University of Calgary

Natural Gas Processing Principles and Technology - Part II

April 2004

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Purpose of this Document

1. To study the physical, chemical and engineering principles used in the handling of natural gas, substitute natural gas, their associated liquids, and the other compounds obtained from them.
2. To apply these principles to actual design and operating problems.

Preface

The gas processing industry in the past 25 years has developed extensively. Many different qualities of gas have been discovered and are being processed. Many different processing schemes are used to accomplish the same ends. This book tries to outline the major ways that this is being done, and to make the reader aware of the main areas of technology that are considered in gas processing and handling.

This book is intended for practising engineers of all disciplines as a guide for them in the gas processing industry as gas engineers come from all branches of engineering.

At the end of most chapters, typical problems that might be encountered in industry are given. As these are practical problems, more than the principles of the particular chapter have to be used to solve the problems.

The Gas Processors Suppliers Association Engineering Data Book1 has been referred to extensively for data and other material as it is the basic reference book for the industry.

This book has been developed from the lecture notes developed by the author for a course given by him at the University of Calgary in the Department of Chemical Engineering over the past 24 years, and at other centres in both North America and Europe. During that period, several people have lectured in various sections of the course and have helped in the preparation of the text. These are, with the address when they participated in the course or their present address are as follows:

Dr. N. Anderson, Kilborn Engineering, Toronto, Ontario

Dr. K. Aziz, Stanford University, Palo Alto, California

Mr. E. Berlie, Western Research & Development, Calgary, Alberta

Dr. A. Chakma, University of Calgary, Calgary, Alberta

Dr. P.R. Bishnoi, University of Calgary, Calgary, Alberta

Dr. K. Goddard, Chevron Standard, Calgary, Alberta

Dr. G.A. Gregory, Neotechnology Consultants Ltd., Calgary, Alberta

Mr. G.E. Handwerk, Consultant, Denver, Colorado

Dr. R.A. Heidemann, University of Calgary, Calgary, Alberta

Mr. G. Kinsep, SKM Consulting, Calgary, Alberta

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[1] Published by the Gas Processors Suppliers Association and the Gas Processors Association, Tulsa, Oklahoma. Latest edition is the 10th edition which is published in English Units only. Page No.'s in the text refer to the Metric edition or the 9th English edition unless otherwise noted.

Mr. M.A. Kyllo, Ralph M. Parsons Company Ltd., London, England

Dr. D. Leahey, Western Research, Calgary, Alberta

Mr. J. Lore, Lore & Associates, Calgary, Alberta

Mr. D. Mcoy, Canterra Energy, Calgary, Alberta

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Mr. J. Same, Western Research, Calgary, Alberta

Dr. W.Y. Svrcek, University of Calgary, Calgary, Alberta

Dr. E.L. Tollefson, University of Calgary, Calgary, Alberta

Mr. E. Wichert, Petro Canada, Calgary, Alberta

Also, as the notes are based on the author's more than 30 years in the gas processing industry, he wishes to acknowledge the many persons not already acknowledged who contributed data and suggestions incorporate in the book.

The more recent revisions have been made to bring the notes up—to—date and to show both metric units and English units whenever appropriate. The problem of using metric units is that the source of much of the data is the 9th edition of GPSA Metric Data book, which is based on 1972 data. There is now available a 10th edition of this book which was published in 1987 that gives English units only. This is a much more complete reference book and as such will be used extensively in class.

10.0 Compression and Refrigeration

- A review of the theory of compression and refrigeration operations.
- The sizing equipment for compression and refrigeration operations.

10.1 Compression

10.1.1 Types

There are several types of compressors. Table 10.1 gives the main types and their general application; ie. reciprocating, rotary centrifugal or, axial flow types. The types and application are shown on Figure 10.2.

10.1.2 Determining Discharge Pressure

This is done as outlined in Chapter 6, Fluid Flow. A diagram of the system is drawn and the total pressure at the discharge of compressor is worked out by calculating the pressure drop in the discharge system of the compressor. The following example shows how this is done.

Example Problem 10.1

Determine the discharge pressure required and the compression ratio for the compressor shown in Figure 10.1.

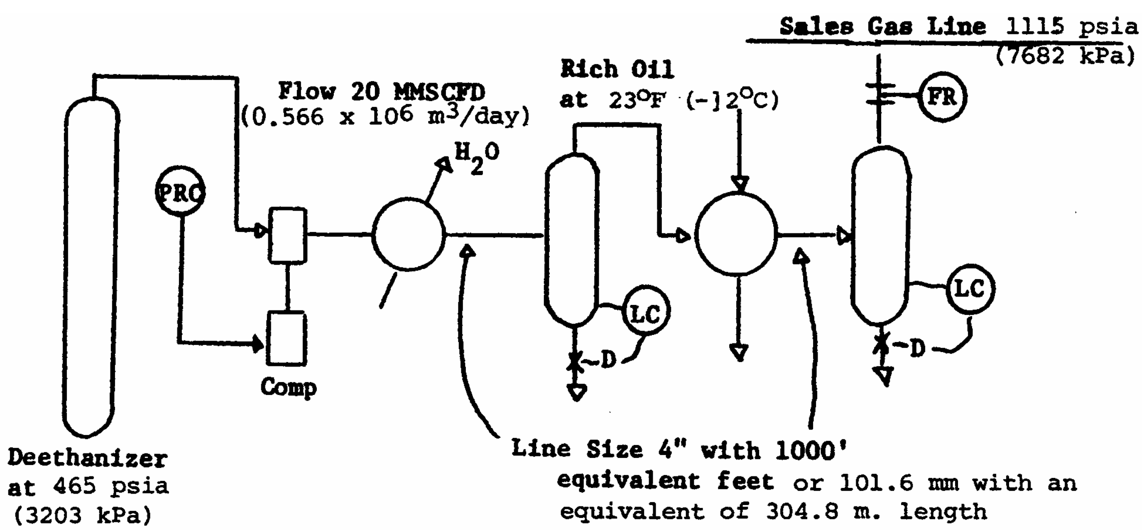


Figure 10.1: Compressor Discharge System

Table 10.1: Compressor Types and Ranges

Type	Maximum Pressure Range PSIG	Approx. Maximum Compression Ratio/Stage	Maximum Compression Ratio	Application	Example In Gas Processing Industry
Reciprocating	30,000 - 50,000	10	As required	All flows & pressures	Reinjection pipeline. Stations

					gathering systems. Small refrigeration loads.
Centrifugal	3,000 - 5,000	3	6	High Flow	Large refrigeration units. Pipe-Trans. Comp.
Rotary Displacement	130	4	4	High Flow Low Press.	Stock Recovery Units
Axial	3,000 - 5,000	1.5	5.65	High Flow Low Press. Differential	Sulphur Plant Blowers. Very large HP in LNG plants
Rotary Screw	800	10	As required	Constant flow medium quantity	Propane Refrigeration Instrument Air

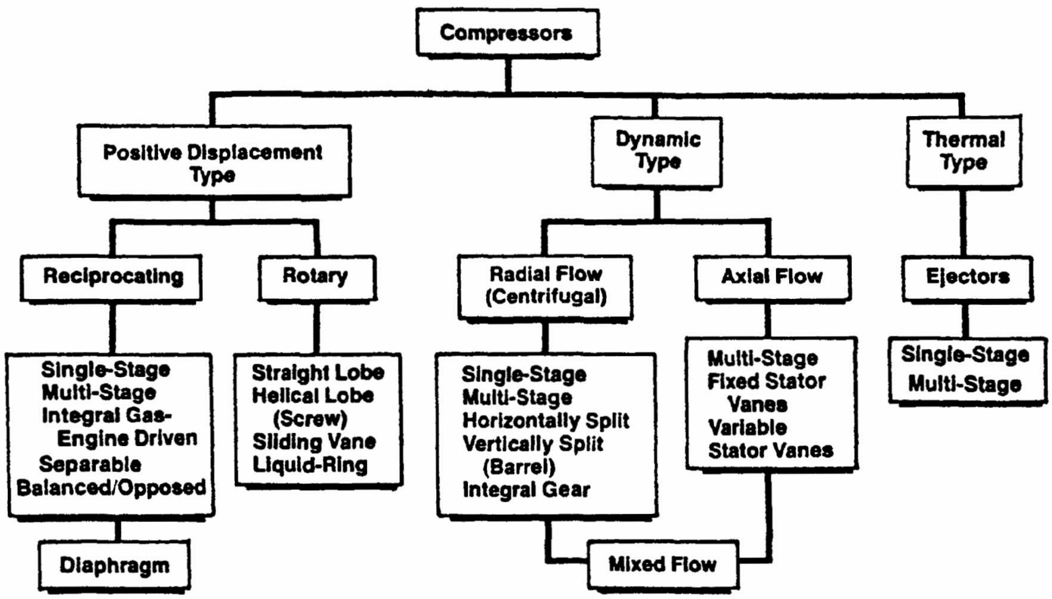


Figure 10.2: Types of Compressors

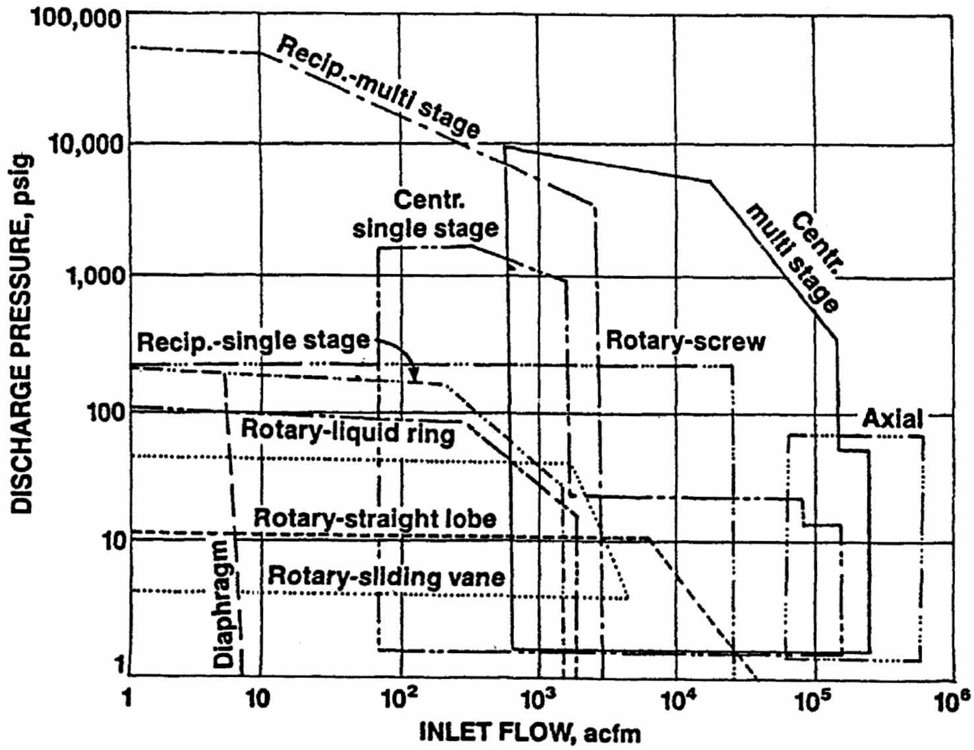


Figure 10.3: Compressor Coverage Chart

	English Units	Metric Units
	PSIA	**kPa**
Suction Pressure	465	3203
Final Pressure	1115	7682
Pressure drop through exchangers (10 psi/exchanger)	20	128
Flow Recorder	3	21
Piping Losses	30	207
Safety Factor 32 of Difference between Suction S Discharge Pressure .03 (700)	21	145
Total Discharge Pressure	1189	8193
Compression Ratio	$\frac{1189}{465} = 2.56$	$\frac{8193}{3203} = 2.56$

10.1.3 Work of Compression

The work required to compress a gas from Condition 1 to Condition 2 can be found from the energy equation which was given in Fluid Flow.

Equation 10-1

$$U_2 + \frac{v_2^2}{2g_C} + Z_2 + P_2V_2 = U_1 + \frac{v_1^2}{2g_C} + Z_1 + P_1V_1 + Q + W$$

or

Equation 10-2

$$\Delta U + \frac{\Delta\left(v_2^2 - v_1^2\right)}{2g_C} + \Delta Z + \Delta PV = Q + W$$

where:

U = the internal energy.

v = the velocity.

Z = the height or potential.

P = the pressure.

W = the work.

Q = the heat added to the system.

V = the volume.

For normal calculations the changes in potential, kinetic energy, etc., are neglected. We then get two cases. These are both based on integrating the change of P and V between points 1 and 2.

Equation 10-3

$$W = \int_{P}^{P_2} VdP$$

Isothermal Compression (at constant temperature)

The ideal gas law equation applies. The work required in terms of HP:

Equation 10-4

$$HP = 0.004363 P_1 V_1 \ln\left(\frac{P_2}{P_1}\right)$$

where:

P_1 = the intake condition.

P_2 = the final conditions.

V_1 = the intake volume in CF/min (actual CFM).

P = the inlet pressure in PSIA.

