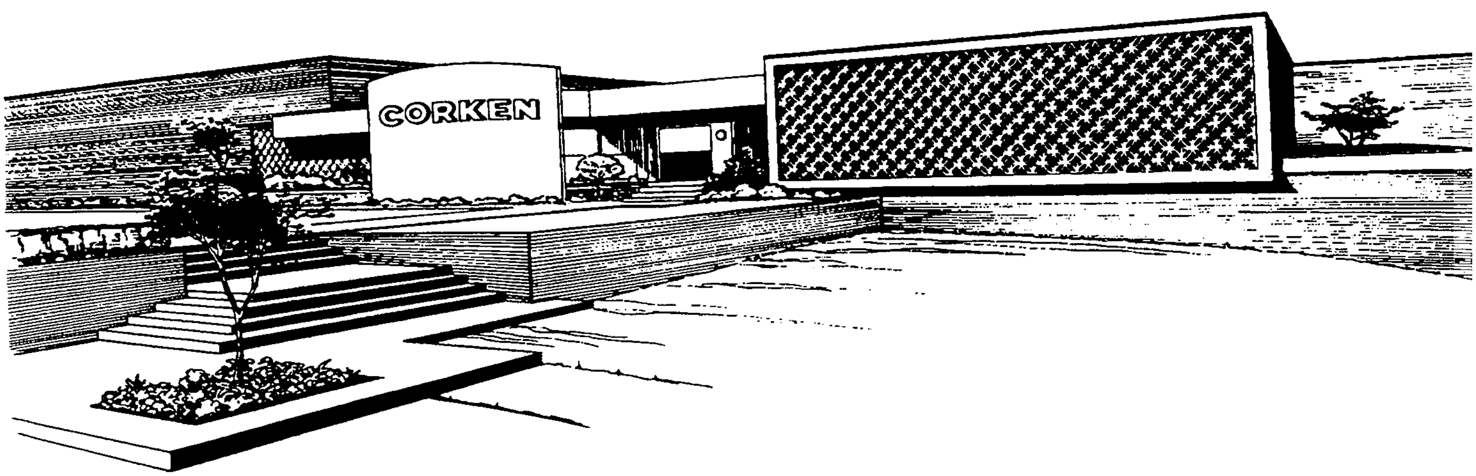
CORKEN

AIR AND GAS COMPRESSOR SELECTION MANUAL



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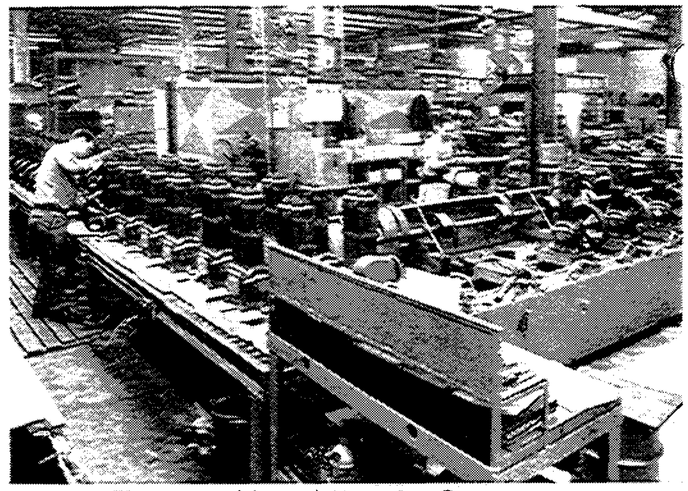
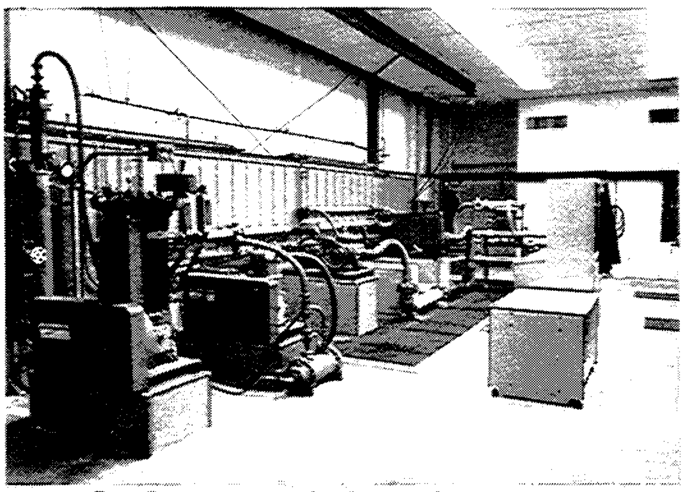
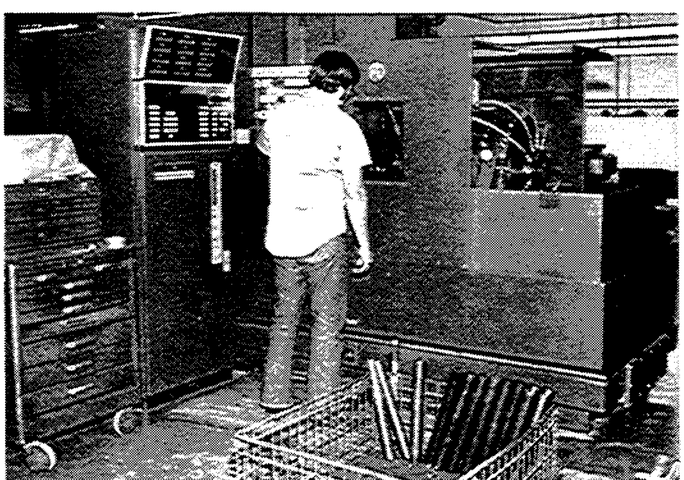


The Assembly and Mounting Department.

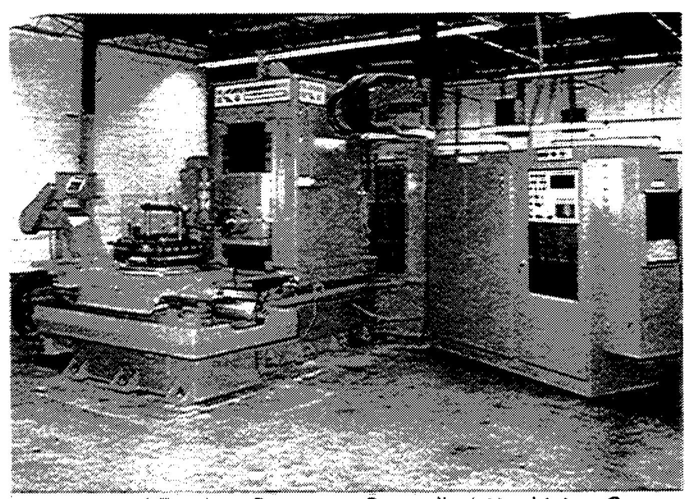


Test Equipment in the Research Department.

Storage Area for finished parts.



Warner and Swasey Numerically Controlled Turning Lathe.



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SELECTION MANUAL

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INDEX

CHAPTER ONE
GAS FUNDAMENTALS

In the first chapter we will review the principles that govern gas behavior. In the following chapters these principles will be used to predict how gases will behave in practical gas compression problems.

1.1 WHAT IS A GAS?

Pure substances can exist in one of three different phases: solid, liquid or vapor (gas). Each of these phases, which we are all familiar with from our everyday experiences, have unique properties the compressor engineer must carefully consider.

The phase a substance is in can be scientifically defined by the presence or absence of two conditions: 1) Fluidity 2) Compressibility

A material is fluid if it flows when subjected to shear stresses. Shear stresses are created by forcing a fluid to flow. Pouring, pumping or blowing are ways of creating shear stresses in a fluid.

A material is compressible if it expands or contracts when the pressure is decreased or increased.

Using these two principals we can construct a simple chart to determine if a certain quantity of matter is solid, liquid or vapor.

	FLUID	COMPRESSIBLE
SOLID	No	No
LIQUID	Yes	No
VAPOR	Yes	Yes

For a substance to exist in a vapor state it must be both fluid and compressible. "Vapor" and "Gas" are interchangeable terms.

The phase of a substance is determined by its pressure and temperature. A chart can be drawn to determine the phase from the pressure and temperature. This chart is called a phase equilibrium diagram (See Fig 1.1A). When the pressure temperature coordinates fall on one of the three lines, the substance can exist as either of the phases on each side of the line. At the "triple point" the substance can exist as a solid, liquid or vapor. We will take a closer look at phase equilibrium in Chapter Five when we examine liquified gas tank car unloading.

Since reciprocating compressors are designed for handling only gases, the compressor engineer must pay careful attention to phase equilibrium. If a gas is pressurized and then allowed to condense inside a compressor, it can result in serious damage the to the machine.

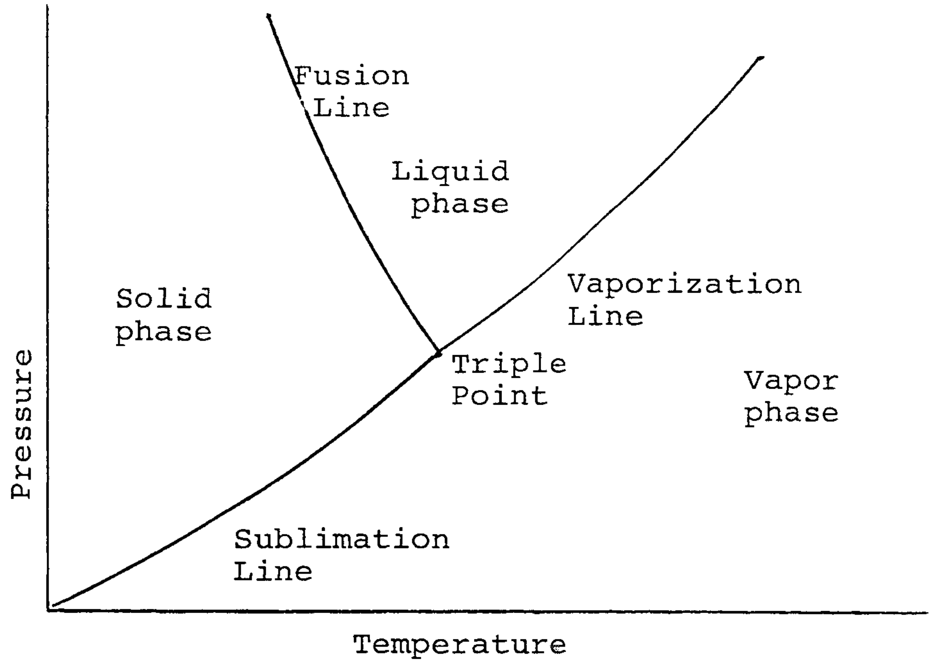


Figure 1.1A Pressure-Temperature diagram for a substance such as water.

1.2 UNITS OF MEASUREMENT

Gas compression problems are defined in terms of pressure, temperature and flow rate. There are several different measurements for each of these three quantities that are frequently confused.

1.2.1 PRESSURE

Pressure is the force per unit area exerted upon a surface. Pounds-per-square-inch (PSI) is the most common unit used to measure pressure in the United States. The metric system normally uses BARs to measure pressure. One BAR equals 14.5 PSIA. Another metric pressure unit is the Pascal. 100,000 Pascals equal one BAR. Don't confuse the BAR with the atmospheric pressure at sea level which is 14.7 PSIA or one atmosphere. It is mere coincidence that the value of the BAR and the atmosphere are so close. The metric system also uses the Kg/cm^2 as a measure of pressure. One Kg/cm^2 equals 14.22 psi or 0.9807 BAR.

Regardless of the unit of measurement, all pressures must be classified as either a gauge pressure or absolute pressure. Pressure gauges can only measure the difference between atmospheric pressure and tank pressure. The absolute pressure must be determined by adding the atmospheric pressure to the gauge pressure. At sea level the atmospheric pressure is 14.7 PSI. Atmospheric pressure decreases as altitude increases. See Fig. 1.2A for the atmospheric pressure at different altitudes. Gauge pressures are designated by placing a "G" after the pressure unit (i.e. PSIG) and absolute pressures are designated by placing an "A" after the pressure unit (i.e. PSIA).

ATMOSPHERIC PRESSURE AND BAROMETRIC READINGS AT DIFFERENT ALTITUDES

Altitude above sea level, ft.	Atmospheric pressure psi.	Barometer reading, in. Hg.	Altitude above sea level, ft.	Atmospheric pressure, psi.	Barometer reading, in. Hg.
0	14.69	29.92	7,500	11.12	22.65
500	14.42	29.38	8,000	10.91	22.22
1,000	14.16	28.86	8,500	10.70	21.80
1,500	13.91	28.33	9,000	10.50	21.38
2,000	13.66	27.82	9,500	10.30	20.98
2,500	13.41	27.31	10,000	10.10	20.58
3,000	13.16	26.81	10,500	9.90	20.18
3,500	12.92	26.32	11,000	9.71	19.75
4,000	12.68	25.84	11,500	9.52	19.40
4,500	12.45	25.36	12,000	9.34	19.03
5,000	12.22	24.89	12,500	9.15	18.65
5,500	11.99	24.43	13,000	8.97	18.29
6,000	11.77	23.98	13,500	8.80	17.93
6,500	11.55	23.53	14,000	8.62	17.57
7,000	11.33	23.09	14,500	8.45	17.22
			15,000	8.28	16.88

Figure 1.2A

1.2.2 TEMPERATURE

In the United States, temperature is generally defined in terms of degrees Fahrenheit (°F). The metric system uses degrees centigrade (°C) which is also called degrees Celsius. The two methods of temperature are related according to the equation -

$$T(°C) * 1.8 + 32 = T(°F)$$

The coolest temperature that is physically possible is called absolute zero. This quantity is equal to -460°F or -273°C. Absolute temperature is the number of degrees a substance is above absolute zero. Absolute temperatures will be used later on to predict how the temperature will change when a gas is expanded or compressed. Absolute temperature can be defined as degrees Rankine (°R) or degrees Kelvin (°K).

°R equals °F + 460 and °K equals °C + 273.

1.2.3 FLOW RATE

Compressor capacities can be defined as either volume flow rates or mass flow rates. Regardless of how the flow rate is specified, the compressor engineer must convert it to the actual volume flow rate at the compressor suction. Actual volume flow rates are measured as Actual Cubic Feet Per Minute (ACFM) or Actual Cubic Meters Per Hour (Actual M^3/Hr). Section 2.5 will review how to convert the different flow rates discussed in this section to actual volume flow rates.

The chemical industry typically specifies gas flow rates in terms of mass flow rates measured as Pounds-Per-Hour (lb/hr) or Kilograms-per-hour (Kg/hr). Another method of measuring mass called the Mole is also extensively used. A mole equals the same number of mass units as a substance's molecular weight. For example, nitrogen has a molecular weight of 28 so a lb-mole of nitrogen equals 28 lb, a Kilogram-mole equals 28 kilograms.

Mass flows are often defined in terms of the equivalent volume flow they would have at a "standard" temperature and pressure. In the United States, "standard" is 60°F and 14.7 PSIA. To differentiate this "standard" flow from actual flow, it is called Standard Cubic Feet Per Minute or SCFM. In metric, "normal" replaces "standard". "Normal" is defined as 0°C and 1.0132 bar, flow is measured as Normal Cubic Meters Per Hour or NM^3/Hr. For compressors with atmospheric inlet pressures, the difference between actual and standard is small. As suction pressure is elevated (or lowered), confusing "standard" with "actual" can lead to very serious mistakes. Be Careful!

The natural gas industry measures flow rates as Thousands of Standard Cubic Feet Per Day (MSCFD or just MCFD, standard conditions are assumed) or Millions of Cubic Feet Per Day (MMSCFD or MMCFD).

1.3 CONSTANT SPECIFIC HEAT

The quantity constant specific heat is the amount of energy it takes to raise the temperature of one unit of mass (such as a pound or kilogram) by one degree. This value is measured as BTU/(lbm°F); in metric it is Kilojoules/(Kg°C) . There are two different methods to determine the constant specific heat, each of which yields a different value. The differences between these two values are extremely useful in predicting gas behavior.

The first method measures the constant specific heat at a constant pressure while volume is allowed to change and is designated C_p. The second method measures the constant specific heat at a constant volume while the pressure is allowed to change and is designated C_v. Both of these values are "constant" properties of a gas that change very little over the pressure and temperature ranges Corken compressors are designed to handle.

Two quantities, K and R, are derived from C_p and C_v and are extremely useful for predicting how a particular gas will behave when it is compressed. The K value is the ratio of constant specific heats, C_p/C_v. "K" is a dimensionless value. R is called the gas constant and equals C_p-C_v. When R is multiplied by the molecular weight of the gas, it will equal a value close to 1544 lbf-ft/(lb-mole°R). This value is called the Universal Ideal Gas Constant and is designated by R_u, an ideal value that can be used to predict the behavior of gases that are within the range of operating pressures and temperature for Corken compressors.

1.4 FIRST AND SECOND LAWS OF THERMODYNAMICS

The scientific principles used to size and select compressors are founded on Thermodynamics, the science of energy.

The first law of thermodynamics states that energy is conserved, i.e. it cannot be created or destroyed. However, energy can be transformed from one form to another. Therefore, all the energy that goes into a compressor must come out, although it will be in a different form. For example, a compressor system converts electrical energy to mechanical energy which is finally stored in the gas as thermal energy and potential energy. Potential energy is the energy stored in the gas by pressurization similar to the energy stored in a spring when it is compressed.

The second law of thermodynamics states that no energy conversion process is 100% efficient . This means that all the energy going into the compressor will not come out stored in a recoverable form in the gas. Some of the energy will change to waste heat created by friction. The less efficient a compression system is, the more entropy it will create. Entropy is a measure of the unavailability of energy. An ideal system that is 100% efficient and creates no entropy is called an Isentropic System. An ideal isentropic compression system would be frictionless, adiabatic and reversible. An Adiabatic system is one that is perfectly insulated. An adiabatic compressor won't allow any of the thermal energy in the compressed gas to escape the system through cooling. A Reversible system is one that allows the energy stored in the gas to be reconverted to mechanical energy at 100% efficiency. A turbine or air motor are two types of devices that would do this if they were frictionless.

It is neither possible nor desirable for a compression system to be isentropic. However, by analyzing a compression system as if it were, many useful predictions can be made.

The predictions made for an isentropic system can be adjusted with "fudge factors" to very closely approximate reality.

1.5 GAS LAWS

Nearly the entire spectrum of industrial gases behave according to a set of mathmatical formulas called Gas Laws. A gas that conforms perfectly to these laws is called an Ideal Gas. Corken compressors are usually utilized within a range of temperatures and pressures where the difference between "ideal" and "real" is insignificant. For the purpose of this text we will assume real gases conform perfectly to ideal behavior.

The Ideal Gas Law relates the pressure, volume, mass, and temperature for a finite quantity of gas according to the mathmatical formula:

$PV = nR_uT$ where
- P = Absolute Pressure
- V = Absolute Volume
- n = Moles of Gas
- R_u = Universal Gas Constant
- T = Absolute Temperature

If molecular weight = MW and mass = m, then $R_u = R * MW$ (See Section 1.3) and $n = m/MW$ (See Section 1.2). By substituting these two relationships into $PV = nR_uT$ we can derive the ideal gas law in a new form: $PV = mRT$ where m = mass and R = R value for specific gas. If we know three of the four properties, 1) pressure, 2) volume, 3) mass and 4) temperature, we can determine the fourth property accordingly:

$P = mRT/V$ $\quad V = mRT/P$ $\quad m = PV/RT$ $\quad T = PV/mR$

or if we use n and R_u instead of m and R:

$P = nR_uT/V$ $\quad V = nR_uT/P$ $\quad n = PV/R_uT$ $\quad T = PV/nR_u$

By analyzing the ideal gas law we can conclude that the amount of space a finite quantity of gas fills up is determined by the number of gas molecules and not their overall weight. For example, a lb-mole of nitrogen contains the same number of molecules as a lb-mole of CO_2, although a lb-mole of nitrogen weights 28 lb and lb-mole of CO_2 weights 44 lb. As long as their temperatures and pressures are the same, a lb-mole of nitrogen fills the same space as a lb-mole of CO_2.

For any quantity of gas where one of the properties pressure, temperature or volume is held constant, a change in one of the other two properties results in a proportional or inversely proportional change in the third quantity according to the following relationships:

ISOBARIC SYSTEM
(If P is constant) $T_2/T_1 = V_2/V_1$ (Charle's Law)

ISOTHERMAL SYSTEM
(If T is constant) $P_2/P_1 = V_1/V_2$ (Boyle's Law)

ISOMETRIC SYSTEM
(If V is constant) $P_2/P_1 = T_2/T_1$ (Amonton's Law)

P = Absolute Pressure T = Absolute Temperature V = Volume

Unfortunately, none of these relations are very useful for predicting gas behavior in a real gas compression system. However, a set of relationships can be derived based on the ideal gas law and the principals of the first and second laws of thermodynamics which can successfully predict how gases behave during compression. These formulas are called the Polytropic Process Equations:

$$T_2/T_1 = (P_2/P_1)^{(n-1)/n} = (V_1/V_2)^{n-1}$$

A polytropic process is one where PV equals a constant value as P is increased. "n" is the polytropic compression coefficient. For an isentropic compression system, n = K. For a watercooled system n is less than K and the more the system is cooled, the lower the n value becomes. For a system that is cooled so much the gas temperature cannot increase as it is compressed , n = 1. If n = 1 is substituted into the polytropic relationships they will reduce to Boyle's Law. Such a system is referred to as "isothermal".

For air-cooled air and gas compressors, the values of n and K will be extremely close. Therefore, pressure, temperature and volume changes can be successfully predicted by assuming the system to be isentropic.

1.6 COMPRESSIBILITY

The vast majority of CORKEN compressor applications are for temperature and pressure ranges where the gas being compressed remains very close to ideal conditions. However, in some applications involving high pessure and/or high molecular weight gases, the gas behavior may vary substantially from ideal conditions.

The deviation of gas behavior from ideal conditions is called the compressibility of the gas and is designated by the letter Z. The Z factor is used as a 'fudge factor' to correct the ideal gas law as follows:

$$PV = ZnR_uT$$

Z is a dimensionless value. For ideal conditions Z = 1. For most CORKEN applications Z = 0.98 to 1.0. It would be extremely rare for a Z value less than 0.8 to be encountered for any CORKEN application.

The Z factor is a function of the temperature and pressure of the gas and varies according to changes in pressure and temperature. Fortunately, the Z factor may be predicted with reasonable accuracy by knowing the critical pressure and critical temperature of the gas in question. If you are interested in a more in depth discussion of critical pressure and temperature, consult a thermodynamics textbook.

Since Z is a function of pressure and temperature, it must be calculated for both the discharge and suction conditions in order to be applied to an actual compression problem. The polytropic process equation may now be rewritten as follows:

$$T_2/T_1 = (P_2/P_1)^{(n-1)/n} = (Z_2/Z_1)(V_1/V_2)^{(n-1)}$$

From this equation one can see that while pressure and temperature are not directly affected by the compressibility of the gas, the capacity of a compressor will be affected.

Applying compressibility to manual compressor calculations is quite cumbersome. However, compressibility can be very easily analysed with a computer. CORKEN INTERNATIONAL's compressor selection software has been written to consider compressibility when desired.

CHAPTER 2
FUNDAMENTALS OF COMPRESSOR OPERATION

This chapter will review the mechanics of compressor operation and examine the use of the gas laws to predict gas behavior in a compressor.

2.1 CRANKCASE

The function of the crankcase is to transform rotary motion from a driver into linear reciprocating motion for moving pistons up and down. (See Fig 2.1 for details) The crankcase contains a crankshaft supported on either end by roller bearings. With one exception, all Corken compressors have two throw crankshafts. A "throw" is a cylindrical lobe that is eccentric to the crankshaft centerline to which a connecting rod is attached. The throws are 180° out of phase to minimize the peak stresses on the crankshaft and to lower vibration.

The connecting rod and crankshaft are separated by a journal bearing. The journal bearing is a cylindrical insert that fits over the throw and the connecting rod is clamped over it. The inside face of the bearing is coated with a low friction material called babbitt. The lubricating oil forms a thin film to separate the bearing face and the throw.

The end of the connecting rod is attached to the wrist pin made of polished chrome-plated steel. The wrist pin and connecting rod are separated by a bronze wrist pin bushing. The surface between the bushing and the pin is also separated by a thin film of lubricant.

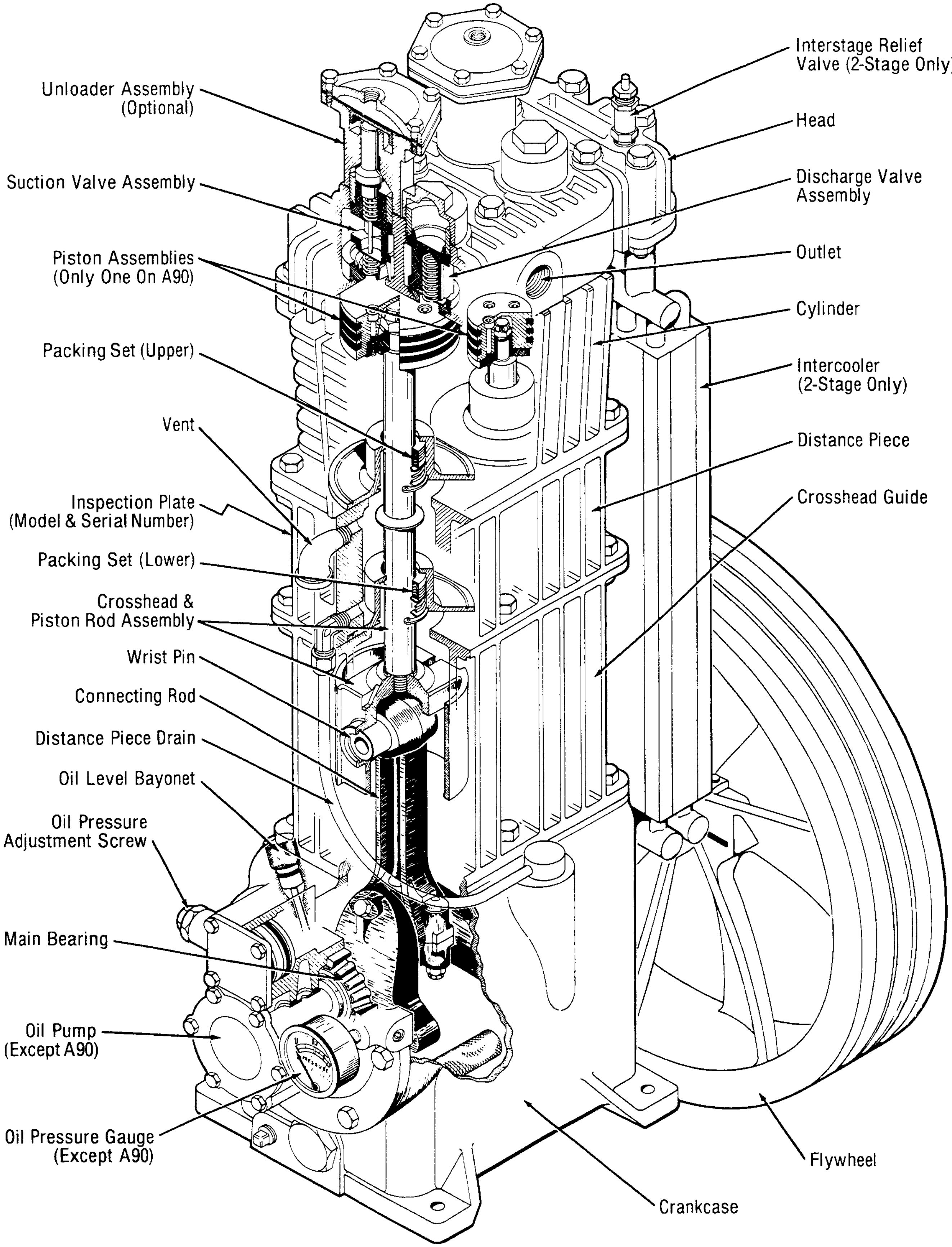
To insure adequate lubricating oil is present at the journal bearing and wrist pin bushing, Corken uses a pressure lubrication system. A small gear pump draws oil from the reservoir and pumps it through passages in the crankshaft and connecting rod to the bearing surfaces. Corken's oil pump is reversible, this allows all Corken units to rotate in either direction. (Note- the size 90/91 crankshaft is splash lubricated rather than pressure lubricated.)

2.2 PISTONS

In simple lubricated air compressors the piston is directly attached to the connecting rod. The compression cylinder is sealed off from the crankcase by only the piston rings. While piston rings seal tight enough to compress air or gas, they still allow a good deal of leakage to take place. When compression is only done on the top side of the piston, enough leakage will take place to pressurize the backside to suction pressure. This is not a problem if the suction pressure is atmospheric but it results in pressurizing the crankcase when elevated suction pressures are encountered on ordinary lubricated air compressors. Since the crankcase is not designed for pressure containment, this type of compressor is limited to only slightly elevated suction pressure (a few PSI).

The loose seal between the compression cylinder and crankcase also allows crankcase oil to seep into the cylinder and contaminate the gas. Furthermore, since the gas being compressed will leak into the crankcase, lubricated air compressors are limited to handling gases that will not contaminate the crankcase oil.

Figure 2.1A X-SECTION OF A TYPICAL CORKEN OIL-FREE VERTICAL COMPRESSOR



In summary, a simple lubricated air compressor has the following limitations:

1) Crankcase oil will contaminate the gas being compressed.

2) Gas will leak into the crankcase, therefore, flammable, corrosive and toxic gases cannot be compressed.

3) Suction pressure must be close to atmospheric pressure to prevent pressurization of the crankcase.

In order to overcome these limitations, a much tighter seal between the compression chamber and crankcase is required. Corken achieves this by mounting the piston on the end of a piston rod which is in turn mounted to a crosshead (See fig 2.1A). Now the piston only compresses the gas while the crosshead transforms rotary motion to reciprocating motion. The piston rod provides a relatively smaller diameter than the piston that can be tightly sealed with one or more sets of rod packing as required.

Since the compression chamber is sealed off, it can compress gas on both sides of the piston. When gas is compressed on one side of the piston the compressor is single-acting, if gas is compressed on both sides it is double-acting.

The side opposite the crankcase is the head end and the side nearest the crankcase is the crank-end.

The piston displacement of a cylinder refers to the volume swept by the piston on 180° of each revolution per unit time.

For single acting compressors:

Piston Displacement = Piston Area * RPM * Stroke

For double acting compressors the space occupied by the piston rod must be considered, so:

P.D. = ((2 * Piston Area) - Rod Area) * RPM * Stroke

Appendix B contains a table showing the displacements of Corken vertical compressors for the complete range of permissible V-belt and sheave combinations.

2.3 VALVES

Compressor valves regulate the flow of gas into and out of the compression cylinder by opening and closing in response to the pressure differential across the valve. Suction and discharge valves are virtually identical except that they are mounted in the compressor in opposite directions.

The four major valve parts are the seat, spring, disc and bumper (See Fig. 2.3A). When the valve is closed, the spring pushes the disc firmly against the seat. The openings in the seat correspond to the solid portions of the disc. The openings in the bumper correspond to those in the disc. When the pressure at the seat exceeds the pressure at the bumper, the differential pressure across the disc will depress the spring. As the disc moves away from the seat, gas flows into the valve through the seat, passes through the disc and exits at the bumper.

The valves are a part of the compressor head (See Fig. 2.3B). The head holds the valves in place, seals off the top end of the cylinder and acts as an inlet and outlet manifold to channel the gas flow through the valves. On the down stroke of the piston the cylinder pressure is lowered below suction pressure as the volume in the cylinder expands. This causes the suction valve to open so gas can be drawn into the compressor. On the up stroke the pressure is increased above suction pressure which pushes the suction valve disc securely against its seat to seal off the chamber. The pressure increases until the cylinder pressure exceeds the discharge pressure at which point the discharge valve opens for the remainder of the stroke.

