Raw Gas Compression

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1. Introduction

"Raw gas" is a compound of gases, liquids, and solids that is unique to its source. Often an entire reservoir will provide a mixture that is roughly homogeneous across the field, but sometimes each formation in each well presents unique mixtures. For example, it is common for a field to go from "sweet" to "sour" over time with the accumulation of sour gas (H_2S). When this "souring" occurs it usually happens one well at a time, so equipment designed for sweet gas must be modified or replaced. Other signs of individuality in gas production include changes in the mix of inert gases like nitrogen and carbon dioxide and the gas-water-ratio of the stream over time.

Reservoir gas is rarely suitable for delivery to end users. First, it is not located where the end user is located, so it must be transported. Reservoir gas is rarely of a suitable quality for end-user equipment, so it must be processed to the required quality-standard with the proper energy content. Every step in the transportation and processing of reservoir gas reduces the pressure of the gas. To boost the gas pressure to the values required for the next step, the gas is compressed. This compression-and-use-of-energy cycle may be repeated several times in the journey of a molecule of gas from the reservoir to the burner tip. As the gas moves through the systems, it evolves towards becoming a commodity that is universally consistent for all end-users. The most appropriate choice for compression technology evolves as well.

Raw well-head gas is quite variable in terms of contaminants, pressures, and temperatures. The best choice from a fluid-mechanics viewpoint might not be the best choice from a space-management, or equipment-availability viewpoint. Preferences must always be modified to accommodate the realities of an operation stage. As the various constraints are prioritized, the requirements of the reservoir must remain the top priority. For example, the movement of fluids within a reservoir is chaotic and constantly changing; no reservoir is going to provide gas at a constant flow rate at a constant pressure. This variability can be "controlled" by putting an aggressive control valve on the suction of a compressor to take a large pressure drop during maximum flow while still taking a significant pressure drop at minimum flow to allow the compressor to be fed at a constant suction pressure. A better solution for the reservoir would be to use a compressor that can adapt to maximum flow and unload as needed to accommodate minimum flow.

The work that a compressor does is the combination of how much material it is lifting (mass flow rate), and how high it is lifting that material (compression ratios). The same amount of work may be done at a low mass-flow rate and high compression ratios as is done at a much higher flow rate and lower compression ratios. The simplest (and most accurate) representation of compressor work is the change in specific enthalpy times mass flow rate (Eq 1). The accuracy of this equation is dependent on the quality of the equation of state that is used to determine the specific enthalpy, and the quality of the gas analysis used to describe the gas.

$$W = \dot{m} \cdot \Delta h \qquad \textit{Eq 1}$$

People throw the word "efficiency" around like it has exactly one mathematical meaning and that meaning is an immutable constant. Poppycock. Efficiency can be defined as "the ratio of work applied to the process divided by the power input to the process" (Eq 2).

$$\eta = \frac{\text{Work applied to the process}}{\text{Power input to the process}} \qquad \textit{Eq 2}$$

Here is where it gets messy. Where is the "power input" calculated? For a compressor driven by an internal combustion engine, is it the specific energy inherent in the fuel times the mass flow rate of the fuel? Or is it the power that is transferred to the input shaft on the compressor, which is generally around 1/5 of the energy in the fuel? On the output side, is it the conditions at the points you take suction/discharge pressures or the skid edge? There are no absolute answers to any of these questions. There are conventions that move us towards a common understanding, but they are far from universal. At the end of the day, a particular compression technology must be selected before it is possible to assess compression efficiency, so this topic will be developed further in sections talking about specific technologies.

After gas is processed through a plant, it is much closer to a manufactured product than the raw-gas from where it came, and it can now be called "commodity gas" as a volume of it purchased from one plant will be operationally identical to a similar volume purchased at any other plant. Compression equipment designed for commodity gas needs to maximize economic efficiency (i.e., the sum of cost of capital plus operating costs divided by throughput) over time. A compressor station designed for commodity gas will generally be expected to have the same physical equipment in 20 years that it started with. Commodity-gas compression will not be discussed further in this course.

Nomenclature

Table 1 Nomenclature Table

Symbol	Name	fps units	SI units
C_{work}	Unit conversion for Eq 5-5	3.301 hp	0.0115741 kW
c_p	Specific heat capacity at constant pressure	BTU/(R×lbm)	J/(K×kg)
dP	Differential pressure	psi	kPa
g	Acceleration of gravity	32.174 ft/s^2	9.81 m/s^2

Symbol	Name	fps units	SI units
h	Specific enthalpy	BTU/lbm	J/kg
ID	Inside diameter	ft	m
k	Adiabatic constant (c_p/c_v)	none	none
$\dot{m}$	Mass flow rate at actual conditions	lbm/ft^3	kg/m^3
n	Polytropic index	None	None
P	Pressure	psi	kPa
q	Volume flow rate at standard conditions unless subscript specifically designates ACF	SCF/day	SCm/day
q_{mmscfd}	Flow rate for Eq 5	MMSCF/day	kSCm/day
q_{liquid}	Liquid flow rate	bbl/day	m^3/day
Q	Energy content	BTU/ft^3	J/m^3
R_c	Compression ratio	None	None
T	Temperature	R	K
VI	Volume index	None	None
V	Volume	ft^3	m^3
W	Work	BTU/ft^3	J/m^3
Z	Compressibility	None	None
γ	Specific gravity	None	None
η	Efficiency	None	None

Subscripts	
approx	Approximate value
atm	Atmospheric conditions
avg	Average
design	Design conditions
disch	Discharge conditions
dischTheo	Theoretical discharge conditions
gas	Parameter applies to gas
gasAct	Parameter applies to gas at actual flowing conditions
gasACFd	Parameter applies to gas at actual flowing conditions in ft^3/day
gasSuperficial	Calculate the parameter as though gas were the only thing in the pipe
gasSTD	Parameter applies to gas at standard conditions
in	Conditions at the input to a process
liq	Parameter applies to liquid

Subscripts	
max	Maximum value
mscfd	Thousands of standard cubic ft per day
min	Minimum value
oil	Parameter applies to oil
out	Conditions at output
std	Standard conditions
suct	Suction conditions
total	Total
water	Parameter applies to water
1	Upstream conditions, or first state
2	Downstream conditions, or second state

2. Families of compressors

Figure 1 shows the families of compressors. Most well-site compressors and booster compressors on gas gathering systems are positive displacement. Of those, reciprocating compressors ("recips," in the lingo) significantly outnumber all other types of compressors, both in numbers of installed units, and installed horsepower. Whether that mix of technologies represents an optimum or not is open for debate, but that mix is the reality as of this writing. Dynamic compressors are quite rare at onshore well-sites for the good reason that they don't handle varying conditions well.

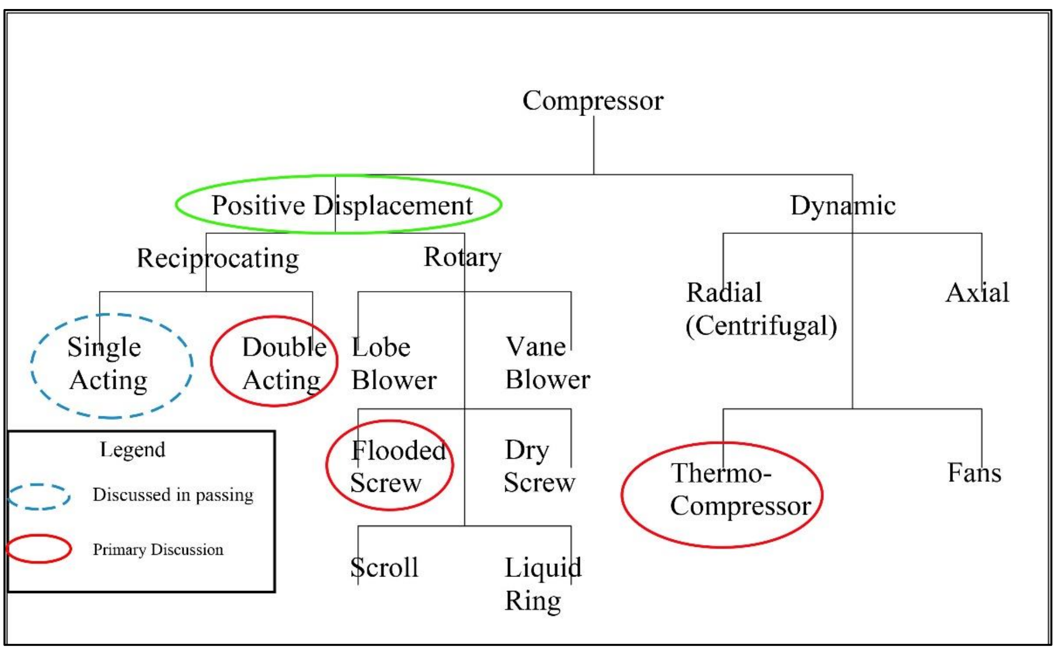

Figure 1: Types of Compression

However, we will introduce a family of dynamic compressors (thermocompressors) in this document that should have a significant role in well-site equipment.

3. **Compressor stations**

Moving from an individual well site to well pads, gathering systems, mid-stream piping, plants, mainstream piping, and local distribution systems, one sees compressor stations that handle the production from multiple wells. Each of these station-types has its own preferred choices, regulations, and limitations.

Well Pads: Multiple-well pads provide the option to measure the gas from all of the wells, and then compress the combined stream with jointly owned equipment. This combination allows the wells to share the capital and operating costs of the compression. For this option to be successful, the wells on the pad must have similar wellhead-pressure requirements and it is generally inadvisable to include a suction pressure controller on the compressor. It is rare for well-pad stations to include dehydration or other treatment. The compressors on well pads are generally single-stage recips, or oil-flooded screws. These stations typically use gas-fired engines to drive the compressors.

Gathering systems: It has become common in the 21st century to place "straddle compression" (i.e., compressor stations that take gas off a system, raise the pressure, and discharge back to the same system, with no change in ownership of either the gas, or the piping from inlet to outlet) amid gathering systems. These stations generally do not include dehydration facilities. Both single-stage and two-stage recips are common on these sites, though large oil-flooded screw compressors are occasionally used. Compressor-suction control is usually needed. Discharge pressure is often too low for dehydration or other treating to be economical. Gathering system compressor stations are increasingly electrified, and they use electric motors to drive the compression. However, the majority of stations are still gas-fired engines. There are times when field gas must be treated to make it suitable for the engines on compressors. Internal combustion engines are very tolerant of poor-quality gas, but not infinitely so which occasionally requires the use of plant-type equipment (e.g., small membrane units, or pressure-swing adsorption units) to get the fuel gas into the proper mixture for engine-use.

Mid-Stream compression (Central Point of Delivery or CPD): At the end of a gathering system, the common-carrier frequently changes to a mid-stream operator. The upstream pipe is owned by the gatherer (increasingly, the producer), and the downstream pipe is owned by the mid-stream operator. The mid-stream operator is a common carrier who moves the gas for a fee. The ownership of the gas does not change from the gathering system to the mid-stream pipe. Mid-stream systems are completely isolated from the reservoir, and it is common (and very good practice) to set the pressure at the end of the gathering system (inlet to the CPD) at a particular pressure with a very narrow operating band (e.g., 100 psig ±5 psig [689 kPag ±34.4 kPag]), and a very specific pressure into the mid-stream piping (e.g., 1000 psig [6.9 MPag] is common). These stations nearly always have suction control and dehydration. Historically, these stations have been powered by gas-fired engines, but government intervention is forcing most new facilities to be electric-motor driven. Mid-stream CPDs nearly always have two-, or three-stage recips. Midstream CPDs are rarely, if ever, ownership-transfer locations since at the midstream outlet, the gas is much closer to reservoir quality than it is to commodity quality.

Plant-outlet compression: The gas at the outlet of the plant is clean, dry, and consistent (much more a manufactured product than a natural product). Design and operation of plant-outlet and subsequent compressor sites is outside the scope of this document.

4. Types of Compression

There are several compressor technologies shown on Figure 1 that we will not discuss in this chapter. In general, these technologies simply lack the reliability and/or operating range to stand up to 24/7 unmanned operations. A machine that needs to be "fiddled with" seldom fits with well-site operating strategies but may be perfect for a different class of station.

4.1 Positive displacement compressors

Positive displacement compressors are devices that physically reduce the space available for the gas to occupy as the gas moves from suction conditions to discharge conditions. This is quite visible in a recip (Figure 3), where one can easily visualize the piston moving up in the cylinder, closing the suction valve, and pushing against the discharge valve to make a fixed mass of gas occupy a progressively smaller space.

Positive displacement compressors can lift the load much higher than dynamic machines, but they require physically larger components to move the quantity of gas that a dynamic compressor moves in smaller components.

4.1.1 Positive displacement compressor work

Eq 1 contains all the information required to determine the work that has been successfully applied to a gas, but specific enthalpy is not a term that is always readily available. Using terms that are more accessible, we can calculate the work using Eq 3. The enthalpy in Eq 1 contains all of the adjustments for adiabatic changes and for compressibility, and if you are rigorous in determining efficiency, density, and compressibility Eq 3 is amazingly close to the results obtained from Eq 1.

$$W = \frac{C_{work}}{\eta_{total}} \cdot P_{suct} \cdot q_{mmscf\,d} \cdot \left(\frac{\rho_{std}}{\rho_{suct}} \right) \cdot \left(\frac{k}{k-1} \right) \cdot \left(\left(R_c \right)^{\frac{k-1}{k}} - 1 \right) \cdot \left(\frac{Z_{avg}}{Z_{suct}} \right) \qquad \text{Eq 3}$$

“Compressor efficiency” is a term with considerable gray area and space for (mis-)interpretation. “Compression ratios” on machines without suction/discharge valves is outlet pressure divided by inlet pressure (in absolute pressure units). We will discuss compression ratios on compressors with valves in Section 4.1.5.4.

4.1.2 Positive displacement compressor efficiency

In simplest terms, “efficiency” is given in Eq 4. Driver output is the power delivered to the shaft of the compressor. The delivered-power could be the reading on a power meter times a motor-efficiency value. Or it could be a value calculated from an engine manifold pressure. Or it could be the reading from a dynamometer.

$$\eta = \frac{W_{out}}{W_{in}} = \frac{\dot{m} \cdot \Delta h}{DriverOutput} \qquad \text{Eq 4}$$

Compressor efficiency can be calculated without knowing driver-input at all. Those techniques are quite different for rotary and reciprocating equipment and will be discussed below.

4.1.3 Impact of changing suction pressure on a positive displacement compressor

$$W = \left(\frac{C \cdot P_{std} \cdot k \cdot T_{suct} \cdot Z_{avg}}{T_{std} \cdot Z_{std} \cdot (k-1) \cdot \eta_{total}} \right) \cdot \left(q_{mmscf\,/\,d} \right) \cdot \left(\left(\frac{P_{disch}}{P_{suct}} \right)^{\frac{k-1}{k}} - 1 \right) \qquad \text{Eq 5}$$

Determining what will happen inside a compressor when suction pressure changes is crucial to the application of positive displacement compression technology. If one takes Eq 3 and rearranges it to group all the terms that remain essentially constant with a change in suction pressure (Eq 5), it is clear that a change in suction pressure per unit volume would change the work required in the opposite direction (suction pressure is on the bottom of the R_c term, so a decrease in suction pressure would raise R_c and therefore, the work). At the same time, positive displacement compressors move the same physical volume per revolution of the shaft regardless of suction pressure. At a lower pressure, the volumetric flow rate at standard conditions would be lower than it would be at higher pressures.