In terms of thermodynamic expressions, for isothermal compression we have the theoretical work expressed as:

Equation 10-5

$$Work = \Delta H - T\Delta S$$

or for 1 MMSCFD of gas

Equation 10-6

$$HP = 0.0432(\Delta H - T\Delta S)$$

where:

ΔH = the enthalpy change in BTU's/lb mole.

ΔS = the entropy change in BTU's/lb mole, ºR.

Adiabatic Compression

No outside heat enters or leaves. In this case gas obeys the expressions PVK = constant. For this the theoretical HP can be expressed by:

Equation 10-7

$$HP = 0.004363 P_1 V_1 \left[\left(\frac{K}{K-1} \right) \left(\frac{P_2}{P_1} \right)^{K-1/K} - 1 \right]$$

where:

V_1 = cu. ft/min at intake conditions (actual CFM).

K = the ratio of specific heats $\frac{C_P}{C_V}$.

P_1 = the suction pressure psia.

P_2 = the discharge pressure psia.

Most compressors operate close to the adiabatic case. For adiabatic compression the work is expressed by:

Equation 10-8

$$Work = \Delta H$$

where:

ΔS = 0, ie. constant entropy expression.

Therefore for 1/MMSCFD:

Equation 10-9

$$HP = 0.0432 \Delta H$$

where:

ΔH is in BTU/lb mole.

Calculating the HP Required

As a quick approximation, use for reciprocating compressors.

$$HP = 22 \left(\frac{Compression\ Ratio}{Stage} \right) (No.\, of\ \ Stages)(F)\, MMCFD$$

or in metric units

$$W = 14\left(\frac{Compression\ Ratio}{Stage}\right)(No.\, of\ \ Stages)(F)\, m^3/hr$$

where:

F = 1.00 for 1 stage.

1.08 for 2 stages.

1.10 for 3 stages.

W = watts.

See pages 13.4 (4-2) and Fig. 13.9 (4.4) of the GPSA Data Book for further details.

Note: Figure no's and pages in booklets are from Metric Data Book. Others are from 1987 English Data Book.

For a more accurate means of calculating the HP charts are given in the GPSA Data Book plotting BHP vs ratio of specific heat (K) and the compressor ratio. The mechanical efficiency is allowed for in the charts. However corrections have to be made for specific gravity, which is given in Fig. 13-15 (4-12) in the GPSA Data Book and correction factor for low intake pressure is given by Fig. 13-12 (4-11). Also other corrections have to be made as shown below.

The horsepower for compression is found from:

Equation 10-10

$$BHP = BHP \Big/ MMSCFD\left(\frac{P_L}{14.4}\right)\left(\frac{T_S}{T_L}\right)(Z_{ave}) \qquad (Flow\ in\ MMSCFD)$$

or

$$W = \frac{W}{m^3/hr}\left(\frac{P_L}{101.325}\right)\left(\frac{T_S}{T_L}\right)(Z_{ave}) \qquad (Flow\ in\ m^3/hr)$$

where:

BHP/MMSCFD or $\frac{W}{m^3/hr}$ is found from charts (4-7) through (4-10) and corrected if necessary by charts 13-12 (4-11) and 13-15 (4-12).

P_L = Pressure base psia or kPa.

T_S = Intake temperature ºR or K.

T_L = Temperature base ºR or K.

MMSCFD = Flow at standard conditions of the base given by $P_L + T_L$.

(m^3 /hr) = Flow at standard conditions of the base given by $P_L + T_L$.

Z_{ave} = Average compressibility factor in the cylinder $\frac{Z_{suction} + Z_{discharge}}{2}$.

Please note this data is for slow speed units. For high speed units make the corrections shown on Page 4.13 of the GPSA Data Book.

10.1.4 Compression Ratio

$$\frac{Discharge\ Absolute\ Pressure}{Suction\ Absolute\ Pressure}$$

Normally do not use a compression ratio of greater than 6 for single stage compression. Generally it is around 3.

10.1.5 Ratio of Specific Heats

In the equation for adiabatic compression the term K is used. It is the ratio of the specific heat at constant pressure (Cp) and specific heat at constant volume (CV). It can be calculated if the value of the specific heat at constant pressure is known from the equation

Equation 10-11

$$MC_P - MC_V = 1.99$$

where:

M = molecular weight.

Therefore

Equation 10-12

$$K = \frac{MC_P}{MC_P - 1.99}$$

in metric units

$$K = \frac{MC_P}{MC_P - 8.3145}$$

The GPSA Data Book, Fig. 4-1, gives the value of MC for most compounds P expected in natural gas at various temperatures so that a K for a mixture can be easily calculated. Some common values of K are:

for Air 1.4

for Natural Gas 1.26

For a quick calculation of "K" for hydrocarbon gases, use Fig. 4-3 of the GPSA Data Book.

The value of K is one of the most important values in determining the compression power required. The following example problem illustrates the calculations required.

Example Problem 10.2

Calculate the HP required to compress 80 MMSCFD of gas of the composition given below from 400 psia at 100ºF to 1100 psia.

First we have to calculate the ratio of specific heat $"K" = \frac{C_P}{C_V}$.

This is found from the molal heat capacity for the mixture.

Component	Mole %	Mole* Heat Cap	Partial Heat Cap	MW	pMW
H_2S	2.4	8.27	0.20	34	0.82
CO_2	4.4	9.28	0.41	44	1.93
C_1	85.4	8.95	77.64	16	13.66
C_2	4.5	13.77	0.62	31	1.35
C_3	1.6	19.52	0.31	44	0.70
iC_4	0.4	25.77	0.10	58	0.23
nC_4	0.7	25.81	0.18	58	0.40
iC_5	0.6	31.66	0.19	72	0.43
			9.65		19.52

* At 150ºF average temperature approximately the average temperature in the compressor. Calculate the discharge temperature using the graph, Fig. 4-6 of the GPSA Data Book and find the arithmetic average between suction and discharge conditions.

You Can also use chart in GPSA Data Book for this calculation.

To find the compression HP, use the graph in the GPSA Data Book and find that the HP per MMCF at a compression ratio of

$\frac{1100}{400} = 2.75$ and a "K" of 1.26 is 59 or in metric units 38.8 W/m3/hr.

$$K = \frac{9.65}{9.65 - 1.99} = 1.26$$

Corrections for intake pressure and for specific gravity are both negligible and multiplying factors are 1.0.

$$\frac{T_S}{T_L} = \frac{560}{520} = 1.076$$

Now find Z_{ave}

$$
\begin{aligned}
& Z_{400} PSIA = .93 \\
& \alpha 100^\circ F \\
& Z_{1100 PSIA \alpha 226^\circ F} = .94 \\
& Z_{ave} = .935 \\
& BHP = 59(1.076)(.935)80 \\
& \quad\quad = 4748
\end{aligned}
$$

By the Cooper Bessemer slide rule the HP required works out to 56/MMSCF or 4500 HP, which is a good check.

10.1.6 Reciprocating Compressor Capacity

The capacity of a reciprocating compressor is a function of the suction pressure, the piston displacement and the strokes per minute. Figure 10.4 below illustrates a problem that occurs.

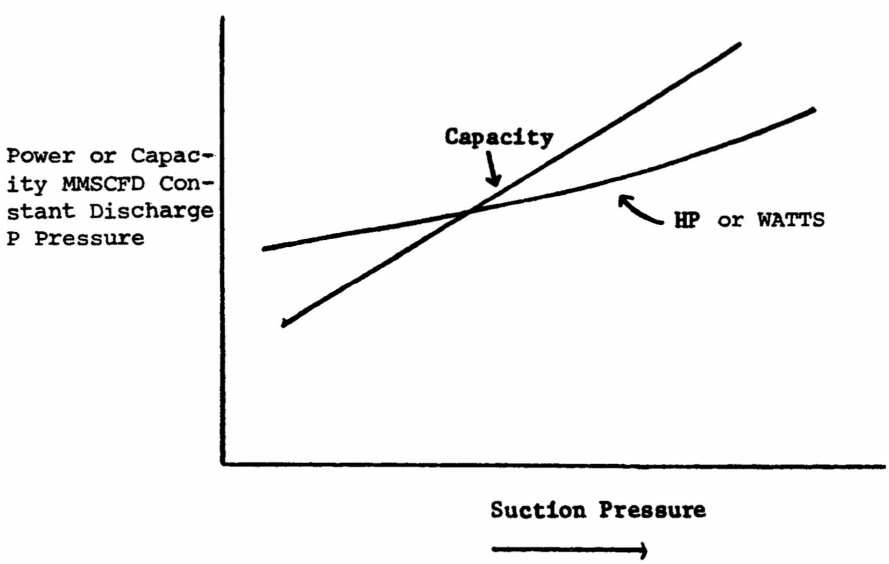


Figure 10.4: Capacity and Power of Reciprocating Compressors

As the suction pressure is dropped the capacity of the machine as represented by the Stroke-Piston displacement and RPM decreases faster than the HP required for the compression ratio, and the volume that can be pumped. The machine does not have enough HP and the compressor stalls or volumetric changes to reduce the capacity have to be made. (i.e., clearance pockets).

The flow as mentioned is a function of Piston Displacement. For a single acting compressor:

Equation 10-13

$$P_D = \frac{A_{HE}(S)(RPM)}{1728}$$

or

$$P_D = \frac{A_{HE}(S)(RPM)(60)}{4\,x\,10^9}$$

and for a double acting compressor:

Equation 10-14

$$P_{DA} = \frac{(S)(RPM)(2)}{1728}\left[A_{HE} - \frac{A_R}{2}\right] = 4.55\,x\,10^{-4}(S)(RPM)(2D^2 - d^2)$$

or

$$P_{DA} = \frac{(S)(RPM)\,2\left(A_{HE} - {}^{A_R}\!/_{2}\right)}{4\,x\,10^9}$$

where

P_D = Piston Displacement in CFM (single acting) or m^3/hr.

P_{DA} = Piston Displacement in CFM (double acting) or m^3/hr.

S = Stroke in inches or mm.

RPM = Revolution per minute.

A_{HE} = Area head end in inches^2 or mm^2.

A_R = Area rod in inches^2 or mm^2.

ACE = Area of crank end ACE = A_{HE} - A_R in inches^2 or mm^2.

D = cylinder diameter in inches^2 or mm^2.

d = piston rod diameter in inches^2 or mm^2.

Now the capacity of a compressor is found in SCFD per day from

Equation 10-15

$$Capacity = P_D\left(E_V\right)\left(\frac{P_1}{14.4}\right)(1440)\left(\frac{520}{T_S}\right)\left(\frac{1}{Z}\right)$$

or simplifying for ordinary temperatures and low pressures in English Units.

Equation 10-16

$$Capacity = P_D E_V P_1(100)$$

in Metric Units

$$m^3 / hr = \frac{P_D E_V P_1}{(100)(101.325)Z_{ave}}$$

where

P_1 = the suction pressure.

E_V = the volumetric efficiency which is found from

Equation 10-17

$$E_V = 96 - R - \%Cl\left[\left(\frac{Z_S}{Z_d}\right)(R)^{1/K} - 1\right]$$

where

$$\%Cl = Clearance\ Volume$$

$$= \frac{V_C\ (clearance\ \ volume)}{P_D\ (displacement)}$$

$$R = compression\ ratio\ \frac{P_2}{P_1}$$

10.1.7 Rod Loading

The net or differential force in pounds exerted upon a piston rod due to gas pressure differences between head end of a piston and the rod is called the rod load. Generally it is calculated as follows:

Equation 10-18

$$Rod\ \ Load\ \ in\ compression = A_{HE}P_2 - A_{CE}P_1$$

The compressor manufacturer states the allowable rod loading for the machine and the calculated value should be well below this.

10.1.8 Specifying Reciprocating Compressors

There are several items that should be specified or obtained from the manufacturer when receiving a bid for the compressor. These include:

Table 10.2: Specifications for Reciprocating Compressors

	Specified By
The capacity required	Operator
The gas being compressed	Operator
The suction pressure	Operator
The suction temperature	Operator
The discharge pressure	Operator
The cylinder material	Operator or Manufacturer
The liner material	Operator or Manufacturer
Unloading Facilities or the method of flow control	Operator or Manufacturer
Lubrication of piston required	Operator
Rod Loading allowed	Manufacturer
Allowable vibration in piping	Operator
Driver type and speed	Operator
No. of stages	Manufacturer
Discharge temperature	Manufacturer
HP required for the service	Manufacturer
Intercoolers required and supplied	Manufacturer
Type of machine	Operator or Manufacturer
Allowable piston speed	Operator
Valve type and material	Manufacturer
Coupling to driver	Manufacturer
Area of installation and elevation	Operator
Cooling requirements	Manufacturer

Considerably more data has to be supplied if the compressor and driver are a single package. Also, it is generally desirable to specify a standard specification such as API 618 which covers reciprocating compressors.

10.1.9 Trouble Shooting Reciprocating Compressors

There are many operating problems that can occur with reciprocating compressors. Table 10.3 from the latest GPSA Data Book (English units version) outlines many of the problems and their causes.