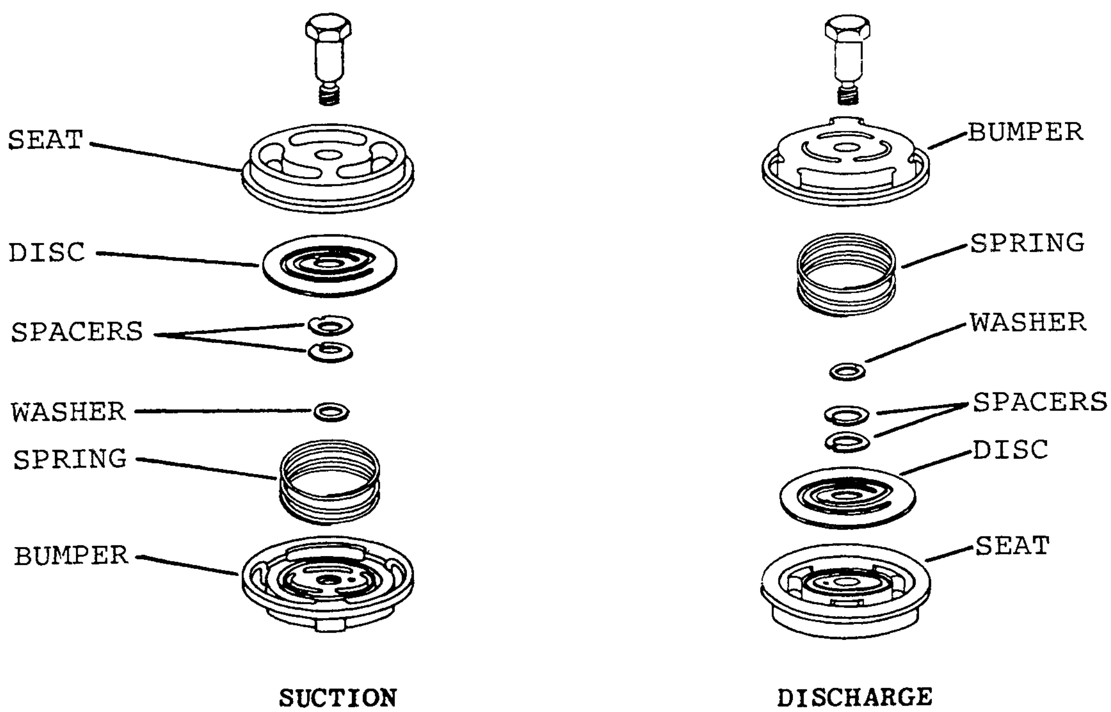


Figure 2.3A CORKEN SUCTION & DISCHARGE VALVES
FOR SIZE 390 & 490 COMPRESSORS

The open areas of the valve through which the gas flows is called the valve area. The pressure drop through the valve is inversely proportional to the square of the valve area and proportional to the masss flow through the valve. A well designed compressor valve minimizes the pressure drop to save on horsepower and maintain volumetric efficiency.

2.4 VOLUMETRIC EFFICIENCY

The amount of gas (ACFM) that a compressor is capable of moving is always less than the piston displacement. The ratio of the ACFM moved by a compressor to its piston displacement is called the volumetric efficiency.

Volumetric efficiency is determined by several factors which include the "k" valve of the gas, the compression ratio, the clearance volume and the pressure drop through the valves.

Clearance volume refers to the unswept volume in the compression chamber. This volume includes the clearance between the piston at top-dead-center and the head. It also includes the inlet and outlet passages in the valves which are within the compression chamber beneath the valve disc. At the end of each stroke a small quantity of gas at the discharge pressure is trapped in the clearance volume. As the suction stroke begins this gas will expand into the swept volume of the compression chamber until its pressure equals the suction pressure, at this point the suction valve opens and fresh gas fills the remainder of the cylinder.

Clearance volume is usually referred to as percent clearance which can be expressed as:

$$\text{Clearance (\%)} = (\text{Unswept Volume} / \text{Piston Displacement}) * 100$$

The volumetric efficiency of a perfect compression process with no friction or slippage past the piston rings can be described by the following equation:

$$VE\% = 100(1 - \text{Clearance})(P_2/P_1^{1/n} - 1)$$

Clearance should be expressed as a % of Cylinder Volume.

By examining the equation it is evident that increasing the clearance volume and compression ratio will decrease the volumetric efficiency while increasing the 'k' value will increase the volumetric efficiency.

The volumetric efficiency equations used in industry are based on this formula. Empirical "fudge factors" are added to account for the other factors that effect volumetric efficiency such as pressure drops through the valves, slippage past the piston ring and cylinder cooling.

The volumetric efficiencies for vertical Corken compressors can be determined by either the curves (See Figs. 2.4A and 2.4B) or by the following equations:

Single Stage Volumetric Efficiency

$$VE = 93 - CR - 8(CR^{1/n} - 1)$$

Two Stage Volumetric Efficiency

$$VE = 89 - CR - 7.75(CR^{1/2n} - 1)$$

CR is the Compression Ratio.

On some CORKEN compressors it is possible to adjust the size of the clearance volume. This offers two advantages: 1) Capacities can be adjusted without changing the V-belt and sheave combination or driver speed. 2) The interstage pressure of two-stage units may be adjusted to produce the lowest possible overall operating temperature.

With the exception of the D791, all the vertical compressors have fixed clearance volumes. The clearance of the first stage of the D791 may be adjusted to help control the machine's interstage pressure. All of the horizontal compressors (Models HG00) may be equipped with adjustable head ends as an option (see Fig. 4.1, item 6).

Figure 2.4A

VOLUMETRIC EFFICIENCY FOR CORKEN SINGLE-STAGE VERTICAL COMPRESSORS

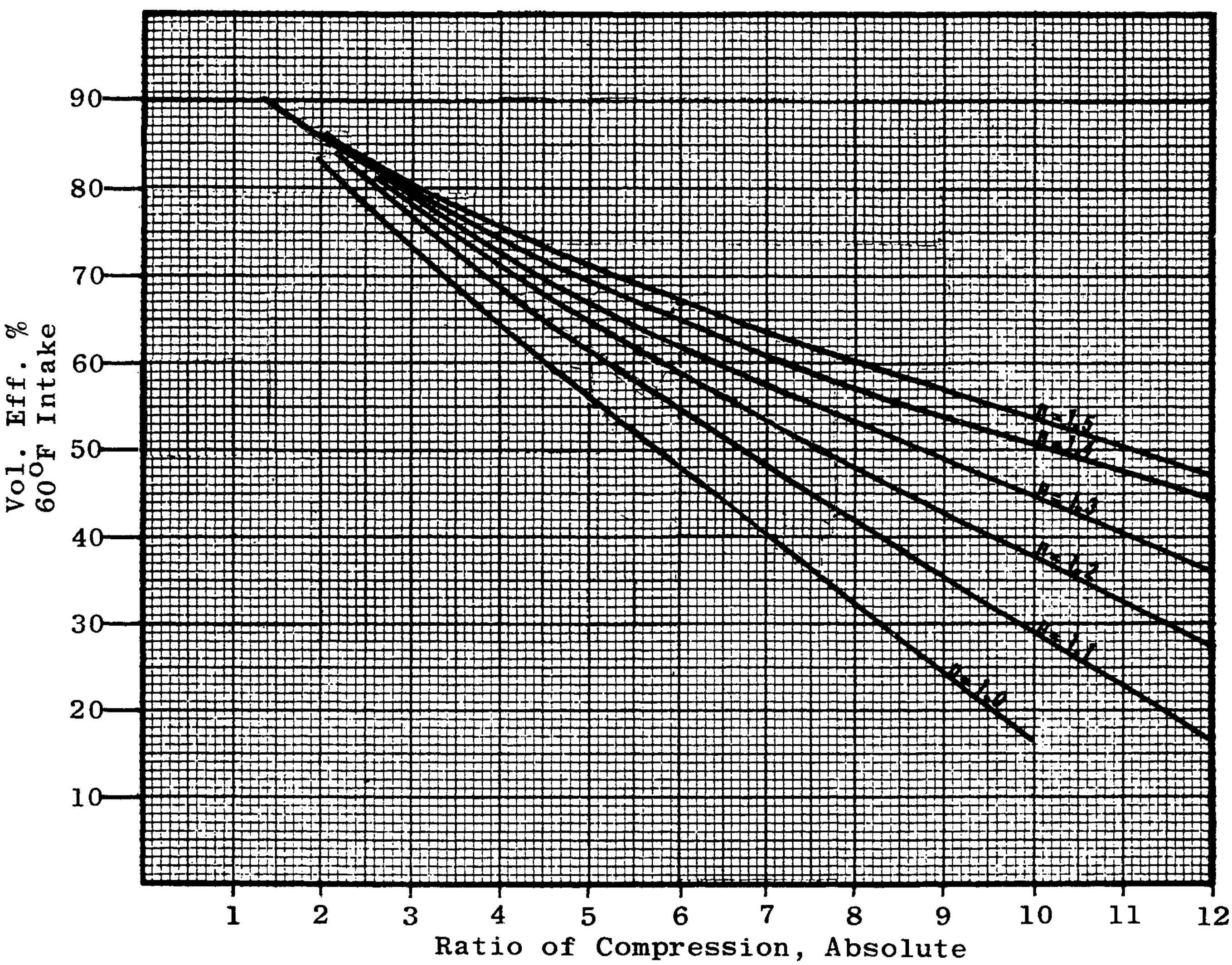
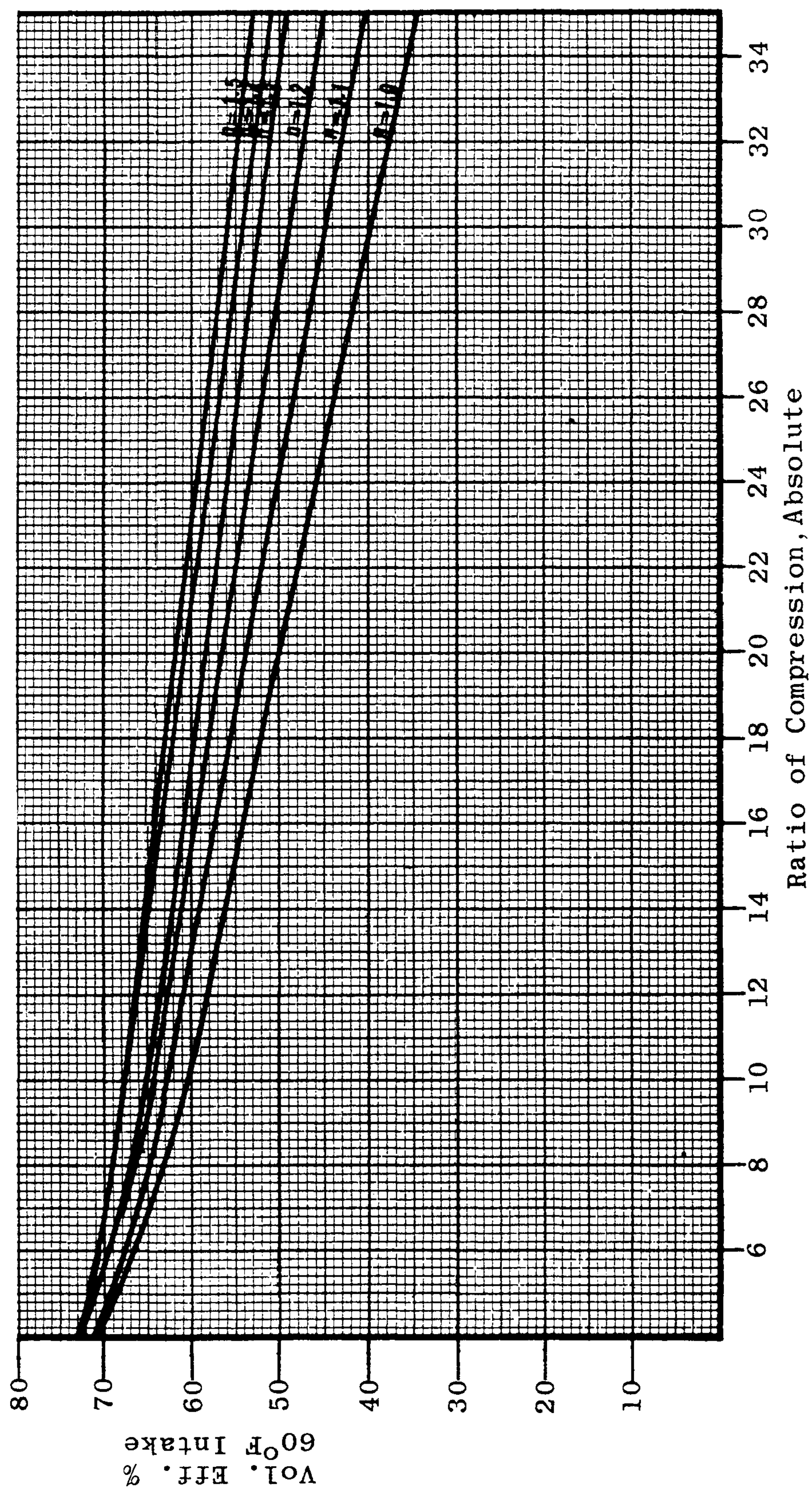


Fig. 2.4B

VOLUMETRIC EFFICIENCY FOR CORKEN TWO-STAGE VERTICAL COMPRESSORS



2.5 CALCULATING CAPACITY

The first step in choosing the right size compressor is to determine the flow rate in ACFM. If the capacity data available is not in terms of ACFM then it must be converted. The following equations are useful for converting some of the commonly used capacity measurements to ACFM.

a) SCFM to ACFM $\qquad ACFM = \dfrac{(SCFM)(14.7\ psia)(T_1\ {}^\circ R)}{(P_1\ psia)(520^\circ R)}$

b) MCFD to ACFM $\qquad ACFM = \dfrac{MCFD\ (14.7\ psia)\ (T_1{}^\circ R)}{1.44\ (P_1\ psia)\ (520^\circ R)}$

c) lb./hr to ACFM $\qquad ACFM = \dfrac{(lb/hr)\ (T_1{}^\circ R)\ (0.1787)}{(P_1\ psia)\ (MW)}$

Note: P_1 = Suction Pressure $\qquad T_1$ = Suction Temp.
MW = Moleculer Weight

Once the required capacity is known in terms of ACFM the required piston displacement can be determined by dividing the ACFM by the volumetric efficiency:

PD(required) = ACFM/VE

2.6 HORSEPOWER

The horsepower required to drive a compressor can be traced to four major sources:
1) Gas Horsepower
2) Valve Losses
3) Crankcase Frictional Losses
4) Slippage Losses

The primary horsepower requirement is for the gas horsepower. This is the actual energy that must be added to the gas to increase its pressure. For a single stage polytropic compression process the gas horsepower can be expressed in terms of ACFM, P_1 (suction pressure), P_2 (discharge pressure), and 'k'.

$$\text{Gas HP} = \frac{ACFM\ (n)\ (P_1\ psia)\ \left((P_2/P_1)^{(n-1)/n} - 1\right)}{229\ (n-1)}$$

The frictional losses in the valves are typically measured as a percent loss of the kinetic energy of the gas as it passes through the valve.

$$\text{Valve HP} = (\%\ \text{Loss})\ (\dot{m})\ (V^2)$$

$\dot{m}$ = Mass Flow $\qquad V$ = Gas Velocity

Decreasing the valve area or increasing the compressor speed results in a greater velocity and higher pressure drop through the valve. This results in higher energy losses through the valve.

The horsepower requirements due to mechanical friction are difficult to calculate with much precision, and are therefore determined by lab tests.

The horsepower for Corken's vertical units can be approximated by calculating the gas horsepower based on the piston displacement rather than the ACFM and then adding 30%. Due to the small horsepower involved, this approximation works very well for most cases. The frictional losses in the valve and crankcase must be less than 30% for the equation to indicate adequate horsepower, which is usually the case. However, when the compression ratio is low (CR < 1.5) and the suction pressure is elevated, the frictional losses may exceed 30% of gas horsepower. In these cases contact Corken to have the motor sized. The horsepower equations are as follows:

SINGLE STAGE VERTICAL COMPESSOR

$$HP = 0.00528\ (PD)\ (P_1)\ (n/n-1)\ (CR^{(n-1)/n} - 1)$$

TWO STAGE VERTICAL COMPRESSOR

$$HP = 0.00528\ (PD)\ (P_1)\ (2n/n-1)\ (CR^{(n-1)/2n} - 1)$$

PD = Piston Displacement (Ft^3/min)
P_1 = Suction Pressure (psia)
n = Polytropic Compression Coefficient
CR = Compression Ratio P_2/P_1

Horsepower calculations for Corken horizontal compressors require a higher degree of accuracy due to the higher horsepower involved. The complexity of these calculations is beyond the scope of this booklet. CORKEN also has a detailed computer analysis programs for its vertical compressors which produce more accurate results than the manual method presented above.

2.7 INTERSTAGE PRESSURES, TEMPERATURES AND INTERCOOLERS

As the compression ratio across a single stage of compression increases, the discharge temperature will increase and the volumetric efficiency will decrease. For the higher "n" value gases the compression ratio is limited by the discharge temperature the compressor can withstand. Corken compressors can operate continuously at 350°F or less. The following relation can be derived from the polytropic relationships for determining the allowable compression ratio per stage for aircooled gas compressors:

$$CR_{max} = \left(\frac{350 + 460°R}{T_1 + 460°R}\right)^{n/(n-1)}$$

T_1 = Suction Temperature (°R)

For a suction temperature of 100°F , air (n = 1.4) can be compressed by a ratio 3.6, helium (n = 1.66) by a ratio of 2.5, and propane (n = 1.13) by a ratio of 25. While low "n" value gases like propane can be compressed to ratios over 20 without getting too hot, the volumetric efficiency will fall to beneath 10%. Therefore, the compression ratio per stage for low "n" value gases is determined by volumetric efficiency rather than discharge temperature. Generally, compression ratios over 8 per stage are impractical due to low volumetric efficiencies.

Whenever a compression ratio is required that is higher than those a single stage compressor can handle, a multistage system must be used. An ideal compression system evenly divides the work done per stage, so for a two-stage system:

$$CR_1 = CR_2 = CR^{0.5}$$

In an actual compression system the compression ratio per stage is a function of the relative displacements and volumetric efficiencies of all the stages. For a two stage system the first stage compression ratio can be determined by the following relationship:

$$CR_1 = \frac{(ACFM_1)\ (T_{21})}{(ACFM_2)\ (T_{12})}$$

CR	$= CR_1 * CR_2$		
T_{12}	= Stage 1 Disharge Temp.	T_{21}	= Stage 2 Suction Temp.
$ACFM_1$	= Stage 1 ACFM	$ACFM_2$	= Stage 2 ACFM

From this relation it is evident that lowering the value of T_{21} lowers R_1. Lowering the suction temperature to the second stage will also increase the maximum allowable compression ratio across the second stage. Therefore it is desirable to cool the discharge from the first stage as much as possible (assuming condensation will not take place) before feeding it to the second stage. This is done by means of an aircooled or watercooled intercooler (See section 3.10 for more on intercoolers).

Solving the above equation requires an interactive process that is best left to a digital computer. However, substituting piston displacement for ACFM yields a reasonable approximation in most cases.

When relative compression ratios per stage can be varied, they should be adjusted to split the compression load as evenly as possible. The relative displacements of Corken two-stage vertical units are fixed. Typically the first stage compression ratio will vary from 2.8 to 3.5.

2.8 ROD LOADS AND ROD REVERSAL

If too great a load is placed on the wrist pins and crankshafts, the lubricant film protecting the wrist pins and crankshaft journals will break down. Break down of the lubricating film leads to rapid failure of the wrist pin bushings or crankshaft journals.

There are two factors that must be considered in order to maintain an adequate lubricating film. These factors are rod load and rod reversal. Rod load is the tensile or compressive force exerted on the rod at a given angle of rotation. The load varies continuously with the angle of rotation. Rod load has two components, gas load and inertial load. The gas load is the actual load created on the rod by the gas being compressed. (See Fig. 2.8A)

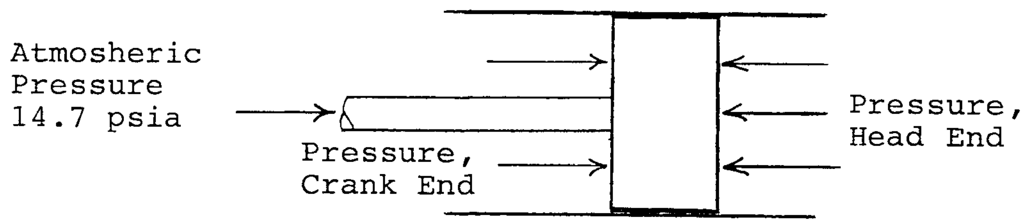


Figure 2.8A

For a double acting compressor, the gas load can be expressed as follows:

$$Fgl = (Ap)(Phe) - (Ap - Ar)(Pce) - (Ar)(14.7\ psia)$$

Ap = Piston Area (in^2)
Phe = Head End Pressure
Fgl = Gas Rod Load (lbf)

Ar = Rod Area (in^2)
Pce = Crank End Pressure (psia)

The inertial rod load is caused by the forces created from accelerating and decelerating the piston according to Newton's Second law, force equals mass times acceleration. When the speed of the compressor is increased, the acceleration and deceleration of the piston is increased which results in higher inertial rod loads. The gas rod load is 180° out of phase with inertial rod load. This is a fortunate coincidence because it causes the two types of load to cancel each other out to some extent. For example, at top dead center on a double acting compressor, the inertial load is tensile and the gas load is compressive. The gas load is nearly always the larger of the two forces, but considering the inertial load may result in a much lower overall rod load.

In order to maintain the lubricating film on the wrist pin and crankshaft, it is desirable for the load on the rod to shift from tension to compression during each stroke. This allows the oil being pumped to the bearing surfaces to spread out more evenly over the entire cylindrical surface. The relation between the degrees of tension and degrees of compression on the rod per rotation is referred to as rod reversal. Ideal rod reversal is 180° of tension and 180° of compression.

The maximum allowable rod load and minimum allowable rod reversal are quantities that must ultimately be determined through lab tests and field experience. For Corken's vertical units (except the WD391), rod reversal may be ignored entirely because the design parameters (pressure, horsepower, etc.) intrinsically limit the machine to safe values for rod reversal and rod load. The HG600 has a maximum rod load of 7500 lbs and should have at least 90° of rod reversal. Corken has computer programs to analyze these factors.

2.9 CRANKCASE LUBRICATING OIL

Proper selection of the crankcase lubricating oil is essential to maintaining an adequate lubricant film at the wrist pin and crankshaft journals. The key to maintaining these lubricant films is selecting an oil with the correct viscosity for the operating temperature of the oil. Too low a viscosity results in a breakdown of the film and too high a viscosity results in unnecessary friction.

The operating temperature of the oil is related to the horsepower (which takes speed and rod load into account) and the ambient temperature. The horsepower used to overcome friction is converted to heat in the lubricating oil. By designing large enough bearings and a large enough reservoir, adequate cooling will occur to maintain the oil at a temperature where it will have adequate viscosity.

If Corken compressors are operated within their horsepower and rod load limitations with the proper lubricant, then maintaining the sufficient viscosity in the oil is no problem. See Fig. 2.9A for the proper selection of crankcase lubricants.

Ambient Temperature At Compressor	SAE Viscosity
Below 0°F (-18°C)	5W, 5W-30
0 to 32°F (-18 to 0°C)	10W, 5W-30, 10W-40
32 to 80°F (0 to 27°C)	20W, 5W-30, 10W-40
Above 80°F (27°C)	30W, 5W-30, 10W-40

Figure 2.9A OIL SELECTION CHART

Corken compressors are designed for non-detergent lubricating oils. Detergents are used to suspend the contaminants to settle out in the reservoir. Corken compressors are too small for filtration to be practical. Therefore, non-detergent lubricants are recommended although detergent lubricants have been successfully used in most applications.

For certain applications, detergents may react with the gas being compressed. This is especially true for ammonia. Ammonia reacts with the detergent and causes the oil to be contaminated, and crankcase failure frequently results. Air and hydrocarbon gases do not present any problems for detergent lubricants.

CHAPTER 3

SPECIFYING VERTICAL CORKEN COMPRESSORS

3.1 INTRODUCTION

Corken International can build over 12,000 different types of vertical compressors from its supply of standard components. The Corken compressor line is specially designed to provide the maximum amount of flexibility while minimizing the number of components required. This strategy allows Corken to provide its customers with a product specifically tailored to their application without the high cost commonly associated with custom engineered machines.

Three limiting parameters must always be considered when selecting a Corken compressor. These parameters are pressure, temperature and horsepower. Fig. 3.1A summarized the maximum limits of these factors for Corken's vertical compressor line. Fig 3.1A also summarizes the bores, strokes and speeds for the line.

The maximum pressure rating is the amount of pressure a unit can safely contain. It is very important not to confuse this with the maximum amount of pressure the unit can generate for a specific application. The actual discharge pressure can equal the maximum rating only if the discharge temperature and total horsepower are beneath their maximum ratings.

For continuous duty applications the discharge temperature must be held to 350°F or less. Corken defines any application with a duty cycle over 30 minutes as a continuous duty application. Intermittent duty cycles are less than 30 minutes, operate less than half of the time and allow for a cooling period equal to the last run period. Intermittent duty cycles allow discharge temperatures up to 400°F. Operation at too high a temperature will result in very short piston ring and packing life.

The size of the crankcase limits the size of motor that can be used on each compressor. The horsepower limit prevents the crankshaft, ball bearings, journal bearings and wrist pins from being overloaded and the crankcase oil from being overheated.

The pressure and horsepower limits inherently limit the rod loads within safe limits for all vertical compressor models except the WD391, D791, and D891.

Figure 3.1A CORKEN VERTICAL COMPRESSOR DATA

SINGLE-STAGE UNITS

MODEL	D91	D290 D291	D490 D491 T491 [3]	D490-3	D690 WD690 [2] D691	D690-4 WD690-4 [2]
Bore in. (cm)	3.0 (7.62)	3.0 (7.62)	4.0 (10.2)	3.0 (7.62)	4.5 (11.4)	4.0 (10.2)
Stroke in. (cm)	2.5 (6.35)	2.5 (6.35)	3.0 (7.62)	3.0 (7.62)	4.0 (10.2)	4.0 (10.2)
Piston Displ. CFM (m^3/hr)						
100 RPM	1.0 (1.7)	2.0 (3.4)	4.3 (7.3)	2.45 (4.2)	7.3(12.4)	5.8 (9.6)
350 RPM	3.5 (5.9)	7.0 (11.9)	15.1 (25.7)	8.6 (14.6)	25.6(43.4)	20.3 (34.5)
825 RPM	8.3 (14.0)	16.5 (28.0)	35.5 (60.3)	20.2 (34.3)	60.2 (102)	48.0 (81.6)
Min. Inlet Pressure psia (Bar-A)	3 (.21)	3 (.21)	3 (.21)	3 (.21)	3 (.21)	3 (.21)
Max. Discharge Pressure psia (Bar-A)	350 (24.1)	280 (19.3) 350 D291	280 (19.3) 350 D491/T491	515 (35.5)	280 (19.3) 350 D691	365 (25.2)
Max. Compression Ratio [5]						
Continous Duty	5	5	5	5	5	5
Intermittent Duty	7	7	7	7	7	7
Max. Driver Size HP (KW)	7.5 (5.6)	15 (11)	15 (11)	15 (11)	30 (22)	30 (22)
Max Discharge Temp. Deg. F (Deg. C)	350 (177)	350 (177)	350 (177)	350 (177)	350 (177)	350 (177)
Approx. Wt. [4] lb. (kg)	265 (120)	265 (120)	520 (236)	520 (236)	745 (338)	745 (338)

Notes:

1. The D290, D490, D490-3, D690, D690-4, WD690, and WD690-4 have a Cast Iron Cyl. & Head.
   The D91, D291, D491, T491, and D691 have a Ductile Iron Cyl. & Head. (Note the higher Max. Dishcarge Pressure rating.)
2. The WD690 and WD690-4 have water-cooled heads.
3. The Model T491 has a double distance piece for precise leakage control and is most useful when handling Chlorine, Vinyl Chloride, etc.
4. Weights are for the compressor with flywheel, baseplate, driver slide base, & belt guard (no motor).
5. The maximum CRs given here should serve as a guideline only since the normal limiting factor is discharge temperature. The maximum CR shown are based on gas with a specific heat ratio of 1.4 and an inlet temp. of 70°F (21°C). Extreme ratios of compression for short periods of time are possible if the load is reduced or eliminated intermittently so compressor temp. can drop. The max. allowable temp. measured at the compressor head is approximately 350°F. Any operating cycle causing temp. rises above this will require special attention: lower ambient temp. if possible, reduced inlet temp., improved interstage cooling, or multi-compressor cycling to hold the operating temp. within recommendations, otherwise premature compressor failure may occur.

Figure 3.1B

CORKEN VERTICAL COMPRESSOR DATA

TWO-STAGE MODELS

MODEL	D190	D390	WD391 [2]	D590	WD590 [3]
Bore in. (cm)					
First Stage	3.0 (7.62)	4.5 (11.4)	4.5 (11.4)	6.0 (15.2)	6.0 (15.2)
Second Stage	1.75 (4.45)	2.5 (6.45)	2.5 (6.45)	3.25 (8.3)	3.25 (8.3)
Stroke in. (cm)	2.5 (6.35)	3.0 (7.62)	3.0 (7.62)	4.0 (10.2)	4.0 (10.2)
Piston Displ. CFM (m^3/hr)					
100 RPM	1.0 (1.7)	2.76 (4.7)	2.76 (4.7)	6.55 (11.1)	6.55 (11.1)
350 RPM	3.5 (5.9)	9.7 (16.4)	9.7 (16.4)	22.9 (39.0)	22.9 (39.0)
825 RPM	8.25 (14)	22.8 (38.7)	22.8 (38.7)	54.0 (91.8)	54.0 (91.8)
Min. Inlet Pressure psia (Bar-A)	3 (.21)	3 (.21)	3 (.21)	3 (.21)	3 (.21)
Max. Inlet Pressure psia (Bar-A)	100 (7.0)	100 (7.0)	100 (7.0)	100 (7.0)	100 (7.0)
Max. Discharge Pressure psia (Bar-A)	515 (35.5)	515 (35.5)	625 (43.1)	365 (25.2)	365 (25.2)
Max. Compression Ratios					
Continous Duty	10	10	12	10	12
Intermittent Duty	35	35	35	25	25
Max. Driver Size HP (KW)	15 (11)	15 (11)	15 (11)	30 (22)	30 (22)
Max Discharge Temp. Deg. F (Deg. C)	350 (177)	350 (177)	350 (177)	350 (177)	350 (177)
Approx. Water Flow GPM (m^3/hr)	-	-	1 (.23)	-	1 (.23)
Approx. Wt. [4] lb. (kg)	325 (147)	520 (236)	520 (236)	780 (354)	780 (354)

NOTES:

1. The D190, D390, D590, and WD590 have Cast Iron cylinders and heads. The WD391 has a Ductile Iron cylinder and head.
2. The WD391 has a water-cooled cylinder and head.
3. The WD590 has a water-cooled head.
4. Weights are for the compressor with flywheel, baseplate, driver slide base, & belt guard (no motor).
5. Refer to the note #5 under the single-stage compressor data.

