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Second Edition

TRAINING FOR PROFESSIONAL PERFORMANCE
IN THE PETROLEUM INDUSTRY

Piston Type Compressors

by

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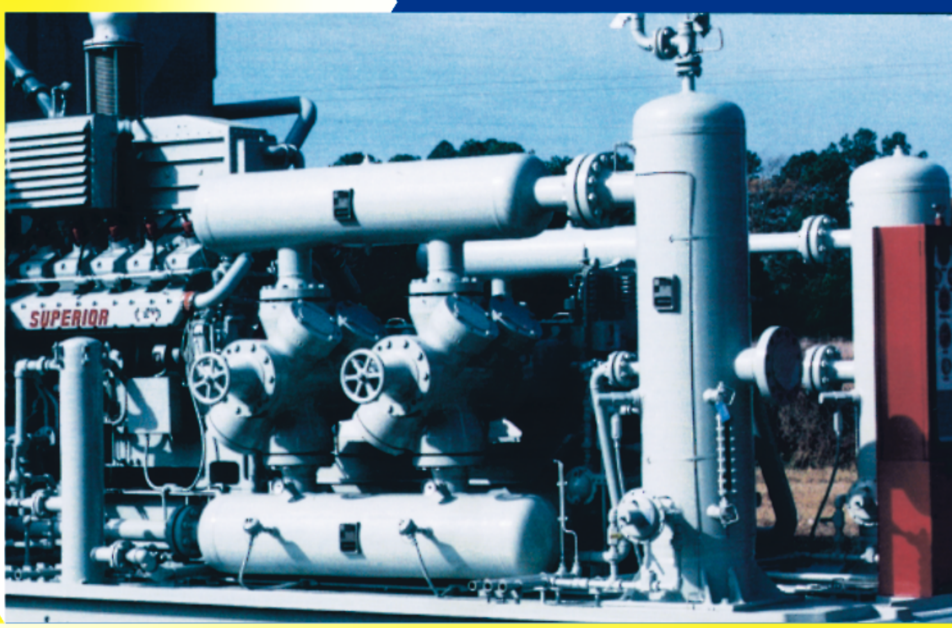
Petroleum Learning Programs



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TRAINING FOR PROFESSIONAL PERFORMANCE

This manual is one of a series for your use in learning more about equipment that you work with in the petroleum industry. Its purpose is to assist in developing your knowledge and skills to the point that you can perform your work in a more **professional** manner.

The manual was prepared so that you can learn its contents on your own time, without the assistance of an instructor or classroom discussion. Educators refer to learning by self-study as **Programmed Learning.** It is a method widely used in all industries as a means of training employees to do their job properly and teach them how to perform higher rated jobs.

You can demonstrate your desire to be a professional by taking a positive attitude toward learning the contents of this manual and others that are applicable to your job.

The author of this manual has years of experience in operating petroleum equipment. He also has the technical knowledge of **how and why** petroleum equipment functions. The text was written for use by personnel with little or no previous experience with petroleum equipment. Consequently, some of the material may be familiar to you if you have experience with oilfield equipment. From such experience, you have observed the effect of making operating changes. The manual will help explain why the changes occurred that you observed. It will also teach you **how and why** equipment functions.

In order for you to learn the contents of the manual, you must dig out the pertinent facts and relate them to the subject. Simply reading the material and answering the questions is not enough. The more effort you make to learn the material, the more you will learn from the manual.

Teaching yourself requires self-discipline and hard work. In order to prepare yourself for the sacrifice you will have to make, you should set goals for yourself. Your ultimate goal is to perform your work in a more professional manner. Training is one step in reaching that goal. Application of what you learn is another. Seeking answers to questions is a third.

Once you have established your final goal, you must determine the means for reaching that goal. You may decide, for example, that you must complete a series of 10 to 15 manuals to get the basic knowledge and skills you need. After you decide which training material is required, you should set a time table for completing each section of the material.

Achieving your final goal may take more than a year, and will require hours of hard work on your part. You will know you have achieved your goal when you understand how and why to operate oilfield equipment in order to obtain the maximum product at the lowest cost. Your sacrifice will have been worth-while from the satisfaction of knowing that you can perform your job in a methodical professional manner, instead of a trial-and-error approach.

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INSTRUCTIONS FOR USING THIS MANUAL

This your manual. You should write your name on the cover. Upon completion you will find it helpful to keep it in an accessible place for future reference.

Problems are included throughout the text. The solutions to the problems are given at the end of the book.

The manual is used in training programs all over the world. In some countries, English units of measurement such as feet, gallons, pounds, etc, are used. In other countries, Systems Internationale (SI) or Metric units, such as meters, liters, kilograms, etc., are used. In order for the manual to be of maximum use, both SI and English units are shown.

The SI unit always appears first, and the English unit follows in brackets [ ]. Example: the temperature is 25°C [77°F]. The English equivalent of the SI unit will be rounded off to the nearest whole number to simplify the text and examples. A distance of 10 m may be shown as 33 ft whereas the exact equivalent is 32.81 ft.

If you are working in English units, you may find it helpful to mark out the parts that are in SI units and vice versa.

Some of the Figures have units of measurement. In such cases, two figures are included. The first one is in SI units, and the second-figure will be on the next page and will have English units. Each figure is identified with SI or English units. Be sure you use the figures in the units you are working in.

The following procedure is recommended for using this manual:

1. Turn to Page 1. Read the material until you come to the first problem or question.
2. Work the first problem or answer the question and enter the answer in the proper space in ink. If the problem or question is shown in both SI and English Units of measurement, answer only the part in units of measurement that you use.
3. Compare your answer with that shown at the end of the book; be sure to use solutions to the problems in the units you are working in.
   If your answer is correct, continue reading until you come to the next problem and work it. If not, restudy the manual until you understand the reason for your error. Rework the problem if necesary. Leave your wrong answer and note the correct one. This will keep you from making the same mistake later on.
4. Proceed stepwise as shown above until you have completed the text.

The above approach will require thought, making mistakes, and rethinking the situation. Concenrate on two things — the **how** and **why**. Do not cheat yourself by taking short-cuts or looking up the answers in advance. It saves time and errors but produces no real understanding. Your future depends on how efficiently you perform your job and not on how rapidly you proceed through this manual. Since this is your manual, any errors you make are private.

A validation or test covering the entire manual is included at the back of the book. Answers will be sent to your supervisor or training director at their request at no cost.

ABBREVIATIONS USED IN THIS MANUAL

SI UNIT ABBREVIATIONS

s, min	second, minute,	time
h, d	hours, day	time
mm	millimeter	length
cm	centimeter	length
m	meter	length
m^2	square meter	area
m^3	cubic meter	volume
m^3/d	cubic meters per day	volume rate
l	liter	volume
g, kg	gram, kilogram	weight
Pa, kPa	pascal, kilopascal	pressure
kPa(a)	kilopascal absolute	pressure
MPa	megapascal	pressure
bar	bar (1 bar = 100 kPa)	pressure
J, kJ	joule, kilojoule	heat, work
MJ	megajoule (J x 10^6)	heat, work
W,kW	watt, kilowatt	power

ENGLISH UNIT ABBREVIATIONS

s, min	second, minute	time
h, d	hour, day	time
in, ft	inch, foot	length
sq in	square inch	area
sq ft	square foot	area
cu ft	cubic foot	volume
gal	gallon	volume
bbl	barrel (42 US gal)	volume
BPD	barrels per day	volume rate
lb	pound	weight
psi	lbs per square inch	pressure
psia	lbs per sq in **absolute**	pressure
Btu	British thermal unit	heat
MBtu	thousand Btu	heat
MMBtu	million Btu	heat
W, kW	watt, kilowatt	electric power
hp	horsepower	mechanical power
cf/d	cubic feet per day	gas flow rate
Mcf/d	thousand cf/d	gas flow rate
MMcf/d	million cf/d	gas flow rate
M	thousand	
MM	million	

UNITS OF MEASUREMENT

SI UNITS OF MEASUREMENT

Most of the SI units of measurement used in the oilfield are traditional metric units. The exceptions we are concerned with are pressure and heat units. The SI pressure unit is kilopascal; the heat units are joule and watt. A watt equals 1 joule per second. Coversions from traditional metric units to SI units are as follows:

	METRIC UNIT	SI UNIT	CONVERSION
Pressure	bar	kilopascal, kPa	kPa = bar x 100
Heat	calorie	joule, J or	J = cal x 4.2
		watt, W	W = J ÷ time, sec.

STANDARD CONDITIONS FOR GAS

Measurement units for gas volume are cubic meters per day (m^3) or thousands of cubic feet per day (Mcf/d). The letters **st** or **s** are sometimes used with the units to designate volume at standard temperature and pressure: **m³d** (**st**) or **Mscf/d**. In this manual, standard volumes are corrected to a temperature of 15°C and a pressure of 101.325 kPa(a), or 60°F and 14.7 psia.

To simplify the text, the letters **st** and **s** are omitted. However, all gas volumes shown are at standard conditions unless specifically stated otherwise.

HEAT CAPACITY AND RELATIVE DENSITY

Specific heat and **specific gravity** are traditional terms that have been used in both Metric and English units for many years. These names are being replaced with the terms: **heat capacity** and **relative density**. The new names are used in this manual. When you see the term heat capacity (**Ht Cap**), it will have the same meaning as specific heat, and relative density (**Rel Dens**) means specific gravity.

PISTON TYPE COMPRESSORS
TABLE OF CONTENTS

DRAWINGS, GRAPHS AND ILLUSTRATIONS

INTRODUCTION

Compressors are mechanical devices used to raise the pressure of gas. Quite frequently, the turbine, engine or motor that drives a compressor is much larger, and requires more operating attention than the compressor itself. In this manual, we will discuss only the compressor, and not the driver.

There are two common types of compressors used in the oilfield and process plants:

1. Centrifugal Compressors
2. Piston-type (positive displacement) Compressors.

A third type of compressor — screw type — is a form of a positive displacement machine that is gaining in popularity in the oilfield.

This manual discusses only piston-type compressors. Centrifugal compressors are described in another two-part manual.

DEFINITIONS

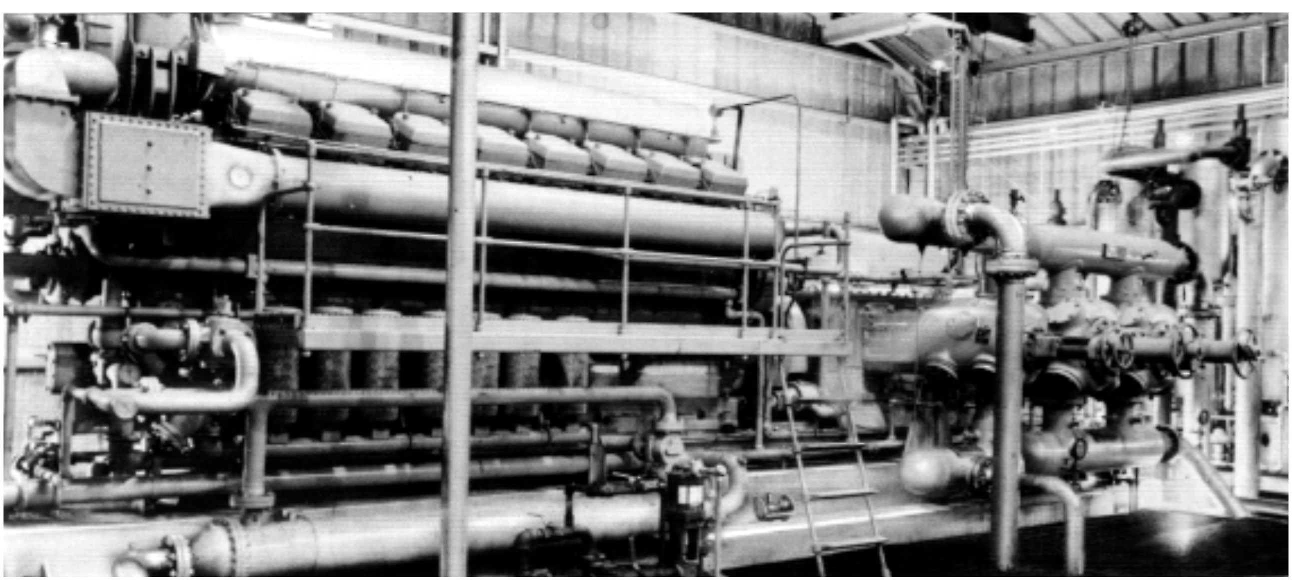
1. **Inboard** or **crank** end. The end of the compressor cylinder nearest the crankshaft. The **inboard stroke** is the movement of the piston toward the crankshaft.

2. **Outboard** or **head** end. The front of the cylinder. The **outboard stroke** is the piston movement toward the front of the cylinder.

3. **Double-acting** cylinder. A compressor cylinder that compresses gas on both the inboard and outboard strokes. It has suction and discharge valves at each end of the compressor cylinder. Most industrial compressor cylinder are double-acting.

4. **Single-acting** cylinder. A compressor that compresses gas on one stroke only, usually the inboard stroke. A double-acting cylinder can be made single-acting by removing suction valves on one end of the piston.

5. **Compression stage**. It is frequently necessary to boost gas pressure from a very low point to a very high point. The pressure rise is more than one cylinder can deliver, so the pressure increase is accomplished with 2 or more compressor cylinders, each boosting the gas pressure in successive increments. The gas pressure is boosted in stages from the first stage to the final stage.



GAS ENGINE DRIVEN COMPRESSOR

I. DESCRIPTION OF COMPRESSORS

A. Compressor Cylinder

There are five pricipal parts in a piston-type compressor cylinder:

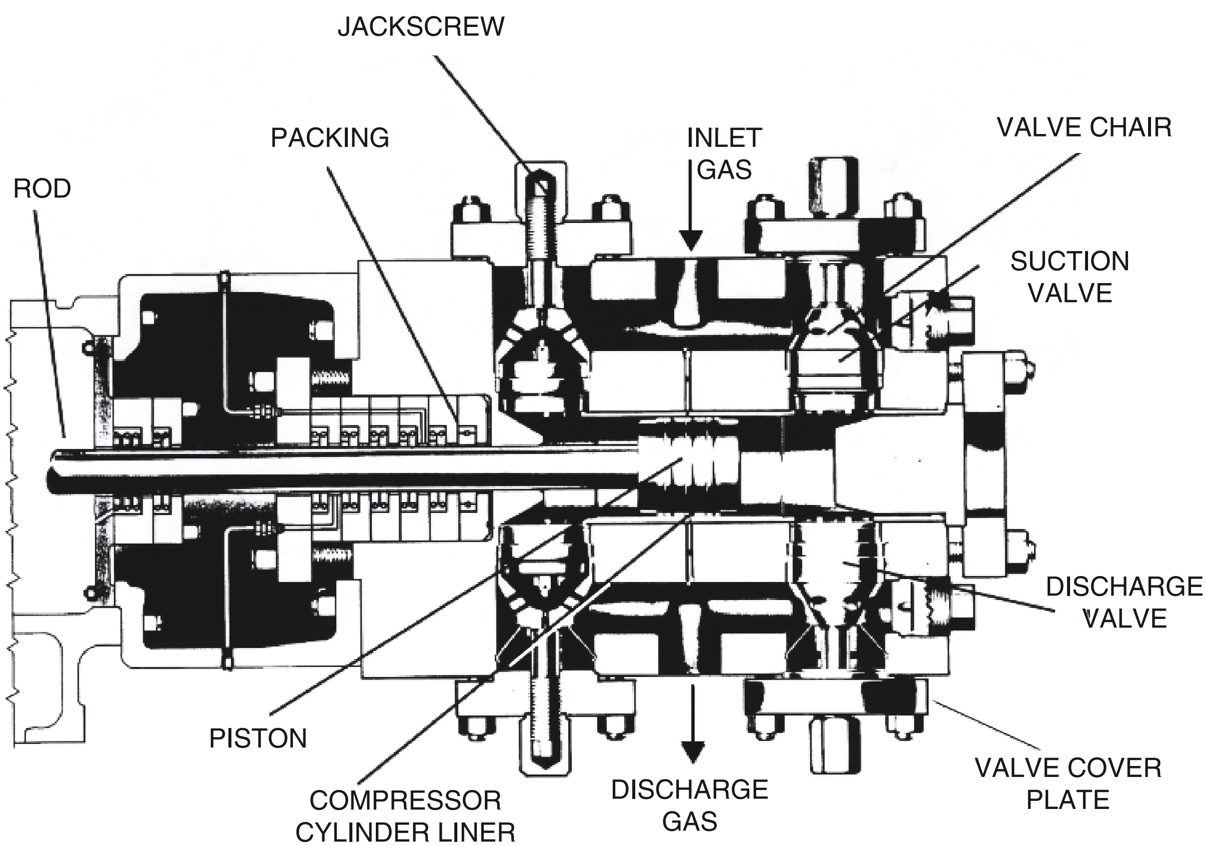
1. Piston
2. Rod
3. Cylinder
4. Valves
5. Packing

Each will be discussed separately.

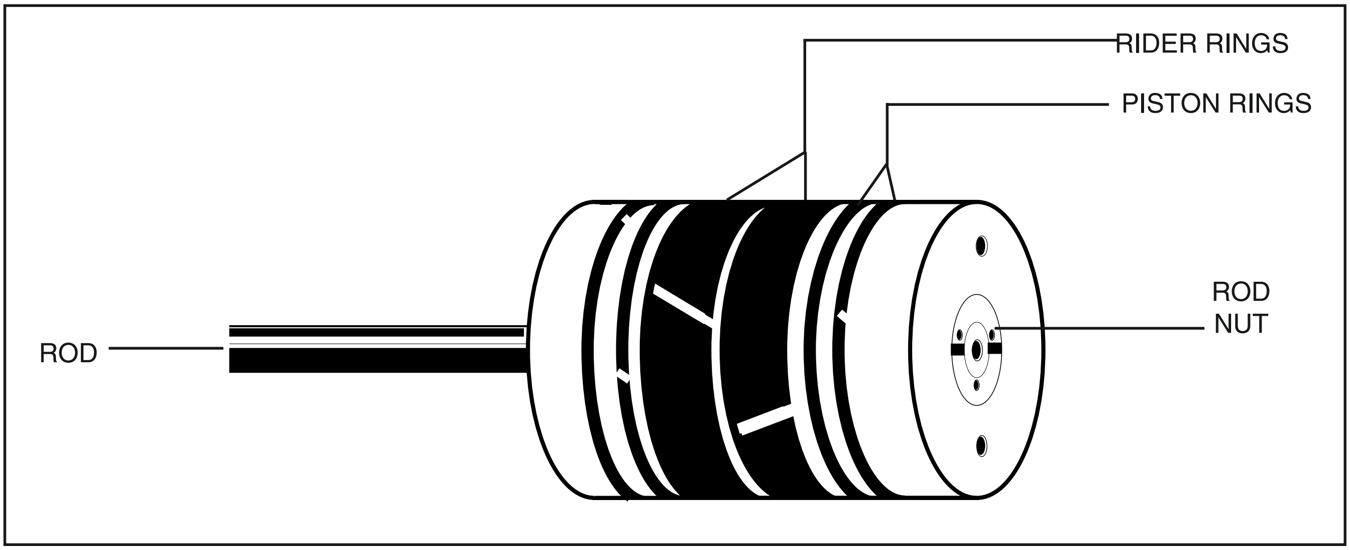
1. Piston

The piston is made of cast iron, aluminum or forged steel, depending upon the size and operating pressure. It is attached to the rod. It slips into the rod until it butts against a shoulder machined on the rod. It is secured to the rod with a nut or with a series of bolts as shown on the following pages.

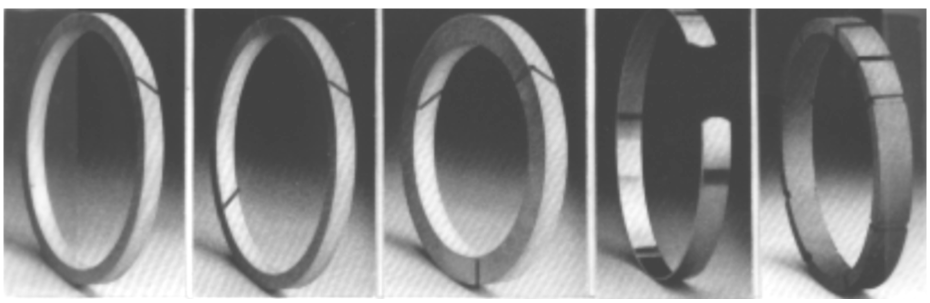
The piston has **grooves** or **lands** which contain **rider rings** and **sealing rings**. The rider rings are slightly larger than the piston. They prevent the piston from touching or "riding on" the cylinder wall.



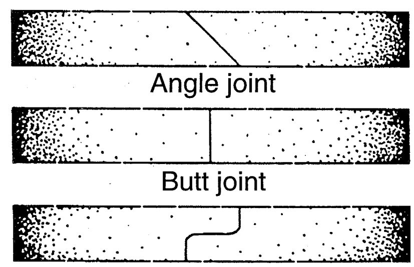
CUTAWAY DRAWING OF COMPRESSOR CYLINDER



COMPRESSOR PISTON AND RINGS



One Piece Two Piece Three Piece Expander Rider Ring



PISTON RINGS AND EXPANDER

Piston rings seal **discharge pressure** gas on one end of the piston from leaking into **suction pressure** gas on the other end of the piston. The rings can be a single piece, or in two or 3 segments. They are made of carbon filled teflon, bronze, cast iron, micarta or other plastic materials.

The number of piston rings usually depends upon the difference in pressure on each end of the piston, which is the difference between suction and discharge pressure in a double acting cylinder. More rings are required for a greater pressure difference.

An **expander spring** is sometimes installed behind the piston rings to push them against the cylinder.

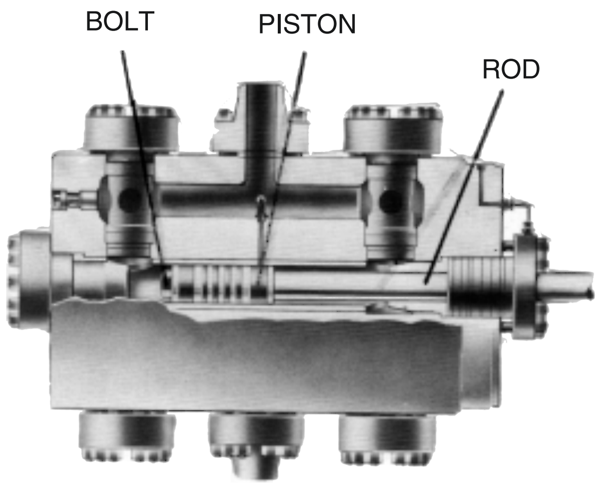
2. Rod

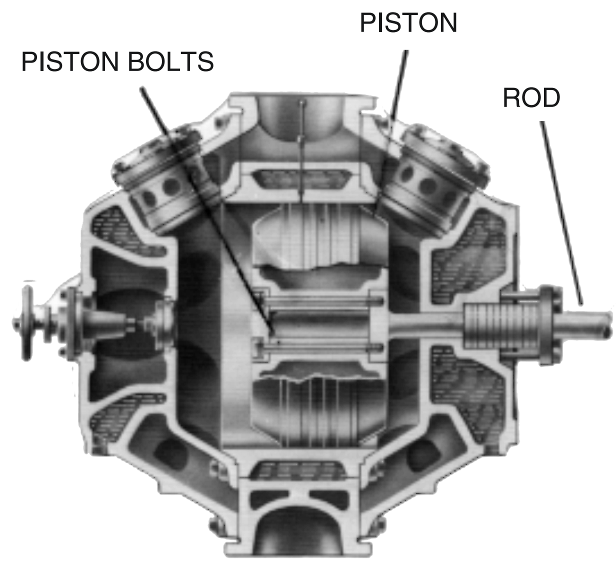
The rod is attached to the **piston** on one end, and the other end screws into a **crosshead**, which is attached to the driver with a **connecting rod**.

The rod usually is made of steel or steel alloy, depending upon the pressure and the corrosiveness of the gas. The section of the rod which travels through the packing often is lined or coated with a hardened material to reduce wear at that point.

The size of the rod is important in designing the compressor. If the diameter is too small, the rod may buckle during the outboard stroke, or pull apart during the inboard stroke. The capacity of the compressor is reduced by the volume of the piston rod. Consequently, a large rod reduces the volume of gas that the compressor can handle.

CYLINDER



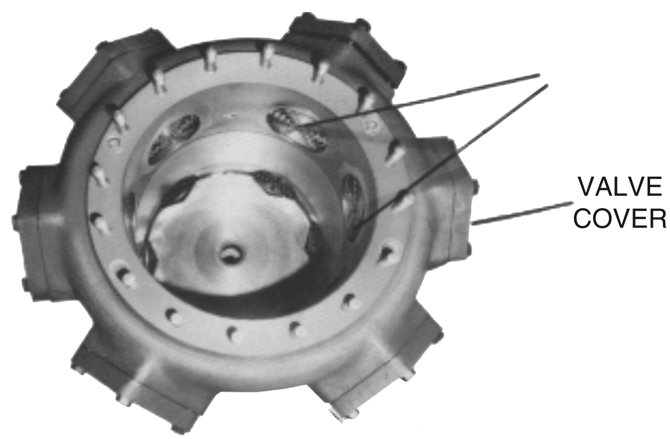


COMPRESSOR CYLINDERS SHOWING PISTON ATTACHMENT TO THE ROD

3. Cylinder

The compressor cylinder is machined to close tolerances, so that the piston will not scrape it during its stroke. Cylinders operating at a pressure below 7000 kPa [1 015 psi] are often made of cast iron. Higher pressure cylinders normally are made of steel. Quite often a liner is inserted in the cylinder, particularly if the gas is corrosive. A liner is shown in the drawing on page 2. The liner can be replaced rather easily if it corrodes or wears. This prevents replacement of the entire cylinder.

The cylinder usually has passages for cooling water to flow.



COMPRESSOR CYLINDER WITH END COVER REMOVED

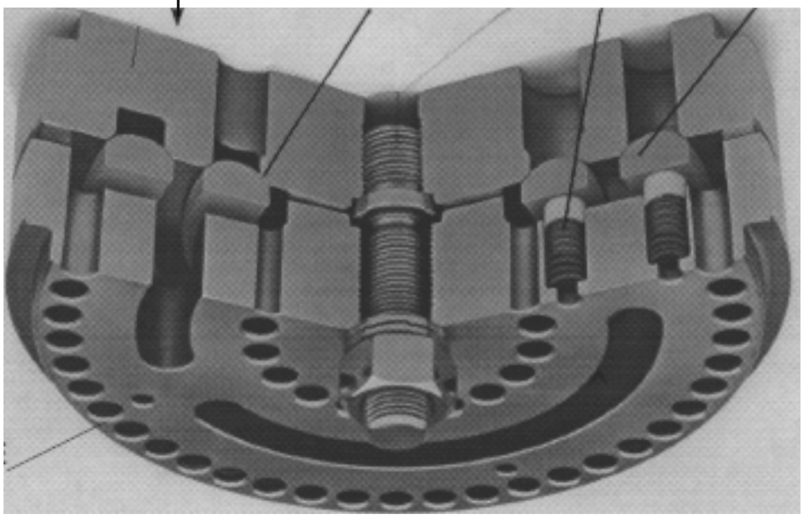
4. Valves

Compressor valves are a form of **check valve**: gas flows in only one direction through the valve. The suction valve opens on the intake stroke as soon as the pressure inside the cylinder is less than the pressure in the suction line. When the piston begins its compression stroke, pressure inside the cylinder exceeds suction line pressure and the suction valve closes. When the pressure exceeds the discharge line pressure, the discharge valve opens.

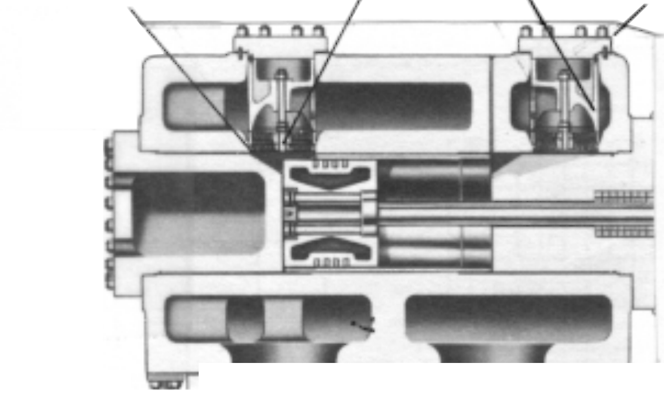
Suction and discharge valves are usually of the same type. In fact, the same valve is often used for either service. Its position in the cylinder as a discharge valve is upside down from its position as a suction valve.

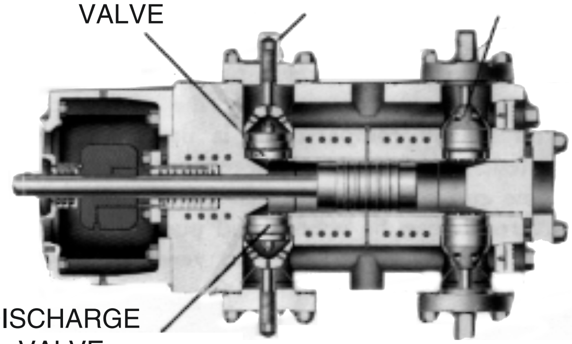
The valves usually sit in the cylinder on a metal gasket. A **retainer** or **chair** centers the valve in its seat. A jackscrew or valve cover plate secures the valve against the gasket.

