AIR COMPRESSORS

This book is produced in full compliance with the government's regulations for conserving paper and other essential materials.

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AIR COMPRESSORS

THEIR INSTALLATION, OPERATION AND MAINTENANCE

First Edition
Second Impression

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AIR COMPRESSORS

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PREFACE

This book is intended primarily for the operating engineer and mechanical engineering student interested in air compressors and compressed air. Although many books on thermodynamics are available, they are highly theoretical and offer little practical help to the man responsible for operating and maintaining air compressors or supplying compressed air for industrial use. Since industry is becoming more air-minded every day, it is only natural that users will want to know the most economical means of supplying this air.

The author has endeavored to present the material in such a way that the reader will become familiar with all types of compressors, will know how they operate, and will learn what points to consider when called upon to purchase one.

Installation, operation, and maintenance information is presented in detail, with special emphasis on engineering practices that generally are known only to those working in the three separate branches. Knowing how to install a machine helps the operator determine what causes poor operation; on the other hand, the erecting engineer is better fitted to install a compressor if he knows what its operating requirements are. Maintenance costs can also be reduced if the repair mechanic is able to point out to the operator faulty operation that causes damage to the machine.

In compiling this information, the author was benefited considerably by twenty three years of plant experience as well as by his present opportunity to review material originally published in *Power*. He is also indebted to the Compressed Air Institute, Factory Management and Maintenance, "Compressed Air Data," and Compressed Air Magazine for material secured from these sources.

The author is indebted to his wife Claire for her help in reading proof, to his associates P. W. Swain, L. N. Rowley, S. A. Tucker, and E. J. Tangerman for their helpful suggestions and to Fred A. Annett for his encouragement and guidance. Without Mr.

Annett's inspiring confidence this book undoubtedly would never have been written.

Since this work deals with all types of available compressors, the author relied heavily on the generosity of equipment manufacturers and wishes to express his appreciation for their wholehearted cooperation which made the book possible. Technical and descriptive material was freely supplied by the following manufacturers:

Air Maze Corp., Cleveland, Ohio
Allen Billmyre Corp., Mamaroneck, N.Y.
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American Air Compressor Corp., North Bergen, N.J.
American Air Filter Co., Louisville, Ky.
American Blower Corp., Detroit, Mich.
The American Brake Shoe & Foundry Co., New York, N.Y.
American Hammered Piston Ring Div., Baltimore, Md.
Andale Co., Philadelphia, Pa.
Armstrong Machine Works, Three Rivers, Mich.
Askania Regulator Co., Chicago, Ill.
Bailey Meter Co., Cleveland, Ohio
Beach-Russ Co., New York, N.Y.
Burgess Battery Co., Chicago, Ill.
Bury Compressor Co., Erie, Pa.
Champion Blower & Forge Co., Lancaster, Pa.
Chaplin-Fulton Manufacturing Co., Pittsburgh, Pa.
Chicago Pneumatic Tool Co., New York, N.Y.
Clark Bros. Co., Inc., Olean, N.Y.
Cleveland Rock Drill Co., Cleveland, Ohio
Cochrane Corp., Philadelphia, Pa.
R. Conrader Co., Erie, Pa.
C. Lee Cook Manufacturing Co., Louisville, Ky.
Cooper-Bessemer Corp., Mt. Vernon, Ohio
Coppus Engineering Corp., Worcester, Mass.
Crane Co., Chicago, Ill.
De Laval Steam Turbine Co., Trenton, N.J.
Double Seal Ring Co., Fort Worth, Tex.
Eclipse Fuel Engineering Co., Rockford, Ill.
The Electric Sprayit Co., Sheboygan, Wis.
Elliott Company, Jeannette, Pa.
Fisher Governor Co., Marshalltown, Iowa
Foster Pump Works, Inc., Brooklyn, N.Y.
Fuller Co., Catasauqua, Pa.
Gardner-Denver Co., Quincy, Ill.
Garlock Packing Co., Palmyra, N.Y.
Hagan Corp., Pittsburgh, Pa.
Ingersoll-Rand Co., New York, N.Y.

The Johnson Corp., Three Rivers, Mich.
The Kraissl Co., Inc., Hackensack, N.J.
Lammert & Mann Co., Chicago, Ill.
Leavitt Machine Co., Orange, Mass.
Logan Engineering Co., Chicago, Ill.
Mahr Manufacturing Co., Minneapolis, Minn.
Maxim Silencer Co., Hartford, Conn.
Nash Engineering Co., South Norwalk, Conn.
National Carbon Co., Inc., Cleveland, Ohio
W. H. Nicholson & Co., Wilkes-Barre, Pa.
Nordberg Manufacturing Co., Milwaukee, Wis.
Norwalk Co., Inc., South Norwalk, Conn.
Oakite Products, Inc., New York, N.Y.
Owens-Corning Fiberglas Corp., Toledo, Ohio
Pennsylvania Pump & Compressor Co., Easton, Pa.
Roots-Connersville Blower Corp., Connersville, Ind.
A. Schrader's Sons, Brooklyn, N.Y.
Schramm, Inc., West Chester, Pa.
The Skinner Chuck Co., New Britain, Conn.
Socony-Vacuum Oil Co., New York, N.Y.
Spencer Turbine Corp., Hartford, Conn.
Staynew Filter Corp., Rochester, N.Y.
B. F. Sturtevant Co., Boston, Mass.
Sullivan Machinery Co., Michigan City, Ind.
Swartwout Co., Cleveland, Ohio
The Texas Co., New York, N.Y.
J. H. H. Voss Co., New York, N.Y.
Walworth Co., New York, N.Y.
Westinghouse Air Brake Co., Wilmerding, Pa.
Worthington Pump and Machinery Corp., Harrison, N.J.
Wright Manufacturing Div., York, Pa.
Yeomans Bros., Chicago, Ill.

Every reasonable precaution has been taken to avoid mistakes. The author will appreciate having errors called to his attention so that corrections may be made in the next revision.

EUGENE W. F. FELLER.

PORT WASHINGTON, N.Y.,
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INTRODUCTION

Air is one of the most valuable and useful tools provided for mankind. From the day when he first discovered that his lungs could be used as a compressor to supply air for enlivening his fire, man has been occupied in devising new ways of furnishing air under pressure. As he progressed through the ages his knowledge increased and he developed improvements on methods previously known. Early contrivances for compressing air were inefficient, but as methods were improved more ways were found to put air to work.

Air-treading bags, the wooden cylinder and piston, and the Chinese wind box, still common primitive methods of producing an air blast, were known at least two thousand years before Christ. The bellows was confined, in those early days, to forcing fires and to operating devices used in priestly incantations. By the time of Hero of Alexandria (150 B.C.) the water trompe, which compresses air by the fall of water in a tube, was in use for blowing forges.

Otto von Guericke made great improvements in both the compressor and the vacuum pump about 1650. William Mann, in 1829, received a patent on the compound compression of air, then called "stage pumping," which effected great economy and lighter compressor construction. Thilorier received a medal a year later from the French Academy for compressing gases to high pressures in stages.

Application kept pace with development. As early as 1683 Papin proposed transmitting power for considerable distances by compressed air. The development of his ideas resulted in such famous plants as that serving the North Star mine near Grass Valley, Calif., where Edward Rix in 1890 transmitted air from a Pelton wheel-driven compressor many miles away to Corliss engines operating the mine's hoisting machines.

Compressed air drove machine drills patented by Colladon at the Mont Cenis tunnel constructed between 1857 and 1870. The air, which was furnished by a machine driven by water power, was compressed by columns of water rising in two vertical cylinders with closed tops, according to a description published in "Appleton's Cyclopaedia of Applied Mechanics" in 1880.

Not until 1872 was cooling during compression adopted. Colladon in the St. Gothard tunnel sprayed water jets into the cylinder. This method was efficient enough but led to other difficulties which soon caused it to be discarded for water-jacketed cylinders, although some wet-type compressors were still in use during the early years of the present century.

The first compressor used on large-scale work in this country was a four-cylinder horizontal unit built in 1866 for use in the Hoosac tunnel. The cylinders were single-acting, the pistons being driven by a turbine wheel; air was admitted through poppet valves in the pistons. Water for cooling was injected through the inlet valves. This compressor was later moved to marble quarries where a stream of water flowing over the cylinder was substituted for injection cooling.

Impelled by the inventive genius of Burleigh, Ingersoll, Sergeant, Rand, Clayton, and others, compressors and compressed-air machinery were accepted at once for quarrying, mining, and tunneling. Success in these fields soon proved the value of compressed air for industrial purposes, and it has continually found new uses, until today it even serves to operate high-speed electrical circuit breakers, a field heretofore reserved strictly for electric power.

EUGENE W. F. FELLER.

AIR COMPRESSORS

CHAPTER I

RECIPROCATING COMPRESSORS

An Essential Machine.—Compressors today are essential in practically every plant where any form of gas is handled. Mechanically, aside from problems such as the type of stuffing box, selection of materials suitable for the gas to be handled, and regulation, there is little difference between an air compressor and a gas compressor. Therefore, a certain amount of standardization has been done on the basis of using available parts of standard air-compressing equipment when applying such machinery for handling gases other than air.

Of the several air-compressor types built, the reciprocating-piston machine finds widest use, largely because of its ability to meet the needs of the greatest number of compressed-air users. The most common of these is the horizontal machine (Fig. 1), although many vertical, V-, and L-angle units are built.

Classification.—Compressors may be classified according to the method of driving (1) steam reciprocating, (2) electric motor, (3) steam turbine, and (4) internal-combustion engine. They may be further classified according to the method of connection to the driving unit (1) direct-connected (close-coupled), (2) direct-connected through flexible couplings, (3) reduction-gear coupled, (4) belt-driven, and (5) en bloc, in which the power cylinders are built into the compressor frame and connect to a common crankshaft.

Common Driving Arrangements.—Steam drive is available for single and duplex horizontal double-acting compressors, with the steam engine usually built as an integral part of the compressor (Fig. 2), although a separate steam engine attached to the compressor may be used. The steam and air cylinders of single horizontal compressors are arranged in tandem on a common

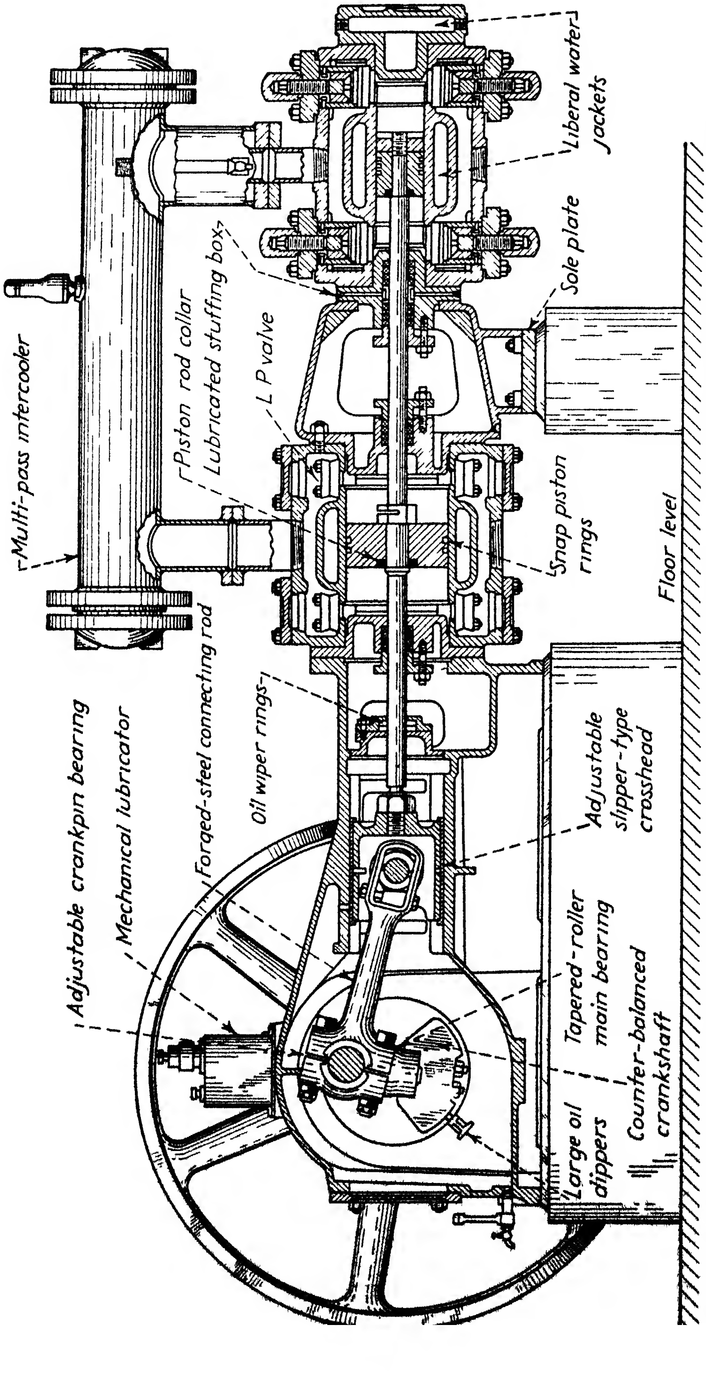


FIG. 1.—Construction details of a horizontal two-stage machine with tandem cylinders (Courtesy of Worthington Pump and Machinery Corp.)

piston rod. Steam valves are of the piston or slide type. Duplex steam-cylinder units are available for low steam pressures, or for relatively high back-pressure operation. For higher initial steam pressures compound cylinders may be used.

Electric-motor drive is available for compressors of any capacity. For driving belted and high-speed direct-coupled compressors the squirrel-cage induction motor is usually used, but occasionally the synchronous motor is employed because of its capacity for power-factor correction.

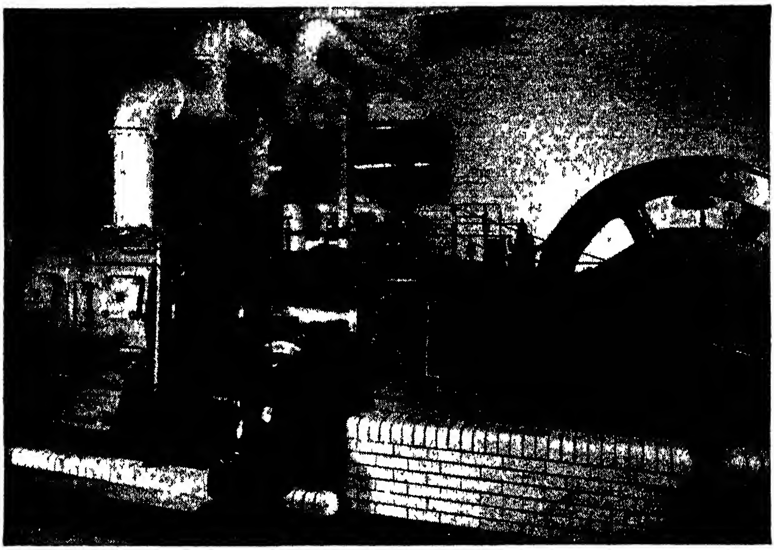


FIG. 2.—Cross-compound steam-engine-driven duplex compressor.

Direct-connected synchronous-motor drive of duplex compressors has become standard in sizes of 200 hp and larger and is quite popular even in sizes as small as 60 hp. High motor efficiency, especially at partial loads, power-factor correction, elimination of belt losses and maintenance, and reduction of floor space are factors that highly recommend the direct-connected synchronous-motor-driven compressor (Fig. 3).

Internal-combustion engines (Fig. 4) are frequently used for driving compressors, where the cost of suitable fuel is favorable or where other forms of power are not available. Small sizes may be driven by gasoline, fuel oil, or gas engines, belted or directly coupled; large sizes by an oil or gas engine built integral

FIG. 3 —Double duplex single-stage compressor driven by a synchronous motor. (*Courtesy of Worthington Pump and Machinery Corp*)

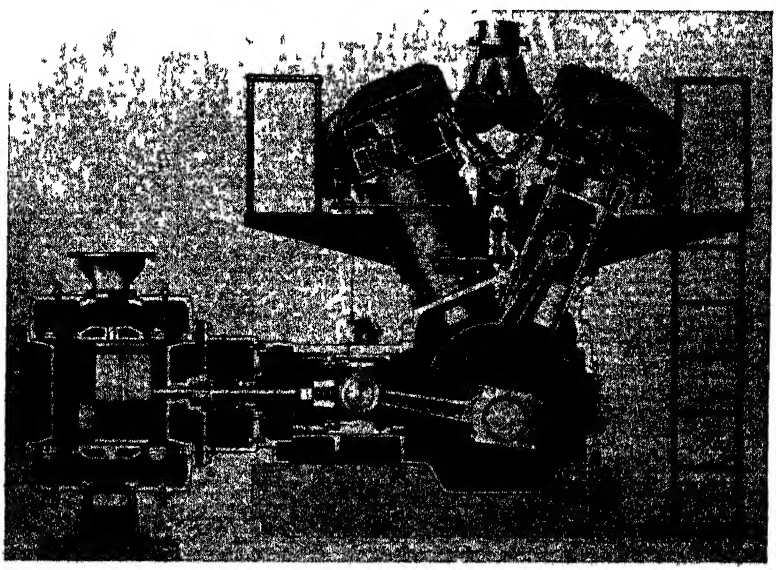


FIG. 4.—Gasoline-engine-driven compressor showing marine connecting-rod construction. (*Courtesy of Ingersoll-Rand Co.*)

or directly coupled to the compressor. The free-piston compressor at the top of Fig. 5 operates by the internal combustion of fuel between the two pistons. The opposed diesel pistons work directly on the compressor pistons so that power passes directly between them without crankshaft and connecting-rod

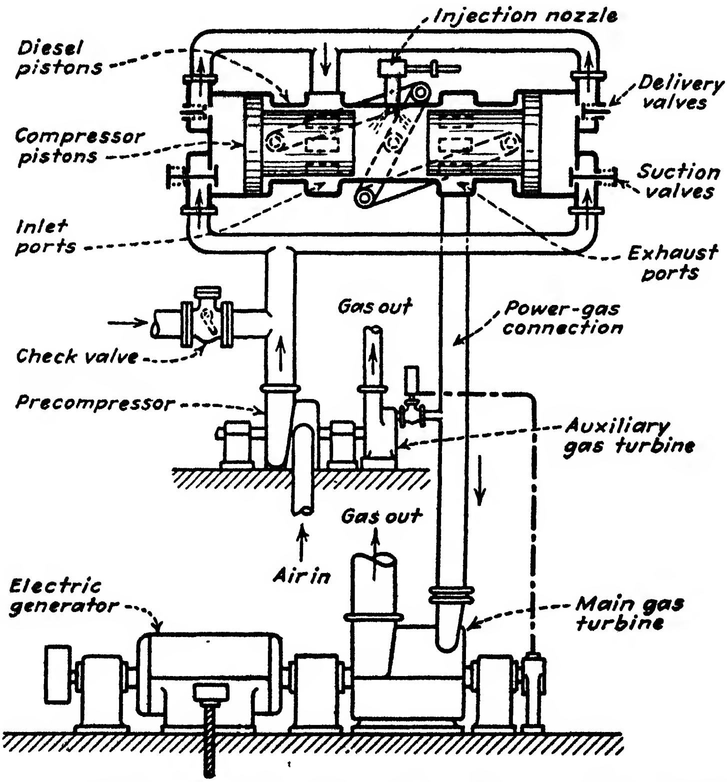


Fig. 5.—Free-piston compressor supplies exhaust gas to drive main gas-turbine generating unit and auxiliary precompressor. (*Courtesy of Power.*)

losses. The compressor pistons furnish supercharging air for the diesel, which in turn supplies exhaust gas to the gas turbine.

Compressor Shapes.—Vertical and V-type single-acting compressors (Fig. 6) may be classified into two groups.

The first group includes single-cylinder and two-cylinder, single-stage and two-stage compressors in sizes varying from fractional horsepower to 15 hp, for maximum pressures usually limited to 250 psi. Such compressors are commonly used by

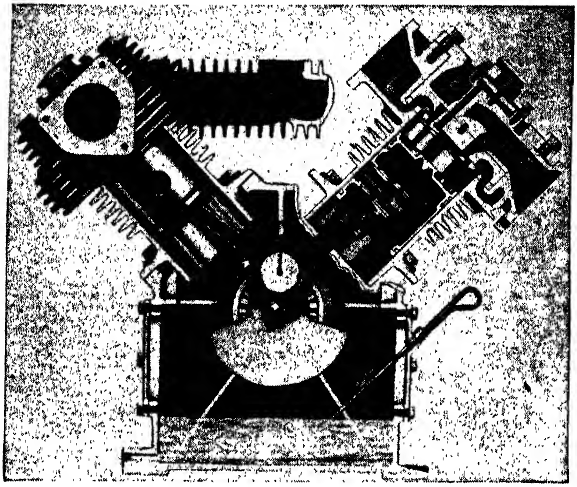


FIG. 6.—V-angle compressor with splash oiling. (*Courtesy of Sullivan Machinery Co.*)

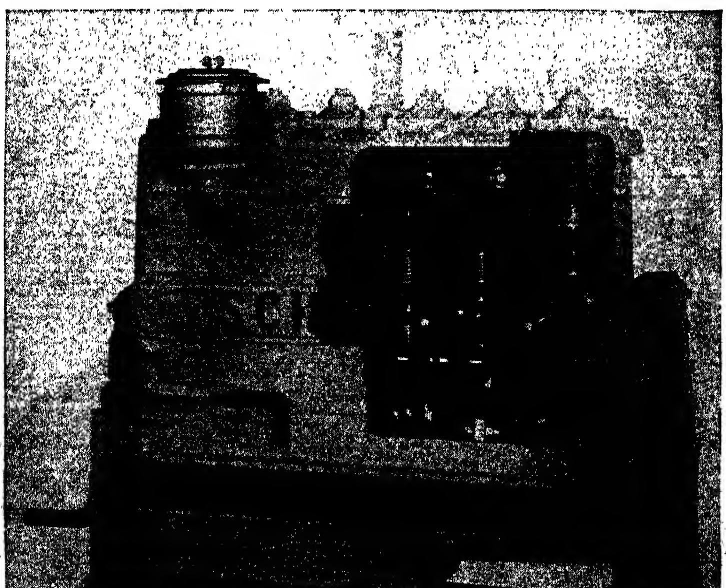


FIG. 7.—Vertical single-acting unit with mechanically operated inlet valves. (*Courtesy of Schramm, Inc.*)

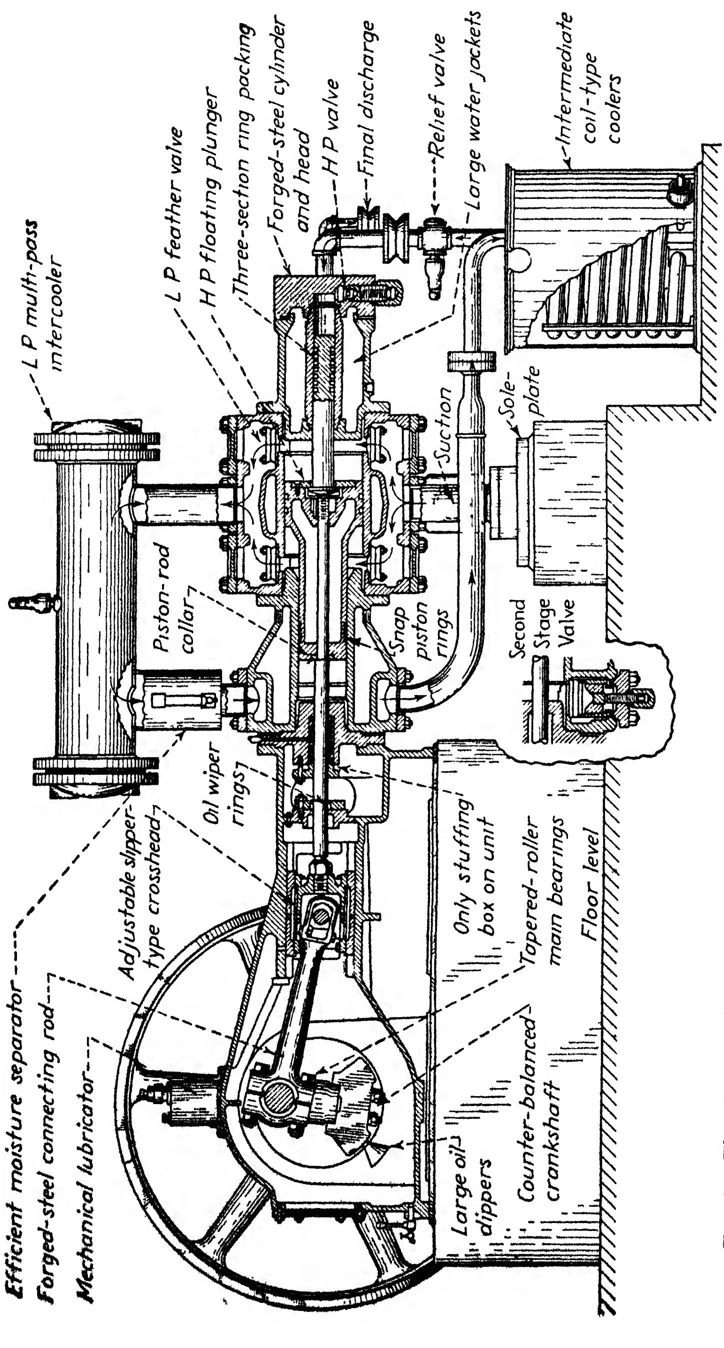


FIG. 8.—Three-stage unit with floating high-pressure piston rod. (*Courtesy of Worthington Pump and Machinery Corp.*)

filling stations and garages, for starting internal-combustion engines, and for operating laundry presses, paint sprayers, and other small jobs

In the second group are compressors of the multicylinder group (Fig 7) in sizes of 15 to 125 hp, with pressures usually limited to 125 psi This group is available either single-stage or two-stage, air-cooled or water-cooled.

These compressors may be belted or directly connected to the electric motor or other driver.

Single horizontal double-acting compressors, (Fig. 1) are of the straight-line type, with all cylinders in tandem on a common

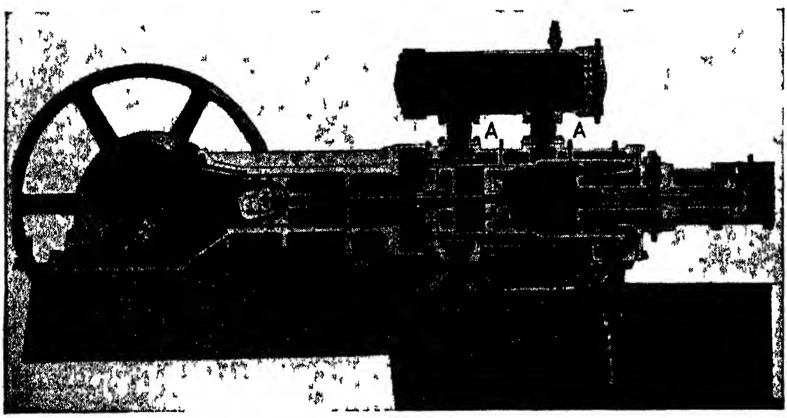


FIG. 9.—A 6,000 psi three-stage compressor with hollow low-pressure piston. (*Courtesy of Norwalk Co., Inc.*)

piston rod. They are built in approximately the same range of sizes as the second group of the vertical and V-type compressors, *i.e.*, 10 to 125 hp, but are considerably heavier and operate at much lower speed. The single-stage type is built for a maximum pressure of approximately 125 psi, the two-stage type for 500 psi, the three- (Fig. 8) for 2,500 psi, and four-stage (Fig. 9) and the five-stage units for pressures up to 15,000 psi. The types of drives regularly employed include belt, direct-connected synchronous motors, and steam end in tandem with the compressor cylinder.

Duplex double-acting compressors are built in standard sizes of 50 hp and up for a wide range of pressures, with compressor cylinders arranged horizontally, vertically, or at an angle (Fig. 10) for a wide variety of special conditions. Compressors of this type may be driven by steam, electric motors, or internal-combustion engines.

Compressor Performance.—Compressor performance guarantees are based on the temperature of the cooling water being not less than 20 F below the temperature of the incoming air,

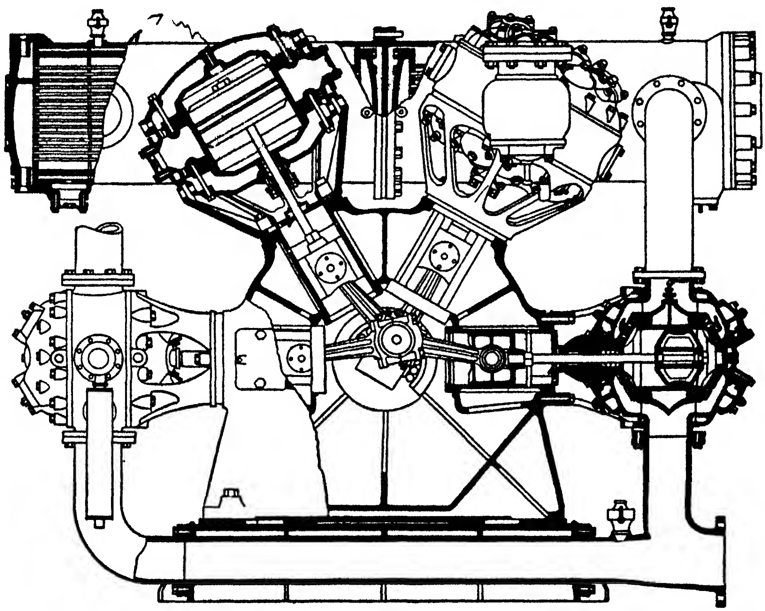


FIG. 10.—Double-acting water-cooled angle construction. (*Courtesy of Sullivan Machinery Co.*)

and the supply of intercooler cooling water being not less than 1½ gal for each 100 cu ft of free air (when a cooler is supplied), with the same quantity of water for the cylinder jacket. Capacity ratings are based on dry air at intake temperature and pressure. Because of varying conditions of installation and operation, any performance specified is guaranteed only within a variation as stipulated by the manufacturer; *e.g.*, the actual capacity for the total power consumption may vary 3 per cent, and, accordingly, the power consumed per ft of air delivered may vary 3 per cent from the performance guaranteed.

When judging the relative performance of compressors, purchasers are frequently confused by such incidental factors as mechanical efficiency and compression efficiency. Only the actual capacity of a compressor and the energy or power it requires per unit of air actually delivered should be considered, for they are the real standard of a compressor's efficiency.

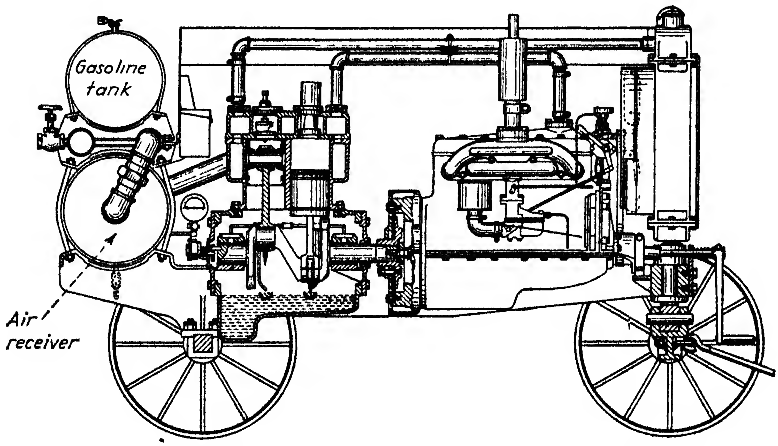


FIG. 11.—Sectional view of gasoline-engine-driven portable unit.

Definitions.—In order to simplify the description of compressors and to give the reader an understanding of various terms in common usage among compressor manufacturers, the following definitions are given:

1. *Vertical compressors* are those that have the compressing element in a vertical plane (Fig. 7).

2. *Horizontal compressors* are those that have the compressing element in a horizontal plane (Fig. 1).

3. *Angle compressors* are those of the multicylinder type having the axes of the cylinders at an angle with each other. The cylinders, mounted at a 45- or 90-deg angle, may have one cylinder vertical and the other horizontal or all cylinders at some angle (Fig. 10) from the horizontal. The V-angle and vertical machines serve for installations with limited floor space; horizontal units need more floor space but less headroom.

4. *Single-acting compressors* are those in which compression takes place on but one stroke per revolution (Fig. 6).

5. *Double-acting compressors* are those in which compression takes place on both strokes per revolution. Practically all stationary units (Fig. 1)

compress air during both the forward and return stroke of the piston. The opposite is true (Fig. 11) in portable and semiportable machines, the majority being of single-acting construction. All other things being equal, and neglecting the volume of the piston rod, the double-acting machine has twice the capacity of the single-acting unit.

6. *Single-stage compressors* are those in which compression from initial to final pressure is completed in a single compressing element.

TABLE 1.—DIESEL- AND GAS-ENGINE COMPRESSOR CAPACITIES: TWO-CYCLE RIGHT-ANGLE COMPRESSORS

Capacity in cubic feet of free air per minute

No. of 13- by 14-in. power cylinders	Rated hp	Single stage, Discharge pressure psi gage			Two stage, Discharge pressure, psi gage			
		60	80	100	60	80	100	125
Diesel-engine-driven compressor								
2	175	1,072	900	792	1,190	1,025	915	820
3	260	1,595	1,330	1,175	1,770	1,520	1,360	1,220
4	350	2,150	1,800	1,584	2,380	2,050	1,830	1,640
5	435	2,660	2,230	1,970	2,960	2,540	2,275	2,040
6	525	3,220	2,700	2,376	3,570	3,075	2,745	2,460
Gas-engine-driven compressor								
2	230	1,410	1,185	1,045	1,565	1,350	1,200	1,080
3	345	2,095	1,750	1,545	2,330	2,000	1,790	1,610
4	460	2,820	2,370	2,090	3,130	2,700	2,400	2,160
5	575	3,500	2,940	2,590	3,900	3,340	3,000	2,690
6	690	4,230	3,555	3,135	4,695	4,050	3,600	3,240

7. *Multistage compressors* are those in which compression from initial to final pressure is completed in two or more stages (Fig. 9).

A choice between single- and multistage compression depends upon many widely varying factors, such as size of compressor, ratio of compression, discharge-temperature limitations, cost of power, continuity of service, and relative permanence of installation.* The Compressed Air Institute suggests that, in general, the dividing line may be drawn as follows, assuming sea-level atmospheric intake pressure: for purposes below 60 psi use single-stage; for 60 to 100 psi, single-stage for capacities below 300 cfm and two-stage for larger sizes.

When compressing air in excess of 300 cfm at a pressure of 100 to 125 psi, it is advisable to use two-stage machines because they use 11 per cent less power than single-stage at 300 cfm capacity, 14 per cent less at 450 cfm, and 19 per cent less at 1500 cfm—all capacities at 100 psi delivery. When one year's power cost for operating a compressor approaches its initial cost, economy of operation is of utmost importance.

Externally some multistage compressors (Fig 9) appear to have only one cylinder because the design is such that the multistage compression takes place inside one common casing. This is an unusual construction, because normally the cylinders mounted separately can be distinguished by their difference in size.

8. *Duplex compressors* have two parallel sets of compressing elements driven by individual cranks on a common crankshaft (Fig. 12). Double

FIG. 12.—Duplex two-stage compressor with synchronous-motor drive. (*Courtesy of Chicago Pneumatic Tool Co.*)

duplex units form the letter H, having a cylinder at the end of each leg with the driving unit forming the crossbar (Fig. 3). Duplex machine cranks placed at an angle of 90 deg reduce power pulsations to a minimum.

9. *Ratio of compression* is the ratio of absolute discharge pressure to absolute intake pressure. This may be further qualified to the ratio for any particular stage or to the compressor as a unit.

10. *Displacement* of a compressor is the volume swept through per unit of time by the first stage piston, or pistons, expressed in cubic feet per minute (cfm). In double-acting compressors, it is the volume swept through by both sides of the piston.

11. *Free air* is air at atmospheric pressure and temperature at the place the compressor is installed. Standard air is air at 68 F, 36 per cent relative humidity, and 14.7 psi abs.

12. *Actual capacity* of a compressor is the quantity of air actually delivered and compressed. For low- and medium-pressure compressors it is expressed

in cubic feet of free air per minute. The actual capacity of a compressor is always less than its displacement.

13. *Volumetric efficiency* is the ratio of actual capacity to the displacement. For any given compressor the volumetric efficiency decreases with

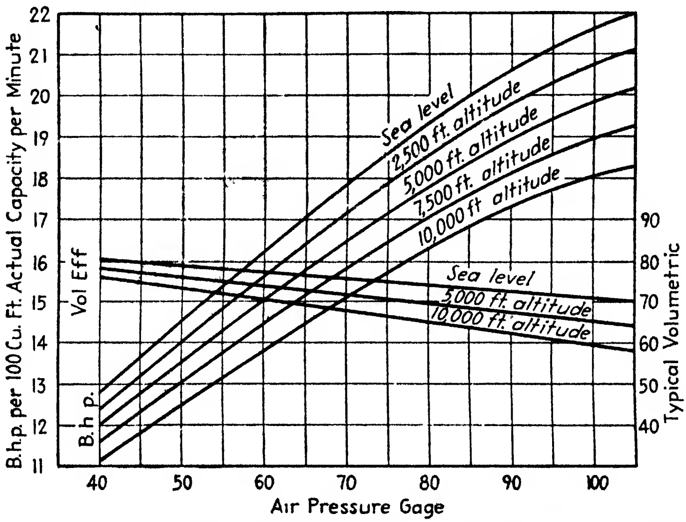


FIG. 13.—Brake horsepower required by single-stage compressor. (*Courtesy of Worthington Pump and Machinery Corp.*)

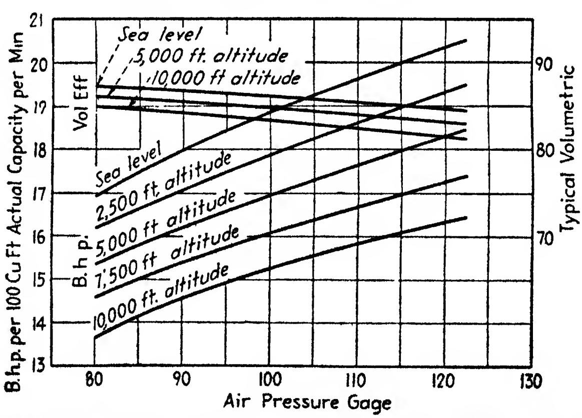


FIG. 14.—Brake horsepower required by two-stage compressor. (*Courtesy of Worthington Pump and Machinery Corp.*)

an increase in the compression ratio. Theoretically, volumetric efficiency has no influence on the power required by the compressor.

14. *Compression efficiency* is the ratio of the theoretical horsepower of compression to the actual indicated horsepower of the compressor cylinders.

This efficiency is affected by the design and proportioning of the inlet and discharge valves, the cooling of the cylinders, the ratio of compression, and the speed of the compressor.

15. *Mechanical efficiency* is the ratio of the indicated horsepower in the compressor cylinders to the brake horsepower in the case of a motor- or turbine-driven unit, or to the indicated horsepower in the power cylinders in the case of steam-driven or internal-combustion-engine-driven compressors. Mechanical efficiency is affected by the power lost in overcoming the friction of bearings, packing boxes, and pistons.

16. *Over-all efficiency* is the product of the compression efficiency and the mechanical efficiency. It is the ratio of total output to the total power input and shows the actual cost per unit output.

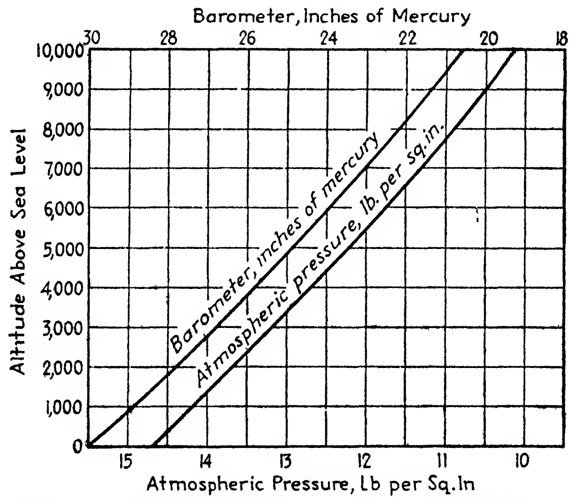


FIG. 15.—Barometric pressures at various altitudes. (*Courtesy of Worthington Pump and Machinery Corp.*)

17. *Theoretical horsepower* (adiabatic base) is the horsepower required to compress air adiabatically through the specified pressure range. In a multistage compressor it is the sum of the horsepower calculated for each stage.

18. *Air-indicated horsepower* is the horsepower calculated from a cylinder-indicator diagram.

19. *Brake horsepower* is the measured horsepower input to the compressor shaft (Figs. 13 and 14).

20. *Absolute pressure* is the total pressure measured from absolute zero. It equals the sum of the gage reading plus atmospheric pressure in psi corresponding to the barometer (Fig. 15).

21. *Absolute temperature* equals the degrees Fahrenheit plus 459.6 or, as more commonly used, 460.

22. *Air-cooled compressors* have radiating fins cast on the cylinder walls to produce cooling of the cylinder by circulation of atmospheric air (Fig. 6).

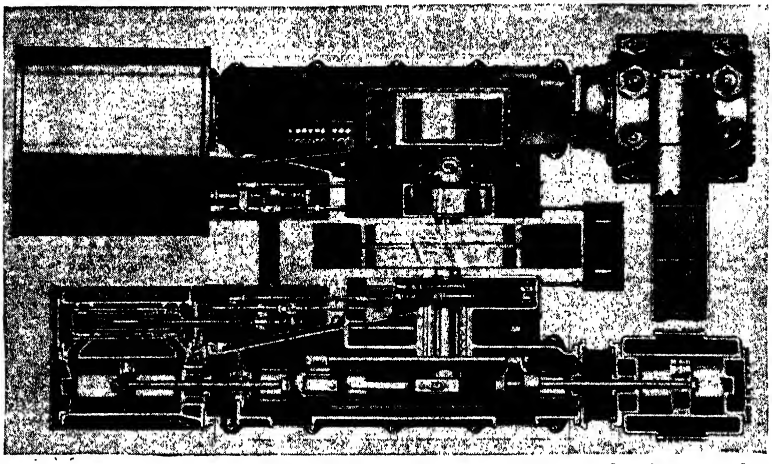


FIG. 16.—Plan view of a steam-driven two-stage compressor showing stretcher rods. (*Courtesy of Ingersoll-Rand Co.*)

FIG. 17.—Hopper-jacketed construction for cylinders and intercoolers. (*Courtesy of Norwalk Co., Inc.*)

23. *Water-cooled* compressors have the cylinders cooled by water circulation through jacketed spaces (Fig. 16).

24. *Hopper-jacketed* compressors are constructed so that the jacketed space forms an open reservoir (Fig. 17).

25. *Adiabatic compression* occurs when no heat is transferred to or from the air being compressed.

26. *Isothermal compression* occurs when heat is removed at such a rate that the air temperature remains constant.

TABLE 2.—SINGLE-ACTING COMPRESSOR SPECIFICATIONS

	Compressor model number		
	85	210	420
Compressor bore and stroke, in.	$3\frac{3}{8}$ by $4\frac{1}{2}$	$4\frac{1}{2}$ by $4\frac{3}{4}$	$5\frac{5}{8}$ by 6
No. compressor cylinders	4	6	6
No. main bearings	5	7	7
Main-bearing diameter, in.	3	3	$3\frac{1}{2}$
Air-discharge fitting—pipe size in.	2	$2\frac{1}{2}$	3
Standard horizontal tank size, in.	16 by 42	18 by 48	20 by 60
Standard vertical tank size, in.	18 by 48	24 by 72	30 by 84
Direct-motor-drive specifications:			
Piston displacement per min	109	294	596
Actual air delivery at 100 lb	75	206	416
Horsepower required for maximum delivery	20	50	100
Operating speed—rpm for maximum delivery	1,170	1,175	1,170
V-belt-drive specifications:			
Piston displacement per min	124	300	601
Actual air delivery at 100 lb	85	210	420
Horsepower required for maximum delivery	20	50	100
Operating speed, rpm—compressor	1,330	1,200	1,180
Operating speed, rpm—motor	1,760	1,765	1,170

Elements of an Air Compressor.—Briefly, compressors consist of the following elements:

1. The compressing element, made up of air cylinders, heads, and pistons, together with inlet and discharge valves.

2. A system of connecting rods, piston rods, crossheads, crankshafts, and flywheel to transmit the power developed by the driving unit to the air-cylinder piston.

3. A self-contained lubricating system for bearings, gears, and cylinder walls, including a reservoir or sump for the lubricating oil, a pump or other

means of delivering oil to the various parts, suitable filters, and coolers. On most compressors a separate force-feed lubricator supplies oil to the compressor cylinders

4 A cooling system for removing heat from the cylinder and heads, intercoolers, and aftercoolers

5 A regulating or control system designed to maintain the pressure in the discharge line and receiver within a certain predetermined pressure range.

6 An unloading system which operates in conjunction with the regulators to reduce the load or eliminate it entirely

Stationary construction follows all the previously discussed designs, with portable and semiportable machines being built mostly in the V and vertical style. Semiportable units, usually

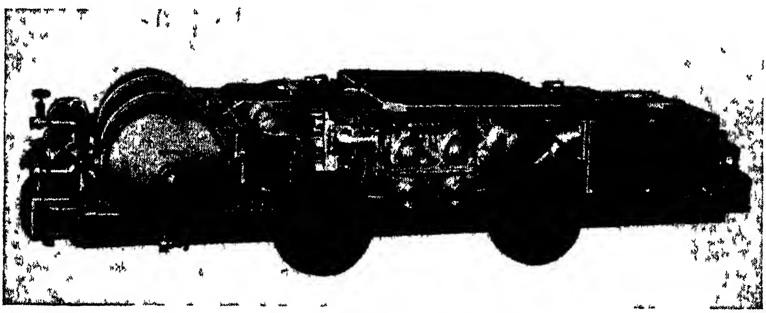


Fig. 18.—Complete compressor station installed on a mine car. (*Courtesy of Ingersoll-Rand Co*)

mounted on skids to serve as a foundation, permit moving the compressor from place to place at infrequent intervals. The portable unit mounted on its own truck (Fig 11) does not require special hoisting facilities for moving; this makes frequent transporting easy. A portable compressing plant for mine service is shown in Fig. 18.

Cylinder Details.—Cylinder design varies among different manufacturers. Some cylinders carry the inlet and discharge valves (Fig. 19) while in others the valves are contained in extensions cast integral with the cylinder heads (Fig. 20). The heads then form part of the cylinder barrel but do not serve as a sliding surface for the piston. Casting the cylinder with triple walls (Fig. 21) provides space for cooling-water circulation, air intake and discharge chambers, and clearance pockets for load control. Cylinder heads having double walls (Fig. 19) form water chambers, sometimes connecting to those in the cylinder walls, for circulating cooling water.

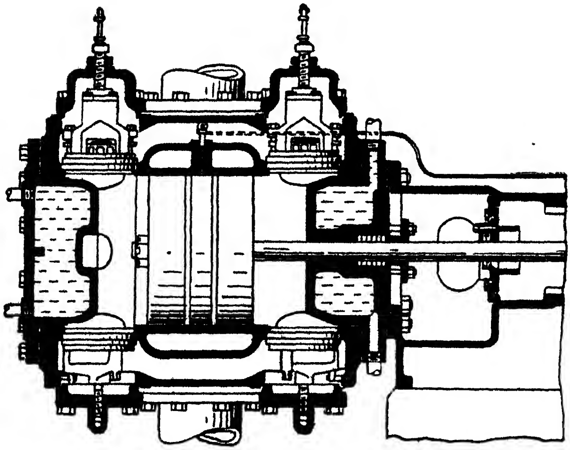


FIG. 19.—Inlet and discharge valves carried in cylinder walls. (*Courtesy of Power.*)

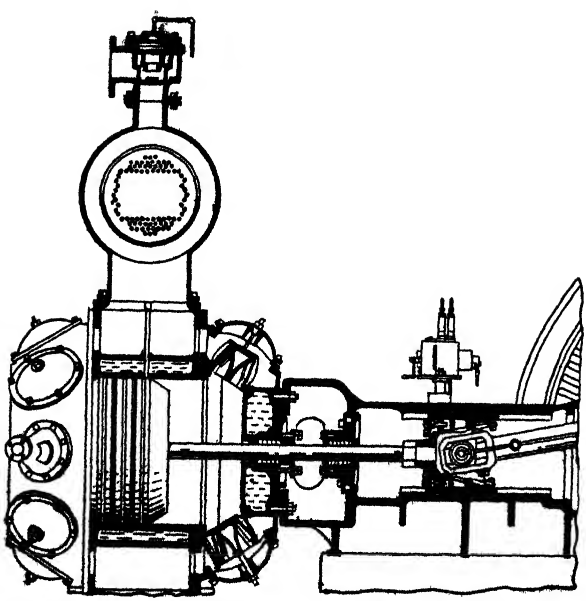


FIG. 20.—Inlet and discharge valves carried in extensions on cylinder heads. (*Courtesy of Power.*)

Many compressors of smaller size, as well as numerous portable and semiportable compressors, use atmospheric-air circulation for cylinder cooling. Fins cast integral with the cylinder furnish additional radiating surface (Fig. 6). The circulation of cooling water between the cylinder walls or, in an air-cooled machine, of air over the cast fins, removes only a small quantity of heat from the compressed air but does serve to keep the cylinders and valves at a reasonably low working temperature.

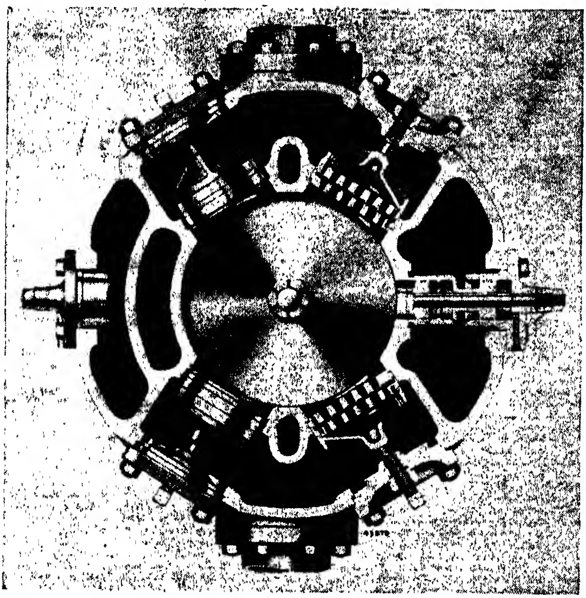


FIG. 21.—End view showing water jackets, inlet and discharge chambers, and clearance pockets. (*Courtesy of Ingersoll-Rand Co.*)

The pressure duty of the compressor determines the kind of metal used for the cylinder casting. Close-grained gray cast iron is commonly used and serves for pressures as high as 1,000 psi. Other metals, including semisteel, nickel alloy, tool steel, cast steel, and forged steel (Fig. 8), serve for higher pressures.

Grinding, honing, or the use of special boring tools produces a satin-finish surface on the cylinder walls. All integral cylinder barrels permit at least one reboring (Fig. 22), or, according to several manufacturers, $\frac{1}{8}$ in. of metal may be removed without danger of weakening the walls. Separate slip-fit barrel inserts

(Fig. 23), being removable, may be replaced with new ones when wear becomes excessive.

Small horizontal compressor units permit overhanging the cylinders from the main frame, while large and heavy construction requires a soleplate cylinder support (Fig. 3).

Valve Design.—Valves are the vital part of a compressor. More advance has been made in valve design and improvement than in any other part of the machine. With but few exceptions

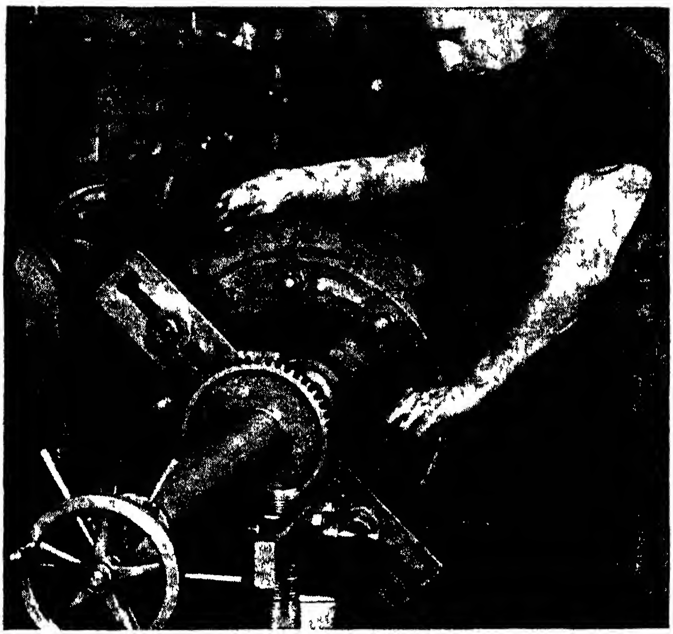


Fig. 22.—Boring-bar setup for truing cylinder walls. (*Courtesy of American Air Compressor Corp.*)

the mechanically operated and heavy automatic poppet valves have disappeared from use. Those still remaining, being much lighter, require little power for their operation. One manufacturer uses a light poppet-inlet valve operated by a camshaft arrangement (Fig. 24).

Old compressors having mechanically operated valves are still in service, although many of them have been modernized by installing new cylinders fitted with plate valves or by putting new plate valves in the old cylinders. Discarding the mechani-

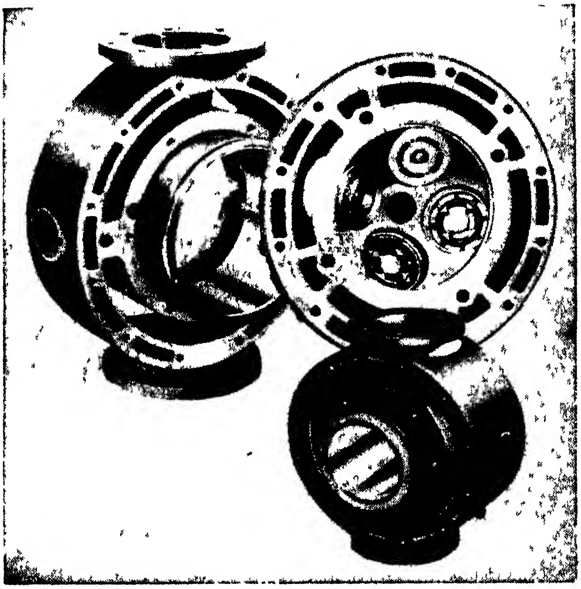


FIG. 23.—Cylinder constructed with removable liner. (*Courtesy of Sullivan Machinery Co.*)

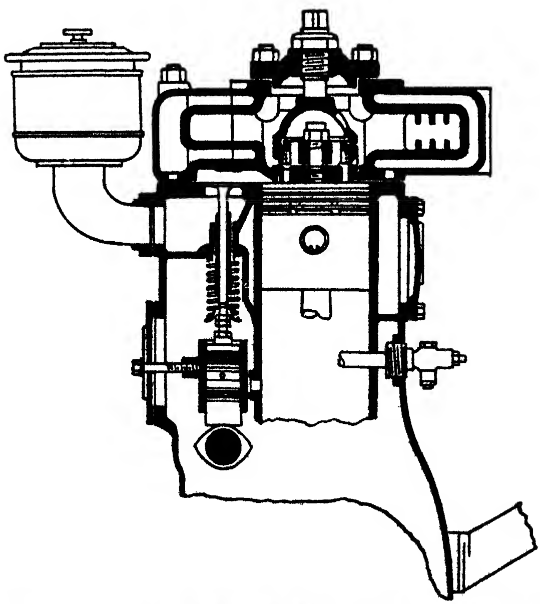
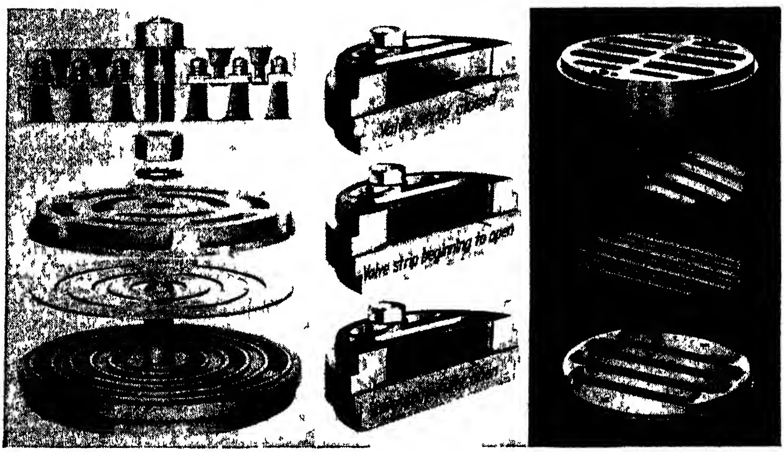


FIG. 24.—Cross section of vertical single-acting machine with mechanical inlet valves. (*Courtesy of Schramm, Inc.*)

cally operated valve did away with the need of indicating the air cylinder to determine compressor performance.

With this old valve arrangement the valve action is controlled by a valve gear operated by the reciprocating or rotating parts of the compressor. The control may be complete, the actual opening and closing of the valve taking place at exactly the same point in the stroke each revolution; or it may be only partial, in which case, at the proper time, the valve starts to close, but the actual closing depends upon the difference in pressure between the cylinder and external air. This action, being similar to that



a *b* *c*

FIG. 25.—Different plate valve constructions. (*a*, *Gardner-Denver*, *b*, *Worthington*, *c*, *Ingersoll-Rand*)

in a steam engine, shows the necessity of using an indicator when checking the valve setting on these older units.

Modern Valves.—Practically all valves in present-day compressors are of the automatic type, consisting of flat steel plate. Plate shapes (Fig. 25) vary with the different manufacturers; some use circular plates, others flat concentric rings, and still others rectangular strips or channels.

The valve in Fig. 26 uses a spring having a number of equally spaced flexures or waves, as has also the reaction plate. There are two waves in the reaction plate for each wave in the valve spring.

The top figure shows the valve closed and the valve spring resting against the bottom of alternate reaction-plate waves.

The unsupported length or effective lever arm of the valve spring is X.

As the valve opens, the valve spring begins to flatten and its reaction points on the reaction plate change from the bottom of the reaction-plate waves to points along the upward slopes.

On further opening of the valve, the spring continues to roll up the slopes of the reaction plate until it rests upon the tops of all the reaction-plate waves as shown in the bottom figure. The unsupported length of the valve spring, or its effective lever arm,

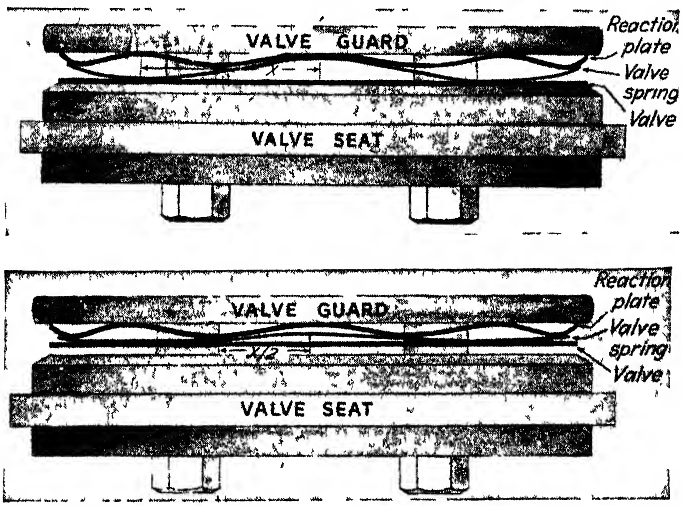


FIG. 26.—Valve construction with reaction plate in addition to regular spring (*Courtesy of Sullivan Machinery Co.*)

has now shortened from the original distance X to a distance $X/2$, or half as much. The result is that the stiffness of the valve spring becomes approximately twice as great after the valve has opened part way as it would have been had the wave-type reaction plate not been present.

Advantage of Thin Plates.—Thin-plate low-lift valves consist of a valve seat, valve plate, valve spring, and valve guard (Fig. 27). They have the following advantages over other valves: (1) Large areas for the passage of air permit a low velocity through the valve and hence improve the compression efficiency. (2) They open with minimum resistance and close promptly, thereby keeping the range of pressure inside the cylinder within limits close to those to which the compressor is designed. (3) Because of their light weight and low lift, they operate with a minimum of noise. (4) They are simpler and can be replaced quickly and cheaply.

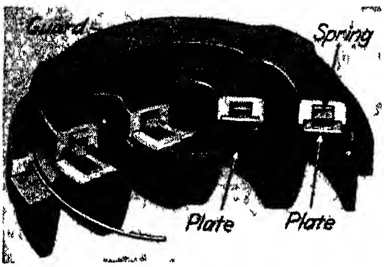


FIG. 27.—Valve using coiled plate springs. (*Courtesy of Pennsylvania Pump & Compressor Co.*)

Some plates are held in place by coil springs (Fig. 28); others use circular or rectangular leaf springs, while still others are held loosely in place without any spring action. Care has been taken to produce a lightweight valve having plenty of passage area; this reduces velocity friction and pressure loss, as well as lowering the pressure needed to force the valve open. The opening and closing of the valve is caused solely by pressure differential between the air within the cylinder and the external air on the opposite side of the valve.

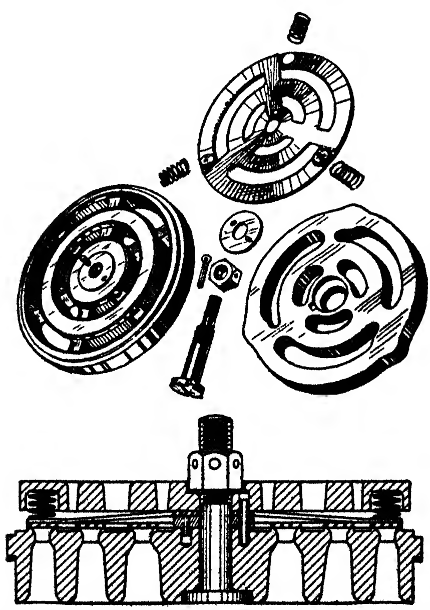


FIG. 28.—Plate valve with coiled-wire springs.

High temperatures require metals able to withstand heat without damage. This means that valve plates must be made from chrome-vanadium steel, stainless steel, or monel metal. Corrosive conditions require valve parts made from composition bronze.

TABLE 3.—TANDEM AND DUPLEX TWO-STAGE UNITS

Displacement, cfm		Rpm		Max pressure continuous service	Cylinder size			Air pressure	Bhp at max speed	Pipe openings, in.		
Max	Min	Max	Min		L-p cyl.	H-p cyl.	Stroke			Suction	Discharge	Water
Tanden two-stage												
26	10 4	870	350	350	4½	1⅞	4	250 300 350 500*	7 3 7 5 7 6 8 0	1¼	½	¾
Duplex two-stage												
40	13 6	870	300	300	5	2¼	4	150– 300	10 3–12 1	2	1	1
40	13 6	870	300	600	5	2	4	300– 600	12 0–14 1	2	1	1
40	13 6	870	300	1,000	5	1⅞	4	600–1,000	14 0–16 8	2	1	1
65 6	22 6	870	300	300	5¾	2¾	5	150– 300	17 0–20 1	2½	1	1¼
65 6	22 6	870	300	600	5¾	2¼	5	300– 600	19 5–22 3	2½	1	1¼
65 6	22 6	870	300	1,000	5¾	1⅞	5	600–1,000	21.9–23 9	2½	1	1¼
92	31 7	870	300	300	6½	3¼	5½	150– 300	24 0–29 4	3	1¼	1½
92	31 7	870	300	600	6½	2¾	5½	300– 600	28 5–34 9	3	1¼	1½
92	31 7	870	300	1,000	6½	2¼	5½	600–1,000	34 0–40 0	3	1¼	1½

* Horsepower shown at 500 lb is for intermittent service only.

Valve tightness depends upon a narrow seat around the periphery of the plate (in most valves $\frac{1}{16}$ in.). This means that the plate and seat faces must be perfectly flat and made from material that will not warp.

Valve area as well as plate lift affects the pressure drop of the air flowing into and from the cylinder. The lift must be kept low to prevent the plate from hammering itself to pieces. Other factors that contribute to compression loss are valve slip and frictional resistance to air flow through valves and passages.

Valve slip is that volume of air which, when the piston has reached the end of the compression stroke, slips back into the cylinder before the valve can close tightly. Whether this occurs on suction or on discharge, a corresponding loss occurs in the amount of air delivered by the compressor. The discharge valve cannot close before the piston has reached the extreme end of its travel, yet the valve must be tightly closed before the piston starts on its return stroke. It is obvious, then, that the valve must close in a small fraction of the time of a compressor stroke.

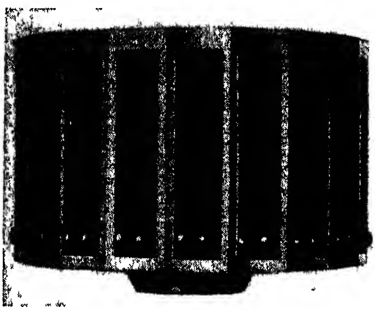


Fig. 29.—Pyramid-box plate valve (*Courtesy of Bury Compressor Co.*)

Speed of Valve Operation.—Consider the case of a compressor operating at 257 rpm (revolutions per min), which is equivalent to $4\frac{1}{3}$ revolutions, $8\frac{2}{3}$ strokes per sec, or one stroke in $\frac{1}{8}$ sec. Assuming that the discharge valve, to be efficient, must function within $\frac{1}{100}$ of the stroke, it follows that the valve must close

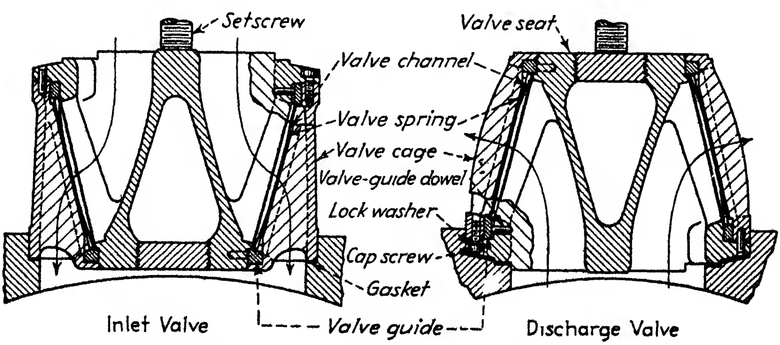


Fig. 30.—Cross section showing turret channel valve construction.

within $\frac{1}{800}$ sec, an exceedingly short period of time. It can be appreciated then that the valve must be instantly responsive, if blowback is to be avoided. Speeds such as this require plates with small inertia yet with strength to withstand terrific punishment.

Another important factor that must be recognized in accounting for the losses which combine to lower compression efficiency is friction or resistance to the flow of air. To make any air or liquid flow through a pipe or restricted passage, a difference in pressure between the intake and outlet points is necessary. Just how great a difference is required depends upon the amount of the restriction and the volume being forced through. When air flows through the valve of a compressor there is some restric-

TABLE 4.—SINGLE AND DUPLEX COMPRESSORS

Displacement cfm: Max speed	Displacement cfm: Min speed	Rpm: Max	Rpm: Min	Max pressure continuous service	Cylinder size: Diameter	Cylinder size: Stroke	*Bhp at max speed, Air pressure: 60	80	100	125	150	175	200	When used as vacuum pump: Max vacuum (30 in barometer)	Min h p motor recommended	Pipe openings in: Suction	Discharge	Water
Single Cylinder																		
32	11	870	300	150	4½	4	4.8	5.3	5.5	5.8	6	**6.2**	**6.4**	28.2	3	1¼	1¼	1
42.6		1160		100	4½	4	6.8	7.4	7.8							1¼	1¼	1
Duplex Compressor																		
64	22	870	300	150	4½	4	9.7	10.5	11.0	11.6	12	**12.4**		28.2	5	1½	1½	1
85.3		1160		100	4½	4	13.5	14.6	15.4							1½	1½	1
79	27	870	300	125	5	4	12.4	13.4	14.2	14.8	**15.4**			28.2	5	2	2	1
105.2		1160		100	5	4	17.4	18.7	19.9							2	2	1
130	45	870	300	125	5¾	5	21.0	23.0	24.3	25.5	**26.7**			28.5	10	2½	2½	1¼
184	63	870	300	125	6½	5½	28.5	31.1	33.3	35.5	**38.**			28.7	15	3	3	1½

* Bhp is at belt wheel and does not include belt loss. Horsepowers shown in heavy type are for intermittent service only.

tion, causing frictional loss, no matter how liberal the valve area or how light the valve may be, since there must be some slight pressure differential or the valves will not open. Naturally, large valve and port areas are very important to obtain high efficiency. The additional energy required to force the air through the compressor valves and passages is energy uselessly expended and reduces the compression efficiency by a corresponding amount.

Piston Construction.—Pistons, usually constructed from semi-steel, close-grained cast iron, or heat-treated aluminum, are ground to a close fit in the cylinder. Clearance between the piston and cylinder averages 0.001 in. per in. diameter on pistons up to 20 in.; above this size clearance increases 0.0005 in. per in. diameter. One manufacturer uses an aluminum low-pressure and an alloy-iron-tinned high-pressure piston (Fig. 10). Aluminum pistons, because of their light weight, reduce the shaking force of the unit in locations where this is undesirable. An unusual occurrence was discovered in a horizontal compressor fitted with aluminum pistons. After several years' operation the piston clearance, rather than increasing, was found to be less than half of the original clearance. Either squeezing or heat action and unequal expansion had caused the piston diameter to increase several mils.

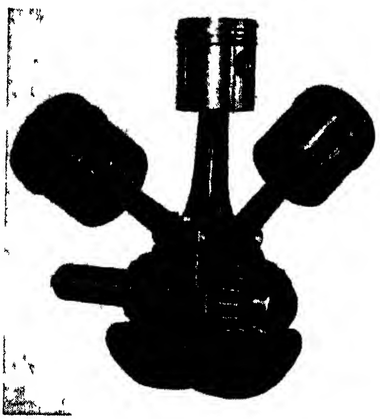


Fig. 31.—Single-acting vertical two-stage crank assembly (*Courtesy of Worthington Pump and Machinery Corp.*)

Trunk pistons (Fig. 31), used mostly on single-acting compressors, are driven directly by the connecting rod. They are fitted with long skirts that transmit the guide pressure directly to the cylinder walls to avoid slap, as the piston itself acts as a crosshead. Differential pistons (Figs. 8 and 9) are modified trunk pistons having two different diameters that fit into special cylinders so arranged that two stages of compression are served by one piston, compression for one stage taking place over the piston crown, and the stage being the annular space between the large and small diameters of the piston.

Double-acting pistons are, as their name implies, required to compress air on both strokes per revolution. These require some form of piston rod and crosshead so that the compressing element nearer the running gear may be properly sealed against leakage. Double-acting pistons are made either solid or hollow with either box or truncated shapes. Truncated pistons (Figs. 10 and 20) permit increasing the valve area and using cylinder shapes, which reduces end clearance and increases the volumetric efficiency. Some manufacturers drill small relief holes through the piston (Fig. 32) to the underside of the rings to equalize air pressure acting on the ring, which permits free action of ring tension against the cylinder walls. Piston ends must conform to the cylinder shape to keep the clearance volume at a minimum. This sometimes means truncated or tapered ends rather than a flat disk.

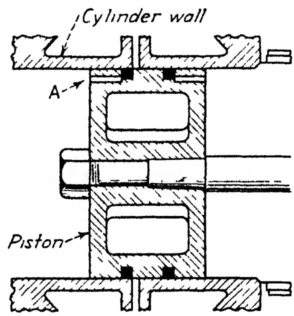


Fig. 32.—Piston with equalizing holes leading to ring grooves.

Built-up pistons are sometimes made with L-section ring carriers, but it is found that the large amount of metal removed in cutting the grooves causes these carriers to twist and warp sufficiently to prevent the sides of the grooves from forming true bearing surfaces for the piston rings. To overcome this twisting of the ring carriers, flat spacers and separators of rectangular cross section which are entirely free from warpage are used. These are assembled on the piston rod into a rugged and effective unit, forming accurate ring-carrying grooves as shown in Fig. 33. The simplicity of construction ensures maximum strength of each part and permits easy and thorough cleaning during overhaul.

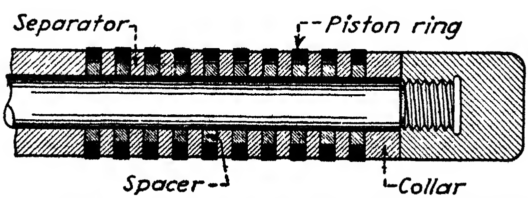


Fig. 33.—Built-up high-pressure piston showing spacers, separators, and rings.

With this type of construction the piston rings may be installed during assembly of the piston, thus eliminating the necessity of springing the rings over the piston. Rings of extremely heavy wall thickness with proportionately longer life can therefore be used.

Spacers and separators are finished with smooth surfaces on the outside and inside, and are accurately ground on the sides to a tolerance of 0.0005 in. Thus, the piston rings may be fitted to an accurate predetermined clearance; consequently, side wear on both rings and grooves will be held to a minimum.

TABLE 5.—DIESEL-ENGINE COMPRESSOR SPECIFICATIONS
125 lb. discharge pressure

Rating, cfm		Engine rating, bhp	Compressor cylinders				Rpm
Piston displacement	Actual capacity		No.	L-p dia., in.	H-p dia., in.	Stroke, in.	
770	662	142	2	13	8	7	720
1,026	880	190	2	15	9	7	720
1,170	1,000	200	2	18	11	10	400
1,450	1,225	250	2	20	12	10	400
1,850	1,563	300	4	16	10	10	400
2,340	2,000	400	4	18	11	10	400

Piston-ring Details.—Metal rings (Fig. 34*A*) fitted in grooves around the piston circumference prevent leakage between the piston and cylinder walls. The pressure requirements determine the number of rings needed; many manufacturers use one- or two-piece snap rings, while others use segmental-section rings with internal expander. Two-piece snap rings sometimes ride on an internal expander. Taper or step joints (Fig. 35) prevent leakage past the ring ends. Clearance between the ring and groove, approximately 0.003 in., is just sufficient to permit the ring to drop into the groove by its own weight. Single-acting compressors have from three to eight compression rings installed adjacent to the compression chamber. Oil-control or scraper rings remove excess oil from cylinder liners and return it to the crankcase.

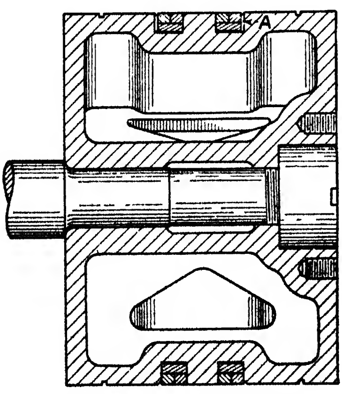


FIG. 34.—Hollow-ribbed piston carrying three-piece rings.

Requires Strong Material.—Piston-ring material should be strong, tough, resilient, and slow wearing, with freedom from scoring tendencies. For normal compressor application, cast iron is almost universally used, although bronze and other materials serve for special conditions. For the sake of heat conductivity, it is desirable to have as thick a ring wall as possible, because this is more effective than ring width. Narrow rings distribute their heat over the full length of travel in the cylinder. The amount of heat passing through the rings depends primarily on the contact between the rings and the piston rather than between the rings and the cylinder. One manufacturer recommends a copper-lead ring when it is desirable to operate the

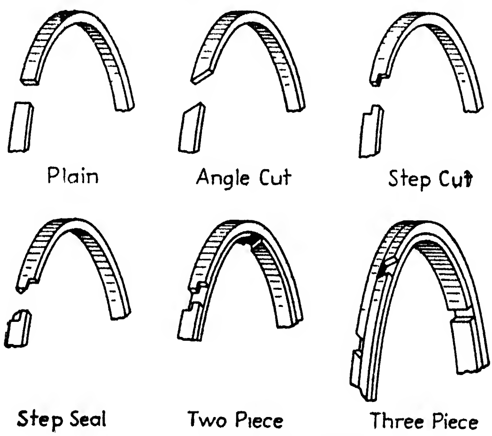


FIG. 35.—Typical ring joints.

compressor with minimum lubrication. Auxiliary copper-lead or bronze rings installed with regular rings serve to polish the cylinder walls. Rings are also available with chemically treated surfaces to promote rapid seating during run-in.

Special Service.—When compressors must furnish oil-free air, no oil is allowed to enter the cylinder; rubbing friction between the piston and cylinder is reduced by replacing the metal piston rings with others constructed from carbon (Fig. 36). In *B* the piston itself is supported by a third carbon ring, while in *A* the piston rod passes through the piston and front head to a sliding bearing outside. The piston fitted with carbon segmental rings then rides between the main crosshead and sliding bearing.

Ring Characteristics.—In order for a ring to function properly, it must be truly flat on the side faces so that there are no irregularities to permit leakage between the face of the ring and the side of the groove. It must also conform to the cylinder so that there is no space for leakage between the outside circumference of the ring and the cylinder bore. These points are well understood,

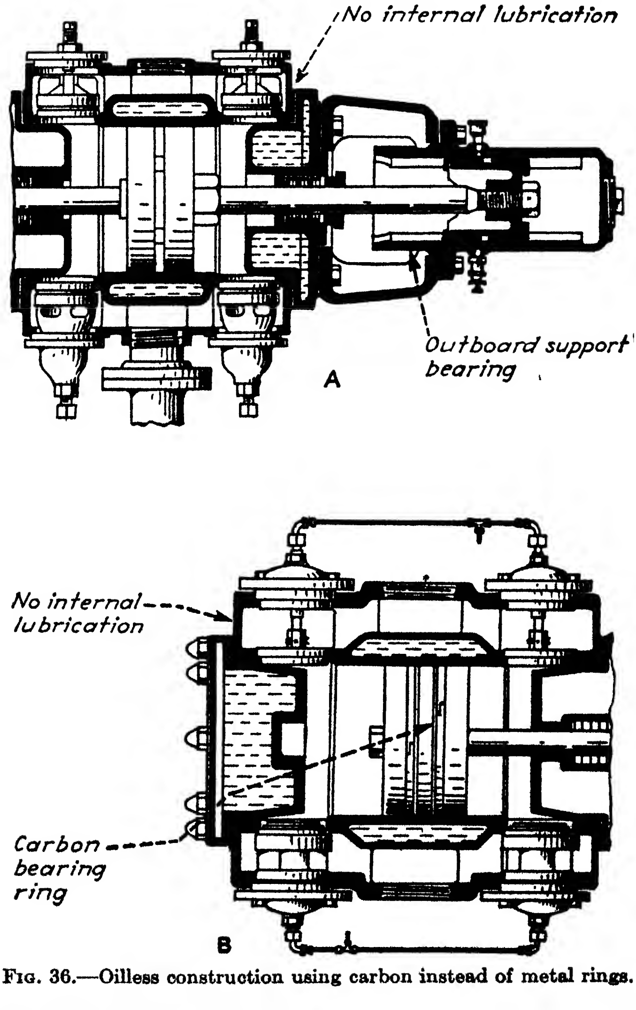


FIG. 36.—Oilless construction using carbon instead of metal rings.

but there is another quality that is not so well known. This is *circularity*. It is measured by closing the ring in a flexible band until the joint is closed to the same opening it will have when placed in the cylinder. With the ring in this position the difference between the diameter through the joint and the diameter 90 deg from the joint is the measure of circularity. If the diameter at the joint is larger than the other, the ring is said to have plus circularity, whereas if it is smaller the ring has minus circularity. It is most important that rings have plus circularity when used at high speeds, so as to avoid flutter or end vibration.

One of the most important qualities of a piston ring is its tension (Fig. 37) or ability to spring outward against the cylinder wall. Some types of service require more tension than others. However, it is essential that the original tension be maintained as long as possible. There are three general methods of producing tension: (1) milling out a relatively large section at the joint, (2) heat-treatment, and (3) hammering. Hammered tension is mechanically produced by peening the inside surface of the ring. Tension produced by milling out a section and then closing the ring stretches the outer ring fibers and compresses the inner ones.

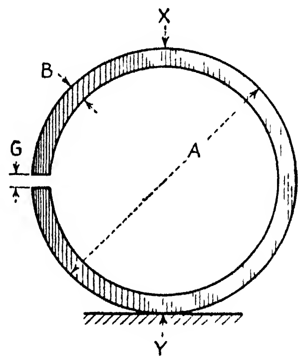


Fig. 37.—To measure ring tension, place ring on scales at Y, apply pressure at X, and close gap G to normal end clearance. Scales reading gives total tension.

It is most important that rings be given ample side clearance to prevent, as far as possible, any sticking of the rings in their grooves. Also, a generous clearance should be allowed at the joint between the ends so as to prevent any danger of the ends of the rings butting while in operation. If too little clearance is allowed on the sides, the rings will stick in the grooves; if too little clearance is allowed at the joint, they will buckle and permit leakage on the outside. This buckling may become so severe as to break the rings.

The straight-cut joint is common, but angle or step joints are also furnished when desired. Tests show that there is little difference between the leakage of the three types. Straight-cut joints are sometimes preferred on narrow rings because they are stronger and can be fitted more easily than either of the other types.

Two-piece Construction.—The two-piece compressor ring (Fig. 38) consists of a tensioned angle-joint inner ring, and an untensioned step-joint outer ring. The inner ring provides the necessary tension and effectually seals the joint of the outer ring.

When subjected to high pressures that may build up behind the ring, causing it to expand, there is a definite restraining action

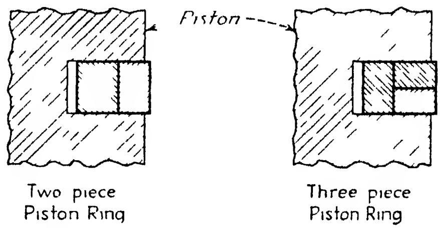


FIG. 38.—Two- and three-piece rings.

obtained from friction between the rubbing surfaces of the two rings which prevents undue cylinder-wall pressure and wear.

The over-all wall thickness of a two-piece compressor ring should always be somewhat thicker than the wall of a one-piece ring of the same diameter. Thickness depends on the diameter, but the average two-piece compressor-ring wall is at least 60

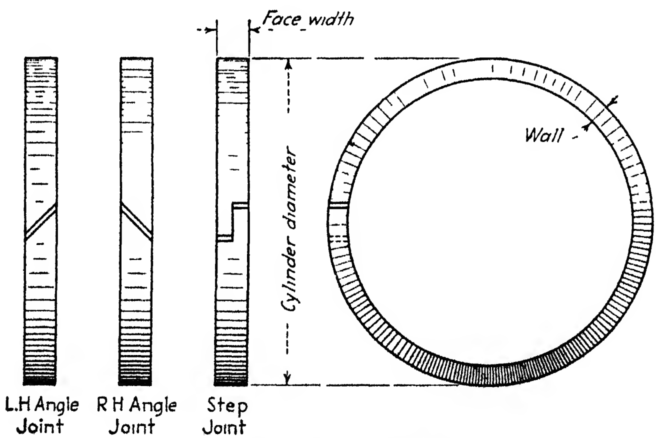


FIG. 39.—Ring nomenclature.

per cent thicker than the corresponding one-piece-ring wall. Piston grooves must therefore be correspondingly deeper.

Three-piece Ring.—The three-piece compressor ring (Fig. 38) consists of a straight-joint inner ring and two outer rings with joints of opposite angles. The inner ring provides the necessary tension and seals the joint of the outer rings. Opposite angle joints line up only momentarily in service (Fig. 39).

Usually the inner ring and both outer rings are made of cast iron, but one or both of the outer rings can be made of bronze or

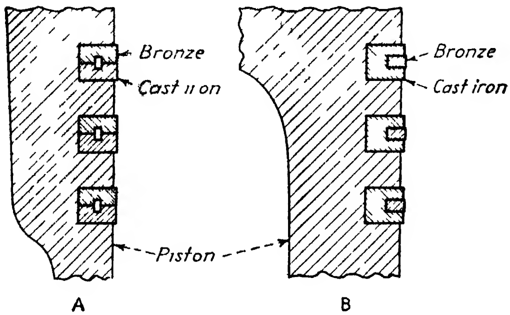


FIG. 40.—Bronze and cast-iron combination rings (*Courtesy of American Hammered Piston Ring Division, Koppers Co.*)

bakelite when desired. The combination of one cast-iron and one bronze outer ring (Fig. 40) has proved satisfactory in prolonging the life of rings and cylinders.

Segmental Rings.—The stepped-seal segmental ring (Fig. 41) consists of several segments with overlapping ends, forming stepped-seal joints, which fit together in such a way that the segments form a complete ring without any possible joint leakage. The joints are designed so that the maximum strength is secured.

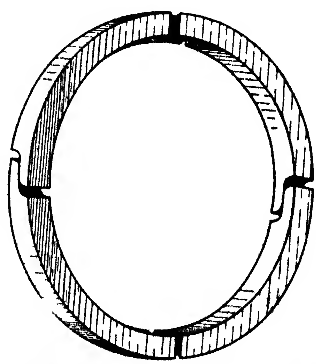


FIG. 41.—Carbon segment ring.

The segmental construction permits a much heavier wall than is possible in a one-piece ring that must be sprung over a piston, thus proportionately increasing the length of service obtained from a set of rings. Incorporating special step-seal joints improves the seal, which prevents leakage and preserves all available lubrication on the cylinder walls.

Bakelite Rings.—Bakelite piston rings are constructed in segmental form, also as one-piece compression rings, or as outer rings for two- and three-piece units. The segmental rings are regularly provided with compression inner rings made of corrosion-resistant steel. The one-piece rings do not require inner rings because they have tension hammered into them. They are used in compressors handling corrosive elements—natural gas and corrosive gas.

Ring Metals.—Bronze piston rings are manufactured from individual castings, which give a dense uniform metal. For general use, a high-quality tin bronze has been found most suitable, the mixture being adjusted to suit the diameter and cross section of the ring.

Bearing bronzes, containing suitable percentages of lead and nickel, are frequently cast into piston rings for use under conditions of poor lubrication. Also, several acid-resisting bronze mixtures have been applied to counteract the corrosive action of both organic and inorganic acids.

One quick-seating piston ring is a one-piece cast-iron concentric ring having either one or two bands of bearing bronze inserted in its periphery. These bands (Fig. 40*B*) normally project approximately 0.002 in. beyond the periphery of the ring.

The ring takes a quick initial seat because of the high-unit pressure obtained on the narrow bronze band. This prevents blow-by in either new or worn cylinders. As the bronze wears it burnishes the cylinder wall to a mirrorlike surface and allows the cast-iron body of the ring to reach a seat gradually. Throughout its life the bimetallic ring tends to eliminate scuffing and reduces wear on both rings and cylinder.

Another one-piece, concentric, quick-seating piston ring has inserted bands made from a special white-metal bearing material. Because of the relatively low melting point of the insert material, this ring is used only in installations free from high temperatures.

A combination bronze and cast-iron ring consists of two ring members, one of cast iron and one of bronze, both of which have internal tension. An interlocking arrangement (Fig. 40*A*) allows the two members a slight radial movement so that they can seat independently of each other, but causes the two to wear uniformly. This two-piece construction combines the well-established bearing qualities of cast bronze with the strength and heat-resisting qualities of cast iron. The bronze member deposits a thin coating of bronze on the cylinder wall. The iron member, having a greater affinity for oil, carries oil along with it and maintains adequate lubrication. In this way the cylinder becomes highly polished over the entire ring travel, and excessive wear is prevented.

Special Designs.—The grooved and drilled ring is of one-piece, concentric construction having a centrally located annular groove in the periphery and a series of equally spaced holes drilled radially from the annular groove to the inside of the ring (Fig. 42). When pressure builds up in the cylinder, some of it passes between the flat face of the ring and the side of the piston groove into the annular space behind the ring. From here it passes through the radial holes and fills the groove on the outside of the ring, exerting a unit pressure in the peripheral groove against the ring approximately the same as the unit pressure exerted by the air over a similar area on the inside of the ring. Thus the outward force of the air trapped in the piston groove behind the ring is to some extent counterbalanced.

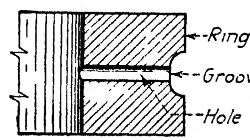


FIG 42.—Ring with center groove and equalizing hole.

A recent development in the piston-ring field is the application of plated deposits on the cylinder-contacting surface of the ring. A vast number of metals, such as silver, copper, tin, zinc, lead, cadmium, and chromium have been used with a view toward increasing useful ring life. Most encouraging results have been obtained with chromium plating. Recently a new surface finish, known as porous-chromium plate, has been developed. To produce a porous finish, the metal is electrodeposited in the usual manner. Reversing the plating current causes a stripping action to take place and removes metal from the outer face producing a porous surface. This porous condition usually extends about 0.001 to 0.0015 in. in depth, leaving the rest of the plate as it was originally. The porosity permits the ring to carry oil for immediate seating purposes and at the same time permits it to wear quickly and produce its own seating surface.

Piston-rod Composition.—High-carbon-steel forged rods connect the piston to its crosshead. Accurate machining and grinding provide a true and uniform periphery over the entire rod length which reduces leakage past the packing to a minimum. In some compressors the piston is hydraulically pressed on the rod, while in others a taper joint is used. However, regardless of how the piston fits on the rod, it is almost always held in place by a lock nut (Fig. 34).

Rods are, in most cases, screwed into the crosshead and then locked in place by a nut. Some manufacturers also slot the rod

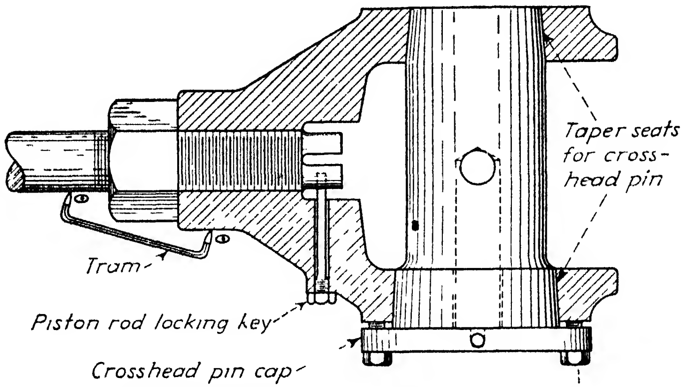


FIG. 43.—Piston-rod locking pin in crosshead.

end (Fig. 43) so that a pin inserted through the slot holds the rod from turning. A screwed connection between piston rod and crosshead simplifies adjustment of the piston clearance at both ends of its stroke.

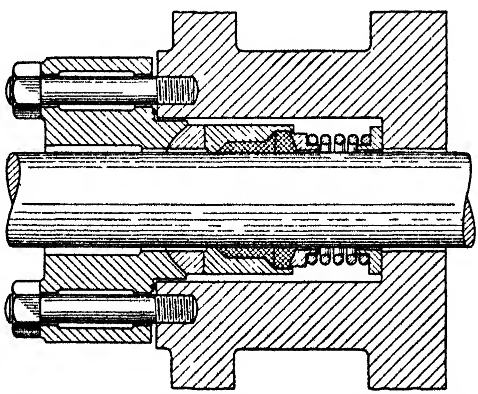


FIG. 44.—Metallic piston-rod packing with helical spring. *(Courtesy of C. Lee Cook Manufacturing Co.)*

Piston-rod Packing.—Openings in the cylinder head for passage of the piston rod must be fitted with packing to prevent air leakage. The operating pressure determines the amount needed. Some manufacturers use metallic (Figs. 44 and 45), composition, or carbon packing, while others prefer fibrous material (Fig. 46). One authority cautions against hemp packing because of its rapid scoring characteristics.

The satisfactory performance of metallic piston-rod packing depends on the mechanical fit between the packing rings and rod.

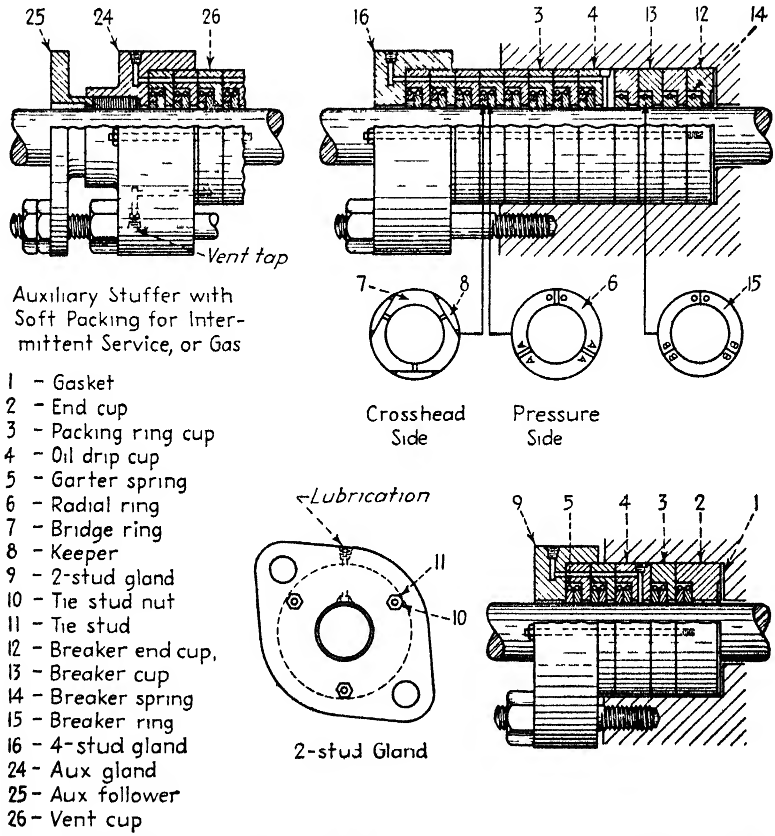


Fig. 45.—Segmental metallic packing rings held in place by garter springs. (*Courtesy of C. Lee Cook Manufacturing Co.*)

Because metal packing damages quite easily, follow the manufacturer's instructions exactly when installing, inspecting, or repairing. The packing, piston rod, and stuffing box must be absolutely clean to prevent injury to the packing rings. Identification marks placed on individual rings assure correct assembly; the markings must coincide when the packing is placed on the rod. Protect the rings when sliding them over the piston rod by covering the threaded portion.

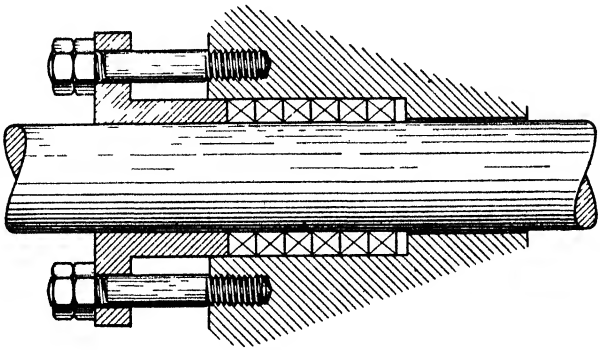


FIG. 46.—Fibrous packing is used in many machines.

To obtain best results with metallic packing, piston rods should be finished true and parallel, preferably by grinding. Scores, scars, and shoulders must be removed from old rods by stoning or grinding before replacing fibrous with metal packing. It is essential to provide adequate lubrication; the oil also helps to seal against air leakage. Wiping rings (Fig. 47) formed from carbon segments held together by garter springs, or circular metal rings, seal against oil leakage from the crankshaft and crosshead chamber.

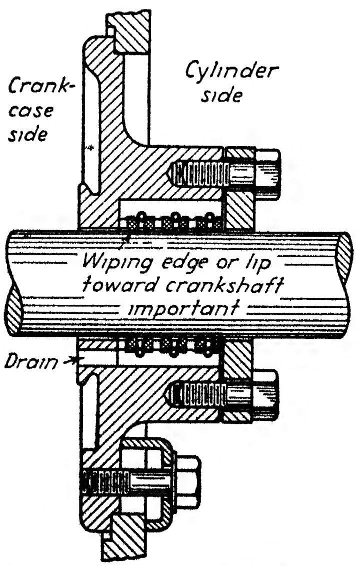


FIG. 47.—Wiping rings installed in crankcase partition.

Crosshead Construction.—The crosshead is usually box-shaped, although some compressors use a U-shaped casting. It serves as a sliding joint between the piston and connecting rod on double-acting machines (Fig. 48). Single-acting compressors do not use a crosshead, the angularity thrust of the connecting rod being absorbed by the trunk piston (Fig. 31).

Since there is no reason to seal the crank end of the cylinder on a single-acting machine, the connecting rod can be connected directly to the piston, which eliminates a crosshead entirely.

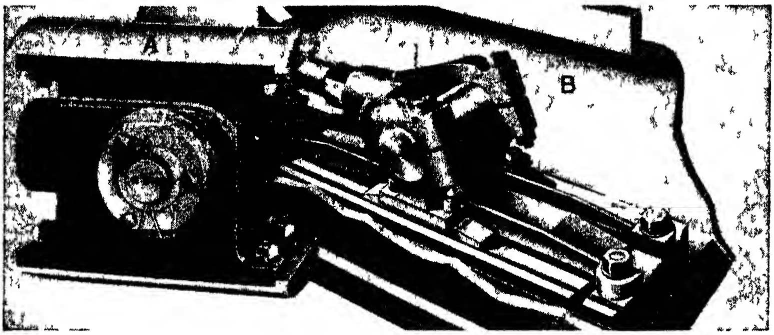


FIG 48.—Typical crosshead construction.

Crossheads are cast from open-hearth steel, charcoal iron, or malleable iron The shoes have either flat or rounded (Fig 48*A*) sliding surfaces riding in contact with heavy stationary guides to prevent transmission of bending strains to the piston rod.

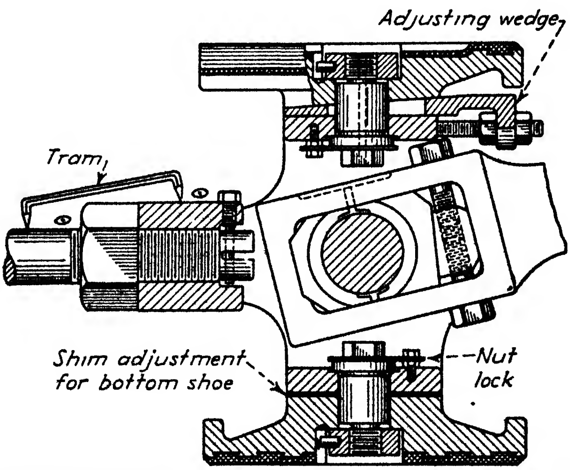


FIG. 49.—Wedge adjustment on crosshead shoe. (*Courtesy of Ingersoll-Rand Co*)

Shoe wear can be compensated for on some machines by shim or wedge adjustment between the shoe and the crosshead (Fig. 49). Other machines have no adjustment.

Crosshead Shoes.—The sliding surface of most cast-iron crosshead shoes is babbitted, although a few machines use bronze shoes. One V-angle unit uses shoes made from high-tensile Meehanite metal with tinned bearing surfaces. The metal used for crosshead guides ranges from cast iron and cast steel to chrome-nickel iron. They are honed to a smooth finish.

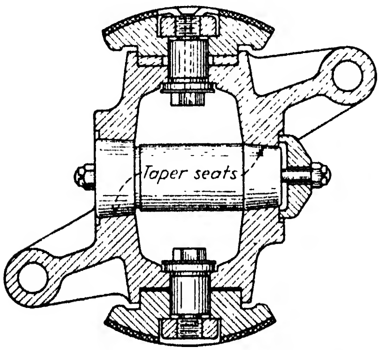


FIG. 50.—Main crosshead with stretcher-rod lugs.

Crosshead Pins.—A heavy pin holds the connecting rod and crosshead together. These pins, made from steel forgings, are casehardened and ground to size. Various methods are used to hold the pin in the crosshead; in some a taper fit between the pinhead and crosshead web with a nut on the free end permits tightening it in place. Several machines use split crosshead webs (Fig. 48*B*) fitted with constricting bolts which draw up the split

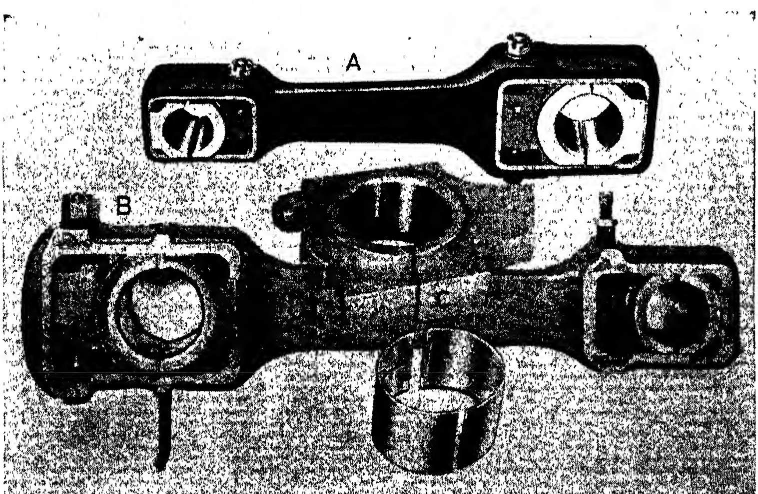


FIG. 51.—Typical connecting rods with both inner and outer adjusting wedges.

and hold the pin tightly in place. Another method (Fig. 50) utilizes the taper head with a plate clamp to hold the pin in place.

Connecting-rod Bearings.—Connecting rods for horizontal machines (Fig. 51) are usually solid-end heat-treated forgings with either I-beam or rectangular cross-sectional shape. The crosshead bearing consists of two pieces of solid bronze with a crankpin bearing being made from babbitt-faced cast steel or phosphor-bronze two-piece castings. A wedge and screw

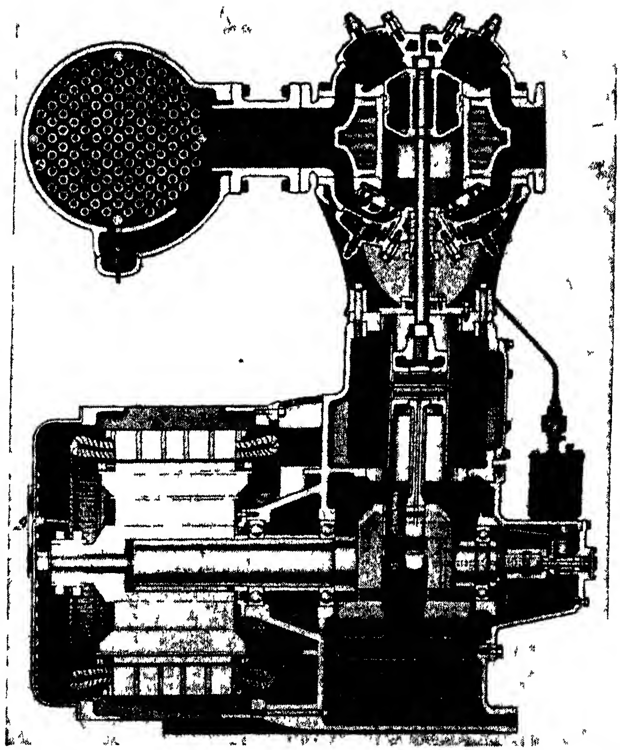


FIG. 52.—U-crank main shaft supported in ball bearings. (*Courtesy of Sullivan Machinery Co.*)

arrangement provides adjustment for bearing wear on both the crosshead and crankpin ends.

Some rods carry both wedges on the inner bearing pads (A), while others (B) have the crosshead-end wedge on the inside and the crank wedge against the outer pad. Repeated adjustment on rods having both wedges inside affects the piston clearance by making it greater at one end than at the other.

Most vertical and single-acting compressors use a marine-type connecting rod fitted with thin-shell babbitt-steel-backed bearings or bushings at both ends (*C*). Bearing adjustment is made by removing shims between the connecting rod and end cap.

The connecting rod produces reciprocating motion of the piston by a crank connection to the main shaft. The type of compressor determines whether this connection will be a U crank (Fig. 52) or a counterweighted disk and pin. Compressors having two cylinders in line operate with one connecting rod and crank by connecting one piston through stretcher rods (Fig. 53) reaching past the crank to the main crosshead. The connecting rod then connects this main crosshead to the crankpin.

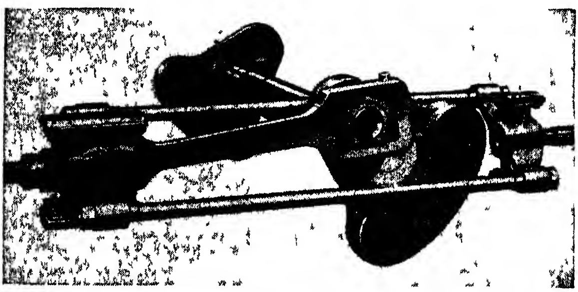


FIG. 53.—Stretcher rods connect crank to two opposed pistons.

Crankpin and Shaft.—Crankpins made from high-carbon steel forgings are hardened and then pressed into the counterweighted disk, after which the ends are riveted over (Fig. 54). Some manufacturers cast the pin and counterweight in one piece from open-hearth alloy steel. On U-crank machines the crank itself is a part of the main shaft, generally a single open-hearth steel forging. Crank counterweighted disks, when made separately from the main shaft, are pressed and keyed to the shaft; duplex machines mount the crankpins at 90 deg to each other (Fig. 53), which prevents maximum compression in all cylinders from occurring simultaneously, reduces load pulsations, and minimizes main-bearing shock.