FIGURE 3.1C CORKEN DOUBLE-ACTING VERTICAL COMPRESSOR DATA

Model	891 SINGLE-STAGE	791 TWO-STAGE	
		Stage 1	Stage 2
Bore in. (mm.)	4.5 (114)	6.0 (152)	3.25 (83)
Stroke in. (mm.)	4.0 (101)	4.0 (101)	4.0 (101)
Head End Clearance % (min.)	9.5	10.4	8.5
Head End Clearance % (max.)	9.5	27.8	8.5
Crank End Clearance %	11.4	10.9	10.4
Piston Area in.^{2} (cm^2)			
Head End	15.9 (103)	28.3 (182)	8.3 (53.5)
Crank End	14.7 (95)	27.0 (174)	7.1 (45.6)
Displacement CFM (M^3/hr)			
Head End at 900 RPM	66.2 (112.5)	59.0 (99.9)	17.3 (29.3)
Crank End at 900 RPM	61.2 (104.0)	56.3 (95.4)	14.8 (25.1)
Total at 900 RPM	127.4 (216.5)	115.3 (195.3)	32.1 (54.4)
Maximum Cylinder Working Pressure PSIG (BAR)	400 (27.6)	350 (24)	600 (41)

Inlet/Outlet Port Size	2" Weld
Maximum Discharge Temperature	350°F (177°C)
Approximate Weight with Flywheel	800 lb (365 Kg)
Maximum Rated Power	45 HP (34 KW)
Maximum Allowable Rod Load	7000 lb. (3175 Kg)
Cylinder Material	Ductile Iron ASTM A536

3.2 CHOOSING THE CORRECT NUMBER OF STAGES AND TYPE OF COOLING.

The first step in determining if multiple compression stages are needed is to calculate the discharge temperature for the application if it were done in a single, air cooled compression stage. This can be done simply by using the gas laws to derive the following equation:

$$T_2 = T_1 \, (P_2 \,/\, P_1)^{(n-1)/n}$$

T_1 = Suction Temperature °R
T_2 = Discharge Temperature °R
P_1 = Suction Pressure psia
P_2 = Discharge Pressure psia
n = Ratio of Specific Heats

If T_2 is greater than 350°F and the compression ratio is greater than 5 then a two-stage unit should be considered. Corken's vertical two-stage units require a compression ratio of 5 or greater in order to adequately balance the load on each stage. Because the ratio of the piston displacements is fixed at about 3.5, the total compression ratio must exceed 3.5 in order for any substantial compression to take place on the second stage. This results in a substantial rod load on the first stage and none on the second. The uneven rod loads result in torsional characteristics that cause unacceptable vibration in some cases. In some instances a larger flywheel will solve vibration problems at low compression ratios. For compression ratios above 5, the rod loads tend to even out and the vibration is greatly reduced.

Low "n" valve gases can be compressed to compression ratios above 10 without exceeding the 350°F discharge temperature limit but the volumetric efficiency can fall below 20%. In these situations a multi-stage compression system should be used to increase the volumetric efficiency back to a reasonable level.

Watercooling is another method to be considered when discharge temperatures become a problem. It is especially helpful when a compression ratio less than 5 is desired but a single stage aircooled compressor generates a discharge temperature over 350°F. Watercooling the A690 single stage compressor allows its continuous duty discharge pressure to increase from 60 PSIG to 100 PSIG when it is used to compress air entering at atmospheric pressure.

Watercooling two stage compressors allows the maximum allowable compression ratio to be increased. The A390 two stage air compressor can continously compress atmospheric pressure air to 135 PSIG when air cooled but it can compress it to 200 PSIG when water cooled.

Figure 3.2A

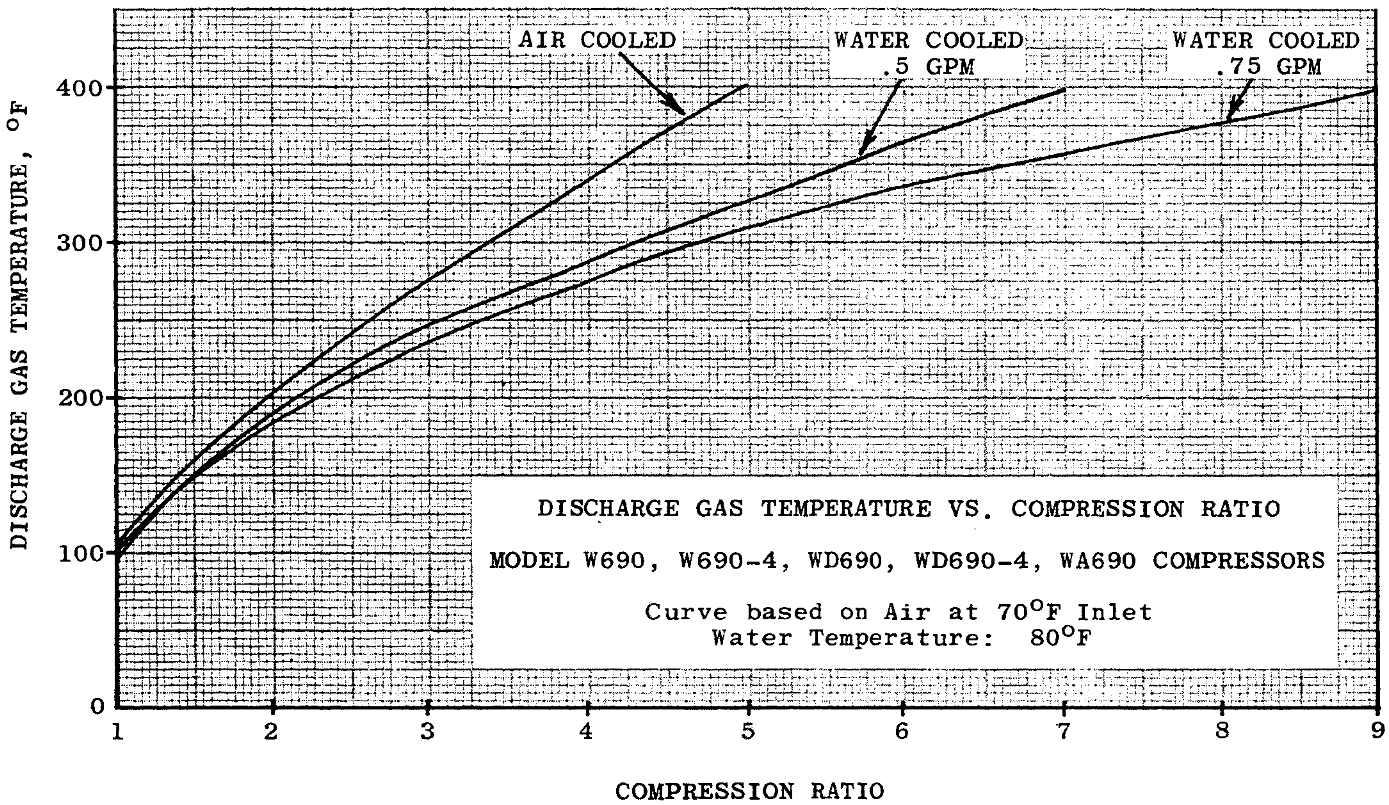
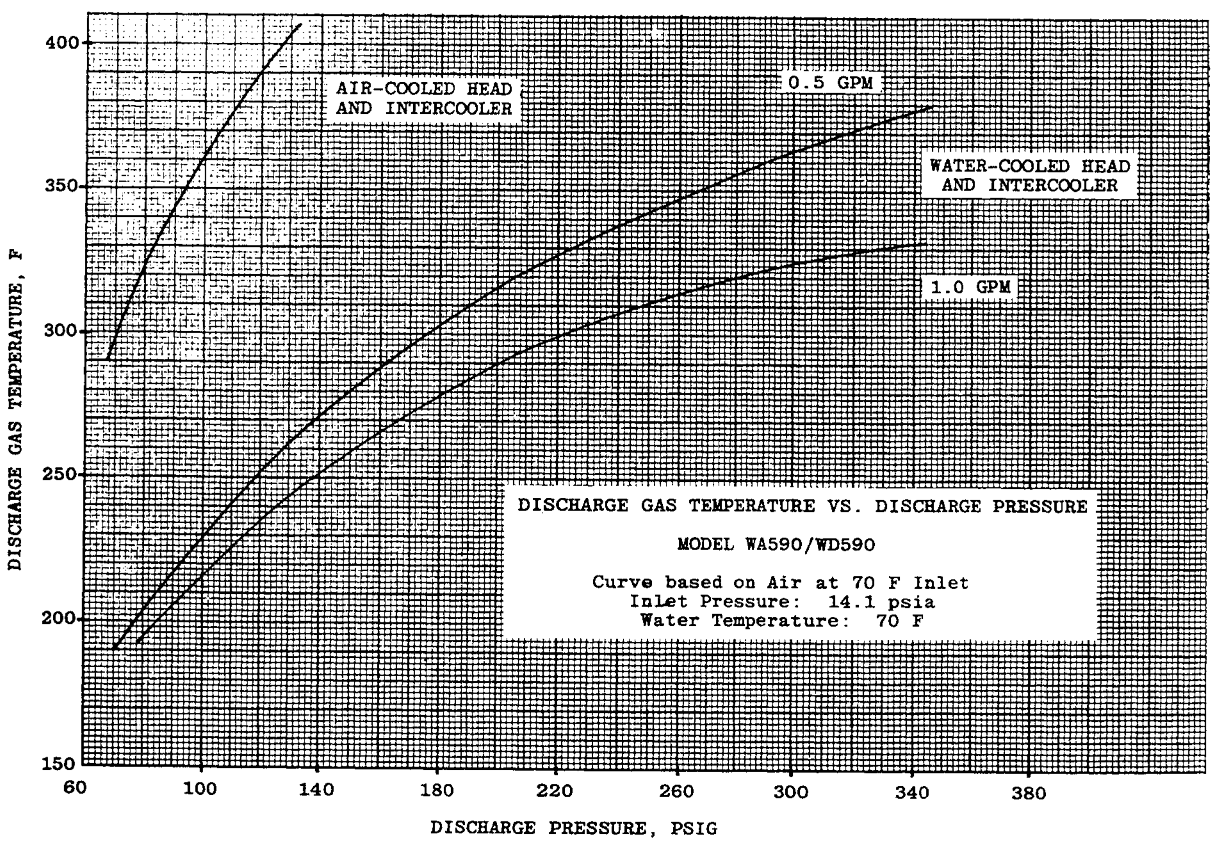


Figure 3.2B



3.3 CHOOSING THE RIGHT SIZE COMPRESSOR

After the required piston displacement has been calculated and the number of stages and the type of cooling have been determined, the compressor size required can be selected from the table in Appendix B. Note that the 91, 290, 291, 490, 491. 490-3, 690, 691, 690-4 size compressors are single stage and the 190, 390, 391 and 590 size compressors are two stage. Appendix B also gives the belts and motor sheaves required to run at the correct speed for the required piston displacement. After choosing a model number, double check to make sure the application will not exceed the pressure and horsepower rating of the machine.

Air compressors may be quickly and easily chosen from Fig. 3.3A rather than going through the calculations.

Figure 3.3A SELECTION CHART FOR CORKEN SINGLE-STAGE COMPRESSORS

		40 PSIG		(3.8 BAR)		60 PSIG		(5.2 BAR)		80 PSIG		(6.5 BAR)		100 PSIG		(7.9 BAR)	
	RPM	SCFM	NM³/HR	BHP	KW	SCFM	NM³/HR	BHP	KW	SCFM	NM³/HR	BHP	KW	SCFM	NM³/HR	BHP	KW
	400	3.0	5.1	.5	.4	2.8	4.8	.7	.5	2.5	4.2	.8	.6	2.3	3.9	.9	.7
	540	4.1	7.0	.7	.5	3.7	6.3	.9	.7	3.4	5.8	1.1	.8	3.1	5.3	1.2	.9
A91	620	4.7	8.0	.8	.6	4.3	7.3	1.1	.8	3.9	6.6	1.2	.9	3.5	5.9	1.4	1.0
	720	5.4	9.2	.9	.7	5.0	8.5	1.2	.9	4.5	7.6	1.4	1.0	4.1	7.0	1.6	1.2
	850	6.4	10.9	1.1	.8	5.9	10.0	1.4	1.0	5.4	9.2	1.7	1.3	4.8	8.2	1.9	1.4
	345	5.2	8.8	.9	.7	4.8	8.2	1.2	.9	4.3	7.3	1.4	1.0	3.9	6.6	1.6	1.2
	390	5.9	10.0	1.0	.7	5.4	9.2	1.3	1.0	4.9	8.3	1.5	1.1	4.4	7.5	1.8	1.3
	435	6.5	11.0	1.1	.8	6.0	10.2	1.5	1.1	5.5	9.3	1.7	1.3	5.0	8.5	1.9	1.4
A290	515	7.7	13.1	1.3	1.0	7.1	12.1	1.7	1.3	6.5	11.0	2.0	1.5	5.9	10.0	2.3	1.7
	585	8.8	15.0	1.5	1.1	8.1	13.8	1.9	1.4	7.4	12.6	2.3	1.7	6.7	11.4	2.6	1.9
	650	9.8	16.7	1.7	1.3	9.0	15.3	2.2	1.6	8.2	13.9	2.6	1.9	7.4	12.6	2.9	2.2
	720	10.8	18.4	1.8	1.3	9.9	16.8	2.4	1.8	9.1	15.5	2.8	2.1	8.2	13.9	3.2	2.4
	785	11.8	20.1	2.0	1.5	10.8	18.4	2.6	1.9	9.9	16.8	3.1	2.3	8.9	15.1	3.5	2.6
	895	13.4	22.8	2.3	1.7	12.4	21.1	2.9	2.2	11.3	19.2	3.5	2.6	10.2	17.3	4.0	3.0
	415	13.4	22.8	2.3	1.7	12.3	20.9	2.9	2.2	11.2	19.0	3.5	2.6	10.1	17.2	3.9	2.9
	470	15.2	25.8	2.6	1.9	13.9	23.6	3.3	2.5	12.7	21.6	3.9	2.9	11.5	19.5	4.5	3.4
A490	515	16.6	28.2	2.8	2.1	15.2	25.8	3.6	2.7	13.9	23.6	4.3	3.2	12.6	21.4	4.9	3.7
	605	19.5	33.1	3.3	2.5	17.9	30.4	4.3	3.2	16.4	27.9	5.1	3.8	14.8	25.1	5.7	4.3
	695	22.4	38.1	3.8	2.8	20.6	35.0	4.9	3.7	18.8	31.9	5.8	4.3	17.0	28.9	6.6	4.9
	740	23.9	40.6	4.0	3.0	21.9	37.2	5.2	3.9	20.0	34.0	6.2	4.6	18.1	30.8	7.0	5.2
	785	25.4	43.2	4.3	3.2	23.3	39.6	5.5	4.1	21.3	36.2	6.6	4.9	19.3	32.8	7.4	5.5
	830	26.8	45.5	4.5	3.4	24.6	41.8	5.8	4.3	22.5	38.2	6.9	5.1	20.3	34.5	7.8	5.8
	895	28.9	49.1	4.8	3.6	26.6	45.2	6.3	4.7	24.3	41.3	7.5	5.6	21.9	37.2	8.5	6.3
	420	23.0	39.1	4.3	3.2	21.2	36.0	5.5	4.1	19.3	32.8	6.5	4.8	17.5	29.7	7.4	5.5
	495	27.1	46.0	5.0	3.7	24.9	42.3	6.5	4.8	22.7	38.6	7.7	5.7	20.6	35.0	8.7	6.5
WA690	550	30.2	51.3	5.6	4.2	27.7	47.1	7.2	5.4	25.3	43.0	8.6	6.4	22.9	38.9	9.7	7.2
	605	33.2	56.4	6.1	4.5	30.5	51.8	7.9	5.9	27.8	47.2	9.4	7.0	25.2	42.8	10.6	7.9
	680	37.2	63.2	6.8	5.1	34.2	58.1	8.9	6.6	31.2	53.0	10.5	7.8	28.3	48.1	11.9	8.9
	735	40.3	68.5	7.4	5.5	37.1	63.0	9.6	7.2	33.8	57.4	11.4	8.5	30.6	52.0	12.9	9.6
	790	43.3	73.6	8.0	6.0	39.8	67.6	10.3	7.7	36.4	61.9	12.3	9.2	32.9	55.9	13.9	10.4
	825	45.2	76.8	8.3	6.2	41.5	70.5	10.8	8.1	37.9	64.4	12.8	9.5	34.3	58.3	14.5	10.8
	860	47.1	80.0	8.7	6.5	43.3	73.6	11.2	8.4	39.6	67.3	13.3	9.9	35.8	60.8	15.1	11.3

(SCFM is at 70°F and 14.7 psia, NM³/HR is at 22°C and 1.01 Bar.)

FIGURE 3.3B SELECTION CHART FOR CORKEN TWO-STAGE COMPRESSORS

		100 PSIG (7.9 BAR)				120 PSIG (9.3 BAR)				200 PSIG (14.8 BAR)				500 PSIG (35.5 BAR)			
	RPM	SCFM	NM³/HR	BHP	KW	SCFM	NM³/HR	BHP	KW	SCFM	NM³/HR	BHP	KW	SCFM	NM³/HR	BHP	KW
	370	2.6	4.4	.7	.5	2.5	4.2	.9	.7	2.4	4.1	1.0	.7	1.9	3.2	1.4	1.0
	435	3.0	5.1	.9	.7	3.0	5.1	1.0	.7	2.8	4.8	1.2	.9	2.2	3.7	1.6	1.2
	470	3.2	5.4	.9	.7	3.2	5.4	1.1	.8	3.0	5.1	1.2	.9	2.4	4.1	1.7	1.3
A190	540	3.7	6.3	1.1	.8	3.7	6.3	1.3	1.0	3.5	5.9	1.4	1.0	2.8	4.8	2.0	1.5
	605	4.2	7.1	1.2	.9	4.1	7.0	1.4	1.0	3.9	6.6	1.6	1.2	3.1	5.3	2.3	1.7
	675	4.7	8.0	1.3	1.0	4.6	7.8	1.6	1.2	4.4	7.5	1.8	1.3	3.5	5.9	2.5	1.9
	740	5.1	8.7	1.4	1.0	5.0	8.5	1.7	1.3	4.7	8.0	1.9	1.4	3.8	6.5	2.7	2.0
	830	5.7	9.7	1.6	1.2	5.6	9.5	1.9	1.4	5.3	9.0	2.2	1.6	4.2	7.1	3.0	2.2
	895	6.2	10.5	1.7	1.3	6.1	10.4	2.1	1.6	5.8	9.9	2.3	1.7	4.6	7.8	3.3	2.5
	370	7.0	11.9	1.9	1.4	6.9	11.7	2.1	1.6	6.5	11.0	2.6	1.9	5.2	8.8	3.7	2.8
	450	8.6	14.6	2.4	1.8	8.4	14.3	2.6	1.9	7.9	13.4	3.2	2.4	6.3	10.7	4.5	3.4
	515	9.8	16.7	2.7	2.0	9.7	16.5	2.9	2.2	9.1	15.5	3.7	2.8	7.2	12.2	5.2	3.9
A390	560	10.6	18.0	2.9	2.2	10.5	17.8	3.2	2.4	9.9	16.8	4.0	3.0	7.9	13.4	5.6	4.2
WA391	630	11.9	20.2	3.3	2.5	11.8	20.1	3.6	2.7	11.1	18.9	4.4	3.3	8.8	15.0	6.3	4.7
	695	13.2	22.4	3.6	2.7	13.0	22.1	3.9	2.9	12.2	20.7	4.9	3.7	9.7	16.5	7.0	5.2
	765	14.5	24.6	4.0	3.0	14.3	24.3	4.3	3.2	13.4	22.8	5.4	4.0	10.7	18.2	7.6	5.7
	830	15.7	26.7	4.3	3.2	15.5	26.3	4.7	3.5	14.6	24.8	5.8	4.3	11.6	19.7	8.3	6.2
	895	17.0	28.9	4.6	3.4	16.7	28.4	5.0	3.7	15.7	26.7	6.3	4.7	12.5	21.2	8.9	6.6

		100 PSIG (7.9 BAR)				120 PSIG (9.3 BAR)				200 PSIG (14.8 BAR)				350 PSIG (25.2 BAR)			
	RPM	SCFM	NM³/HR	BHP	KW	SCFM	NM³/HR	BHP	KW	SCFM	NM³/HR	BHP	KW	SCFM	NM³/HR	BHP	KW
	405	18.3	31.1	5.5	4.1	18.0	30.6	6.0	4.5	17.0	28.9	7.4	5.5	15.1	25.7	9.3	6.9
	495	22.4	38.1	6.7	5.0	22.0	37.4	7.3	5.4	20.7	35.2	9.1	6.8	18.5	31.4	11.4	8.5
	550	24.8	42.1	7.4	5.5	24.5	41.6	8.1	6.0	23.0	39.1	10.1	7.5	20.5	34.8	12.6	9.4
A590	605	27.3	46.4	8.1	6.0	26.9	45.7	8.9	6.6	25.3	43.0	11.1	8.3	22.6	38.4	13.9	10.4
WA590	640	28.9	49.1	8.6	6.4	28.5	48.4	9.4	7.0	26.8	45.5	11.7	8.7	23.9	40.6	14.7	11.0
	680	30.7	52.2	9.1	6.8	30.3	51.5	10.0	7.5	28.5	48.4	12.5	9.3	25.4	43.2	15.6	11.6
	735	33.2	56.4	9.9	7.4	32.7	55.6	10.8	8.1	30.8	52.3	13.5	10.1	27.4	46.6	16.9	12.6
	790	35.7	60.7	10.6	7.9	35.2	59.8	11.6	8.7	33.1	56.2	14.5	10.8	29.5	50.1	18.1	13.5
	860	38.8	65.9	11.6	8.7	38.3	65.1	12.6	9.4	36.0	61.2	15.8	11.8	32.1	54.5	19.7	14.7

(SCFM is at 70°F and 14.7 psia, NM³/HR is at 22°C and 1.01 Bar.)

The above charts show conditions for both continuous and intermittent duty services. When continuous duty is required, check Fig. 3.3C to make sure continuous duty limits are not exceeded.

Figure 3.3C Duty Cycle Specifications

	Single-Stage					Two-Stage				
	A90	A290	A490	A690	WA690[2]	A190	A390	WA391[2]	A590	WA590[2]
Maximum Outlet Pressure, psig (bar)										
Continuous Duty[1]	60(5.2)	60(5.2)	60(5.2)	60(5.2)	100(7.9)	135(10.3)	135(10.3)	225(16.6)	135(10.3)	225(16.6)
Intermittent Duty[1]	100(7.9)	100(7.9)	100(7.9)	100(7.9)	120(9.2)	500(35.5)	500(35.5)	500(35.5)	350(25.1)	350(25.1)

[1]A definition of duty cycle is helpful: whether it is continuous or intermittent duty, the limiting factor is temperature. Extreme discharge pressures for short periods of time are possible if the load is reduced or eliminated intermittently so compressor temperatures may go back down to ambient.The maximum allowable gas temperature measured at the compressor head is approximately 350° F. Any operating cycle causing temperatures above this will require special attention: lower ambient temperature, lower interstage cooling temperature or provide multi-compressor cycling means to hold the operating range within recommendations – otherwise premature compressor failure may occur. Intermittent'may be defined, gene as 30 minutes on and 30 minutes off.

[2]Models WA391, WA590 and WA690 have water-cooled heads

3.4 DISTANCE PIECES AND ROD PACKING

Piston rod packing serves two simple functions: 1) Containing the gas in the compression chamber. 2) Containing the crankcase oil in the crankcase in order to prevent it from contaminating the gas being compressed.

The piston rings do not form a perfect seal with the cylinder wall. Enough "blow by" will occur to pressurize the backside of the piston to the suction pressure to the compressor. Rod packing is used to form a very tight pressure containing seal to hold the gas on the backside of the piston in the compression cylinder. Corken uses a chevron or "V-ring" type packing made of Teflon that is suitable for both lubricated and non-lubricated service. Depending on the type of compressor, 3 to 8 V-rings are used. The rings are held in place by a male and female retainer ring made of glass-filled Teflon A spring holds the packing in place between the two snap rings in the packing cartridge.

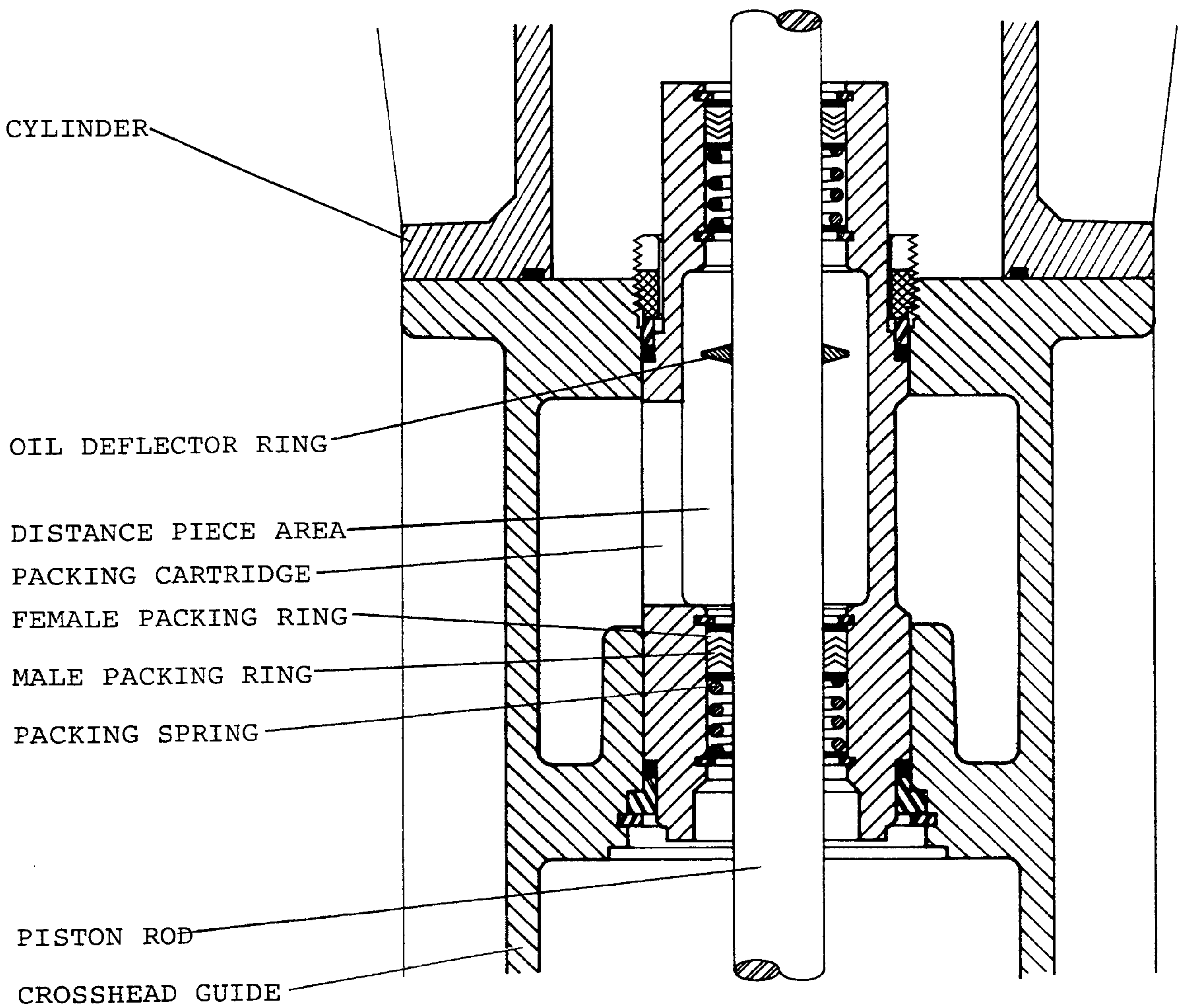
Air compressors use 3 V-rings while the gas compressors use 4 to 8 rings. Gas compressors require more packing because the elevated suction pressures they are designed to handle result in higher pressures on the back side of the piston.

Depending on the degree of isolation required between the crankcase and compression chamber, one, two or three sets of packing may be used per rod. When the gas being compressed is non-toxic, non-corrosive, and a minute amount of oil contamination is acceptable, one packing set will suffice. Corken LP gas compressors and sweet natural gas compressors belong to this category. When totally oil-free service is required, two sets of packing at either end of the distance piece are used. A distance piece adds additional distance between the crankcase and compression chamber. The distance piece is long enough that no "overtravel" occurs on the piston rod, i.e. no part of the piston rod contacting the bottom packing set can contact the top packing set. A deflector ring mounted on the piston rod between the packing sets prevents the minute amount of oil that passes the bottom packing set from working its way up the rod to the top of the packing set. Highly corrosive gases such as chlorine require three packing sets to hold leakage of chlorine into the crankcase to an absolute minimum.

For V-ring packing to form an effective seal, the pressure on the open end of the 'V' must be higher than the pressure on the pointed end. For gas applications with positive gauge pressures, the 'V' should point down. For suction pressure in the vacuum range, the 'V' points up. In order to prevent leakage from the cylinder if the suction pressure should increase from a negative to a positive gauge pressure, it is necessary to pressurize the distance piece. This assures the open end of the 'V' will see a higher pressure than the pointed end.

For compressing highly corrosive gases, purging of the distance piece with nitrogen provides an additional buffer to prevent contamination of the crankcase. Oxygen compressors are supplied with open distance pieces to eliminate the possibility of forming an explosive mixture of oxygen and crackcase oil in the distance pieces.

Figure 3.4A DISTANCE PIECE DETAILS



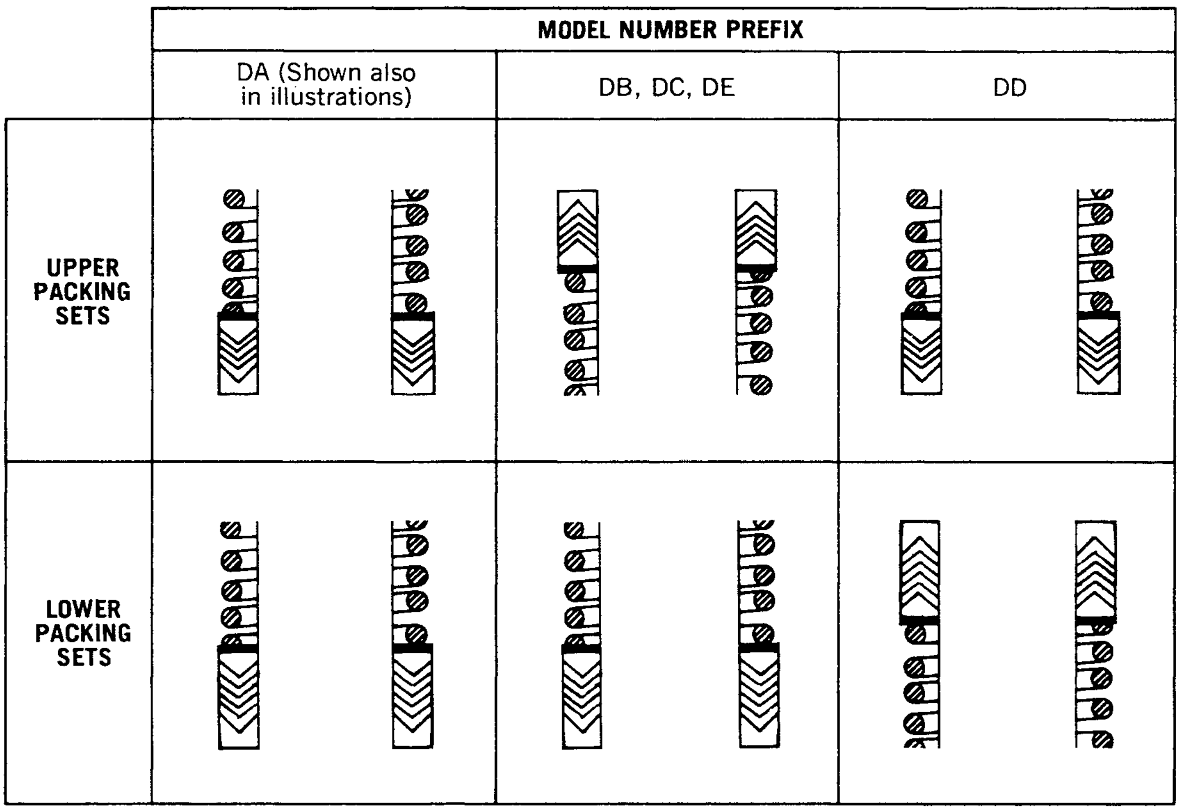
For a summary of the different types of distance piece tubing arrangements see Fig. 3.4B.

