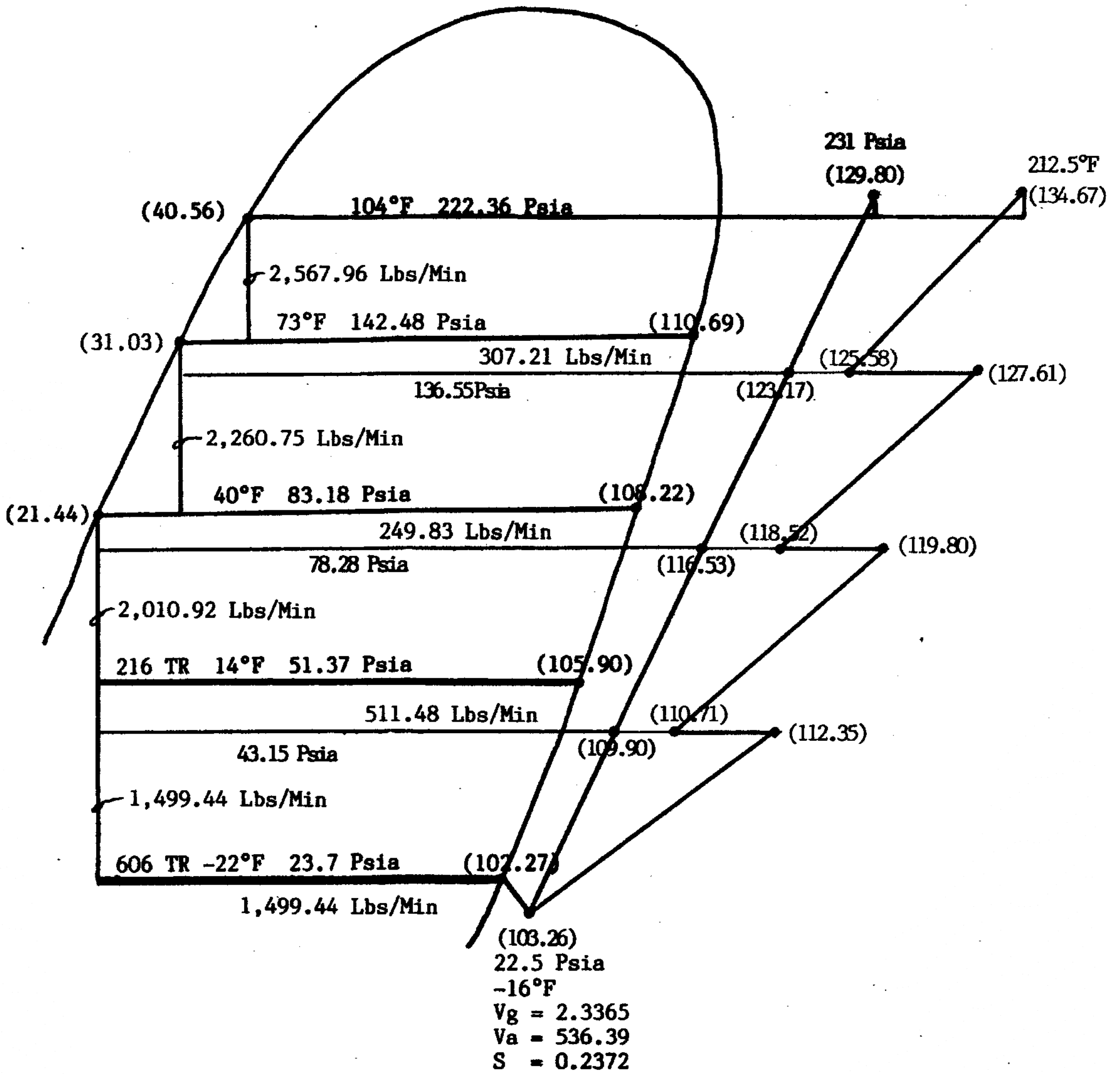


Figure 12-10 Final System P-H Diagram

<u>Step No. 19</u> – Compressor Discharge Temperature:

Compressor discharge temperature is calculated at 212.5°F

Compressor discharge with PRV closed $= -16 + \{212.5 - (-16)\} \times 1.3 = 279.75°F$
$\cong 280°F$

Step No. 20 – Oil Cooler:

Oil cooler = FHP + F x (Tdisch. – 275)

TDisch. = 280°F

Oil Cooling = 17 + 0.08 x (280 – 275) = 17.4 HP

Therefore, Oil cooling = 18 FHP

Step No. 21 – Impeller Material:

T_s = 588.46 ft/sec.