When heavy flywheels or motor rotors are mounted between bearings on the main shaft, its middle cross section is enlarged (Fig. 54*A*) to reduce bending action. Oil-throwing and bearing seal collars are sometimes machined on the shaft at bearing journals.

Main Bearings.—Both roller and split-block bearings are used to support the main shaft. Ball and roller bearings were formerly employed only in small compressors where the loading was light, but today many heavy units use roller bearings and give excellent service. Choice between roller and block or shell bearings is largely a matter of personal preference, because when every important point is considered the fact remains that each of them gives perfect or satisfactory results.

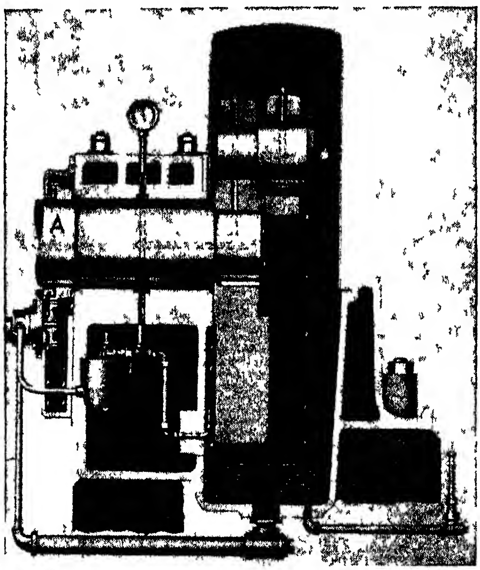


FIG. 54.—Oil passages drilled in main shaft and crankpin.

The split-block bearing consists of two or sometimes three separate babbitt-faced castings (Fig. 55). One three-piece design uses a bronze bottom section and babbitt-faced sidepieces. Bronze is not well suited for sidepieces because its coefficient of expansion is high; since these pieces are closely confined in the frame having a lower expansion coefficient, bearing heat would expand the bronze toward the shaft, which would aggravate the trouble. Babbitt-faced boxes have the same coefficient of expansion as the frame. Some babbitt-faced bottom pieces carry a bronze band insert which will support the shaft if the babbitt fails. Dismantling is a simple operation of removing the bearing cap and lifting out the sidepieces The bottom section can be rolled out of place after the shaft has been lifted slightly

In practically all machines a movable wedge *A* provides adjustment for main-bearing wear Spacing shims *B* located between bearing sections prevent movement of the adjustable shoe. Some manufacturers build smooth-faced bearing shoes; others groove the shoes for lubricating-oil passage.

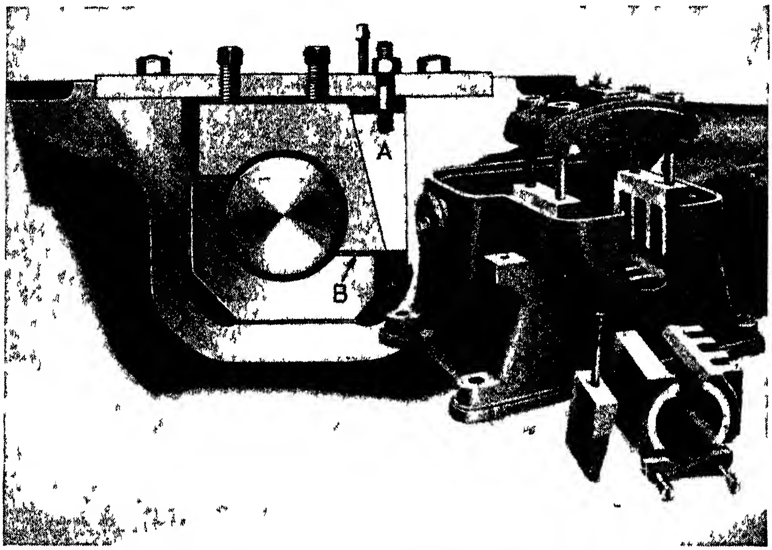


FIG 55 —Typical main-bearing construction

Frame Construction.—Main frames, sometimes of the open-top construction, heavily ribbed and reinforced, are generally completely enclosed to keep out dirt and dust. Both semisteel and cast iron are used to make these castings which support the moving parts. Cast with a solid bottom, they also form an oil reservoir for lubricating oil.

Strength of Frame.—The maximum load allowable on a compressor is determined more frequently by the strength of the frame and the running gear than by the strength of the air cylinder. As a general rule, air cylinders have quite a large safety margin and are actually used over a large pressure range.

Three points in the frame which are under heavy strain while transmitting power from the shaft to the air cylinder are (1) main bearings, (2) crankpin, and (3) crosshead pin.

The projected area of any bearing is the diameter of the bearing times its length; *i.e.*, if a crankpin is 4 in. in diameter and 5 in. long, its projected area is 4 × 5 = 20 sq in. The maximum frame load is found by multiplying the cylinder area by the maximum pressure acting on the piston. This total load divided by the projected area of the bearing gives the pounds pressure per square inch of area.

A 10- by 11-in. compressor, operating at 100 psi pressure, has a cylinder area of 78.5 sq in. The maximum pressure of 100 psi gives a maximum frame load of 7,850 lb. Now assume that the two main shaft bearings are 4 by 4 in. each, or a total projected area of 32 sq in. for the two bearings. Then 7,850 ÷ 32 = 245 lb maximum load per sq in. If the crankpin is 3.5 by 3 in., the projected area is 10.5 sq in., equivalent to a maximum pressure of 750 psi projected area on the pin. If the crosshead pin is 3 by 3 in., the maximum pressure on this pin becomes 873 psi.

Bearing Lubrication.—Oil contained in the crankcase or special reservoirs furnishes lubrication for the moving parts. Rotation of the crank disk splashes or carries the oil in scoops up and over the connecting rod and crosshead or into an overhead pan from which it flows to the connecting rod and main bearings by gravity. Some large stationary compressors use a pump (Fig. 54) to force the oil through piping or drilled channels in the shaft and crank disk to the crankpin, and through a rifle-drilled connecting rod to the crosshead pin and crosshead guides. Single-acting portable and semiportable compressors generally use the splash system (Fig. 56) for cylinder walls as well as for all bearings.

Cylinder Lubrication.—Double-acting machines require a force-feed supply of cylinder lubricating oil, generally furnished by a mechanically driven lubricator. Pipes fitted with check valves carry the oil into the top of each cylinder (Fig. 9*A*); as the piston passes, oil gathered up and carried between the piston rings serves for lubrication and also helps to seal the piston against leakage. Being mechanically driven, the lubricators feed oil to the cylinder only during operation, this feed being constant regardless of whether the machine is running loaded or unloaded.

Load Control.—Load-control arrangement varies, but all manufacturers use some form of the simple methods of either holding open the inlet valves, by closing the intake, or opening the valves to clearance pockets cast in the cylinder walls. When the compressor is to run fully loaded, a centrifugal or time-delay governor operates the unloading mechanism at a point just under full speed. Unloading control, to be described later, depends to some extent on the method and type of compressor drive.

Driving Arrangements.—Although reciprocating compressors give a pulsating load, they can be driven by any of the available prime movers. Connection between the driving unit and compressor is accomplished by various means.

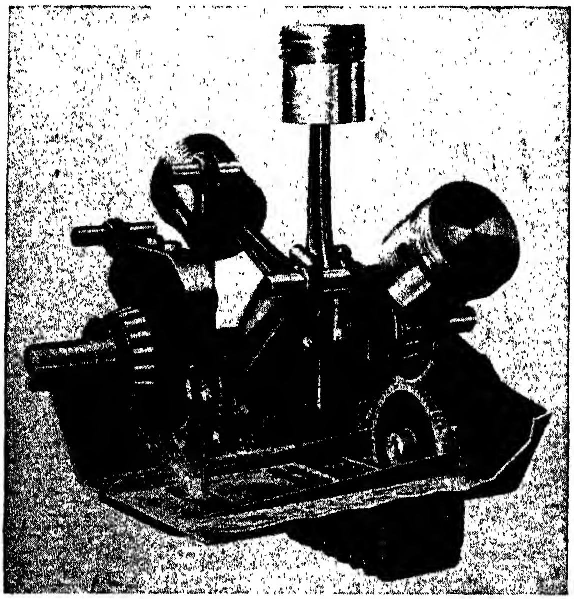


FIG. 56.—Splash oiling for single-acting compressor.

Flexible and rigid couplings are employed with electric-motor or internal-combustion-engine drive in which the speeds of the compressor and driving unit can be made the same, due consideration being given to space, weight, and economy.

Reduction-gear drive, sometimes used for electric-motor and internal-combustion-engine drive, is mandatory for turbine-driven units because of their speed.

Belt drive is used for small, low-pressure, motor-driven compressors only. The belts usually consist of a number of V-belts running on special

When driven by steam, gas, or oil engines having a similar pulsating power-impulse characteristic, drive connection must be made so that maximum power input occurs at the time of

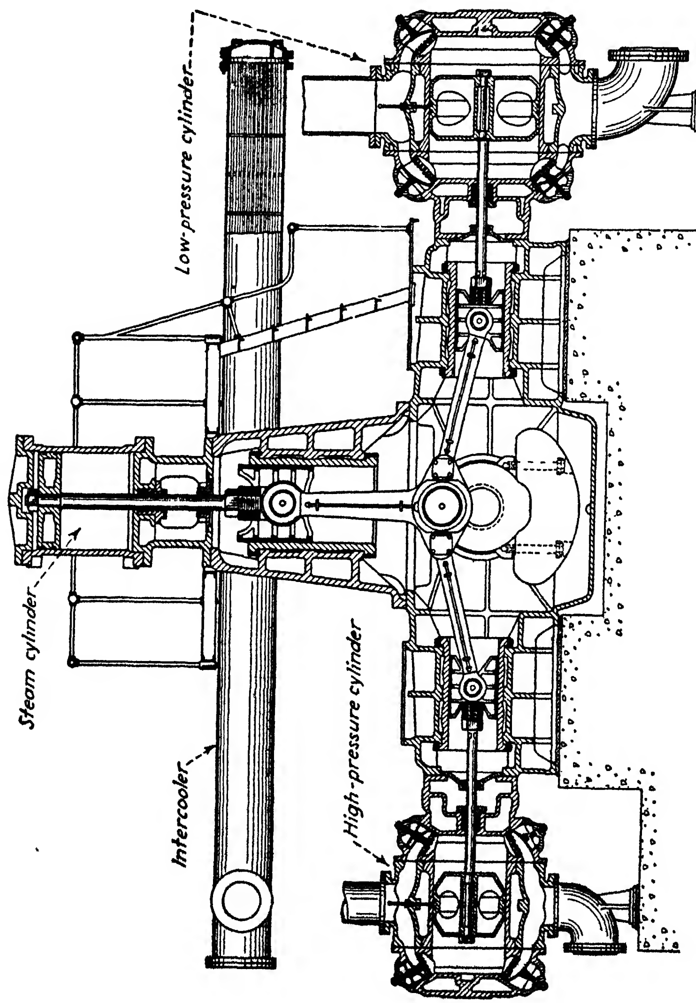


Fig. 57.—Large vertical steam-engine-drive for two-stage compressor. (Courtesy of Sullivan Machinery Co.)

greatest air-compression resistance (Fig. 57). Otherwise, an excessively large flywheel must furnish momentum to carry through the compression stroke.

Flywheels are necessary on all reciprocating compressors to keep the fluctuations of speed within specified limits, because a reciprocating compressor is a variable torque machine. In reduction-gear-driven units the customary practice is to provide

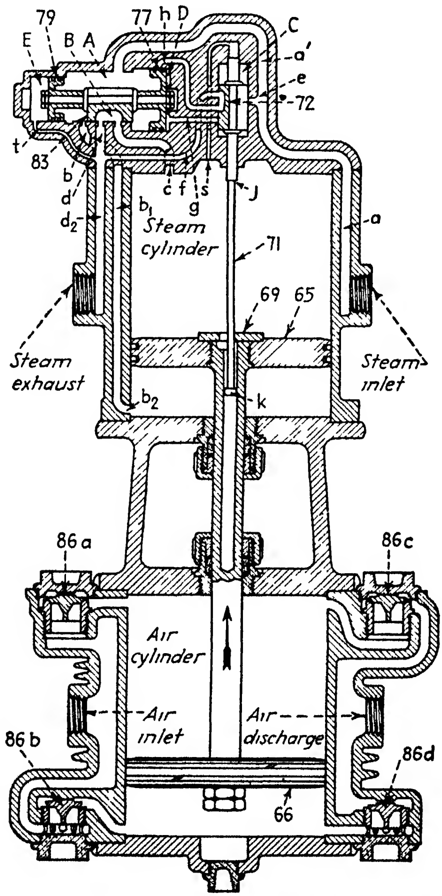


FIG. 58.—Cross section of a steam-driven compressor. (*Courtesy of Westinghouse Air Brake Co.*)

all the required flywheel effects in the gear wheel. On belted compressors the belt wheel and flywheel are frequently combined, and occasionally flexible-coupled and rigid-coupled compressors use the flywheel as part of the coupling. Flywheels, of both one piece and split construction, are clamped and keyed to the crankshaft. Those mounted on a shaft taper are further secured by a shoulder and lock nut, in addition to being keyed.

The size of the flywheel depends on the position of the driving cylinders and pistons in relation to those of the compressor. Prime movers having more than one cylinder and cranks 90 deg apart bring the greatest power impulse in time with the maximum compression resistance. Another method (Fig. 57) keeps the cranks together but mounts driven and driving cylinders at some angle with each other, this angle being 45 deg in some designs and 90 in others. Gasoline-engine-driven compressors connected by flanged couplings have marked flanges to prevent miscoupling.

Locomotive Compressor.—This unit has its steam cylinder located above the air cylinder, the two being separated by a distance piece; steam and air pistons mount on a common piston rod, which also operates the pilot valve controlling steam admission and exhaust. The compressor is double-acting, steam being admitted alternately on either side of the steam piston.

Consider the steam end of the compressor shown in Fig. 58. Supply steam flows through the passageway aa^1 to chamber A above main valve 83 and between pistons 77 and 79 and through passage e to chamber C, feeding reversing valve 72. Main valve 83 controls steam flow to and from the cylinder. It is operated by the two pistons 77 and 79 of unequal diameter and connected by the through stem. The pistons in turn are controlled by reversing valve 72, which is operated by the main steam piston 65 through reach rod 71 and the reversing plate 69.

Reversing valve 72 alternately admits live steam to or discharges it from chamber D, at the right of piston 77, thus alternately balancing or unbalancing this piston. Rod 71 is alternately moved up and down by plate 69 which engages reversing shoulder j on the upward stroke of the piston and button k, at the end of the rod, on the downward stroke.

Chambers A and C always communicate freely with each other and with the steam inlet through passages a^1 and e. Live steam is therefore always present in the chambers A and C. Chamber E, at the left of small piston 79, is always open to the exhaust passage d_2, through the port t. Exhaust steam, practically at atmospheric pressure, is therefore always present in chamber E.

A balancing port s runs from the reversing-valve cap down the outside of the valve bushing and enters the upper cylinder head to equalize the pressure above and below the reversing roc regardless of whether there is live or exhaust steam in the uppe end of the cylinder.

With reversing slide valve 72 in its lower position, chamber I is connected through port *h*, reversing-valve exhaust cavity, an port *f*, with main exhaust passage d_2. There is therefore onl atmospheric pressure at the right of piston 77.

As chamber *E*, at the left of piston 79, and chamber *D*, at th right of piston 77, are then both connected to exhaust, the stean pressure in chamber *A* has moved the larger piston 77 to th right and has pulled the smaller piston 79 and main valve 8: with it to the position shown. Main valve 83 then admits stean below main piston 65 through ports b, b_1, and b_2. Piston 65 i thereby forced upward, and the steam above it passes througl port *c*, exhaust cavity *B* in main valve 83, port *d*, and passag d_2, where it discharges into the atmosphere.

When piston 65 reaches the upper end of its stroke, reversing plate 69 strikes shoulder *j* on rod 71, forcing it and reversing slide valve 72 upward to open port *g*. Steam from chamber *C* then enters chamber *D* through port *g*. The pressures upon the two sides of piston 77 are thus equalized, and since chamber *E* is always open to the exhaust, the pressure in chamber *A* forces piston 79 to the left, drawing with it piston 77 and main valve 83.

With main valve 83 in this position, steam then flows from chamber *A* through port *c* above piston 65, forcing it down; at the same time steam below this piston exhausts to atmosphere through ports b_2, b_1, and b. When piston 65 reaches the lower end of its stroke, reversing plate 69 engages button *k* and draws rod 71 and reversing valve 72 down to their starting position.

In starting, always run this compressor slowly until it becomes warm, permitting the condensate to escape through the drain cocks and exhaust, until sufficient pressure builds up in the main reservoir (25 to 30 psi) to provide an air cushion. Then close the drain cocks and open the steam valve sufficiently to run the compressor at normal speed. Never run the compressor faster than is necessary to do the work required. A pressure governor automatically controls starting and stopping.

Lubrication.—The air cylinders are lubricated by an automatic oil cup. Its construction and operation are illustrated in the sectional view (Fig. 59). The oil chamber *a* is filled from the top when the cap nut 5 is removed. The cap nut carries a vent hole *f*, so located that when the seal between the cap and body is broken the air pressure vents to atmosphere, thereby permitting filling while the compressor is running. The stem portion 3 of the body 2 has a central passage *b*, communicating at the bottom with the pipe connection leading to the air cylinder, and at the top with chamber *a* through the cavity in the cap This passage has its top outlet on the side so as to permit filling chamber *a* without the possibility of pouring oil directly into the passage to the compressor.

An oil port *d* in the stem connects passage *b* to an annular feeding cavity *e*, which is formed by a recess in the stem and the neat-fitting sleeve around it. This sleeve has two diametrically placed notches *c* at its lower end, which connect chamber *a* with cavity *e*. When the compressor makes its upward stroke, air is forced up through passage *b* and into the space above the oil in chamber *a*.

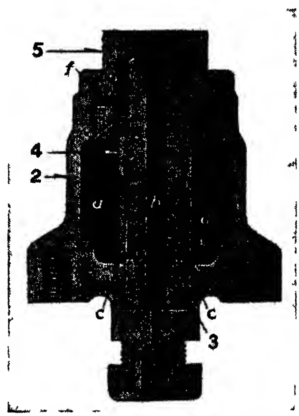


FIG 59 —Lubricator for Westinghouse compressor.

The lubricant in the cup flows through the notches *c* into the space between stem 3 and sleeve 4, rises in space *e* by capillary attraction, and then enters opening *d* to passage *b*, from which, on the downward stroke of the compressor, it is carried with the flow of air from the chamber above the oil through passage *b* into the compressor cylinder. This small amount of oil supplied regularly and reliably is ample to lubricate the air cylinder adequately. Use only a good grade of standard locomotive saturated-steam valve oil in the air cylinder.

Operating these compressors continually at high speeds or against excessive pressure inevitably results in high temperatures which destroy lubrication, causing the air cylinders to cut, besides filling the discharge passages with deposits from burnt oil and in general reducing the over-all compressor efficiency.

Under normal conditions, the speed should not exceed 140 strokes per min, and such a speed should not be maintained continuously for any considerable time. Even this speed will eventually cause excessive heating.

TABLE 6.—RING END CLEARANCE FOR COMPRESSORS

Cylinder dia, in.	Min side clearance, in.	Joint end clearance, in., min	
		Straight and step	45-deg angle
Below 6.	0.0015	0.003 × diam	0.002 × diam
6 and over.	0.0025	0.003 × diam	0 002 × diam

Courtesy of American Hammered Piston Ring Division, Koppers Co.

Operating Cycle.—Reciprocating compressors compress the air by moving a piston back and forth inside a closed cylinder. Figure 60 shows in cross section a double-acting compressor with piston, piston rod, inlet valves *A*, and discharge valves *B*.

For maximum capacity and efficiency the piston should approach the cylinder ends closely and leave minimum clearance volume (including volume of valve passages).

See what happens on the right side of the piston. Starting from the right end of the stroke, the piston moves toward the left. As it moves, the discharge valve closes and air remaining in the clearance space under discharge pressure expands, aiding movement of the piston until the internal pressure is below atmospheric. Free air then flows into the cylinder until the piston reaches the end of the suction stroke, leaving the cylinder filled with air at slightly below atmospheric pressure.

The reverse stroke of the piston starts compressing the entrapped air, forcing the inlet valve closed. The piston movement to the right continues to reduce the air volume until the resulting pressure forces the discharge valve open against receiver pressure. The air then flows into the system during the remainder of the piston stroke. Air remaining in the clearance space then reexpands as the piston travel reverses, and the process repeats itself.

An ideal compressor would take in a cylinder full of air at maximum atmospheric pressure without the incoming air absorbing heat from the metal as it enters. The piston would then compress the air to its final volume and pressure without any increase in temperature and force the entire volume of air from the cylinder. This ideal condition does not prevail in practice. Internal pressure must be slightly below atmospheric or the air will not enter. Traveling through the hot inlet passages raises the air temperature, increases the internal pressure, and reduces the total weight of air flowing in. As compression takes place the air temperature rises, building up the pressure prematurely and extending the length of time the piston must act against this

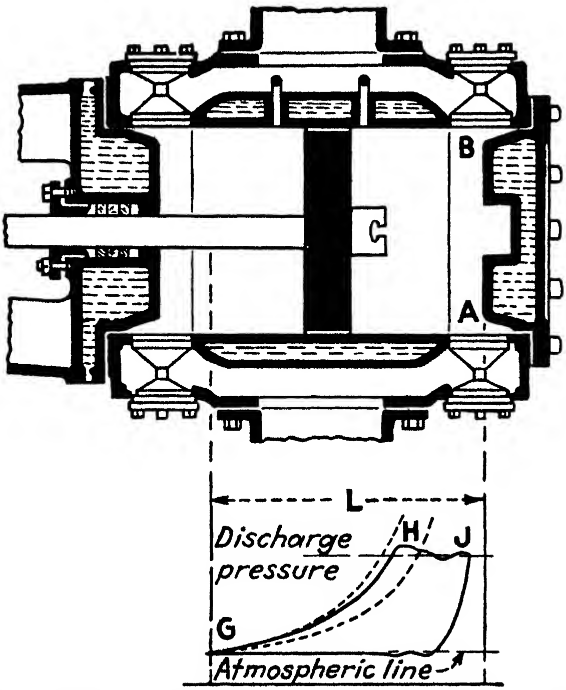


FIG. 60.—Indicator card showing conditions inside compressor cylinder.

pressure in forcing the air from the cylinder. Another output loss occurs because of clearance space. The piston cannot force all the air from the cylinder; this air occupies space that cannot be filled with new air during the inlet stroke.

How the Pressure and Volume Change.—The changes in pressure and volume occurring in one end of the cylinder can be followed on the indicator card (Fig. 60). The distance L represents the volume displaced by the piston during one stroke. Starting with the piston at G and the cylinder filled with air at atmospheric pressure, the pressure of the confined air gradually rises as the piston reduces the volume, and the heat of compression acts until a pressure *H*, slightly above receiver pressure, builds up. Completing the stroke, the piston forces the compressed air from the cylinder. The line *HJ* represents this expulsion.

Upon completion of the stroke, the clearance space remains filled with air at pressure *J*. The higher the compression ratio or the larger the space, the more air it will hold and the farther the piston must move to the left before this confined air can expand to below intake pressure. A compressor cannot be operated without some clearance, but the smaller it is the better.

Effects of Volumetric Efficiency.—The clearance-space or volumetric loss, although reducing the capacity of the compressor, represents little loss of mechanical power because most of the energy used in compressing air trapped in this space returns during the expansion period in helping to move the piston. The longer the stroke the less percentage of clearance is required for any given volume. If a machine has a 30- by 60-in. air cylinder with a 1½ per cent clearance, and the stroke is shortened to 30 in., or one-half, with the length of clearance remaining the same, the percentage of clearance would be doubled, or 3 per cent.

Volumetric loss in single-stage compressors, resulting from clearance, means that the single-stage compressors furnish less actual free air than do two-stage machines for the same discharge pressure and piston displacement. All comparisons must be based on actual capacity and not on piston displacement; furthermore, the horsepower required must be compared on the basis of the brake horsepower (bhp) per 100 cu ft actual capacity.

Comparing Output.—To illustrate, let us compare two standard compressors. A 14 by 13 single-stage belt-driven unit has a piston displacement of 628 cfm and requires about 95 bhp to compress air to 100 psi. A 13½ by 8 by 8 duplex two-stage machine has a displacement of 593 cfm and requires about 95 bhp for 100 psi.

Now consider the single-stage compressor. If we divide 95 hp by 628, we find that the machine requires 0.151 hp per cfm piston displacement. Similarly, we find that the two-stage machine requires 0.16 hp per cu ft piston displacement. This indicates that the single-stage machine takes less horsepower per cubic foot displacement. Where, then, is the economy of two-stage compression?

TABLE 7.—THEORETICAL HORSEPOWER FOR 100 CFM OF FREE AIR TO VARIOUS PRESSURES

Discharge pressure, psi	Ratio of compression	Theoretical hp				
		Single- or multistage isothermal	Single-stage adiabatic	Two-stage adiabatic	Three-stage adiabatic	Four-stage adiabatic
50	4.4	9.5	11.9			
100	7.8	13.2	18.0	15.4		
150	11.2	15.5	22.5	18.6		
200	14.6	17.2	26.3	21.0		
300	21.4	19.7		24.7		
500	35.0	22.9		29.8	27.3	
1,000	69.0	27.2		37.3	33.6	
1,250	86.0	28.6			35.7	33.6
3,000	205	34.1			44.5	41.6
3,500	239	35.1			46.4	42.8

We are told that because of cooling in the intercooler, two-stage compression requires about 12 per cent less horsepower when compressing to 100 psi than single-stage does. Suppose we investigate the volumetric efficiency and consider the amount of air actually delivered by each machine. We know that volumetric efficiency, which is a function of compression ratio, is much higher in a two-stage than in a single-stage machine. The volumetric efficiency of the 14 by 13 is actually about 71 per cent, so that this compressor delivers only 445 cu ft. By dividing 95 hp by 445 we find that the single-stage machine requires 0.213 hp per cu ft actual capacity. The volumetric efficiency of the two-stage compressor is about 85 per cent, so that it delivers 505 cu ft, equivalent to 0.19 hp per cu ft actual capacity; *i.e.*, the two-stage machine requires 0.023 hp less per cu ft discharge, or a saving of about 11 per cent.

In any two-stage compressor, the number of compressions in the low-pressure cylinder is theoretically equal to the ratio of cylinders. To illustrate, consider a two-stage machine with a 17-in. low-pressure cylinder and a 12-in. high-pressure cylinder, in which the cylinder-area ratio is 2:1. As the high-pressure cylinder has only half the volume of the low-pressure cylinder, it follows that, as the same amount of air must pass through this cylinder, the absolute pressure of the air must be doubled if the high-pressure volume is reduced one half. If the ratio of cylinders is 3:1, the absolute pressure must be trebled if the volume is reduced to one third. In the former case we double the atmospheric pressure of 14.7 psi or 29.4 psi abs, which is equivalent to 14.7 psi gage. In the latter case, we treble the atmospheric pressure of 14.7 psi to 44.1 psi abs, or 29.4 psi gage.

These pressures are theoretical and could result only if the intercooler reduced the temperature of the air to that of the low-pressure intake. Actually, however, the high-pressure suction temperature depends upon the cooling-water temperature and may, therefore, be higher than that of the low-pressure suction. To determine the intercooler pressure accurately the theoretical value must be corrected for the difference in temperature between the low- and high-pressure intake; it is computed thus:

$$\text{Intercooler pressure} = \frac{R \times P \times HPT}{LPT}$$

where R = ratio of cylinders.
P = absolute intake pressure.
HPT = h-p intake temperature abs.
LPT = l-p intake temperature abs.

In calculating the cylinder ratio, use care to ascertain whether the low- and high-pressure cylinders have approximately the same cylinder clearance, and hence about the same volumetric efficiency. If one cylinder has an abnormal amount of clearance, this will accordingly reduce its effective size. Therefore, when the volumetric efficiency of the two cylinders is not approximately equal, the cylinder ratio should be obtained thus:

$$\text{Cylinder ratio} = \frac{LPC \times LPV}{HPC \times HPV}$$

where LPC = l-p cylinder area.
HPC = h-p cylinder area.
LPV = l-p volumetric efficiency.
HPV = h-p volumetric efficiency.

Now, to find the volumetric efficiency of this compressor, we have only to find the volumetric of the low-pressure 15½ by 10-in. single-stage cylinder, discharging at 26.6 psi. The discharge pressure of 26.6 psi is equivalent to 41.3 psi abs which, divided by the absolute pressure of 14.7 psi, gives 2.66 compressions.

Reexpansion of clearance air follows closer to the adiabatic line than to the isothermal. Ten per cent clearance, expanded adiabatically through 2.66 compressions, would occupy twice its original volume. Other volumetric losses, such as heating the air as it enters the intake, must be taken into consideration.

We know that air expands when it is heated. Naturally, a cubic foot of heated air does not weigh as much as one of cool air, the weight varying directly as the absolute temperature. Therefore, we cannot confine as much air at high temperature in 1 cu ft volume as we can at low temperature. If the air comes in contact with hot surfaces on entering the cylinder, it will take up some of the heat and expand in volume. Thus, if 70 F air becomes heated 20 deg while entering the cylinder, the weight of air within the cylinder will have been reduced in the ratio of 530:550, or approximately 4 per cent when the cylinder volume is filled. Roughly, each 5 F increase in the air's temperature results in a 1 per cent loss in weight. Since the actual capacity of a compressor is measured on the basis of the intake-air temperature, any increase in temperature that occurs as the air passes into the cylinder becomes a volumetric loss and contributes to an increase in horsepower required per unit of air delivered.

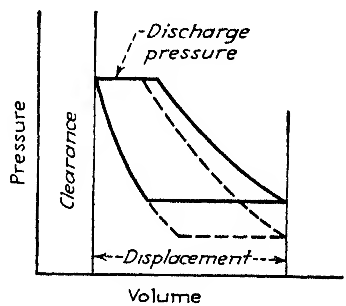


FIG. 61.—Effects of low intake pressure.

Further heat losses affecting compressor power input may be sustained in the intercooler. The hot air is discharged from the low-pressure cylinder into the intercooler, where the temperature of the air is reduced before entering the second stage. For best performance, cooling-water temperature should be at least 20 F below the low-pressure intake-air temperature.

Cooling-water temperature above that of the low-pressure intake will increase compressor power requirements at a fairly rapid rate. When air leaves the intercooler at a high temperature, work done within the high-pressure cylinder is appreciably increased. Air confined within a given volume, as in an intercooler, cannot expand when heated and consequently the pressure increases as the temperature rises.

Effects of Leaks.—When leakage occurs past valves or piston it results in a definite volumetric loss. Not only the compressed air on which energy has been expended to compress it is lost, but the hot air leaks from one end of the cylinder to the other which, at that time, contains cool air and is just taking suction. The incoming air is thereby heated, and compression starts at a higher temperature. The final air temperature, therefore, is higher. During the next revolution the leakage again occurs, but the leakage air's temperature is slightly higher this time because the discharge temperature of the previous stroke is higher. This higher temperature results in a still higher discharge temperature on the next stroke, and thus the effect of the leakage is cumulative.

✓**Effect of Altitude.**—When the rarefied air at an altitude is compressed, slightly less power is required than when compressing the same volume of free air at sea level. This is because the air is less dense, or a smaller weight of air is being compressed but at a lower intake pressure and therefore through a greater number of compressions.

If it were not for the fact that as the altitude increases, the ratio of compression also increases for the same discharge pressure, the horsepower required would fall as the altitude increased. Actually, the horsepower at altitudes is slightly greater because of the increase in the compression ratio.

Since the volumetric efficiency is largely determined by the cylinder clearance and compression ratio, let us see what the effect of this is. When operating at an altitude the piston will travel a greater percentage of its stroke before the inlet valve can open, as the reexpansion must extend to a lower pressure (dotted line Fig. 61). It follows, then, that the volumetric efficiency of a single-stage compressor is lower at an altitude than at sea level.

As the volumetric efficiency of a two-stage compressor is determined by the compression ratio in the low-pressure cylinder, this latter is affected only slightly when operating at an altitude, therefore its volumetric efficiency is also affected very little—less than 1 per cent.

Since the total ratio of compression is increased at an altitude, and the ratio in the low-pressure cylinder remains practically the same, it follows that virtually all the increased compression must occur in the high-pressure cylinder, thus reducing its volumetric.

With the high-pressure volumetric slightly reduced, when operating at an altitude as compared with sea level, the effective cylinder ratio increases, thus slightly increasing the intercooler pressure and also the compression in the low-pressure cylinder.

While the cubic feet of actual free-air capacity may be approximately the same as at sea level, the real amount of air discharged, measured either by compressed volume or by weight of air, is appreciably smaller, being in fact smaller by ratio of the absolute inlet pressures. It follows then that the horsepower per given weight of air discharged is greater when operating at an altitude than at sea level.

Free Air Handled.—The volume of free air handled by a compressor is a function of the piston displacement, the volumetric

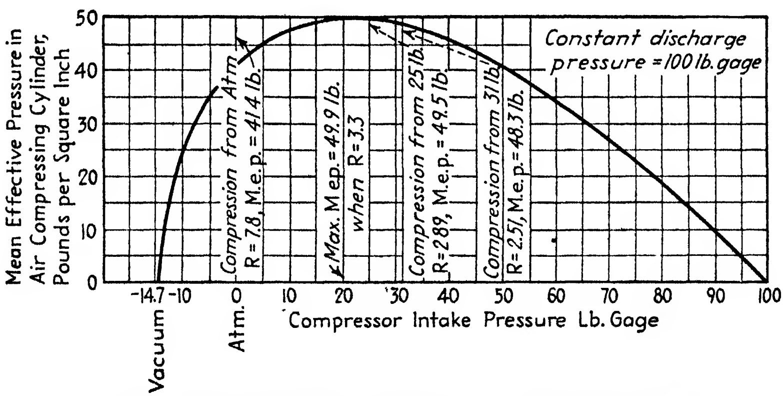


Fig. 62.—Intake-discharge pressure curve. (*Courtesy of Power.*)

efficiency, and the rpm. The motive power provided is based on the above factors with little or no spare power included in the motor. Attempting to secure more capacity by increasing the speed increases proportionately the power needed, which would result in overloading the motor. Increasing the rpm means a greater piston speed, with its lubrication problems and greater stresses imposed on the flywheel, reciprocating parts, and valves.

More air capacity can also be obtained by raising the intake pressure for a given piston displacement. However, raising the intake pressure increases the power required by the compressor up to a compression ratio ranging from 2.0 to 3.3, the actual peak depending on machine characteristics. As the ratio decreases the mean effective pressure (mep) inside the cylinder drops and power falls off. The curve (Fig. 62) gives a 3.3 compression-ratio relation between intake pressure and cylinder mep. Zero mep occurs at two points, when intake is at absolute vacuum and when the intake pressure equals that of the discharge. A decrease in compression ratio improves the volumetric efficiency and increases the output.

Assume a machine designed to compress air from atmospheric pressure to 100 psi. This gives a compression ratio of 7.8 and a mep of 41.4 psi. Suppose we supply air from a 20 psi source to the compressor intake. The total quantity of air forced into the cylinder at this pressure is $20 + 14.7 \div 14.7 = 2.4$ times the quantity held at atmospheric pressure. This larger quantity of air will reach discharge pressure quicker and the piston will have to act against it longer in expelling it from the cylinder. Although the compression ratio falls to 3.3, the mep increases to 49.9 psi, which calls for more power.

Increase the intake pressure to 31 psi, and the compression ratio falls to 2.51 with a mep of 48.3 psi. Raising the intake pressure from atmospheric to 20 psi would overload the unit 20 per cent, whereas raising it to 31 psi causes only a 16.7 per cent overload.

Booster Compressors.—The term booster compressor applies to a machine having its intake at some pressure above atmospheric and discharging at a higher pressure. Both suction and discharge pressures might be under 100 psi or they might both be several thousand psi pressure.

Assume the compressor develops its maximum horsepower at compression ratios between 2.00 and 2.50. The curves (Fig. 63) will be quite flat between these two points, so that the exact point where the peak occurs will not matter materially.

As the intake pressure rises to decrease the compression below this figure, the horsepower required begins to recede; likewise, as the intake pressure is reduced, increasing the compression beyond the peak, the maximum horsepower required to drive will again decrease. Compressors that operate through this range of suction pressure should be motored for the peak power requirements occurring between 2.00 and 2.50 compression.

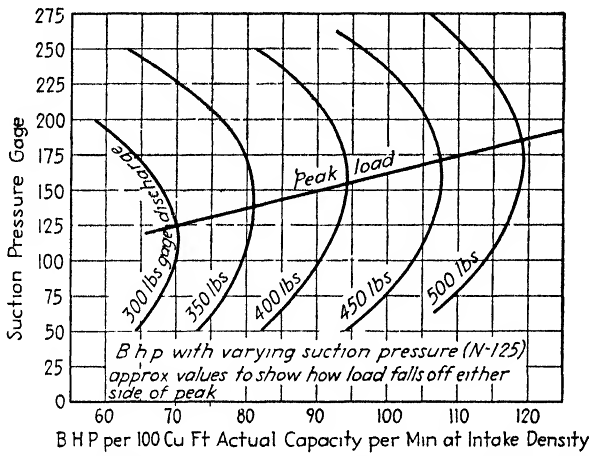


FIG. 63.—Characteristic peak-load curves. (*Courtesy of Worthington Pump and Machinery Corp.*)

Comparison of Actual and Theoretical Horsepower.—Water jacketing brings the compression curve somewhat below the adiabatic, but the excess pressure required to force the air through valves makes the work done, in compressing and discharging, close to that calculated on the assumption of adiabatic compression for the cylinder of air.

Several factors that cause the actual to be greater than the theoretical horsepower are

1. Heating of the air during compression, resulting in a steep compression line.
2. Heating of the instake air as it contacts heated parts of the compressor.
3. Impracticability of removing all the heat in the intercooler.
4. Leakage past the valves and pistons.
5. Friction of the air flowing through valves and ports.
6. Valve slip at the end of a stroke.

The normal compression line always falls somewhere between the true isothermal and adiabatic, and its position is governed by several factors (Fig. 64). A slow-speed compressor with a cylinder of small diameter will have a compression line more nearly approaching isothermal than one with a large-diameter cylinder or one rotating at high speed. The faster the air is compressed, the greater the heat developed. The larger the cylinder, the poorer the heat transfer between air and cooling water. This does not mean that large-cylinder or high-speed compressors are inefficient or wasteful, but it does show that these

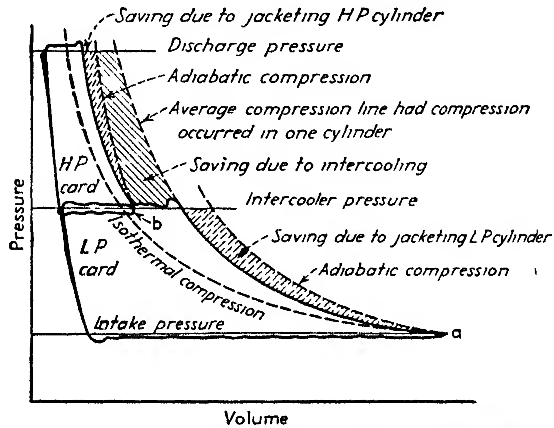


FIG. 64.—Combined card for two-stage compressor. (*Courtesy of Worthington Pump and Machinery Corp.*)

points must be considered when judging compression lines in comparison to true isothermal or adiabatic. If a cylinder indicator diagram is made when the machine is new and the theoretical lines plotted in, all later diagrams should be similar if the machine remains in good condition.

Power Pulsations.—Further study of the action inside a compressor cylinder shows how adding a flywheel on a single-cylinder compressor or setting the cranks at 90 deg on a duplex machine smooths out power pulsations. Although a steam-driven compressor is used as an example, the same conditions would apply to any internal-combustion-engine drive and to a lesser degree with motor drive.

Consider Fig. 65*A*, a single-cylinder steam-driven unit. During the first part of the stroke the energy in the steam cylinder (hatched area) is at its greatest, while in the air cylinder compression is just beginning, and consequently the resistance is least. As the pistons advance, the steam and air pressures

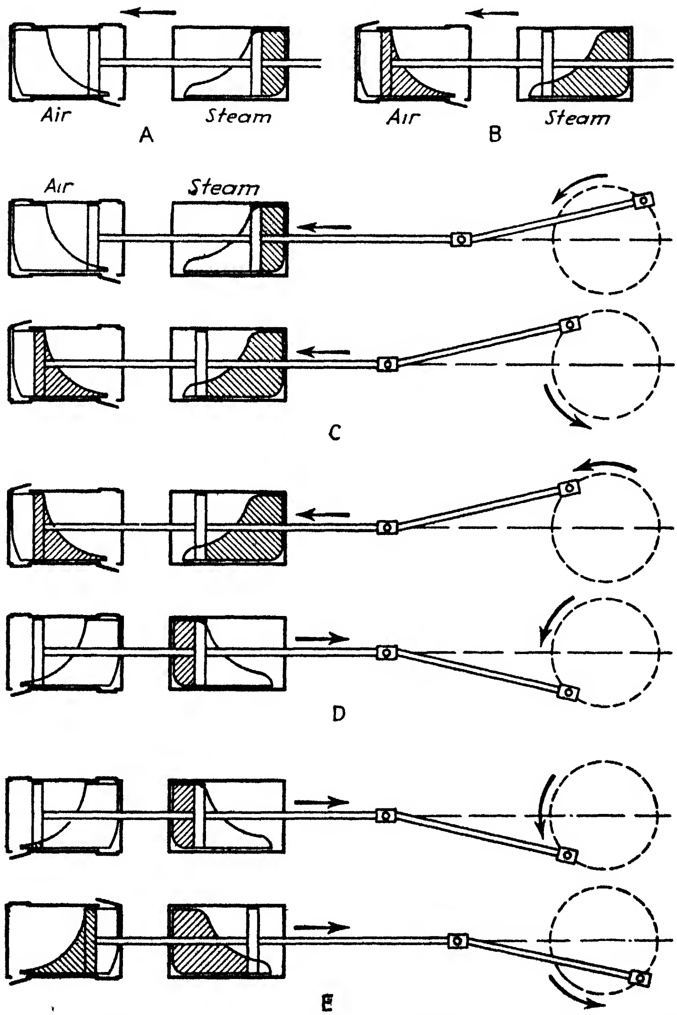


FIG. 65.—Duplex-compressor operation with cranks at 90 deg. (*Courtesy of Power.*)

approach a balance until at some point in the stroke steam power and air resistance are exactly equal.

After this neutral point is passed, the ratio of power to resistance decreases until finally, at the end of the stroke *B*, steam power is minimum and air resistance is maximum. Excess power in the steam cylinder at the beginning of the stroke is stored up in the flywheel, increasing its momentum, and returns to the shaft during the latter part of the stroke, helping the machine over each dead center.

The duplex compressor is, in effect, two straight-line compressors placed side by side and connected by a common crankshaft on which is mounted the flywheel. Putting the cylinders *A* and *B* together on a common crankshaft with the cranks 90 deg apart as in *C* shows the steam pressures and opposed air

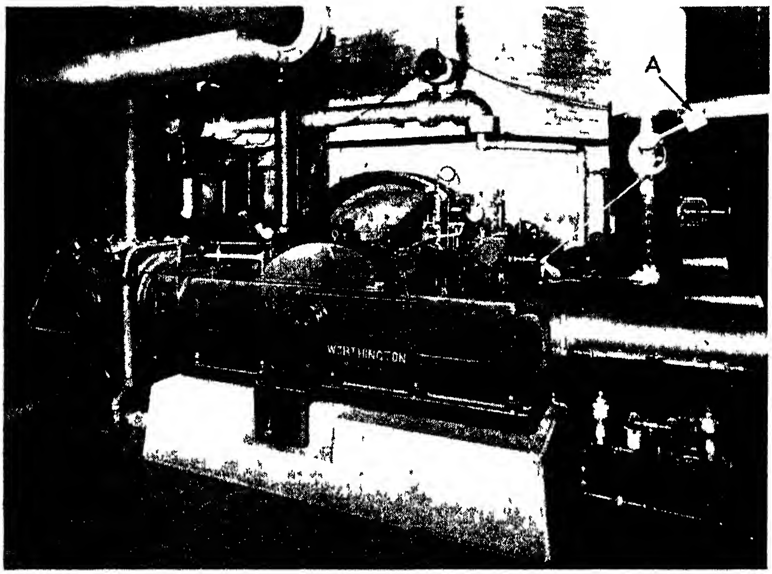


Fig. 66.—Weighted butterfly valve in steam line is tripped by overspeed device (*Courtesy of Worthington Pump and Machinery Corp.*)

pressures at the beginning of a revolution. Power is in excess of resistance in the upper set of cylinders, while in the lower cylinders the reverse exists. The surplus power in the upper steam cylinders is applied through the crankshaft to carry the lower half over center.

In *D* the instantaneous steam and air pressures appear a quarter of a revolution later. Here, power is in excess of resistance in the lower cylinders, and minimum power is opposed to maximum resistance in the upper cylinders. Assistance is now received by the upper half from the lower half by the transmission of the surplus power through the crankshaft as before.

A quarter of a revolution later *E*, the compressor, has begun its return stroke, and the conditions are exactly the same as those shown in *C*, but reverse in direction.

This sequence of operation shows the advantage of using a flywheel on a single-cylinder compressor, connecting duplex-machine cranks at 90 deg, or mounting steam and internal-combustion-engine cylinders at 90 deg with the air cylinders.

Mechanical Efficiency.—The mechanical efficiency of any compressor is at its best when the machine is fully loaded. Assume that a 12 by 11 single-stage belted machine, operating at 100 psi, requires 62 bhp. Assuming a mechanical efficiency of 88 per cent, the air-indicated horsepower would be 54.5, leaving 7.5 hp as mechanical friction. If the same compressor operated at only 40 psi, the air-indicated horsepower would be about 37.5. The mechanical friction changes little, regardless of the load. It remains 7.5 hp, which, added to the air horsepower, results in a total of 45 bhp. It will thus be seen that the mechanical efficiency is 83.5 per cent when operating at the reduced load of 40 psi. Thus the mechanical efficiency falls off with a reduced load because of mechanical friction.

Large steam-driven compressors (Fig. 66), when in first-class condition, show a mechanical efficiency, when fully loaded, of approximately 90 per cent. Smaller steam-driven machines may show slightly lower results, 87 to 90 per cent. Naturally, belt-driven or direct-connected compressors show higher mechanical efficiency than steam-driven machines because there are not so many running parts to give friction. Such compressors, of medium and large size, should show a mechanical efficiency of 92 to 93 per cent.

In belt-driven compressors the mechanical efficiency does not allow for belt friction or slip. As bhp is measured at the compressor pulley or shaft, the belt loss is not a part of the compressor friction but is beyond the compressor. It may be stated here that in selecting a motor to drive a belted compressor it is accepted practice to add about 5 per cent to bhp for belt loss.

The mechanical efficiency of a direct-connected compressor, with the motor mounted on the compressor shaft, is identical with that of a belt-driven machine of corresponding size. How-

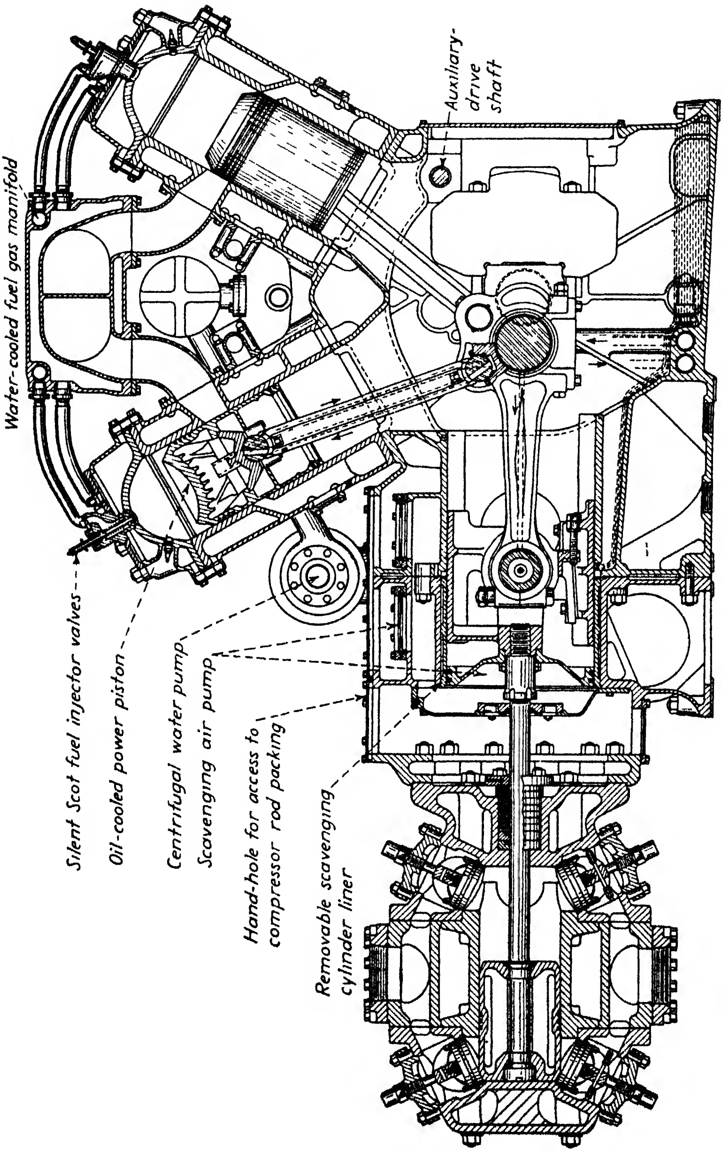


FIG. 67.—Gas-engine drive with oil-cooled power pistons. Compressor uses metallic piston-rod packing. (*Courtesy of Cooper-Bessemer Corp.*)

ever, the over-all mechanical efficiency of the compressor and motor of a direct-connected set is somewhat better than that of a belted set, because in the direct-connected machine, belt loss and main-bearing friction are somewhat reduced. The belted unit has two sets of bearings, one on the compressor and one on the motor. The direct-connected set has only one set of bearings, as the compressor bearing also serves for the motor. Figuring the belt loss at 5 per cent and the reduced friction of extra bearings as 1 per cent, it can be shown that the over-all mechanical efficieny of a direct-connected compressor is about 6 per cent higher than a belt-driven one. In addition, the synchronous motors, usually used on direct-connected machines, are more efficient than induction motors used on belted units.

The over-all economy of an air-compressor unit depends on (1) mechanical construction—the size and proportion of bearings and wearing surfaces, the lubrication system, and general design of parts; (2) the length and volume of ports in the air cylinder—long and restricted inlet-air passages increase losses by throttling and heating the incoming air before compression begins; (3) the cooling surfaces in water jackets and intercoolers, and the temperature of the cooling water; (4) surrounding conditions—the altitude at which the machine is operated and the atmospheric temperature; (5) clearance spaces and slippage; (6) economy of the power end.

CHAPTER II

SELECTING AIR-COMPRESSOR DRIVE

First Considerations.—Before purchasing an air compressor, determine from an economic standpoint what kind of driving equipment will best serve plant needs, whether motive power will consist of gas (Fig. 67), steam, gasoline (Fig. 68) or diesel-engine, turbine, or electric drive. This also means giving full

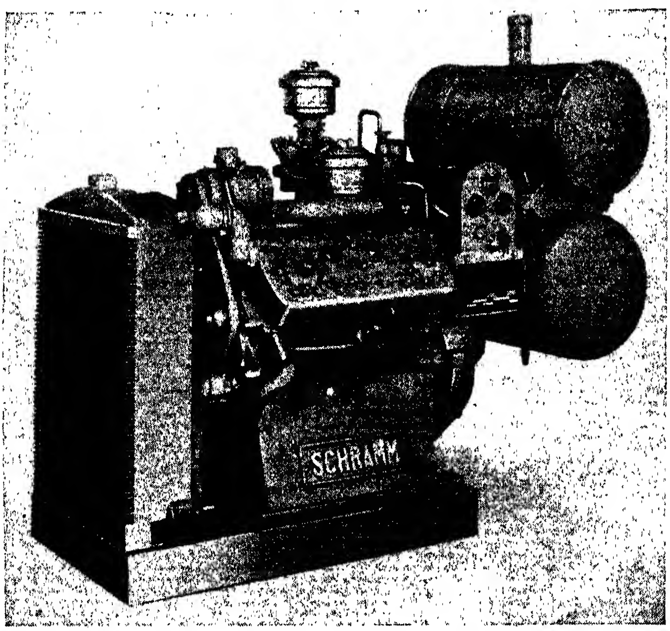


FIG. 68.—Gasoline engine with four cylinders used for compressing air. (*Courtesy of Schramm, Inc.*)

consideration to labor, maintenance and depreciation costs, interest on investment, type of compressor, accessory installation, and piping and connections.

Over-all calculations must include future as well as present requirements. The first consideration is the amount of air needed for expected operations. The manufacturer of the air-operated equipment will furnish this information. The plant operating cycle will then determine the load factor of these tools.

Install Adequate Capacity.—When all air requirements are known, increase the figure 50 per cent, then divide the total load between a minimum of two compressors. The 50 per cent increase represents spare capacity—a reserve that protects the production schedule because output can be increased later by using air in operations that previously had not been air-operated. Since air compressors are mechanical equipment and naturally require some maintenance, the plant with at least two compressors is never totally without air during outage of one unit for repairs.

TABLE 8.—COST OF COAL AT $1 PER TON TO COMPRESS AND DELIVER 100 CU FT OF FREE AIR

Multiply figures by price of coal in dollars per ton; ton = 2,000 lb

Ihp to deliver 100 cfm of free air	Lb of steam per ihp-hr required by the engine									
	24	26	28	30	32	34	36	38	40	42
16	0 0457	0 0495	0 0533	0 0572	0 0610	0 0634	0 0680	0 0724	0 0762	0 0800
18	0 0514	0 0557	0 0600	0 0643	0 0686	0 0714	0 0772	0 0814	0 0857	0 0900
20	0 0570	0 0619	0 0667	0 0714	0 0762	0 0793	0 0857	0 0904	0 0952	0 1000
22	0 0627	0 0681	0 0733	0 0785	0 0838	0 0872	0 0943	0 0995	0 1047	0 1100
24	0 0684	0 0743	0 0800	0 0857	0 0914	0 0951	0 1029	0 1085	0 1142	0 1200
26	0 0741	0 0805	0 0866	0 0929	0 0990	0 1030	0 1114	0 1175	0 1238	0 1300
28	0 0798	0 0867	0 0933	0 1000	0 1066	0 1109	0 1200	0 1266	0 1334	0 1400
30	0 0855	0 0929	0 1000	0 1071	0 1142	0 1189	0 1285	0 1357	0 1429	0 1500

Selecting the Prime Mover.—Many factors enter into the selection of a prime mover. One of the first should be the price of fuel and its availability. Generally, the first thought is for a motor-driven installation, but it is best not to arrive at a hasty decision. Perhaps a better plan lies in first considering the various other kinds of motive power. For each of the following drivers there is a niche where it is the best possible selection from considerations of first cost, operating economy, and reliability: (1) steam—either direct-connected steam engine or turbine, (2) oil—either diesel or gasoline engine, and (3) gas—either natural or manufactured-gas engine.

A plant served with electrical energy at reasonable rates, having no problems of frequent or prolonged current interruptions and requiring no steam for process work, usually finds electric-motor drive best. As a rule, compressors that require drivers larger than 100 hp are synchronous-motor-driven, with the motor mounted directly on the compressor crankshaft (Fig. 3). This conserves space and eliminates belt maintenance. Usually, smaller units use belt drive to reduce initial investment when belt replacements are not extremely expensive.

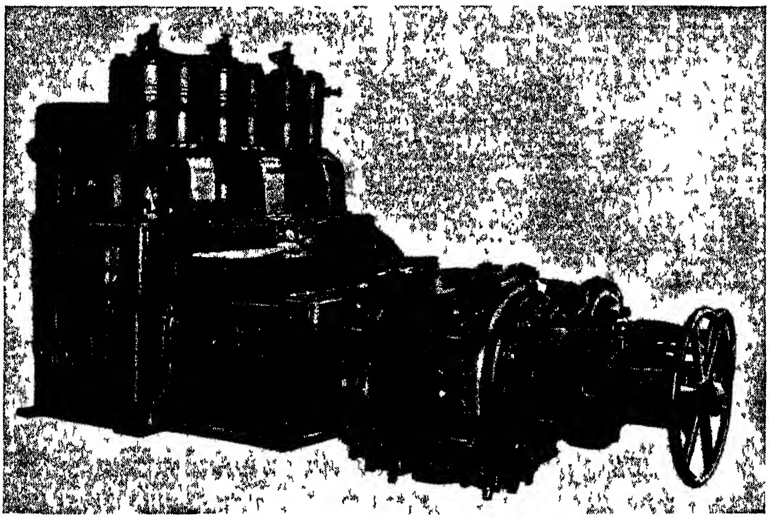


FIG. 69.—En bloc construction showing hand-operated clearance valves (*Courtesy of Clark Brothers Co, Inc*)

In localities where fuel oil is cheap and readily available, a diesel drive (Fig. 69) may be best If there is a plentiful supply of coal at a low price or a surplus of available steam, or if considerable steam is needed for process work, steam might be the economic answer. An abundant supply of natural gas at reasonable cost would indicate using a gas engine as a prime mover.

A fair idea of the relative operating cost of various drives may be obtained from studying Tables 8, 9, 10, 11, and 12. Power-consumption rates of a two-stage unit having an actual delivery of approximately 1,500 cfm, compressing from atmosphere at sea level to 100 psi are

Motor........................	372 cu ft per kwhr
Steam*........................	13,600 cu ft per 1,000 lb
Diesel........................	5,600 cu ft per gal
Gas engine....................	30 cu ft per 1,000 Btu

* Based on 150 psi steam, 50 F superheat, and 5 psi back pressure. Increase or decrease 6 per cent per 25 psi change in steam pressure.

Study the Fuel Situation.—When studying the fuel situation, compute the comparative cost of transforming energy stored up in these sources into a given quantity of compressed air, say 100 cfm at the required pressure.

Before using the cost tables, determine the bhp required to compress and deliver 100 cfm from curves (Figs. 13 and 14). This horsepower varies with the discharge pressure and altitude. Mechanical and compression efficiencies of the machine also affect the power input, but to a smaller degree. The time required to compress this given quantity of air need not be considered when determining cost from the tables, but the figures given must be adjusted to the prevailing prices where the compressor plant is located, as follows:

Steam-engine Drive.—To obtain the cost of coal, first determine the steam consumption and the ihp (indicated horsepower) required to compress 100 cfm at the necessary pressure.

TABLE 9.—COST OF FUEL OIL IN CENTS TO COMPRESS 100 CU FT OF FREE AIR

Oil consumption ½ pt (about 0.45 lb) per bhp-hr

Bhp to deliver 100 cfm of free air	Based on fuel oil with effective heat value of 18,500 Btu per lb Price of fuel oil, cents per gal							
	3	4	5	6	7	8	9	10
16	0.050	0.067	0.083	0.100	0.117	0.133	0.150	0.167
18	0.056	0.075	0.094	0.113	0.131	0.150	0.169	0.189
20	0.063	0.083	0.104	0.125	0.146	0.167	0.188	0.210
22	0.069	0.092	0.115	0.138	0.160	0.183	0.206	0.230
24	0.075	0.100	0.125	0.150	0.175	0.200	0.225	0.251
26	0.081	0.109	0.136	0.163	0.190	0.217	0.244	0.272
28	0.088	0.117	0.146	0.175	0.204	0.233	0.262	0.293
30	0.094	0.125	0.156	0.188	0.219	0.250	0.281	0.314

Suppose coal costs $1 per ton and each pound evaporates 7 lb of water. Refer to Table 8, which gives the cost of coal, based on \$1 per ton, to compress and deliver 100 cu ft of free air. If the compressor requires 16 hp and uses 24 lb of steam per ihp, then the figure found in column 2, line 1 is 0.0457 cent.

TABLE 10.—COST OF GASOLINE IN CENTS TO COMPRESS AND DELIVER 100 CU FT OF FREE AIR

Gasoline consumption 1 pt (about 0.75 lb) per bhp-hr

Bhp to deliver 100 cfm of free air	Price of gasoline, cents per gal							
	8	9	10	11	12	13	14	15
16	0.267	0.300	0.333	0.367	0.400	0.433	0.467	0.500
18	0.300	0.338	0.375	0.413	0.450	0.488	0.525	0.563
20	0.333	0.375	0.416	0.458	0.500	0.542	0.584	0.626
22	0.367	0.413	0.458	0.504	0.550	0.596	0.642	0.688
24	0.400	0.450	0.500	0.550	0.600	0.650	0.700	0.750
26	0.433	0.488	0.542	0.596	0.650	0.705	0.759	0.813
28	0.467	0.525	0.584	0.642	0.700	0.759	0.816	0.875
30	0.500	0.563	0.626	0.688	0.750	0.813	0.875	0.938

TABLE 11.—COST OF GAS IN CENTS TO COMPRESS AND DELIVER 100 CU FT OF FREE AIR

Based on fuel consumption of 10,000 Btu per bhp-hr

Bhp to deliver 100 cfm of free air	Rate based on 1,000 Btu gas Cost of gas in cents per 1,000 cu ft							
	10	15	20	25	30	35	40	45
16	0.026	0.039	0.052	0.065	0.078	0.091	0.104	0.117
18	0.029	0.044	0.059	0.073	0.088	0.102	0.117	0.132
20	0.033	0.049	0.065	0.081	0.098	0.114	0.130	0.146
22	0.036	0.054	0.072	0.090	0.107	0.125	0.143	0.161
24	0.039	0.058	0.078	0.097	0.117	0.136	0.156	0.176
26	0.042	0.063	0.085	0.106	0.127	0.148	0.169	0.190
28	0.045	0.068	0.091	0.114	0.137	0.159	0.182	0.205
30	0.049	0.073	0.098	0.122	0.146	0.170	0.195	0.220

This gives a basic figure that can be multiplied by the actual coal price and the evaporation factor of the particular steam generator. For example, if coal is \$4 a ton the cost would be $4 \times 0.0457 = 0.1828$ cent. Then, if the boiler evaporates 8 lb of water per lb of coal, $\frac{7}{8} \times 0.1828 = 0.1599$ cent, the cost per 100 cfm.

Steam-driven compressors generally prove most economical in plants that generate steam for process or for electrical energy. Table 13 shows the economics of installing steam-driven compressors to supply the air requirements of a large airplane-propeller-manufacturing plant. This plant needed process steam that, if generated at higher pressure, would supply, at practically no cost, at least half of the compressed air in summer and all of it in winter. Because exhaust steam from the compressor supplies heating and process, only 10 per cent of the steam cost, or 19.1 cents per hr, is chargeable to the compressor. Therefore, the cost of 1,000 cu ft of air is $19 \div (800 \times 60) = 4$ mills.

In summer, the plant requires steam only for process work up to a maximum of 3,500 lb per hr at 10 psi. At this exhaust pressure, the compressor water rate falls to approximately 27 lb per hp-hr. Then,

Total hourly steam	4,450 lb
Process requirements	3,500 lb
Steam wasted	950 lb
Cost of steam wasted	$0 35
10 per cent of process steam cost	0 12
Total hourly operating cost for 48,000 cu ft of air	$0 47
Cost per 1,000 cu ft in summer	$0 01

These figures are based on full compressor load; at partial load, there would be no waste steam in summer, thereby further cutting operating costs. Considering electric-motor drive on a basis of 20.6 hp per 100 cu ft and 92 per cent motor efficiency, 6,000 cu ft delivered requires 16.70 kwhr. At 1.4 cents per kwhr, the total hourly cost equals 23 cents, or 3.8 cents per 1000 cu ft.

Final Selection.—Based on this analysis, two 800 cfm steam-driven compressors operating at 165 psi initial steam pressure and exhausting to 35 psi were installed, together with one 900 cfm motor-driven unit. During 7 months of the year the two steam-driven machines operate; during the summer months one steam-driven unit supplies the base air load and process-steam requirements, with the motor-driven unit supplying the balance of the air.

Diesel-engine Drive.—Assume that the compressor is to be driven by a diesel engine requiring approximately the same ihp. Table 9 gives the fuel-oil cost to compress 100 cfm of free air at an oil consumption of $\frac{1}{2}$ pt per bhp-hr. With an oil price of 3 cents per gal and an ihp of 16, the total cost, obtained from the table, is $\frac{1}{2}$ cent per 100 cfm.

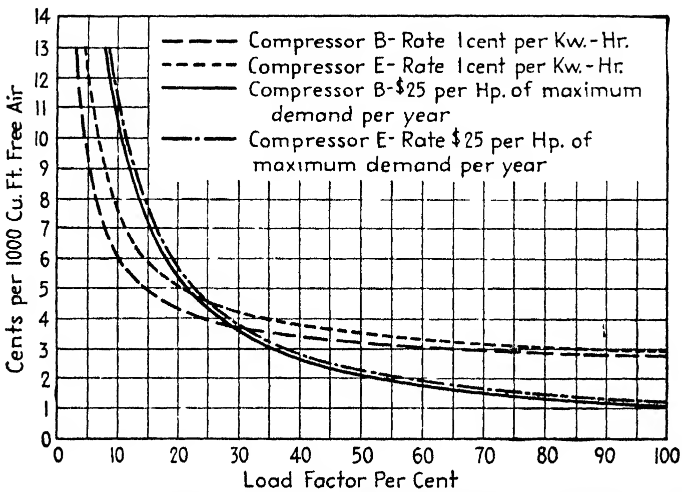


FIG. 70.—Cost of electricity for compressing air at different load factors. (*Courtesy of Power.*)

Gasoline-engine Drive.—Table 10 gives the cost of gasoline based on a consumption of 1 pt per bhp-hr. This amount will not vary greatly, but where there is a variation, the figures given in the table should be changed accordingly. If gasoline costs 8 cents per gal, then line 1, column 2 indicates a cost of 0.267 cent to deliver 100 cfm of free air at an ihp of 16.

Gas-engine Drive.—Table 11 is based on gas having a heating value of 1,000 Btu per cu ft. If the gas has a different heat content, the figures should be changed proportionately. For 16 ihp and gas at 10 cents per 1,000 cu ft, the cost would be 0.026 cent.

In the operation of the internal-combustion engine, consideration should be given to energy dissipated in the cooling system and through the exhaust and lubricating oil. Some of this otherwise waste energy may be reclaimed by heat exchangers for certain manufacturing processes or for heating.

Oil- or gas-engine-driven equipment serves best when (1) these fuels are available at economical rates, (2) it is desirable to have a compressed-air system independent of other utilities, or (3) installation is in a remote section, such as an oil field or a construction job not served by utilities.

TABLE 12.—COST OF ELECTRIC CURRENT IN CENTS TO COMPRESS AND DELIVER 100 CU FT OF FREE AIR

Based on 100 per cent motor efficiency

Bhp to deliver 100 cfm of free air	Price of electric current per kwhr, cents								
	1	1.5	2	2.5	3	3.5	4	4.5	5
16	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
18	0.225	0.337	0.45	0.562	0.675	0.787	0.9	1.012	1.125
20	0.25	0.375	0.5	0.625	0.75	0.875	1.0	1.125	1.25
22	0.275	0.412	0.55	0.687	0.825	0.962	1.1	1.237	1.375
24	0.3	0.45	0.6	0.75	0.9	1.05	1.2	1.35	1.5
26	0.325	0.487	0.65	0.812	0.975	1.137	1.3	1.46	1.625
28	0.35	0.525	0.7	0.875	1.05	1.225	1.4	1.575	1.75
30	0.375	0.562	0.75	0.937	1.125	1.312	1.5	1.687	1.875

TABLE 13.—WINTER OPERATING COST OF 11 × 18/11 × 14 COMPRESSOR

Total air delivery	800 cfm at 125 psi
Total horsepower	165
Steam per hp per hr (165 psi and 35 psi exhaust)	33 lb approx
Total steam per hr	5,450 lb
Air delivered per hr	48,000 cu ft
Cost of steam, basis 35 cents per 1,000 lb	$1.91 per hr

A large manufacturer who needed a compressed-air system that was not dependent on the public utility—from which he normally bought power, but which could not guarantee uninterrupted service during electrical storms—installed diesel-engine-driven compressors. These units also have generators mounted on the engine shaft. On complete power failure, the air compressor may be either wholly or partially unloaded; the engine-driven generator then keeps the plant operating until utility service is restored.

Electric-motor Drive.—In selecting electric motors, give careful study to starting torque, starting current, and suitable flywheel effect. The squirrel-cage induction motor usually drives belted and high-speed direct-coupled compressors. The synchronous motor is used occasionally, however, even on these smaller machines because of its capacity for power-factor correction.

The cost of electricity obtained from Table 12 is based on 100 per cent motor efficiency. With power at 1 cent per kwhr and a compressor requiring 16 bhp, the cost of delivering 100 cfm of air would be 0.200 cent. For example, if the current costs 6 cents per kwhr, then $6 \times 0.20 = 1.2$ cents to deliver 100 cfm. If the motor operates at 90 per cent efficiency, $1.2 \div 0.90 = 1.33$ cents per 100 cfm.

TABLE 14.—COMPRESSOR PLANT TEST RESULTS

Compressor	Size	Speed, rpm	Piston displacement, cfm	Discharge air pressure, psi	Volumetric efficiency, per cent
A	26 by 15½ by 18	185.5	2,042 5	100	85.1
B	26 by 15½ by 18	185.3	2,039.7	100	86.5
C	26 by 15½ by 18	188	2,070	100	88.0
D	25¼ by 15¼ by 21	188	2,117	100	79.2
E	22 by 14 by 18	157.8	1,241.2	100	88.6

Compressor	Hp input per 100 cfm of free air	Kwhr per 1,000 cu ft of free air	Hp input, no load	Drive
A	22.85	2.85	53.1	Direct-connected
B	22.75	2.83	51.0	Direct-connected
C	23.26	2.91	56.6	Direct-connected
D	23.89	2.99	60.3	Direct-connected
E	23.7	2.94	46.0	Belted

Because all motor-driven compressors are constant-speed machines, the cost of electricity depends on (1) power efficiency or power input per cu ft of air, (2) plant load factor, and (3) rate charged for electricity.

A value for power efficiency must be assumed; it may be secured from the manufacturer's guarantee or taken from test data on an identical compressor.

The load factor is the ratio of actual air output in a given period to the maximum amount the machine could produce in that time. Since motor-driven machines run at constant speed whether loaded or unloaded, the energy consumed during idling periods increase the energy input per cu ft of air as the load factor decreases.

Effect of Electric Rates.—Electric rates have the most important bearing of all upon operating cost. Usually current costs so many cents per kwhr, so many dollars per horsepower or kilowatt of maximum demand, or a combination of the two. Table 14 gives the results of tests made on five compressors and the curves (Fig. 70) illustrate the effect of the load factor on air

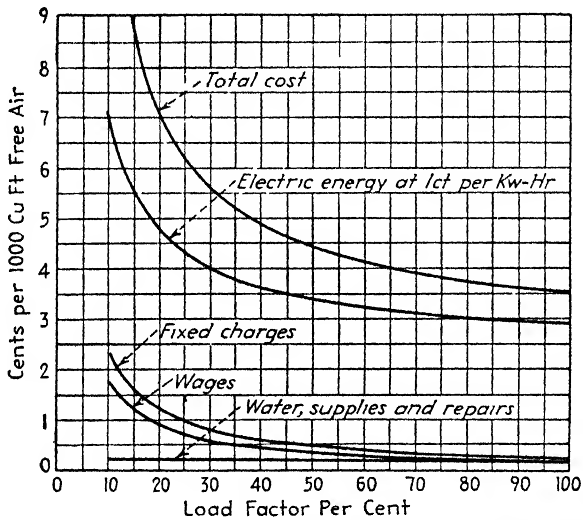


FIG. 71.—Total cost of compressing 1,000 cfm of free air at different load factors. (*Courtesy of Power.*)

cost, based on two methods of energy charge. At low load factors the 1-cent rate is better, but it is not so good at higher factors. Table 15 gives cost data on another plant consisting of two-stage duplex belt-driven compressors having a total actual output of 5,065 cfm. The curves (Fig. 71) break these costs down in relation to load factors.

First Cost Smallest.—The question of what kind of air-compressor drive to use depends on a great many variable quantities. Each must be worked out individually, taking all local conditions into careful consideration as well as striking a working balance between first cost and operating cost. The first cost of a compressor is the smallest; operating expenses far exceed the entire first cost of the equipment. Careful selection, thorough knowledge of compressor capacity, and constant attention to maintenance details all add up to better performance and greater all-over economy.

TABLE 15.—FIRST COST COMPARED TO YEARLY COST

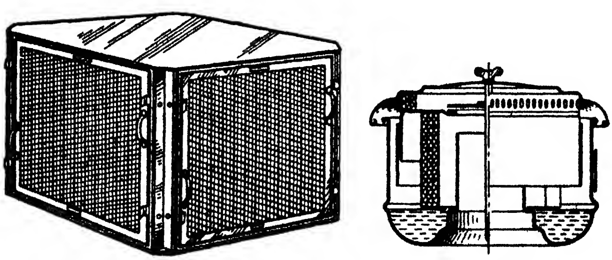
First cost		
Cost of the five compressors, including motors and switchboard		$31,000
Piping, aftercooler and air receiver		3,000
Building and foundations (land not included)		2,700
Installing		2,300
Total		$39,000
Yearly cost		
Fixed charges		
Interest on investment $39,000 × 6 per cent =	$2,340	
Taxes, 1 per cent on land and building	200	
Insurance at 20 cents per $100 per year	78	
Depreciation, at 10 per cent per year	3,900	
Total yearly fixed charges		$ 6,518
Operating charges:		
Power on basis of load curve giving load factor of 78½ per cent		$63,422
Water, average daily consumption 7,000 cu ft or $200 per month at 10 cents per 100 cu ft		2,400
Wages, average $400 per month for three 8-hr shifts		4,800
Supplies, such as oil, waste, etc., average per month $103		1,236
Repairs average per month, $49		588
Total operating charges		72,446
Total yearly cost		$78,964

CHAPTER III

COMPRESSOR ACCESSORIES

Accessories such as (1) intake filters and silencers, (2) cooling systems, (3) receivers, (4) strainers and separators, (5) moisture traps, and (6) protective devices are not usually included in the purchase of a compressor.

Importance of Filters.—Dust and other foreign matter entering a compressor cause sticking valves, scored cylinders, and abnormal wear of moving parts. Passing on into distribution lines, dirt and grit cause similar damage to tools and valves and contaminate certain process work. Filters on the compressor intake eliminate these troubles.



Ⓐ Viscous Coated Wire Ⓑ Oil Filled

FIG. 72.—Sectional panels and unit air filters. (*Courtesy of Air-Maze Corp.*)

From 85 to 95 per cent of the so-called "carbon" found in air-compressor cylinders is dirt saturated with oil and baked into a hard mass. One would not consider throwing a handful of dirt into the cylinder, yet this is about what is done on machines operating without air filters. For example, consider a 2,000 cfm air compressor and assume it operates at full capacity on an average of 10 hr per day, 6 days per week. In one week there would pass into the compressor 7,200,000 cu ft of air.

Air in industrial districts contains from 1 to 4 grains of dirt per 1,000 cu ft. If the air contained only 1 grain per 1,000 cu ft, 7,200 grains of dirt would pass into the compressor in 1 week's operation. Assuming the higher figure of 4 grains per cu ft, 28,800 grains, or over 4 lb, of dirt would be carried into the compressor during a week's operation. Frequently much of the dirt carried in air is of an abrasive character. If allowed to get into the cylinder it mixes with the lubricating oil and causes rapid wear of piston rings, cylinder walls, valves, and other parts.

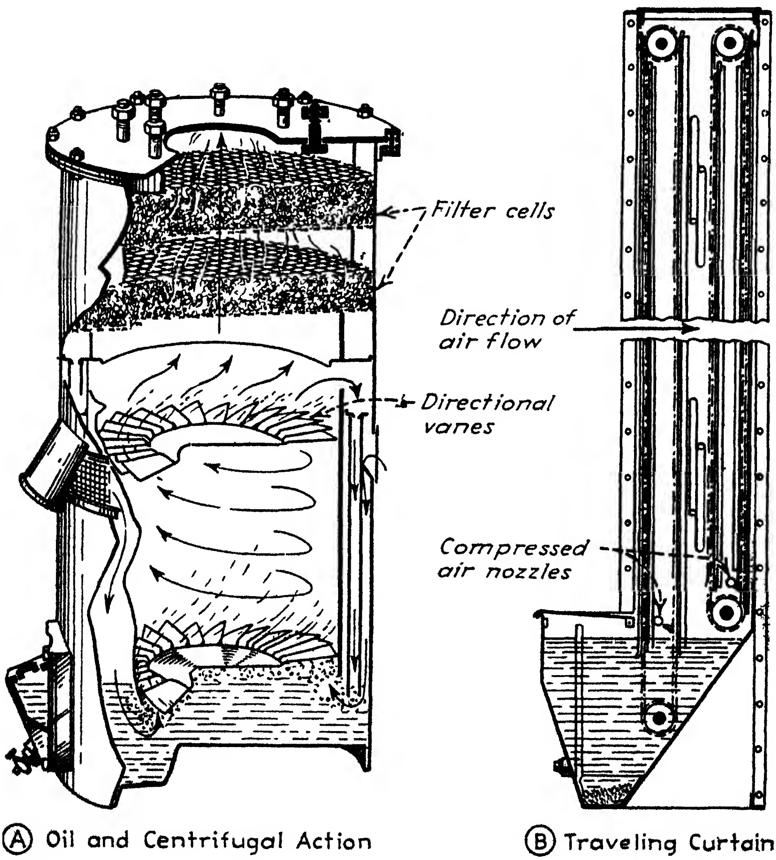


FIG. 73.—*A*, Centrifugal air-flow unit also acts as a silencer. *B*, Traveling curtain cleans itself in oil bath. (*Courtesy of American Air Filter Co., Inc.*)

Two new identical gasoline engines, one equipped with an air filter and the other without, were operated 240 hr.

After operating in severe dust conditions, careful measurements were made of the wear on the pistons, cylinders, and piston rings, and an analysis was made of the carbon deposits. The unprotected engine showed nine times the wear on cylinders, four times the wear on the pistons, and ten times the wear on the piston rings that the protected engine did. The carbon deposits in the engine run without an air filter contained five times more dirt than was found in the engine operated on filtered air. The unprotected engine was practically worn out, whereas the other was in good condition.

Dirt in the air also causes increased carbon deposits in the cylinders and on the piston rings and valves. This, combined with increased wear, reduces efficiency and causes increased maintenance cost and outage time for repairs. For example, the valves on a 5,000 cfm compressor operating in a railroad shop required cleaning every 2 weeks when the air intake was unprotected by a filter. After an air filter was installed the only attention required was to wipe the valves off once every 6 months.

Of course, the value of air filters on any installation depends upon local conditions. If operation is continuous for 24 hr per day where the air is extremely dirty, air filters will more than pay for themselves in a single year by the saving in maintenance and lubrication cost, increased efficiency, and reduced outage time. On the other hand, if operation is intermittent and the air is fairly clean, the saving in operating cost made by the installation of an air filter may be small. However, experience with the protection of many machines has shown that it pays to protect.

Filters and Cleaners.—Viscous-coated filters catch dust by passing the air around a series of viscous-coated-metal baffles or through closely packed crimped wire (Fig. 72*A*). Viscous impingement filters should not be used where the air contains large amounts of lint or in localities subject to heavy dust storms.

Another type of filter consists of perforated aluminum plates with a film on their surfaces. The plates are arranged in frames so that the air is passed through the holes in one plate to impinge upon the metal between perforations in the following plate. In this way the dirt is projected against the film and thus retained, while the air changes its direction to flow through orifices in the next plate. By this process the air is cleaned as it passes through successive plates. The so-called "dry filters" consist of either glass wool or special felt cloth supported by wire mesh (Fig. 74).

Small centrifugal filters remove dirt by whirling the air around the inside wall of the circular intake chamber. Combinations of the viscous-impingement and centrifugal principle are also used. In these the air receives a primary cleaning by centrifugal action, the dirt collects in an oil pool, and the air then passes through an impingement filter (Fig. 72*B*).

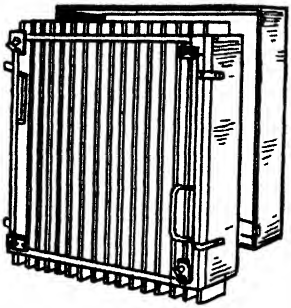


Fig. 74.—Dry-felt filter.

The large design (Fig. 73*A*) takes air in through a screened opening and passes it down between two outer walls to strike the oil surface at the bottom. Heavy dust particles settle on the oil as the air continues upward through vane openings that give it and the entrapped oil a whirling motion. Centrifugal force throws the lighter dirt with oil to the side wall. Passing through another series of vane openings, the air whirls faster, throwing the remaining dirt and oil into a basin around the side. Oil and dirt flow down an inner wall passage and air continues up through the filter cells to the compressor inlet. Changing the direction of air flow and expanding and contracting the air stream as it flows through the filter silence any intake noises made by the compressor.

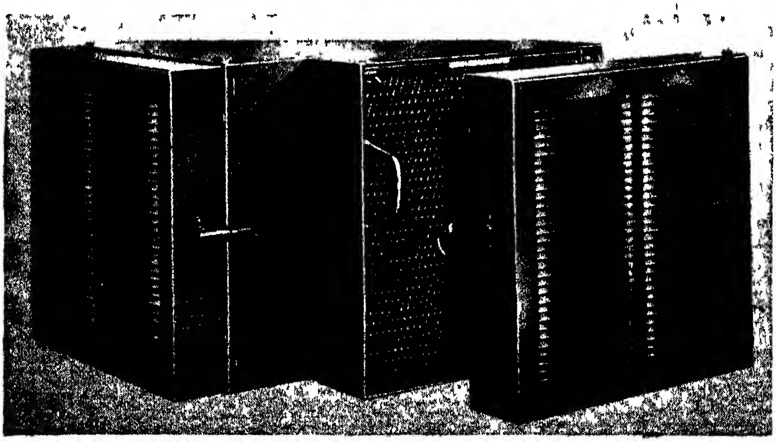


Fig. 75.—Large-capacity units composed of group panels.

The self-cleaning filter (Fig. 73*B*) consists of two endless curtains made from panels of bronze cloth, backed by successive layers of metallic mesh. The front curtain moves down into an oil bath, allowing large particles of dirt to settle to the bottom of the basin. Emerging from the oil, the curtain moves up to act as a second filter while a compressed-air spray removes excess oil from it. The third filter (second curtain) moves down but does not enter the oil. Continuing around the lower sprocket, another air spray cleans the second curtain as it moves up to the final or fourth filter stage.

Each panel *A* (Figs. 72 and 74) forms a complete unit. The number of units used depends on the filtering capacity needed (Fig. 75); *e.g.*, if it requires 1 sq in. of panel surface for every 2 cfm of air, a 20,000 cfm compressor plant will require 10,000 sq in. of filter surface.

Air washers can also be used for large installations. In these the air first passes through atomized water sprays. Leaving the spray chamber, the air passes between corrugated-metal sheets kept flooded with a film of water, so that the dirt which has been wet by the spray is thrown against the sheets and washed to the settling basin below. Any unabsorbed moisture in the air leaving the washer is removed between dry corrugated plates.

Cleaning.—Intake filters must be cleaned often enough to remove dirt before restriction of the intake occurs. Local conditions determine the frequency of cleaning; weekly inspections expose conditions before they become serious. The method of cleaning depends on the type of filter.

Air Cooling.—If the free air is effectively cleaned before it enters the compressor, there usually remains only the problem of removing moisture and atomized oil.

The amount of water vapor that can be present at a given temperature in a given space is independent of the other gases present. For example, 1,000 cu ft of free air at 70 F and 100 per cent relative humidity contains 1.15 lb of water vapor (Fig. 76). If this air is compressed to 100 psi gage pressure and cooled to its original temperature, it will then contain saturated water vapor weighing only 0.15 lb, because the air volume has been reduced from 1,000 to 128 cu ft. The difference in the weight of water vapor under the two conditions, $1.15 - 0.15 = 1$ lb, is precipitated from the air as water.

A 2,000 cfm compressor at rated capacity, operating 8 hr per day, handling free air of 64 per cent relative humidity, passes

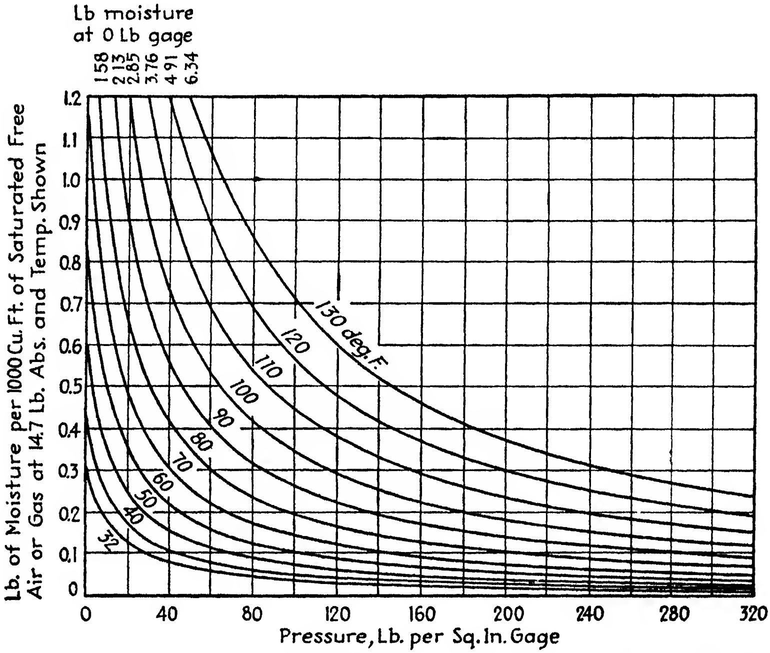


FIG. 76.—Moisture remaining in saturated air when compressed isothermally to different pressures. (*Courtesy of Ingersoll-Rand Co.*)

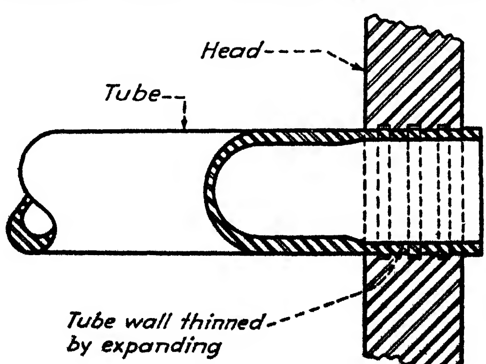


FIG. 77.—Intercooler tubes expanded into tube sheet.

about 710 lb of water vapor each day (0.64 × 1.15). If the air is compressed to 100 psi and cooled to its original temperature, only about 16 per cent of the vapor remains in the air, so that about 596 lb will be deposited as water in the different parts of the compressed-air system, such as intercoolers, aftercoolers, receivers, and pipe lines.

When air carries water into tools and other devices that require lubrication, the water washes away the lubricant and causes excessive wear and maintenance. Because of inadequate lubrication, the equipment operates unsatisfactorily and its efficiency decreases. Water in compressed air also causes water hammer in pipe lines and reduces their carrying capacity by collecting at low points. It may freeze and burst the pipes in cold weather.

Dry-air Delivery.—The only way to deliver dry air to pipe lines and tools is to prevent the formation of liquid particles

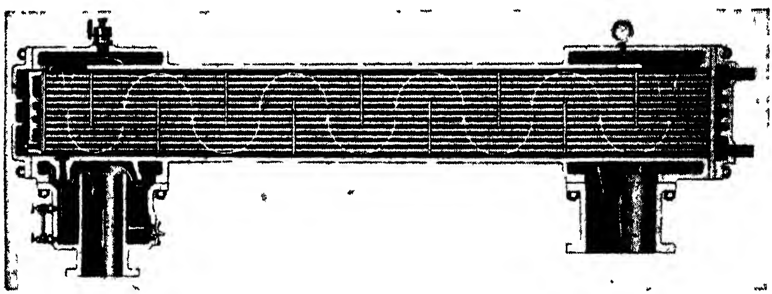


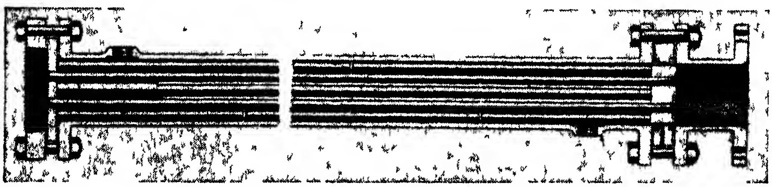
Fig. 78.—Intercooler equipped with float-operated condensate drain. (*Courtesy of Ingersoll-Rand Co.*)

during transit by precooling the air to such a temperature that further cooling in the line is impossible. For such complete cooling and drying, the air must be cooled to a temperature as low as or lower than it will reach in the line.

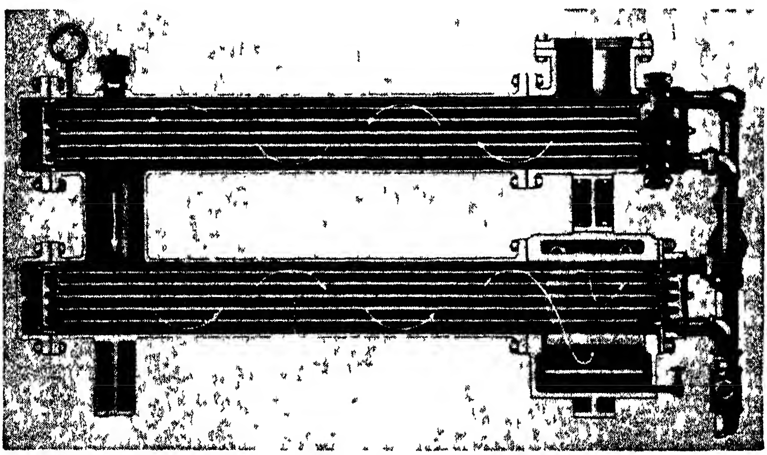
Air as it passes through the compressor takes up atomized oil and carries it along. The oil, if not removed, goes into pipe lines and comes in contact with rubber gaskets and hose lines, which it will damage. Oil is also objectionable in compressed air used for paint spraying, cleaning electrical machinery, and agitating liquids.

The place to begin removing oil and moisture is at the compressor inter and aftercooler. (Intercoolers are not classed as accessories but since they belong in the cooling system they will be discussed here.) Coolers consist of a nest of tubes expanded into two heads (Fig. 77) supported inside a steel shell. Figure

78 illustrates a typical unit intercooler, Fig 79A a single-, and Fig. 79B, a double-shell aftercooler One tube head is fastened solidly in the shell while the other floats free to allow for tube expansion and contraction. The arrangement (Fig 80) eliminates the floating head by inserting an inlet tube, leading from one tube head concentrically into a larger outlet tube. The latter is closed at its far end, then connects to a different tube sheet, allowing one end of each tube to float free.



A



B

Fig 79 —A, Single-pass and B, double-pass aftercoolers.

Atmospheric air coolers consist of finned tubes to give greater cooling surface. Used mostly on air-cooled compressors, they eliminate the need of a cooling-water supply.

Cooler Operation.—Air coolers serve (1) to cool the air and condense water vapor and oil, and (2) when used as intercoolers on multistage units, to reduce the total power required to compress the air.

Economies of cooling depend on (1) the temperature and quantity of the cooling medium; (2) the amount of and cleanliness

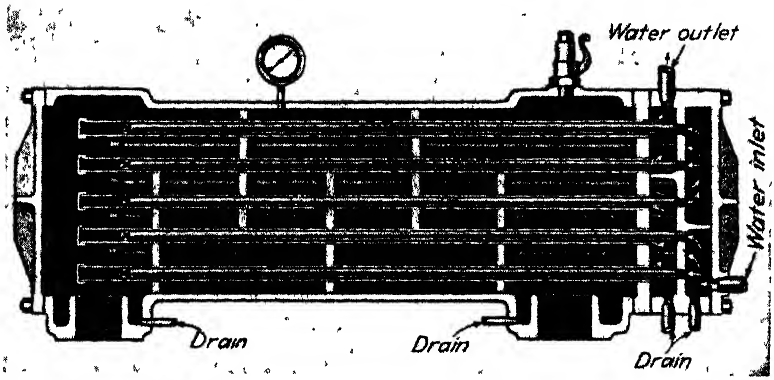


FIG. 80.—Double-tube construction eliminates expansion and contraction troubles. (*Courtesy of Bury Compressor Co.*)

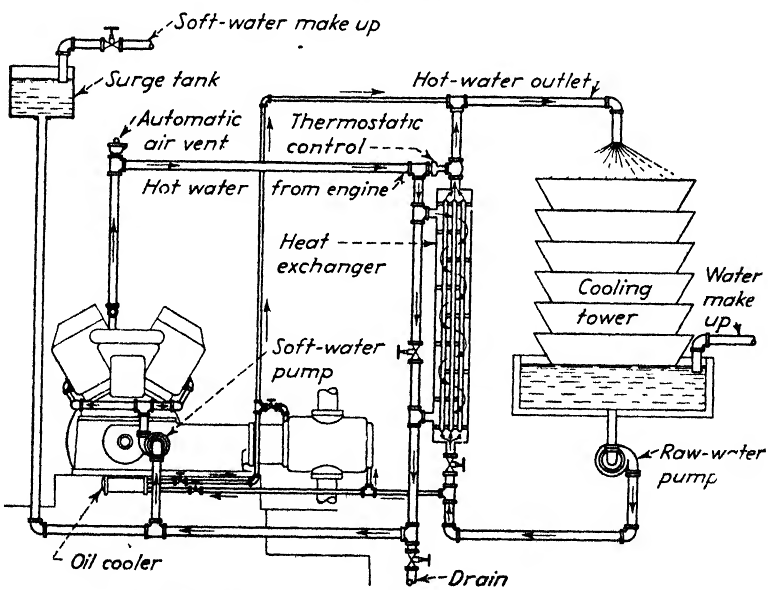


FIG. 81.—Heat exchanger for power-cylinder cooling and raw water for compressor. (*Courtesy of Ingersoll-Rand Co.*)

of the cooling surface; and (3) the degree of intimate contact between the air on one side of the separating barrier and the cooling medium on the other side. Coolers are available that cool the air down to temperatures from 15 to 2 F above that of the cooling water.

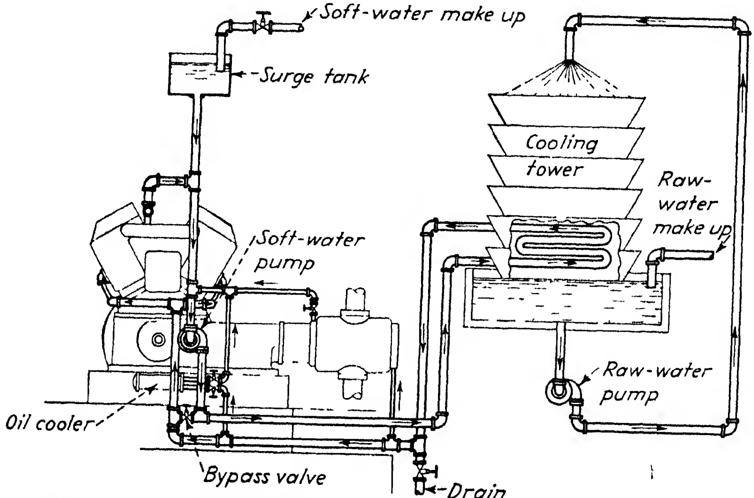


FIG. 82.—Closed system for power and air cylinders utilizing pipe coil in cooling tower. (*Courtesy of Ingersoll-Rand Co.*)

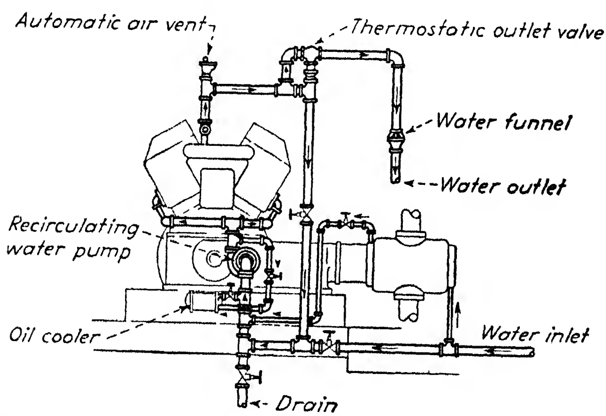


FIG. 83.—Raw water goes to air cylinders; power cylinders use closed system protected by thermostatic valve. (*Courtesy of Ingersoll-Rand Co.*)

Water, under turbulent flow, swirls and boils in passing through the tubes or jacket and sweeps the warm water away from the metal surfaces, replacing it with cooler water; this mixing action continues until the water flows out the discharge. Baffles inserted in compressed-air passages produce turbulence by destroying any tendency to stratified layer flow. Countercurrent flow, or feeding the water into the cooler in a direction opposite to that of the air, produces the coolest discharge air because it brings the coolest water in contact with the outgoing air.

Division of the tube nest into groups of tubes passes the water back and forth through the cooler several times before discharging it. Bulkheads or partitions cast integral with both shell heads bear against respective tube sheets and serve as circulating pockets for backflow through the next group of tubes.

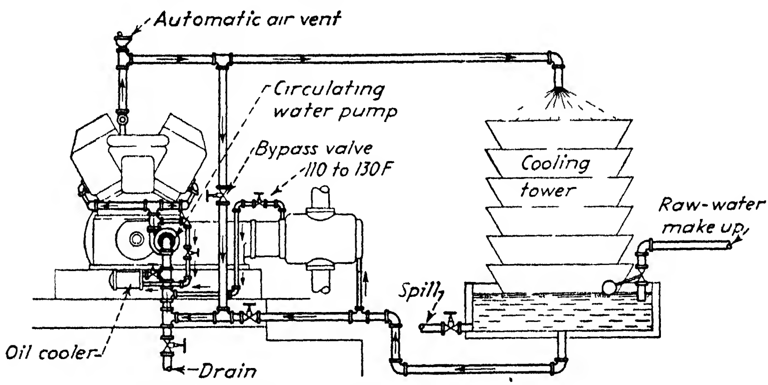


FIG. 84.—Cooling-tower recirculating system is not recommended. (*Courtesy of Ingersoll-Rand Co.*)

Good cooler design includes a water leg and drain connection to catch and discharge moisture condensed as the air cools. Connections to the air compartment should permit installing a pressure gage, a safety valve, and an interstage unloader if one is used.

When the cost of cooling water is high, it is frequently advisable to put in a cooling system that permits reusing the water. This may be in the form of an evaporative aftercooler that occupies a relatively small space, or a closed cooling system may be used. Figure 81 illustrates a closed system with a heat exchanger furnishing soft water for power cylinders with raw water going to the compressor; Fig. 82, a closed system with a pipe coil in the cooling tower furnishing soft water to both power and compressor cylinders; Fig. 83, a regular open system; and Fig. 84 an open system with an atmospheric cooling tower.

Air Receivers.—The chief functions of a receiver are to dampen pulsations or pressure waves created by the intermittent compressor discharge and to furnish reserve air capacity. If the receiver is too small, pressure waves will acquire considerable amplitude and increase the power consumption.

A large receiver will not assure minimum pressure waves if the discharge pipe is too small or contains many bends. Experience dictates that, for double-acting compressors running between 200 and 400 rpm, the combined receiver and discharge-pipe volume should be approximately twenty times the high-pressure cylinder displacement per revolution. Recommended sizes of receivers are given in Table 16.

TABLE 16.—ASME STANDARD AIR RECEIVERS

Dia, in.	Length, ft	Actual compressor capacity for which receiver is suited, cfm of free air*	Safety valves		Volume, cu ft	Types of opening for cleaning
			No.	Dia, in.		
14	4	60	1	¾	4½	Openings for inspection and cleaning are required and must meet specifications of ASME code for unfired pressure vessels
18	6	95	1	1	11	
24	6	185	1	1½	19	
30	7	305	1	2	34	
36	8	450	1	2½	57	
42	10	640	1	3	96	
48	12	1,275	2	3	151	
54	14	1,900	3	3	223	
60	14	3,000	3	3	275	
66	18	4,500	3	3	428	

* For automatic start-and-stop service, extra-large receivers are recommended to avoid starting too frequently.

Since most air-compressor plants furnish air for intermittent use, there will be periods of peak demand when simultaneous use at all points requires more air than the compressor can provide. Without a receiver, the pressure would fall almost instantly, causing serious interruption to service.

Large receivers installed at the end of long or undersized distribution systems, where the maximum air use exceeds the line capacity, improve operating conditions considerably.

Regulation or control of compressor output depends upon slight pressure changes alternately to load and unload the unit,

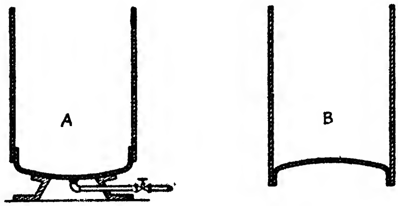


FIG. 85.—Stand vertical receivers so that moisture does not lie in the head seams.

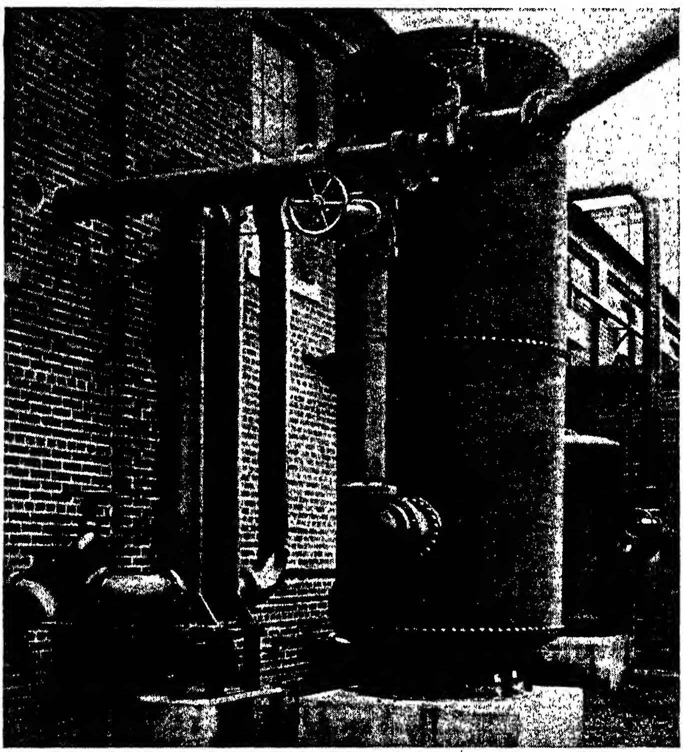


FIG. 86.—Air-receiver foot keeps it away from moisture lying on the concrete. (*Courtesy of Worthington Pump and Machinery Corp.*)

or to vary the operating speed, which in turn varies the compressor output. Without sufficient receiver capacity these changes occur so frequently as to cause unnecessary wear on regulating devices.

Considerable danger arises from using air receivers of unsound or questionable construction. The Compressed Air Institute endorses only those air receivers that meet ASME code requirements and has standardized on the sizes and capacities given in Table 16 for discharge pressures up to 125 psi. Receivers up to 42 in. by 10 ft, inclusive, have screw openings; 48 in. by 12 ft and larger have flanged openings. Vertical receivers must have a base to raise them 6 in. above the floor to permit the inspector's hammer test. Always mount vertical receivers as shown in Fig. 85*A* to permit complete drainage. Figure 85*B* is incorrect because it permits moisture to deposit at the seam between the side wall and head.

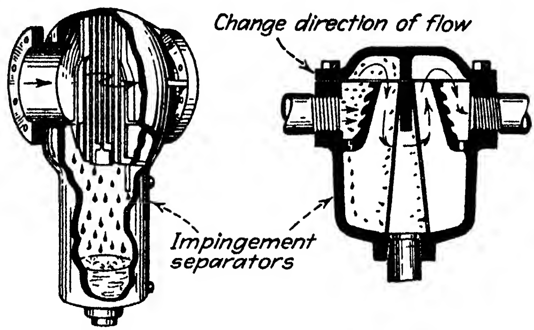


Fig. 87.—Separators contain impingement plates that stop moisture particles. (*Courtesy of Cochrane Corp. and W. H. Nicholson & Co.*)

Receivers are furnished complete with spring-loaded safety valves, pressure gage, handholes or manhole, drain valve and nipple, and base for vertical mounting. Figure 86 shows a vertical-mounted receiver complete with aftercooler and piping.

Strainers and Separators.—Aftercoolers remove excess moisture and oil only in proportion to their ability to lower the air temperature. Air leaving the aftercooler still remains saturated; any further cooling causes additional precipitation of moisture.

To remove the dirt and moisture not taken out by the intercoolers and aftercoolers, a separator must be inserted in the lines where the air is used. The first precaution is to design and install the pipe line properly. All air-piping systems should be uniformly sloped in the direction of flow insofar as possible, and low points should be properly drained. If the lines are not properly drained, water will accumulate, restrict the flow, and cause an unnecessary pressure loss. In cold weather it may freeze.