Table 10.3: Probable Causes of Reciprocating Compressor Double

Trouble	Probable Causes
Compressor will not start	1. Power supply failure.
	2. Switchgear or starting panel.
	3. Low oil pressure shut down switch.
	4. Control panel.
Motor will not Synchronize	1. Low voltage.
	2. Excessive starting toque.
	3. Incorrect power factor.
	4. Excitation voltage failure.
Low oil pressure	1. Oil pump failure.
	2. Oil foaming from counter weights striking oil surface.
	3. Cold oil.
	4. Dirty oil filter.
	5. Interior frame oil leaks.
	6. Excessive leakage at bearing shim tabs and/or bearings.
	7. Improper low oil pressure switch setting.
	8. Low gear oil pump by-pass/relief valve setting.
	9. Defective pressure gauge.
	10. Plugged oil sump strainer.
	11. Defective oil relief valve.
Noise in cylinder	1. Loose piston.
	2. Piston hitting outer head or frame end of cylinder.
	3. Loose crosshead lock nut.
	4. Broken or leaking valve(s).
	5. Worn or broken piston rings or expanders.
	6. Valve improperly seated/damaged seat gasket.
	7. Free air unloader plunger chattering.
Excessive packing Leakage	1. Worn packing rings.
	2. Improper lube oil and/or insufficientlube rate (blue rings).
	3. Dirt in packing.
	4. Excessive rate of pressure increase.
	5. Packing rings assembled incorrectly.
	6. Improper ring side or end gap clearance.
	7. Plugged packing vent system.
	8. Scored piston rod.

Trouble	Probable Causes
Packing over heating	1. Lubrication failure.
	2. Improper lube oil and/or insufficient lube rate.
	3. Insufficient cooling.
Excessive carbon on valves	1. Excessive lube oil.
	2. Improper lube oil (too light, high carbon residue).
	3. Oil carryover from inlet system or previous stage.
	4. Broken or leaking valves causing high temperature.
	5. Excessive temperature due to high pressure ratio across cylinders.
Relief valve popping	1. Faulty relief valve.
	2. Leaking suction valves or rings on next higher stage.
	3. Obstruction (foreign material, rags), blind or valve closed in discharge line.
High discharge temperature	1. Excessive ratio on cylinder due to leaking inlet valves or rings on next higher stage.
	2. Fouled intercooler/piping.
	3. Leaking discharge valves or piston rings.
	4. High inlet temperature.
	5. Fouled water jackets on cylinder.
	6. Improper lube oil and or lube rate.
Frame knocks	1. Loose crosshead pin, pin caps or crosshead shoes.
	2. Loose/worn main, crankpin or crosshead bearings.
	3. Low oil pressure.
	4. Cold oil.
	5. Incorrect oil.
	6. Knock is actually from cylinder end.
Crankshaft oil seal leaks	1. Faulty seal installation.
	2. Clogged drain hole.
Piston rod oil scraper leaks	1. Worn scraper rings.
	2. Scrapers incorrectly assembled.
	3. Worn/scored rod.
	4. Improper fit of rings to rod/side clearance.

Courtesy of Ingersoll-Rand Co.

10.1.10 Centrifugal Compressors - HP or W Calculations

Centrifugal compressors are used for high flow and relatively low compression ratios. Figure 10.6 shows the different characteristics of centrifugal axial and reciprocating compressors. In considering compression for these machines, the compression law follows an equation similar to the reciprocating machines. To calculate the BHP the approximate equation can be used:

Equation 10-19

$$BHP/MMSCFD = 240.5\left[(R)^{\frac{K-1}{K}} - 1\right]$$

where

R = the compression ratio per unit or case.

This is assuming an efficiency of 73%. A more general equation is:

Equation 10-20

$$BHP = \frac{1545(Z_a)T_1 W}{33000\,E(MW)}\left(\frac{K}{K-1}\right)\left[(R)^{\frac{K-1}{K}} - 1\right]$$

where:

Za = average compressibility factor.

T_1 = suction temperature ºR.

W = gas flow rate lb/min.

E = overall efficiency.

MW = molecular weight.

In metric units the power in KW is given by:

$$Power = \frac{1000W\left(\frac{8.314}{M}\right)\left(\frac{K}{K-1}\right)(Z_{ave}T_1)\left[\left(\frac{P_2}{P_1}\right)^{\frac{K-1}{K}} - 1\right]}{3600000(E)}$$

where:

W = flow in kg/hr.

M = mole wt kg.

K = ratio of specific heats.

Z_{ave} = average compressibility factor.

P_2 = final pressure kpa.

P_1 = initial pressure kpa.

T_1 = initial temperature ºk.

E = efficiency.

10.1.11 Centrifugal Compressors - General Operating Characteristics

The centrifugal (and the axial compressor) are generally like a centrifugal pump. They develop pressure from rotating blades or impellers and they have a performance curve as shown in the figure below.

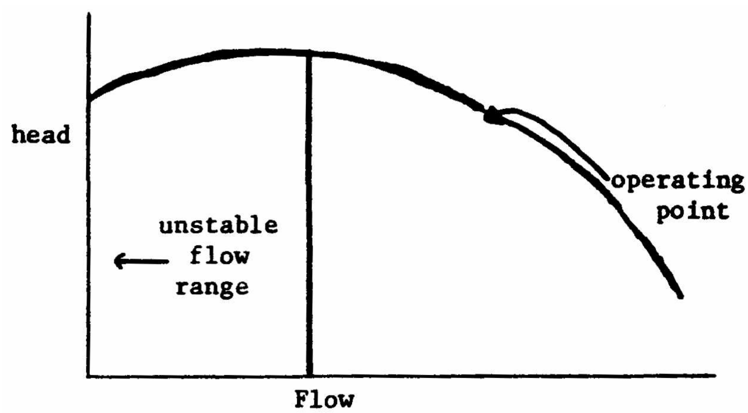


Figure 10.5: Head Capacity Curve for Centrifugal Compressors

The system and the compressor characteristics have to meet. However the system can change by changing pressures and of course the compressor curve can be changed in many ways such as changing the speed, etc. The head required by the system is calculated from the following equation:

Equation 10-21

$$H_{adiabatic} = ZRT\left(\frac{K}{K-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{\frac{K-1}{K}} - 1\right]$$

The compressor head developed is found from the equation

Equation 10-22

$$H_{adiabatic} = \psi \frac{u^2}{g}$$

where

ψ = head coefficient and is a function of blade angle, efficiency, etc. It is normally 0.5 for industrial compressors and goes to 0.63 with radial blading.

u = tip speed ft/sec.

g = gravity.

Each stage of a compressor normally runs between 8000 to 12,000 feet of adiabatic head.

Figure 10.7 is a typical performance map which shows the basic shape of performance curves for a variable speed centrifugal compressor. The curves are affected by many variables, such as desired compression ratio, type of gas, number of wheels, sizing of compressor, etc.

With variable speed, the centrifugal compressor can deliver constant capacity at variable pressure, variable capacity at constant pressure, or a combination variable capacity and variable pressure.

Basically the performance of the centrifugal compressor, at speeds other than design, is such that the capacity will vary directly as the speed, the head developed as the square of the speed, and the required horsepower as the cube of the speed. As the speed deviates from the design speed, the error of these rules, known as the affinity laws, or fan laws , increases.

Fan Laws

Equation 10-23

$$Q \ \alpha \ N; \ i.e. \frac{Q_{110}}{N_{110}} = \frac{Q_{100}}{N_{100}} = \frac{Q_{90}}{N_{90}}$$

$$H \ \alpha \ N^2; \ i.e. \frac{H_{110}}{(N_{110})^2} = \frac{H_{100}}{(N_{100})^2} = \frac{H_{90}}{(N_{90})^2}$$

$$Bhp \ \alpha \ N^3; \ i.e. \frac{Bhp_{110}}{(N_{110})^3} = \frac{Bhp_{100}}{(N_{100})^3} = \frac{Bhp_{90}}{(N_{90})^3}$$

By varying speed, the centrifugal compressor will meet any load and pressure condition demanded by the process system within the operating limits of the compressor and the driver. It normally accomplishes this as efficiently as possible, since only the head required by the process is developed by the compressor. This compares to the essentially constant head developed by the constant speed compressor.

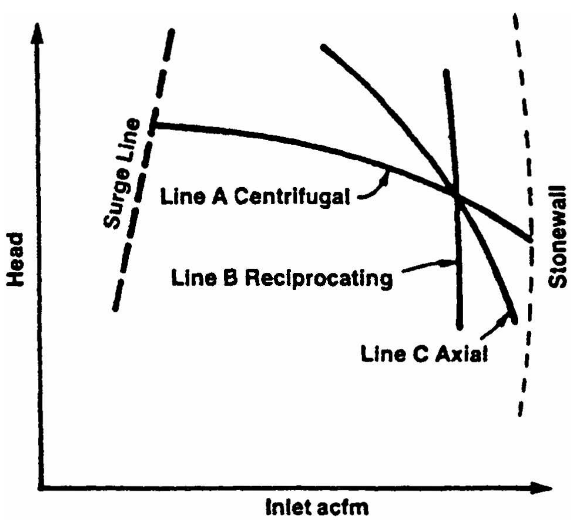


Figure 10.6: Compressor Head

10.1.12 Centrifugal Compressor Controls

Compress performance control may be effected by:

1. change of speed
2. throttling at the discharge
3. throttling at the suction
4. Guide Vane control
5. by pass control

Figure 10.5 shows a typical range of control and the effect of some of the above variables.

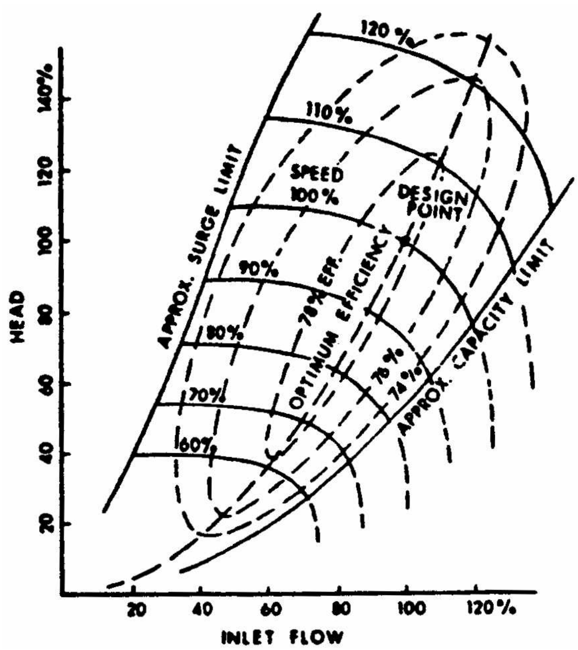


Figure 10.7: Typical Performance Map of Centrifugal Compressors

Speed changes the head of the compressor without much change in capacity. Discharge and suction throttling also changes the head but this time the system conditions.

Guide Vane control changes the shape of the performance curve and it can change significantly the surging condition.

By pass control is to prevent surging.

Surging is one of the most undesirable features of a centrifugal compressor. This occurs at a flow below which the compressor does not operate stably. The condition usually occurs when the flow is to the left of the maximum head point shown in Figure 10.7.

Figures 5.3 and 5.4 in the GPSA Data Book show how power and head are affected by rpm.

10.1.13 Specifying Centrifugal Compressors

There are also several items that should be specified for a centrifugal compressor. These include the following:

Table 10.4: Specification for Centrifugal Compressors

	Specified By
The maximum capacity required	Operator
The minimum capacity the machine is expected to operate at	Operator
The gas being compressed	Operator
The suction pressure	Operator
The discharge pressure	Operator
The driver type and speed	Operator
Operating curve showing head vs flow	Manufacturer
Number of stages	Manufacturer
Coupling to driver	Manufacturer
Compressor discharge temperature	Manufacturer
Case materials	Manufacturer
Rotating Element	Manufacturer
Control System	Operator

Considerably more data on the driver has to be supplied if the unit is to be a package of compressor and driver. Also the specification API 617, which covers centrifugal compressors, should be made part of the overall specifications.

10.1.14 Trouble Shooting Centrifugal Compressors

Operational troubles occurring in service may be due to a variety of causes.

If the trouble cannot be traced to adverse gas flow conditions or liquid "slugs" present in the system, Table 10.5 can be used as a guide for trouble shooting frequently encountered problems.

Careless operation and maintenance needs little comment. Lack of proper care of any machine is bound to result in a succession of minor troubles eventually leading to a major breakdown.

10.2 Expansion of Gases

There are many processes that use an expander as part of the process. The process is generally considered adiabatic and is the reverse of compression. The power generated is found from the equation:

Equation 10-24

$$Power = (H_1 - H_2)E$$

where

$H_1 - H_2$ = the enthalpy change of the stream assuming that the fluid is expanded from 1 to 2.

E = the efficiency which about 80 to 85% is usually run in the expander itself. Also then in the machine it is driving, such as a centrifugal compressor, efficiency must be taken into account.

The machines used are similar to steam or gas turbines but run at very high speeds. A good reference is an articles by J.S. Sweargin.