It can be shown [Simpson, 2017] that energy input requirements change in the same direction as the change in suction pressure because the specific work change is smaller than the change in flow rate. It is common to look at the specific work per MMSCF and draw conclusions [Lea, Table 6-1], which generally leads to poor decisions and even worse results.

4.1.4 Adiabatic heat of compression for positive displacement compressors

Gas compression using positive displacement equipment can be approximated as an "adiabatic" process. "Adiabatic" means that the flow exhibits no appreciable heat transfer from or to the environment (which tends to imply is both isentropic and "reversible"). Touching the hot discharge pipe from a reciprocating compressor will confirm that there is heat lost to the environment. However, in terms of the total thermal energy in the compressed gas, the heat that is transferred to the ambient is an insignificant portion of the total heat generated, so the adiabatic assumption is a reasonable approximation for practical purposes.

For any adiabatic compression process, the discharge temperature is shown in Eq 6. The temperatures and pressures are in absolute units (the Adiabatic Constant is not actually constant and is in fact a weak function of pressure and temperature for real (non-ideal) gases, it is often worthwhile to use a program like REFPROP from NIST to calculate the adiabatic constant at suction and discharge conditions and use the average in Eq 6).

$$T_{disch} = T_{suct} \cdot \left(R_c \right)^{\frac{k-1}{k}} \qquad \textit{Eq 6}$$

4.1.5 Reciprocating compressors

Reciprocating compressors have been used since the 1800s, originally in steam-driven air service (Figure 2). Pistons moving inside of cylinders draw gas in through suction valves, then the

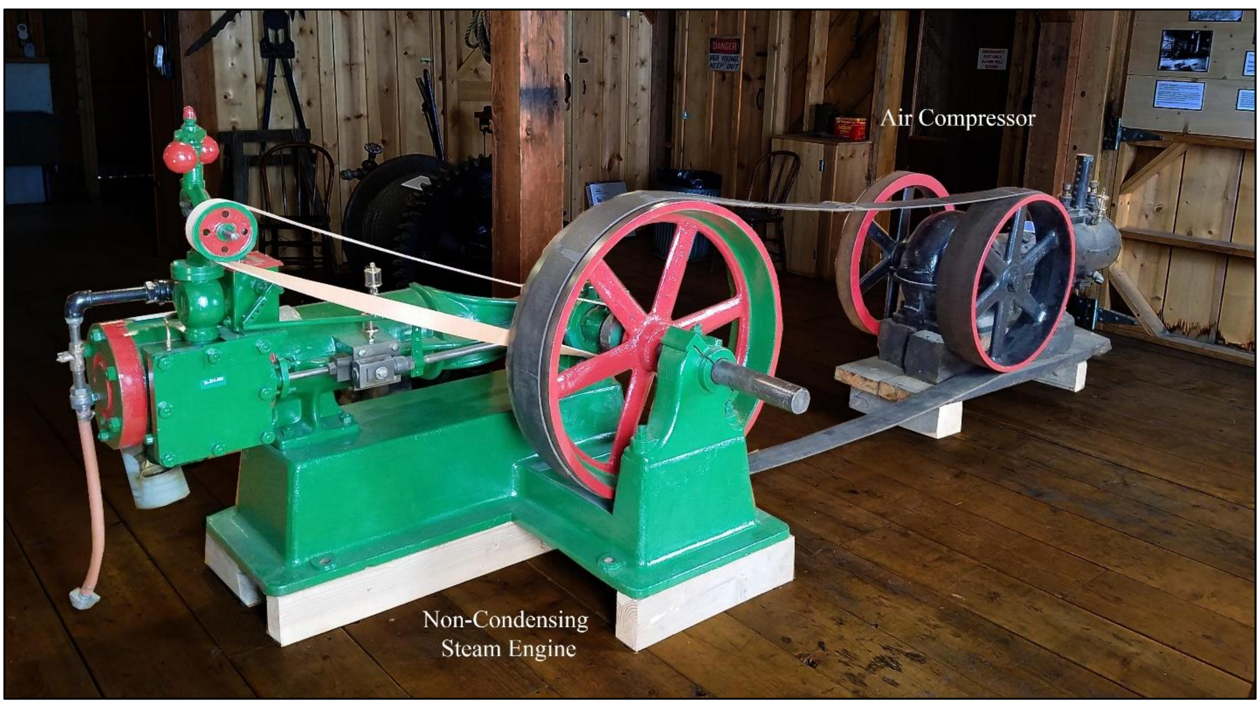

Figure 2: Steam driven air compressor

pressure is increased by reducing the physical volume inside the cylinder. The higher-pressure gas is exhausted out the discharge valves.

4.1.5.1 Operating principles

In its simplest configuration, a reciprocating compressor has a cylinder, piston, connecting rod and suction/discharge valves. Figure 3 shows that the connecting rod cannot be directly connected to a crank shaft in a double-acting cylinder since the rod must physically enter the compression chamber through a seal. The transition from the distinctive motion of a rod that accommodates a crank shaft to a rod that accommodates entering a compression chamber is made in a "crosshead" which converts a two-dimensional motion to a linear motion.

The "throw" in Figure 3 is a fairly rare configuration; it is much more common for the cylinder diameter to be the same on the crank-end (first stage in Figure 3) as on the head-end (second stage in Figure 3).

We categorize recips by: (1) number of "throws"; (2) "action"; (3) number of stages; (4) separable or integral; and (5) high speed or low speed.

Compressor throw. An industry term for a cylinder/piston pair (Figure 3 is one throw).

Action. An industry term indicating if a cylinder has one compression chamber ("single acting") or if it has two compression chambers ("double acting') (Figure 3 is double acting).

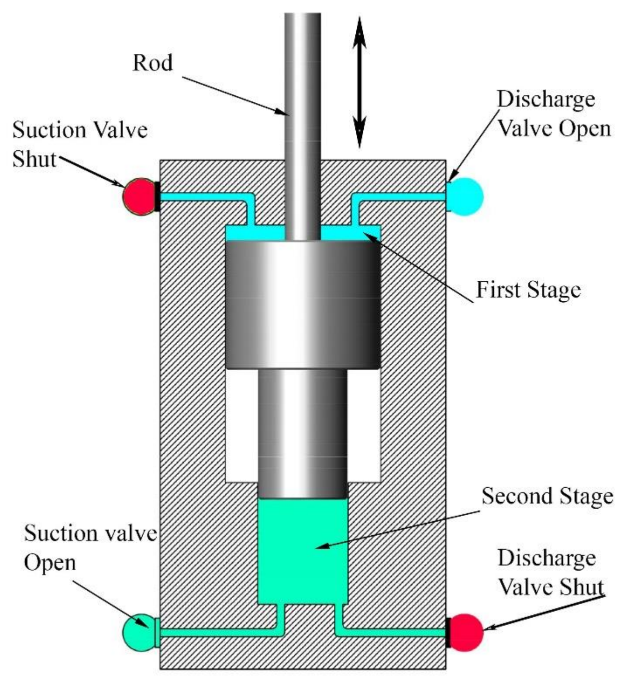

Figure 3: Double acting recip cylinder mockup

Stages. The gas leaving a compressor cylinder can go many places. On some skids the gas goes directly to an end-use; on other skids the gas goes to an aftercooler before going on to an end use. Yet other skids send the gas to an inter-stage cooler and then to another cylinder on the same skid. In Figure 3 the gas goes from the upper right-hand valve (first-stage discharge) to an inter-stage cooler, then back to the lower right-hand valve (second-stage suction). So, this double-acting throw would be two stages, meaning that the gas is compressed twice. It is important (and often difficult) to balance the load between stages of compression. The main tool used to balance stages is adjusting "clearance" to have the early stages

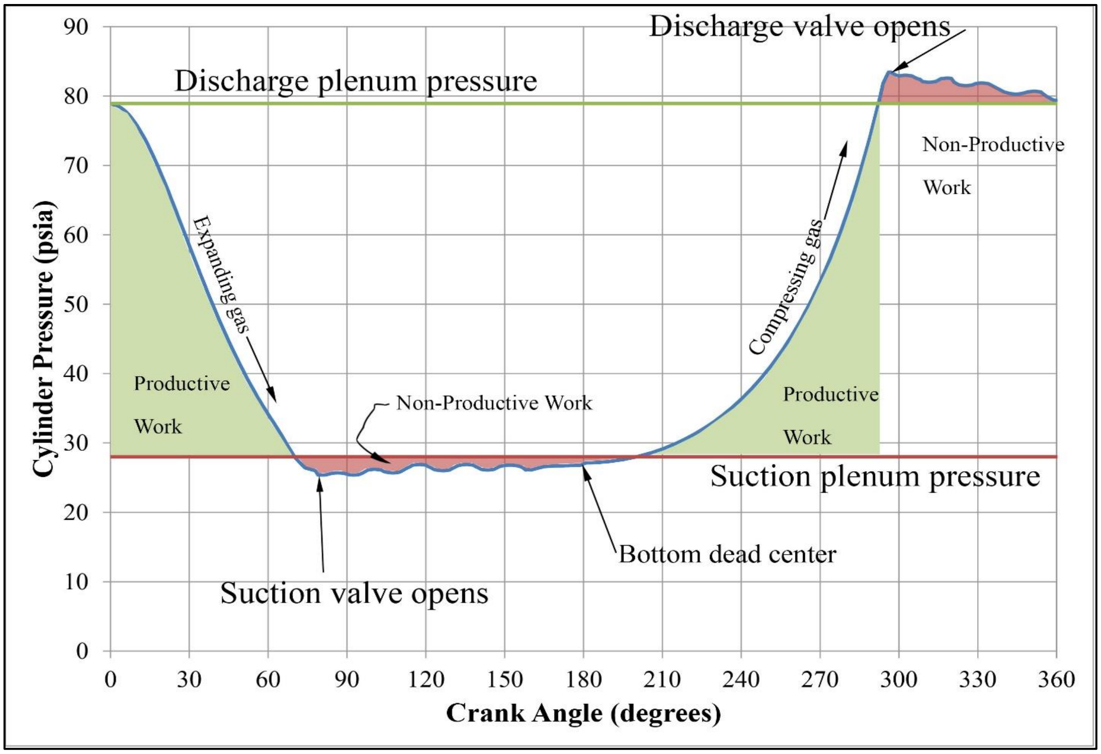

Figure 4: Recip compressor cylinder pressure vs. crank angle

to do more (or less) work to transfer the work towards (or away from) later stages. As discussed below, "clearance" is the portion of the cylinder volume that is in the compression cylinder when the discharge valve closes.

Integral vs. Separable. In an integral compressor there is a single crankshaft shared by the driver and the compressor. A separable compressor is designed for the compressor to be matched up with a driver by the packager, and each component stands alone. Integral compressors were quite common in the early days of internal-combustion-engine driven compressors (i.e., the end of the dominance of steam power), but they have fallen out of favor as separable machines have come into their own.

High speed vs. low speed. Originally, recip compressors were all designated as "low speed," and typically had a maximum speed less than 300 rpm. Recip compressors that can operate well above 1000 rpm were developed to better utilize the load curve of electric motors and industrial engines. Today, people tend to equate "integral" with "low speed" and "separable" with "high speed," and the high-speed/low-speed designation is largely meaningless.

Recip cylinders are lubricated by injecting oil into the gas stream on the suction side of the compressor. If the lubricators are adjusted to be too lean, the pistons and cylinders experience excessive wear and short time to failure. If the lubricators are adjusted to be too rich, excess oil is pumped into the downstream gas (typically as an emulsion that can be very difficult to remove from piping). When the lubricators are properly adjusted, very little (but not zero) oil is injected into the discharge gas.

4.1.5.2 Compressor valves

In a recip, a single space is used for both suction and discharge, so a mechanical barrier is required between the inlet/outlet plenums and the compression chamber. In most recips this mechanical barrier is provided by spring-loaded check valves. When the differential pressure across the valve is high enough (and in the right direction) to overcome spring tension, the valve opens. As the piston descends in the cylinder, the bypassed gas (i.e., the gas in the clearance) must be expanded. As the pressure drops, the expanding gas is trapped in the clearance, eventually the differential pressure gets high enough to open the suction valve and allow suction gas into the cylinder. When the piston reaches the lower limit of travel ("bottom dead center" or BDC) and starts back up, the gas in the cylinder begins to compress, and the differential pressure between the compression chamber and the suction plenum falls below the spring tension, and the suction valve closes. Further up the cylinder, the pressure within the cylinder gets high enough for the differential pressure between the cylinder and the outlet plenum to exceed the spring tension on the discharge valve, and the discharge check valve opens to allow the gas to exit into the outlet plenum. This process repeats every revolution of the crank (Figure 4).

The "stiffness" of the valve springs (i.e., the amount of dP required for them to open) is selectable from a list of valve options. The stiffer the spring, the greater the dP required to open them. It is common to use stiffer springs on the discharge valves than the suction valves because the volume flow rate into the cylinder is greater than the volume flow rate out of the cylinder (since $\dot{V} = {\dot{m}}/{\rho}$ and gas density is higher on the compressor discharge than on the suction) and the gas has more momentum., Thus, on the discharge side one can tolerate a shorter valve-open time.

The shaded areas on Figure 4 represent the work done by the piston on the gas (right hand shaded area), or the work by the gas on the piston (left hand shaded area). Though the shaded areas appear similar in size, far more work is done on the gas during compression than the gas does on the piston during expansion. This is primarily because there a greater density of gas in the cylinder during compression than expansion. Any work done between the suction plenum pressure and the discharge plenum pressure is either input work, or productive work. Any work done outside of these limits is non-productive, or "lost work," which represents some of the irreversibility of the process. Graphically, the efficiency of a recip can be seen as the shaded area between the plenum pressures divided by the total shaded area.

A recip compressor is intolerant of varying suction pressure. If you lower suction pressure, the suction valve opens later (for the same clearance and discharge pressure) which means that it doesn't have as much time for gas to flow into the cylinder. More importantly, the lower pressure inside the cylinder when the suction valve closes requires a higher compression ratio, which causes the temperature of the discharge gas to increase. It is very common for a seemingly small decrease in suction pressure to result in a large increase in discharge temperature.

In general terms the suction pressure on a recip should be held within ±5 percent of design conditions. This means that if the compressor is designed for 30 psia [207 kPaa], the suction pressure should remain in the range of 28.5 psia [197 kPaa] to 31.5 psia [217 kPaa]. This represents less than 7 ft [2.13 m] of liquid level change in a well-bore. If the design suction pressure were 12 psia [83 kPaa], then the allowable liquid level change drops to 1.2 ft [366 mm], which doesn't allow for much variation in wellbore inflow. For suction pressures much over 145 psig, the impact of this limitation is significantly reduced. Pressures that high are common in second and third stage of multi-stage compressors and most mid-stream compressors.

The ±5 percent limitation can be circumvented by installing a pressure-control valve on the compressor suction that is set to take a pressure drop that is larger than expected variation in flowing pressure. In general, this practice has a detrimental effect on reservoir performance, but some see the benefits of recips to be greater than the harm of throttling the reservoir.