Devices that remove this water before it reaches tools or process include water legs, separators, and strainers. The amount of moisture that can be tolerated determines which of the three

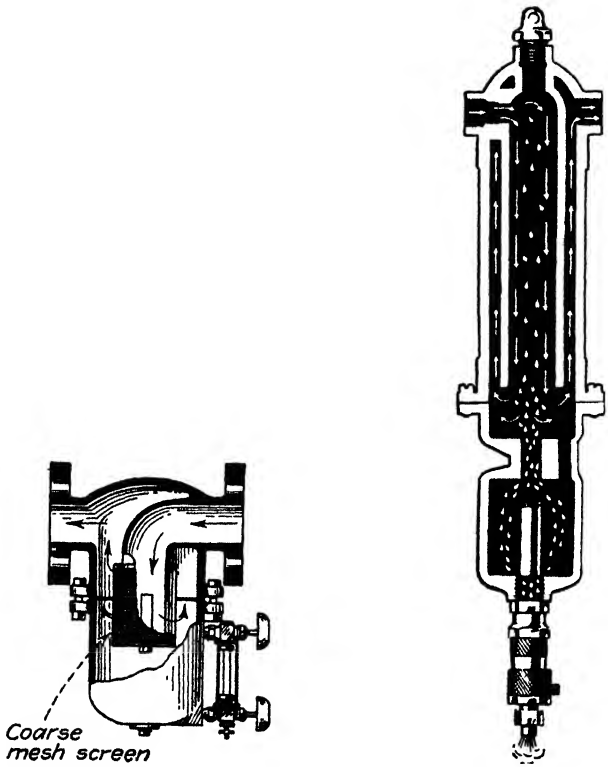


FIG. 88.—Changing the direction of flow throws out moisture. (*Courtesy of The Johnson Corp. and Leavitt Machine Co.*)

devices to use: water legs and separators remove heavy liquid particles; strainers, depending on internal design, take care of foreign matter and small particles of suspended moisture and oil.

A separator cannot lower the air temperature and therefore cannot cause it to release water vapor. Before a separator can function the vapor must condense. In some conditions of temperature and velocity a mist of vapor particles may be carried suspended in the air with little or no tendency to fall, until a

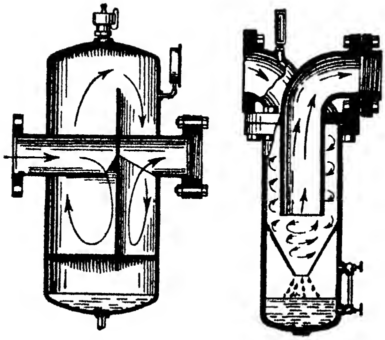


FIG 89 —Whirling action throws moisture from air stream (*Courtesy of Ingersoll-Rand Co and Worthington Pump and Machinery Corp*)

FIG 90 —Impellers rotate in opposite directions to remove moisture (*Courtesy of Logan Engineering Co*)

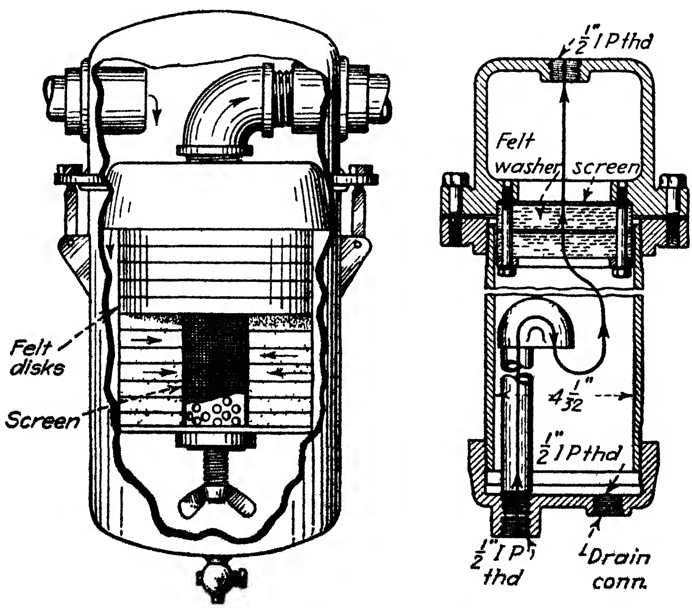


FIG. 91.—Felt disks strain the air and remove moisture. (*Courtesy of Staynew Filter Corp. and A. Schrader's Sons.*)

change in conditions causes them to unite as flowing liquid. Separators produce these changed conditions through centrifugal force or velocity of flow.

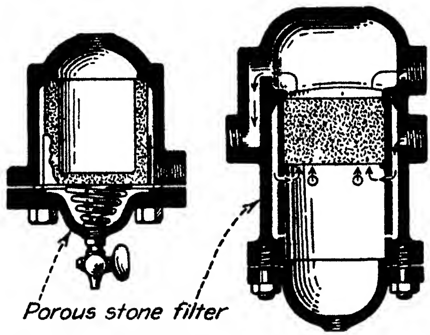


FIG. 92.—Porous-stone filters for removing moisture and oil. (*Courtesy of Fisher Governor Co.*)

Changing the direction of air flow (Figs. 87 and 88) while moisture globules continue straight ahead to strike on a baffle, or giving the air a whirling motion (Fig. 89) throws the moisture particles out of the air stream. Some designs use a converging nozzle to increase the air's velocity as it follows a whirling path.

The unit in Fig. 90 employs centrifugal force through four multiblade rotors. Air velocity rotates the impellers in opposite directions at high speed, as indicated by the arrows. Moisture, dirt, and oil are thrown to the outer wall where they fall to the chamber below. Filter units employ felt disks (Fig. 91) or porous stone cups or disks (Fig. 92).

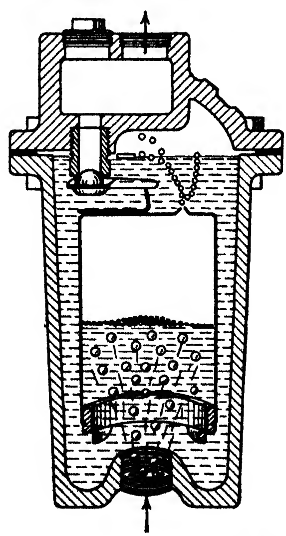


FIG. 93.—Inverted-bucket trap.

Moisture Traps.—Traps are nothing more than automatic valves placed at separators and drain pockets to discharge condensed moisture. The operating device consists of an inverted bucket (Fig. 93) or a float (Fig. 94). Discharge can be either continuous or intermittent.

Since considerable air may be lost through continuous discharge and since condensation is not as rapid as in steam lines, an intermittent-discharge trap serves best for compressed air. Always fill the inverted-bucket trap chamber with water before placing it in service. When the air contains only small amounts of moisture, the trap may require a periodic priming to prevent air blowing through.

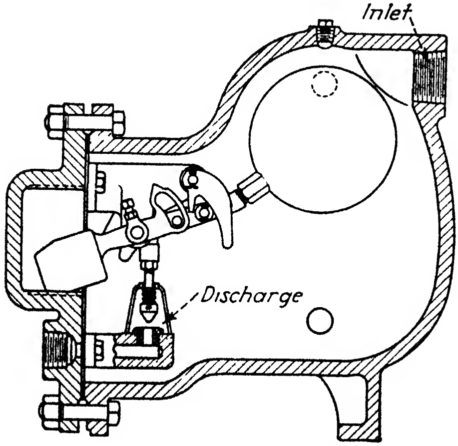


FIG. 94.—Intermittent float.

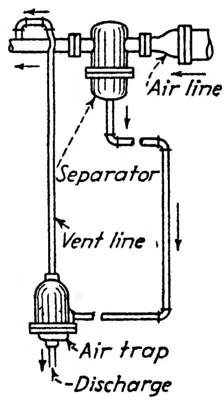


FIG. 95.—Vent line relieves air binding in trap.

When a considerable length of horizontal pipe, with insufficient pitch for good gravity drainage, must be installed between the trap and the equipment it drains, or when this pipe must run below trap-inlet level, water may be unable to reach the trap because of air pocketed in the trap body. If this occurs, run a vent line from the trap body to a high point in the main air line (Fig. 95).

The inverted-bucket trap (Fig. 93) should be equipped with a check valve on the inlet to prevent loss of prime. Locating the discharge valve at the top of the trap body keeps it clear of dirt that settles in the bottom. Air loss through the vent hole is small—5 to 7 cu ft of free air per hr.

The snap-action float trap (Fig. 96) has an intermittent instantaneous action by a float connection to a short valve lever through a flat strip of stainless spring steel. In the closed position this spring is bowed downward. As water enters, the float rises and stores up energy in the spring. Just before the float reaches the upper limit of travel, the spring bends past dead center and the stored energy snaps the valve wide open. As the water level drops the cycle is reversed and the valve snaps shut.

Water entering the trap (Fig. 94) raises the float to its highest point of travel, where it releases the weight latch. The weight then falls and raises the valve from its seat instantly. The valve

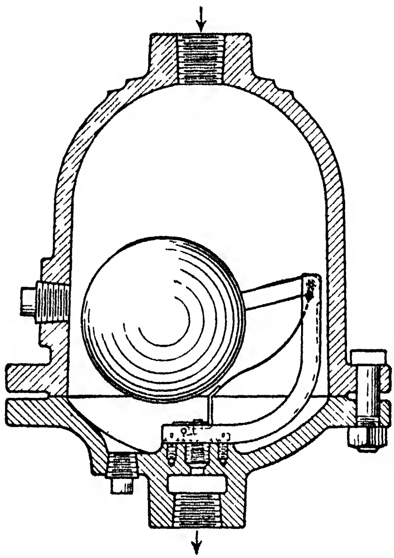


Fig. 96.—Snap-action float.

remains open until water discharge lowers the float to its lowest position. The latch then engages the weight, which has been lifted by the falling float, and the link latch disengages, permitting the discharge valve to close instantly.

Protective Devices.—These devices, discussed more fully in Chap. VI, Installing the Compressor, consist of (1) safety and relief valves, (2) oil filters, (3) overspeed shutdown, (4) oil-pressure shutdown, (5) automatic water valves, (6) high-pressure shutdown, (7) high-temperature shutdown, (8) main-bearing temperature alarm, and (9) recording discharge-pressure gage.

CHAPTER IV

COMPRESSOR LOAD CONTROL

Variations of Load Control.—Since the demand for compressed air usually varies widely, the compressor must be provided with

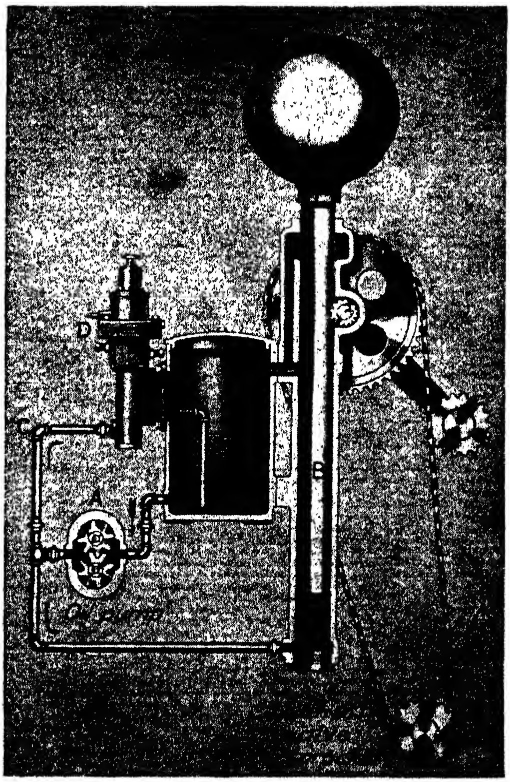


FIG. 97.—Pump forces oil under plunger to raise rack and move cutoff valve. (*Courtesy of Ingersoll-Rand Co.*)

means for changing the volume of air compressed in order to maintain a relatively constant discharge pressure. This is accomplished by (1) operating at varying speeds to suit the demand (steam and internal-combustion-engine or d-c-motor drive); (2) operating at constant speed, with automatic or hand-operated unloaders built into the compressors; (3) variations or combinations of these general forms of control.

Steam- and internal-combustion-engine-driven compressors are usually provided with combination speed and pressure governors which vary the capacity by changing the compressor speed. Such governors are available in two forms:

1. Throttling type, which adjusts the compressor speed by varying the steam or fuel admitted to power cylinders, throttling it as the discharge air pressure rises.
2 Automatic-cutoff type, which varies compressor speed by varying the point of cutoff in the steam valve.

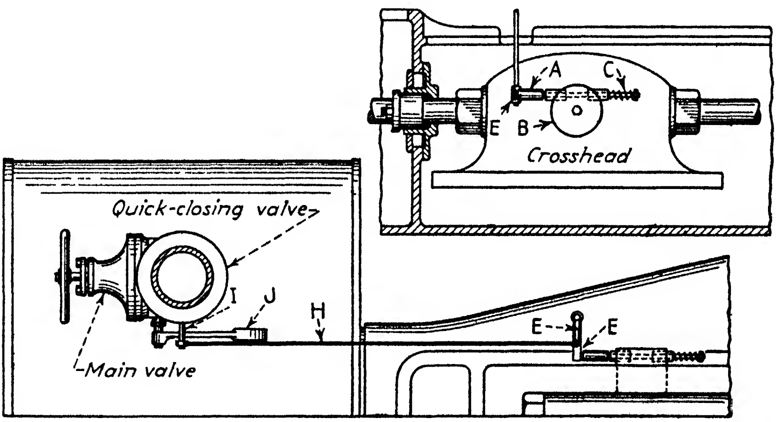


FIG. 98.—Crosshead overspeed trips weight-operated butterfly valve. (*Courtesy of Ingersoll-Rand Co.*)

Motor-driven and other constant-speed compressors are usually provided with either of these general types of capacity control:

1. Constant-speed unloaders, which decrease the compressor capacity automatically in one or more steps while the compressor continues to operate at normal speed.
2. Automatic start-and-stop control, in which an automatic starter and pressure switch start and stop the motor within definite pressure limits. This control is recommended only when there is no demand for air for fairly long periods of time, because it does not regulate the pressure closely.
3. Dual control, a combination of constant-speed and automatic-start-and-stop control.

Constant-speed load regulation can take either of these general forms:

1. Intake control, unloading the cylinder by closing the intake
2. Free-air control, holding the inlet valves off their seats, thus permitting air to pass into and out of the cylinder without being compressed.
3. Clearance control, varying compressor output by changing the amount of clearance volume in the cylinder.

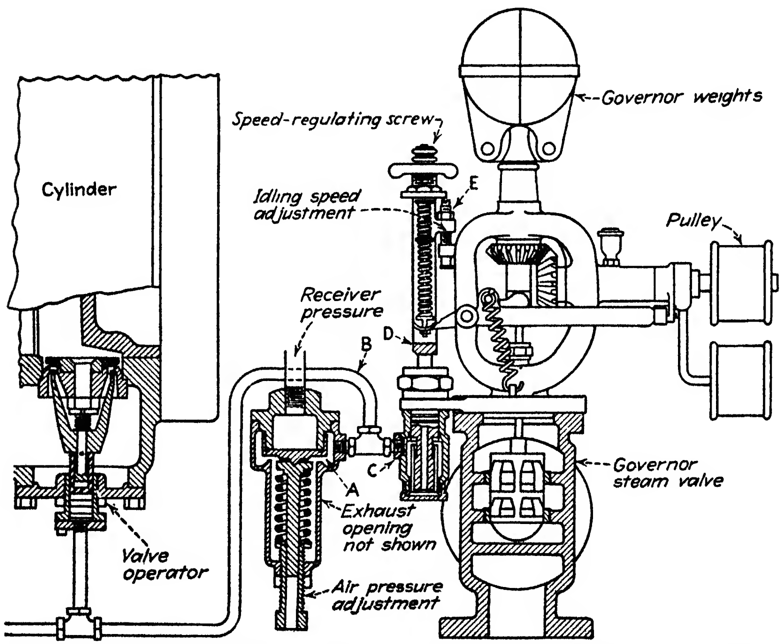


FIG. 99.—Flyball speed governor with air-operated pilot attachment for unloading. (*Courtesy of Chicago Pneumatic Tool Co.*)

Regardless of how a constant-speed compressor loads or unloads while running, its control always incorporates some means for unloading during the start-and-stop period.

Variable-speed Control.—The governor (Fig. 97) rotates the cutoff steam-valve stem, which changes the cutoff and thus varies the compressor speed. This is accomplished by raising or lowering the rack and weight *B* by a varying oil pressure under the plunger. The amount of this pressure variation is the resultant of two separate controls acting simultaneously.

Oil supplied by pump *A*, chain-driven from the main shaft, produces a pressure acting under the governor plunger. Any variation in steam pressure tending to change the compressor speed is instantly reflected in an increase or decrease in this oil pressure, causing a movement of the weight and rack which rotates the cutoff-valve pinion and changes the cutoff to restore the speed

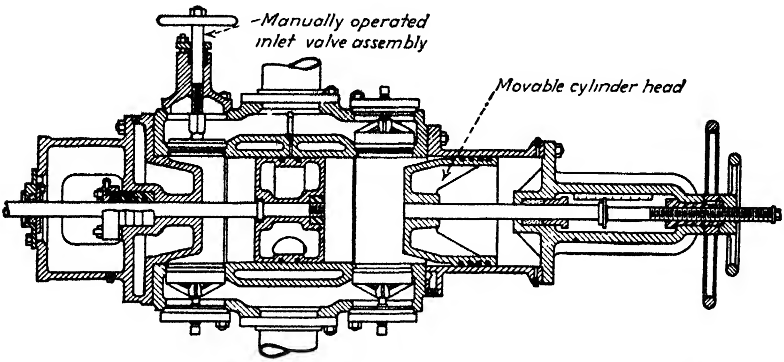


FIG 100 —Hand-operated inlet valve and adjustable cylinder head.

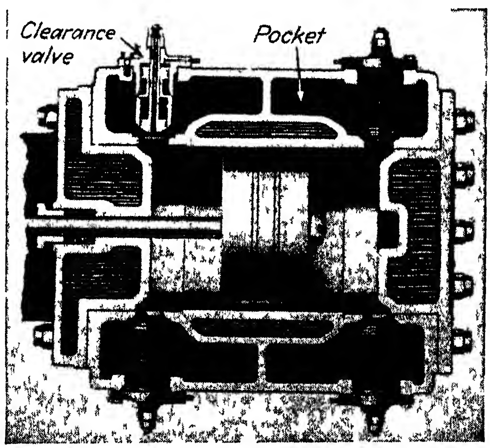


FIG. 101.—Automatic clearance-pocket valves.

Variations in air demand affect the oil pressure by means of a diaphragm-operated valve *D* which by-passes a portion of the oil flow. The change in oil pressure, effected by these two distinct methods of regulation acting in unison, controls the speed according to the load demand and compensates for varying steam conditions.

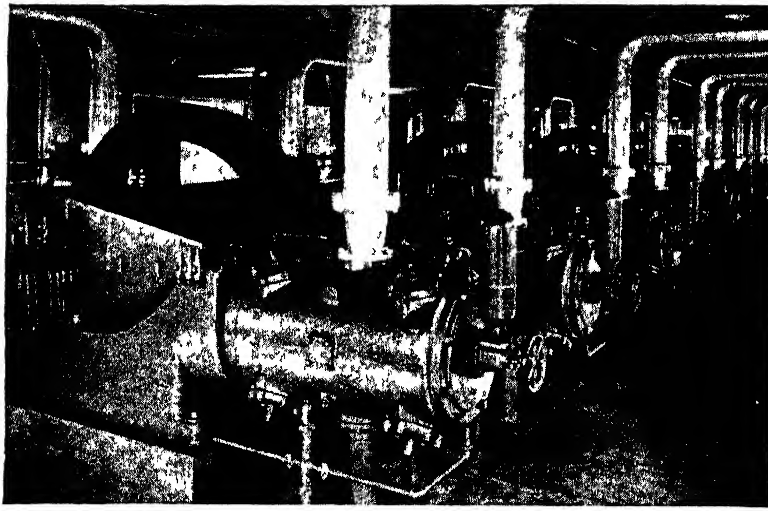


FIG. 102.—Hand-operated external clearance pockets. (*Courtesy of Cooper-Bessemer Corp*)

FIG. 103.—Inlet by-pass valve fitted with hand-closing screw. (*Courtesy of Bury Compressor Co.*)

A positive weight-actuated safety stop, similar to Fig. 66*A*, located ahead of the throttle valve and operated by an inertia slide on the crosshead (Fig. 98), is used with automatic-cutoff governors to shut off the steam supply should the compressor exceed a safe speed. The trip rod *A* passes through and is supported by two bosses on the crosshead-pin cap *B*. The trip-rod shoulder is held against the left-hand boss by tension of the spring *C*. Thus the trip rod can move, in relation to the crosshead, in one direction only when some force overcomes the spring tension.

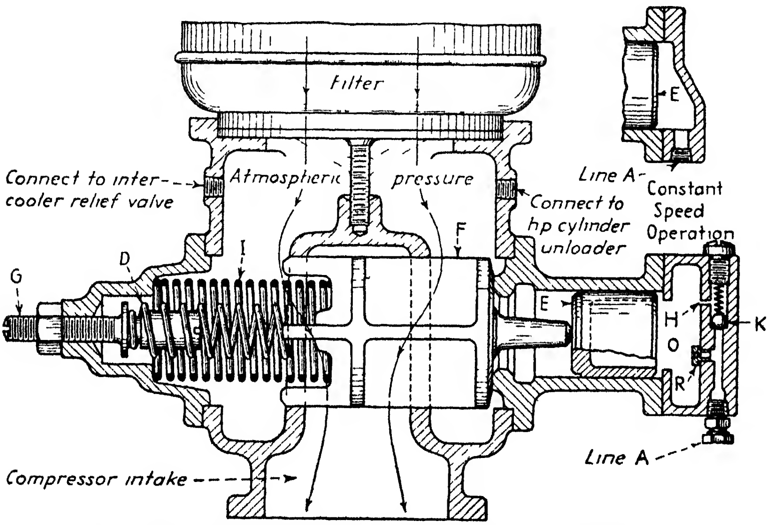


FIG. 104.—Unloader operates to close compressor intake.

When the compressor reaches the maximum speed for which the spring *C* is set, the trip rod, because of its inertia, moves and strikes the lower part of the arm *E*. This motion is transmitted through the shaft to the upper arm *F* and thence through latch rod *H* to the latch *I*. Movement of latch *I* allows the outside lever *J* to drop, thereby closing the valve and shutting off the steam supply.

The cross-compound-engine-driven compressor (Fig. 2) uses three separate governors: a mechanical one for overspeed protection; another controlled by air pressure for speed regulation, both controlling the high-pressure-cylinder throttle valve; and a third governor, also actuated by air pressure, which controls live-steam admission to the low-pressure cylinder. The latter governor prevents the unit from stopping on dead center when extremely low system requirements result in low compressor speeds.

Connections for throttling control of a steam-driven compressor are shown in Fig. 99. The unloader valve A is held against its seat on the unloader cap by tension of the spring.

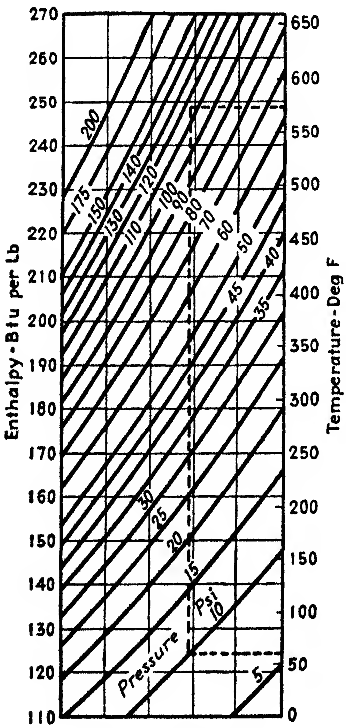


FIG. 105.—Temperatures of air when compressed adiabatically. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

When receiver pressure reaches the unloading point, the valve is forced away from its seat, allowing air to flow into the pipe connection on the side of the unloader. Here the air divides, part passing through pipe B to the compressor inlet valves, and the other part passing into cylinder C to raise yoke D. This forces the governor-valve stem downward, closing the valve and throttling the steam supply.

The extent of yoke travel and the amount which the governor valve closes during unloaded periods can be regulated by the position of stop screw E attached to the yoke and traveling up and down through a lug on the governor frame.

When the air pressure falls, valve A is forced to its seat by the spring, allowing air from pipe B and cylinder C to escape to the atmosphere through the exhaust opening (not shown) in the unloader body. As air escapes from cylinder C, springs draw the yoke and piston downward, opening the governor valve and increasing the compressor speed. Cylinder C should be kept filled with steam-cylinder oil to a height just below the air-entrance pipe so that it will act as a dashpot to dampen governor action.

Constant-speed Control.—When compressors operate at constant speed, the output must be controlled by starting and stop-

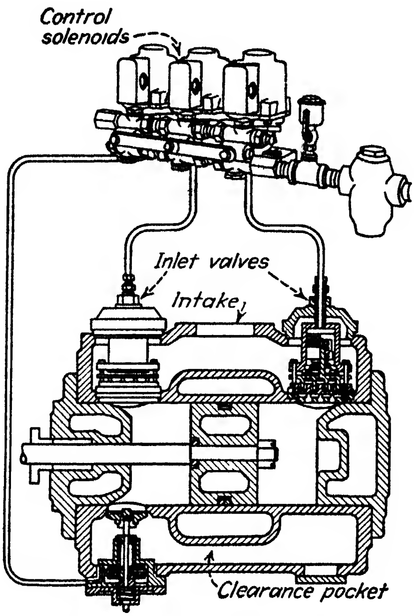


Fig. 106.—Solenoid-controlled inlet and clearance-pocket valves.

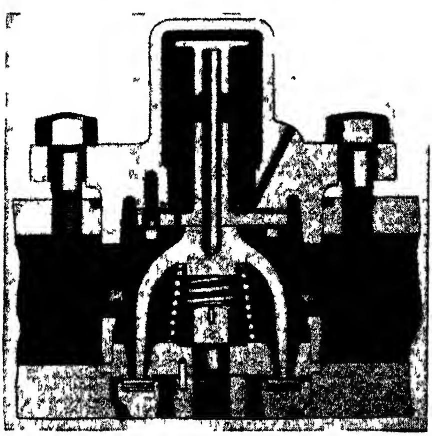


Fig. 107.—Bellows-operated valve lifter.

ping, or by automatically regulating the amount of free air entering the cylinder.

Most compressors supply a changing load and, to hold a constant delivery pressure, the machine must be loaded in proportion to air demand. Most loading-control systems act to operate the

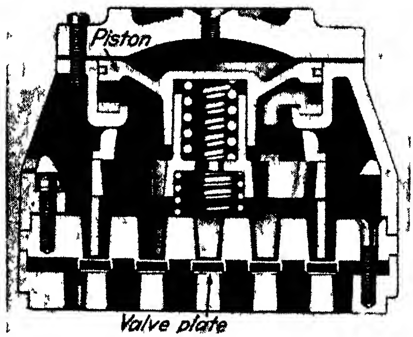


FIG 108 —Plunger-operated valve lifter

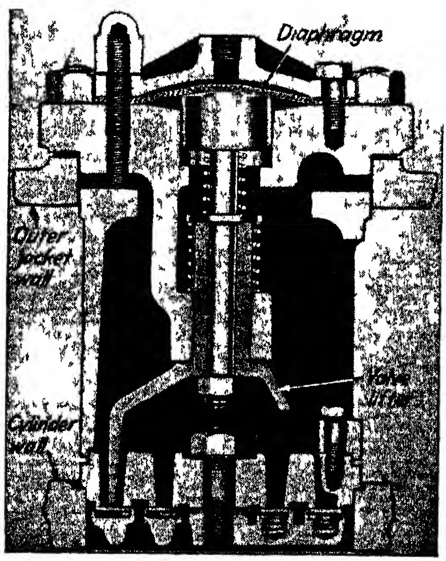


FIG. 109 —Diaphragm-operated valve lifter.

compressor at 100, 75, 50, 25, and 0 per cent of full load, or at one or more combinations of these percentages. To regulate through all these steps naturally requires more control equipment than for one- or two-step unloading. The actual unloading is accomplished by closing the intake, holding open the inlet valve or opening an auxiliary inlet valve, and increasing the cylinder clearance

Cylinder clearance is increased by installing an adjustable cylinder-head plunger (Fig 100), or by forming pockets (Fig. 101), in the cylinder walls when making the casting A suitable valve cuts off clearance pockets from communication with the cylinder, and the adjustable cylinder head initially occupies a position of least clearance Opening the valve to the clearance

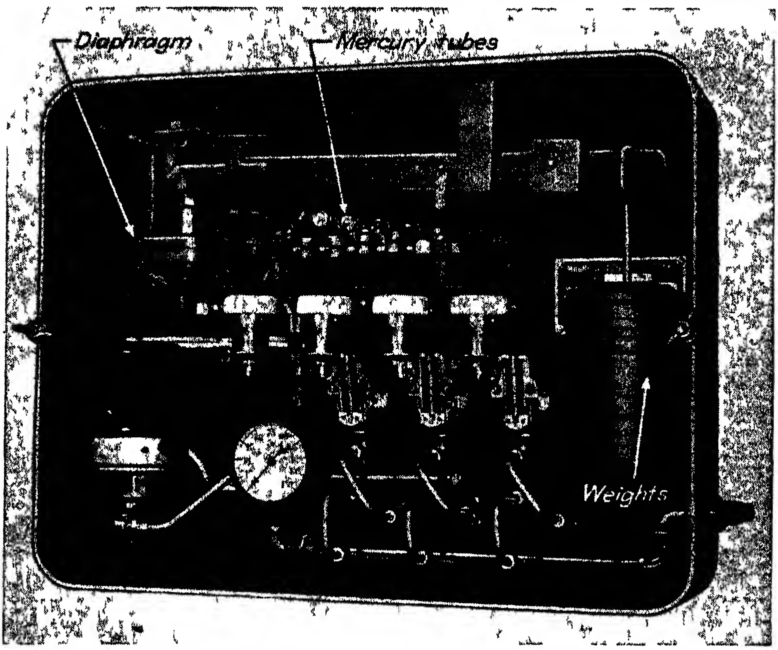


FIG 110—Weight-loaded diaphragm operates mercoid unloading switches (*Courtesy of Gardner-Denver Company*)

pocket or moving the adjustable head away from the cylinder bore increases the cylinder volume to such an extent that full output can no longer be discharged A clearance pocket gives a marked reduction in output (note the battery of manually operated clearance-pocket compressors in Fig 102); a movable cylinder head gives almost micrometer adjustment of discharge.

Inlet-valve unloading gives the greatest change because the output falls practically to zero when the valve or valves are held open. A special inlet by-pass valve (Fig. 103), reduces output as well as holding the inlet valve open.

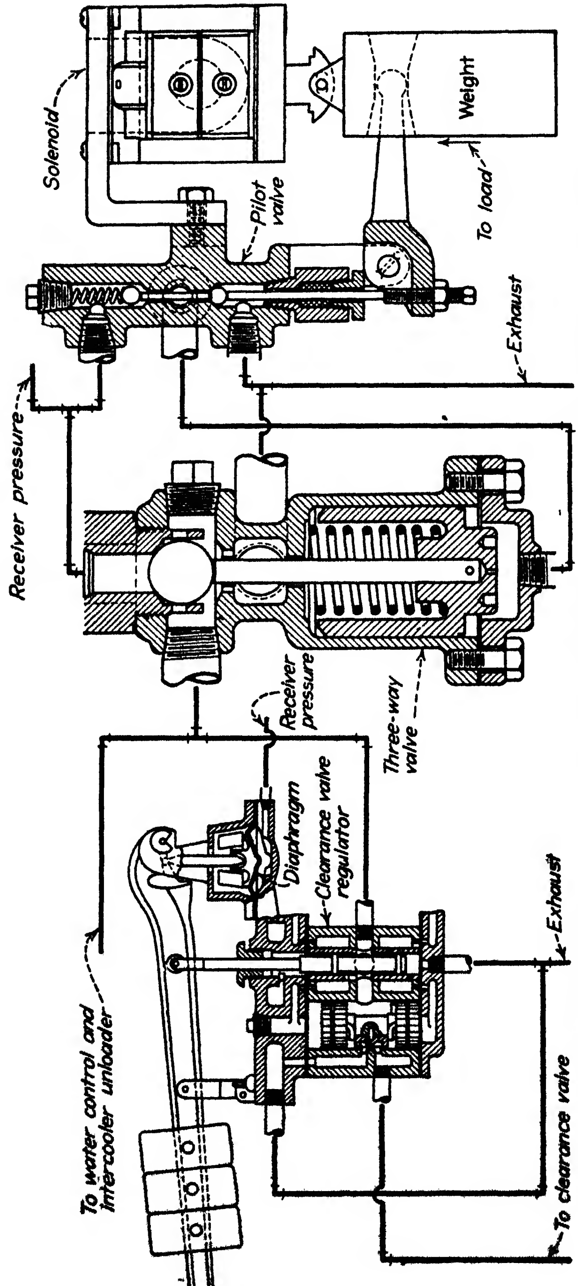


FIG. 111.—Solenoid controlled from motor starter operates unloader pilot valve for starting and stopping.

Total closure of the inlet cuts off all air flow (Fig. 104), forming a vacuum inside the cylinder and reducing the power input to that needed for sweeping the piston back and forth against the friction load. Partial closing of the intake has the same effect as moving the compressor to a high altitude, and manufacturers

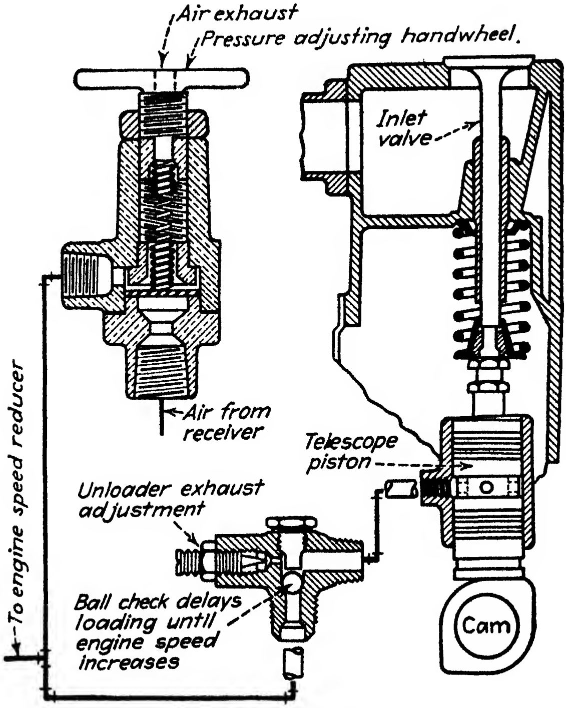


FIG. 112.—Pilot valve controls air to compressor inlet-valve unloader and engine-speed reducer.

caution against it because throttling the incoming air causes low suction pressure, increases the compression ratio, and results in high internal temperatures. Assume the intake throttled so that the cylinder is filled with 70 F air at 10 psi abs at the start of compression. Figure 105 gives the final temperature (adiabatic) as 574 F if the air is compressed to 115 psi abs.

Adaptation of Methods.—Methods of load control used by different manufacturers vary; frequently one builder uses all possible methods, adapting a different one for each compressor design. A combination of clearance pocket and open inlet-valve control (Fig. 106) gives a flexible loading on a compressor of some particular size. In another machine, manually raising the inlet-valve assembly adjusts the crank end, and the movable cylinder head changes front-end clearance.

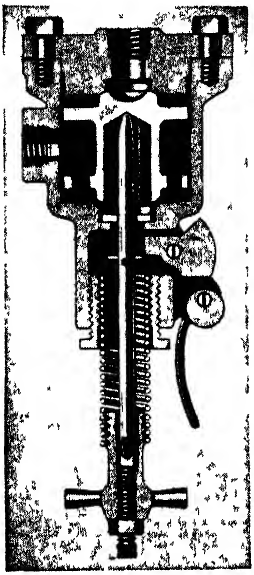


FIG. 113.—Pilot valve with hand trip.

Although a few machines incorporate manually operated load control, automatic pneumatically operated unloading systems predominate. Actuating operators, mechanisms that force the valves open or closed, receive actuating force from an air-operated bellows (Fig. 107), piston (Fig. 108), or diaphragm (Fig. 109). Either electrically or air-operated valves control the air supply to compressor-valve operators. Over-all governing takes place in a pressure switch or air relay. Electrical governing usually requires a separate switch (Fig. 110) to energize each individual solenoid-operated valve for every loading step.

The master control (Fig. 110) consists essentially of a weighted lever balanced by receiver-air pressure under a diaphragm. This lever, through a link, oscillates a shaft

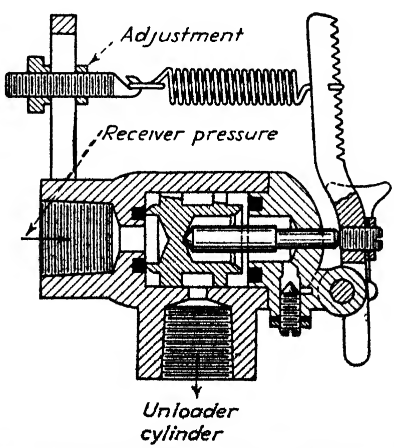


FIG. 114.—Spring and lever pilot valve. (*Courtesy of R. Conrader Company.*)

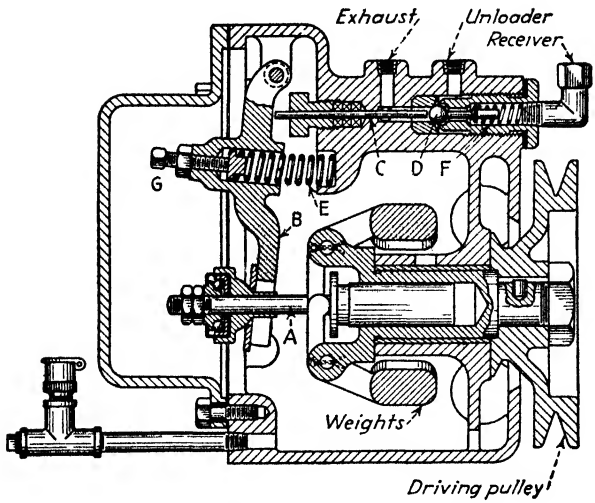


FIG. 115.—Belt-driven centrifugal unloader.

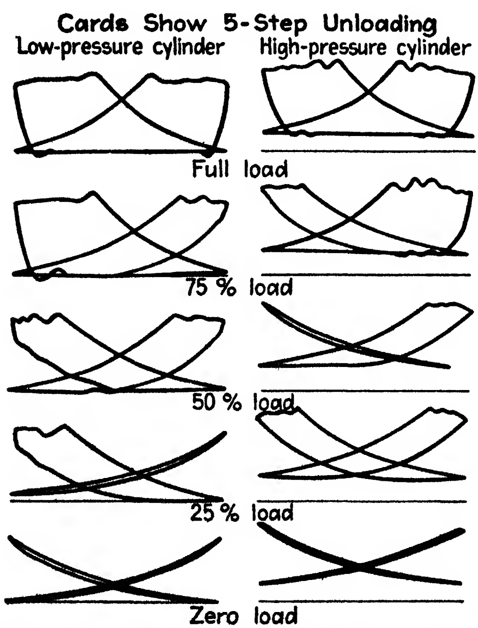


FIG. 116.—Indicator cards illustrate five-step unloading. (*Courtesy of Worthington Pump and Machinery Corp.*)

that carries mercury tubes to make and break electrical circuits for the solenoids.

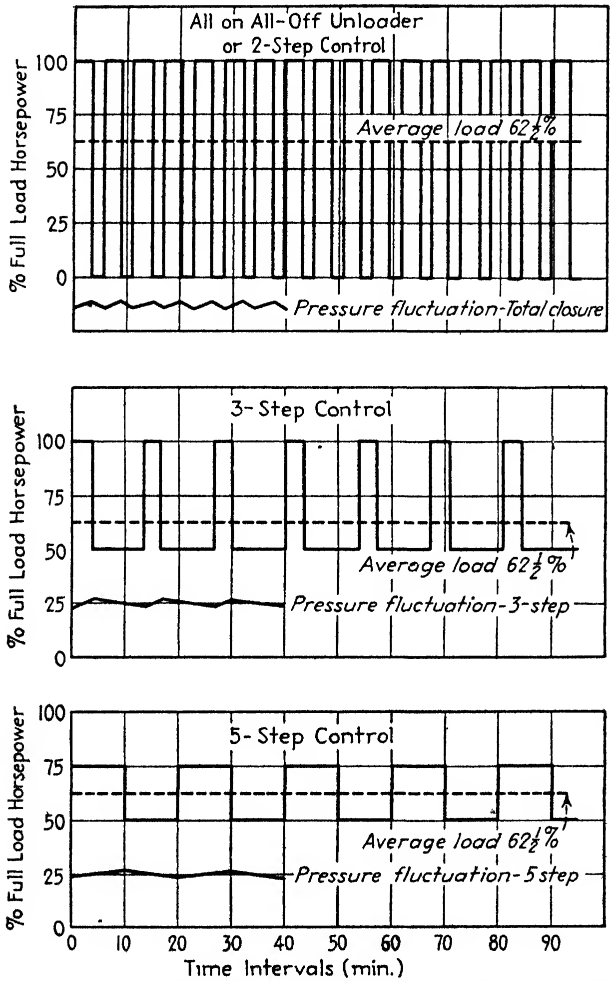


FIG. 117.—Typical power curves based on 62½ per cent compressor load. (*Courtesy of Worthington Pump and Machinery Corp.*)

Each solenoid operates a three-way valve that admits air to operate one set of suction unloading valves (for three-step control) or one set of clearance-pocket valves (for five-step control). If the lever is in balance at a certain air pressure, a slight increase in pressure causes the mercury tube to break the solenoid circuit, which causes a reduction in the compressor output. Conversely, a slight decrease in pressure will cause an increase in compressor output. This controller incorporates a recycling switch to provide a fairly even distribution of operation and wear on all clearance-pocket valves.

Diaphragm or bellows governors move a pilot piston (Fig. 111) to uncover progressive ports supplying air to each step-control

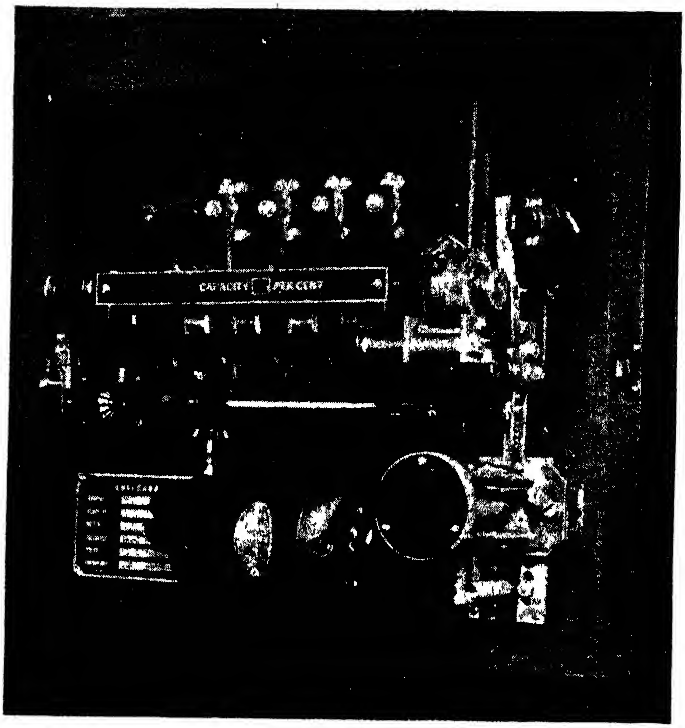


FIG. 118.—Pressure diaphragm engages ratchets that rotate camshaft and close the contacts.

valve operator. The main governor may be a contact-making Bourdon tube, a bellows-operated pressure switch, a diaphragm and weighted arm, a spring-opposed pilot valve (Fig. 112), or an unbalanced spring-loaded plunger (Figs. 113 and 114).

The pilot valve (Fig. 113) used to operate an inlet-valve unloader is simple in construction and easily adjusted. The piston is a loose fit in the body and has but a slight movement as it flips from one seat to the other. Pressure adjustment is made by screwing up on a setscrew at the extreme lower end of the pilot valve for higher pressures. The range between unload-

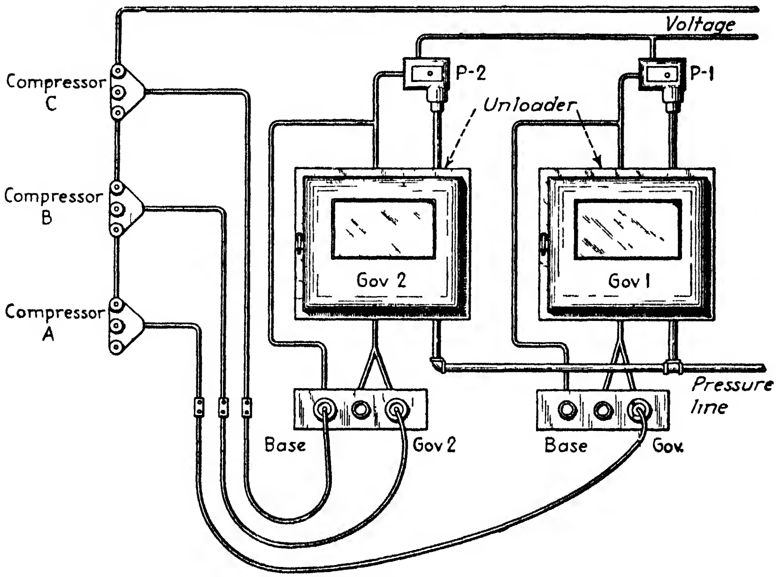


FIG. 119.—Two controllers, properly wired, serve to unload three compressors.

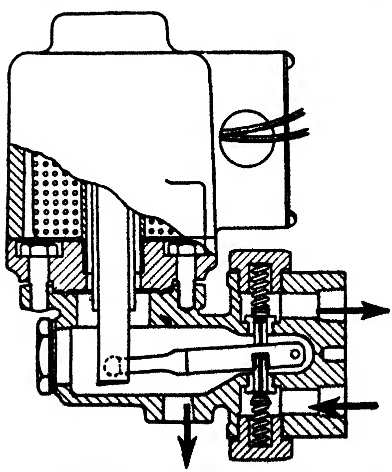


FIG. 120.—Cross section of solenoid valve in Fig. 106.

ing and loading pressures is regulated by varying the active number of working coils in the spring. Turning up on the adjuster which engages this spring gives close range corresponding to an unloaded period of 3 to 4 psi drop, and screwing out the adjuster gives a range of 10 psi or more. The small lever provides manual operation so that the compressor can be unloaded for starting when air pressure is available.

Manual operation can be eliminated by fitting the pilot with a diaphragm connected to a stem that throws the pilot piston to the unloaded position. Air from a solenoid-operated valve,

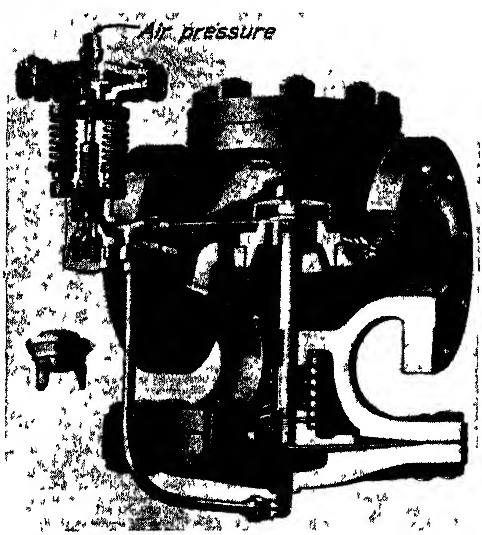


Fig. 121.—Cooling-water-control valve. Air pressure depresses small diaphragm which admits water pressure to large diaphragm. (*Courtesy of Ingersoll-Rand Co.*)

energized at the time of starting the motor, supplies the air to these diaphragms.

Practical Applications.—To make an unloading system practical, loading must be prevented until the machine approaches or reaches normal speed. Load release upon interruption of driving power permits the unit to coast smoothly to rest. Means to accomplish this range from a centrifugal-operated pilot valve (Fig. 115) to numerous electrical devices. A definite-time delay relay, a current relay (for induction motors), or a field relay, connected in the closed-field circuit of synchronous motors during the starting period, controls a solenoid-operated pilot (Fig. 111) supplying air to the unloading system. Compressors operating fully loaded on a start-and-stop basis use the primary controllers

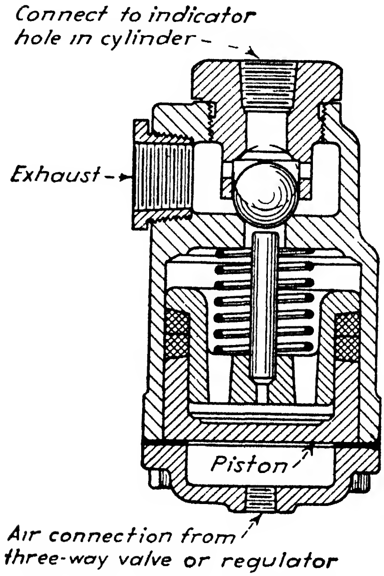


FIG. 122.—Cylinder relief valve.

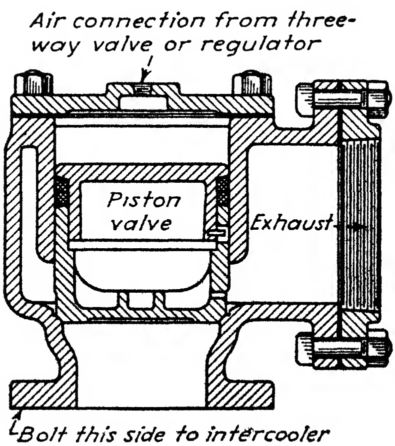


FIG. 123.—Intercooler unloader.

just mentioned to prevent loading until full speed is reached and to assure unloading at shutdown.

The flyball regulator (Fig. 115) is driven from a pulley on the compressor crankshaft. The rotating spindle A carries the pulley and flyballs. The bearing at the outer end of the spindle shaft transmits flyball action to lever B. By means of this lever and pin C, the flyballs control the single ball valve D.

When the compressor is idle, the flyballs are held in by the spring E and the ball valve is pushed onto its seat by the spring F. This closes the exhaust connection and puts full receiver pressure on the unloader connection.

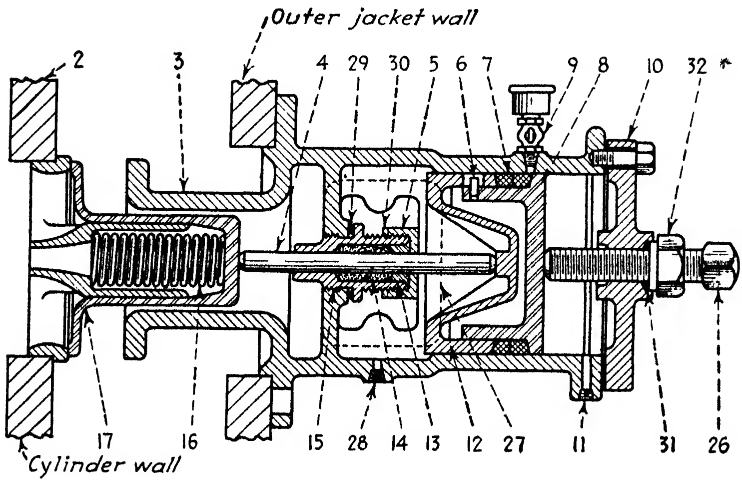


FIG. 124.—Clearance-pocket valve.

When the compressor comes up to speed, the flyballs are thrown outward and push the ball valve off its seat, thereby shutting off receiver pressure and exhausting pressure from the unloaders. The regulator operates at 500 rpm and the pulleys must be arranged accordingly to fit different compressor speeds. If the compressor loads before it comes up to speed, turn the adjusting screw G clockwise; if it fails to load at full speed, turn the adjusting screw counterclockwise.

Many plants use synchronous-motor-driven compressors to gain the advantage of power-factor correction by overexciting the motor. The machine then operates continuously and the compressor load varies according to system requirements. While continuous running does require a certain amount of power to overcome running friction, no great amount of power loss appears

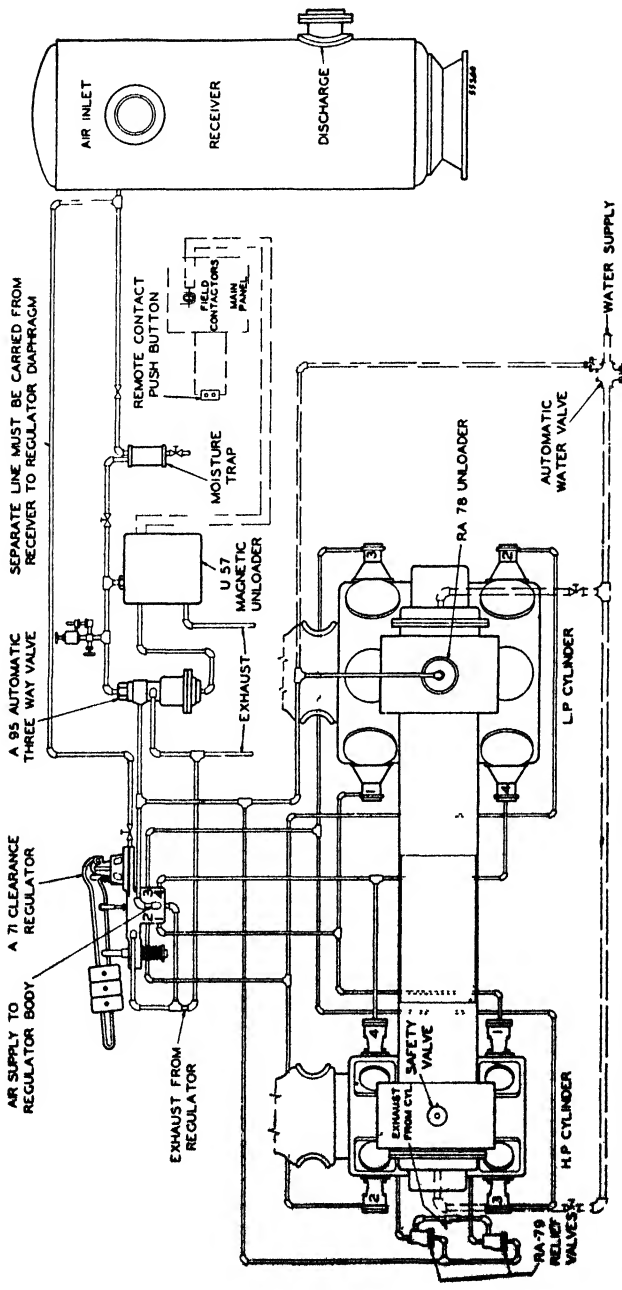


FIG. 125.—Complete unloading system for cylinders and intercooler for automatic start-and-stop operation. (*Courtesy of Ingersoll-Rand Co.*)

in the air cylinder when the machine operates unloaded. Loss of efficiency occurring during partial and full unloaded operation varies with the size of the machine because of the friction of moving parts. The series of indicator diagrams taken with a five-step clearance-pocket unloader (Fig. 116) shows how the air load falls off when the machine unloads and indicated the small air-power loss at zero load.

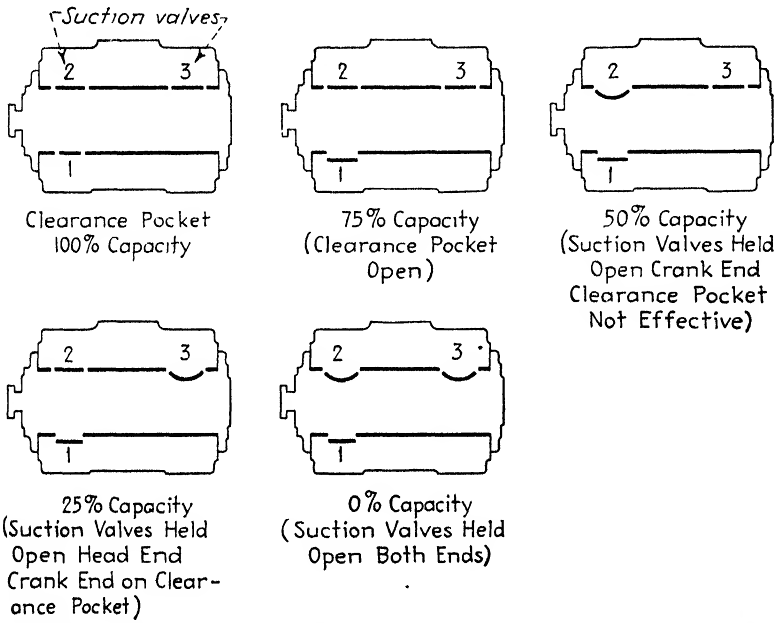


FIG. 126.—Five-step unloading that utilizes inlet valves and one clearance pocket. (*Courtesy of Worthington Pump and Machinery Corp.*)

Choice of an unloading system depends entirely on the job to be done. Operating results, as far as power economy is concerned, differ so little in the three basic methods of control that they are scarcely worthy of consideration.

Motor-driven Compressors.—The mechanism usually provided for volume control of motor-driven compressors reduces the compressor capacity by steps or increments; thus, five-step control means that, as the pressure rises, the capacity of the compressor is reduced in steps of 25 per cent. With widely varying air requirements, the compressor might operate in five steps (Fig. 116), or with three-step control it might operate at 100 per cent, 50 per cent, or zero capacity. A two-step control which would operate at either 100 per cent or zero capacity, is usually called all-on or all-off control, there being no intermediate steps.

The ideal control, of course, would compress with each stroke only that amount of air needed. This gives a mean average power requirement for the prime mover driving the compressor.

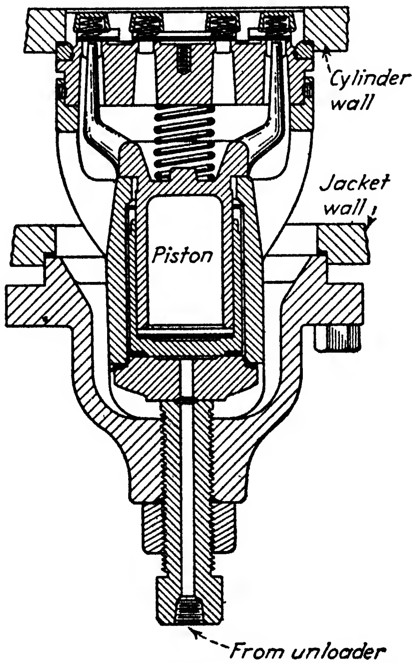


FIG. 127.—Plunger-operated valve lifter.

Obviously, widely fluctuating power requirements are to be avoided whenever possible, whether power is being purchased or being made on the premises.

In actual operation the compressor control does not run the entire range from 100 per cent to zero capacity; instead the machine may, and does, float for some considerable period of time back and forth between two capacity steps. Only the most violently fluctuating air demand would cause a step-control compressor to vary its power requirements over extremely wide limits. If, for example, the compressor is 40 per cent larger than air requirements, the capacity control will float it between the

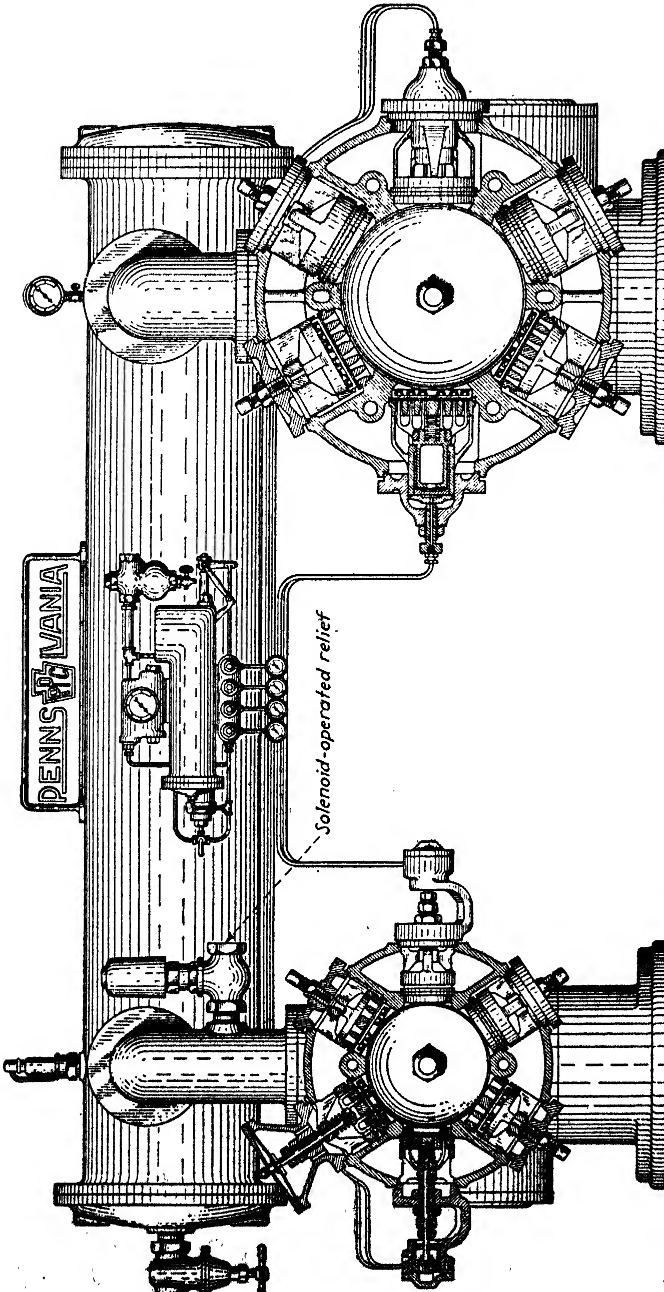


FIG. 128.—Five-step control using four clearance pockets and diaphragm-operated high-pressure inlet valve.

50 and 75 per cent steps, thus providing an average-load curve only as heavy and as uniform as the demand dictates. Since the machine operates at a capacity step somewhere near the actual requirements, it takes a relatively long time interval for the load to change to another step. It is obvious that the smaller the number of capacity steps, the greater the amount of capacity change between steps and the shorter will be the time between load changes. In other words, if the machine with five-step

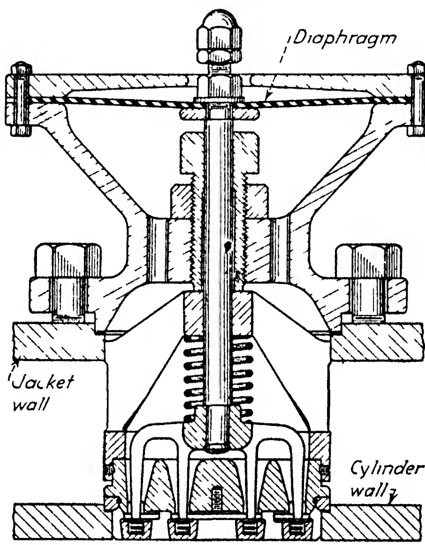


FIG. 129.—Cross section of high-pressure inlet valve in Fig 128

control operates 10 min between load changes (Fig. 117), the machine with three-step control would operate approximately 5 min between load changes, whereas the machine with the all-on-all-off control would be changing load every minute or two with the same variation in discharge pressure.

Five-step Clearance Control.—The main controllers (Figs. 110 and 118) operate the unloading valves electrically. In Fig. 118 a ratchet wheel and pawls rotate the cam-driving shaft in either direction. A small instrument motor runs continuously to furnish driving power. Pressure changes on a diaphragm operate a linkage to engage either one or the other ratchet pawl.

With the instrument at 100 per cent load, all contacts are closed and all solenoids energized. As the receiver pressure increases the instrument moves to the 75 per cent position and opens one of its contacts to reduce compressor load.

Controls of this kind interlock the solenoid-valve circuits with the motor starter so that, regardless of the governor-instrument position, all solenoids are deenergized during starting and stopping. With a synchronous-motor starter the interlocking connects with the automatic field control so that the compressor loads only after the motor is synchronized.

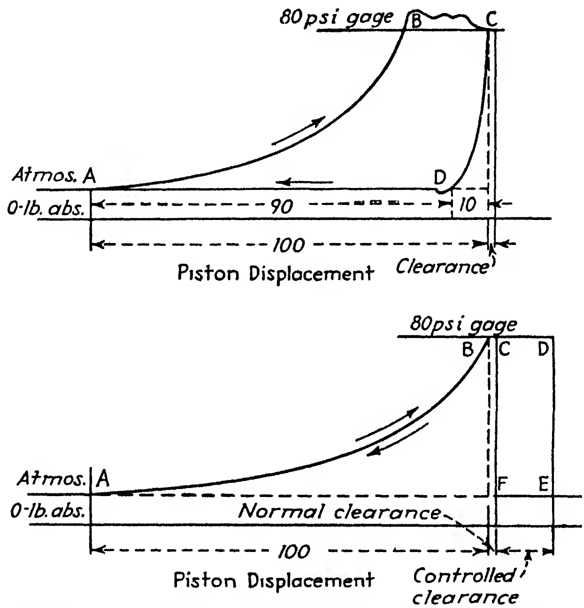


FIG. 130.—Effect of clearance pockets on compressor indicator diagrams. (*Courtesy of Power.*)

Repositioning of the pressure element takes place after each governor operation, and each successive capacity-reduction step occurs at precisely the same pressure.

These instruments are adaptable to multiple compressor control (Fig. 119), using a plugging system arranged to control several compressors.

With this hookup, solenoid valves (Fig. 120) mounted on the compressor are not wired permanently to the governor contacts but are connected to polarized plugs that fit into sockets wired to the governor instrument. Two compressors can then be connected to the same governor and they will load and unload in parallel as though they were one machine of larger capacity.

Figure 111 is a five-step unloader using a solenoid-operated pilot valve energized from the motor-starting circuit. After the compressor reaches normal speed, the pilot valve exhausts air from beneath the three-way-valve piston, allowing it to fall away from the ball valve. Receiver-air pressure then flows to the clearance-valve controller; clearance-pocket valves then close or open, according to the position of the weight-loaded diaphragm arm. Other devices included in this arrangement are a cooling-water supply valve (Fig. 121), a high-pressure-cylinder relief valve (Fig. 122), an intercooler unloader (Fig. 123), and clearance

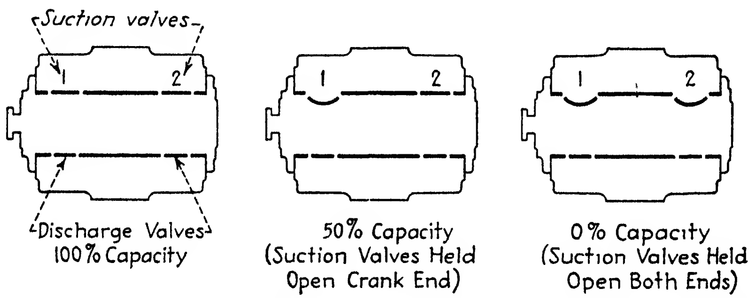


FIG. 131.—Three-step unloading by controlling inlet-valve operation. (*Courtesy of Worthington Pump and Machinery Corp.*)

valves (Fig. 21 or 124). A complete piping layout with all control devices is shown in Fig. 125.

Five-step control can be secured by using either four clearance pockets or one clearance pocket combined with inlet-valve lifters on each cylinder. Figure 126 shows five-step unloading with the latter method.

When air pressure within the cylinder tends to close the clearance-pocket valve (Fig. 127), no air pressure is needed in the control system to load the compressor. The opposite is true (Fig. 124) as cylinder pressure tends to open the valve. To overcome this condition the valve cap carries a setscrew for holding the valve closed during initial starting or when receiver pressure is zero.

Since the clearance-pocket valves (Fig. 127) will not unload the compressor until air pressure is built up in the receiver, unloading must be taken care of by other means. The system (Fig. 128) equips the intercooler with a solenoid-operated relief controlled from the motor starter. In addition, one high-pressure-cylinder inlet valve (Fig. 129) is equipped with a diaphragm operated by receiver pressure. After the compressor is up to speed, the intercooler solenoid-relief valve closes and pressure builds up in the intercooler and under the high-pressure inlet-valve dia-

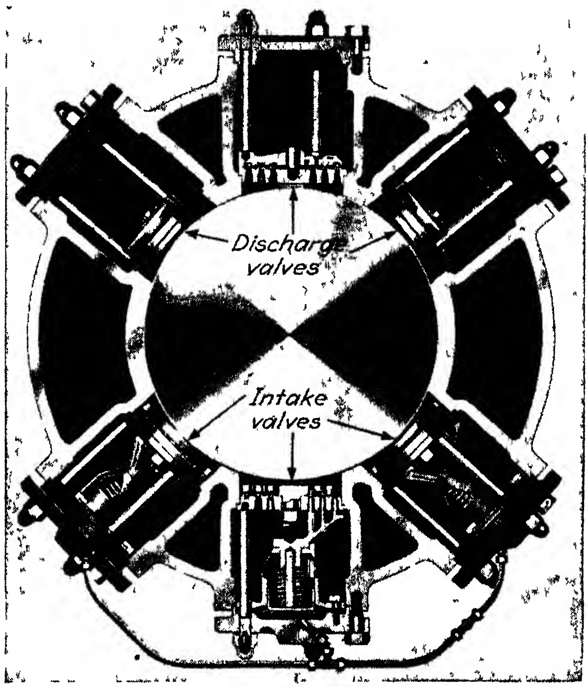


FIG. 132.—Inlet-valve unloading with diaphragm-operated valve lifters. (*Courtesy of Chicago Pneumatic Tool Co.*)

phragm, thus raising lifting fingers away from the valve and allowing it to close.

An idea of how clearance control affects the compression curve may be secured from Fig. 130. The top curve shows a typical diagram. Suction does not depart from the atmospheric line, indicating no restriction to the flow of air into the cylinder.

The lower card is from a compressor unloaded by means of clearance-pocket control. In order to begin the compression stroke at atmospheric pressure, the normal clearance is increased by the volume represented by the rectangle *CDEF*. The compression and expansion line would be the single line shown instead of double lines (Fig. 116), were it not for heat transfer to the jacket water.

Three-step Regulation.—This method operates the compressor at 100 per cent, 50 per cent, and zero load. The most common method (Fig. 131), holds the inlet valves (Fig. 132) off their seats. With all inlet valves functioning, the machine operates at full load; holding inlet valves on one end open reduces the load to 50 per cent, and holding the valves on both ends open reduces the load to zero.

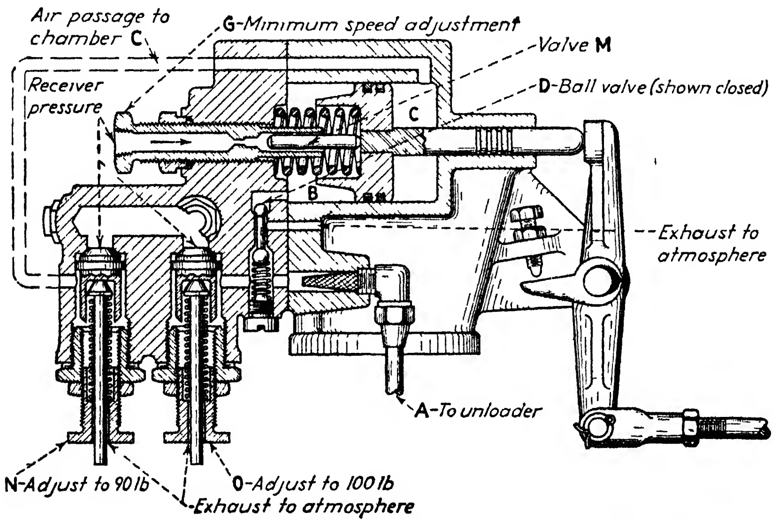


FIG. 133.—Unloading regulator that operates intake valve (Fig. 104) and controls engine speed.

Combinations of Free-air Unloading and Variable Speed.—The unloader in Fig. 112 controls the compressor inlet valve and at the same time increases the speed of internal-combustion-engine drive by changing the fuel-throttle setting.

Figure 133 shows a governor used for internal-combustion-engine drive with the unit normally operating at three-quarters speed. As pressure builds up in the receiver, air flows past valve *M* into chamber *B*, forcing the regulator piston and governor linkage to the maximum speed position.

At 90 psi the auxiliary valve *N* opens to admit receiver pressure to chamber *C* and the piston returns to the three-quarters-speed position. When the pressure reaches 100 psi, the auxiliary valve *O* opens to unload the compressor and open ball valve *D*, exhausting chamber *B* to atmosphere. The piston moves back to the half-speed position and at the same time closes valve *M*. When pressure drops to 90 psi, auxiliary valve *O* closes to load the compressor and relieves the pressure from ball *D*, but the engine continues at half speed because valve *M* remains closed.

At 80 psi auxiliary valve *N* closes and exhausts pressure from chamber *C* to atmosphere, a spring moves the piston to open

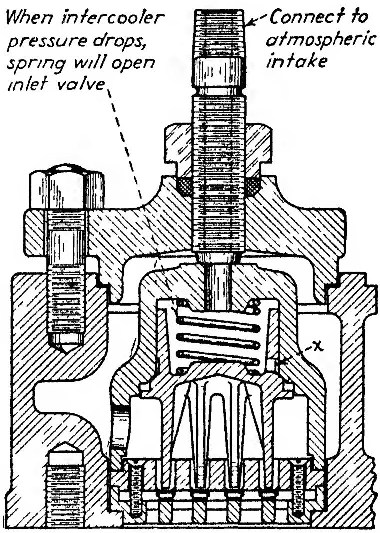


Fig. 134.—High-pressure-cylinder relief valve.

valve *M* and the receiver pressure admitted to chamber *B* moves the piston forward and increases the speed. In actual practice, the compressor finds the speed that corresponds closest to air demand and operates at this level, thus avoiding continuous high-speed operation and wide changes between loading and unloading.

The unloader valve (Fig. 104) mounts on the intake manifold and unloads the compressor when the receiver pressure reaches 100 psi by shutting off the air intake. This unloader, working in conjunction with the high-pressure-cylinder relief valve (Fig. 134) and the intercooler unloader (Fig. 135), unloads the compressor completely.

Figure 104 is actuated by air from auxiliary valve O (Fig. 133), flowing out line A. This pressure enters and flows freely past

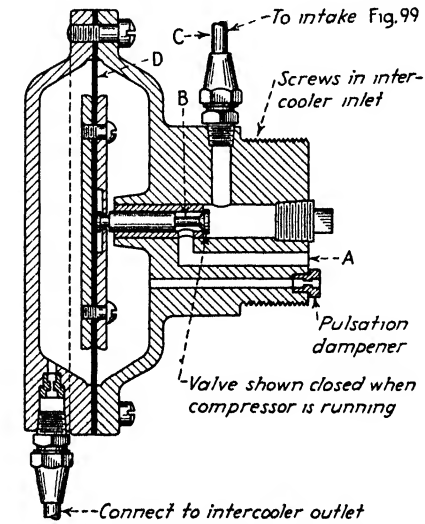


FIG. 135.—Cross section of intercooler relief valve.

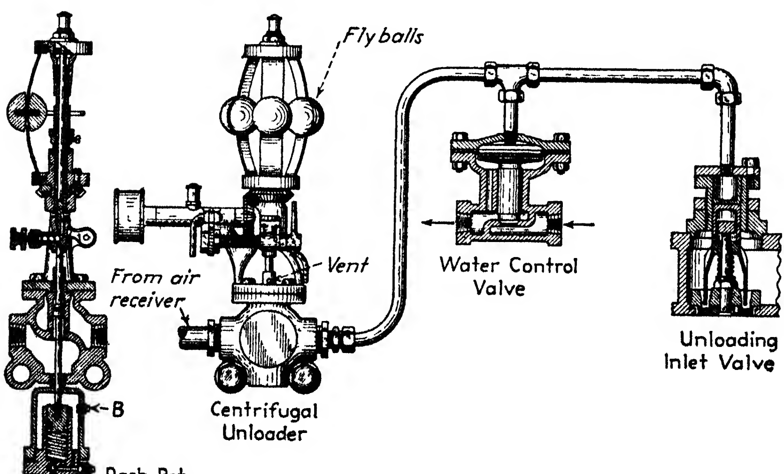


FIG. 136.—Belt-driven flyball unloader piping and cross section views.

ball check K to act on piston E, overcoming springs D and I and forcing valve F closed. The check valve permits a free flow of air from the pressure switch through line A past check K and port *H* to piston *E*, so that the compressor unloads simultaneously with shutting off the engine. To load the compressor after it is started, a three-way valve at the pressure switch opens line *A* to atmosphere, permitting the pressure in *O* to leak out slowly through orifice *R* and delay loading until the compressor is up to speed. The point of loading actually occurs when spring pressure *D* and *I* exceeds the air pressure in *O* and forces valve *F* open. This time delay is set by adjusting screw *G*.

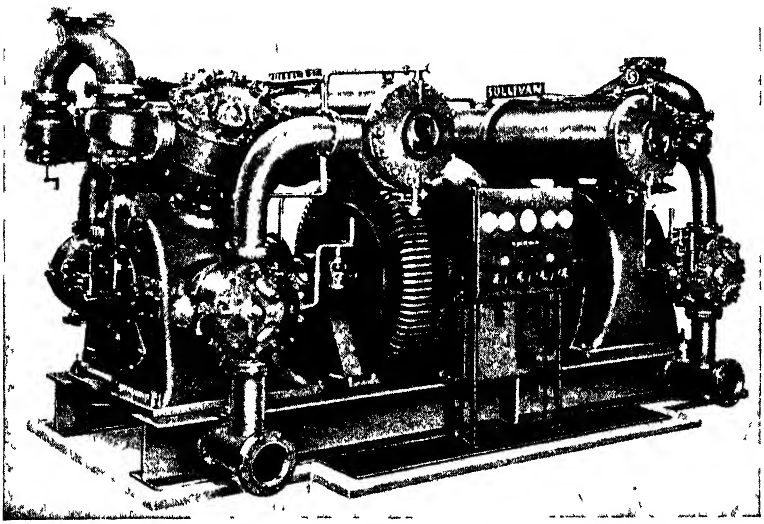


Fig. 137.—Unloading control panel that operates intake valves.

The diaphragm valve (Fig. 135) relieves the intercooler pressure when the compressor stops. The diaphragm chamber connects to the intercooler inlet and outlet as shown. The slight pressure drop through the intercooler creates an unbalanced pressure on the diaphragm *D* which holds the valve *B* on its seat as long as the compressor discharges air through the intercooler. When the compressor cuts off, pressure on the diaphragm equalizes, the intercooler pressure acting on valve *B* opens it, and the intercooler pressure at *A* exhausts to atmosphere through line *C*.

Immediately after the compressor unloader closes the low-pressure intake passage, the pressure in the intercooler and connecting pipes drops rapidly to 10 or 15 psi, at which pressure the spring in the high-pressure-cylinder relief valve (Fig. 134) unseats the plunger. This uncovers ports X and exhausts all pressure in these passages. The plunger then holds the high-pressure inlet valve open, completely unloading the compressor.

When the compressor loads, holes X are inadequate to relieve all the air being compressed in the low-pressure cylinders and pressure builds up in the high-pressure intake passage. As soon

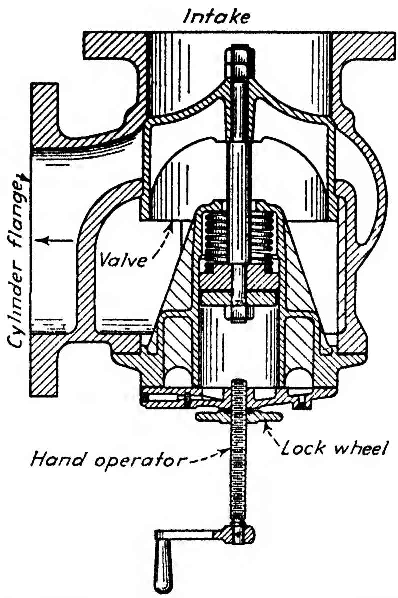


FIG. 138.—Cross section of intake valve on compressor in Fig. 137.

as it reaches approximately 15 psi, the upward air pressure on the bottom of the plunger overcomes the spring resistance and lifts the plunger. This closes ports X to atmosphere and permits the intake valve to function, bringing the compressor to normal operation.

Centrifugal Controllers.—The action of a centrifugal unloader (Fig. 136) is similar to that of a flyball steam governor. When the unloader head is at a standstill, the flyballs spring in toward the spindle, causing it to rise and open a port in the unloader-valve cage. Air then passes to the inlet valves, holding them open and removing the load from the compressor.

When the compressor starts and its speed increases to just under full operating speed, the flyballs pull outward, drawing the spindle and valve down, cutting off air from the receiver, and opening a vent which allows air to escape from the unloading inlet valves. The vent at the top of the unloader body should always be kept open.

The dashpot, furnished only when the compressor is to be driven by a constant-speed motor, delays compressor loading until the motor reaches full speed or has operated a short period at full speed. Remove plug *B* and fill the chamber with dashpot oil. Adjust the regulating screw *C* until the compressor loads at the proper speed.

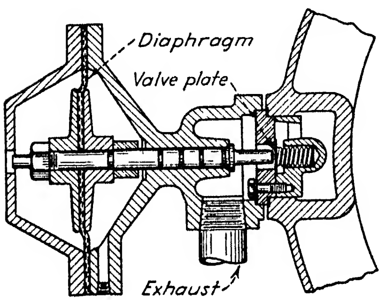


FIG. 139.—High-pressure-cylinder relief valve for compressor in Fig. 137.

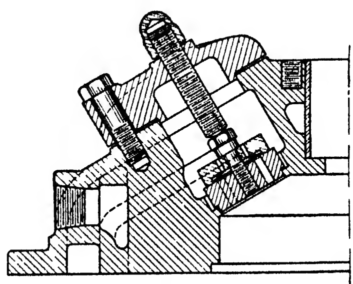


FIG. 140.—High-pressure-cylinder check valve for compressor in Fig. 137.

The compressor (Fig. 137) uses selective electropneumatic control consisting of low-pressure-cylinder unloaders (closed intake) (Fig. 138), automatic high-pressure-cylinder relief valves (Fig. 139), high-pressure-cylinder check valve (Fig. 140), mercoid switches, and solenoid valves. When air demand is less than full load capacity, the compressor automatically regulates its output between 50 and 100 per cent of rated capacity. If the demand falls below half-load capacity, the control automatically regulates output between 50 per cent and zero load. During each of these two distinct unloading steps each half of the machine, namely, each low-pressure and its corresponding high-pressure cylinder, operates independently of the other half of the unit.

A main controller unloads the compressor, and the relief valve keeps the high-pressure cylinder from compressing during the unloaded period by exhausting any air that may leak into the cylinder.

The unloader valve (Fig. 138) attached to the main intake consists of a double-seat valve controlled by air from the main pilot. When receiver pressure rises to a predetermined point, the valve closes to shut off all incoming air to the compressor. When the receiver pressure falls below the unloading-device setting, the valve again opens fully and the compressor resumes full load until the pressure again rises to the unloading point.

The high-pressure relief valve (Fig. 139) consists of a small wafer check in each high-pressure-cylinder head, in combination with a vacuum-operated relief. Vacuum in the system operates the valve.

Failure of the high-pressure relief valve will be indicated by excessive temperature and also by the absence of a sharp hiss of air shortly after the compressor unloads. Generally the trouble will be caused by the vacuum-operated diaphragm wearing out. Replacement of this part should be made at least every 2 years.

Always choose the unloading system that best suits plant needs. Mechanical losses in the machine remain practically the same regardless of whether it runs at full or part load. Operating for long periods at part load markedly increases the power cost per cubic foot delivered. Therefore, operating costs are reduced when the compressor operates the greatest part of its time at full load.

CHAPTER V

RECIPROCATING-COMPRESSOR LUBRICATION

Bearing Lubrication.—Bearings of reciprocating air compressors are generally oiled by a splash, gravity-circulation, force-circulation, ring, or drop-feed system. In some instances combinations of two or more of these methods are used. Bearing lubrication serves two purposes: (1) it reduces friction between the moving parts, and (2) it acts as a cooling medium.

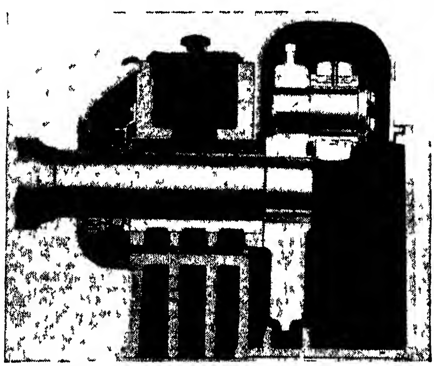


FIG. 141.—Main-bearing oil seal (*Courtesy of Pennsylvania Pump and Compressor Co*)

Splash System.—The splash method, generally used to lubricate all parts of small single-acting compressors (Fig. 56), is also widely used for the bearings of many horizontal double-acting machines (Fig. 141). The crank disk and connecting rod dipping into the oil reservoir splash lubricant to all bearings.

Because the crankcase of splash-lubricated compressors generally serves as an oil reservoir or sump, it is closed except for a breather or vent pipe. The cylinder end of the crankcase on double-acting machines contains a partition or bulkhead *O* (Fig. 142), fitted with wiper rings that prevent oil from working along the piston rod into the cylinder. The machine (Fig. 143) carries a scoop *A* on the crank disk that throws oil into the basin

B from where it travels to the crosshead. The connecting rod *B* (Fig. 51) carries its own oil scoop. Another splash system (Fig. 144) feeds oil to the crankpin by centrifugal force.

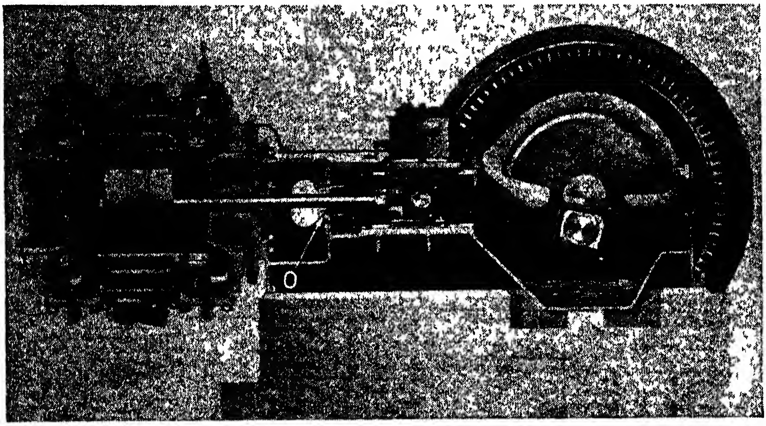


Fig. 142.—Partition fitted with wiper rings prevents oil from flowing along piston rod.

The tapered-roller crankshaft bearings (Fig. 145) are lubricated by oil splashed into pocket *A*, from which it flows down into the bearing and then back to the reservoir. Oil flowing to the outside of the bearings is prevented from leaking along the shaft by oil slinger *B*.

Table 17.—Specifications for Compressor Running-gear Oil

Room temperature, F	Pour point, F max	Viscosity at 100 F, min SSU	Steam emulsion value min, sec
−10 to +30	−15	190	75
30 to 60	25	250	75
Above 60	30	300	75

Gravity Circulation.—In this system, oil is pumped from a bottom tank or sump to an overhead reservoir from which it flows to all bearings through pipes equipped with feed-regulating valves. Regulating valves serve to assist in adjusting the feed to the requirements of each individual part.

Force-circulation Oiling.—In this system, a flood of oil is pumped under pressure directly from a reservoir to all bearings (Figs. 146 and 147). Oil returns to the sump by gravity. Figure

54 shows a gear-driven pump on a compressor having drilled oil passages in the main shaft and crank disk.

On the horizontal compressor (Fig. 146), a chain-driven rotary gear pump takes oil through a strainer from a reservoir below the crankshaft bearings. The oil then goes to the crankshaft and crosshead pin and guides in both the low- and high-pressure side of the compressor. On its way the oil passes through a filter and cooler to ensure a clean cool supply of lubricant to the bearings. In case of oil-pressure failure, a pressure switch connected into the oil line shuts down the compressor to prevent bearing damage from lack of lubricant. Suitable piping carries the oil

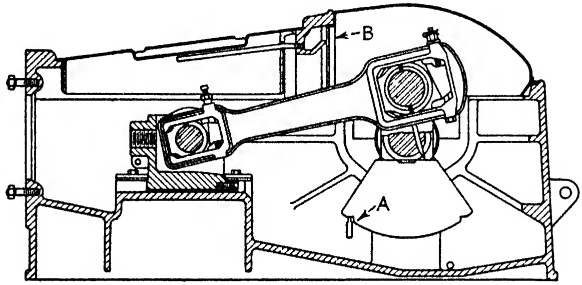


Fig. 143.—Oil scoop *A* throws oil into basin *B*.

to the main bearings (Fig. 148) and to the crosshead pin (Fig. 149). The best location for the radial hole supplying oil to the crankpin bearings is just ahead of the pressure area when the crank is on either dead center. Thus, as the crank advances, a film of oil spreads over the surface just before it is subjected to pressure (Fig. 150). The machine (Fig. 151) uses a sheet-metal baffle and slinger ring to stop oil leakage around the crankshaft. If leakage occurs, it will be found that either the slinger ring is not bearing squarely against metal baffle plate *A*, oil fog is leaking through gap *X*, or the oil-return pipe is clogged.

Both the gravity- and force-circulation systems may be equipped with an oil filter, or cooler, or both, as in Fig. 152. Here the cylinder lubricator, a combination unit, feeds both the steam and air cylinders with additional lines going to the metallic piston-rod packing.

Ring-oiled and Drop-feed Systems.—Ring-oiled bearings are in common use on many types of air compressors. In these bearings, all of which are similar in principle, oil is contained in a reservoir in the bearing housing just below the journal. The ring or chain turns with the shaft and brings oil up and distributes it on the journal where it spreads over the bearing surfaces. Surplus oil drains back into the reservoir to be reused.

Drop-feed oil cups furnish lubrication to small parts, or to those not subject to severe operating conditions. An adjustable needle valve regulates the feed and a sight glass makes it possible to see the oil drops as they are fed.

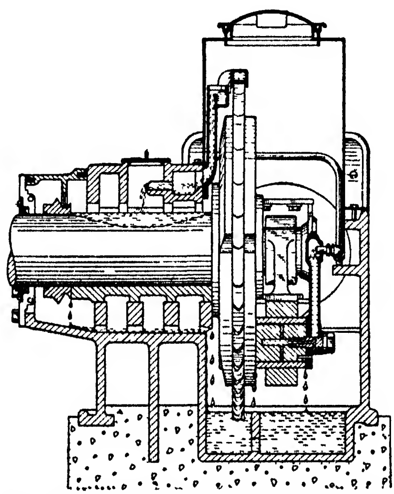


FIG. 144.—Crank disk supplies oil to main bearing and crankpin.

Oil Required.—The quantity of oil used depends on the capacity of the compressor. It should be sufficient to permit the oil to remain quiescent for a period of time each cycle, thus permitting impurities and water to settle. Water is especially troublesome and if not quickly separated will produce emulsion and sludge that collect in the pipes and prevent oil from reaching the bearings. Water and other impurities should be filtered out so that only a clean supply of lubricant remains in the system.

Because it is used repeatedly, the oil must possess, in addition to correct viscosity, high resistance to oxidation and emulsification. Constant agitation and churning over long periods in the presence of air causes oxidation. Oxidized products combine with atmospheric impurities and water, if present, to form sludge. This sludge acts as a catalyst to promote additional oxidation.

Most compressors are operated in comparatively warm surroundings, but some, particularly portable units, are often exposed to low temperatures. Under these circumstances, the oil should have a pour point at least as low as the surrounding air temperature. Otherwise, the oil will not flow freely to the pump suction when starting up, or through the orifices of drop-feed cups. Congealed oil will also tend to channel in splash systems.

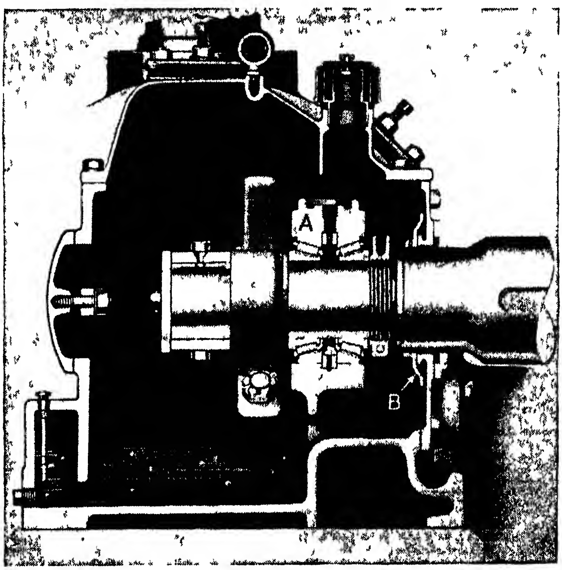


Fig. 145.—Oil slinger *B* throws oil off main shaft.

In most single-acting compressors, one oil is required to serve both the bearings and cylinders. Here the oil viscosity is fixed by the cylinder requirements and not by those of the bearings. Viscosity requirements for cylinders are generally somewhat higher than those of bearings in order to provide adequate piston seal. For reasons of simplicity and to minimize the danger of mixing, use as few oils as possible, especially on the same machine. Consequently, whenever possible, one oil should be used for lubricating both the cylinders and bearings.

The Compressed Air Institute suggests that "for compressors which are provided with independent means for cylinder lubrication, oil for the running gear or crankcase should be selected in general accordance with the specifications given in Table 17."

Always consult the compressor manufacturer and follow his advice in selecting lubricating oils.

Cylinder Lubrication.—Regardless of the make, construction, and operating conditions, lubrication of reciprocating-compressor cylinders resolves itself into protection of the piston, cylinder walls, piston rod, and valves. These internal parts are much more exacting in their lubrication requirements than the external running gear. In the cylinder the oil must not only minimize wear and reduce friction but it must also provide a seal between piston rings and cylinder walls.

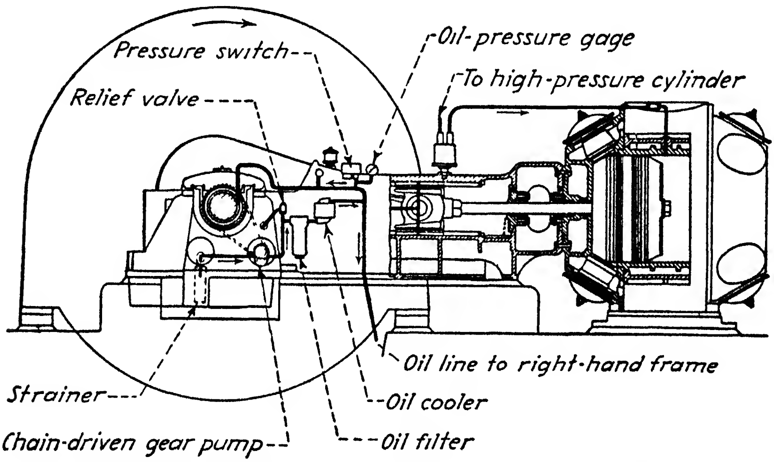


FIG. 146.—Force-circulation oiling for running gear. (*Courtesy of Power.*)

Effective lubrication can be obtained only when the cylinder-wall temperatures are low enough to preclude vaporization of the oil film, yet not so low that condensed moisture will wash away the lubricant.

Oil feed generally goes to the cylinder through one or more pipes leading from separate elements on a multiple force-feed lubricator (Fig. 153). These pipes fitted with check valves connect to the cylinder bore; as the piston passes, the oil gathered up and carried between the piston rings serves for lubrication and also seals the piston against leakage. A separate line feeds the metallic piston-rod packing (Figs. 152 and 154).

The transparent liquid in some sight-feed mechanical lubricators, usually glycerin, is heavier than oil and permits the lighter lubricant to travel up through the sight feed. This provides a solid column of oil from the top of the sight feed to the compressor cylinder. Each drop coming into sight displaces a like amount at the top. Judgment must be exercised when adjusting the feeding rate of this device, because one drop of oil in a glycerin sight-feed lubricator is the equivalent of several drops in another type (Fig. 155) of combination lubricator and unloader.

In time the glycerin mixture will be carried out with the oil, and, although the compressor will still be lubricated properly,

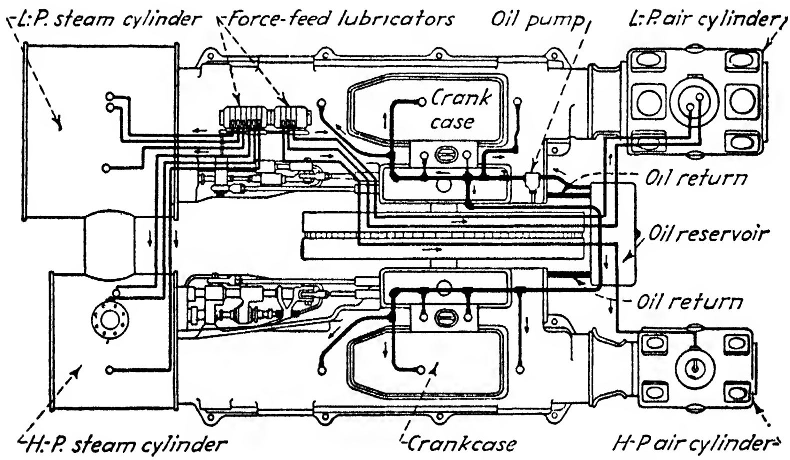


FIG. 147.—Circulating oiling system to all external parts. (*Courtesy of Power.*)

the sight feed must be refilled to observe the feeding rate. A mixture of half glycerin and half distilled water or all glycerin or all water can be used for refilling.

One suggested arrangement for lubricating the cylinder connects oil-feed lines from the lubricator to the intake pipe instead of to the cylinder (Fig. 156). Atomized oil entering with the air covers all moving parts with a fog blanket.

The force-feed lubricator assures a controlled flow of oil under pressure to each cylinder and is an efficient and economical method of lubrication. This device eliminates the possibility of overfeeding because oil feed starts and stops with the compressor (Fig. 157). Force-feed lubricators are not affected by variations in air pressure and will feed at the desired rate according to their adjustment, the compressor speed, and the oil viscosity.

Feeding Rate.—It is not advisable to establish any hard and fast rule in regard to the theoretically proper amount of oil that should be supplied to an air-compressor cylinder. There are too many variables involved, such as the size of the machine, its speed, condition of the piston and cylinder walls, and the amount of dirt in the air. The oil will probably remain in an air-compressor cylinder considerably longer than in the cylinders of either a steam or internal-combustion engine, because there is less atomization and little or no washing action or dilution of the oil film.

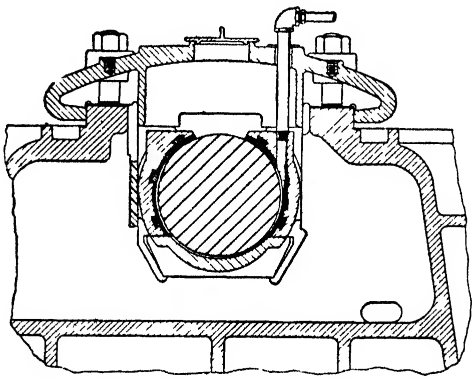


Fig. 148.—Pressure oiling to main bearing.

A time-honored rule states that from 1 to 2 drops of oil per 500 to 1,500 sq ft of cylinder area covered by the piston per minute should assure adequate protection. However, the size of the drops varies with the viscosity, temperature, service conditions, diameter and shape of the lubricator orifice, and the type or design of the lubricator—*i.e.*, whether it is mechanical force-feed or sight-feed. Also, the number of drops per pt will vary. As a result, the number of drops secured per min from an oil having a Saybolt viscosity of 200 sec at 100 F would differ from the number obtained from an oil of 300-sec viscosity.

Two compressors of the same design and size, built by the same manufacturer, may be operating in a room under identical conditions. Yet it will be practically impossible to have the same piston-ring and valve fit and the same polished cylinder surfaces in the two machines, and they may require a surprisingly different amount of oil for lubrication. Any increase in the feed above that necessary to provide an effective piston seal and reduce friction to the minimum tends to increase carbon deposits and the danger of fire and explosion.

It is therefore not considered good practice to make an absolute recommendation as to the number of drops per min that should be used because of the great variation in operating conditions that will be encountered. Since there are so many variables to contend with, the Compressed Air Institute recommends that

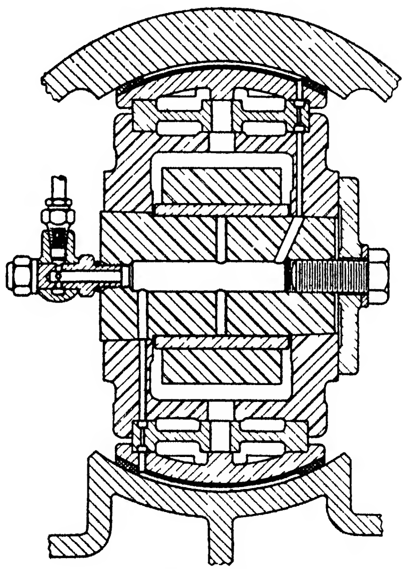


FIG. 149.—Force-feed oiling to crosshead pin and shoes.

the minimum allowable feed under ideal conditions should not be less than that given in Table 18.

Lubricating-oil Characteristics.—A mineral oil that is especially refined to resist the effect of temperatures and one that has the volatile elements thoroughly removed is necessary. This oil should have the correct body and characteristics to meet the requirements of the work. It should not contain compounds that will cause gumming of the valves, or sticking of the piston rings. Neither should it contain elements that will carbonize and necessitate frequent cleaning.

Oil too heavy in body will not spread readily over the rubbing surfaces, and there will be considerable drag on the piston, resulting in excessive friction loss and air leakage, because the oil film is not complete. Furthermore, impurities in the intake air will more readily cling to the heavy oil and bake into a carbonaceous deposit on the piston and discharge valves. If the oil is too light in body and is used in excess to form a satisfactory film, much of it will be carried over into the intercoolers and receiver and will form explosive mixtures. Carbon deposits on the discharge valves interfere with their operation to such an extent that they fail to close completely or quickly enough, permitting the leakage of air back into the cylinders where recompression causes additional heating.

The most important point, however, is that the oils should not contain, as a result of improper refining, any elements that give off highly explosive vapors even above the ordinary operating temperatures which are encountered in air-compressor operation.

Flash-point Considerations.—Flash point is not an all-important consideration when selecting compressor cylinder oil. The flash point is the temperature to which the oil must be heated to give off vapors in sufficient quantity at atmospheric pressure (the flash point increases with pressure) to produce an inflammable mixture with air. The amount of oil evaporated while making the flash test is too small to be measured.

Petroleum oils, being complex chemical compounds or mixtures, have a range of boiling points. The lowest boiling point occurs at a far higher temperature than the flash point for any given oil. A high-grade compressor oil boils at 782 F and flashes at 385 F. Slow evaporation takes place at any temperature and the rate of change from a liquid to a gas depends on time, temperature, and chemical composition. Laboratory tests show that the evaporation rate of a high-grade air-compressor oil is 0.0004 gram per sq cm of exposed surface when kept at 250 F for 5 hr.

This theoretical evaporation cannot occur under normal operation because of water jacketing, cooling effect of intake air, the relaying of a new oil film at each piston stroke, and the fact that high temperatures occur only toward the end of the stroke. Since the discharge valves, being in the path of the hot discharge air, represent the hottest part of the compressor, oil evaporated on the cylinder walls could not condense on them. The presence of liquid oil on the hot valve surfaces proves that no great amount leaves the cylinder as a gas and that carbon deposits come from this liquid oil adhering to the valve. Leaking discharge valves quickly reach temperatures that quite possibly can exceed the

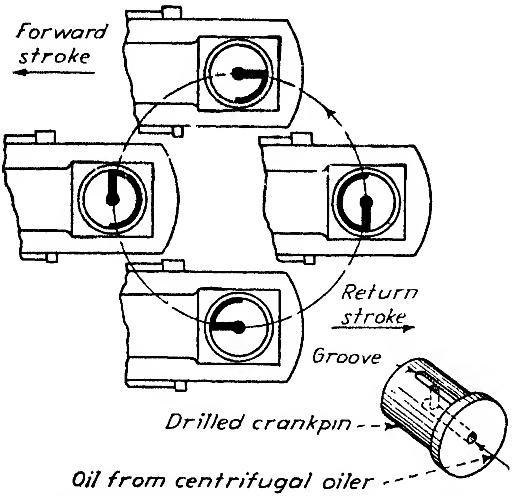


FIG. 150.—Oil spreads over the bearing before load is applied. (*Courtesy of Socony-Vacuum Oil Co.*)

highest flash point of any lubricating oil in existence. This may cause the carbon deposits to become incandescent, forming oil vapors from the carbon and excess oil into an explosive mixture.

TABLE 18.—AIR-CYLINDER OIL-FEEDING RATES

Cylinder dia, in.	Piston displacement per cylinder, cfm	Swept surface per cylinder, sq ft per min	Oil feed per cylinder	
			Drops per min	Pt per 10 hr*
Up to 6	Up to 65	Up to 500	2 d in 3 m	0.05
6–8	65–125	500–750	1	0.075
8–10	125–225	750–1,100	4 d in 3 m	0.10
10–12	225–350	1,100–1,500	1–2	0.112
12–15	350–600	1,500–2,000	2–3	0.188
15–18	600–1,000	2,000–2,600	3–4	0.262
18–24	1,000–1,800	2,600–3,600	4–5	0.338
24–30	1,800–3,000	3,600–4,800	5–6	0.412
30–36	3,000–4,500	4,800–6,000	6–8	0.525
36–42	4,500–6,500	6,000–7,500	8–10	0.675
42–48	6,500–9,000	7,500–9,000	10–12	0.825

* Figures are based upon 8,000 drops per pt at 75 F.