VALVES



CUTAWAY OF COMPRESSOR DISCHARGE VALVE





VALVE INSTALLATION IN COMPRESSOR CYLINDER

VALVE TYPES

5. Packing

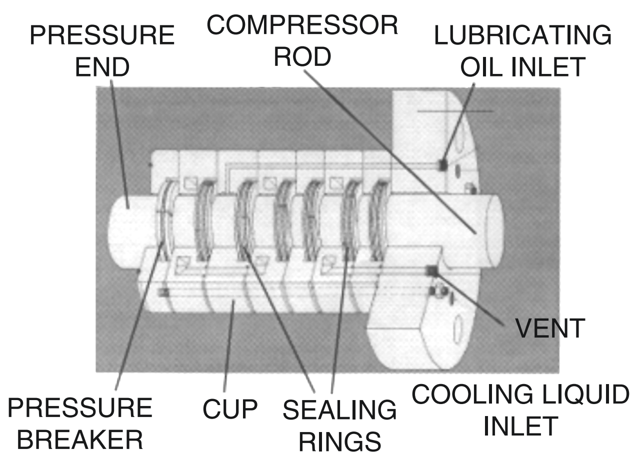
The packing is used to seal the pressure inside the cylinder from atmospheric pressure outside. The packing is contained in a **packing cage**. The cage is made up of a series of **cups**. Each cup contains **segmented packing rings**.

The packing material is usually the same as that used for piston rings - carbon filled teflon, bronze, micarta, etc.

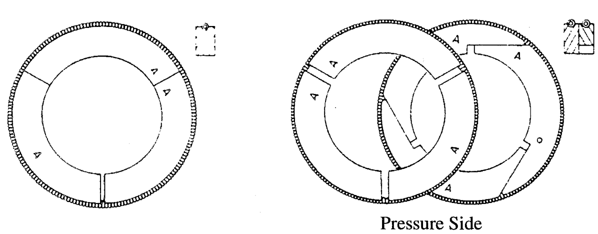
The number of packing rings will depend upon the pressure difference between that inside the cylinder and outside. More rings will be required for high pressure compressors.

The packing continuously rubs against the rod. This creates friction and heat. Lubricant is usually forced into the packing to minimize this friction. In some applications, water flows around the packing cage to prevent it from overheating. The packing is mounted on the crank end of the cylinder.

Gas under pressure inside the cylinder will attempt to flow down the rod and leak to the atmosphere. The head end of the cylinder is plugged, so no leakage will occur on it. Consequently, only the crank end must be sealed. The pressure in the crank end of the cylinder is suction pressure on the intake stroke, and discharge pressure on the compression stroke. The pressure changes from suction to discharge over 5 times a second. This rapid pressure change will result in discharge pressure gas inside the packing cage to backflow into the cylinder during the intake stroke. The flexing of the packing rings when the pressure changes from suction to discharge will shorten the life of the packing. To prevent this from occurring, one or two **pressure breaker rings** are installed on the front end of the cage. These are **limited leakage** type rings. They limit the amount of discharge pressure gas that leaks into the packing cage; and limit the backflow of high pressure gas from the cage to the cylinder during the intake stroke. The net effect of the pressure breakers is to hold a fairly constant pressure in the packing cage that is between suction and discharge pressure.



CUTAWAY DRAWING OF PACKING CAGE



Single Cut Pressure Breaker

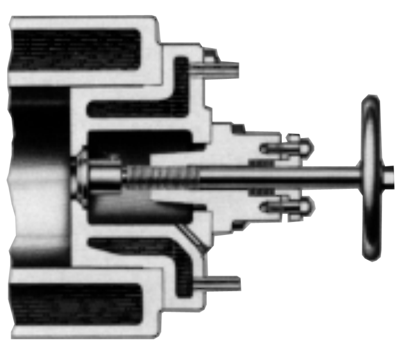
Ring Set That Seals From One Direction

PACKING TYPES

6. Miscellaneous Parts of Compressors

a. Clearance Pocket

Piston type compressors are designed for a fairly constant gas flow rate at a given speed. As we will see later, the power (and fuel) required to drive the compressor is about the same whether the compressor is operating at 50% or 100% of its capacity. The clearance pockets are a means for **lowering** the power required when the gas flow rate is **less** than the capacity of the compressor. They are also used to limit the capacity of a compressor so that the driver will not overload.



FIXED VOLUME CLEARANCE POCKET

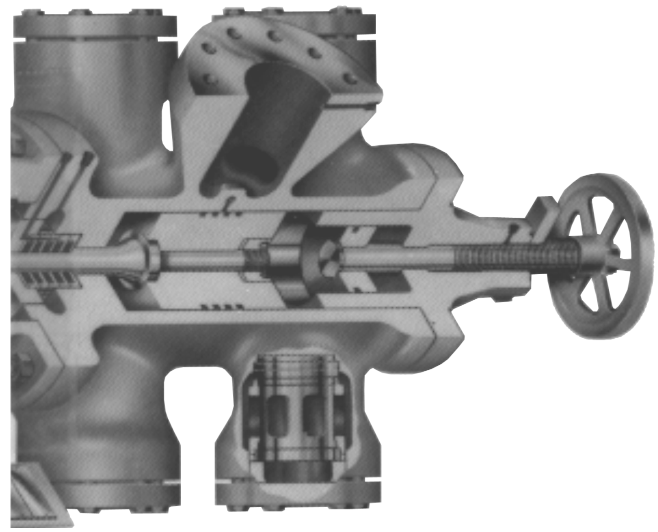
The clearance pocket may take the shape of a piston which is moved inboard or outboard, or may be a valve which is connected to a chamber. The valve can be fitted with an actuator which automatically adjusts the clearance to hold a constant suction pressure or other process condition. In any case, the effect is the same — to adjust the capacity of the compressor so that it is the same as the volume of gas entering the unit.

b. Valve Unloaders

Valve unloaders are used to 'unload' or take out of service the inboard or outboard suction valves. Most gas compressors are double-acting machines: they compress gas on both the inboard and outboard strokes. A valve unloader unseats a suction valve, so that the compressor will not compress gas in that portion of the cylinder where the valve has been unloaded. If valves are unloaded on both ends of the cylinder, that cylinder is completely out of service.

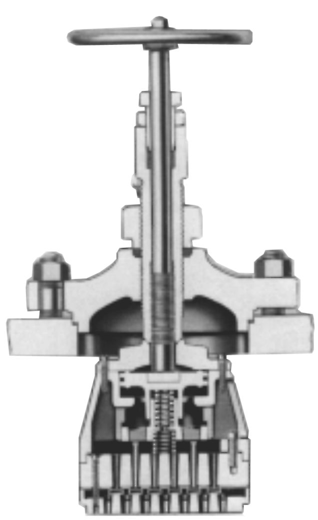
Valve unloaders are most frequently used on instrument air compressors to allow the compressor to run in a non-compressing mode rather than shut it down, and then restart it a few minutes later when the demand for air is present. Frequent starting and stopping the unit can cause mechanical failures.

The most common type of unloader has "fingers" which unseat valve discs as shown below.

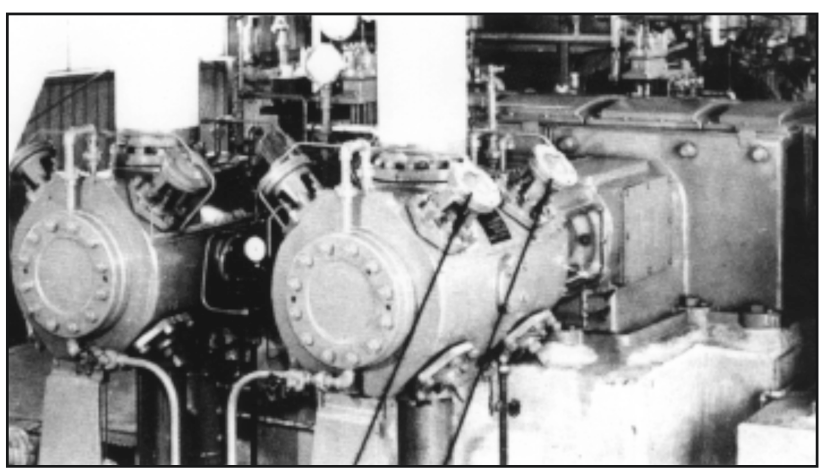


COMPRESSOR CYLINDER WITH ADJUSTABLE CLEARANCE POCKET

Valve unloaders can be operated by hand, or they can be equipped with air or hydraulic actuators that automatically operate them. The latter type are often used on pipeline compressors that are started and stopped from a remote control point. Valves are unloaded at startup; after the machine warms up and reaches operating speed, the valves are loaded, and the compressor starts compressing gas.



MANUAL VALVE UNLOADER



COMPRESSOR CYLINDERS WITH AIR ACTUATED VALVE UNLOADERS

7. Lubrication System

Proper lubrication of the piston and packing is probably the most important factor in compressor operation. Without lubrication, piston rings and packing will become hot and eventually break, melt, or distort.

During the past 20 years, technological improvements in lubricants and ring and packing material have significantly reduced the quantity of lubricating oil required. However, the service life of so called "non-lubricated" rings and packing usually can be extended several fold if some lubrication is provided. One of the non-lubricated materials is a substance called **molybdenum disulfide**. It is a solid material, rather than a liquid, and has excellent lubricating qualities. It is most frequently used in rings and packing made of a plastic material in which the molybdenum disulfide is dispersed on the outer surface of the material.

Manufacturers of certain plastic materials, such as teflon, also claim that little or no lubrication is required. These non-lubrication claims are probably valid if the gas contains no dirt, solid or liquid particles or mist. Unfortunately, perfectly clean gas seldom occurs in the oilfield; consequently, lubrication is almost always required.

As we said, lubrication is used on the piston rings and the rod packing. These are the only parts in the compressor cylinder that continually rub. With proper lubrication, an oil-film coats the cylinder so that the piston rings and cylinders never touch. The piston floats on a film of oil. In the same manner, an oil-film coats the rod where it contacts the packing. The film actually separates the rod from the packing. So long as the lubrication film is present, very little wear will occur on the packing or the piston rings. If the film is destroyed, the packing or rings will make direct contact with metal and wear will occur.

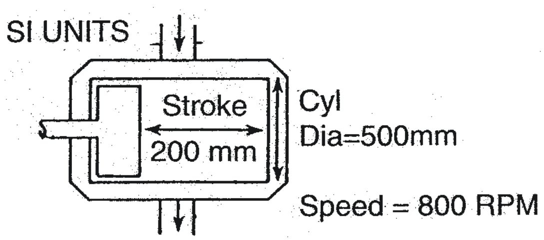
On each stroke of the piston, some of the oil film on the cylinder wall and rod is wiped off. It must be continually replaced to prevent piston rings and packing from rubbing against metal. The amount of oil that must be added depends upon the area that is swept by the piston rings and packing. One liter will lubricate 4000 000 m^2 of swept area. A liter equals 25 000 drops. [One gallon will lubricate 16 000 000 square feet of swept area. A gallon equals 100 000 drops.]

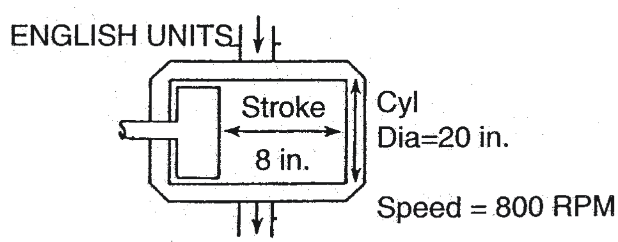
The swept area is determined from the cylinder diameter, length of the stroke, and compressor speed. The graphs on the following pages indicate the quantity of oil required to lubricate the cylinder wall for various sizes and speeds. The oil rates from the graphs do **not** include the rate of oil injected in the packing. Cylinders having a diameter of 200 mm [8 in.] or less often have one injection point at the top of the cylinder. Larger cylinders may have 2 or more injection points.

The graphs show the total oil to lubricate the cylinder wall. The rate per injection point is the total from the graph divided by the number of injection points. If there is only one injection point in the cylinder, the rate from the chart is obviously the rate required for the cylinder wall.

The rate of oil injection in the packing will be the same as the injection rate at each point in the cylinder wall. If the cylinder has one injection point, the rate to the packing will be the same as that in the cylinder wall. If the cylinder wall has 2 injection points, the rate to the packing will be one half of the total to the cylinder wall.

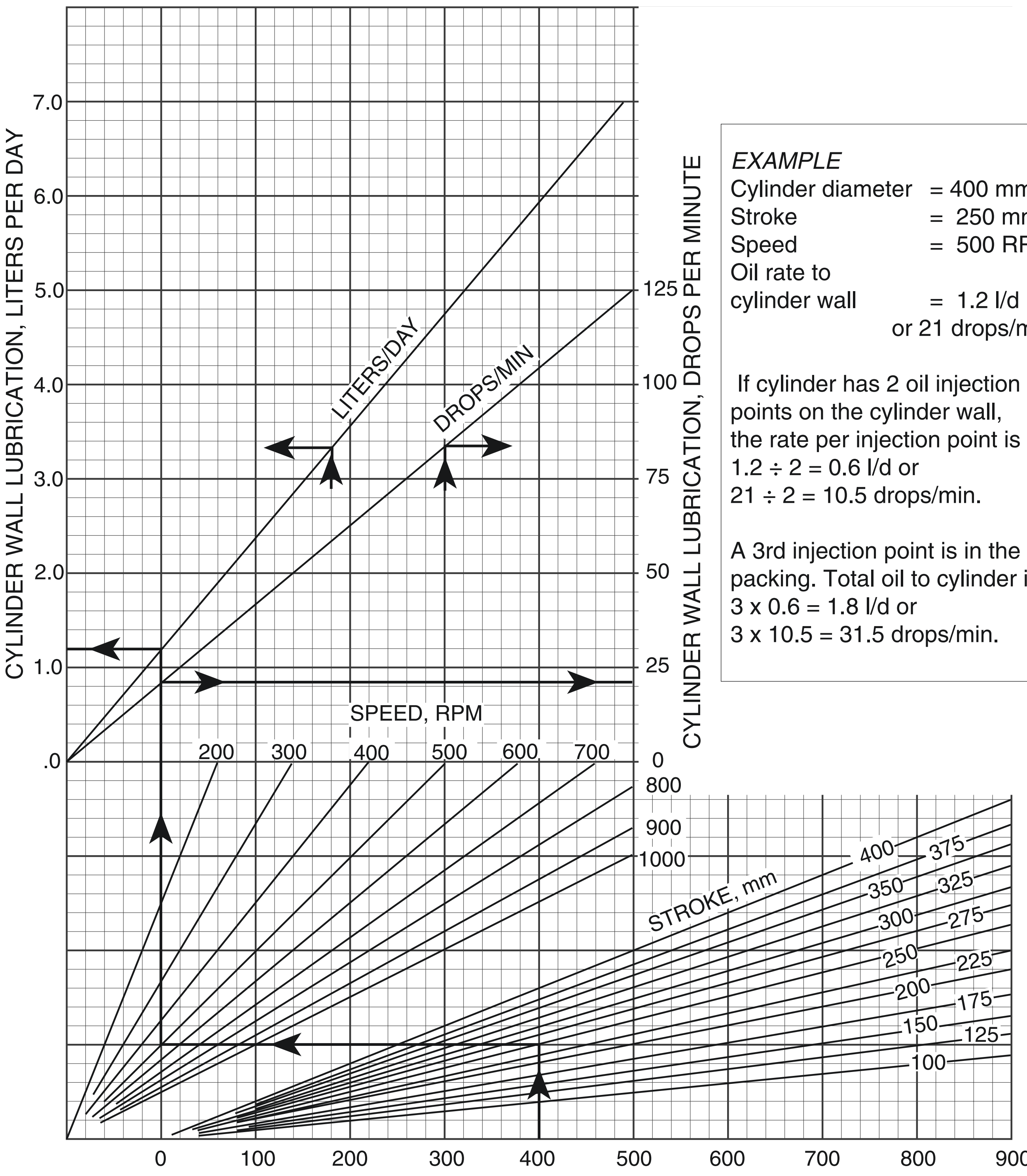
Example: Determine the oil injection rates in the cylinders shown below. There is 1 injection point on the cylinder wall and 1 in the packing.





	SI Units	English Units
Oil rate to cylinder wall	= 2.0 l/d or 34 drops/min.	0.50 gal/d or 35 drops/min.
Oil rate to packing	= 2.0 l/d or 34 drops/min.	0.50 gal/d or 35 drops/min.
Total oil rate	= 4.0 l/d or 68 drops/min.	1.0 gal/d or 70 drops/min.

COMPRESSOR CYLINDER LUBRICATION - SI UNITS

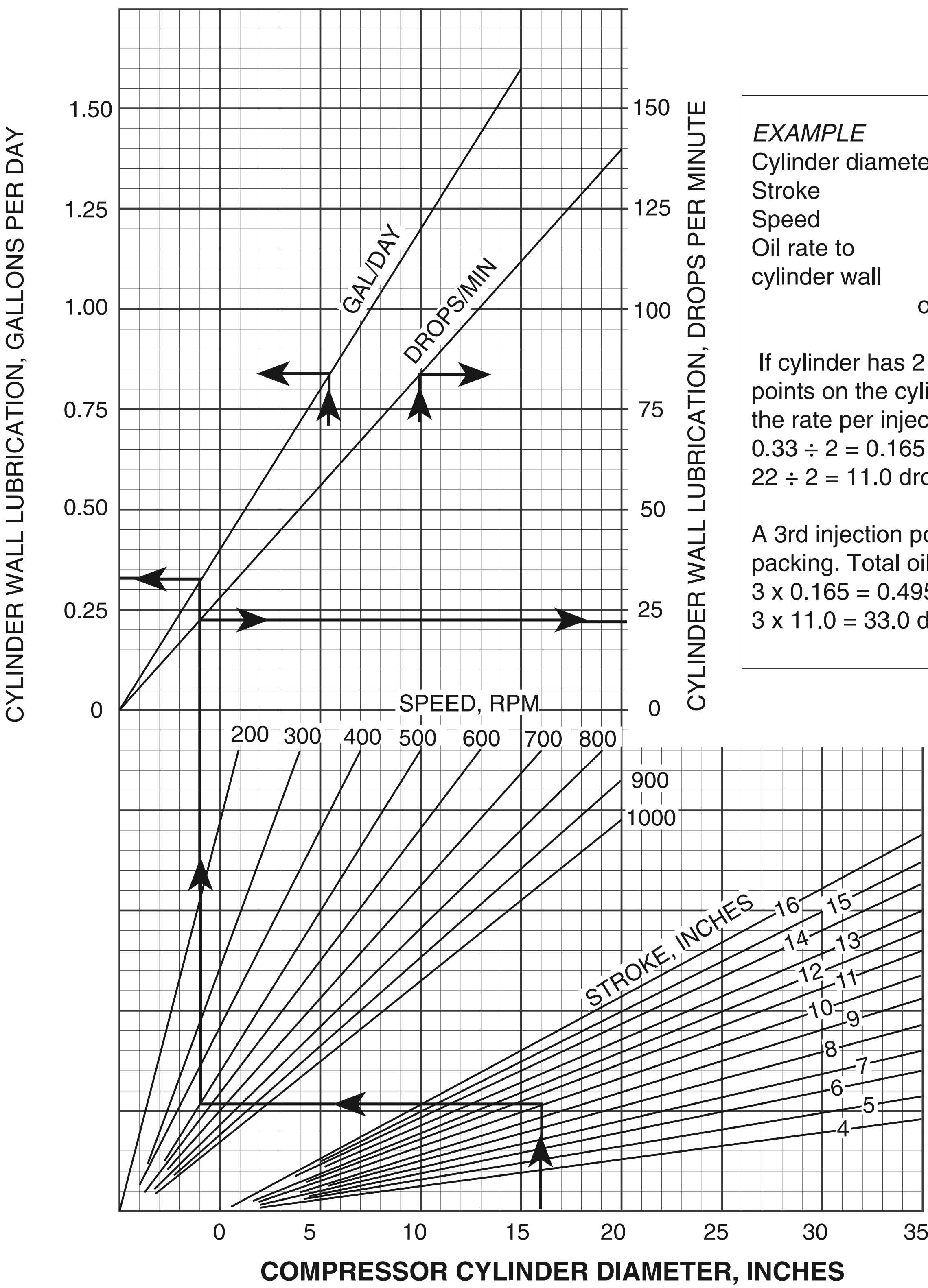


EXAMPLE
Cylinder diameter = 400 mm
Stroke = 250 mm
Speed = 500 RPM
Oil rate to cylinder wall = 1.2 l/d or 21 drops/min

If cylinder has 2 oil injection points on the cylinder wall, the rate per injection point is 1.2 ÷ 2 = 0.6 l/d or 21 ÷ 2 = 10.5 drops/min.

A 3rd injection point is in the packing. Total oil to cylinder is 3 x 0.6 = 1.8 l/d or 3 x 10.5 = 31.5 drops/min.

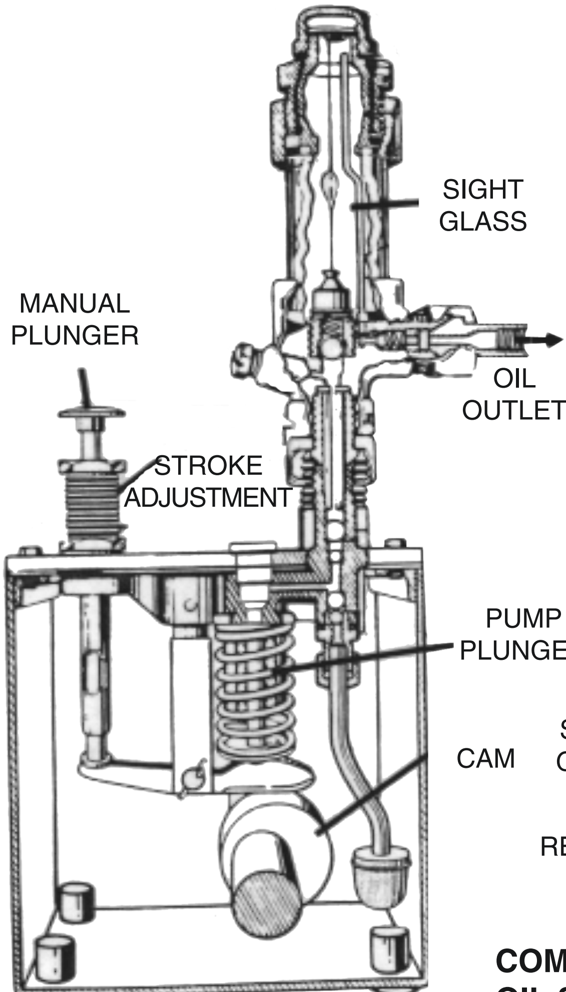
COMPRESSOR CYLINDER LUBRICATION - ENGLISH UNITS



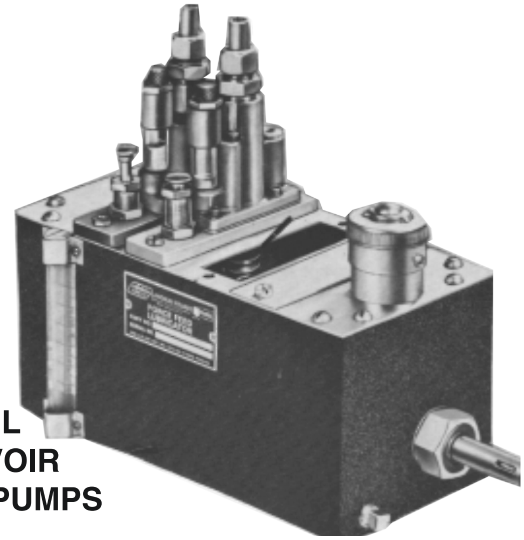
EXAMPLE
Cylinder diameter = 16 in.
Stroke = 10 in.
Speed = 500 RPM
Oil rate to cylinder wall = 0.33 gal/d or 22 drops/min

If cylinder has 2 oil injection points on the cylinder wall, the rate per injection point is 0.33 ÷ 2 = 0.165 g/d or 22 ÷ 2 = 11.0 drops/min.

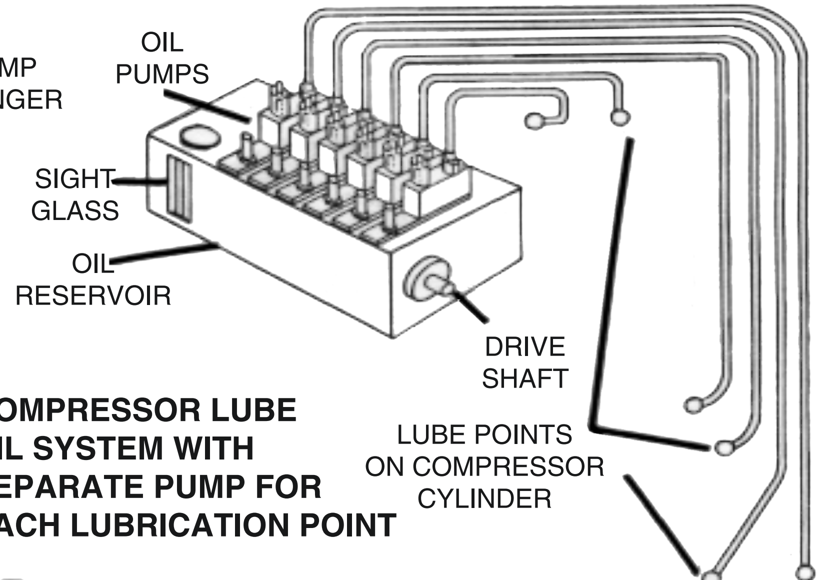
A 3rd injection point is in the packing. Total oil to cylinder is 3 x 0.165 = 0.495 g/d or 3 x 11.0 = 33.0 drops/min.



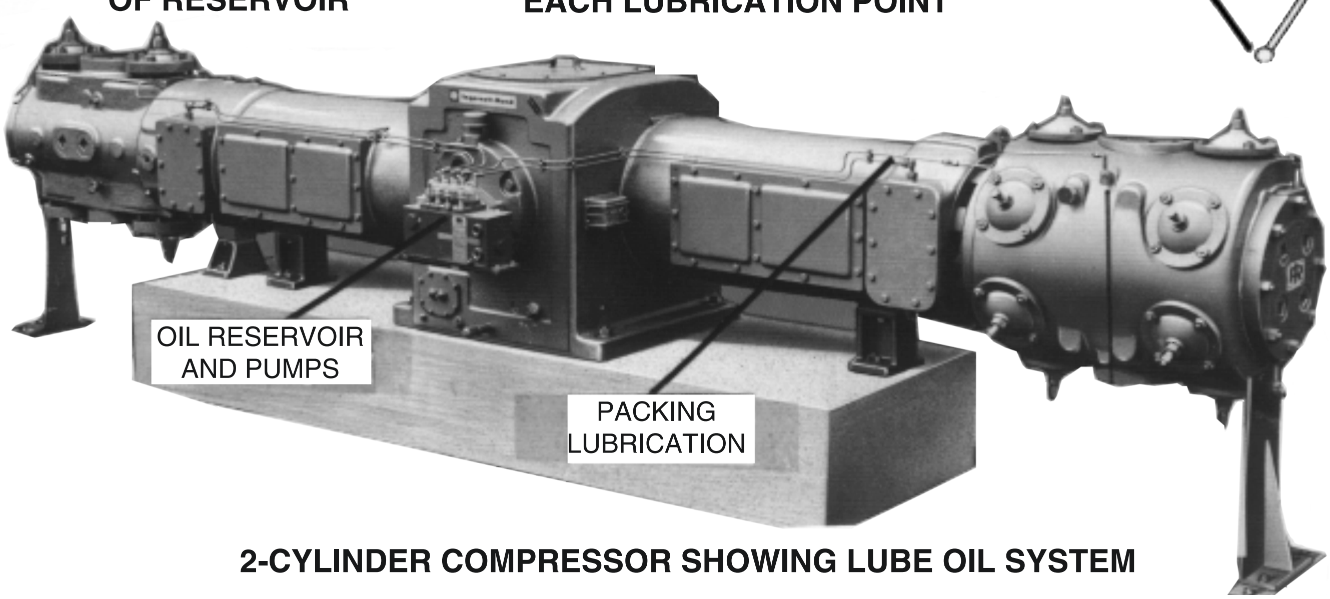
CROSS SECTION OF OIL PUMP OF RESERVOIR



LUBE OIL RESERVOIR WITH 2 PUMPS



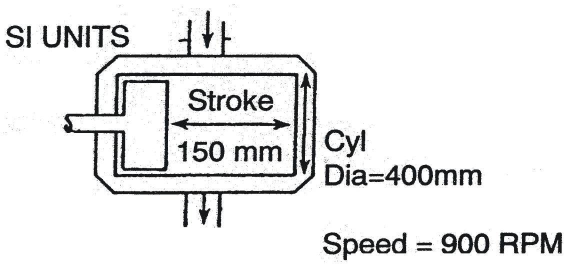
COMPRESSOR LUBE OIL SYSTEM WITH SEPARATE PUMP FOR EACH LUBRICATION POINT

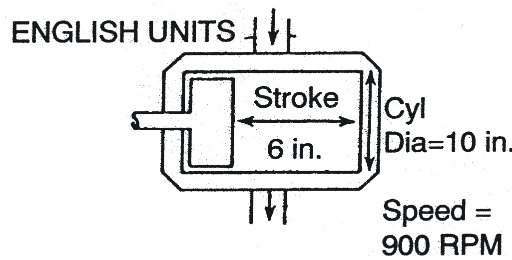


2-CYLINDER COMPRESSOR SHOWING LUBE OIL SYSTEM

Problem 1:

The following cylinder has 2 oil injection points on the cylinder wall and one in the packing. Determine the oil injection rate in the cylinder and packing.





Cylinder Lubrication ______ Packing Lubrication ______

The type of lubricant used in the system will depend upon the quality of the gas and material of construction. If the gas contains corrosive materials such as H_2S, CO_2, sulfur compounds, or oxygen, corrosion resistant lubricants may be required. A tallow-base oil is often used when the gas contains liquid particles that dilute other lubricating oils.

If the compressor is driven with a gas engine, the lubricating oil used in the engine crankcase is often suitable for use in the compressor. Lubrication in the engine continually circulates through the system and deteriorates with time. It must be replaced at periodic intervals. Frequently, oil from the crankcase of the driver is used for makeup oil to the compressor lubricating system. With this arrangement, fresh oil is added to the driver instead of the compressor. This reduces the frequency of changing oil in the driver crankcase as fresh oil is continually added to it.