4.1.5.3 Compression Ratios

For compressors with suction/discharge valves, the pressure drop (cracking pressure) of the suction/discharge valves must be considered. Since it is rare to have a pressure transducer on the piston-side of the valves, cracking pressure of the valves must be estimated to correct the measured pressures for this cracking pressure (Eq 7). If no manufacturer's data are available to help estimate the valve cracking pressure, using 5 psi [34.5 kPa] for the suction and 10 psi [69.0 kPa] for the discharge provides reasonable answers in most well-site and gathering applications.

$$R_c = \frac{P_{disch} + dP_{disch}}{P_{suct} - dP_{suct}} \qquad \text{Eq 7}$$

Example Calculations. For a specific compressor, the conditions are:

- Atmospheric pressure: 12 psia [82.7 kPaa]
- Suction: 14.5 psig [100 kPag] (26.5 psia [182.7 kPaa]) at 70 °F [21.1 °C] (529.7 R [294.3 K])
- Discharge: 74.5 psig [514 kPag] (86.5 psia [596 kPaa])
- Techniques used:
    - Ratio of gauge pressures: 74.5/14.5 = 5.14
    - Ratio of absolute pressures: 86.5/26.5 = 3.26
    - Ratio of cylinder pressures: (86.5+10)/(26.5-5) =4.49
- Discharge Temperatures (from Eq 5-4, using n=1.3)
    - Ratio of gauge pressures: 314 °F [157 °C]

- Ratio of absolute pressures: 237 °F [114 °C]
- Ratio of cylinder pressures: 290 °F [143 °C]

It is common for a discharge temperature over 300 °F [149 °C] to result in an automatic compressor shut down, so if the correct compression ratio was "gauge pressure," then the compressor would be down on high discharge temperature. If the ratio of absolute pressures was right, then it would be reasonable to run the compressor harder to try to move more gas (the normal design limit for recips is 4.5 ratios per stage). However, that would almost certainly shorten the life of the compressor. Using the ratio of cylinder pressures in this case allowed the calculated value to exactly match the measured value in the field, as one might expect.

For design conditions, engineers try to keep compression ratios above 3.5 and below 4.5. Below 3.5 the efficiency decreases (see below), and above 4.5 the temperatures and mechanical stresses become too great.

4.1.5.4 Recip Compressor efficiency

For recips it is necessary to account for: (1) unswept volume (also called clearance); (2) leakage around valves, piston rings, and rod packing; (3) friction losses; and (4) number of stages.

Volumetric efficiency. The manufacturer of the compressor will provide a clearance number as a percentage of the cylinder volume. If a compressor is configured to adjust clearance in the field, then the manufacturer will have a conversion from number of turns on the variable volume pocket to total clearance.

Eq 8 provides the volumetric efficiency. In the compression ratio example above, with 5 percent clearance the volumetric efficiency would be 0.891.

$$\eta_{volumetric} = 1 - \left(\left(R_c \right)^{\frac{1}{k}} - 1 \right) \cdot \left(\frac{V_{clearance\,\%}}{100\%} \right) \qquad \textit{Eq 8}$$

Compression efficiency. At low compression ratios, the time that the valves are open becomes too large a proportion of the total time, and the adiabatic assumption becomes less valid. Consequently, compression efficiency is:

- Below 1.5 compression ratios → 0.50
- Above 3.5 compression ratios →0.92
- Between 1.5 and 3.5 compression ratios → $0.21 \times R_c + 0.185$

For the compression ratio example above you would use 0.92.

Mechanical efficiency. Friction losses are very difficult to assess. If there is adequate mass flow rate to carry off the heat generated by friction, then its impact is typically small. Mechanical efficiency is generally taken to be 0.95.

Stage efficiency. These three factors are combined in Eq 9 to get a total efficiency for each stage. Each stage will have a different stage efficiency, but for a properly balanced multi-stage compressor (rare), the values will be similar. It is common to assume that all stages are equal to the first stage conditions, and this is probably "close enough" in most cases.

$$\eta_{stage} = \frac{\eta_{volumetric} \cdot \eta_{compression}}{\eta_{mechanical}} \qquad \textit{Eq 9}$$

Total efficiency. The efficiency of the compressor is shown in Eq 10. If the example above is for a single stage machine, then it would have 86.2 percent efficiency. If there are two additional stages with similar compression ratios and interstage cooling, then the compressor would have 64 percent efficiency.

$$\eta_{total} = \prod \eta_{stage} \approx \eta^{n}_{FirstStage} \qquad \textit{Eq 10}$$

Meaning of efficiency. Ideally, compressor efficiency should be a clear indicator of fuel consumption, and a compressor with higher efficiency would use less fuel (with the same driver, same gas, and the same conditions) than a compressor with lower efficiency. Unfortunately, this is rarely the case.

The compressor skid in Figure 5 is reasonably representative. The operator was having a problem loading the second stage (due to high first stage temperature), so he installed a backpressure valve.

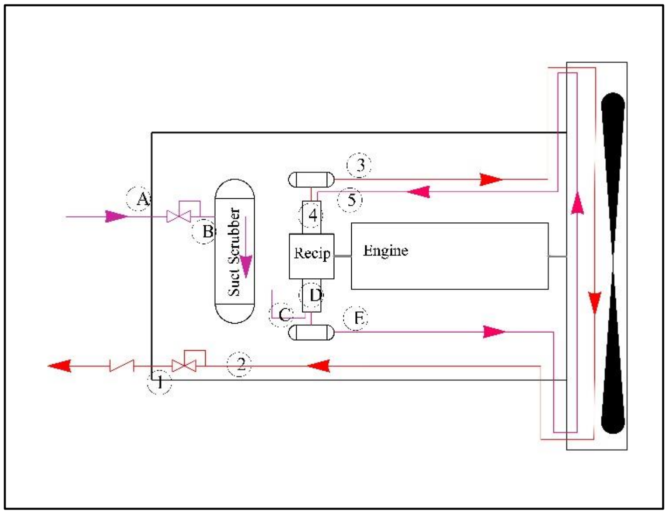

Figure 5: Recip efficiency example

Flowing conditions were:

- q_{gas} → 1 MMSCF/d [28.3 kSCm/d]
- q_{water} → 10 bbl/MMSCF [56.1 m^3/MSCm]
- c_p →0.1925 BTU/lbm/R [0.8060 kJ/kg/K]
- c_v → 0.1473 BTU/lbm/R [0.6167 kJ/kg/K]
- k → 1.306

The conditions at each point in Figure 5 are tabulated in Table 2.

Table 2: Efficiency example data points

	Map point	Press (psig)	Temp (°F)	Specific Enthalpy (BTU/lbm) [REFPROP]
Well-head	A	60.0	70	353.13
After suction controller	B	20.0	68	353.31
Upstream of first stage	C	15.0	68	353.45
In 1st stage cylinder when suction valve opens	D_s	13.3	77	357.72
In 1st stage cylinder at end of discharge stroke	D_d	71.4	251	443.45
Downstream of discharge valve	E	66.9	251	443.72
Upstream of 2nd stage suction valve	5	65.5	120	376.98
In 2nd stage cylinder when suction valve opens	4_s	61.4	127	380.49
In 2nd stage cylinder at end of discharge stroke	4_d	264.8	332	485.31
Downstream of discharge valve	3	255.2	332	485.43
Before backpressure valve	2	250.0	120	372.59
Line pressure	1	160.0	115	372.27

The change in energy from one point to the next will either be $W=m\times\Delta h$, $W=m\times c_p\times\Delta T$, or $W=m\times c_v\times\Delta T$ depending on what is happening in the step.

The energy traverse is shown in Table 3. The normal way to calculate compression hp is to ignore the heat lost to the atmosphere in the inner and after coolers since that is energy that is not usable.

Table 3: Efficiency example energy traverse

	Equation	ΔT or Δh	BTU/hr	hp
A→C	$m\times c_v\times\Delta T$	-2 R	-593	-0.2
C→E	$m\times\Delta h$	90.3 BTU/lbm	181,612	71.4
E→5	$m\times c_p\times\Delta T$	-118 R	-45,700	-18.0
5→3	$m\times\Delta h$	108.5 BTU/lbm	218,187	85.8
3→2	$m\times c_p\times\Delta T$	-192 R	-74,359	-29.0
2→1	$m\times c_v\times\Delta T$	-5 R	=1,482	-1.0
Traverse total (including heat transferred to ambient in cooler)				108.4
Traverse total (without heat transferred to ambient in cooler				155.4
A→1	$m\times\Delta h$	19.14 BTU/lbm	3,697	15.1

Using the excellent Performance.EXE program from Ariel Corporation [Ariel], this compressor should have used 74 hp in the first stage, and 81.4 hp in the second stage for a total of 155.4 hp. (There is about ±5 percent difference in each cylinder from the energy traverse, probably due to a slight difference in the unswept volume of the two cylinders; total energy was the same as the energy traverse calculation).

The compressor skid in this example had a Waukesha F-18 LE driver. Output horsepower calculated from manifold pressure was 220 hp. Fuel usage was 2.54 MMBTU/hr [744 kW-hr/hr] (73 MSCF/day [2.1 kSCm/day]). This would make skid efficiency one of the following:

- Theoretical compressor efficiency→81%
- Compressor net (E→3) vs. engine output → 70%
- Compressor net (E→3) vs. energy in fuel → 22%
- Compressor skid (A→1) vs. engine output→7%

In other words, due to the limitations of the technology and the design choices, this unit applies only ~7 percent of the energy from the engine to compressing the gas up to line pressure. Thus, it uses twice as much fuel as a skid that could transfer 14 percent of the engine output into the gas.

4.1.5.5 Limiting capacity

A recip will move about the same volume with every revolution of the crank. At a constant suction pressure, it will move the same mass of a constant-composition gas each revolution. If the compressor is not keeping up with inflow, then suction pressure will increase along with the mass flow rate per revolution. This means that the work required to compress the gas will increase, and this can cause the driver to stall. It is common to address inadequate compressor capacity by using suction pressure controllers that keep the pressure at the compressor inlet plenum at an acceptable level.

If the compressor is trying to move more mass than is being supplied, then suction pressure will drop, which will lower the mass flow rate per revolution. If the compressor keeps out-running the inflow, the suction pressure will continue to drop until either the mechanical forces get so high the allowable load on the rods is exceeded, or the discharge temperature gets so high that the compressor controls stop the machine. On a recip one tries to prevent over-running supply by progressively:

- Reducing compressor speed
- Increasing clearance
- Removing some of the suction valves (called "crippling the cylinder")

These actions are progressively more intrusive. Changing compressor speed (on an engine driven compressor) is a minor adjustment to a governor that can be done with the engine running. For an electric-motor driven compressor, if the motor has a variable speed drive, then changing speed is a simple adjustment. For very small compressors that are belt driven, one can adjust the size of the sheaves to change the compressor speed.

Increasing clearance is sometimes just a matter of turning the compressor off and turning an adjustment wheel. In other cases, one must remove the valve(s) and add a spacer (called a "chair") under the valve(s).

Completely removing suction valves turns the cylinder into a place in the line that does no work. Suction gas flows into the cylinder as it descends, and then flows back into the inlet plenum as the cylinder ascends.

4.1.5.6 Other recips

The most common compressors in Oil & Gas are high-speed, separable, double-acting recips. There are a couple of manufacturers that still make integral machines (still double acting), but they are fairly rare and tend to be used in locations where the decision maker "has always used" integral machines from a specific manufacturer. They tend to be massive machines that require more extensive foundations and more site work than separable machines do.

The other frequently-discussed recip is a modified industrial internal-combustion engine. Generally, these will be a V-8 configuration with one bank of four cylinders converted from internal combustion to compression by changing the heads. Obviously, these compressors are integral (single crank shaft) and single-acting (traditional connecting rod below the piston can't be sealed). These compressor/engine combinations have a water-cooled block, so the lion's share of the heat of compression is carried away by the coolant. These hybrid machines have reported long-term performance at over 10 compression ratios without excessive gas-discharge temperatures.

4.1.6 Oil-flooded screw compressors

Work began on developing a positive-displacement rotary compressor to overcome surge problems with dynamic compressors in the 1930s at Svenska Rotor Maskiner AB and proceeded through the 1960s (see reference [SRM] for an interesting discussion). In the reference there is a list of companies that have licensed SRM technology, and it reads like the Who's Who of the screw-compressor industry. Initial development was for a dry screw where the male rotor was driven by a motor or engine, and the female rotor was driven by a timing chain to minimize wear on the lobes. Dry screws are available today, but they tend to have a poor reputation in Oil & Gas because of changes in fluid composition can push the machine out of its operating envelope, poor lubrication, and problems with rotor timing.

In the late 1940's, basic patents were issued to SRM for an "oil injected screw" that is functionally identical to the units commonly called "oil flooded screw compressors" or "oil injected screw compressors". Oil-flooded screws still have the male rotor driven by the driver, but the female rotor is driven by the male rotor. The oil injection was intended to: (1) prevent metal-

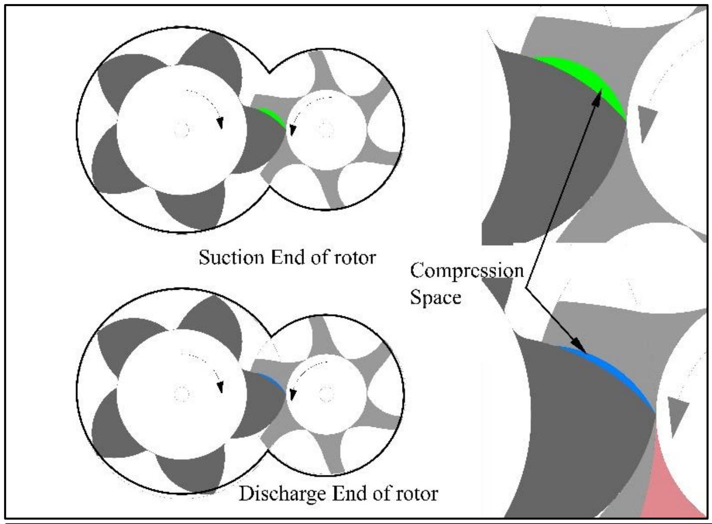

Figure 6: Flooded Screw Rotors

to-metal contact between the rotors; (2) seal the area around the rotors to minimize leak-back; (3) lubricate the rotors; and (4) cool the process.

Initially, oil-flooded screw compressors were used within plants either for refrigeration (generally called "process derivative"), or for compressed air (generally called "air derivative"). Plant compressors usually don't have to be very flexible:

- The oil only has to be compatible with one gas.
- The gas typically has a very low water vapor content (even air is rarely more than 20 percent RH when compressed to useful values).
- The process is consistent enough to allow the differential pressure across the skid to pump the oil.
- Compressors are set on a rigid foundation, so the skids do not have to provide all the structural support.

In the 1990's, oil-flooded screw compressors started being deployed to well sites where the gas was subject to significant change from day to day, let alone year to year. There is not much control over either suction or discharge pressure. To top that off, the gas generally had a very large water vapor content.

Unmodified plant packages (i.e., pre-assembled units) turned out to be an unmodified disaster on well sites. It was immediately found:

- Oil selection had to be compatible with a range of condensable hydrocarbons, not just one.
- The oil-temperature control resulted in the oil being too cool (more about this below), and the control method was too inflexible.
- Differential pressure across the skid was too inconsistent to allow safe use without an oil pump.
- Accumulated solids were quite common, and the package did not anticipate having to open pressure vessels to shovel the solids out.
- The plant skids were too light to allow rough field handling (things like tail-rolling the skid off a truck, or total lack of a prepared foundation).

Over time, packagers have become progressively better at designing packages that are robust enough for field use, but even today one occasionally sees a plant package (i.e., a package with a light skid frame, no clean-outs on suction vessels, no oil pump, simplistic temperature control, etc.) being deployed to a well-site. The second most-important lesson learned on the very steep learning curve with screw compressors was that the packager was far more important than the compressor manufacturer or the derivative industry. It was found that a well-packaged process-derivative machine from one manufacturer would have about the same life expectancy and field performance as a well-packaged air-derivative machine from another manufacturer. We found that a well-packaged compressor from either industry from a third manufacturer would perform just as well. We also found that poorly-packaged machines broke frequently and horribly regardless of the industry or manufacturer.