			Distance Piece Opening, 1/4 Inch NPT	
Model	Conditions	Service	Upper	Lower
D190A, D390A, WD391A, D590A WD590A	Inlet Pressure: Above 15 psia	1) General Gas Transfer	Plugged - allow pressure to reach its own level	Pipe to drain or vent with a shut-off valve. Distance piece must be drained weekly to prevent an accumulation of oil or condensate.
		2) Highly toxic gasses	Purge with an inert gas at a pressure below inlet but above 15 psia or use atmospheric vent to a safe area.	
D190B, D390B, WD391B, D590B, WD590B	Inlet Pressure: Limited vaccum to 25 psia or Distance Piece Pressure greater than Inlet Pressure.	General gas transfer or evacuation	Tubed to second stage inlet	
DC190, DC390		Obsolete	Tubed to inlet	
D190D, D390D, WD391D, D590D, WD590D	Inlet Pressure: 15 psia to 50 psia	Oxygen transfer	None - Distance piece milled open	None
D190E, D390E, WD391E, D590E, WD590E	Inlet Pressure: 3 psia to 25 psia	General gas transfer or evacuation	Tubed to outlet	Pipe to drain or vent with a shut-off valve. Distance piece must be drained weekly to prevent an accumulation of oil or condensate.
D90A, D91A, D290A, D291A, D490A, D490-3A, D491A, D690A, D691A, D690-4A, WD690A, WD690-4A	Inlet Pressure: Above 15 psia	1) General gas transfer	Plugged - allow pressure to reach its own level	
		2) Highly toxic gases	Purge with an inert gas at a pressure below inlet but above 15 psia or use atmospheric vent to a safe area.	
D90B, D91B, D290B, D291B D490B, D491B, D690B, D691B, WD690B	Inlet Pressure: 3 psia to 25 psia or Distance Piece Pressure greater than Inlet Pressure.	General gas transfer or evacuation	Tubed to outlet	
DC290, DC490		Obsolete	Tubed to inlet	
D90D, D91D, D290D, D291D, D490D, D491D, D690D, D691D, WD690D	Inlet Pressure: 15 psia to 50 psia	Oxygen transfer	None - Distance piece milled open	None

Figure 3.4B DISTANCE PIECE TUBING DETAILS

Over a long period of time a small amount of oil will collect in the distance piece. On Corken air compressors the distance piece is drained to the crankcase. Gas compressor distance pieces are sealed so it is necessary to periodically drain the distance piece to maintain oil-free compression. For continuous duty oil-free compression, draining the distance piece once a week is recommended.

Figure 3.4C PACKING INSTALLATION



PACKING INSTALLATION

3.5 PISTON RODS AND CROSSHEAD GUIDES

For Chevron piston rod packing to seal properly, it is essential that the piston rod be precisely positioned in the center of the packing. To insure this level of precision, Corken machines the crosshead after attaching it to the piston rod. This method insures that the crosshead and rod will be perfectly concentric, and that no metal-to-metal contact will occur between the piston and cylinder wall.

Corken piston rods are made of steel plated with a polished chrome surface. Since some chemicals, such as chlorine, attack chrome plated surfaces, Corken offers chrome-oxide plated rods as well. The black chrome-oxide coating offers a much higher level of corrosion resistance than the chrome plate. (Note - the chrome-oxide rods require special break in procedures.)

3.6 CASTINGS

A complete list of material specifications for Corken compressors is given is Appendix D.

Corken's standard material for the major structural castings is ASTM 48, class 30 grey iron. Corken also has several models available with ASTM A536 ductile iron cylinders. Ductile iron is a stronger material which allows higher pressures to be contained. It should be noted that to take advantage of the higher strengths offered by ductile iron that a stronger gasketing system is required as well. Corken uses O-ring gaskets in place of compressed asbestos gaskets in order to provide a stronger seal.

On single stage models, the pressure rating is 280 PSIA for grey iron and 350 PSIA for ductile iron models. Corken also has two higher pressure, single stage models made of grey iron, the 490-3 and 690-4. These compressors use a smaller piston bore that allows for a heavier casting. The 690-4 is rated for 365 PSIA (25.2 BAR-A) while the 490-3 is rated for 515 PSIA (35.5 BAR-A).

3.7 PISTON RINGS

Corken piston rings use a step-cut design made of glass-filled Teflon. This design and material have been proven to be the best reasonably priced design and material available for most non-lubricated piston rings. Carbon-filled Teflon rings are available in some sizes for oxygen or nitrogen applications.

For applications with high discharge temperatures (350°F) and operating speeds (825 RPM) a ring life of 2200 hours is typical in non-lubricated service. Lower discharge temperatures and speeds result in much longer ring life. 10,000 hour ring lifes are not unusual on cool, intermittent applications such as propane tank car unloading.

When a cylinder lubricator is used, the ring life will usually be quadrupled (See Sec. 3.11).

Extremely dry gases such as nitrogen vaporized from a cryogenic liquid storage tank cause extremely poor lubricity which results in shorter ring lifes. For dry nitrogen applications it is recommended to hold the discharge temperature below 250°F.

The piston ring only partially fills the space between the piston ring groove and the cylinder wall. In order for the piston ring to seal, it must be held against the cylinder wall be a ring expander. The expander is made of a flat strip of stainless steel bent in a circular shape that is fit into the piston ring groove before installing the piston ring.

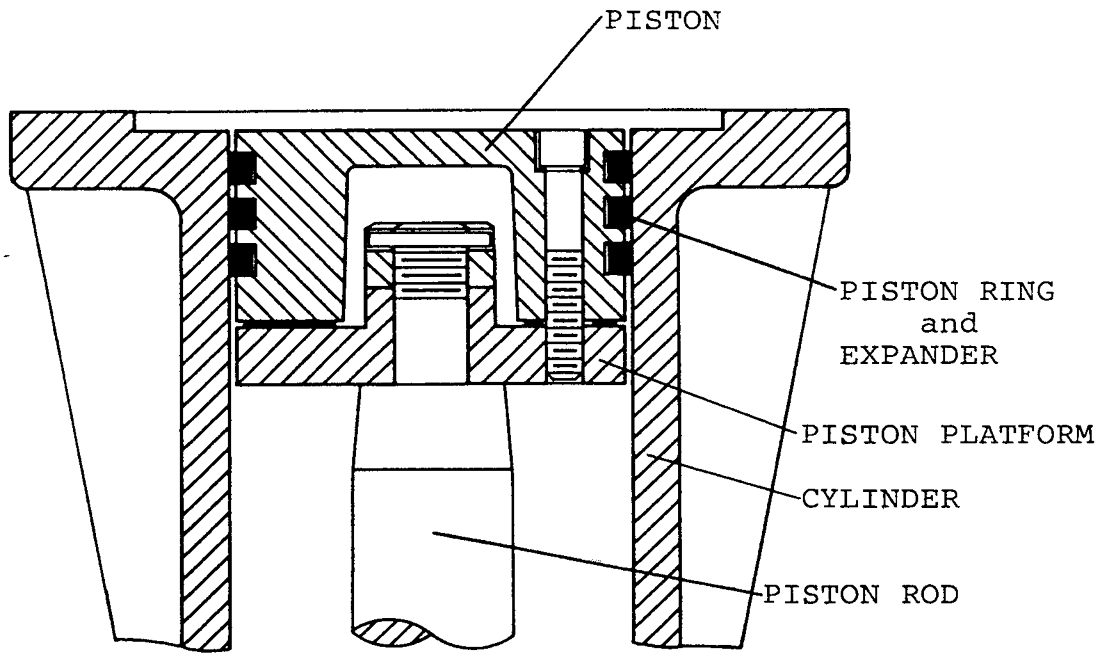
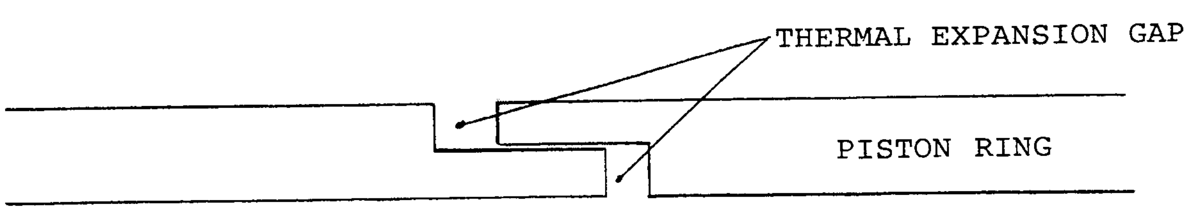


Figure 3.7A CROSS SECTION OF PISTON ON CORKEN VERTICAL COMPRESSOR

All of Corken's vertical compressors use 3 piston rings per piston, exept the D791 and D891 which use 4 rings per piston.

Figure 3.7B STEP-CUT DESIGN



3.8 GASKETS

Most metal-to-metal contact surfaces on the non-moving parts of Corken compressors are sealed with O-ring gaskets. O-rings are the preferred method of sealing because they are simple to install, readily available and provide the tightest seal. Buna-N is the standard O-ring material used while Teflon, Viton, and Neoprene are also available for more corrosive applications.

Corken grey iron compressors use flat compressed asbestos head gaskets for the head-to-cylinder seal. For light gases such as helium and hydrogen, Grafoil head gasket is used. Grafoil is a non-porous carbon material made by Union Carbide which is more leak tight than compressed asbestos. The ductile iron compressors use O-rings for head-to-cylinder sealing.

3.9 TRIM

Trim designates the materials of the metal gaskets used for the valve-to-head seal and for the tubing used for pressurizing the distance piece ("DB" models only), relieving pressure from the back side of the pistons (two-stage models) and connecting the unloaders.

The three types of trim are:

- "Aluminum Trim" - Aluminum valve gaskets with copper tubing.
- "Copper Trim" - Copper valve gaskets with copper tubing.
- "Iron-Lead Trim" - Lead valve gaskets with steel tubing.

3.10 INTERCOOLERS AND AFTERCOOLERS

Corken's standard intercoolers are air-cooled (except on the water-cooled W391 and W590 series). The standard intercoolers are made of brass tubes with aluminum fins. All-steel intercoolers are available for use with gases that corrode copper. The intercooler is covered by a shroud (not on A190 or D190) that channels air from the fan type flywheel across the surface of the cooling tubes. The Corken air-cooled intercooler typically cools the gas to within 30 to 50°F of the ambient temperature.

When lower interstage temperatures are desired, the air-cooler may be replaced by a water-cooler. Water-cooling typically cools the gas to within 20 to 30°F of the cooling water temperature. Custom sized water-cooled intercoolers may be used when extra cooling is required. The standard intercooler is a shell and tube design of all brass construction, an all stainless steel model is available as an option. For unusual applications requiring special cooling systems, Corken can supply the compressor without the intercooler, so specially engineered intercoolers can be used.

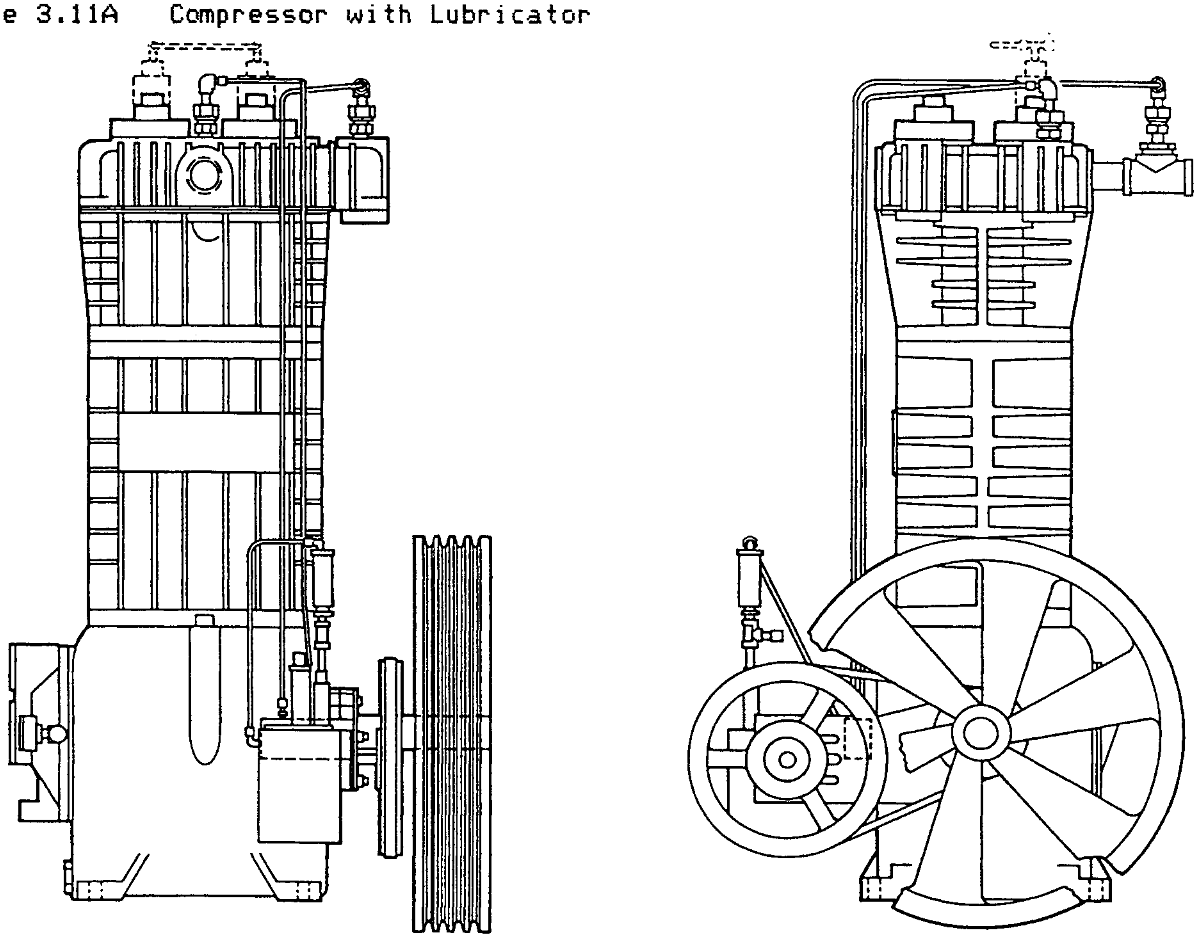
In some cases the gas discharged from the compressor will be too hot for use by the system it is supplying. In these situations an aftercooler must be used. All of Corken's vertical compressor packages are available with optional air or water-cooled aftercoolers. The air-cooled aftercooler is a free-convection, static cooler made of brass tubes with aluminum fins. Small shell and tube heat exchangers are available in brass and stainless steel when water-cooled aftercoolers are required. These can be directly attached to the baseplate at the factory with the interconnecting piping between the pump and compressor.

3.11 CYLINDER LUBRICATORS

Corken's "A" and "D" type compressors are capable of providing 100% oil-free compression while the standard style compressor comes very close. However, in many cases oil-free compression is neither required nor desirable. The best example of this is oil-field natural gas compression. Corken compressors make excellent oil-field gas compressors due to their ability to handle elevated suction pressures, however the ability to do oil-free compression is of no value since the gas will need to be treated later anyway. By injecting lubricating oil into the gas at the compressor suction, the ring and packing life can be quadrupled. Therefore, all of Corken's gas compressors are available with cylinder lubricators. On the vertical compressor the lubricator is attached to the crankcase and driven by a V-belt connected to a small sheave behind the flywheel (See Fig. 3.11A). The lubricator is gravity-fed by a reservoir of lubricant. The lubricator pumps the lubricant at a metered rate through a quill in the suction piping that atomizes the oil into the suction gas stream. The lubricant is carried by the gas to coat the valves and cylinder wall. A typical rate is 6 drops/min per lube point (1 pint = approximately 14,000 drops). The reservoir type lubricator should not be confused with the sump-feed type lubricator which draws oil out of the lubricator sump. The oil in the Corken lubricator sump is strictly for the internal lubrication of the lubricator parts.

For a more in-depth review of the complete lubrication system see section 4.5 on the HG600 lubrication system.

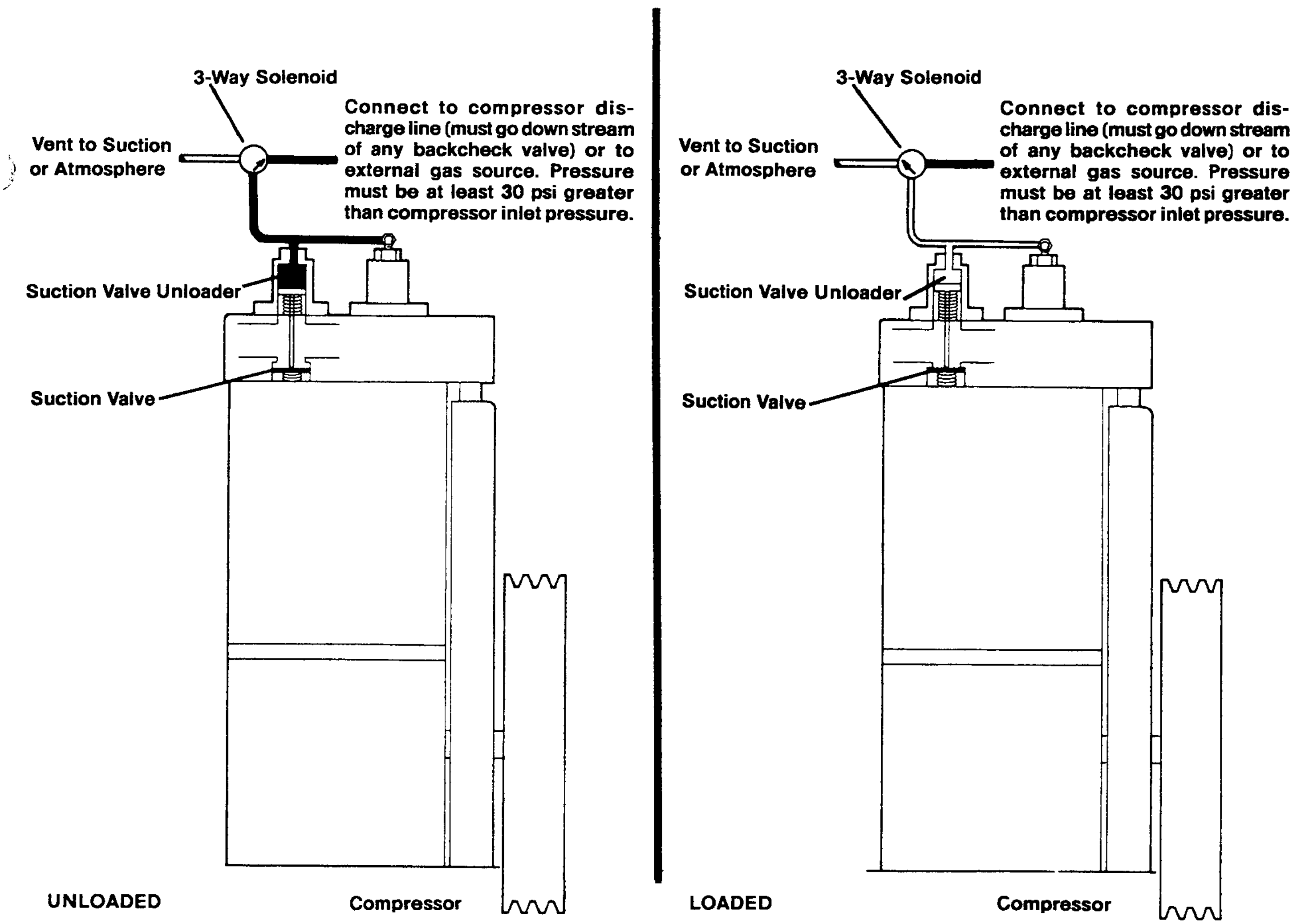
Figure 3.11A Compressor with Lubricator



3.12 SUCTION VALVE UNLOADERS

A suction valve unloader is a piston actuated device used to hold the suction valve in an open position (See Fig. 3.12A). When the suction valve is held open, the compression piston forces gas in and out through the suction valve and no compression takes place. Since no compression takes place, no load is exerted on the crankcase. The unloader consists of a small cylinder and a piston which is attached to a long rod which holds down the suction valve when the machine is unloaded. When the machine is loaded the rod is held clear of the suction valve by a spring pushing up on the unloader piston. To unload the compressor, high pressure gas is bled into the upper chamber of the unloader cylinder. If this gas is 30 PSI or greater than the suction pressure, the spring resistance will be overcome and the unloader piston assembly will be pushed down. To deactivate the unloader and reload the compressor, the high pressure gas in the upper unloader cylinder must be vented to a lower pressure point such as the atmosphere or the compressor suction.

Figure 3.12A SOLENOID CONTROLLED SUCTION VALVE UNLOADERS



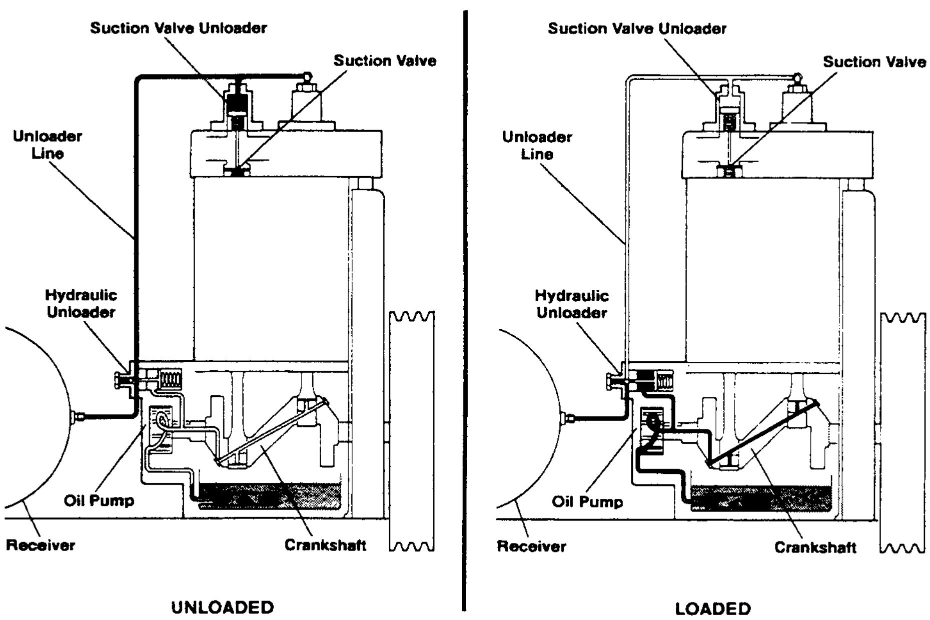
Unloaders are utilized for two primary functions. The first is loadless start which allows the compressor to start up under no load. This allows the load bearing parts in the crankcase to become thoroughly lubricated before being exposed to full load. Loadless start should always be used whenever the differential pressure at start-up exceeds 100 PSI. When differential pressure at start-up ranges from 50 to 100 PSI, it should be considered. For differential pressures below 50 PSIG it should not be used. Note that most liquid transfer compressors nearly always start at zero differential pressure (although the suction pressure may be quite high), so suction valve unloaders are not normally used because the discharge pressure of the compressor is not high enough to activate the unloaders.

The second common function of suction valve unloaders is for constant speed unloading. Whenever a compressor pumps gas into a finite volume such as a receiver tank, the pressure will steadily rise. When the pressure reaches the desired point no more gas should be pumped in. One way to do this is to stop the compressor but this is impractical if short cycle times require frequent starting and stopping. In these situations constant speed unloading is used to stop the gas compression process without stopping the compressor. Constant speed unloading is ideal for any application which requires more than five cycles an hour.

Fig. 3.12B LOADLESS STARTING WITH HYDRAULIC UNLOADER ASSEMBLY

LOADLESS STARTING

For Air or Non-contaminating or Non-hazardous Gas



Application:

This system is designed for compressors which are required to start up, under load, at frequent intervals. In order to reduce the stress on the compressor and motor during start up, the compressor valves are opened and held open just long enough for the compressor and motor to reach normal operating RPM.

Operation:

When the Compressor is not running, the Suction Valves are held open by the pressure in the Unloader Line from the Receiver. This is referred to as the "Unloaded" Condition.

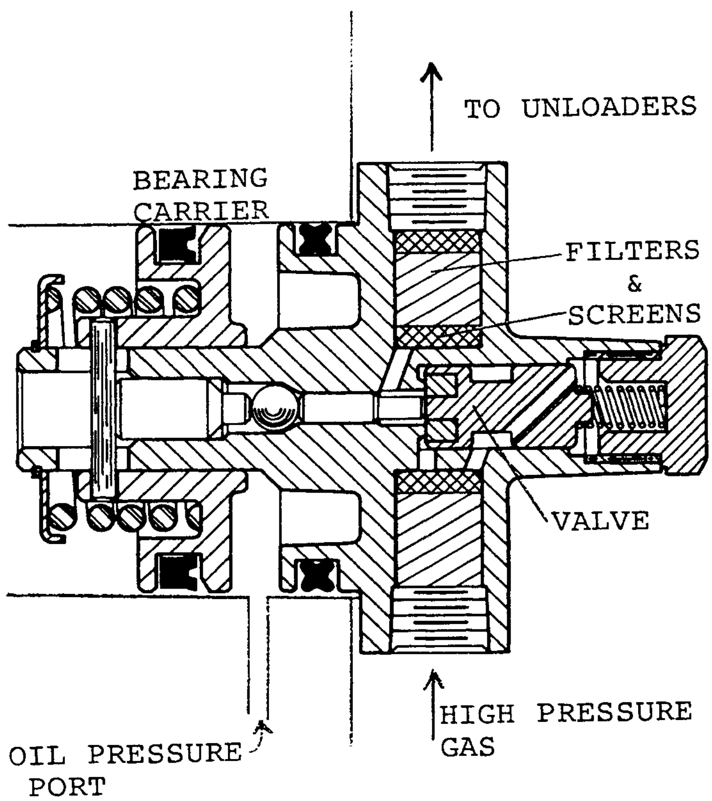
As the Compressor starts running, the crankcase oil pressure rises and closes the Hydraulic Unloader, this vents the Unloader Line and the Suction Valves may begin normal operation. The Compressor is now running in a "Loaded" condition.

Further protection to the Compressor is afforded due to the Hydraulic Unloader closing in the event of low oil pressure. The unit will run "Unloaded" thus reducing damage to the bearings.

The Key to controlling the suction valve unloaders, is controlling the flow of gas pressure to and from the top of the unloader cylinder. This can be done with either mechanically or electrically controlled valves.

Air compressors typically use mechanical devices for unloader control. While mechanical devices are less versatile than electrical ones, they are much less expensive and easier to install because no complex control wiring is required. Because mechanical devices vent the high pressure gas out of the unloader cylinder to atmosphere, they are limited to air and clean, non-toxic and non-corrosive gases. Mechanically controlled loadless starting is achieved via a hydraulic unloader assembly, a valve that is hydraulically controlled by crankcase oil pressure. The valve is connected to a high pressure gas source (at lease 30 PSI greater than suction) and to the top of the unloader cylinder. Before the compressor is started, the valve is open. Since the valve is open, the unloader cylinder is filled with high pressure gas and is in an activated state (See Fig. 3.12B). As the oil pressure builds, the valve closes (See Fig. 3.12C). The high pressure gas in the line between the hydraulic unloader valve and the suction valve unloaders bleeds back through passageways in the assembly until the pressure drops to a point where the unloader deactivates.

Figure 3.12C HYDRAULIC UNLOADER ASSEMBLY



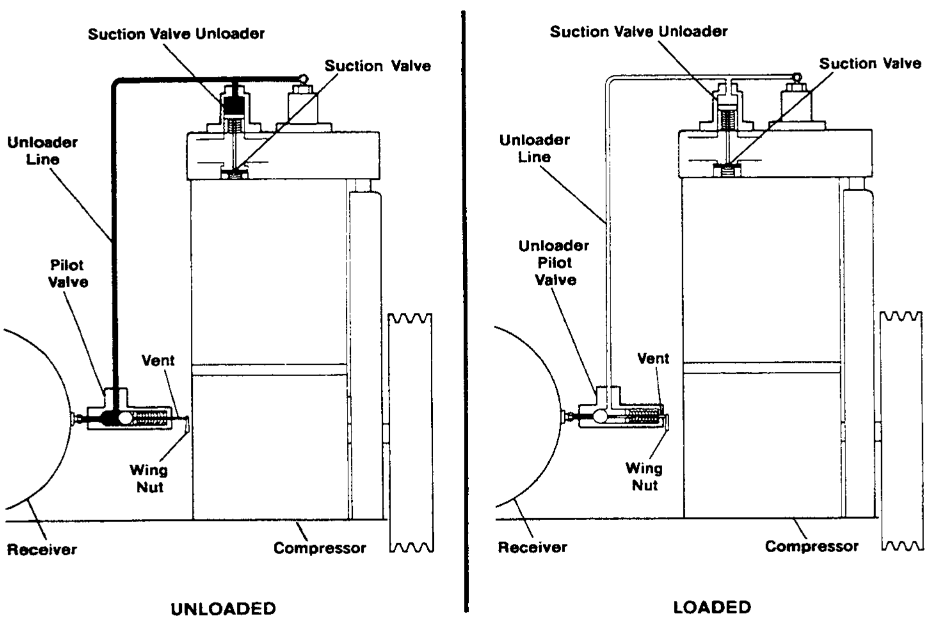
Mechanical control for constant speed unloading is provided with a pilot valve. A pilot valve is an adjustable ball and spring valve which is closed until the inlet pressure rises to a point where the spring is compressed and the valve opens to supply high pressure gas for unloader activation. (See Figs. 3.12D & E). As the inlet pressure to the valve falls, the spring reseats the ball seal and the gas trapped in the unloader bleeds out through threads on the adjusting nuts. Both the activation pressure and differential pressure can be adjusted using adjustment nuts on the pilot valve (See Fig. 3.12E).

When both loadless starting and constant speed unloading is required, "dual control" should be specified. The tubing arrangement used to achieve dual control is shown in Fig. 3.13F. When one of the two controlling devices is open, the dual control check valve seals off the second device so the high pressure gas is routed to the unloader. The block valve in front of the pilot valve allows the pilot valve to be eliminated from the control scheme. This allows the user to easily switch from constant speed unloading to start/stop operation.

Figure 3.12D CONSTANT SPEED UNLOADING WITH A PILOT VALVE

CONSTANT SPEED UNLOADING

For Air or Non-contaminating or Non-hazardous Gas



Application:

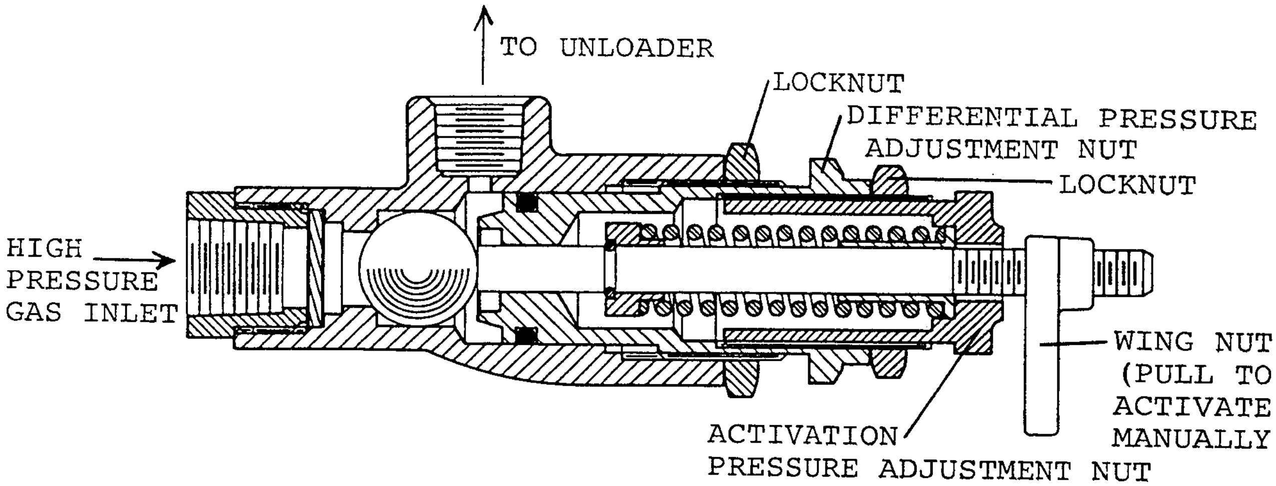
This system is designed for compressors in continuous run operation or engine driven units.