A lubrication system for compressor cylinders is shown on the previous page. A separate lubricating pump is provided for each point that oil is injected. The pumps are immersed in a common reservoir. Each pump contains a plunger that discharges a few drops of oil on each stroke. A camshaft in the reservoir strokes the pumps. It is connected to the compressor driver with a reduction gear that turns the cam 5 to 10% of the speed of the compressor. In other words, the camshaft on a 300 rpm compressor would rotate at 15 to 30 rpm. Each pump would stroke every 2 to 4 seconds.

This lubricating system has a separate pump for each oil injection point. Large cylinders may have 5 or more injection points on the cylinder wall and packing. A unit having 8 such cylinders could require up to 40 lubricating pumps. Considerable attention would be required to check each pump for proper operation.

The number of pumps can be reduced if the oil output from a pump is divided among several injection points. However, the pressure required to inject oil at each different lubrication point may be slightly different. Consequently, if the flow form one pump were divided, most of the oil would flow to the point of lowest pressure.

This objection can be overcome by the se of **divider blocks.** Each divider block splits the incoming oil stream into two equal outlet streams at the same oil pressure as that of the inlet oil. Several blocks can be installed in the same assembly, so that the inlet stream can be split into a number of outlet streams. When four divider blocks are used, eight outlet streams are available.

In addition to reducing the number of pumps required, the driver blocks can also be installed so that two pumps feed one divider block. This provides a safety factor in the event that one or the pumps fails.

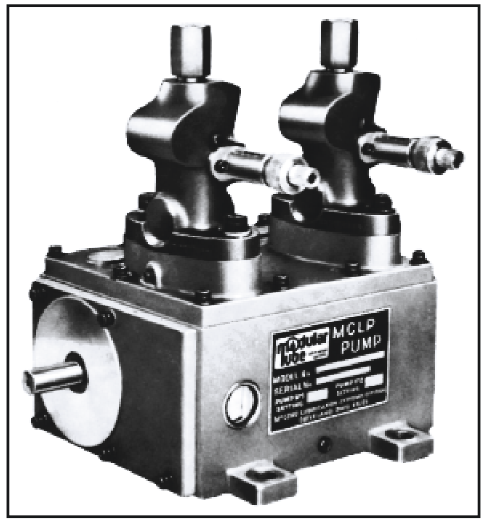
The divider assemblies can be arranged so that the output from one assembly enters a second one. The arrangement shown below splits the inlet oil stream into 32 parts. In this case, the blocks in the master assembly usually have a single outlet rather than a double one.

A panel is included in the system which includes an oil pressure gauge, and a counter. The name plate on the panel will indicate the number of counts equivalent to a liter or pint of oil.

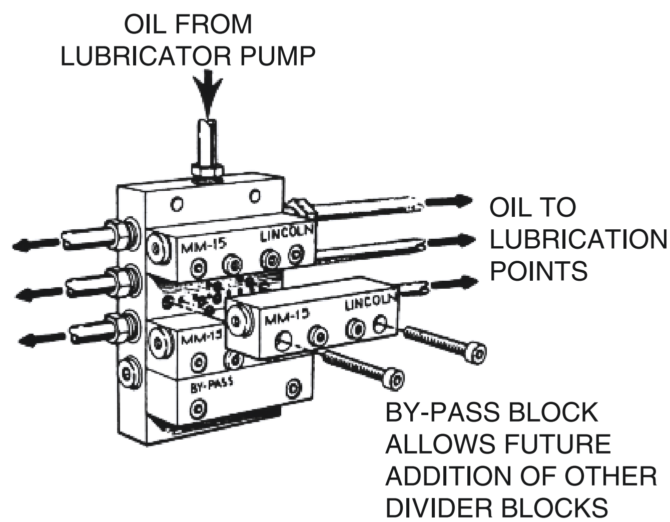
The drawings on pages 14 and 15 indicate the operation of the plungers in a 3-block system. Because of the interaction of flow between the blocks, a flow stoppage at any point will stop flow for the entire system. This triggers a low flow shut-down.

Divider blocks can be supplied with plungers of different sizes to provide different oil injection rates. Consequently, the system can be designed to provide the proper injection rate to each injection point. However, if a system having different divider blocks ever requires maintenance or replacement, it is likely that the wrong size of replacement block will be installed. Thus, most divider block systems have only one size of block. Its capacity is the rate required at the point of maximum oil injection. This is usually in the largest diameter cylinder.

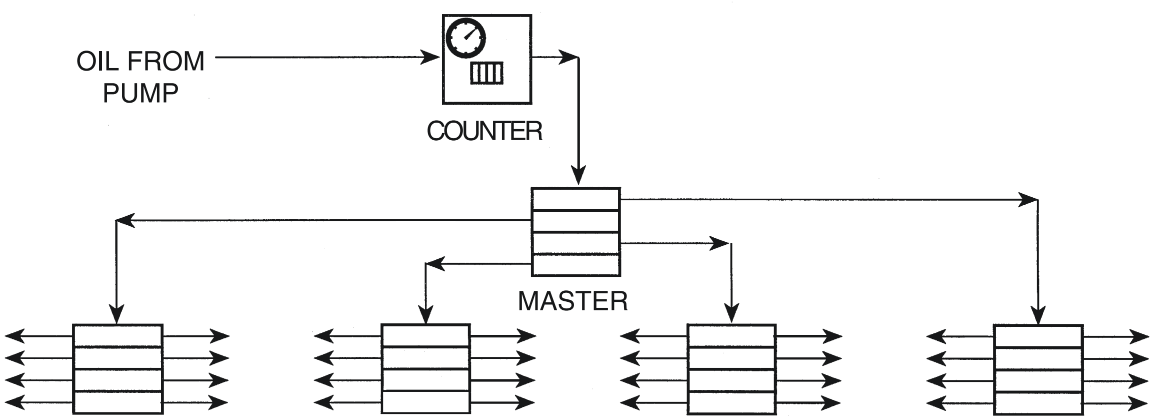
Quite often, oil is injected at 2 or more points on the wall of a compressor cylinder. In this case, the quantity of oil required at each injection point is the amount calculated by the graphs on pages 9 and 10 divided by the number of injection points.



LUBRICATION PUMPS FOR MULTI-POINT DIVIDER



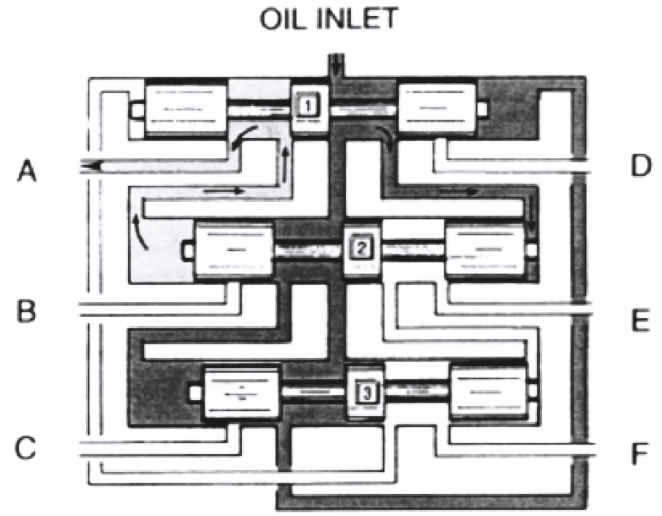
SIX POINT DIVIDER BLOCK ASSEMBLY



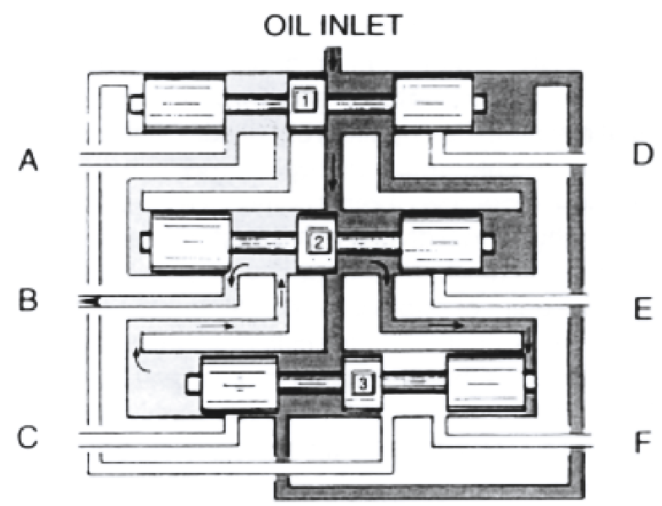
32 POINT OIL DIVIDER BLOCK SYSTEM

OPERATION OF DIVIDER BLOCK OIL LUBRICATORS

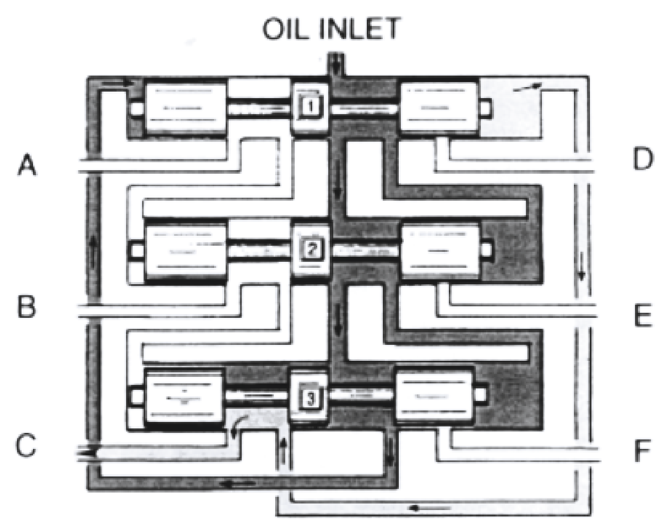
Inlet oil flows through block #1 to the right end of the piston in block #2, pushing the piston to the left. Oil on the left end of the piston is pushed through the passageway on block #1 to injection point A.



When the piston in block #2 nears the end of its stoke to the left, a passageway is opened for inlet oil to flow through blocks #1 and #2 and enter the right end of the piston in block #3. the piston moves to the left and pushes oil from the left end of the piston through the passageway in block #2 to injection point B.

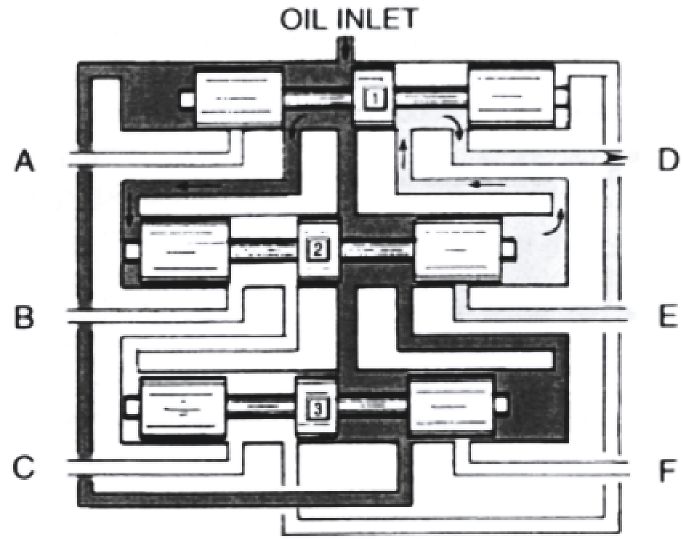


When the piston in block #3 nears the end of its stroke to the left, passageways are open for inlet oil to flow through all 3 blocks to the left end of the piston in block #1. The piston moves to the right and pushes oil through a passageway on block #3 to injection point C.

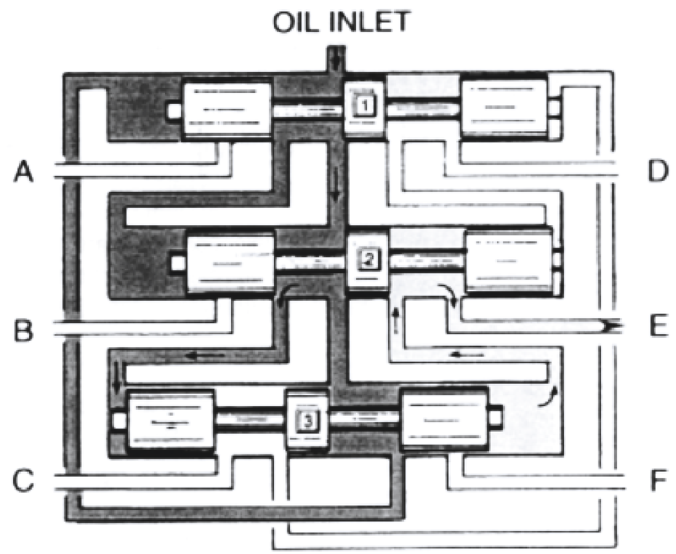


INLET OIL
OIL TO INJECTION

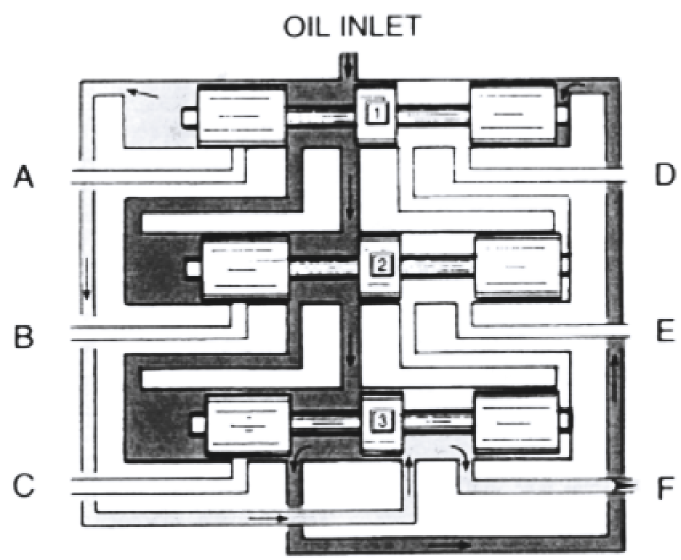
When the piston in block #1 nears the end of its stroke to the right, a passageway is open for inlet oil to flow through block #1 to the left side of the piston in block #2. The piston moves to the right, pushing oil through a passageway in block #1 to injection point D.



When the piston in block #2 nears the end of its stroke to the right, passageways are open for inlet oil to flow through blocks #1 and #2 to the left side of the piston in block #3. The piston moves to the right, pushing oil through a passageway in block #2 to injection point E.



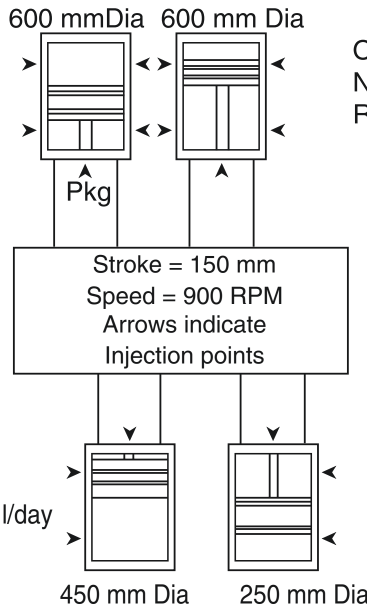
When the piston in block #3 nears the end of its stroke to the right, passageways are open for inlet oil to flow successively through blocks 1, 2 and 3 to the right end of the piston in block #1. The piston moves to the left, pushing oil to the left of the piston through the passageway in block #3 to injection point F.



INLET OIL	(dark)
OIL TO INJECTION	(light)

Example: A compressor with a divider block lube system has 4 compressor cylinders with sizes and calculated oil injection rates as follows. If each divider block is identical, determine the capacity of each and the total oil rate.

SI UNITS



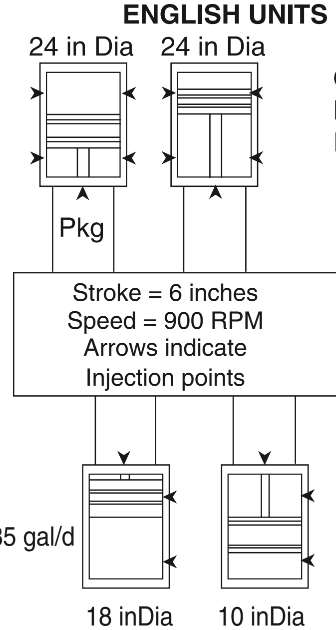
Oil to cyl wall = 1.9 l/day
No. injection points on cyl wall = 4
Rate per inj point = 1.9 ÷ 4 = 0.475 l/d

Oil to cyl wall = 1.4 l/day
No of injection points on cyl wall = 2
Rate per injection point = 1.4 ÷ 2 = 0.7 l/day

Oil to cyl wall = 0.8 l/day
No of injection points on cyl wall = 2
Rate per injection point = 0.8 ÷ 2 = 0.4 l/d

Max injection point rate = 0.7 l/d in 450 mm cylinder
Total number of injection points = 5 + 5 + 3 + 3 = 16
Total oil injection with divider block system = 16 x 0.7 = 11.2 l/day

ENGLISH UNITS



Oil to cyl wall = 0.5 gal/d
No. injection points on cyl wall = 4
Rate per inj point = 0.5 ÷ 4 = 0.125 gal/d

Oil to cyl wall = 0.37 gal/d
No of injection points on cyl wall = 2
Rate per injection point = 0.37 ÷ 2 = 0.185 gal/d

Oil to cyl wall = 0.23 gal/d
No of injection points on cyl wall = 2
Rate per injection point = 0.23 ÷ 2 = 0.115 gal/d

Max injection point rate = 0.185 gal/d in 18 inch cylinder
Total number of injection points = 5 + 5 + 3 + 3 = 16
Total oil injection with divider block system = 16 x 0.185 = 2.96 gal/d

The previous example illustrates the procedure for determining the oil rate in a divider block system when each block has the same capacity. In this case, all blocks have to deliver at the rate of the highest injection point. This results in an excess rate to other injection points. The excess oil is almost 40%.

The excess can be reduced by using divider blocks having capacities to match the rate at each injection point. This will obviously reduce the volume and cost of lube oil. It will also require a higher spare parts inventory. The main drawback to having blocks of different capacities is that of installing a low capacity block in a high capacity application. This can result in piston ring or rod failure.

8. Compressor Cylinder Cooling System

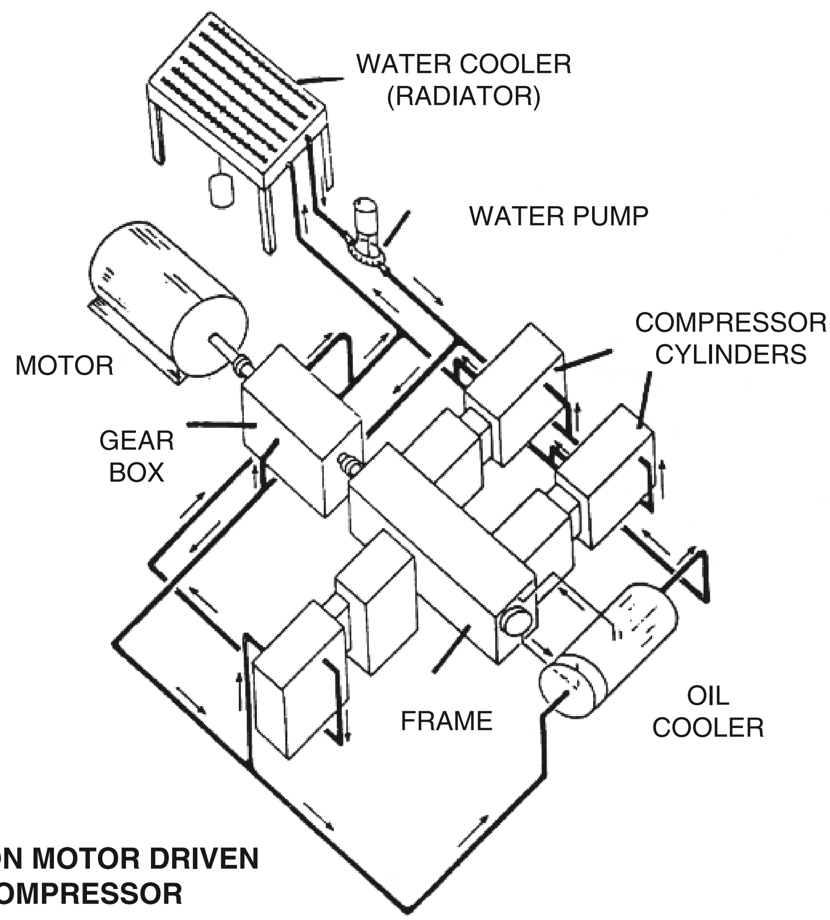
As gas is compressed, its temperature rises. This in turn, heats the compressor cylinder, piston, valves, etc., which can weaken them to the point of failure. Most compressor cylinders have passages for circulating cooling water to prevent overheating.

A cooling system for a motor-driven frame type compressor is shown opposite. The system is a "closed loop" type similar to that in an automobile. The pump continuously circulates water in the system.

The temperature of water out of the cooler is regulated with a temperature controller or thermostat which is usually set at 38° to 54°C [100° to 130°F].

If the unit is installed in locations where the temperature gets below freezing, antifreeze is added to the circulating water to prevent it from freezing when the compressor is shutdown. A rust inhibitor should be added to the water to prevent corrosion in the system.

Almost all compressors made before 1975 had cooling water in compressor cylinders. Some of the newer machines have no cooling water in the cylinders. No significant change in maintenance has been experienced with the non-cooled cylinders.



COOLING SYSTEM ON MOTOR DRIVEN FRAME TYPE COMPRESSOR

B. Crosshead and Connecting Rod

Compressor cylinders can be mounted directly on an engine, or on a frame that is driven by an engine, motor, or turbine. In either case, a **cross-head** and **connecting rod** are used to transfer power and motion from the drive shaft to the compressor piston.

The compressor piston rod attaches to one end of the crosshead. The connecting rod is also attached to the crosshead and to the drive shaft.

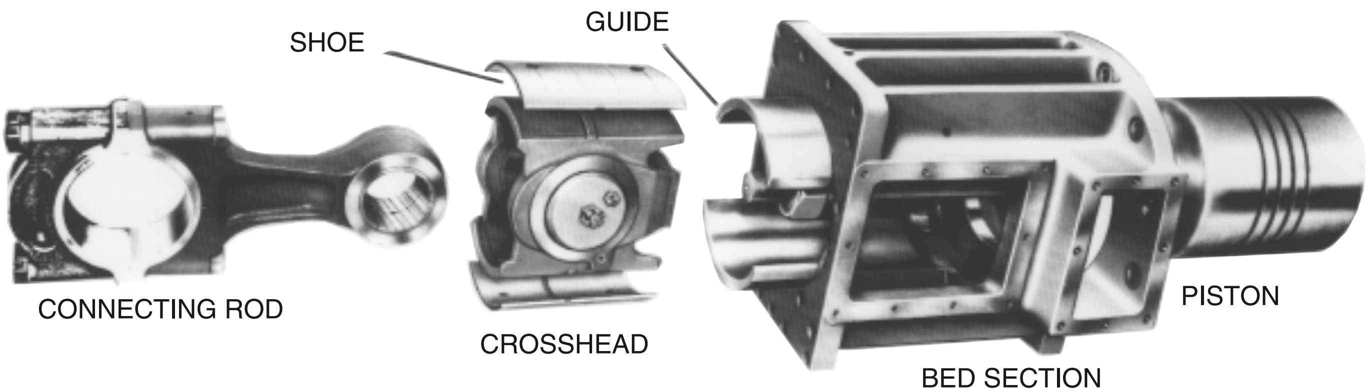
The function of the crosshead and connecting rod is to convert circular motion at the drive shaft into horizontal motion at the piston rod and piston. The piston rod continuously strokes back and forth through the packing. If this stroke is not in a straight line, the packing will quickly wear out and start to leak. The crosshead assures straight line movement of the piston rod.

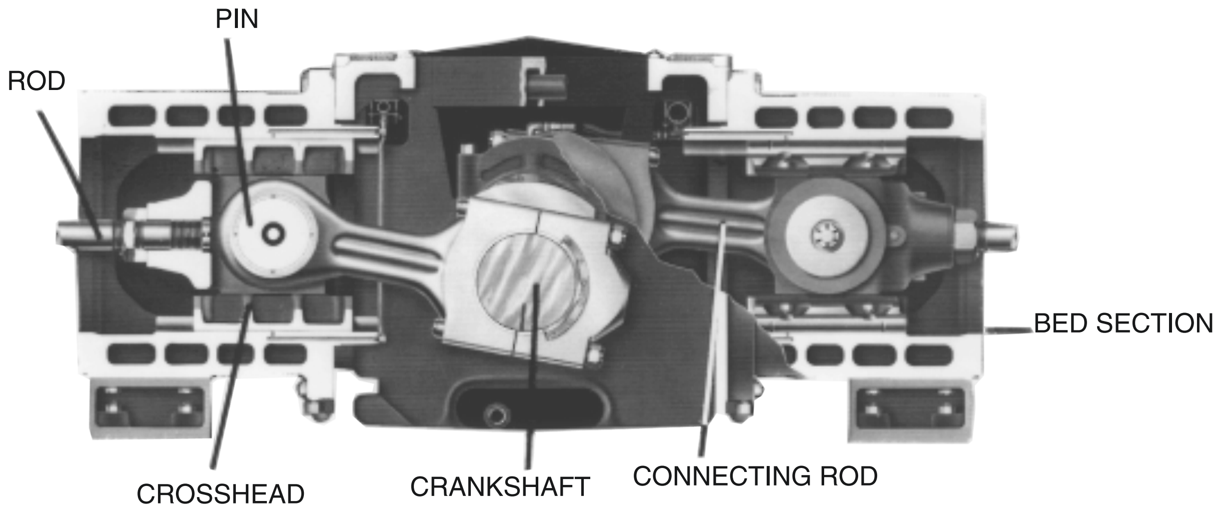
The crosshead actually is a moving **bearing**. The upper and lower plates or "**shoes**" are made of bearing material — usually babbitt. The shoes travel between **guides** which are usually installed in a bed section or distance piece. The compressor cylinder attaches to the bed section.

The **piston rod** screws into the crosshead. A lock nut prevents it from unscrewing while the unit is operating.

The **connecting rod** is the link between the crosshead and crankshaft. It is held in the crosshead with a **pin**. The interior faces of the connecting rod are made of bearing material.

The crosshead is lubricated by the same lubricating system that supplies the crankshaft. This is a separate system from the compressor cylinder lubrication system.





CROSS-SECTION SHOWING CROSSHEAD INSTALLED

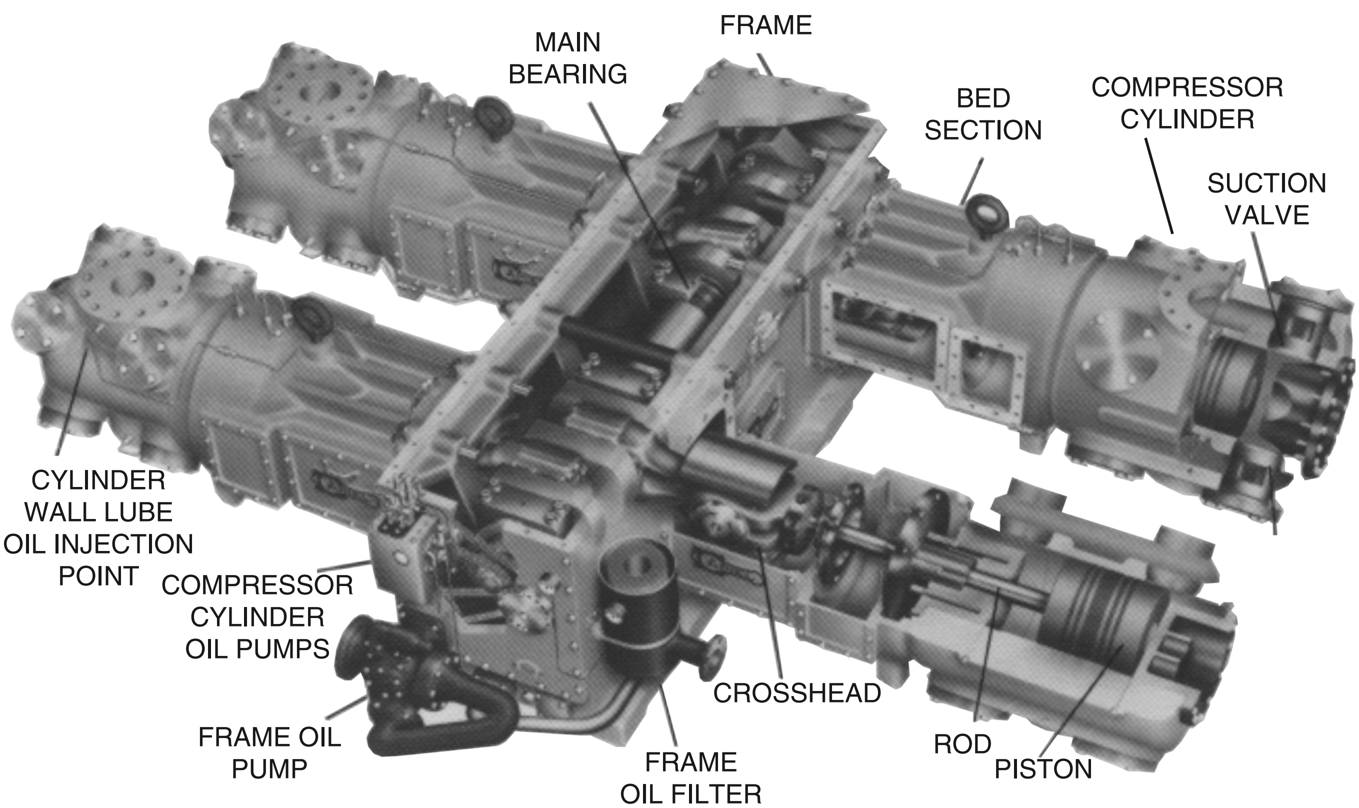
Problem 2

Match each item on the left that describes the item on the right.