4.1.6.1 Configuration

The basic shape and relative size of the rotors (Figure 6) was developed mostly through trial and error and has been refined by many companies over the years. Computer modeling has been used to make the rotor both more efficient and less expensive to manufacture.

Notice in Figure 6 that the compression space is much larger at the suction than at the discharge. This relationship is called the "Volume Index" or "VI". A useful representation of the VI relationship is shown in Eq 11. This says that one will reach maximum efficiency if the discharge pressure (in absolute units) is equal to the suction pressure (also in absolute units) times VI raised to the adiabatic constant. Other values for discharge pressure are possible (see below) at a lower efficiency.

$$VI = \left(\frac{P_{disch}}{P_{suct}}\right)^{1/k}$$

$$P_{disch} = P_{suct} \cdot VI^{k} \qquad \textit{Eq 11}$$

Most compressors have VI established in the factory, and it is not field-adjustable. (Some manufactures sell a replacement "VI-plate" that can be changed by mechanics in the field with about a day's work). Some compressors have adjustable VI configurations. These "adjustable VI" or "variable VI" machines often allow the VI to be changed by field-automation on the fly. Many prefer to never allow the field-automation to adjust the VI at all. I have operated compressors where changing the VI was the first step in an unloading scheme, and the results were universally poor (increased failures and machines operating at a bad place in their load curves at all times).

In addition to the VI, we also speak of the "unloader". This is a variable port to take suction gas from near the front of the rotor and dump it back to the suction, thus only doing minimal work on the gas. In process-derivative machines this unloader is nearly always a "slide valve". In air-derivative machines it may be a turn valve, a poppet valve, or missing altogether. There is no real

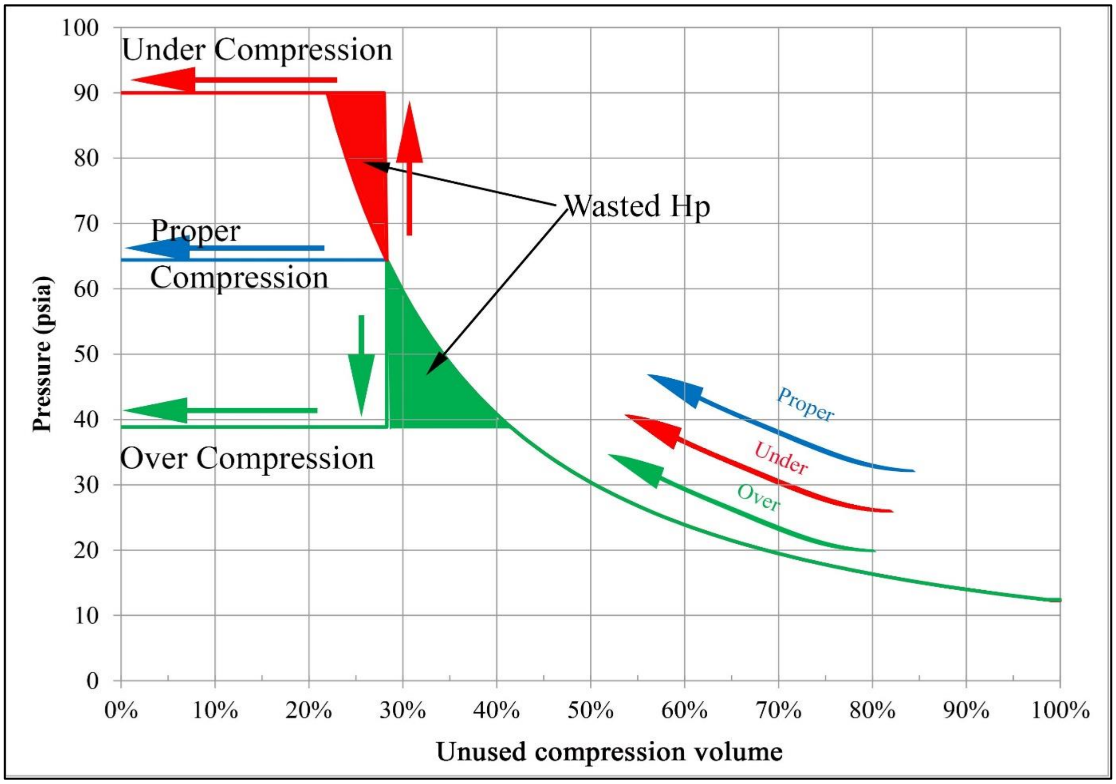

Figure 7: Example Screw Compressor PV optimization

operational difference between turn valves and slide valves. Poppet valves have poor throttling characteristics, so an unloading scheme with poppet valves can be too abrupt.

If unloaders (or variable VI's, for that matter) are included in an automation design, the actuators should be driven with oil instead of gas. Pneumatic actuators on these devices tend to overshoot and often move independent of the control signal.

Some compressor manufacturers include reduction gears in some of their frames that have either step-up or step-down gearing. These integral gears can be a very effective way to maximize driver/compressor performance. Trying to accomplish the same result with 3rd party gear sets has not been as effective since the lubrication requirements of general-purpose gears are not compatible with the characteristics of the oil available on the skid.

Finally, we mention "rotor diameter" and "length to diameter ratio (L/D)". The rotor diameter refers to the maximum diameter of the male rotor. All other things being equal, a larger rotor will move more gas. The larger the L/D ratio is, the larger the compressor capacity, but it's at the cost of lower maximum differential pressure. Shorter L/D ratios move less gas, but at a higher differential pressure. Oil flooded screw compressors are positive displacement machines and the work is calculated using Eq 3.

4.1.6.2 Efficiency

As the gas moves from the suction plate (100% point in on x-axis in Figure 7) to the discharge plate (which is located at about 28 percent in Figure 7), the pressure changes in a predictable manner. A compressor configured to satisfy Eq 11 will smoothly compress the gas up to the discharge plenum pressure, and the gas will flow out the end of the rotor with maximum efficiency.

If Eq 11 predicts a higher pressure than the actual discharge plenum pressure, then the compressor will actually satisfy Eq 11 and then dump pressure to get to actual outlet pressure (wasting energy). This is called “over compression”.

If the actual discharge pressure is higher than Eq 11 then the compressor has to “stuff gas” into the outlet plenum which results in gas compressing gas (the vertical up-arrow in Figure 7) instead of steel compressing gas (the smooth line), again resulting in wasted hp. This is called “under compression”.

The two wedges marked “wasted hp” on Figure 7 are different size. “Over compression” wastes more hp than the same amount of “under compression”. Figure 8 shows why. Notice that

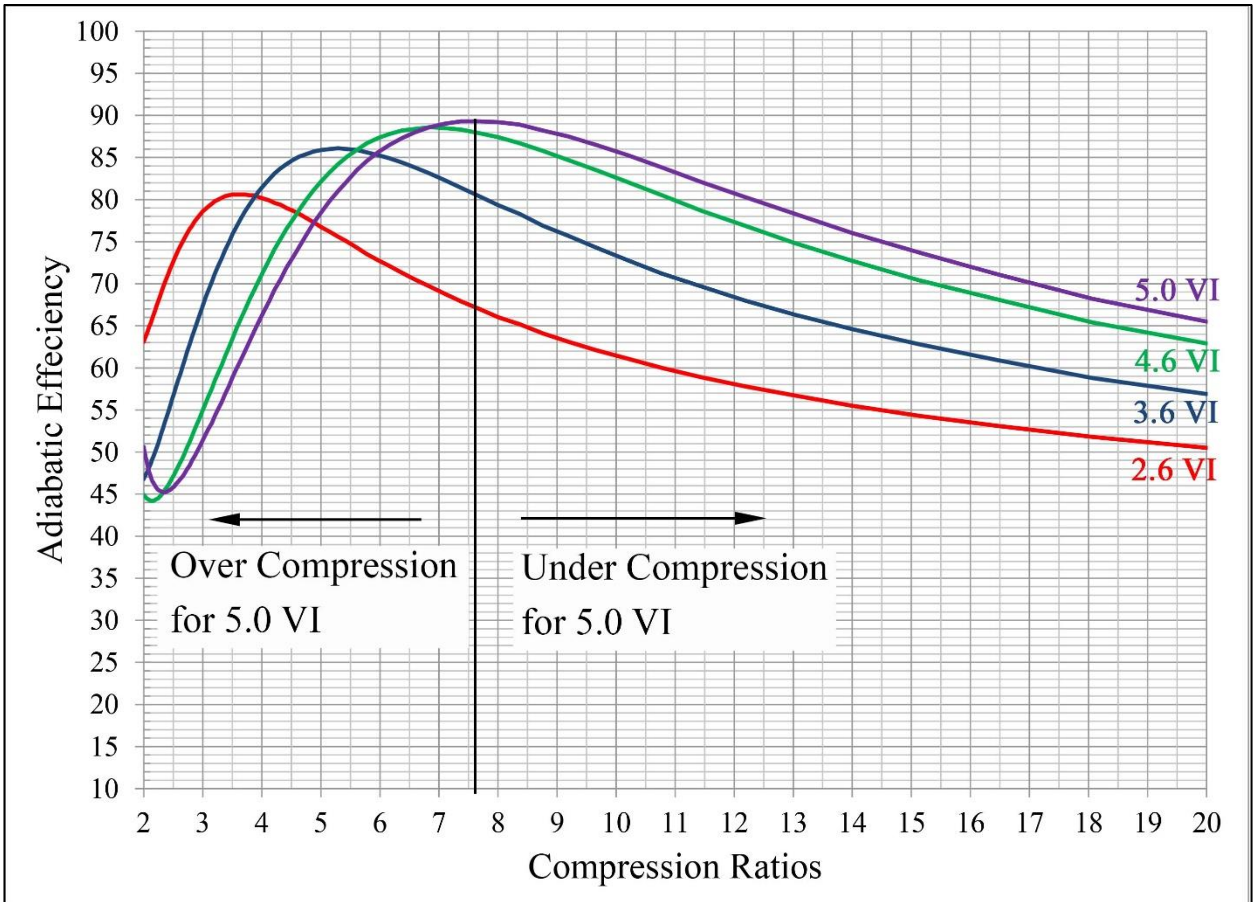

Figure 8: Screw Compressor efficiency

all the curves are much steeper to the left of the peak and then taper off slowly to the right. This is an indication that a slight under compression has a smaller impact on unit efficiency than the same magnitude of over compression. The curves in Figure 8 are based on Eq 12. The peak enthalpy of the gas is calculated at target discharge temperature and theoretical discharge pressure (using the second equation in Eq 11 to calculate discharge pressure). As seen from Eq 12, that peak efficiency is determined by the work done on the oil. If the mass flow rate and the change in enthalpy of the oil are both high, then maximum efficiency is very low.

$$\eta_{screw} = \frac{\dot{m}_{gas} \cdot \left(\Delta h_{gasPeak} - \left| \Delta h_{gasPeak} - \Delta h_{gasActual} \right| \right)}{\dot{m}_{gas} \cdot \Delta h_{gasPeak} + \dot{m}_{oil} \Delta h_{oil}} \qquad \textit{Eq 12}$$

Calculating total work is a two-step process. First, use Eq 3 to calculate the work done on the gas, then you must calculate the work done on the oil and sum the two.

$$W_{oil} = \dot{m} \cdot \left(h_{oilDischarge} - h_{oilSuction} \right) \qquad \textit{Eq 13}$$

It is common for the work done on the oil to be 30-40% of the total work required.

4.1.6.3 Compressor Oil

The compressor oil is key to successful screw compressor operation. The kinds of oil that are considered include: (1) mineral oil; (2) synthetic oil; and (3) semi-synthetic oil. Each has strengths and weaknesses, and each has a place where it excels.

Mineral oil. Least expensive, but it tends to be an excellent solvent for heavier hydrocarbons, which is not desirable. When mineral oil takes on propane and butanes-plus, the characteristics of the oil change until it stops performing per specifications. Mineral oil also tends to begin breaking down above 210 °F [99 °C]. When it has been exposed to a high-temperature transient, the oil does not return to original performance when cooled.

Synthetic oil. Most expensive (usually by a good margin), but it tends to not act as a solvent for heavier hydrocarbons. Synthetic oils are very stable at high temperatures, and values above 350 °F [177 °C] have been reported without permanent degradation. As always, one must follow manufacturer's recommendations for maximum temperature.

Semi-Synthetic oil. Mineral oil and synthetic oil can be blended to achieve specific intermediate properties. These blends are often very cost effective.

Compatibility with water vapor. All the oil types are significantly hygroscopic (i.e., "Readily absorbing moisture, as from the atmosphere"), and have a prodigious capacity for absorbing water vapor. When the oil absorbs water vapor, it: (1) becomes more viscous (i.e., harder to pump); (2) loses lubricity (i.e., need to pump more of it); (3) increases surface tension (i.e., bigger droplets fail to coalesce in the outlet separator vessel); and (4) raises the oil level in the reservoir (i.e., increases foaming and carryover).

There is simply no way to prevent the oil from absorbing water vapor that is present in the gas, and removing the water vapor from the suction stream is very expensive. The only way to successfully operate an oil-flooded screw compressor in raw-gas service is to manage the temperature in the outlet separator vessel to cook the water vapor out of the oil. Figure 9 shows an example of how a small change in discharge temperature can change the process. The inlet gas to this compressor is saturated with water vapor and contains 5,061 lbm/MMSCF [81 gm/SCm]. Inside the compressor the gas is heated from 110 °F [43.3 °C] at 12 psia [82.7 kPaa] to 192 °F [88.9 °C] at 112 psia [772 kPaa].

At these discharge conditions, the gas can only hold 4,225 lbm/MMSCF [70.9 gm/SCm], so 826 lbm/MMSCF [13.2 gm/SCm] stays in the oil, which represents a volume of nearly 100 gal [375 L] of water that remains in the oil for every MMSCF [28.3 kSCm] of gas. If the temperature out of the compressor is raised just 13 °F [7.2 °C] to 123°F [50.5°C] then the gas can carry more water vapor than is present in the system, so it completes the cycle with no water left in the oil, and the gas at 89 percent relative humidity.