Maximum temperature for Aluminum impeller allowed is 388°F from Data Sheet.
Discharge temperature is 280 < Max. temperature 388°F allowed.
Therefore, all Aluminum wheels are Ok.

Step No. 22 – Last Wheel Q/ND^3:

Last wheel flow = 2,567.96 Lbs/Min
Last wheel inlet Vg = 0.5014

Last wheel ACFM = 2,567.96 x 0.5014 = 1,287.5 CFM

T_s = 588.46 ft/sec.

$$Q/ND^3 = \frac{7.54 \times 1{,}287.5}{588.46 \times 219} = 0.075$$

Last wheel $Q/ND^3 = 0.075 >$ Minimum 0.02 Limit, Ok

Step No. 23 – Check Critical Speed:

See Table 12-4, the first critical speed of all Aluminum wheel of M426A compressor is 18,400 RPM. The compressor speed is 9,105 RPM. The critical speed is above the operation speed and it should be at least 20% above the operating speed.

1.2 x 9,105 = 10,926 RPM, the critical speed > 10,926 rpm, above 20% of the operating speed. Ok.

Step No. 24 – Check Compressor Suction Pressure Drop:

M426A suction connection is 10"

$$FPS = \frac{CFM}{60 \times FT^2} \qquad CFM = 3,503.44 \quad FT^2 = 0.548$$

$$= 106.55$$

$$FVH = \frac{(FPS)^2 \times k}{64.4} \qquad k = 1.5 \text{ for suction}$$

$$= 264.44$$

$$PD\ Psi = \frac{FVH}{144 \times V_g} \qquad V_g = 2.3365$$

$$= 0.79\ Psi$$

Suction Inlet PD = 0.79 Psi < 0.9 Psi assumed, Ok.

Step No. 25 – Check Compressor Discharge Pressure Drop.

$$\Delta P = \frac{W^2 \times V_g}{C} + 0.25$$

W = Compressor discharge flow, Lbs/Min
= 2,567.96

V_g = Specific volume of the gas, Ft^3/Lb
= 0.3196

C = 309,000 for M26

$$\Delta P = \frac{(2{,}567.96)^2 \times 0.3196}{309{,}000} + 0.25$$

$$= 7.07 \text{ Psi}$$

Discharge PD = 7.07 Psi < 8.0 Psi assumed. Ok.

<u>Step No. 25</u> – Check Side Load Connection:

The side load of 14°F is the largest flow of all the side connections.

Flow = 511.48 Lbs/Min. Vg = 1.047, CFM = 536

Try two (2) standard size side connection 3”ϕ, FT^2 = 0.0513 x 2 = 0.1026

$$FPS = \frac{CFM}{60 \times FT^2} \qquad CFM = 536 \text{ cfm}, \quad FT^2 = 0.1026$$

$$= 87.1$$

Inlet velocity = 87.1 FPS<1/3(Ts) = (1/3) x (588.46) = 196 fps, Ok.