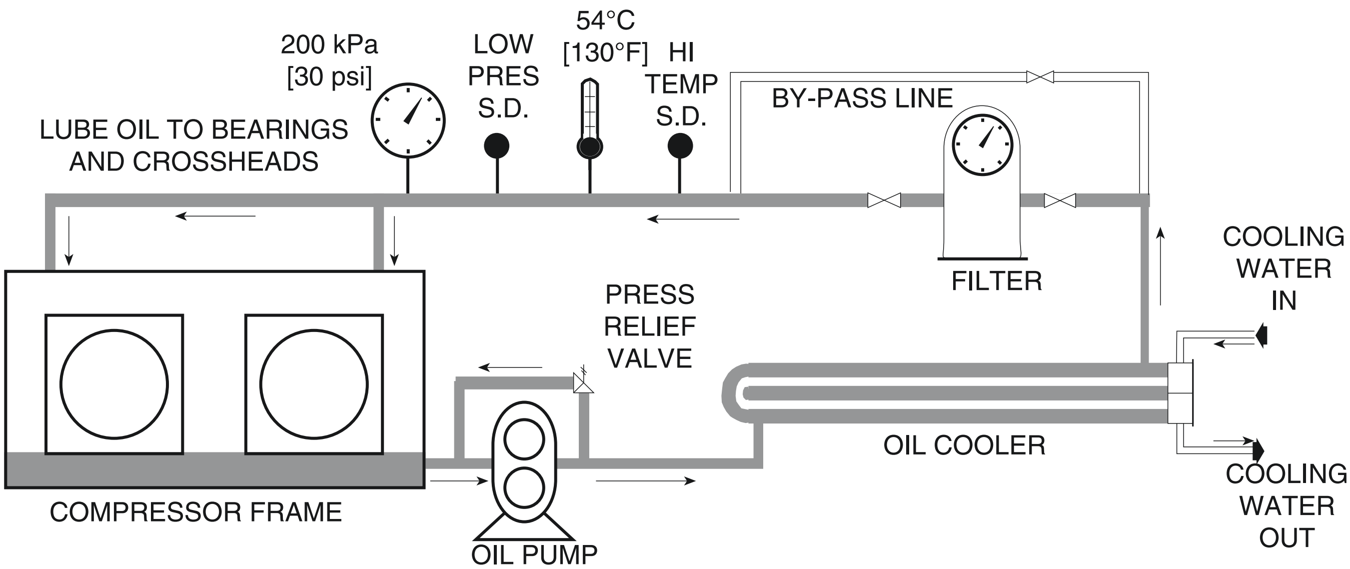
_______ 1.	Rod	a. Suction and discharge.
_______ 2.	Piston	b. Holds rod in straight line.
_______ 3.	Cylinder	c. Attaches to piston and crosshead.
_______ 4.	Valves	d. Often has a liner.
_______ 5.	Packing	e. Lubricates packing and cylinder wall.
_______ 6.	Crosshead	f. Keeps gas from leaking out cylinder.
_______ 7.	Oil system	g. Has grooves for sealing rings.

C. Compressor Frame

The compressor cylinder can be mounted on a frame, or mounted directly to an engine. When it is mounted on an engine, the drive shaft is an integral part of the engine. When compressor cylinders are mounted on a frame, the frame is part of the compressor unit.



CUT AWAY OF FRAME TYPE COMPRESSOR



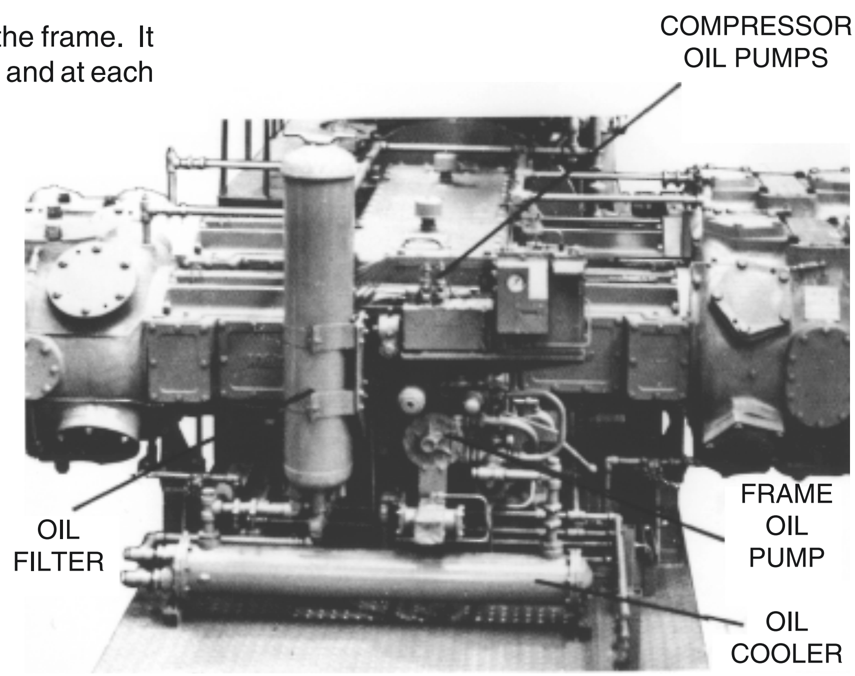
FRAME LUBE OIL SYSTEM

Frames come in a variety of sizes and power ratings. They vary from a low power single cylinder type to ten cylinder frames in excess of 10 000 KW [15 000 hp]. The main parts of the frame are similar for all sizes of units. The frame housing is made of cast iron or steel and contains reinforcing ribs so that it can withstand continual pounding from compressing gas on each stroke. The frame is machined to a smooth flat surface where a compressor cylinder, inspection plate, or other connection attaches to it.

The crankshaft fits in the center of the frame. It is supported with bearings on each end and at each throw. Bearings are installed so that the axis of the crankshaft remains almost perfectly straight while it is rotating in the frame. A slight misalignment can cause expensive shaft failures.

The bottom of the frame is an oil reservoir. An oil pump is usually located on the outboard end of the frame, and is driven by a chain or gears from the crankshaft. The oil pump circulates oil through the cooler, filter, and to each bearing and crosshead as shown above. Oil pressure is usually controlled by adjusting spring tension in a relief valve. The relief valve is often built into the pump body.

The oil system provides lubrication at each metal-to-metal bearing surface and also removes heat from such surfaces. The oil cooler removes heat that the oil picked up in the system. The oil also removes worn metal and dirt from the bearing surfaces. This oil system is for the frame only, and not the compressor cylinders, although makeup oil to the cylinder lubrication pumps may come from the frame oil system.



END VIEW OF COMPRESSOR FRAME

From an operating standpoint, the lubricating oil system is the key to trouble-free operation of the frame. It should be closely observed for proper oil level, oil pressure, and temperature. Typical temperature and pressure are shown in the opposite drawing.

The filter elements should be replaced as they become plugged. A differential pressure gauge (P) is usually included on the filter which indicates its condition. As the filter becomes plugged, more pressure is required to push oil through it. The differential pressure gauge should be observed when new filter elements are installed, and when the pressure drop increases about 70 kPa [10 psi], the elements should be changed. The filter is equipped with a by-pass line in which oil flow is diverted to isolate the filter for changing the elements.

The crankcase oil should be tested periodically for degradation. Degradation is indicated by a change in viscosity of 10% or more, and a rise in the acid content. Checking the oil for metal content is also helpful in anticipating bearing failures. The presence of zinc or aluminum in the lube oil is an indication that bearing surfaces are wearing.

The lube oil system usually has at least two shutdown devices that stop the compressor when they trip:

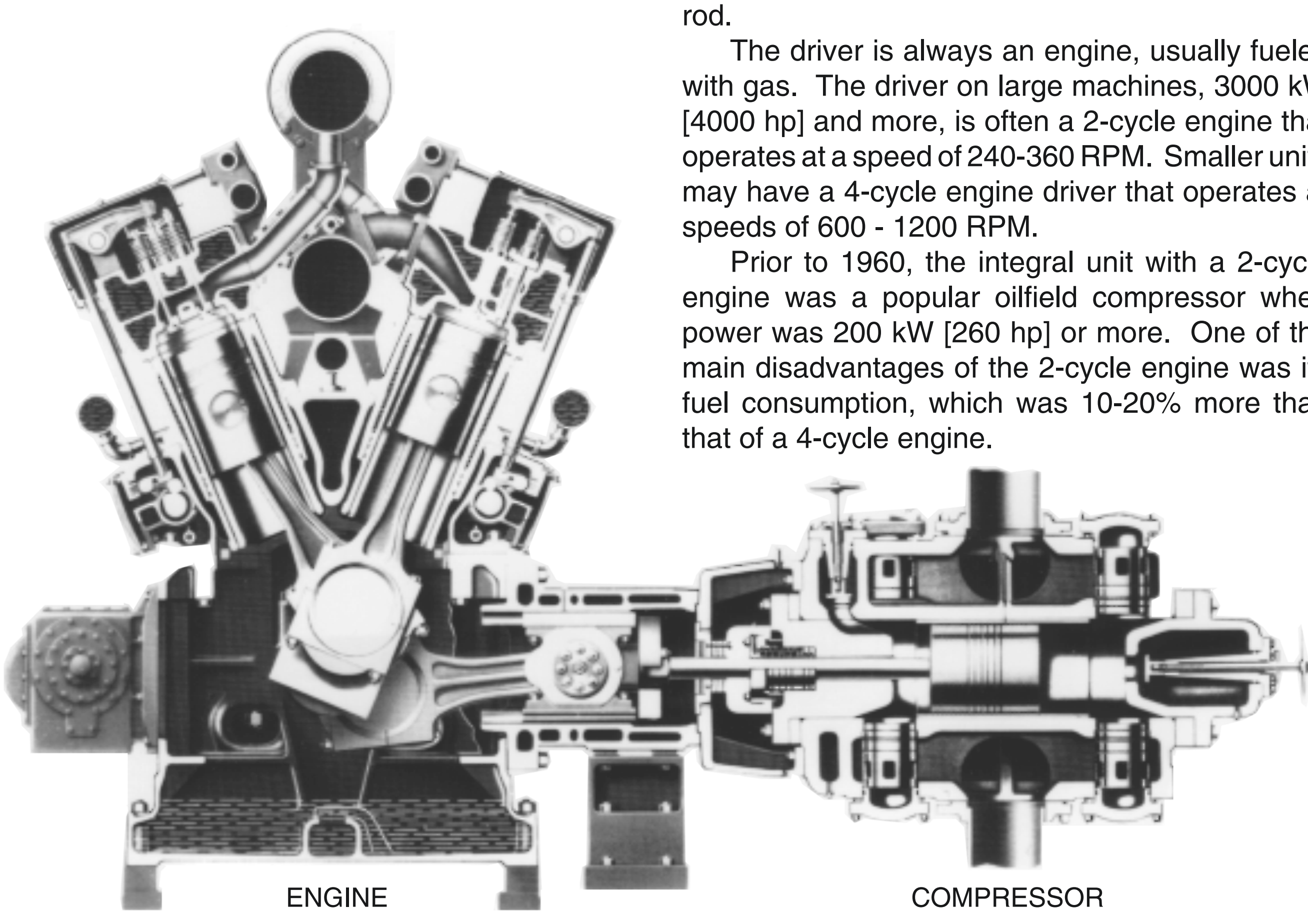
1. Low oil pressure
2. High oil temperature

D. Integral Compressor Unit

On integral compressor units, compressor cylinders are mounted on the crankcase of the driver; and the compressor pistons are connected to the driver crankshaft via a crosshead and connecting rod.

The driver is always an engine, usually fueled with gas. The driver on large machines, 3000 kW [4000 hp] and more, is often a 2-cycle engine that operates at a speed of 240-360 RPM. Smaller units may have a 4-cycle engine driver that operates at speeds of 600 - 1200 RPM.

Prior to 1960, the integral unit with a 2-cycle engine was a popular oilfield compressor when power was 200 kW [260 hp] or more. One of the main disadvantages of the 2-cycle engine was its fuel consumption, which was 10-20% more than that of a 4-cycle engine.



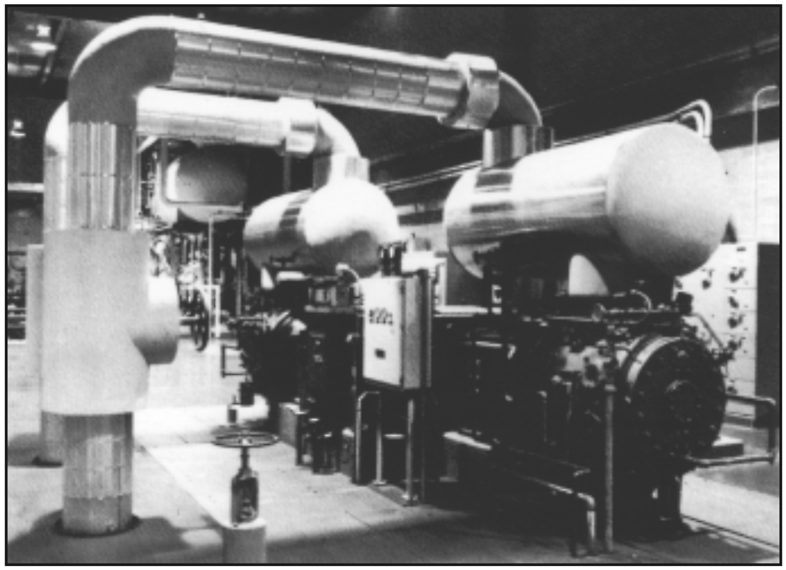
CROSS-SECTION OF INTEGRAL COMPRESSOR UNIT

II. APPLICATION

During the 1950's, industrial gas turbines and improved 4-cycle engines were developed for use with frame type compressors that are less expensive to purchase and operate than integral units.

The main advantage of the integral unit is that it is a rugged machine that will operate 50 or 60 years with proper maintenance. The basic design of a 2-cycle engine requires more fuel than a 4-cycle machine. The difference in fuel consumption of the two can be reduced by adding high compression power cylinders and turbochargers to the 2-cycle engines.

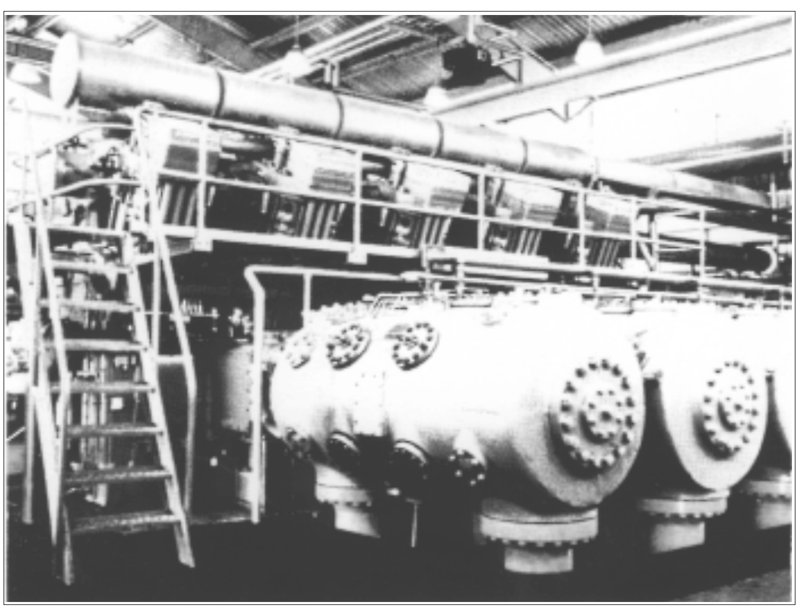
The most common applications of piston-type compressors in the oilfield are those of compressing gas prior to its entry into a pipeline, and supplying high pressure gas to a gas lift system. These compressors vary in size from a 75 kW [100 horsepower] to thousands of kW [hp]. They may have one or two small cylinders, or eight or ten large cylinders.



TWO CYLINDER FRAME TYPE COMPRESSOR USED FOR REFRIGERATION SERVICE

Another frequent use of piston-type compressors is that of refrigeration service in a gas processing plant. In this application, a refrigerant — usually propane — is the gas compressed in the unit.

In some production fields and process plants, gas flows to the compressor from equipment that operates at various pressures. In this situation, gas is compressed in **stages** from the lowest pressure source up to the final pressure required. Each stage of compression is a cylinder or cylinders to compress the gas from one pressure level to the next higher one. It is not unusual to have a single compressor with three or four cylinders, each of which boosts the gas pressure to a successively higher pressure.



THREE CYLINDER INTEGRAL COMPRESSOR USED ON GAS PIPELINE

In gas process plants, compressor cylinders on the same frame may be in entirely different services. For example, a four cylinder machine may use two cylinders in refrigeration service, a third cylinder compressing ethane for delivery into a pipeline, and the fourth cylinder compressing process gas which enters other equipment in the plant.

Instrument air compressors are also a common application of piston-type compressors. These units are often single-acting; they pump gas on the outboard stroke only.

Piston-type compressors are also used in petrochemical plants and ammonia plants to compress various process gases.

III. PRINCIPLES OF COMPRESSION

A. FLOW DESCRIPTION

Gas entering a compressor should not contain solid particles or droplets of liquid. Consequently, a scrubber is provided on the inlet line to remove liquid and solid materials from the gas. The solid and liquid matter drop to the bottom of the suction scrubber and are withdrawn with a level control system to a liquid disposal facility.

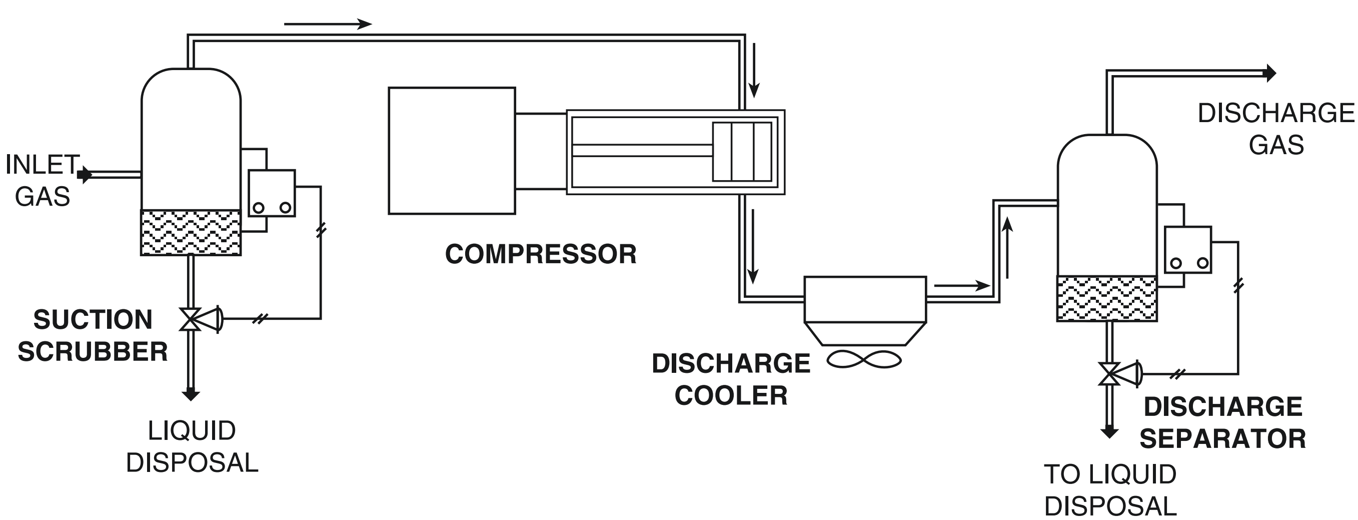
Gas flows out the top of the suction scrubber and enters the inlet side of the compressor. The compressed gas leaving the compressor is usually hot, and normally requires cooling before continuing on its normal flow path. The cooler can be an aerial type as shown below, or it can be a water-cooled exchanger. In either case, some hydrocarbon and water usually condense from the gas as it is cooled. Consequently, a separator is required to remove liquid that condenses in the cooler. The liquid falls to the bottom and is withdrawn to a liquid disposal facility with a level control system. Gas leaving the discharge separator then flows to process equipment or a pipeline.

Quite frequently, the compressor receives gas at a rather low pressure, and compresses it to a higher pressure. This may take two or three stages of compression to boost the pressure. The drawing on the next page shows the flow path for a 2-stage compression system. Flow is as follows:

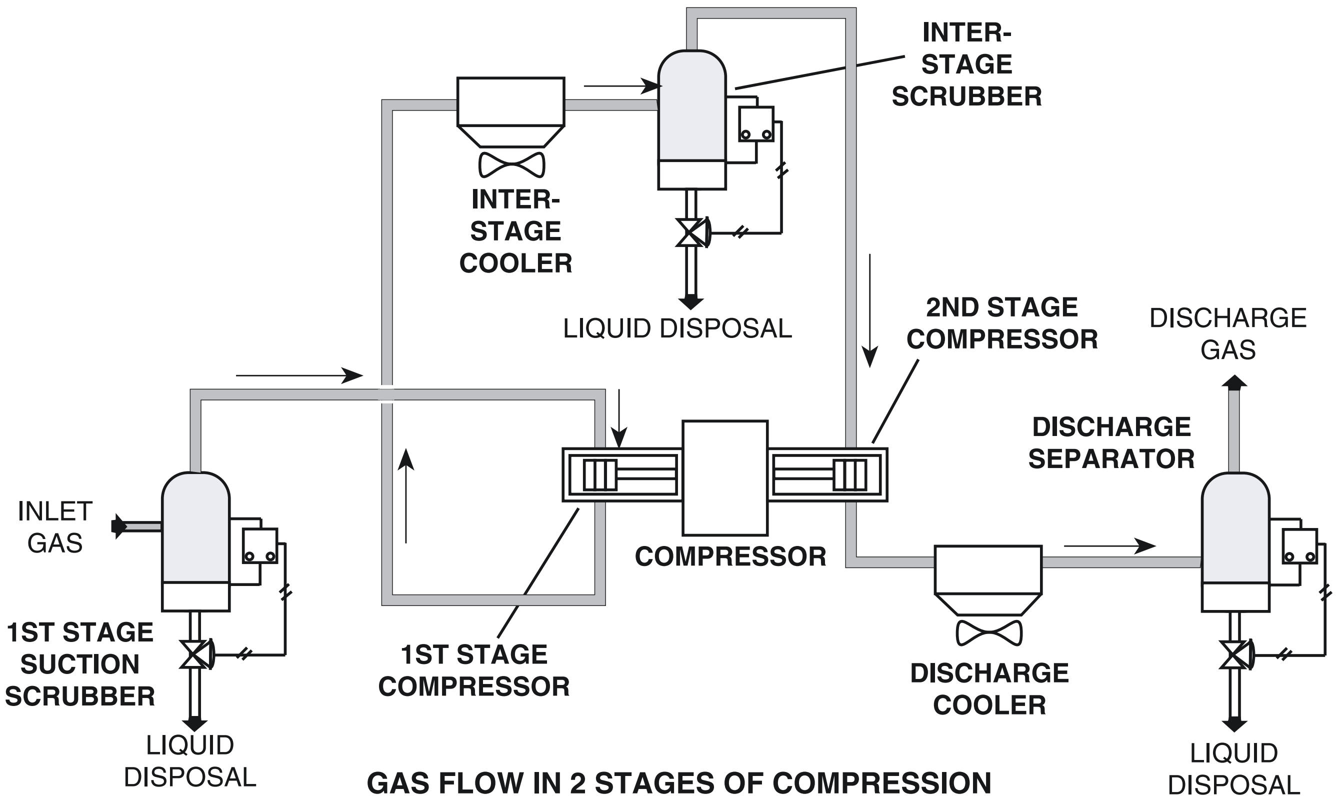
Inlet gas passes through a scrubber to remove liquid and solid materials, which are withdrawn at the bottom. Gas leaves the top of the scrubber and enters the 1st stage compressor, which boosts its pressure about half of the total required. Hot gas leaves the 1st stage cylinder and flows through a cooler and scrubber. Liquid which condenses as the 1st stage gas is cooled falls to the bottom of the interstage scrubber and is withdrawn with a level control system. Gas leaves the interstage scrubber and flows to the 2nd stage compressor cylinder which boosts its pressure to the desired point. Hot gas leaving the 2nd stage compressor is cooled and flows through a discharge scrubber to remove liquid that condenses in the discharge cooler. Gas leaving the discharge scrubber then flows to its normal disposition.

In some situations, 3 or more compression stages are required to boost the gas pressure to the required level. Each stage will usually have a cooler and scrubber on the discharge gas.

The gas scrubbers used in the suction and discharge lines can be either vertical or horizontal, providing they are properly sized for the volume of gas flowing.



TYPICAL GAS FLOW THROUGH A COMPRESSOR



GAS FLOW IN 2 STAGES OF COMPRESSION

B. Theory of Positive Displacement Compression

Before continuing our discussion on compression, let us look at the compressor unit. It consists of a driver and one or more compressor cylinders. The unit is an **energy transfer device**. Energy consumed by the driver is transferred to gas inside the compressor cylinders in the form of pressure and temperature. The driver may be an electric motor, an engine or a turbine using gas or oil for fuel.

Regardless of the type of driver, it consumes energy from electricity or by burning fuel. The energy consumed by the driver is transferred by the compressor to **pressure** and **temperature** energy in the gas.

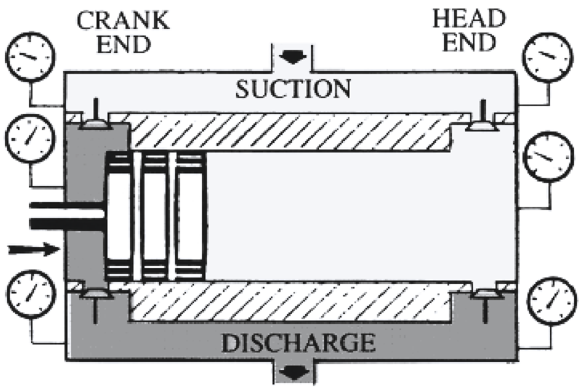
It is important we recognize that the compressor is an energy transfer device in order to understand how the compressor functions.

The main purpose of the compressor is to raise gas pressure. Ideally, the compressor would transfer driver energy into pressure energy in the gas. However, it is one of Nature's laws that gas temperature will rise as it is compressed.

Another way of looking at the energy transfer concept is that driver energy has to go somewhere, and its only outlet is into the gas. It enters the gas in two forms: pressure and heat.

Positive displacement compression is simply squeezing gas to a smaller volume with a piston, meshing screw, or other mechanical device. A piston-type compressor is most commonly used in the oilfield.

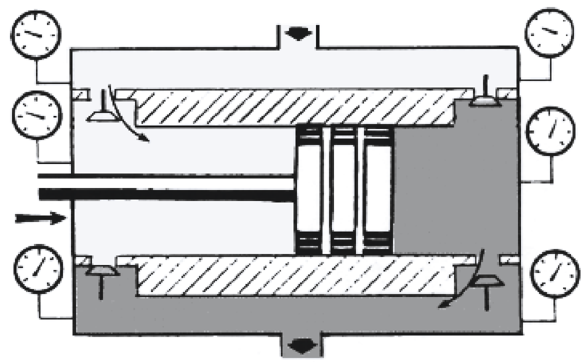
A piston-type compressor has two strokes: the **intake** or **suction** stroke in which gas at suction pressure enters the cylinder; and a **compression** stroke in which the gas is compressed and flows out the cylinder. The strokes are explained in the drawing on the next page.

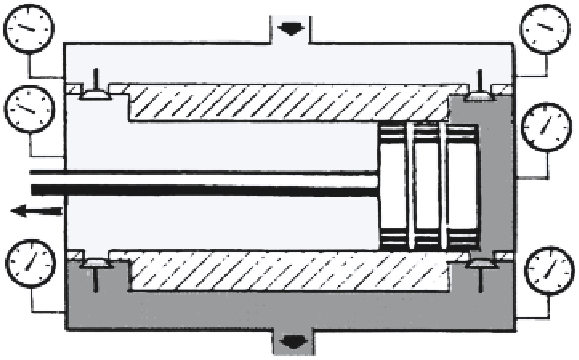


Piston is at the end of its crank end stroke and is ready to move toward the Head End. Discharge pressure is on the crank end of the piston, and the cylinder on the head end of the piston is full of gas at suction pressure. All valves are closed.

When piston has traveled about 2/3 rds of its Head End stroke, pressure in the Head End reaches discharge pressure, and the discharge valve opens. Gas flows out of the cylinder for the remainder of the stroke.

Pressure in the Crank End drops as soon as the piston begins its travel toward the Head End. After it travels a short distance, pressure on the Crank End falls to suction pressure, and the suction valve opens and suction gas flows into the cylinder. Flow continues for the remainder of the Head End stroke.

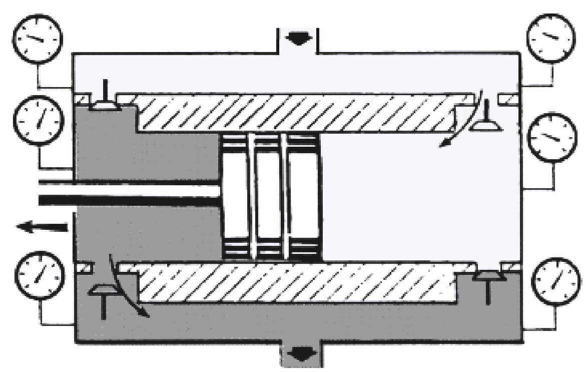




When piston reaches the end of its Head End stroke, gas in head end is at discharge pressure, and the Crank End is full of gas at suction pressure. All valves are closed. Piston begins its travel toward crank end.

When piston travels about 2/3 rds of its stroke toward the Crank End, pressure in the Crank End reaches discharge pressure. Discharge valve opens and gas flows out of the cylinder for the remainder of the stroke.

Pressure in the Head End drops as soon as the piston begins its travel toward the Crank End. After it travels a short distance, pressure on the Head End falls to suction pressure, and the suction valve opens. Gas flows into the cylinder for the remainder of the stroke.