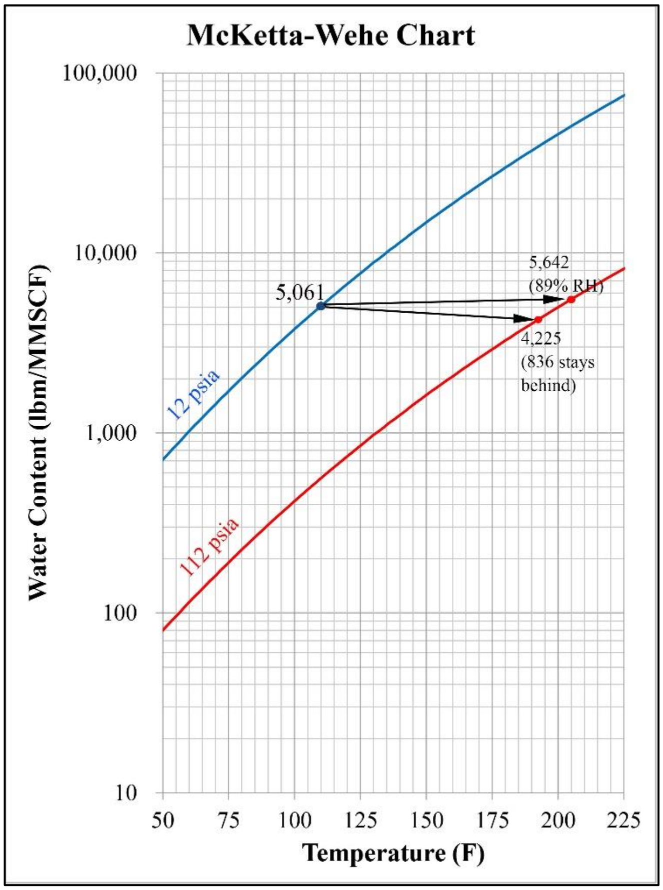

Figure 9: Screw compressor outlet water content

The goal of oil temperature control is to be able to cook this relative humidity out of the oil. Empirical evidence has shown that a temperature above 185°F out of the compressor is generally effective at cooking water out of the oil. Mineral oil usually has a maximum temperature of around 210°F before it experiences chemical changes. Synthetic oil maintains its chemical properties at much higher temperatures (the exact maximum temperature is specific to each oil blend). For any oil, compressor-outlet temperature in the range of 195-210°F has shown to be effective.

4.1.6.4 Outlet temperature

A screw compressor compresses gas. The process can be assumed to be adiabatic. That means that Eq 6 should have some role in predicting the discharge temperature. If a compressor has a suction pressure of 12 psia [82.7 kPaa] at 80 °F [26.7 °C], and a discharge pressure of 112 psia [772 kPaa], Eq 6 predicts the outlet temperature to be 405 °F [207 °C] for k=1.28. That temperature is a very difficult problem for metallurgy, and for operations in general. The injected oil must have a role to play in that temperature. First one should use Eq 6 to calculate the heat of compression. Then one must convert the flows to mass flow rate. Eq 14 is used to determine the thermal energy that the act of adiabatic compression added to the process.

$$Q_{gas} = \dot{m}_{gas} \cdot c_{pGas} \cdot \left(T_{dischTheo} - T_{suct} \right) \qquad \textit{Eq 14}$$

The discharge temperature can then be determined using Eq 15.

$$T_{disch} = \frac{Q_{gas} + T_{suctGas} \cdot \dot{m}_{gas} \cdot c_{pGas} + T_{oilIn} \cdot \dot{m}_{oil} \cdot c_{pOil}}{\dot{m}_{gas} \cdot c_{pGas} + \dot{m}_{oil} \cdot c_{pOil}} \qquad \textit{Eq 15}$$

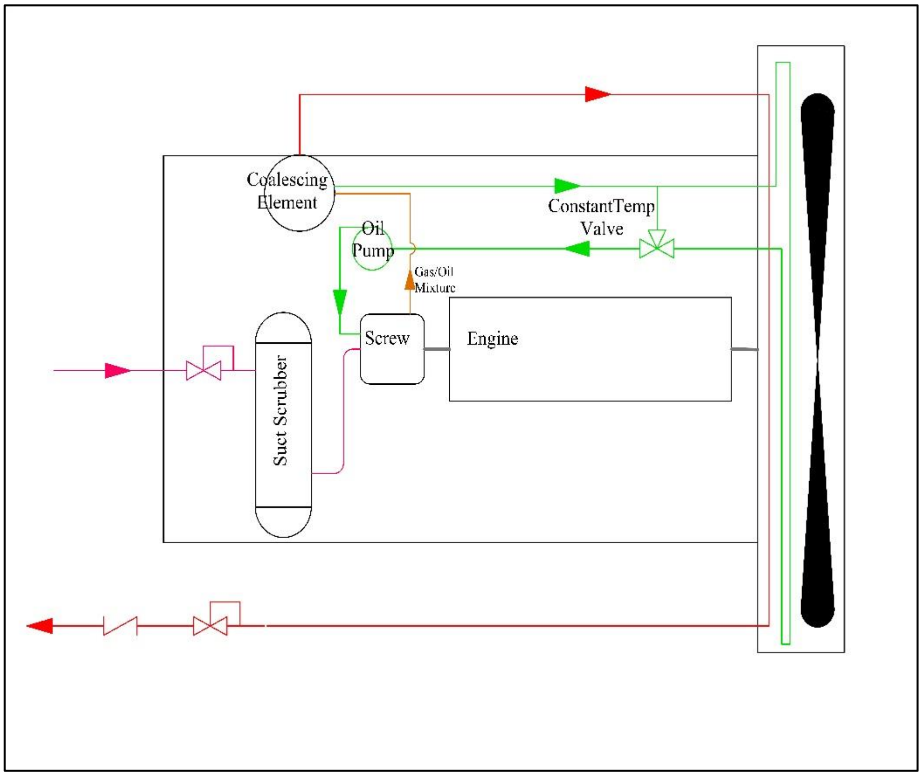

Figure 10: Screw compressor "standard" temperature control

4.1.6.5 Oil temperature control

The “standard” way to control oil temperature is shown in Figure 10. In this common scheme, the 3-way “temperature control valve” controls the temperature of the oil into the screw by bypassing a portion of the oil going to the cooler to try to maintain a constant input temperature. The set point for this parameter is optimal for one exact set of conditions; however, if the gas flow-rate increases, or the discharge-pressure decreases, etc., that set point will no longer be optimal. The long and the short of the story is that controlling the inlet temperature is an unwise practice that came from experience with plant compressors (where water-vapor in the gas is much lower), and it really should have stayed there.

The packagers of the first screw compressors deployed for coalbed methane failed to understand field use to the extent that the 3-way valve was set for 140 °F [60 °C] into the screw, but the oil flow rate was expecting about 3 compression ratios (the machines actually had 12 compression ratios). Therefore, the oil was flowing so quickly that the temperature out of the compressor was only 142 °F [61.1 °C], and there were a considerable number of serious issues, and very high oil-replacement/replenishment costs.

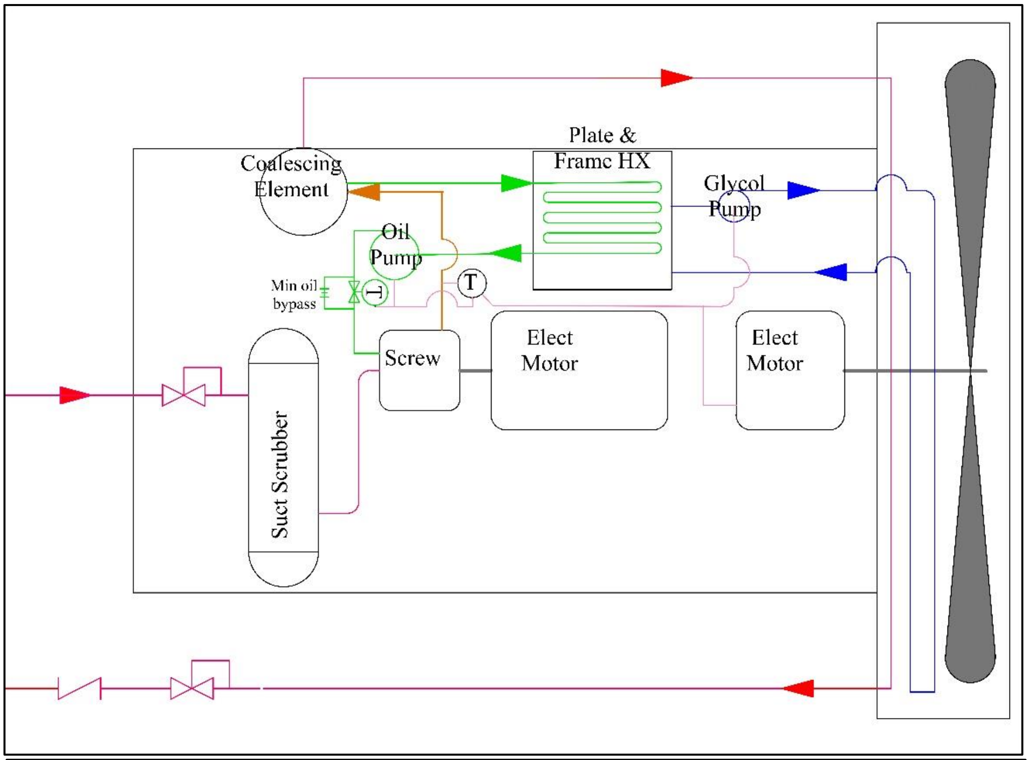

Figure 11: Flooded screw compressor effective temperature control

These skids were so basic that they didn't even have a temperature gauge anywhere on the skid let alone on the compressor outlet.

After several months of fighting with controlling the wrong parameter, it was realized that life gets easier if you try to control the right thing. The "right thing" in this case is to control the compressor discharge temperature in the face of frequently changing conditions. In Figure 11, there is a temperature sensor measuring the temperature out of the screw. Based on that temperature, the PLC controls the temperature of the oil to the screw (it goes through the following steps in sequence with a 2-minute pause before going to the next step):

1. Adjust the speed of the glycol pump (can be adjusted to a minimum, and then on the next time the PLC looks at the glycol pump it can be turned off)
2. Adjust the setting on the thermostatic valve on the oil pump discharge (this valve can go fully shut, the orifice in the bypass is sized for required lubrication and minimum oil-injection)
3. Adjust the speed of the fan (can go to a minimum, but not zero)
4. Adjust the speed of the oil pump (can go to a minimum, but not zero)
5. Repeat

A group of skids designed with this temperature control scheme was able to run for 3 years with zero unscheduled downtime and nearly zero added oil. This skid had variable speed drive on all electric motors. Before this skid was designed, the operating company was reluctant to set an electric-motor- driven compressor on a well site because that puts the weakest link (e.g., the power supply) under someone else's control—a hard concept for a well-site engineer. After reviewing the results of this compressor skid, I am prepared to set a genset to run site facilities with electric power. Genset's combined with variable speed electric motors have a high capital cost, but low operating cost. Payout occurs in the first couple of years.

4.1.6.6 Oil pressure management

As mentioned above, plant machines historically did not have oil pumps, but instead rely on the differential pressure across the skid to move oil. This can work in steady-state operations, but well-site and gathering operations are rarely steady state. Experience has shown that field compressors without oil pumps will tend to be on the top of down-time lists and failure reports. On engine driven compressors, oil pumps can be run off the pony shaft. On electric skids the oil pump should have its own variable speed drive.

4.1.6.7 Coalescing element

We often call the big lump of steel on the backside of the skid a "coalescing filter". It is not a filter. The coalescing elements look somewhat like filter elements, but they serve a different function. The coalescing element is intended to force small droplets (that are buoyant in the gas stream) to crash into other droplets and coalesce into larger drops that are not buoyant in the gas. As is normal with any piece of equipment, there is a range where they are more effective. It is recommended to try to get the same magnitude of velocity in a coalescing element as the target velocity for a separator mist pad.

4.1.7 Positive Displacement Compressor Selection Criteria

An oil-flooded screw compressor will generally require more energy input than the same task with a recip. More energy relates directly to higher operating costs (either fuel that you cannot sell or electricity that you must buy). The strengths and weaknesses of Recips and Oil-Flooded Screw compressors are described in Table 4.

Table 4: Positive Displacement Compressor Comparison

Recip		Oil-Flooded Screw	
Strengths	Weaknesses	Strengths	Weaknesses
When everything is perfect, best use of hp	Narrow suction range	Wide suction range	Moving oil requires energy
Operating staff is familiar with them	Not tolerant of changing conditions	Very tolerant of changing conditions	Technology unfamiliar to operating staff
Few consumables	Valves are very high maintenance	No rods, no valves	Oil is expensive

Some packagers do field machines well	Difficult to balance stages	No stages to balance	Few packagers understand field requirements
High operating pressures	Low compression ratios per stage	High compression ratios per “stage”	Limited operating pressure
Rugged and reliable	High temperatures	Very low temperatures	
	High maintenance cost	Low maintenance cost	
	High capital cost	Lower capital cost	

It would be easy to assume from Table 4 that recips are junk, and oil-flooded screws are wonderful. That would be a mistake. It is appropriate for the first compressor on a wellhead to be an oil-flooded screw because of the flexibility on the suction side. However, after that first compression it makes a lot of sense for the remainder of raw-gas compression to be recips. Moreover, there are good reasons for commodity gas to be moved with dynamic compressors (very high throughput in a single frame and no injected oil).

4.2 Dynamic Compressors

A dynamic compressor is a device that uses added kinetic energy to accelerate a gas at nearly constant pressure. A common type of dynamic compressor is the “centrifugal”, where Figure 12 illustrates that the ribs on the impeller are farther apart at the rim of the impellor than at the eye in the center. Consequently, the flow in the impeller is not precisely isobaric, but it is reasonably close. The velocity of the gas is rapidly increased in the impeller. Once the gas enters the volute, Bernoulli’s Law describes how pressure and velocity change as a function of the cross-sectional area of the volute.

For well-defined flows, dynamic compressors can move a lot of gas. Joining a gas turbine to a centrifugal compressor can compress many times the volume of gas as a positive displacement machine can for the same footprint. This characteristic is vitally important on offshore platforms where available real estate is in short supply. A centrifugal compressor can do fewer compression ratios per stage than a PD compressor, but it is possible (at skid design time) to add stages to the compressor, like stages are added to ESP downhole pump.

Dynamic compressors are the preferred technology on offshore production platforms (small footprint), plant outlets, and mainline compression (no oil injection). Their use in raw gas is very limited.