Operation:

The pressure in the receiver is set at a predetermined maximum and minimum by adjusting the Unloader Pilot Valve. When the Compressor is running and the pressure reaches the maximum setting, the Unloader Pilot Valve opens, the line to the Suction Valve Unloader is pressurized and the Suction Valves are held open. The Compressor is now running "Unloaded" thus reducing stress on the compressor and motor. As the pressure in the receiver drops to the preset minimum, the Pilot Valve closes, the line to the Suction Valve Unloader is vented, and the Suction Valves return to normal operation.

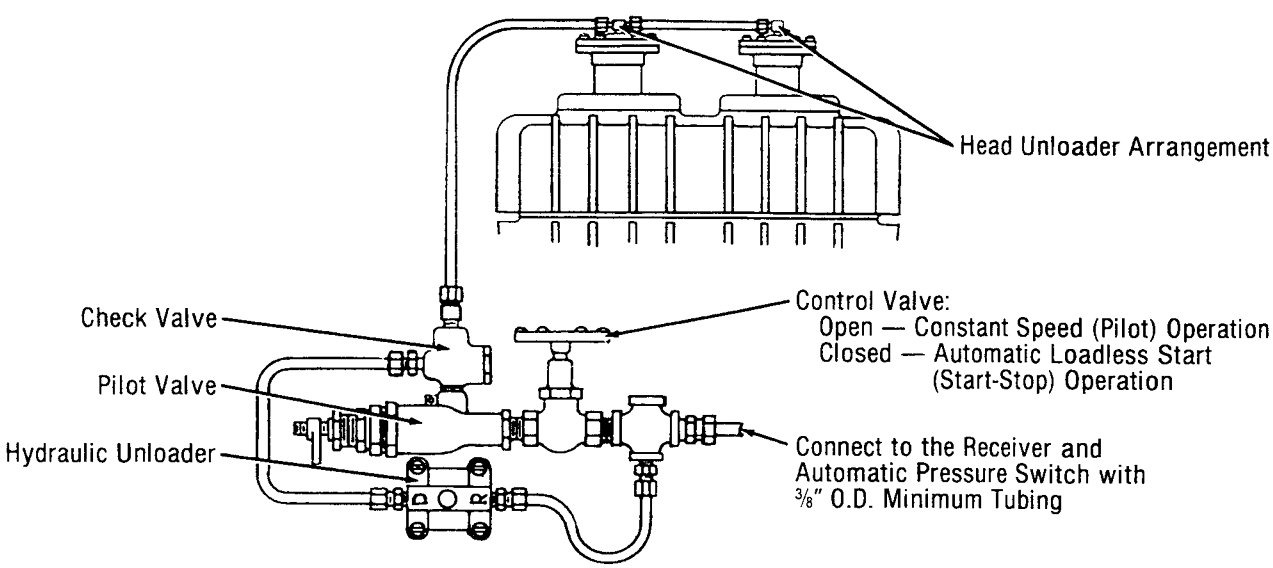
The Compressor can be manually "Unloaded" at any time by adjusting the wing nut on the Pilot Valve.

Figure 3.12E PILOT VALVE CUTAWAY



Move DIFFERENTIAL PRESSURE NUT IN to increase differential.
Move ACTIVATION PRESSURE NUT IN to increase activation pressure.

Figure 3.12F DUAL CONTROL



Tubing Installation for Combination Control System (Specification 78).

In some cases it is desirable to use mechanical unloader control devices on compressors handling flammable, toxic or corrosive gases. In these situations a separate, inert gas such as nitrogen must be used for injection into the unloader. Air may be used for non-flammable gases. Never use a toxic, corrosive or flammable gas for unloader activation when mechanical unloader control devices are being used.

Whenever it is necessary to use a toxic, corrosive or flammable gas for unloader activation, a closed loop, electrical control system such as the one shown in Fig. 3.12G must be used. When power is being applied to the solenoid, high pressure gas will flow into the unloader. When the power to the solenoid is shut off, the high pressure gas will be shut off and the gas in the unloader will vent into the compressor suction. This scheme allows gas from the compressor discharge to be used to activate the unloader while maintaining zero leakage.

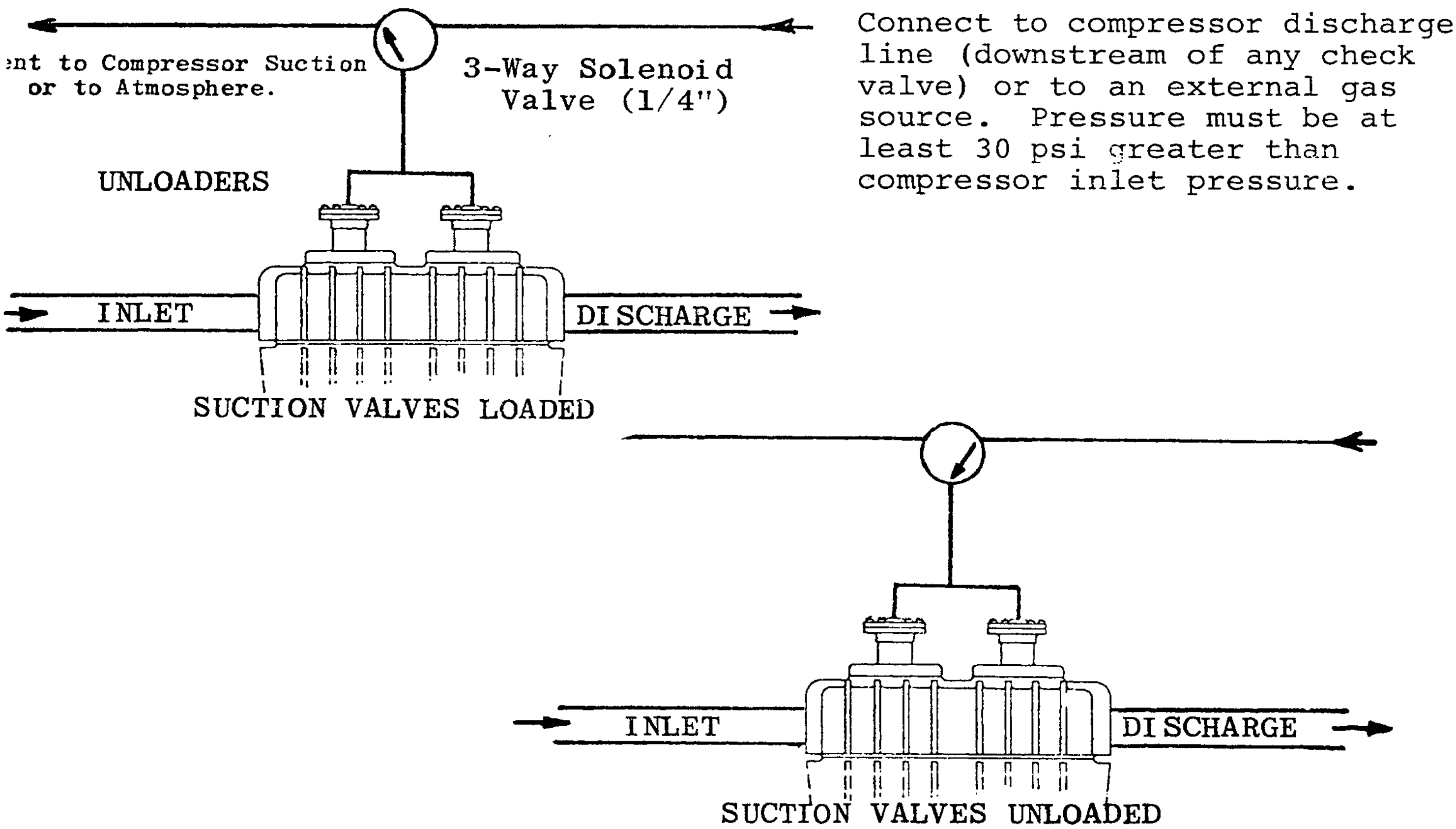


Figure 3.12G 3-WAY SOLENOID VALVE CONTROL OF SUCTION VALVE UNLOADERS

Loadless start for solenoid controlled unloaders is achieved by using a 10 - 30 second time delay to activate the solenoid and unloader for the first few seconds of operation. Low oil pressure switches typically have a thirty second time delay that can be used for this function. If no low oil pressure switch is available, a time delay relay connected to the motor starter can be used.

Constant speed unloading for solenoid controlled unloaders employs a pressure switch to activate the solenoid and unloader whenever the set pressure is exceeded.

An extra advantage of solenoid controlled unloaders is that virtually any instrument which sends out an electrical signal can be used to activate the unloader. Any array of temperature switches, pressure switches, timers, etc. can be used to operate the suction valve unloaders.

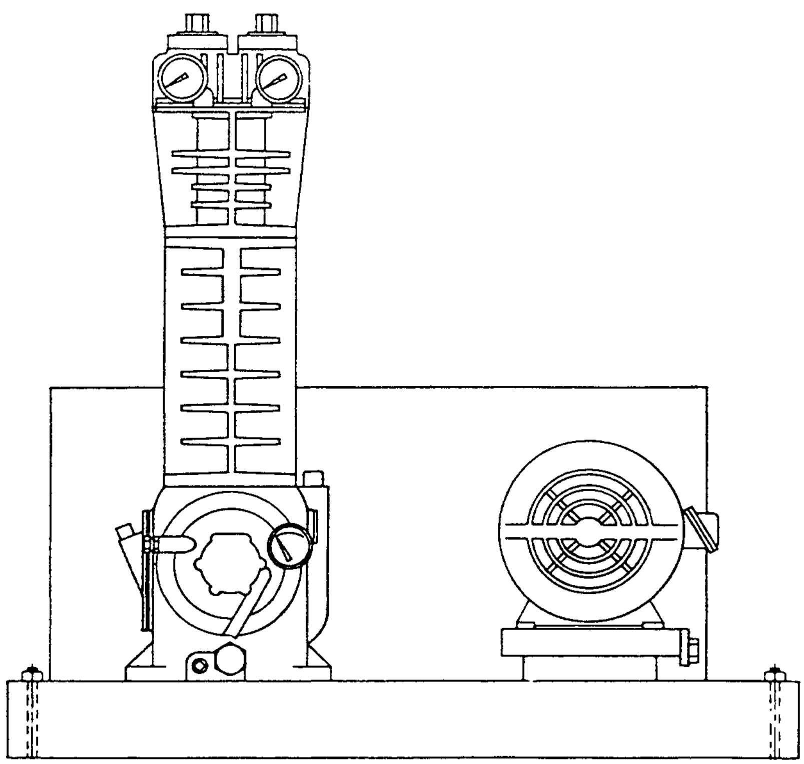
A note of caution: unloaders should not be cycled more than about once a minute to prevent excess heat build up. Also, if the compressor is expected to run unloaded for extended periods of time (say, greater than 5 - 10 minutes), it should probably be shutdown and restarted when required.

There is one final method of compressor unloading which doesn't even require suction valve unloaders. This method is called by-pass unloading. When the compressor starts up the discharge gas is either vented to atmosphere (for air) or looped back to the suction. After start up, the by-pass loop is shut off and the appropriate valves are open so the compressor discharges against full system pressure. Care must be taken in the design of such a system that the by-pass gas be properly cooled before reaching the compressor inlet.

3.13 MOUNTING CONFIGURATIONS AND LIQUID TRAPS

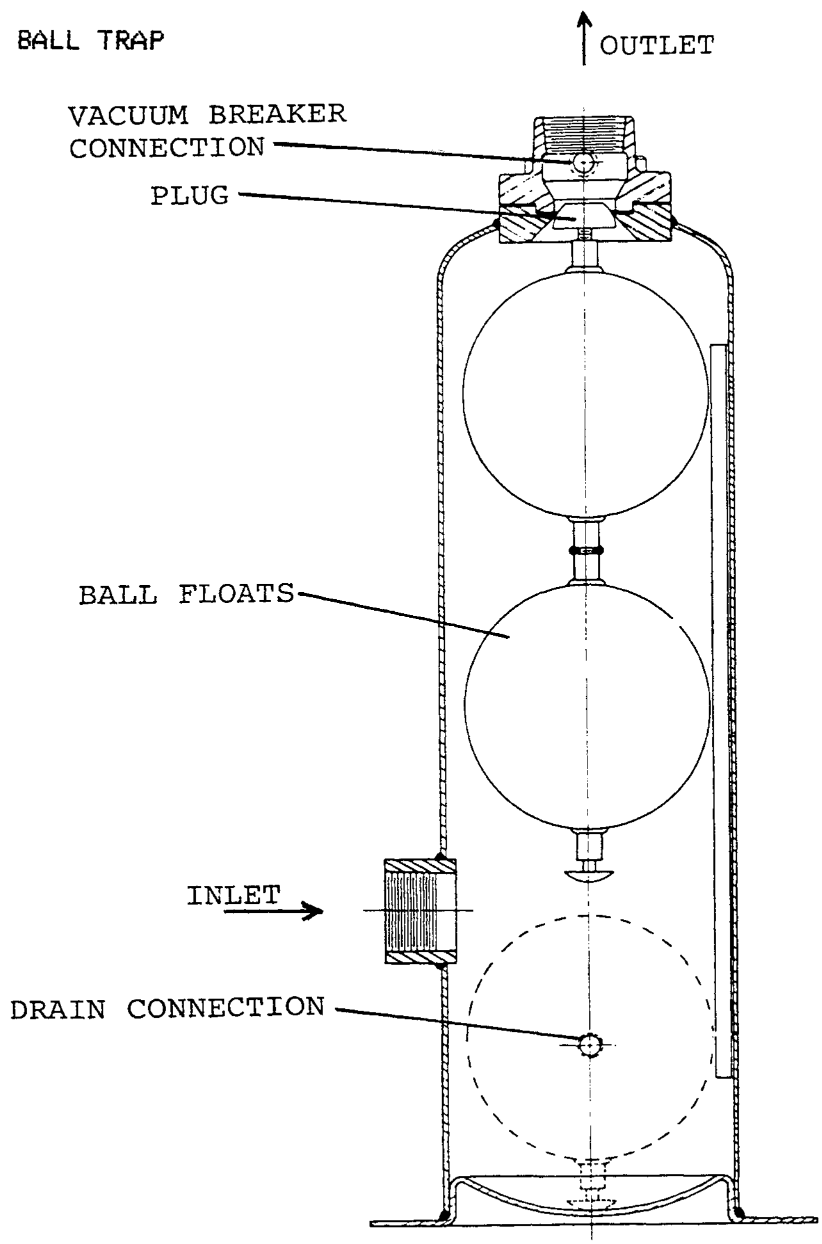
Corken compressors are available in a number of standard packages. The simplest is the "103" mounting which includes the baseplate, V-belt drive, beltguard and adjustable motor base (Fig. 3 13A). Most single-stage models also come equipped with pressure guages.

Fig. 3.13A 103 MOUNTING



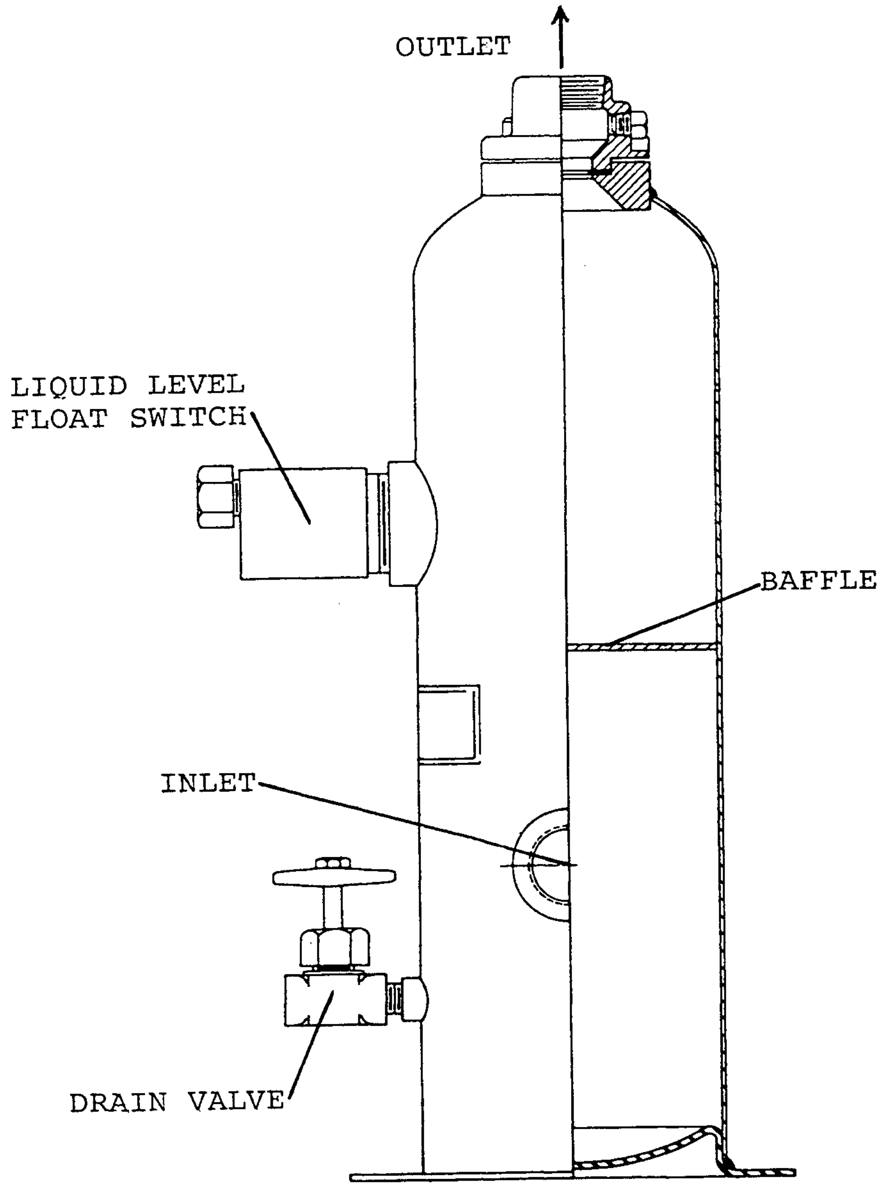
For many applications the suction gas stream will contain entrained liquids. Since liquids are incompressible, it is absolutely essential that they be removed to prevent "liquid slugging" which will quickly result in the failure of the compressor. Corken offers three types of liquid traps for removal of entrained liquids. The simplest is a ball float trap. As the liquid enters the trap the gas velocity is greatly reduced which allows the entrained liquid to drop out. If the liquid level rises above the inlet, the ball floats will plug the compressor suction. The compressor creates a vacuum in the inlet piping and continues to operate until it is manually shut down by the operator. Before restarting the compressor, the trap must be drained and the vacuum-breaker valve opened to allow the ball float to drop back to the bottom of the trap. This type of trap is only appropriate for use where the compressor is kept under fairly close observation by the operator. Typically, this trap is only used in the LP-gas and agricultural ammonia business and is included in the -109 and -107 mountings.

Figure 3.13B BALL TRAP



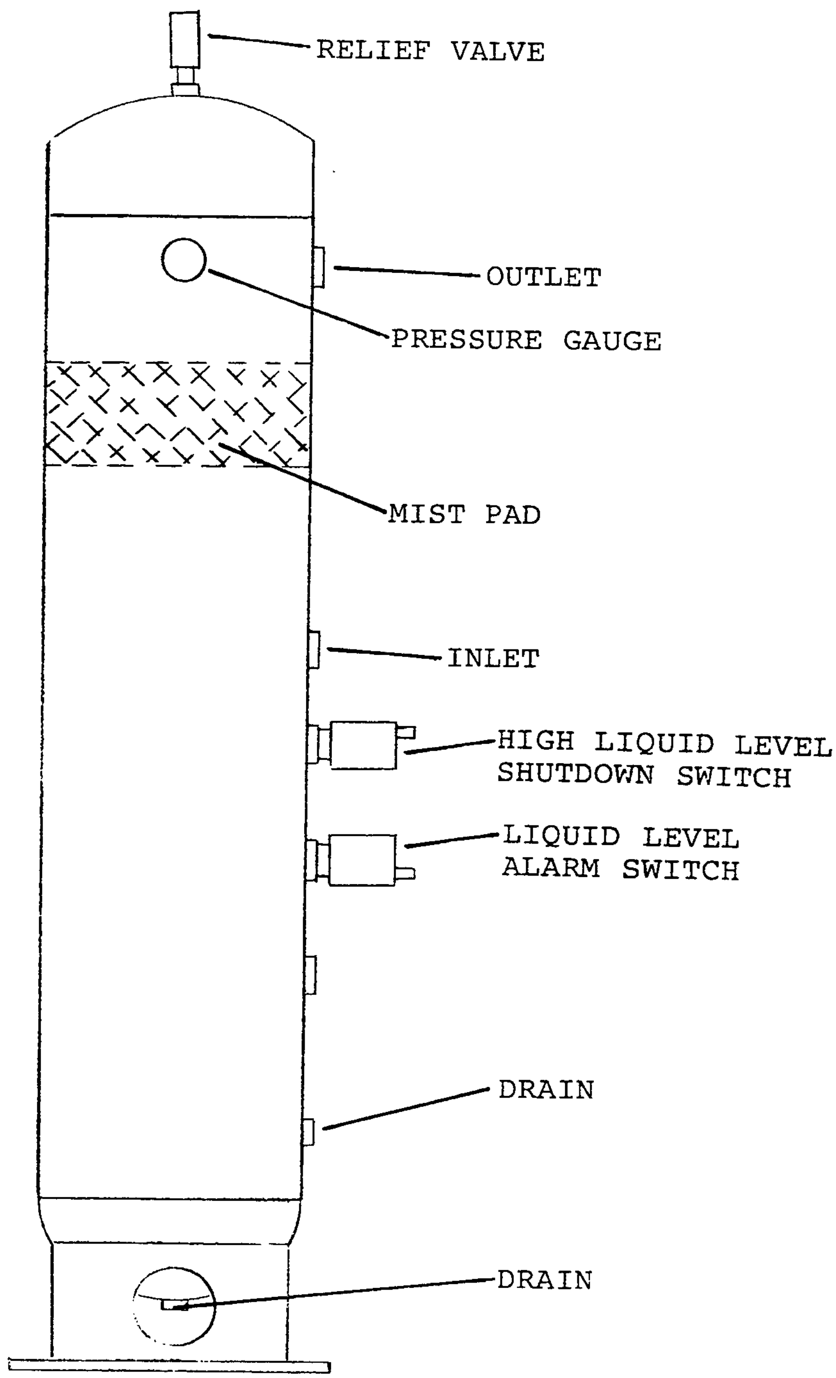
For continuous duty operation applications such as those found in the chemical industry, an automatic trap should be used (see Fig 3.13C). The automatic trap replaces ball floats with electrical float switches. If the liquid level should rise to too high a level, the level switch will open up and disconnect the power to the motor starter which will stop the compressor. This design insures the machine will be protected even when it is not under close observation and is standard in the -109A and -107A mounting configurations.

Fig. 3.13C AUTOMATIC LIQUID TRAP



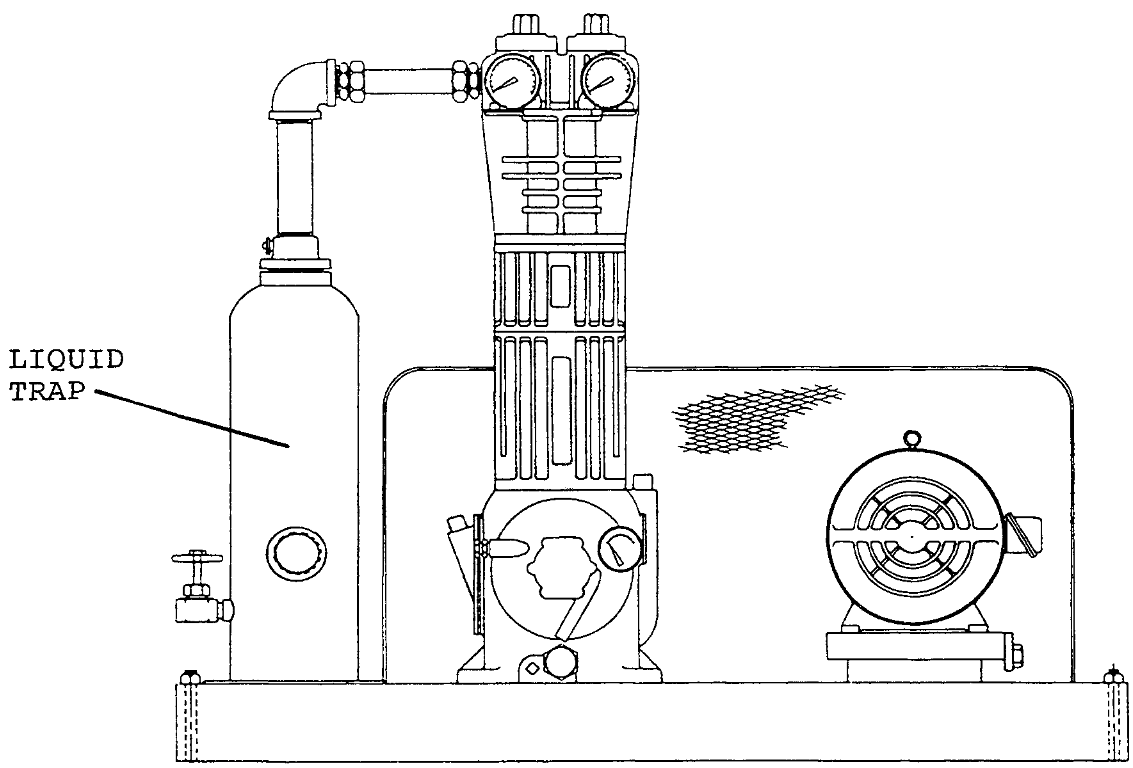
Corken's most sophisticated trap provides the most thorough liquid separation. This trap is larger than the other two traps and is ASME code stamped. It contains two level switches, one for alarm and one for shutdown. In some cases the alarm switch is used to activate a dump valve (not included with trap) or sound an alarm for the trap to be manually drained by the operator. This trap also contains a mist pad. A mist pad is a mesh of interwoven wire that is extremely effective at disentraining fine liquid mists. The ASME code trap is standard in the -109B and -107B mounting configurations.

Fig. 3.13D ASME AUTOMATIC TRAP



Any of Corken's three types of liquid traps can be purchased as part of a -109 or -107 type package. The -109 package (Fig. 3.13E) includes a liquid trap with interconnecting piping along with the standard features of the -103 type package. The -107 package (Fig. 3.13F) includes all the features of the -109 package as well as a strainer, a relief valve, and piping connecting the suction and discharge of the compressor through a four way valve. This allows the suction and discharge sources to be reversed for unloading of liquified gas tanks (this will be explained in Chapter 5).

Figure 3.13E 109 MOUNTING



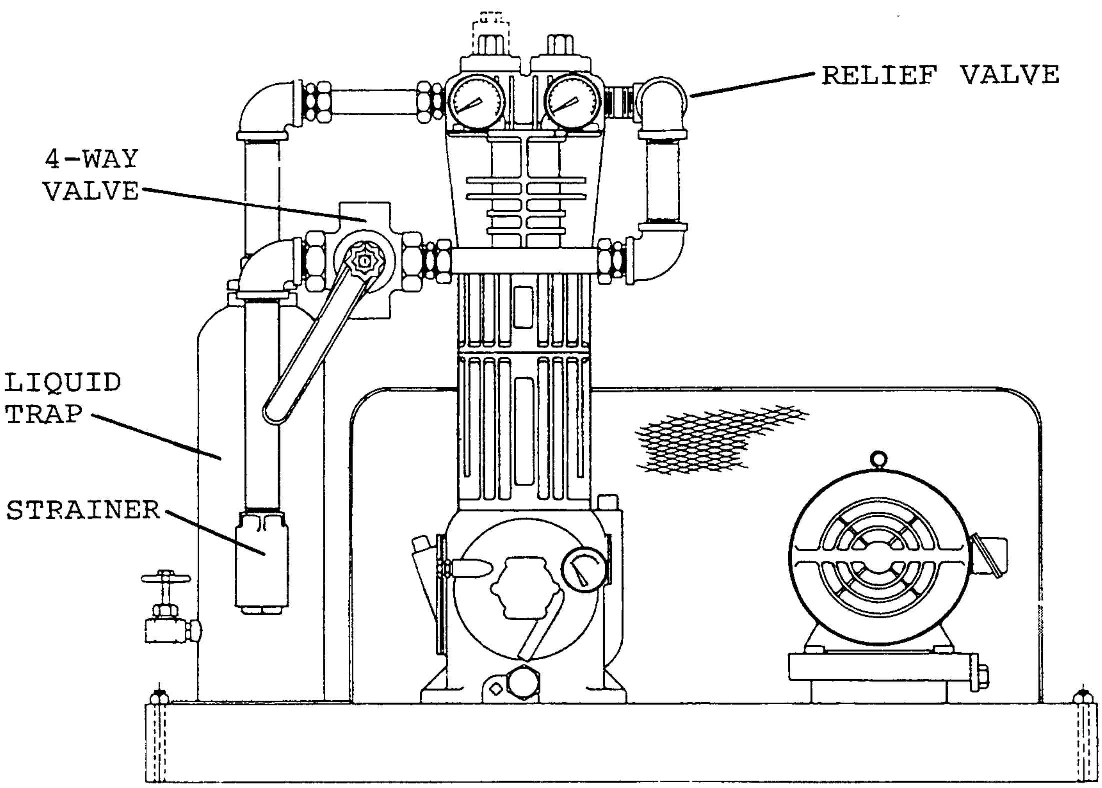


Figure 3.13F 107 MOUNTING

Corken air and gas compressors are available mounted on tanks up through the 490 size compressor. The 590 and 690 are too large for tank mounting to be practical.

3.15 MODEL D791 AND D891 COMPRESSORS

The latest additions to CORKEN's vertical compressor line are the D791 and D891 compressors. These units are essentially double-acting versions of the single-acting D590 and D690, and are the only double-acting vertical CORKEN compressors. The double-acting design requires several design modifications that make these models different from CORKEN's other vertical compressors.

The most notable difference is that the valves have been moved from the compressor head to the compressor cylinder. The suction and discharge monifold for each stage is cast into the cylinder. This design allows a clearance pocket to be added to the first stage of the D791 compressor for controlling interstage pressure and temperature. The second stage of te D791 and the single-stage D891 have fixed clearance volumes.

The double-acting models can provide almost double the capacity of the D590/D690 with virtually no increase in the units' physical size. The double-acting design also results in much better rod revesal than is possible on the single-acting models.

The top packing set of the D791/D891 is exposed to the full variation of suction to discharge pressure, while the top packing set of a single-acting model is exposed to a pressure close to the suction pressure with very little fluctuations. Because the fluctuating pressure on the D791/D891 result in much larger loads on the packing than the loads seen on the single-acting models, the D791/D891 require a different type of packing. Cheveron packing is not suitable for pulsating loads, so the D791/D891 use segmented packing instead. Segmented packing handles these pulsating loads well but does not provide as tight a seal as the cheveron packing. This makes the D791/D891 unsuitable for vacuum service.

Since the D791/D891 will be used extensively in oilfield applications, the crankcase is equipped with a mounting flange and an internal chain drive for the cylinder lubricator. Of course, a lubricator can be mounted or not, depending upon the application.

Please refer to the specification chart (Fig. 3.1c) for in depth information on materials, bore and stroke dimensions, piston displacements, etc.

It should also be noted that the D791 is typically provided <u>without</u> an intercooler, and that an integral, air-cooled intercooler like those on the D390 and D590 is not available. Arrangements for intercooling must be made as part of the packaging for the compresssor.