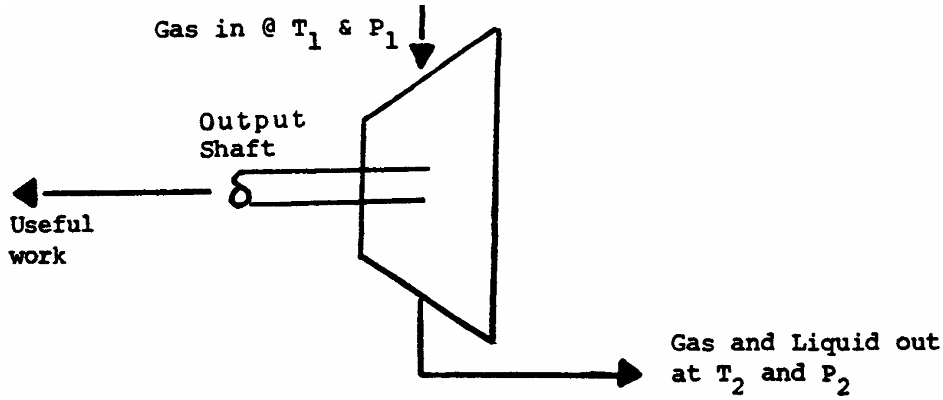


Figure 10.8: Simple Expander

Conditions for an expansion are generally set as follows: P_1 and T_1 are set by process conditions. P_2 and T_2 are set by conditions for liquid recovery and overall horsepower requirements. Generally P_2 is set and T_2 calculated and thus the expected liquid condensation and power out of the machine calculated.

Table 10.5: Probable Causes of Centrifugal Compressor Trouble

Trouble	Probable Causes
Low discharge pressure	1. Compressor not up to speed.
	2. Excessive compressor inlet temperature.
	3. Low inlet pressure.
	4. Leak in discharge piping.
	5. Excessive system demand from compressor.
Compressor surge	1. Inadequate flow through the compressor.
	2. Change in system resistance due to obstruction in the discharge piping or improper valve position.
	3. Deposit build-up on rotor or diffusers restricting gas flow.
Low lube oil pressure	1. Faulty lube oil pressure gage or switch.
	2. Low level in oil reservoir.
	3. Oil pump suction plugged.
	4. Leak in oil pump suction piping.
	5. Clogged oil strainers or filters.
	6. Failure of both main and auxiliary oil pumps
	7. Operation at a low speed without the auxiliary oil pump running (if main oil pump is shaft driven).
	8. Relief valve improperly set or stuck open.
	9. Leaks in the oil system.
	10. Incorrect pressure control valve setting operation.

Trouble	Probable Causes
High bearing oil temperature Note: Lube oil temperature leaving bearings should never be permitted to exceed 180°F.	1. Inadequate or restricted flow of lube oil to bearings.
	2. Poor conditions of lube oil or dirt or gummy deposits in bearings.
	3. Inadequate cooling water flow lube oil cooler.
	4. Fouled lube oil cooler.
	5. Wiped bearing.
	6. High oil viscosity.
	7. Excessive vibration.
	8. Water in lube oil.
	9. Rough journal surface.
Excessive vibration Note: Vibration may be transmitted from the coupled machine. To localize vibration, disconnect coupling and operate driver alone. This should help to indicate whether driver or driven machine is causing vibration.	1. Improperly assembled parts.
	2. Loose or broken bolting.
	3. Piping strajn.
	4. Shaft misalignment.
	5. Worn or damaged coupling.
	6. Dry coupling (if continuously lubricated type is used).
	7. Warped shaft caused by uneven heating or cooling.
	8. Damaged rotor or bent shaft.
	9. Unbalanced rotor or warped shaft due to severe rubbing.
	10. Uneven build-up of deposits on rotor wheels, causing unbalance.
	11. Excessive bearing clearance.
	12. Loose wheel(s) (rare case).
	13. Operating at or near critical

	11. Bearing lube oil orifices missing or plugged.
Shaft misalignment	1. Piping strain.
	2. Warped bedplate, compressor or driver.
	3. Warped foundation.
	4. Loose or broken foundation bolts.
	5. Defective grouting.

	speed.
	14. Operating in surge region.
	15. Liquid "slugs" striking wheels.
	16. Excessive vibration of adjacent machinery (sympathetic vibration).
Water in lube oil	1. Condensation in oil reservoir.
	2. Leak in lube oil cooler tubes or tube-sheet.

Courtesy of Ingersoll-Rand Co.

A very simplified method to give a order of magnitude of the power out of any expansion is to use the Mollier Diagram for methane and assume a constant entropy expansion from P_1T_1 to P_2 (and T_2). This will give you $H_1 - H_2$ in equation (24). Then assuming 80% efficiency, the approximate power available for driving the expander compressor is found.

A more accurate method is to find the amount of liquid that is condensed and thus the expected liquid recovery in a process, a trial and error type of calculation has to be done in which the outlet temperature of the expander (T_2) must be assumed if the outlet pressure is given.

Generally the method is as follows:

Given P_1, T_1 and P_2 assume 3 values for T_2 using values for T_2 less than that found from a methane Mollier diagram. At these three P_2T_2', P_2T_2'', and P_2T_2''' conditions, calculate the equilibrium conditions for liquid and vapour present, the enthalpy and entropy at each condition and plot the entropy and the change in enthalpy from 1. (i.e. $H_1 - H_2$ or ΔH). All as shown on the figure below.

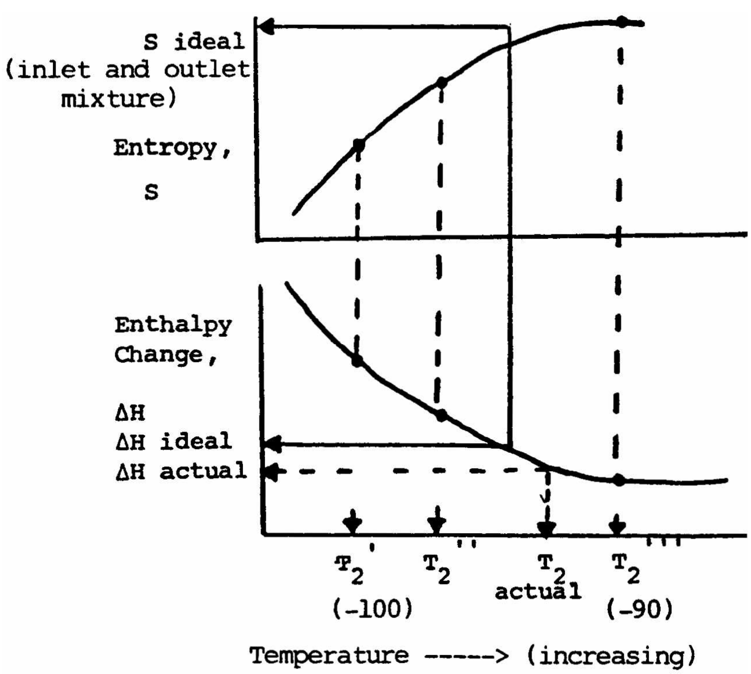


Figure 10.9: Entropy and Enthalpy Plot

Find entropy at the outlet conditions which is to equal the inlet condition entropy and then the corresponding H which is the ideal H. Assume an efficiency and calculate the actual H and thus find the actual T_2. Run a flash calculation at this temperature to see how much liquid condenses so as to have an estimate of the liquid recovery.

Pages 5-11, 5-12 and 5-13 of the GPSA Manual go into more detail of this calculation and give an example calculation.

Figure 10.10 shows the steps of the calculation more explicitly.

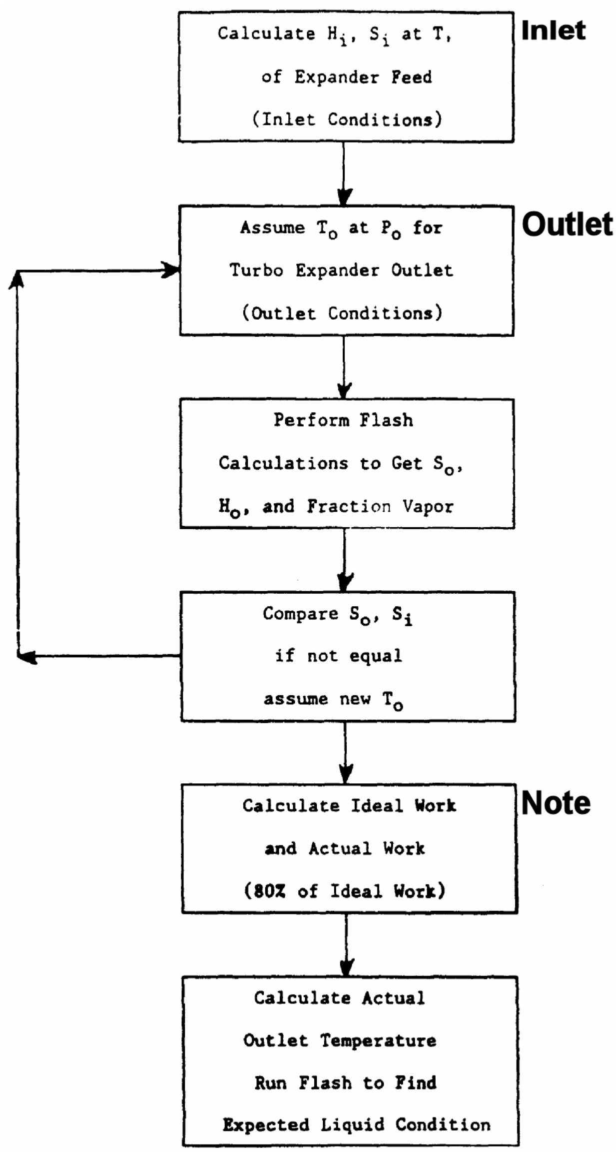


Figure 10.10: Finding the Amount of Condensed Liquid

10.3 Refrigeration

We are concerned mostly with the general theory of operation of the refrigeration system and not all the mechanical details.

10.3.1 General

To start let us consider the following simple system (Figure 10.11).

The process is as follows:

Liquid refrigerant is vaporized in the evaporator to remove heat. It is recompressed by the compressor to high pressure, then condensed at the high pressure and collected in the surge drum. The liquid so formed is then throttled through the throttling valve into the chiller where all the liquid is evaporated and the cycle is repeated.

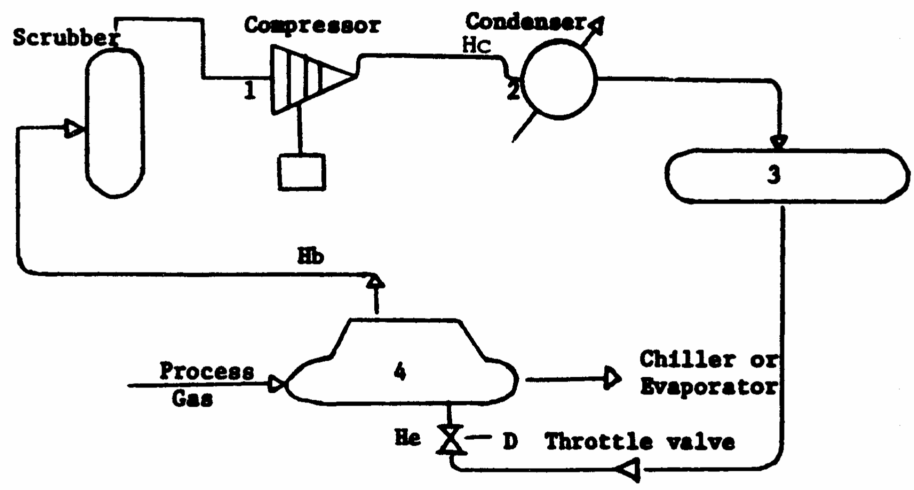


Figure 10.11: Typical Refrigeration System

The prime principle of a refrigeration machine is the absorbing of heat at a low temperature in the evaporator and rejecting this heat at a higher temperature in the condenser. It also depends on the Joule Thompson effect of constant enthalpy expansion.

Basically the cycle can be broken down into four steps as discussed above i.e. Expansion, Evaporation, Compression and Condensation and as shown on Fig. 10.12.

In Figure 10.11 let us define the heat conditions as follows:

Hb = heat content of refrigerant leaving chiller.

Hc = heat content of refrigerant leaving compressor.

He = heat content of refrigerant entering the chiller.

Also we define tons of refrigeration = amount of heat removed such that

1 ton = 200 BTU/min or 12,000 BTU/hr

In metric units we measure the KW of refrigeration and the brake power per KW of refrigeration.

The heat abstracted is

Hb - He

The work done is (assuming adiabatic compression) the change in enthalpy

Hc - Hb

The coefficient of performance β is defined as follows:

Equation 10-25

$$\beta = \frac{Hb - He}{Hc - Hb}$$

Now to find HP required:

Equation 10-26

$$HP\ per\ ton\ of\ refrigeration = \frac{4.713}{\beta}$$

This is theoretical HP of course. To find actual amount we have to use an efficiency factor. Take about 75-80%. Thus actual HP is

Equation 10-27

$$\frac{Theoretical\ HP}{0.80}$$

It is also possible to find theoretical HP by finding

(Hc - Hb) (W) and converting BTU's to HP.