A high-flash-point oil actually may be detrimental to good lubrication because it must necessarily be heavy-bodied, and as a result it will form more objectionable carbon deposits than a lighter bodied oil even if both are of equal quality. The heavier bodied oil also has greater tendency to catch and hold dust particles and is more sluggish in its distribution. The comparatively low cylinder-wall and oil-film temperature eliminates the need of a heavy-bodied oil for lubrication. The most important point is high quality; the oil must not break down and form gummy deposits.

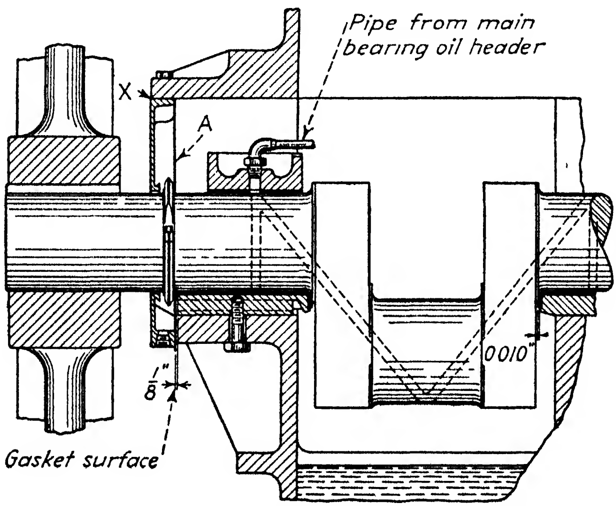


FIG. 151.—Drilled crankshaft with oil-seal ring.

To connect the flash point with the possibility of explosion is irrelevant, unless some adverse condition of construction or operation has prevailed. For example, the use of dirty air, along with an unsuitable lubricating oil, may have built up restrictions at fittings or bends in the air line.

All lubricating oils will be subject to more or less fractional distillation or separation under high temperatures. As this continues, the lighter portions evaporate and are carried off with the discharge air, the heavier fractions meanwhile remaining in the cylinder or collecting in the heads or on the valves.

Carbon residues are caused by exposing lubricants to direct heat. This involves a chemical change in their structure. If this heat is sufficiently intense, it causes abnormal vaporization of the more volatile constituents to result in an accumulation of nonlubricating matter which may be of a tarry or gummy nature. Continued exposure of the latter to heat causes coking or baking

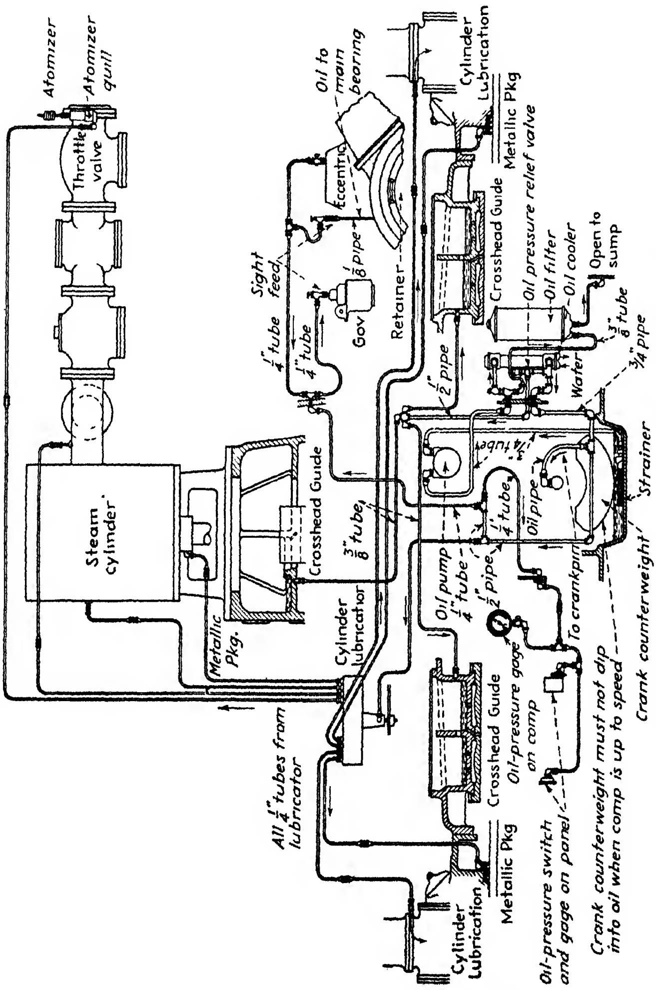


FIG. 152.—Schematic oil piping for cylinders and running gear.

in piston-ring grooves and valves. The cleaner an oil will vaporize, the lower will be the amount of objectionable residue. Normally, carbon residue will vary directly with the body or viscosity of an oil, as well as with the flash point. A wholly distilled compressor oil such as is used for normal conditions should show a carbon-residue content well below 0.50 per cent by the Conradson test.

Compounded Oils.—It is not always practicable, however, to use a wholly distilled oil. In marine air compressors or those installed in salt-air localities, a small amount of fixed or animal oil, such as lard oil, must be added to develop a lathering effect with moisture and prevent corrosion or rusting during stand-by. By adding this compound it is possible to maintain lubrication with normal oil feeds; otherwise, were a straight mineral used, the drops per min would have to be increased. Even then there would be insufficient lathering or emulsification with moisture to assure a protective film that would remain on the contact surfaces when the machine is shut down.

TABLE 19.—AIR-CYLINDER OIL SPECIFICATIONS

Flash point, F.	350 (min)
Viscosity, SSU at 100 F.	245 (min)
Viscosity, SSU at 210 F.	45 (min)
Pour point, F.	+35 (max)
Neutralization number	0.10 (max)
Conradson carbon residue, per cent	2.0 (max)

Either procedure will in time lead to increased residual deposits. On vertical or marine compressors in damp atmospheres this is, however, of less importance than positive lubrication. Even so, any fixed-oil compound should be used as sparingly as possible, for fixed oils are not subject to distillation under normal conditions. Instead, they will break down or decompose to a tarry or gummy residue when exposed to high temperatures.

Causes of High Temperatures.—Excessive temperatures may be caused by any one or more of the following faults: (1) an inadequate supply of cooling water; (2) cooling water too hot to permit proper heat transfer from the air to the water; (3) intercoolers and jackets blocked with dirt, mud, and scale, preventing heat absorption by the water; (4) air baffles in the intercoolers broken, permitting the air to short-circuit to the discharge with very little heat removal; (5) partitions in the waterheads of the intercoolers not tight, permitting the cooling water to short-circuit between the inlet and outlet without passing through all intercooler tubes; and (6) leaking valves or piston rings that permit reentry of compressed air to the cylinders.

That fires and explosions have occurred cannot be disputed. Another indisputable fact shows that even if an oil has a flash point of 400 F, the vapors will not spontaneously ignite at this temperature. Since a good compressor oil has a boiling temperature of approximately 782 F, the spontaneous-ignition temperature must be still higher. Vaporizing any great quantity of oil requires considerable exposed surface and the only way of getting this in a compressor is by failure of cooling-water action. The small amount of vapor produced from a hot valve would not

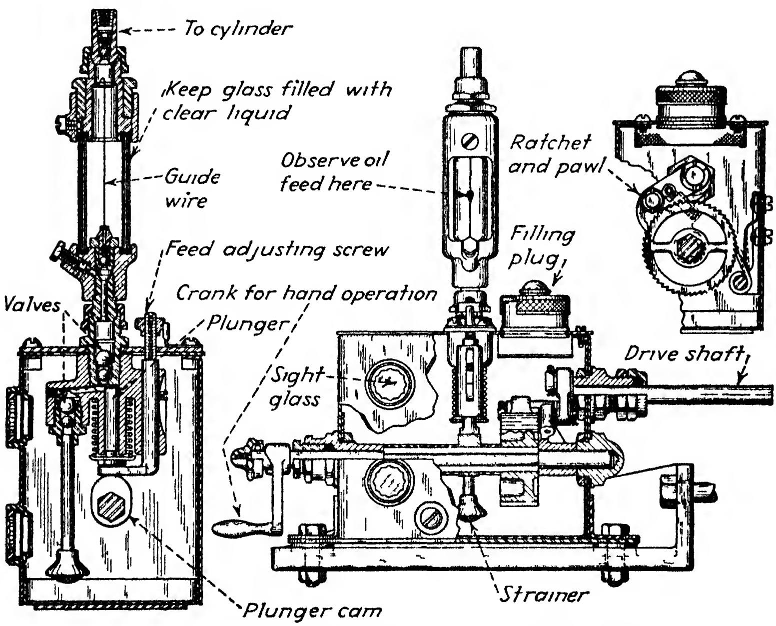


FIG 153.—Cross section of cylinder force-feed lubricator.

produce an explosive mixture of any great quantity, but if this small quantity did ignite, sufficient heat to start vaporizing atomized particles in suspension could be generated.

What makes a leaky discharge valve hot? Many authorities say that recompression of the hot air causes cumulative heating or that friction of the leaking air heats the valve. It is well known that vibration of metal parts causes heating, and fluid leaking past loose or flexible restrictions causes vibration of the obstruction. Therefore, constant vibration may also contribute toward extremely high temperatures in a leaking valve plate.

TABLE 20.—AIR-CYLINDER OIL SAMPLES

	Sample *A*	Sample *B*
Flash point, F	365	410
Fire point, F	410	470
Viscosity, SSU at 100 F	312	403
Viscosity, SSU at 210 F	48	59
Pour point, F.	−25	+25
Neutralization number	0 02	0 04
Carbon residue, per cent	0 04	0 40
Gravity, deg API	20 4	24 0

Packed accumulations of oil and dust lying in piping and other pockets mixed with concentrated oxygen can under certain con-

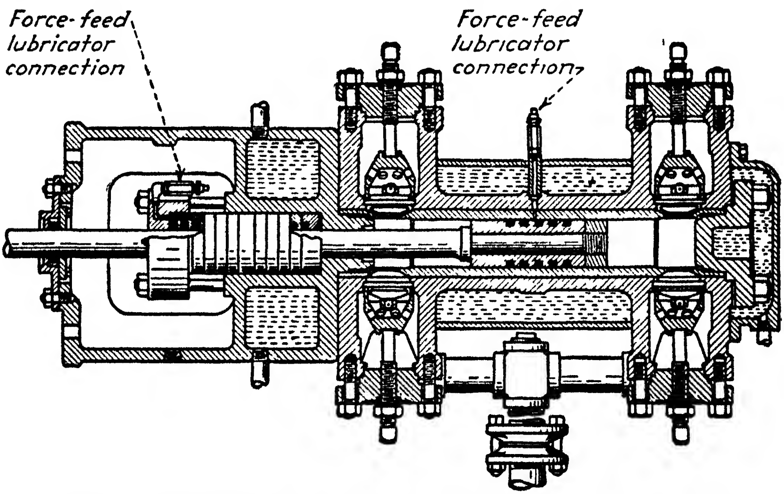


FIG. 154.—Lubricating connections to cylinder and piston-rod packing.

ditions ignite from spontaneous combustion. This trouble appears quite frequently in coal-storage piles when sufficient oxygen is present to cause slow oxidation but not enough air circulates to carry off the heat. Atomized oil mixing with dirt particles and absorbing oxygen from the air may settle to form accumulations favorable for spontaneous combustion.

To prevent explosions adopt the following operating rules:

1. Use only a high-grade compressor oil in the air cylinders, and feed the minimum amount that gives satisfactory compressor operation.

2. Maintain the air filters in good condition to prevent dust and dirt from entering the air cylinders.

3. Clean the intercoolers and jackets at regular intervals according to a schedule dictated by local water conditions.

4. Provide cold circulating water for intercoolers and aftercoolers.

5. Maintain tight air baffles and water partitions in intercoolers.

6. Make periodic inspection and regrind or replace valves and springs as needed.

7. Periodically clean all air passages between the cylinders and receiver, and also pistonheads and piston rings

8. Maintain accurate intercooler gages so that abnormal intercooler pressures may be noted.

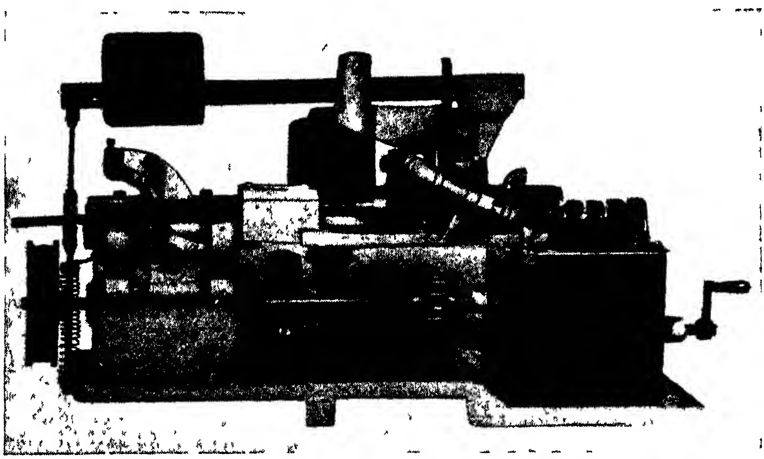


FIG. 155.—Combination unloader and force-feed lubricator.

Oil Selection.—Because of the wide variation in factors influencing selection, it is possible to establish only the minimum or maximum (as the case may be) permissible limits of general requirements for air-compressor cylinder oils. Values recommended by the Compressed Air Institute are given in Table 19. Two representative oils, both falling within the limitations outlined in this table, and both suitable under certain conditions for compressor cylinder lubrication, may have physical characteristics as given in Table 20.

TABLE 21.—STEAM-CYLINDER OIL SPECIFICATIONS

Flash point, F	500 (min)
Viscosity, SSU at 210 F	100 (min)
Pour point, F	80 (max
Conradson carbon residue, per cent	4.5 (max)
Compounding, per cent	0–12 (max)

Steam-cylinder Oil.—Conditions under which steam-cylinder oil must perform depend on initial pressure, percentage of moisture in the steam, amount of superheat, exhaust pressure—whether condensing or noncondensing, and the kind of lubricating system used.

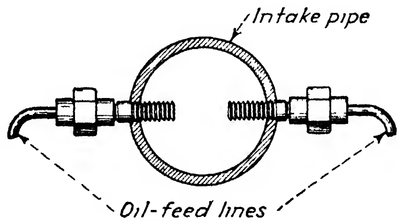


FIG. 156.—Feed lines supply atomized oil into intake pipe (*Courtesy of Double Seal Ring Co*)

For wet saturated steam, compounded oils are usually necessary, with compounding between zero and 12 per cent. For superheated steam, straight mineral oil is sometimes preferred and compounded oils should be limited to those in which com-

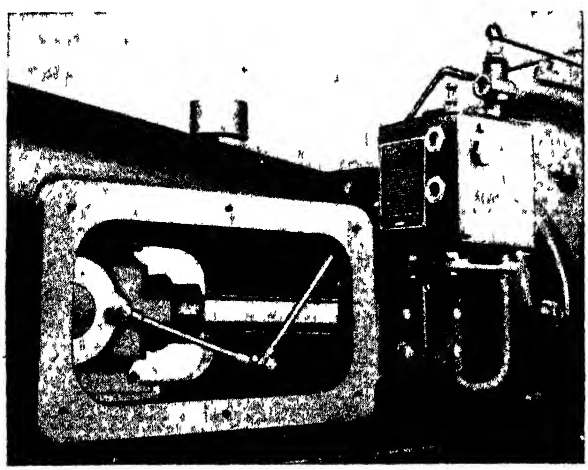


FIG. 157.—Force-feed drive connected to crosshead.

pounding is between zero and 5 per cent. When exhaust steam goes to a surface condenser, it is good practice to specify an oil having a pour test sufficiently low to prevent accumulation of congealed oils within the condenser. This means a minimum amount of compounding.

Because of variable factors affecting the choice of steam-cylinder oils, the Compressed Air Institute recommends permissible limits for the requirements as given in Table 21. Two

TABLE 22.—STEAM-CYLINDER OIL SAMPLES

	Sample *A*	Sample *B*
Flash point, F	565	610
Fire point, F	650	680
Viscosity, SSU at 210 F	210	235
Pour point, F	+50	+45
Conradson carbon residue, per cent	1.0	2.8
Compounding, per cent	5	0

representative oils, both falling within the limitations outlined in this table, and both suitable under certain conditions for steam-cylinder lubrication, may have physical characteristics similar to those given in Table 22.

CHAPTER VI

INSTALLING THE COMPRESSOR

Compressor Room.—Locate the compressor in a clean, light room with ample space around it for making repairs, cleaning, and inspection. Be sure to allow room for removing the pistons, rods, and intercooler tube nests. Provide a support of sufficient strength overhead to handle the machine parts. Making it easy for the operator to inspect a machine gives it a better chance of receiving proper attention.

Foundation.—The foundation is usually of concrete, which is preferable to any other material, although brick or stone may be

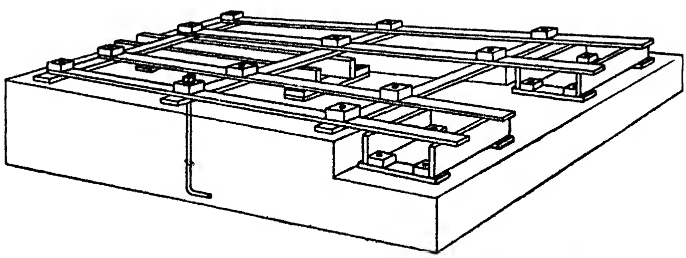


FIG. 158.—Build a substantial template and anchor it securely.

used if they are more convenient; in this event always use cement mortar. Avoid the use of lime mortar.

A reciprocating compressor causes a certain amount of vibration which the foundation must absorb; unless it rests on bedrock or hardpan, a subfooting should be made of such depth and area as to provide a solid bottom. Reinforcing steel bars inbedded in the concrete lengthwise, vertically, and crosswise add considerable strength. In extreme cases of poor subsoil, piling may be required; this generally calls for the services of a competent foundation engineer.

Installing and operating compressors on floors of light-manufacturing loft buildings where noise and vibration are prohibited often present unusual problems.

In such locations the major shaking forces, which exist in all reciprocating compressors, should be so balanced that their greatest effects are in the vertical, with the minor forces acting in the horizontal plane. As these forces not only depend on the weights of the reciprocating parts, but also vary as the square of the speed, a slight change in operating speed often corrects serious vibration. When no shaking forces in either the horizontal or vertical planes can be tolerated, a so-called "balanced" opposed unit fills the requirements. Acquainting the com-

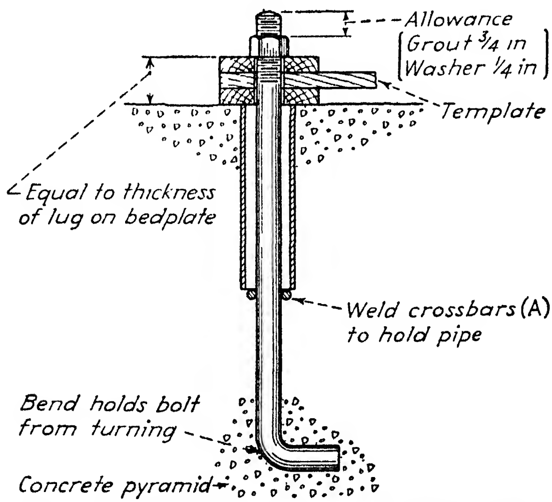


FIG. 159.—Support the anchor bolts and sleeves solidly.

pressor manufacturer with all particulars covering soil conditions encourages him to make suitable recommendations for foundation construction.

In one plant the compressors were fitted with aluminum pistons to reduce the inertia of moving parts and the driving motor was relocated to bring the center of gravity nearer the mid-point of the foundation. One driving-motor pulley was made slightly smaller to keep the machines out of step. To isolate the remaining shaking forces from the building a special foundation was built.

A frame of 8-in. steel angles welded together formed a rectangle of the size required for the foundation. The horizontal faces of the angles, which extended inward toward the center, rested directly upon and were bolted to the floor. The enclosed area was then covered with a 4-in.-thick layer of formed cork. A cork ring of the same thickness carried upward inside the vertical faces of the angles extended for 3 in. beyond their top edge. Concrete poured on this cork bed formed a 10,000-lb block foundation.

Formwork.—When making the foundation forms, make allowance for all air, water, and oil piping and prepare a template to

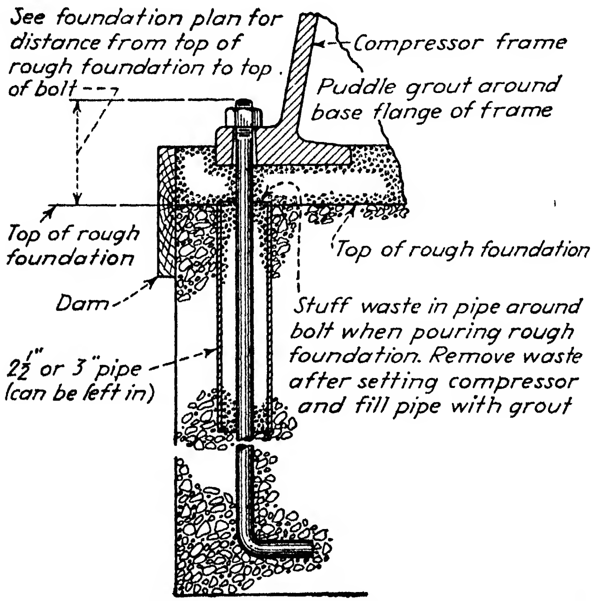


FIG. 160.—Build a substantial dam around foundation to hold grout.

hold the foundation bolts in position. Spacing of these bolts will be found on the foundation plan or can be measured directly from the compressor bedplate. Make a wooden template from heavy planks rigidly braced and supported on the foundation formwork. Lay out the bolt position as shown in Fig. 158 and check the measurements accurately before proceeding with the work.

Set the template in the exact position to be occupied by the compressor with the lower board surface even with the proposed height of rough concrete, and hang the foundation bolts in place. Bolts should not be solidly embedded in the concrete; a sleeve of wood or steel pipe slipped over the bolt (Figs. 159 and 160) will prevent this. The wooden box can be tapered, say 1 in. smaller at the bottom than at the top, to facilitate its removal after the concrete has set. Steel sleeves remain in the foundation.

To ensure the proper height of bolts above the top of the finished foundation, place blocks upon the top of the template boards, so that the top of the bolt will be the required distance

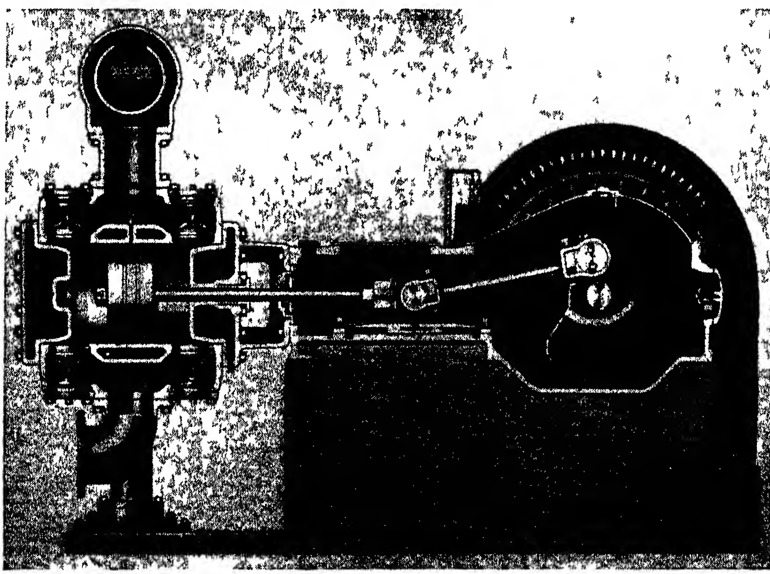


FIG. 161.—Force grout up under bedplate as at *A*.

from the foundation top before grouting The bolts can then be hung with their sleeves and the nuts placed on top. Wooden sleeves can be held in place by nailing to the template; crossbars *A* welded to the bolts below the end of metal sleeves will pull them tight against the template when the bolt nuts are tightened. Do not permit the bolts to swing loosely from the template; weld them to the reinforcing rods or build pyramids of concrete from the foundation floor high enough to embed the rod end (Fig. 159). Allow this concrete to set before the main batch is poured. A loose-swinging bolt becomes misplaced easily. After the template and foundation bolts are substantially supported, preparations can be started toward pouring the concrete.

Most manufacturers recommend a concrete mixture of one part Portland cement, two parts clean sharp sand, and four or five parts well-washed sharp broken stone. Mix the sand and cement together thoroughly before adding the stone. Add only enough water to make a stiff and plastic mixture and work it through the reinforcing and around the formwork into a compact mass. Allow the foundation to set at least 1 week before erecting the machine. During this time keep the concrete wet and, if out in the sun, lay wet burlap over the surface.

After the concrete has hardened, remove the template and check the foundation-bolt spacing again. Then clean the foundation face by chipping away $\frac{1}{4}$ in. of concrete to a rough uneven surface, except for a small area around each bolt. These small areas will support leveling wedges. Protect the cleaned surface during machine erection and keep away all oil and grease.

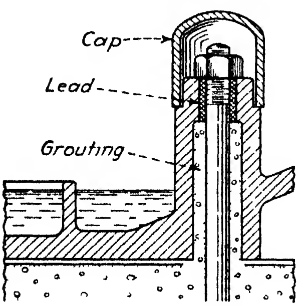


Fig 162.—Seal for anchor bolt that extends into oil reservoir.

Preparing for Grout.—When the machine is ready for grouting, build a wood dam around the foundation just high enough to contain the required amount of grout. Then sweep or brush the concrete surface absolutely clean and wet it thoroughly. This prevents rapid absorption of water from the grout. Mark the wedge locations on the dam so that they can be removed after the grout is poured. Some authorities recommend filling the bolt sleeves with grout, others say fill them with sand, and still others suggest leaving them empty. Many foundations have been poured without using sleeves, and since they have served their purpose during erection they may as well be filled with grout.

Prepare the grout from one part Portland cement and two parts clean sharp sand mixed with water to the consistency of thick cream. Pour it quickly and work it into the sleeves and well under the machine frame Fig. 161*A*. As soon as it has set sufficiently, remove the dam, trim the grouting, pull out the leveling wedges, and fill the resulting holes. After the grout has thoroughly set and hardened (4 days to a week) pull the foundation-bolt nuts down tight. If the compressor must be placed in service sooner than this, use a quick-setting grout.

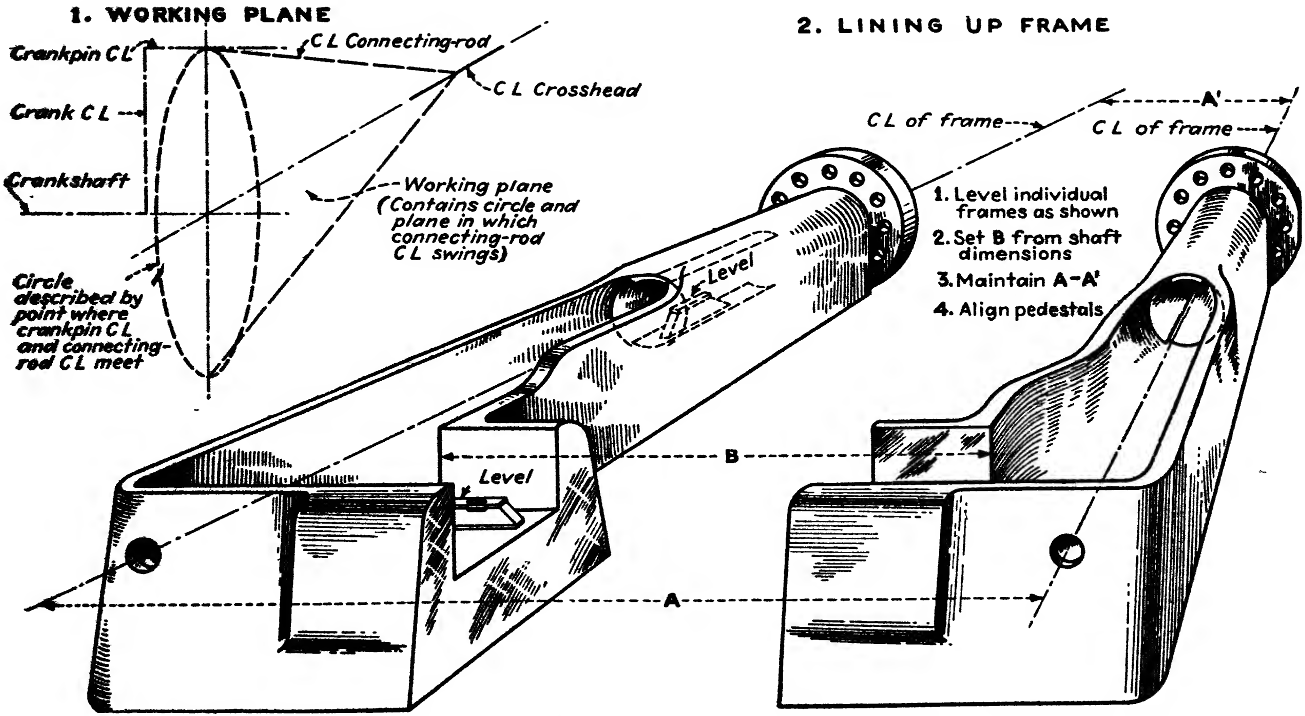


FIG. 163.—Working plane of reciprocating machine as affected by frame leveling. (*Courtesy of Power.*)

Grouting sometimes upsets alignment, and it is extremely important that leveling be rechecked after the anchor bolts have been pulled down tight. It is much better to reset the compressor frames and regrout than to repair the damage caused by operating a unit out of line.

On some machines one foundation bolt comes on the inside of the crankcase. This bolt is covered by a loose bolt cap, but as crankcase oil is liable to work in around the cap and down into the concrete, care must be taken to see that the grout is carried up along the foundation bolt to within 2 in. from the face of the bolt lug and then leaded into the underside of the nut as shown in Fig. 162. If this is overlooked and too much oil placed in the crankcase, the oil will run down under the bolt and finally loosen the bed from the foundation. Never operate the compressor or connect external piping to it until the foundation has set and is positively hard and strong.

Erecting the Compressor.—Compressor erection consists of assembling the unit and leveling it on the foundation. Before proceeding to details of lining up the compressor, let us see what is meant by correct alignment. In a reciprocating compressor the shaft converts rotary into reciprocating motion.

Looking at 1 in Fig. 163, where only center lines are shown, the point where crankpin and connecting-rod center lines meet describes a circle around the shaft center line. The plane of this circle must be perpendicular to the shaft. Likewise, the connecting-rod center line should also swing in a plane perpendicular to the shaft. This plane is the same as that of the circle, or the so-called "working plane." Proper alignment of a piston compressor requires a working plane perpendicular to the shaft, in which piston, crosshead, and connecting rod must move.

Prepare to level the frames by providing metal wedges and shims. At each anchor bolt put one wedge and shims enough to hold the frame 1 in. above the concrete. Maintain this 1-in. clearance throughout leveling operations to provide plenty of room for the grout.

Two-piece frames must be individually leveled and then leveled exactly with each other by planed surfaces in bearing jaws and specially located leveling pads (Figs. 163 and 168). First remove the pistons and crossheads, then lower one frame into position over the anchor bolts and support it on the wedges.

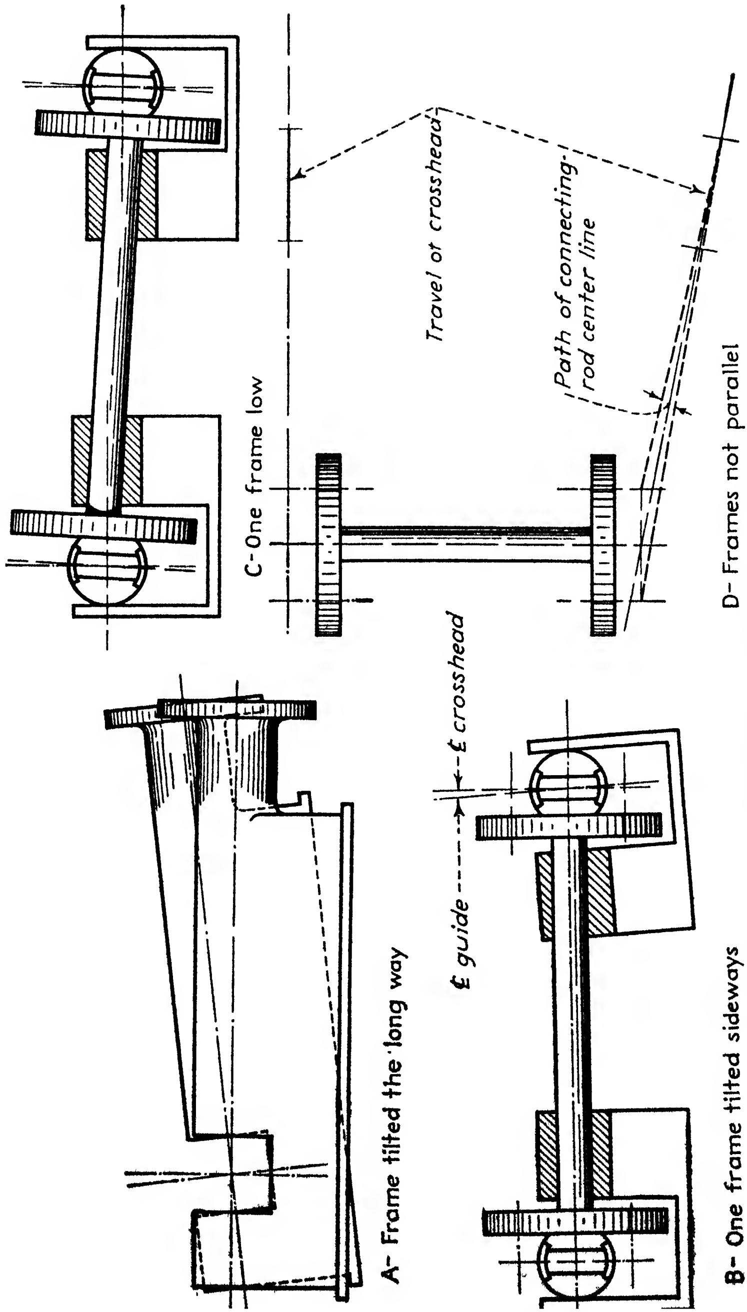


FIG. 164.—Four possible kinds of misalignment. (*Courtesy of Power.*)

Adjust this frame with the wedges, putting the level on the crosshead guides or in the cylinder bore, if it is attached, to level the frame lengthwise and across the bearing jaws or special leveling pads to get it true crosswise. Tighten the anchor

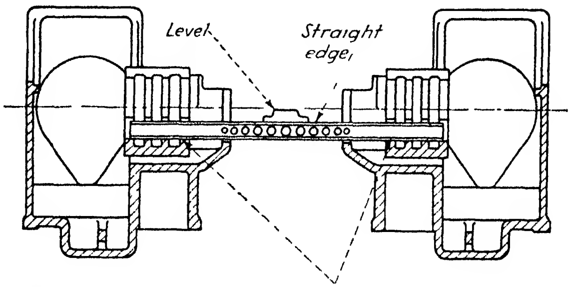


FIG. 165.—Allowance for compressor shaft that carries a heavy rotor and flywheel. (*Courtesy of Ingersoll-Rand Co.*)

bolts firmly. (Slightly pitched frames such as Fig. 164*A* do not interfere seriously with bearing performance because this out-of-level condition does not affect the working plane. If a frame goes slightly out of level in the long direction and remains undis-

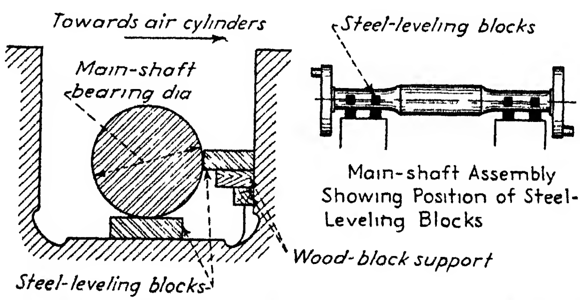


FIG. 166.—Use steel leveling blocks to center the main shaft. (*Courtesy of Pennsylvania Pump & Compressor Co.*)

covered until after grouting, do not consider the condition alarming.) Now lower the opposite frame on its anchor bolts and put it in an approximately level position.

Level this side to the other by placing a line or straightedge across the bottom of the main-bearing jaws (Fig. 165), and across the cylinder flanges of the two frames. Sometimes this head-end leveling can be done by laying the straightedge across the two crosshead guides. Line up the two main-bearing-jaw faces next to the cylinder so that they are directly in a straight line, using a good straightedge or piano wire, preparatory to putting in the crankshaft.

Check the over-all distance over the outside ends of the main bearings or bearing housing against the distance between the disks on the crankshaft to make sure that the center distance of the two frame halves is correct and that the end clearance between the disks and main bearings is sufficient to let the shaft into position. This measurement can be checked with a wood stick 1 in. shorter than the disk span. Fit a nail in each end to make up the remaining distance. A light tapping on either nail will adjust the stick to the accurate length.

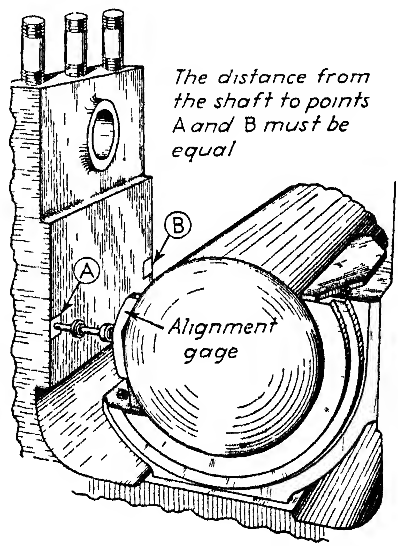


FIG. 167.—Checking shaft in main bearing with alignment gage.

The final alignment of frames and the main shaft can be done either with leveling blocks or by using the regular main bearings. Leveling blocks, eight in number, support the shaft and its bearings as shown in Fig. 166. Place two blocks on the floor of each main-bearing jaw and adjust the frames until the main shaft rests evenly on all four blocks. For lengthwise alignment of the frames, place two blocks between the main shaft and the two upright ribs of each main-bearing jaw. Then move the frames until the blocks fit snugly; if all four are in contact with the shaft accurate lengthwise alignment is assured. These side blocks can be supported at the center of the shaft by wood strips.

Leveling with Main Bearings.—Frames also can be leveled with the bottom sections of the main bearings in place. Carefully wipe the bottom of the bearing blocks and the inside of the bearing jaws so that specks of dirt do not adhere to any of the surfaces that go together. Set the blocks in the frame. Check the over-all distance over the outside ends of the bottom halves of the bearings against the distance between the crank disks on

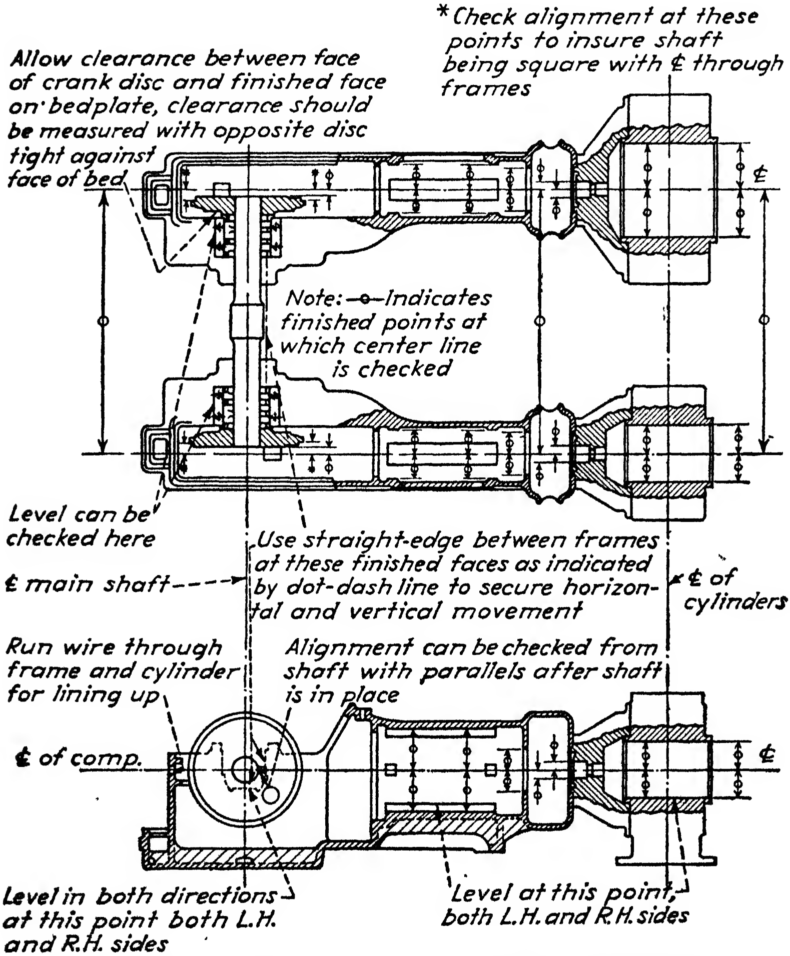
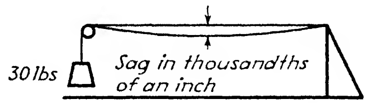


Fig. 168.—Follow these directions when leveling compressor frames.

the shaft to make sure that the center distance of the two frame halves is correct and that the end clearance between the shaft and bearings is sufficient to let the shaft into position. Lower the shaft into position and slide endwise until the clearance between crank disks and bearings is all at one end of the shaft. The total end clearance usually runs from 0.006 to 0.035 in., depending on the size of the machine.

Select a bolt and its nut long enough to place in a horizontal position between the shaft and the center of each bearing jaw to act as a jackscrew. Put a piece of brass or copper between the bolt and the shaft to prevent marring its surface and jack the shaft back against the bearing block and the block against the



Sag table for 0.016-in. piano wire when used with a 30-lb weight

Length of Wire in Feet	2	4	6	8	10	12	14	16	18	20
4	.001"									
8	.001"	.002"								
12	.002"	.004"	005"							
16	.003"	.006"	.007"	008"						
20	.004"	.008"	.011"	.012"	.013"					
24	.006"	.010"	.014"	.017"	.019"	.020"				
28	.007"	.013"	.017"	022"	.024"	.025"	.026"			
32	.008"	.015"	.021"	.026"	.030"	.033"	.034"	.035"		
36	.009"	.017"	.024"	.030"	.035"	.039"	.042"	.043"	.044"	
40	.010"	.020"	.028"	035"	.040"	.046"	.050"	.052"	.053"	.054"

Distance From End of Wire

FIG. 169.—Sag table for piano wire. (*Courtesy of Ingersoll-Rand Co.*)

frame jaws. Very little pressure is required to force the shaft and bearings to position.

Check with a 0.0015-in.-thick feeler gage to see that there are no spaces between the back and bottom faces of the bottom halves of the bearing and the bottom and sides of the bearing jaws. An adjustable alignment gage serves to check shaft and frame alignment. Some bearing jaws carry oblong spots *A* and *B* scribed near the outer edges of bearing jaws for applying the gage as shown in Fig. 167. Should the gage show that the shaft is not the same distance from both spots on either jaw or with both jaws, move one frame either forward or backward just enough to allow the alignment gage to read the same on both frames. Should only one jaw now fail to line up with the shaft, this indicates that the tramming should be rechecked because one of the frames is out of parallel with the other. During erection at the factory, the maximum allowable difference between the two spots and the shaft in any one jaw is only 0.0005 in.

Next run a piano wire through the frame and cylinder bore (if the cylinder barrel is in place), the packing gland, and the hole through the rear end of the frame as shown in Fig. 168. Stretch this wire taut (Fig. 169) and center it accurately over its entire length. Turn the main shaft so that the crankpin on each disk touches the wire, and measure the distance from the disk face to the wire (Fig. 168). Turn the shaft 180 deg and take another measurement. If the shaft is perpendicular to the cylinder center lines, these distances will be equal. With the wires still in place measure the horizontal distance between them at the front of the frame or cylinders, at the crossheads, and behind the rear end of the frame. If the frames are parallel these measurements will be equal.

When the frame halves are center-punched to show the distance of separation as shown in Fig. 170, they also carry two punch marks *A* and *B* to use as a gage for the over-all length of the tram. If the tram is not furnished, make one from a $\frac{1}{4}$- or $\frac{1}{2}$-in. rod long enough to match the two standard center-punch marks.

Alignment Difficulties.—Difficulty in aligning a two-frame compressor arises from the fact that each adjustment quite often throws the unit out of level in some other direction. This causes considerable confusion unless every point is rechecked after each individual change. To sum up: Each frame must be level in the direction of its center line and at right angles to this line; the two frames must be level with each other in the same horizontal plane, and their center lines must be parallel and the correct distance apart; and the main-shaft center line must be at right angles to both cylinder center lines while passing through the centers of both bearings.

If a compressor frame with flat crosshead guides tilts sideways, the crosshead cannot adjust itself to this position. However, this will not affect alignment of a compressor with bored guides

(Fig. 149) because the crosshead can shift to remain perpendicular to the shaft (Fig. 164*B*) without disturbing the working plane, although the main bearing will have to be refitted.

The misalignment (Fig. 164*C*) shows one frame lower than the other. The bored guides allow the crosshead to turn slightly and the working plane stays perpendicular to the shaft, but the shaft thrusts toward the low bearing and both bearings will

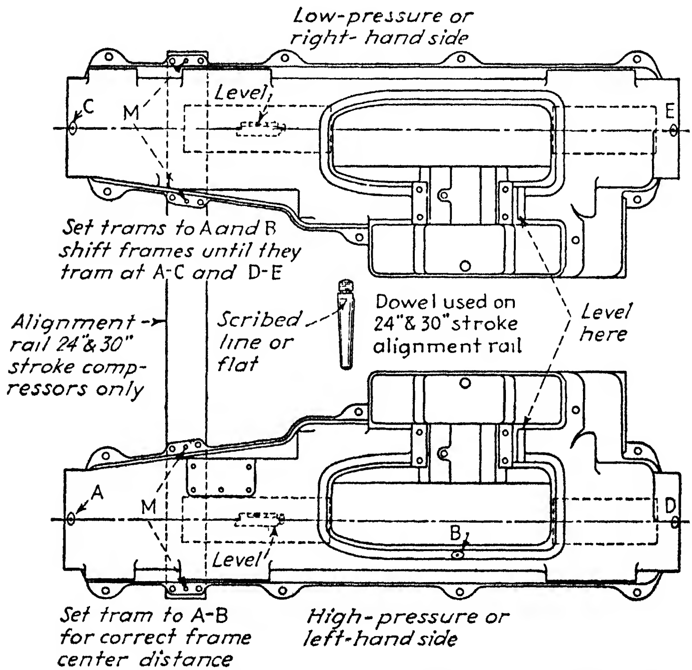


FIG. 170.—Leveling points are generally marked on compressor frames.

require special fitting. This misalignment, if not too great, can be corrected by shimming up the low bearing. In *D* out-of-parallel frames cause the connecting rod to move in three dimensions, which produces a side thrust on the crosshead.

Most compressors with 12-in. cylinders and larger are usually fitted with a cylinder foot-piece support (Fig. 161). These frequently consist of three parts—an elbow, a distance piece, and a footplate. The elbow is held to the foot piece with four tap bolts. Jackscrews between the elbow and the footplate provide temporary adjustment during line-up, after which this space is filled with shims. A $\frac{1}{8}$-in. filler should always be inserted between the elbow and footplate to provide clearance for breaking the pipe joint when the cylinder is to be removed.

Aligning the Cylinder.—To align a cylinder with the frame, place the pedestal in position under the cylinder and remove its outer head. Then raise the cylinder to the proper height in a

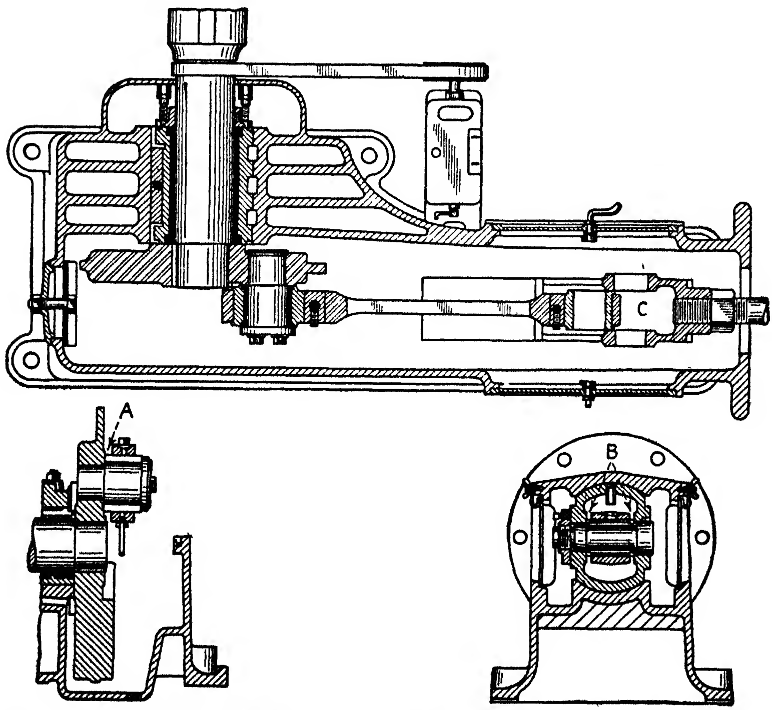


FIG. 171.—Checking parallelism of connecting-rod bearings. (*Courtesy of Power.*)

level position and move it into the frame joint of the bore. Tighten the stud nuts temporarily to hold it in this position. Clean the intercooler or piping flange and place a machinist's level on it parallel to the main shaft and crosswise of the cylinder. Roll the cylinder until it is level with the frame and tighten the frame head bolts, meanwhile watching to see that the level condition is not disturbed.

Next place the level on the bottom of the cylinder bore at right angles to the main shaft or parallel to the piston travel. Put wedges under the cylinder pedestal and raise the far end until the bore is level with the crosshead guide. The cylinder must be level with the frame. When using a machinist's level on a curved surface, always check the crosslevel to be certain that the level lies at the extreme bottom or top of the curve and at right angles to the curve.

Checking Piston and Connecting Rods.—Manufacturers usually put two center-punch marks a given distance apart somewhere on the compressor frame as a setting for trams used to couple the piston rods to the crossheads. Corresponding punch marks will be found on the piston rods and crossheads. After these parts are in place align the connecting rod with its crank and crosshead.

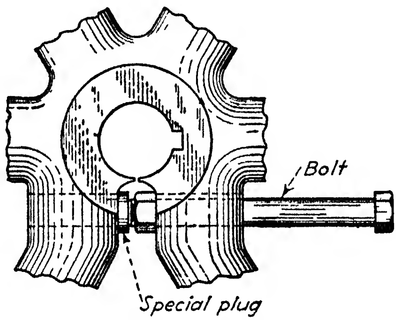


FIG. 172.—Steel plug and bolt opens flywheel bore.

Place the machine on dead center with the crosshead pin removed (Fig. 171*C*) and move the crosshead forward so that the connecting-rod boxes clear the inside faces of the crosshead. Then draw up the adjusting screws on the crankpin box so that the connecting rod is held firmly. If the crosshead-pin boxes remain equidistant between the inside faces of the crosshead (*B*) when this test is made at all four 90-deg crank positions, the crankpin and its bearing faces are true.

Check crankpin alignment in the same manner by replacing the crosshead pin and setting the machine on dead center. Loosen the adjusting screws on the crankpin boxes, remove the retaining washer and cap screw, and tighten the adjusting screws on the crosshead-pin box. Check the position of crankpin boxes relative to the crank disks at all quarter positions of the crank. If the crankpin boxes are just free of the crank-disk face (*A*), the connecting rods are in proper alignment.

Attaching the Flywheel.—Two-piece flywheels are usually shipped separate from the shaft. In preparation for assembling turn the main shaft until the keyway faces upward. Lower the unkeyed half into the wheel pit and block it up into position against the shaft. Lower the other half over the shaft and key and carefully match the fit. Be careful to see that one half of the wheel has not been turned end for end.

When the flywheel is properly set in place on the shaft, bring the hub and rim faces even. If the hub faces will not stay even, hold them by wedging halves of the flywheel. The hub bolts must be made tight while the hub faces are even. After tightening the first bolt in place, next tighten the one adjacent to it, then follow with the other two. The rim links can be inserted in their slots by heating them to a cherry-red color; do not overheat. Keep the rim faces even while inserting the links.

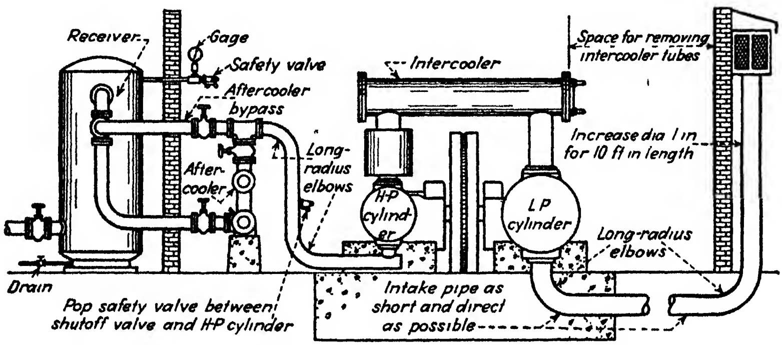


FIG. 173.—Practical pointers for installing compressors. (*Courtesy of Ingersoll-Rand Co*)

Flywheels with solid rims and split hubs must have their hubs opened up to slip on the shaft freely. The key is usually a drive fit in the shaft and a sliding fit in the hub. First put the key in the shaft and then loosen the hub bolts. Drive wedges in the hub split to expand the bore enough to slip over the shaft. Do this expanding carefully; if the wheel does not slip on easily look for an obstruction of some kind. Wheel hubs having sufficient gap space can be expanded with the bolt and plug or washer shown in Fig. 172.

After running the machine a short time, if the flywheel has been found to run true, remove the through bolts one at a time and heat until the section between threads and head is a cherry red. Dip the threaded end into water and cool until its color disappears. Quickly insert it into the flywheel hub and tighten the nuts as much as possible. The bolts shrink as they cool, and the increased tension thus produced ensures a wheel that should never loosen.

Lining Up the Motor.—Since the rotor of a direct-drive motor usually depends on the compressor bearings for support, it is obvious that the stator of the motor must be exactly in line with the compressor shaft. After locating the compressor and firmly securing it in position, put the motor bedplate or slide rails carefully in place and level in both directions. Do not grout in the bedplate until after the stator and rotor are located and accurately aligned. Before assembling the rotor on the shaft, remove all protecting grease or paint. It is extremely important

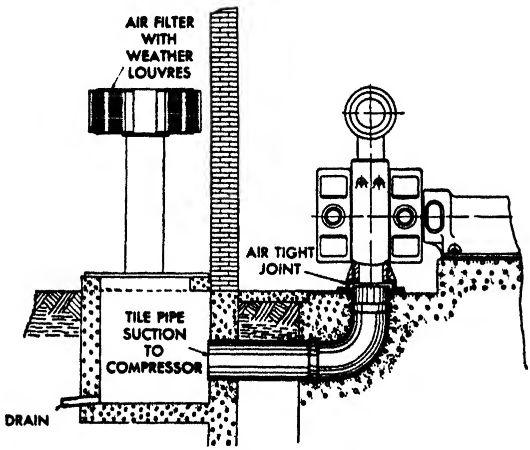


Fig. 174.—Glazed-tile intake.

that the rotor be tight on the shaft and that it run true. The final position of the stator must be such that its center line coincides with that of the rotor.

This point is of great importance, since in any other position there will be a strong end thrust which will overheat and may seriously damage the compressor bearings. In addition to being located with respect to the rotor along the shaft, the stator must be set so that the air gap is equalized around the circumference and on both sides. After the air gap has been equalized in one position of the rotor, rotate it one quarter turn and check at four equally spaced points on both sides. The air gap on the horizontal must be exactly equalized. Any large error in adjusting the air gap will cause severe strain on the shaft and excessive wear in the bearings. Moreover, it frequently causes the motor to be noisy.

Provision is made for sliding the stator along its foundation bedplate to provide easy access to the rotor and stator coils. When putting conduit, potheads, or other equipment in the pit, be careful that they do not interfere with this movement.

Collector rings must be truly centered on the shaft so that they run true and without eccentricity. Fasten the brush rigging so that the brush holders clear the rings by about $\frac{1}{8}$ in. Sand in the brushes until each one makes maximum contact with the ring.

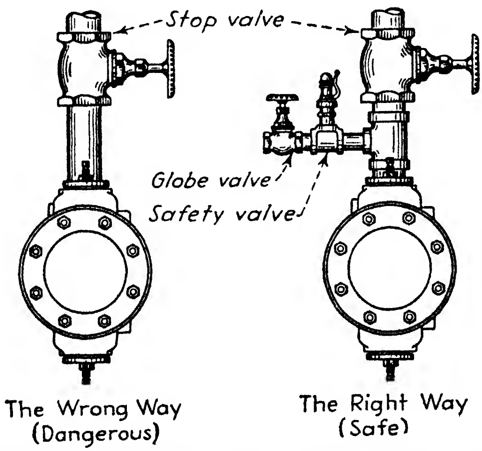


FIG. 175.—Always install a safety valve between cylinder and stop valve. (*Courtesy of Compressed Air Institute.*)

After the unit has run several days, recheck the motor alignment and then dowel the stator to its bedplate. This running time permits any errors in alignment to show up. Doweling serves to hold the stator in the correct position and facilitates returning it to its proper position after removal.

Adjusting Belt Drive.—After setting a belt-driven compressor and its driver in position, move the driving pulley so that its V-groove centers line up with those in the sheave wheel. Slide the motor or engine toward the compressor so that the belts can be mounted without stretching them over groove ridges. Adjust the belts so that with one side tight, the loose side can be depressed in or out about 1 in. at a point midway between the two sheaves. Screws are provided on the engine or motor frame to facilitate this movement. To tighten the belts, loosen the motor or engine hold-down bolts so that they are free, but do not remove them. Take up evenly on the two sliderail screws to move the engine or motor away from the compressor. In this way the driver will not move askew, with consequent misalignment of sheaves. When the belts are adjusted, retighten the hold-down bolts. Never allow the belts to run too tight, as this will cause them to rupture and also places an excessive load on the bearings. Slippage is indicated by a high-pitched squeal or by burning of the belt.

Air Intake.—Run the compressor intake (Fig. 173) to the outside of the building and keep it at least 8 to 10 ft above ground level. Attach a good filter to the air inlet and locate it so that steam, water, excessive dust, or other waste discharged from near-by vents cannot be blown or drawn into the intake. Install a hood covering to keep out rain and snow. Locating the intake at the cool shady side of the building is much better than placing it directly over a roof where the air may be 5 to 10 F hotter than at the side (Table 23).

TABLE 23.—HOW INTAKE TEMPERATURE AFFECTS COMPRESSOR CAPACITY

Temperature of intake, F	Relative intake volume required, cu ft*	Temperature of intake, F	Relative intake volume required, cu ft*
30	925	80	1,019
40	943	90	1,038
50	962	100	1,057
60	981	110	1,076
70	1,000	120	1,095

* Intake volume required to produce 1,000 cu ft of free air at 70 F.

If it is not possible to place the intake outside, the air can be taken directly from the compressor room. However, as the room temperature is usually much higher than the outside air, considerably more power will be required to compress the same amount of air by weight. The lower the intake-air temperature in relation to the temperature of the air at the tools, the greater the amount of energy delivered to the tools or work.

It is equally important to safeguard against drawing moisture into the cylinders. Arrange the piping so that the moisture will drain off before it reaches the cylinder. Never run intake lines near steam or hot-water lines, for this radiating heat raises the intake-air temperature and causes considerable loss in efficiency of the unit. For every 5 deg reduction in the intake-air temperature there is an actual capacity gain of approximately 1 per cent.

The intake-pipe diameter must be equal to or larger than the connection on the cylinder and never less than 25 per cent of the piston area. It should be increased 1 in. in diameter for every 10 ft in length measured from the compressor. A low

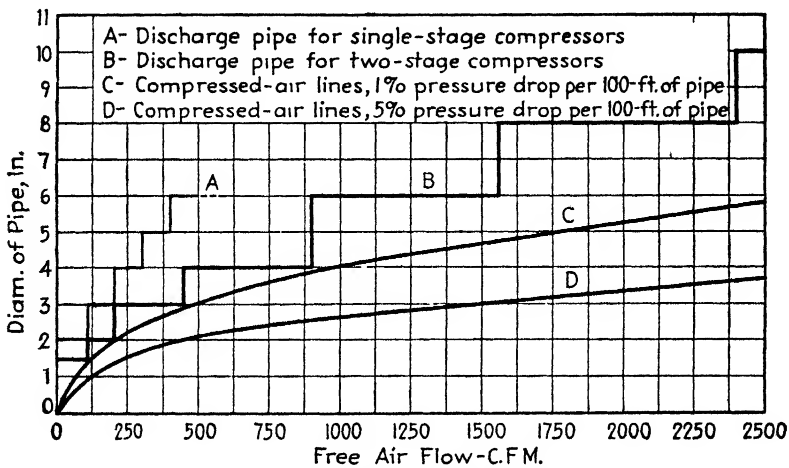


FIG. 176.—Pipe sizes for transmitting compressed air at 100 psi. (*Courtesy of Power.*)

intake-pipe velocity (which should not exceed 1,200 ft per min) reduces friction and consequent pressure loss, and also tends to reduce air pulsations. Make the intake pipe as short and direct as possible and use long-radius bends instead of short elbows where a change in direction occurs. When one intake pipe or duct serves for more than one compressor, make the cross-sectional area of the main duct at least as large and preferably larger than the combined areas of all the individual intake pipes. In branching from the main pipe avoid sharp corners.

The intake pipe can be of cast iron, black or galvanized steel pipe, or any other material which will not crack and which will keep out moisture and dirt. Glazed vitrified pipe with cemented joints (Fig. 174), all embedded in concrete, makes good construction. If a concrete duct is used, coat the inside surface with a good waterproof paint. Avoid intake pipes of rectangular cross section, particularly when using wood or metal for the duct. Rectangular cross section gives large flat sides, which are very susceptible to vibration. Wood is not a good material for intake purposes unless lined with metal because cracks soon develop and dirt and moisture are free to enter.

Should a shutoff be necessary in the intake, install a gate-type valve. Inserting a valve in this line calls for a small receiver between the valve and compressor to reduce intake pulsations.

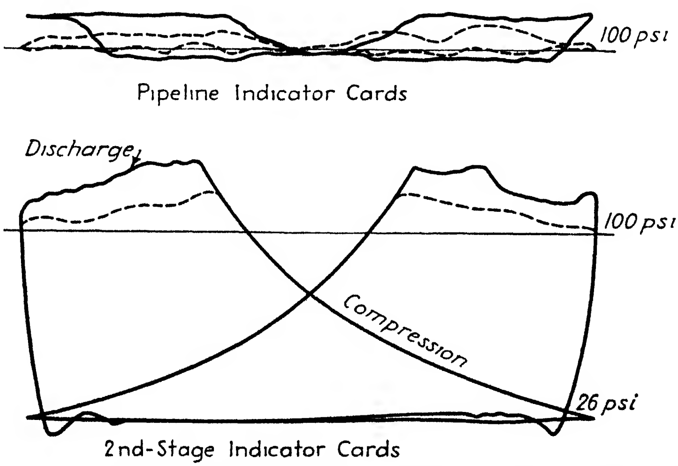


FIG. 177.—Extra power required because discharge line was too small Dotted lines show reduction when larger pipe was installed. (*Courtesy of Power.*)

The receiver also collects dirt and moisture and should therefore be drained periodically. The intake pipe must be thoroughly cleaned and absolutely free from all scale and dirt before the compressor is started. When the line runs underground, pull a swab through it occasionally as a check against surface-water leakage or condensation.

Discharge Line.—The discharge pipe connects the compressor cylinder to the aftercooler and the aftercooler to the air receiver. It must be at least as large as the flanged or threaded outlet on the cylinder, and should be short and direct with as few bends as possible, these being made with long-radius elbows. All joint gaskets should be made from oil-resisting material; otherwise, oil will soon destroy them. Connect a safety valve, capable of discharging compressor capacity to hold the pressure to 10 per cent above normal, in the discharge line between the cylinder outlet and the first valve in this line (Fig. 175). Also provide a drain valve in the lowest point on the line so that accumulated oil and moisture can be drained. Insert an aftercooler between the compressor and the receiver.

The piping must be well supported so that it imposes no strain on the compressor cylinder, aftercooler, or receiver. Any shutoff

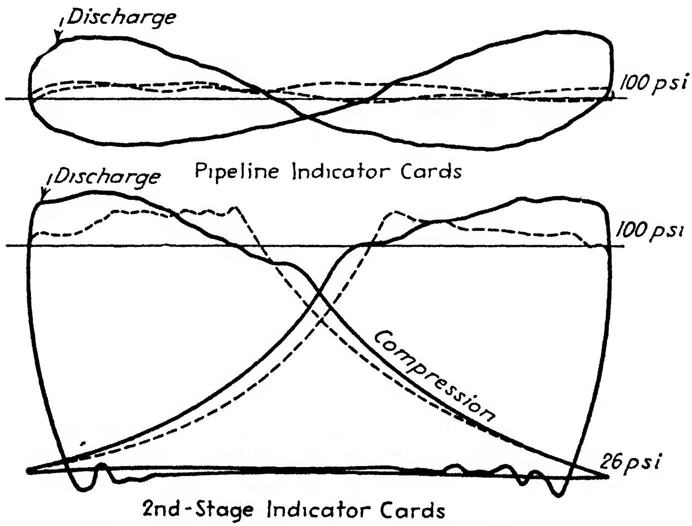


FIG. 178.—Extra power required because of discharge-pipe pulsations. Dotted lines show reduction after installing surge chamber. (*Courtesy of Power.*)

valves installed in the air line should be of gate design to minimize friction loss.

Line losses come from air friction on the inside of pipe and fittings, pulsation of air pressure, and air leakage. Pipe lines that are too small for the air volume handled cause excessive pressure drop and power loss. The size or length of pipe between the compressor and receiver, or between the compressor and a right-angle bend in the discharge, may be such that it will produce surges in resonance with the frequency of the cylinder discharge. When such surges occur, discharge pressure goes higher than normal, with a corresponding power loss. In Fig. 176, curve *A* gives the recommended discharge-pipe sizes for single-stage compressors, and curve *B* for two-stage units.

Indicator diagrams (Fig. 177) show a condition where the line from the compressor to the receiver is so small that it causes an excessive pressure increase each time the compressor discharges. These cards were taken simultaneously on the high-pressure cylinder and on the discharge line, the indicator drive in both cases being connected to the high-pressure crosshead. Normally, this compressor would require 250 hp for continuous operation against 100 psi, but in this case it required 265 hp, which caused the driving motor to overheat. The dotted line shows the cards after a larger pipe was installed.

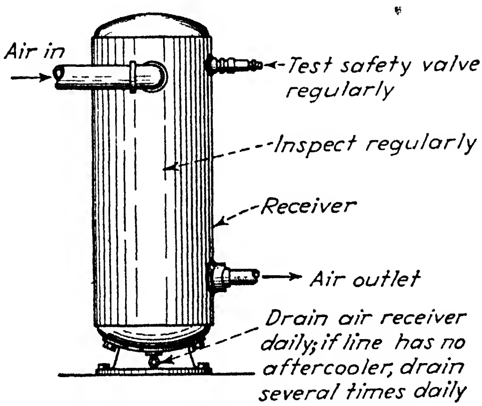


FIG. 179.—Suggested safety measures.

The effects of pipe-line surges are best illustrated by a typical compressor and compressor-discharge pipe-line indicator card (Fig. 178). Pulsations set up in the pipe line were of such frequency that the waves returned to the compressor at the instant of discharge. This caused a build-up in pressure for part of the cycle and increased the work required to deliver air to the line. During the rest of the cycle, pressure fell below normal to give an average of 100 psi, as indicated on a damped gage connected to this line.

The cards show about 30 hp excess load due to pipe-line pulsations, which was corrected by connecting a pipe tee and a short length of pipe, equal to half the cylinder volume, to act as a surge chamber. Pipe-line and compressor indicator diagrams, taken after the change was made, are shown in dotted lines. The lower compression lines were probably caused by the lower temperature of the cylinders as the discharge-air temperature dropped from 240 to 225 F after installing the surge pipe.

Air pulsations may be suspected if the motor is overheating or if power requirements are higher than expected. To determine the existence of these surges, take indicator cards on the line; always take diagrams when a new machine is installed.

Air Receivers.—Using air receivers of unsound or questionable construction is asking for serious trouble. Most states now incorporate in their laws the ASME standard of construction for unfired pressure vessels. Therefore, compressor manufacturers comply with these regulations. Table 16 gives ASME

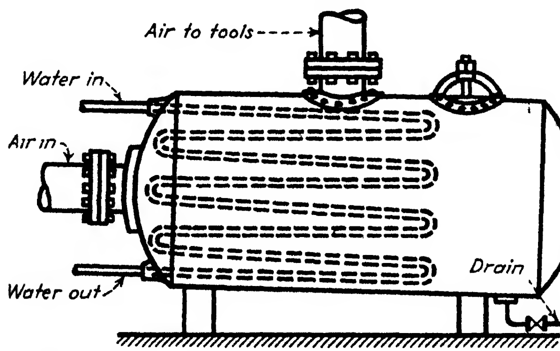


Fig. 180.—If receiver or cooler must lie on its side, tilt it toward drain line. (*Courtesy of Power.*)

recommendations for the minimum-size receiver that will dampen discharge pulsations and give a steady flow of air. This volume usually equals one sixth to one tenth the free-air capacity of the compressor. Larger receivers furnish more storage capacity for start-and-stop operation or to supply heavy instantaneous loads.

Mount the receiver so that no areas contact wet surfaces, to protect it against external corrosion. When located outside, set the receiver on a suitable foundation. Keep the metal surfaces and seams cleaned and well painted.

If one head is convex and the other concave on a vertical unit, have the concave (pressure side) head at the bottom (Fig. 179) to promote complete drainage. Connect a drain line in the middle of the head and attach an automatic drain trap. A horizontal receiver should not rest directly on brick, concrete, or wooden supports. Cradle it on metal shims to prevent corrosion from moisture in these surfaces. Give a horizontal unit (Fig. 180) a slight pitch and connect the drain line at the low point.

Always install receivers in such a position that all drains, handholes, and manholes are easily accessible; allow clearance at all points to permit a complete external inspection. Never bury an air receiver underground or place it in an inaccessible location.

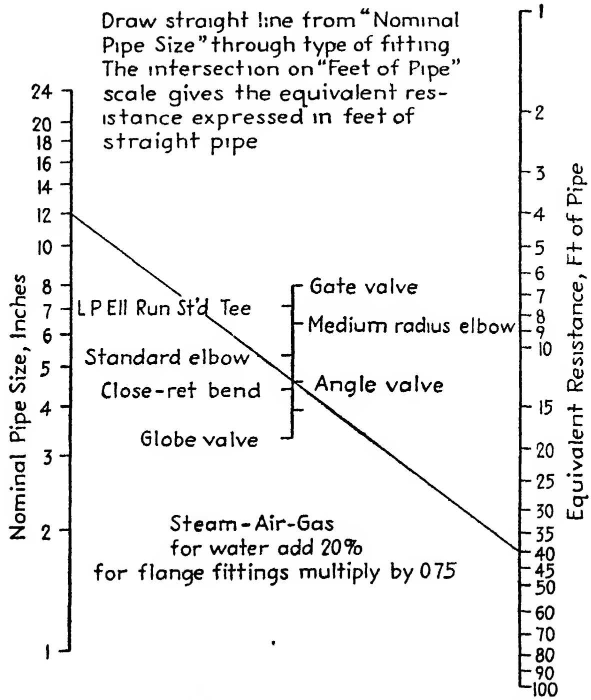


FIG. 181.—Equivalent resistance of screwed fittings. (*Courtesy of Power.*)

Provide each receiver with an accurate pressure gage and one or more spring-loaded safety valves, set not to exceed the vessel's safe working pressure. The valves must be able to discharge full compressor capacity without allowing the pressure to exceed normal by more than 10 per cent. When connecting a new compressor to an existing receiver, be sure to install safety-valve capacity to take care of the new unit. Connect service lines for unloaders to the receiver rather than to the compressor-discharge line. Insert a moisture and sediment trap in this line.

Compressor manufacturers recommend connecting the air-discharge line into the receiver near the top and taking the plant

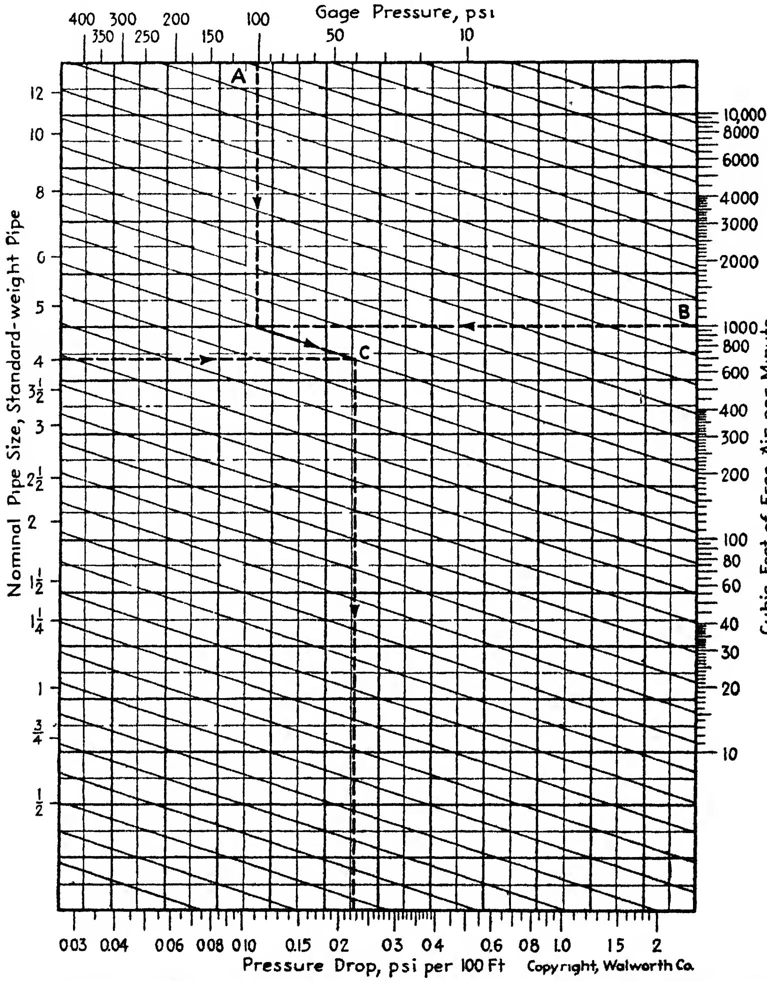


FIG. 182.—1,000 cfm of free air flowing through a 4-in. standard-weight pipe under an initial pressure of 100 psi gage loses 0.225 psi for every 100 ft length of run. Enter the chart at *A* and *B*, follow the point of intersection diagonally to size of pipe *C*, and then drop down to the bottom scale. (*Courtesy of Walworth Co.*)

load supply out near the bottom (Fig. 179). With this hookup the coldest air will be near the bottom, but it may not be any drier because moisture carry-over coming in at the top naturally falls through the lower air. High outlet velocity can carry this moisture out into the main line, whereas with the outlet from the top the air would be warmer but would not carry much excess moisture. Having the inlet at the bottom prevents incoming moisture from passing the outlet as it falls to the receiver bottom.

Distribution Lines.—The pipe line from the receiver to the point of distribution can be smaller than that from the compressor to the receiver, since, for the former, air flow is steady and the air volume reduces in cooling. Compressed-air transmission of necessity entails pressure drop. Curve *C* (Fig. 176) shows pipe sizes on the basis of 1 per cent loss per 100 ft at full line capacity.

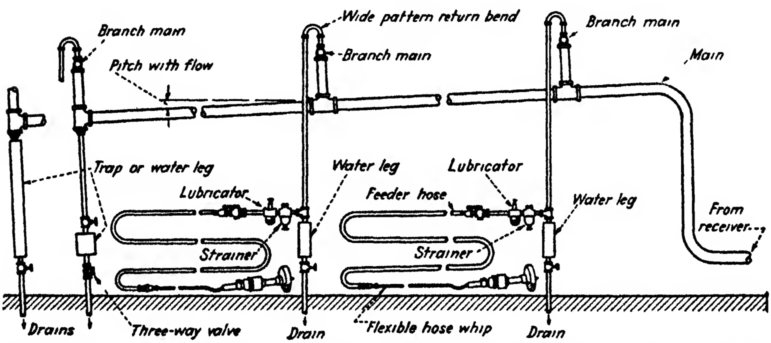


FIG. 183.—Take all outlets from the top and slope the main to a drain pocket. (*Courtesy of Factory Management and Maintenance.*)

Since demand varies, the average loss will be less than 1 per cent. For comparison, pipe sizes giving a calculated 5 per cent loss per 100 ft at a corresponding rate of air flow are shown by curve *D*.

The length of pipe includes an equivalent length for elbows, valves, and other fittings. Complete data on the proper equivalent to use for different fittings will be found in Fig. 181.

As an example of how to use the curves in Fig. 176, assume that a typical industrial plant uses a maximum of 1,000 cfm of air and has a 200-ft main supply pipe, including necessary fittings, between receivers and branch lines leading off to points of use. According to curve *C*, the distribution line should be at least a 4-in. pipe. If 2.5-in. pipe were used, the loss in 200 ft would be 10 per cent if fully loaded, 8 per cent at 750 cfm, and 2.5 per cent at 500 cfm. Assume 750 cfm and a power input of 112 kw to the compressor motor at this capacity. Taking the average loss at 8 per cent, power loss is 8.96 kwhr per hr, which, at 2 cents per kwhr, represents a cost of $0.18 per hr.

Only the air volume and pressure that actually arrive at the point of use can do any useful work. Energy lost through pressure drop (Fig. 182), and air lost through leaks cannot be recovered. Here is how to avoid them.

1. Plan pipe sizes generously, to keep pressure drop between the receiver and the point of use to 5 psi or less.

2. Where possible, use a loop system around the plant and within each shop or building. This usually costs somewhat more but has the advantage of giving two-way feed to the point of greatest air demand. If an existing installation is overloaded, remarkable improvement can be made by installing an additional line parallel with the first and providing connections between them at several points. If part of the new feeders can be taken off the new line, so much the better.

3. Place good-sized receivers near the far ends or at points of heavy use on long systems. Without storage capacity near such points of use, the pressure will drop excessively.

4. Provide numerous outlets on each header or main for hose-attached air-operated tools and devices. Always put the outlet at the top of the pipe so that moisture will not be carried along out of the header. If several tools are to be connected to one outlet, be sure to make it sufficiently large. The main (Fig. 183) is sloped to a drain with all outlets coming out at the top. Such a system with its drop-legs, traps, and screens prevents moisture and dirt from getting into the tools and eliminates waste of time and air for the morning blowdown to clear the lines of water.

Practically, it may be assumed that the loss in power of a pneumatic tool is directly proportional to the drop in pressure at the throttle. Pressure drop increases with added rate of flow through pipe, hose, and connections. A large part of this loss comes from the use of improper hose and fittings.

The curves (Fig. 184) show the horsepower developed by a tool with 90 psi at the hose entrance at various speeds and with different hose arrangements. Actual air pressure at the tool throttle is shown in Fig. 185. These data represent a test of a typical pneumatic tool requiring from 75 to 85 cfm of free air. The horsepower is shown in per cent of the maximum for more ready indication of production losses.

Curve *A* (Fig. 184) using 12½ ft of ¾-in. hose, is considered basic. This is the minimum hose length that could be used. Most tools of larger variety will require at least a 50-ft length. To make it easier for the operator to handle the tool, a short section or whip of lightweight flexible hose frequently is put between the main hose and the tool.

The cumulative total of many small leaks is surprising. It is probable that few plants would show less than 10 per cent, and

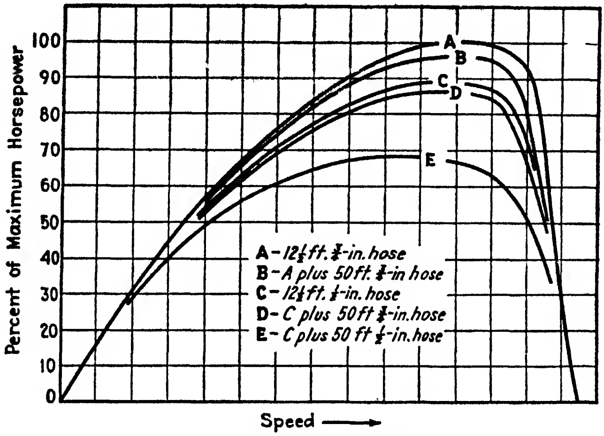


FIG. 184.—Horsepower developed by a tool with 90 psi air at various speeds. (*Courtesy of Factory Management and Maintenance*)

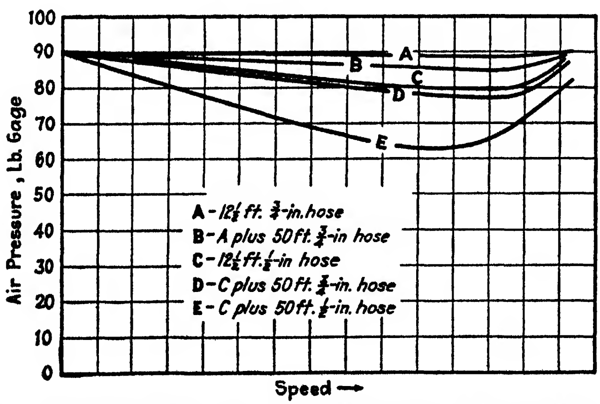


FIG. 185.—Actual air pressure at throttle of a tool requiring from 75 to 85 cfm of free air. (*Courtesy of Factory Management and Maintenance.*)

most would probably waste from 15 to 20 per cent. A plant-leakage test is not difficult to run. First, see that all valved outlets are closed. Bring the pressure up to the operating point with one compressor and permit it to load and unload, maintaining that pressure for an hour. Keep a record of the time the unit operates at each load point, so that the load factor over the test period may be calculated. From the latter and the known full-load capacity, the leakage may be determined. Figure 186

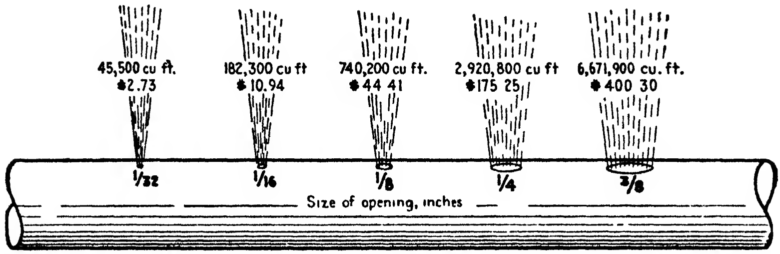


FIG. 186.—When air goes to waste, dollars go with it. The values above are based on a cost of 6 cents per 1,000 cu ft. (*Courtesy of Factory Management and Maintenance.*)

gives some indication of the volume and value of air wasted through leaks.

Large leaks can be checked by sound when the plant is quiet, but an application of soapsuds or oil to the joints is best for detection of small ones. An aromatic material such as essence of peppermint injected into the compressed air sometimes helps, because the odor becomes most pronounced at the leak.

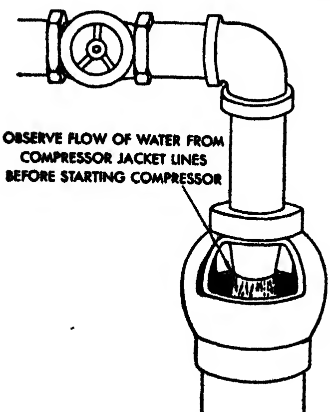


FIG. 187.—Sight-flow funnel for cooling water. (*Courtesy of Compressed Air Institute.*)

Cooling-water Piping.—Protect the cooling-water system by installing a suitable strainer on the main supply line. Provide flanges or unions on all pipes near the compressor and arrange them so that the various parts can be removed with the least disturbance to the piping. Insert drain valves at all low points so that the entire system can be drained during shutdown in freezing weather.

Even though an automatic valve starts and stops water flow (Fig. 121) to the compressor, each cylinder and the intercooler should have its own regulating valve set to maintain a constant inflow. Put these control valves on the inlet, with the outlet free, allowing the water to fall into an open pipe or funnel (Fig. 187) so that the water flow can be checked at a glance.

Feeding the cooling water into the lower connection on cylinders and coolers and discharging it from the top, (1) in Fig. 188

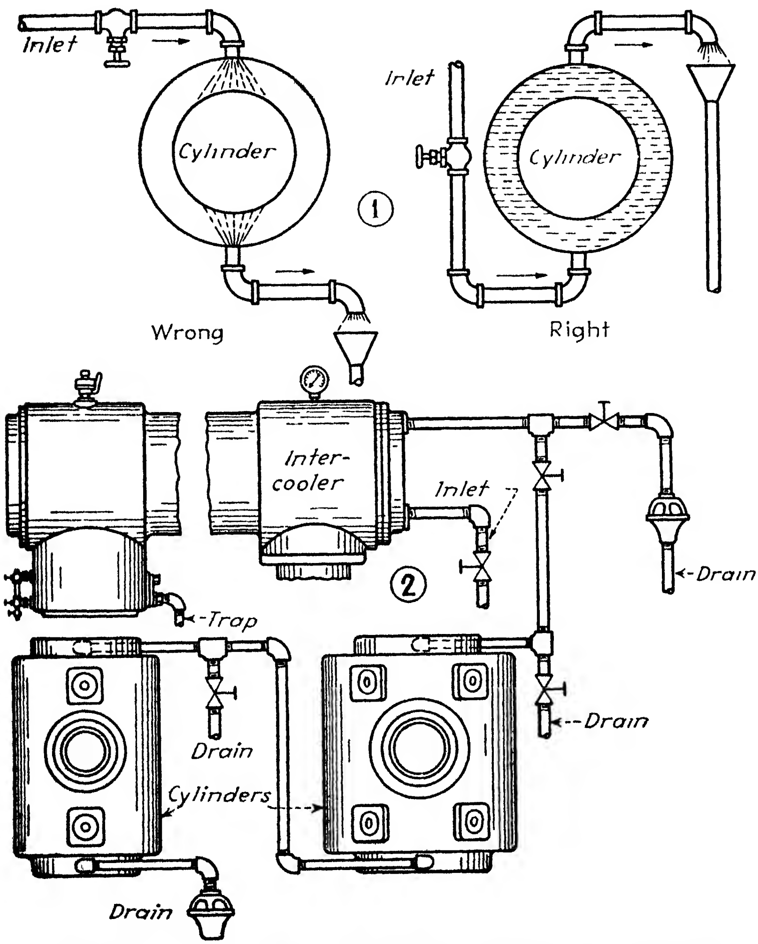


FIG. 188.—Correct cooling-water flow for cylinders and intercoolers.

keeps the jackets full at all times and helps to eject entrapped air. Insert check valves in supply lines branching from a cylinder to the intercooler above to prevent the intercooler's draining down through cylinder passages when the water is turned off. Extremely cold cooling water, having a temperature lower than the incoming air and flowing freely through the cylinder jackets, causes moisture condensation on the cylinder walls. This condensation destroys the lubricant and causes rapid wear. One suggested piping arrangement feeds the cold water through the intercooler to the low-pressure cylinder and then to the high-pressure cylinder, (2) in Fig. 188. Because the intercooler requires more water than the cylinders, a tee, inserted at the low-pressure-cylinder inlet, leads to a discharge valve through which the excess water goes to waste.

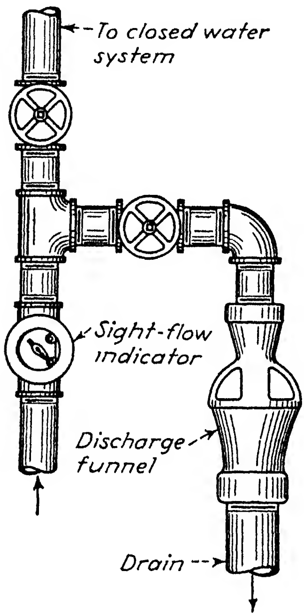


FIG. 189.—Use this piping arrangement for a closed cooling system. (*Courtesy of Compressed Air Institute.*)

On a closed cooling-water system the jacket control valve should be placed on the inlet side; however, if it is located on the outlet side, install a relief valve between it and the cylinder to prevent excessive pressure from building up in the jackets. Jacket water pressure should never exceed 40 psi unless approved by the manufacturer. Install sight-flow indicators on the water discharge to show positively that the water is circulating. Also bypass the outlet to an open funnel (Fig. 189) so that the cylinders and coolers can be tested at frequent intervals to make sure that there are no leaks in the jacket or in the intercooler tubes. If the water discharge is open to an overflow funnel and the valve on the discharge line is shut off, any air leakage into the water spaces will be discovered at once by air flowing out with the water.

To facilitate cleaning cylinder jackets and cooler tubes, insert a tee between the control valve and the cylinder or cooler inlet to serve as an air-hose connection. Discharging alternate slugs of water and air through the passages stirs up and flushes out sediment. Never apply more than 40 psi air pressure to either the cylinder jackets or cooler tubes.

Complete Installation.—Make your compressor installation complete by providing adequate protective devices and indicating or recording instruments (Fig. 190). They are especially needed on a compressor in a remote spot without regular attendants, where well-chosen appliances working together can provide trouble-free operation. Local conditions may justify using all the following devices:

Filters.—On suction lines, filters prevent dust and dirt from the atmosphere getting into the machine. On gas compressors, filters or scrubbers eliminate the possibility of sand and grit being carried into the compressor and damaging valves, piston rings, piston-rod packing, and cylinder walls. Obviously, since it is vital to filter air or gas passing into the machine, it is equally important to clean all welding shot, scale, and other foreign matter from a new suction line; otherwise, such dirt would be a source of trouble shortly after starting up. Far too many compressors are damaged during the first weeks of operation; even tools and overalls sometimes find their way inside machines.

Safety and Relief Valves.—Install a spring-loaded safety valve on the compressor side of the by-pass to avoid damage in case the valves are not properly opened. Relief valves, of course, are required on steam-driven compressors at both ends of the steam cylinders, as protection against the gradual accumulation of condensation. These relief valves, however, are not large enough to take care of water slugs, such as occur from poorly designed steam lines having low spots without proper condensate drains.

A steam-driven compressor having compound cylinders needs a safety valve on the steam receiver, set for a pressure slightly greater than normal. Any valve popping gives ample warning of steam-cutoff misadjustment or of something decidedly wrong with the engine.

Drains.—Compressor intercoolers condense considerable moisture in the air, which must be drained instead of being allowed to pass to the next cylinder. Larger compressors frequently carry automatic drain traps as standard equipment for carrying away this condensate. Purchased traps for smaller machines and aftercoolers make it unnecessary to depend upon the operator to drain the condensate manually. Drains from the steam receiver and cylinders of steam-driven compressors require automatic drain traps.

Oil Filters.—Most of the larger compressors use force-feed lubrication for the running gear, in which all frame bearings are

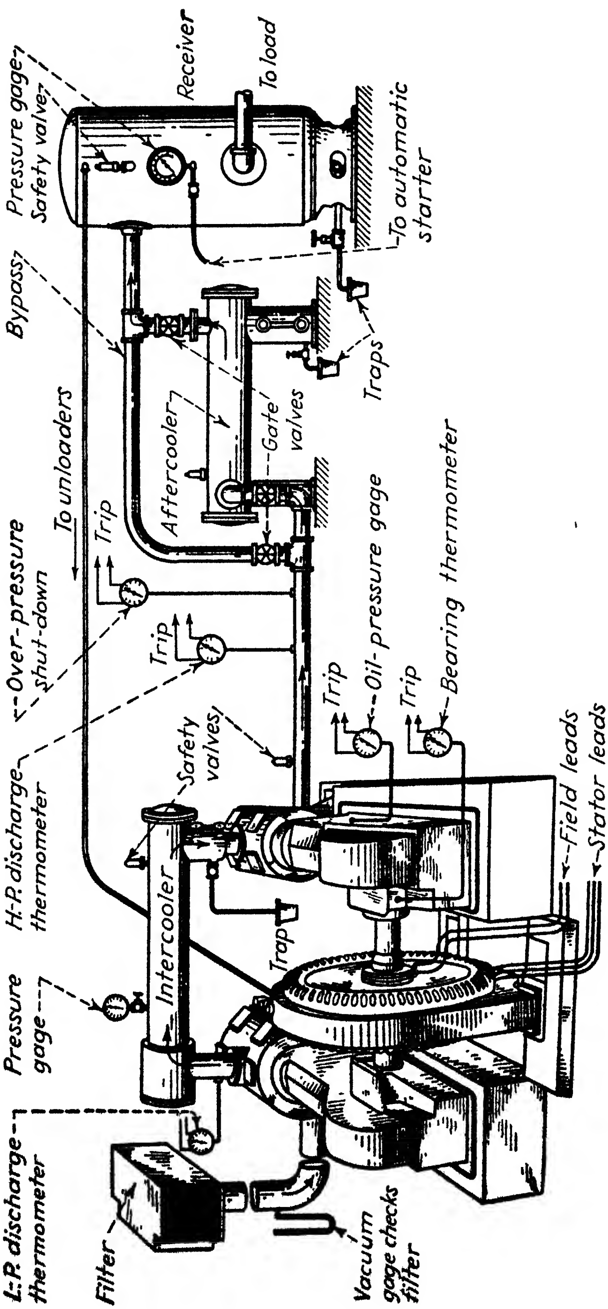


FIG. 190.—Complete protection is provided by these safety devices. (*Courtesy of Power.*)

lubricated under pressure through drilled oil channels. Passing the oil through an approved filter keeps dirt out of the bearings Such filters are frequently standard equipment on compressors having full-pressure oiling systems that provide definite protection to these parts.

Overspeed Shutdown.—A steam- or gas-engine-driven compressor, being a variable-speed unit, is frequently provided with an overspeed trip. Such a machine has a load-control governor, but if this should stick and the machine overspeed, the tripping

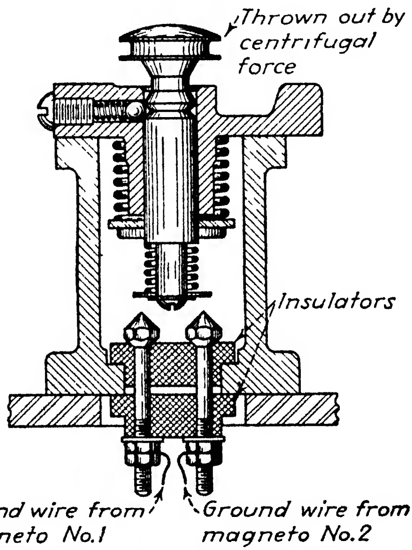


Fig. 191.—Overspeed safety stop opens magneto circuit.

device would shut down the compressor. In most gas-engine and duplex steam-driven compressors, this device is standard equipment. It either opens the magneto (Fig. 191), shuts off the gas supply to the engine, or releases a latch and closes a butterfly valve in the steam-engine supply line. Similar devices for single-cylinder and smaller duplex steam-driven compressors may be purchased separately.

Oil-failure Shutdown.—Larger compressors fitted with pressure oiling are frequently provided with an oil alarm or shutdown system. Oil-pressure failure shuts the unit down and thus injury to the bearings is avoided. When shutdown is undesirable for installations that serve process work, an alarm can warn the operator that something is wrong with the oiling system. Figure 192 includes an overspeed and oil-failure trip.

Jacket-water Valve.—In many plants, jacket and intercooler water must be turned on manually, just as a steam-driven compressor is turned on or an electric motor started manually. An automatic water valve, to shut down the unit if water pressure fails, may be operated by either pressure or temperature. Figure 193 shows a homemade water-failure alarm.

Excessive-pressure Shutdown.—A pressure switch shuts down the compressor if the governor sticks and allows excessive dis-

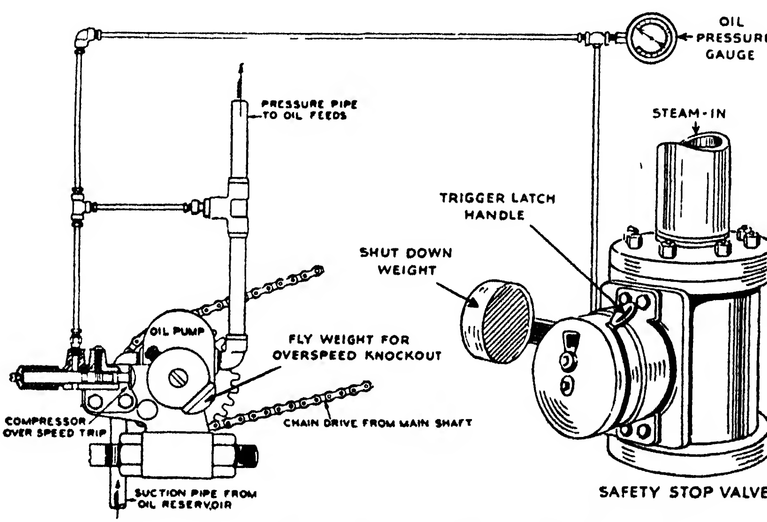


FIG. 192.—Loss of oil pressure or overspeed closes the steam valve.

charge pressure to build up. For a motor-driven machine, this simply means that a pressure switch is set to trip at a higher pressure than usual for a standard compressor regulator. For a gas-engine-driven compressor, the switch, set at a higher pressure than the governor, grounds the magneto. You can provide protection for a steam-driven compressor in connection with the overspeed-shutdown device.

Excessive-temperature Shutdown.—Under normal conditions, a compressor seldom requires protection against possible excessive discharge temperature other than the operating crew's usual inspection and maintenance. However, in many abnormal operating or pressure conditions, protection from excessive discharge temperatures is desirable. Where a compressor is visited by an attendant at infrequent intervals, a thermostat in the discharge line to shut it down is advisable. On a motor-driven compressor, this device opens the control circuit of the driving motor; on a gas-engine compressor, it may ground the ignition or operate to close the valve in the gas-supply line. On a steam-driven compressor, the thermostat can function through the over-speed shutdown, if one is provided.

Main-bearing Protection.—When motor-driven compressors do not have a full-time attendant, the main compressor bearings can be protected by thermal shutdown devices. These open

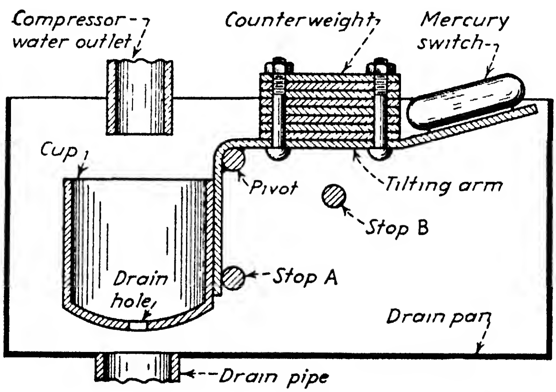


FIG. 193.—Homemade cooling-water-flow alarm. (*Courtesy of Power.*)

the control circuit and shut the compressor down if the temperature of the bearings becomes excessive. This device may be installed regardless of the type of oiling system.

Discharge Thermometers.—On the discharge of each compressor cylinder, thermometers warn of excessive temperature within the cylinder and thus indicate that something is wrong with valves or piston rings. This requires a thermometer well in the pipe near the compressor cylinder. The thermometer may be either an indicating one, which must be read at intervals, or a recording one, which provides a permanent temperature record throughout the operating period. Thermometers are not so necessary on low-pressure compressors but are valuable when machines operate at high pressures.

Multistage Temperature Protection.—On multistage compressors of two, three, four, five, or six stages for high pressures, a thermometer on each cylinder discharge is the best means of determining whether the compressor is functioning normally and safely. Recording the operating temperatures and pressures of each stage on an engine-room log chart at definite intervals serves the same purpose as taking the temperature and pulse of the human body. Indicating thermometers are useful, but recording ones give a definite and permanent record.

Recording Discharge Gage.—A record of the pressure variation in the discharge line throughout the day is often desirable. A recording pressure gage provides this.

CHAPTER VII

COMPRESSOR OPERATION

Careful Attention Prevents Damage.—Before placing a new or repaired compressor in operation take these precautionary measures to prevent machine damage. If you do not already have instruction booklets, secure them from the compressor manufacturer.

First, inspect the intake filter and see that all essential parts are in place. Clean the intake piping of all foreign matter such as dust, welding beads, and rust particles. Clean the interior of the frame, crankcase, and oil reservoirs, using an air jet to blow out oil passages and clean rags to wipe out basins (Fig. 194). Do not use cotton waste, as linters will adhere to metal parts.

Next fill all bearings, lubricators, and reservoirs, pouring the oil over the bearing journals so that every moving part receives a good wetting. A careful mechanic applies a light oil film to all bearings, cylinder walls, and piston rods during assembly. After filling the cylinder lubricator, disconnect the oil piping at the cylinder and operate the lubricator until oil issues from the open end (Fig. 195). Then reconnect the pipe and force several drops of oil into the cylinder. After the compressor operates and the lubricating pump and piping fills, bring the oil in the reservoir up to normal level.

Remove all tools and blocking and inspect the unit thoroughly for loose parts and foreign objects. Turn on the cooling water and check its flow through all jackets and coolers. Trace out the piping. The water supply must enter each cylinder and cooler at such a point that complete filling occurs before any water discharges. Look for cross connections in the piping that might allow some part to empty during normal water shutoff.

Turn the unit over several times by hand, this will reveal tight bearings as well as obstructions interfering with compressor movement. If no trouble appears during hand turning, apply driving power momentarily and let the machine coast to rest.

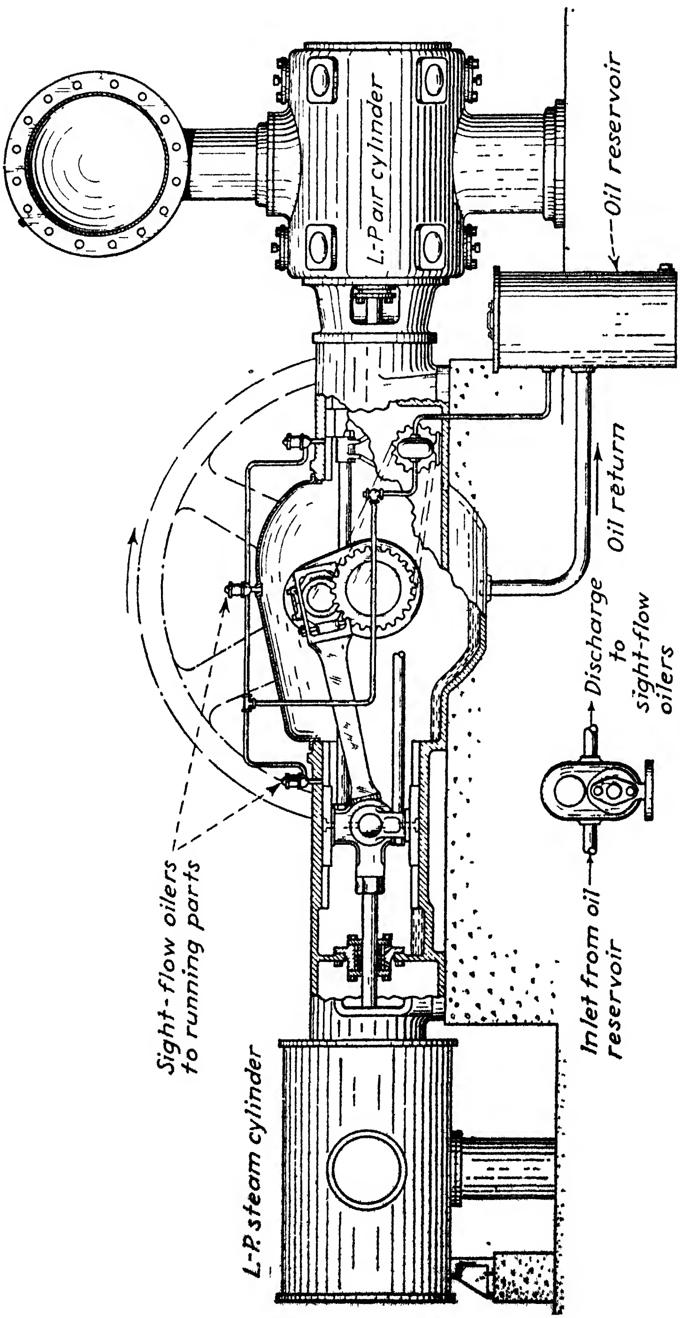


FIG. 194.—Clean the oil piping and reservoirs before filling with new oil.

(The unit should be completely unloaded on the air end and the piston-rod packing should be adjusted loosely.) Close observance during the coasting period will reveal any excessive tightness in the moving parts. The time that the unloaded machine continues to roll after pressing the motor-stop button gives a fair indication of no-load friction; if no trouble is in evidence, the unit can be run without load.