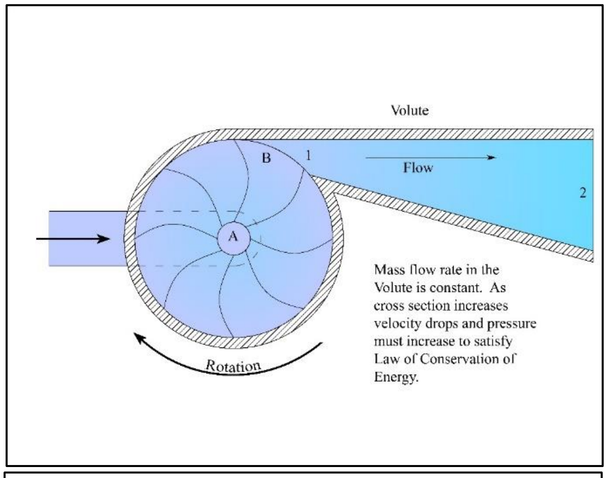

Figure 12: Centrifugal Compressor

4.2.1 Thermocompressors

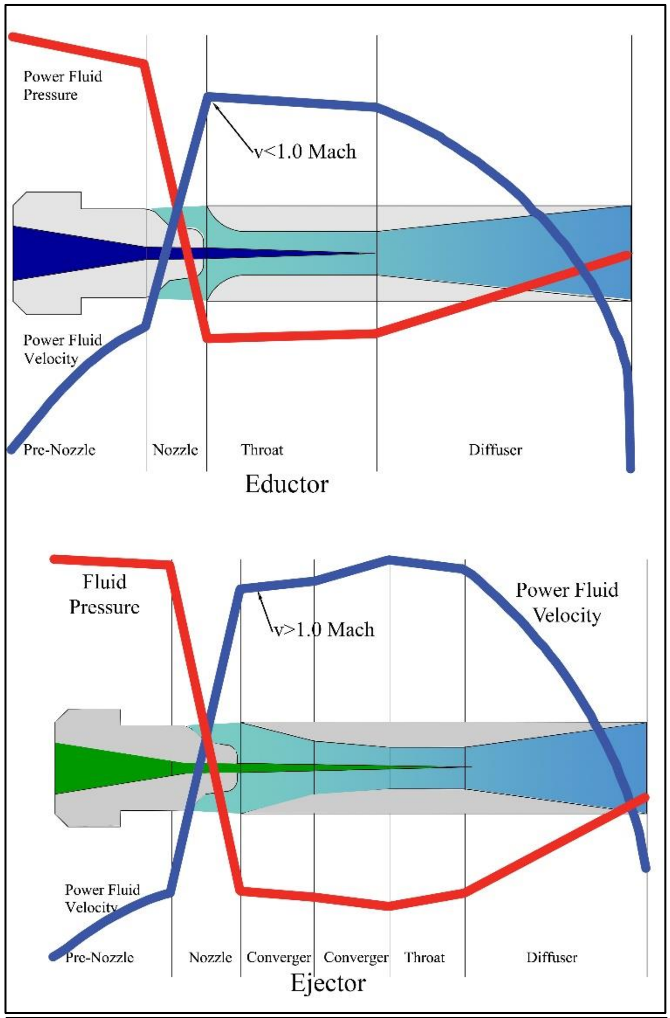

Figure 13: Thermocompressor pressure/velocity map

"Thermocompressors" are devices that exploit the laws of thermodynamics to allow a high-velocity fluid to transfer flow energy to another fluid in a manner that leaves the second fluid at an intermediate pressure between the pressure of the power fluid and the pressure of the suction fluid. The terminology is a bit confusing:

- Venturi: a constricted area of a pipe, not a thermocompressor
- Eductor: a thermocompressor designed for a liquid power fluid
- Ejector: a thermocompressor designed for a gaseous power fluid

Most terminology doesn't matter, but in this case, using an eductor with a gaseous power fluid or vice versa has poor results. As seen from the notation on Figure 13, an ejector can reach supersonic velocities, but an eductor cannot. The ejector has a choke point near the end of the nozzle where a shock wave is created, and the gas velocity reaches sonic velocity. That choke point is followed by a divergent section that increases velocity at a constant pressure. This high-speed fluid has significant dynamic pressure and is quite dense. The combination of high speed and enhanced density supplies significant momentum to transfer to the suction fluid. An eductor does not provide these enhancements to flow because flow of a largely incompressible fluid is not amenable to very high velocities.

4.2.2 Eductor vs. Ejector

Both eductors and ejectors are in the family of equipment that includes air ejectors, evacuators, sand blasters, certain kinds of paint sprayers, hose-end sprayers, and jet pumps. In both cases the high pressure/high velocity power fluid entrains the suction fluid at the no-flow boundary between the two fluids, which causes energy to transfer from the power fluid to the suction fluid. The combined stream is left at an intermediate pressure.

The exhaust pressure for an eductor is limited to about 1.5-3 times the suction pressure (in absolute units). The exhaust pressure for an ejector can be as much as 10 times suction pressure, but there is a strong relationship between compression ratios and mass flow rate of the power fluid. Higher ratios require significantly more power fluid. Except in very specific cases, a compression ratio greater than three requires uneconomic quantities of power fluid. Oddly, the more power fluid you use, the higher the overall efficiency of the unit will be. Low power-fluid usage is associated with efficiencies as low as 30 percent. High power fluid usage is associated with efficiencies as high as 70 percent.

4.2.3 Cases

It can be difficult to visualize where it makes sense to take 100 psig [690 kPag] gas and pump it through a device that ends up with 40 psig [276 kPag] gas that must be compressed to usable pressures. We will present four cases where eductors and ejectors have been used to provide a real benefit.

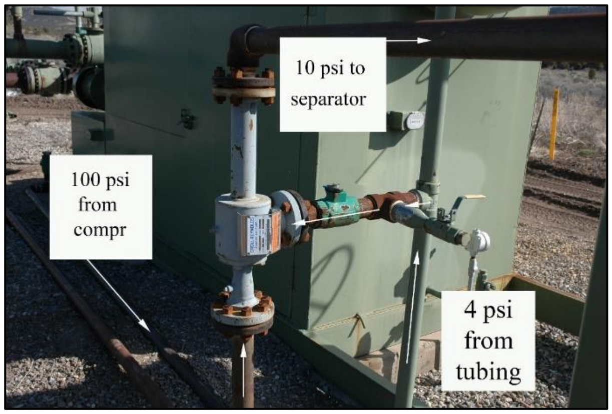

Figure 14: Critical flow ejector

4.2.3.1 Critical flow

Fluid flow in a well-bore is a complicated system, especially at low pressures. The target pressure for optimizing gas flow may not be close to the optimum pressure for flowing liquid. A common way to solve this problem is to restrict casing flow to force an adequate amount of flow up the tubing. However, when the target pressure gets very low, managing that differential becomes impossible in real time. One solution is to install a second compressor to manage the flowing tubing pressure independent from the flowing casing pressure to optimize each one. Figure 14 shows an application of this concept. The ejector takes power gas from the oil-flooded screw compressor that is drawing from the production separator at 10 psig [68.9 kPag] suction. At the well's flow rate, this separator pressure results in a flowing bottom-hole pressure of 12 psig [82.7 kPag]. This corresponds to zero flow rate up the tubing (because of the cracking pressure of the check valve on the tubing). The ejector provides just enough compression horsepower to pull the tubing to 4 psig [27.6 kPag] with 250 MSCF/day

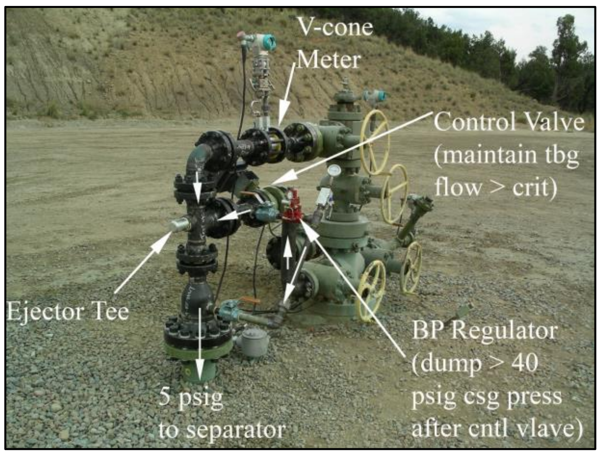

Figure 15: Tubing flow control ejector

[7.1 kSCm/day] flowing up the tubing. It was determined that the actual critical flow rate at these conditions was 220 MSCF/day [6.2 kSCm/day]. Thus, this configuration was able to keep the well unloaded for over 4 years.

The ejector in Figure 14 used 28 hp [20.9 kW] of the 500 hp [372 kW] oil-flooded screw compressor output. Boosting 250 MSCF/day [7.1 kSCm/day] from 4 psig [27.6 kPag] to 10 psig [68.9 kPag] is a 10 hp [7.5 kW] job, so the ejector was 36 percent efficient while avoiding the cost of a downhole pump.

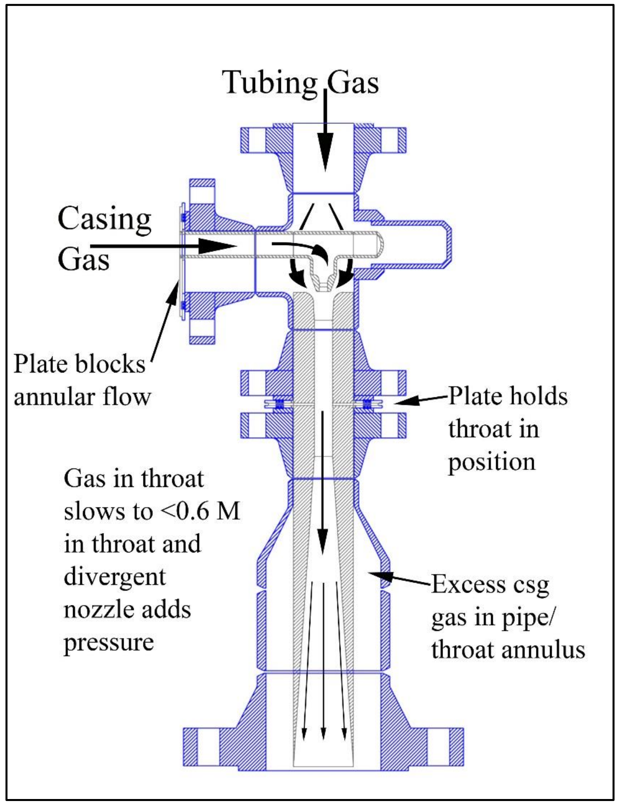

Figure 16: Ejector tee internals

In the first four years of this project, critical flow ejectors were installed on 32 CBM wells, and all of them showed flatter declines and reduced variability. The installed cost (for sites that already had well-site compression) was $4.5 k/site, and the contribution to net income of this $144 k investment was $16.2 million over four years. There was a change in staff in 2003, and all the ejectors were replaced with nodding donkey sucker rod pumps, so long term performance of the ejectors was not assessed.

4.2.3.2 Tubing flow control

One of the techniques for deliquifying gas wells using reservoir pressure is *Tubing Flow Control*. Under this method, a control valve is placed on the casing, and a flow-measurement device on the tubing. While the tubing flow is above critical (i.e., the gas flow rate that is required to reliably drag water along with it in vertical flow), the casing is allowed to flow. At very low pressure, the amount of throttling required on the casing can cause the gas flow to become unstable, and even log the well off. The well in Figure 15 was experiencing this problem. There would be periods of high flow rates (on the order of 3-4 MMSCF/day [85-113 kSCm/day]) followed by several days of no flow or slugging flow. The ejector tee in Figure 16 was designed to use the pressure drop up the tubing to allow the casing valve to be opened further (while still maintaining critical flow in the tubing). The tubing-flow control equipment and software were not changed; everything done was after the control valve on the casing.

The ejector tee (Figure 16) was designed to have very high efficiency (upwards of 65 percent) with 40-45 psig [276-310 kPag] power gas. This is at the cost of poor efficiency outside that range.

After the installation:

- If tubing flow was greater than the pre-set value, then the program bumped the casing control valve toward open.
- If the pressure downstream of the control valve is less than 40 psig [276 kPag], the casing gas just goes through the ejector nozzle without doing much work.
- If the pressure downstream of the control valve is in the design range, then the ejector sucks on the tubing.
- If the pressure downstream of the control valve is greater than 45 psig [310 kPag], then the backpressure valve allows some of the excess gas to bypass the ejector.
- The compressor maintains the exhaust pressure at 5 psig [34.5 kPag].

The initial daily cumulative production was just over 25 percent higher after installing the ejector tee, and the project paid out in 8 days. Everyone was so happy with these results that they got greedy. They surmised that if 45 psig [310 kPag] would give them a 25 percent uplift, that 85 psig would have to give them a 50 percent uplift (see below for a discussion of the effects of changing power gas pressure; they are not good). When that caused tubing pressure to increase dramatically, they decided that the tubing-flow control ejector was a failed idea and removed it. That project was chalked up to the importance of making sure that field staff understands the technology being deployed.

4.2.3.3 Add-a-stage Compressor

A CBM well was making 600 MSCF/day [17 kSCM/day] into 9 psig [62.1 kPag] suction pressure, and the well was experiencing significant slugging of liquid. Efforts to lower that well-head pressure failed, and we were unable to lower the suction pressure in spite of the fact that the compressor driver was running at only 35 percent of rated power. This was a 2-stage integral recip with a poured concrete foundation that would cost upwards of a million dollars to replace. Reconfiguring a 2-stage integral recip for three stages would be nearly as expensive.

Our solution was to pull the tubing out of the well-bore for vacuum operations, and install the ejector in Figure 17 between the well-head and the separator. What the compressor now saw was:

- Recip compressor suction pressure: remained the same at 9 psig [62.1 kPag]
- Recycled power gas: 1,700 MSCF/day [48.2 kSCm/day]
- Well-head gas: 900 MSCF/day [25.4 kSCm/day]
- Engine load: 65 percent of rated power (engine-fuel increased 10 MSCF/day [288 SCm/day])
- Ejector efficiency 48 percent

At the same time the flowing well-head pressure was -5 psig (6.75 psia [46.5 kPaa]), and the well stopped slugging. This unit (which cost 20 k USD) has delayed the requirement to replace the well-site compressor for 20 years so far.

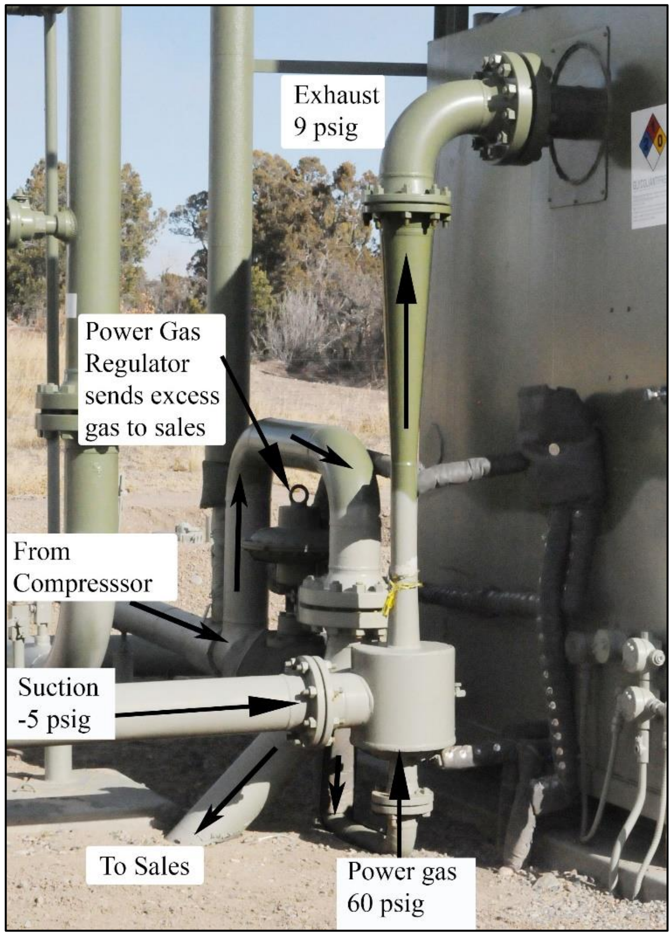

Figure 17: Add-a-Stage thermocompressors

4.2.3.4 Add a compressor

There are few things that will turn a cooperative neighbor into an adversary faster than installing a compressor in their front yard. In the example depicted in Figure 18, "Well 2" had been free flowing for nearly 10 years when the adjacent plot of land was sold to someone who built a house sharing both a million-dollar view and a fence line with Well 2. Wellhead pressure inevitably declined, and the well needed a compressor. However, installing a noisy compressor would spoil a good relationship with the adjacent property owner. The solution was to modify the compressor on Well 1 (it was a single-stage recip which was designed to allow conversion to a 2-stages in a couple of hours of site work) to get high enough pressure to supply an ejector that could pull on Well 2, and to exhaust into the pre-existing line pressure. The necessary ejector was of a similar size to the "Add-a-Stage" ejector with a different nozzle and throat.

This project had passed the design stage when I stopped working in that basin, and I never heard if it was ever installed or not. However, as of this writing, there is still no compressor on the well with the million-dollar view.