W = lbs of refrigerant circulated/unit of time.

10.3.2 Thermodynamics

To understand the refrigeration process let us examine a pressure enthalpy diagram and point out the conditions that occur in the cycle.

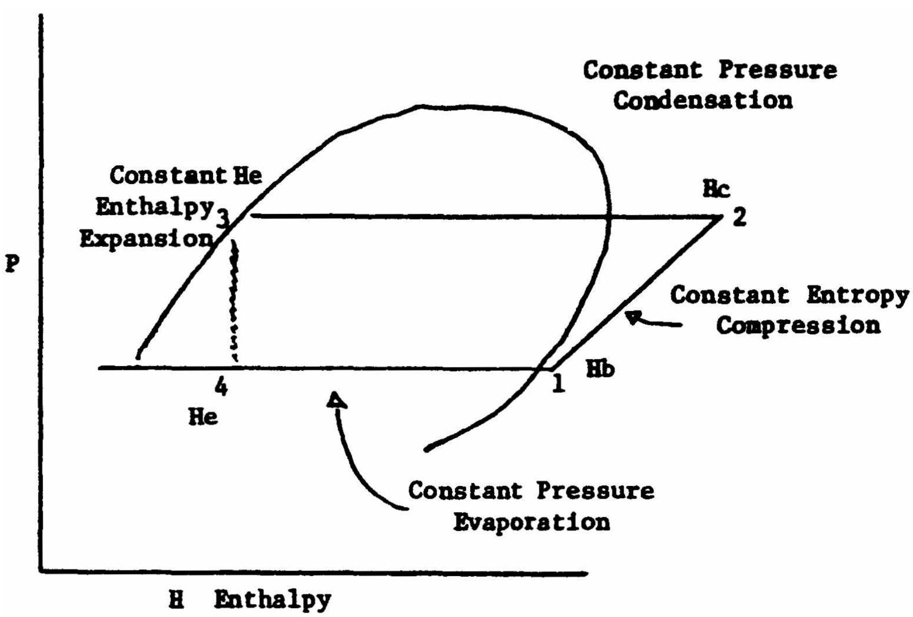


Figure 10.12: Refrigeration Thermodynamic Cycle

At Point 1 we have a refrigerant vapour at low temperature and ready to be compressed adiabatically (enthalpy Hb). At Point 2 we have a compressed gas at the outlet of the compressor (enthalpy Hc). It is then cooled and condensed to enthalpy "He". It is then throttled (i.e. expanded) at constant enthalpy to Point 4. Finally the refrigerant is evaporated and returned to the compressor suction.

There are several modifications to the simple refrigeration system that can reduce the horsepower required in the system. The most common system is to use stages as described below.

10.3.2.1 Refrigeration Stages

Refrigeration systems utilizing one, two, three, or four stages of compression have been successfully operated in various services. The number of levels of refrigeration generally depends upon the number of compression stages required, interstage heat loads, economics, and the type of compression.

One-Stage Refrigeration System

A typical one-stage refrigeration system is shown in Fig. 10.13 where the data are for pure propane refrigerant.

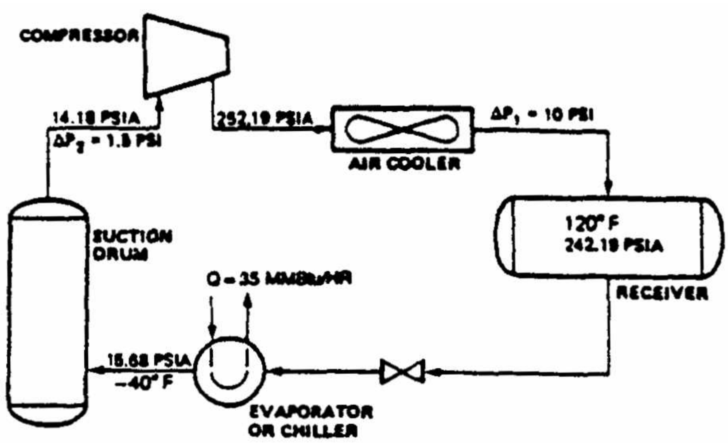


Figure 10.13: One-Stage Refrigeration System

Two-Stage Refrigeration System

Savings in the 20% range can often be achieved with a two-stage refrigeration system and interstage flash economizer. Additional savings can be realized by removing process heat at the interstage level rather than at the low stage level. A typical two-stage system with an intermediate load is shown in Fig. 10.14 with data for pure propane.

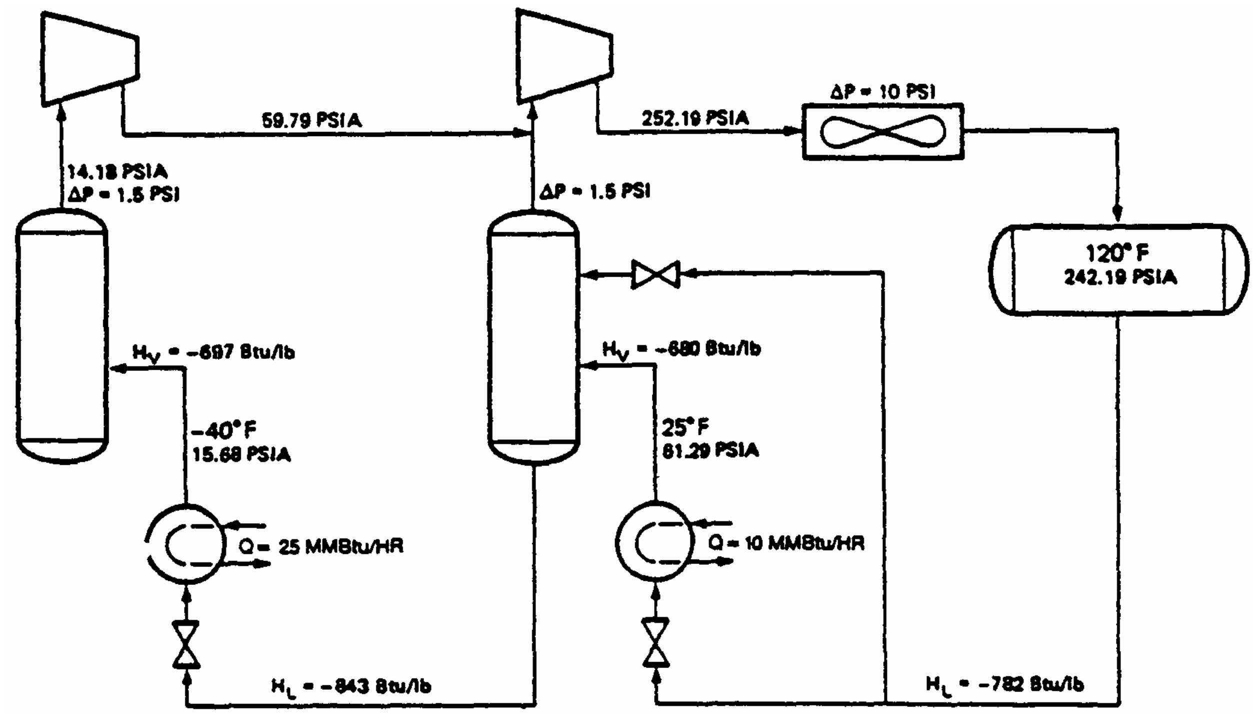


Figure 10.14: Two-Stage Refrigeration System

Three-Stage Refrigeration System

Additional horsepower savings can be achieved by using a three-stage compression system. As with a two-stage system, flash economization and/or an intermediate heat loads can be used. The savings, while not as dramatic as the two stage versus one-stage, can still be significant enough to justify the additional equipment. A typical three stage propane system is shown in Fig. 10.15.

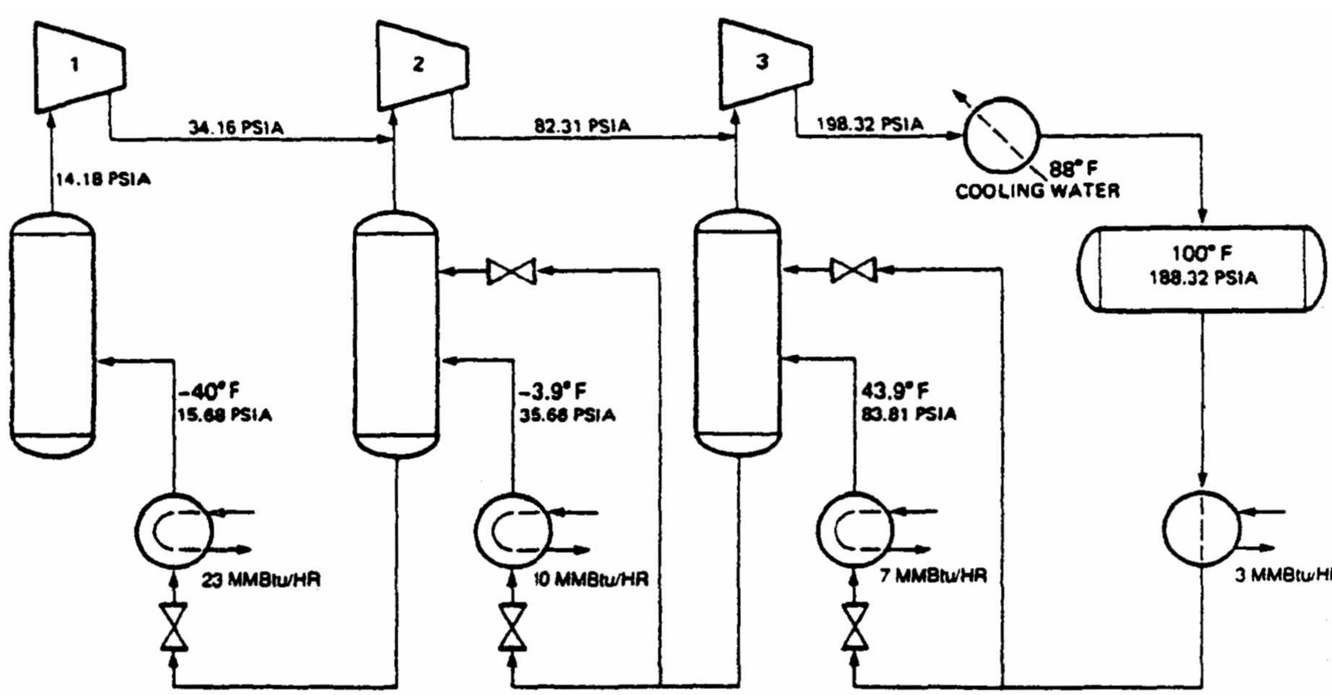


Figure 10.15: Three-Stage Refrigeration System

Refrigeration System Configuration

Energy consumption is frequently reduced as the number of stages is increased. For a propane refrigeration system, Table 10.6 illustrates the effect of interstages without using refrigeration at intermediate levels. However, the installation cost of such refrigeration systems increases as the number of stages increases. The optimum overall cost will be a function of the specific system and has to be determined for a set of economic criteria.

Table 10.6: Effect of Staging on a Propane Refrigeration System

	Stages, n		
	1	**2**	**3**
Refrigeration Duty, MMBtu/hr	1.0	1.0	1.0
Refrigeration Temperature, ºF	- 40	- 40	- 40
Refrigerant Condensing Temperature, ºF	100	100	100
Compression Requirements, hp	292	236	224
Reduction in hp, %	Base	19.2	23.3
Condenser Duty, MMBtu/hr	1.743	1.600	1.575
Change in condenser duty, %	Base	-8.2	-9.6

10.3.3 Refrigerants Used

The common refrigerant used in gas processing is propane as it is normally readily available in a plant. However propylene, freon and ammonia can also be used in the same service. For colder temperatures in LNG cycles ethylene and methane are the normal refrigerants.

10.3.4 Power Required

The GPSA Data Book gives a comparison of the atrinonia, propylene, propane and freon in regards to HP/ton of refrigeration (or Brake power/KW of refrigeration), etc. on Fig. 5.18.

To obtain a more accurate power requirement Figure 10.11, Figure 10.12 and Figure 10.13 can be used to calculate this value and the condenser duty depending on the evaporator temperature and the refrigerant condensing temperature and the number of stages.

10.3.5 Equipment Required

a) Compressors - Both reciprocating compressors and centrifugal used. Centrifugal are preferred because of lower maintenance generally not used when HP requirements are below 1,000.

b) Condensers - The condenser must remove all the heat taken out of the evaporator plus the heat of compression. The condenser can be either water cooled or air cooled.

c) Water Cooled - These are standard, well-designed shell and tube exchangers. They have the advantage of making the system have less HP than a system with an air cooled condenser. Figure 10.14 shows the difference of HP required with various condensing temperatures for a propane refrigerant system against various evaporation temperatures.

d) Air Cooled - These are standard air cooled condensers. The lowest condensing temperature that probably can be expected is about 20ºF above the design ambient temperature and thus the system requires more HP.

e) Economizer - This is an intermediate flash vessel that gets rid of some of the vapours that would be formed on direct throttling. Two stages are often used when more than two compression stages are used.

f) Throttling Valve - This is a standard control valve mostly operated on level control from the evaporator of chiller.

g) Evaporator (or Chiller) - A standard U-tube heat exchanger in which the refrigerant is normally on the outside of the tubes is used. A good disengaging space should be provided in the top of the refrigerant side to prevent liquid entrainment.

h) Suction Scrubber - This is a large vessel complete with entrainment separators, automatic liquid level dumps, etc. It is there to prevent any liquid carryover into the compressors.

i) Auxiliaries - Such units as refrigerant driers, oil removal facilities, purges, etc., are also generally provided.