Operate variable-speed machines slowly, gradually increasing at reasonable intervals to full speed; constant-speed machines must come to full speed immediately. Bring the unit to rest

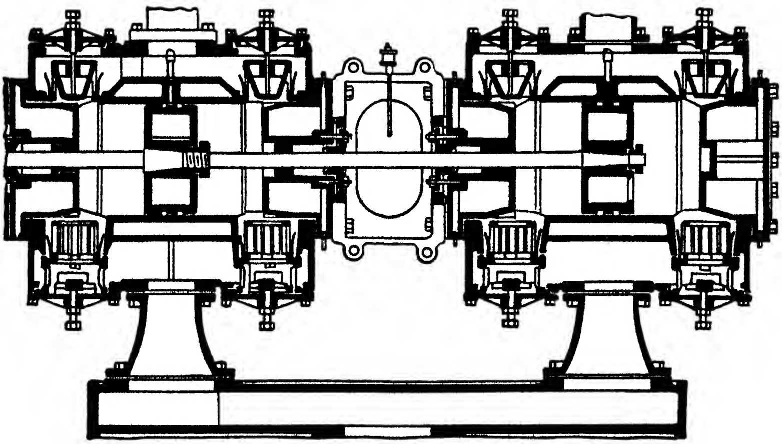


FIG. 195.—Oil supply for tandem cylinders and piston rod.

within 5 min and inspect the bearings and other moving parts for heating. Adjust the cylinder lubricator to feed not less than 2 drops per minute during the breaking-in run.

After running from 1 to 2 hr with inspection stops at 15-min intervals, apply a partial load. Continue the periodic shutdowns for inspection with a slight increase in load on each restart. During the running-in periods the crossheads, main and connecting-rod bearings, and piston-rod packings may require some adjustment. Use particular care with metallic or carbon rod packing (Fig. 196). Remove bearings that run hot, examine them, and scrape them to a proper fit.

The entire breaking-in run should consume a minimum of 4 hr., and to do the job right requires a run of several days at light load; larger machines require a longer time. The initial loading of automatic clearance-pocket-regulated compressors requires blocking the clearance valves shut unless sufficient air pressure from an outside source is available to hold them closed. After several hours' operation, remove the cylinder heads and inspect cylinder-wall lubrication; continue these inspections daily until the correct oil feed is determined. After the first day's run, drain and renew the bearing and reservoir oil.

The importance of a break-in run cannot be stressed too strongly. The time and care spent in giving the running surfaces a polished finish pay dividends by increasing compressor life. After the initial run, compressor operation resolves itself

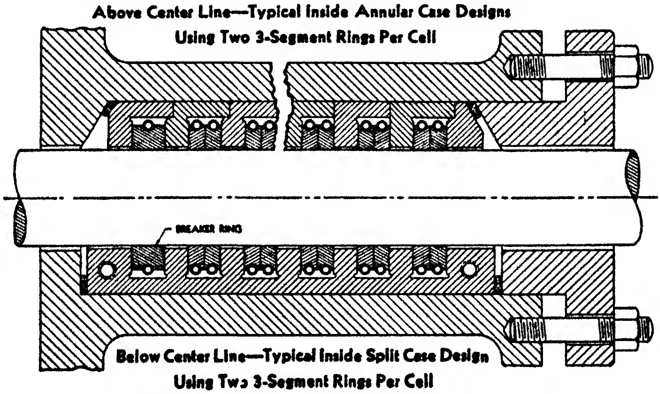


FIG. 196.—Cross section of metallic packing.

into maintaining a clean air supply, feeding sufficient cooling water, and supplying adequate lubrication.

Vertical Locomotive Compressor.—Refusal to start may be caused by insufficient oil, failure of the lubricator to feed properly, or the fact that the oil has been washed away by condensation (Fig. 197). Leaky piston rings in the small end of the main-valve piston or the accumulation of rust during the time the compressor has lain idle may also cause failure to start.

If the compressor groans, the air or steam cylinder needs oil, or the piston-rod packing is dry, causing the rod to bind.

Uneven strokes of the compressor can come from sticky air valves, improper lift of the air valves, plugged discharge-valve passages, leaky air valves, or binding of the reversing rod.

Lack of capacity comes from leakage past the air piston rings caused by poor fit or wear in the cylinder or ring, dirty valves and air passages, or a clogged air-suction strainer. To determine which is causing the trouble, obtain about 90 psi air pressure, reduce the speed to 40 or 60 single strokes per min, then listen at the air inlet and note if air is drawn in only during a portion of each stroke, or if any blows back. If the latter, an inlet valve is leaking. If the suction does not continue until each stroke is

FIG. 197.—Vertical steam-driven compressors mount on engine-room wall.

nearly completed, air is leaking past the piston rings or back from the main reservoir through the discharge valves. Erratic compressor action indicates worn valve parts.

Compressor heating comes from clogged air passages, leakage past air piston rings, or the discharge valves having insufficient lift. Pounding will be caused by a loose air piston or loose reversing plate or the compressor may not be well secured to its foundation, or the reversing rod or plate may be so worn that the compressor motion is not reversed at the proper time.

The compound compressor (Fig. 198) combines maximum capacity and highest efficiency by compounding both the steam and air ends to the extent that, while this compressor has a capacity over three times greater than the single-stage unit, it consumes but one third the steam per 100 cu ft of air compressed. The cross-compound compressor is an arrangement of two standard single-stage compressors, actuated by the same controlling mechanism, with pistons moving uniformly in opposite directions.

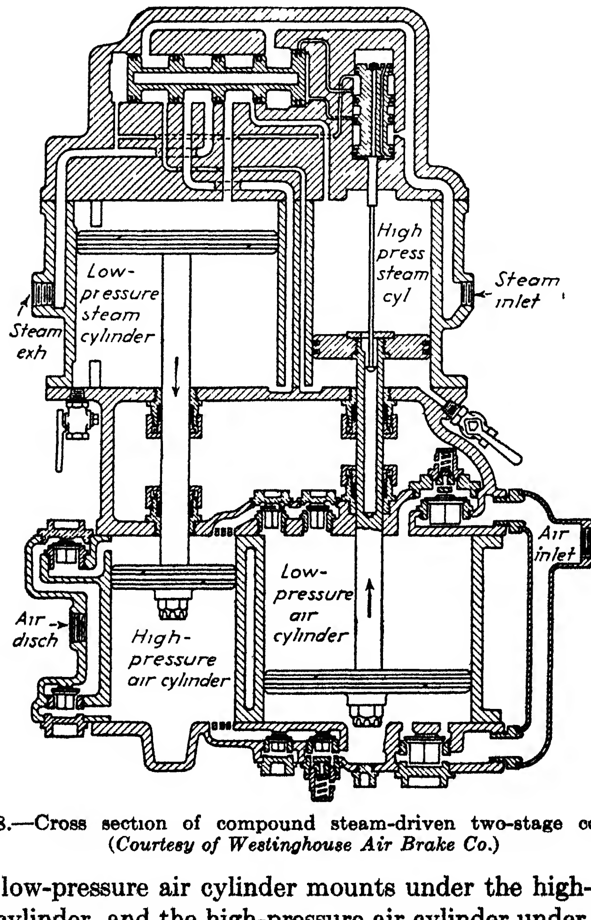


FIG. 198.—Cross section of compound steam-driven two-stage compressor. (*Courtesy of Westinghouse Air Brake Co.*)

The low-pressure air cylinder mounts under the high-pressure steam cylinder, and the high-pressure air cylinder under the low-pressure steam cylinder.

Compressor Lubrication.—Mechanical force-feed oiling, splash oiling, gravity feed, or drop oiling provides general lubrication for bearings, crossheads, and metallic piston-rod packing. Whatever the method, hold wear to a minimum by keeping the oil level at the proper height in the splash reservoir (Fig 199) and lubricators. Changing the oil at frequent intervals (monthly or quarterly) and cleaning the reservoirs, pumps, oil passages, and piping add to the bearing life by furnishing a good supply of clean lubricant

Single-acting compressors use crankcase oil to lubricate the cylinder walls as well as the crankshaft bearings Cylinder lubrication seals the piston and rings against air leakage and

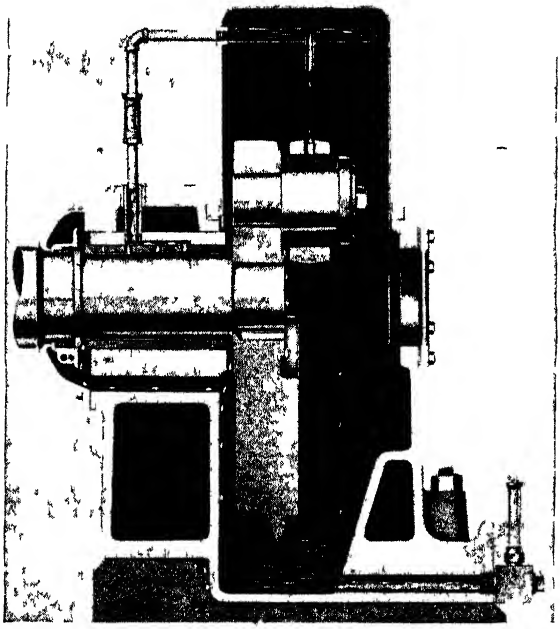


FIG 199 —An overhead basin supplies oil to the main bearing and crank

reduces sliding friction and wear Carrying too high a level splashes excessive oil up into the cylinder cavities and increases oil leakage past the rings. Continual splashing action of the connecting rods causes aeration of the oil, with resulting oxidation.

Double-acting compressor cylinders are lubricated by oil fed from pressure lubricators. Cylinder-oil feeding rates depend on compressor size. While approximate rates serve as a guide, the only accurate way of determining how much oil to feed is to inspect the cylinder and valves. If these parts carry enough oil to give the metal a wet appearance or if a slight amount remains on the fingers after wiping them across the cylinder surface, the lubricator is set properly. Sizable oil pools lying in cylinder recesses, clearance pockets, and valve cages show that the oil feed can be reduced. In checking cylinder appearance, do not overlook the fact that oil normally carried on the piston between rings will drain down on the bottom of the cylinder during extended shutdowns. Excess oil in the compression chamber of a singleacting machine lubricated by crankcase splash oil indicates worn and leaking piston rings.

Buy a good grade of cylinder oil recommended by the compressor manufacturer and do not overfeed, because excess oil and dirt under high pressures in the presence of concentrated oxygen oxidizes rapidly. Internal deposits do not necessarily indicate an excess of oil; dust and dirt in the intake air may be partly to blame. Laboratory tests show that only about 10 per cent of a good compressor oil evaporates in a well-cooled compressor cylinder. This produces approximately 0.90 oz of carbon during a year's operation of a large compressor. If heavy deposits form, look for dirty intake air or check on the kind of oil being used.

TABLE 24.—GALLONS OF WATER PER MINUTE PER 100 CFM OF FREE AIR

Water temperature	60 F	80 F
Aftercooler or intercooler separate (80–125 lb 2 stage compression)	1 0 –1.2	1.4–1.6
Intercooler and jackets in series (80–125 lb two-stage compression)	1.4 –1.6	1.9–2.1
Aftercooler (80–125 lb single-stage compression)	1.0 –1.2	1.4–1.6
Both low- and high-pressure jackets with water supply separate from intercooler (80–125 lb two-stage compressor)	1.85–2.0	1.2–1.4
Jacket for single-stage compressor		
40 lb air pressure	0.5 –0.6	0.7–0.9
60 lb air pressure	0.6 –0.7	0.8–1.0
80 lb air pressure	0.7 –0.8	0.9–1.1
100 lb. air pressure	0.8 –0.9	1.0–1.2

Eliminate possible explosions by using high-quality oil in correct quantities, by frequently cleaning the valves, clearance pockets, pipe lines, and receivers, and by supplying sufficient cooling water to the compressor and coolers.

Cooling-water Requirements.—Cooling water circulated through the cylinder jackets serves to hold cylinder-wall temperatures below the oil vaporization point, helping to maintain an oil film. Cooling action in intercoolers reduces the power required for compression and causes condensation of moisture. Table 24 gives the recommended amounts of water for cooling purposes. Always turn on a liberal supply when starting the compressor.

Water that carries sediment or scale-forming material will soon clog pipes and cylinder water jackets, causing high air temperatures. Scored cylinders and cracked cylinder heads have resulted from lack of cooling water when the machine was started. The water should discharge through a sight-feed glass or into an open funnel in plain sight of the operator. Portable and other compressors subjected to freezing weather need antifreeze solutions in their cooling systems. If the manufacturer's recommendations are not at hand, use the percentages given in Table 25. Check the system for leaks, especially at the pump packing (Fig. 200), before adding the solution.

TABLE 25.—RADIATOR ANTIFREEZE SOLUTIONS

Pure methyl wood alcohol	Denatured wood alcohol	Ethylene glycol	Radiator glycerin	Freezes at deg	
				F	C
Percentages by volume					
13	17	16	37	20	−7
20	26	25	55	10	−12
27	34	33	70	0	−18
32	40	39	81	−10	−23
37	46	44	92	−20	−29
40	53	48	100	−30	−35

When using parallel piping connections, regulate the water flow to the cylinders so that the discharge-water temperature does not go below 100 F or above 160 F. Supply the maximum quantity of water possible to the intercooler; sufficient water should flow through the aftercooler to lower the air temperature as much as possible. Some aftercoolers will cool the air to within 2 F of the cooling-water temperature. Maximum cooling should take place in the intercooler and aftercooler. Do not allow water to circulate through the cylinder jackets and intercooler when the compressor shuts down, because condensation will destroy the cylinder oil film.

Inspect the Air Intake.—Clean intake air reduces wear on cylinder walls and pistons, minimizes dirt deposit on valves, and causes less contamination in process work. Cool air compresses with less power than warm air. Excess moisture must be separated after compression takes place. As these conditions

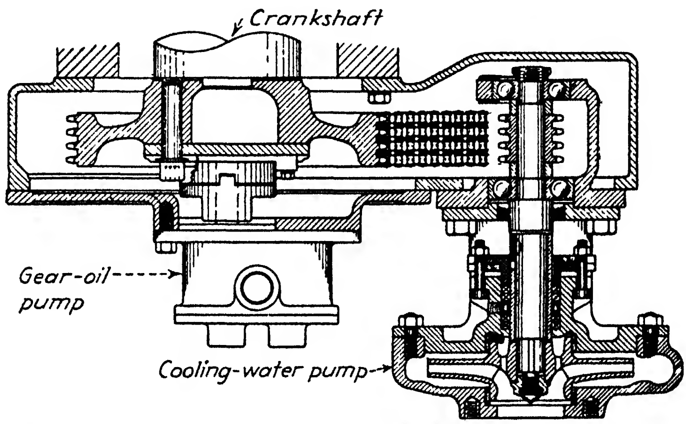


FIG. 200.—A chain-driven built-in cooling-water pump for a diesel-engine-driven compressor.

all affect the operation of a compressor, select the air source carefully. Figure 201 shows a homemade moisture separator.

Excessive moisture and corrosive gases in the air cause valve trouble, a condition generally localized in the second stage of a two-stage compressor. The air, after passing through the first stage of compression and then being cooled, becomes saturated with water vapor. Corrosive gas absorbed by this water has increased corrosive action. Consult the manufacturer about protection for unusual conditions.

Dirt or grit in the air not only causes valve trouble, but combines with lubricating oil to form a grinding compound which rapidly wears cylinder liners, pistons, and piston rings. Some of this foreign matter will be carried in the air and cause rapid wear of pneumatic tools. Air-filtering equipment must be kept clean and in place to ensure complete protection to the compressor Several examples will show the value of air filters.

The valves on a 5,000 cfm compressor operating in a railroad shop required cleaning every 2 weeks when the air intake was unprotected. After installing a filter, cleaning frequency was reduced to once every 6 months.

A large compressor-plant engineer saved $2,200, equal to a 79 per cent reduction in previous repair costs, by installing filters.

TABLE 26.—TEST RESULTS OF STEAM-ENGINE-DRIVEN AIR COMPRESSOR

Actual free air delivered, cfm	4,551
Piston displacement, cfm	5,108
Volumetric efficiency, %	89 08
Ihp (steam cylinder)	779 36
Ihp per 100 cfm (steam cylinder)	17 13
Steam consumption, lb per ihp-hr	12 73
Steam consumption, lb per 100 cfm	3 635
Over-all efficiency (isothermal)	70 34
Steam conditions throttle, psi gage	150 92
Steam conditions superheat, F	106 50
Steam conditions exhaust, in Hg abs	4 22
Average intake barometer, psi	12 60
Air discharge pressure, psi gage	101 58
Speed, rpm	179 43
Steam-cylinder bore, in	25
Steam-cylinder stroke, in	20
Low-pressure air-cylinder bore, in	28
High-pressure air-cylinder bore, in.	17
Air-cylinder stroke, in	20
Weight of machine, lb	185,000

Another maintenance saving on two small compressors equaled 62 per cent of the filter cost the first year.

Where to Look for Trouble.—It is vitally important to find and repair compressor troubles before a machine reaches the breakdown stage. Here are practical pointers that tell how to detect the cause of such troubles and guard against extensive damage from them.

1. After a new machine has been well broken in and you know it is in good shape, read the intercooler pressure when the machine is operating at its normal intake and discharge pressure. If the normal intake and discharge pressures vary, take a number of readings at different pressures covering the range of operation Make a record of these readings for future reference. Table 26 shows test results of a steam-driven compressor.

The intercooler air pressure is a good indication of internal trouble in the cylinders. For every multistage compressor there is one correct intercooler pressure if the unit is operating properly. This correct pressure in psi abs can be determined to an approximate value by three methods, as follows:

a.
$$P \times R \times 1.09$$

where P = intake pressure abs
R = low-pressure cylinder diameter squared, divided by the high-pressure cylinder diameter squared.

b. Assume that a compressor takes air at atmospheric pressure and discharges at 250 psi gage. The ratio of compression is

$$\frac{250 + 14.7}{0 + 14.7} = 18$$

Then the intercooler pressure is

$$(0 + 14.7) \times \sqrt{18} = 62.5 \text{ psi abs, or } 47.8 \text{ psi gage}$$

c. Obtain the same result by taking the square root of the product of the initial and final pressures. Then the intercooler pressure is

$$\sqrt{264.7 \times 14.7} = 62.4 \text{ psi abs, or } 47.7 \text{ psi gage.}$$

2. A change from normal intercooler pressure indicates trouble within the machine. In multistage compressors, intercooler pressure when operating at normal full load should equal the theoretical calculated pressure. If there is no explanation for pressure rise, leaky or broken valves in the high-pressure cylinder are indicated. If the pressure falls, look for leaky valves in the low-pressure cylinder. Unloading may change the normal intercooler pressure.

Most compressors today have unloading devices that operate simultaneously on all cylinders, thus maintaining fairly constant intercooler pressure. Process compressors, operating with either a variable suction or discharge pressure, have a fluctuating intercooler pressure. You may use hand-operated clearance pockets on either or both cylinders of such machines, to adjust within a reasonable amount an otherwise varying intercooler pressure resulting from variable intake or discharge pressures.

3. When intercooler pressure indicates a leaky or broken compressor valve, you can tell which one is defective without removing all the valves. Experienced operators usually identify the offending valve by the fact that its cover is somewhat hotter than the others, or by a slight difference in the valve-operating sound, or by both these symptoms.

4. The discharge temperature indicates the compressor's mechanical condition because a compressor, operating under the same air pressure and cooling-water temperature, should register constant discharge-air temperature. If the latter increases while the other conditions remain constant, leakage is definitely indicated—usually of the hot discharge back to an intermediate pressure. This may be either valve or piston-ring leakage.

In multistage compressors, a power-plant log record of interstage pressures and temperatures should be a regular practice, because it definitely notifies the operator when to examine piston rings and valves. An increase in discharge temperature also puts him on guard concerning the mechanical condition of a single-stage compressor.

5. Perhaps the discharge-air temperature increases gradually and the cylinder becomes somewhat hotter than formerly, even though piston rings and valves are in first-class condition and jacket-water temperature is normal.

After years of operation, if jacket-water quality is not particularly good, the jacket cooling effect may have deteriorated considerably. When you use lake or river water that contains sufficient silt or mud to precipitate in the jackets and restrict free circulation, or when the water is very hard, the inside surface of the jacket walls may be covered with a heavy lime or mag-

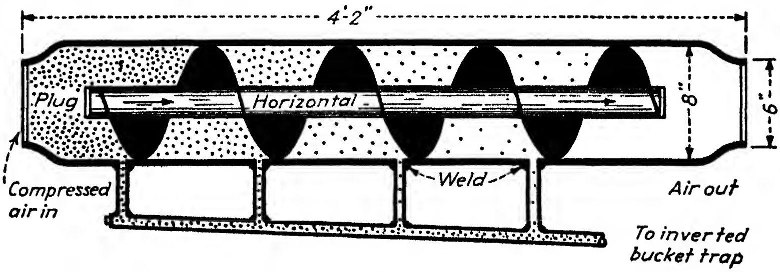


FIG. 201.—Homemade moisture separator for distribution line. (*Courtesy of Power.*)

nesium deposit which interferes with normal heat transfer. The answer, of course, is to clean the jackets thoroughly. In the case of lime deposits, soaking the jackets while hot with caustic or weak muriatic acid may make it possible to scrape out the deposit.

6. You can locate the cause of a loud knock in the cylinder if you can isolate the noise to one end of the cylinder where it does not register back to the crankpin. It may be that the jam screws are not holding a valve case tightly and as a result allow the entire case to pound on its seat (Fig. 195). A valve cage allowed to hammer will break, with the possibility of disastrous results if a piece falls into the cylinder and is hit by the piston.

If the valve cages are tight, the piston may be loose on the rod, or the piston may be hitting the head, either because clearance between them has not been properly adjusted at the two ends or because the clearance space may be filled with a carbon deposit or condensate. Usually, such a pound can be heard at the crankpin, fooling the operator into thinking that the main bearing or crankpin is loose.

7. If the cylinder wall appears dry with a slightly rusty appearance some 15 min after the cylinder head is removed, the cause is lack of lubrication or

possibly washing of the lubricant from the walls. If a normal amount of lubricant is fed, check the cylinder's circulating-water temperature. If the cooling water is much colder than the entering air or gas, sufficient moisture may be precipitated on the cylinder wall to wash off the lubricant. Or, incomplete separation of condensate at the high-pressure-cylinder inlet may cause the trouble. The intercooler, of course, has reduced air temperature to the dew point so that the air is saturated at the high-pressure intake. Any liquid passing to the high-pressure cylinder wears the suction valves seriously and causes rapid cylinder-wall wear by washing off the lubricant. In other cases, sufficient mist to affect lubrication may be carried by the air into the high-pressure cylinder.

8. Use aftercoolers with compressors to remove excess moisture from the air before it reaches distribution lines. As compression takes place, rising

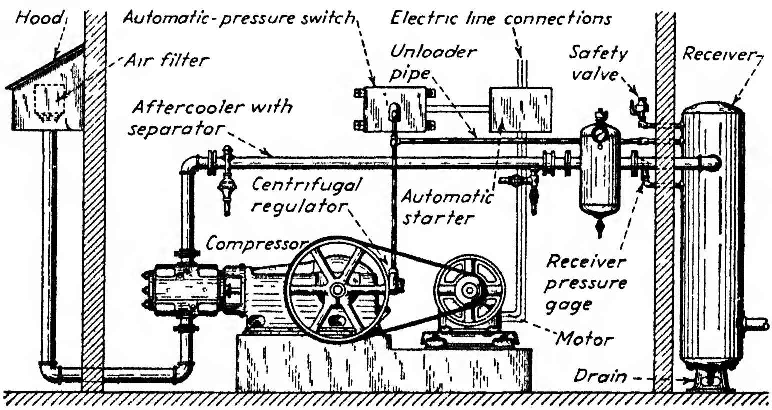


FIG. 202.—Piping arrangement for start-and-stop control.

temperature greatly increases the air's moisture-carrying capacity; discharge air at 100 psi and 300 F can carry many times the moisture in air at atmospheric conditions. As the compressed air cools to approximately atmospheric temperature, this moisture deposits in pipe lines and, aside from being objectionable in the line, is destructive to pneumatic tools or other air-operated mechanical equipment. Besides, there is a great loss in man-hours and power if operators must blow down air lines each time they use air.

Reduction in the temperature of the air and, therefore, its moisture-carrying capacity, is limited by the cooling-water temperature. If the air cannot be cooled to a very low temperature in the aftercooler but subsequently cools to a lower temperature by atmospheric radiation in the system, more water will precipitate. However, most of the water is removed at the aftercooler, and later condensation may be trapped out by placing receivers in the system where the air is used.

Installing an aftercooler as an accessory to the compressor reduces the compressed-air temperature immediately, so that the moisture precipitates

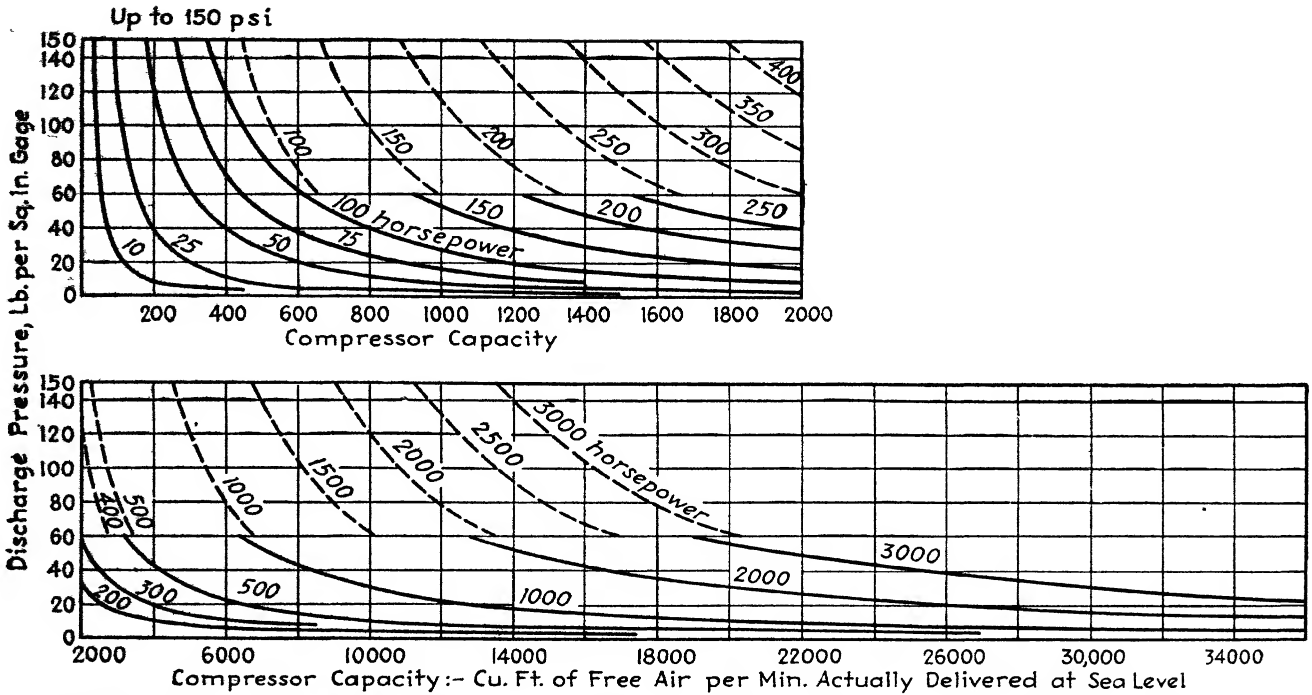


Fig. 203.—Approximate horsepower required to compress air to pressures up to 150 psi. The curves are lines of constant horsepower and are based on compression of air from 14.7 psi abs intake pressure to the final discharge pressure indicated. Allowance has been made for average compression and mechanical efficiencies of reciprocating compressors. Dotted lines are for two-stage water-cooled units. High-pressure units (Fig. 204) are two-, three-, four-, five-, or six-stage, depending on discharge pressure. (*Courtesy of Ingersoll-Rand Co.*)

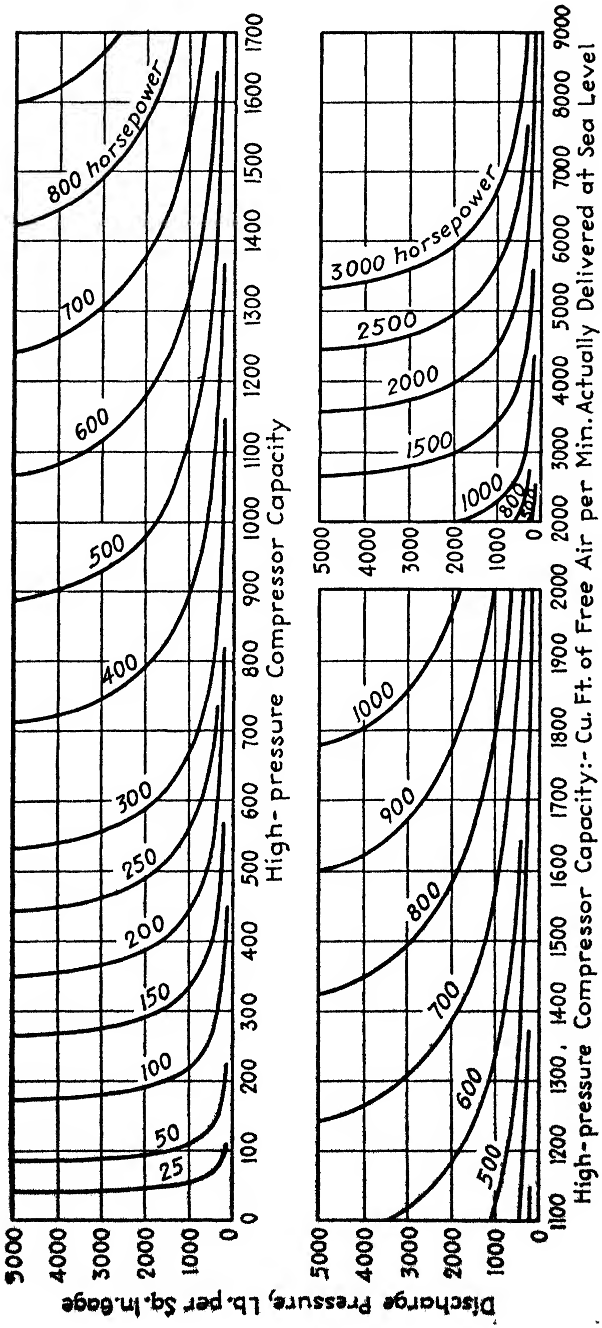


FIG. 204.—Approximate horsepower required to compress air to high pressures. (*Courtesy of Ingersoll-Rand Co.*)

in the aftercooler and receiver, for removal before compressed air enters the system.

9. Plant conditions vary, but usually the best place to locate the aftercooler is directly at the compressor discharge, as near the compressor as conditions permit (Fig. 202). Placing it between the compressor and receiver allows condensate not removed by the aftercooler trap to precipitate out in the air receiver. The aftercooler should be located indoors to prevent freezing, but the air receiver can be placed outdoors.

A by-pass around the aftercooler permits dismantling for cleaning or repair without shutting down the compressor. In all such cases you must, of course, install a relief valve on the compressor side of the by-pass to avoid damage from improper valve manipulation.

10. It sometimes appears that compressor capacity has apparently fallen below that formerly obtained (Figs. 203, 204, and 205). When compressor valves, either suction or discharge, have been removed for cleaning or inspection, take extreme care not to install them upside down. These valves, frequently made interchangeable between suction and discharge, should be stamped to indicate which side must be toward the piston when used in either location. If valves are incorrectly installed, the compressor may unload itself, or it may even be possible that air cannot get out of the cylinder. In the latter case, serious damage results.

11. Cylinders may become too hot for the hand to be held against them because compressing air heats them rapidly. When compressed to 100 psi, the temperature may exceed 300 F, depending upon the cylinder's size. This is not excessive. The hand cannot be held in water at 150 F; so this condition does not in itself mean that the cylinder is too hot or that anything is wrong internally.

A study of the following list of trouble symptoms will assist in diagnosing defective operation.

BROKEN OR LEAKY VALVE STRIPS

The location and amount of leakage may be determined while the compressor is in operation as follows:

1. High-pressure discharge valve
   a. Excessive leakage.
   Intercooler safety valve blows continuously.
   b. Moderate leakage.
   Intercooler safety valve blows intermittently.
   Compressor unloads infrequently.
   c. Light leakage.
   Intercooler pressure runs above 30 psi, but not enough to blow safety valve.
   Compressor unloads almost normally.
2. High-pressure suction valve
   a. Excessive leakage.
   Intercooler safety valve blows continuously.

b. Moderate leakage.
Intercooler safety valve blows during loaded period only.

c. Light leakage.
Intercooler pressure runs above 30 psi, but not enough to blow safety valve.

Compressor unloads almost normally.

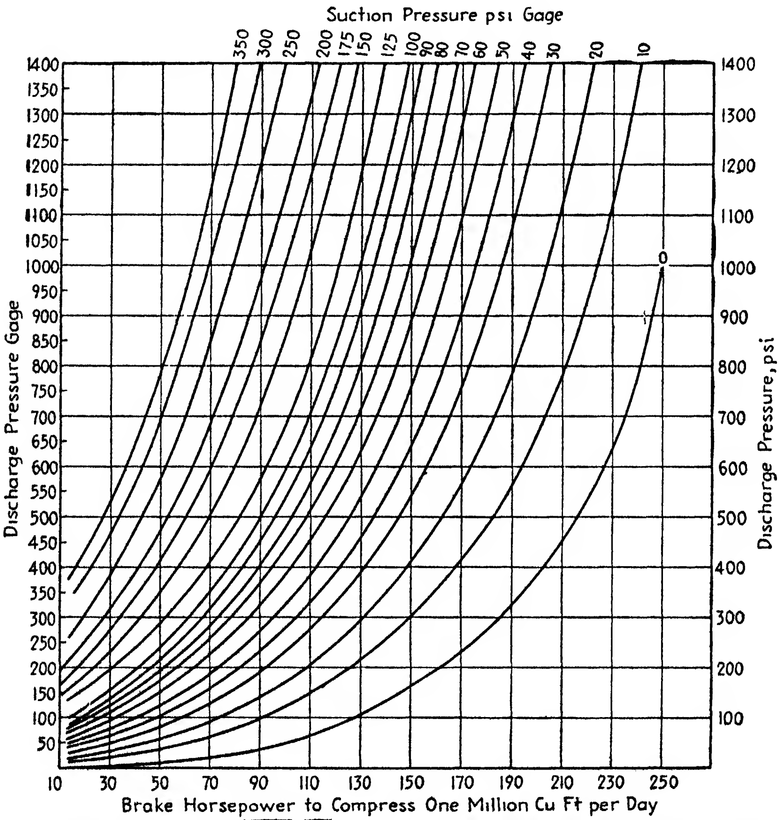


FIG. 205.—Brake horsepower required to compress 1,000,000 cu ft of air per day. (*Courtesy of Cooper-Bessemer Corp.*)

3. Low-pressure discharge valve

a. Excessive leakage.
Little or no intercooler pressure during loaded and unloaded period.
High-pressure cylinder and discharge pipe run abnormally hot. Low-pressure-cylinder head runs abnormally cool.

b. Moderate leakage.
Intercooler pressure runs below 26 psi during loaded periods, and rapidly drains to zero during unloaded periods.

c. Light leakage.
Intercooler pressure runs below 26 psi during loaded periods, and slowly drains to zero during unloaded periods.

4. Low-pressure suction valve
   a. Excessive leakage.
   Same as in 3*a*.
   Air is being blown out through air filter during loaded periods.
   b. Moderate or light leakage.
   Intercooler pressure runs low during loaded periods, but does not drain during unloaded periods. Outward puffs through air filter during loaded periods.

SERVICE CHECK CHART

Should a derangement of some mechanical part of the compressor appear, it can generally be traced to the common causes listed below:

1. Low oil pressure
   a. Low oil level.
   b. Plugged strainer.
   c. Leak in suction or pressure lines.
   d. Worn-out bearings (connecting rod or crosshead pin).
   e. Defective pump.
   f. Dirt in filter check valve.
   g. Broken spring in filter check valve.
2. High oil pressure
   a. Plugged oil-pressure line.
   b. Defective filter mechanism.
   c. Excessive spring tension on filter check valve.
3. Incorrect delivery of lubricator
   a. Dirty or gummed valves.
   b. Broken spring or dirt in check valve at cylinder.
   c. Leak in lines or sight feed.
   d. Low oil level.
   e. Plugged vent in lubricator reservoirs.
4. Overheated low-pressure cylinder
   a. Insufficient cooling water.
   b. Scored piston or liner.
   c. Broken discharge valves or valve springs.
   d. Excessive carbon deposits.
   e. Packing too tight.
   f. Insufficient lubrication.
   g. Deposits of foreign material in cooling jackets.
5. Overheated high-pressure cylinder
   a. Insufficient cooling water.
   b. Scored piston or liner.

   c. Broken high-pressure discharge valves or springs.
   d. Broken high-pressure check valves or high-pressure relief valve.
   e. Packing too tight.
   f. Excessive carbon deposits.
   g. Insufficient lubrication.
   h. Leak in low-pressure inlet-valve unloader.
   i. Deposits in cooling jackets.

6. Water in cylinders
   a. Leak in head gaskets.
   b. Cracked cylinder head.
   c. Condensate caused by too cold cooling water.
7. High intercooler pressure
   a. Broken high-pressure discharge valves or springs.
   b. Defective gages.
   c. Broken high-pressure head gasket.
8. Low intercooler pressure
   a. Broken low-pressure inlet valves or springs.
   b. Leak in intercooler.
   c. Defective or dirty intercooler drain check valve.
   d. Inlet-valve unloader partially closed.
   e. Leak in rod packing.
9. Low intercooler vacuum
   a. Leak in low-pressure unloader.
   b. Broken high-pressure (or low-pressure) inlet valves or springs.
   c. Leak in intercooler or connections.
   d. Defective intercooler check valve.
   e. Leak in rod packing.
10. Knocks
    a. Excessive carbon deposits.
    b. Scored cylinder or liner.
    c. Faulty lubricator.
    d. Foreign material in cylinder.
    e. Incorrect head clearance.
    f. Loose piston.
    g. Burned-out or worn rod bearings.
    h. Worn-out or scored crosshead liner or crosshead
11. Scored cylinder liner and piston
    a. Foreign material.
    b. Lack of lubrication.
    c. Too cold cooling water causing condensate and washing out lubrication.
    d. Excessive heat.
    e. Plugged water jackets.
12. Broken valves and springs
    a. Insufficient lubrication.
    b. Rust.

c. Condensation, or water.
d. Carbon deposits.
e. Foreign material.
f. Incorrect assembly.

13. Control trouble
    a. Inlet-valve unloader stuck open or closed.
    b. Mercoid pressure switches faulty.
    c. Foreign material in three-way solenoid valves.
    d. Vibration of control panel.
    e. Voltage drop or loss of power.
    f. Plugged air-control line or control-line strainer.
    g. Incorrect voltage.
14. Incorrect operation of inlet-valve unloader
    a. Leak caused by excess weight of inlet piping.
    b. Foreign material in guides or seats.
    c. Worn plunger seal.
    d. Broken spring.
    e. Manual shutoff partly closed.

CHAPTER VIII

RECIPROCATING-COMPRESSOR MAINTENANCE

Maintain Full Output.—New reciprocating compressors have a known full-load capacity in cubic feet of free air per minute and require a known power input (Figs. 203, 204, and 205); the ordinary wear of internal parts adds up to one thing—a gradual reduction in capacity. Compressors can be kept in perfect

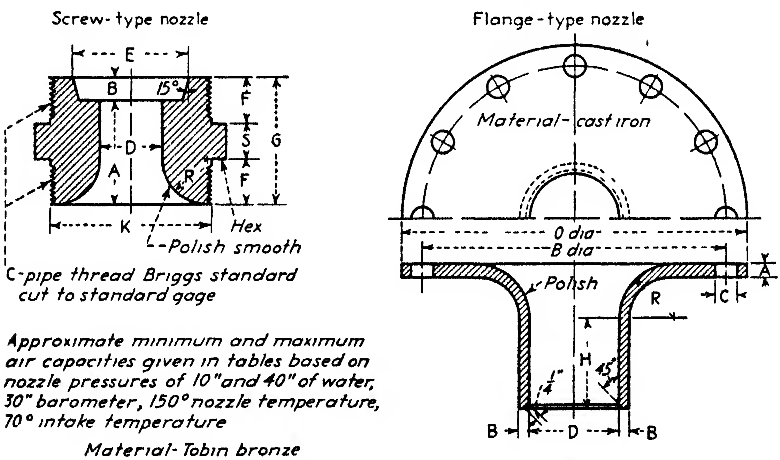


FIG. 206.—Recommended dimensions for standard nozzles taken from the ASME power test code. (See page 216 for dimension tables.)

condition by making frequent inspections and following up with necessary repairs.

Although finding the output of a compressor involves a complicated setup when based on acceptance-test standards, several practical tests will disclose any considerable capacity loss. Some of the methods give only comparative results, and, to get a true picture of performance, one day's test must be compared with previous results.

Pump-up Capacity Test.—A capacity check may be made by the so-called "pump-up method." The air delivered by the compressor is discharged into a closed tank of known capacity (see Chap. XIV, Theory of Compressing Air), the pressure rise in a given time is noted, and temperatures at the beginning and end of the tests are observed. From these data the weight of air pumped into the tank can be computed and expressed as volume delivered by the compressor per unit of time. It must be understood that the results obtained are only approximate, because of errors in computing the volume of the tank and attached piping, and of the difficulty in obtaining the true temperature within the tank.

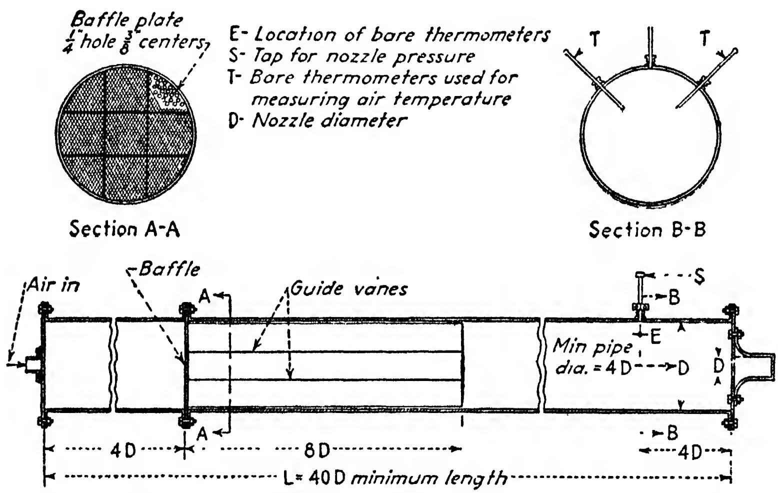


FIG. 207.—Dimensions of nozzle tank and instrument location, ASME power test code.

Knowing that a positive-displacement compressor handles approximately a constant volume of air regardless of discharge pressure, it becomes a simple matter to time the compressor while filling the receiver to a given pressure. Cylindrical tank and elliptical head volumes are given in Tables 27 and 28.

Nozzle Check.—When air to the distribution system can be turned off, a correctly designed orifice (Fig. 206), properly connected to the receiver, checks compressor output. For best results install the orifice at the end of a straight pipe having a length forty times the orifice diameter and a pipe diameter four times that of the orifice (Fig. 207). Use a gate shutoff valve

TABLE 27.—CAPACITIES OF CYLINDRICAL TANKS
Contents of Cylinder per Inch of Length

Dia., in.	Volume, cu ft	Dia., in.	Volume, cu ft	Dia., in.	Volume, cu ft	Dia., in.	Volume, cu ft	Dia., in.	Volume, cu ft
10	0.04545	26	0.3072	42	0.8017	58	1.529	78	2.765
11	0.05499	27	0.3313	43	0.8404	59	1.582	81	2.982
12	0.06545	28	0.3563	44	0.8798	60	1.636	84	3.207
13	0.07683	29	0.3822	45	0.9204	61	1.691	87	3.440
14	0.08908	30	0.4091	46	0.9617	62	1.747	90	3.682
15	0.1022	31	0.4367	47	1.004	63	1.804	93	3.931
16	0.1163	32	0.4654	48	1.047	64	1.862	96	4.189
17	0.1313	33	0.4950	49	1.091	65	1.920	99	4.455
18	0.1472	34	0.5254	50	1.136	66	1.980	102	4.729
19	0.1641	35	0.5567	51	1.182	67	2.040	105	5.011
20	0.1818	36	0.5891	52	1.229	68	2.102	108	5.301
21	0.2004	37	0.6222	53	1.277	69	2.164	111	5.600
22	0 2200	38	0.6563	54	1.325	70	2.227	114	5.906
23	0.2404	39	0.6913	55	1.375	71	2.291	117	6.221
24	0.2618	40	0.7272	56	1.425	72	2.356	120	6.545
25	0.2841	41	0.7640	57	1.477	75	2.557		

ASME DIMENSIONS FOR STANDARD NOZZLES
SCREW-TYPE NOZZLE—(See Fig. 206)

D	*A*	*B*	*C*	*E*	*F*	*G*	*J*	*K*	*R*	*S*	Min cap free air cfm	Max cap free air cfm
1/8	7/32	1 23/32	1 1/2	1 5/16	3/4	2	2 1/4	1 9	3/32	1/2	.988	1.98
3/16	5/16	1 11/16	1 1/2	1 5/16	3/4	2	2 1/4	1.9	1/8	1/2	2.235	4.45
1/4	11/32	1 23/32	1 1/2	1 5/16	3/4	2	2 1/4	1.9	5/32	1/2	3.95	7.91
3/8	1/2	1 5/8	1 1/2	1 5/16	3/4	2	2 1/4	1.9	3/16	1/2	8.89	17.8
1/2	13/16	1 5/16	1 1/2	1 5/16	3/4	2	2 1/4	1 9	5/16	1/2	15.8	31.6
3/4	1 1/4	1 5/8	2	1 5/8	13/16	2 3/8	2 1/2	2 3/8	1/2	1/2	35.6	71.2
1	1 5/8	3/4	2	1 5/8	13/16	2 3/8	2 1/2	2 3/8	5/8	1/2	63.2	127
1 1/2	2 1/2	1/2	3	2 1/2	1	2 3/4	3 3/4	3 1/2	7/8	3/4	119	239
2	3 1/2	1/2	4	3 1/2	1 1/2	3 3/4	4 3/4	4 1/2	1 1/2	3/4	253	506

FLANGE-TYPE NOZZLE—(See Fig. 206)

D	*A*	*B*	*C*	*H*	*R*	*OD*	*BC*	Number of bolts	Standard flange size	Max cap free air cfm	Min cap free air cfm
2 1/2	5/8	1/2	7/8	2 1/2	1.5	13.5	11.75	8	8	780	390
3	5/8	1/2	7/8	3	1.8	13.5	11.75	8	8	1,140	570
4	5/8	1/2	1	4	2 4	16.0	14.25	12	10	2,020	1,010
5	5/8	1/2	1	5	3.0	19.0	17.0	12	12	3,160	1,590
6	3/4	1/2	1 1/8	6	3.6	21.0	18.75	12	14	4,510	2,260
8	7/8	5/8	1 1/4	6	4.8	27.5	25.0	20	20	8,100	4,050
10	1	3/4	1 3/8	8	6.0	32.0	29.5	20	24	12,600	6,350
12	1 1/4	3/4	1 5/8	8	7.2	38.75	36.0	28	30	18,200	9,100

at the receiver. Discharge capacities of different-sized orifices at various pressures are given in Table 29.

TABLE 28.—LENGTH OF CYLINDER WITH VOLUME EQUAL TO ELLIPTICAL HEAD OF SAME DIAMETER

Depth of head, in.	Equivalent cylinder, in.	Depth of head, in	Equivalent cylinder, in	Depth of head, in	Equivalent cylinder, in	Depth of head, in	Equivalent cylinder, in
1	0 66	11	7 33	21	14	31	20 66
2	1 33	12	8	22	14 66	32	21 33
3	2	13	8 66	23	15 33	33	22
4	2 66	14	9 33	24	16	34	22 66
5	3 33	15	10	25	16 66	35	23 33
6	4	16	10 66	26	17 33	36	24
7	4 66	17	11 33	27	18	37	24 66
8	5 33	18	12	28	18 66	38	25 33
9	6	19	12 66	29	19 33	39	26
10	6 66	20	13 33	30	20	40	26 66

If extreme accuracy is desired, conduct the orifice test according to ASME test code requirements. If the test is run during humid summer weather, or if the intake is warm and moist, corrections must be made to compensate for shrinkage in air volume, because moisture condenses out in the aftercooler. When the intake's relative humidity is below 30 per cent, correction for condensation may be omitted.

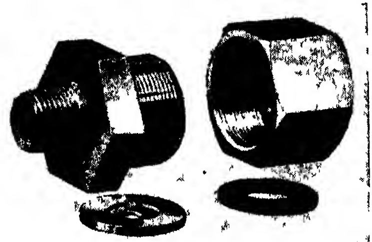


FIG. 208.—Orifice used for testing Westinghouse compressor

Orifice tests on Westinghouse compressors are made at 60 psi receiver pressure. The sharp-edged orifice (Fig. 208) connects to the receiver fitted with a pressure gage. With the orifice open to atmosphere, throttle the steam supply to the compressor until it maintains 60 psi receiver pressure.

The 9½-in. single-stage compressor using a $\frac{11}{64}$-in. orifice should not make more than 120 single strokes per min, and the 11-in.

Table 29.—Discharge of Air through an Orifice

In cfm of free air at standard atmospheric pressure of 14.7 psi abs and 70 F.

Gage pressure before orifice, psi	Diameter of orifice, in.										
	$\frac{1}{64}$	$\frac{1}{32}$	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
	Discharge, cfm of free air										
1	0 028	0 112	0 450	1 80	7 18	16 2	28 7	45 0	64 7	88 1	115
2	0 040	0 158	0 633	2 53	10 1	22 8	40 5	63 3	91 2	124	162
3	0 048	0 194	0 775	3 10	12 4	27 8	49 5	77 5	111	152	198
4	0 056	0 223	0 892	3 56	14 3	32 1	57 0	89 2	128	175	228
5	0 062	0 248	0 993	3 97	15 9	35 7	63 5	99 3	143	195	254
6	0 068	0 272	1 09	4 34	17 4	39 1	69 5	109	156	213	278
7	0 073	0 293	1 17	4 68	18 7	42 2	75 0	117	168	230	300
9	0 083	0 331	1 32	5 30	21 2	47 7	84 7	132	191	260	339
12	0 095	0 379	1 52	6 07	24 3	54 6	97 0	152	218	297	388
15	0 105	0 420	1 68	6 72	26 9	60 5	108	168	242	329	430
20	0 123	0 491	1 96	7 86	31 4	70 7	126	196	283	385	503
25	0 140	0 562	2 25	8 98	35 9	80 9	144	225	323	440	575
30	0 158	0 633	2 53	10 1	40 5	91 1	162	253	365	496	648
35	0 176	0 703	2 81	11 3	45 0	101	180	281	405	551	720
40	0 194	0 774	3 10	12 4	49 6	112	198	310	446	607	793
45	0 211	0 845	3 38	13 5	54 1	122	216	338	487	662	865
50	0 229	0 916	3 66	14 7	58 6	132	235	366	528	718	938
60	0 264	1 06	4 23	16 9	67 6	152	271	423	609	828	1,082
70	0 300	1 20	4 79	19 2	76 7	173	307	479	690	939	1,227
80	0 335	1 34	5 36	21 4	85 7	193	343	536	771	1,050	1,371
90	0 370	1 48	5 92	23 7	94 8	213	379	592	853	1,161	1,516
100	0 406	1 62	6 49	26 0	104	234	415	649	934	1,272	1,661
110	0 441	1 76	7 05	28 2	113	254	452	705	1,016	1,383	1,806
120	0 476	1 91	7 62	30 5	122	274	488	762	1,097	1,494	1,951
125	0 494	1 98	7 90	31 6	126	284	506	790	1,138	1,549	2,023

Courtesy of "Compressed Air Data."

Table is based on 100 per cent coefficient of flow For a well-rounded entrance, multiply values by 0.97. For sharp-edged orifices a multiplier of 0 65 may be used for approximate results.

Values for pressures from 1 to 15 lb gage calculated by standard adiabatic formula.

Values for pressures above 15 lb gage calculated by approximate formula proposed by S. A. Moss.

$$W = 0.5303 \frac{aCP_1}{\sqrt{T_1}}$$

where W = discharge, lb per sec.
a = area of orifice, sq in.
C = coefficient of flow.
P_1 = upstream total pressure, psi abs.
T_1 = upstream temperature, F abs.

Values used in calculating above table were $C = 1.0$, P_1 = gage pressure + 14.7 psi, $T_1 = 530$ F abs.

Weights (W) were converted to volumes using density factor of 0.07494 lb per cu ft. This is correct for dry air at 14.7 psi abs pressure and 70 F.

Formula cannot be used where P_1 is less than two times the barometric pressure.

unit with a $\frac{3}{16}$-in. orifice should single-stroke not more than 100 per min.

The $8\frac{1}{2}$-in.–150 cross-compound compressor, with a $\frac{9}{32}$-in.-orifice and the same receiver pressure, should not make more than 100 single strokes per min; use a $\frac{15}{64}$-in. orifice with the $8\frac{1}{2}$-in.–120 unit at the same number of strokes.

For altitudes over 1,000 ft, the compressors can make five additional strokes per min for each increase of 1,000 ft.

Test the steam end of these compressors according to instructions given in Fig. 209. If the compressor strokes less than 75 per cent of the values shown in the curves it needs repairs.

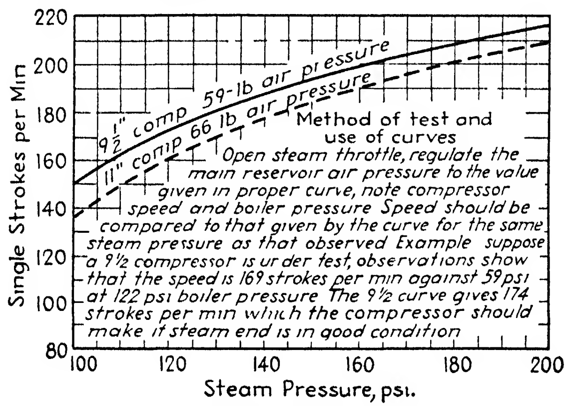


FIG. 209.—Speed curves for testing the steam end of Westinghouse compressor.

Vacuum Method.—Where each compressor has its individual intake gate valve, a practical method is to check for vacuum pulled against the closed intake. Connect a vacuum gage to the intake line and close the intake valve. Note the vacuum that the cylinder will pull when in operation. All compressors having cylinders of the same size and make, if they are in good shape, will pull nearly the same vacuum, while any cylinders that are under capacity will be indicated by the smaller amount of vacuum pulled. Table 4 shows the vacuum pulled by one make of compressor.

When an under-capacity cylinder is located, feel the intake-valve covers. A warm or hot intake cover indicates a leaking valve. While the machine operates on closed intake with the vacuum gage attached, adjust all the valve jam screws, noting any change in vacuum. The valves may not be clamped into the cylinder tightly enough, or they may be clamped so tight as to distort the valve seat and cause leakage.

If the vacuum pulled is still below normal, shut down the unit and take out the questionable valve, or all of them. Note whether or not the intake gate valve leaks. If the gate valve is tight and the trouble is not in any of the compressor valves, then it is no doubt caused by bad piston rings.

Leaking Valves.—Leaking discharge valves can be detected by their extremely high temperature during operation, or by

FIG 210.—True up the valve plates so that they fit the seats perfectly.

opening the cylinder indicator connection valves when the compressor is at rest and subjected to normal operating pressure while the unit is still hot. Test the inlet valves by connecting an air hose to the indicator connection valves and admitting normal operating air pressure to the cylinder, one end at a time. Before applying the air turn the machine until the piston reaches one end so that it cannot move when air flows in behind it.

Leaking clearance-pocket valves may be detected by opening the clearance-pocket drain valves during full-load operation. Air issuing from the drain valve indicates a leaking pocket valve.

These valves can be reseated by applying grinding compound to the seat and disk and then lapping until the score marks disappear.

Cleaning.—Remove all valves for cleaning and inspection every 500 hr or once a month; at this time remove all deposits and check seat leakage. Seats may be tested by clamping the valve assembly in a portable test chamber capable of withstanding full operating air pressure from a hose connection. This method permits checking each individual valve strip or ring after cleaning and assures return to its respective location and position in the valve assembly.

The most satisfactory way to clean compressor valves and air passages is to remove the valves and pistons and wipe the cylinder

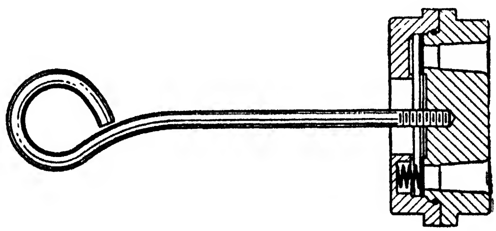


Fig. 211.—Handy valve remover screws into valve seat.

and air passages with clean rags. *Never use gasoline, kerosene, or other volatile solvents inside the compressor or air piping,* because these vapors are highly explosive even at moderate temperatures. Scrape heavy accumulations from air passages.

Dirt on the valves can be removed by brushing, scraping, or light grinding (Fig. 210). Be careful not to scratch the valve seat or plate. Before closing the cylinder apply a light film of oil to the piston and the cylinder wall.

Oil tends to collect in clearance pockets; blow these out regularly when the compressor runs unloaded. Open receivers at least once a year, remove all accumulated oil, inspect them thoroughly, and test the safety valves. Always swab the entire discharge pipe out during the general inspection.

Circulating a soap solution through the compressor, although of uncertain value, is suggested as one way to remove excess oil, carbon, and dirt deposits. This solution can be made from soap powder containing a high percentage of alkali. Introduce the solution through the lubricator or directly into the suction of the compressor while it is running. Ordinarily 0.5 lb of soap powder to 10 gal of water is sufficient, but the soap content may be increased without being harmful. Particular care should be exercised not to feed too rapidly, as the compressor piston clearance is small and too much water will crack a piston or cylinder head. Restore normal oil feed to the compressor for several minutes before shutting it down, as a slight amount of rust pitting caused from water lying in the cylinder will greatly overshadow any benefits from this cleaning.

Valve Wear.—Wear between the valve plate and seat appears as indentations in the seat and shoulders on the plate. These formations start as soon as the compressor first operates, which

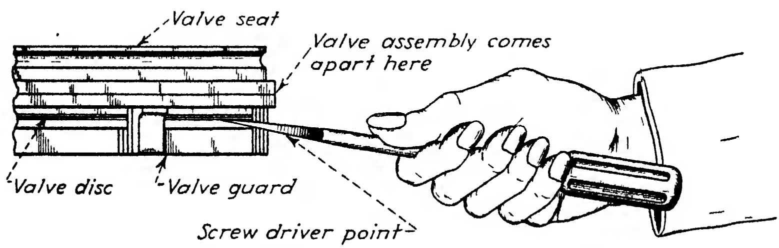


FIG. 212.—Screw driver opens Pennsylvania valve assembly that is held together by a snap ring.

makes necessary the return of each individual plate to its original location after dismantling for inspection.

Inasmuch as compressor valves operate hundreds of times each minute, it will readily be seen that these are subject to considerable wear. When a plate wears to the point where it is less than half its original thickness, it should be changed. The valve seats may also wear slightly after several years. The seat can be refaced and in some cases the recess in which the plate fits must also be cut down an equal amount. Failure to do this increases the valve lift and results in rapid valve wear, with short life.

Valve lift must be kept normal; always measure it before doing any seat machining. Some valve guards bear directly on the seat surface and take up automatically for seat machining; others require removal of metal from the guard-bearing shoulder or spacing washers.

Whenever a valve has overheated, replace all the valve strips and springs when convenient, because excessive temperature resulting from this heat may reduce the life of these parts and result in breakage.

Replacing Assemblies.—The gaskets under the valve seats and also under the valve cover are of a special type, usually consisting of a copper-covered asbestos core. This makes an ideal gasket and generally can be used over many times. Should they become damaged and others not be available, substitutes can be made out of soft copper wire with the ends soldered, or out of $\frac{1}{8}$-in. cord packing.

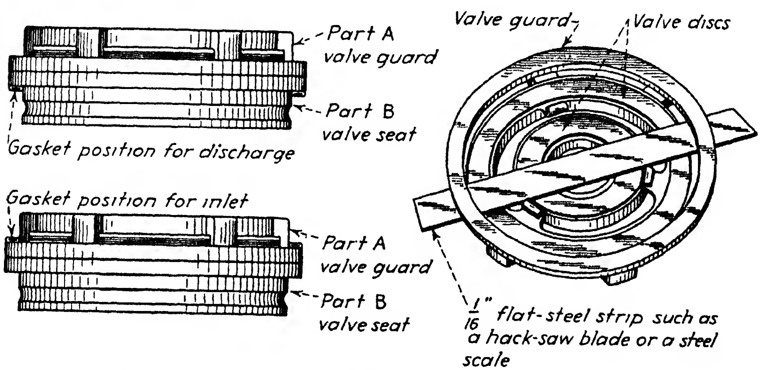


FIG. 213.—Slide a metal strip through guard openings to hold the plate and springs until seat is snapped into place.

When replacing valve assemblies in the cylinder, see that protruding cap-screw heads are not adjacent to the piston and that the valve cage bears uniformly around its gasket; tighten the valve assembly cap (with jam screws backed out $\frac{1}{2}$ in.) and then bring the jam screws tight against the valve assembly. Forcing the jam screw may spring the valve assembly or crack the cylinder inner web.

When taking valves apart it is important that the same parts be reassembled as a unit and not mixed up with other valve parts of the same size; otherwise they will not seat properly.

The joints between seat and stop plates are machined smooth and must not be damaged in handling. Make sure they are clean and smooth, as leaks at some of the joints will cause overheating and cut down the compressor efficiency. If a leak persists at this joint, apply a thin coating of red or white lead. This will stop the leak but will make later dismantling quite difficult.

Valve seats, which have become worn or damaged to a point where they are no longer tight, should be refaced to a new surface. If the valve plates cannot be resurfaced, install new ones.

Some valve plates can be resurfaced on a flat grinder but, when grinding, do not reduce the thickness below that permitted by the manufacturer. A plate ground too thin may break and fall into the cylinder, causing considerably more damage than the

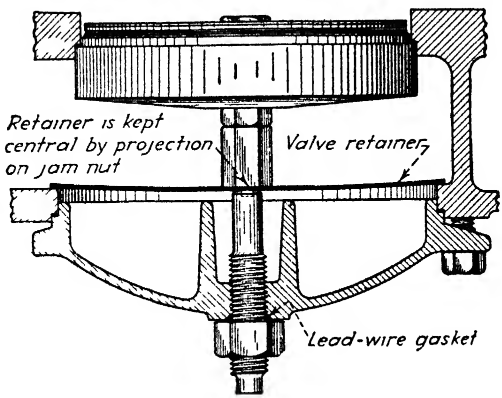


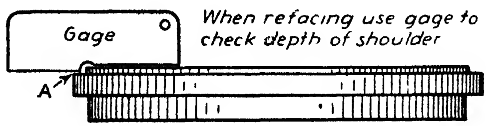
FIG 214 —Retainer plate holds Ingersoll-Rand valve assembly in cylinder until jam screw is tightened.

cost of these small parts. Shoulders on the valve seat can be machined off in a lathe. Lapping the seat and plate will produce an airtight fit.

Dismantling.—A valve extractor (Fig. 211) simplifies removing valve assemblies from the cylinder. One end of this extractor threads into the small hole tapped in the center of the valve seat. No extractor is needed for some inlet valves, which are easily removed from the cylinder when the valve cover is taken off.

To take the Pennsylvania assembly (Fig. 212) apart, lay it on a bench with the valve seat up, then place the end of a screw driver or similar tool between the top of the valve disk and the edge of the guard and pry open.

When assembling the valve, lay the guard flat on a bench with the open side up; place the valve springs into the grooves provided for them, and lay the valve disks on the springs in as central a position as possible. Then place a flat strip of steel, about $\frac{1}{16}$ in. thick, under the valve guard ring and over the disks as shown in Fig. 213 (a hack-saw blade minus its teeth or a steel scale will serve the purpose). The pressure of the steel strip on the valve disks depresses them in their individual pockets and a slight pressure on the valve seat will cause the retaining



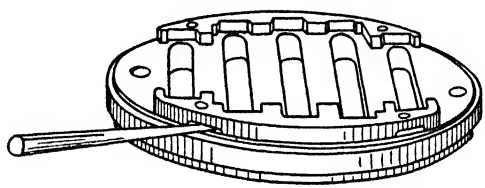


FIG. 215.—Remove guides with a screw driver before machining channel-valve seats.

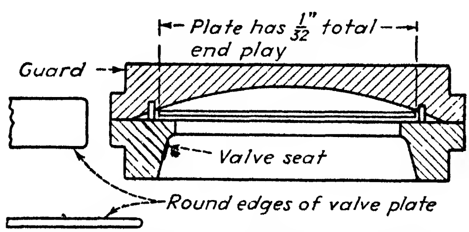


FIG. 216.—Check plate and guard clearance on Worthington valves.

ring to snap into place and lock the guard and seat together. The steel strip may then be withdrawn.

When taking the Ingersoll-Rand channel valve (Fig. 25) apart, proceed as follows:

1. Take out the fillister-head screws and lift off the crab.
2. Take out the two flathead screws, which hold the seat and stop plate together.
3. Lay the valve on a clean flat surface with the valve seat down.
4. Turn the stop plate one-quarter turn to loosen from the seat and lift it off.

5. Remove each spring and its channel separately, clean them, and replace exactly as removed, without turning end for end.

6. Replace the stop plate and flathead screw and tighten. Replace the crab and fillister-head screws. See that lock washers are under the fillister-head screws and that the screws are well tightened.

7. Press each channel separately back against the stop plate to make sure each works freely.

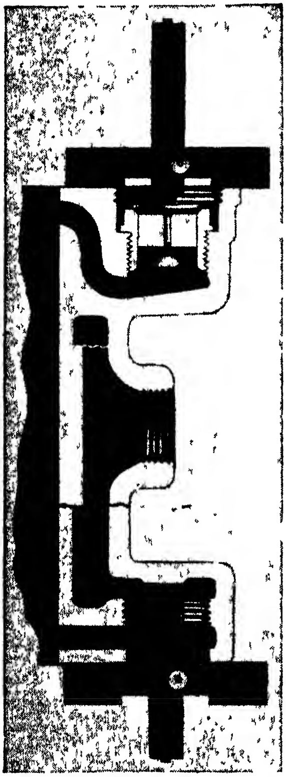


FIG. 217.—Valve gage for Westinghouse compressors.

To replace the valve assembly in the cylinder, proceed as follows:

1. Put each valve assembly in the same cylinder hole from which it was removed, being sure that the gasket is in good condition and properly placed so that the valve will rest squarely on the cylinder seat. When putting channel valves in the cylinder, it is good practice to turn the valve assembly so that the valve ports are parallel with the piston rod.

2 Place the valve cover on the cylinder, being sure that the gasket is squarely in place, and draw the cover nuts down evenly and in rotation. Do not tilt the cover in tightening.

3. Tighten down on the valve-cover jam screw to hold the valve on the seat. Use only a standard-length wrench. Lock the cover setscrew by means of the lock nut.

4. To aid in placing a channel valve in the lower side of the cylinder, a spring retainer (Fig. 214) is bolted to the assembly by means of the fillister-head screw on the crab. This retainer holds the valve assembly in place until the cover and jam screw are put into position.

5. After all valves have been replaced, turn the compressor over one complete revolution by hand to see that everything is clear.

When channel seats must be refaced, proceed as follows:

1. Remove the guides and dowels (Fig. 215) by lightly tapping a sharp-edged flat chisel under the guide to loosen. The dowels are a snug fit in the seat, but are purposely left free enough for easy removal of the guides. Mark the guides for replacing.

2. Using a micrometer (or make a gage, as shown in the figure), measure the distance from the face of the seat to shoulder *A* on which the stop plate

rests. This dimension is important later, as it must be reestablished and held to a close tolerance.

3. Place the seat in a lathe, being careful to grip it just tight enough to hold it without distortion. Remove enough stock from the seat face to provide a smooth surface at the port edges for the channel rest. Take a light final cut and use emery cloth (stretched over a flat true plate) to remove all tool marks.

4. Face off the shoulder *A* on which the stop plate rests to the micrometer reading (or gage) so that this distance will be the same as originally made.

5. Place the guides on the seat and turn in the same position as before removal. Drive in the dowels to hold the guides snug. If one of the dowels has been damaged, it may be replaced by a piece of drill rod or cold-rolled steel of the same length.

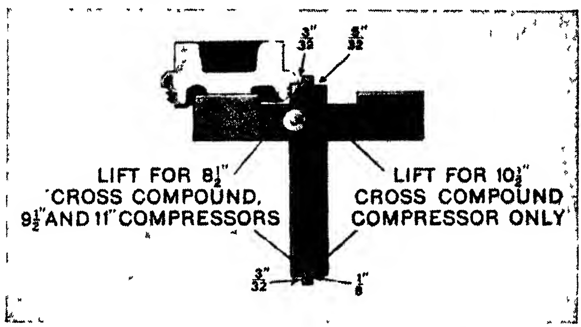


FIG. 218.—Determining lift of upper valve.

6. If the guides are worn, new ones should be put on after the seat is refaced. For best performance, the total side clearance between the side of the channel and the slot in the guide should not be less than 0.006 in. or greater than 0.010 in. The total end clearance of the channel in the guide should be between 0.010 in. and 0.015 in.

When assembling valves, use a string to hold the valve strips and springs in their respective positions against the guard. Check to make sure that the strips and springs are free to move and are not pinched between the seat and the guard.

Worthington valve strips (Fig. 216) are of the correct length and width to fit freely without bagging in the recesses of the guard. No part of any strip should project above the ground face of the guard. Each strip should have a lengthwise lost motion of about $\frac{1}{32}$ in. By holding a straightedge firmly against the ground face of the guard and directly over the strip in its slot, the amount of lost motion can be determined by moving the strip back and forth with the fingers. When in contact with the seat or when flexed against the curved backing of the guard, the strip should move freely and should not bind either lengthwise or sideways.

Westinghouse Compressor.—If it is necessary to replace a broken air valve in this machine, be sure that the new valve has the required lift. When the combined valve and seat wear increases the lift more than $\frac{1}{16}$ in. above standard, the seat is liable to be injured and the valve broken.

To measure the lift of the upper air valve, apply the lift gage to the top flange of the air cylinder, as illustrated in Fig. 217, and adjust the sliding arm until its ends rest against the top of the stop on the valve. Lock it in this position by means of the thumb nut. Then apply the gage to the valve cap, as illustrated in Fig. 218; if the valve has the proper lift, the under side of the valve-cap collar will just rest upon the shoulder of the sliding arm. If the gage arm fails to touch the stop on the valve when the sliding-bar shoulder rests upon the face of the collar, the valve lift is greater than standard by the distance between the gage arm and stop.

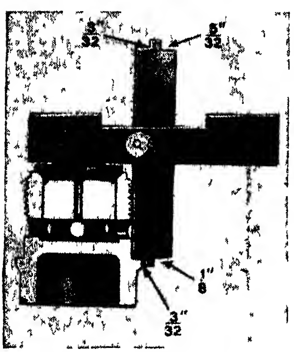


Fig. 219.—Applying gage to lower air valve and cage.

To determine the lower-valve lift, apply the gage to the bottom flange of the air cylinder (Fig. 217) and adjust the sliding arm until its end rests against the stop in the port. Then lock it in position by means of the thumb nut. Then apply the gage to the air-valve cage and air valve (Fig. 219); if the valve has the proper lift the sliding-arm shoulder will just rest upon the upper side of the valve-cage collar. If the gage arm fails to touch the stop on the valve when the sliding-bar shoulder rests on the cage collar, the valve lift is greater than standard by the distance between the stop and gage arm.

Never remove or replace the top steam-cylinder head with the reversing-valve rod in place, as this will almost invariably bend the rod.

To remove or replace the reversing valve and rod, place the steam and air pistons at half-stroke, either by using steam or by removing the bottom cylinder-head plug and forcing them up with a rod or bar. The reversing valve and rod can then be raised so that the button on the rod end can be disengaged

FIG. 220.—Adjust end clearance carefully when fitting new rings. (*Courtesy of American Air Compressor Corp.*)

from the piston plate through an offset hole, and so entirely removed.

Testing Piston-ring Leakage. To check piston-ring leakage in a double-acting compressor, turn the unit by hand until the piston reaches the front end. Then remove the front cylinder head and apply normal air pressure behind the piston through a hose connected to the rear indicator-valve connection. Excessive leakage around the periphery of the piston shows the need of new rings. Distinguish between this leakage and that through ring joints, because air will leak here even when it does not occur between the ring and the cylinder wall.

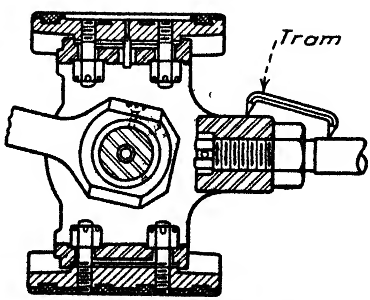


FIG. 221.—Tram marks check piston-rod position.

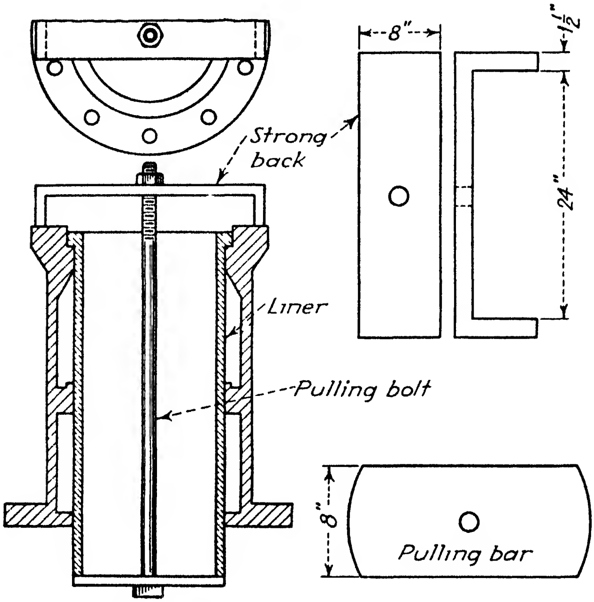


FIG. 222.—Simple pulling device for removing cylinder liners. (*Courtesy of Power.*)

FIG. 223.—Roll new rings around groove to determine fit.

The amount of cylinder out-of-roundness or piston wear will determine whether it is worth while to install new rings without reboring the cylinder and installing a new piston. Most cylinders permit reboring $\frac{1}{8}$ in. on the diameter; consult the manufacturer for anything above this. When the cylinder carries a renewable liner, wear can be corrected by installing a new one.

Ordering New Piston Rings.—Before ordering new piston rings, take an accurate measurement of the cylinder to determine its exact diameter. Furnish this information with the order and specify the ring tension wanted. Standard-size rings may be used in oversize cylinders if the oversize does not exceed 0.003 in. per in. of cylinder diameter.

In other words, a 10-in. ring, which is light tight in a standard gage, will be light tight in a gage which measures 10.030 in., and a 20-in. ring, which is light tight in a standard gage, will be light tight in a gage of 20.060 in. diameter.

The following requirements will cover the installation of standard rings in oversize cylinders:

1. Ring diameters up to 12 in., inclusive: Standard rings are to be used in cylinders up to and including $\frac{1}{32}$-in. oversize.
2. Ring diameters 13 to 20 in., inclusive: Standard rings are to be used in cylinders up to and including $\frac{3}{64}$-in. oversize.
3. Ring diameters 21 in. and over: Standard rings are to be used in cylinders up to and including $\frac{1}{16}$-in. oversize.

When fitting new rings, push them into the cylinder before removing the piston, until they back squarely against the piston face. When in this position measure the ring-joint clearance (Fig. 220); file the ends until the clearance equals 0.003 in. per in. of cylinder diameter. Rings expand under operating temperature, and insufficient clearance causes breakage. After fitting new rings to the cylinder bore, prepare to remove the piston from the cylinder.

Removing the Cylinder Head.—To remove a cylinder head, first turn off the cooling water and drain all water from the jacket. Remove a valve cover from the end of the cylinder and take out the valve. Through the valve opening and between the piston and head place two sticks of wood, one on each side of the piston rod or nut. These sticks should be $\frac{1}{2}$ in. thick and long enough to reach beyond the center of the piston. Back the nuts off $\frac{1}{4}$ in. on the head studs, allowing the head to come loose and yet preventing throwing it off completely. Have a helper turn the compressor wheel by hand and bump the piston very gently against the wood to start the head.

Do not use power to bump the head; turn the wheel slowly by hand. Be sure the wood reaches beyond the center of the piston, as otherwise there is a possibility of bending the piston rod. The less the head is moved by this bumping process, the more chance there is to cut the gasket free without tearing where it may stick to the iron. A knife blade carefully run in between while the head stands away not over $\frac{1}{4}$ in. may save the gasket. If the

FIG. 224.—Constrict ring in groove while inserting piston. (*Courtesy of American Air Compressor Corp.*)

cylinder head remains tight after all visible bolts are loosened, remove the valves and look for additional bolts inside the head.

Checking the Piston Rod and Crosshead.—Check tram marks between the piston rod and crosshead before loosening the crosshead nut (Fig. 221). (Some piston-rod ends are slotted to fit a lockpin in addition to the lock nut.) The piston rod can then be unscrewed from the crosshead. Use a spanner wrench on the piston and milled flats on the piston rod or rod lock nut for turning; do not attach a pipe wrench to the round surfaces of the piston rod. After screwing the piston rod from the crosshead, turn off the lock nut and remove all packing from the gland. Metallic and carbon-ring packing, usually removed by sliding over the end of the rod, must be protected by a sleeve when passing over the threaded portion. Now push the piston to the front of the cylinder to a convenient position for installing the rings. Substantial blocking, level with the lower cylinder surface, placed on the floor at the cylinder opening, catches the piston if it emerges too far.

Removing the Piston from the Rod.—Where pistons must be removed from the rods, it is much simpler to remove the piston while it is inside the cylinder than to take the piston and rod to an outside workbench. This removal can be done in a manner similar to that used when removing the cylinder head by bump-

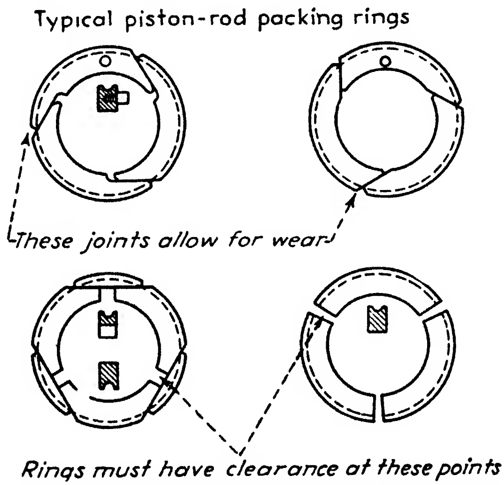


Fig. 225.—Typical piston-rod packing-ring joints.

ing. To remove pistons from the rods, remove two valves from the frame end of the cylinder and insert two pieces of wood on either side of the piston rod. These pieces should be of the same thickness. Turn the compressor over by hand and jar the piston loose. (The piston lock nut should have been previously loosened.)

In removing trunk pistons from vertical compressors, it is first necessary to remove the cylinder head; in the case of three- or four-stage compressors, the third- and fourth-stage cylinders must also be removed. Next turn the compressor by hand to top center. Then remove the lower half of the crankpin bearing, or on some designs the entire crankpin bearing box. The piston and connecting rod can then be pulled up through the cylinder. In removing the pistons from compressors fitted with crossheads and piston rods, the general procedure is to loosen the piston-rod lock nut adjacent to the crosshead, then unscrew the piston rod from the crosshead and lift the piston and rod out of the cylinder.

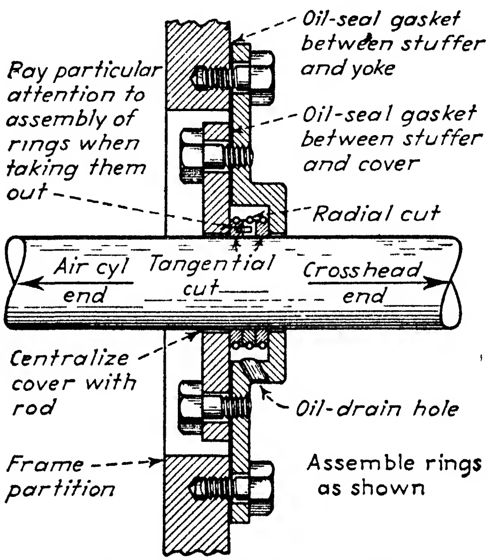


FIG. 226.—Piston-rod oil-wiper rings in frame partition.

Removing Cylinder Liners.—To remove the cylinder liners, first disconnect the lubricator connection and remove the cylinder head by taking out all of the holding bolts. Next, remove the piston and rod by unscrewing the piston rod from the

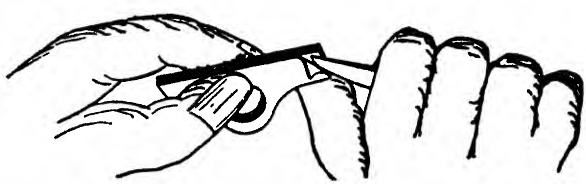


FIG. 227.—Laminated shims facilitate connecting-rod bearing adjustment.

crosshead. The liner can then be pulled out with a homemade puller (Fig. 222). If it has become rusted in place, the cylinder must be removed from the frame head. The liner can then be pressed out from the frame end. If a cylinder bore has been damaged, hone the damaged area smooth or rebore it; otherwise piston or piston-ring scorings will result and serious damage to the compressor may follow.

Replacing Piston Rings.—Replace piston rings when they become badly worn or when blow-by exceeds normal. When putting on new rings, keep the piston in the cylinder for easiest manipulation with only the ring-groove section protruding. New snap rings can be slid to their respective grooves by bridging the piston at several points with thin metal strips long enough

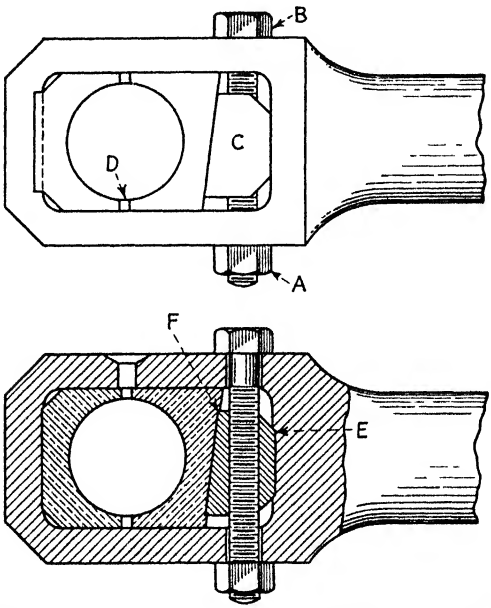


Fig. 228.—Shoulder wear on connecting-rod bearing. (*Courtesy of Power.*)

to span the foremost grooves. Clearance between the ring and its groove should just allow the ring to fall into place of its own weight (Fig. 223). Scrape carbon accumulation from the grooves before dropping the rings into place. Constrict the rings with several turns of tightly drawn cord (Fig. 224) or by a clamping band until they enter the cylinder easily; center the rings with the cylinder bore, and the piston can be pushed in place by hand with a light bar for leverage. If this force will not do the work look for some obstruction; do not use a jack. After pushing the piston into place, reassembly can begin.

Piston-rod Packing.—Before replacing piston-rod packing check its condition; discard old fibrous material; metallic and carbon-ring sections should be completely separated to measure end clearance. Exercise extreme care when handling this material; any scoring of ground splits, faces, ball joints, or seats will make the packing useless. Babbitt packing rings are especially vulnerable to damage at the thin areas coming in contact with the rod or casing.

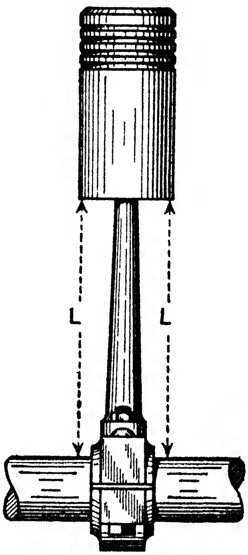


FIG. 229.—Measure distance L to check alignment of connecting-rod bearings. (*Courtesy of Power.*)

Metallic packing requires little attention other than cleaning as often as operating conditions require. When the rings have worn until the ends butt, their life may be prolonged by filing a small amount from each end of each segment (Fig. 225); it is extremely important that the same amount be removed from both ends. This is especially true of the bridge segments, where removal of unequal amounts, or removal of all clearance from one segment, will cause the keepers to rock to one side, permitting a leak. When removing material from the ends of segments follow the manufacturer's instructions implicity, because too much clearance is as bad as not enough. Tangential ring joints need no filing because the joint shape automatically adjusts for wear. If abnormal conditions make it necessary to reseat the packing to the rod, do this by scraping. Do not use a file or round off the bore edges.

Metallic packing must be kept sufficiently tight to prevent the escape of air from the cylinders. When the compressor is running unloaded, a vacuum is formed in the cylinder and there is a tendency to draw into the cylinder the oil which collects on the piston rod. Also, if the packing is sufficiently loose, some of the oil from the crankcase will be drawn up into the cylinder through the drain tubes. If the packing is kept sufficiently tight to prevent the escape of air from the cylinder, it will be tight enough to prevent the entrance of oil into the cylinder when the compressor is running unloaded.

Cylinder stuffing boxes are readily accessible through large openings in the frame-head yoke. They should be inspected frequently to make sure that there is no leakage. When adjusting these glands set up the adjusting nuts only enough to prevent

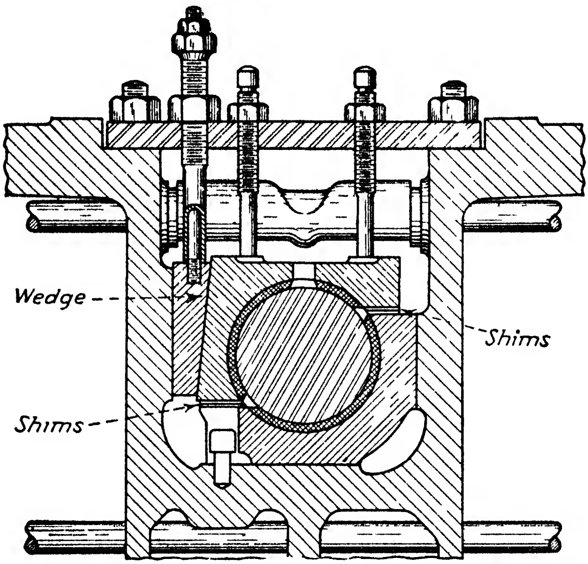


FIG. 230.—Wedge and screw main-bearing adjustment on Ingersoll-Rand compressor

leakage. Further tightening results only in undue friction, wear on the packing, and possible scoring of the rod. Set up the nuts an equal amount so that the glands will be square with the rod, for otherwise they will bear on the rod and score it. Always adjust the stuffing-box glands while the compressor is in operation.

Extended Shutdowns.—If the compressor is to be shut down for more than 2 or 3 days, fibrous packing should be removed from the stuffing boxes to prevent damage to the piston rod by corrosion. If the stuffing boxes are fitted with metallic packing, it is advisable to remove the packing rings and the partition plate oil-wiper rings. Never allow machines to lie idle with

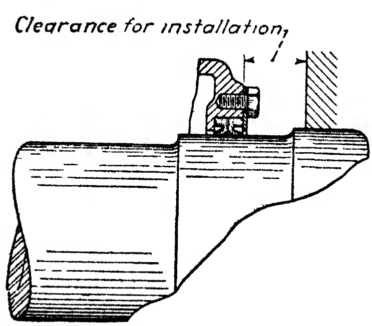


FIG. 231.—Main-bearing shaft-seal ring (*Courtesy of Garlock Packing Co.*)

condensed moisture in their cylinders and do not expose cylinders to the weather. The piston rod should, in such cases, be given a protective coating of oil to prevent pitting or corrosion. Remove all grease before replacing the packing.

The stuffing box in the partition between the crankcase and frame-head yoke usually contains a set of metallic or carbon wiper rings (Fig. 226). These rings are split and held to the piston rod by means of garter springs. The rings have a special edge for wiping the oil off the piston rod and have grooves for draining. The oil then flows back into the crankcase. The oil-wiper rings are designed to wipe the rod practically dry, leaving just enough oil film to lubricate the cylinder packing.

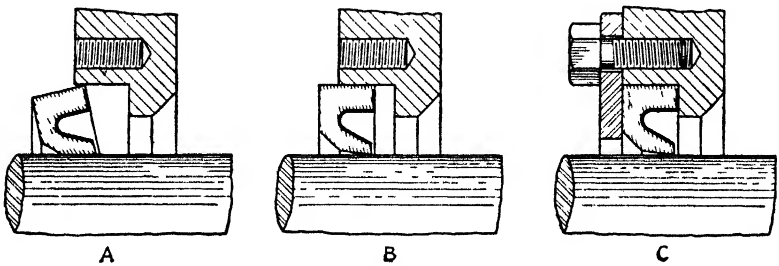


FIG. 232.—Installing Ingersoll-Rand shaft-seal rings

Clean the wiper rings periodically. Nicks or dents in the rings or a scored rod prevents a tight seal. Broken springs should be replaced immediately.

Installing Wiper Rings.—When the assembling wiper rings, stagger the joints. Newly installed assemblies should be fitted to the piston rod to ensure a good seal. They are usually numbered, and there is only one correct way to install them. The total end clearance of the wiper ring should not be more than 0.003 in. More clearance will cause the rings to act as a pump instead of wiping oil from the rod.

With forced-feed lubrication the partition plate stuffing box is fitted with tangential cut, cast-iron oil-scraper rings. These rings are specially designed to take care of the larger amount of oil carried by the piston rods on machines under pressure lubrication.

After putting the packing on the rod, screw on the nut and enter the rod in the crosshead. Continue to screw the piston rod into the crosshead until tram marks show that the original position has been reached, then tighten the lock nut.

Cylinder-head Gaskets.—Any graphited asbestos-body sheet packing can be used for gaskets between air cylinders and heads. Rubber packing is not satisfactory, as the heat and oil in time will soften it. If an old head gasket shows damage, install a new one. Measure the thickness of the old gasket; if there is any doubt as to the original thickness, use material $\frac{1}{16}$ in. thick, cutting all necessary ports in the gasket to fit openings in the cylinder and head. After bolting the head securely in place, remove a valve case from each end of the cylinder to check piston clearance.

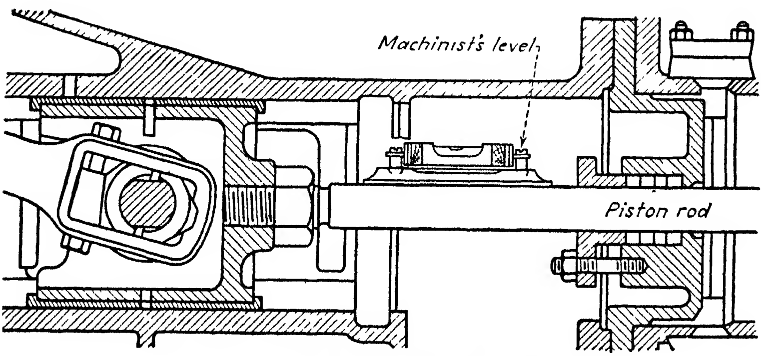


FIG. 233.—Checking and adjusting for piston wear.

Adjusting Piston Clearance.—Insert a lead wire through each valve opening and turn the compressor by hand through one complete revolution. Remove the wires and measure their flattened thickness; adjust for any inequality by screwing the piston rod in or out of the crosshead. When cold, head-end clearance can be 50 per cent greater than that of the crosshead end because the piston rod expands when reaching operating temperature.

The method of adjusting piston clearance varies with different compressor designs. In a vertical machine it is accomplished sometimes by adding or removing shims between the lower end of the connecting rod and the crankpin-bearing boxes, at other times by adding or removing shims between the piston and socket-type wrist-pin boxes. In some compressors the piston end clearance is not adjustable; in this case taking up on the crankpin bearing on account of wear will continue to unequalize the clearance to such a point that it becomes necessary to replace or rebabbitt the crankpin bearing. In double-acting pistons fitted with a piston rod and crosshead, the clearance is adjusted by turning the piston rod in or out of the crosshead.

Installing Connecting Rods.—With flood lubrication, the upper side of the connecting rod is provided with small grooves directly over the center of the bearing to receive oil for lubricating the pins. Be sure that the rod is replaced in the machine with the oil groove on top, as connecting rods are usually outwardly symmetrical with this exception; if it is not placed in this position the pins will not receive proper lubrication. Laminated shims are used between the halves of these crankpin boxes to prevent loss of oil pressure.

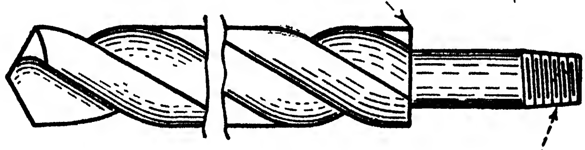


FIG. 234.—Drill head facilitates removal of hard deposit from cooler tubes. (*Courtesy of Power.*)

Adjusting Connecting-rod Bearings.—To make adjustments, loosen the bearing, remove the shim, and peel off one thickness at a time from both top and bottom shims, until proper clearance for the bearing is obtained (Fig. 227). After proper shim thickness is inserted, pull the box up tight. It is important that the same number of laminations be removed from both top and bottom shims to prevent distortion of the box.

Adjust connecting-rod bearings with considerable care, not only because of the effect on bearing life but also because these adjustments can affect piston clearance. Rods having both adjusting wedges located so that they tend to increase the distance between the crosshead and crankpin cause a lengthening of the connecting rod and reduce front-end piston clearance. Before adjusting, examine the bearing box where it contacts the wedge for worn shoulders (Fig. 228*F*).

Since considerable pressure can be exerted through a wedge and screw, adjustments must be made carefully or the bearings will be too tight. When adjusting connecting rods draw the wedges up snug, then loosen the bolt just enough to permit sliding the rod back and forth on the pins by hand. When it is impossible to detect rod movement, loosen the wedge-adjusting

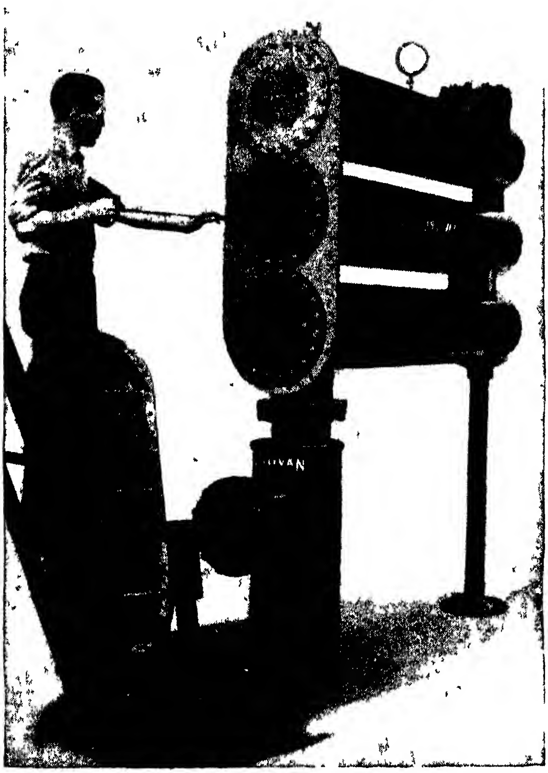


FIG. 235.—Using the scraper shown in Fig. 234.

screw one-half turn and then check the running temperature before making further adjustments.

Connecting-rod wedge bolts should never be made to take the load stress in operation, which occurs if the wedges are allowed to become loose. Unless they are kept tight to prevent this, broken wedge bolts will result from hammering of the loose boxes. Tapping the wedge on its side will tell whether it is loose.

When it becomes necessary to remove connecting-rod bearings, these must always be replaced in their original position. Put check marks on each half of the bearing and rod so that the halves will not be turned end for end or upside down.

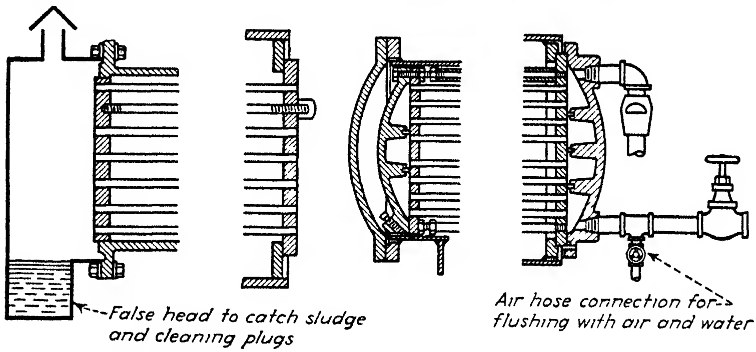


FIG. 236—Convenient arrangements for cleaning intercoolers (*Courtesy of Power.*)

Parallelism must be maintained between the face and back of each connecting-rod bearing shoe or the connecting rod will operate under a bending stress. Uneven scraping may destroy

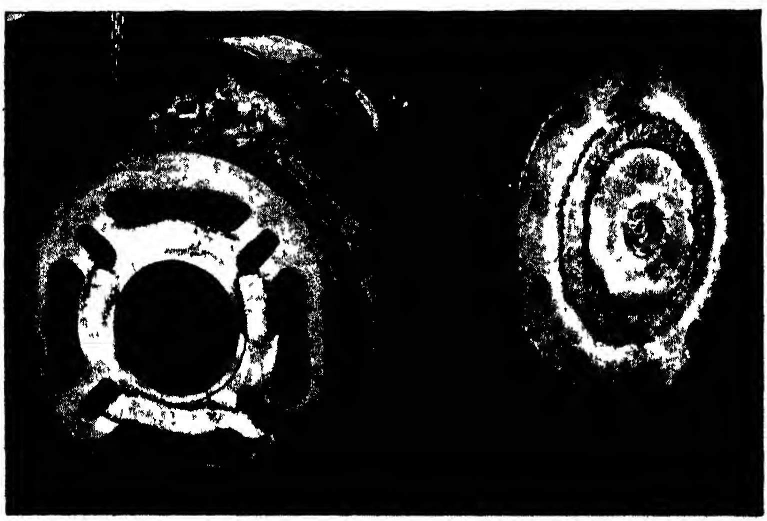


FIG 237.—Cracked cylinder water jackets can be repaired by brazing.

this original parallel contact; check each end of the rod as described under Fig. 171. Check single-acting rods, as shown in Fig. 229.

Adjusting Main Bearings.—Main-bearing adjustment is important and calls for good judgment. First, take up on the adjusting screws (Fig. 230) uniformly and tightly, then back off the setscrews a half flat. Second, hold the adjusting screw and tighten the lock nut. This will give a somewhat loose bearing, and there should be a slight pound when the machine is first started up. If the bearing fails to become quiet after the machine is warmed up, take up very slightly on the adjusting screws, being careful to take up the same amount on each. After any such adjustment watch the bearing carefully to make sure it does not heat up.

When necessary to inspect or renew the boxes, loosen the setscrews and take off the caps. Jack up the shaft just enough to remove pressure from the bearing. The side boxes can then be moved sideways, turned, and lifted out. The bottom box should be moved in the same manner until the ribs line up with slots in the bedplates; then turn the box around the shaft, and lift it out. Usually there is a plate in front of the quarter box against which the setscrews bear. This plate is to fill up the gap between the bedplate and bearing box so that if a setscrew should get loose no excessive movement of the shaft can occur and damage the motor winding.

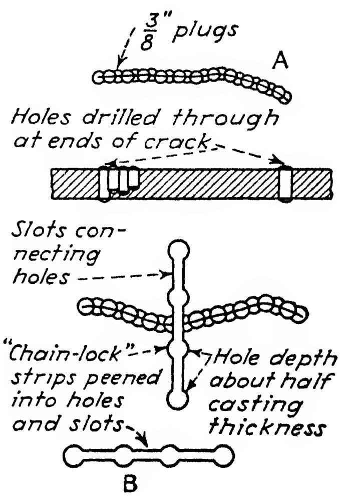


FIG. 238.—Jacket cracks can be sewed with plugs. (*Courtesy of Power.*)

Liners are placed between the main-bearing cap and bearing jaws on the bedplate. The bearing cap should be adjusted by means of these liners so that there is about 0.004-in. clearance between the cap and the top of each side box.

Value of Shims.—The value of installing shims between the bearing sections on both main-shaft and connecting-rod bearings is illustrated in Fig. 228, using a connecting rod for an example. Assume that a 0.004-in. running clearance is required. This will allow the bearing parts to move, causing wear adjacent to both sides of the wedge, as is shown in somewhat exaggerated form in the lower view. Unless these worn surfaces on the rod and taper half of the bearing are trued up before adjustments are attempted, satisfactory results will not be obtained, because when the wedge is moved up it has only a small bearing surface at points *E* and *F*. These points soon wear away and leave the bearing in as slack a condition as it was before.

When this trouble is encountered it can be eliminated by installing shims between the bearing sections. They can be held in place and away from the journal by dowel pins, or they

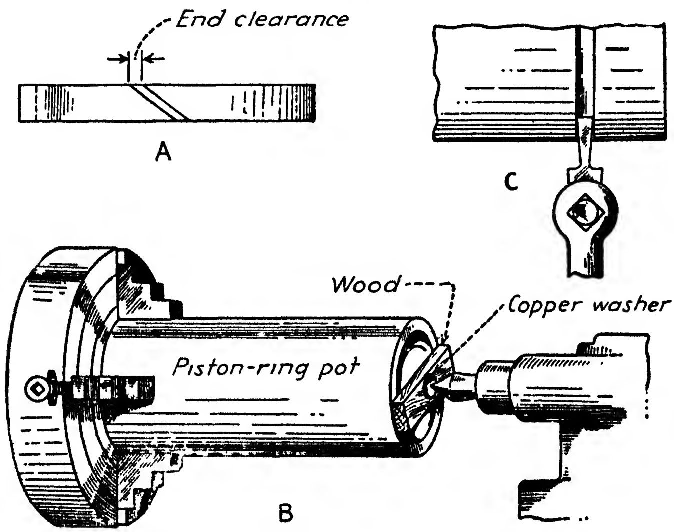


FIG. 239.—Cutting rings from a pot casting. (*Courtesy of Power.*)

can be cut away at their center edges to prevent their touching the journal. With this arrangement all parts of the bearing can be drawn up tight and there will be no looseness at the wedge.

A good rule for bearing adjustment allows 0.003-in. running clearance for shafts up to 3 in., with a 0.00098-in. increase in clearance for each inch increase in shaft diameter. Crosshead-pin clearance on large machines varies between 0.003 and 0.004 in., and crankpin clearance from 0.004 to 0.006 in. Main bearings, depending on shaft size, have running clearances from 0.008 to 0.010 in. To check the tightness of main bearings more easily, remove the connecting rods and then turn the unit by hand.

Tapered Roller Bearings.—Units using tapered roller main bearings usually have these bearings carefully and properly set up at the factory. If unusual operating conditions necessitate adjustment, proceed as follows: Loosen the clamp bolt and insert a suitable bar in the cast notches to rotate the adjusting collar (Fig. 145). It is easy to tighten up the bearing. A fine thread on the nut permits close adjustment. The bearing

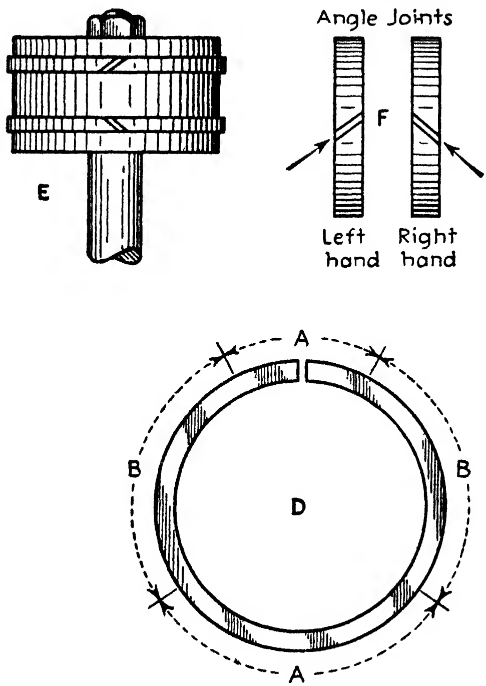


Fig. 240.—One-cut rings leak at *B*. (*Courtesy of Power.*)

is properly adjusted when there is no perceptible end play and the compressor turns over quietly and freely by hand.

Should it ever become necessary to remove a tapered roller bearing, it is so fitted that this can be done by drawing it off the shaft without injuring either the bearing or the crankshaft. In replacing one of these bearings, heat it in hot oil (not over 275 F) and then slip the bearing over the shaft. When the bearing is cool it will fit the shaft properly.

Main-bearing End Covers.—Split end covers at main bearings are fitted with sealing rings to prevent oil leakage along the main shaft (Fig. 231). Rings should be installed with the bevel joint at the top of the shaft. To ensure proper application and to prevent damage to the sealing lip of the rings, install them in the following steps (Fig. 232).

1. Do not heat oil-seal rings before application. Preferably they should be installed at room temperature (70 to 80 F).
2. Clean the recess thoroughly and remove all burrs and sharp-cutting edges.
3. Apply grease or oil to the shaft.
4. Place the ring around the shaft at a point near the recess into which it is to be placed.
5. Compress the ring slightly by squeezing the sealing lip to the body of the element (*A*).
6. Start the compressed end of the sealing element into the recess at the top or upper side of the housing, inserting the lip first; gradually work the heel or back of the element into place, keeping the seal compressed while doing so (*B*).
7. Continue this process around the entire periphery of the shaft until the sealing element is inserted in the recess.
8. Seat the element by tapping, using care to prevent damage to the lip.
9. Apply the cover plate, bolting it tightly into position to compress the oil-sealing ring into the recess (*C*).
10. It is important that the oil-sealing ring should not be too tight on the shaft, as this would cause excessive heating. If rings are found to be too tight, relieve them slightly with a smooth file.