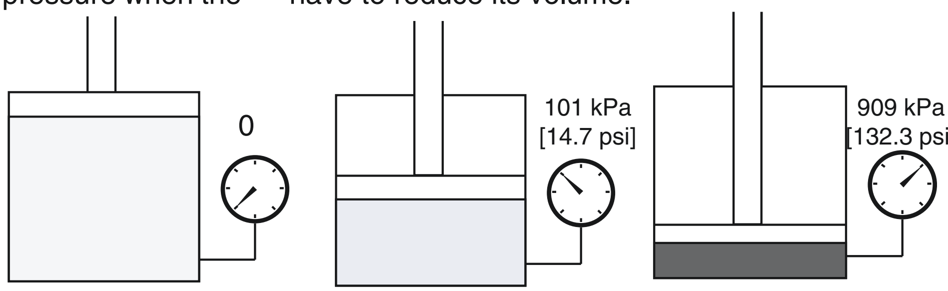
C. Effect of Change in Gas Volume on Pressure

The purpose of a compressor is to increase the pressure of gas. Gas pressure is increased by squeezing it together to reduce its volume. When its volume is reduced, its pressure rises. This can be visualized by thinking of a room that has a piston for a roof. The room contains air at normal atmospheric pressure. If the roof is lowered halfway down the wall, and it is sealed so that no air escapes, the volume of air will be half of what it was when the roof was all the way at the top. The same weight of air is contained in the room, but its volume has been reduced by one half. If a pressure gauge was installed on the room, it would read 101 kPa [14.7 psi] of pressure.

The gauge would read zero pressure when the roof was at the top. However, atmospheric pressure — 101 kPa [14.7 psi] — was present in the room when the roof was at the top. The absolute pressure inside the room was 101 kPa [14.7 psi].

When the roof was moved halfway down, the absolute pressure inside increased from 101 kPa to 202 kPa [14.7 psi to 29.4 psi]. In other words, as the air was compressed to half of its original volume, the absolute pressure increased by a factor of two.

Suppose the roof is lowered until the volume remaining is one tenth of the original. At this point, the absolute pressure will be ten times atmospheric pressure, or 10 x 101 = 1010 kPa(a) [10 x 14.7 = 147 psia]. To increase the pressure of gas, we simply have to reduce its volume.

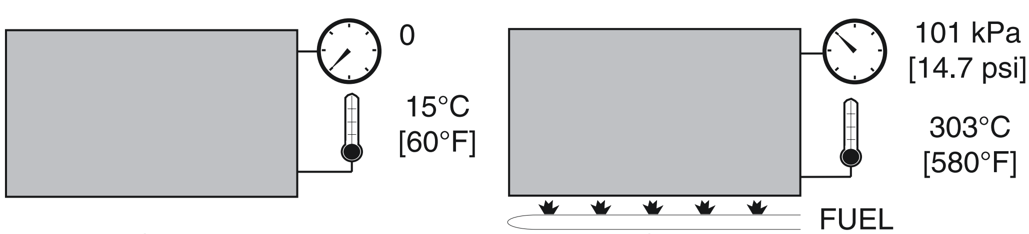


Gas Volume	100%	50%	10%
Pressure Gauge Reading	0 0	101 kPa [14.7 psi]	909 kPa [132.3 psi]
Atmospheric Pressure	+ 101 kPa [14.7 psi]	+ 101 kPa [14.7 psi]	+ 101 kPa [14.7 psi]
Absolute Pressure	101 kPa [14.7 psi]	202 kPa [29.4 psi]	1010 kPa [147.0 psi]
Relative Volume	1	1/2	1/10th
Relative Pressure	1	2	10

GAS ABSOLUTE PRESSURE INCREASES AS VOLUME IS REDUCED

D. Effect of Temperature on Pressure

When gas in a confined chamber is heated, its pressure rises. The amount of pressure rise is proportional to the increase in **absolute** temperature. In other words, if gas absolute temperature is doubled, its absolute pressure will also double.



Gas Temperature	15°C [60°F]	303°C [580°F]
Absolute Zero	+ 273 [460]	+ 273 [460]
Absolute Temperature	288 [520]	576 [1040]
Absolute Pressure	101 kPa (a) [14.7 psia]	202 kPa (a) [29.4 psia]

ABSOLUTE PRESSURE DOUBLES WHEN ABSOLUTE TEMPERATURE DOUBLES

When we talk about increasing or reducing the gas volume, the question arises: at what volume do we start? The answer to this question is: at **standard conditions**. Standard conditions of gas volume are **atmospheric pressure** at sea level, and a temperature of **15°C [60°F]**. Standard volume of measurement is 1 m^3 or 1,000 cu ft (1 Mcf).

When we read a flow meter that shows the gas flow rate through a pipe, we are actually reading the **theoretical** volume of the gas if it was flowing at atmospheric pressure (0 gauge pressure) and 15°C [60°F].

Gas seldom flows at atmospheric pressure and 15°C [60°F]. The pressure is almost always considerably higher than this. However, we convert the volume at its flowing pressure to the volume it would occupy at standard conditions.

Remember: When we speak of gas flow measurement, we are referring to the volume the gas would occupy if its pressure was reduced to atmospheric and its temperature is 15°C [60°F].

Since gas pressure is almost always above atmospheric, the **actual** volume of gas at operating pressure will be less than that shown on the gas flow meter.

We can calculate the actual volume of gas at flowing temperature and pressure from the following equations:

		SI UNITS	ENGLISH UNITS
Actual Gas Volume	=	$\frac{0.35 x (Std\ Vol) x (273+T)}{P + 100}$	$\frac{0.028 x (Std\ Vol) x (460+T)}{P + 15}$
Where,			
T	=	Temperature °C	Temperature, °F
P	=	Gauge Pressure, kPa	Gauge Pressure, psi

Examples

1. Gas flowing at a rate of 1.3 million m^3/d [50 MMcf/d] enters a compressor at a pressure of 2700 kPa [391 psi] and a temperature of 40°C [104°F]. What is the actual volume of gas flowing to the compressor?

	SI UNITS	ENGLISH UNITS
Flowing Temperature, T	*40°C*	*104°F*
Flowing Pressure, P	*2700 kPa*	*391 psi*
Flowing Volume, Std.	*1.3 million m^3/d*	*50 MMcf/d*
Substitute in Equation	$\frac{0.35x(1.3)x(273+40)}{2700 + 100}$	$\frac{0.028x(50)x(460+104)}{391 + 15}$
Actual Volume	*0.051 million m^3/d or 51 000 m^3/d (act)*	*1.94 MMcf/d (act)*

2. The gas is compressed to a pressure of 6900 kPa [1000 psi], and the compressor discharge temperature is 145°C [293°F]. What is the actual volume leaving the compressor?

Flowing Temperature	*145°C*	*293°F*
Flowing Pressure	*6900 kPa*	*1000 psi*
Flowing Volume, Std	*1.3 million m^3/d*	*50 MMcf/d*
Subst. in Equation	$\frac{0.35x(1.3)x(273+145)}{6900 + 100}$	$\frac{0.028x(50)x(460+293)}{1000 + 15}$
Actual Volume	*0.027 million m^3/d or 27 000 m^3/d (act)*	*1.04 MMcf/d (act)*

Problem 3

Gas flows to a compressor at a rate of 1 100 000 m^3/d [40 MMcf/d]. The suction pressure is 2100 kPa [305 psi] and its temperature is 32°C [90°F]. What is the actual volume of gas flowing? ____________

From the previous example, we see that the actual volume of gas flowing to the compressor was 51 000 m^3/d [1.95 MMcf/d]. A flow meter, on the suction line, would read the standard volume of gas or 1.3 million m^3/d [50 MMcf/d]. The actual volume of gas leaving the compressor was 27 000 m^3/d [1.04 MMcd/d]. A flow meter on the discharge line would also read 1.3 million m^3/d [50 MMcf/d].

Another way of looking at this is that if we design our compressor to reduce the volume from its suction volume to its discharge volume, then its discharge pressure will be 6900 kPa [1000 psi].

The actual gas volume is used for sizing piping, control valves, vessels, and other process equipment. One of its most important applications is that of sizing compressor cylinders. The compressor diameter is selected so that the displacement equals the actual volume of gas entering the compressor.

F. Compressor Cylinder Displacement

The capacity of the cylinder will depend upon its **displacement**. The displacement is simply the volume that the piston displaces as it moves from one end of the cylinder to the other. It is equal to the cross-sectional area of the cylinder times the length of the stroke. This volume is displaced on each stroke of the piston. A unit operating at 300 RPM would have 300 displacements per minute.

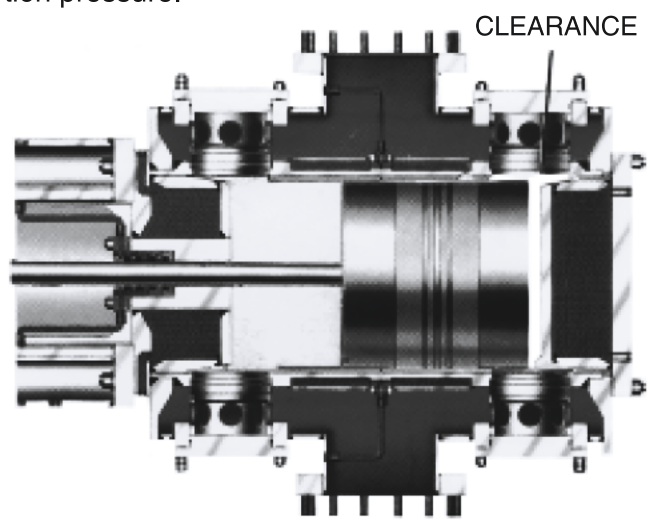
Since most pistons displace on both the inboard and outboard stroke, the total capacity is the displacement multiplied by two. However, the volume occupied by the rod must be subtracted from the displacement on the inboard stroke to get the actual displacement on the inboard stroke.

The displacement of various sizes of double-acting compressor cylinders is shown on the graphs on the pages 30-33.

The displacement shown in the graphs is the **actual** volume of gas that the compressor will handle at suction conditions. It is not the volume at standard pressure and temperature. In the examples we worked on page 27, the standard volume of gas flowing to the compressor was 1.3 million m^3/d [50 MMcf/d]. The actual volume at suction pressure was 51 000 m^3/d [1.94 MMcf/d]. We would design our compressor cylinder for a displacement equal to the **actual** volume entering at suction temperature and pressure.

G. Volumetric Efficiency and Clearance

When the piston reaches the end of its stroke, there is a small volume in front of the piston and around valves that is not swept by the piston. This volume is called **clearance**; it is 4 - 15% of the total volume swept by the piston. The gas in the clearance volume is at **discharge pressure**. No gas will flow into the cylinder on the suction stroke until pressure inside the cylinder is slightly below suction pressure. This will require movement of the piston so that the clearance gas can expand. The amount of piston movement required for clearance gas pressure to drop to suction pressure equals the clearance volume times the compression ratio. For example, a cylinder having a clearance volume of 4%, and a compression ratio of 2.5 would require a piston movement of : 4% x 2.5 = 10% of its stroke before the clearance gas pressure dropped to suction pressure.



CUTAWAY OF CYLINDER WITH PISTON AT THE END OF ITS OUTBOARD STROKE

This piston movement represents a reduction in the capacity of the compressor. The **volumetric efficiency** is 100% less the % of piston movement required for clearance gas pressure to drop to suction pressure. In the previous example, the volumetric efficiency is 100 -10= 90%.

Clearance is a loss of compressor capacity, so it is usually less than 20% of compressor displacement. The volumetric efficiency includes loss due to clearance.

Although clearance results in a loss of capacity, it does not result in a significant loss of driver power. Most of the energy used to compress the clearance gas transfers to the gas, and it is recovered when the piston reverses its stroke. The high pressure clearance gas pushes the piston until its pressure equals suction pressure.

As noted earlier, the actual gas volume at suction temperature and pressure are used to determine the displacement and diameter of compressor cylinders. In order to determine the capacity of a cylinder, we must convert actual displacement volume to standard volume. The formulas are:

	SI UNITS	ENGLISH UNITS
Actual Gas Volume, Va =	$\frac{0.35\,(\text{Std Vol})(273+°C)}{(P + 100)}$	$\frac{0.028\,(\text{Std Vol})(460+°F)}{(P + 15)}$

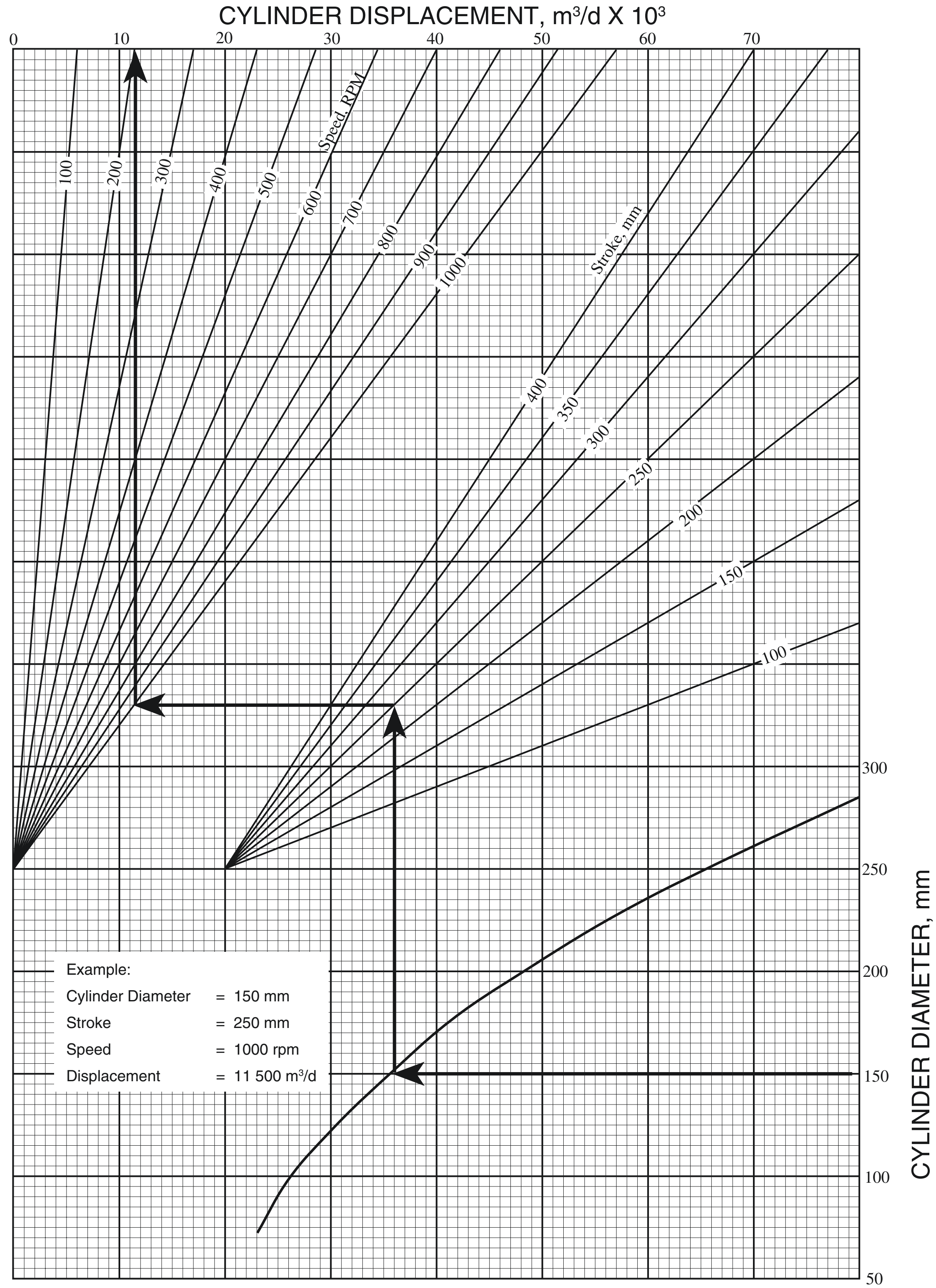
Rearranging the equation, the Standard Volume equals:

	SI UNITS	ENGLISH UNITS
Standard Volume, V_S =	$\frac{2.84\,(\text{Act Vol})(P+100)}{(273 + °C)}$	$\frac{35.4\,(\text{Act Vol})(P+15)}{(460 + °F)}$

Example:

Determine the standard volume capacity of the following 3-cylinder compressor which has a volumetric efficiency of 90%.

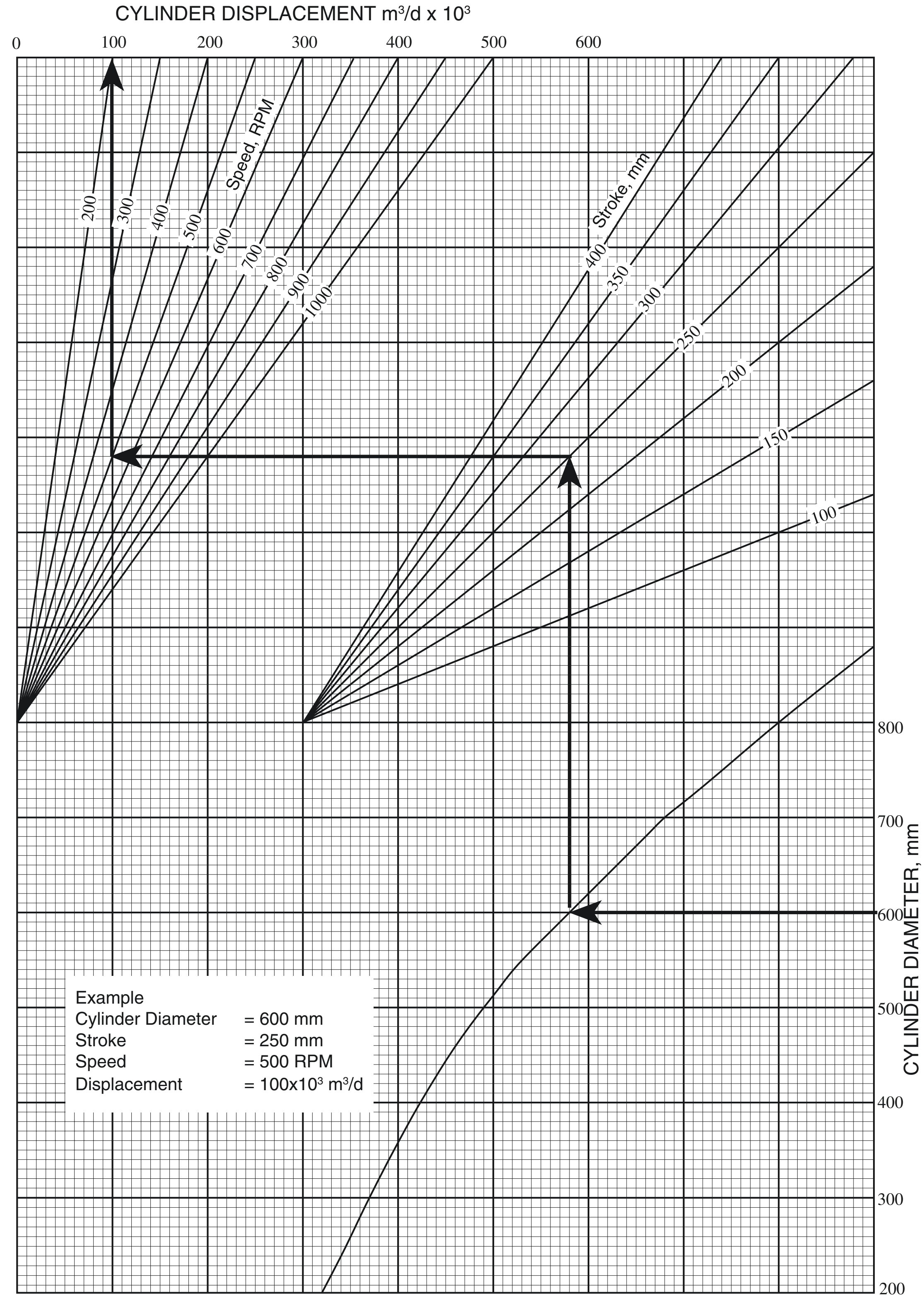
	SI UNITS	ENGLISH UNITS
Cylinder diameter	200 mm	8 in
Stroke/Speed	300 mm/800 RPM	12 in/800 RPM
Volumetric Efficiency	90%	90%
Suction Pressure/Temperature	2700 kPa/40°C	391 psi/104°F
Theoretical Cylinder Displacement, from Fig 1A & B	19 500 m^3/d	760 Mcf/d or 0.76 MMcf/d
Cylinder Displacement at 90% Efficiency	$19\,500 \times \frac{90\%}{100}$	$0.76 \times \frac{90\%}{100}$
	= 17 550 m^3/d	0.684 MMcf/d
Standard Volume Equation	$\frac{2.84(\text{ActVol})(P+100)}{(273 + °C)}$	$\frac{35.4(\text{Act Vol})(P+15)}{(460 + °F)}$
Substitute in Equation, V_s	$\frac{2.84(17\,550)(2\,700+100)}{(273 + 38)}$	$\frac{35.4(0.684)(391+15)}{(460 + 104)}$
Cylinder capacity, Std Vol	446 000 m^3/d	= 17.43 MMscf/d
Capacity of 3 cylinders	446 000 x 3	17.43 x 3
	= 1 338 000 m^3/d	= 52.29 MMcf/d



COMPRESSOR CYLINDER DISPLACEMENT CURVES FOR SMALL DIAMETER CYLINDERS - SI UNITS

Figure 1A

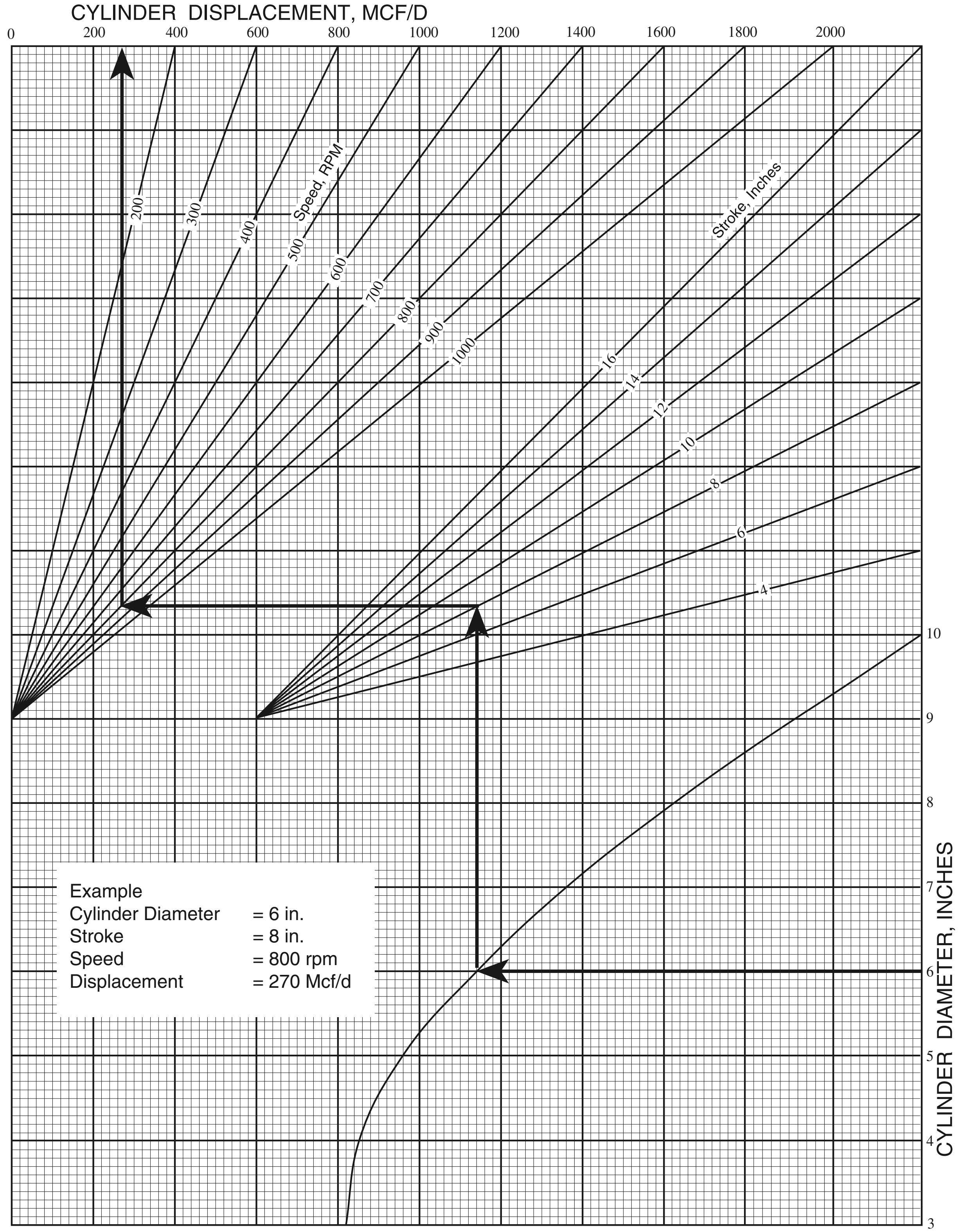
LARGE DIAMETER COMPRESSOR CYLINDER DISPLACEMENT - SI UNITS



COMPRESSOR CYLINDER DISPLACEMENT CURVES FOR LARGE DIAMETER CYLINDERS - SI UNITS

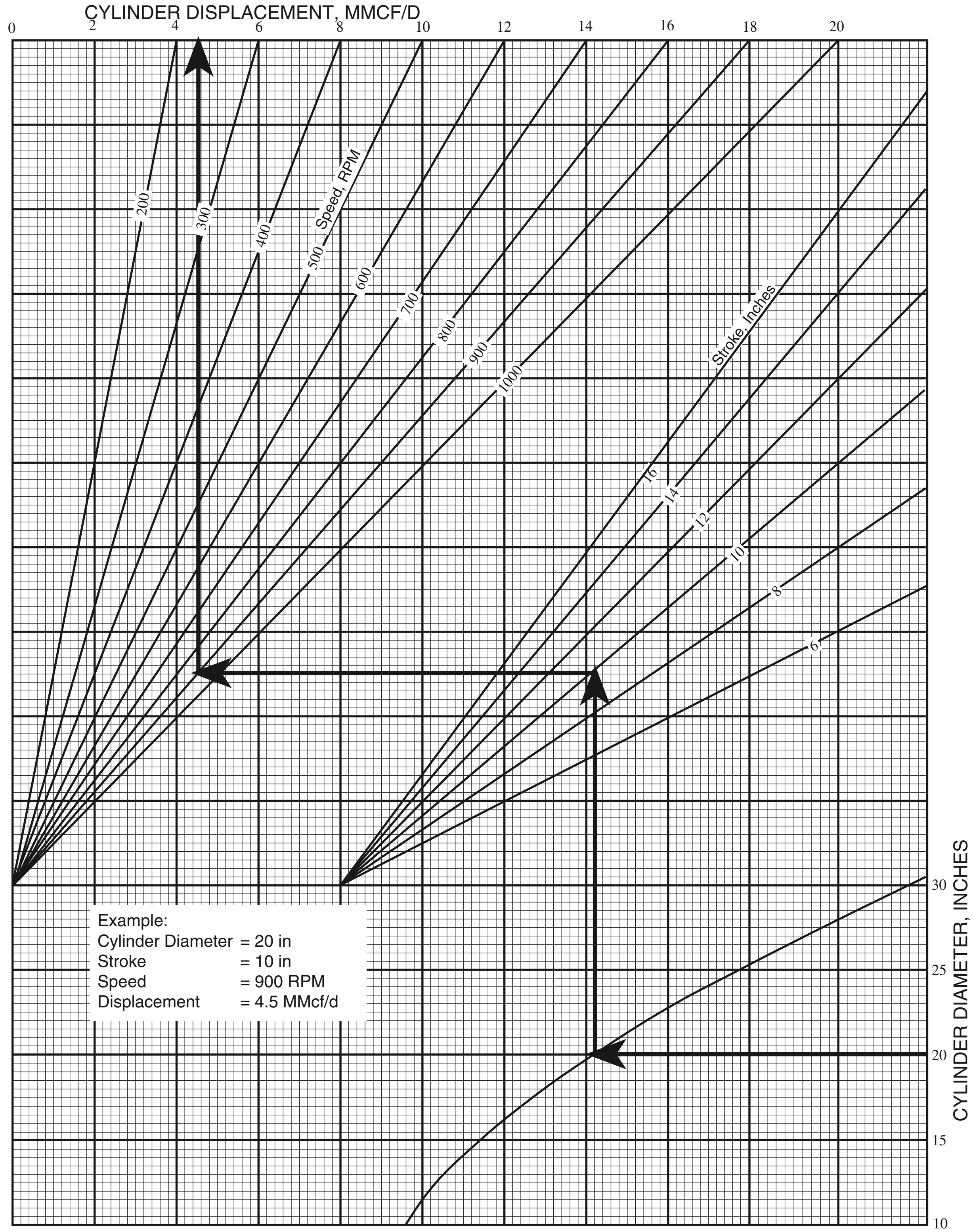
Figure 2A

SMALL DIA. COMPRESSOR CYLINDER DISPLACEMENT — ENGLISH UNITS



COMPRESSOR CYLINDER DISPLACEMENT CURVES FOR SMALL DIAMETER CYLINDERS - ENGLISH UNITS

Figure 1B



COMPRESSOR CYLINDER DISPLACEMENT CURVES FOR LARGE DIAMETER CYLINDERS - ENGLISH UNITS

Figure 2B

Problem 4

A compressor cylinder is 250 mm [10 in] in diameter and the piston stroke is 200 mm [8 in]. Its speed is 600 RPM. The suction pressure gauge reads 2100 kPa [305 psi] and its temperature is 32°C [90°F]. Compressor efficiency is 90%. What is the displacement and capacity of the cylinder in standard volume?