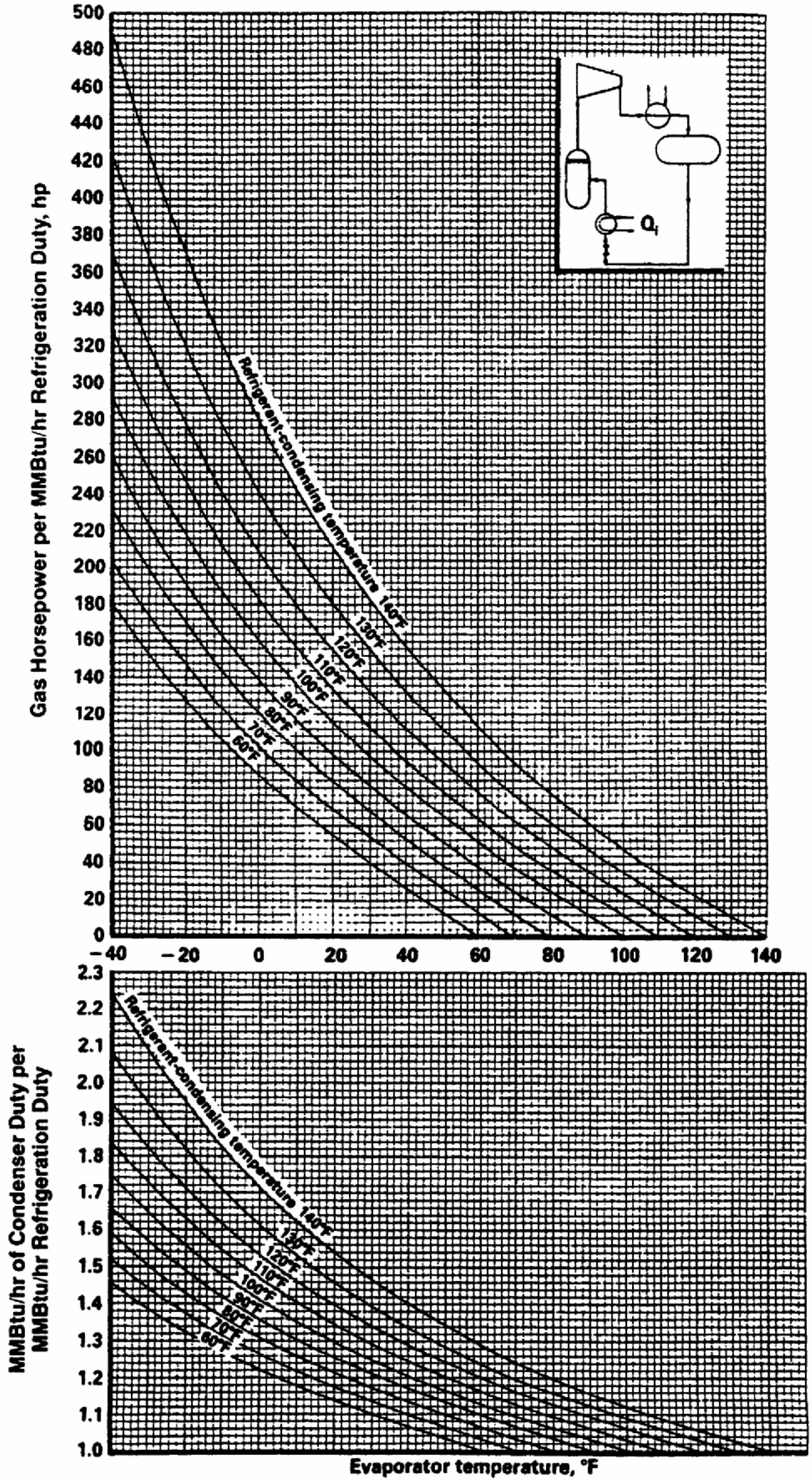


Figure 10.16: Single-Stage Propane Refrigeration System

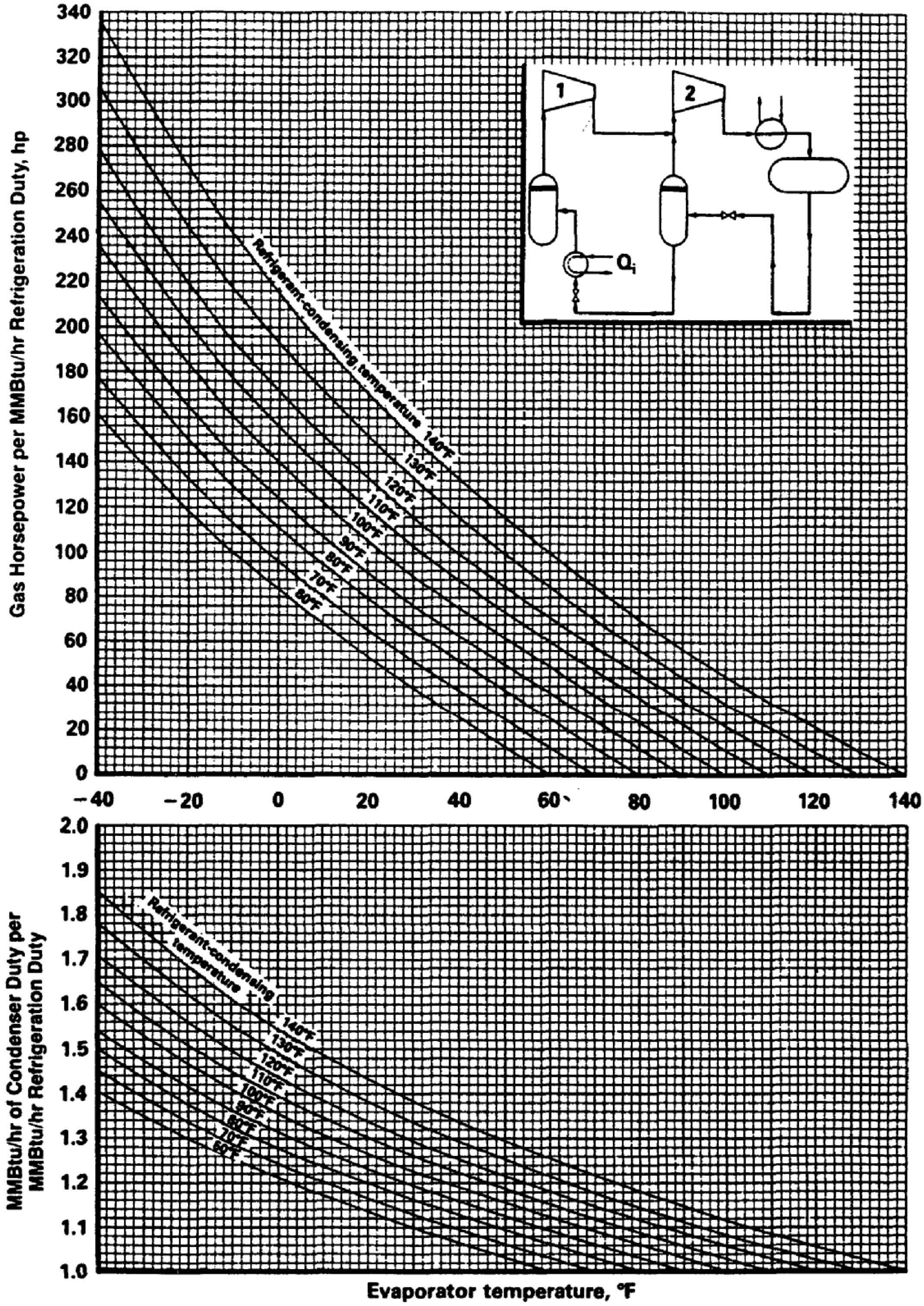


Figure 10.17: Two-Stage Propane Refrigeration System

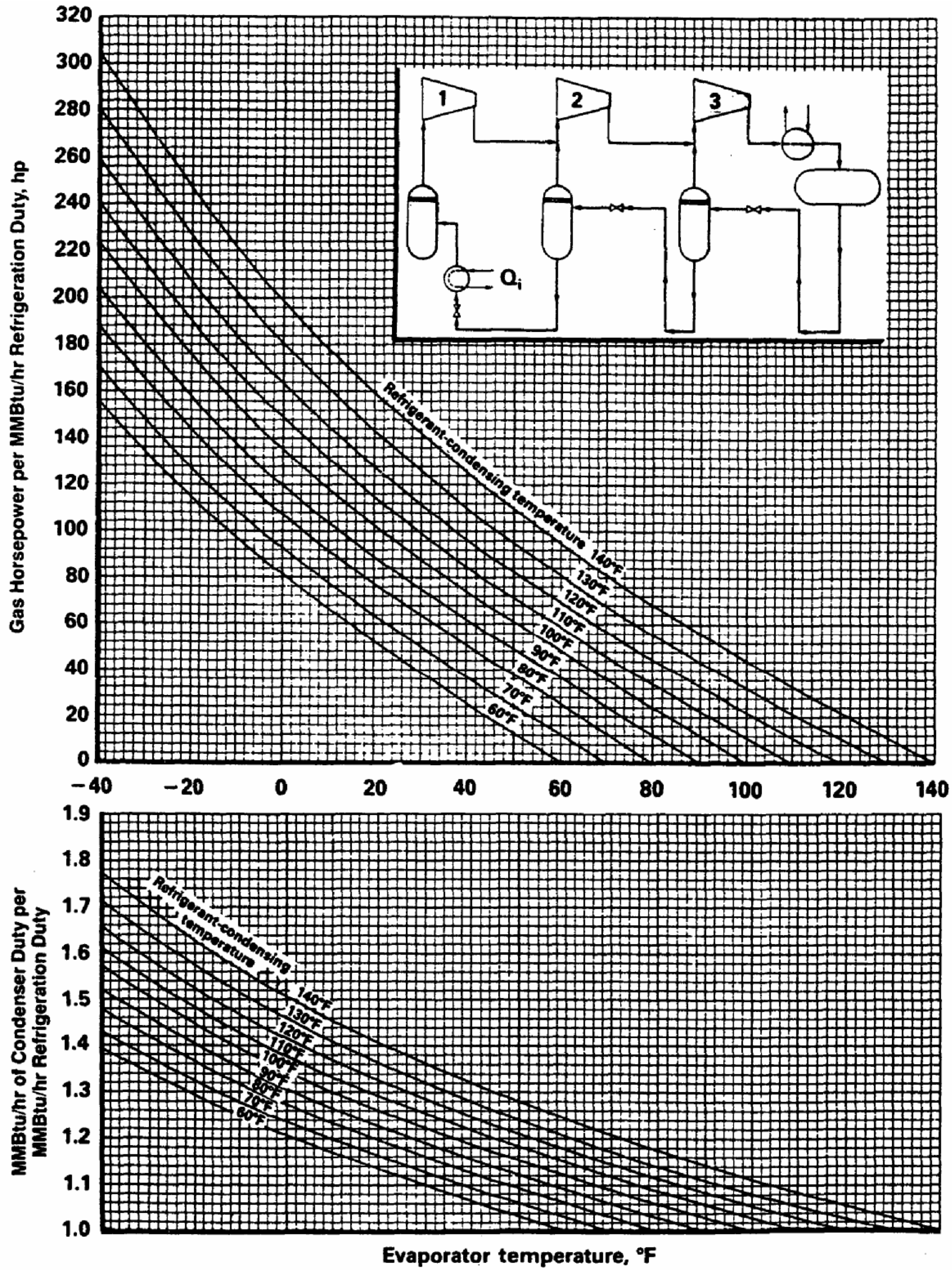


Figure 10.18: Three-Stage Propane Refrigeration System

10.3.6 Conditions for a Chilling System

Evaporator Temperature

This should be set at about 10ºF lower than the process outlet temperature. This sets the evaporator pressure and thus the suction pressure to the compressor.