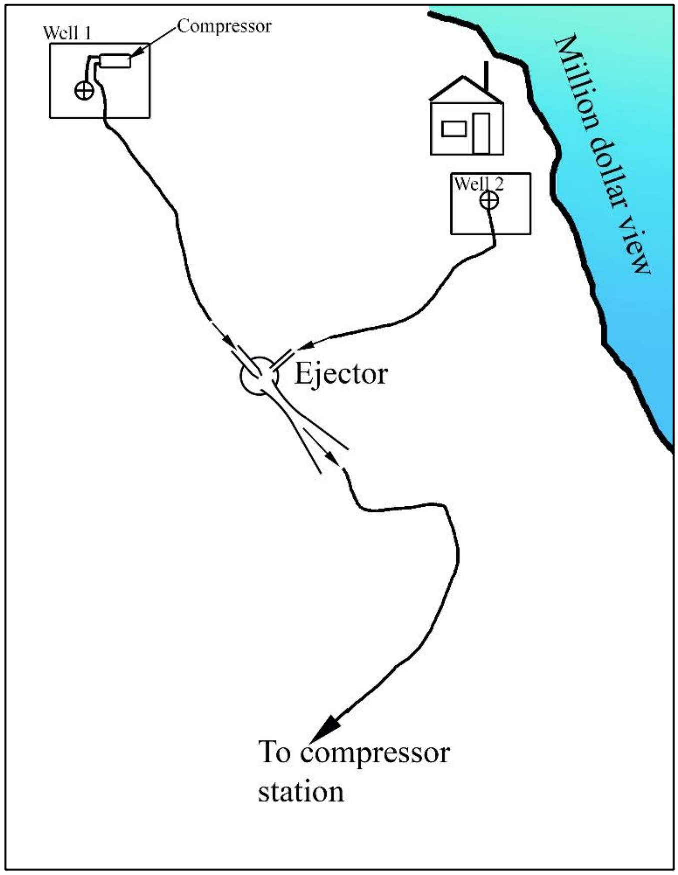

Figure 18: Add-a-Compressor

The most famous "add-a-compressor" ejector application [Caltec] was in the North Sea. The BP Indefatigable 23A platform handled its own gas, plus 11 satellite platforms using 120,000 hp [89.4 MW] of centrifugal compression. This equipment, plus necessary separators, dehydrators, valves and piping took most of the available space on the platform. Shell proposed bringing the gas from two of their platforms to the 23A platform, but these wells needed 50 psig [345 kPag] line pressure instead of the 80 psig [552 kPag] that the compression was designed for. Caltec, Ltd. designed an ejector able to boost the gas from the Shell platforms from 50 psig to 80 psig using the installed compression on the platform. Everything on a North Sea platform is very expensive, but even at those prices this project paid out very quickly.

4.2.3.5 Case studies conclusion

These four cases were included to provide reasons for why one should care about thermocompressors. Every time I get a new problem that has resisted solution (because no one ever calls me first, everyone looks for cheaper alternatives themselves before agreeing to my exorbitant hourly rate), I ask myself "is there any application for eductors or ejectors as part of the solution set?" The answer is usually a resounding "NO", but occasionally I can solve a problem with an ejector that cannot be solved economically any other way.

4.2.4 Rule of twos

The design of an ejector or eductor involves some complicated arithmetic (the printout of the FORTRAN program for sizing is close to 100 pages each for eductors and ejectors), but for scoping purposes one can use the "Rule of Twos". If one can satisfy these bullet points, then there is a reasonable chance that a thermocompressor could be part of the solution (all pressures are in absolute terms, and all volumes are mass flow rate, or volume flow rate at standard conditions):

- Achievable suction pressure is more than half of maximum exhaust pressure (i.e., if the exhaust pressure is expected to be less than 50 psig [345 kPaa] then the lowest suction pressure you can reach is 25 psia [170 kPaa]).
- Minimum power gas pressure is twice exhaust pressure.
- It takes twice as much power fluid as you are planning on pulling through the suction.
- You are going to get something like 50 percent of the work out that you put in.

Each these rules are useful for answering the question "can an eductor/ejector solve part of this problem?" However, specific designs can be significantly different from one another. The "add-a-stage" compressor does a 3:1 compression ratio with a 1.89:1 mass flow ratio. It is 48 percent efficient but requires the power-gas to exhaust ratio to be 3.4:1. The tubing-flow-control ejector tee is designed for up to 8:1 power gas to suction gas ratio to ramp up the efficiency, but the design trade-offs made the operating range for the power gas very narrow.

In general terms one must remember:

- Only a limited amount of mass can flow through the throat.
- Increasing compression ratio requires increasing the mass flow rate of the power gas (more work requires more power input).
- Adding power fluid decreases the amount of suction-mass that can flow through the throat (see first bullet point).
- To increase compression ratio generally requires a design change, not an operational change.

4.2.5 Ejector response to changing conditions

There are four parameters that can be adjusted in the ejector system: (1) power fluid pressure; (2) exhaust pressure; (3) suction pressure; and (4) power fluid flow rate (which changes with power fluid pressure, and thus is not independent). Table 5 summarizes the expected response to these potential changes.

The two highlighted cells in the table may appear counterintuitive, so some explanation is in order. First, increasing the power gas pressure increases the power-fluid flow rate into the throat, leaving less space available for suction gas. Thus, suction flow rate must decrease. Second, the fluid dynamics within an ejector are quite complex, and the transition to and from compressible flow creates shock waves that limit communication upstream and downstream. In short, the steam chamber and the throat do not "know" what the exhaust pressure is, as long as it is low enough to allow the compressible/incompressible transitions to occur (i.e., "choked flow"), which means that decreasing the exhaust pressure will not change the suction flow rate at all.

Table 5: Ejector response to changes

		Power fluid pressure	Exhaust Pressure	Suction Pressure	Suction flow rate
Power fluid pressure	Decrease		Constant	Increase	Decrease
	Increase		Constant	Constant	Decrease
Exhaust pressure	Decrease	Constant		Increase	Unchanged
	Increase	Constant		Constant	Decrease
Suction pressure	Decrease	Constant	Constant		Decrease
	Increase	Constant	Constant		Increase

4.2.6 Thermocompressor conclusion

Thermocompressors are not magic, nor any facsimile thereof. They are a tool that should be in every facilities engineer's toolbox. There are thousands of potential applications across a world with millions of wells. Being the engineer that finds the next *Indefatigable* application may turn one into a rock-star for a minute, but dang, a minute is far better than never. Look for wasted horsepower, for example:

- A compressor that can't be loaded to a good place on the driver load curve.
- Well-head chokes.
- A pump that is too big for the application.

Then look for someplace that needs a little horsepower:

- Well-site too close to housing for a compressor.
- VRU requirements.
- Focused power required.
- A need for a sump pump.

Finally, use the rule of twos to try to match the wasted power to the power requirement. It is surprising how often there is a good fit, and if power is being wasted anyway, the costs can be scandalously low. In fact, thermocompressors can be so inexpensive that it can be economic to take advantage of temporary conditions. For example, if a well's tubing flow must be choked for the first 1 or 2 years of production, and if an eductor project has an 8 day payout, there is "free money" for 722 days by using that wasted power for two years.

4.3 Vacuum operations

The amount of water vapor that a gas can carry as water vapor in vacuum conditions can be enough to deliquify a reservoir. Additionally, requirements to remove the vapors from atmospheric tanks forces vacuum operations in many cases. On the other hand, there is significant resistance from gathering companies to operating wells in a vacuum due to (largely irrational) fears of oxygen corrosion. Sometimes the gathering companies have a concern about ingesting enough air to support combustion, but this is a threat without any basis in the physical world. There has never been a compressor fire or explosion that could be traced to vacuum operations. To support combustion, six times as much air must be ingested as the gas being aggregated, which is simply not possible. Regardless of resistance, the economic benefit of operating in vacuum conditions will increasingly overrule the resistance.

When we talk about vacuum operations, the first technology that comes to mind is liquid ring compressors. These machines are similar to a fan or dynamic blower operating in an asymmetrical housing mostly filled with liquid. The blades sling the seal-liquid to the casing-wall, and as a chamber between blades moves from the near casing-wall to the far casing-wall, the liquid moves outward and creates a low-pressure area at the suction. As the blades move towards the discharge side, the casing-wall gets closer and raises the pressure in that chamber.

The choice of seal liquid is important (e.g., a water seal will tend to evaporate, and most glycols and oils have compatibility issues with heavier hydrocarbons). With a maximum compression ratio of 4.5, if the target pressure is 2 psia [13.8 kPaa], the maximum discharge is only 9 psia [82 kPaa]. Therefore, an additional stage is required just to get above atmospheric pressure. These compressor casings tend to be limited to about 20 psig [138 kPag] operating pressure. On the face of it, liquid ring compressors seem to be a reasonable choice for vacuum operations, but maintenance and make-up seal liquid make them a poor choice for well-site use.

The greatest strength of centrifugal and axial compressors is the ability to move large volumes of gas at very constant conditions. There are no technical reasons why one could not use this technology for vacuum operations; however, the lack of mass in the flow stream would be an issue for maintaining internal temperatures (high rotational speeds and low mass flow rate is a recipe for high friction losses and high discharge temperatures).

Thermocompressors can have a role in vacuum operations, but subsequently removing the power fluid from the combined stream can be a problem. There are a couple of vapor-recovery-unit manufacturers who use an ejector to pull on the tank(s), and then use recips to boost the ejector-exhaust (positive suction pressure on the recip) up to line pressure while providing power gas back to the ejector. Eductors tend to lessen the chance of sucking in the tanks since direct connection to a recip can pull too hard, whereas the choked flow in an eductor helps limit the mass rate that can be withdrawn, especially in cold weather.

Recip compressors can certainly work in vacuum service, but that requirement of being within ±5 percent of design conditions creates a real problem as suction pressure moves into a vacuum. If design suction pressure is 2 psia [13.8 kPaa], then we want to control the suction pressure to ±0.1 psi [689 Pa]. This is quite unlikely to be consistently doable, and temperature control will be very difficult.

The best choice I have found for well-site vacuum operations is oil-flooded-screws. With the ability to provide compression ratios up to 20, it is a single step from a deep vacuum to a positive pressure that is high enough to be useful. Also, the injected-oil mass-flow is adequate to control temperatures and friction losses.

There is an apocryphal story (circulated by salesmen of process-derivative screws) that air derivative screws are inappropriate for vacuum operations. This story goes something like: air-derivative screws have the low-pressure end of the screw at the shaft-penetration end of the casing (true), and the shaft seals are inadequate to keep air out (false). They go on to say that process-derivative screws have the high-pressure end at the shaft penetration and will not suck in air. The only true part of this story is that the typical air-derivative screws have the suction on the end of the casing with the shaft penetration. In fact, both the shaft seal and the journal bearings are pressurized with oil, and every air-derivative screw manufacturer will provide an affidavit that certifies that their compressors are rated for vacuum service without modification.

4.4 Fuel gas

Gas compression needs a reliable and economic power source. Most well-site compressors are engine driven, or electric-driven with an on-site engine-driven genset. The engines require clean, dry gas, but well-site gas tends to be dirty and wet. The typical solution is to use something called a "fuel gas dryer" and a great deal of wishful thinking. These units are characterized by:

- Smaller than 6.0 in [152 mm] ID to circumvent the requirements of the Boiler and Pressure Vessel Code (BPVC).
- Fairly short (usually less than 18 in [457 mm] long).
- No mist pad.
- No level control.
- No pressure safety valve.
- Manual water drain.

These units are far worse than worthless. They do collect some water in spite of themselves, and a person must manually drain them often. If the water content of the gas is 4,000 lbm/MMSCF [64 gm/SCm], and the engine burns 75 MSCF/day [2 kSCm/day], then there is 36 gal/day [136 L/day] of water going through the dryer. However, the unit holds only 2.1 gal [8.2 L], so it doesn't take much condensation to fill the dryer and start carrying over liquid water instead of the water vapor that was inherent in the gas.

Several field techs were asked how often they drained their fuel scrubbers, and they all said "about once a week" which likely meant "monthly". They were then asked to drain all of their dryers into a bucket daily for a week. They grumbled but did it. Every one of them reported that on wells with compressors or pump jacks, the scrubber was full of liquid every day on every unit.

That means that the dryers were probably full within a few hours of being drained. Collecting water in a place that requires manual intervention to drain was a bad idea for gathering systems, and it is an even worse idea for fuel-gas systems. Water in fuel gas is the number-one cause of compressor, genset, and pump downtime on well sites. I recommend that if a device does not have an automated dump-valve, then remove the device. A straight piece of 1-inch [25 DN] pipe is far better than blowing the gas through a water bath. If the dryer does have an automated dump, then frequently confirm that it is still working (these little dump valves have a terrible track record for plugging off).

For very cold-weather operations, it can be economical to use a deliquescent (salt) dryer that can actually remove water vapor, but this is a serious commitment to maintaining the salt level and keeping the brine drained off.

4.5 Compressor control

When we talk about "compressor control," we are really asking "how do we match inflow at our target pressure to the capacity of the compressor?". If the capacity of the compressor is less than the well can produce, then inlet pressure will increase until the two match. Unless one can install additional compression, there is not much one can do about the compressor having less capacity than the supply, except to install a suction-control valve to keep the flow into the compressor in line with the compressor capacity.

The other end of the spectrum is more interesting. When the compressor has more capacity than the well, one must "shed rate", or reduce the flow capacity of the compressor to match the inflow. Compressor control schemes are almost exclusively rate-shedding programs.

There are three common compressor "control valves" that should **not** be part of program logic to control the compressor, since all three types are external to the compressor. The valves are: (1) suction pressure control valve; (2) discharge back-pressure control valve; and (3) recycle valve.

Suction pressure control valve. The ability of a compressor to protect itself from too much suction pressure is quite limited. Consequently, a suction pressure control valve can be installed in front of the compressor to establish the maximum pressure that is allowed to come into the compressor suction. Ideally, these valves should have a near-zero dP in "steady state" operations to minimize the amount of reservoir energy that is wasted. When a suction controller is locally controlled, it is called a "Max-inlet control". When a suction controller is controlled within a range of values (called "range control"), it requires a PLC. Range control is sometimes used to keep multi-stage machines in balance, and it will always have a non-zero dP. The need for this type of logic is a positive indication that the wrong compressor is being used for the job.

Discharge back-pressure control valve. It is occasionally necessary or desirable to increase the dP across a compressor skid. This can be done to increase the compression ratio on a screw compressor to heat up the oil, for example. Discharge back-pressure control valves can be used on recips to balance stages. There is almost always a better way to accomplish one's goals than to install a discharge back-pressure control valve, but people do it as a stop-gap, or because they don't know any better. If one is going to use a discharge back-pressure control valve, it should always be locally controlled. PLC control does not add value and can risk having the valve non-responsive at key times.

Recycle valve. A compressor going out of service due to low suction-pressure is undesirable. To prevent this, recycle valves are often used to send discharge gas back to the suction. This is a common practice on machines without speed control or unloaders, but recycle valves are occasionally seen on machines with speed control. On one skid, I was distressed to see that the PLC controlled the recycle valve, and the priority in the PLC was to operate the recycle valve before it activated speed control. This is the least efficient option.

One of the instructors at Ariel's Advanced Service School said "Gas you never compress is infinitely efficient, compressed gas you throw away is zero percent efficient." This sentiment has stuck with me over the years. There are times when one has no option but to install a recycle valve, but it needs to be locally controlled and never part of the program logic.