$$FVH = \frac{(FPS)^2 \times k}{64.4} \qquad k = 3.0 \text{ for suction}$$

$$= 353.4$$

$$PD\ Psi = \frac{FVH}{144 \times V_g} \qquad V_g = 1.047$$

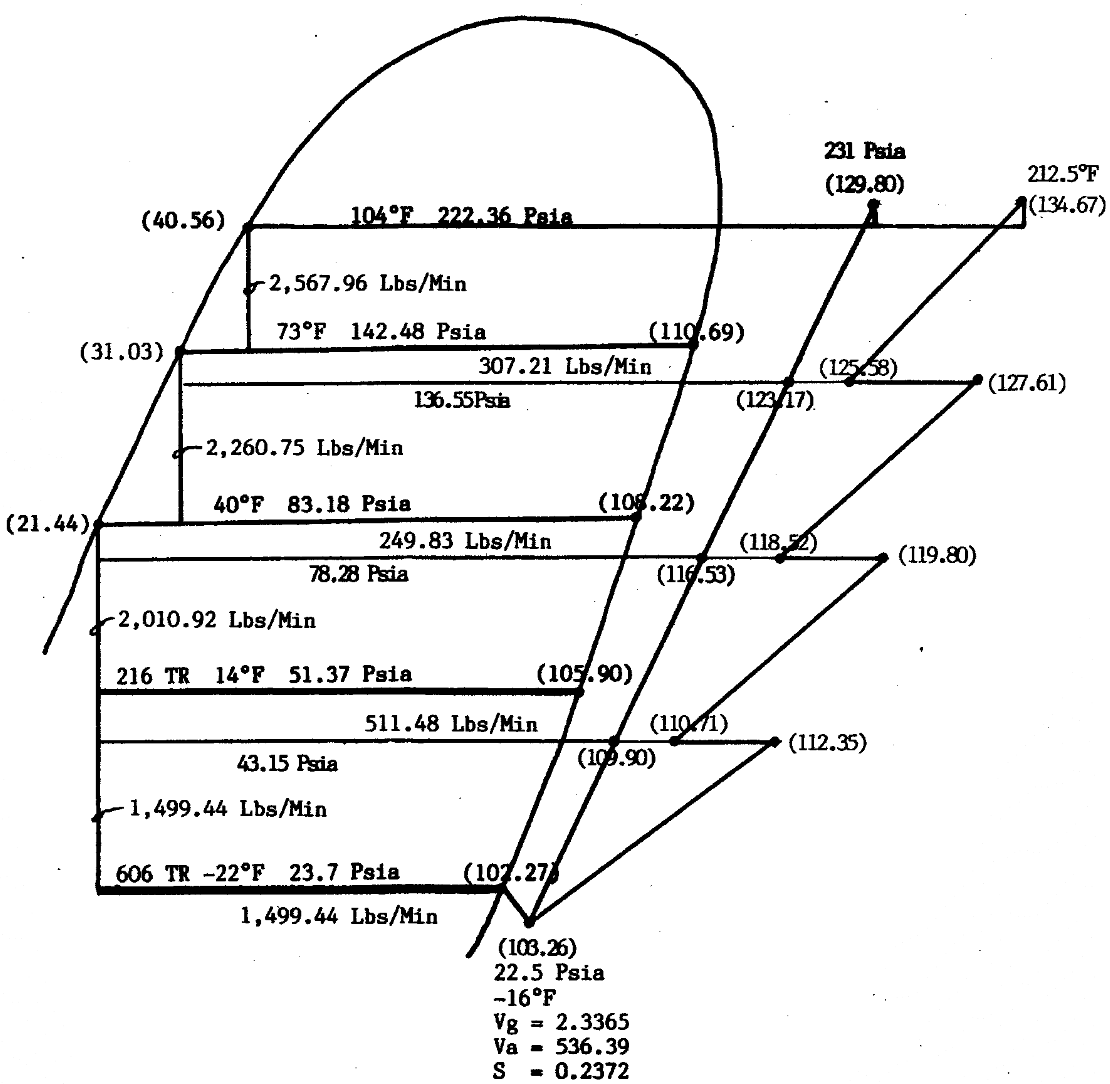
$$= 2.34 \text{ Psi}$$

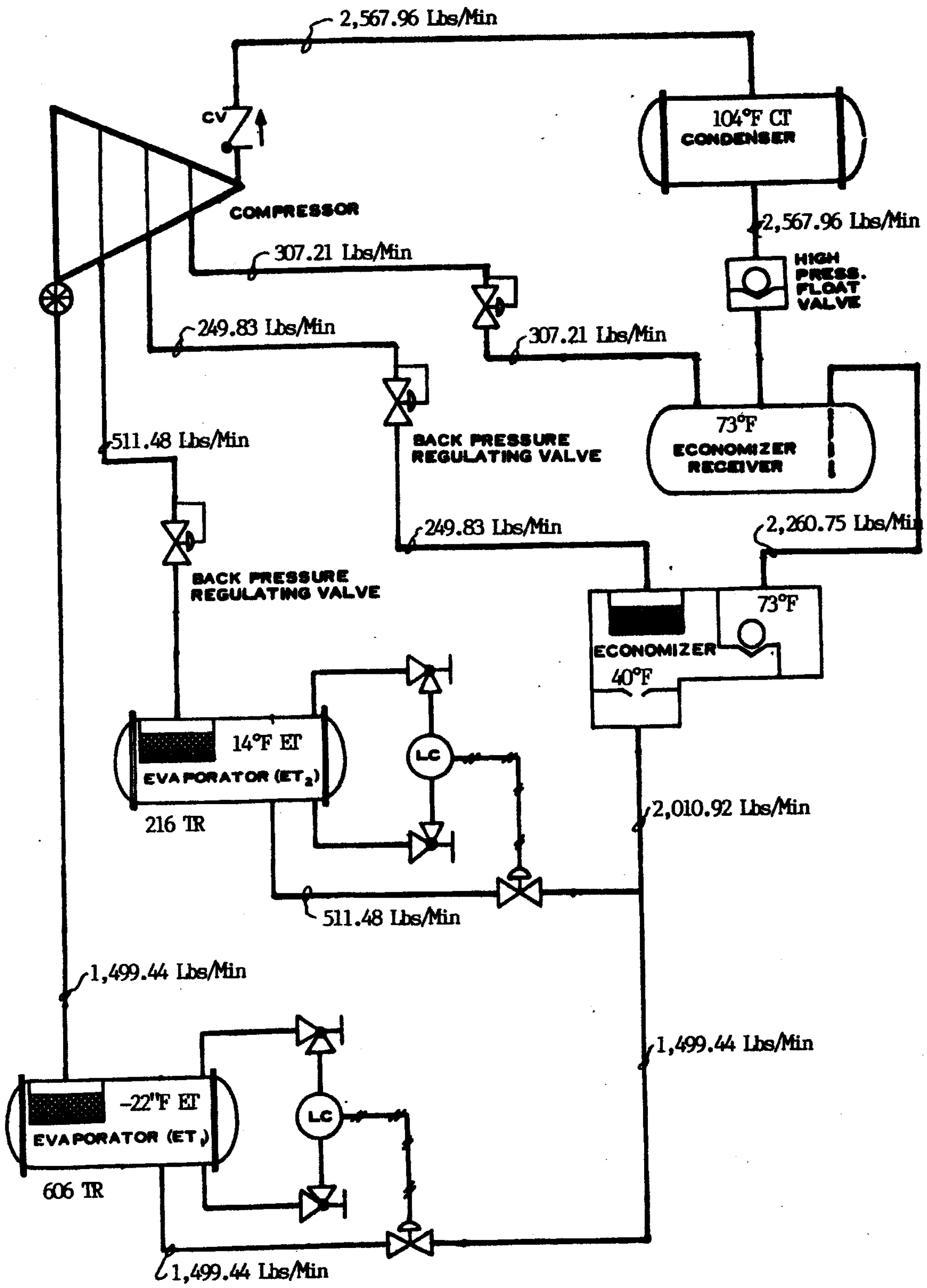
Side load connection PD = 2.34 Psi < 3.92 Psi allowed. Ok.

Conclusion:

Compressor:	M426A
Compressor Casing:	Cast Iron
DWP, Casing:	300 Psig
Shaft HP:	1,858 BHP
Compressor Speed:	9,105 rpm
Compressor Coupling:	1-7/8"ϕ
Oil Cooler:	18 HP
Driving HP:	1,913 BHP with estimated gear loss of 3%.

The system P-H diagram and the refrigerant flow diagram are shown below:




2,567.96 Lbs/Min
CV
104°F CT
CONDENSER
COMPRESSOR
2,567.96 Lbs/Min
307.21 Lbs/Min
HIGH PRESS. FLOAT VALVE
249.83 Lbs/Min
307.21 Lbs/Min
73°F
ECONOMIZER RECEIVER
511.48 Lbs/Min
BACK PRESSURE REGULATING VALVE
249.83 Lbs/Min
2,260.75 Lbs/Min
BACK PRESSURE REGULATING VALVE
73°F
ECONOMIZER
40°F
14°F ET
EVAPORATOR (ET2)
216 TR
LC
2,010.92 Lbs/Min
511.48 Lbs/Min
1,499.44 Lbs/Min
1,499.44 Lbs/Min
-22°F ET
EVAPORATOR (ET1)
LC
606 TR
1,499.44 Lbs/Min