Condenser Refrigerant Outlet Temperature

This is set by the type of cooling medium. With water it is about 10ºF above the water outlet temperature and thus runs about 90ºF to 100ºF. With an air cooled condenser, again the propane will be 10ºF above the outlet temperature of the aerial cooler outlet temperature which will probably run about 100ºF to 110ºF. Thus a propane temperature of 110ºF to 120ºF will be obtained. This information sets the compressor discharge pressure. From the above data the compressor HP can be calculated. The following example shows how a typical system is calculated:

10.3.7 Refrigeration System Checklist

Indication	Causes
High Compressor.	Check accumulator temperature.
Discharge Pressure.	If the accumulator temperature is high, check: 1. Condenser operation for fouling. 2. High air or water temperature. 3. Low fan speed or pitch. 4. Low water circulation. If condensing temperature is normal, check for: 1. Non-condensables in refrigerant. 2. Restriction in system which is creating pressure drop.
High Process Temperature.	Check refrigerant temperature from chiller. If refrigerant temperature is high and approach temperature on chiller is normal, check: 1. Chiller pressure. 2. Refrigerant composition for heavy ends contamination. 3. Refrigerant circulation or kettle level (possible inadequate flow resulting in superheating of refrigerant). 4. Process overload of refrigerant system. If refrigerant temperature is normal, and approach to process temperature is high, check: 1. Fouling on refrigerant side (lube oil or moisture). 2. Fouling on process side (wax or hydrates). 3. Process overload of chiller capacity.
Inadequate Compressor Capacity	Check: 1. Process overload of refrigerant system. 2. Premature opening of hot gas bypass. 3. Compressor valve failure. 4. Compressor suction pressure restriction. 5. Low compressor speed.
Inadequate Refrigerant Flow to Economizer or Chiller	Check: 1. Low accumulator level. 2. Expansion valve capacity. 3. Chiller or economizer level control malfunction. 4. Restriction in refrigerant flow (hydrates or ice).

Example Problem 10.3

Calculate the HP required for refrigeration to chill 100 MMSCFD of a gas at 937 psig of the composition listed below from 40ºF to 0ºF using propane as a refrigerant. The evaporator temperature is -10ºF and the propane condenser operates to condense the propane to a liquid at 95ºF.

Composition	Mole %
N_2	0.00
C_1	95.62
C_2	3.01
C_3	0.67
iC_4	0.18
nC_4	0.20
iC_5	0.09
nC_5	0.09
C_6	0.14

Solution

First find the amount of heat to be removed in the chillers. This is found by enthalpy calculations.

This calculation is shown in Example Problem 2 in Chapter 2. The duty is equal to 5,727,028 BTU/HR or $\frac{5{,}727.028}{12000} = 418$ tons of refrigeration.

In metric this equals 1679 KW.

Next calculate the refrigeration HP, finding the coefficient of performance:

$$\frac{H_b - H_e}{H_c - H_b}$$

For this condition, using Propane Mollier Diagram in GPSA Data Book:

H_b at -10ºF and saturated vapour = -690 BTU/lb

H_e at 95ºF and saturated liquid = -800 at (175 psia-condensing pressure)

H_c at 118ºF and 175 psia = -652

$$\beta = \frac{-690 - (-800)}{-652 - (-690)} = 2.90$$

HP per ton of refrigeration

$$\frac{4.713}{\beta} = \frac{4.713}{2.90} = 1.62$$

This is a theoretical value. Divide by efficiency of 80% to get brake HP

$$\frac{1.62}{.80} = 2.02\ HP/ton$$

Compare this with value obtained from Fig. 5-14 of the GPSA Data Book.

2.03 HP/ton for conditions:

HP required using calculated HP/ton = 2.02 (481) = 971.6

Therefore use 1000 HP.

In metric units the Brake Power/KW of refrigeration is 0.43 at -23.3°C from Fig. 5.18 in metric GPSA Data Book. Therefore the Brake Power required: 1679.2 (.43) = 722 KW.

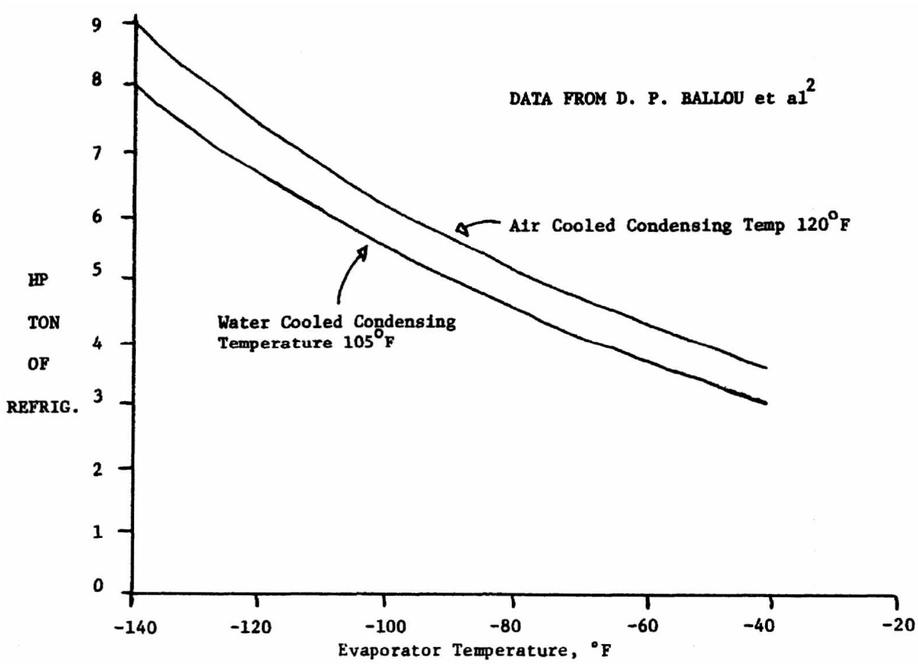


Figure 10.19: HP Required for Different Condensing System

10.4 Example Problems

Example Problem 10.4

A problem has arisen in an absorption plant that has an average flow of 600 MMSCFD. Too much ethane is being absorbed such that at maximum recoveries of propane of 85% of the stream it is necessary to flare approximately 4 MMSCFD of a mixture of 10% C_1 and 90% C_2. This is not allowed by governmental regulations. The flaring can be stopped by cutting back the absorption rate so that only 83% of the propane is recovered. Instead it is proposed to install a compressor to reinject the gas into the sales gas line. The capital cost of the compressor will be $650/HP installed and it will cost to operate $10/HP/year including fuel. The value of gas to the plant is $1.05/MMBTU. The value of propane is $7.00/barrel when sent out of the plant. The ethane gas is available at dethanizer overheat at 400 psig and the sales gas line operates at 925 psig. The pressure drop in any piping and controls to the sales gas line will be 15 psi. The gas composition to the plant is:

Composition	Mole %
C_1	93.0
C_2	4.8
C_3	1.4
iC_4	0.3
nC_4	0.5

The ethane overhead stream is colled to 60ºF before it is compressed. Assume an overall efficiency to the compressor of 85%. Determine the payout in years before taxes on the installation of this facility to see if it is worthwhile. Also determine the DCF rate of return if the installation has a 25 year life.

Example Problem 10.5

For a special deep chilling process for lpg recovery it is planned to use ethylene as the refrigerant. The gas in the process is to be chilled to -110ºF using ethylene with a 10ºF approach in the chiller.

The ethylene is condensed by propane to 0ºF in the ethylene compressor condenser.

If the ethylene compressor has to take out 600 tons of refrigeration, calculate the HP required by the ethylene refrigeration machine. Assume the efficiency of the machine is 75%.

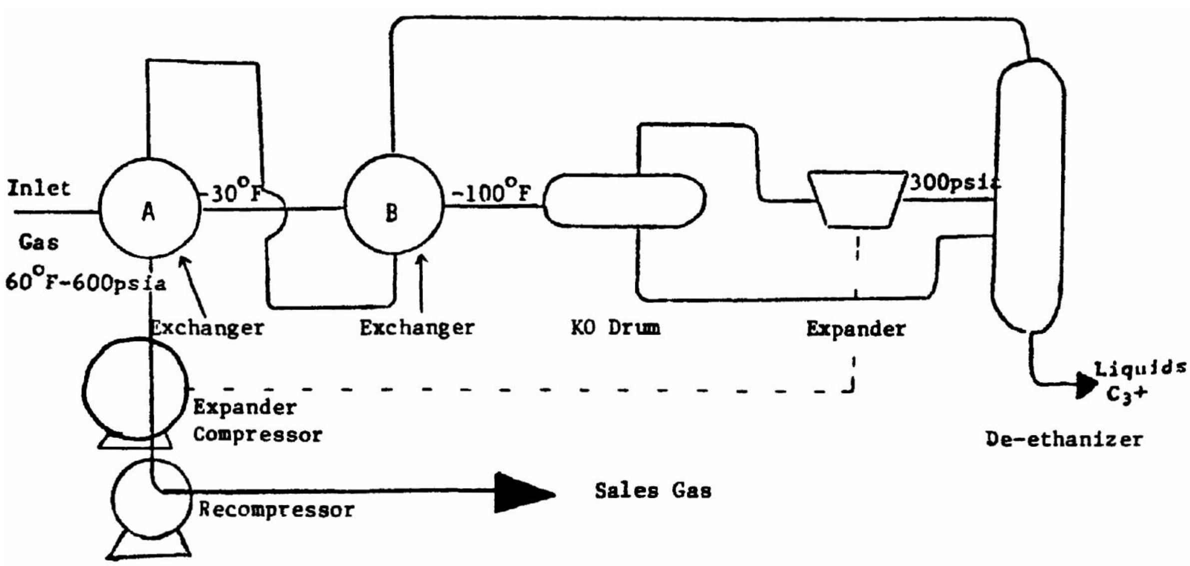
Also calculate the HP (assume same efficiency) for the propane compressor if it is used to cool the gas flow at 200 MMSCFD (0.6 SpG at 1000 psig) from 40ºF to 0ºF as well as condense the ethylene.

Example Problem 10.6

A plant that is on a sales gas line is designed to recover lpg's (C_3^+) uses the expander process. The design volume of gas is 400 MMSCFD. The inlet gas analysis is:

Composition	Mole %
C_1	92.00
C_2	5.40
C_3	1.81
iC_4	0.29
nC_4	0.40
iC_5	0.10
Average MW	17.5

A simple flow sheet of the plant is shown below.



Exchanger B ruptures and is out of service for 4 months. You are to decide what is the best way to continue processing the gas from the following choices:

1. Bypass exchanger B on both sides and handle 400 MMSCFD in the expander. In this case the outlet process gas temperature from A is reduced to -40ºF because A has colder stripped gas entering it.
2. Bypass B and the expander and reduce the pressure to 300 psia by means of a throttle valve. The outlet of exchanger A is reduced to -80ºF in this case.

Assume that the reduction in flow is proportional to loss in HP from the expander being shut down, that is:

$$400\ MMSCFD\left(\frac{HP\ to\ system\ \ from\ Expander}{\mathrm{Re}compressor\ HP + HP\ to\ system\ \ from\ Compressor}\right)$$

The recompressor HP is 24,000.

The expander system is 60% overall efficient.

Assume 5 psi drop per exchanger.

Assume pure methane for all expander enthalpy, entropy calculations.

Assume all C_3^+ that is liquefied is recovered.

Example Problem 10.7

One of the schemes for moving gas from the Arctic involves chilling the gas initially to remove the liquids and then keeping it cold so not to melt the permafrost.

Please calculate the approximate HP required at a compressor station that handles 5 MMSCFD of gas (measured at 14.4 psia and 60ºF) and the following takes place.

- Gas is compressed from 400 psia to 800 psia. The suction temperature is +25ºF.
- The gas is chilled from +80ºF to -5ºF so as to keep the gas cool enough to reach the next station at not greater than +25ºF.

Compare this with the case in which no chilling is put in the station and the gas is compressed at a suction temperature of +75ºF (summer conditions) from 400 psia to 800 psia.

Assumptions

a) Gas properties

   MW = 18

   c_P/c_V = 1.26

b) Enthalpy difference of cooling gas from 80ºF to -5ºF at 800 psia is 43 BTU's/lb.

c) Use propane as the chilling medium. Assume a 5ºF approach in the chiller and assume a 95ºF propane liquid condenser temperature.

d) Assume pipeline compressors are 85% efficient and that reciprocating compressors are used.

References

3. J. S. Sweargin, Hydrocarbon Processing, April 1970, P. 97.

4. D. P. Ballou et al., Hydrocarbon Processing, June 1967, P. 119.

5. GPSA Data Book 1987 Edition Chapters 13 & iii.

11.0 Economics

- A review of the types of calculations used in justifying a project, ie. payouts, average annual rates of return.
- A look at the data required for these calculations.

11.1 Introduction

The determining of whether a project is commercially feasible is, of course, very important. There are several methods of determining whether a project or venture is economically desirable. These methods and their results are also used to decide which of many alternates are the most desirable.

11.2 Economics Indicators

a) Payout - This is the time from initial operation it takes for the project to generate enough cash to recover the investment made in the project. The cash flow is the sum of the net income after income tax plus the allowable income tax depreciation and, if allowed, the depletion allowance.

b) Discount Cash Flow Rate of Return - This is the interest rate which the actual investment is earning over the life of the project.

c) Number of Times Investment Return - This is the number of times that the earnings recover the investment. It is found by dividing the cash flow for the life of the project by the investment made.

d) Present Worth - This is the present day value of all future earnings. This is found by discounting the future earnings at a prescribed discount rate. Generally both a) and b) and often c) and d) should be worked out for each project. If alternates are being considered, all should be obtained.

e) Present Value Profit Ratio - This is the present value of all earnings at prescribed discount rate divided by the capital cost. Often companies set the minimum DCF rate of return acceptable and then calculate the present worth at this rate. The measure of a project is the present worth of a project above this minimum.

f) Others - There are several economic indicators such as the equivalent rate of return, simple rate of return, etc.