Older machines have oil-box covers with cast grooves for felt packing around the shaft. This felt packing will in time become oil-soaked and should be replaced when necesssary for satisfactory operation.

Removing Crosshead Pins.—To remove tapered crosshead pins, unscrew the cap screw and remove the cap. Place this cap over the large end of the crosshead pin and insert the cap screw. Tighten the screw snugly against the cap. Tap the outside end of the crosshead pin enough to jar it loose and remove it by drawing up on the cap screw. Other tapered pins can be removed with a jack bar. Make the steel bar more than long enough to extend across the end of the crosshead pin, and drill two holes in it to match corresponding holes in the crosshead pin. Insert cap screws through the bar to the two tapped holes in the end of the pin. Separate the bar from the crosshead by blocks at each end so that when the two cap screws are turned down the pin will come loose.

Some of the larger compressors use a tap bolt and washer to hold the crosshead pin in place. To remove the pin, remove the washer, screw the tap bolt in the pin again, and then move the crosshead so that by unscrewing the tap bolt the latter will jam against a rib so provided in the side of the bedplate. Put a light strain on the tap bolt by unscrewing it. Then hit the crosshead with a lead hammer, causing the crosshead pin to spring loose.

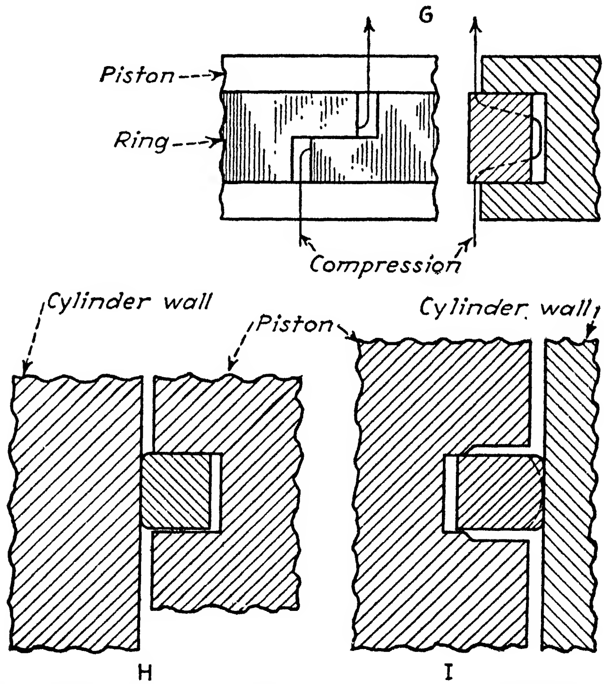


FIG. 241.—Ordinary step-joint leakage path *G*. Groove shoulders wear the ring face round *I*. (*Courtesy of Power.*)

Before crossheads have been worn in, care should be taken to allow enough clearance to keep shoes from heating. Start with about 0.012-in. clearance between the crosshead and guide, and, when the bearing surfaces have been worn in, take up the clearance to approximately 0.008 in.

Adjusting for Piston Wear.—When wear occurs at the piston and cylinder wall, the piston naturally takes a lower relative position. To compensate for this wear and to reestablish the original alignment, the crosshead should be lowered an amount equal to the wear. In making this adjustment, first loosen the cap bolt and back off the wedge, or remove shims at the bottom crosshead shoe until the piston rod is level with crosshead guide. Be sure the cap bolts are tightened up securely after adjustments

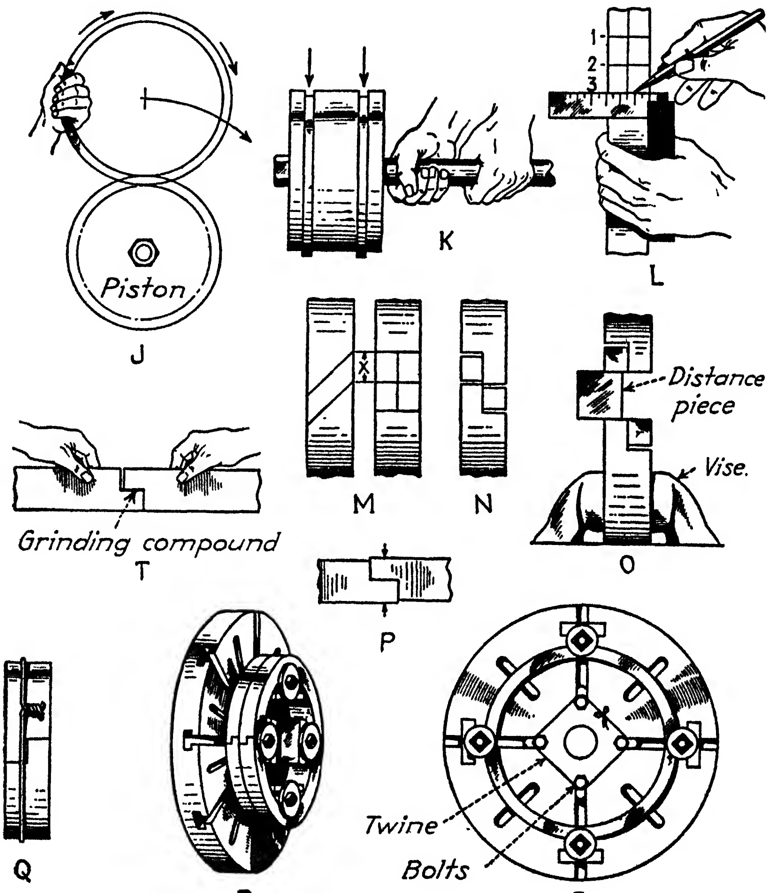


FIG. 242.—Cutting the step joint. Making the ring into a true circle produces perfect fit to the cylinder wall. (*Courtesy of Power.*)

have been made. When the adjustment has been completed, check the piston rod to make certain it runs true by placing an indicator on it and barring over the compressor a few turns. This can also be checked by laying a machinist's level (Fig. 233) on the piston rod and comparing this reading with that taken with the level on the bottom guide.

When making bearing and crosshead adjustments it is far better to make two or three individual adjustments to get proper bearing clearance than to pull a bearing up too tight and burn it out.

Cooling Systems.—Another maintenance job on reciprocating compressors consists of cleaning silt and scale from cylinder water jackets.

If the circulating water is dirty, mud deposited in cylinder jackets will ultimately obstruct the flow of water entirely unless

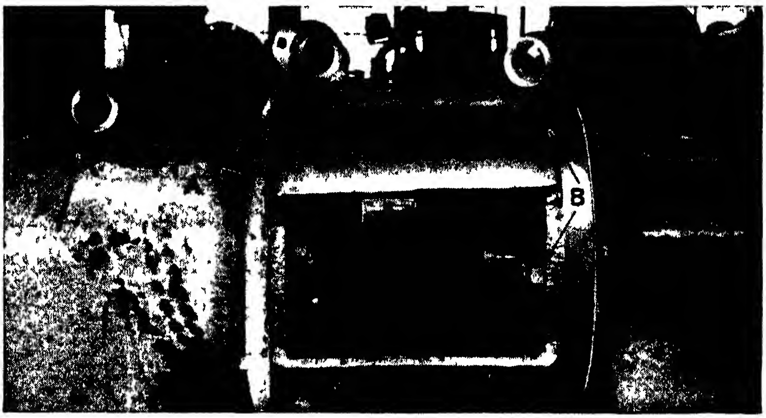


FIG. 243.—Distance piece for oilless compressor. (*Courtesy of American Air Compressor Corp*.)

care is used to prevent such an accumulation. Clogged passages will interfere with proper cooling, which will result in possible damage to the cylinders and pistons. Cylinder heads should be removed occasionally and the water jackets and passages inspected. If any mud deposit is found it should be thoroughly cleaned out and the jackets completely flushed with water applied through a nozzle made of ¼- or ½-in. pipe, pointed at the end. With this nozzle water can be sprayed directly on the bottom and in all corners of the water jackets. Loose silt also can be removed by admitting alternate slugs of air and water through the regular water lines to stir up sediment and flush it out. Scale proves more difficult and must be removed with acid.

Removing Scale.—To remove scale from the water jacket, a solution of hydrochloric acid and water is most effective. The solution should be made up of 80 per cent water and 20 per cent acid, mixed in a steel or wooden barrel or tank of a size comparable to the water-jacket volume. Do not use a galvanized vessel, for acid attacks these metals readily; this reduces the effectiveness of the solution.

Disconnect the cooling-water line and pump the solution from the barrel into the cooling-water inlet, through the water jacket, then from the water outlet to the barrel. If a suitable pump is not available, a short length of pipe may be attached to the water inlet with a funnel elevated so that the acid solution will circulate through the water jacket and flow by gravity back to the mixing barrel. Circulate the acid solution for 10 to 15 min, then let the jacket stand full for 30 min. After this recirculate for another 10 to 15 min. Repeat this process until the jacket is clean or the solution spent. For extremely heavy scale deposits it will be necessary to add more acid to the solution after it becomes weak. After the jacket is clean, flush it with fresh water to remove the solution, and inspect the cylinder-head gaskets for possible damage. When handling this solution take all the usual precautions; wear goggles to protect your eyes from it, and do not spill it on your body or clothing.

A cross connection between the water inlet and outlet will permit reversing the cooling-water flow for a few minutes of the day, thus working out much of the sediment that would otherwise settle and cause trouble.

Cleaning After- and Intercoolers.—The amount of heat removed from air as it passes through a cooler depends to a greater extent than is usually appreciated on the cleanliness of the cooling surfaces. Silt or sludge in water passages or oil scum on air surfaces greatly retards heat transfer. Slushing with a hot-water and washing-soda solution, in the form of a solid stream or coarse spray, will remove oil film adhering to the tubes. Flushing the water side with alternate shots of water and compressed air (air pressure not over 40 psi) stirs up the silt and flushes it out of the discharge. A convenient air-hose attachment, connected in the water line, permits a quick change-over from water to air. Slime and sludge removal requires hosing with a strong stream of water or shooting elastic cleaning plugs through the tubes with compressed air and then flushing them with water. This requires loosening one cooler head and removing the one at the working end. The tool (Fig. 234) can be used to remove hard deposits from tubes (Fig. 235).

When shooting tubes, keep the far head in place, supported by bolts of such length as to permit a 1-in. opening between the cooler and the head. The head stops the cleaning plugs and the free opening allows air exhaust without blowing back through other tubes onto the operator. A pan placed under the rear-head opening catches heavy sludge and a canvas thrown loosely around the end prevents spattering the surrounding walls. If

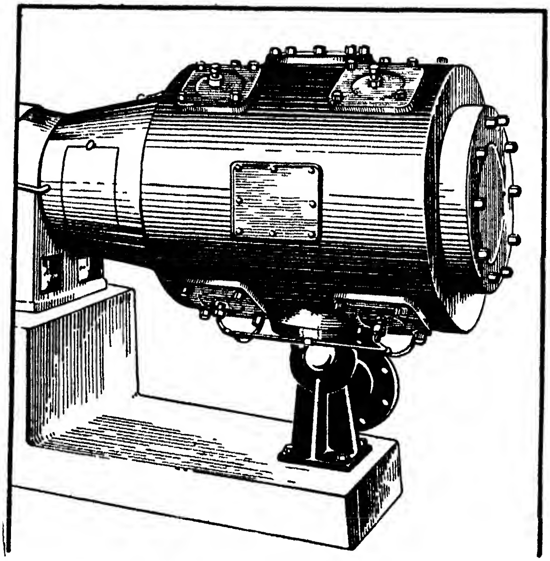


FIG. 244.—Supporting foot piece.

the frequency of cleaning warrants fabrication, a false head and pan (Fig. 236) constructed from sheet metal serve as a plug stop and catch basin for sludge.

As this cleaning requires removing the heads, attention must be given to the head gaskets. When partitions cast integral with the heads divide water flow into passes, gasket extensions pass between these partitions and the tube sheet. Use extreme care when replacing the heads and locate the gasket properly, or the gap between the partition and tube sheet will short-circuit the water flow and lower the cooler's efficiency.

Cooling-system Leaks.—Inspect the coolers and cylinder jackets frequently for leaks. When the cooling-water pressure is high, water will find its way into the high-pressure cylinder and, even in small amounts, will wash off the oil and cause wear of the cylinder, piston rings, and valves With low water pressure there will be a loss in capacity because air can escape through the cooling passages Leaking tubes must be replaced immediately. New ones can be expanded in place with a roller or swaged tightly by driving a round pin, tapered ½ in. per ft, into the tube end.

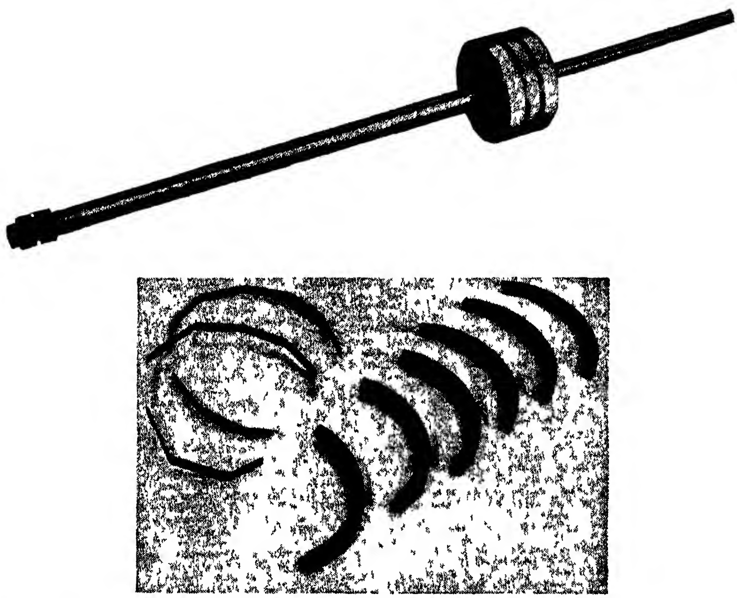


FIG 245 —Aluminum piston, carbon rings, and expanders (*Courtesy of American Air Compressor Corp*)

Cracked cylinder jackets can be brazed (Fig. 237). "Sewing" offers a method of repair when the crack is on the outside surface (Fig. 238). Start by drilling a hole at each end of the crack to prevent further development. Then drill and tap a series of blind holes along the crack line. Into these thread cast-iron or steel plugs and cut them smooth with the surface. Between these plugs insert a second row, overlapping the first plugs. Hammer the entire line of plugs smooth (*A*).

Another method depends on the use of special patented low-coefficient-of-expansion steel strips, shaped as shown at *B*. At right angles to the crack, cut blind holes and slots into which the strips fit. After the first strip is inserted and peened, follow the same process with succeeding strips until the holes and slots are filled. When the cracked casting becomes heated in operation, as would a cylinder head or jacket casting, the "chain-locked" strips expand less than the surrounding metal and tend to pull the crack together.

Small cracks can be mended temporarily by filling the jacket with a solution of sal ammoniac and water or ground flaxseed and water. After the leak is plugged, remove the solution and flush with clean water. Consider this method as an emergency repair only.

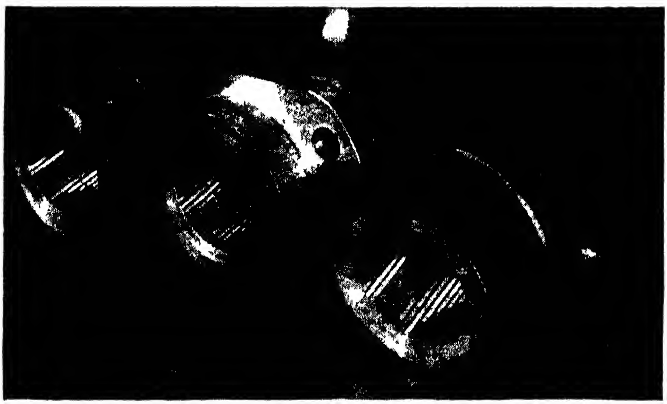


Fig. 246.—Carbon and bronze bushing in packing follower.

Intake Filters.—Clean intake filters often enough to remove dirt before the intake is restricted. The cleaning method depends on the filter. Dust dry-felt units by blowing air through them in a reverse direction or by washing; wash viscous filters—never use a volatile solvent—and recoat with the viscous fluid; drain and filter the oil from oil-filled cleaners to remove sediment, then scrape and wipe off the accumulations adhering to the internal surfaces. Certain dry filters and strainers cannot be cleaned; they must be replaced with new ones.

Distribution Lines.—Indicator cards taken on discharge piping with indicator motion actuated from the compressor crosshead will show whether or not excessive pulsations in the discharge line are causing power losses. On one installation indicator cards showed a pulsation in the discharge line which built up terminal pressures to 130 psi when the average pressure in the line was 100 psi. These pulsations represented about 30 hp excess load on a machine requiring normal power of 350 hp; they were eliminated by placing a surge chamber in the discharge pipe near the compressor.

Test the pipe lines with liquid soap to make sure that the joints are tight. Large power losses result from leaky air lines. The amount of time the compressor operates on a closed line indicates leakage. These lines are often installed without much planning and are extended as demand for air arises in other locations. When extended several times with more or less temporary piping or with air hose, the entire section should be rebuilt with pipe of adequate size.

MAKING COMPRESSOR PISTON RINGS

Good ready-made rings usually outlast and outperform the homemade variety, but the operating engineer should know enough about this subject to enable him to design and make his own rings when necessary.

Select Materials Carefully.—The first step involves selecting the proper material. Most ring manufacturers have discontinued machining rings from cylindrical pots and instead cast each ring individually, leaving just enough excess material for machining. Development of the individual casting introduced a higher standard of quality and uniformity than could be obtained with the old pot or cylindrical casting.

However, it is more practical to machine homemade rings from pots than to attempt the more difficult individual casting procedure. The pot should be composed of hard, homogeneous, close-grained iron containing a minimum amount of free carbon. Have the material heat-treated at the foundry.

After obtaining a proper casting, the operator must design the ring. Whether it is to be a *one-cut* ring, that is, a ring that is not returned and rebored into a true circle after it is split, or a *two-cut* ring, the dimensions and tension should be the same.

Securing Proper Tension.—In a homemade ring, tension is secured by turning the ring larger than the cylinder diameter and then collapsing it into the cylinder after a section is cut out. Pressure exerted between the ring and cylinder depends on the kind of material used, the finished radial thickness, and the axial width. Since the operating engineer rarely has all this information at hand, he must use a general formula which disregards the nature of the material. Almost every handbook gives a different formula for figuring tension; the following method is both simple and reliable.

Add 0.012 in. for every inch of cylinder diameter to get the total ring diameter. Therefore, in making a ring for a 10.5-in. cylinder, multiply 10.5 by 0.012, which gives 0.126 in. Adding this to 10.5 gives a total of 10.626 in., which will produce a pressure between the ring and the cylinder wall of from 2 to 5 psi

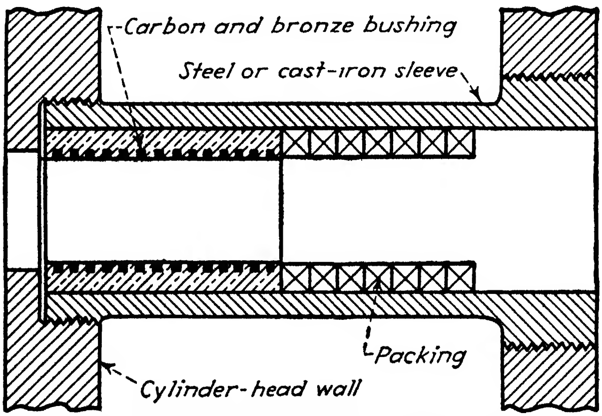


FIG. 247.—Carbon and bronze bushing in steel or cast-iron sleeve for front head.

depending on the material and the other dimensions. Some handbooks recommend adding $\frac{1}{72}$ of the cylinder diameter, which is about 0.15 in. for a 10.5-in. ring.

The end clearance, shown in Fig. 239*A*, represents the amount of opening the ring has when it is in the cylinder. This clearance is for circumferential expansion. A standard allowance is 0.003 in. per in. of cylinder diameter. Thus, a ring made for a 10.5-in. cylinder will have $10.5 \times 0.003 = 0.0315$ in. end clearance.

The end clearance is only part of the amount that must be cut out of the ring. The diameter was increased to 10.626 in. to provide tension, which is the same as saying that the circumference was increased by $0.126 \times \pi = 0.396$ in. This much, plus the 0.0315 in. end clearance, or a total of 0.428 in., must be removed from the circumference to ensure a well-fitted ring.

Concentric snap rings which are to be sprung over the piston may be made about one thirtieth of the diameter in thickness, or about 0 035 × diameter. For a 10 5-in cylinder, this gives $10\ 5 \times 0\ 035 = 0\ 368$ in, or about $\frac{3}{8}$ in A ring of this size is not likely to become distorted or to break when springing it over the piston.

Since rings are usually fitted to old ring grooves, the ring-face width cannot be controlled by the operating engineer If new

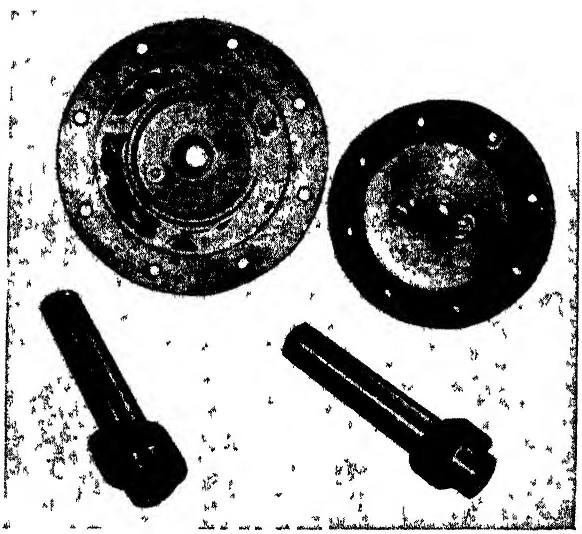


Fig 248—Front heads fitted with tail-rod bushings and guards. (*Courtesy of American Air Compressor Corp*)

pistons are machined, then the groove width can be made equal to the thickness of the ring wall. While standard rings are usually a little wider across the face than through the wall, the above proportion will make a serviceable ring if high-grade cast iron is used. For rings from 4 to 16 in. in diameter, some manufacturers use a face dimension $\frac{1}{16}$ to $\frac{1}{8}$ in. wider than the ring thickness.

Machining Operation.—In machining, grind the tool to a blunt point with little clearance or rake and take a heavy cut at slow turning speed. The tool will last longer if the point digs in under the scale, which is usually hard and sandy. Chattering due to overhang may be minimized by running the tailstock against a piece of wood and a copper sheet washer as shown in Fig. 239*B*. The copper prevents the center from digging into the wood. The pot may also be supported this way while cutting off the ring. A narrow, square-nosed cutting-off tool with sides tapering back from the edge (Fig. 239*C*) does the best job. Locking the carriage to the lathe bed prevents rings from being cut off too narrow, which often happens because the carriage moves toward the ring. Before cutting off, face the side of each ring.

After the rings are split and the proper amount cut out, it will be seen that the rings are egg-shaped when placed in the cylinder (Fig. 240*D*). The rings will bear against the cylinder along surfaces *A* while compression will blow past surfaces *B*. This will continue until surfaces *A* wear sufficiently to allow surfaces *B* to seat against the cylinder wall. This is one of the most undesirable features of one-cut rings. Other disadvantages include wearing cylinders out of round and excessive end clearance when they are finally worn in.

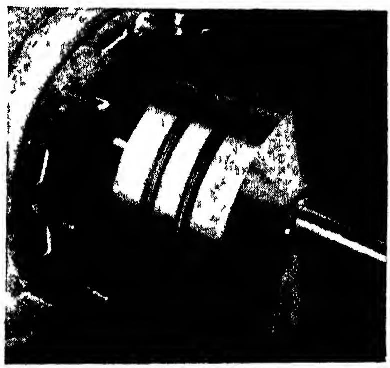


Fig. 249.—Aluminum piston made from solid casting. (*Courtesy of American Air Compressor Corp.*)

Perhaps the best reason for making thin, flexible rings lies in the fact that they adjust themselves quickly to worn cylinder outlines. Keeping sliding friction between the rings and cylinder walls reduced to the lowest possible point for efficient operation is also of major importance. For worn cylinders, the outside of the ring should never be machined to a smooth finish. By leaving tool marks on the outside face, a more rapid sealing action is secured.

Whether piston rings are made with an angle or step joint is of little importance. The ordinary step-cut ring does not make a perfect seal, just because the ends overlap as shown in Fig. 241*G*. It can be seen that air enters the gap on the compression side of the ring, goes under the ring, and out the other side. So long as there is space between the inside of the ring and the groove, compression will leak through the same as through an angle joint.

Figure 240*E* shows a piston with right- and left-hand angle-joint rings. Some engineers state that rings having opposite angle joints turn in opposite directions and the joints, therefore, will not line up for any appreciable length of time. A right-hand joint is one that slopes upward from right to left when the ring is in a vertical position, while a left-hand joint slopes upward from left to right (Fig. 240*F*). Some authorities recommend that rings made for lubricated cylinders have the outside edges rounded as in Fig. 241*H*. This causes the ring to slide over the oil film rather than to scrape it off.

Preparing Piston Grooves.—The adage that a ring is only as good as its groove should be kept in mind; unless the sides of the grooves are faced each time new rings are installed, there is little sense in replacing rings. Figure 241*I* shows a new ring riding on the old groove shoulder which puts the ring under severe strain, causing wear, as shown by the dotted line, and perhaps breakage. The importance of a good joint between the sides of the ring and the groove cannot be overemphasized.

The best way to assure a proper fit is to machine the ring several thousandths wider than the groove, then place the piston in the lathe and face off the groove sides until the ring rolls freely around the groove, as shown in Fig. 242*J*. This assures a better job than that obtained by the usual method of facing the groove first and then filing the ring to fit. To determine the proper amount of side clearance, some engineers hold the piston as in *K*. If the rings slide to the bottom of the groove, there is enough clearance, assuming they have not been sprung out of line in stretching over the piston.

Two-cut Rings Preferred.—Experience shows that it is well worth the extra time required to make two-cut rings instead of the more common one-cut egg-shaped variety. The calculations are the same for both types except that from $\frac{1}{16}$ to $\frac{1}{8}$ in., depending on the ring size, is left on both inside and outside dimensions for later machining. Rings made in this way should be step-cut, or they cannot be clamped properly against the lathe faceplate.

To lay out the step, chalk the ring and scribe three lines across the axial width, as shown in Fig. 242*L*. The distance *X* in *M* is equal to the total amount to be removed and is the same for either angle or step-cut rings. Split the ring with a hack saw, through the center line, and saw halfway through from opposite sides as shown in *N*. Put the ring in a vise with a distance piece holding the ends apart just wide enough to saw out the square along vertical center line *O*. File the steps so that the distance indicated by arrows in *P* is slightly greater than the ring width.

Making the Second Cut.—Hold the ring in the closed position by twisting a wire around the circumference, *Q* Then clamp the ring to a faceplate by means of bolts, washers, and distance

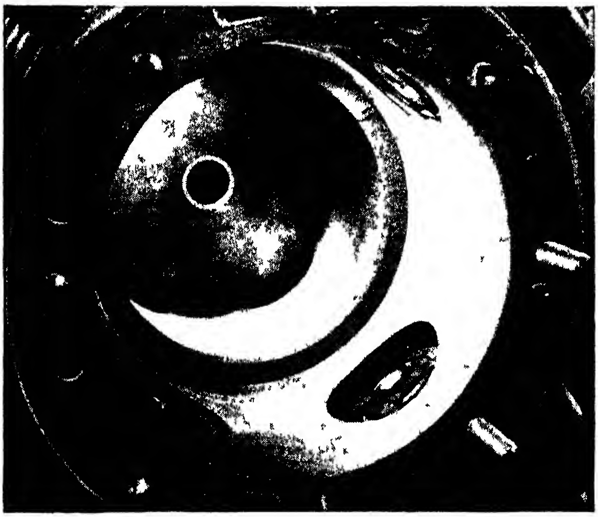


Fig. 250.—Fit the carbon rings carefully

pieces *R*. A heavy piece of paper between the ring and faceplate protects the latter from the lathe tool bit. Now it can be seen that a few thousandths left in the step joint will ensure a ring staying closed while machining. After the rings are trued and clamped securely against the plate, the wires can be removed and the outside diameter turned to the exact cylinder dimensions.

Next twist wires around the rings as before and place another set of clamps around the outside of the rings, opposite the inside clamps. When secure, loosen the inside bolts, and, if they cannot be removed, hold them together as shown in *S* so that they will be out of the way of the boring tool. After the rings have been bored to the proper wall thickness and removed from the faceplate, the step joint can be filed or ground to conform with ring width T.

Rings made in this way will be as nearly perfect as possible, and the operator will be repaid by the efficient and satisfactory operation of his compressor. It is assumed, of course, that the shoulders are removed from the cylinders before installing new rings. Figure 22 shows a boring bar set up for boring the cylinder.

RINGS FOR OILLESS OPERATION

Manufacturers build reciprocating compressors that operate without oil lubrication, but if you need one, cannot buy a new one, and have a double-acting machine available, these pointers show how to convert it to oilless operation.

Material Needed.—To do the job right you will need (1) a distance piece the same length as the compressor stroke to hold the cylinder clear of oil carry-over from the crosshead by the piston rod; (2) a new one-piece piston rod with tail extension to carry the piston clear of the cylinder walls; (3) carbon-insert bronze bushings for both front and rear cylinder heads and packing followers; (4) an additional packing gland in the front head; (5) a tail-rod guard mounted on the front head; (6) a set of carbon piston rings with metal expanders; (7) solid aluminum casting for the piston; (8) a substantial foot piece to support the cylinder weight; and (9) lubricated flexible-metal or carbon packing for the piston rod.

Distance Piece Required.—Purchase a distance piece from the manufacturer or fabricate one from boiler-plate steel (Fig. 243). It must be strong enough to resist piston-thrust load and carry the overhung cylinder weight. Assume that you have a 12-in. cylinder that compresses to 100 psi. Then $6^2 \times 3.14 = 113$ sq. in. area of piston, and $100 \times 113 = 11{,}300$ lb total thrust load. The overhanging weight can be determined with beam formulas. Proceed with caution at this point, because extending the cylinder away from frame studs A stresses them considerably more than originally. The stress in studs B remains approximately the same. (Use a liberal safety factor and always provide a foot piece as shown in Fig. 244, to support the cylinder.) Machine the distance-piece end disks parallel to each other and with accurate fit into the cylinder head and frame joints; otherwise, the cylinder will not line up with the frame center line. The distance piece for this 12- by 12-in. compressor was fabricated from $\frac{3}{4}$-in. boiler plate welded to $1\frac{1}{4}$-in. end disks.

As the distance piece moves the cylinder farther away from the frame, a longer piston rod is required. Make it long enough to reach through the front head.

Piston Rod.—Machine the piston and tail rod in one piece with a tapered section to match the piston (Fig. 245). The

Fig. 251.—Tie rings in the grooves and push the piston in. (*Courtesy of American Air Compressor Corp.*)

tail section may be smaller in diameter because it carries no thrust load. This permits sliding the piston lock nut up to the threaded section ahead of the piston and screwing it home. The extension or tail rod slides through a carbon lubricated bushing in the front cylinder head. This floats the piston and prevents it from touching the cylinder walls.

Bronze Bushings.—Carbon-insert bronze bushings in each cylinder head (Figs. 248 and 250) and crankcase partition carry the sliding rod. Packing-gland followers are also fitted with carbon and bronze bushings (Fig. 246). As the rear head already contains a bushing, enlarge the opening just enough to press the new one in place, then key it with setscrews.

The front head presents a different problem because the water-jacket pocket must be penetrated. If boring will weaken the head, purchase a new one cast with a center core; otherwise, bore a hole perpendicular to the head-gasket face large enough to pass the tail rod. Then, leaving a substantial shoulder on the inner wall (Fig. 247), enlarge the hole to fit a full-length

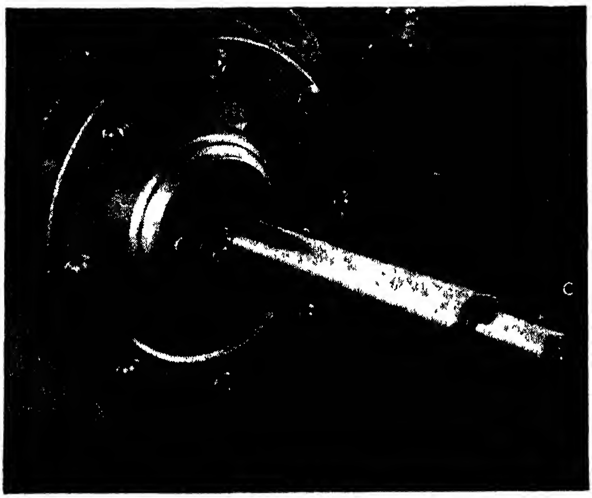


Fig. 252.—Cut piston rod off at *C* after clearance is adjusted.

sleeve containing the bronze bushing. Thread both ends of the hole and sleeve, making each complete thread-cut without releasing the lathe-carriage split nut from the lead screw. Bronze bushing inside the sleeve also serves as a packing stop. The heads (Fig. 248, left) show packing gland and bronze bushing, (right) flexible metallic packing in place.

Aluminum Piston.—Since the regular cast-iron piston is quite heavy, the load on the piston-rod bushings can be reduced by using an aluminum piston (Fig. 249). Make one from a solid block, to eliminate trouble from unequal expansion that may occur in a hollow casting having walls of uneven thickness.

After machining the piston to correct size, cut grooves for carbon rings. Make them deep enough to allow for expander rings of $\frac{1}{32}$-in. spring steel, bent as shown in Fig. 245.

Fitting Piston Rings.—Purchase a set of carbon rings and fit them to the cylinder bore. After fitting them correctly (the total joint clearance should be about 0.001 in. per in. diameter), scribe reference marks on both sides of each joint to assure correct mating when fitting the rings in the piston grooves (Fig. 250).

Put the expander rings and carbon segments in place and tie with cord (Fig. 251). Push the piston in the cylinder slowly so that the ring edges do not chip. The cord will push off as the piston slides in. Screw the piston rod in the crosshead but do not tighten the crosshead lock nut until the piston clearance has been adjusted.

Button up the job by putting on the head and gasket. Insert rod packing, put the gland follower and tail-rod guard in place, and draw up the gland. After adjusting the piston clearance, measure and cut off the excess part of tail rod *C* in Fig. 252. A pipe cap then closes the end of the tail-rod guard.

CHAPTER IX

ROTARY COMPRESSORS

Field of Application.—Rotary-compressor applications overlap those of both reciprocating and centrifugal machines. These machines are built for pressures up to and including 100 psi and capacities above 12,000 cfm.

Classification.—The ASME test code classifies rotary compressors as *sliding vane*, in which longitudinal vanes slide radially

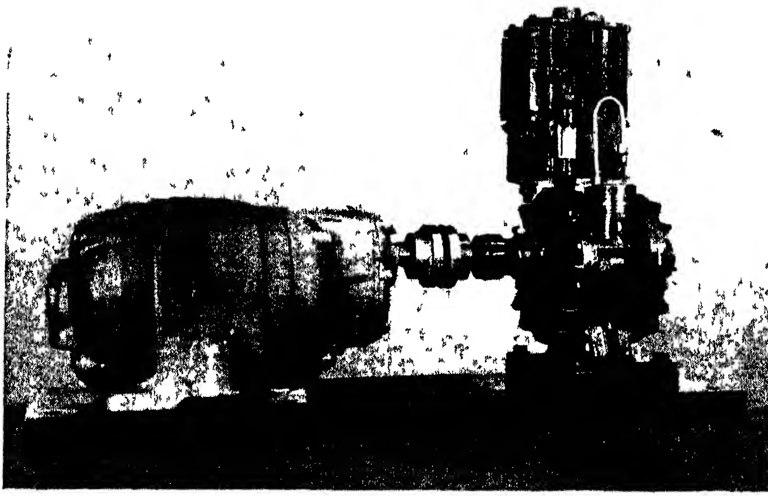


FIG. 253.—Air-cooled sliding-vane compressor. (*Courtesy of Foster Pump Works Inc.*)

in a rotor mounted eccentrically in a cylinder; *two impeller*, in which two mating lobed impellers revolve within a cylinder; and *liquid piston*, in which a liquid serves to displace the air within a rotating element.

SLIDING-VANE COMPRESSORS

General Characteristics.—These machines are made for pressures as high as 125 psi and capacities up to 2,000 cfm.

Generally, single-stage machines (Table 30) are used for pressures up to 50 psi, and two-stage machines (Table 31) for higher pressures. Operating speeds vary from 3,600 rpm for small to 450 rpm for large units.

TABLE 30.—SINGLE-STAGE SLIDING-VANE COMPRESSOR CAPACITIES

60 cycle rpm	Cfm actual free air delivery				Hp rating of nearest commercial size squirrel-cage motor			
	Lb gage discharge pressure				Lb gage discharge pressure			
	20	30	40	50	20	30	40	50
1,160	32	31	.....	.....	5	5		
1,160	44	42	.....	.....	5	7½		
1,160	52	50	.....	....	5	7½		
1,160	76	74	72	70	7½	10	15	15
1,160	83	82	80	78	7½	10	15	15
1,160	112	109	107	105	10	15	20	20
1,160	129	127	124	122	15	15	20	25
1,160	154	152	149	146	15	20	25	25
1,160	197	194	190	186	20	25	30	30
1,160	232	228	224	220	20	25	30	40
870	284	280	275	270	25	30	40	50
870	339	334	329	324	30	40	50	50
870	377	370	365	360	30	40	50	60
870	403	396	390	385	40	50	60	60
870	482	473	467	460	40	50	75	75
690	534	526	519	512	50	60	75	100
690	607	598	592	585	50	75	100	100
690	685	675	665	656	60	75	100	100
690	773	763	754	745	75	100	100	125
575	890	878	866	855	75	100	125	150
575	1,050	1,037	1,023	1,010	100	125	150	150
575	1,410	1,392	1,374	1,355	125	150	200	200
575	1,610	1,592	1,572	1,554	125	200	200	250

Operating Principle.—Construction consists of a cylindrical or cam-shaped casing in which a cylindrical rotor, smaller in diameter than the casing bore, is arranged eccentrically (Fig. 254). A

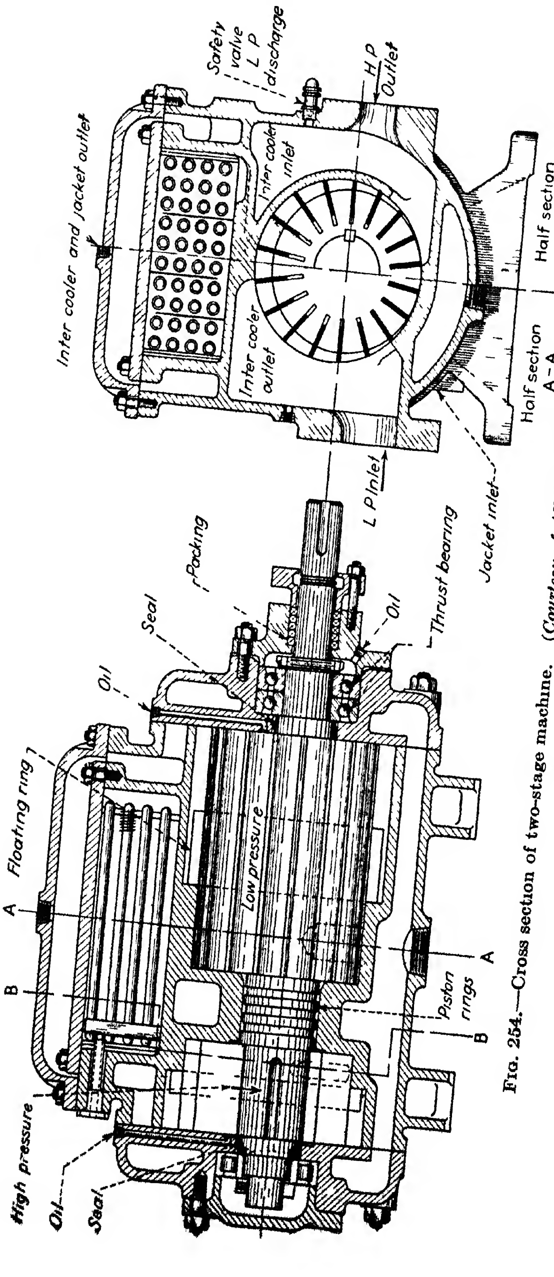


FIG. 254.—Cross section of two-stage machine. (*Courtesy of Allis-Chalmers Manufacturing Co*)

number of radial slots cut along the entire length of the rotor hold the sliding vanes. In some units these vanes are made of metal; in others they are nonmetallic. As the rotor revolves, the vanes are held out by centrifugal force, by pins, or by springs, forming a number of cells in the space between the rotor surface and casing wall. Because of the eccentric rotor position, the cells vary in capacity during each revolution from a maximum to a minimum, and again to a maximum.

Air enters the cell between vanes *A* and *B* (Fig. 255); as vane *A* passes the inlet, trapped air is carried around to the discharge

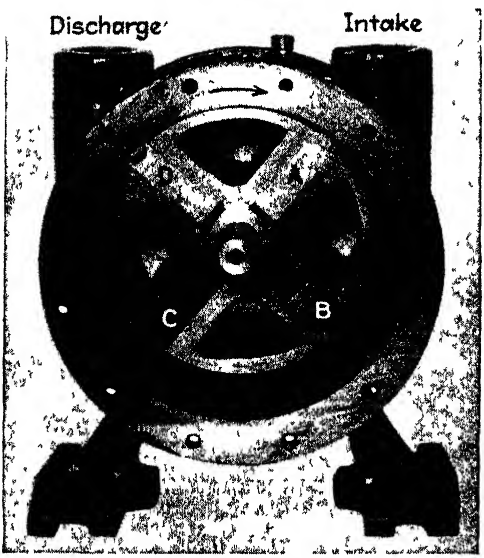


Fig. 255.—Cam-shaped cylinder bore. (*Courtesy of Foster Pump Works Inc.*)

port. The air is then forced out of the discharge as the cell area decreases. The volumetric efficiencies of two different sliding-vane compressors, when used as vacuum pumps, are shown by curves in Figs. 256 and 257.

Compare Cam-shaped and Circular Bore.—In the cam-shaped bore (Fig. 258) the center of the rotor is offset from the true center of the cylinder bore; nevertheless, lines drawn from one side of the cylinder, passing through the offset-rotor center to the other side of the cylinder, are all equal. This means that the vanes are always in contact with the cylinder walls at their outer edges, while their inner edges are always in contact with each other. Thus a true piston action is maintained without depending on springs or centrifugal force to extrude the vanes. In this design AB equals CD.

TABLE 31.—TWO-STAGE SLIDING-VANE COMPRESSOR CAPACITIES

60 cycle rpm	Cfm actual free air delivery				Hp rating of nearest commercial size squirrel-cage motor			
	Lb gage discharge pressure				Lb gage discharge pressure			
	60	80	100	120	60	80	100	120
1,750	115	114	113	112	25	25	30	30
1,750	124	123	122	121	25	30	30	40
1,160	154	153	152	151	30	30	40	40
1,160	197	196	194	192	40	40	50	50
1,160	233	231	229	227	40	50	50	60
870	284	282	281	279	50	60	60	75
870	341	338	336	334	60	60	75	75
870	377	375	373	371	60	75	75	100
870	403	401	399	397	75	75	100	100
870	482	480	477	473	75	100	100	125
690	535	533	530	526	100	100	125	125
690	609	606	603	600	100	125	125	150
690	686	683	680	676	125	125	150	150
690	776	772	768	764	125	150	150	200
575	892	888	884	880	150	150	200	200
575	1,053	1,047	1,041	1,035	150	200	200	250
575	1,408	1,402	1,396	1,390	200	250	250	300
575	1,613	1,606	1,598	1,592	250	250	300	350

In the round-cylinder bore (Fig. 259) diameter $A'B'$ is greater than a line drawn through the center of the rotor when it is offset from the true center $C'D'$. Therefore, the maximum over-all length of any two-complementing vanes could not be more than $C'D'$. As this is less than $A'B'$, it is necessary to depend on centrifugal force, or springs, to press the vanes against the cylinder when they are in a vertical position.

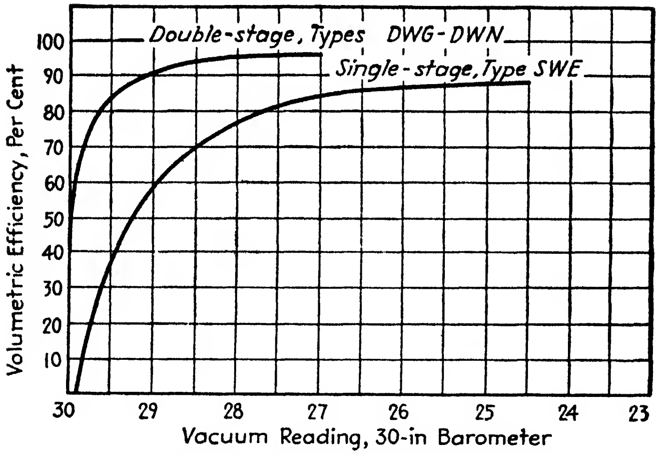


FIG. 256.—Volumetric efficiency of single- and two-stage sliding-vane machines when used as vacuum pumps. (*Courtesy of Lammert & Mann Co.*)

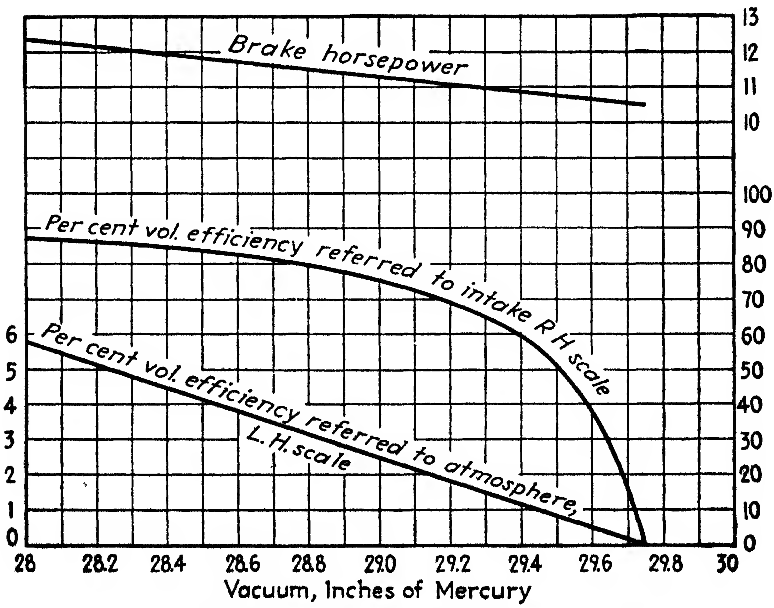


FIG. 257.—Characteristics of a two-stage machine used as a vacuum pump. (*Courtesy of Fuller Co.*)

Rotor End Clearance.—Close clearance between the rotor and end caps, together with an oil film, serves as a seal against leakage. If the rotor floats freely between the end caps, actual rubbing between rotor ends and the casing occurs. Normal wear occurs mostly on the end caps, and worn depressions must be machined out to bring the clearance back to normal. When rotors float freely between the two heads, alignment of driving power greatly affects rotor end wear. Some machines use a thrust bearing (Fig. 254) to prevent end rubbing; another design (Fig. 260) uses a special bearing arrangement which holds the rotor centered in the casing.

Vane Wear.—Sliding contact between the vanes and the cylinder wears the vane edges or shoes. Seemingly, this wear

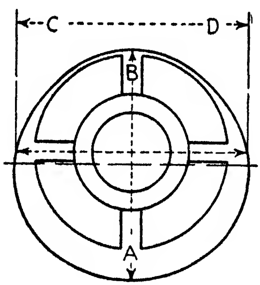


FIG. 258.—Cam-shaped cylinder.

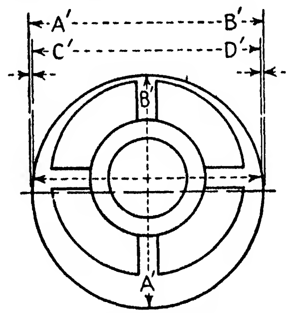


FIG. 259.—Circular cylinder.

would be considerable, but actual wear in one compressor was only 0.004 in. after 13,000 hr operation.

Normal wear will in time require replacing the vanes. Depending on compressor design, some may need replacing after they have worn ⅛ in., while others may permit ¼-in. wear. Reasonable wear does not cause leakage. Infrequent inspection will permit the vanes to shorten until they begin to cock and bind in their slots, and bending or breakage will soon occur.

To reduce wear, the vanes in some machines do not bear on the casing wall. Instead, floating rings (Fig. 261), having an inner diameter slightly less than the casing bore, are used. The vanes bear on these rings and cause them to rotate with the rotor. The outer surface of the floating rings is separated from a recess (Fig. 254) in the casing by a small clearance. The dragging action of the rings in relation to rotor movement may cause indentations in the vane edges. Vanes held out by centrifugal force must be replaced when wear reduces their weight to such an extent that centrifugal action no longer exerts sufficient force to seal against leakage. Low-speed machines having spacer rods (Fig 255) to hold opposite blades in position have no compensation for wear; either new shoes or longer spacer rods must be installed when wear increases the total spacer-rod clearance 0.002 or 0.003 in. Consult the manufacturer about the amount of wear permitted.

Cooling Arrangements.—Water jackets and finned cylinder walls provide for either water or atmospheric-air cooling of the compressor cylinder. Water cooling usually begins for compression of 35 psi and above. Figure 262 illustrates one method where water passages are cast integral with the casing. Some units may have water-cooling passages only in the casing heads (Fig. 255). The piping arrangement (Fig. 263) follows the method used on reciprocating machines with an overflow sight funnel in the discharge line coming out of the casing top. Figure 253 is an air-cooled machine.

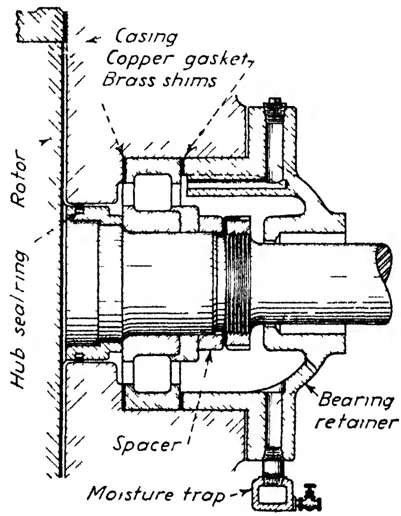


Fig 260.—Bearing construction and shaft seal ring (*Courtesy of Fuller Co*)

Two-stage compressors employ intercoolers to cool the air before it enters the high-pressure unit. Figure 264 shows an intercooler built integral with the compressor base, and Fig. 265 illustrates a separate unit.

Shaft Packing.—Packing methods vary with the different manufacturers according to the service for which the machine is to be used. Sometimes soft packing (Fig. 266) is used, while in other machines a shaft-seal ring is utilized. In the machines in Figs. 254 and 266 only one end of the shaft extends through the casing; therefore, only one set of packing is required. Metallic seal rings require oil for lubrication and to assist in preventing leakage.

Lubricated rotary seals on one or both ends of the shaft have automatic spring adjustment. In Fig. 267, the shaft-seal assembly consists of a hardened and ground steel or composition seal ring running on a seat formed on the inner surface of the end cap. The seal ring moves with the shaft by means of the driving pin pressed into the set collar and projecting into a slot in the ring periphery. The seal ring has a conical bore in which is inserted the end of a movable seal sleeve, forming a ball-and-socket joint so that the ring is free to make perfect contact with its seat. The seal sleeve fits the shaft accurately so that axial movement is possible without noticeable air leakage.

The spring between the sleeve flange and the set collar serves to maintain a tight joint between the sleeve and seal ring and to

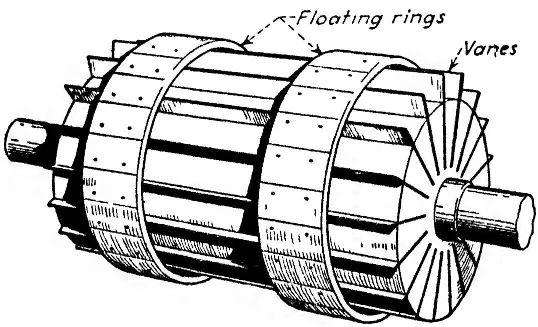


Fig. 261.—Floating rings prevent contact between vanes and cylinder.

keep the seal ring perfectly seated. The set collar serves also to keep the roller-bearing inner race in proper position.

In Fig. 268, soft packing is provided between the seal ring and the shaft in place of the seal sleeve and ball-and-socket joint. This packing rotates with the shaft and is, therefore, not subject to wear.

Lubrication Requirements.—The sliding-vane compressor will normally require a heavier oil than the reciprocating machine for the same compression range because of the scraping action of the vanes against the cylinder wall. Force-feed lubrication delivers oil to the compressor; this oil must not only maintain a suitable film on the cylinder walls to provide a running surface for the vanes, but it must also lubricate the slot in which they slide.

Lubrication requirements are different for each machine and the manufacturer's instructions should be followed carefully. Table 32 gives one set of specifications. Internal lubrication is

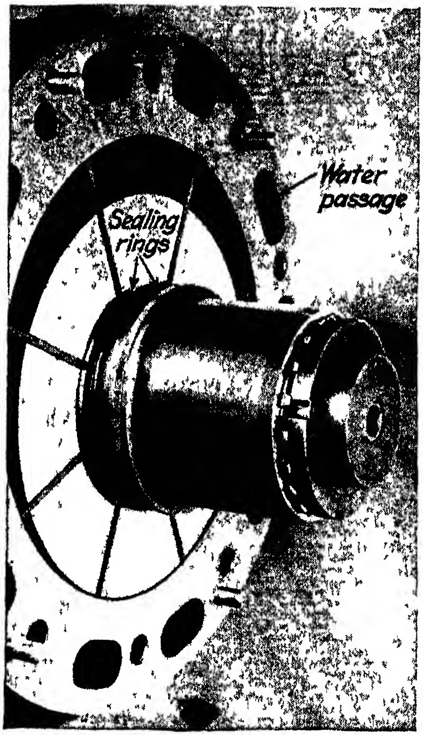


Fig. 262.—Cooling-water passages in cylinder walls. *(Courtesy of Fuller Co.)*

generally supplied by means of automatic oilers (Fig. 269) or mechanical force-feed lubricators (Fig. 270*L*).

Table 32.—Characteristics of an Oil Suitable for Sliding-vane Compressors

For machines operating with room temperature 60 to 90 F

Characteristics	A oils	B oils
Flash minimum	450 F	450 F
Fire minimum	500 F	500 F
Saybolt at 210 F	70–90 sec	55–70 sec
Recommended maximum carbon	0.50% Conradson	0.35% Conradson

The oil economizer (Fig. 271) has a needle valve V at the bottom of the oil container with its stem extended through the operating cylinder on top of the container. The valve is opened and closed by pressure from the compressor discharge exerted under a spring-loaded cup-leather piston P in the cylinder. A seal around the needle-valve stem prevents leakage of pressure into the oil container. The feeding rate is regulated by the knurled nut N at the top of the stem which limits the opening

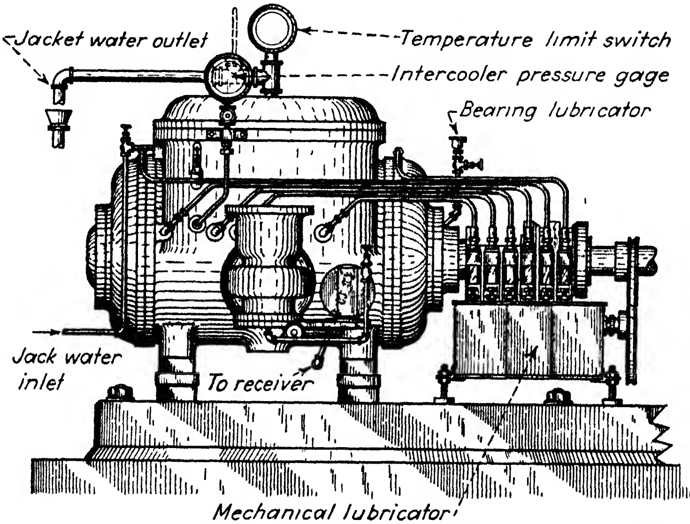


FIG. 263.—Pipe the cooling water so that the passages fill completely before any discharges. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

of the needle valve and which can also be used to open the valve by hand in case the compressor is operated without pressure on the discharge. A minimum discharge pressure of 5 psi is required to operate the oiler satisfactorily. Adjust the feeding rate to 4 or 5 drops of oil per minute.

The most reliable method of determining whether or not the feed is correct is by inspecting the compressor interior. If the surfaces are dry and show signs of rust, the feed is too small and should be increased; if lubricant has accumulated in large quantities, the feed is excessive and should be reduced.

The feeding rate on the mechanical unit (Fig. 272) is adjusted by screw E. To decrease the feed, turn screw E clockwise.

Nine complete turns are necessary to adjust from minimum to maximum feed.

To refill the sight-feed glass, fill the gun X with equal parts of glycerin and distilled water. Remove plug F and screw the gun in its place. Turn the gun handle until all oil is forced out of the sight glass and the glass is full of clear liquid. Then give the gun handle 40 additional turns to fill the space C above the sight glass completely. Then remove the gun and insert the plug.

Water and Oil Separators.—Some manufacturers use a combination separator and oiler. These devices connected in the air-discharge line remove oil carry-over and return it to the compressor. A cross section of the unit in Fig. 273 is shown in Fig. 274.

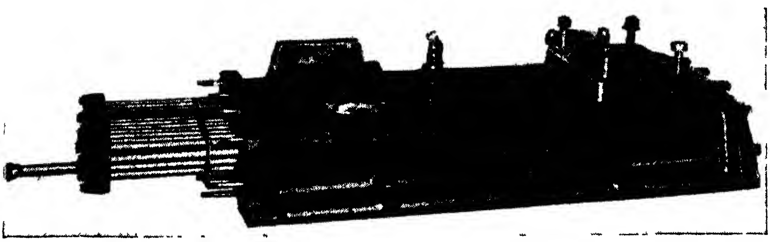


Fig. 264.—Intercooler formed inside the compressor base.

To check up on oil flow, remove the lower end of the copper tube from its connection on top of the compressor cylinder, loosen the coupling nut on the brass elbow on the oiler, and turn the tube to one side. Open the suction and close the discharge by the valve, or by pressing the palm of the hand against it. Start the compressor. After about 1 min operation, oil should drop from the tube at the rate of 1 or 2 drops a sec. If no oil flows after several minutes' operation, it is possible that the flow-reducing valve is clogged. In this case, unscrew the brass elbow that contains the check valve and clean it thoroughly.

There are three internal strainers in the oiler, so that the chance of the filter's becoming clogged is somewhat remote. However, after the oiler has been in service for about a year it is advisable to clean the oil filter. To do this, remove the small pipe plug in the front of the oiler below the copper tube and pull out the wire mesh with a pair of small pliers, or a small hook, which will connect with the eye in the middle of the roll of wire screen.

After cleaning, make sure that the filter screen is pushed back far enough not to get caught in the threads of the pipe plug; otherwise it will be impossible to tighten the plug.

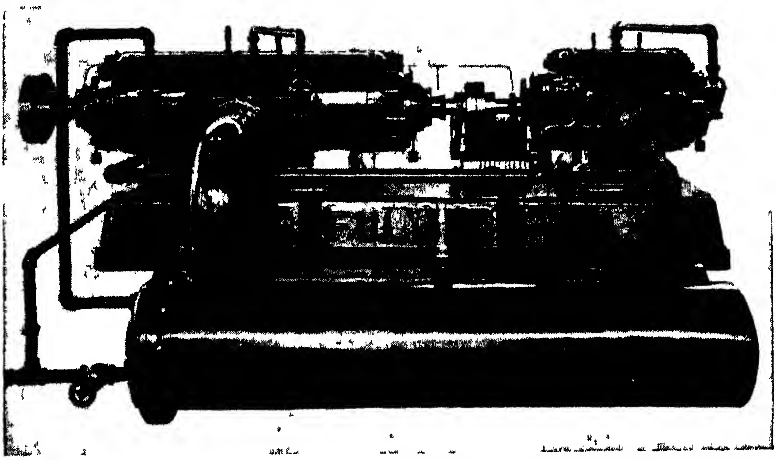


FIG 265 — Two-stage machine using a separate intercooler

The unit in Fig 275 automatically separates oil and water from the discharge air. Water discharges automatically and the oil feeds back to the compressor.

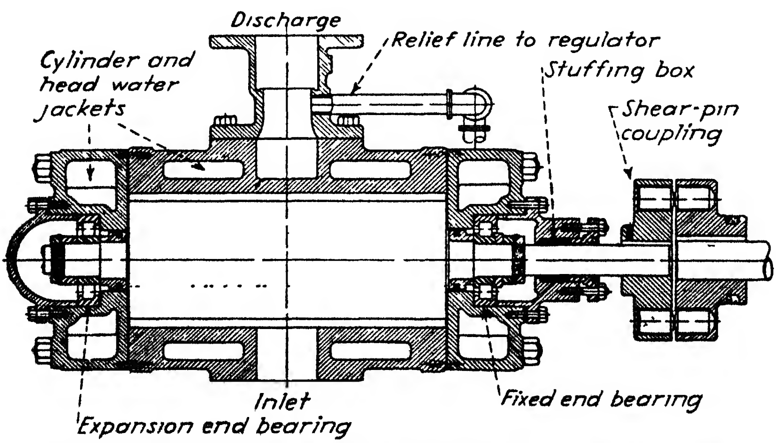


FIG. 266.—Cross section showing bearing arrangement and shaft-coupling shear pins. (*Courtesy of Fuller Co.*)

Unloading Methods.—Sliding-vane compressors are positive-displacement machines, require a safety valve on the discharge

(Fig. 276), and are normally equipped with an automatic unloader consisting of a spring-loaded, air-operated intake valve controlled by a pilot valve. When maximum-discharge pressure is reached, the inlet valve closes, a check valve in the discharge line (Fig. 277) closes, and a by-pass opens between the discharge and inlet that completely unloads the machine. When excess air must be by-passed for any considerable length of time, it is advisable to install a by-pass cooler.

On small compressors or when air demand is intermittent, the automatic unloader is exchanged for a pressure switch and automatic motor starter. The compressor then starts and stops according to the air demands.

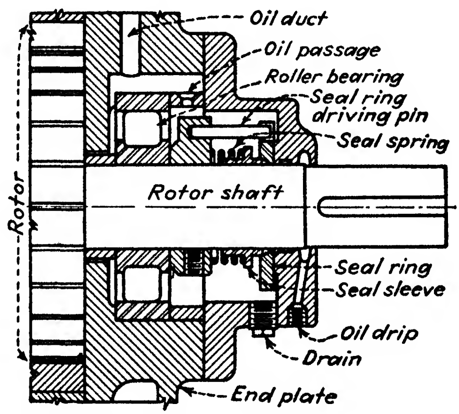


FIG. 267.—Shaft-seal assembly. (*Courtesy of Yeomans Brothers Co.*)

In the unloader (Fig. 278) incoming air moves down through ports in the main valve as indicated by arrows. Control pressure from the receiver enters the pilot valve at X.

When air pressure exceeds the setting of the pilot-valve spring, the pilot valve lifts and uncovers a port leading to piston Y. The piston then moves to the right, closes the main valve, and opens the relief valve Z. This valve relieves the pressure between the outlet check valve and the compressor cylinder.

The intake unloader (Fig. 281), shown in cross section in Fig. 279, closes the compressor inlet and opens a by-pass to unload the machine. When the compressor is loaded, valve A is held open by its spring. A pilot or solenoid-operated valve (Fig. 280) admits air pressure to close the intake valve.

When the solenoid is deenergized, port C is open, allowing air to pass from C to A and then to B (Fig. 279). This pressure forces the main valve up, closes the intake, and opens the by-pass to relieve pressure inside the cylinder.

Starting the Compressor.—Before starting the machine, check the following points:

1. The lubricator should contain an adequate amount of oil. Turn the hand crank to ensure that all oil lines are clear of air and are filled with oil so that lubrication starts at once. Loosen tubing connectors at the cylinder to determine when tubes are filled with oil. Do not neglect to retighten as soon as all air is forced out.

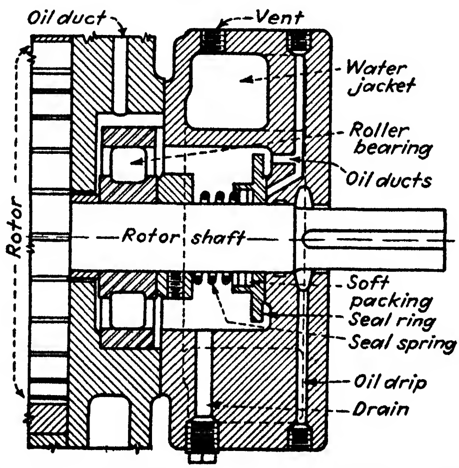


FIG. 268.—Shaft seal contains auxiliary soft packing. (*Courtesy, Yeomans Brothers Co.*)

2. Turn the cooling water on and adjust the cylinder supply so that the jackets are never less than 70 F, or above 100 F. Adjust water flow to the intercooler of two-stage machines so that air discharge from the intercooler is as cool as possible.
3. Open the main air-line-cutoff valve.
4. Before starting a machine fitted with an intake regulator, which is to discharge into a system already under pressure, set the regulator to the unloaded position. Unload the regulator (Fig. 278) by turning the handwheel to screw the valve all the way in. Be sure to unscrew it all the way out to reload after starting and opening the line-cutoff valve.
5. In the case of a new machine, check the direction of motor rotation with the coupling opened. This is important. The direction of rotation is marked by an arrow on the cylinder.

6. Should the compressor have been shut down during freezing weather and cooled off thoroughly, turn it over by hand for at least one complete revolution before starting.

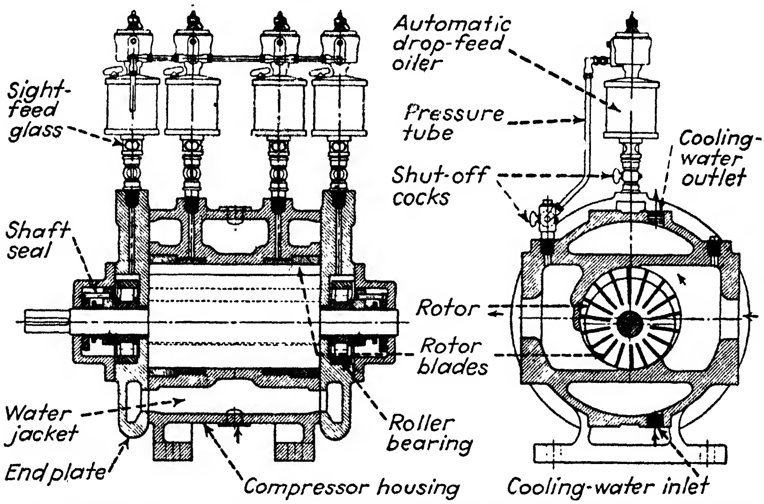


FIG. 269.—Automatic drop-feed oilers supply lubrication. (*Courtesy of Power.*)

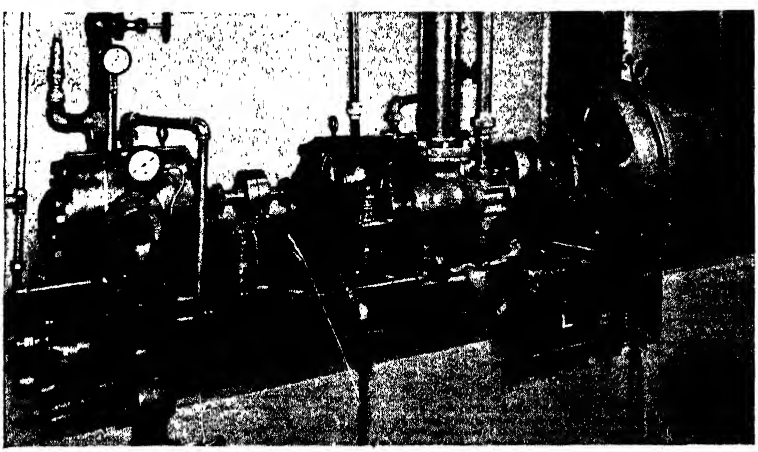


FIG. 270.—Force-feed lubricator *L* feeds oil as required.

Intercooler Drainage.—Drain all condensed moisture from the intercooler air passages of two-stage compressors regularly. This helps dry the air and adds to the life of the compressor. Condensate traps must be watched; do not assume that they are working. All drains should have visible outlets. Do not attempt to drain the intercooler when the compressor is running unloaded, because at that time it is under vacuum. Keep the moisture trap under the cylinder head drained of water condensed in the bearing chamber.

Running Unloaded.—Any compressor having an intake regulator should not run in the unloaded condition (either automatically or otherwise) more than ½ hr at any one time. If conditions require a greater length of unloaded time, the compressor must be shut down or a blowoff provided so as to increase the total proportion of full-load running time. To run unloaded for any considerable length of time also wastes power.

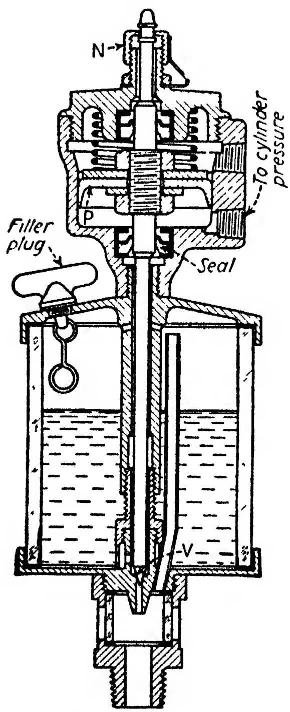


FIG. 271.—Automatic oil economizer.

Stopping the Machine.—When a compressor is equipped with a regulator, unloading the machine by hand before stopping will prolong the life of coupling shear pins. Shut the jacket water off as soon as the compressor stops, and open the drains to empty the jacket of water. This prevents condensation of moisture on the cylinder walls when the compressor is idle. If there is any possibility of freezing, always drain the cylinder jacket and intercooler.

The machine should start easily and run quietly with a minimum of vibration, normally operating with a smooth, low-pitched hum. (The pitch is higher when the machine is unloaded.)

Maintenance Requirements.—Maintenance consists of inspecting the bearings, shaft packing, and rotor vanes and cleaning the lubricating system, water jackets, air filter, unloading valve, and pilot valve.

When inspecting bearings, tap them lightly to see that the race is tight on the shaft. Flush oil-libricated bearings by pouring about 1 pt of light oil into the housing through the filling line. Clean the lubricator, oil reservoir, and piping at regular intervals.

The unloader, pilot valve, and supply lines require regular cleaning to prevent accumulations of carbonized oil and pipe scale.

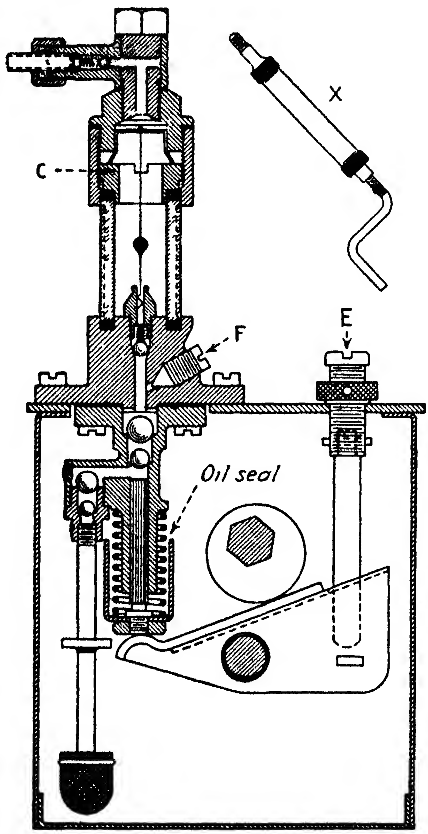


Fig. 272.—Cross section of force-feed lubricator.

Here is one suggested maintenance time schedule:

Daily:
1. Drain moisture from the air lines.

Weekly:
1. Oil the shaft packing.
2. Flush oil-lubricated bearings.

Monthly:
1. Inspect and grease the bearings.
2. Clean the cooling-water system.
3. Clean the unloader and pilot valve.
4. Clean the intake filter.
5. Check shaft-coupling bolts and shear pins.

Yearly:
1. Dismantle the machine to inspect rotor vanes and cylinder walls.
2. Clean the lubricating system thoroughly.

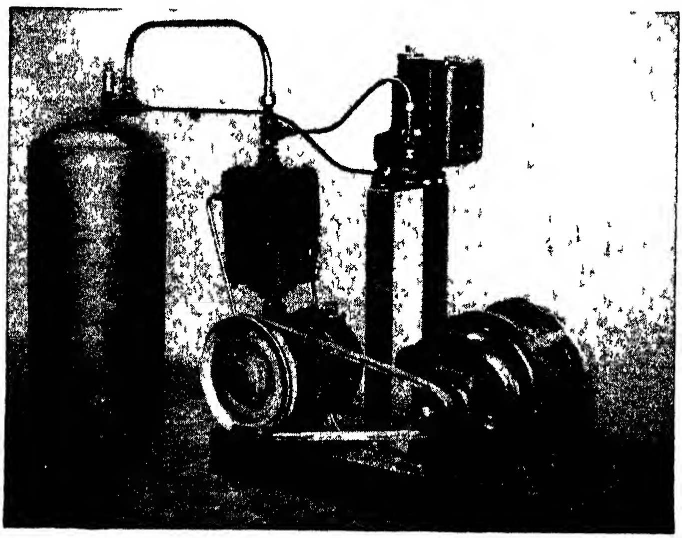


FIG. 273.—Automatic device removes oil from the discharge and returns it to compressor (*Courtesy of Foster Pump Works, Inc.*)

These faults indicate the need of internal inspection and attention:

I. Excessive oil sludge indicates
  1. Insufficient lubrication or poor quality lubricating oil, causing:
     a. Excessive vane wear.
     b. Excessive wear of cylinder walls.
  2. Dirty inlet air, due to:
     a. Excessively dusty location.
     b. Clogging of the inlet-air filter.

II. Excessive discharge-air temperature indicates
  1. Ineffective water cooling from:
     a. Inlet-water temperature too high.
     b. Insufficient water pressure.
     c. Dirt and scale in the water jacket.

2. A defective bearing, causing metallic contact of the rotor and cylinder
3. A dirty air filter, causing a reduction in the amount of air delivered.

III. Undue noise, vibration, or periodic knocking may be caused by
1. A defective bearing.
2. Excessive vane or cylinder wear.
3. Insufficient lubrication.
4. Rotor scraping the bottom of the cylinder. (Knocking will be more apparent when the machine is unloaded.)

IV. Faulty operation of the unloading valve may be caused by
1. Oil sludge, dirt, or scale in the pilot valve or line.
2. Gummy oil causing the power piston to stick.
3. Failure of the pilot-valve diaphragm.
4. Breakage of the three-way valve in the pilot valve.
5. A leaky valve.
6. A solenoid coil burned out.
7. Failure of the pressure switch and/or electrical connection.

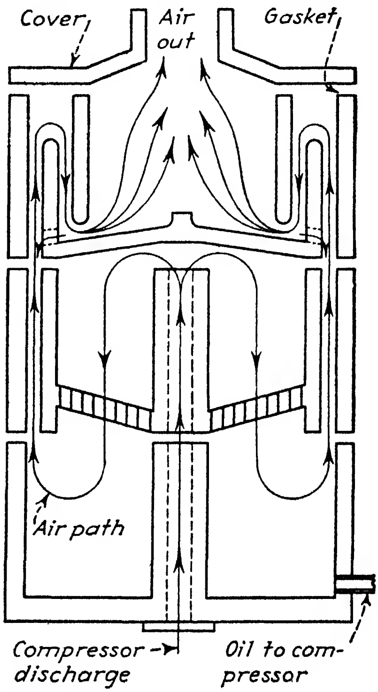


Fig. 274.—Cross section of automatic oiler in Fig. 273.

Dismantling Two-stage Compressors.—To dismantle the compressor in Fig. 254, proceed as follows:

1. Lift the machine from the base plate and remove the half coupling from the rotor shaft.
2. Slide the stuffing box and gland from the shaft.
3. Remove the lock nut and washer holding the inner bearing race. Be sure to mark the location of the lock nut with respect to the shaft before removal, so that it can be drawn up to its original position when reassembling.
4. Remove the low-pressure cylinder head (motor end) and bearing as a single unit as follows:

a. Insert studs in the cylinder head that extend beyond the shaft end.

b. Place a drilled plate or bar over the studs and against the shaft end. Tighten the stud nuts to remove the head and thrust bearing from the cylinder. Note carefully the relative position of the two bearings. They must be replaced in exactly the same position.

5. Remove the high-pressure cylinder head (outboard end) together with the outer bearing race and roller assembly.

6. Remove the high-pressure rotor. It is a slide fit on the shaft and can be easily removed with eye bolts inserted into tapped holes in the rotor face. Tie a cord around the rotor to hold the vanes in place.

7. Remove the low-pressure rotor and shaft as a single unit. Tie a cord around the rotor to hold the vanes in place.

8. Remove the retaining rings by inserting two long threaded studs in tapped holes in the ring edge. Drill a plate to pass over the studs and block

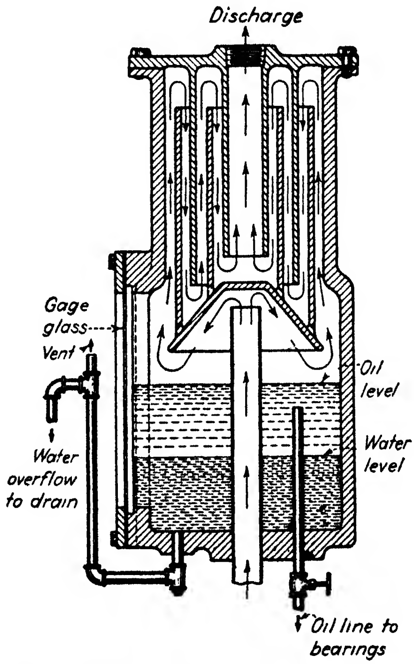


FIG. 275.—Oil and water separator. (*Courtesy of Beach-Russ Co.*)

it against the cylinder. Keep the rings square with the cylinder bore while pulling. Note the relative location of markings on the rings with respect to the cylinder so that they can be replaced in their original position.

9. Remove the floating rings by hand, checking reference marks so that they can be replaced correctly. An arrow shows the direction of rotation. Tie a cord around the ring to hold the sealing blades in place.

Reassembling Two-stage Compressors.—See that all internal surfaces are free from dirt, oil holes are open, and water jackets and intercooler-water passages are clean. Remove burrs and stone down all rough spots in the cylinder and on the rotors.

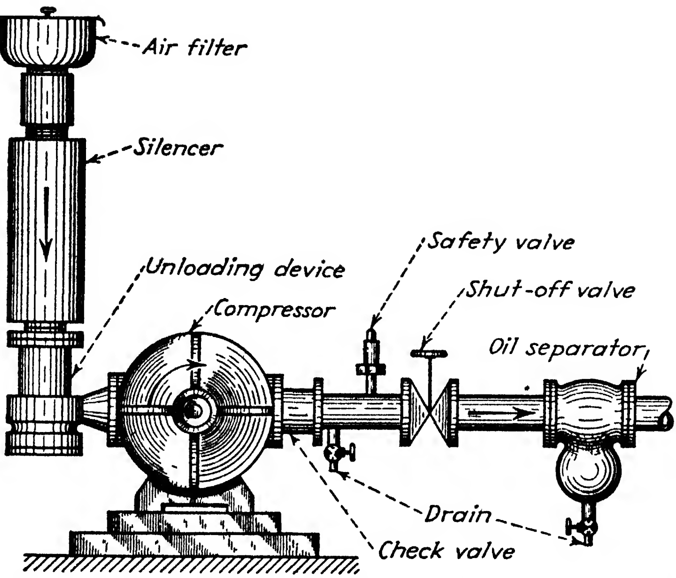


FIG. 276.—Typical air-piping arrangement. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

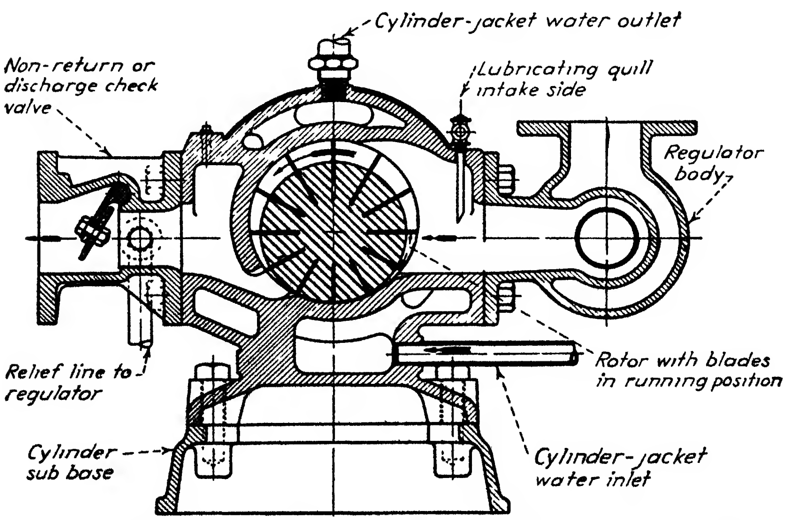


FIG. 277.—Cross section showing water passages and lubricating quill. (*Courtesy of Fuller Co.*)

Wash the bearings in clean oil and cover each part with lubricant as it is replaced.

I. Replace the floating rings and their blades in the cylinder recesses. Be sure the rotation arrows point in the right direction.

II. Install the retaining rings. Drive them lightly to position, keeping the ring square with the cylinder. Be sure the rings are in their correct position—the oil hole in the cylinder and ring must line up. After the retaining rings are in place, see that the floating rings turn freely by hand.

III. Assemble the low-pressure rotor and shaft, low-pressure cylinder head, bearing, and stuffing box as follows:

1. Block up the rotor and shaft on end and replace the low-pressure shaft-sealing ring in its groove.
2. Place the cylinder head on the shaft. Put thin metal strips over the sealing ring so that it will not be pinched when the head slips over. Then remove the strips.
3. Press the thrust bearing on the shaft as follows:,
   a. Transfer the scribed lines on inner and outer races from open face to back face on one of the bearings.
   b. Place this bearing on the shaft so that the back face (printed side) is visible. Note the position of the scribed line on the outer race by making a chalk mark on the face of the cylinder-head flange so that the chalk mark is in line with the scribed line on the outer race.
   c. Turn the inner bearing race so that its scribed line coincides with the shaft keyway.
   d. Place the second bearing on the shaft with the back face (printed side) in contact with the first bearing. (Bearings are in the back-to-back position.) The scribed line on the inner race must line up with the shaft keyway, and the line on the outer race must coincide with the chalk mark on the cylinder head.
   e. Press the bearings into position, applying pressure to the inner races only (Fig. 282). Do not hammer the bearings or apply pressure to the outer races.
4. Replace the lock nut and washer and draw the nut up tightly.
5. Attach the stuffing box, insert the packing, and draw the gland up lightly.
6. Measure the clearance between the rotor and the cylinder head. It should be from 0.004 to 0.006 in. and must be the same at all points on the rotor circumference Fig. 283*A*.
7. Place cup grease in the piston-ring grooves, thus preventing the rings from dropping too far out of central position. Put the rings in the grooves, being careful not to spring them more than necessary.
8. Slide on the cylinder-head gasket. If a new gasket is required, use one of the same thickness as that stamped on the old gasket

name plate. If the name plate is not legible, use gasket material 0.001 to 0.002 in. thicker than the old one to provide for compressing the material. Since sliding-vane machines can compress to fairly high pressures, leakage past the rotor and at the shaft seals can be a major item. An overthick gasket between the case and end cap permits serious air slippage. Apply a film of cylinder oil and graphite or cup grease to the gasket face next to the cylinder head to facilitate later removal of the head.

9. Slide the rotor and shaft into the cylinder, taking care not to scratch the cylinder or pinch the piston rings in the interstage partition. See that the rotor vanes are returned to their exact position in the slots.

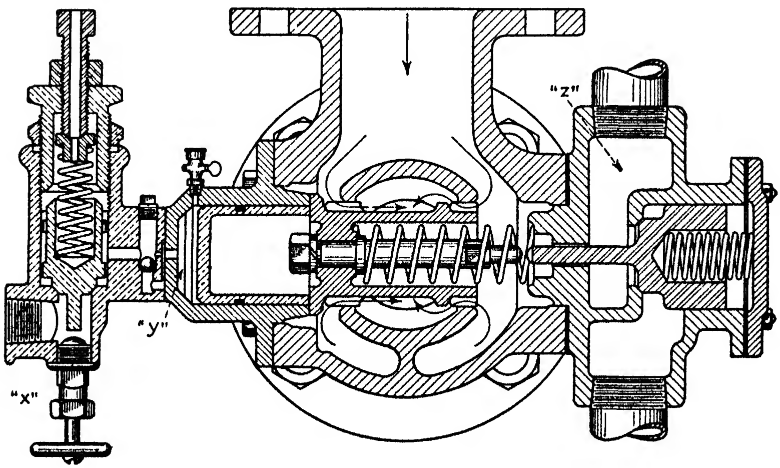


FIG. 278.—Intake-valve unloader. (*Courtesy of Fuller Co.*)

10. Before tightening the cylinder head, raise it slightly until the dowel holes line up. The dowels determine the clearance between the rotor and the cylinder wall, and care must be taken not to damage them or their reamed holes.

IV. Slide the high-pressure rotor on the shaft and replace the vanes. Note that the shaft is fitted for two keys. The key marked *A* must fit in the rotor-key slot marked *A*.

V. Replace the high-pressure sealing ring.

VI. Attach the high-pressure gasket and cylinder head. Place thin metal strips over the sealing ring so that the cylinder head will not pinch the ring as it slips over. Install dowels as described on the low-pressure cylinder head.

VII. Replace the bearing and bearing cover.

VIII. Attach the shaft half coupling, heating it in oil to 250 F if necessary. (Do not use excessive force to drive it to place.)

IX. Return the machine to its bedplate and line up.

TWO-IMPELLER COMPRESSORS

General Characteristics.—These compressors are primarily suited to pressures from 10 to 15 psi and for higher pressures by operating two or more units in series.

They are built in a wide range of sizes for capacities up to 50,000 cfm, and with the rotors running in either sleeve (Fig. 284), or roller bearings (Fig. 285). New sleeve bearings can be installed by taking off the cap and rolling out the bottom bearing shell. New bearings automatically restore the impellers to their

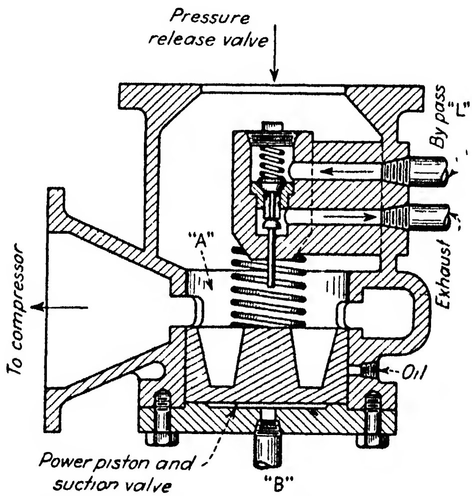


FIG. 279.—Power piston operated by solenoid valve. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

original proper clearance in relation to the case. Double-row roller bearings are considerably oversized to keep bearing pressures lower than those used in ordinary practice.

Operating Principle.—Two impellers (Fig. 286) mounted on parallel shafts rotate in opposite directions. Their contour and finish are such that a clearance of a few thousandths of an inch is precisely maintained at all points of a revolution by a pair of accurately cut timing gears.

In Fig. 286, impeller *A*, rotating in a clockwise direction, is in a position where its bottom tip has just cut off the opening to the inlet port, and the top tip is ready to open to discharge. Air is trapped between the left-hand side of the impeller and the casing, which is not open to either the suction or discharge. As the impeller continues to rotate, its top tip opens the discharge and

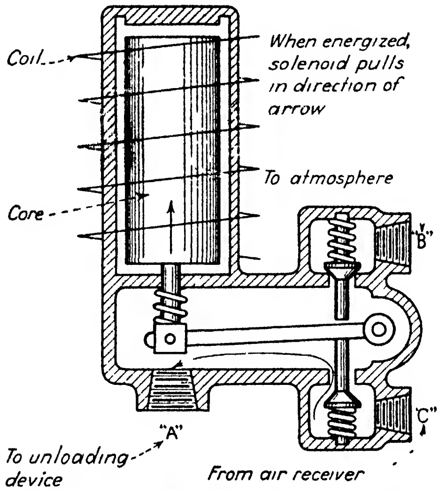


FIG. 280.—Solenoid-operated pilot valve. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

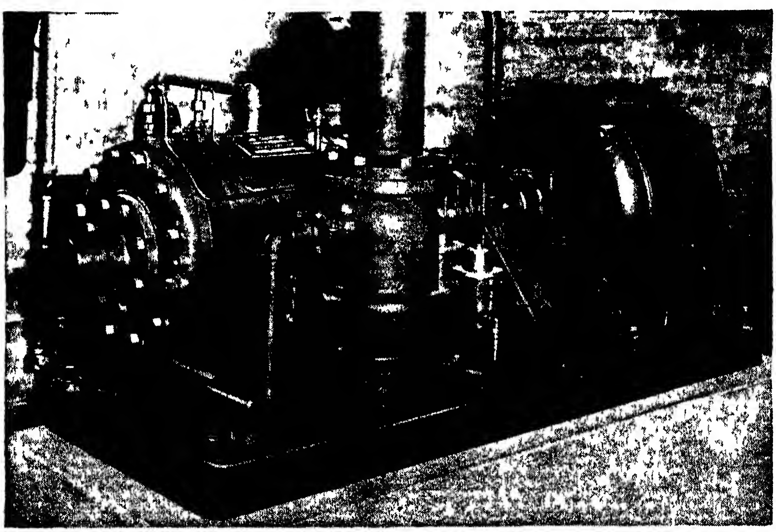


FIG. 281.—Intake unloader with its pilot valve.

the bottom tip pushes the enclosed air into the discharge system. The action is repeated twice for each impeller revolution, or four times for each turn of the driving shaft. The speed determines the capacity in cubic feet of air delivered, while the generated pressure is that needed to force the air into the system. Because there is no internal rubbing between lobes, no internal lubrication is required, except a small amount of grease at the shaft shoulders. Air is therefore delivered free of oil.

The curve (Fig. 287) illustrates constant-speed operation, as when driven from induction or synchronous motors. The capacity remains practically constant regardless of pressure changes. The horsepower varies with the discharge pressure.

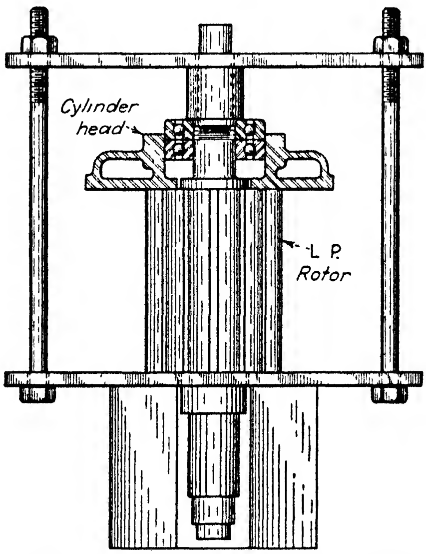


Fig. 282.—Pressing thrust bearing in position.

Figure 288 gives the characteristic curve of a three-speed motor-driven unit. This arrangement gives three efficient volume points and shows the horsepower saving that can be made with this drive when variable volumes are handled.

Figure 289 illustrates the characteristic curve of variable-speed operation, as when the machine is driven from a steam engine, steam turbine, or gas engine. In this case horsepower varies directly with volume and pressure. Figure 290 shows efficiency curves of a 588-rpm, 12,500-cfm motor-driven machine.

Driving Gears.—The multiple-contact tooth gears (Fig. 291) have three or four pairs of teeth actually in working contact at all times. Because of this, intensity of tooth pressure is low and a more effective oil film is maintained with consequent smooth operation and long life. Gears are made from close-grained, hard semisteel castings with teeth cut from solid blanks.

As clearance must be maintained between the lobes, provision must be made in the driving gears to adjust for tooth wear. The gib key in one gear (Fig. 291) permits accurate location of the lobes with relation to each other and obviates the necessity of using an offset key to shift the gear when setting the lobes or taking up for tooth-surface wear. Wear occurs at shaft shoulders only if excessive end-thrust or out-of-level conditions exist.

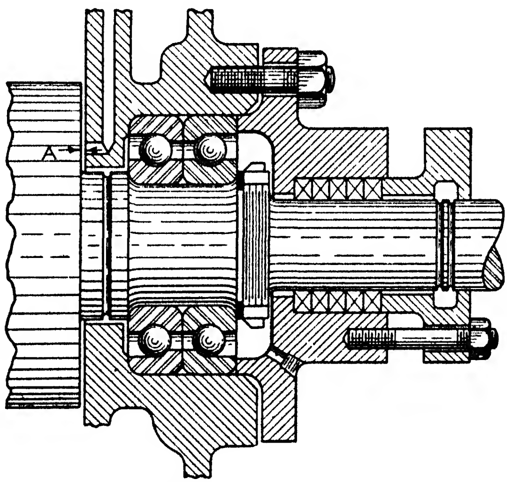


FIG. 283.—Sectional view of rotor end clearance.

End Clearance.—Axial clearance between the impeller ends and casing head is extremely small and is just sufficient for expansion caused by the heat of compression. On sleeve-bearing machines the rotor must be free to center itself between the heads.

For this reason it is important that there be enough clearance between the coupling flanges so that the driving shaft will not exert any thrust on the compressor shaft. If the casing head should be removed, the same thickness of gaskets must be used in reassembling to ensure correct clearance. Compression of material must be considered when measuring gasket thicknesses. Excessive end clearance will cause reduced capacity because of increased slippage.

Air Slip.—Efficiency depends largely on the amount of friction and slip. Slip is the leakage back to the inlet side which occurs because of clearance between the casing and rotating parts. The quantity of air leaking back can be shown as cfm, but for convenience is usually expressed in rpm. Thus it may be said that the slip in rpm is the number of revolutions the machine must make to hold the pressure against leakage or to make up for leakage air.

Pressure and speed control the slip to a great extent, as is apparent by the efficiency curves in Fig. 294. A machine

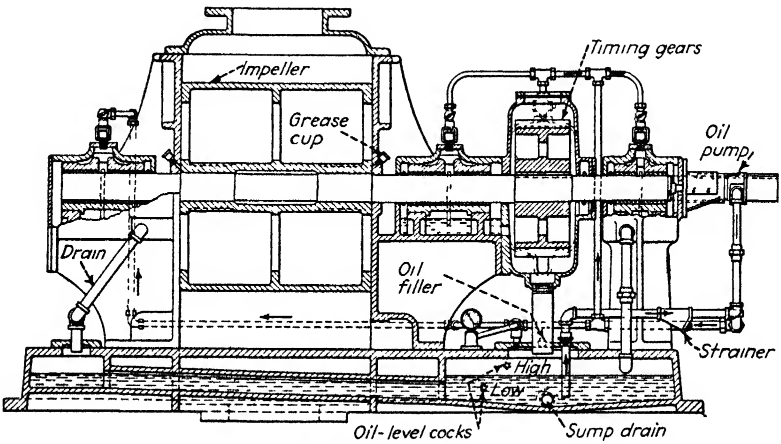


FIG. 284.—Ring-oiled sleeve bearings and grease-lubricated shaft shoulders. (*Courtesy of Power.*)

developing a given pressure at low speed has a greater slip than it would have if operated at a higher speed of the same pressure. Since these machines are positive-displacement compressors, they require a safety valve in the discharge line.

Load Control.—The two-impeller unit operating at constant speed, as when driven by electric motors, delivers constant volume. Excess capacity is relieved by a hand-adjusted or automatic by-pass valve set to maintain the desired suction or pressure on the inlet or discharge side. A diaphragm-actuated by-pass valve, generally loaded with weights, serves on installations where extremely close regulation is not required.

When close regulation is needed, a Huntoon-actuated by-pass valve is used. This regulator maintains constant suction or inlet pressure down to a ½-in. water column. The governor consists of a bowl in which a float, weight-loaded and subjected to the inlet-air pressure, operates in either water or mercury as determined by the pressure condition.

Diaphragm-actuated relay governors using a hydraulic cylinder on the by-pass valve are used where closer regulation than a ½-in. water column is required. On steam- or gas-driven units, control may be by either hand throttling or automatic governor—either

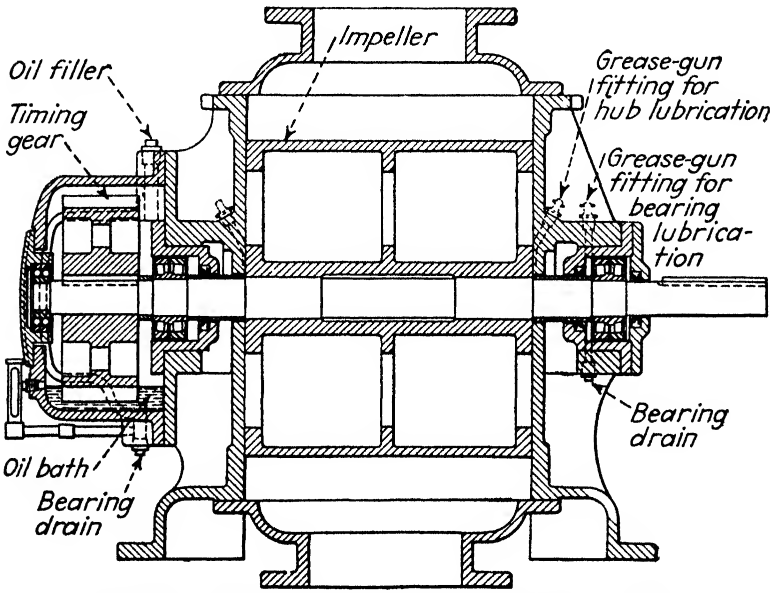


FIG. 285.—Grease- and oil-lubricated ball and roller bearings.

of the diaphragm or Huntoon design—mechanically connected to the throttle valve controlling the unit speed to maintain the desired pressure conditions.

On large steam-engine-driven units, controls frequently function by changing the cutoff, or, working through pilot valves, operate a hydraulic-power cylinder. Regulators can be furnished that will completely unload large constant-speed motor-driven units for reduced volumes by by-pass-valve operation to maintain the desired pressure or suction for which the regulator is set.

Shaft Packing.—The improved packing gland (Fig. 295) serves to prevent air leakage where the shafts pass through the head-

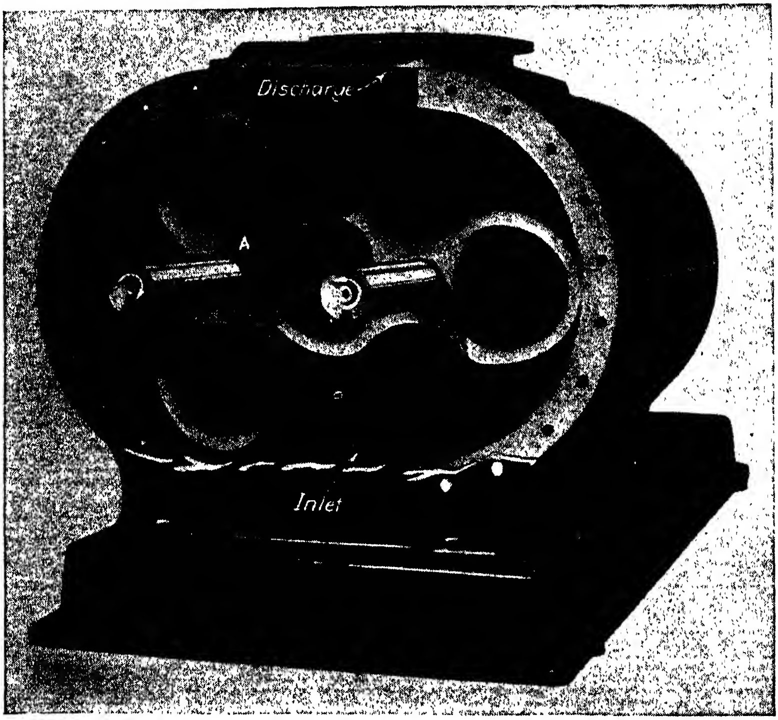


Fig. 286.—Two-lobed impellers rotate in opposite directions and carry the air in pockets between them and the case. (*Courtesy of Roots-Connersville Blower Corp.*)

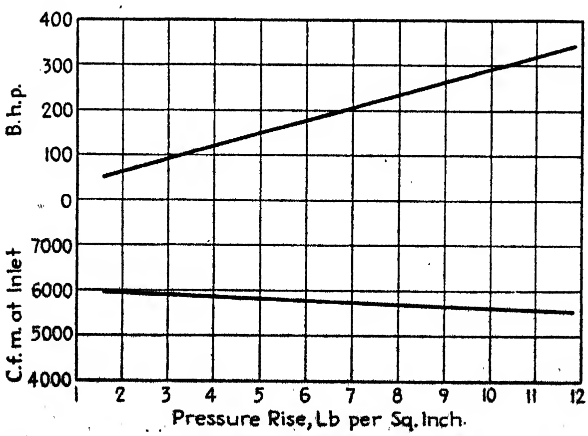


Fig. 287.—Characteristic curves of a constant-speed two-impeller compressor. (*Courtesy of Roots-Connersville Blower Corp.*)

plates. Packing is vented to the suction side by a lantern ring in the back of the gland so that it is only necessary to pack against suction pressure.

Cooling Arrangements.—The two-impeller compressor does not require cylinder cooling because of its low-pressure discharge. When two units operate in series to make a two-stage compressor, the air usually goes through an intercooler (Fig. 296) before entering the high-pressure machine.

Field of Application.—Even though the two-impeller compressor is classed as a positive-displacement machine it more

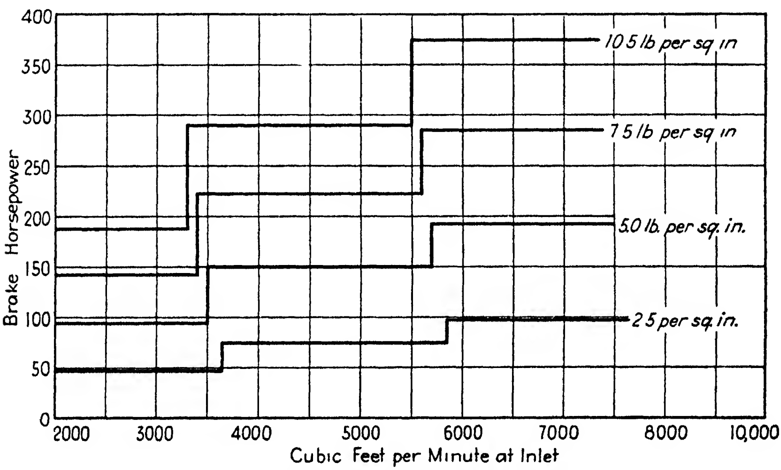


FIG. 288. -Characteristic curves of a multispeed two-impeller machine. (*Courtesy of Roots-Connersville Blower Corp.*)

nearly approaches the single-stage centrifugal compressor in its field of application than do any of the other rotary compressors. These swing, to some extent, to the reciprocating field where higher pressures and lower capacities predominate.

LIQUID-PISTON COMPRESSOR

General Characteristics.—This machine consists of a round multiblade rotor revolving in an elliptical casing (Fig. 297) partly filled with water. A continuous supply feeds into the casing, any overflow passing out with discharged air to settle in the separator (Fig. 298). If the seal-water temperature is sufficiently low, air delivered to the distribution system can be maintained above the dew point. This feature obviates the

TABLE 33.—SIZE, CAPACITY, HEAD, AND POWER REQUIREMENTS OF A LIQUID-PISTON COMPRESSOR

Size and speed		Pressure, psi 5	10	15	20
K-4	Capacity	340	339	310	
870	Motor size	15	20	25	
K-5	Capacity	540	509	480	350
680	Motor size	25	30	40	50
K-6	Capacity	1,020	990	870	650
570	Motor size	40	60	75	100
K-7	Capacity	1,480	1,380	1,050	
360	Motor size	50	75	125	
K-8	Capacity	2,080	1,980	1,530	
306	Motor size	75	125	150	

Size and speed		Pressure, psi 5	10	15	20	25
L-1	Capacity	33	30	24	14	
1,750	Motor size	2	3	3	5	
L-2	Capacity	68	64	55	37	
1,750	Motor size	5	5	7½	7½	
L-3	Capacity	154	146	138	127	102
1,750	Motor size	10	15	15	15	20
L-4	Capacity	210	210	180	160	
1,150	Motor size	10	15	20	20	
L-5	Capacity	425	410	400	385	340
1,150	Motor size	30	40	40	50	60
L-6	Capacity	745	725	695	685	480
870	Motor size	40	50	60	75	100
L-7	Capacity	1,275	1,200	1,150	1,075	
680	Motor size	75	100	100	125	
L-8	Capacity	1,930	1,780	1,625	1,300	
450	Motor size	100	125	150	200	

Size and speed		Pressure, psi 10	15	20	25	30	35	40	45
H-2	Capacity	65	62	60	58	58	56	54	51
3,500	Motor size	10	10	10	15	15	15	15	15
H-3	Capacity		123	122	121	120	120	118	111
3,500	Motor size		20	25	25	25	25	30	30
H-4	Capacity			116	104	89	68		
1,750	Motor size			15	20	20	25		
H-5	Capacity			305	294	285	274	260	
1,750	Motor size			40	50	50	60	60	
H-6	Capacity			510	500	490	480	470	
1,440	Motor size			75	75	100	100	100	
H-7	Capacity			945	930	900	875		
1,150	Motor size			125	150	150	150		
H-8	Capacity			1,125	1,090				
720	Motor size			150	150				

necessity of installing an aftercooler. For exacting services the compressor seal water can be chilled to eliminate any possibility of condensation in the pipe lines.