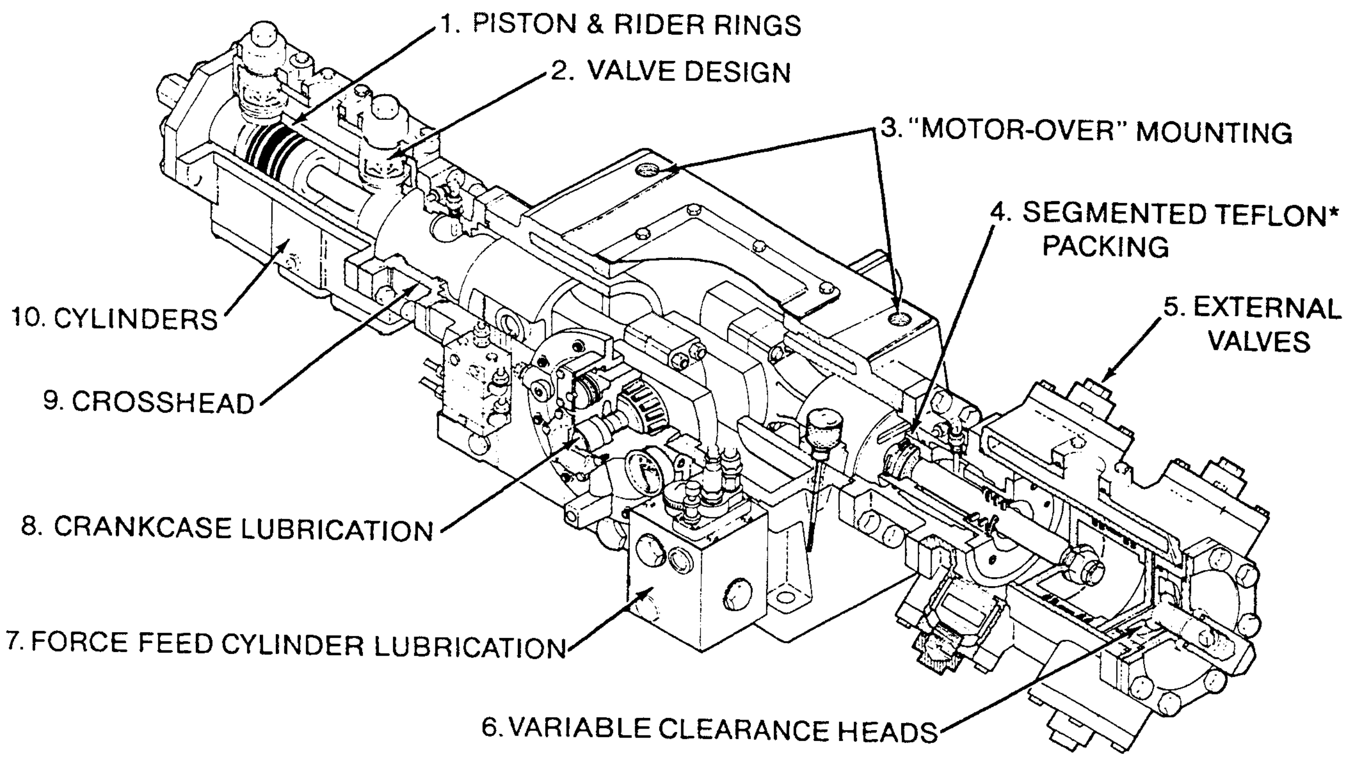
CHAPTER FOUR

SPECIFYING HORIZONTAL CORKEN COMPRESSORS

4.1 INTRODUCTION

Corken's newest compressor line is the HG600 Horizontal Balanced Opposed Compressor. At present, the compressor is primarily being used for hydrocarbon gas compression in the oilfield and in chemical plants. As more cylinder and rod packing designs become available, the line will be expanded to cover the entire range of gas compression applications already handled by Corken's vertical compressors.

Figure 4.1 HG600 HORIZONTAL BALANCED OPPOSED COMPRESSOR



The horizontal balanced opposed design concept has existed for quite some time; but it has only recently been applied to compressors under 100 HP. The design utilizes a single crankcase with several different cylinder sizes. This allows a wide range of single and two stage compressors to be built with a minimum number of basic components.

'Balanced Opposed' means that the cylinders are parallel and opposite each other and that the "reciprocating mass" (i.e. the piston) of the two cylinders are the same, even if the cylinders are different sizes.

This design concept has several distinct advantages:

1) Low vibration: Both pistons of the HG600 always move at exactly the same velocity and acceleration, but in opposite directions. This, plus the fact that they are of the same weight means that virtually all inertial forces are canceled which results in very low vibration.

2) Optimum utilization of crankcase horsepower capacity: Situations can arise on machines which have fixed cylinder sizes where the piston displacement is fully employed, but the horsepower load on the crankcase is a fraction of its rated horsepower. Since a range of cylinder sizes are available on the HG600, cylinders may be chosen for a given application to fully utilize the rated horsepower of the machine. This greatly expands the range of applications that can be covered with a single crankcase design.

3) "Tunability": The choice of cylinders allows relative cylinder sizes to be chosen for two-stage applications which result in minimum overall operating temperatures. The ideal is for the compression ratio per stage to equal the square root of the overall compression ratio (it is assumed the gas is intercooled to close to the suction temperature). This results in the same compression ratio per cylinder and the same discharge temperature from each cylinder. When several different relative cylinder combinations are available it is possible to approximate this ideal situation. Variable volume pockets may be utilized to get even closer to this ideal situation (See section 4.2).

The selection procedure used for choosing HG600 compressors is quite complex and is best left to a computer. Of course the theories discussed earlier still apply and the sizing procedure for vertical compressors can still be used for making rough approximations.

A complete analysis of HG600 model numbers is given in APPENDIX E.

4.2 CRANKCASE AND CYLINDERS

The HG600 series utilizes one crankcase design for all models. The running gear is rated for 60HP for continuous duty service and 75HP for intermittent duty service. The crankcase has two throws with a 3" stroke, and the running gear is pressure lubricated by a reversible, gear-type oil pump.

CHAPTER 5

SPECIAL COMPRESSOR APPLICATIONS

5.1 STORING GASES UNDER PRESSURE

One or the most typical applications for small compressors is the filling of tanks with compressed air or gas for later usage. The time required to fill a tank from an initial pressure of P_i to a final pressure of P_f using gas supplied to the compressor suction at P_1 can be calculated with the following equations:

$$tf = \frac{V\ (P_f - P_i)}{PD\ (E_a)(P_1)}$$

tf = tank fill time (min)
V = Tank Vol. (ft^3)
P_1 = Compressor suction pressure (psia)
P_i = Initial (cutin) tank pressure (psi)
P_f = Final (cutout) tank pressure (psi)
PD = Compressor piston displacement
E_a = Average Vol. Eff. $(VE_1 + VE_2)/2$

Compressor receivers are necessary for handling two classes of applications:

1) Applications with erratic or intermittent usage rates. This includes applications where a large flow of gas is required for a short period of time and it would be uneconomical to buy a compressor with a capacity equal to the maximum flow rate. The tank provides surge capacity to allow a compressor supplying gas at a constant rate to supply a system which has "peaks" and "valleys" on its demand curve that are both below and above the compressor capacity.

2) Applications where the average usage is so small compared to the compressor capacity that is uneconomical or unfeasible to run the compressor all the time.

A compressor/receiver combination should be controlled by either a constant speed unloading or a start/stop control system. Constant speed unloaders have already been reviewed in section 3.12. A start/stop control system uses a pressure switch to start the compressor whenever the tank pressure falls below a certain set pressure ("cut in pressure") and turns the compressor off when the tank has been pressurized to its maximum set pressure ("cut out pressure"). Start/stop control systems are usually used with loadless starting (see Section 3.12).

When a compressor system will be cycled less than five times an hour, a start/stop control system is recommended. When a system cycles more than five times an hour, constant speed unloading should be utilized. When the system cycles more than five times an hour, the wear and tear on the motor and compressor outweigh the energy savings from stopping the compressor.

At a usage rate of Q at a pressure of P_0, it will take te minutes to reduce a tank of volume V from pressure P_i to pressure P_n. This is expressed as:

$$te = \frac{(P_i - P_f)}{P_0}\frac{V}{Q}$$

P_i = Initial (cut in) Pressure (psi)
P_f = Final (cut out) Pressure (psi)
P_0 = System Usage Pressure (psia)
V = Tank Volume (ft^3)
Q = ACFM usage rate at P_0

If the time-to-empty (te) and the time-to-fill (tf) are combined and the gas usage during the time the compressor is operating is taken into consideration, the cycle time can be calculated with the following equation:

$$t(cycle) = \frac{V(P_f - P_i)}{PD(E_a)(P_1) - Q(P_0/P_1)} + \frac{V}{Q}\frac{(P_f - P_i)}{P_0}$$

$$= V(P_f - P_i)\left(\frac{1}{PD(E_a)(P_1) - Q(P_0/P_1)} + \frac{1}{Q\,P_0}\right)$$

P_i and P_f = Tank initial and final pressure (psi)
P_1 = Suction pressure (psia)
P_0 = System usage pressure (psia)
Q = ACFM usage rate at P_0
PD = Piston displacement (ft^3)
E_a = Average Volumetric Efficiency

It can be seen from this formula that both the tank size and the difference between the cut-in (P_i) and cut-out (P_f) points have a significant effect on the cycle time.

5.2 LIQUIFIED GAS TRANSFER AND VAPOR RECOVERY

One of the most clever applications for a small gas compressor is for the unloading of tank cars of liquified gases. Common liquified gases include propane, ammonia and butane. For a complete list of common liquified gases with their physical properties see Appendix G.

5.2.1 LIQUIFIED GASES

Any pure substance and a large number of chemical compounds may exist as a solid, liquid or gas (See Sec. 1.1). The phase that a certain substance exists in is a function of its pressure and temperature. A liquified gas is defined as a substance that exists as a gas at normal atmospheric pressures and temperatures but can be liquified at moderate pressures. A gas is liquified by being compressed and then condensed into a liquid by cooling it to remove the heat of compression. The liquified gases commonly handled by Corken equipment do not require cooling beyond normal ambient temperatures (-25 to 110°F). Several common gases such as nitrogen, oxygen and helium can only be liquified by cooling to extremely cool temperatures well below -25°F. These liquified gases are known as cryogenic liquids. Corken compressors and pumps are not designed to handle liquids or gases in a cryogenic state (less than -25°F).

A liquified gas storage vessel is never completely filled with liquid. 10% to 15% of the storage volume is left for vapor space. The vapor space gives the liquid room to expand and contract as the temperature increases or decreases. The vapor space will be filled with gas formed from evaporated liquid. The pressure this vapor exerts on the storage volume is called the vapor pressure.

To demonstrate the concept of vapor pressure, imagine a cylinder closed at one end with a movable piston at the other. The cylinder is designed to quickly cool or heat up to match the surrounding ambient temperature. Initially, the piston will be extended for maximum volume and the cylinder will be filled with propane gas. Next, the piston is slowly pushed into the cylinder to compress the gas. When the gas has been compressed to the vapor pressure corresponding to the ambient temperature (See Appendix G), some of the gas will begin to condense into liquid. Up to the point where the gas starts to condense, the gas is said to be in a superheated state. After condensation begins, the gas is in a saturated state, and the further the piston is pushed into the cylinder, the more condensation will take place. The cylinder pressure will remain at the vapor pressure as the cylinder is pushed further into the cylinder and more condensation takes place. Near the end of the stroke, all the vapor will be condensed and the cylinder will be filled with incompressible liquid. At this point, no further volume reduction could take place without rupturing the cylinder.

If the piston is slowly withdrawn to its original position, the process would reverse its self. The liquid would revaporize and return to a gaseous state.

Now imagine the piston locked in position during the intermediate phase where the cylinder contains both liquid and vapor. If the ambient temperature were increased, the vapor pressure in the cylinder would increase. If enough heat were added, all the liquid would revaporize and the gas would be superheated. By the same token, if the ambient temperature were lowered, the vapor pressure would decrease.

It can be concluded that whenever the cylinder contains a two-phase mixture, heating the cylinder or increasing the containment volume will cause some of the liquid to 'boil off' into vapor. This leads us to the simplest definition of a liquified gas: "a liquid stored at its boiling point".

5.2.2 LIQUIFIED GAS TANK CAR UNLOADING

Safety regulations forbid railroad tank cars designed for transporting liquified gases from having bottom unloading connections. Chlorine truck tanks are also forbidden to have bottom unloading connections. In cases where bottom connections are allowed on liquified gas tanks, such as propane trucks, a pump is generally used to unload the tank. However, if a pump is used to unload a tank through a top connection, a negative NPSH is nearly always unavoidable. Even if siphoning effects are considered, pressure drops through excess flow valves and emergency valves exceed whatever suction head is available.

Negative NPSH's result, of course, in a considerable amount of cavitation which results in high wear on the pump and greatly reduced flow rates. (If you are unfamiliar with NPSH and cavitation, refer to any standard reference on pumps). While liquid pumps have been used for tank car unloading, their performance has been extremely poor.

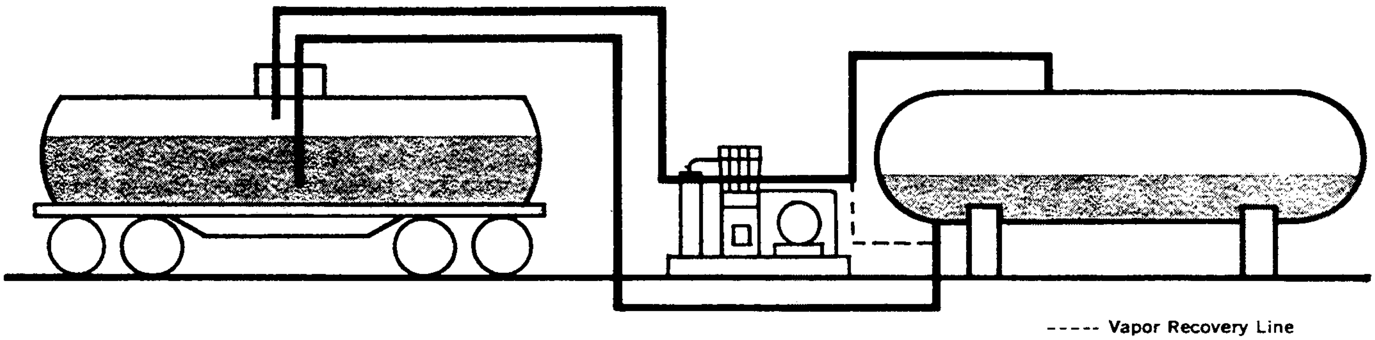


Figure 5.2.2A TANK CAR UNLOADING SCHEMATIC

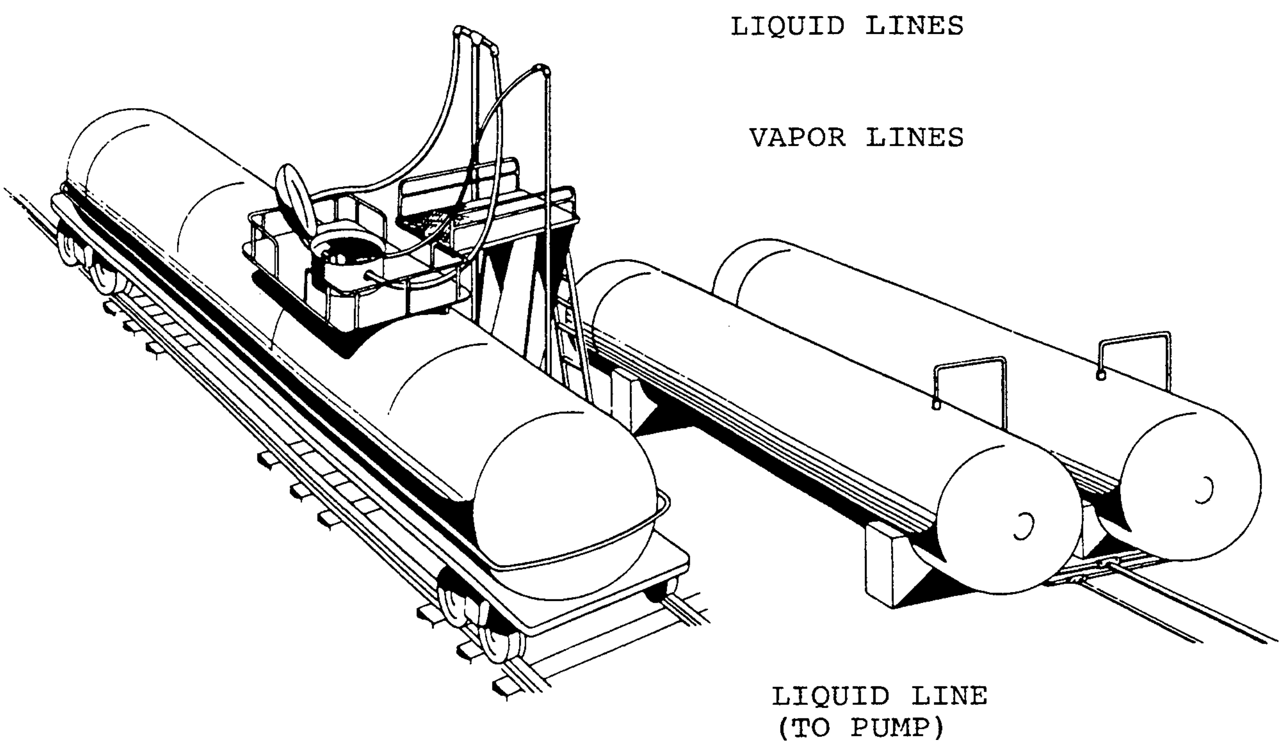


Figure 5.2.2B TYPICAL LIQUIFIED GAS TANK CAR UNLOADING TERMINAL

A gas compressor offers a simple and reliable method to overcome the difficulties encountered with pumps. The compressor suction is connected to the storage tank vapor space, the discharge is connected to the tank car vapor space, and the liquid spaces of the two tanks are connected thru a separate liquid flow line (See Fig 5.2.2A & B). When the compressor is started, it will attempt to evacuate the vapor space of the storage tank and lower its pressure. At the same time, the compressor will increase the pressure in the tank car, since it is on the discharge side of the compressor.

When the pressure difference between the vapor spaces exceeds the pressure drop in the liquid piping, liquid will begin to flow out of the tank car into the storage tank. The induced liquid flow rate equals the actual volumetric flow from the discharge of the compressor. Some vapor will be boiled off in the storage tank and some will condense in the tank car. However, the boil-off and condensation is so small compared to the volumetric flow rate that it has very little effect on the induced liquid flow.

To properly size a compressor for tank car unloading it is necessary to know four things: 1) The gas to be transferred, 2) The required liquid flow rate, usually given in GPM, 3) The temperature at which this flow rate is required, and 4) The pressure drop in the liquid and vapor piping (the pressure drop in the vapor piping is usually small and may be ignored).

At this point the required piston displacement may be calculated using the following equation:

$$PD = \frac{(GPM/7.48)\ CR}{VE}$$

PD = Piston displacement (ft^3/min)
GPM = Gallon per Minute
CR = Compression ratio
VE = Volumetric efficiency

This equation is based on isothermal compression to take account for the contraction that takes place in the discharge gas as it cools to ambient temperature. The compression ratio is determined by adding the pressure drop to the vapor pressure and dividing the sum by the vapor pressure:

$$CR = \frac{\text{Vapor Pressure + Pressure Drop}}{\text{Vapor Pressure}}$$

The volumetric efficiency may be found in Fig. 2.4A using the compression ratio calculated above. From the above equation it is evident that lower vapor pressure caused by lower temperatures result in higher compression ratios which ultimately cause a lower overall flow rate. In many cases it is necessary to analyze the flow rates at several different temperatures to assure adequate flow rates at all temperatures. This may be done by calculating a flow rate for the highest and lowest operating temperatures. For the most economical installation, the required flow rate should be specified at the midrange of the expected ambient temperatures. Corken uses a computer analysis program to quickly calculate liquid flow rates over a range of ambient temperatures in 10°F increments. See Fig. 5.2.C for a sample print out.

TANK CAR UNLOADING
PROPANE

LIQUID TRANSFER PHASE   TANK VOLUME= 33000 GALLONS   25 PSI PRESSURE DROP IN LIQUID TRANSFER SYSTEM   MOLECULAR WEIGHT= 44.1
TOTAL LIQUID VOLUME TO BE TRANSFERED= 29535 GALLONS OR 89.5 % OF TOTAL TANK VOLUME   TANK IS 90.0 % FULL

690   N=1.13 ACTUAL RPM= 730 PD= 53.8 CFM MAWP=265 PSIA

T1 F	V.P. PSIA	P2 PSIA	T2 F	CR	VE %	GPM	ACFM IN	ACFM OUT	BHP	TIME MIN	LB/HR LIQUID
0	38	63	28	1.7	87.5	212	47.0	28.4	10.5	139	54137
10	46	71	34	1.5	88.7	231	47.7	30.9	11.1	128	58934
20	55	80	41	1.5	89.6	248	48.1	33.1	11.7	119	63182
30	66	91	48	1.4	90.3	263	48.6	35.2	12.4	112	67228
40	78	103	56	1.3	90.9	277	48.9	37.0	13.1	107	70651
50	92	117	64	1.3	91.4	289	49.1	38.6	13.8	102	73750
60	107	132	73	1.2	91.8	299	49.3	40.0	14.6	99	76334
70	124	149	81	1.2	92.1	308	49.5	41.2	15.4	96	78626
80	144	169	90	1.2	92.3	316	49.6	42.3	16.3	93	80724
90	165	190	99	1.2	92.5	323	49.7	43.2	17.3	91	82443
100	189	214	108	1.1	92.7	329	49.8	44.0	18.4	90	83986
110	215	240	117	1.1	92.8	334	49.9	44.7	19.5	88	85301

BOIL-OFF PHASE   HEEL VOLUME= 165 GALLONS OF LIQUID OR 0.50 % OF TOTAL TANK VOLUME   5 PSI PRESSURE DROP IN VAPOR RECOVERY SYSTEM

T1 F	V.P. PSIA	P2 PSIA	T2 F	CR	VE %	BHP	ACFM IN	EQUIV.VAP. VOLUME(CFM)	TIME MIN
0	38	43	7	1.1	93.5	7.5	50.3	2066	41
10	46	51	16	1.1	93.7	7.9	50.4	1744	35
20	55	60	25	1.1	93.8	8.4	50.4	1490	30
30	66	71	34	1.1	93.9	8.9	50.5	1267	25
40	78	83	44	1.1	94.0	9.5	50.5	1094	22
50	92	97	53	1.1	94.0	10.2	50.5	946	19
60	107	112	63	1.0	94.0	10.9	50.6	829	16
70	124	129	72	1.0	94.1	11.7	50.6	729	14
80	144	149	82	1.0	94.0	12.7	50.5	640	13
90	165	170	92	1.0	94.0	13.7	50.5	569	11
100	189	194	102	1.0	94.0	14.9	50.5	506	10
110	215	220	112	1.0	94.0	16.3	50.5	452	9

VAPOR RECOVERY PHASE   40 PSIA FINAL EVACUATION PRESSURE

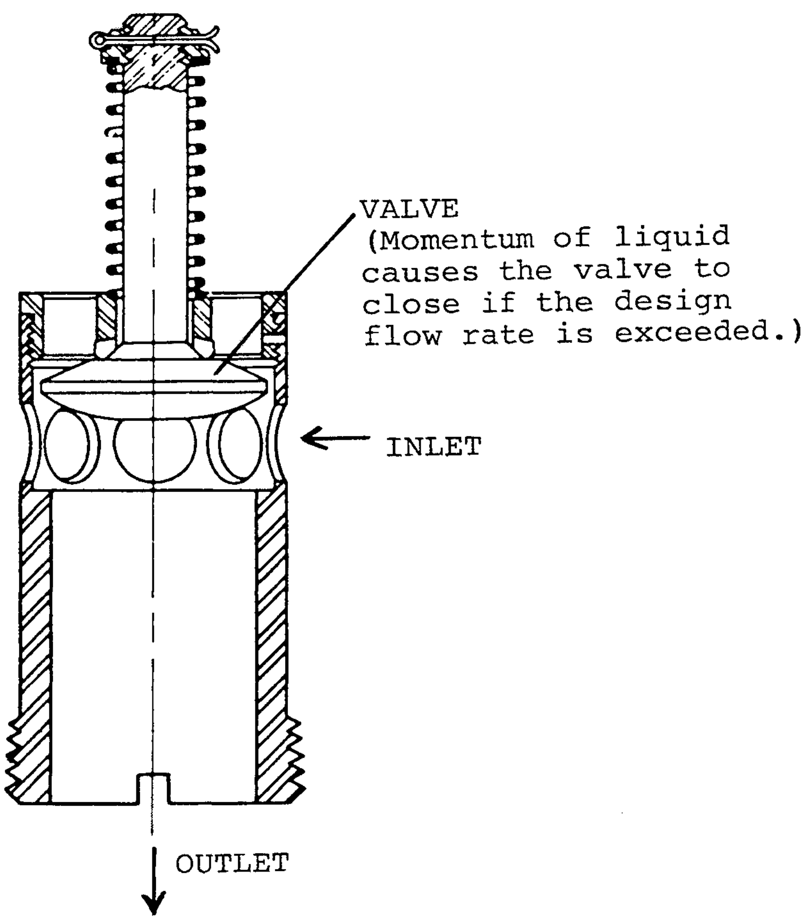
T1 F	V.P. PSIA	P2 PSIA	T2 F	VE(%) INT	VE(%) FINAL	VE(%) AVG	ACFM INT	ACFM FINAL	ACFM AVG	BHP MAX	CRITICAL CR	CRITICAL P1	TIME MIN	EQUIV LIQUID GAL	TOTAL TIME HRS
10	46	51	23	93.7	91.8	92.8	50.4	49.4	49.9	8.7	1.3	40	12	54.4	2.9
20	55	60	43	93.8	89.2	91.5	50.4	48.0	49.2	10.0	1.5	40	29	133.2	3.0
30	66	71	63	93.9	86.1	90.0	50.5	46.3	48.4	11.4	1.8	40	46	226.2	3.0
40	78	83	84	94.0	82.8	88.4	50.5	44.5	47.5	12.6	1.8	45	62	324.1	3.2
50	92	97	105	94.0	78.9	86.5	50.5	42.4	46.5	13.9	1.9	50	79	434.8	3.3
60	107	112	125	94.0	74.9	84.5	50.6	40.2	45.4	15.3	1.9	60	96	549.4	3.5
70	124	129	146	94.1	70.3	82.2	50.6	37.8	44.2	17.0	1.9	70	113	675.8	3.7
80	144	149	168	94.0	65.1	79.5	50.5	35.0	42.8	18.9	1.9	77	132	821.2	4.0
90	165	170	190	94.0	59.6	76.8	50.5	32.0	41.3	20.9	1.9	89	151	969.1	4.2
100	189	194	212	94.0	53.4	73.7	50.5	28.7	39.6	23.1	1.9	102	173	1134.5	4.5
110	215	220	234	94.0	46.8	70.4	50.5	25.2	37.8	25.6	1.9	116	196	1309.1	4.9

NOTE:ASSUMPTIONS FOR CALCULATIONS 1)PRESSURE DROPS REMAIN CONSTANT 2)INDUCED FLOW BASED ON ISOTHERMAL COMPRESSION 3)BHP AND TEMP BASED ON ADIABATIC COMPRESSION

One final consideration must be taken into account in sizing liquified gas transfer compressor systems - the excess flow valve. The excess flow valve is a safety device used to stop the liquid flow from a tank caused by a ruptured hose or pipe. A cross section of an excess flow valve is shown in Fig. 5.2.2D. The valve remains open as long as the flow rate is less than the design rate of the valve. If the liquid flow rate through the valve is excessive (as would be the case in the event of a hose rupture), it will close, stopping all liquid flow. To reopen the valve, the pressure at the inlet and outlet must be equalized.

Because of the flow limitations created by an excess flow valve, it is important not to use too large a compressor.

Figure 5.2.2D EXCESS FLOW VALVE CROSS SECTION



5.2.3 AIR AND NITROGEN PADDING

In some situations it is desirable to unload liquified gas tank cars by pushing the liquid out with a second, different type of gas. The two situations where this technique is typically used are: 1) Highly corrosive gases like chlorine, and 2) liquified gases or volatile liquids with low vapor pressures that must be unloaded through top end connections. Ethylene oxide is an example.

Highly corrosive gases such as chlorine tend to wear out gas compressors quickly. By compressing air instead of chlorine, the compressor life can be greatly extended. The primary disadvantage of this method is that a much larger compressor is required if an air pad is used than if a chlorine gas pad is used. Since a much larger intake volume is required on an atmospheric air compressor than on a chlorine compressor to achieve the same discharge volume. This is because the chlorine compressor takes its suction pressure at vapor pressure while the air compressor takes its suction pressure at atmospheric conditions which are well below the vapor pressure of chlorine.

Air padding is effective on products with vapor pressures less than 30 PSIA. A tank pressure of 30 PSIA will result in the compressor operating at 2 compression ratios. Products with higher vapor pressures may often be padded with nitrogen. Nitrogen can often be supplied to the compressor inlet at an elevated pressure so that the ratio between tank pressure and compressor suction is under 2. Nitrogen may also be desirable since it is relatively inert and will not pose the fire hazard that air might.

5.2.4 LIQUID HEEL RECOVERY

The liquified gas in a railroad tank car is removed through a dip tube which extends from the top of the tank towards the bottom. When the liquid level of the tank drops below the bottom of the dip tube, no more liquid can be removed from the tank. The "liquid heel" is typically 1% or less of the tank volume.

The remaining liquid and vapor has substantial economic value and may be recovered by removing it from the tank in a vapor rather than a liquid state.

To switch the compressor system from liquid transfer to a vapor recovery operation, the liquid line is blocked and the suction and discharge of the compressor are reversed by means of a four-way valve (to be discussed in section 5.2.6). Now the compressor suction will be taken from the vapor section of the tank car and discharged into the storage tank. The vapor is fed into the bottom of the storage tank so it will bubble up through the liquid. This helps to cool the vapor and condense it into liquid more quickly than if it were fed to the vapor space on top of the storage tank.

Removing vapor from the tank car will cause the remaining liquid to boil into vapor in order to replace the vapor being removed by the compressor.

The liquid heel boils more readily in summer than winter. The higher temperatures result in higher vapor pressures, which according to the ideal gas law, result in a denser gas. Since the gas is denser in hot conditions, more liquid must be boiled off in the summer than in the winter to replace an equivalent amount of vapor.

The first step in calculating the boil off time is to determine the mass of the liquid heel.

$$\text{Mass of heel} = \text{Vol. of heel } (ft^3) \times \text{Density } (lbm/ft^3)$$
$$= \text{Vol. of heel } (ft^3) \times 62.4\ (lbm/ft^3) \times \text{Specific Gravity}$$

The ideal gas law is used to convert the mass of the liquid to the equivalent volume of vapor it will create at the system vapor pressure:

$$\text{Equivalent vapor volume} = \frac{m\ R\ T}{P}$$

The boil off time is determined by dividing the equivalent vapor volume by the ACFM of the compressor.

$$\text{Boil off time} = \frac{\text{Equivalent vapor volume } (ft^3)}{\text{ACFM}}$$

During the boil-off phase the compression ratio will be quite low. Since the two tanks are at virtually the same pressure (vapor pressure) the differential pressure across the compressor will equal the pressure drop through the vapor piping. The pressures drop through the vapor piping seldom exceeds 5 PSI.

5.2.5 VAPOR RECOVERY FROM LIQUIFIED GAS TANKS

The vapor remaining in the tank after the liquid boil-off operation is completed has substantial economic value which justifies at least partial recovery. Vapor recovery is not really a distinct operation from the boil-off phase since none of the valving needs to be changed from where it was set for the boil-off phase. The separation is made because different methods of calculation are required to determine the length of time required for each phase. The vapor recovery phase begins immediately after the liquid heel is completely vaporized. The pressure in the tank will begin to decrease since no liquid remains to replace the vapor as it is withdrawn by the compressor. The discharge pressure remains near the vapor pressure.

The vapor recovery phase continues until the tank has been evacuated to a final predetermined pressure. The final pressure should be based on the value of gas, the energy to run the compressor and any rental or demurrage costs on the tank car. There are three other limitations on the final evacuation pressure: 1) The final compression ratio must result in a final discharge temperature less than 350°F. 2) The compression ratio must be low enough to produce a reasonable volumetric efficiency, 20% or greater. 3) The final pressure must be above atmospheric pressure. The distance piece tubing/rod packing arrangements used for vacuum service are not suitable for the high suction pressures present through most of the liquid transfer/vapor recovery operation.

For the majority of applications, it is economically viable to reduce the tank pressure to only about 25-30% of vapor pressure (so long as the final pressure is above atmospheric pressure). Reduction of tank pressure to atmospheric pressure (or lower) should only be done where environmental concerns or government regulations absolutely require it.