11.3 What is Considered a Good Investment

This, of course, depends a great deal on the size and type of project and on the risks involved in the project. The risks involved can be outlined as follows:

a) Markets Changing - Markets for products can disappear. This is unlikely with gas and by-products, which are primarily fuel. However, if they are not priced competitively, they can be reduced drastically as has happened to Canadian natural gas in the United States. For example, sulphur may be replaced if the price is too high. Also, there is the possibility of reduced market prices. This was not so significant with gas, but recently it has become a real possibility and is very much the case with LPG's and sulphur, but not condensates. For example, in 1967 sulphur was being sold at \$50/LT f.o.b. plants and in 1971 was down as low as \$4/LT. It has subsequently risen to over \$150 in 1985.

b) Change of Costs - Plant costs can change, both capital and operating; these changes can generally be reasonably forecasted. However, with the shortages of labour and material, the cost of plants, pipelines, etc. go up, then with a recession they go down again as witnessed by 1983 and 1984 costs as compared to 1981 and 1982 costs. See the Nelson Index table later in this Section showing the changes with time. Inflation is generally a very big factor in capital costs. The risk in regard to plant capital costs are discussed later with the accuracy expected in cost estimating. (Also see reference #5 on this.)

c) New Technology - New plants become cheaper because of new technology. This change is not so important in gas processing as compared to petrochemical plants because the main product, gas, is often contracted for the life of the plant. Also, there has not been a major technical change in gas processing in the last ten years.

d) Government Regulations - This change is very difficult to forecast, as witnessed by the many changes in 1974, 1975 and 1981 in the petroleum industry via taxes, royalty rates, PIP grants, etc. This has become the most variable and unpredictable factor in economic evaluations.

All these factors have to be taken into account when evaluating projects. The higher the risk, the better the economic factors should be.

An article by D. Stahlberg discusses how to identify some of the risks. A formula that identifies the risk is as follows:

Equation 11-1

$$\%\ Risk = \frac{(Original\ Net\ Cost + Cost\ of\ Way\ Out - Net\ Cost\ of\ Safe\ Plan)}{Net\ Cost\ of\ Safe\ Plan - Original\ Net\ Cost} \times 100\%$$

This % risk is compared with the chance of success, and if the success chance greatly outweighs the risks, then the riskier method should be tried.

Here are some guidelines for projects.

Minimum Payouts

For gas processing projects:

Before Taxes	3 - 4 years
After Taxes	6 - 7 years

For petrochemical projects where the risks are probably greater:

Before Taxes	2 - 3 years
After Taxes	5 years

Minimum Discounted Cash Flow After Taxes

For Gas Processing Plants	11% - 15%
For Petrochemicals	20%
For Pipelines	12%

11.4 Capital Costs

There are many types of capital cost estimates. The accuracy of these vary a great deal depending on the amount of information obtained and work done by the estimator. There is a great deal of information and data on the type and scope of estimates. A whole course could be given on making proper estimates. Also, there is a professional society devoted to cost engineering, the American Association of Cost Engineers. Figure 11.1 shows the approximate accuracy of estimates depending on the status of the project. Also Figure 11.2 gives a graphical outline of some of the types of estimates and the information needed for them. Each type of estimate requires a different amount of data to obtain.

11.4.1 Preliminary Feasibility Estimate

The preliminary feasibility estimate (order of magnitude).

As a very general guide the following can be used for gas plant costs:

Type of Gas	Capital Cost - $/MMSCF
Very sour and rich	300,000 - 500,000
Medium sour	100,000 - 300,000
Sweet but rich	50,000 - 100,000
Dry and sweet	10,000 - 50,000

or you can use the method of E. Berlie.

11.4.2 Preliminary Budget Estimate

The concept or preliminary budget estimate can be done if you have the overall data on gas analysis, volumes, etc. This estimate is done by breaking the plant down into various process units and using general data for gathering pipelines, etc. The following table gives the curves that are included.

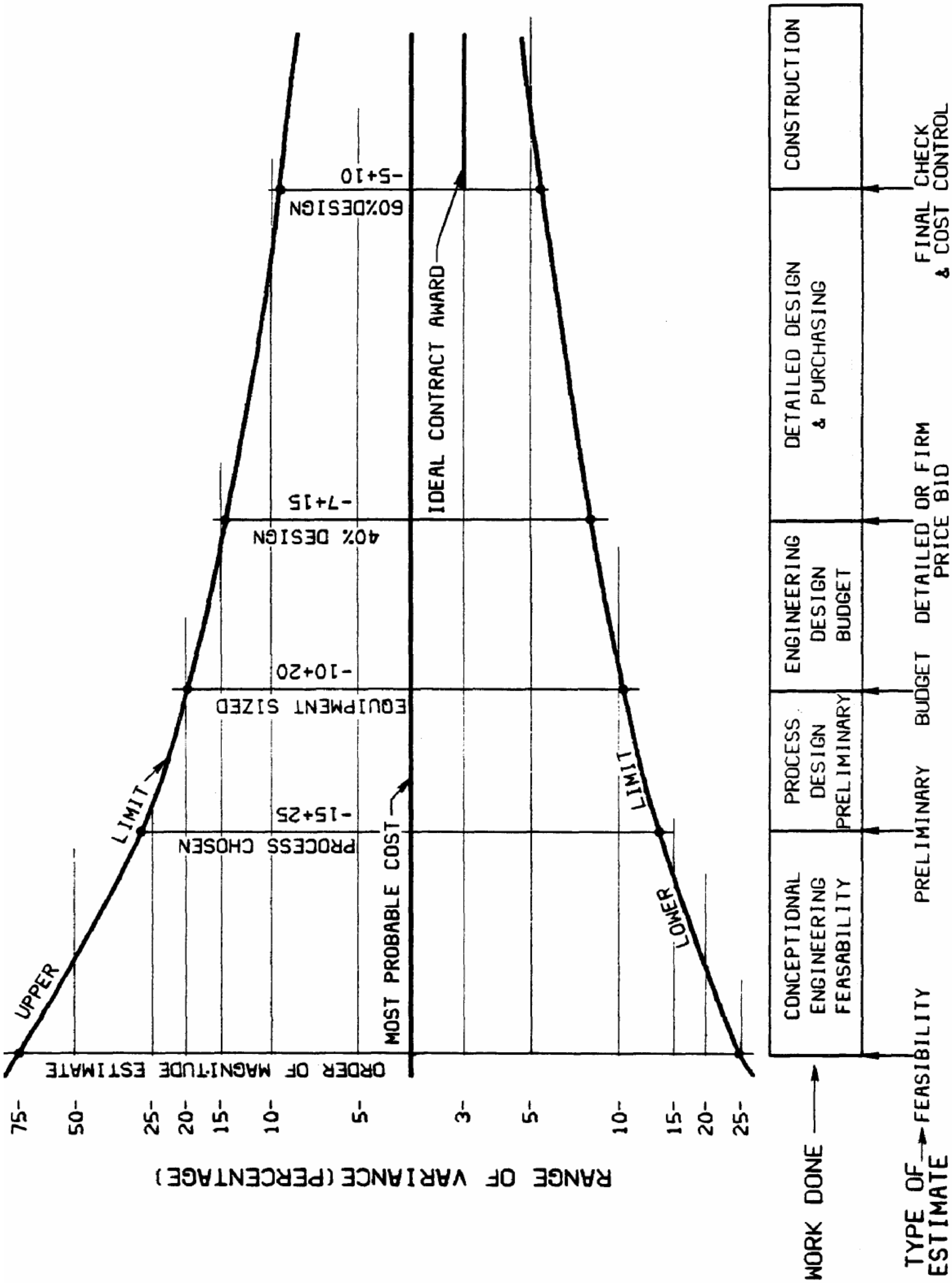


Figure 11.1: Estimate Accuracy

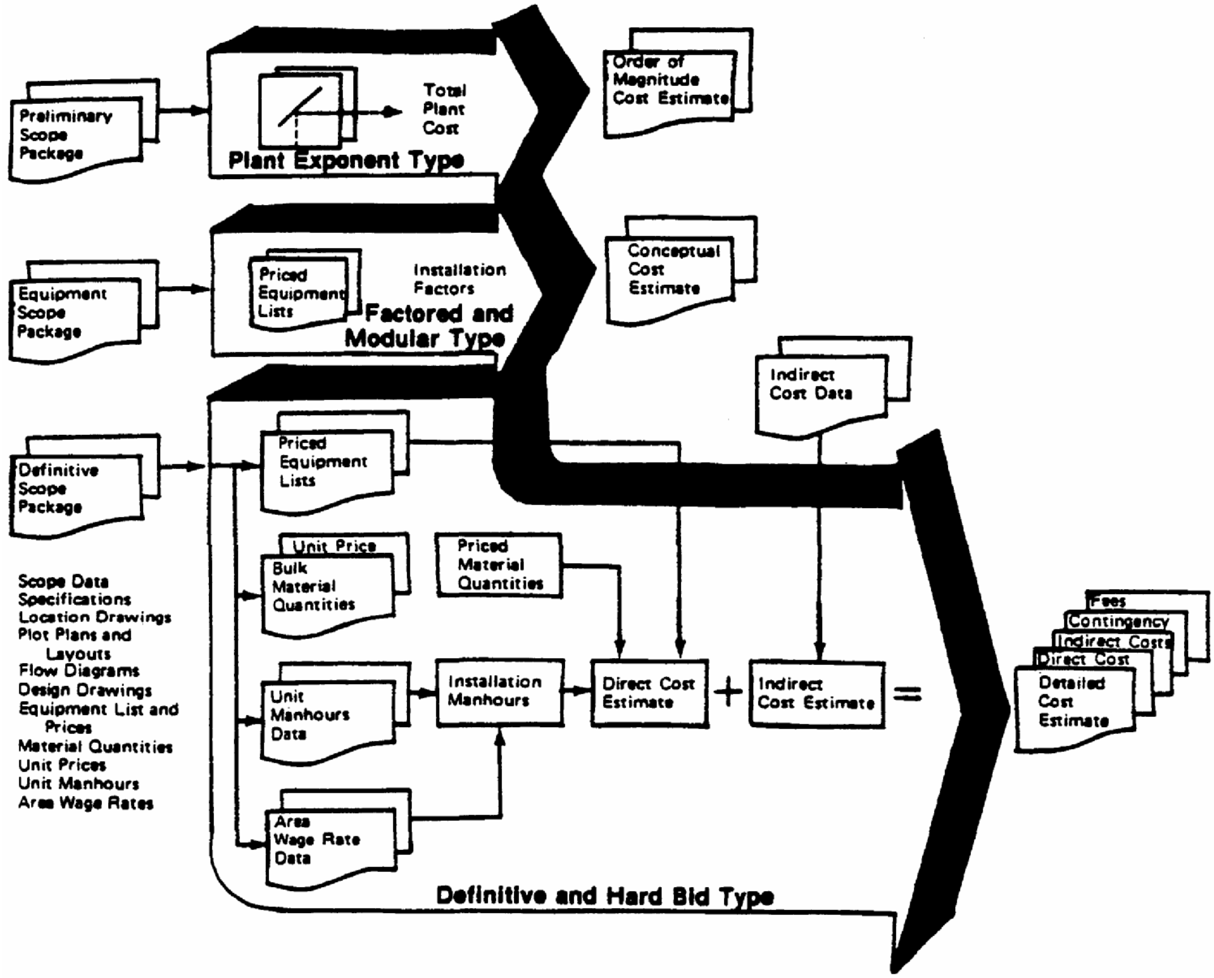


Figure 11.2: Categories of Estimates

Table 11.1: Process Unit Cost Curves

Figure No.	Process Unit	Variable
Figure 11.3	Sweetening	US gpm solvent circulated
Figure 11.4	Sulphur Recovery	LTD sulphur product
Figure 11.5	Dewpoint Control	MMSCF Feed Gas
Figure 11.6	Glycol Dehydration	MMSCF Feed Gas
Figure 11.7	Fractionation	Feed MBPD
Figure 11.8	Compressor Installation	BHP
Figure 11.9	Gathering System Cost	Diameter inch mile
Figure 11.10	Wellsite Cost	Flow MMSCFD
Figure 11.11	NGL's Facilities	Recovery of Liquids BPD

Note: This information is presented courtesy of G. E. Handwerk, Consulting Engineer, Denver, Colorado, Fish International Canada Ltd., Calgary, Alberta, Dome Petroleum Limited, Calgary, Alberta and SKM Consulting Ltd., Calgary, Alberta.

The year of information in the curves is given and should be adjusted to present day costs using an inflation index as indicated below.

The cost of various facilities (particularly plants) changes with time. A good way to update costs is to multiply the cost of the facility in a certain year by the ratio of the "Nelson" construction index of today's date and divide by the Nelson construction index of the date the plant is built. Therefore, if you know that a plant costs $10,000,000 in 1964, the cost for the same plant in 1976 would be:

$$\$10{,}000{,}000\,\frac{626.1}{257} = \$24{,}362{,}000$$