Cylinder Displacement____________________ Capacity __________________________

H. Compression Ratio

One of the important factors in the design and operation of a compressor is that of its **compression ratio**. It equals:

Compression Ratio, C. R. = $\frac{\text{Absolute Discharge Pressure}}{\text{Absolute Suction Pressure}}$

Absolute Pressure = Gauge Pressure + Atmospheric Pressure

		SI UNITS	ENGLISH UNITS
Compression Ratio	=	$\frac{\text{(Discharge Pres, kPa)+100}}{\text{(Suction Pres, kPa)+100}}$	$\frac{\text{(Discharge Pres, psig)+15}}{\text{(Suction Pres, psig)+15}}$

Atmospheric pressure at sea level is 101 kPa(a) [14.7 psia]. To simplify the calculations, we round it off to 100 kPa [15 psi]. At elevations above 600m [2000 ft] atmospheric pressure at the higher elevation should be used instead of using 100 kPa [15 psi].

The compression ratio is used to estimate the temperature rise in the compressor and the compressor power required. The table on the next page shows power and temperature rise at different compression ratios. The temperature rise is the difference between discharge temperature and suction temperature. In other words, it must be added to suction temperature in order to calculate the discharge temperature.

The power and temperature rise shown are based on average conditions of gas composition, pressure and temperature. Their accuracy is 90% or better for most natural gas compressors.

The table is useful in estimating the driver load and the discharge temperatures of compressor cylinders on an operating machine.

Example

Gas enters a compressor at 2700 kPa and 40°C [391 psi and 104°F]. Discharge pressure is 6900 kPa [1000 psi]. Flow rate is 1.34 million^3/d [52.29 MMcf/d]. Determine the discharge temperature and power required.

	SI UNITS	*ENGLISH UNITS*
Absolute Discharge Pressure	*6900+100 = 7000 kPa(a)*	*1000+15 + 1015 psia*
Absolute Suction Pressure	*2700+100 = 2800 kPa(a)*	*391+15 = 406 psia*
Compression Ratio	$\frac{7000}{2800} = 2.5$	$\frac{1015}{406} = 2.5$
Temperature Rise @ 2.5 C.R. (From Table)	*65°C*	*117°F*
Suction Temperature	*+40°C*	*+104°F*
Calculated Discharge Temperature.	*105°C*	*221°F*

COMPRESSOR POWER REQUIREMENTS AND TEMPERATURE RISE AT VARIOUS COMPRESSION RATIOS

COMPRESSION RATIO	POWER REQUIRED		TEMPERATURE RISE	
	kW/million m^3/d	HP/MMcf/d	°C	°F
1.5	690	26	28	50
1.6	790	30	32	58
1.7	900	34	36	65
1.8	970	37	40	72
1.9	1050	40	44	79
2.0	1130	43	48	86
2.1	1210	46	52	93
2.2	1260	48	55	99
2.3	1320	50	58	105
2.4	1400	53	62	111
2.5	1480	56	65	117
2.6	1530	58	68	123
2.7	1580	60	71	128
2.8	1660	63	74	134
2.9	1720	65	77	139
3.0	1790	68	80	144
3.1	1840	70	83	149
3.2	1900	72	85	153
3.3	1950	74	88	158
3.4	2000	76	90	162
3.5	2060	78	93	167
3.6	2110	80	95	171
3.7	2160	82	97	175
3.8	2210	84	99	178
3.9	2240	85	101	182
4.0	2290	87	104	186
4.1	2360	89	106	190
4.2	2410	91	108	194
4.3	2440	92	110	197
4.4	2490	94	112	201
4.5	2540	96	114	205
4.6	2570	97	116	208
4.7	2600	98	117	211
4.8	2640	100	119	214
4.9	2700	102	121	217
5.0	2730	103	122	220
5.1	2750	104	124	223
5.2	2810	106	126	226
5.3	2830	107	127	229
5.4	2860	108	129	232
5.5	2910	110	131	235
5.6	2930	111	132	238
5.7	2950	112	134	241
5.8	3000	114	136	244
5.9	3030	115	137	247
6.0	3060	116	139	250

Power Required/Unit Volume @ 2.5 C.R.	*1480 kW/million m^3/d*	*56 hp/MMcf/d*
Total Gas Volume	*1.34 million m^3/d*	*52.29 MMscf/d*
Total Power Required	*(1.34) (1480) = 1983 kW*	*52.29 x 56 = 2928 hp*

We compare the calculated power requirement with the power rating of the driver to determine whether the driver is overloaded. Comparing the calculated discharge temperature with the actual gives an indication of the condition of the compressor…a temperature above the calculated temperature indicates leaking valves or piston rings.

Problem 5

Gas enters a compressor at 32°C and 2100 kPa [90°F and 305 psig]. Discharge pressure is 6500 kPa [945 psig]. Flow rate is 1.1 million m^3/d (Std) [40 MMcf/d]. Calculate the following:

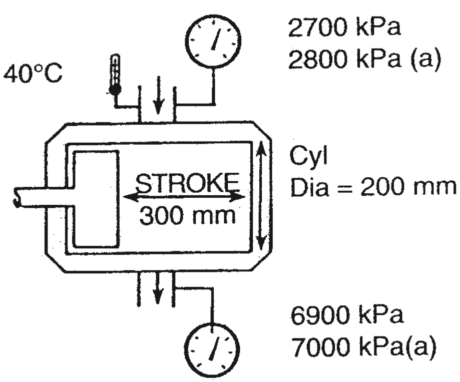
a. Compression Ratio _______

b Discharge Temperature _______

c. Power Required _______

Let us review how the theoretical discussion we covered applies to compressor operation. The compressor cylinder we sized in previous examples is shown below.

SI UNITS



Speed = 800 rpm

Vol Eff = 90%

Capacity 1-Cylinder = 446 000 m^3/d (std)

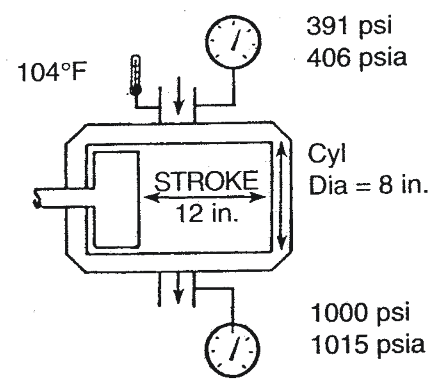
Capacity 3-Cylinders = 1.34 million m^3/d

Compression Ratio = $\frac{7000}{2800}$ = 2.5

Unit Power = 1480 kw/million m^3/d

Total Power = 1480 x 1.34 = 1983 kw

ENGLISH UNITS



Speed = 800 rpm

Vol Eff = 90%

Capacity 1-Cylinder = 17.43 MMscf/d

Capacity 3-Cylinders = 52.29 MMscf/d

Compression Ratio = $\frac{1015}{406}$ = 2.5

Unit Power = 56 hp/MMcf/d

Total Power = 56 x 52.29 = 2928 hp

The design capacity of the 3-cylinder compressor is 1.34 million m^3/d [52.29 MMcf/d]. Suppose in actual operation, the flow drops until suction pressure is only 1900 kPa [275 psi]. Discharge pressure and speed remain the same. At the lower suction pressure, we calculate the capacity, compression ratio, and power required and get the following:

	SI UNITS	ENGLISH UNITS
Suction Pressure, gauge	1900 kPa	275 psi
Capacity, standard volume	954 000 m^3/d	37.4 MMcf/d
Compression ratio	3.5	3.5
Total power required	1971 kW	2917 hp

Suppose the gas flow rate increases until suction pressure reaches 3400 kPa [492 psi]. Calculating capacity and power at the higher suction pressure gives the following:

	SI UNITS	ENGLISH UNITS
Suction Pressure, gauge	3400 kPa	492 psi
Capacity, standard volume	1 671 000 m^3/d	65.31 MMcf/d
Compression ratio	2.0	2.0
Total power required	1888 kW	2808

Summary of the above:

SI UNITS

Suction Pres kPa	Compr Ratio	Cylinder Capacity m^3/d(std)	Cylinder Capacity % of Design	Power Reqr'd kW	Power Reqr'd % of Design
1 900	3.50	954 000	71	1 965	99
2 700	2.50	1 340 000	100	1 983	100
3 400	2.0	1 671 000	125	1 888	95

Design

ENGLISH UNITS

Suction Pres psig	Compr Ratio	Cylinder Capacity MMcf/d	Cylinder Capacity % of Design	Power Reqr'd hp	Power Reqr'd % of Design
275	3.50	37.35	71	2 913	99
391	2.50	52.29	100	2 928	100
492	2.0	65.31	125	2 808	96

Look at the above figures for a moment. First of all, check the power required at the different suction pressure conditions. It remains fairly constant over a wide range of suction pressures. The cylinder capacity at the low suction pressure condition is 71% of its design capacity. Suppose we slow the unit down to 71%. The cylinder will handle the volume of gas available to it, but the suction pressure will rise to its design point. By slowing the unit to 71% speed, the power requirement will be only 71% of its design, or 1408 kW [2079 hp]. This compares with a power requirement of 1965 kW [2913 hp] if the unit operates at 100% speed. The power difference between 100% speed and 71% speed requires additional fuel to the driver that costs about 150,000 U.S. dollars a year.

The point is that compressors should always run at the **lowest** speed required to compress the gas. When the gas flow rate to the compressor drops, the speed should be reduced to maintain a constant suction pressure, if speed control is possible.

We can summarize what we have calculated as follows:

1. Power required to drive a compressor at constant speed is almost constant at any suction pressure.
2. The capacity of a compressor is directly related to the suction pressure.
3. Significant fuel savings result from reducing compressor speed when the gas rate is below design.

The preceding discussion assumes that the discharge pressure remains constant. This is a valid assumption because in actual operation, discharge pressure is usually fairly constant.

A change in the discharge pressure has very little effect on the cylinder capacity. It may change the volumetric efficiency a few percentage points. Its main effect is on the compression ratio, which in turn affects the power required to drive the compressor.

I. Pulsation

Most piston-type compressors operate at speeds of 250 to 1200 revolutions per minute. On each stroke, the compressor must fill with gas on the suction stroke, and compress the gas and discharge it on the discharge stroke.

If the cylinder does not completely fill with gas on the suction stroke, it obviously will not compress its design volume of gas. A compressor operating at 300 rpm makes five cycles each second. A cycle includes a suction stroke and a discharge stroke. The suction stroke lasts a tenth of a second. In other words, gas has a tenth of a second to flow from the suction piping through the suction valve into the compressor cylinder. A slight restriction in the suction piping, suction passageway in the cylinder, or the suction valves can prevent the cylinder from filling with gas on the suction stroke. This reduces the capacity of the cylinder.

The discharge stroke on a 300 rpm unit also lasts a tenth of a second. Gas inside the cylinder is compressed and must flow out the discharge valve in a tenth of a second. If there is any restriction in the discharge valve, discharge passageway in the cylinder, or discharge piping, more pressure will be required to push the gas out the cylinder. The net effect is the pressure inside the cylinder will rise above normal, which will raise the power and discharge temperature. It may also cause the cylinder to pound or pulsate because of the higher pressure that is building up inside the cylinder.

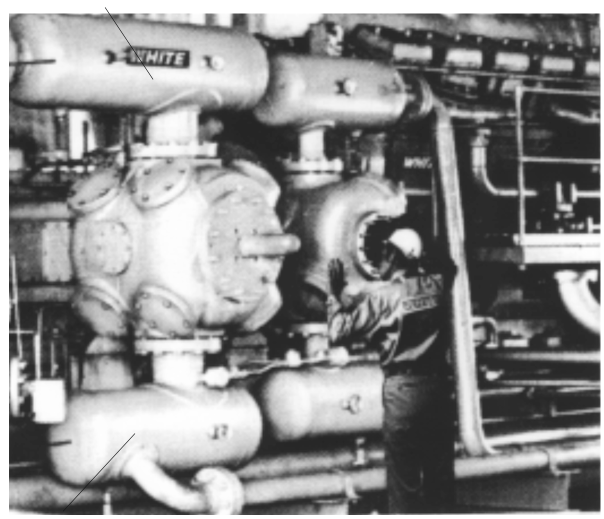
The situation is compounded when two or more cylinders are operating in parallel. This is when each cylinder takes suction from the same line and discharges into a common line. In fact, when several cylinders operate in parallel, critical vibrations may develop which can destroy piping.

Designing piping systems is beyond the scope of this manual. However, it is helpful for the compressor operator to recognize the signs of pulsation, so that he can take corrective action to prevent mechanical failure.

The effect of pulsation is to reduce the capacity of the compressor and to increase the power of the driver. Discharge temperature is higher than normal. Pulsation on the discharge side is usually accompanied by pounding and vibration. Suction pulsation may be audible as a series of "gasping" sounds as gas flows in slugs into the compressor cylinder.

The permanent solution to pulsation is to install pulsation dampeners or bottles in compressor piping as shown below.

A pulsation dampener is usually a pipe that has several times the volume of normal suction or discharge piping. It acts as a surge tank to smooth out the flow of gas entering or leaving the compressor. The discharge bottle may contain baffles or other internal devices to cushion or dampen the flow of gas. In some situations, pulsation can be helped by installing high capacity suction and discharge valves.



PULSATION DAMPENERS

Problem 6

Match terms in the column on the left with those on the right.

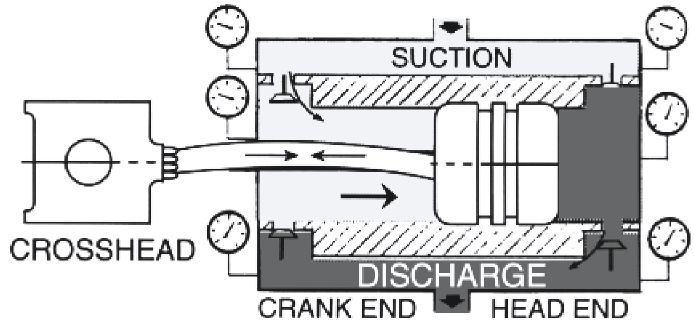
____	1.	Pulsation	a.	Volume remaining in cylinder at the end of stroke.
____	2.	Clearance	b.	% of displacement usable for compression.
____	3.	Volumetric Efficiency	c.	Pounding, vibration and loss of capacity.

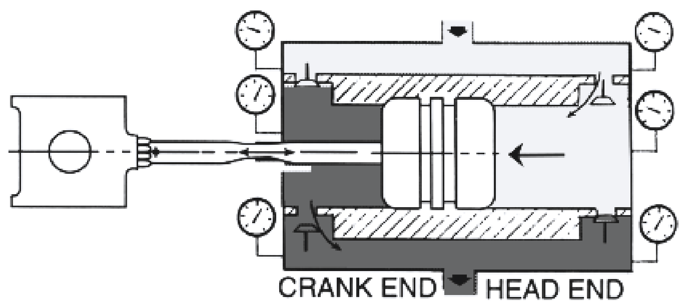
J. Rod Load

The piston in the drawing to the right is compressing gas in the head end. Pressure on the head end of the piston is discharge pressure. The total force on the head end of the piston equals discharge pressure times the area of the piston. Pressure on the crank end of the piston is suction pressure. Total force acting against the piston equals suction pressure times the crank end area of the piston. The difference between the forces is the force required to push the piston toward the head end. This force is called the ***compression rod load***. The rod is **pushing** the piston. If the pushing force becomes excessive, the rod will buckle.

When the piston is moving toward the crank end, discharge pressure is exerted on the crank end of the piston, and suction pressure is on the head end. In this case, the rod is **pulling** the piston toward the crank end. If the pulling force is excessive, the rod will pull apart. The rod load in this case is in ***tension***.

The rod load is of concern for much more than the forces on the piston rod. The rod load is also the force exerted on the crankshaft by each compressor connecting rod. The size of the crankshaft, bearings, counterweights, connecting rods and bolts, piston, and crosshead are affected by the rod loads.





The compressor manufacturers select rod sizes that will safely withstand rod loads at design conditions. The tables on the next page indicate allowable rod loads for various rod diameters.

The formulas for calculating rod load are:

	SI UNITS	ENGLISH UNITS
Compression Rod Load	kNewtons= $0.785 \times 10^{-6}\,[D_P^2(P_D-P_S)+D_R^2P_S]$	lbs = $0.785[D_P^2(P_D-P_S)+D_R^2P_S]$
Tension Rod Load	kNewtons= $0.785 \times 10^{-6}[D_P^2(P_D-P_S)-D_R^2P_D]$	lbs = $0.785[D_P^2(P_D-P_S)-D_R^2P_D]$

Where:

		SI UNITS	ENGLISH UNITS
D_P	=	Piston Diameter, mm	Piston Diameter, inches
D_R	=	Rod Diameter, mm	Rod Diameter, inches
P_D	=	Discharge Pressure, kPa	Discharge Pressure, psi
P_S	=	Suction Pressure, kPa	Suction Pressure, psi

ROD LOAD
ALLOWABLE ROD LOADS

SI UNITS, KILONEWTONS

ROD DIAMETER	COMPRESSION	TENSION
50 mm	130	110
63 mm	215	180
75 mm	265	225
87 mm	345	295
100 mm	430	365
113 mm	520	440
125 mm	600	505

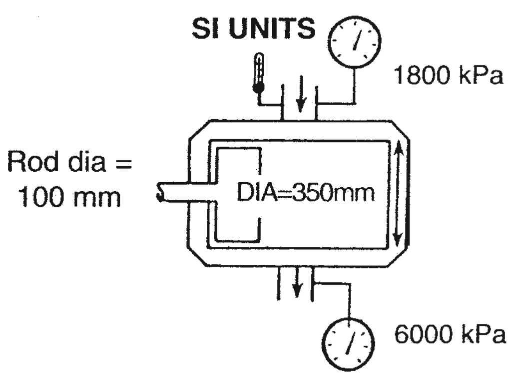
ENGLISH UNITS, POUNDS

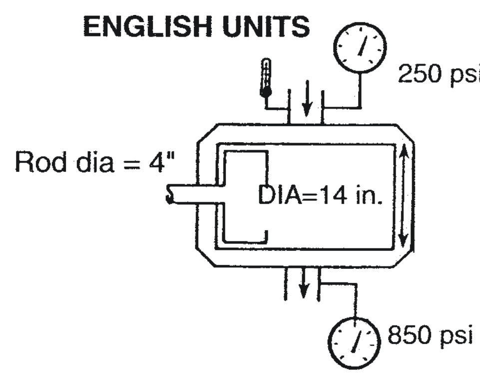
ROD DIAMETER	COMPRESSION	TENSION
2.0 in.	30 000	25 000
2.5 in.	50 000	42 500
3.0 in.	60 000	50 000
3.5 in.	80 000	70 000
4.0 in.	100 000	85 000
4.5 in.	120 000	100 000
5.0 in.	140 000	120 000

If the rod loads shown above are exceeded for a prolonged period (several days), failure of rods and/or connecting rod bearings may occur.

High rod loads usually result from low gas flow rates. In this situation, the suction pressure will drop; discharge pressure remains the same, and the pressure difference (discharge — suction pressure) increases, which increases the rod load. Drop in suction pressure is prevented by reducing the speed, or opening clearance pockets. It is covered in more detail in the next chapter.

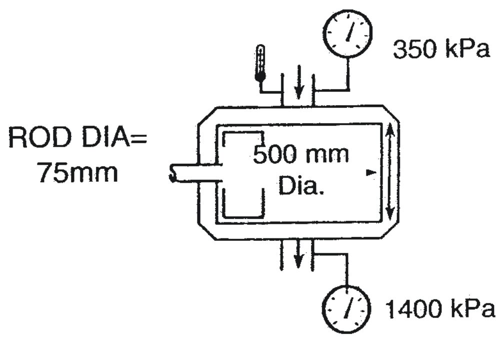
Example: Determine rod loads in tension and compression for the following compressor.

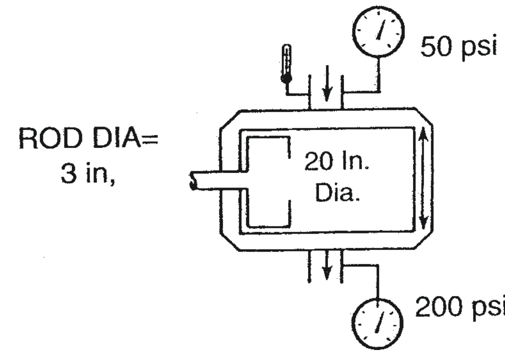




	SI	English
Rod Load, Compression	$0.785 \times 10^{-6}[D_P^2(P_D-P_S)+D_R^2P_S]$	$=0.785\,[D_P^2(P_D-P_S)+D_R^2P_S]$
Substitute in Equation	$0.785 \times 10^{-6}[350^2(6000-1800)+100^2 x1800]$	$=0.785\,[14^2(850-250)+4^2 x250]$
	$= 0.785 \times 10^{-6}[122\,500\,(4200)+18\,000\,000]$	$= 0.785\,[196(600+4000]$
	= 418 kNewtons	= 95 456 lbs
Allowable Compression Load for rod diameter	430 kN	100 000 lbs
Rod Load, Tension	$= 0.785 \times 10^{-6}[D_P^2(P_D-P_S)-D_R^2P_D]$	$=0.785\,[D_P^2[P_D-P_S)-D_R^2P_D]$
Substitute in Equation	$= 0.785 \times 10^{-6}[350^2(6000-1800)-100^2 x6000]$	$= 0.785\,[196(600)-4^2 x850]$
	= 356 kNewtons	= 81 640 lbs
Allowable Tension Load for Rod Diameter	= 365 kN	85 000 lbs

Problem 7





Compression rod load ______________________

Tension rod load _________________________

IV. COMPRESSOR CONTROL

A compressor should be controlled so that it handles the gas flowing to it with the least amount of energy consumed by the driver. This means that the capacity of the compressor cylinders is adjusted so that it is the same as the volume of gas flowing to it, and the driver consumes the lowest amount of energy to run the unit.

Most oilfield and process compressors operate at a fairly constant discharge pressure. The **suction pressure** is the control point for the unit. When the suction pressure drops, the compressor capacity is more than the flow of gas to it. Some of the driver energy is being wasted. When the suction pressure rises, gas flow is more than the capacity of the compressor cylinders. This condition may increase the energy required by the driver — possibly to an overload condition.

The upper limit of the control range of a compressor is the power rating of the driver. The driver has a nameplate which shows the power rating - kilowatts or horsepower. This rating should not be exceeded for extended periods. In fact, most engine and turbine manufacturers rate their machines on the high side, and continuous operation at 100% rating often results in excessive maintenance of the driver. Consequently, it is often desirable to operate below maximum ratings.

We said the suction pressure is our control point, and we want to operate the unit so that it consumes the least amount of power. The fuel cost for 1 kW of driver power is about $250 U.S. dollars a year. [1 hp of fuel costs about $200 per year]. A power reduction of a few per cent will often result in substantial fuel savings.

The power required depends on the compression ratio and the volume of gas that is compressed. A higher compression ratio takes more power.

You recall the compression ratio equals:

$$CR = \frac{\text{Absolute Discharge Pressure}}{\text{Absolute Suction Pressure}}$$

Since the discharge pressure is usually constant, the lowest compression ratio and power consumption will occur at the highest suction pressure.

So, The first step in controlling our compressor is that of determining the highest suction pressure we can operate, and attempt to hold it at that point. The suction pressure will depend upon the source of gas to the compressor.

If gas flows to the compressor from a production gathering system, suction pressure should be the highest pressure that the system can tolerate.

In some fields, pressure in the gathering system is held as low as possible in order to make the maximum oil production. In this situation, the compressor operator has no control over the suction pressure so he can do very little optimizing of his machine. The compressor is operated at maximum speed in order to keep the suction pressure as low as possible, even to the point of operating under a vacuum.

Compressors in process plants often take gas from a process separator or other equipment that operates at a constant pressure. In this case, compressor suction pressure should be held 100 to 175 kPa [15 to 25 psi] below the operating pressure of the process vessel. Care must be taken in controlling the suction pressure to be sure that it is always less than the operating pressure of the equipment that it is taking gas from.

When the source of gas to a compressor allows the compressor operator to control suction pressure within a certain pressure range, the suction pressure should be controlled at the highest possible point. Once this point has been determined, it is up to the compressor operator to hold it at that point.

There are 4 common ways of controlling suction pressure:

1. Speed Control
2. Clearance pocket adjustment
3. Unload suction valves
4. By-pass discharge gas back to the suction (recycle)

A. Speed Control

Adjusting the speed of the driver (and compressor cylinders) is the preferred method of controlling suction pressure for 2 reasons:

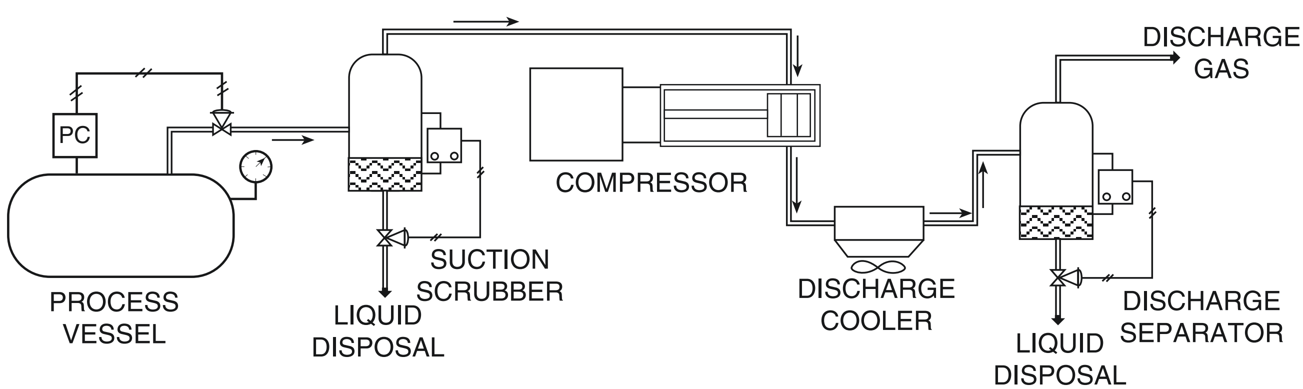
1. The driver will operate at the lowest speed, so it will consume the least fuel.
2. Less maintenance of the driver and compressor cylinders will be required at lower operating speeds.

Compressor speed is controlled by changing the speed of the driver. If the driver is a constant speed machine, we obviously cannot use speed control. If the driver is a variable speed engine or turbine, speed is adjusted by changing the setting on the governor. A variable speed electric motor is changed by adjusting the frequency on an alternating current (AC) motor; or by adjusting the voltage on a direct current (DC) motor.

The operating speed range of most drivers is 70-110% of rated speed. This is the maximum speed control range available. If the volume of gas is less than 70% of the compressor capacity, speed control alone will not provide the ideal operating point.

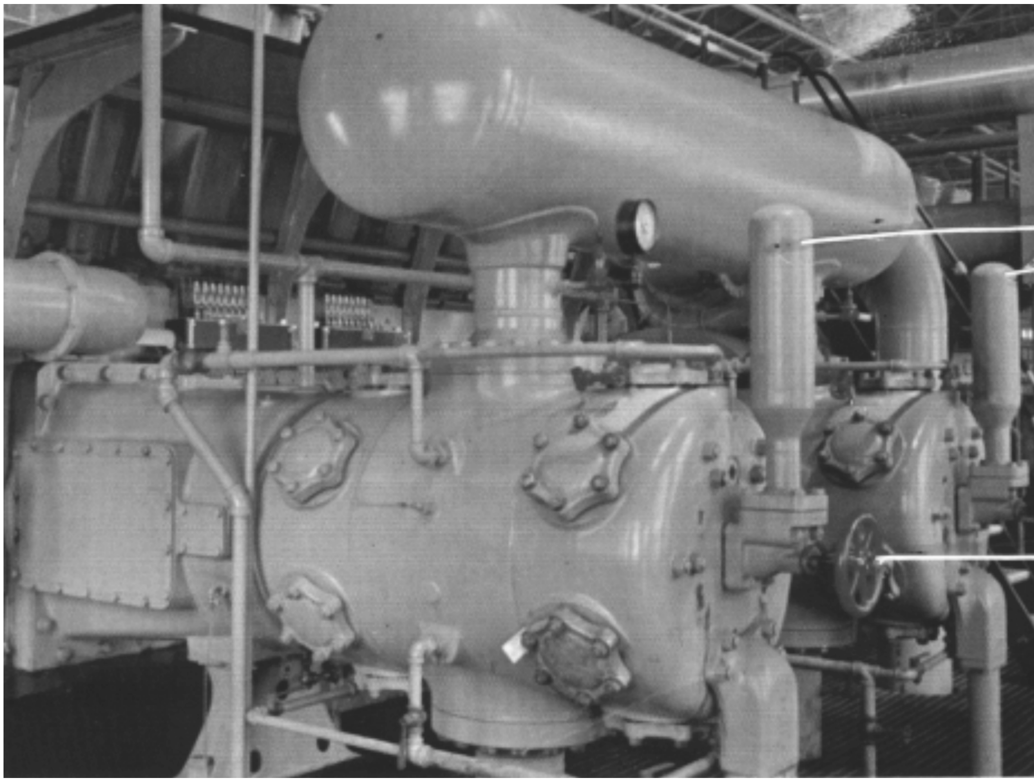
One other point about speed control: it should be done gradually and as infrequently as possible. Frequent speed changes can result in more wear and fuel consumption than constant speed operation at a higher speed.

To summarize: controlling suction pressure by varying the speed will result in the lowest operating cost provided speed changes are gradual and not too frequent.