The time required to evacuate a tank from one pressure to another may be calculated as follows:

$$t = \frac{V}{PD\ E_a} \ln(P_i/P_f)$$

V = Tank Volume (ft^3)
P_i = Initial Suction Pressure (psia)
P_f = Final Suction Pressure (psia)
E_a = Average Volumetric Efficiency

The initial compression ratio is 1 and it increases as the suction pressure falls. The horsepower will increase initially as the compression ratio rises, but as the compression ratio rises, the mass flow drops. This causes the horsepower to peak out and then fall as the compression ratio continues to rise. Therefore, DO NOT DETERMINE THE BHP OF A VAPOR RECOVERY COMPRESSOR BASED ON THE FINAL EVACUATION PRESSURE!

The compression ratio at which the peak horsepower occurs can be approximated by the following relation:

$$CR\ (peak) = n^{n/n-1}$$

n = Ratio of specific heats

The vapor recovery phase is completed most quickly in the winter since the lower vapor pressure results in a lower mass of gas in the tank.

Note that Section 5.2.2 showed that the liquid flow rate is maximum in the summer and lowest in the winter. The vapor recovery operation is completed most quickly in the winter and most slowly in the summer. The net result is that the total time for liquid transfer and vapor recovery stays relatively constant most of the year.

5.2.6 LIQUIFIED GAS TRANSFER SYSTEMS

To simplify the installation of a liquified gas transfer system, Corken supplies its compressors prepackaged with an inlet suction scrubber, four way valve, pressure gauges, baseplate, V-belt drive, beltguard, strainer and motor base as part of its 107 mounting arrangement.

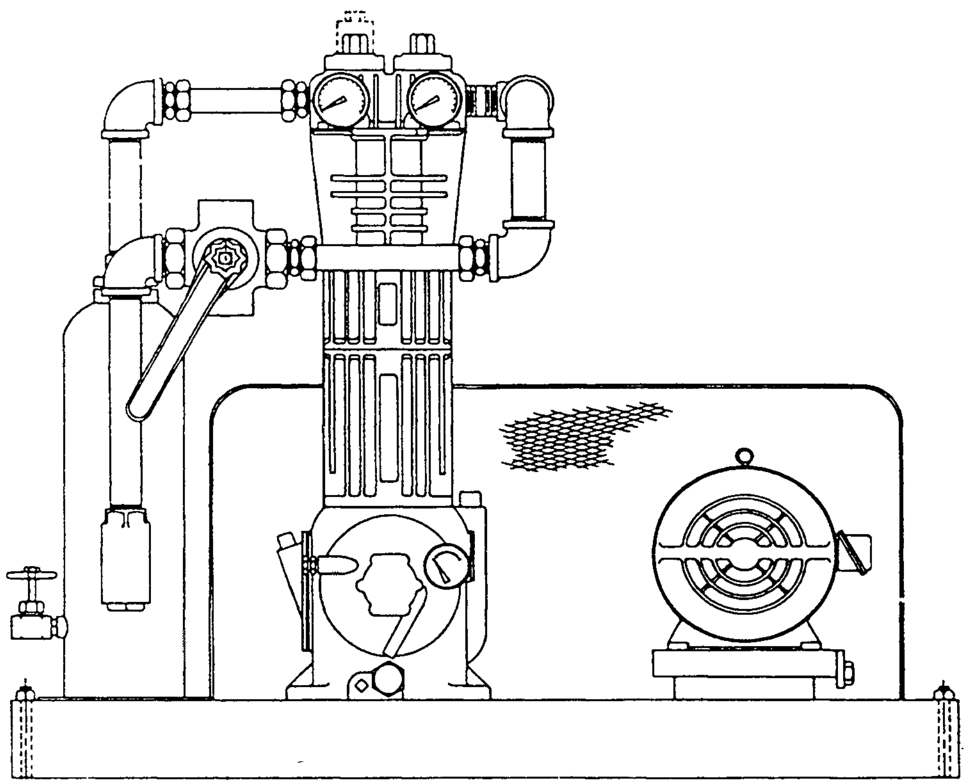
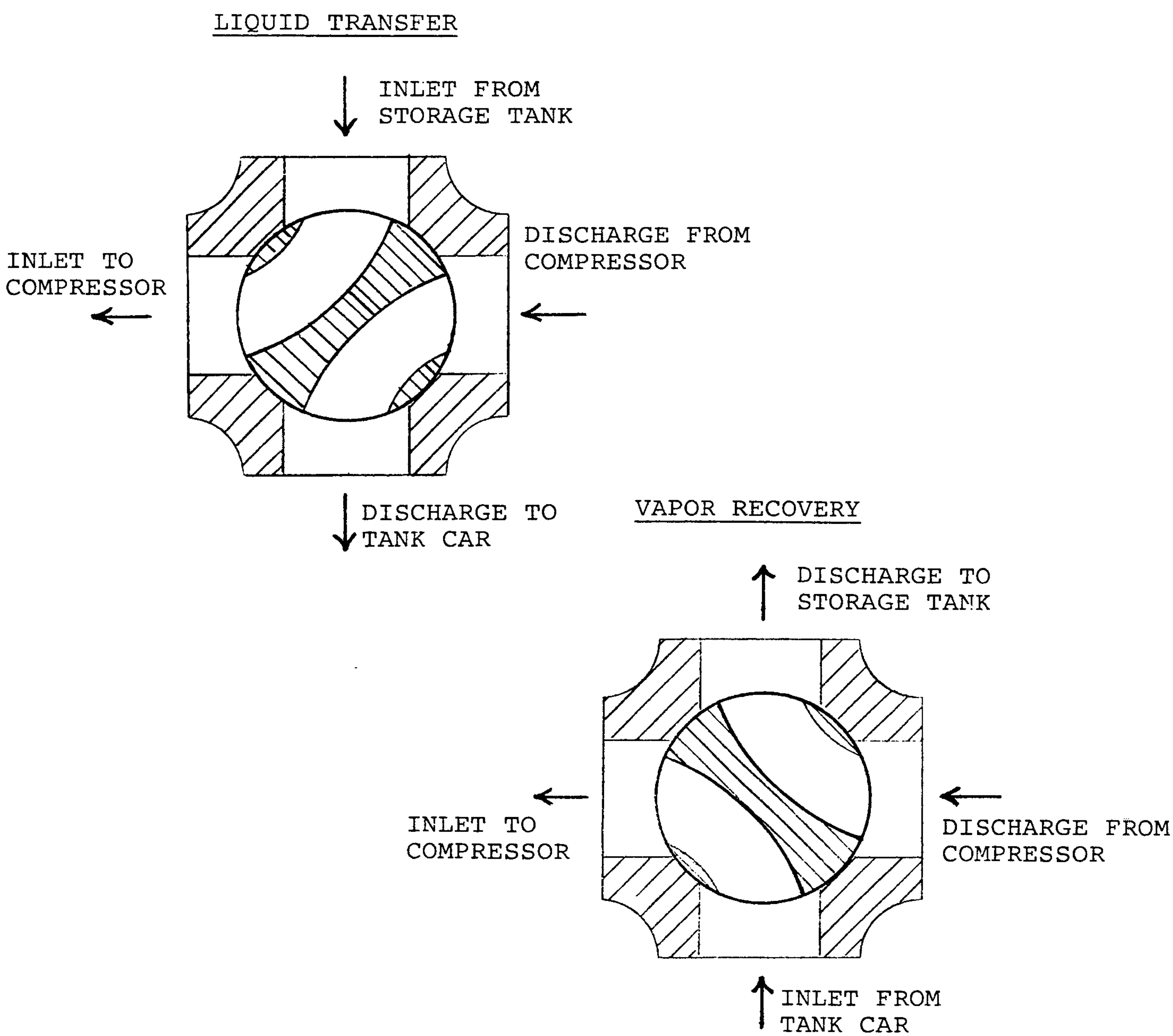


Fig. 5.2.6A 107 COMPRESSOR MOUNTING

Liquid transfer systems are only available for single stage compressors. During most of the liquid transfer/vapor recovery process the compressor operates at very low compression ratios - less than 1.3. Two-stage compressors are not practical for such low compression ratios (see section 3.2).

Inlet suction scrubbers are ALWAYS required on liquid transfer compressors to prevent entrained liquid from entering the machine (see section 3.13). The four-way valve is used to simplify the change over from liquid transfer to vapor recovery. As shown in Fig. 5.2.6B, the four-way valve reverses the compressor's suction and discharge by being rotated 90°. The valve is sealed with grease.

FIG 5.2.6B CROSS SECTIONAL VIEW OF 4-WAY VALVE OPERATION



CHAPTER 6

EXAMPLE PROBLEMS

6.1 SINGLE STAGE GAS BOOSTER

The manufacturing engineer for XYZ Semiconductor Corporation needs to supply a small quantity of high pressure air to the air bearings in his Perkin-Elmer machine, a device used to make etching masks. He has oil-free air available from his existing plant air system but it is not available at high enough pressure. Suggest a solution using a Corken compressor.

Step 1 - Ask the right questions! The key to solving a customer's problem quickly is asking the right questions the first time so all the necessary information required is available to solve the problem.

Here are the questions with the customer's answers:

Q: What's the inlet pressure?	A: 100 PSI
Q: PSIG or PSIA	A: PSIG
Q: What's the required discharge pressure?	A: 200 PSIG
Q: What's the inlet temperature?	A: 100°F maximum
Q: What capacity do you need?	A: 100 CFM
Q: ACFM or SCFM?	A: SCFM

(Note-this is an important question, be adamant about getting an answer. In this case 1 ACFM will equal 7 SCFM. Customers frequently don't understand the difference so be prepared to explain it to them.)

a) From Appendix A find the n value (same as k value) for air:

$$n = 1.4$$

b) Determine absolute conditions: (Sec. 1.2)

$$P_1 = 100 + 14.7 = 114.7 \text{ psia}$$
$$P_2 = 200 + 14.7 = 214.7 \text{ psia}$$
$$T_1 = 100 + 460 = 560°R$$

c) Determine discharge temperature: (Sec. 3.2)

$$CR = P_2/P_1 = 214.7/114.7 = 1.9$$

$$T_2 = T_1(CR)^{(n-1)/n} = 560\ (1.9)^{(1.4-1)/1.4} = 672°R = 212°F$$

Discharge temperature is O.K. for continuous duty operation since $T_2 < 350°F$.

d) Determine ACFM: (Sec. 2.5)

$$ACFM = \frac{(SCFM)(14.7\ psia)(T_1\ ^oR)}{(P_2\ psia)\ (520^oR)}$$

$$= \frac{(100)(14.7)(560)}{(114.7)\ (520)}$$

$$= 13.8\ ACFM$$

e) Determine volumetric efficiency

VE = 86% at Compression Ratio = 1.9, and n = 1.4 (Fig. 2.4A)

f) Determine required piston displacement

$$P.D. = ACFM\ /\ VE = 13.8\ /\ 0.86 = 16\ ft^3/min$$

g) Choose proper size single stage compressor

From Appendix B, the model 290 produces 16.1 ft^3/min. of piston displacement at 805 RPM

h) Determine HP (Sec. 2.6)

$$HP = 0.00528\ (n/n-1)\ (P_1)\ (PD)\ (CR^{(n-1)/n} -1)\ (1.1)$$

$$= 0.00528\ (1.4/0.4)\ (114.7)\ (16.1)\ (1.9^{0.4/1.4} -1)\ (1.1)$$

$$= 7.5\ BHP$$

Use a 10 HP Motor

Note: This equation provides a good approximation in most cases but it is not intended to be an exact BHP prediction. Also, small changes in either suction or discharge pressures can drastically affect the BHP required. Therefore, the application engineer should always be conservative in choosing motor sizes when this method is used. Do not rely on motor service factors when BHP approaches the nameplate rating! Use the next size motor.

i) Determine the model number

From Appendix C the model number required is D290AP9FBA.

6.2 TWO STAGE GAS BOOSTER

ABC Chemical needs to compress 60 lbs/hr of ethane from 5 PSIG to 225 PSIG. The ethane is supplied to the compressor at 100°F. Select the proper Corken compressor for the application. Determine proper RPM and motor si/e.

a) From Appendix A, find the physical properties of ethane.

$n = 1.19$ Molecular Weight = 30

b) Determine the discharge temperature for single stage compression. (Sec. 3.2)

$$CR = P_2 / P_1 = (225 + 14.7) / (5 + 14.7) = 12.2$$

$$T_2 = T_1 \; CR^{(n-1)/n}$$

$$T_2 = (460 + 100)\;(12.2)^{(1.19-1)/1.19} = 835°R = 375°F$$

The discharge temperature is too high for continuous duty operation since T2 > 350°F. Therefore, use two stage compression.

Assume that the gas will be cooled to 130°F by the interstage cooler, and the compression ratio per stage to be the square root of the total compression ratio.

$$T_2 = T_i \; CR^{(n-1)/2n} \qquad (T_i \text{ is interstage Temperature})$$

$$T_2 = (460 + 130)\;(12.2)^{(1.19-1)/1.19} = 720°R = 260°F$$

A two stage compressor will be O.K. for continuous duty operation since T2 < 350°F.

NOTE: Exact interstage pressures are calculated by a complex iterative procedure beyond the scope of this text. However, for vertical, two-stage CorKen compressors, the square root of the total compression ratio times the suction pressure provides a reasonable approximation of interstage pressure in most cases.

c) Calculate ACFM required: (Sec. 2.5)

$$ACFM = \frac{(lb/hr)\;(T_1\;°R)\;(0.1787)}{(P_1\;psia)\;(MW)}$$

$$= \frac{60\;(560)\;(0.1787)}{(19.7)\;(30)} = 10.2\;ACFM$$

d) Calculate volumetric Efficiency (Fig. 2.4B)

VE = 63% for CR = 12.2 and n = 1.19

e) Calculate required piston displacement:

P.D. (required) = ACFM / VE = 10.2 / 0.63 = 16.1 CFM

f) Choose correct size and speed for two-stage model.

From Appendix B, the Model 390 gives 16.6 Ft^3/min. P.D. at 605 RPM

g) Calculate HP (Sec. 2.6)

$$BHP = 0.00528\ (2n/n-1)\ (P_1)\ (PD)\ (CR^{(n-1)/2n} - 1)\ (1.1)$$

$$= 0.00528\ (2.38/0.19)\ (19.7)\ (16.6)\ (12.2^{0.19/1.19} - 1)\ (1.1)$$

$$= 5.25$$

Use 7.5 HP Motor

h) Determine model number.

From Appendix C, choose model D390AM9FBAB

i) Check horsepower and pressure rating to make sure they have not been exceeded. (Fig. 3.1A)

HP Rating = 15 HP
Pressure rating = 515 PSIA

6.3 FILLING A TANK

Determine the time required for an A390 compressor running at 805 RPM to pressurize a 30 ft^3 tank from 80 PSIG to 120 PSIG. The compressor takes suction from atmosphere. Also determine maximum BHP and discharge temperature (T_1 = 100°F).

a) Determine k or n value of the gas.

From Appendix A, n= 1.4

b) Determine Piston Displacment required: (Appendix B)

P.D. = 22.15 ft^3/min for an A390 at 805 RPM

c) Determine initial and final compression ratios.

$$CR_i = (80 + 14.7) / 14.7 = 6.4$$

$$CR_f = (120 + 14.7) / 14.7 = 9.2$$

d) Determine initial and final volumetric efficiencies.
From Fig. 2.4B,

VE_i = 70% VE_f = 68%

e) Determine the average volumetric efficiencies

$$E_a = (70\% + 68\%)/2 = 69\% = .69$$

f) Determine the pressurization time: (Sec. 5.1)

$$T = \frac{V\ (P_f - P_i)}{P.D.\ (E_a)\ (P_1)}$$

$$= \frac{30\ (120 - 80)}{(22.15)\ (0.69)\ (14.7)} = 5.3 \text{ min.}$$

g) Since the suction pressure remains constant, the maximum BHP and temperature occur at P_f = 120 PSIG. (Sec. 2.6 & 3.2)

$$BHP = 0.00528\ (2.8 / 0.4)\ (22.15)\ (14.7)\ (9.2^{0.4/2.8} - 1)\ (1.1)$$

Use a 7-1/2 HP motor (see problem 6.2)

$$T_2 = (460 + 130)\ (9.2)^{0.4/2.8} = 350°F$$ (see problem 6.2)

O.K. for continuous duty operation

h) Check specifications to make sure horsepower and pressure ratings have not been exceeded. (From Fig. 3.1A)

HP Rating = 15 HP
Pressure rating = 515 PSIA

6.4 LIQUIFIED GAS TANK CAR UNLOADING

XYZ Chemical receives a 33,000 gal. tank car of propane once a week and are interested in using a D490 compressor to unloading it. Calculate how long it will take to unload a car that is 90% full. Also calculate how long it would take to boil off a 0.5% liquid heel and evacuate the tank car vapors to 40 PSIA. The pressure drop in the liquid transfer system is 30 PSI and the pressure drop in the vapor piping alone is 2 PSI. The temperature the calculation is to be based upon is 70°F. The compressor will run at 715 RPM.

LIQUID TRANSFER PHASE

a) Determine the physical properties of the gas at 70°F from Appendix G.

$k = n = 1.13$
Specific Gavity of liquid = 0.51
MW = 44
Vapor Pressure = 124 PSI at 70°F.

b) Determine piston displacement at 715 RPM from Appendix B

P.D. = 30.8 CFM

c) Calculate the ACFM at the discharge assuming the gas has cooled to close to ambient temperature:

Compression Ratio (Sec. 5.2.2.):

CR = (Vapor Pressure + Liquid Pressure Drop) / Vapor Pressure

= (124 +30) / 124 = 1.2

From Fig. 2.4A, VE = 90.1%

Discharge CFM = PD (VE) (P_1 / P_2)

= 30.8 (0.901) (124 / 154) = 22.3 CFM

d) Calculate the induced liquid flow rate convert discharge CFM to GPM.

GPM = 22.3 Discharge CFM x 7.48 Gal/Ft^3 = 167 GPM

e) Calculate time to unload liquid from tank car.

% of tank vol. to be moved = liquid vol. - liquid heel vol.
= 90% - 0.5%
= 89.5% = 0.895

$$\text{Unloading time} = \frac{33{,}000 \text{ gal } (0.895)}{167 \text{ gal/min}} = 177 \text{ min}$$

f) Check maximum temperature and horsepower for liquid transfer.

$$HP = 0.0528\ (1.13/0.13)\ (30.8)\ (124)\ (1.2^{0.13/1.13} - 1)\ (1.1)$$

$$= 4.9\ HP$$

$$\text{Max Temp.} = (460 + 70)\ (1.2)^{0.13/1.13} = 541^{o}R = 81^{o}F$$

Temperature and HP are O.K. for D490 compressor

BOIL OFF PHASE

g) Calculate the mass of the liquid heel

(Sec. 5.2.4)
m = (vol. of heel in ft^3)(Specific Gravity)(62.4 lb/ft^3)

Liquid heel vol. = 33,000 Gal (.005) = 165 Gal

= 165 Gal (0.1337 ft^3/Gal) = 22.1 ft^3

m = 22.1 ft^3 (0.51) (62.4) = 703 lbm

h) Calculate equivalent vapor volume (Sec. 1.5)

$$V = \frac{m\ R_u\ T}{P\ MW}$$

$$= \frac{703\ lbm\ (1544\ lbf\text{-}ft/lbmole\text{-}^{o}R)\ (460 + 70\ ^{o}R)}{(124\ lbf/in^2)\ (44.1\ lbm/lbmole)\ (144\ in^2/ft^2)} = 729\ ft^3$$

i) Calculate the time required to boil out the liquid heel.

CR = (Vap. Press. + Press. loss in Vapor system) / Vapor Pressure

= (124 + 2) / 124 = 1.016

From Fig 2.4A, VE = 93% = 0.93

ACFM = PD (VE) = 30.8 ft^3/min (0.93) = 28.7 ft^3/min

From Sec. 5.2.4, T = (Equivalent Vol) / ACFM

= 729 ft^3 / 28.6 ACFM

= 25 min

VAPOR RECOVERY PHASE

j) Calculate average volumetric efficiency for vapor recovery.

$$CR_i = \frac{\text{Vapor Pressure + Pressure Drop in Vapor Piping}}{\text{Vapor Pressure}}$$

$$= \frac{124 + 2}{124} = 1.016$$

$$CR_f = \frac{\text{Vapor Pressure + Pressure Drop in Vapor Piping}}{\text{Final Evacuation Pressure}}$$

$$= \frac{124 + 2}{40} = 3.15$$

From Fig.2.4A,
VE_i = 92% VE_f = 75%

Average VE = E_a

$= (0.92 + 0.75)/2$

$= .84$

k) Calculate the vapor recovery time.   (Sec. 5.2.5)

$$T = \frac{(V\ ft^3)}{E_a\ (PD\ ft^3)}\ \ln\frac{(P_i\ psia)}{(P_f\ psia)}$$

$$= \frac{(33{,}000\ /\ 7.48)}{0.84\ (30.8)}\ \ln(124/40)$$

$= 194$ min

l) Calculate the total time.

Total time = $T_1 + T_2 + T_3$

$= 177 + 25 + 194 = 396$ min.

$= 6.6$ hr

APPENDIX A
PHYSICAL PROPERTIES OF GASES

GAS NAME	TRADE NAME	K VALUE	MOL. WT.
ACETYLENE		1.234	26.038
AIR		1.40	28.960
AMMONIA		1.307	17.031
ARGON		1.669	39.948
BORON TRICHLORIDE		1.145	117.170
BORON TRIFLUORIDE		1.201	67.805
BROMOTRIFLOUROETHYLENE		1.103	160.921
BROMOTRIFLUOROMETHANE	R-13B1	1.134	148.910
1.3-BUTADIENE		1.113	54.092
BUTANE		1.09	58.124
1-BUTENE		1.104	56.108
2-BUTENE		1.115	56.108
CARBON DIOXIDE		1.316	44.011
CARBON MONOXIDE		1.404	28.010
CARBON TETRAFLUORIDE	R-14	1.157	88.005
CARBOYL SULFIDE		1.241	60.070
CHLORINE		1.308	70.906
CHLORODIFLUOROMETHANE	R-22	1.171	86.470
CHLOROTRIFLUOROMETHANE	R-13	1.14	104.459
CYANOGEN		1.166	52.035
CYANOGEN CHLORIDE		1.214	61.417
DEUTERIUM		1.40	4.032
DIBORANE		1.179	27.688
DIBROMODIFLUOROMETHANE	12B2	1.116	209.820
DICHLORODIFLUOROMETHANE	R-12	1.127	120.914
DICHLOROFLUOROMETHANE	R-21	1.15	102.923
1,2 DICHLOROTETRAFLUOROETHANE	R-114	1.08	170.922
1,1-DIFLUORO-1-CHLOROETHANE		1.108	100.496
DIMETHYLAMINE	DMA	1.149	45.085
DIMETHYL ETHER	DME	1.11	46.069
2,2-DIMETHYLPROPANE		1.076	72.151
ETHANE		1.192	30.070
ETHYLACETYLENE		1.117	54.092
ETHYLENE		1.244	28.054
ETHYLENE OXIDE		1.20	44.054
FLOURINE		1.358	37.966
HELIUM		1.63	4.002
HYDROGEN		1.405	2.016
HYDORGEN BROMIDE		1.42	80.912
HYDROGEN CHLORIDE		1.41	34.461
HYDROGEN CYANIDE		1.31	27.026
HYDROGEN IODINE		1.40	127.912
HYDROGEN SULFIDE		1.33	34.076

GAS NAME	TRADE NAME	K VALUE	MOL. WT.
ISOBUTANE		1.097	58.124
ISOBUTYLENE		1.106	56.108
KRYPTON		1.667	83.800
METHANE		1.305	16.043
METHYLACETYLENE		1.156	40.065
METHYL BROMIDE		1.227	94.939
3-METHYL-1-BUTANE		1.077	70.135
METHYL CHLORIDE		1.244	50.488
METHYL FLUORIDE		1.278	34.033
METHYL MERCAPTAN		1.20	48.103
MONOMETHYLAMINE	MMA	1.202	31.058
NEON		1.642	20.183
NITRIC OXIDE		1.40	30.006
NITROGEN		1.407	28.013
NITROUS OXIDE		1.303	44.013
OCTOFLUOROCYCLOBUTANE	R-318	1.055	200.031
OXYGEN		1.414	31.998
PERCHLORYL FLUORIDE		1.12	102.450
PROPANE		1.142	44.097
PROPYLENE		1.154	42.081
SULFUR DIOXIDE		1.29	64.063
SULFUR HEXAFLUORIDE		1.08	146.054
TRIMETHYLAMINE	TMA	1.184	59.112
XENON		0.66	131.300

PISTON DISPLACEMENTS WITH V-BELT & SHEAVE SELECTIONS

SIZE 90, 91

Std. center distance is 14".
Std. flywheel is A13.2 or B13.6, 2 groove.

1750 RPM DRIVERS

RPM	SHEAVE	BELTS	PISTON DISPLACEMENT (CFM) 90/91
400	A3.0	2A54	4.00
425	A3.2	2A54	4.25
450	A3.4	2A54	4.50
480	A3.6	2A54	4.80
505	A3.8	2A55	5.05
515	B4.0	2B56	5.15
540	B4.2	2B56	5.40
565	B4.4	2B56	5.65
590	B4.6	2B56	5.90
620	B4.8	2B56	6.20
645	B5.0	2B58	6.45
670	B5.2	2B58	6.70
695	B5.4	2B58	6.95
720	B5.6	2B58	7.20
800	B6.2	2B60	8.00
825	B6.4	2B60	8.25
850*	B6.6	2B60	8.50

SIZE 190, 290, 291

Std. sheave center distance is 16".
Std. flywheel is A15.2 or B15.6, 3 groove.

1750 RPM DRIVERS

RPM	SHEAVE	BELTS	PISTON DISPLACEMENT (CFM) 190	290/291
345	A3.0	3A60	3.45	6.9
370	A3.2	3A60	3.70	7.4
390	A3.4	3A62	3.90	7.8
415	A3.6	3A62	4.15	8.3
435	A3.8	3A62	4.35	8.7
450	B4.0	3B64	4.50	9.0
470	B4.2	3B64	4.70	9.4
490	B4.4	3B64	4.90	9.8
515	B4.6	3B64	5.15	10.3
535	B4.8	3B64	5.35	10.7
560	B5.0	3B64	5.60	11.2
580	B5.2	3B65	5.80	11.6
605	B5.4	3B65	6.05	12.1
625	B5.6	3B65	6.25	12.5
645	B5.8	3B66	6.45	12.9
670	B6.0	3B66	6.70	13.4
695	B6.2	3B66	6.95	13.9
715	B6.4	3B66	7.15	14.3
735	B6.6	3B66	7.35	14.7
760	B6.8	3B68	7.60	15.2
780	B7.0	3B68	7.80	15.6
805	A7.0	3A68	8.05	16.1
825	B7.4	3B68	8.25	16.5
895*	B8.0	3B69	8.95	17.9

* "A" STYLE UNITS ONLY!

SIZE 390 391 490 491 490-3

Std. sheave center distance is 18".
Std. flywheel is A15.2 or B15.6, 3 groove.

RPM	SHEAVE	BELTS	PISTON DISPLACEMENT (CFM) 390/391	490/491	490-3	
345	A3.0	3A64	9.5	14.9	8.5	
370	A3.2	3A64	10.2	15.9	9.1	
390	A3.4	3A66	10.7	16.8	9.6	
415	A3.6	3A66	11.4	17.9	10.2	
435	A3.8	3A66	12.0	18.7	10.7	
450	B4.0	3B66	12.4	19.4	11.1	
470	B4.2	3B68	13.0	20.2	11.5	
490	B4.4	3B68	13.5	21.1	12.0	
515	B4.6	3B68	14.2	22.2	12.6	1750 RPM
535	B4.8	3B68	14.7	23.0	13.1	DRIVERS
560	B5.0	3B68	15.4	24.1	13.7	
580	B5.2	3B68	16.0	25.0	14.2	
605	B5.4	3B68	16.7	26.0	14.8	
625	B5.6	3B71	17.2	26.9	15.3	
645	B5.8	3B71	17.8	27.8	15.8	
670	B6.0	3B71	18.5	28.8	16.4	
695	B6.2	3B71	19.1	29.9	17.1	
715	B6.4	3B71	19.7	30.8	17.5	
735	B6.6	3B71	20.2	31.6	18.0	
760	B6.8	3B71	20.9	32.7	18.6	
780	B7.0	3B71	21.5	33.6	19.1	
825	B7.4	3B71	22.7	35.5	20.2	
895*	B8.0	3B71	24.6	38.5	21.9	

SIZE 590 690 691 690-4

Std. sheave center distance is 20".
Std. flywheel is B19.1, 4 groove.

RPM	SHEAVE	BELTS	PISTON DISPLACEMENT (CFM) 590	690/691	690-4	
400	B4.4	4B78	26.2	29.2	23.3	
420	B4.6	4B78	27.5	30.7	24.5	
440	B4.8	4B78	28.8	32.1	25.6	
455	B5.0	4B78	29.8	33.2	26.5	
475	B5.2	4B78	31.1	34.7	27.7	
495	B5.4	4B78	32.4	36.2	28.8	
510	B5.6	4B78	33.4	37.5	29.7	
530	B5.8	4B80	34.7	38.7	30.9	1750 RPM
550	B6.0	4B80	36.1	40.2	32.0	DRIVERS
565	B6.2	4B80	37.0	41.3	32.9	
585	B6.4	4B80	38.3	42.7	34.1	
605	B6.6	4B80	39.7	44.2	35.2	
620	B6.8	4B81	40.6	45.3	36.1	
640	B7.0	4B81	41.9	46.7	37.3	
675	B7.4	4B81	44.2	49.3	39.3	
730	B8.0	4B83	47.8	53.3	42.5	
785	B8.6	4B83	51.4	57.3	45.7	
820	TB9.0	4B85	53.7	59.9	47.7	
860*	B9.4	4B85	56.4	62.8	50.1	

* "A" STYLE UNITS ONLY!

APPENDIX B

PISTON DISPLACEMENTS WITH V-BELT & SHEAVE SELECTIONS

SIZE 90 91

Std. center distance is 14" (356mm).
Std. flywheel is A13.2 or B13.6, 2 groove.

1450 RPM DRIVERS

RPM	SHEAVE	BELTS	PISTON DISPLACEMENT (M3/HR) 90/91
330	A3.0	2A54	5.61
350	A3.2	2A54	5.98
375	A3.4	2A54	6.34
395	A3.6	2A54	6.71
415	A3.8	2A55	7.09
425	B4.0	2B56	7.24
450	B4.2	2B56	7.61
470	B4.4	2B56	7.97
490	B4.6	2B56	8.33
510	B4.8	2B56	8.70
535	B5.0	2B58	9.06
555	B5.2	2B58	9.41
575	B5.4	2B58	9.79
595	B5.6	2B58	10.14
620	B5.8	2B58	10.50
640	B6.0	2B60	10.88
660	B6.2	2B60	11.23
680	B6.4	2B60	11.59
705	B6.6	2B60	11.96
725	B6.8	2B61	12.32
745	B7.0	2B61	12.68
790	B7.4	2B62	13.41
855	B8.0	2B62	14.49

SIZE 190 290 291

Std. sheave center distance is 16" (406mm)
Std. flywheel is A15.2 or B15.6, 3 groove.