4.6 Local vs. PLC options

Programmed Logic Control (PLC) devices become more powerful and more capable every year. It is common to REALLY want to use the latest "Gee-Whiz" device, but sometimes that is not the best-ever idea. I recently reviewed a compressor-skid design that had taken the "Gee-Whiz" to amazing lengths. In addition to every external valve being PLC-controlled, this skid had a blow-case with an electric level switch on the suction scrubber that went to the PLC. The PLC told the dump cycle to begin (this involves opening the dump valve and opening the power gas valve). The problem that they wanted to look at was that, at least once a day, the compressor went down on high-suction-scrubber level. I'd never seen the compressor or the design, but the downtime report told all: the time stamp on every level transient was within 90 seconds of the top of the hour. The PLC was reporting to the central database at the top of the hour, and when it received a notice to drain the blow-case, it had to wait until the data transfer was completed (in a queue with every other field device trying to update the database at the same time). While waiting for the data transfer to end, the scrubber filled with water. This was an easy fix because they had used electric level switches that were compatible with the electric dump and power valves, and it was reasonable to terminate (and power) the level switches at the valves. However, it is not always that easy.

Some of the other functions on that skid were harder to correct, so we installed a second PLC to handle data transfer while the functions were being re-assessed.

Well-site automation has historically been one-controller-one-end-device, and that is an appropriate technology mix for nearly everything on a compressor skid. A PLC is likely required for capacity control, but that function needs to limit the levers it can pull to speed and unloader valves that need more complex logic than can be economically provided with a local device.

4.7 Capacity control

It is common in rate-shedding situations to include early use of unloaders (on screw compressors), but this rarely works as well as people hope it will. Rate-shedding schemes have a variety of costs, primarily involving reduced efficiency:

- Reducing speed within a design range that honors the driver torque curve is the most efficient (i.e., it costs zero dollars to not compress gas).
- Adjusting engine manifold pressure to keep rpm constant under changing loads is next most efficient.
- Unloader valves are less efficient, but still viable.
- Recycle valves are least efficient and should be a last resort.

The three most common control sequences that are in general use are: (1) suction control; (2) fuel-manifold pressure control; and (3) suction control with manifold pressure override.

Suction control. This is different from a suction controller. In this scheme, the PLC checks the suction pressure, and adjusts the driver speed to keep suction pressure within a prescribed range. This control must adjust the engine fuel-manifold pressure as the load changes in response to the speed change.

Fuel-manifold pressure control. This applies only to engine-driven compressors. An internal combustion engine gains or loses speed in response to changing load. Controlling fuel-manifold pressure allows the compressor to maintain a constant speed under changing load. If suction pressure is constant (usually accomplished by taking a large dP across a suction controller), then constant rpm is good. However, a large dP across a suction controller is not good.

Suction control with manifold pressure override. This scheme is a combination of the two control schemes described above. When manifold pressure is less than the maximum, the PLC controls on suction pressure. If the manifold pressure approaches the maximum, then the priority in the PLC shifts to the fuel-manifold pressure and begins ignoring suction pressure.

The most effective load shedding sequence I've ever seen is for screw compressors:

- Adjust compressor speed between a minimum and a maximum with the unloader closed
- If the speed reaches the minimum and suction pressure is still dropping, operate the unloader (suction pressure control)

- If the speed reaches the maximum and suction pressure is still increasing, operate the unloader to keep the driver from exceeding maximum power

This scheme is called "Suction pressure control with power override" and it works well for screw compressors. For recips, the only lever one can pull is speed control.

Where to put sensors? Typically, we sense suction pressure after the suction controller, and sometimes after the suction scrubber. When the dP across both the suction controller and the suction scrubber is near zero, then this is just fine. But, what about start-up conditions? Say a well has been shut in for a week and near well-bore pressure has built up above both normal operating pressure and the design suction pressure of the compressor. One will be able to draw this pressure off eventually, but how long will it take? With the usual placement of the suction-pressure transducer, the compressor does not have any indication that there is a huge dP across the suction controller, and the program logic does not place a priority on pulling the head pressure off the well. If you move the sensor to the well side of the suction controller, the PLC sees the head pressure and shifts into "DRAWDOWN" mode to lower it.

In a competitive reservoir this difference is worth thousands of dollars. It doesn't take many of these transients to pay for sending an automation tech to the site to move the transducer.

4.8 Lease vs. Buy

The discussion concerning owning compression vs. leasing it comes up more often in compressor decisions than in any other upstream capital decision. The economists say that the only differences are: (1) cost of acquiring capital; and (2) salvage value of equipment.

Large production companies typically have a very low cost of capital because they tend to have a very good credit rating (not always, but often enough to be significant). Compressor leasing companies tend to have a higher cost of capital than large production companies. Based on cost of capital alone, it is rare for a decision to be "lease" instead of "buy". But what about salvage value? If one can re-deploy a piece of gear immediately, then the "salvage value" is very good. If one can't find a place within one's operation to re-deploy it, then the salvage value could easily be less than zero (i.e., one must pay rental on the space where the machine is sitting and rotting away). Some of the supposedly second-order effects that should go into the lease-vs.-buy decision are:

1. How long is the compressor expected to be on the original site? The longer it is there, the more the analysis should favor purchase.
2. What will you do with it when you are done with that site? Does that disposition have a reasonable chance of still being viable in 2 years? 10 years? 30 years?
3. What value will it transfer at? This can be a major stumbling block when a machine that has been running for several years needs an overhaul prior to being redeployed. Does it

leave the lease at salvage value, or at "used" value? If it leaves at salvage value, is there a place to charge the overhaul cost before it is "sold" to the next lease?

4. What partner permissions are needed in order to authorize disposal and/or replacement? A leased machine does not become part of the Oil & Gas-lease equipment, so the monthly cost of the lease and the removal cost generally do not require an "Authorization for Expenditure (AFE)". Moving a purchased compressor certainly does require an AFE, and any business partners may have the right to take possession of the compressor rather than paying for the overhaul.

Having managed fleets that: i) leased compressors from a third party; ii) compressors were owned by the company and leased to the wells; iii) compressors jointly owned by another production company and leased to the wells; and iv) compressors owned by the wells (which was the worst of all worlds). It can easily take six months or more from the time it is decided to move a compressor from one well to another, until the move can actually be accomplished.. On too many occasions, by the time the move was authorized, the machine was no longer right for the target location.

The other fleet options had similar lead times to each other and were all better for the reservoir than for a well to own a compressor. For fields where one lacks certainty on the duration of the decline period, leasing from a third party (generally higher cost) has the positive characteristic of being able to just let the third party take the compressors back after a (fairly short) initial contract period. By that point, the wells' decline rates will be better understood to make informed decisions about compressor placements going forward.

Whatever convoluted capitalization scheme the economists come up with, one needs to make very certain that one completely understands the terms of the "lease agreement." This includes understanding what is required before one removes, replaces, or reconfigures any, or all of the machines in the fleet. Before agreeing to the lease agreement, one should run some scenarios. For example, one may ask, "if we need to take a 'Size A' compressor off joint-interest Lease XYZ, send that compressor to Lease MNO that we own 100 percent, and replace the 'Size A' compressor with a 'Size B' compressor that came from the yard, what reporting and accounting processes must be triggered?" Then one must go through the contract to see if the terms are not too onerous. I did this on one contract, and found that the fleet compressors we were setting up (jointly owned 50/50 with a working interest owner who was in *most* of our operated wells 50/50) could sit only on wells that were owned 50/50 with that owner. This limitation certainly was not in the best interests of the field as a whole.

4.9 Compressor conclusion

Table 6 briefly describes some key elements of selected compressor technology. The last column is a general assessment of the applicability of the technology for unattended wellsite use. Unattended is the important concept, since centrifugal compressors are used on many offshore platforms all over the world with good success—those facilities tend to have manpower available 24 hours/day, and even if a compressor is running on an unmanned facility, the platforms are mostly very well automated with call-outs to manned facilities.

Table 6: Compressor technology comparison

	Eff	Limiting parameter	Max ratio/ stage	Typical use	Well site use?
Liquid Ring	40-50%	Discharge press	5	Deep vacuum into lp	No
Dry Screw	50-65%	Discharge temp	4	Control air	No
Eductor/Ejector	40-70%	Power fluid flow rate	10	Focus hp	Yes
Axial	60-70%	Discharge press	1.1	Large volume, low head	No
Centrifugal	65-75%	Discharge temp	2.5	Large volume	No
Oil-flooded screw	70-88%	Differential pressure	20	Varying suction	Yes
Recip	78-92%	Rod load or discharge temp	4.5	Varying discharge	Yes

Compressors are everywhere in Oil & Gas, and far too often technology and operating decisions are based on "what the last guy did" rather than "what the reservoir needs." I have spent many hours in conversations with people who were certain that the only viable technology was "recips" or "Ajax recips" or "Gas Jacks" or "flooded screws" trying to filter the fear and superstition from actual facts. Sadly, the discussions have always been very short on facts. I've had a PhD mechanical engineer say "reciprocating compressor technology is the highest efficiency, and the only rational choice is based on fuel consumption related to efficiency." Then, when I showed them an efficiency example, their response was "you manufactured a case that showed recips in a bad light, the real world is not like that". I had admittedly selected a real-world example that illustrated my point, but I could have selected any one of several hundred similar cases.

There is one rational basis for selecting compressor technology—what optimizes ultimate profitability of the reservoir? If that answer is to install a recip for a few years, and then install a flooded screw (including reasonable capital costs for the swap out and reasonable redeployment assumptions), then that is what one should do. The idea that the first compressor set on a site will be the last compressor set there is rarely consistent with the best interests of the reservoir. When I designed a compressor fleet for a group of CBM wells, the one position from which I never deviated was that the machines had to be interchangeable. The relative position of every suction flange relative to the position of the discharge flange and to the edge of the skid was the same, and every machine had the same size flanges from suction to discharge and from skid to skid. This allowed me to move compressors from site to site with very low costs, and we made 362 compressor moves in seven years. The design allowed for optimizing the impact of compression on the reservoir while keeping operating costs for compression very low (about $0.04/MSCF [$1.41/kSCm]). When this philosophy was abandoned (that same fleet had only about 10 compressor moves in the following 14 years), the operating costs started increasing while production dropped rapidly. Few things are more expensive than a “cost avoidance” policy.

Choice of technology is key, but close behind it is the choice of packager. There are examples of every manufacturer’s compressor working well. There are also examples of the same models of machine that work poorly. The difference is almost always the packager. One of the best examples of packager differences is seen in Figure 19. In both pictures, the package is the same: Both skids had the same 5,000 hp [3730 kW] engine driving the same 4-stage, 6 throw separable compressor (with the same size cylinders in the same places on the frame). Both packagers were working from the same company specifications (including a requirement that there be no threaded connections on the skids). The requirements included a sight glass on the suction scrubber.

The company on the left-hand picture took all the specifications literally. Without critical analysis of those specifications, they hung a sight glass on the vessel with flanged valves. The packager on the left-hand side did not consider what vibration would do to a rigidly mounted, cantilevered load with only two points of support that could move independently. This put enough force on the structure to rip the sight glass off the vessel, which it did several times.

The packager on the right-hand side saw a potential problem and obtained an exception to the "no threaded connections" mandate. This packager did several other things properly as well. The sight glass weight is carried by a beam tied into the skid base. The piping from the vessel is heavy wall, 2-in [50 DN] swedged down to 1-inch [25 DN] after the flange, and then turned 90° so that vibration would translate to independent torque in the two pipes that could not set up harmonic vibrations. Finally, if the repeated cyclical torque did result in a failure, it would be in the piping, not in the BPVC certified part of the vessel structure.

The same producer engineering team reviewed the design for the two skids and never questioned the layout of the sight glass. There were simply too many details on the drawings for anyone who had never had a nozzle fall off a pressurized vessel to question the attachment of sight

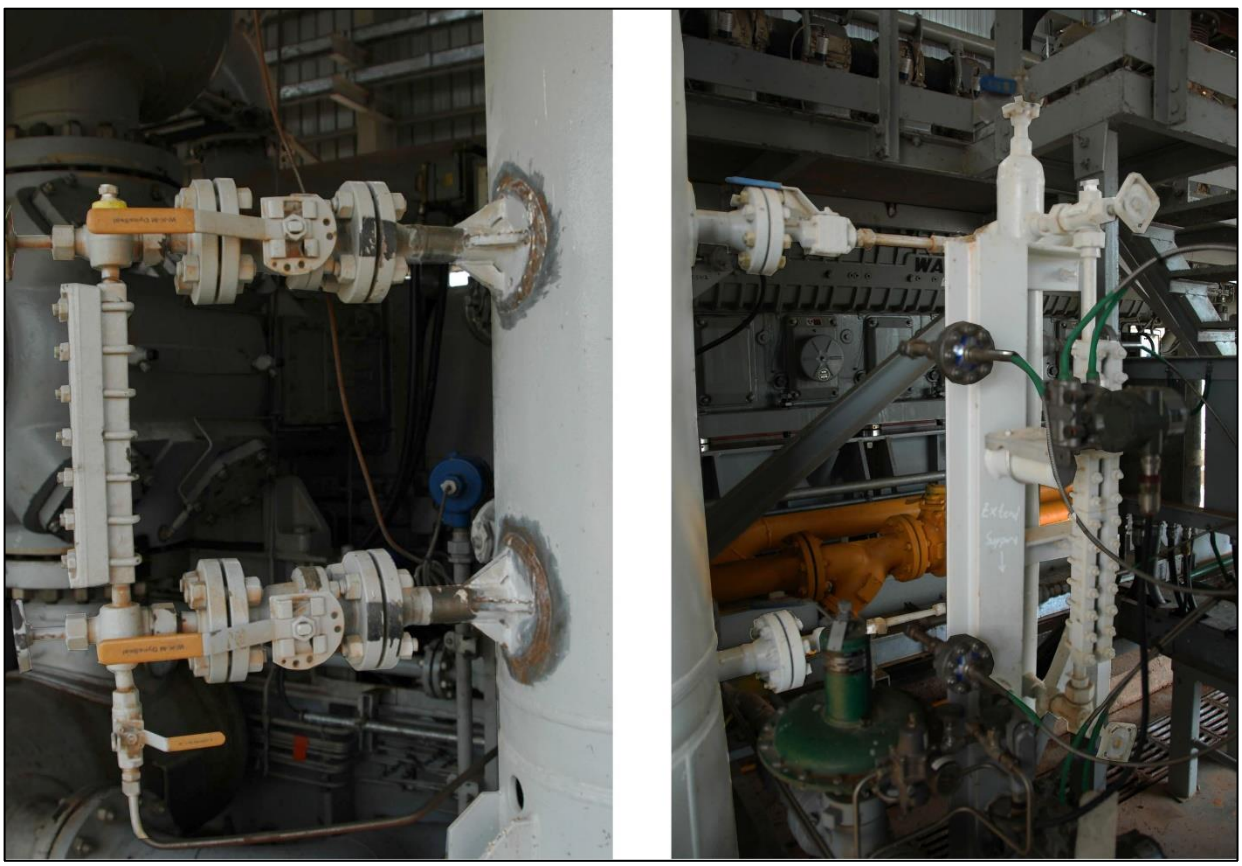

Figure 19: Example of Compressor packaging

glass. Obviously, the packager on the right-hand side in Figure 19 had a design engineer who had seen an unsupported nozzle fall off a vessel, and thus designed the sight glass to prevent that. While there are also many examples of the packager in the right-hand picture getting it wrong, they tend to do a workmanlike job, but so does the packager on the left-hand side. I prefer the packager on the right-hand side and have used them often, but I still spend considerable time reviewing design documents and make sure that I visit the fab shop several times during the fabrication process. Further, if I find an error that I approved, I accept my mistake and pay for the fix.

4.10 References

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