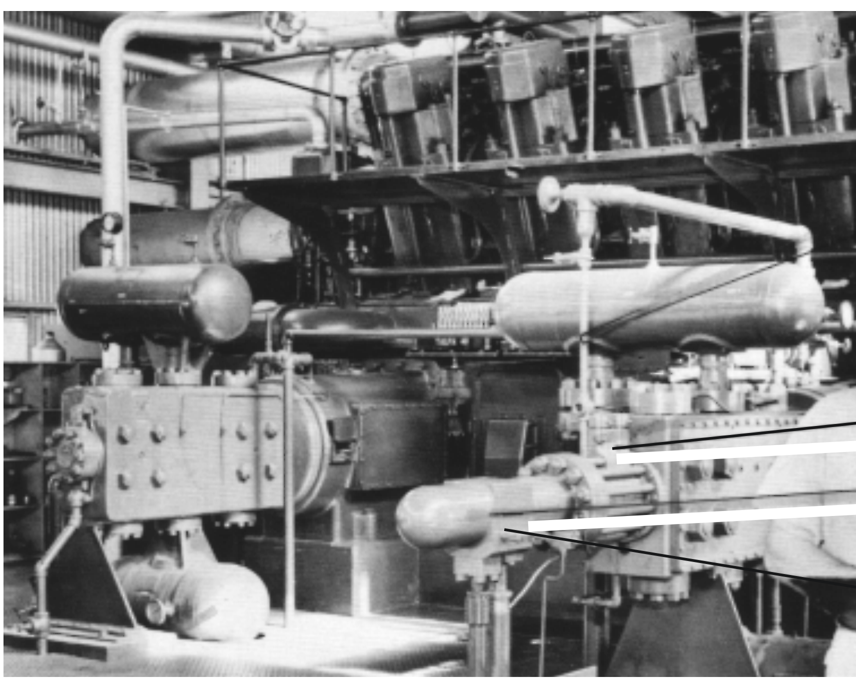


GAS FLOW TO COMPRESSOR FROM PROCESS VESSEL
Control compressor suction pressure 100 to 175 [15 to 25 psi] below pressure in process vessel

CAPACITY CONTROL WITH CLEARANCE POCKETS



COMPRESSOR CYLINDERS WITH FIXED VOLUME CLEARANCE POCKETS



COMPRESSOR CYLINDER WITH CLEARANCE POCKET AUTOMATICALLY ADJUSTED BY SUCTION PRESSURE

B. Control With Clearance Pockets

Quite often, compressor units run at a constant speed, or other control modes are not sufficient to hold the desired suction pressure. In these situations, clearance pockets are often used for controlling suction pressure.

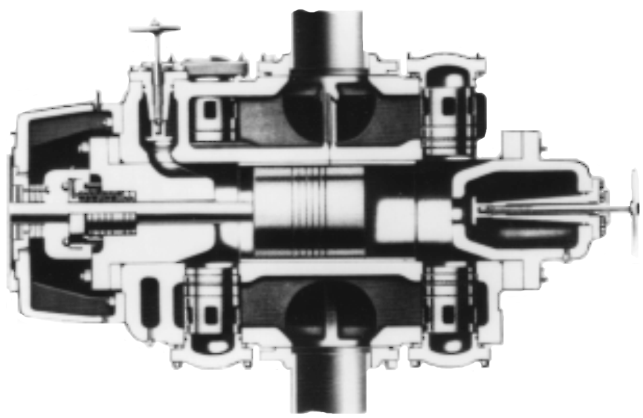
When a clearance pocket is opened, some of the gas on the discharge stroke flows into the clearance chamber instead of flowing out the compressor. Consequently, the capacity of the cylinder is reduced. Most of the energy used to compress the gas that enters the clearance chamber is recovered when the piston reverses its stroke, and the clearance gas at discharge pressure pushes the piston until its pressure drops to suction pressure.

The use of clearance pockets to control capacity is almost as cost effective as that of speed control. On variable speed machines, clearance pockets should not be opened unless the unit is operating at minimum speed and its capacity is still greater than the volume of gas flowing to it, i.e. the suction pressure is below normal.

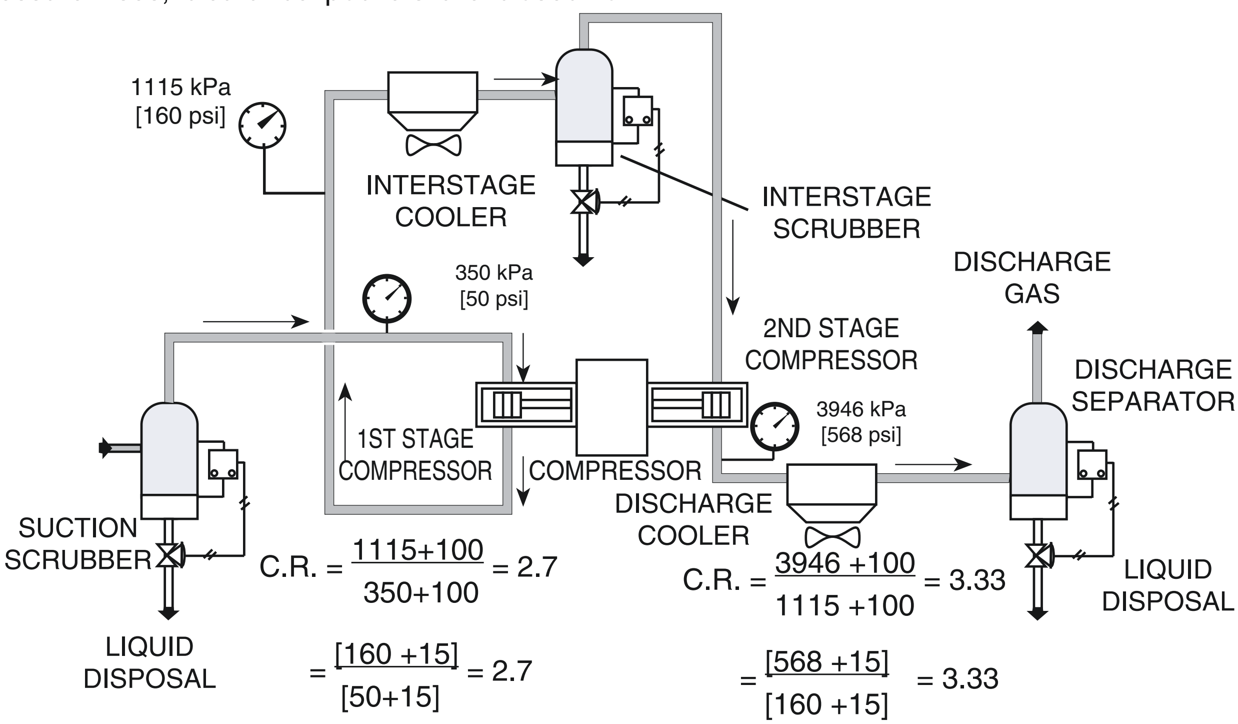
When the suction pressure **falls**, the compressor capacity is more than the volume of gas flowing to it. Clearance pockets are opened to reduce the compressor capacity. Conversely, when the suction pressure rises, clearance pockets are closed to increase capacity.

Clearance pockets can be operated by hand, or they can be equipped with actuators that open or close automatically from a suction pressure controller.

Another application of clearance pockets is that of equalizing the loads on multi-stage compressors. In the 2-stage unit shown below, the compression ratio in the second stage is 3.33, whereas it is only 2.7 in the first stage. The imbalance can be corrected by unloading the second stage — open clearance pockets — or loading the first stage —



COMPRESSOR CYLINDER WITH CLEARANCE POCKETS ON HEAD AND CRANK ENDS



close clearance pockets. The compression ratio should be the same on each stage for optimum operation of the compressor.

The question at hand is: in which stage should pockets be adjusted? The answer is: **close pockets on low loaded cylinders first**.

Ideally, the machine has all pockets closed and speed is changed to control capacity. Ideal conditions seldom exist in the oilfield, particulary in multistage units.

In the machine opposite, pockets on the first stage should be closed. This will raise the volume of gas the cylinder will pump. Suction pressure will drop, and discharge pressure will rise. The speed can be reduced to raise suction pressure to normal. When speed is lowered, the discharge pressure will rise. First stage compression ratio (and load) will rise; 2nd stage C.R. and load will fall.

The volume of gas a multistage compressor will pump is determined by the capacity of the 1st stage cylinder. Whatever volume it pumps must obviously flow through the other stages. Thus, adjusting pockets on 2nd, 3rd or 4th stage will not affect the volume of gas that stage will compress. It will affect the suction pressure and power that the cylinder requires. The effect of adjusting pockets on various stages of a 3-stage compressor is shown in the table below. The table assumes a constant discharge pressure from the 3rd stage.

C. Control With Valve Unloaders

In some operating situations, the volume of gas flowing to a compressor is less than 50% of the compressor capacity. In this case, suction valves on one end of the compressor — usually the outboard end — can be unloaded and the capacity of the compressor is reduced by 50%.

Unloading a suction valve has the same effect as removing the valve from the compressor. It makes the piston single-acting rather than its usual double-acting role.

The piston normally compresses gas on both the outboard and inboard strokes. The compressor unit is designed so that it is balanced when the piston compresses gas on the inboard and outboard strokes.

When a suction valve is unloaded, the piston does not work during half of its cycle. This may

EFFECT OF ADJUSTING CLEARANCE POCKETS ON 3-STAGE COMPRESSOR

When Clearance Pocket is	Effect on 1st Stage Cylinder				Effect on 2nd Stage Cylinder				Effect on 3rd Stage Cylinder			
	Driver Power	Cylinder Capacity	Suction Pressure	Discharge Pressure	Driver Power	Cylinder Capacity	Suction Pressure	Discharge Pressure	Driver Power	Cylidner Capacity	Suction Pressure	Discharge Pressure
Opened on 1st Stage	Less	Less	Rises	Falls	More	Less	Falls	Falls	More	Less	Falls	Same
Closed on 1st Stage	More	More	Falls	Rises	Less	More	Rises	Rises	Less	More	Rises	Same
Opened on 2nd Stage	More	Same	Same	Rises	Less	Same	Rises	Falls	More	Same	Falls	Same
Closed on 2nd Stage	Less	Same	Same	Falls	More	Same	Falls	Rises	Less	Same	Rises	Same
Opened on 3rd Stage	Rises	Same	Same	Rises	More	Same	Rises	Rises	Less	Same	Rises	Same
Closed on 3rd Stage	Less	Same	Same	Falls	Less	Same	Falls	Falls	More	Same	Falls	Same

adversely affect the balance of the unit, and result in serious mechanical failures. Approval by the compressor manufacturer should be obtained before unloading suction valves unless the machine was designed to operate with suction valves unloaded.

Suction valves can be unloaded by turning a hand wheel, or they may have automatic actuators that raise and lower them. In either case, they should be checked to see that they are unloaded **all the way**. If a valve is only partially unloaded, it will rapidly overheat, and a valve failure is almost a certainty. Illustrations of unloaders are shown on page 7.

D. Recycle Control

A decrease in the gas flow to a compressor results in a drop in suction pressure. This may increase the rod load to the point of failure. A recycle line is often installed to prevent the suction pressure from dropping to the point of failure. A drawing of a typical recycle system is shown below.

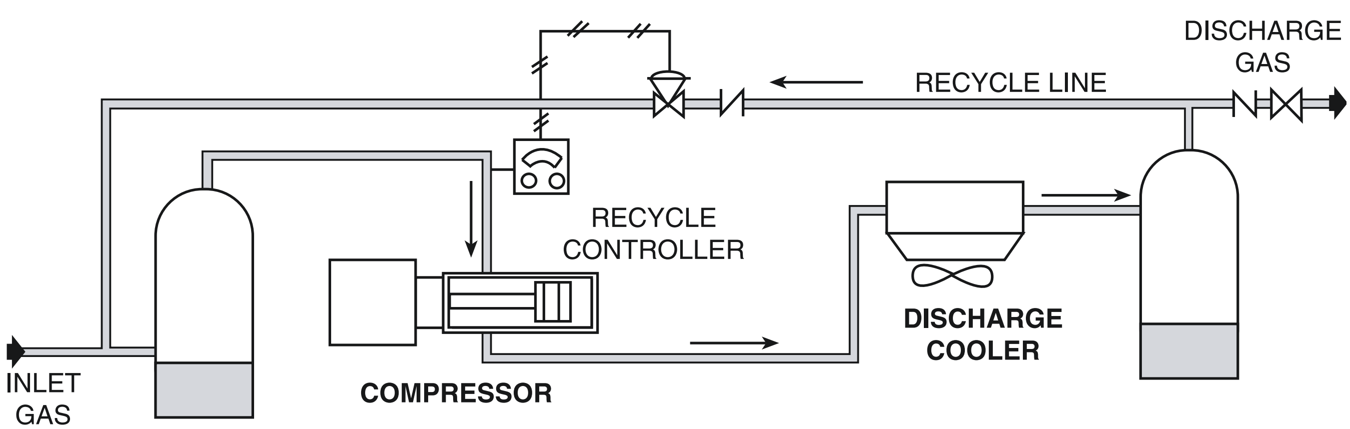
The recycle system by-passes some of the discharge gas back to the suction side of the compressor. Discharge gas for recycle is withdrawn downstream of the discharge cooler. If hot discharge gas was recycled, it would overheat the compressor valves, rings and packing and result in failures.

Flow of recycle gas is regulated with a recycle controller, which is a pressure controller. It senses suction pressure, and when it drops, it signals a control valve in the recycle line to open.

If flow of incoming gas rises, the suction pressure will increase, and the recycle controller will signal the control valve to close.

The recycle control may be the primary capacity control method for machines operating at constant speed that do not have clearance pockets. It is frequently installed on other units as a safety control to prevent low suction pressure from occurring. These units will use speed, clearance or valve unloading as primary capacity control: but use recycle as a safety control to protect the machine in the event of a sudden drop in inlet gas flow.

Flow of recycle gas does not significantly effect the compressor driver power. As we learned on page 37, driver power drops only 1% when suction pressure drops almost 30%. In fact, driver power may increase slightly due to recycle. However, the protection against rod load failure more than offsets the minimal expense of recycle.



FLOW OF RECYCLE GAS FOR CAPACITY CONTROL

E. Other Factors Affecting Control

Up to this point, we have assumed that suction pressure was the primary control concern. There are other factors that must also be considered.

In setting the compressor speed, excessive vibration and pulsation must be avoided. Identical compressor units located side-by-side often have different vibration and pulsation characteristics. One unit may purr like a kitten at 90% speed, whereas the other one rattles the windows at that speed. Although vibration should not be a significant factor within the normal speed range of a compressor, it often is. Consequently, speed that results in vibration should be avoided.

Vibration from a compressor is transmitted through piping directly connected to it as well as through nearby piping. These vibration waves, either by themselves, or in combination with waves from other sources, can cause ruptures in piping located far beyond the compressor unit.

Vibration of piping can be seen visually. Pulsation is an audible knocking sound. It may also cause piping to vibrate. When vibration or pulsation occur, the speed should be adjusted to find the point that results in the "smoothest" operation.

SUMMARY OF COMPRESSOR CONTROL

1. A compressor is controlled to hold a constant suction pressure at the highest pressure the system will tolerate.
2. The ideal control point is that which results in the lowest operating cost. The major factors affecting operating cost are fuel or electricity to the driver, and maintenance.
3. Lowest operating costs will occur when the unit runs at the lowest speed.

Following are procedures for controlling compressor suction pressure.

OPERATING CONDITION	COMPRESSOR CONDITION	PREFERRED OPERATING SEQUENCE TO CORRECT CONDITION
Low Suction Pressure	Compressor capacity is more than gas flow rate.	1. Reduce speed until suction pressure rises to normal. 2. Open clearance pockets until suction pressure returns to normal. 3. Unload suction valves if gas flow is continuously less than 50% of design. 4. By-pass some discharge gas back to the suction until suction pressure rises to normal.
High Suction Pressure	Gas flow rate is more than compressor capacity.	1. Reduce flow of by-pass gas until pressure drops to normal. 2. Load suction valves that are unloaded if gas rate will continually be more than 50% of design. 3. Close clearance pockets until suction pressure falls to normal. 4. Raise speed until suction pressure drops to normal. Do not exceed maximum speed.

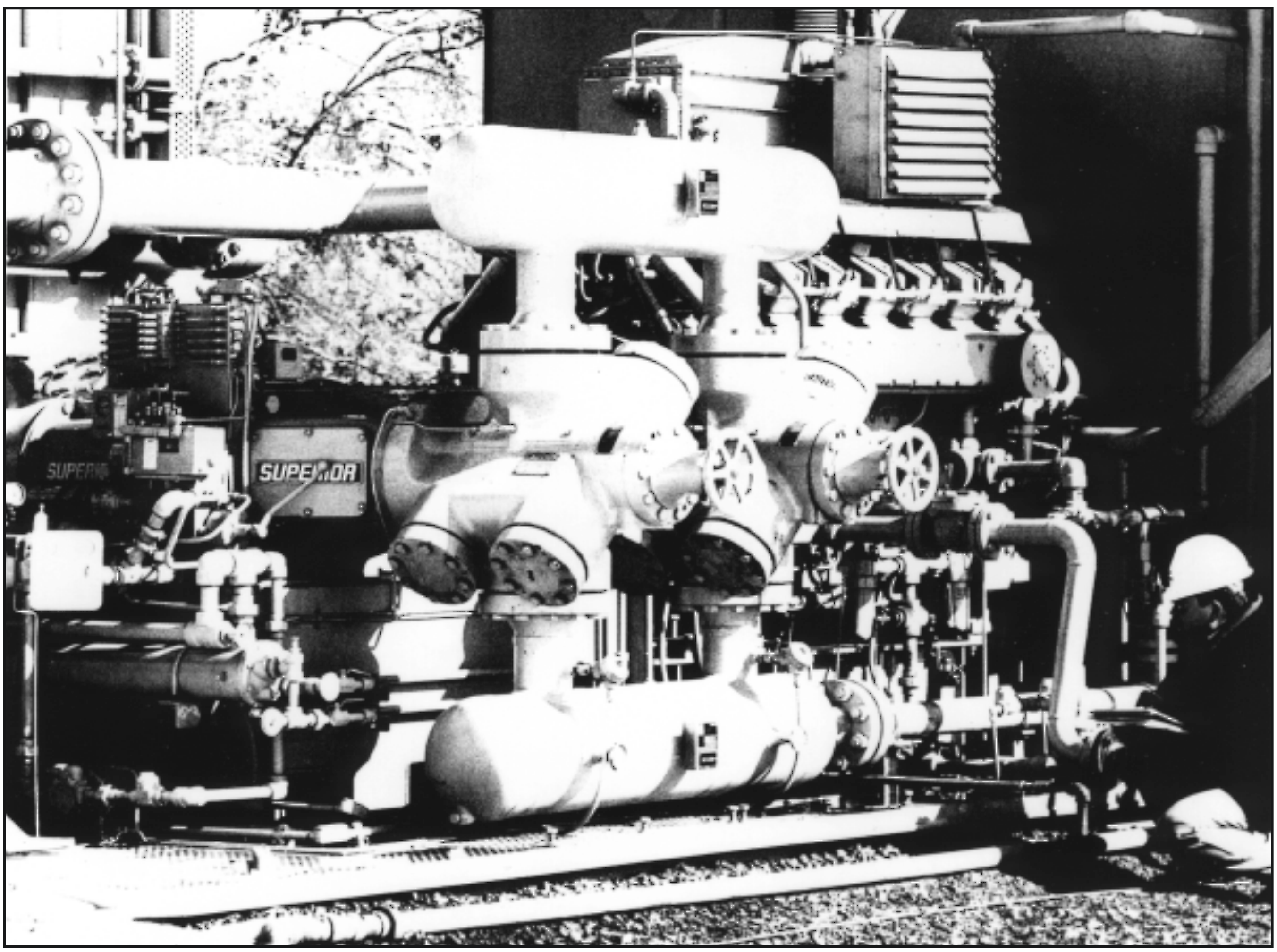
Problem 8

A. A compressor is designed to operate at a suction pressure of 3450 kPa [500 psig]. Operating suction pressure is 3100 kPa [450 psig]. List the proper sequence of the following items for the operator to perform:

________	Open clearance pockets	________	Lower compressor speed
________	Close clearance pockets	________	Do nothing
________	Raise compressor speed		

B. Suction pressure rises to 3800 kPa [551 psig]. List the proper sequence of the following items for the operator to perform:

________	Open clearance pockets	________	Lower compressor speed
________	Close clearance pockets	________	Do nothing
________	Raise compressor speed		



V. OPERATION

A. Start-up

Two factors are essential in starting a compressor unit:

1. The gas piping must be thoroughly purged of air to prevent the possibility of a combustible mixture occurring in the compressor and its adjoining piping.
2. The unit must be started in an unloaded condition until the driver gets up to operating speed. The compressor load on the driver is then slowly increased.

1. Purging Air

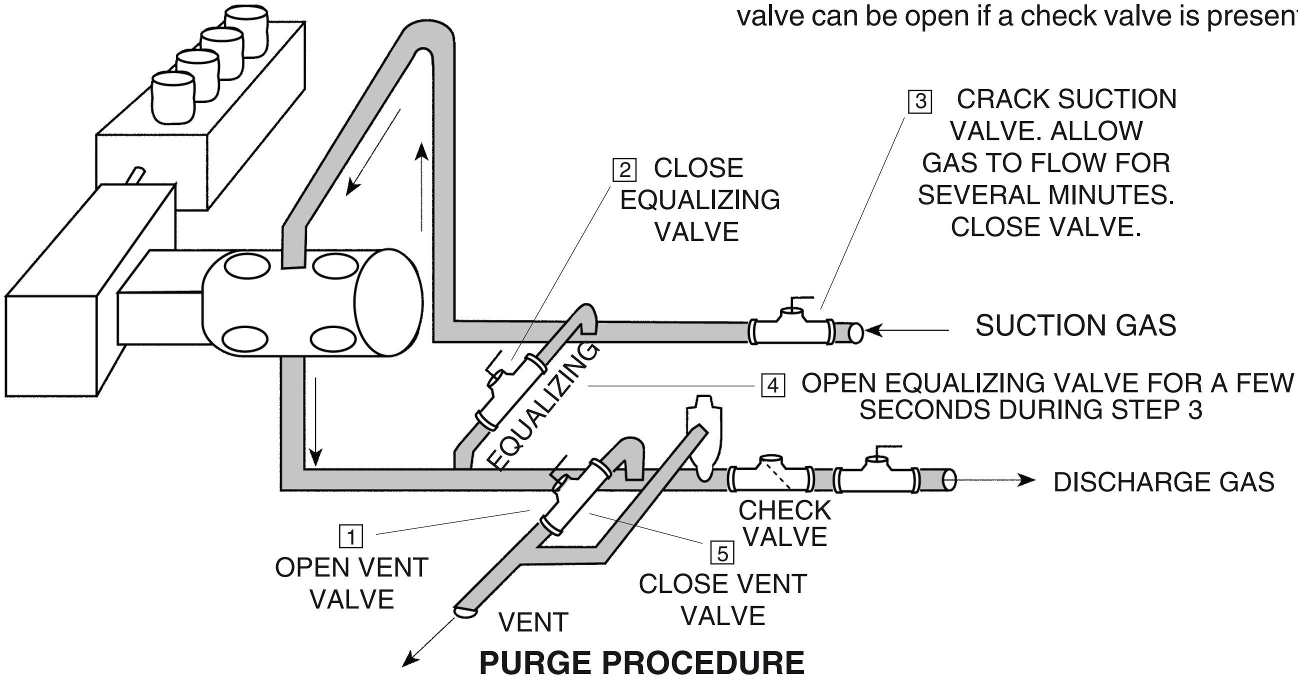
Purging is only necessary when a compressor has been depressured to the atmosphere when it was out of service. It is not necessary if the unit was not depressured when it was shut down.

Three elements are essential for a fire to occur: fuel, air and heat. Fuel is always present when a compressor handling natural gas is started. Furthermore, a spark from moving valves or other parts of the unit is always likely. The third element for a fire — air — must be removed from a compressor system before it is started up.

At the time a compressor is started, the vent valve is usually open if the unit was depressured. This enables air to enter the piping in the compressor. If the compressor has been dismantled for repair, air is certain to be present. Consequently, the first step in starting a compressor is to remove the air by purging the unit with gas.

The unit should be purged before the driver is started. One procedure to purge is as follows:

1. Open the vent valve.
2. Close the equalizing valve if present.
3. Crack open the suction valve. Gas will flow through the compressor and out the vent line. Allow purge to continue for several minutes. At the end of this time, open the equalizing line to purge any air that may be trapped in it.
4. If there is no check valve in the discharge piping, crack the discharge valve and purge air from the outer end of the discharge line. Omit this step if the discharge line has a check valve.
5. Close the vent valve.
6. Check to be sure the valve in the equalizing line is open, and valves in the suction and discharge lines are closed. The discharge valve can be open if a check valve is present.



PURGE PROCEDURE

Operation of the valve in the equalizing line is important to safely start up a compressor. If the valve is closed when the compressor starts running, a vacuum will occur in the suction piping, and may pull air into the system. The valve in the equalizing line must always be open when a compressor is started.

If there is no equalizing line, the vent valve remains open at all times that the compressor is down, and during purge.

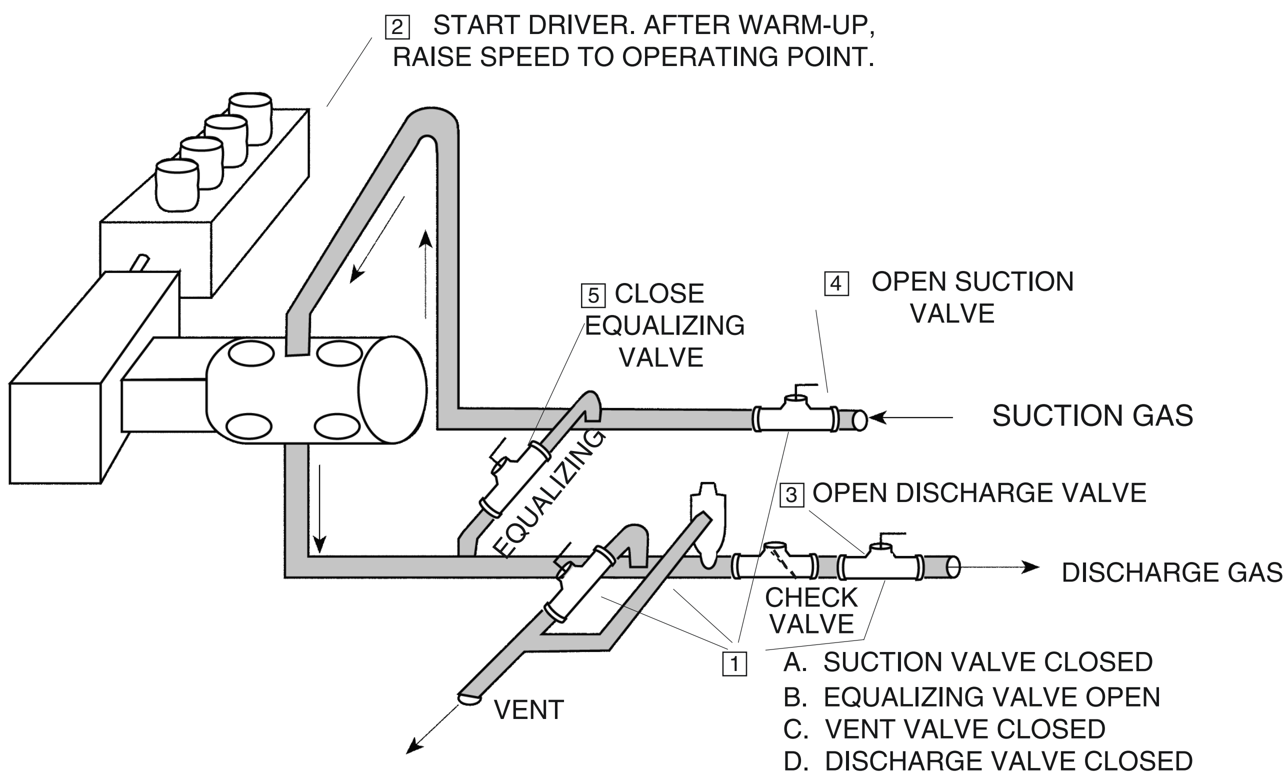
After purge is completed, the driver is started. If the driver is an engine, it is usually desirable to allow it to idle a few minutes to warm up. If the unit has no equalizing line, a vacuum will be pulled in the suction piping. To prevent the possibility of air entering the system, the suction valve should be cracked open during engine warm up to allow enough gas to enter the unit to prevent a vacuum. The vent valve must obviously be open during this time.

When the driver is warmed up, the compressor is put in service. The procedure is as follows:

1. Open the valve in the discharge line.
2. Slowly open the valve in the suction line.
3. Slowly close the valve in the equalizing line. If there is no equalizing line, close the valve in the vent line.

Quite often, compressor units have several cylinders in different services, or in second or third stage compression service. If the cylinders are in different services, the order of putting each cylinder in service is not too important. Generally, the cylinder requiring the least power is put in service first, and the cylinder requiring the most power is put in service last.

If the unit has two or more stages of compression, i.e. discharge from one or more cylinders enters the suction of other cylinders, the highest compression stage cylinders are put in service first; and the lowest compression stage is put in service



PROCEDURE TO START-UP AFTER PURGING

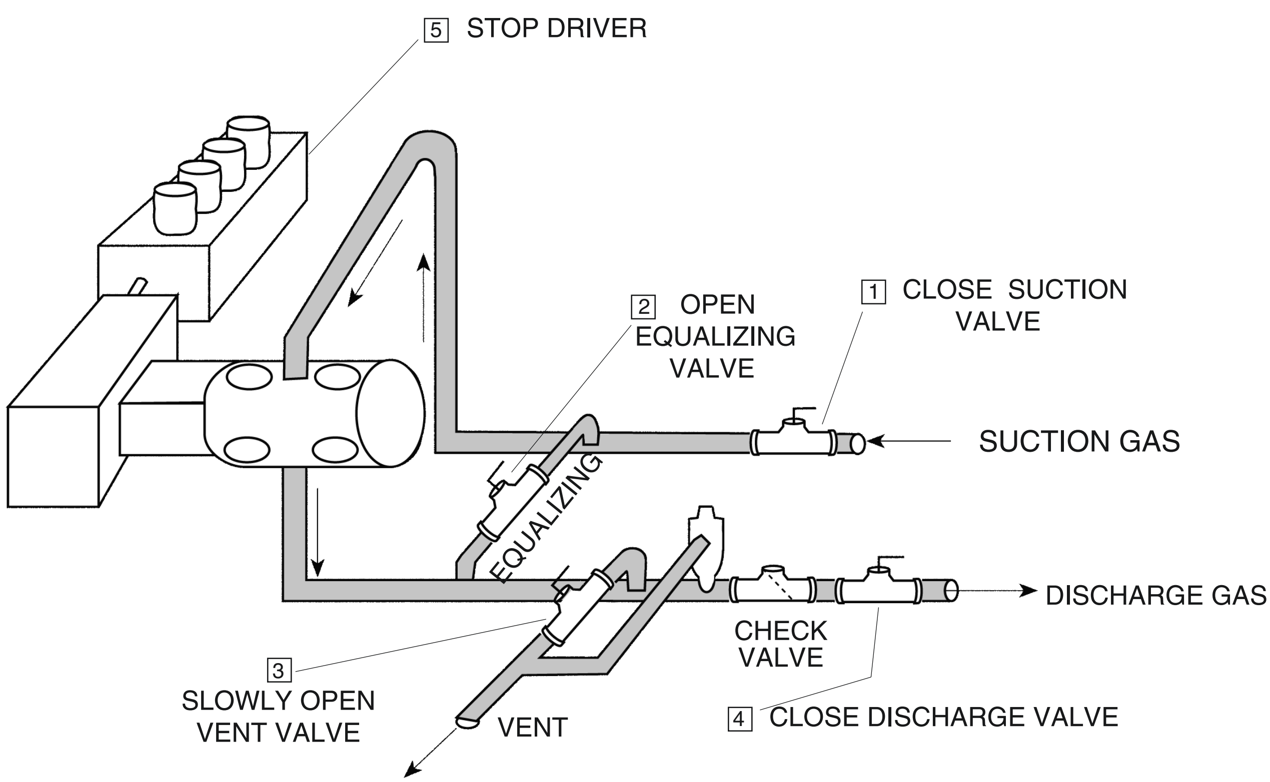
last. If the first stage cylinders are put in service first, there will be no place for the discharge gas to go (it normally flows to the second stage cylinders, but they are not in service yet.) Discharge pressure from the first stage will rise and open the relief valve on the discharge line. The gas will flow out a vent stack to the air. This not only wastes gas, but may also result in a leaking relief valve, as they often fail to seat after opening.