1450 RPM DRIVERS

RPM	SHEAVE	BELTS	PISTON DISPLACEMENT (M3/HR) 190	290/291
305	A3.2	3A60	5.18	10.4
325	A3.4	3A62	5.51	11.0
345	A3.6	3A62	5.83	11.7
365	A3.8	3A62	6.17	12.3
370	B4.0	3B64	6.32	12.6
390	B4.2	3B64	6.63	13.3
410	B4.4	3B64	6.95	13.9
430	B4.6	3B64	7.27	14.5
445	B4.8	3B64	7.58	15.2
465	B5.0	3B64	7.90	15.8
485	B5.2	3B65	8.21	16.4
500	B5.4	3B65	8.53	17.1
520	B5.6	3B65	8.85	17.7
540	B5.8	3B66	9.16	18.3
560	B6.0	3B66	9.48	19.0
575	B6.2	3B66	9.79	19.6
595	B6.4	3B66	10.11	20.2
615	B6.6	3B66	10.42	20.8
630	B6.8	3B68	10.74	21.5
650	B7.0	3B68	11.06	22.1
670	A7.0	3A68	11.35	22.7
690	B7.4	3B68	11.69	23.4
745	B8.0	3B70	12.64	25.3
800	B8.6	3B71	13.58	27.2
835	B9.0	3B71	14.22	28.4
875	B9.4	3B71	14.85	29.7

SIZE	Std. sheave center distance is 18" (457mm).						
390	Std. flywheel is A15.2 or B15.6, 3 groove.						
391							
490				PISTON DISPLACEMENT (M3/HR)			
491	RPM	SHEAVE	BELTS	390/391	490/491	490-3	
490-3	305	A3.2	3A64	14.3	22.3	12.7	
	325	A3.4	3A66	15.1	23.7	13.5	
	345	A3.6	3A66	16.0	25.1	14.3	
	360	A3.8	3A66	16.9	26.5	15.1	
	370	B4.0	3B66	17.4	27.2	15.5	
	390	B4.2	3B68	18.2	28.5	16.2	
	410	B4.4	3B68	19.1	29.9	17.0	
	430	B4.6	3B68	20.0	31.3	17.8	
	445	B4.8	3B68	20.8	32.6	18.6	
	465	B5.0	3B68	21.7	34.0	19.4	
	485	B5.2	3B68	22.6	35.3	20.1	
	500	B5.4	3B68	23.5	36.7	20.9	1450 RPM
	520	B5.6	3B71	24.3	38.1	21.7	DRIVERS
	540	B5.8	3B71	25.2	39.4	22.4	
	560	B6.0	3B71	26.1	40.8	23.2	
	575	B6.2	3B71	26.9	42.1	24.0	
	595	B6.4	3B71	27.8	43.5	24.8	
	615	B6.6	3B71	28.6	44.8	25.5	
	630	B6.8	3B71	29.5	46.2	26.3	
	650	B7.0	3B71	30.4	47.6	27.1	
	690	B7.4	3B71	32.1	50.3	28.6	
	745	B8.0	3B75	34.8	54.4	31.0	
	800	B8.6	3B75	37.3	58.4	33.3	
	835	B9.0	3B75	39.1	61.2	34.8	
	875	B9.4	3B75	40.8	63.9	36.4	

SIZE	Std. sheave center distance is 20" (508mm).						
590	Std. flywheel is B19.1, 4 groove.						
690				PISTON DISPLACEMT (M3/HR)			
691	RPM	SHEAVE	BELTS	590	690/691	690-4	
690-4	364	B4.8	4B78	40.5	45.2	36.0	
	380	B5.0	4B78	42.3	47.1	37.6	
	395	B5.2	4B78	44.0	49.0	39.1	
	410	B5.4	4B78	45.6	50.9	40.5	
	425	B5.6	4B78	47.3	52.7	42.0	
	440	B5.8	4B80	49.0	54.6	43.5	
	455	B6.0	4B80	50.6	56.4	45.0	
	470	B6.2	4B80	52.4	58.4	46.6	1450 RPM
	485	B6.4	4B80	54.1	60.3	48.1	DRIVERS
	500	B6.6	4B80	55.8	62.1	49.5	
	515	B6.8	4B81	57.4	64.0	51.0	
	530	B7.0	4B81	59.1	65.9	52.5	
	560	B7.4	4B81	62.6	69.7	55.6	
	605	B8.0	4B83	67.6	75.3	60.0	
	655	B8.6	4B83	72.7	81.0	64.6	
	685	TB9.0	4B85	76.0	84.7	67.5	
	715	B9.4	4B85	79.5	88.6	70.6	
	835	B11.0	4B90	92.9	103.6	82.6	

GAS COMPRESSORS - NON LUBRICATED — EE 150C

MODEL NO. AND MOUNTING IDENTIFICATION CODE

JANUARY 1988
SUPERSEDES EE150B

EXAMPLE

MODELS INVOLVED	DESCRIPTION	CODE	FEATURE	EXAMPLE (MODEL NUMBER)
ALL EXCEPT 391 391,590,690,690-4	AIR WATER	NONE W	HEAD COOLING	W
ALL EXCEPT 791,891 ALL 390,391,491,490-3	NO DISTANCE PIECE SINGLE DISTANCE PIECE DOUBLE DISTANCE PIECE	NONE D T	DISTANCE PIECE DESIGN	D
	SINGLE-STAGE, GRAY IRON	290 490 490-3 690 690-4	BASIC MODEL	590
	SINGLE-STAGE, DUCTILE IRON	91 291 491 691 891		
	TWO-STAGE, GRAY IRON	390 590		
	TWO-STAGE, DUCTILE IRON	191 391 791		
ALL EXCEPT 791,891	PACKING ARRANGEMENTS SEE SERVICE MANUAL PAGE E270	A B D E G H	PACKING ARRANGEMENT	A
791,891		J,K		
91	SPLASH	J	CRANKCASE LUBRICATION	M
ALL EXCEPT 91	PRESSURE-GEAR OIL PUMP	M		
ALL ALL 'L EXCEPT 91 .L EXCEPT 91 ALL ALL	LIQUID RELIEF STANDARD LOADLESS START COMBINATION CONTROL CONSTANT SPEED UNLOADING UNLOADERS ONLY	3 4 7 78 8 9	VALVE/UNLOADER TYPE	4
ALL	TEFLON*	F	PISTON RING MATERIAL	F
ALL	ALUMINUM (STANDARD) COPPER IRON-LEAD	B C D	TRIM MATERIAL	B
ALL	BUNA-N (STANDARD) TEFLON* VITON* NEOPRENE*	A B C D	O-RING MATERIAL	A
TWO-STAGE	AIR COOLED-IRON AIR COOLED-COPPER (STANDARD) WATER COOLED-BRASS (STANDARD ON WD391,WD590) WATER COOLED-STAINLESS STEEL HEAVY DUTY AIR COOLED-COPPER NO INTERCOOLER-FLANGES ONLY	A B C D E F	INTERCOOLER (TWO-STAGE ONLY)	C
ALL**	DENOTES SPECIAL MATERIALS USED OR SPECIAL CONSTRUCTION.			Z

*TEFLON, VITON AND NEOPRENE ARE REGISTERED TRADEMARKS OF DUPONT.
**CONTACT THE FACTORY WITH COMPLETE MODEL NO. AND SERIAL NO. FOR DETAILS OF SPECIAL CONSTRUCTION.

—

MODELS INVOLVED	DESCRIPTION	CODE	FEATURE	EXAMPLE (MOUNTING)
ALL	BARE COMPRESSOR WITH FLYWHEEL	NONE	MOUNTING	109
SINGLE-STAGE	PRESSURE GAUGES MOUNTED ON HEAD***	100		
ALL	EXTENDED CRANKSHAFT	101		
SINGLE-STAGE	EXTENDED CRANKSHAFT WITH PRESSURE GAUGES MOUNTED***	102		
ALL	V-BELT DRIVE (PRESSURE GAUGES INCLUDED WITH SINGLE STAGE ONLY)***	103		
SINGLE-STAGE	V-BELT DRIVE, PRESSURE GAUGES MECHANICAL LIQUID TRAP, 4-WAY VALVE, STRAINER	107		
SINGLE-STAGE	SAME AS 107 WITH ELECTRIC LIQUID TRAP	107A		
ALL	SAME AS 107A WITH LARGER ASME LIQUID TRAP	107B		
ALL	V-BELT DRIVE, MECHANICAL LIQUID TRAP (PRESSURE GAUGES INCLUDED WITH SINGLE STAGE ONLY)***	109		
.L	SAME AS 109 WITH ELECTRIC LIQUID TRAP	109A		
ALL	SAME AS 109A WITH LARGER ASME LIQUID TRAP	109B		

***PRESSURE GAUGES CANNOT BE MOUNTED ON ANY TWO STAGE UNIT OR ON MODELS W690, WD690 OR WD690-4.

MATERIAL SPECIFICATIONS

JANUARY 1988
SUPERSEDES EE151B

PART	STANDARD SIZE	STANDARD MATERIAL	OPTIONAL SIZE	OPTIONAL MATERIAL
HEAD, CYLINDER	290,390,490,490-3, 590,690,690-4	GRAY IRON ASTM A48,CLASS 30		NONE
	91,191,291,391,491, 691,791,891	DUCTILE IRON ASTM A536		
DISTANCE PIECE, CROSSHEAD GUIDE CRANKCASE, FLYWHEEL BEARING CARRIER	ALL	GRAY IRON ASTM A48,CLASS 30		NONE
FLANGE	590,690,691,690-4	DUCTILE IRON ASTM A536	590,690,691,690-4	STEEL WELDING
	791,891	ASTM A36 CARBON STEEL		
VALVE SEAT AND BUMPER	91,191,290,291,D891	17-4 PH STAINLESS STEEL		NONE
	390,391,490, 491,490-3	DUCTILE IRON ASTM A536		
	590,690,691,690-4	GRAY IRON ASTM A48,CLASS 30		
	D791(1ST STAGE)	17-4 PH STAINLESS STEEL		
	D791(2ND STAGE)	DUCTILE IRON ASTM A536		
VALVE PLATE	91,191,290,291,D891	410 STAINLESS STEEL		NONE
	390,391,490, 491,490-3	17-7 PH STAINLESS STEEL		
	590,690,691,690-4	STEEL, ROCKWELL 50C		
	D791(1ST STAGE)	410 STAINLESS STEEL		
	D791(2ND STAGE)	17-7 PH STAINLESS STEEL		
VALVE SPRING	91,191,290,291,590 690,691,690-4,D891	17-7 PH STAINLESS STEEL		NONE
	390,391,490,491, 490-3	INCONEL		
	D791(1ST STAGE)	17-7 PH STAINLESS STEEL		
	D791(2ND STAGE)	INCONEL		
VALVE GASKETS	ALL	SOFT ALUMINUM	ALL	COPPER, IRON-LEAD
PISTON	91,191,290,291,390, 391,490,491,490-3, 590,690,691,690-4	GRAY IRON ASTM A48,CLASS 30		NONE
	D891,D791(1ST STAGE)	DUCTILE IRON ASTM A536		
	D791(2ND STAGE)	D4512 DURA-BAR		
PISTON ROD	ALL	C1050 STEEL HARD CHROMIUM PLATED ROCKWELL 60C	D91,D191,D290,D291, T390,WT391,T491, T490-3	CHROME OXIDE COATING
CROSSHEAD	ALL	GRAY IRON ASTM A48,CLASS 30		NONE
PISTON RINGS	ALL	TEFLON*, GLASS AND MOLY FILLED		SPECIAL ORDER MATERIALS AVAILABLE
PISTON RING EXPANDERS	ALL	302 STAINLESS STEEL		NONE
HEAD GASKET	390,490,490-3,590, 690,690-4	CELLULOSE FIBER	390,490,490-3,590, 690,690-4	GRAFOIL
	91,191,290,291,391, 491,691,791,891	O-RING (BUNA-N)	91,191,290,291,391, 491,691,791,891	TEFLON,VITON,NEOPRENE*
ADAPTER PLATE, PACKING CARTRIDGE, CONNECTING ROD	ALL	DUCTILE IRON ASTM A536		NONE
PACKING RINGS	ALL	TEFLON*, GLASS AND MOLY FILLED		NONE
CRANKSHAFT	ALL	DUCTILE IRON ASTM A536		NONE
CONNECTING ROD BEARING	ALL	BIMETAL D-2 BABBIT		NONE
WRIST PIN	ALL	C1018 STEEL, ROCKWELL 62C		NONE
WRIST PIN BUSHING	ALL	BRONZE SAE 660		NONE
MAIN BEARING	ALL	TAPERED ROLLER		NONE
INSPECTION PLATE	ALL	ALUMINUM		NONE
O-RINGS	ALL	BUNA-N	ALL	TEFLON,VITON,NEOPRENE*
RETAINER RINGS	ALL	STEEL		NONE
MISCELLANEOUS GASKETS	ALL	COROPRENE		NONE

*TEFLON, VITON AND NEOPRENE ARE REGISTERED TRADEMARKS OF DUPONT.

APPENDIX E

MODEL NUMBER AND MOUNTING INDENTIFICATION CODE

OCTOBER 1984
SUPERSEDES VJ150

FEATURE	DESCRIPTION	MODEL NUMBER HG60
NUMBER OF STAGES	BARE SINGLE TWO	0 1 2
FIRST STAGE CYLINDER	8″ 6″ 4″ 3¼″ NONE	A B D E X
SECOND STAGE CYLINDER	8″ 6″ 4″ 3¼″ NONE	A B D E X
PACKING ARRANGEMENT		0
CRANKCASE DESIGN	WITH EXT. LUBRICATOR WITHOUT	L M
VALVE UNLOADER TYPE	PLAIN SUCTION VALVE UNLOADER	4 9
PISTON RING MATERIAL	TEFLON	F
TRIM	ALUM COPPER IRON	E F G
O-RING MATERIAL	BUNA-N TEFLON VITON NEOPRENE	A B C D
INTERCOOLER	NONE	F
VARIABLE/ CLEARANCE POCKET HEADS	NONE 1st STAGE 2nd STAGE BOTH	0 1 2 3

MATERIAL SPECIFICATIONS

Part	Size	Standard	Optional
Crankcase	All	Gray Iron ASTM A48, Class 30	
Adapters	All	Ductile Iron ASTM A536, Grade 65-45-12	
Cylinders	All	Ductile Iron ASTM A536, Grade 65-45-12	
Cylinder Heads (Adjustable & Std)	All	Ductile Iron ASTM A536, Grade 65-45-12	
Valve Caps	All	Ductile Iron ASTM A536, Grade 65-45-12	
Crankcase Bearing Carrier	All	Gray Iron ASTM A48, Class 30	
Flanges	All	Steel Slip-on Welded	
Valve Guard	All	Stainless Steel	
Valve Seat	4" 3¼", 6", 8"	Steel 4140 Stainless Steel	
Valve Plates And Springs	All	Stainless Steel	
Valve Gaskets	3¼", 4" 6", 8"	Steel Soft Aluminum	 Copper, Iron-Lead
Pistons	3¼", 4" 6", 8"	Steel Aluminum A356-T6	 Ductile Iron A536 Grade 65-45-12 (Single Stage Only)
Piston Rod	All	C-1050 Steel Hard Chrome Plated R_C60	
Crosshead	All	Gray Iron ASTM A48, Class 30	
Connecting Rod & Packing Cartridge	All	Ductile Iron ASTM A536, Grade 65-45-12	
Piston Rings	All	Teflon*, Glass & Moly Filled	
Rider Rings	All	Teflon*, Glass & Moly Filled	
Packing: Segmented Sealing Rings	4" 3¼", 6", 8"	Teflon*, Carbon Filled Teflon*, Glass & Moly Filled	
Segmented Wiper Rings	4" 3¼", 6", 8"	Brass Teflon*, Carbon Filled	
Connecting Rod Bearing	All	Bimetal D-2 Babbit	
Crankshaft	All	Ductile Iron ASTM A536, Grade 80-55-06	
Wrist Pin	All	C-1018 Steel R_C62	
Wrist Pin Bushing	All	Bronze SAE 660	
Main Bearings	All	Tapered Roller	
Inspection Plate	All	Carbon Steel	
O-Rings	All	Buna N	Teflon*, Viton*, Neoprene
Retainer Rings	All	Carbon Steel	
Lubricator Tubing	Lube Models	Steel	
Lubricator	Lube Models	Force Feed, w/ Divider Block No Flow Shutdown	

*Registered Trademark of DuPont

LIQUIFIED GAS DATA

GAS	'n'	MW	Specific Gravity Liquid Water=1	Liquid Visc. cp @ °F	Mat'l. to use ICA BNVT
Anhydrous Ammonia NH_3	1.31	17.0	0.62	.2 @86	AAA AAAA
Butadiene	1.12	54.1	0.63	.15@60	AAA ABAA
1-Butene C_4H_8	1.10	56.1	0.60		
N-Butane C_4H_{10}	1.09	58.1	0.58	.2 @60	AAA ABAA
Carbon Dioxide CO_2	1.28	44.0	0.82		AAA ABXA
Chlorine Cl_2 *	1.36	70.8	1.40	.4 @32	AXX XXXA
Dimethylamine DMA	1.15	45.1	0.65	.19@77	AXX XBBA
Dimethylether	1.11	46.1	0.63	.3@100	AXX XXXA
Ethane C_2H_6	1.19	30.1	0.37		AAA AAAA
Ethyl Chloride C_2H_5Cl *	1.19	64.5	0.92	.8 @60	AAA AAXA
Ethylene/Ethene C_2H_4	1.22	28.1	0.57		AXX XXXA
Ethylene Oxide C_2H_4O	1.20	44.1	0.87	.3 @32	BXX XXXA
Hydrogen Chloride HCl *	1.40	36.5	1.05		AXX XXXA
Hydrogen Sulfide H_2S	1.32	34.1	0.99		AXB XXXA
Isobutane	1.09	58.1	0.56	.2 @60	AAA ABXA
Isobutylene	1.10	56.1	0.60		AXX XXAA
Isopentane	1.07	72.2	0.72	.2 @68	AXX AXXA
Methyl Acetylene Propodiene MAPP			0.56	.1 @77	AXA XBAA
Methyl Chloride CH_3Cl *	1.28	50.5	0.94	.2 @86	AAX XXAA
Monoethylamine MEA		45.1	0.68		
Monomethylamine MMA	1.20	31.1	0.66		
N-Pentane C_5H_{12}	1.07	75.2	0.63	.2 @68	AXX AXAA
Propane C_3H_8	1.13	44.1	0.51	.1 @60	AAA AAAA
Propylene/Propene C_3H_6	1.15	42.1	0.52		AXX XXAA
Refrigeration Gases:					
R11 (MF)	1.11	137.4	1.50	.4 @86	AAA AXAA
R12	1.14	120.9	1.35	.2 @86	AAA AAXA
R13	1.17	104.5	0.94	.4 @95	AAA AAAA
R21	1.18	102.9	1.42	.4 @77	
R22	1.16	86.5	1.23	.2 @86	AAA XAXA
R112 (BF)		204.0	1.59	1.2 @77	AAA A AA
R113 (TF)	1.09	187.4	1.59	.6 @86	AAA AXAA
R114	1.08	170.9	1.49	.4 @86	AAA AAAA
R115	1.08	154.5	1.25	.3 @77	AAA BAAA
R1301 / R13B1 HALON	1.18	149.0	1.42	.2 @77	AAA AAXA
R502	1.13	111.6	1.24	.2 @80	AAA AAXA
Sulfur Dioxide SO_2 **	1.29	64.1	1.39	.3 @86	ABB BBBA
Sulfur Hexaflouride SF_6	1.07	146.1	1.37	.35@70	AAA AAXA
Trimethylamine TMA	1.18	59.1	0.62	.2 @77	
Vinyl Chloride VCM CH_2:CHCl	1.18	62.5	0.91	.2	AXX XBAA

* Chrome Oxide Piston Rods should be considered.

** Ethylene Propylene O-rings should be used.

I - Iron
C - Copper
A - Alum

B - Buna N
N - Neoprene
V - Viton
T - Teflon

A - Good
B - OK
X - Do not use

APPROXIMATE VAPOR PRESSURE (PSIA)

TEMPERATURE	°C	-29	-23	-18	-12	-7	-1	4	10	16	21	27	32	38	43
	°F	-20	-10	0	10	20	30	40	50	60	70	80	90	100	110
Anhydrous Ammonia	NH_3	18	24	30	39	48	60	73	89	108	129	153	181	212	247
Butadiene		5	7	9	11	13	17	20	25	30	36	43	51	59	69
1-Butene	C_4H_8					15	18	22	27	32	38	45	53	62	72
N-Butane	C_4H_{10}						15	17	21	26	31	37	44	52	61
Carbon Dioxide	CO_2	215	257	306	360	422	491								
Chlorine	Cl_2	18	23	29	35	43	51	61	73	85	100	117	135	157	178
Dimethylamine	DMA						10	13	17	21	26	31	38	45	52
Dimethylether		12	16	20	25	30	37	46	55	65	77	90	105	121	140
Ethane	C_2H_6	160	188	220	255	294	337	385	438	494					
Ethyl Chloride	C_2H_5Cl						8	11	13	17	21	25	30	36	42
Ethylene/Ethene	C_2H_4	290	335	385	445	510									
Ethylene Oxide	C_2H_4O								15	18	22	27	32	39	46
Hydrogen Chloride	HCl	155	186	221	260	304	358	407	475						
Hydrogen Sulfide	H_2S	54	66	80	96	116	138	162	172	223	258	278	340	385	438
Isobutane				12	15	18	22	27	32	38	45	53	62	73	84
Isobutylene				9	12	15	18	22	27	33	39	46	54	64	74
Isopentane												15		20	
Methyl Acetylene Propodiene	MAPP										109				
Methyl Chloride	CH_3Cl	12	15	19	24	29	36	43	52	62	73	86	100	117	135
Monoethylamine	MEA							9	11	14	17	21	26	33	39
Monomethylamine	MMA	5	6	9	11	15	19	24	29	36	44	53	64	76	90
N-Pentane	C_5H_{12}														
Propane	C_3H_8	25	31	38	46	55	66	78	92	107	124	144	165	189	215
Propylene/Propene	C_3H_6	32	39	48	58	73	82	97	113	131	152	175	200	228	258
Refrigeration Gases:															
R11	(MF)							7	9	11	13	16	20	24	28
R12		15	19	24	29	36	43	52	61	72	85	99	114	132	151
R13		126	149	177	205	240	278	320	366	417	473				
R21						8	10	12	15	19	23	28	34	40	47
R22		25	31	39	48	58	70	84	99	117	137	160	185	213	243
R112	(BF)	BOILS AT 199 °F (93 °C)													
R113	(TF)											7	9	11	13
R114					8	10	12	15	19	23	28	33	39	46	54
R115		22	28	34	42	51	61	73	86	101	118	137	155	181	210
R1301 / R13B1	HALON	49	59	71	84	100	117	137	160	185	211	242	275	312	355
R502		30	36	46	56	67	80	95	112	130	152	175	200	230	260
Sulfur Dioxide	SO_2	6	8	10	13	17	21	27	33	40	49	59	71	84	99
Sulfur Hexaflouride	SF_6	79	97	110	130	150	180	208	245	288	325	370	410	465	505
Trimethylamine	TMA						12	16	19	23	28	33	39	46	54
Vinyl Chloride CH_2:CHCl	VCM				16	19	23	28	34	40	48	56	65	76	88

CHEMICAL RESISTANCE CHART

FLUID	IRON	BRONZE	ALUM.	BUNA	NEOP.	VIT.	TEF.
Acetic Acid, Diluted	-	-	A	-	B	-	A
Acetone	A	A	A	-	B	-	A
Acetylene (Do not handle w/ Corken equip.)	A	-	A	-	-	-	A
Acids, General							
Air	A	A	A	A	A	A	A
Alpha-Methyl-Styrene	A	A	A	B	B	A	A
Ammonia, Anhydrous	A	-	A	B	A	-	A
Ammonia, Aqueous	A	-	A	A	A	-	A
Ammonium Carbonate	A	-	A	-	A	B	A
Ammonium Chloride	B	-	-	A	A	A	A
Ammonium Nitrate	B	A	A	A	A	A	A
Ammonium Phosphate, Dibasic	-	-	-	A	-	A	A
Ammonium Sulfide				A	A	-	A
Ammonium Sulfate	A	-	B	A	A	-	A
Amyl Acetate	A	A	A	-	-	-	A
Amyl Alcohol	A			B		B	A
Amyl Chloride	A	-	-		-		A
Aniline	A	A	A	-	-	B	A
Benzene (Benzol)	A	A	A	-	-	B	A
Benzyl Alcohol				-	B	A	A
Butadiene	A	A	A	A	B	A	A
Butane	A	A	A	A	B	A	A
Butyl Acetate	A	A	A	-	-	-	A
Butyl Alcohol	A	A	A	A	-	A	A
Butylene	A	A	A	B	-	A	
Butylene Glycol							
Carbolic Acid (Phenol)	A	-	A	-	-	A	A
Carbon Dioxide	A	A	A	A	B	-	A
Carbon Disulfide	B	B	B	-	-	A	A
Carbon Monoxide				A	B	A	
Carbon Tetrachloride	A	B	B	A	-	A	A
Carbonic Acid	A	-	A		-		A
Chlorinated Water	-	-	-		-		A
Chlorine Gas (Dry)	A		-	-	B	A	A
Chlorine Liquid (Dry)	A	-	-	-	-	A	A
Chloroethane (Ethyl Chloride)	A	-	-	B	-	A	
Chloroform	B	A	-	-	-	A	
Chlorothene	A	A	A	-	-	A	A
Creosote	A	A		A		A	A
Cyclohexane	A	A	A	B	B	A	A
Cyclohexanol				B	B	A	
Cyclohexanone				-	-	-	
Diethyl Ether	B	A	A	-	-	-	
Dibutyl Ether	B	A	A	-	-	B	

A: Satisfactory   Blank: Unsufficient Data
B: Fair   Dash: Unsatisfactory

FLUID	IRON	BRONZE	ALUM.	BUNA	NEOP.	VIT.	TEF.
Dichlorobenzene				-	-	A	
Dichloroethane (Ethylene) (Dichloride)	A	A		-	-	B	A
Dichloromethane (Methylene) (Chloride)	A			-	-	B	
Dichloropentane					-		
Diesel Fuel Oils	A	A	A	A	B	A	A
Diethanolmine	A				A		A
Diethylamine	A			A	-	-	
Diethylene Glycol	A			A	A	A	A
Diethyl Ether	A			-	-	-	A
Dimethylamine (DMA)	A	-	-	-	B	B	A
Dimethyl Ether (DME) Methyl Ether	A	A	A	A	-	A	A
Ethane	A	A	A	A	A	A	A
Ethanolamine	A		A	B	B	-	
Ether	A	A	A	B		-	A
Ethyl Acetate	A	A	A	-	-	-	A
Ethyl Alcohol	B	A	B	A	A	A	A
Ethyl Benzene	A			-	-	A	
Ethyl Chloride	A	A	A	A	A	-	A
Ethyl Ether	A	A	A	B	-	-	A
Ethyl Mercaptan	A			-	-	B	
Ethylene (Ethene) (Ir.&Tef.)	A	-	A	A	-	-	A
Ethylene Chloride			-	-	-	B	
Ethylene Dichloride	A	A		-	-	B	A
Ethylene Glycol	A	A	A	A	A	A	A
Ethylene Oxide	B	-	-	-	-	-	A
Fluorine			A			B	
Formaldehyde		A	A	A	A	-	A
Formic Acid	-	-	-	-	A	-	A
Fuel Oils	A	B	A	A	B	A	A
Furfural	B	A	A	-	B	-	A
Gasoline (Auto)	A	A	A	B?	B	A	A
Glycerine	B	B	A	A	A	A	A
Glycol (see specific type)							
Helium	A	A	A	A	A	A	A
Heptane	A	A	A	B	B	A	A
Hexane	A	A	A	B	B	A	A
Hydrogen	A	A	A	A	A	-	A
Hydrogen Chloride (Ir.&Tef)	A						A
Hydrogen Sulfide (Dry)	A	-	A	-	-	-	A
Isobutane	A	A	A	A	B	-	A
Isobutyl Acetate	A	A	A		-		A
Isobutyl Alcohol	A			A		A	A
Isobutylene	A	-	-		-	A	A
Iso-Octane					A		
Isopentane	A	-	-	A	-	-	A
Isopropyl Alcohol	A	A	A	B	-	A	A

FLUID	IRON	BRONZE	ALUM.	BUNA	NEOP.	VIT.	TEF.
Isopropyl Ether				B	-	-	
Isotane				A	-	A	
Jet Fuel: JP 3	A	A	-	A	-	-	A
JP 4	A	A	-	A	-	A	A
JP 5	A	A	-	A	-	A	A
Kerosene	A	A	A	A	B	A	A
Lacquer	A	A	A	-	-	-	A
Linoleic Acid	-	A	A	B	-	A	A
Magnesium Chloride	-	A	-	A	A	A	A
Magnesium Sulfate	A	A	B	A	A	A	A
Mapp Gas (use Viton)	A	-	A	-	A	A	A
Methane	A	A	A	A	A	A	A
Methyl Alcohol (Methanol)(MEOH)	A	A	A	A	A	-	A
Methylamine Mono- (MMA)							
Di- (DMA)	A	-	-	-	B	B	A
Tri- (TMA)							
Methyl Butyl Ketone				-	-	-	
Methyl Chloride (dry)	A	A	-	-	-	A	A
Methylene Chloride	A	A	A	-	-	B	A
Methyl Ethyl Ketone	A	A	A	B	B	-	A
Methyl Isobutyl Ketone	A			-	-	-	A
Mineral Oil	A	A	A	A	B	A	A
Monoethanolamine (MEA)	A			-	-	-	A
Naphtha	A	A	A	B?	-	A	A
Naphthalene	A	A	A	-	-	A	A
Natural Gas	A	A	A	A	A	-	A
Nitrogen	A	A	A	A	A	A	A
Nitroethane	A	A	-	-	-	-	A
Nitromethane	A	A		-		-	A
Nitrous Oxide	-	-	-	-	A	A	A
N-Pentane	A			A	A	A	
n-Octane							
Oleic Acid	-	A	A		-		
Oxygen (CONSULT FACTORY)							
Pentane	A	A	-	A	A	A	A
Perchlorethylene	A	A	A	B	-	A	A
Phenol (Carbolic Acid)	A	A	A	-	-	A	A
Phenyl Chloride	A			-		A	A
Phosgene							
Propane	A	A	A	A	A	A	A
Propene (Propylene)	A			-	-	A	A
Propylene Dichloride	A	A		-	-	A	A
Propylene Glycol	A			A	-	A	A
Propylene Oxide	A			-	-	-	A
Pyride	-	-	-	-	-	-	A
Refrigerant Gas: R11	A	A	A	A	-	A	A
R12	A	A	A	A	A	-	A

FLUID	IRON	BRONZE	ALUM.	BUNA	NEOP.	VIT.	TEF.
Refrigerant Gas R13	A	A	A	A	A	A	A
R1301 or R13B1	A	A	A	A	A	–	A
R14	A	A	A	A	A	A	A
R21	A	A	A	–	B	–	A
R22	A	A	A	–	A	–	A
R31	A	A	A	–	A	–	
R32	A	A	A	A	A	–	
R112	A	A	A	B	B	A	
R113	A	A	A	A	–	A	A
R114	A	A	A	A	A	A	A
R114B2	–	–	–	A	A	A	A
R115	A	A	A	B	A	A	A
K142B				A	A	B	
K152A				A	A	–	
C318	A	A	A	A	A	A	A
R502	A	A	A	A	A	–	A
Sodium Hydroxide (Cold)	A	A	–	A	A	B	A
Sodium Silicate	A			A	A	A	A
Sodium Sulfate	A	A	A	A	A	A	A
Sodium Sulfide	B	–	–	A	A	A	A
Stoddard Solvent	A	A	A	A	–	A	A
Styrene	A	A	A	B	B	A	A
Sulfur Dioxide (Dry) (Use Ir. & E/P)	A	B	A	A	B	A	A
Sulfur Hexaflouride	A	A	A	A	A	–	A
Tetrachlorethene (Perchlorethylene)				–	–	A	
Tetrachloroethylene	A			–	–	A	A
Tetraethyl Lead	A			A	–	A	A
Toluene (Toluol)	A	A	A	–	–	A	A
1,1,2 Trichloroethane	A			–	–	A	
Trichloroethylene (Dry)	A	A	A	–	–	A	A
Trichloropropane	A			A	A	A	
Triethylene Glycol	A			A		A	A
Turpentine	A	B	A	–	B	A	A
Vinyl Acetate	A	A			A		
Vinyl Chloride (Ir. & Tef)	A	–	–	–	B	A	A
Vinyl Fluoride	A	–	–	B	B	–	A
Water - Fresh	A	A	A	A	A	A	A
Water - Salt	–	B	–	A	A	–	A
Xylene	A	A	A	–	–	B	A

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