Low-pressure units are good for 35 psi (Table 33). Special designs are available in single stage for pressures up to 75 psi. For higher pressures one or more units operate in series. Processes requiring a large quantity of air at low pressure and smaller

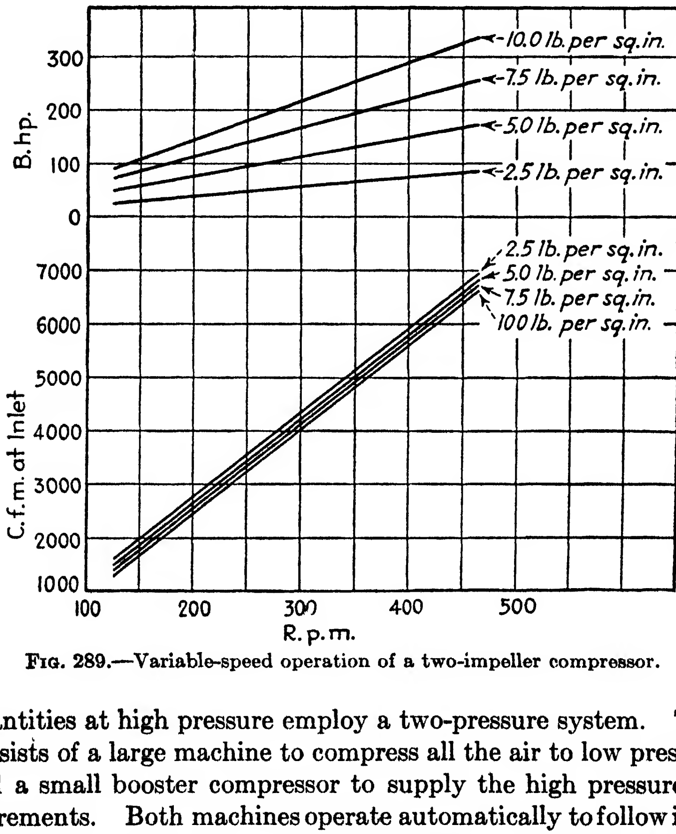


FIG. 289.—Variable-speed operation of a two-impeller compressor.

quantities at high pressure employ a two-pressure system. This consists of a large machine to compress all the air to low pressure and a small booster compressor to supply the high pressure requirements. Both machines operate automatically to follow individual load fluctuations. Capacities range as high as 5,000 cfm. Figure 299 gives performance curves.

The shafts ride in roller or ball bearings mounted in dirtproof housings. The single impeller (Fig. 300) overhangs, both bearings being located between the rotor and driving motor, whereas the double-impeller unit (Fig. 301) carries the rotor between bearings. Some machine bearings are oil-lubricated, while others use grease.

Operating Principle.—Curved rotor blades (Fig. 302) project radially and form, with their supporting side shrouds, a series of pockets or cells around the rim. Rotation at a high speed throws the water out from the center by centrifugal force, resulting in a solid ring of water revolving in the casing at the same speed as the rotor, but following the elliptical contour of the casing. This

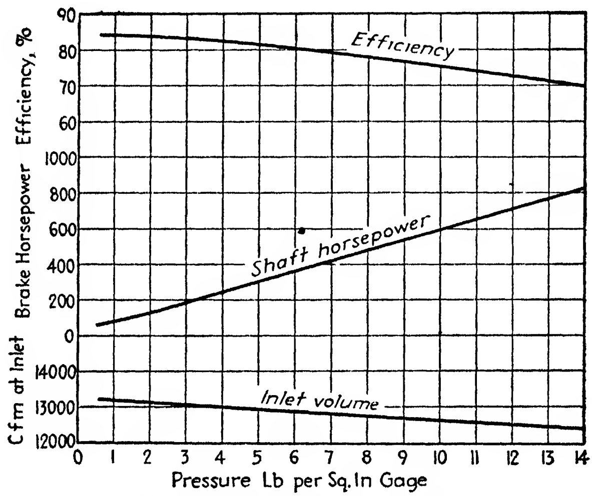


FIG. 290.—Volume, horsepower, and efficiency curves of a 588 rpm, 12,500 cfm, 12 psi, two-impeller compressor. (*Courtesy of Roots-Connersville Blower Corp.*)

alternately forces the water to enter and recede from the cells in the rotor at high velocities.

The four-pocket chamber E, containing the inlet and discharge ports (called the cone), surrounds the shaft; being cast integral with the end bell, it remains stationary. The impeller blades, held and supported by a shroud ring, revolve as a unit around this chamber.

Starting with the cell at A, which is now filled with water and moving in a clockwise direction, we can follow its movement. As the cell moves over inlet port B, centrifugal force causes the liquid to move out into the ellipse so that air takes its place. This action continues until the cell is filled with air, which it carries along until it reaches point C. Here the restricted space causes the water to reenter the cell, forcing the air out from the discharge port.

Forcing the water gradually to enter and recede from the rotor chambers twice each revolution gives an even flow of air with minimum pulsating effect. Efficient operation requires close running clearance between the impeller and cone. A small quantity of water supplied continuously removes the heat of compression.

A certain amount of wear, caused by water erosion and foreign matter entering with the air, increases the rotor clearance and must be compensated for by adjusting at the outer end bearing and at cone flanges. Wear takes place slowly, making frequent internal inspections unnecessary. The manufacturer does not

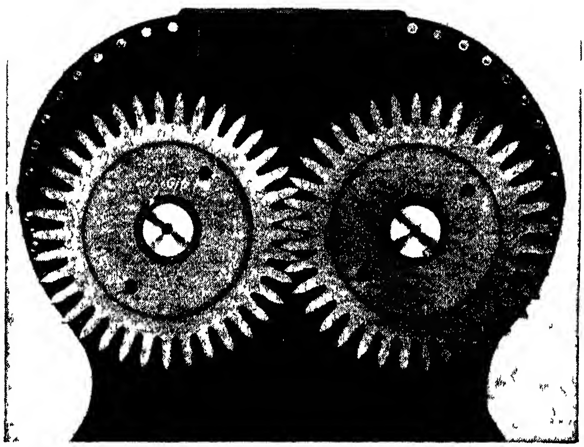


FIG. 291.—Gib key allows gear adjustment to maintain impeller clearance.

recommend frequent dismantling unless output tests show a definite drop in capacity.

Starting and Operation.—Never start the compressor before the sealing water is turned on, or it will be seriously damaged. Before applying power, turn the unit over by hand to see if the rotor is free. If it will not turn, loosen the head and rotate several times to remove any rust adhering to the rotor, then retighten the head. Once the machine runs there is no danger of its binding.

After freeing the rotor, fill the casing about half-full of water, then start the machine, and adjust the cooling-water flow. The water-adjusting needle valve is set at the factory for a supply

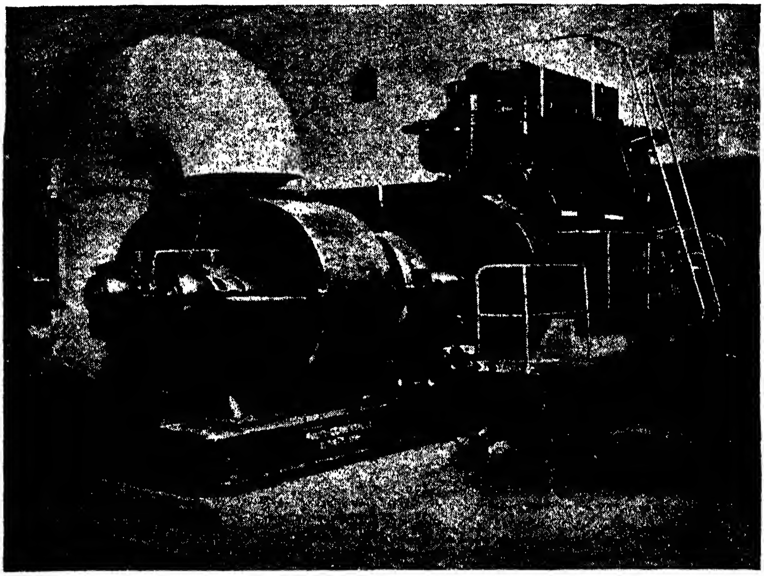


Fig. 292.—A 10,000 cfm compressor driven by an engine using sewage gas as fuel. (*Courtesy of Compressor Air Institute.*)

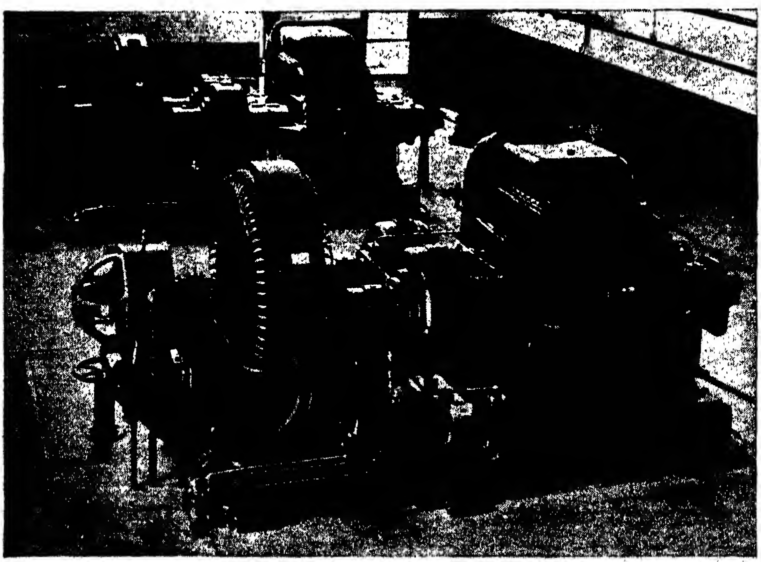


Fig. 293.—Synchronous-motor drive for constant-speed service. (*Courtesy of Power.*)

pressure of from 30 to 75 psi. Check the water pressure before changing the needle-valve adjustment. If the water pressure is below 30 psi loosen the lock nut 200 (Fig. 300) and unscrew the needle valve 171 slightly. Screw the needle valve in for pressures above 75 psi. If local water pressure fluctuates excessively, install a pressure-regulating valve.

Do not reduce seal-water flow to the extent that the compressor runs warm. A warm machine means warm discharge air that contains excess vapor that will condense in the lines. If the compressor heats up, inspect the seal-water strainer and check the water pressure.

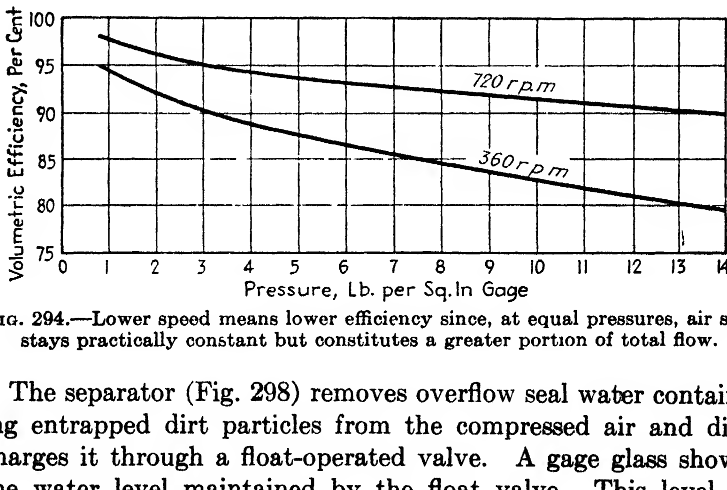


FIG. 294.—Lower speed means lower efficiency since, at equal pressures, air slip stays practically constant but constitutes a greater portion of total flow.

The separator (Fig. 298) removes overflow seal water containing entrapped dirt particles from the compressed air and discharges it through a float-operated valve. A gage glass shows the water level maintained by the float valve. This level is normally low, and, if it shows any appreciable rise, clean the float valve and separator immediately. Clean the inlet-water strainer 189 (Fig. 300) regularly.

Locating Trouble.—If trouble is experienced in securing full output or in maintaining full pressure, check these points before dismantling:

1. Proper amount of seal water.
2. Rotation—check against the arrow mounted on the compressor head or bearing bracket.
3. Correct speed.
4. Incorrect assembly—parts may not have been assembled correctly if the compressor was dismantled.
5. An obstructed inlet.

The compressor operates at best efficiency with the clearance between the rotor and cone as adjusted at the factory. Under constant use, this clearance may increase to such an extent that the compressor must be dismantled. Uniform wear may be compensated for by removing gaskets between the head and body to bring the compressor back to capacity. If cones are pitted and grooved, new ones must be installed or the old ones reclaimed by machining. This involves cutting a taper of 8 deg that must be extremely accurate.

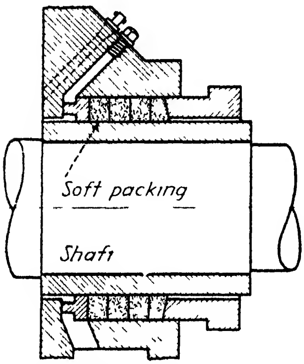


FIG. 295.—Soft-packing shaft seal for two-impeller compressor.

Dismantling the Compressor.—Before dismantling the unit, mark each part accurately to ensure correct assembly.

All ring shims located under the face of outboard bearing outer cap *A* (Fig. 301) should be retained. Measure the thickness of each set so that they will not be interchanged. Small machines (Fig. 301) have the heads and cones in one piece.

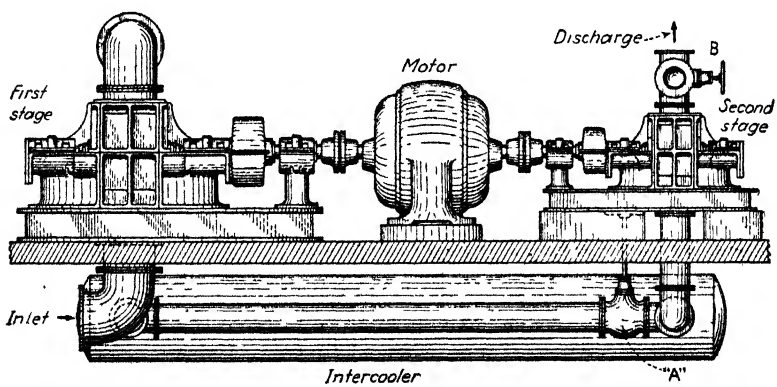


FIG. 296.—Two-stage arrangement of two-impeller compressors with intercooler.

Before removing either head, take off the outer bearing cap and then replace the nuts to hold the inner cap in place. Take off the shaft-bearing lock nut and remove the head, cone, and bearing assembly as one unit.

If it becomes necessary to replace the separately assembled cones (Fig. 303), set the head and cone assembly on the floor with the small end of the cone pointing downward. Then lay blocking around the cone up to within 1 in. of the head and lightly hammer around the head-side wall with a wooden or rawhide mallet until it comes free.

With one head removed from the compressor, the rotor and shaft assembly can be withdrawn. The rotor has a press fit on the shaft; when removing it apply pressure against the rotor

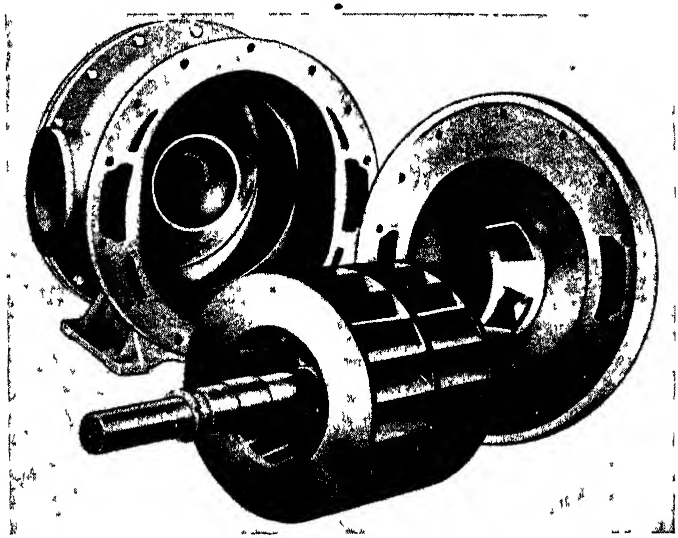


FIG 297.—Component parts of a liquid-piston compressor. (*Courtesy of Nash Engineering Co*)

hub after removing the shaft-sleeve setscrew. Do not apply pressure to the impeller shrouds. The packing sleeves also have a shrink fit on the shaft. Always mark the rotor position on the shaft so that it can be returned to its exact former position. This is important.

Reassembling the Machine.—When fitting a new cone, set the head on blocks with its outer surface facing up and slip the cone into place. Then remove it and apply a thin, even film of fine grinding compound where the conical surfaces meet. Return the cone and rotate it several times in each direction. Do not apply downward pressure; it is necessary only to polish the surfaces to make them tight.

After fitting, relieve the outside edge of the head's conical bore by scraping. Center the cone in its proper position in the head. If all parts have been marked as previously outlined, no difficulty

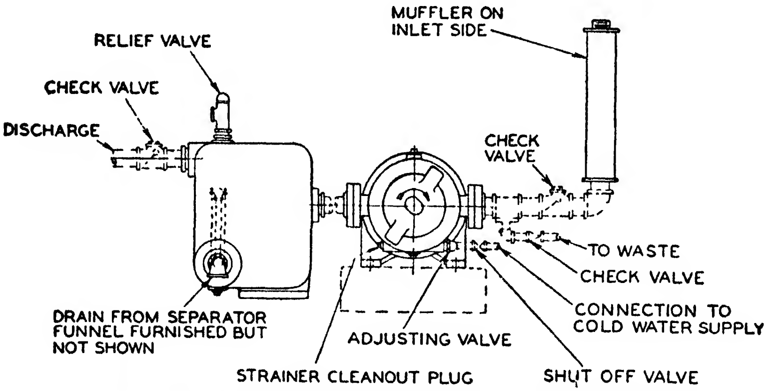


FIG. 298.—Piping arrangement for a liquid-piston compressor plant.

will be experienced. Tighten the cone nuts evenly and use a dial indicator to check squareness with the gasket face of the head.

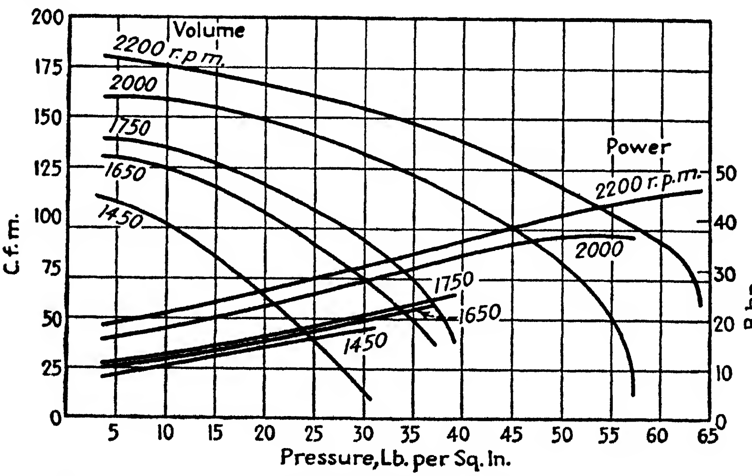


FIG. 299.—Typical performance curve of a single-stage liquid-piston compressor. (*Courtesy of Nash Engineering Co.*)

Rotor end travel is determined by a thickness gage after both ball bearings are installed and the outboard bearing caps are assembled without shims. The end caps of this bearing are held together by through bolts that move freely in clearance holes drilled through the bearing bracket. Therefore, moving the

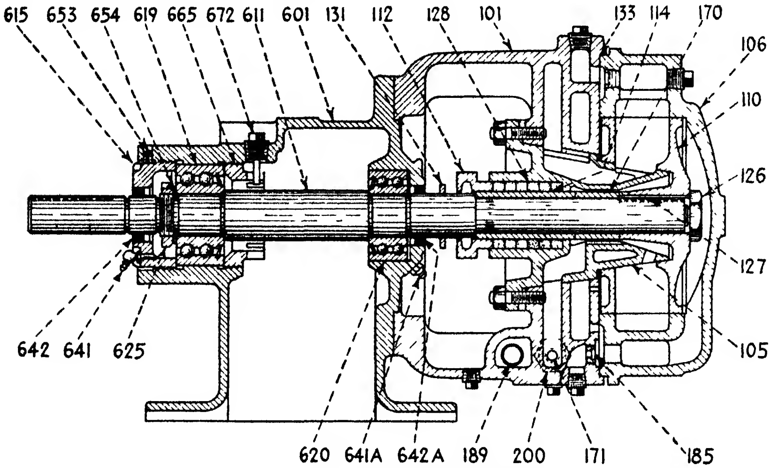


FIG. 300.—Cross section of an overhung single-impeller compressor.

101	Bracket	133	Gasket lobe	620	Bearing—pump end
105	Cone	170	Spacer (471 only)	625	Bearing nut
106	Lobe	171	Needle valve	641	Grease connection
110	Rotor	185	Orifice plug	641A	Grease connection
112	Gland	189	Strainer	642	Grease seal ring
114	Shaft sleeve	200	Needle-valve nut	642A	Grease seal ring
126	Lock screw	601	Pedestal bracket	653	Setscrew
127	Key rotor	611	Shaft	654	Lock washer
128	Packing	615	Bearing cap	665	Notched adjusting nut
131	Water slinger	619	Bearing—drive end	672	Lock plug

end caps and bearing race longitudinally moves the rotor and shaft as well.

TABLE 34.—END-TRAVEL ADJUSTMENT OF NASH LIQUID-PISTON COMPRESSOR

Machine Size	End Travel of Rotor, In.
L1 and H2	0.010–0 015
L2 and H3	0.015–0.025
K2, L3, and H4	0.020–0.030
K3, L4, and H5	0.030–0.040
K4, L5, and H6	0.040–0.050
K5, L6, and H7	0.050–0.060
K6, L7, and H8	0.060–0 070
K7, L8, and H9	0.070–0.080
K8, L9, and H10	0.080–0.090

Referring to Fig. 301, cross section of the outboard-bearing assembly, determine rotor end travel as follows: Tighten the three cap screws *B* in the outer cap *C* until the rotor binds in the cone on the drive end. With a thickness gage measure the distance at *A*. Then slack off the cap screws and insert three setscrews *D* in holes provided in the outer cap. Tighten the setscrews until the rotor binds in the cone on the outer end. Again measure distance *A*. The rotor endwise movement is the difference between the two measurements taken at *A*. If the end travel is greater than standard (Table 34), remove some of the body gaskets; if it is less, add to the body gaskets. Too much end travel lowers compressor efficiency, while too little may cause the rotor to bind.

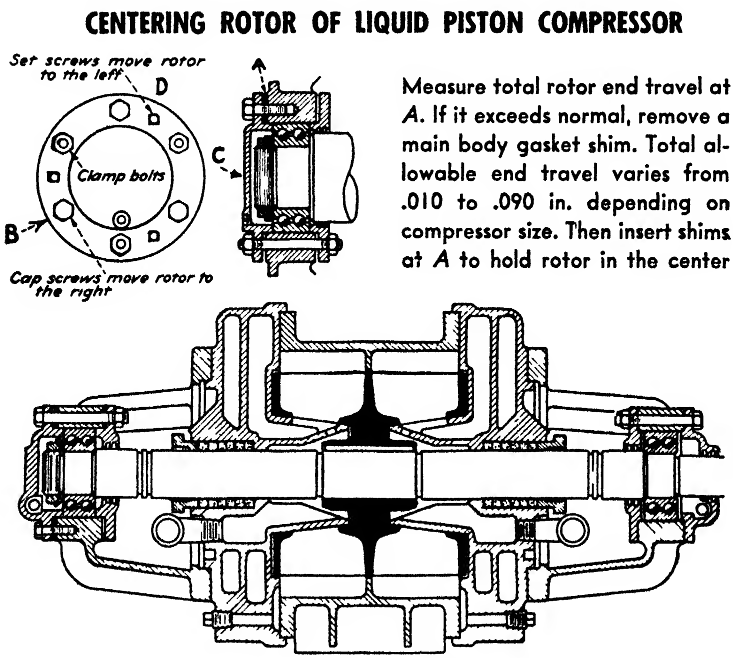


FIG. 301.—Rotor clearance affects efficiency; keep it adjusted properly. (*Courtesy of Power.*)

Adjust the outer end bearing assembly to centralize the rotor in the casing as follows: Tighten setscrews *D* until the rotor binds on its outer end. Then loosen the setscrews and tighten cap screws *B* until the outer cap moves half the amount of endwise movement as previously measured. Now carefully measure the distance *A* to learn the shim thickness to be installed between the face of the outer cap and bearing bracket at *A*. Then tighten the cap screws *B* and remove setscrews *D*. The rotor should now revolve without contact.

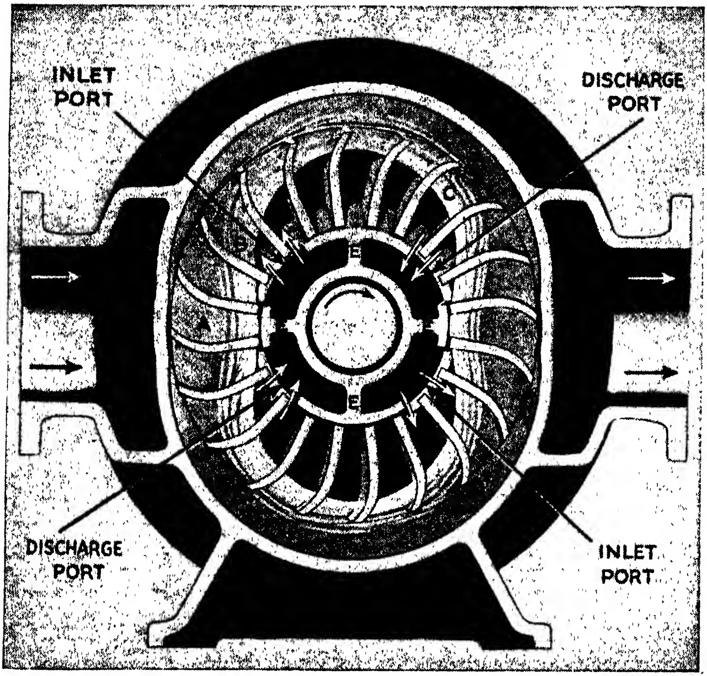


FIG. 302.—Shrouded vanes rotate around inlet and discharge chambers. (*Courtesy of Nash Engineering Co.*)

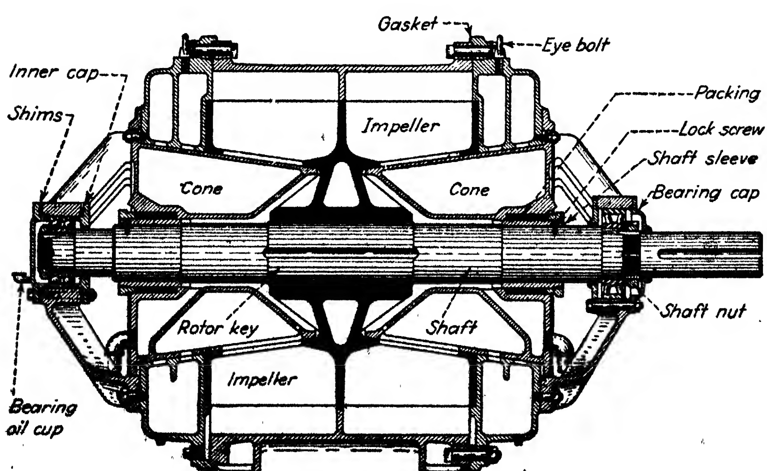


FIG. 303.—Cross section of removable cones and shaft packing.

ELLIOTT-LYSHOLM COMPRESSOR

Development.—Development of the Elliott-Lysholm rotary compressor was undertaken in Sweden in 1943 and is covered by a number of foreign and domestic patents. Approximately thirty machines have now been built and tested, varying in size from 15 to 10,000 cfm.

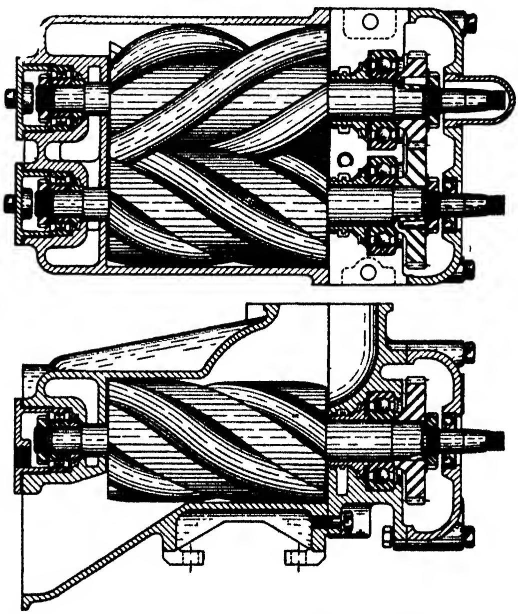


FIG. 304.—Longitudinal section of an Elliott-Lysholm compressor.

Operating Characteristics.—In this compressor (Fig. 304) the air is tranported diagonally by the pair of helically lobed rotors which interact to provide an axial as well as a cross seal. The distinguishing feature lies in its action on each charge of air after it is sealed off from the inlet and before it is brought into communication with the discharge. A charge is initially enclosed in the space bounded by the tooth flanks, casing bore, and end walls.

The rotor helices are so chosen that a particular thread space is completely filled and sealed off from the inlet just as it is entered at its opposite end by a coacting lobe on the other rotor.

Further rotation establishes an axial seal which separates this charge from the charge in the succeeding grooves. As rotation proceeds, this seal moves axially, effecting a reduction in the charge volume and a substantially adiabatic compression. When leading lobes of the grooves pass the boundaries of the discharge port, the compressed air is forced into the discharge. The location of the port thus determines the *built-in* compression ratio.

CHAPTER X

CENTRIFUGAL COMPRESSORS

Classification.—The ASME test code classifies centrifugal compressors and exhausters as those which, if used to compress air initially at standard atmospheric conditions, would increase the pressure by more than 1 psi (27.73 in. of water).

General Characteristics.—Centrifugal machines (Fig. 305) have a pressure range up to and above 100 psi, depending on the number of impellers used. For pressures under 35 psi they

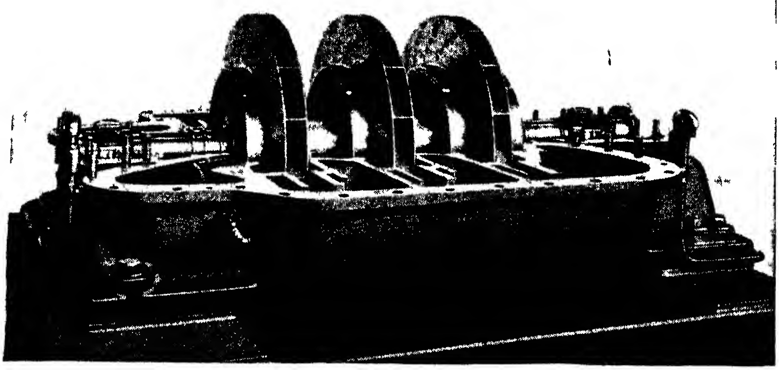


FIG. 305.—A three-stage centrifugal blower with fabricated impeller wheels. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

are classified as blowers, and above that are called compressors. They consist essentially of a casing in which revolve one or more impellers mounted on and supported by a shaft. Centrifugal force causes compression of the air and induces its flow to the periphery of the impeller. This tends to create a low-pressure area at the inlet which causes additional air to enter. As the air passes through the impeller it is accelerated to a high speed; this velocity is then converted into additional pressure by gradual deceleration in the diffuser or volute which surrounds the impeller.

The centrifugal force depends on impeller speed as well as on the density of air being compressed. An impeller, be it a pump or a compressor impeller, will, at a given speed, generate a certain head in feet, no matter what material is handled. For instance, a pump supplying a head of 100 ft of water will, at the same speed, when handling air, give a head of 100 ft of air. However, the pressures in psi will be entirely different because of the difference between the densities of water and air.

The capacity range is practically unlimited; machines have been built to handle volumes as high as 130,000 cfm. The operating-speed range varies from 3,000 to 10,000 rpm, which is considerably higher than that of most centrifugal pumps. They are available in the following types: Single-stage, single- or double-inlet (Figs. 306 and 307); multistage, single- or double-inlet uncooled (Fig. 308); and single-inlet water-cooled (Fig. 309). Water cooling usually starts at pressures above 35 psi.

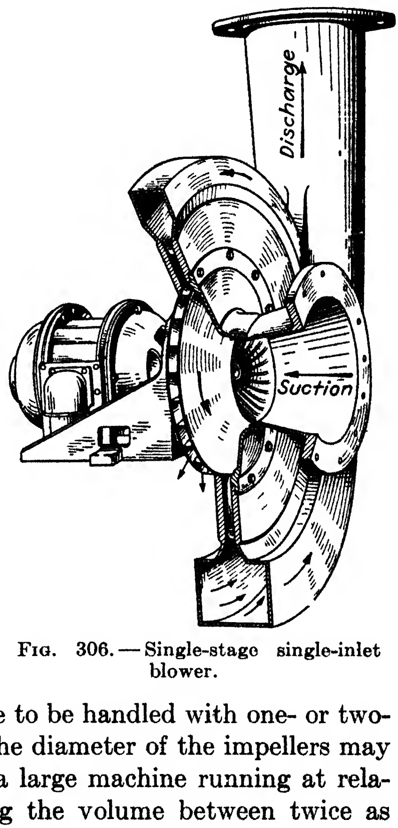


FIG. 306. — Single-stage single-inlet blower.

When large volumes of air are to be handled with one- or two-stage centrifugal compressors, the diameter of the impellers may be unduly great, necessitating a large machine running at relatively slow speed. By dividing the volume between twice as many impellers (Fig. 307) their diameter is kept to a minimum and the operating speed raised. This raises the efficiency and permits using a steam-turbine drive.

Since these machines rotate at high speeds, the impeller must be perfectly balanced. Any slight unbalance causes vibration which may become serious enough to wreck the machine. Repeated painting of the impeller may cause unbalance as it is impossible to spread the paint evenly. Compressors hand-

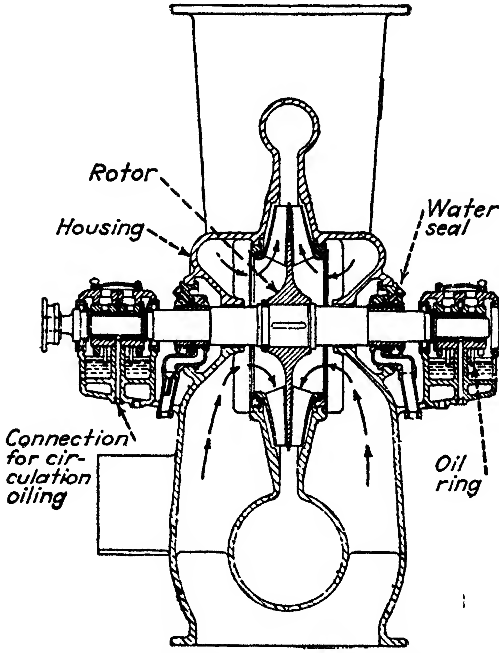


FIG. 307.—Single-stage double-inlet. (*Courtesy of Power.*)

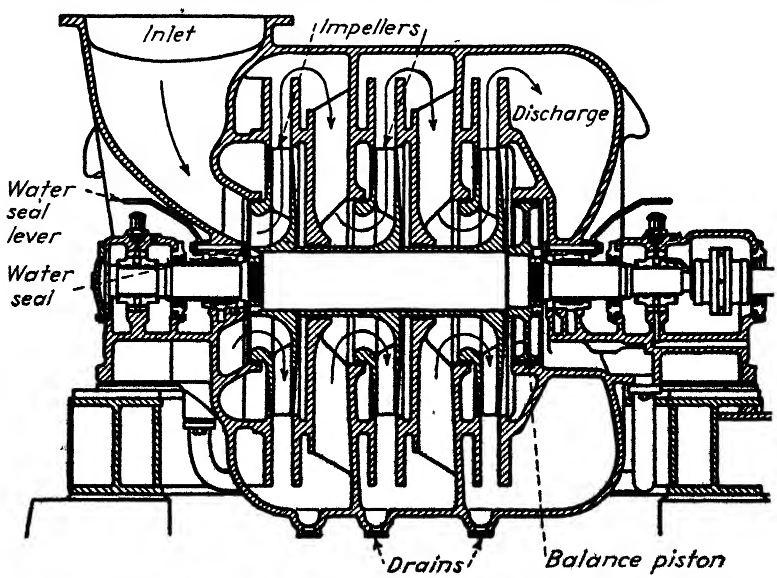


FIG. 308.—Multistage uncooled machine with balance piston. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

ling dirty air may become unbalanced if particles collect on the impellers.

Delivery of air from a centrifugal blower or compressor is without pulsations and produces no disturbance in piping systems. Automatic regulation for constant volume, or for constant inlet or delivery pressure, is easily accomplished. Figure 310 shows a hand-operated discharge valve. Dangerous overpressures are impossible.

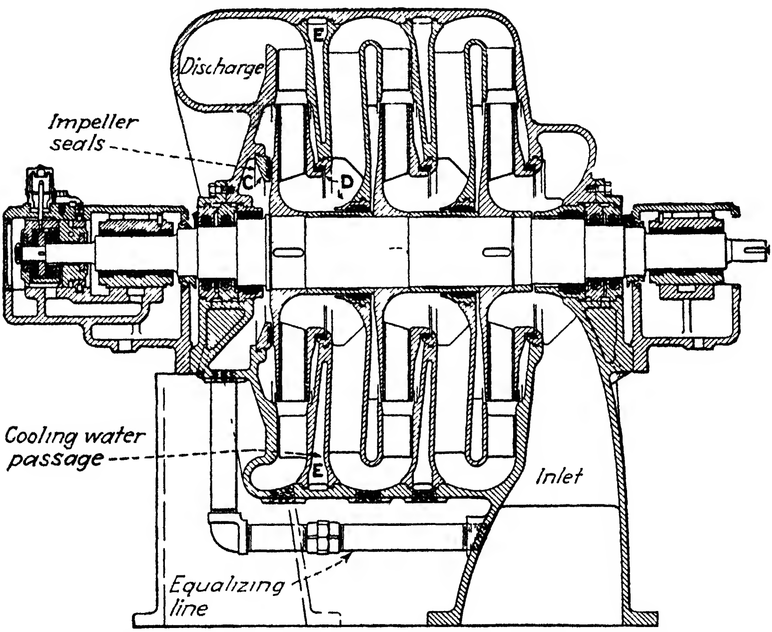


FIG. 309.—Multistage water-cooled unit with balancer seal on last stage impeller. (*Courtesy of De Laval Steam Turbine Co.*)

These machines are compact and simple, have no internal valves, require no internal lubrication, and do not introduce oil or grease into the air. The absence of rubbing parts, except bearings, keeps repair costs fairly low.

Application.—Centrifugal compressors have their principal application where extremely large volumes of air are required at pressures up to 100 psi. Although they have been built for pressures as high as 125 psi, their greatest range of usefulness is in the pressure ranges below 100. Figure 311 shows a group of 25,000 cfm blowers used in copper converter service.

For applications where this machine has been adopted and where the pressure requirements are constant regardless of size, ratings have been standardized. Most manufacturers have a regular line of units developed in convenient capacity steps for blowing blast furnaces, steel converters, cupolas, and exhausting coke ovens.

Operating Principle.—The operating characteristics of a centrifugal compressor can be explained through a study of Fig. 312.

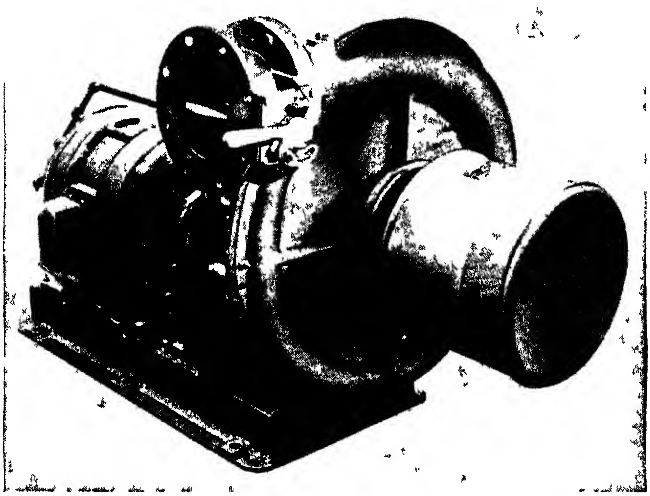


FIG 310.—Single-stage blower using a hand-operated discharge valve. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

Air enters the intake opening and is drawn into the first impeller A_1. Centrifugal action of the impeller produces a pressure increase as shown by the curve *BC*, essentially at a high velocity as shown by *EF*, which is then converted into pressure *CD* in the diffuser B_1 and passage C_1. Curve *FG* shows the drop in velocity as energy is converted to pressure *CD*. The diffuser is a stationary passage of gradually increasing cross section which surrounds the impeller and in which the kinetic energy of the air is converted into pressure as its velocity is reduced.

If the head or pressure built up by a centrifugal compressor is plotted as a function of the delivery volume, curves such as *A* and *B* of Fig. 313 are obtained. Each curve is for a different compressor speed so that we can see what relation speed has to pressure and discharge. Let one unit, when running at speed B, deliver a volume V_2 at pressure P_1. Then let the speed be increased to a higher value as shown by curve A. The unit at this higher speed will then deliver the greater volume V_1 against the old pressure P_1, or it can deliver the old volume V_2 against the higher pressure P_2. At speed B, if we wish to reduce the rate of delivery from V_2 but cannot change the operating speed, the total head will have to be increased. If, for instance, the

FIG. 311.—Four-stage, 25,000 cfm, 18 psi blowers serving copper converters. (*Courtesy of Ingersoll-Rand Co*)

rate of delivery is reduced from V_2 to V_3 and the pressure P_1 maintained, then the pressure P_3 must be dissipated by throttling.

Various relations of the operating factors that govern the action of a centrifugal compressor are shown in somewhat exaggerated form in Fig. 314. In studying these characteristic curves, we see that, although difference in speed affects the pressure and volume output, the shape of the curves is identical. Two different speed curves are shown; as they are similar in shape, we will discuss only the lower curve B. At zero air discharge the pressure P gradually increases to P_1 as the volume increases. Increasing the volume still further causes the pressure to fall to P_2, then to P_3, and finally to zero at maximum discharge.

Assume that the compressor discharges through apparatus which creates a constant resistance regardless of the rate of flow. This constant resistance is represented by the line RP_3, while friction losses through pipes and valves in the distribution system, together with the velocity head, increase with the volume handled. Consequently, the total pressure follows a curve such

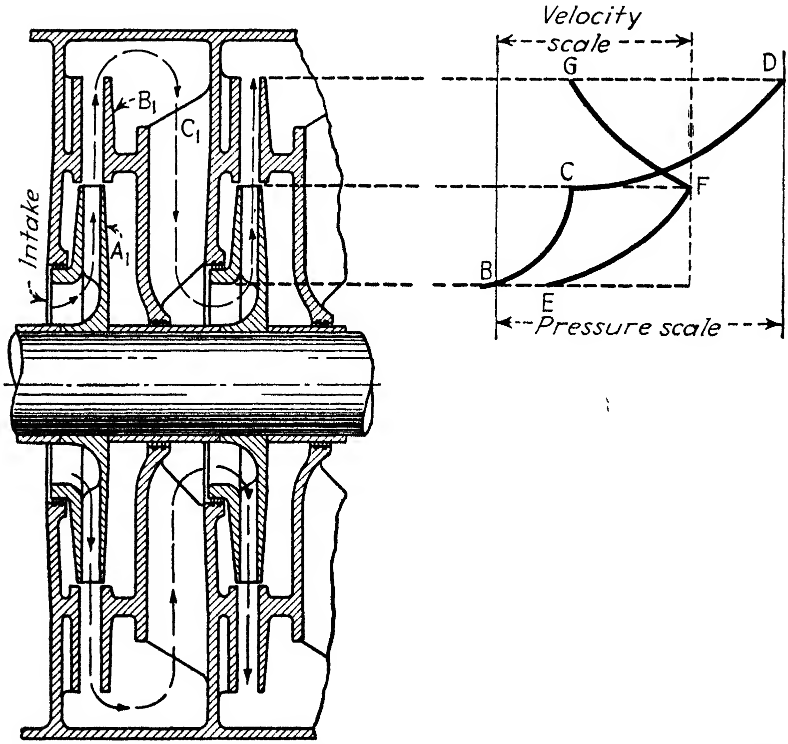


FIG. 312.—Pressure-velocity characteristics of air as it passes through a centrifugal impeller. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

as RP_2. A pressure-characteristic curve of this compressor consists of that part of curve B between points P and P_2. Studying this part of the curve, we find that if the volume is reduced we work back on the curve to point P_1 of maximum pressure. Any further reduction in output causes a reduction in pressure. The section between points P_1 and P is the area of unstable operation.

Between these points the machine no longer delivers a steady flow because a reduction in quantity delivered may mean a machine discharge pressure lower than the system pressure, which causes air to flow back through the compressor. If air is still being used, the system pressure then falls below that at P, the machine immediately begins to deliver air, and its operating point jumps from P to P_2. If this volume exceeds requirements, the pressure rapidly reaches the maximum P_1 and delivery again ceases abruptly. This intermittent delivery or surging is accompanied by a great deal of noise and occurs along the curve from P_1 to P.

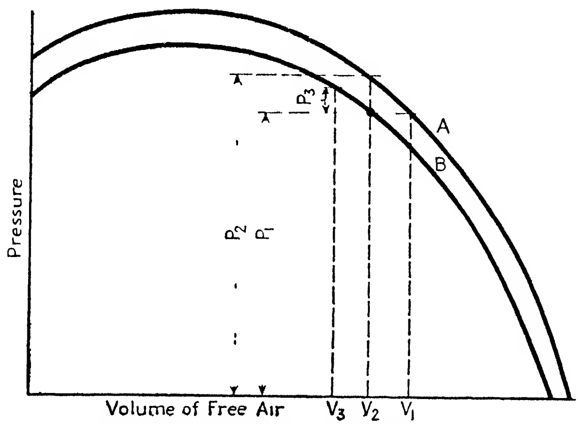


FIG. 313.—Pressure-volume relations of a centrifugal compressor (*Courtesy of Roots-Connersville Blower Corp*)

Long-continued operation below the surging limit P_1 heats up the compressor.

An increase in operating speed increases the compressor's discharge pressure over its entire operating range. In other words, an increase in speed moves the characteristic curve upward on the graph. Also, a decrease in inlet temperature causes the machine to develop a higher pressure over its entire operating range. This is because cold air is heavier and therefore denser. An increase in the density of the air will always move the characteristic curve upward.

Performance curves (Fig. 315) show conditions that affect all centrifugal compressors. The pressure generated is dependent on air temperature because this temperature is one controlling factor of the density or specific gravity of the air. From these curves it is apparent that a machine designed to handle air at a given pressure will not maintain the same pressure if it is used to compress some other gas not having the same specific gravity as air. For this same reason inlet air to the compressor should always be taken from the coolest location in or outside the plant. Another interesting feature is the slope of the power-input curve, indicating the increase of power required as the volume output increases. This introduces the danger of overspeed if the inlet becomes obstructed while the unit is operating at full capacity. For this reason engine- and turbine-driven unit governors should be kept in good operating condition. Overspeed does not generally occur with motor-driven units because motors operating

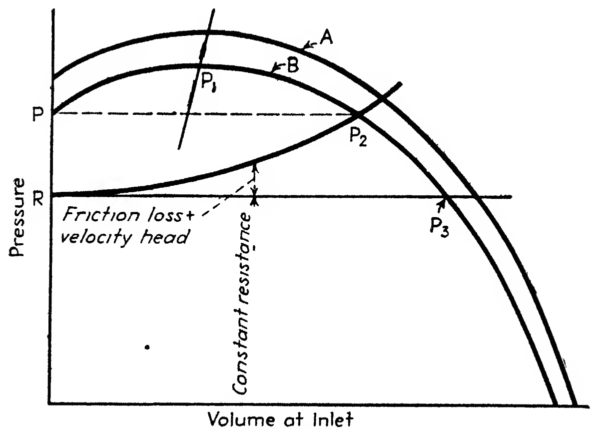


FIG. 314—Operating factors governing the action of a centrifugal machine. (*Courtesy of Roots-Connersville Blower Corp.*)

under normal conditions cannot exceed the speed for which they are designed.

As a rule, centrifugal compressors are seldom guaranteed to maintain stable operation at less than one-half to one-third load, but they can be made to have a flat pressure characteristic that gives stable operation back to 15 to 20 per cent of full load.

The performance of a compressor can best be observed by considering the curves obtained by plotting the increase in head or pressure against the volume of air delivered, usually expressed in cfm at inlet conditions. On the same curve the shaft horsepower may be plotted against inlet volume (Fig. 316).

Since the electric motors used to operate compressors are practically always a-c machines, they may be considered as constant-speed units and only the design speed (indicated at 100 per cent speed on Fig. 317) need be considered. However, steam-turbine-driven machines are subject to operation at a great variety of speeds, making a flexible application and calling for a number of curves, one at each of several different operating speeds, to show the complete picture of change of pressure rise

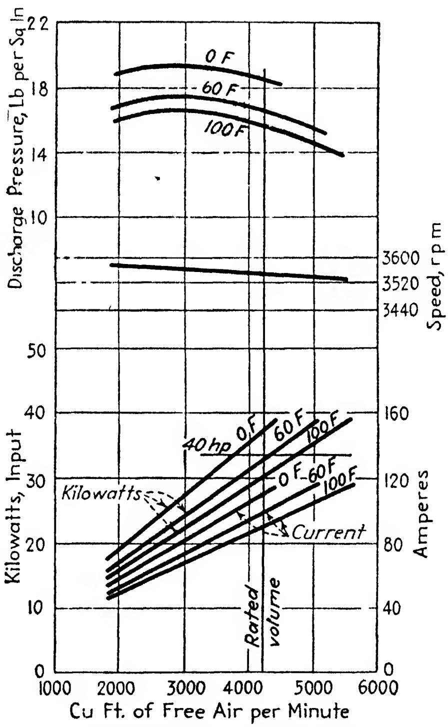


Fig. 315.—Effects of air temperature on discharge pressure and power input. (*Courtesy of Ingersoll-Rand Co.*)

and the resulting change in horsepower when the compressor operates at various inlet volumes at each of these several speeds.

Generalizing, it may be said that, within the limits of compressor design, if it is operated at the peak of its efficiency curve (or at any line of constant efficiency) the following relationships hold approximately true:

1. The volume handled varies directly in proportion to the speed.
2. The pressure rise across the compressor varies with the square of the speed.
3. The shaft horsepower varies with the cube of the speed.

Actually, while the first two relationships are quite reasonably true over a broad range, the third begins to show up in considerable error for high-compression ratios.

Again referring to the performance curves (Fig. 317), it will be observed that for a reduction in flow, down to the minimum stable flow, the pressure tends to rise, and conversely, as the flow increases, the pressure drops off, provided the unit is operated at

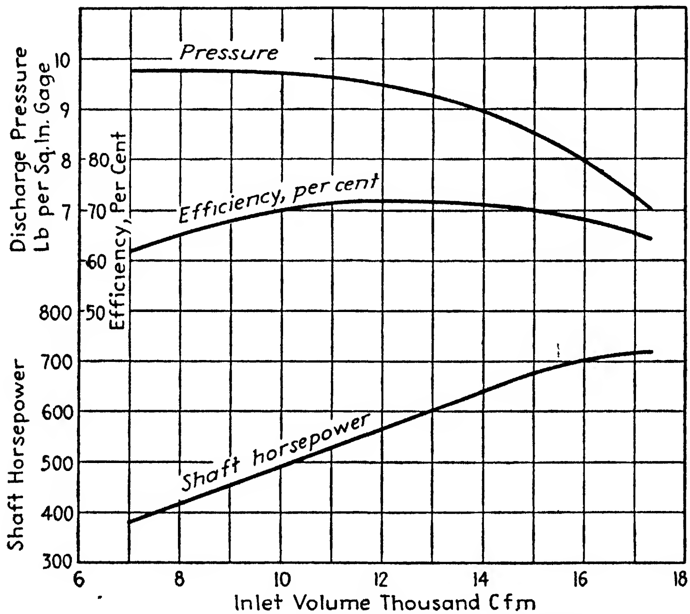


FIG. 316.—Typical characteristic curve of a constant-speed machine. (*Courtesy of Roots-Connersville Blower Corp.*)

constant speed. This is no accidental occurrence, because the drooping characteristic is purposely designed into the compressor in order to improve its stability and operating characteristics. It is apparent that were this characteristic to rise instead of to droop with increase of flow, it would be impossible to control air flow by means of valves since an increase in flow would cause an increase in pressure, which would further cause another increase in flow, and so on in a vicious circle until the driving machine would be entirely overloaded.

With a drooping characteristic, an increase in flow tends to lower the discharge pressure, which in turn reduces the flow approximately back to the point held before the fluctuation occurred. While it is possible to operate a blower alone that has been designed with very little droop in its pressure curve, it is practically impossible to operate two units in parallel without a pronounced drooping characteristic (reduction in pressure with increased flow) to prevent one machine from *hogging* all the load. The stable characteristics are a direct result of the physical proportioning of the rotor and casing passages.

When such a compressor suddenly surges the roar may be terrific, and many an operator has been surprised, upon first hearing it, to find that his machine has come out of a surge absolutely undamaged. Actually, this surging is the result of a sudden, turbulent reversal of air flow with startling acoustic effects. It very seldom causes any mechanical damage.

Surging occurs when restriction at the blower discharge (or inlet) increases, either by closing off a valve or by increasing obstruction from the process being served. The point is finally reached where the pressure developed by the compressor is insufficient to force air out the discharge. Flow may again be reestablished by removing the restriction at the discharge (or inlet) or, in the case of a variable-speed machine, by speeding it up to develop more pressure.

Mechanical Details.—Impeller wheels are designed according to their service requirements as follows: (1) radial-bladed, straight blades extending radially; (2) backward-bladed, with blades curved away from the direction of rotation; open, without an enclosing cover and hub disk; (3) semiclosed, closed on one side only; and (4) closed, closed on both sides. Several designs are shown in Fig. 318.

The cutaway view (Fig. 306) shows a shrouded, single-stage impeller in a volute casing. This is a typical low-pressure unit built to supply air in volumes up to 15,000 cfm at pressures up to 3 psi. The full-depth diffuser and ample-size volute assure maximum conversion of velocity energy to pressure. Leakage is kept at a minimum by the labyrinth seal between the casing and impeller eye. This is a single-suction impeller which requires that the end thrust be taken by properly designed motor bearings.

The enclosed impeller wheel (Fig. 305) eliminates the necessity for close clearance between wheels and casing to prevent leakage past the blade edges. Blades are made from rolled, heat-treated chrome-vanadium steel plate cut to rough size and milled out on the edges to form rivets.

The hub disk and cover disk of the impeller are machined from single-piece forgings of chrome-vanadium steel, heat-treated to give maximum strength consistent with proper ductility. After machining, the disks are drilled to receive the blade rivets.

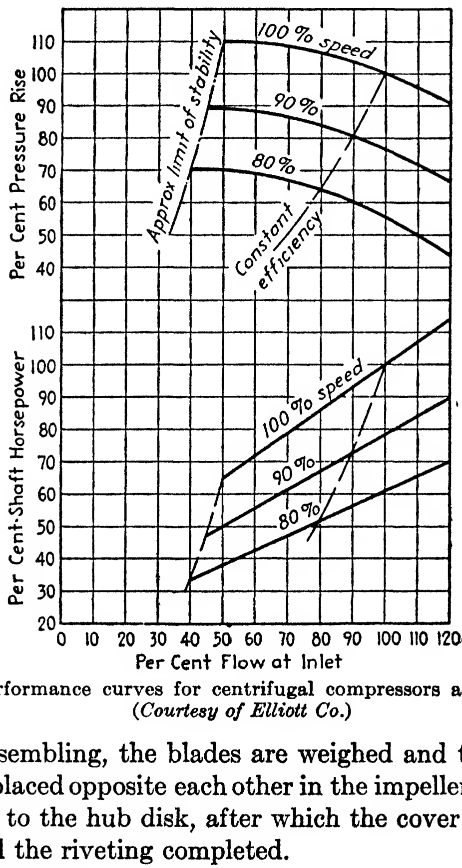


FIG. 317.—Performance curves for centrifugal compressors at various speeds. (*Courtesy of Elliott Co.*)

Before assembling, the blades are weighed and those of equal weight are placed opposite each other in the impeller. Blades are first riveted to the hub disk, after which the cover disk is put in position and the riveting completed.

Shafts are usually made from special hammer-forged, open-hearth steel of high tensile strength and uniform density, ground and polished over the entire surface to assure accurate dimensions, alignment, and finish. To ensure freedom from vibration, the shaft is of relatively large diameter so that its first critical speed is above the operating speed. Connection is made to the driver shaft through solid or flexible couplings.

Shaft Assembly.—The impeller hubs are fastened to the shaft with keys and are separated from one another by shaft-protecting sleeves which form one side of the labyrinth seal between the casing diaphragm and shaft (Fig. 319). The entire series of wheels and sleeves is held against a shaft shoulder by a lock nut or locking collar.

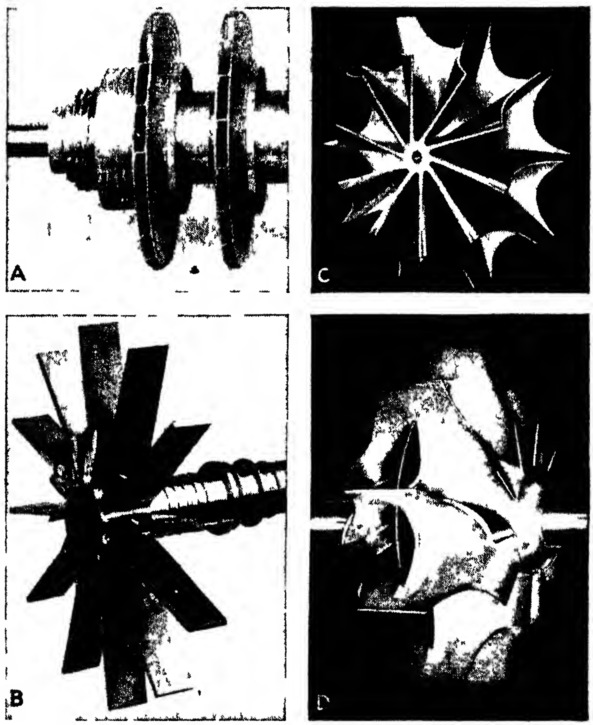


FIG 318.—(*A*) Closed; (*B*) radial-bladed open; (*C*) semi-closed, (*D*) double-inlet impeller. (*Courtesy of Roots-Connersville Blower Corp. and B. F. Sturtevant Co.*)

Short-circuit leakage between stages is prevented by labyrinth seals such as Fig. 309*D*. They form a running joint between the shaft, impellers, and casing diaphragms. Clearance at the seal (Fig. 320) is very small; some seals are actually runin, *i.e.*, the protruding seal ring cuts itself a clearance in the soft seal metal (Fig. 321). They can easily be damaged if the shaft is not held properly centered by the thrust bearing; when this happens they must be replaced.

Pressure or suction at the casing ends determines whether or not seals are needed at this point. If the compressor handles gas they are required, regardless of the pressure present. Sealing glands take various forms, such as carbon or babbitt rings (Fig. 322). Some manufacturers use water as the sealing agent (Fig. 323), while others use steam. Some machines carry a movable lever-operated bushing which renders the seal gastight when the unit is shut down (Fig. 324). A protective device prevents damage if the compressor is started before the lever is released. Figures 325 and 326 illustrate these hand-operated sealing rings.

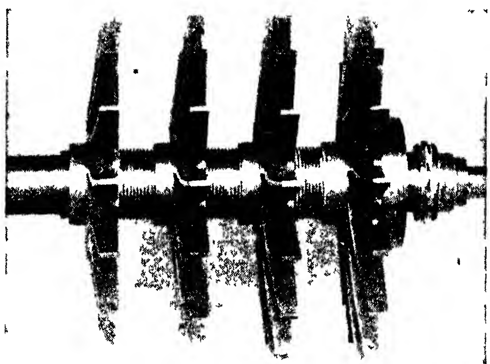


FIG. 319.—Four-stage radial-blade impeller rotor showing labyrinth rings on the shaft.

Axial End Thrust.—All multistage rotors are designed so that most of the axial thrust is eliminated. In the case of single-inlet machines this is accomplished by sealing off the discharge pressure with one face of the last impeller (Fig. 309*C*), or a balance piston of the correct diameter (Figs. 308 and 324). The seal used for this purpose consists of a labyrinth gland. An equalizing passage or pipe then keeps the pressure behind the balance disk at or near inlet pressure, which eliminates practically all the unbalanced rotor thrust (Fig. 324).

In a double-inlet machine (Fig. 307), the thrust is inherently balanced without the aid of a balance piston, because the air flows in opposite directions from the two inlets to a common outlet. In both single- and double-inlet machines, a thrust bearing positions the shaft and absorbs any residual thrust.

Thrust Bearing.—Most manufacturers use the Kingsbury thrust bearing as illustrated in Figs. 327, 328, and 329. Figure

330 shows how leveling plates equalize the load on the thrust shoes of Fig. 329.

Since these bearings all operate on the principle of pivoted segmental shoes, a study of Figs. 329 and 330 shows how the bearing operates. Each shoe covers an arc of about 50 deg and is free to pivot slightly, permitting the oil film between the shoes and the thrust collar to assume a natural wedge-shaped form. This film completely separates the shoes from the thrust collar; hence there is no wear in normal operation.

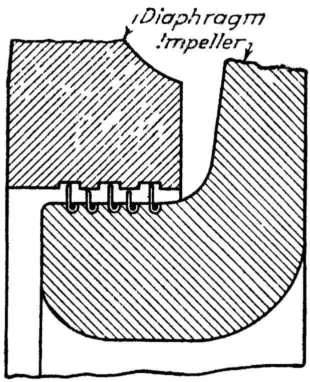


Fig. 320.—Impeller labyrinth seal.

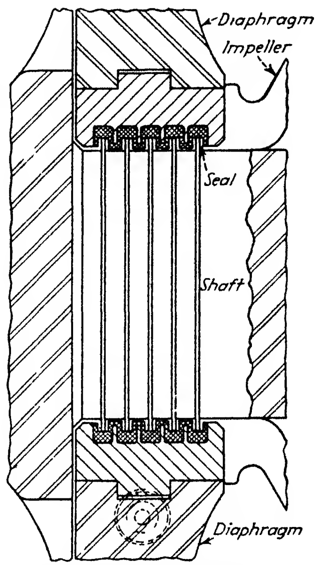


Fig. 321.—Diaphragm labyrinth seal.

In the back of each shoe is a hardened button, which bears against a hardened-steel leveling plate. These leveling plates are interlocked as shown in the developed diagram (Fig. 330); thus the load is equalized between shoes.

The leveling plates are mounted in steel base rings, which also direct the flow of incoming oil. These rings are split to permit radial assembling. The entire bearing is enclosed in a steel cage, which likewise is split for radial assembling. This permits the bearing to be handled as a unit with the shaft and runner when the machine is opened. A split filler piece gives adjustment for end play or oil clearance.

Oil enters both thrust cavities at the back of the base rings and flows inward to the shaft; it then flows toward the thrust collar and moves outward between the shoes. At the collar rim it is thrown off, as if by the impeller of a centrifugal pump, into two circular grooves machined in the cage surrounding the collar.

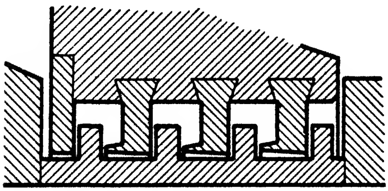


FIG. 322.—Shaft labyrinth seal.

At the top there are tangential outlets like the discharge from a pump. The escaping oil runs down an annular discharge passage in the housing and returns to the pump. Thus churning and loss of power are minimized.

Cleaning Precautions.—Take the bearing apart and thoroughly clean it before assembling. Remove antirust coatings with kerosene. Use rags or cloth, as waste leaves lint which clings

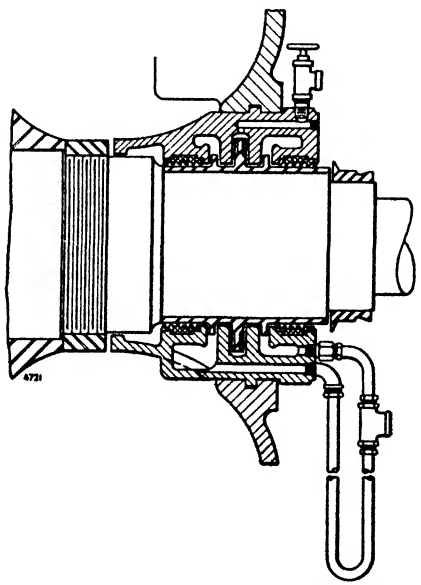


FIG. 323.—Water-sealed shaft packing. (*Courtesy of De Laval Steam Turbine Co.*)

to minute burrs and may cause trouble later. Inspect all bearing parts after cleaning and cover with a film of oil. Remove with a scraper any bruises on the babbitt faces, and with a fine oilstone slight bruises or rust on collar surfaces. Deep rust requires refinishing. Never leave file or scraper marks on collar surfaces. The thrust collar must be exactly square with the shaft. Remove any bruises on the shaft shoulder before assembling.

Inspection and Repair.

1. To inspect the shoes and collar, drain the cage and lift the upper halves of the cage and base rings. Remove the shoes by turning the base rings.

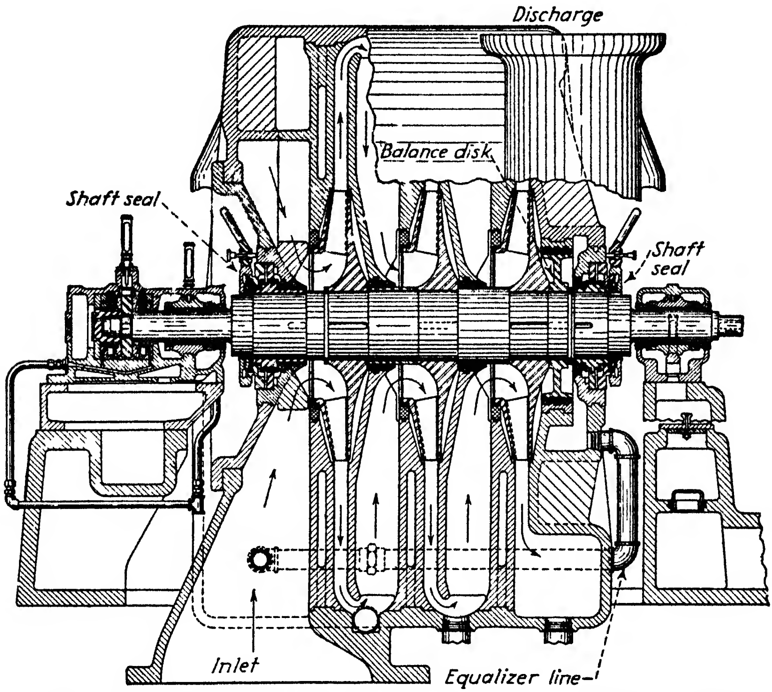


FIG. 324.—Hand-operated shaft seal and equalizing line leading to balancing disk. (*Courtesy of Roots-Connersville Blower Corp.*)

2. Slight damage to the babbitt faces may be corrected by scraping. The filler piece, however, must then be shimmed or replaced by a thicker one to restore the end play; or, it may be possible to insert a thin one-piece shim at the outer end of the bearing cavity. The shoes should be scraped to a surface plate and the radial edges slightly rounded. The leveling plates will accommodate themselves to inequalities from scraping. Shoes seriously damaged should be sent to the manufacturer for repairs.

3. Scoring of the thrust collar may be corrected by regrinding and smooth lapping. The new surfaces must be exactly square with the bore. The collar is a slip fit on the shaft, and may be drawn by using two $\frac{5}{16}$-in. tapped holes near the bore.

The thrust bearing in Fig. 327 has six tilting shoes on each side of the collar. The shoes are made of steel forgings and the bearing surfaces of high-grade babbitt securely bonded to the steel. The shoes have a projection which bears on a groove in the base rings. They are retained on the ring by U bolts extending over lugs projecting from the sides of the shoes. They are free to tilt, thus allowing a wedge-shaped oil film to build up and be maintained continuously between the shoes and the thrust collar. Shoes are prevented from rotating by the end plate on the upper half, which projects past the circular bore retaining the base rings into a slot in the bearing housing.

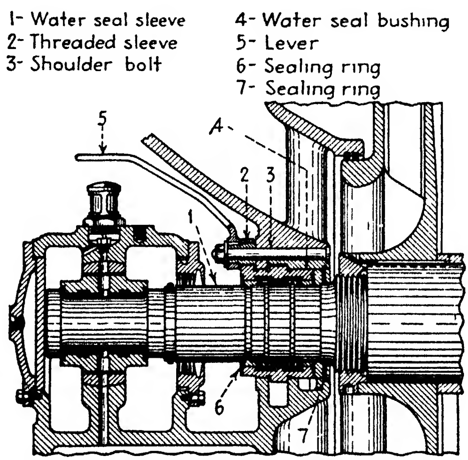


FIG. 325.—Water seal and hand-operated sealing rings. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

These shoes cannot be replaced individually because if a new shoe were to be installed, the remaining ones, being worn, would not touch the thrust collar. Therefore, the shoes must be replaced in sets of six on the same side of the thrust collar.

Axial clearances are maintained by removing or adding shims between the base rings and end plates on both sides of the thrust bearing. When making adjustments, take care that the same shim thickness is added or removed from the upper and lower halves of the thrust bearing.

Journal Bearings.—Most journal bearings are of the conventional sleeve design, made of cast-iron or cast-steel shells, to which a high-grade hard babbitt is securely bonded. The bearings are split on the horizontal center line and the two halves are positioned, one half relative to the other, by an accurately machined joint and dowel pins. They are accurately bored to close tolerances.

Bearings are usually designed for pressure lubrication, the oil under pressure entering the bearing through a drilled hole. They are held in an axial position by a close-fitting tongue-and-groove joint. The radial position of the bearing, which locates the rotor radially, is adjusted by shims placed under the bearing pads. Oil guards provided where the journal emerges from the

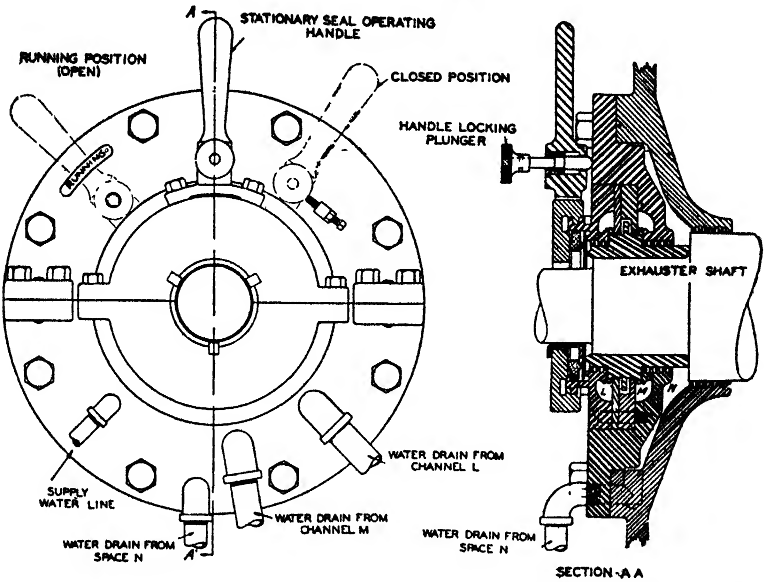


FIG. 326.—Combination running and hand-operated stationary seal. (*Courtesy of Roots-Connersville Blower Corp.*)

bearing prevent oil from creeping along the shaft and prevent air from entering the bearing housing. Figure 331 shows a schematic lubricating system for a turbine-driven compressor.

Bearing temperatures are governed largely by the type of oil used and other operating conditions, as well as by the compressor speed. A fair operating temperature is from 140 to 180 F for the oil leaving the bearings. When the compressor is driven by a steam turbine, or if it is provided with steam seals, there is the possibility of water's getting into the oil. Since with most turbine oils water remains in suspension at temperatures below approximately 130 F, the temperature of oil in the reservoir should be held to a minimum of 130 F.

Referring to the schematic oil-piping diagram (Fig. 332), the main oil pump takes oil from the tank and discharges it through the cooler. A relief valve 2 spills any excess oil back to the tank. After flowing through the cooler the oil enters the bearing supply lines which are fitted with metering plugs 4 to control the supply to the bearings.

When starting the unit for the first time, and also after long periods of shutdown, vent the system by opening both cocks 8 on the main pump before starting the auxiliary oil pump. If

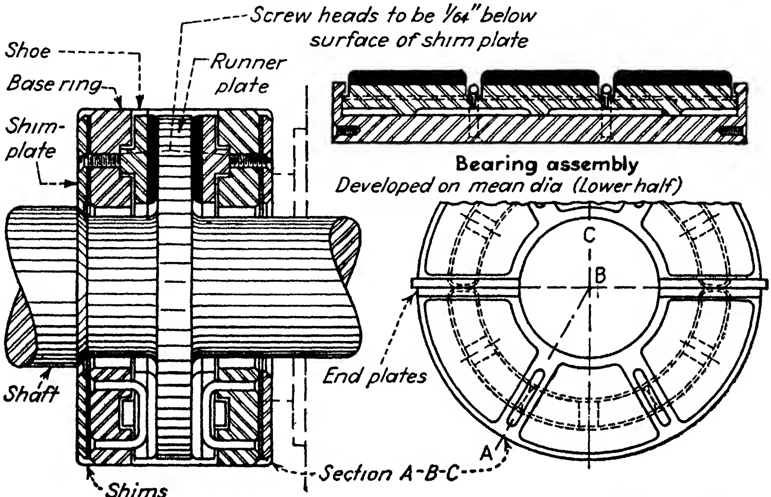


FIG. 327.—Thrust-bearing assembly with developed section showing absence of load equalizer.

necessary, prime the suction line of the main pump by removing the vent cock on the suction side and pouring oil into the line. Inspect and clean the strainers 3 at frequent intervals.

Lubricating-oil Recommendations.—The importance of using the highest grade oil for lubricating centrifugal machines cannot be too strongly emphasized. Although the first cost of the best oil may be considerably greater than that of an inferior product, the additional cost is more than compensated for by longer service, less danger of damage to bearing surfaces, and elimination of serious shutdowns.

Consult with representatives of reliable refiners regarding their lubricating oils and insist that the oil purchased be suitable for the purpose intended.

Here are the basic requirements of a good oil, as recommended by one manufacturer.

1. Properly refined, highly filtered pure mineral oil, free from acid, alkali, water, sediment, soap, resins, or any substance which in service will prove injurious to the oil or to the parts coming in contact with it.

2. Best possible capacity to separate rapidly from water and least tendency to emulsify or foam when agitated with air or water.

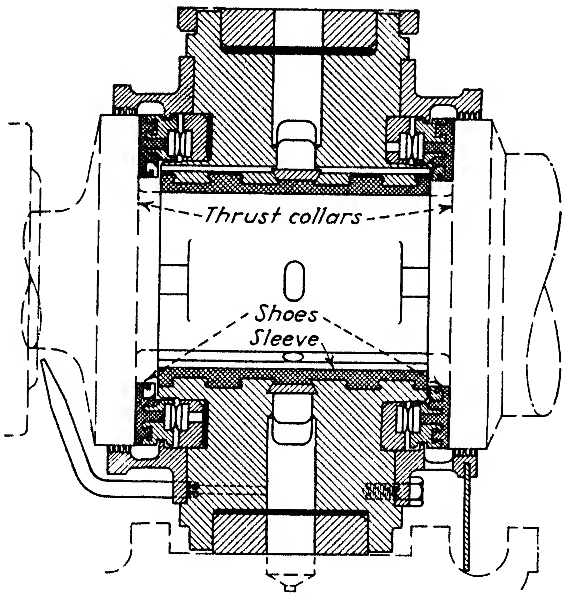


Fig. 328.—Combined sleeve and thrust-bearing assembly. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

3. Least tendency to break down or form sludge when agitated at normal operating temperature and mixed with air or water.

4. Reasonably high flash and fire points, bearing in mind the probable operating temperatures and unavoidable proximity to high-temperature steam surfaces in a turbine-driven unit.

5. Viscosity and viscosity index (relationship between viscosity and temperature changes) suitable to the service intended.

Viscosity (SSU) at 100 F	140 to 170
Viscosity (SSU) at 210 F	42 to 44
Minimum operating oil-tank temperature	130 F
Minimum oil temperature before starting	50 F
Normal bearing temperature	140 to 180 F

Maintain the oil in first-class condition at all times. This involves the elimination of moisture, dirt, sludge, and other impurities. To accomplish this, filter the oil or run it through a centrifuge at regular frequent intervals; the continuous by-pass purifying system may also be used.

The oil reservoir, bearing housings, and piping should be thoroughly cleaned when the machine is shut down for inspection. In case of heavy sludging, accumulations may block off

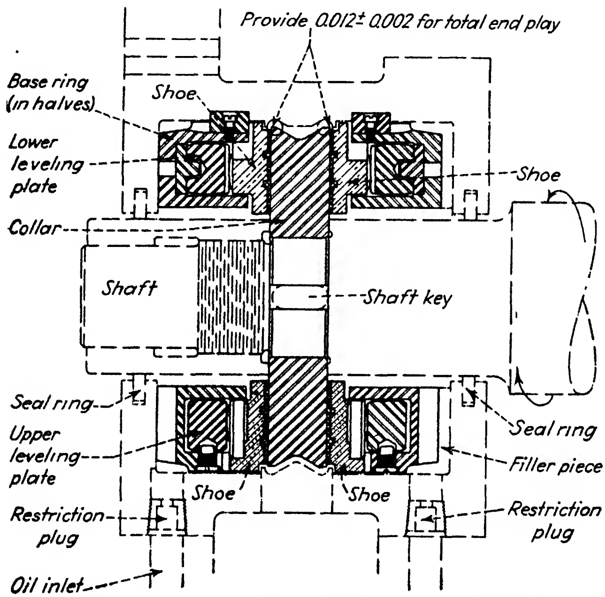


FIG. 329.—Seven-inch double-equalizing six-shoe thrust bearing. (*Courtesy of Kingsbury Machine Works, Inc.*)

the oil passages in the bearings and cause serious trouble. When cleaning, use kerosene or a safety solvent made expressly for this purpose. After cleaning, remove the residual kerosene or solvent by wiping dry with clean lint-free rags.

Load Control.—Centrifugal compressors adapt themselves readily to automatic regulation. This may take the form of constant volume, constant discharge-pressure, or constant suction-pressure regulation, depending upon the service involved. The most economical method of accomplishing automatic regulation with steam-turbine-driven machines is by means of speed variations, in which case the special regulator actuates the turbine-speed governor. Figure 333 shows the relation between output and speed on a variable-speed turbine-driven three-stage centrifugal exhauster handling coke-oven gas. Steam consumption is also plotted against output.

Partly closing the intake lowers the discharge pressure, since the pressure ratio of the compressor is fixed. This moves the operating point back on the characteristic curve to give reduced volume. Throttling the intake also lowers the pumping or surging point.

When the speed cannot be varied on motor-driven compressors, a master regulator (Fig. 334) can be made to actuate a suction

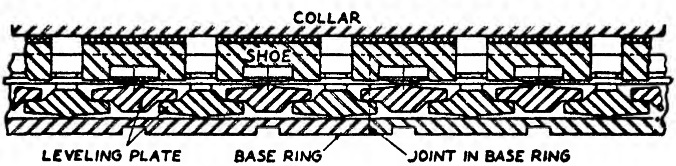


Fig. 330.—Development of equalizing thrust bearing showing leveling plates.

throttle valve or prerotation vanes. With variable-speed drives, the regulator can control the oil flow to a hydraulic coupling or secondary resistance of a wound-rotor motor. The choice of regulation most suitable for motor-driven units depends largely on the service for which the machine is intended. The motor-control system (Fig. 335) consists of automatically adjusted prerotation vanes, a blowoff valve, and a check valve. Prerotation or movable vanes installed near each stage intake (Figs. 336 and 337) change the direction of air flow to the impeller, which reduces the pressure rise through that stage.

When two or more centrifugal compressors operate in parallel, a check valve must be placed in the discharge line to prevent air under pressure from backing up into the blower and causing it to run in reverse direction. This is also necessary when the unit furnishes air to a large pipe line; a sudden slowing down of the compressor allows the air to back up into the machine.

In order to prevent pumping at low loads, a valve A can be installed in the discharge line (Fig. 338), which wastes air to the atmosphere when the volume falls below the pumping limit, thus keeping the compressor operating in its stable range. The valve operates by oil pressure which moves a piston against a spring, the oil pressure being varied by a governor connected to the air lines.

The simplest method of controlling flow from a compressor, when operating at constant speed, is to use a hand valve in the discharge (Fig. 310). Opening the valve increases air flow up to design capacity, and closing it conversely decreases the flow down to the point of instability. The regulating valve may be placed in the blower inlet with similar results.

On turbine-driven units, the flow or discharge pressure may be controlled by regulating steam admission to the turbine to

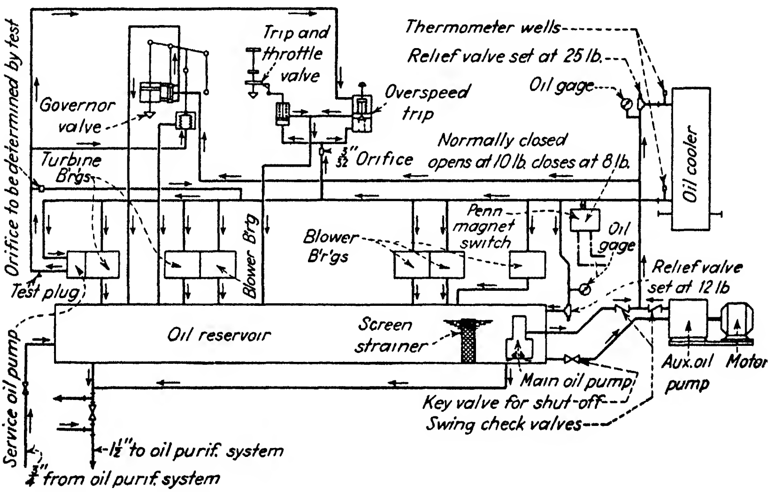


FIG. 331.—Lubricating-oil system for a centrifugal compressor. (*Courtesy of Elliott Co.*)

vary the speed (Fig. 339). For example, if a turbine-driven compressor serves a chemical process which requires, at the moment, 70 per cent of the design air flow, and if the pressure required to force this amount through the process is 86.5 per cent of the design pressure, then it may be seen from the pressure curve (Fig. 317) that a speed 90 per cent of normal will be required. It can further be determined from the power curves that, at 90 per cent of normal speed and 70 per cent flow, the shaft horsepower will be 61 per cent of that required to operate the machine under design conditions.

The constant-speed governor (Fig. 340) is of the oil-relay type. The impulse for speed regulation is oil pressure produced by the governor oil impeller located on the main rotor shaft. Pressure produced is a function of the speed. Thus, the governor is a regulator that holds substantially constant oil pressure, and in turn, constant speed.

The impulse from the governor oil impeller is applied to a bellows in oil-relay chamber 8. Force developed by the bellows is balanced by tension spring 80. When this force balance is disturbed by a slight speed change, movement of the bellows is

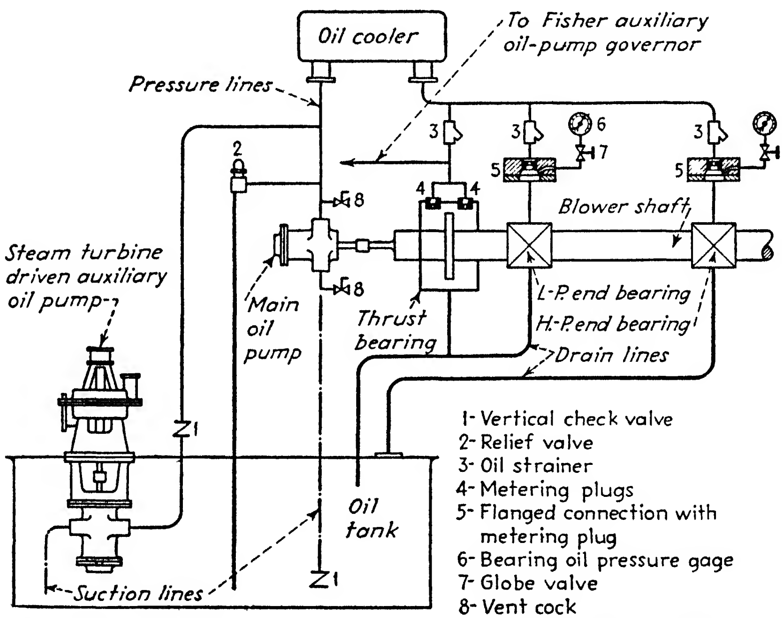


FIG. 332.—Schematic oil system showing use of metering plugs. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

transmitted through linkages to the pilot valve 56. Oil at about 25 psi gage pressure from the main pressure header feeds point A, external oil pressure inlet, located between the pilot-valve ports. When the pilot valve moves from its central position, oil is admitted to one side of power piston 7, and at the same time oil is drained from the opposite side through a drain. This movement of the piston rod is transmitted through linkage to governor valve 81. Moving the pilot valve in one direction causes piston movement in the opposite direction, which brings the pilot valve back to its central position.

A damping action to prevent hunting is provided by an external dashpot. Adjustment of the dashpot is provided in by-pass valve 61, the tapered opening in the lower end regulating the passage of oil across dashpot piston 63. Screwing down on this by-pass valve decreases the governor's speed of response and increases its stability; the opposite effect is gained by screwing upward.

Manual adjustment for the speed to be maintained is provided by changing tension in spring 80. An increase in spring tension

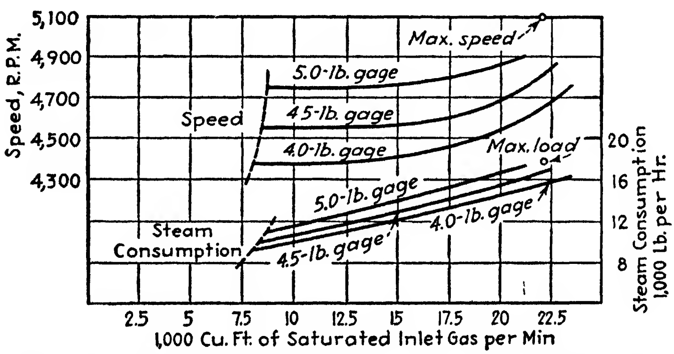


FIG. 333.—Speed and steam-consumption curves for a turbine-driven gas exhauster. (*Courtesy of Power.*)

increases the speed by requiring a higher oil pressure to balance its force; the opposite is true for a decrease in tension.

Normally, the unit's speed is under control of pressure (or volume) regulator 90. This is accomplished by the linkage through which the regulator positions the governor valve. However, if the speed increases to that which the governor has been set to hold, it will limit the speed to this value regardless of the regulator's action.

Soft packing is provided at the upper end of the pilot valve, and molded packing at both ends of the power piston and throttle valve. Do not tighten these glands, particularly on the pilot valve, any more than is necessary to prevent oil leakage, since excessive pressure causes friction and results in faulty governor operation.

The constant-speed governor (Fig. 341) permits a speed variation of 10 to 20 per cent. It incorporates a hand speed changer that permits changing the speed setting while the unit is running.

The overspeed trip (Fig. 342) mounts on the unit. The oil-pressure line from the governor oil pump to the oil-relay chamber of the constant-speed governor (Fig. 340) contains a tee, from which a line connects to point *A* on the overspeed trip. Point *B* is jointly connected to the diaphragm of the trip and throttle valve and the bearing oil-pressure header through a line containing a small orifice. Point *C* is a drain connection to the oil reservoir.

When the oil pressure developed by the governor impeller reaches a value corresponding to the tripping speed, valve 5 is

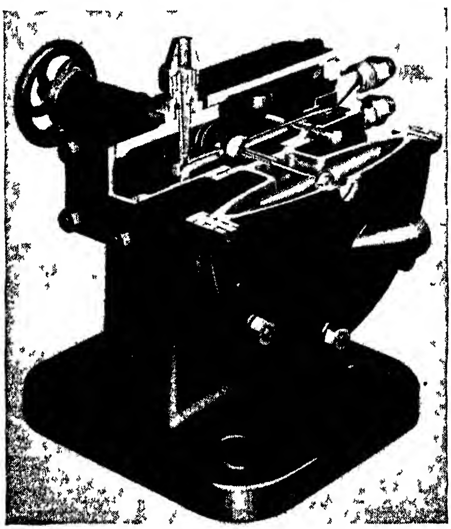


FIG 334—Regulator that controls suction-valve prerotation vanes, hydraulic coupling, or motor resistance.

moved from its seat. When valve 5 opens, oil pressure on the diaphragm of the trip and throttle valve is released, thereby closing the trip and throttle valve and shutting down the unit.

The tripping speed can be changed by placing or removing spacers beneath spring 12, or by substituting a stronger or weaker spring. The unit can be tripped manually by pressing down on the push button mounted on top of the overspeed-trip body. It is good practice to trip in this manner, rather than by closing the throttle valve by hand whenever the unit is to be shut down. Figure 343 illustrates a mechanical overspeed trip that mounts adjacent to the compressor shaft. Overspeed and its attendant increase in centrifugal force throw out the plunger, release the trigger, and shut down the unit.

An exhauster control (Fig. 344) regulates speed to maintain constant suction. It comprises a small reversing motor controlled by suction, which operates a regulating valve in the oil line leading to the sylphon bellows of the variable-speed governor. Oil pressure on the bellows depends upon the amount of oil allowed to flow through the regulating valve by the exhauster

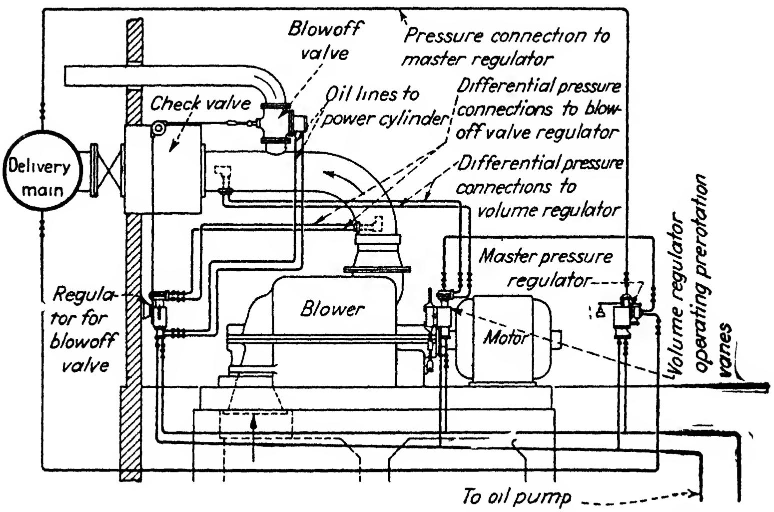


FIG. 335.—Piping control for prerotation vanes, blowoff valve, and check valve. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

control. The position of the bellows in turn controls the oil-relay position which actuates the turbine-governor valve. The mechanical flyball-speed governor driven from the turbine shaft is so arranged that it will not influence the oil-relay position until the speed reaches a maximum allowable value. At that speed the governor nullifies any attempt of the exhauster control to increase speed.

The gear oil pump discharges into a header, one branch of which goes to the oil cooler and the bearings and another to the relay cylinder. A third path is through an adjustable orifice into the chamber surrounding the sylphon bellows. Oil pressure on the bellows is regulated by the valve under control of the exhauster control. An increase in suction causes the control to reduce its valve opening, and a decrease in suction tends to increase its opening.

When speeding up the turbine, the sylphon raises the left-hand end of lever L_1. If the speed exceeds normal, the centrifugal governor comes into action and raises the right-hand end of this lever. In doing so it moves the left-hand end downward and tends to close the throttle valve and offset any tendency of the

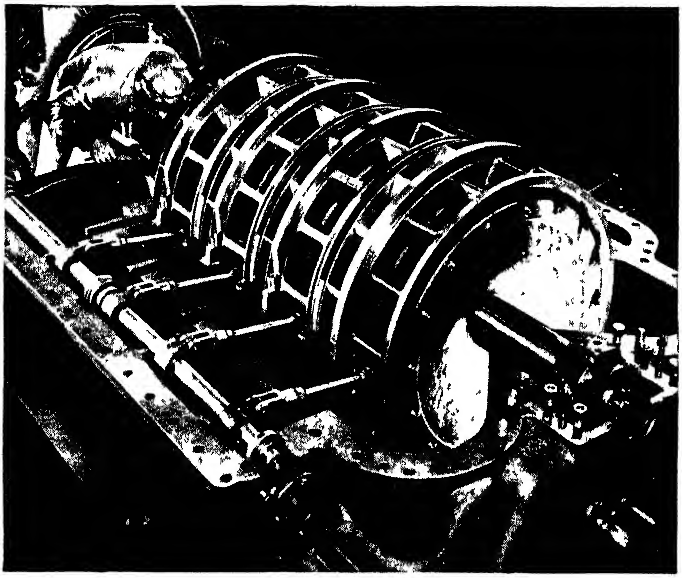


FIG. 336.—Prerotation-vane-operating mechanism on multistage machine. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

exhauster control to raise the turbine speed above normal. Opening and closing the hand-control valve on the sylphon chamber has the same effect as opening or closing the exhauster-control valve.

Cooling Arrangements.—The amount of heat to be dissipated is considerably more than that due to adiabatic compression, as the mechanical energy lost to internal air leakage, unavoidable eddy currents, air impact, and air friction reappears as heat energy stored mostly in the air itself. Heat is also generated through the air's opposition to rotor motion.

Internal water-cooling passages cast in the compressor casing have extensions down inside the diaphragms (Fig. 309 E) and are fitted with suitable cleanout plugs for flushing purposes. For high compressions the air is cooled by passing it through intercoolers. In this arrangement the air discharges from a series of stages through the intercooler and back into another series of stages of the compressor. The work diagram (Fig. 345) shows the power saved. Aftercoolers can be used to remove the remaining heat of compression before the air discharges into the distribution system.

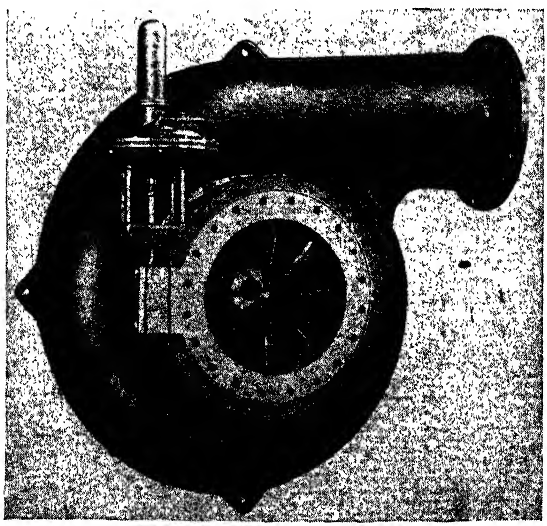


FIG. 337.—Prerotation-vane arrangement on single-stage blower. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

Installation.—Erect the compressor in accordance with foundation and general-arrangement plans furnished by the manufacturer. The unit must be carefully leveled, with steel wedges and shims placed under the base near each foundation bolt. Machined surfaces on top of the base or pipe flanges serve for leveling pads. It is always advisable to check shaft alignment during bedplate leveling so that any warping in the latter will be discovered before grouting is done. When the bedplate is properly leveled and shaft alignment checked completely, fill the bedplate with grout. (Grouting procedure is discussed in Chap. VI.)

The shaft alignment should be checked after the grout is set and again after the machine has a preliminary run and reaches operating temperature. Even though the compressor employs a flexible coupling, use the same care in aligning it as though it were a solid coupling.

The sleeve A should be slipped over the shaft end before hub B is forced on and keyed. Keys for this coupling must fit on their sides for the entire length of the hub; they must not bear on top or bottom or project beyond the hub ends. The hubs must be

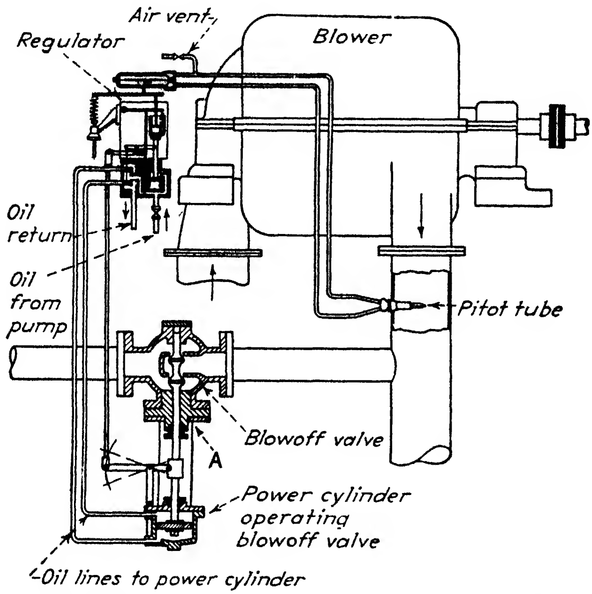


Fig. 338.—Automatic blowoff system prevents surging at low loads. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

separated by the dimension found stamped on their alignment faces.

Then line up the two shafts with a straightedge and thickness gage as shown in Fig. 346, meanwhile rotating each shaft to a new 90-deg position. If the coupling is oil-lubricated, put a good gasket between the coupling sleeves, bolt them together, and add the required amount of oil. When pillow dowels (Fig. 347) are to be inserted under the compressor and turbine casing follow the manufacturer's instructions carefully when drilling and fitting the holding-down bolt shims.

Piping connected to the turbine and compressor must be arranged and supported so that it imposes no strain on the respective casings. Provide adequate means for piping expansion (Fig. 348). Clean and inspect all lines before making the final connection. This is important because any tools or foreign matter left in the pipes will cause serious damage to the compressor or turbine.

When several machines are connected in parallel to one main, it is essential that an atmospheric by-pass and a reliable non-

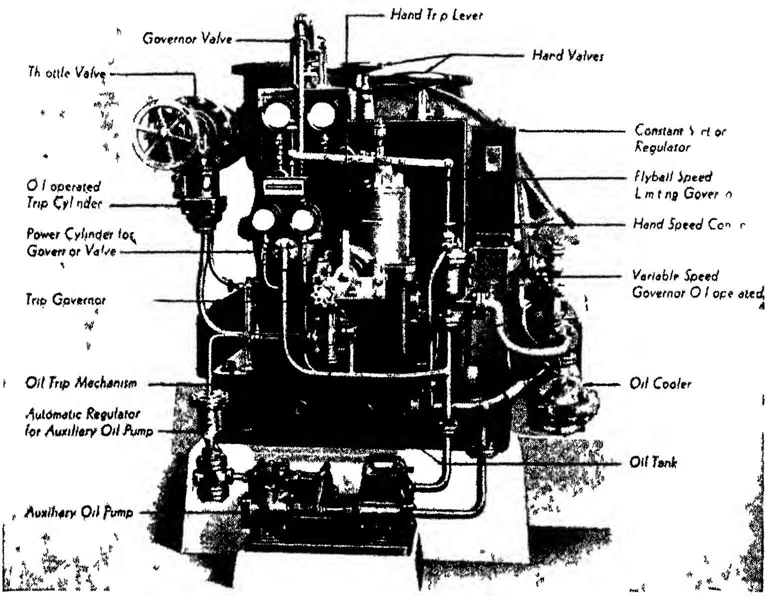


FIG 339.—Complete control system mounted on turbine-driven compressor

return valve be placed in the discharge line. Operation of the nonreturn valve should be checked occasionally, because if it fails to close properly, reverse flow will turn the rotor in a reverse direction, with possible damage to the bearings.

Reverse rotation, while not injurious to certain types of thrust bearings, can cause damage from lack of lubrication if the main oil pump operates in only one direction. The normal direction of rotation marked on the casing serves to check the revolving shaft before bringing the machine to full speed. Auxiliary oil pumps, sometimes used during the starting and stopping interval, should be controlled by oil pressure from the main pump so that they start immediately on failure of the main oil supply.

Initial Starting.—In preparation for the initial start, check impeller clearances at the points designated on Fig. 347 or those marked on the manufacturer's drawings. When taking these measurements, push the rotor hard against the thrust bearing in first one direction and then the other. Always allow an additional several thousandths of an inch clearance in the direction of maximum thrust for springing of the thrust bearing.

Make certain that the lubricating system is functioning properly and circulate oil through the bearings for several hours. Then remove the bearing caps and see that no foreign matter has been deposited in the bearings.

Break the coupling and check the rotation of the driving unit. This prevents any possible damage to the compressor because of reverse operation. Close the coupling and fill it with the recommended grade of oil. Holding a white sheet of paper near the coupling during operation will reveal leakage. If this precaution is not observed, lubricant may leak out and the coupling parts will wear seriously.

The auxiliary oil pump is usually provided with an automatic pressure switch that stops it when the main pump builds up sufficient pressure. Sometimes a flow switch connected in the main-oil-pump discharge controls the auxiliary pump. A relief valve on the bearing oil line discharges into the oil reservoir.

Check the operation of the auxiliary oil pump and see that it goes in and out of operation at the proper oil pressure. Under no condition should the auxiliary-pump motor-starting switch be thrown off while the main unit is turning over. Leave the starting switch on automatic to provide protection against oil failure while the unit is in operation.

Start the unit with closed intake, but do not run it this way very long unless a close watch is maintained on the casing temperature. A centrifugal compressor should start discharging after reaching full speed and before it gets hot or it will not develop rated discharge pressure. After becoming heated the machine must be shut down and cooled before it can be put in service.

The temperature of all bearings should be observed until they become constant. Normal temperature varies from 130 to 160 F, depending on cooling-water and room temperature. Oil leaving the bearings should never exceed 175 F. Oil temperature can be controlled by regulating the quantity of water going to the oil cooler. Grease-filled roller bearings should not exceed a temperature of 130 F.

If there is any noticeable vibration at full speed, shut the unit down at once and determine the cause. Among the more common causes of vibration are contact between the rotor and labyrinth packing ring, misalignment, improper bearing clearances, or rotor unbalance. A rotor vibrates approximately in proportion to the amount of unbalance. Since rotors are balanced both statically and dynamically and tested at full speed before shipment, trouble from this source is rarely experienced. It is possible, however, to destroy rotor balance by mishandling, deposits of foreign material, or uneven erosion of the impellers.