B. Shutdown Procedure

1. Close the valve in the suction line.
2. Open the equalizing valve if it is present.
3. Open the vent valve if the unit will be depressured.
4. Close the valve in the discharge line. This step may not be necessary if a check valve is installed in the discharge line.
5. Shutdown the driver.

Different start-up and shutdown procedures can be used on compressors equipped with suction valve unloaders. These units do not have to be depressured when they are shutdown. Furthermore, unless the unit is shut down for repair work, the piping remains under pressure at shutdown, so purging is not required prior to start-up.

Most units of this type have automatic start and stop control systems that are activated by pushing a START or STOP switch. An automatic start system has a permissive circuit that checks the position of valves and valve unloaders, and performs the same steps that an operator would if he were present.



SHUT-DOWN PROCEDURE

C. Routine Operating Checks

1. If the compressor is equipped with a gas flow meter, check it to see that the unit is compressing the proper quality of gas.
2. Check the suction and discharge pressure to see that they are within normal operating ranges.
3. Check suction and discharge thermometers for readings within the normal operating ranges.
4. Feel each suction valve cover for a hot cover which indicates a leaking valve.
5. Listen for a change in sound or noise level made by the unit.
6. Check the water cooling system to be sure that water is flowing through the jacket on the cylinder.
7. Check the oil lubricators to see that oil is flowing to each point in the cylinder and packing.
8. Feel the inboard end of the cylinder for an indication of hot packing.
9. Check for packing leaks. If the distance piece inspection covers are removed, place your hand near the packing and feel for gas flowing from it. Caution: the rod may be hot so use caution in making the check. If the inspection plates on the distance piece are secured, check the distance piece vent line for gas blowing out of it. The line will be hot if gas is flowing in it.

Problem 9

List the proper sequence for the following start-up and shutdown events for a manually operated compressor that does not have valve unloaders.

Start-up Sequence:

_______ Close vent valve
_______ Start driver and bring it up to operating speed.
_______ Purge air from unit.
_______ Close equalizing valve.
_______ Open valve in discharge piping.
_______ Open valve in suction piping.

Shutdown Sequence:

_______ Close valve in suction piping.
_______ Close valve in discharge piping.
_______ Stop driver.
_______ Open equalizing valve.
_______ Open vent valve.
____

VI. TROUBLESHOOTING

Most operating problems on a compressor fall into two categories:

1. The unit will not handle its design volume of gas due to a process condition — low speed, clearance pocket open, etc.
2. A mechanical failure has occurred in the unit.

Failure of a compressor to handle its design volume of gas is often due to a mechanical problem.

TROUBLESHOOTING REDUCED COMPRESSOR CAPACITY CAUSED BY PROCESS IRREGULARITY

CAUSE	INDICATION	CORRECTIVE ACTION
Driver speed is low	Low tachometer reading	Increase driver speed
Clearance pocket open	Clearance valve position indicator shows valve open	Close clearance pocket.
Valve unloader is in service	Unloader handle or position indicater shows it is in service	Take unloader out of service.
Equalizing or recycle valve is open	Valve handle is in open position	Close recycle or equalizing valve.

The most frequent mechanical failure on a compressor is that of valves — particularly discharge valves. The first indication of trouble is usually an increase in discharge temperature. When this is noted, a hot discharge valve cover should be looked for with a temperature gun. If it is not available, a noise detector, similar to a doctor's stethoscope can spot the valve with an unusual sound. As a last resort, a screwdriver held against the valve cover with your ear firmly against the handle may allow you to hear a different sound from the leaking valve.

Another means of determining which discharge valve is leaking is to shut down the unit and bar the crankshaft until the piston is at the crank end. Remove a head end suction valve and reach into the cylinder and pour water into the discharge valves on the head end. The leaking valve will obviously not hold water. If the head end valves are okay, repeat the procedure on the crank end. Caution: be sure to drain water from good valves by depressing a valve disc with a screwdriver before starting the compressor.

A suction valve failure is spotted by feeling valve covers. The leaking one will have a hot cover.

The symptoms of leaking piston rings are the same as those for a leaking valve except all valve covers will be at about the same temperature. High discharge temperature is the first indication. It is confirmed by the sound of gas leaking across piston rings. If rings have broken, there may be rattling or banging sounds.

Leaking rod packing will result in gas flowing into the distance piece and out the vent line from the distance piece. The vent line will be hot. If the wind is not blowing, gas can be seen flowing out the vent line.

Mechanical failure inside the compressor will also reduce the volume of gas it will pump. The most frequent mechanical failures and the method for checking them are as follows:

TROUBLESHOOTING MECHANICAL FAILURE

TYPE OF FAILURE	SYMPTOMS OF FAILURE
Valve Failure	1. Hot valve cover plate 2. Valve is noisy 3. Gas discharge temperature is high 4. Capacity of cylinder is reduced a. Suction pressure rises
Piston Rings Worn or Broken	1. High gas discharge temperature 2. Change in noise from gas leaking around rings, or clanking sound from broken ring in cylinder 3. Capacity of cylinder is reduced a. Suction pressure rises
Packing Leaking	1. Packing cage is hot. 2. Gas leaks around the rod or out the distance piece vent. The vent line is hot 3. The rod is hot. 4. The crank end of the cylinder is hot.
Piston Loose on Rod	1. Hammering sound as piston hits the end of the cylinder.
Broken Rod	1. Discharge valve covers are cool because no gas is flowing. 2. Cylinder does not make any noise. 3. Remove distance piece inspection plate. If rod break is near the crosshead, the rod will not be moving.

One of the most important aspects of troubleshooting a mechanical failure is to find the **cause** of failure. Most failures occur in valves, piston rings, or packing. When these parts are properly assembled and installed, and receive sufficient lubrication, they should last 6-12 months. If frequent failures are occurring, they will probably continue until the cause is corrected.

The lubrication system is the first place to look, particularly in units that have individual pumps to each lubrication point. If this system checks out, look for dirt or liquid in the gas. Dirt is usually obvious from visual inspection of valves. The compressor will pound if liquid is present, and discharge valves will fail. A high liquid level in suction gas scrubbers can result in liquid carryover in the gas.

The final check is that of assembly, and proper selection of materials. This is usually a maintenance function rather than one for the operator. However, the operator should satisfy himself that the proper material is used for packing, rings, and valves; and that they are properly assembled and installed. The compressor manufacturer or the supplier of packing and rings should be consulted to verify selection of material and maintenance procedures.

One of the most useful tools for troubleshooting is that of a hand held thermometer. It is particularly helpful in spotting a failed discharge valve. The valve cover on a leaking valve will be hotter than the others. Discharge valve covers are usually to hot to touch, so a hotter leaking valve cover can not be detected by touch. The hand held thermometer can be pointed at each valve cover and spot the hot one in a matter of seconds. The devices can be purchased for as little as $350 U. S. dollars.

The devices are portable, light weight, and 98 - 99% accurate. In addition to spotting leaking valves on compressors, they are helpful in checking exhaust manifold temperatures on engines, bearing temperatures on pumps and motors, and other high temperature applications.

Problem 10

Match items in the two columns.

_________	1. Broken compressor valve	a. Hammering sound
_________	2. Leaking packing	b. Hot valve cover
_________	3. Broken rod	c. Cool discharge valve cover plates
_________	4. Piston not secure on rod	d. Hot vent line on distance piece

A. Performance Analyzer

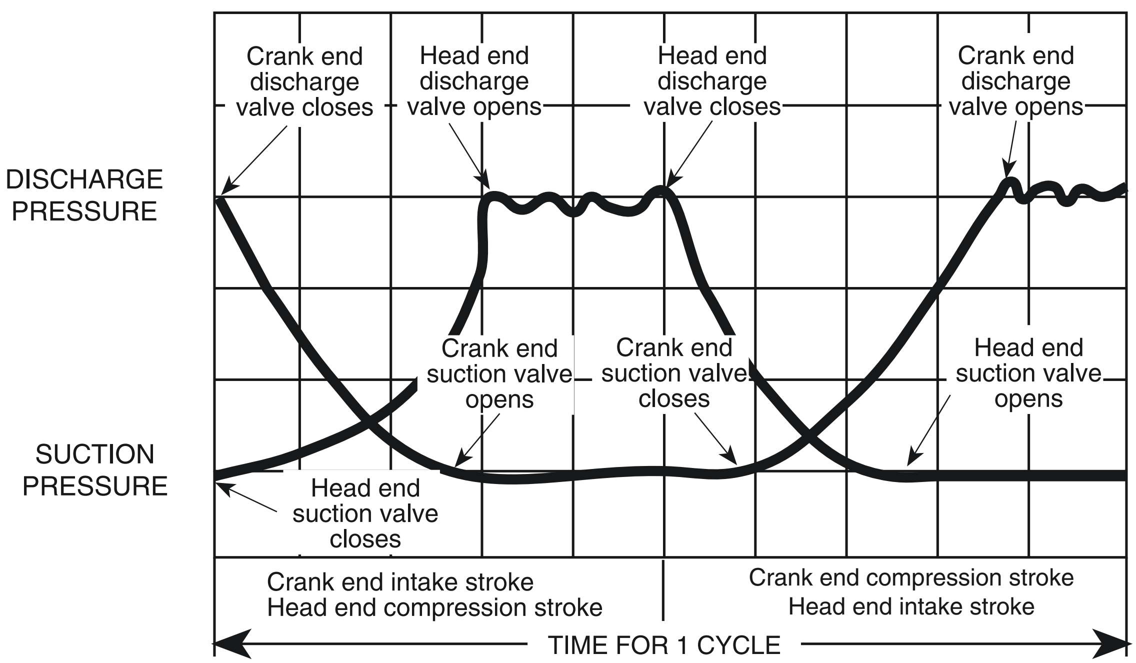
Use of a **performance analyzer** takes most of the guess work out of troubleshooting mechanical problems on a compressor. The analyzer is an electronic device which receives a signal from the compressor crankshaft indicating its speed, and pressure signals from inside the compressor cylinder and suction and discharge bottles. A television type-screen displays 2 types of information;

1. A pressure-time pattern for each cylinder. It shows the pressure rise (during the compression stroke) and pressure decline (during the intake stroke) during one stroke of the piston for head and crank ends.
2. A pressure-volume pattern which indicates the volume of gas being compressed in a cylinder on both ends of the piston.

The shape of these displays can be used by a trained technician to diagnose leaking valves, worn piston rings, leaking packing or other problems in a cylinder.

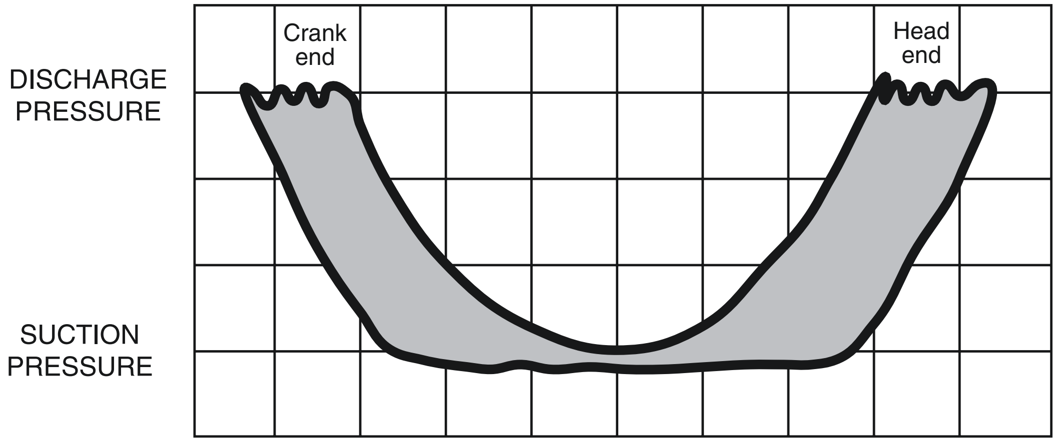
The analyzer display may be slightly different for identical compressor cylinders. Consequently, a knowledge of the appearance of the display of a cylinder when it is in good condition is necessary in order to recognize a change caused by some mechanical failure

A detailed description of the performance analyzer and procedures for using it is beyond the scope of this manual. As we said, an experienced technician can look at the display and quickly pinpoint a mechanical problem. However, an inexperienced technician can compare a picture of a display of a cylinder in good condition with the display for that cylinder while the unit is operating, and quickly tell if some mechanical problem is present. He may not be able to diagnose the problem, but he can see a difference in the display for the compressor, and call for assistance in pinpointing the failure.



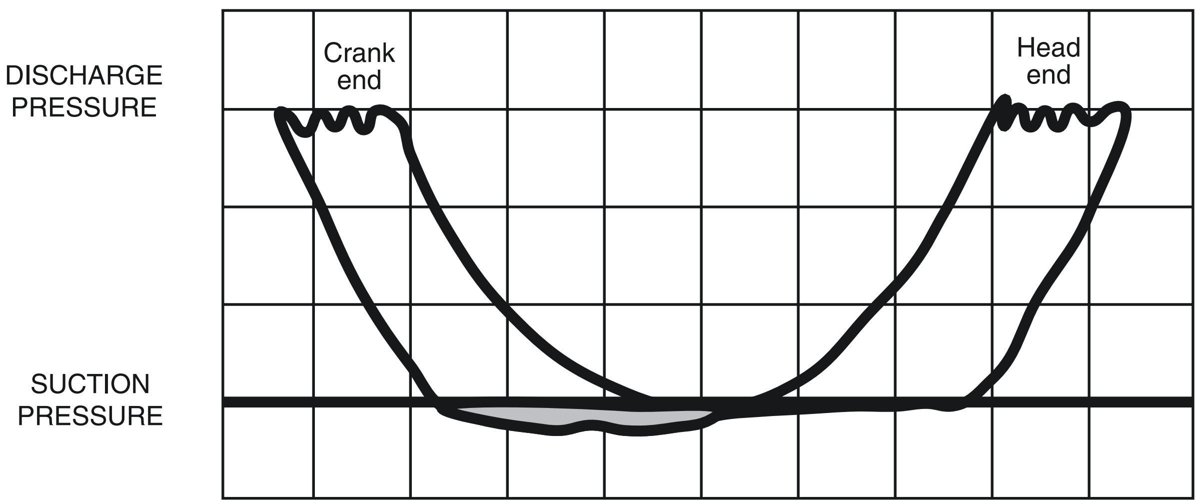
PRESSURE - TIME DISPLAY OF PERFORMANCE ANALYZER

PERFORMANCE ANALYZER



PRESSURE - VOLUME DISPLAY OF PERFORMANCE ANALYZER

Shaded area indicates volume of gas compressed

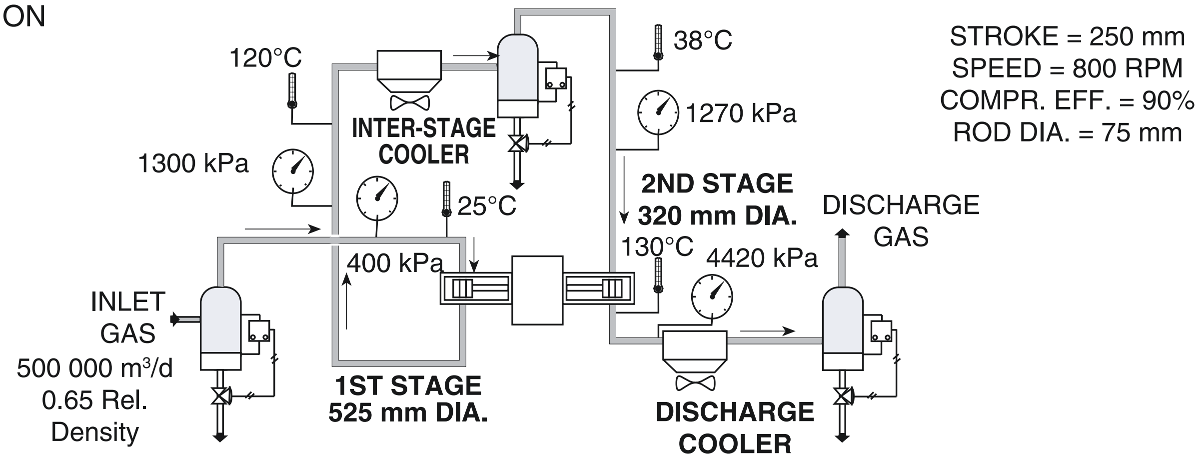


PRESSURE - VOLUME DISPLAY FOR CYLINDER WITH LEAKING SUCTION VALVES ON CRANK END

Shaded area indicates volume of leakage

PISTON TYPE COMPRESSORS
2ND EDITION

Trainee ______________________________



CLEARANCE POCKET ON EACH CYLINDER

1.

	1st STAGE		2nd STAGE	
Compression Ratio	_________		_________	
Power Required	_________	kW	_________	kW
Calculated Discharge Temperature	_________	°C	_________	°C
Actual Inlet Gas Flow Rate @ suction temp & pres	_________	m^3/d	_________	m^3/d
Displacement of Compressor Cylinder @ 90% eff.	_________	m^3/d	_________	m^3/d
Rod Load in Compression	_________	N	_________	N
Allowable Rod Load in Compression	_________	N	_________	N
Lube Oil Required on Compressor Cylinder Wall	_________	l/d	_________	l/d
No. oil injection points on cyl wall	4		2	
Injection rate at each point	_________	l/d	_________	l/d

Each Packing has 1 injection point. Total number of injedtion points ________.
Rate per point _________ l/d. Total injection _________ l/d.

2. Is each cylinder equally loaded? _______ If not, what should be done? ________________________
____
3. What process irregularities (if any) are indicated from operating conditions shown above?________
____
4. Compressor is running at 100% speed. Discharge pressure is 4420 kPa. Gas flow rate is 330 000 m^3/d.
Check each correct item below:
Suction pressure will go up/down/stay same   Discharge temp. will go up/down/stay same
Driver speed should be increased/reduced/stay same
5. The effect of opening a clearance pocket is:
Compressor capacity is more/less/unchanged   Driver power is more/less/unchanged
Discharge gas temperature will go up/down/not change   Suction pressure will go up/down/not change
6. List the order of the following events at start-up.
_______ Open valve in suction line   _______ Start driver
_______ Purge air   _______ Open valve in discharge line.
7. The vent line from the compressor distance piece on 2nd stage is hot. What is indicated? _________
____
8. List 3 indications of a leaking compressor valve ____
____.

SOLUTIONS TO PROBLEMS - SI UNITS

1. Cylinder wall lubrication = 1.2 l/d or 21 drops/min (P-9)
   No of cylinder wall injection points = 2
   Oil rate per injection point = 1.2 ÷ 2 = 0.6 l/d or 21 ÷ 2 = 10.5 drops/min
   This is also the injection rate in the packing.

2.
   1. c
   2. g
   3. d
   4. a
   5. f
   6. b
   7. e

3. Act Vol $= \dfrac{0.35(\text{Std Vol})(273 + T)}{P + 100}$ (P-27)

   $= \dfrac{0.35(1\ 100\ 000)(273 + 32)}{2100 + 100} = \underline{53\ 375\ m^3/d}$

4. Displacement @ 250 mm dia, 200 mm stroke,
   600 RPM = 15.5 x 10^3 = 16 000 m^3/d (Fig 1A P-30)
   Displacement @ 90% efficiency = 16 000 x 0.9 = 14 400 m^3/d
   Std Vol $= \dfrac{2.84\ (\text{Act Vol})(P + 100)}{(273 + °C)}$ (P-29)

   $= \dfrac{2.84(14\ 400)(2100 + 100)}{(273 + 32)} = \underline{294\ 988\ m^3/d}$

5. a. CR $= \dfrac{6500 + 100}{2100 + 100} = \underline{3.0}$ (P-34)

   b. Disch. Temp = 32° + 80 = 112°C (P-35)

   c. Power = (1790 kW/million m^3/d) 1.1 = 1969 kW (P-35)

6.
   1. c
   2. a
   3. b

7. Compression load = 0.785 x 10 - 6 $[D_P^2(P_D - P_S) + D_R^2 P_S]$ = (P-39)
   0.785 x 10^{-6} $[500^2(1400-350) + 75^2 \times 350]$ = 207.6 kN

   Tension load = 0.785 x 10 -6 $[D_P^2(P_D - P_S) - D_R^2 P_D]$ =
   0.785 x 10^{-6} $[500^2(1400 - 350) - 75^2 \times 1400]$ = 199.9 kN

8. A. 1. Lower speed
   2. Open clearance pockets

   B. 1. Close clearance pockets
   2. Raise speed

9.

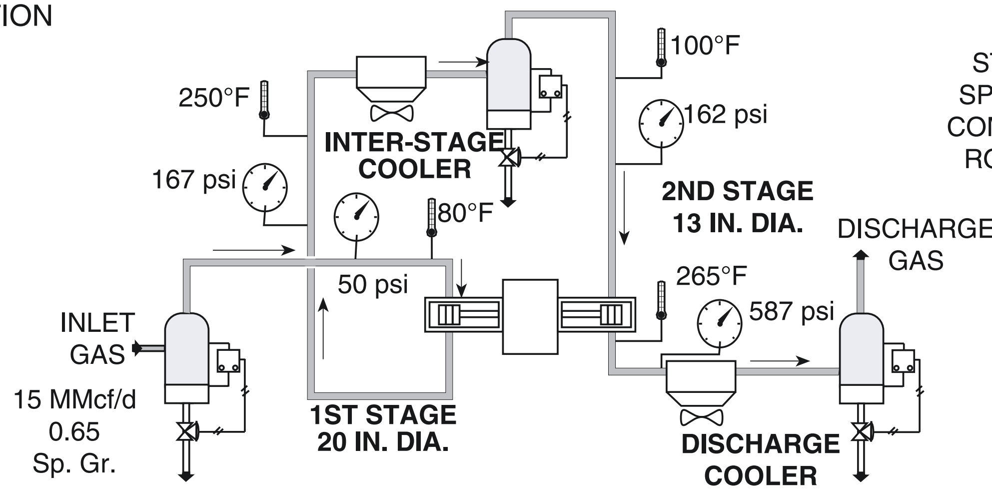
	Start-up Sequence:
3	Close vent valve
2	Start driver and bring it up to operating speed.
1	Purge air from unit.
6	Close equalizing valve.
4	Open valve in discharge piping.
5	Open valve in suction piping

	Shutdown Sequence:
1	Close valve in suction piping.
4	Close valve in discharge piping.
5	Stop driver.
2	Open equalizing valve.
3	Open vent valve.

10.
    1. b
    2. d
    3. c
    4. a

PISTON TYPE COMPRESSORS
2ND EDITION

Trainee ____________________________



STROKE = 10 in.
SPEED = 800 RPM
COMPR. EFF. = 90%
ROD DIA. = 3.0 in.

CLEARANCE POCKET ON EACH CYLINDER

1.

	1st STAGE		2nd STAGE	
Compression Ratio	_________		__________	
Power Required	_________	HP	__________	HP
Calculated Discharge Temperature	_________	°F	__________	°F
Actual Inlet Gas Flow Rate @ Suction Temp & Pres	_________	MMcf/d	__________	MMcf/d
Displacement of Compressor Cylinder @ 90% off	_________	MMcf/d	__________	MMcf/d
Rod Load in Compression	_________	Lbs	__________	Lbs
Allowable Rod Load in Compression	_________	Lbs	__________	Lbs
Lube Oil Required on Compressor Cylinder Wall	_________	gal/d	__________	gal/d
No. oil injection points on cyl wall	4		2	
Injection rate at each point	_________	gal/d	__________	gal/d

Each packing has 1 injection point. Total number of injection points _________.
Rate per point ________. gal/d. Total injection _________ gal/d.

2. Is each cylinder equally loaded? _______ If not, what should be done? ________________________
____
3. What process irregularities (if any) are indicated from operating conditions shown above?________
____
4. Compressor is running at 100% speed. Discharge pressure is 587 psi. Gas flow rate is 10 MMcf/d. Check each correct item below:
   Suction pressure will go up/down/stay same Driver speed should be increased/reduced/stay same
   Discharge temp. will go up/down/stay same
5. The effect of opening a clearance pocket is:
   Compressor capacity is more/less/unchanged Driver power is more/less/unchanged
   Discharge gas temperature will go up/down/not change Suction pressure will go up/down/not change
6. List the order of the following events at start-up.
   _________ Open valve in suction line __________ Start driver
   _________ Purge air __________ Open valve in discharge line.
7. The vent line from the compressor distance piece on 2nd stage is hot. What is indicated?
____.
8. List 3 indications of a leaking compressor valve. ____
____.

SOLUTIONS TO PROBLEMS - ENGLISH UNITS

1. Clinder wall lubrication = 0.25 gal/day or 17 drops/min (P-10)
   No of cylinder wall injection points = 2
   Oil rate per injection point = 0.25 ÷ 2 = 0.125 gal/d or 17 ÷ 2 = 8.5 drops/min
   This is also the rate in the packing

2. 1. c
   2. g
   3. d
   4. a
   5. f
   6. b
   7. e

3. Act Vol $= \frac{0.028(\text{Std Vol})(T + 460)}{P + 15}$ (P-27)

   $= \frac{0.028(40)(460 + 90)}{305 + 15} = \underline{1.92 \text{ MMcf/d}}$

4. Displacement @ 10 dia, 8" stroke,
   600 RPM = 600 Mcf/d (Fig 1B P-32)
   Displacement @ 90% efficiency = 600 x 0.9 = 540 Mcf/d
   Std Vol $= \frac{35.4 (\text{Act Vol})(P + 15)}{(460 + °F)}$ (P-29)

   $= \frac{35.4(540)(305 + 15)}{(460 + 90)} = \underline{11\ 122 \text{ Mcf/d}}$ or 11.122 MMcf/d

5. a. CR $= \frac{945 + 15}{305 + 15} = \underline{3.0}$ (P-34)
   b. Disch. Temp = 90° + 144° = 234°F (P-35)
   c. Power = (68 HP/MMcf/d)(40 MMcf/d) = 2720 HP (P-35)

6. 1. c
   2. a
   3. b

7. Compression load = $0.785[D_P^2(P_D - P_S) + D_R^2 P_S]$ = (P-39)
   $0.785[20^2(200-50) + 3^2 \times 50] = \underline{47\ 453 \text{ lbs}}$

   Tension load = $0.785[D_P^2(P_D - P_S) - D_R^2 P_D]$ =
   $0.785[20^2(200 - 50) - 3^2 \times 200] = \underline{45\ 6871 \text{lbs}}$

8. A. 1. Lower speed
      2. Open clearance pockets

   B. 1. Close clearance pockets
      2. Raise speed

9. Start-up Sequence:
   - 3 Close vent valve
   - 2 Start driver and bring it up to operating speed.
   - 1 Purge air from unit.
   - 6 Close equalizing valve.
   - 4 Open valve in discharge piping.
   - 5 Open valve in suction piping

   Shutdown Sequence:
   - 1 Close valve in suction piping.
   - 4 Close valve in discharge piping.
   - 5 Stop driver.
   - 2 Open equalizing valve.
   - 3 Open vent valve.

10. 1. b
    2. d
    3. c
    4. a

PLP TRAINING MANUALS

NUMBER	TITLE
F-1	Basic Units of Measurement
F-2	Measurement of Energy
F-3	Hydrocarbons
F-4	Fluid Flow
E-1A	Centrifugal Compressors Part 1
E-1B	Centrifugal Compressors Part 2
E-2	Piston Type Compressors
E-3	Centrifugal Pumps
E-4	Reciprocating Pumps
E-5	Gas Engines
E-6	Fractionators
E-7	Heat Exchangers
E-8	Indirect Fired Heaters
E-9	Pneumatic Process Instruments
E-10	LACT Units
E-11	Lean Oil Absorbers
E-12	Separators
P-1	Cryogenic Gas Plants
P-2	Glycol Dehydration Process
P-3	Contactor in Dehydration Plant
P-4	Stripper in Dehydration Plant
P-5	Molecular Sieve Dehydration Process
P-6	Adsorber in Dehydration
P-7	Crude Oil Emulsion Treating
P-8	Hydrate Inhibition
P-9	Mechanical Refrigeration
P-10	Amine Sweetening Process
P-11	Contactor in Sweetening Process
P-12	Stripper in Sweetening Process
P-13	Stabiblizing Crude Oil & Condensate
M-1	Flow Measurement
M-2	The Gas/Oil Well
M-3	Oilfield Safety

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