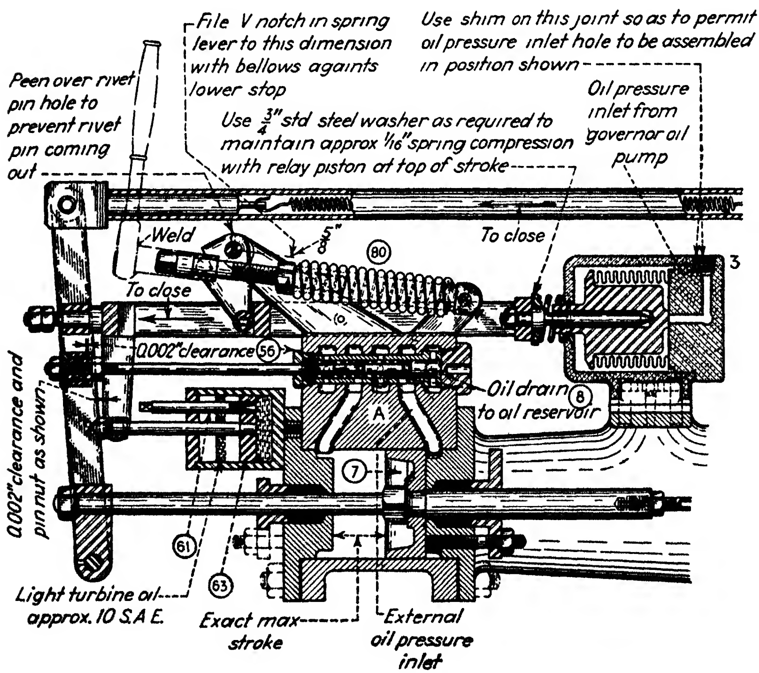
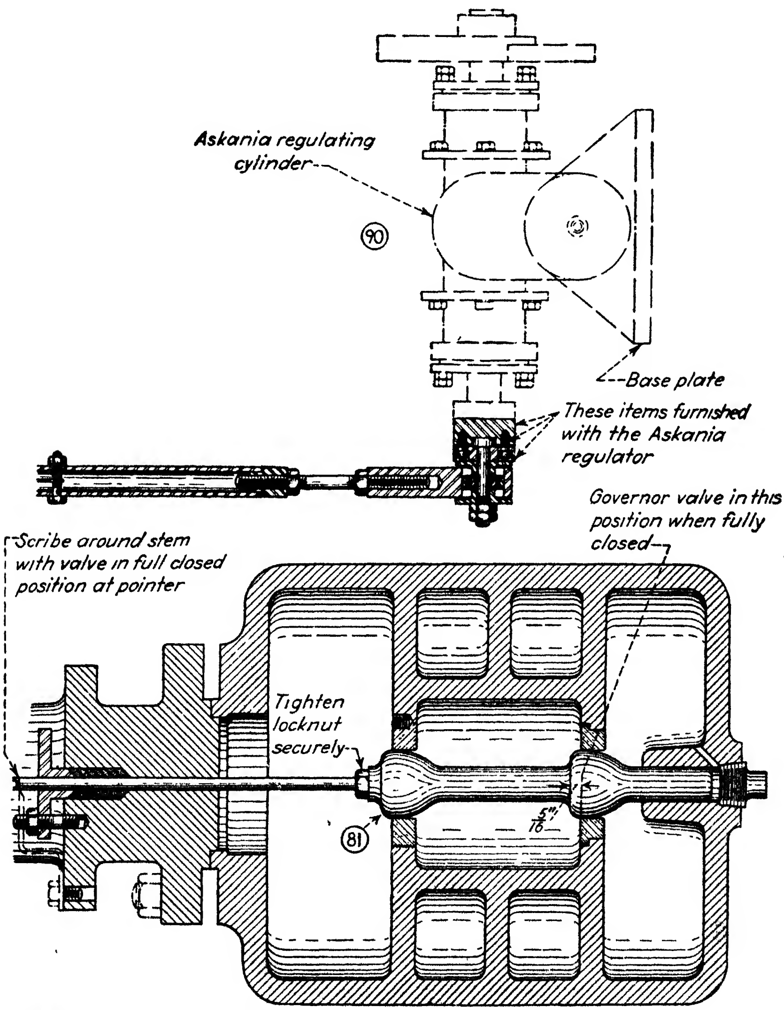


FIG. 340.—Constant-speed governor value controlled from oil-pump

Test the emergency overspeed trip on steam-turbine-driven units at regular intervals; the tripping speed is normally about

10 per cent above rated speed. Bring the unit up to tripping speed by throttling the compressor inlet or discharge. If it does not trip at normal tripping speed, shut down and readjust the overspeed-trip mechanism. Repeat this procedure until the



governor driven from compressor shaft. (*Courtesy of Elliott Co.*)

proper adjustment is obtained. The compressor should never operate at speeds in excess of the tripping speed designated by

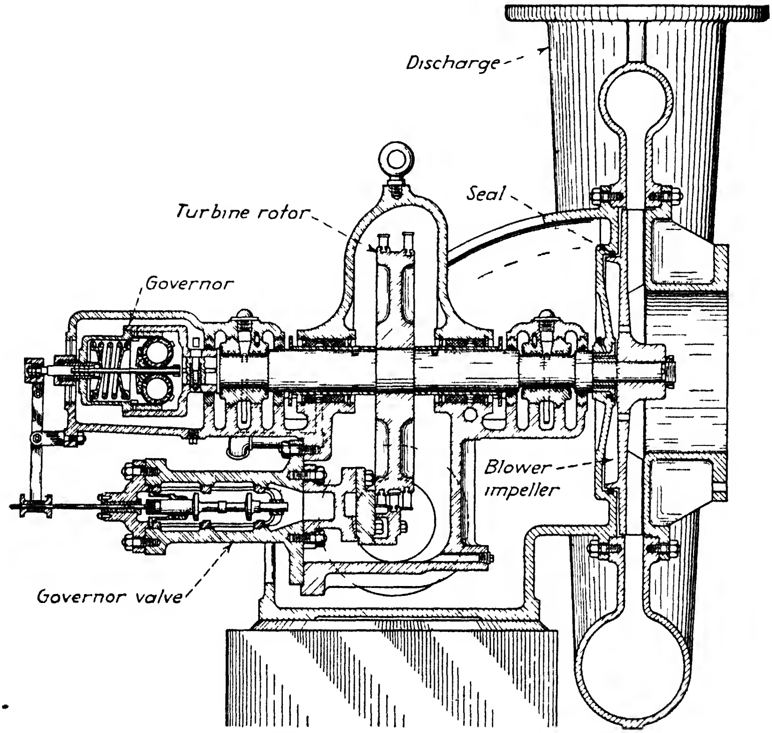


Fig. 341.—Mechanical constant-speed governor controls impulse-turbine-driven blower (*Courtesy of Allis-Chalmers Manufacturing Co.*)

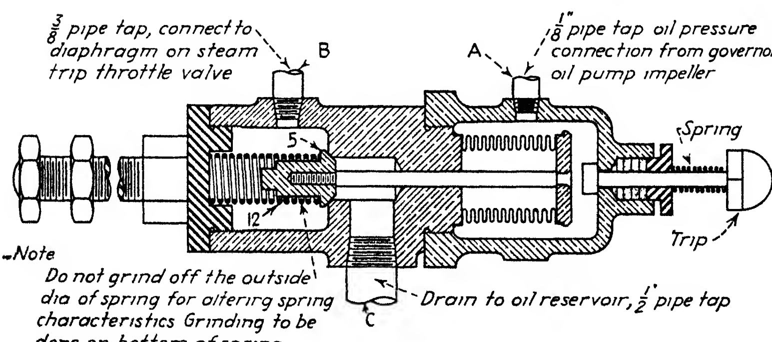


Fig. 342.—Overspeed trip used in connection with constant-speed governor of Fig. 340. (*Courtesy of Elliott Co.*)

the manufacturer. Compressors equipped with movable inlet vanes should have these vanes in their wide-open position during the overspeed test.

To prevent possible overloading of an electric motor, do not operate it with the compressor discharge open to atmosphere or with a piping system in which the normal pressure differential

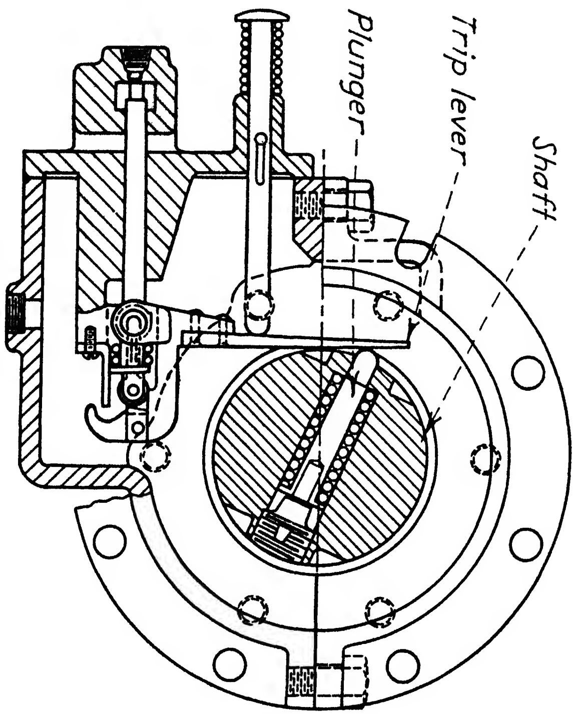


Fig. 343.—Mechanical overspeed trip mounted adjacent to blower shaft. *(Courtesy of B. F. Sturtevant Co.)*

builds up very slowly. In such cases, throttle the inlet so as to limit the motor input to its normal value. No such precautions are necessary with a turbine-driven machine, as an overload merely reduces the turbine speed.

The operating speed of a turbine-driven compressor should be checked at the initial start-up, and periodically thereafter. A vibrating-reed tachometer may be used for speed measurement, if a direct-driven tachometer is not furnished with the unit. It is advisable to check the overspeed governor at regular intervals.

This can be done conveniently at the time of a shutdown, by closing a blower inlet or discharge valve, at the same time holding the turbine governor valve open. If the overspeed governor fails

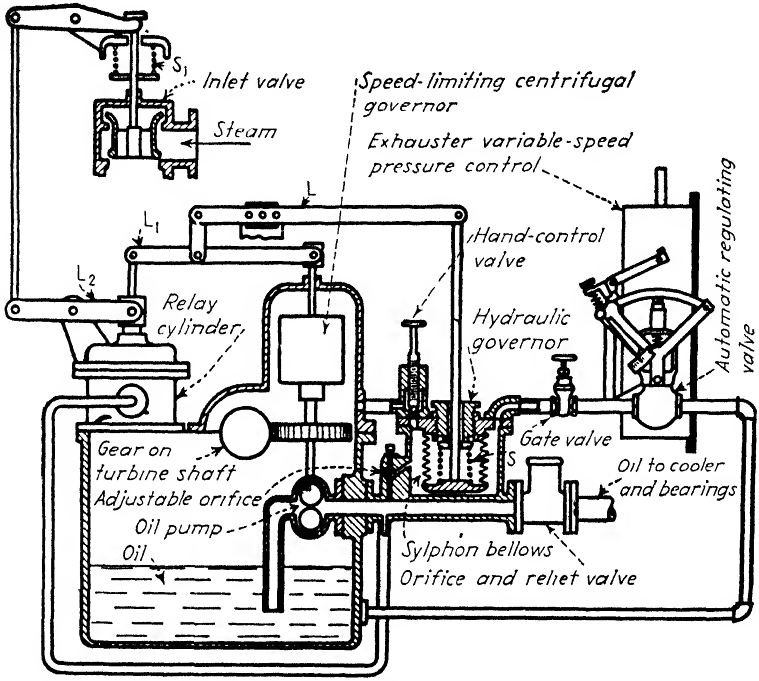


FIG. 344.—Schematic diagram of gas-exhauster governing system. (*Courtesy of Power.*)

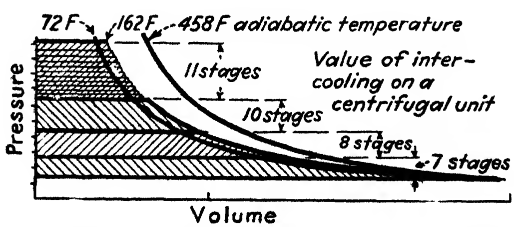


FIG. 345.—Power saved by intercooling a multistage centrifugal compressor. (*Courtesy of Power.*)

to operate at about 10 per cent above maximum operating speed, it should be inspected and the condition corrected.

Routine Operation.—The various operations required to put a centrifugal compressor on the line in parallel with other units cannot be fully understood without some knowledge of the machine's characteristics.

Assume that the compressor operates at constant speed and its capacity must be reduced by throttling the discharge valve from normal to approximately 35 to 40 per cent of normal. Discharge pressure will then increase from 8 to 10 per cent. If the capacity is reduced further, the flow will become intermittent. Unless a check valve is used, the flow actually reverses intermittently and creates violent surges. If the discharge valve is completely

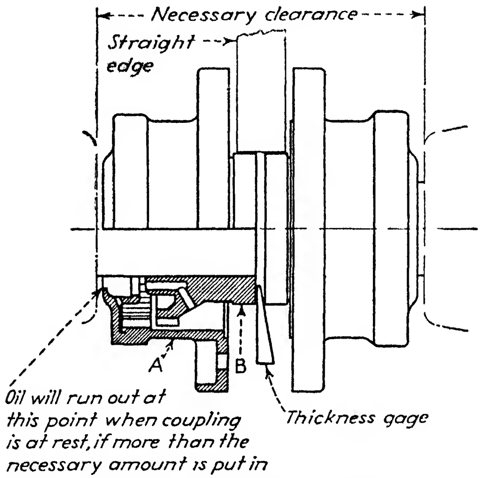
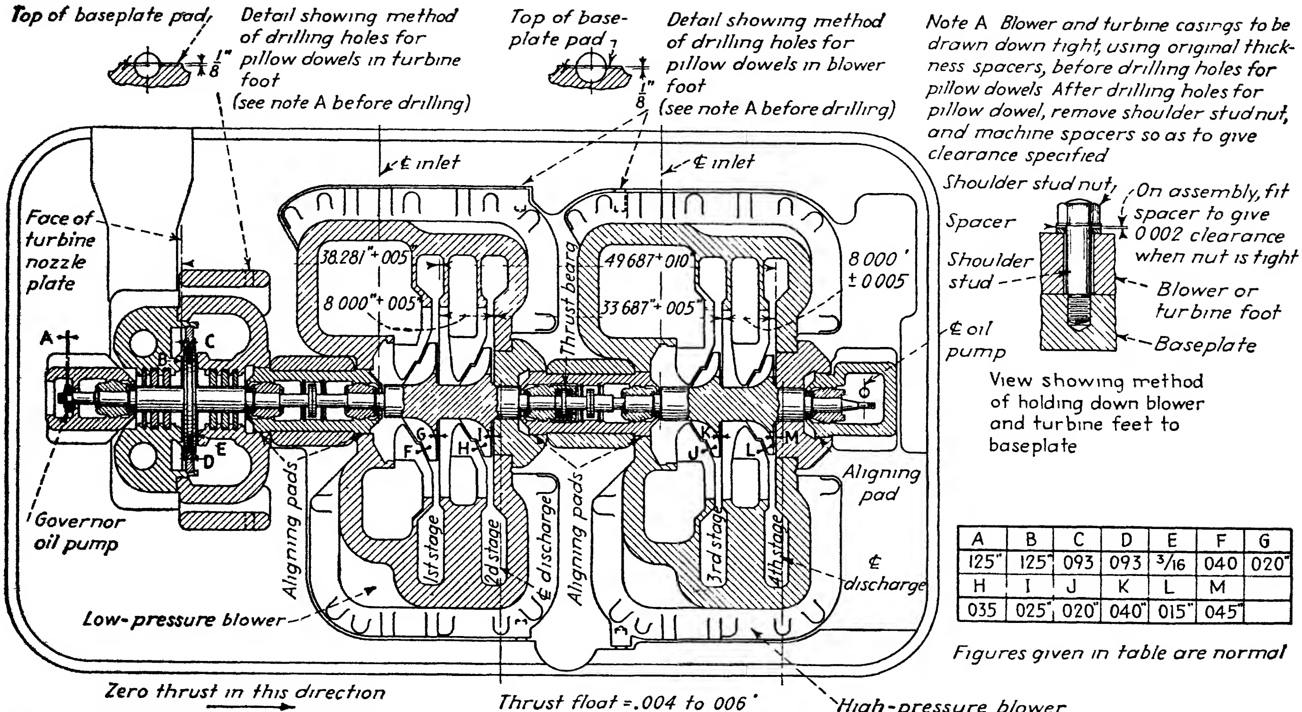


FIG. 346.—Aligning a flexible coupling with straightedge and feeler gage.

closed, the pulsations become less violent but of higher frequency and the pressure may drop as much as 15 to 20 per cent of the peak pressure at 35 to 40 per cent capacity.

From this it is apparent that if two units operate in parallel and the air demand is reduced, one of them will break down and the other will take the load. Moreover, the one forced off the line will not be able to open its check valve because of the drop in pressure at shutoff mentioned above.

It is obvious that, to parallel a centrifugal compressor with other units, it is necessary either to drop the line pressure by throttling the intake of the unit on the line or to by-pass sufficient air to bring the incoming unit's capacity to its stable operating range. To do this bring the unit up to speed as described under starting with the intake valve closed. The blowoff valve will



A	B	C	D	E	F	G
125"	125"	093	093	3/16	040	020"
H	I	J	K	L	M	
035	025"	020"	040"	015"	045"	

Fig. 347.—Instructions for drilling pillow-block holes, fitting casing-bolt washers, and measuring impeller clearance. (*Courtesy of Elliott Co*)

automatically open because the check valve is held closed. Open the discharge valve (the check valve will prevent backflow) and open the intake valve until the desired pressure or capacity is obtained. The check valve automatically opens and the blowoff valve automatically closes if the demand for air is sufficient.

If demand drops below the compressor's stable range, the blowoff valve automatically opens and prevents backflow or surging. The opening of the blowoff valve is adjustable to permit a flow of air equal to or slightly greater than the minimum stable capacity of the compressor. When the demand for air increases, the blowoff automatically closes in response to the opening of the check valve.

Since every operation of load control is performed automatically, the compressor requires very little attention. An occasional check on the temperature, pressure, and quantity of lubricating oil going to the bearings and the quantity of cooling water is all that is required.

Before stopping the compressor, transfer the load to other units. Do this by reducing the capacity to a point where the blowoff valve opens and the check valve closes. As soon as the check valve closes, the unit can be shut down. The discharge valve should then be closed, as there is always the possibility of a slight feedback through the check valve. As the unit slows down, the automatic control maintains pressure on the lubricating system by starting the auxiliary oil pump. Do not stop this pump until the main unit has come to a dead stop.

Maintenance.—Dismantle the compressor once a year, inspect the bearings, and clean the entire lubricating-oil system. Remove grease-packed bearings, clean out the old lubricant, and replenish with new. Do not fill the bearing cavity more than half full because too much grease causes the bearing to run unusually warm. One manufacturer recommends a grease of a relatively high melting point, compounded with a soda base and free from harmful ingredients or solid fillers. Excessive temperatures in grease-filled bearings can be caused by (1) insufficient grease, (2) too much grease, or (3) a defective bearing.

Take the rotor out, scrape off or dissolve all caked dust or other deposits on the impeller, and wipe the casing. When cleaning the impeller, inspect it closely for eroded or corroded areas and small individual deposits of foreign matter. A small deposit causes air swirls and eddies which in turn precipitate additional material on the deposit. (When removing the upper half of the casing, use long guide studs to prevent damage to impellers and seals.)

Inspect the labyrinth packing; if any sections have rubbed the shaft or impellers, scrape the seal to give proper running clear-

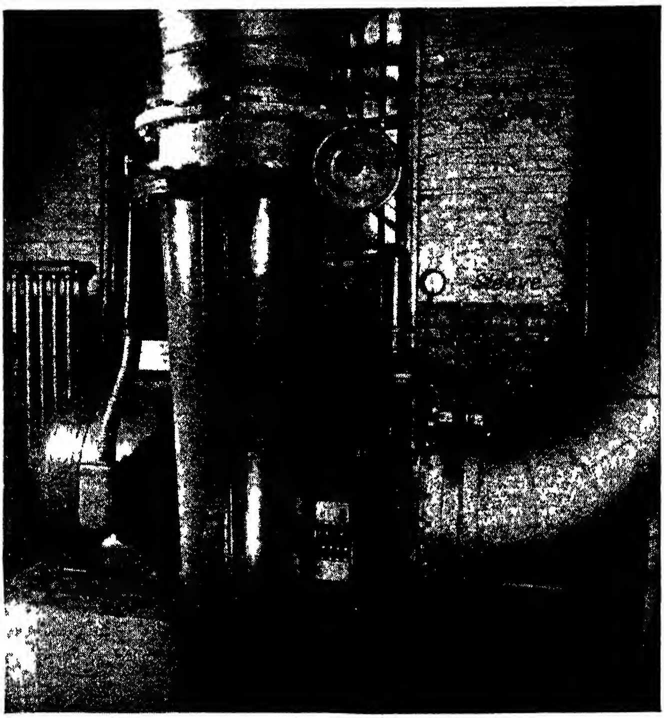


FIG. 348.—Flexible sleeves prevent piping strains on blower casing. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

ance. Before scraping, consult the manufacturer's instructions, as some seals using cast-iron rings in babbitt recesses require a running fit. To reduce air slippage, replace seals showing serious wear and excessive clearance. Seals with twice the initial clearance do not produce a noticeable reduction in output.

Shaft level and alignment sometimes change because of bearing wear; this, as well as end play, must be corrected immediately to prevent serious damage. The thrust bearing restrains unbalanced end thrust, keeping the rotor centered so that excessive end play does not damage labyrinth seals.

Should an impeller become unbalanced for any reason, consult the manufacturer before attempting to do any welding or add or remove any weight from it.

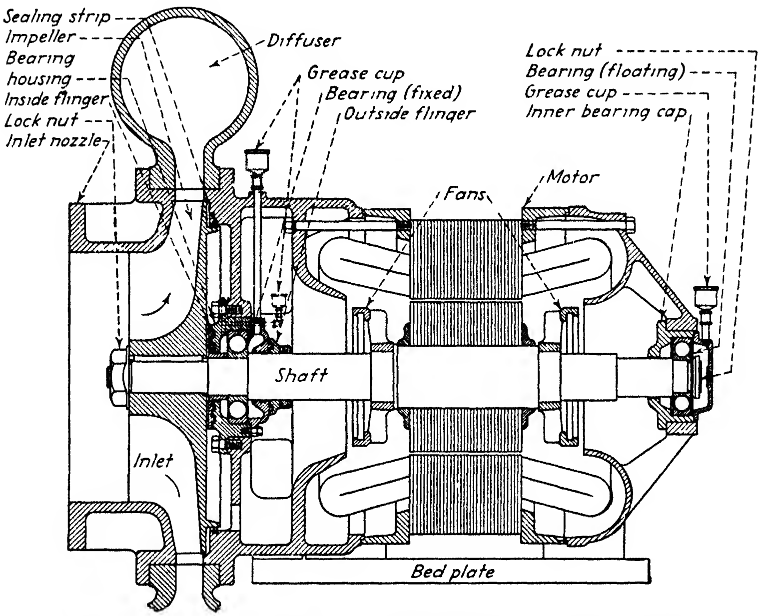


FIG. 349.—Mechanical details of a single-stage motor-driven blower.

When dismantling a motor-driven single-stage compressor having its impeller wheel mounted on the extended motor shaft (Fig. 349), proceed as follows:

1. Remove the blower inlet nozzle.
2. Remove the lock nut and lock washer from the end of the shaft.
3. Pull the impeller from the shaft with a pulling rig (Fig. 350). (The impeller usually has a taper bore to facilitate removal.) If the impeller is made of aluminum, do not heat it with a torch, because the heat is too intense; use steam instead.
4. Remove the inside flinger.
5. Remove the bolts from the bearing-housing cover.
6. Remove the bearing housing.

If it is necessary to remove the blower end bearing, continue as follows:

7. Loosen the setscrews in the outside flinger and slide the flinger back on the shaft.

8. Unfasten the grease fittings from the bearing-housing cover and slide it back on the shaft.

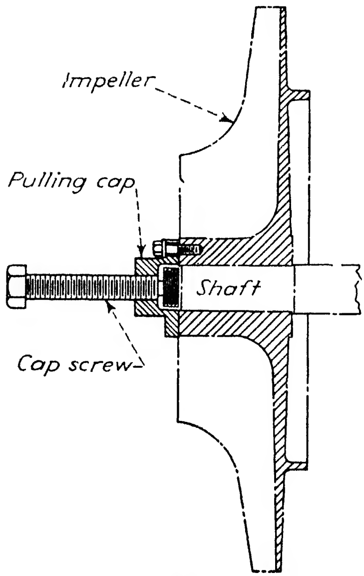


FIG. 350.—Pulling impeller from shaft.

9. When pulling a bearing off the shaft, insert a shim (6) between the bearing and puller (Fig. 351), so that force will be applied only to the inner race.

When reassembling, reverse the procedure outlined for dismantling, observing the following precautions:

1. Heat the bearing in oil to 250 F before sliding it on the shaft.

2. Pack the bearings with grease.

3. When replacing the bearing housing, be sure to bend over the lock washer's prongs.

4. Leave approximately 0.030-in. clearance between the outside flinger and the bearing-housing cover.

5. Press the impeller to place; do not sledge it home, because this may damage the shaft and ball bearings. The backplate of the impeller should bear against the inside flinger when the impeller is in its proper position. Bend one prong of the lock washer over so that the nut will not turn off.

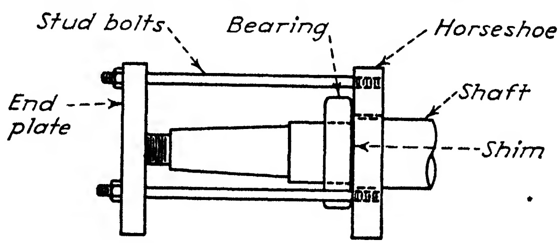


FIG. 351.—Use shim between horseshoe and inner bearing race to prevent strain on outer race.

6. Be extremely careful to use gaskets having the same thickness as those that were removed. This is important, because the clearance between the impeller and inlet nozzle is small.

CHAPTER XI

AXIAL-FLOW COMPRESSORS

Early History.—The axial compressor was first applied commercially in England early in this century to blow blast furnaces and boost gas pressures. It was soon superseded by the centrifugal compressor, which had higher efficiencies and more favorable operating characteristics that made it better suited for general applications.

About 1930, interest was revived in the axial-flow compressor by Brown, Boveri in an endeavor to find a less expensive and more efficient supercharging set for the Velox boiler (Fig. 352).

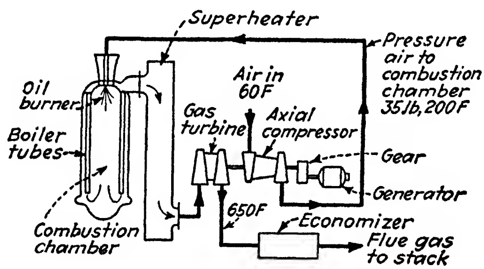


FIG. 352.—Velox steam generator supplied by axial compressor. *(Courtesy of Power.)*

In this boiler, the combustion chamber is supercharged, the supercharging compressor being driven by a turbine operated by exhaust gas from the boiler. Increased knowledge of air flow obtained from aerodynamic experiments has resulted in bringing the axial compressor to a high state of perfection.

Application.—Operating at high speeds with multistaging, the axial-flow machine supplies large volumes of air up to 60 psi in a steady stream free from pulsations. This makes it particularly suitable for supercharging the Velox boiler and producing high-velocity air flow in wind tunnels. Although axial compressors can be driven by steam turbines, their main application will probably remain in those fields where they can be driven by continuous-combustion gas turbines. At present this field lies mainly in the chemical processes—particularly the catalytic cracking of oil. In all such supercharging processes, the energy for compression comes largely from the process itself (Fig. 353) by utilizing the exhaust gas in a turbine.

When the energy available in the exhaust gas is greater than that needed to drive the compressor, excess energy can supply electric-power needs. Obviously, the less power required for compression, the greater the amount available for other purposes. Compressor efficiency is, therefore, an important item. Figure 354 shows the present adiabatic efficiencies of multistage axial and centrifugal compressors operating under the same conditions of 60 psi.

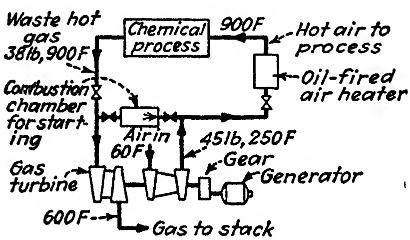


FIG. 353.—Axial compressor furnishes air for process. (*Courtesy of Power.*)

Operating Characteristics.—Like the centrifugal compressor, this machine requires no internal lubrication and does not build up dangerous pressures. One disadvantage of the axial compressor is its inherently high pumping limit. Figure 355 shows the comparative characteristics of centrifugal and axial machines. While the centrifugal compressor can operate at full speed down to 50 per cent volume stably, the axial machine begins to surge at approximately 80 per cent volume.

This characteristic prevents its application in many commercial services where load changes are considerable. This is not a serious fault in supercharging applications where it operates continuously at full capacity. For supercharging service it offers these advantages:

1. Compresses large volumes of air to 60 psi with high efficiency.
2. Requires no water cooling.
3. Operates at high speed and is therefore small and inexpensive. High speed permits a reduction in the dimensions and weight of the driving turbine.

4. Does not build up dangerous pressures.
5. Delivers air in a steady stream free from pulsations.
6. Requires no internal lubrication, and so delivers oil-free air.

Compressor Construction Details.—The axial compressor resembles a reaction steam turbine where each stage consists of a row of moving and a row of stationary blades. The rotor consists of a hollow steel forging, the moving blades being set in grooves in the drum. Stationary blades fit in grooves in the casing. Blade shape (borrowed from aviation practice) makes possible the high efficiencies obtained. The rotor turns in pressure-lubricated bearings, and, when coupled to a gas turbine,

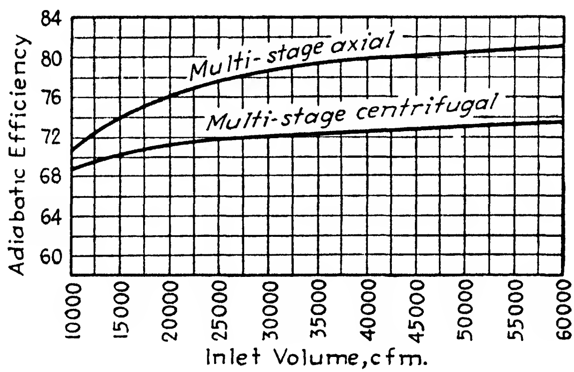


FIG. 354.—Comparative efficiencies of centrifugal and axial compressor working under similar conditions. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

axial thrusts oppose each other (Fig. 356). Figure 357 shows a cross section of a 20-stage, 40,000 cfm, 45 psi machine.

A cast-iron cylinder, split horizontally, has the inlet and outlet openings directed vértically upward and cast together with the upper half. The casing has 22 rows of stationary guide vanes which are calked in place and held in grooves, following general steam-turbine practice.

The forged-steel spindle consists of two parts—a drum with one shaft end, and a shaft end having a hollow cylinder for the rotor proper. The hollow cylinder, shrunk onto the drum and locked, is grooved to receive the rotor blading, which is wedged and calked in place (Fig. 358) similar to the cylinder blading. Labyrinth sealing strips are provided where the shaft emerges from the casing.

The spindle rides in roller bearings that have more internal clearance between the rollers and races than standard bearings. The inner races have been given a stabilized heat treatment to prevent growth.

Each bearing consists of either a single, or two duplicate roller bearings, as required to carry the load. The inner race has a shrink fit on the shaft and is locked against a shoulder by a lock nut. The outer race mounts in a spring ring which fits in a groove in the low-pressure end bearing cover. On some compressors, where two duplicate bearings are used, the spring rings are mounted in a bearing retaining ring, which separates the two bearings. A Kingsbury thrust bearing holds the compressor spindle axially in its casing.

Gas-turbine Construction.—The gas turbine is of the straight reaction type with rows of stainless-steel stationary and moving blades (Fig. 359). The turbine casing is made of molybdenum cast steel and is split at the horizontal center line. Both inlet and outlet nozzles are located in a vertical plane, the inlet nozzle being cast in one piece with the upper cylinder half and the gas-exhaust nozzle being cast in one piece with the lower half. A by-pass safety valve connects the inlet and exhaust passages.

The turbine spindle consists of a solid chrome-nickel-steel forging. Five serrated grooves hold the stainless-steel blades. Labyrinth glands are provided where the spindle ends pass through the casing. Sealing air is taken from the compressor exhaust and injected at a suitable point along the labyrinth to prevent gas from leaking to atmosphere. Separate valves are provided in the sealing lines to both glands.

The outer races of the turbine bearings fit in spring rings of a special alloy steel mounted in grooves machined in the turbine bearing covers. The spring rings aid in aligning the bearings and allow for radial expansion. The inner races are pressed on the shaft sleeves and are locked by retaining rings.

A rigid coupling connects the compressor and turbine shafts. Coupling faces are accurately ground and the two halves fasten together with fitted bolts. A continuous lubricated flexible coupling connects the compressor and speed-reducing gear, which connects to a generator and starting machine. Each half is keyed, pressed, and locked on a shaft extension. Lubrication is furnished through a supply line from the compressor bearing supply. Another flexible coupling connects the low-speed gear shaft with the power-recovery unit. The center section may be removed without disturbing the coupling halves.

Complete Unit.—The complete unit (Fig. 360) consists of an axial compressor direct-connected to a gas turbine at one end and coupled to the high-speed shaft of a reduction gear at the other end. A starting turbine or motor is connected to the low-speed shaft of the gear, and on some units a generator is connected to the same shaft between the gear and the starting unit.

The unit mounts on a base plate with the compressor and turbine arranged so that the cold end of each machine is fixed and the other end is free to expand under the influence of heat.

Beginning from the outboard end of the turbine, the bearings are numbered 1, 2, 3, and 4, respectively. No. 1 bearing is mounted on the outboard end of the turbine cylinder. No. 2 and 3 bearings are mounted on the compressor cylinder, one on each side of the coupling. No. 4 bearing is mounted on the compressor cylinder at the outboard end. A Kingsbury thrust bearing, located adjacent to No. 4 bearing, maintains the axial position of the two spindles and takes any thrust which may be present.

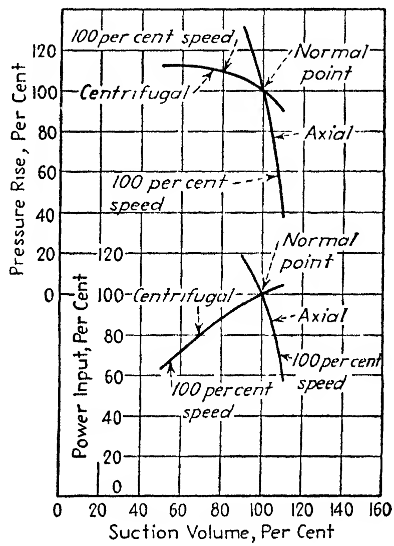


FIG. 355.—Comparative operating characteristics of centrifugal and axial compressor. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

The turbine and compressor cylinders are each supported at their four corners on keys that lie in transverse keyways cut in the bedplate. The ends of the cylinders which lie adjacent to one another are keyed to the bedplate in a manner which prevents their moving axially at that point. The opposite ends of the cylinders rest on keys but are free to slide, thereby taking care of expansion. Transverse movement of the cylinders is prevented by guides fastened to the bedplate along the center line of the unit. Two guides are provided for each cylinder, and on some units an additional guide is provided for the No. 1 bearing pedestal. After the unit has been aligned, fit the guides to their keyways in the cylinders with the gib keys furnished with the machine.

Operating Principle of the Gas Turbine.—The gas turbine as a primary power unit has been little more than an inventor's dream until recently. It offers advantages parallel to those of the steam turbine over a steam engine, in that direct conversion of fuel energy to power on a rotating shaft is possible, eliminating the reciprocating motion, difficult problems of fuel injection and cylinder lubrication, and all need for cooling water. It is the latter, and certain characteristics of use in chemical processes, that have provided the incentive for development and offer the best possibilities for future applications.

The simplest form of gas-turbine compressor is a complete unit with a fuel-burning chamber and generator (Fig. 361). Operation of this unit is continuous. Air from the room or outdoors is compressed to 30 to 50 psi, mixed with oil fuel, and burned in the combustion chamber. The products of combustion, including excess air added to keep the gas temperature down to a predetermined limit, expand through the turbine blading to the exhaust flue or stack. About 75 per cent of the output of the turbine goes to drive the compressor; the remainder is available for electric generation.

In the combustion-gas-turbine cycle there is no boiler, no feed pump, no condenser or cooling water. The only auxiliaries required are a fuel pump and a means of starting. For the latter, a small motor or steam turbine brings the unit up to about one-quarter speed. At 1000 F gas temperature, an over-all thermal efficiency of 15 to 18 per cent is possible (19,000 to 23,000 Btu per kwhr at the generator coupling). Considerably better efficiency could be obtained for direct power production by raising the inlet-gas temperature to 1200 F.

Another method of increasing efficiency is to preheat the compressed air with the turbine exhaust (Fig. 362). A surface of 2½ sq ft per kw of net output increases efficiency about 2½ points. Better efficiency at partial load can be secured by a two-shaft arrangement, one gas turbine driving the compressor at best compressor speed, and another running at constant speed driving the generator. One compressor and combustion chamber supplies both turbines (Fig. 363). Figure 364 shows the improvement in thermal efficiency.

A cycle embodying regeneration, reheating, and intercooling is shown in Fig. 365. In this arrangement air from the atmosphere enters the compressor where its pressure is increased to an intermediate value. It is then passed through an intercooler that lowers its temperature and specific volume. The cooled air

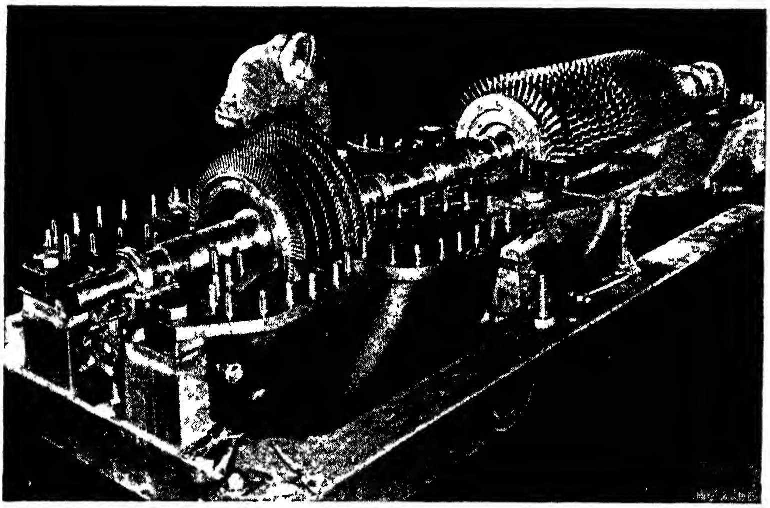


FIG 356 —Gas-turbine and axial-compressor rotors connected for opposed thrust. (*Courtesy of Allis-Chalmers Manufacturing Co*)

then enters the high-pressure element of the compressor where it attains final pressure. In addition to reducing the power required for compression, the intercooler also lowers the temperature of the air leaving the compressor, thus making the heat exchanger more effective.

The discharged air is preheated in a heat exchanger by the exhaust gas from the low-pressure turbine and is then further heated to a satisfactory turbine-inlet temperature by the injection of fuel in a high-pressure combustion chamber. Expansion of this high-temperature gas in the turbine furnishes the necessary power to drive the compressor.

Gas leaving the high-pressure turbine is reheated to inlet temperature and then expanded in a low-pressure turbine to furnish useful power. Exhaust gas from the power turbine in traversing

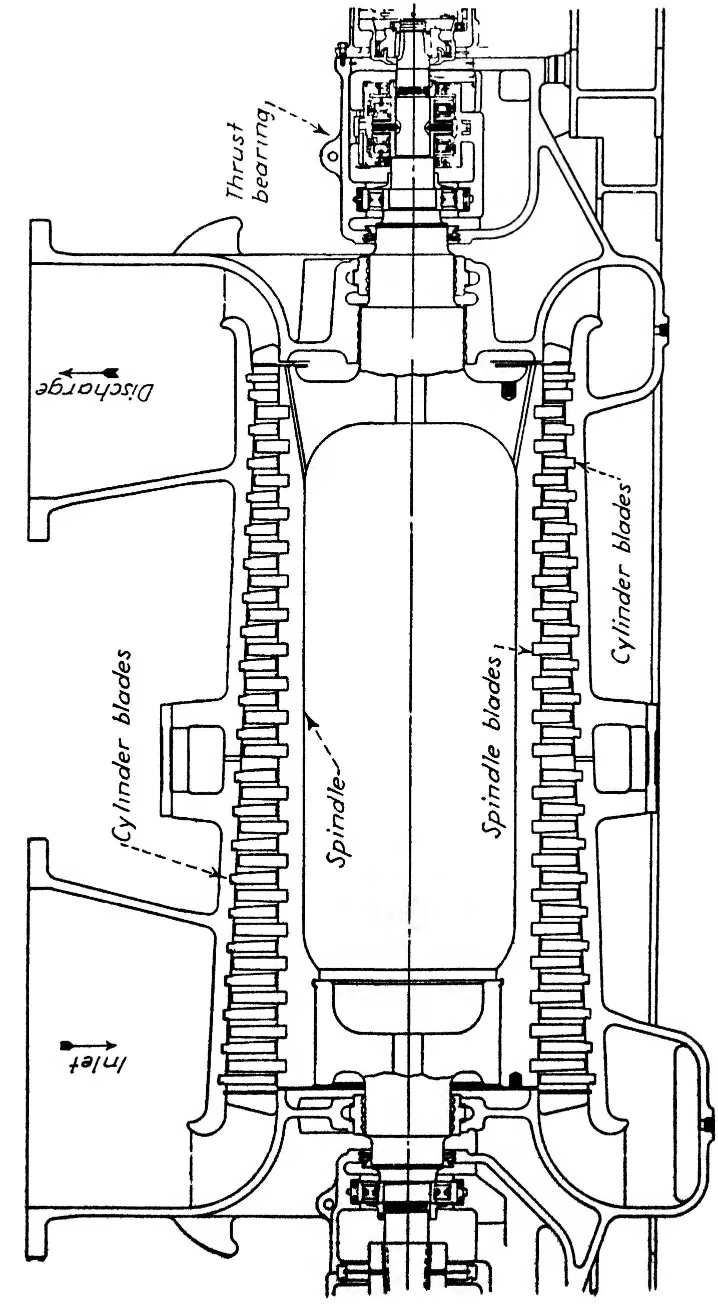


FIG. 357.—Cross section of 20-stage axial-flow compressor with roller-bearing-supported spindle.

the heat exchanger preheats new air leaving the compressor and then discharges to atmosphere.

When this cycle operates with a turbine-inlet temperature of 1200 F, a compressor pressure ratio of 5, and surfaces in the heat exchanger of 5,000 and 1,500 sq ft, respectively, a thermal efficiency of 30 per cent is obtained at the turbine coupling. Such an efficiency is equivalent to or even better than that achieved in both marine steam installations and small land steam-power plants.

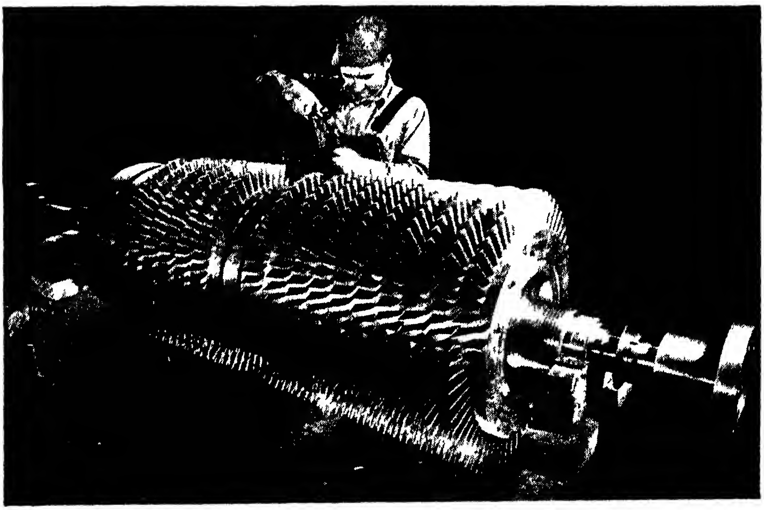


FIG. 358.—Blades are wedged and calked in place as in steam-turbine practice. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

The 1000 F limit should not be taken as the ultimate for operation of turbine parts. There is a reasonable expectation of increasing this temperature limit, which, when it materializes, will open up still broader fields of application.

Power generation by the gas turbine is limited at present to liquid or gaseous fuels, as it is too early to predict the outcome of experiments with pulverized coal. However, the lowest grades of bunker *C* oil which cost only about half as much as diesel fuel oil can be used.

Machines similar to the one in Fig. 360 have been developed to a high degree of refinement and applied to the Houdry process of oil refining to supply large quantities of compressed air.

Power to drive the compressor is generated in the gas turbine by waste products of the process. Since the power produced is in excess of that required by the compressor, a relatively small amount of by-product electric power is generated. The generator is geared, 5,200 to 1,800 rpm, and a separate starting steam turbine is provided at the extreme right.

Pressure drop in the cracking process reduces the amount of by-product power available below what could be made if the unit were operated for power production only. In one typical unit the gas-turbine output is 5,300 kw; the compressor requires 4,400 kw, leaving 900 kw for electric generation.

The gas turbine as a power producer has the advantage over a diesel that the lowest grades of fuel oil can be fired, and waste gas of any quality or heating value is suitable. The difference in fuel price in many cases makes up for the inherently lower efficiency of the gas turbine in comparison with the diesel.

A gas-turbine unit for 2,000 kw net output weighs about 40 tons without the generator, and the cost compares favorably with a diesel. Since no cooling water or other auxiliary apparatus is required for the turbine, comparison of the investment on a complete plant basis is considerably in favor of the turbine.

Load Control.—A by-pass safety valve protects the gas turbine from overspeeds above 10 per cent in case the electrical load is suddenly removed. This valve (Fig. 366), when actuated by the emergency stop, permits the gas to pass directly from the turbine inlet to the exhaust, thereby depriving the turbine of its motive power.

This single-seated unbalanced valve is provided with spring-loaded power piston 13. Unbalance of the valve is in the opening direction. The downward thrust of the piston (depending on the oil pressure) is such that the valve is kept shut during normal operation.

Orifice 21 supplies oil on top of spring-loaded pilot valve 16. Pressure in dome 18 builds up until supporting spring 15 on the pilot is compressed and the pilot valve is seated. From that moment on, oil supplied to main piston 13 through opening 21 cannot escape and the piston travels downward, compressing main spring 12 and closing by-pass valve 5.

When the emergency governor responds to overspeed, oil escapes from the top of pilot valve 16. Spring 15 lifts the pilot

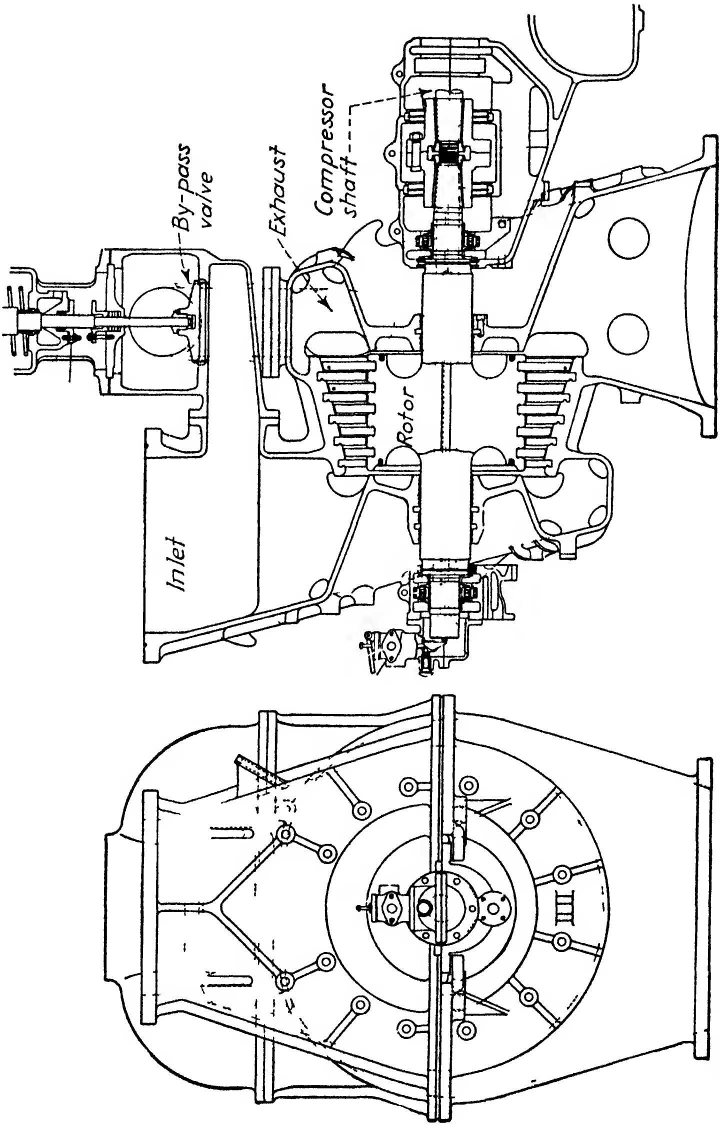


FIG. 359.—Cross section of a five-stage gas turbine that drives the compressor in Fig 357. (*Courtesy of Allis-Chalmers Manufacturing Co*)

valve, and the oil on top of power piston 13 drains to the oil tank through outlets 20 and 14. The unbalanced force under the valve and main spring 12 opens the by-pass valve.

Upon resetting the emergency-stop plunger mounted on the bearing housing, the oil pressure is built up and pilot valve 16 and main valve 5 are closed.

Provision is made to trip the by-pass valve manually. A pipe connecting the dome with the drain line is furnished with a spring-loaded hand-operated valve. Releasing the oil pressure in the dome through this valve opens the by-pass valve.

Emergency Governor.—The emergency governor (Fig. 367) provides against overspeed. A branch of the governing oil line that connects to the dome of the by-pass valve also connects to the emergency governor. Oil pressure is applied on piston 6 at 28. If emergency trip 12 operates on overspeed, the governor spindle strikes lever 7, which provided the support for piston 6. The piston is released, and spring 19 pushes it down. Slots provided in the cylinder wall and connecting to drain 16 are uncovered when the piston moves down, and the oil pressure in the line to the dome of the by-pass valve vanishes, thereby operating the latter valve in an opening direction. After the turbine speed has decreased, the governor spindle of emergency stop 12 returns to normal position and the trip piston can be reset by lifting handle 5. Provision is also made to trip the turbine by hand by striking button 10 lightly.

Speed-control Governor.—The overspeed-governor mechanism combines simplicity, ruggedness, and positive control of the turbine with great sensitivity for close speed regulation. The governor mechanism and blowoff valve (Fig. 368) provide protection against overspeed. The governor is so adjusted as to be inactive below 3 per cent overspeed, at which point the blowoff valve begins to open and air from the compressor discharges to atmosphere. Handwheel 28 adjusts the speed at which blowoff occurs. Full opening is obtained at 7 per cent overspeed.

The governor mechanism consists of speed-governor assembly 19, pilot-valve assembly 25, and oil-relay-controlled blowoff valve 31 with suitable restoring mechanism to provide an instantaneous response to speed changes.

Speed governor 19 is a standard spring unit. Governor weights are secured to two cylindrical rollers rolling on guides provided in the governor casing, these guide surfaces being parallel to the governor center line. Flat flexible springs connect the rollers with the governor casing and central stem. A coil spring loads the governor and this is readily adjustable by means of nuts on the spring casing. The governor spindle connects to pilot valve 25, which controls the oil pressure operating the blowoff valve.

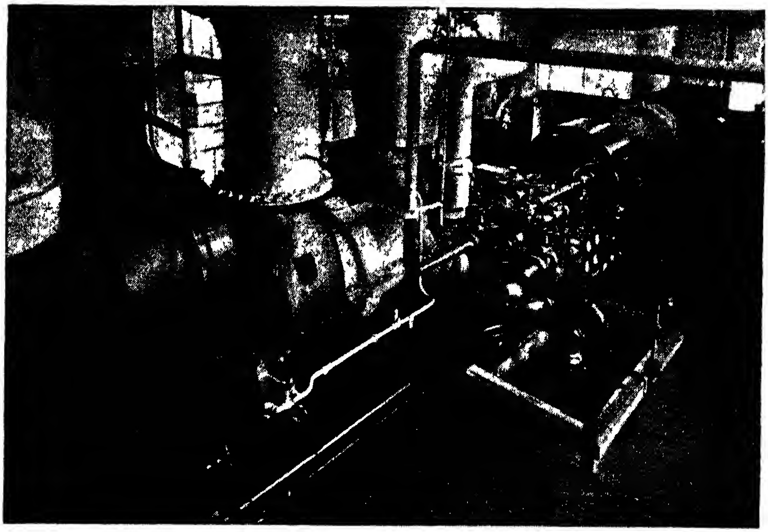


Fig. 360.—Complete unit—gas turbine, axial compressor, speed reducer, induction generator, and starting steam turbine. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

As turbine speed increases, the weights secured to the rollers tend to move apart owing to centrifugal force. When these forces become greater than the combined force of the flat and coil spring, the rollers roll upward and force the spindle up.

This motion is transmitted to pilot valve 25, which moves up and releases the oil pressure in line 29. The resulting motion causes the oil relay on the blowoff valve to admit oil pressure from connection 42 through port 44 above piston 40. The piston moves down and forces the oil below it out through port 43 into the oil relay where it is drained through connection 45. The downward motion of the piston is transmitted to the restoring mechanism, thus adjusting the opening of the blowoff valve to the desired point.

With any decrease in speed, the reverse action takes place, and in this manner the unit's speed is effectively controlled. The governor head rotates in sleeve bearings above and below spiral gear 14. The weight of the governor-head assembly is taken on a sleeve-type thrust bearing (17 and 18) located directly above the spiral gear. Although the governor is extremely sensitive, it is still quite rugged and not subject to variation due to wear.

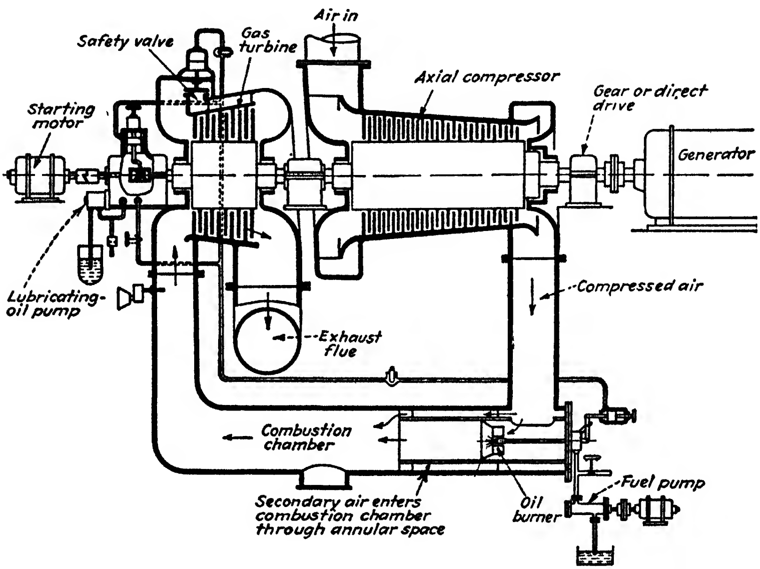


FIG. 361.—Continuous-combustion gas turbine with its compressor and combustion chamber. (*Courtesy of Power.*)

Lubricating-oil System.—The main oil pump is a gear unit driven from the low-speed shaft of the speed reducer; the auxiliary pump is a motor-driven rotary unit mounted on top of the oil tank.

The oil cooler consists of a tube bundle, suitably baffled and enclosed in a cast-iron shell. The shell ends are provided with removable heads to facilitate inspection and cleaning without disturbing existing pipe connections. Water flows within the tubes and the oil is directed across their outside.

The oil tank is of welded construction with its cover reinforced to provide a mounting for the auxiliary oil pump. A magnetic liquid level gage, float operated, is mounted on the tank.

The main oil pump takes oil from the tank and delivers it through the cooler to the speed reducer and compressor and turbine bearings (Fig. 369). A relief valve maintains pressure on the bearing supply lines and by-passes excess oil back to the tank. The bearing and gear-case drains return to the tank. Straight-through oil-flow sights and thermometers in the drain lines check oil flow and temperature.

The auxiliary pump takes oil from the tank and discharges it through the relief valve to the cooler and bearing supply line. This pump also supplies oil to the governor system.

Lubricant is sprayed on the roller bearings by pipes which contain a small orifice through which a jet of oil discharges. This oil strikes a flinger on the shaft which breaks it up into a spray to lubricate the bearing. Gas-turbine bearings are equipped with double jet pipes, one on each side of each bearing; compressor bearings have one jet pipe located on the coupling side of each bearing.

On some compressors fitted with double-roller bearings, the oil is introduced through a passage fitted with a metering orifice in the bearing retainer. Oil enters the bearing through this orifice and flows outward in both directions between the inner and outer bearing races.

Oil goes to the thrust bearing through two inlet pipes near its bottom. Each inlet pipe contains a metering orifice plug and dial gage which indicates the oil pressure on the bearing side of the orifice.

The coupling between the compressor and the speed reducer is lubricated by a jet pipe that delivers oil to collector rings on the coupling. This jet pipe is located on the compressor side of the coupling. Oil for lubricating the speed reducer comes from the bearing supply line.

Governor Oil System.—High-pressure oil required for the by-pass safety valve and overspeed-trip valve is supplied by a second gear pump driven from the low-speed shaft of the speed reducer.

The pump takes its suction from the tank and delivers oil through an adjustable needle valve to the by-pass valve and overspeed trip. Drains are brought back to the oil tank.

Oil supplied to the relay and power piston used to operate the blowoff valve comes from the bearing oil supply line. Pressure in this line is regulated by a reducing valve which by-passes oil to the drain when the pressure gets too high.

Auxiliary Oil-pump Control.—The auxiliary oil pump provides oil to both the governor system and the bearings during starting and also serves as a stand-by for the main governor oil pump and bearing pump.

Both the governor oil line and bearing line have pressure-operated switches adjusted to close and start the auxiliary oil pump when oil pressure falls slightly below normal (Fig. 370). A pilot light on the auxiliary pump push-button station immediately informs the operator that the auxiliary pump is running.

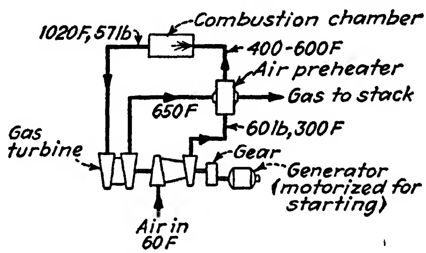


FIG. 362.—Air preheater increases the efficiency of one-shaft units. (*Courtesy of Power.*)

Erecting the Unit.—When erecting the unit, put the bedplate upon its foundation, then place the lower half of each cylinder on the bedplate and line up the bearing bores. When lowering the spindles into place, use the lifting rigs provided and be careful not to damage the labyrinth sealing strips. Guide pins (if any) in the bearing spring rings should be pointing vertically upward when the bearings are in place.

Machines are carefully aligned and given a test run at the factory. To check alignment, separate the rigid coupling slightly and with a thickness gage check to see if the two faces are parallel. After these faces have been made parallel, attach a dial indicator to the compressor half of the coupling, with the indicator following the hub of the turbine half of the coupling. Rotate the compressor and check to see if the shaft centers are in line. By using a small mirror in the bottom of the coupling housing, the dial can be read in any position.

After the first alignment is obtained, pull the foundation bolts up fairly tight and recheck the alignment. When the alignment is completed, fill the bedplate with grout.

The thrust bearing and its housing may be assembled to the compressor shaft before the spindle is lowered into the cylinder. The bearing should be assembled with the filler piece located at the outboard end. The thrust-bearing housing has a tongue

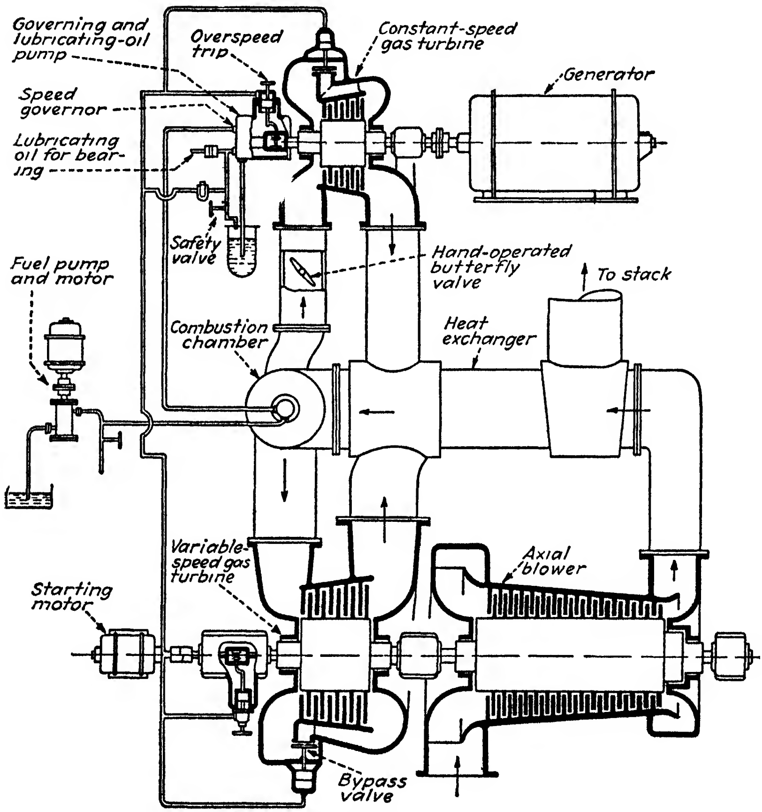


FIG. 363.—Two-shaft arrangement gives better efficiency than a one-shaft unit. (*Courtesy of Power.*)

that fits between the lock rings in a groove cut in the lower half of the compressor cylinder. These lock rings were fitted at the factory and hold the housing in its correct position. They are marked to correspond with letters stamped in the compressor cylinder to indicate their correct positions.

A line has been scribed on the surface of the compressor cylinder in the plane where the faces of the rigid coupling meet. This reference line is to be used when checking the radial clearances of the turbine blading. It may also be used as a guide during erection until the thrust-bearing housing is locked in place.

The radial clearance of all compressor blades should be equal to or slightly more than 0.035 in. Experience has shown that the compressor has ample clearance when a 0.035-in.-thick shim can be shoved through each row of blading between the tips of the spindle blading and the lower half of the cylinder, and between the tips of the cylinder blading and the spindle. Clearance for the upper half of the cylinder can be obtained with lead wire.

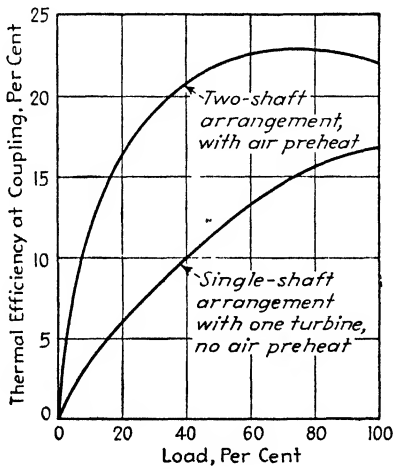


FIG. 364.—Comparative thermal efficiencies of one- and two-shaft machines. (*Courtesy of Power.*)

When taking these clearances, bring the face of the coupling up to the line scribed on the surface of the compressor cylinder. Check this position by laying a straightedge on the line. Make a record of the original clearances of the turbine and keep it for possible future use.

Piping to and from the compressor and turbine *must* be properly supported to prevent weight strains from being transmitted to the compressor or turbine. Proper allowance must be made for compressor and turbine expansion to avoid casing distortion. Expansion joints are recommended for both the inlet and discharge lines of the compressor and turbine.

Aligning the Speed Reducer.—The speed-reducer gear unit located between the compressor and starting unit must run in the direction indicated by the arrow on the gear-case cover. Follow these pointers when installing it:

1. Set the unit to the correct horizontal position by building up with broad flat shims. Do not use wedges.

2. Remove the inspection cover and lay a machinist's level on the machined surface, first lengthwise and then crosswise of the reducer, and, if necessary, change the shimming to bring the reducer into a level position. Be careful not to drop any dirt down into the reducer, and replace the inspection cover immediately after making this check.

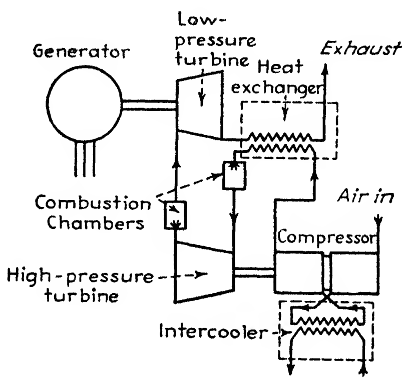


FIG. 365.—Regenerative cycle with reheating and intercooling on a two-shaft unit. (*Courtesy of Power.*)

3. Swing the reducer sufficiently to bring its shafts into horizontal and vertical alignment with the driving and driven shafts.

4. When coupling hubs are mounted in the field, they should be heated and shrunk on, not pressed or driven on unless the shafts are removed, because an endwise force or blow on the shaft may cause serious damage to gears and bearings.

5. Where a gear unit is mounted on a subbase, be sure to check the alignment at the couplings after the base has been bolted down. If it has been disturbed, true alignment should be restored by shimming.

6. Tighten all bolts on the gear unit after it has reached normal operating temperatures.

Drain and filter the oil at regular intervals of from 3 to 6 months. It is especially desirable to filter after the first week of operation in order to remove metallic particles which are a result of the initial wearing-in period. Clean the spray pipes at regular intervals.

A steady increase in oil temperature with normal operation indicates that the oil is dirty or contains sludge. The lubricant should be drained and, if not too dirty, refiltered; otherwise, new oil should be used. To avoid possibilities of explosion, keep open lights away from gear-case openings when the oil is hot and vaporous.

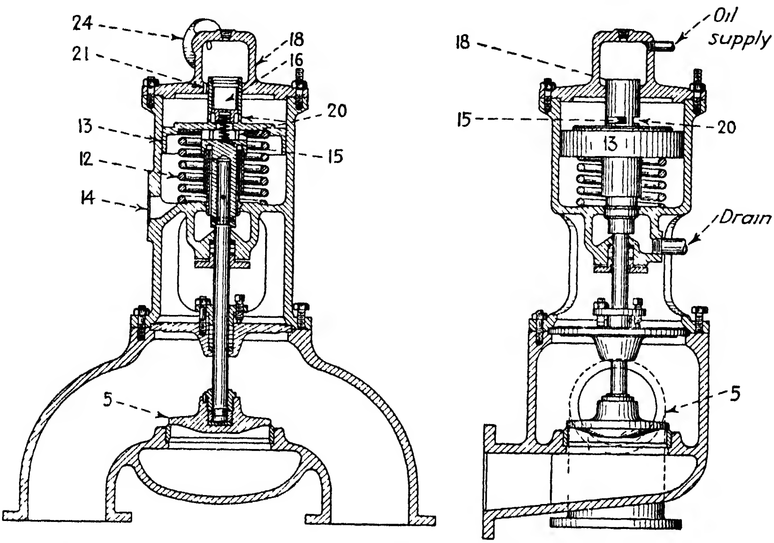


FIG. 366.— By-pass safety valve shunts gas from turbine inlet to the exhaust on overspeed. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

Over-all Erection Pointers.—During erection, check the following points and make necessary adjustments before attempting to start up the unit:

1. Check the alignment carefully after air and gas piping have been installed.
2. Thoroughly clean all bearing housings and all parts of the lubrication system before installing. The parts should be flushed with kerosene and blown out with an air blast. Do not use waste to wipe them dry.
3. Fill the oil tank with a light oil to flush the system. Check the direction of rotation and operation of the auxiliary oil pump. Circulate oil through the system. When the pressure drop across the filter starts to increase, turn the filter handwheel once or twice to clean it. Continue circulating the oil until the system has been thoroughly flushed, then drain, and clean the tank.
4. Refill the tank with a good grade of clean turbine-quality lubricating oil, which is suitable for high-speed bearing service. The oil should have a viscosity of 350 SSU at 100 F.

5. Check the oil and cooling-water system for leaks. Adjust the lubrication- and governing-system control units according to the manufacturer's instructions.
6. Roll the unit slowly and check carefully to see that everything is free.
7. After satisfactory mechanical operation has been obtained, make the

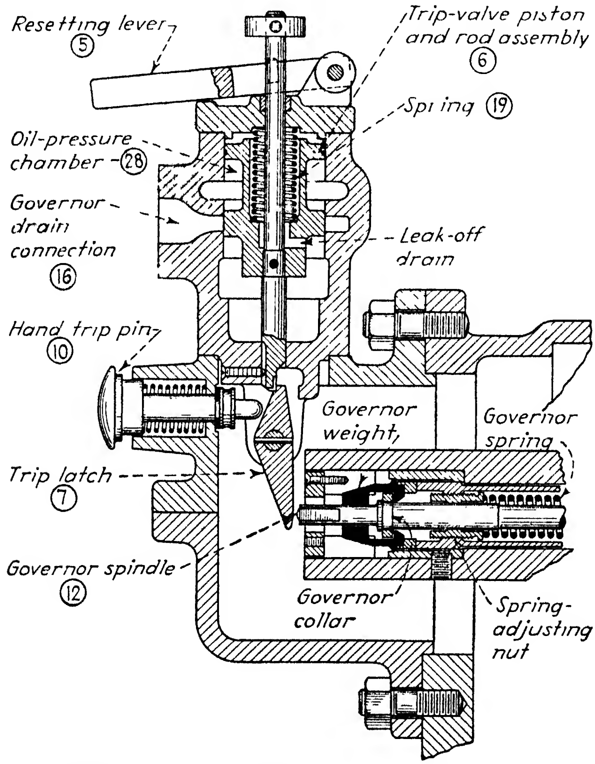


FIG. 367.—Emergency overspeed governor shuts unit down on overspeed. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

turbine- and compressor-casing horizontal joints permanent with a mixture of triple-boiled linseed oil and graphite, of a consistency such that it will run off the brush when being applied.

Starting.—When the unit is fitted with a steam turbine for starting purposes, follow this procedure:

1. Start the auxiliary oil pump.
2. Check the oil pressures and oil-sight feeds.
3. Check the oil level in the starting turbine.

4. Test the gas-turbine constant-speed governor by screwing the handwheel of the main-governor linkage all the way down, then all the way up. Leave the wheel all the way *down*.

5. Test the gas-turbine overspeed valve by tripping the overspeed latch. Reset.

6. Test the steam-turbine overspeed trip. Reset.

7. Open the cooling water to the steam-turbine bearings.

8. Open all steam-turbine drain valves.

9. Open the steam-turbine exhaust valve.

10. Crack the steam-turbine throttle valve and bring it up to part speed. The operating instructions furnished by the turbine manufacturer should be carefully followed. Check immediately to see that the bearing oil rings are turning.

11. Start the cooling water when the return-oil temperature from the bearings reaches 125 F. Regulate the flow to keep it from 125 to 130 F.

12. Check the bearings for noise and temperature. Check the glands for noise. Open the valves to the turbine gland seals. Check the sound of the rotors for rubbing.

13. When water is out of the steam system, close all drain valves.

14. Open the steam throttle to bring the unit up to approximately 40 per cent speed.

15. Check again as in step 12.

16. Light the burners if necessary to bring the inlet-gas temperature up to normal.

17. Cut down on the steam throttle as the gas turbine comes in under its own power.

18. Shut down the auxiliary pump manually. If the governor and main oil pump do not maintain full operating pressure, the auxiliary pump will continue to run. This condition should be corrected before putting the unit into service.

19. Close the steam throttle valve.

20. Keep the steam-turbine exhaust valve open.

21. Open all steam-turbine drains.

22. Regulate the gas-turbine speed with the handwheel on the main governor. Adjust the inlet gas to the maximum continuous allowable temperature.

23. Pay close attention to the temperature of the oil discharged from the bearings. In case of abnormal bearing temperatures, shut down the unit, determine the cause of overheating, and correct it before again placing the unit into operation.

Shutting Down.—When shutting down, proceed as follows:

1. Reduce the generator load to zero and take the generator off the line (for units driving a generator).

2. Put out the burners.

3. Start the auxiliary oil pump when the oil pressure begins to decrease. Check the auxiliary-pump discharge pressure.

NOTE: If the auxiliary pump is equipped with automatic control, the pump should start up in case of low oil pressure.

4*a*. For starting motor: When the unit has slowed down sufficiently, start the motor and engage the clutch.

4*b*. For starting turbine: Crack the throttle valve sufficiently to keep the unit rotating at slow speed.

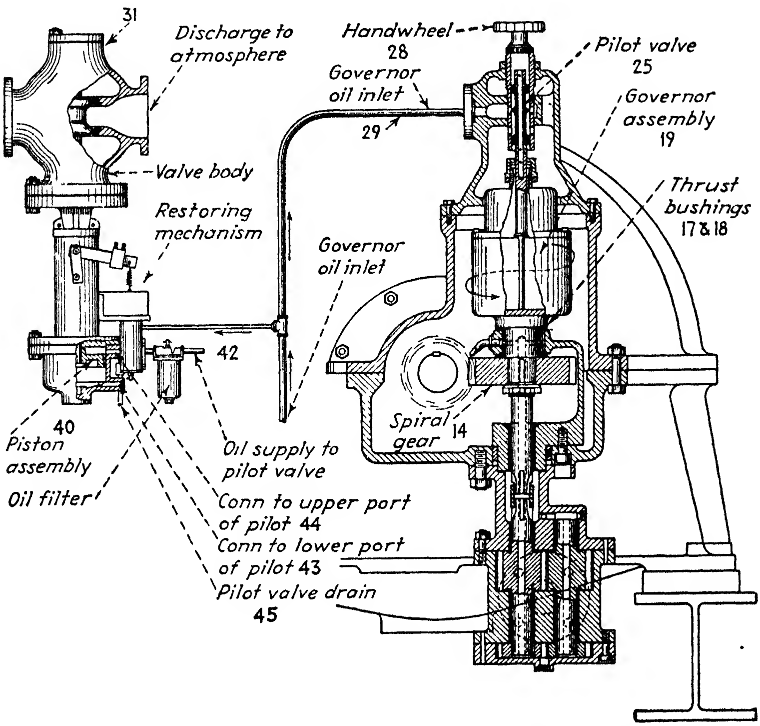


FIG. 368.—Overspeed protective governor regulates compressor by-pass to atmosphere to control turbine speed. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

5. Keep the unit turning over until the gas-turbine-cylinder temperature has dropped to 250 or 300 F.
6. Shut down the driving motor or turbine. Open the turbine drains.
7. Shut off the cooling water.
8. Shut down the auxiliary oil pump.

On an emergency shutdown, trip the overspeed lever at the outboard end of the turbine and start the auxiliary oil pump. Then proceed as above if the unit can be kept running. If the machine cannot be kept rotating, let the auxiliary oil pump run and turn the rotor by hand one-half turn every 15 min.

Normal Operation.—The compressor is essentially a constant-volume machine and delivers a steady flow of air when operating in its stable zone. If the discharge pressure is raised sufficiently by increasing the resistance in the discharge line, the compressor

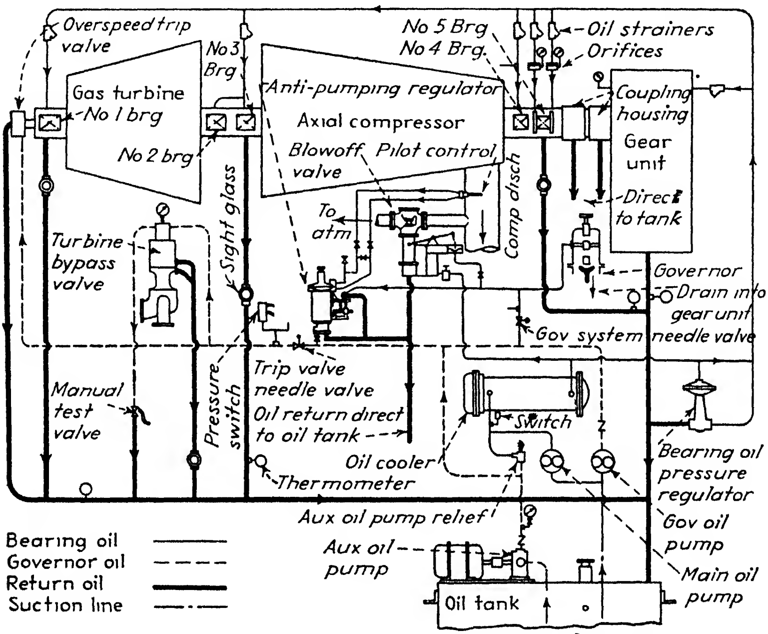


FIG. 369.—Lubricating and governor oil system served by separate pumps.

will begin to surge. This condition should be avoided. Surging can be prevented with a regulator arranged to increase the volume handled by blowing off air at the compressor discharge when the surging zone is reached.

The unit runs with a minimum of vibration at all speeds, except when passing through its critical-speed zone. In case of excessive vibration, shut the machine down and correct the fault at once.

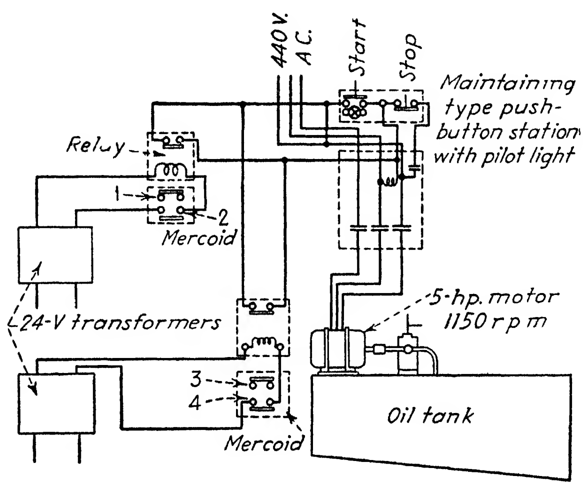
Excessive vibration may be caused by

1. Misalignment.
2. Rubbing of the rotor or stationary blades from
   a. Distortion of casing.
   b. Excessive heating.

3. A defective bearing.
4. Defective gears.

Bring the unit up to speed slowly, and allow it to cool slowly when shutting down. Check the alignment frequently when it is first placed in operation. Oil discharged from the bearings should be maintained below 150 F.

Dismantling.—The compressor and turbine upper-half casings may be lifted after removing the connecting piping and bolts.



1-3 Alarm contact, closes with drop in pressure
2-4 Aux. oil pump contact, closes with drop in pressure

FIG. 370.—Wiring diagram of auxiliary oil-pump motor.

Lift the casing slowly and evenly to avoid binding on the cylinder-guide studs.

To remove the compressor or turbine rotor

1. Remove the bearing covers.
2. Disconnect the couplings.
3. Lift the rotor, using the lifting rig furnished with the unit. (Separate rigs are provided for turbine and compressor rotors.)

NOTE: When the coupling between the compressor and turbine has a male and female fit, the turbine rotor *must* be moved axially to clear this shoulder before lifting.

The rotors are statically and dynamically balanced and given a mechanical test before shipment. Balance is obtained by adjusting screws located at each end of the rotor. Should unbalance occur, consult the manufacturer.

Remove the upper half of the compressor, turbine, and gear casings to inspect the rotating parts at regular intervals, usually once a year. Bearings should also be inspected at this time.

Filter the lubricating oil frequently and clean water and sludge from the reservoir. The oil should be renewed if its appearance or excessive sludging indicates decomposition. Check the filters monthly, and clean the oil-cooler tubes whenever rising oil temperature indicates inadequate cooling.

CHAPTER XII

ALIGNING ROTATING MACHINES

Importance of Correct Alignment.—A rotating machine must be in accurate alignment to prevent overheated bearings and the setting up of bending stresses in the shaft. When two or more machines are connected together, the shafts must not only be correctly aligned in their own bearings, but they must be in alignment with each other. A rough alignment can be made with the units at existing atmospheric temperature, but the final check must be made at each machine's normal operating temperature to correct any inequalities caused by the expansion or contraction of individual parts. For instance, if a steam turbine drives a cold-water pump, both must be at operating temperature for a final alignment check.

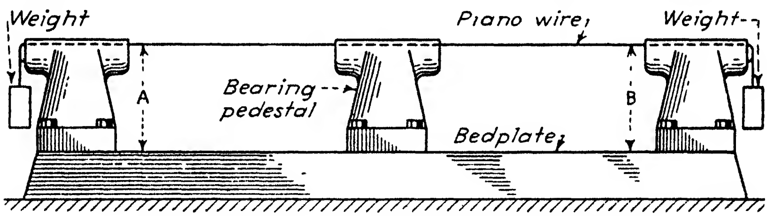


FIG. 371.—Aligning bearing bores with a steel-wire center.

Machines connected by flexible couplings should be aligned as accurately as those units with solid couplings. Alignment of the two shafts can be adjusted by checking the clearance at the coupling halves. Alignment at this point must be as nearly perfect as possible; coupling faces should be kept to less than 0.002 in. out of parallel per 12 in. of face diameter.

Three-bearing Machines.—Three-bearing machines that have a one-piece shaft are the most difficult to align. Here the only solution lies in aligning the bearing bores with a steel wire (Fig. 371). Stretch the wire between weights and determine its sag from Fig. 169, using the size wire and weights conforming to the sag data.

Locate the wire accurately in the center of each end-bearing bore with a micrometer or battery and bell tester. A simple micrometer can be made by pushing pins in the ends of a thin wood stick. Adjust the penetration of one pin until the over-all correct length is found.

Since each bearing pedestal is level with its bedplate, any difference in their heights will show up in micrometer measurements at *A* and *B*. Adjust the end bearings until they are level with each other. Then, taking into account the wire sag, adjust the center bearing to its proper height. If the shaft rotors are heavy, indicating some sag in the shaft itself, the middle bearing may be set slightly lower (0.001 to 0.002 in.) than the other two.

When the shaft contains a coupling, alignment is considerably easier if normal care is exercised. First install suitable blocking under the unsupported half of the shaft before breaking the coupling. Loosen the coupling bolts, remove one of them at a time, and replace it with a slightly smaller one. These temporary bolts hold the shaft from falling apart. Next separate the coupling approximately $\frac{1}{8}$ in., but not enough to separate the male and female joint. The alignment checks may then proceed.

Sagging Shafts.—Shafts that carry heavy rotors generally show a slight sag between bearings which must be taken into account when aligning the couplings. Figure 372 shows an exaggerated view of this bending and how it appears at the coupling.

Bolting the coupling together under this condition forms another bend in the shaft, as at *A*. The general practice is either to lower the two inboard bearings or to raise the outboard bearings so as to form a uniform bend in the shaft, as at *B*.

Shaft deflection in a large synchronous-motor generator, measured with a sensitive machinist's level 18 in. long, is shown in Fig. 372. Deflection at the generator outboard bearing was 0.014 in. with the exciter end high. Deflection at the generator coupling bearing was 0.007 in. with the generator end of the journal low. This deflection represents the amount that one end of the 18-in. level had to be raised to show level on the shaft. This corresponds to a deflection of 0.0046 in. per ft length of shaft. Deflections at the motor bearings were 0.007 and 0.013 in., respectively.

The coupling flanges were found separated 0.018 in. at the top and 0.0 in. at the bottom, while at each side they opened 0.009 in. They were therefore parallel with respect to each other in a horizontal plane, but were out of parallel vertically.

Liners 0.055 in. thick were added under the pedestal at both outboard bearings. This tipped the ends of both shafts upward and restored the coupling flanges parallel vertically.

Several checks were then made by rotating the motor rotor through 90-deg turns while the generator shaft remained stationary. This operation is necessary to determine whether there are any permanent bends in the shafts. It might happen

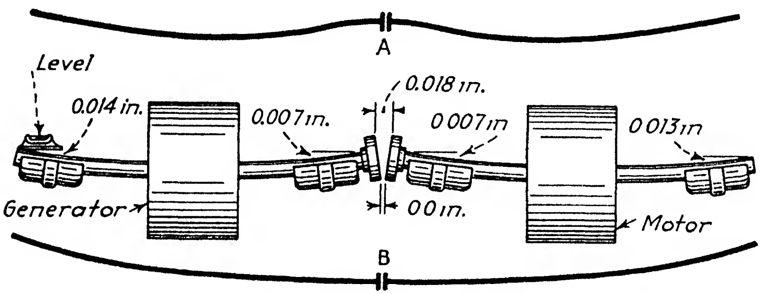


FIG. 372.—Eliminate bend at couplings by forming a uniform bend in shaft.

that the coupling flanges would be parallel even though both shafts were bent in opposite directions. For instance, if one shaft was permanently bent up and the other down, the flanges might still show parallel from top to bottom. Turning each shaft a complete revolution and checking at 90-deg positions while the other remains stationary determines whether either shaft is permanently bent.

Homemade Clamps.—Figure 373 illustrates the construction of several convenient homemade clamps to hold dial indicator 1 and test indicator 2 on shafts and couplings while checking alignment. With a scale graduated in thousandths of an inch and 15 units on each side of zero, the test indicator is sometimes easier to use than the dial unit when checking a shaft after it has been approximately aligned with a straightedge and feeler gage.

To support the test indicator on certain coupling designs, make a 2-in. C clamp from a piece of $\frac{1}{2}$-in. key steel forged to approximately the correct shape and then finished (4). A standard 2-in. C clamp furnishes the screw. To support the indicator, braze a 1-in. piece of $\frac{1}{2}$-in. key steel, with two $\frac{5}{16}$-in. holes drilled and tapped in it, to the back of the clamp.

To mount the dial indicator on shafts connected by flexible couplings, use two standard 4-in. C clamps. Cut a section out of the web on the back of each clamp and braze in a piece of $\frac{1}{2}$-in. key steel fitted with a $\frac{5}{16}$-in. tapped hole (3). For mounting the dial and test indicators, cut from a $\frac{5}{16}$-in. drill rod the following lengths: one 2-in., two 6-in., and one 8-in. At one end, thread each rod to screw into the C clamp fitting. At the other end, file a flat surface for a wrench to tighten the rod or remove it from its socket (5, 6, and 7). Block 8, made from a piece of $\frac{1}{2}$-in. key steel, serves for alignment checks with the dial indicator, or as at *A* in (26) of Fig. 375.

To mount the dial and test indicators, make two clamps (9), each from a $\frac{1}{2}$-in. bolt, a piece of $\frac{3}{8}$-in. pipe, and a piece of key steel. Chuck the $\frac{1}{2}$-in. bolt in a lathe and make a thumb nut for tightening the clamp, then make the clamping bolt.

Machining operations and parts are indicated in (10) (Fig. 374). Machine a piece of $\frac{3}{8}$-in. pipe on the inside to fit snugly over the clamping bolt, and on the outside to obtain a wall $\frac{3}{32}$ in. thick and a length $\frac{3}{32}$ in. longer than the clamping-bolt body. With the sleeve on the body of the clamping bolt and their outer ends flush, drill a $\frac{5}{16}$-in. hole through both the bolt and pipe, as in (11). To prevent the sleeve from slipping up off the bolt when not in use, peen the inner edge of its lower end, as at *A* in (13).

The construction of the other half of the clamp, made from a $1\frac{1}{8}$-in. piece of $\frac{1}{2}$-in. key steel, is clearly indicated in (12). The clamp is assembled as in (13). Two rods, one in hole *B* and the other in hole *C*, can be moved to any position with relation to each other and provide a handy tool for mounting dial and test indicators. Tightening the clamping-bolt thumb nut locks both rods in place at one operation.

Check Each Coupling Half and Its Shafts.—Different couplings will require different dial-indicator setups, or slightly different fittings from those in (14) to (17). The first procedure is to level each machine's bedplate and shaft and make the two shafts level with each other.

To check bearings for vertical clearance (14): Support the test indicator on each of the coupling halves with its contact resting on top of the bearing, preferably with the pointer at zero. Then, if the shaft is lifted to the top side of the bearing, movement of the pointer shows shaft clearance.

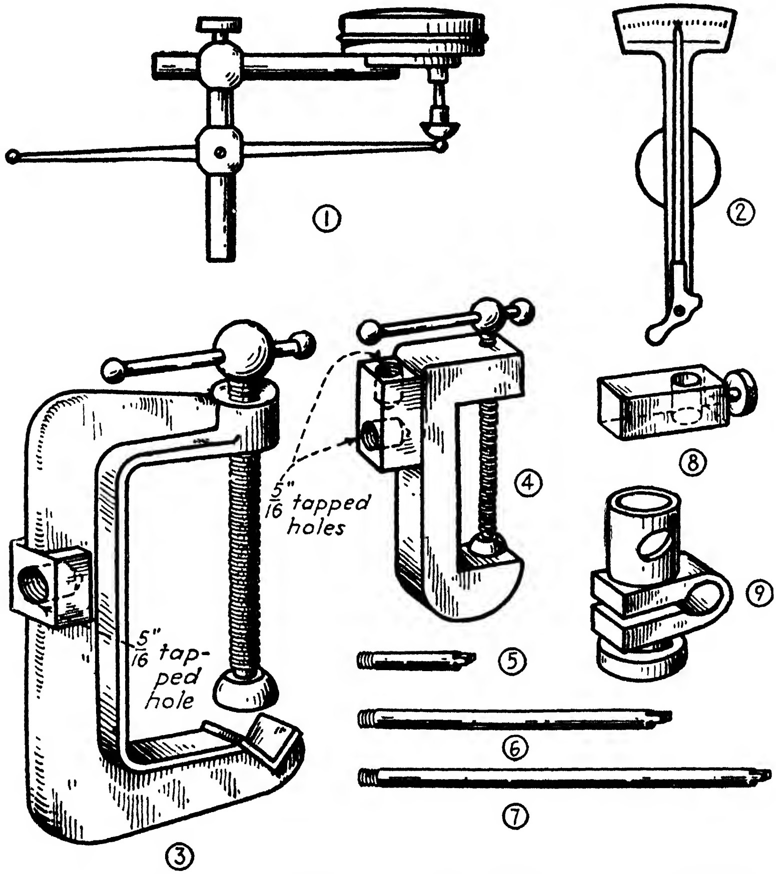


FIG. 373.—Dial and test indicator clamps and extension rods. (*Courtesy of Power.*)

To measure shaft end play (15): Move half the coupling and its shaft to the right to take out all clearance. Then mount the indicator with its pointer contact against the bearing-housing end when the pointer is on zero. When the shaft and coupling are pushed to the left as far as possible, movement of the pointer indicates shaft end play.

To test for a bent shaft or a coupling half not square on its shaft, or for both (16): Mount the test indicator on half the coupling with its contact against the vertical face of the outer half. Then turn the coupling half with the indicator contact resting on it slowly through one revolution. The indicator shows any out-of-true of this half as a result of the coupling not being square on the shaft or the shaft being bent.

To test the coupling for eccentricity (17): Mount the indicator on half the coupling with its contact resting on the circumference of the other half. If the right-hand half is turned through one revolution, any eccentricity shows on the indicator.

When a hot unit drives a cold one, or vice versa, allowance for temperature difference must be made in alignment of the coupling halves. For example, when aligning a steam turbine with a cold-water centrifugal pump, set the turbine, when it is cold, from 0.003 to 0.015 in. lower than the pump, depending upon its size and shape. Recheck at normal operating temperature.

Before using the test indicator for the final check, correct the radial alignment until all points on the circumference of the coupling halves align with a steel straightedge placed against them (18). Also correct the axial alignment until the faces of the coupling halves are parallel, as determined by feeler gages (19).

If the coupling is of the pin-and-rubber bushing type, put one worn pin with its bushing in the coupling. This keeps the coupling halves in the same relation without binding. Force the two shafts away from each other to obtain the maximum space between the coupling halves. Jam a piece of rubber, such as a section of inner tube, between the coupling flanges (23) (Fig. 375), to keep them apart without binding. Then set up and turn the indicator as a unit with the coupling through one revolution. The indicator shows the total axial misalignment between the two machines. Taking indicator readings at each 90-deg point in the revolution shows both horizontal and vertical axial misalignment.

Assume that when the coupling is turned the indicator reads as in (20). This indicates that the axes of the shafts are in alignment in the vertical plane, because the readings are equal and in the same direction on both sides of the coupling. At the bottom, the coupling halves are 0.004 in. closer together than at

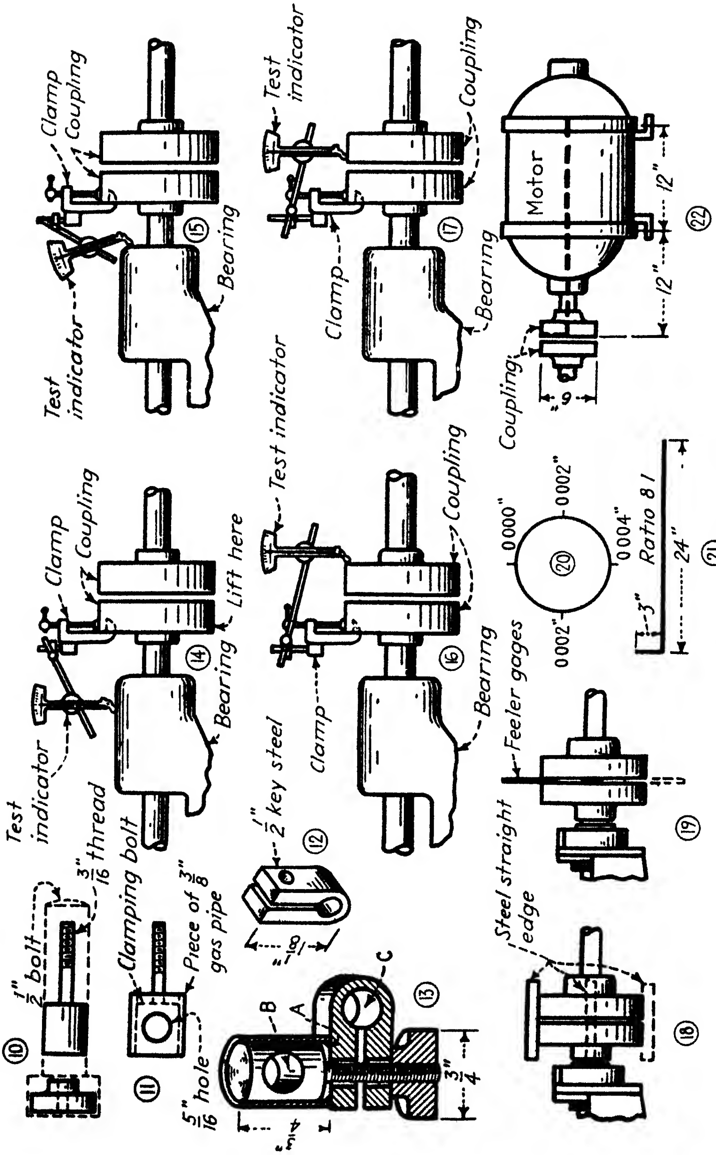


FIG. 374.—Adjustable swing support holds indicator in any position (*Courtesy of Power.*)

the top, which shows axial misalignment in the horizontal plane. In other words, the outer end of one or both shafts is low. To

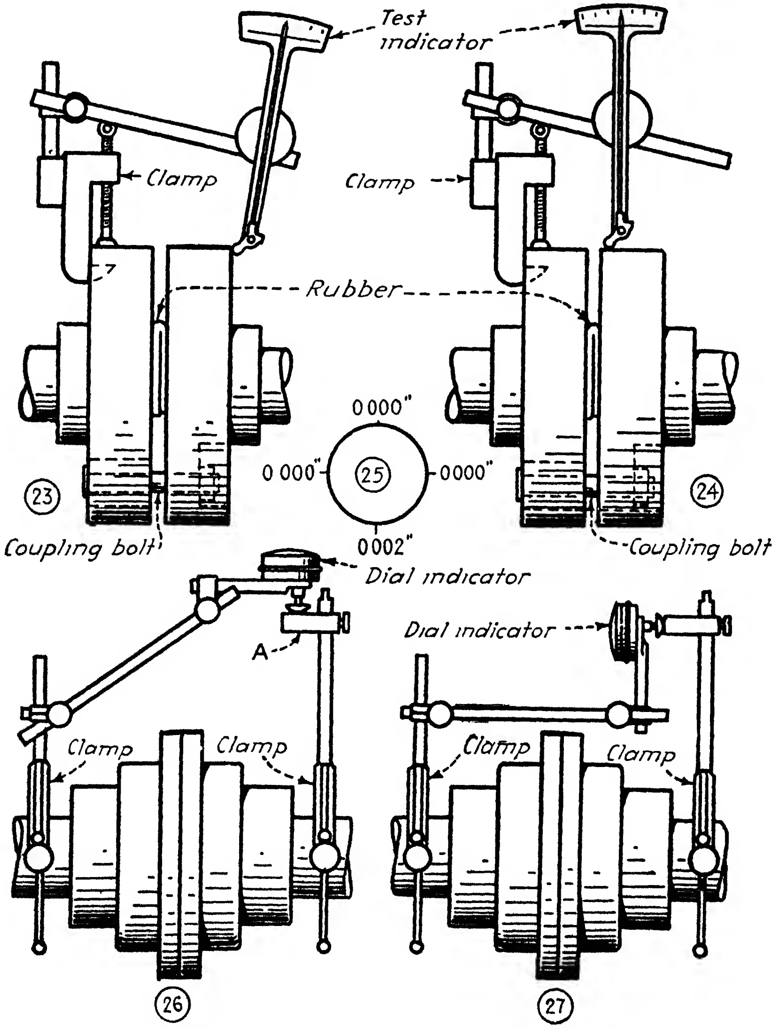


FIG. 375.—Indicator setup for checking axial and radial alignment. (*Courtesy of Power.*)

correct this misalignment, raise the outer end of one machine an easily determined amount.

If the bottom sides of the coupling halves are 0.004 in. closer than the top, this means the top sides are 0.002 in. too far apart and the bottom sides 0.002 in. too close. To bring the coupling faces parallel, raise the outer end of one machine enough to separate the bottom sides of the coupling halves 0.002 in. Assume we are aligning a motor with its machine, and that the coupling has a 3-in. radius and the motor feet are located 12 and 24 in. from the coupling face, as in (22). This gives a ratio of 24:3, or 8:1, between the coupling radius and the distance between the face of the motor coupling half, and the center of the outboard feet of the motor, as in (21). To get 0.002-in. movement of the free end of the 3-in. radius in a vertical plane, the free end of the 24-in. line has to be moved vertically $0.002 \times 8 = 0.016$ in.

In (22) the inner feet of the motor are halfway between the coupling face and the outboard feet. Then if the latter are raised 0.016 in., the coupling face will be lowered by that amount because the motor moves around its inner feet as a fulcrum. To bring the coupling halves into true axial alignment, the inner feet must be raised half the distance that the outboard ones are, or 0.008 in.

When the motor and its load are in true axial alignment, check for radial alignment by mounting the indicator, as in (24) (Fig. 375). Rotate the coupling and indicator as a unit and read the indicator at each 90-deg position. Assume that the indicator reads as in (25). This shows that the motor is 0.002 in. low. To correct this and bring the motor in true radial alignment with its machine requires placing a 0.002-in. shim under each of its feet.

If the indicator in the horizontal positions reads differently on opposite sides of the coupling, it shows radial misalignment in the horizontal plane. Correct this by moving the whole motor horizontally the amount necessary to obtain true alignment.

When not convenient to use the test indicator, mount the dial indicator as in (26) and (27). Place two clamps (3) (Fig. 373) in a horizontal position on the two shafts to support the indicator. The setup in (26) measures radial alignment the same as (24); (27) measures axial alignment, similar to (23). In these tests, the coupling halves must be free to turn with their shafts without binding, and they must be kept the maximum distance apart to obtain a true reading not affected by outside influences.

CHAPTER XIII

HYDRAULIC COMPRESSION OF AIR

Adaptation.—Compressing air hydraulically requires a falling head of water or a volume of water under pressure capable of supplying the desired quantity of air at the necessary pressure. One of the oldest hydraulic methods of compressing air, called a trompe or water bellows, led water from a higher to a lower level

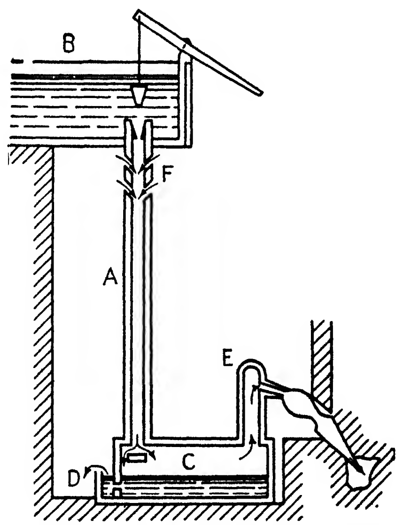


FIG. 376.—Trompe hydraulic compressor.

through a pipe or hollow bamboo pole having openings in the side through which air entered to mingle with the descending water (Fig. 376). This air, later separated from the water, served for blowing forges. Many improvements have been made on this early apparatus and several distinct types developed from it.

With this method low waterfalls, otherwise useless, are made productive, the horsepower being determined by the diameter of the down-flow pipe and the height and volume of water in the fall. Air pressure depends solely upon the depth of the well. Any required pressure can be produced isothermally up to the capacity of the water power; the delivered air is at the same temperature as the water and free of excess moisture. The initial cost compares favorably with mechanical means, and the low cost of attendance makes the method ideal for certain purposes. Investigations show that the compressed air contains less oxygen than the atmosphere, making it undesirable for some uses.

Operating Characteristics.—A study of Fig. 376 shows how falling water compresses the air. Pipe A carries water from reservoir B to chamber C. Just beyond the reservoir in pipe A, openings F project into the descending line. Air flowing in at this point mixes with the descending water and is carried down to chamber C. Air leaves through pipe E to the delivery system, and the water rises up the well D and overflows. The air pressure depends on the height of the water in well D. The distance between the level in reservoir B and well D represents the height or fall of water available to produce water flow.

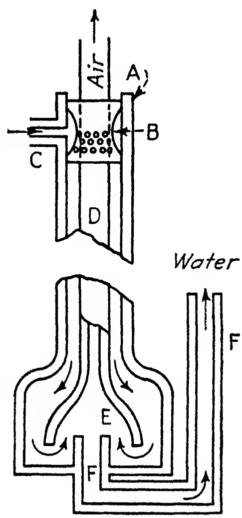


FIG. 377.—Arthur's compressor.

Since bubbles rise in water at a velocity depending on their size, air drawn into a current of water moving downward at a velocity in excess of that at which the bubbles rise will be carried down and subjected to a pressure corresponding to the depth attained. Compression takes place isothermally, a process not accomplished by many of the mechanical methods. The length or shape of the horizontal passage C must be designed to allow air to escape from the water beyond the siphon pipe.

The quantity of intake air regulates itself, being neither more nor less than the flowing quantity of water can carry. If the descending column is so loaded with air that its weight drops to that of the ascending discharge column, the water in the former will rise and less water will enter. In the opposite case the water falls and more air enters.

Typical Designs.—Another device (Fig. 377), patented by Thomas Arthur in 1888, feeds a stream of water directly into the top of a vertical pipe *A*. Inserted into the mouth of this pipe is a double cylindrical cone *B*, forming an annular air passage between it and the walls of pipe *A*. Owing to the increase in the water velocity as it passes through the narrow throat of the double cone, air is inhaled through the pipe *C* into the annular-space perforations in the cone and is entrained with the falling water.

A vertical air-delivery pipe *D*, having its lower end *E* enlarged and open at the bottom, rises through main pipe *A*. Projecting upward into this enlarged air-delivery pipe is an escape pipe *F* through which the water passes after parting with the air.

The Taylor system (Fig. 378) uses a series of small pipes placed vertically in the upper end of a falling water column. These tubes *A* terminate in the conical entrance to the down-flow pipe *B*. Water entering the cone from flume *C* carries air from pipes *A* down to chamber *D*. The air separates and enters pipe *E* and the water passes to discharge through well *F*.

Test Results.—One installation utilizing a head of 19.5 ft required 4,292 cfm of water to produce 1,148 cfm of free air at 53.3 psi. The compressor consumed 158.1 gross hp and produced 117.7 hp of effective work in compressing the air, giving an efficiency of 71 per cent. This air was then used in an engine which developed 81 hp, or a falling-water efficiency of 51.2 per cent.

A three-unit installation of Taylor compressors (Fig. 379), working under a 70-ft head, produced 35,800 cu ft of air per min at 118 psi with a water flow of 44,000 cu ft per min. The waterfall developed 5,400 hp and produced an equivalent air hp of 4,300, or an efficiency of approximately 80 per cent.

In this plant the water, diverted from a dam, runs through a canal 400 ft downstream where there is a net drop of 71 ft to the river. The compressors are located at this point. Three 5-ft cement shafts, each sunk 330 ft through rock, end in the roof of a large compression chamber. Steel tubes flared to 7 ft 4 in. at their outlet ends project from the concrete shafts 16 ft into this chamber. Directly under each discharge tube is a cone-shaped concrete spreading pier.

At the tube end, the compressing chamber is 21 ft high and 57 ft wide. These dimensions continue 50 ft toward its other end, where it reduces to 18 ft wide and 26 ft high for the remainder of the 281 ft. At this end of the chamber a 10-ft tunnel *A*, 40 ft long, leads into an inclined shaft *B* which goes to the surface, through which the water discharges.

The compression chamber, constructed in solid rock, has an air capacity between the water line and the roof of 80,264 cu ft. The distance from the forebay water level to that in the chamber

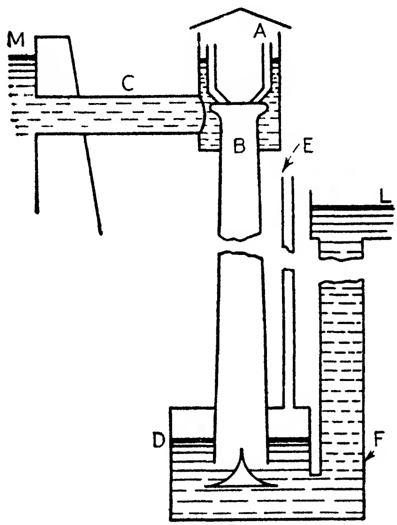


FIG. 378.—Taylor's compressor.

is 343 ft, and the water-power head from forebay to tail water is 72 ft, thus giving an air-pressure head of 271 ft.

From the upper right end of the chamber *C*, a 30-deg incline tunnel runs into the main discharge shaft *B*. This contains a 24-in. pipe that conveys air to the mine and mill and also a 12-in. relief pipe. The space not occupied by the pipes is filled with concrete. The lower end of the relief pipe maintains a water level in the compression chamber 12 ft below the roof, and its discharge end extends 5 ft above water level in the river.

At the upper end of each concrete inlet shaft is a steel tube that extends 6 ft above the bottom of the forebay. This tube, with its telescoping head that moves freely up and down in the pipe projection, is shown in detail in Fig. 379. The telescope joint permits moving the head above the water level, thus cutting off all water flowing to the compression chamber.

A header, 10 ft in diameter, designed to be under water when the system is in operation, carries eight 7-in. vertical tubes that project above the water and serve as air intakes. When water flows into the submerged head, downward flow produces a suction that draws air in through the atmospheric tubes into the water column. Air volume is controlled by adjusting the space between

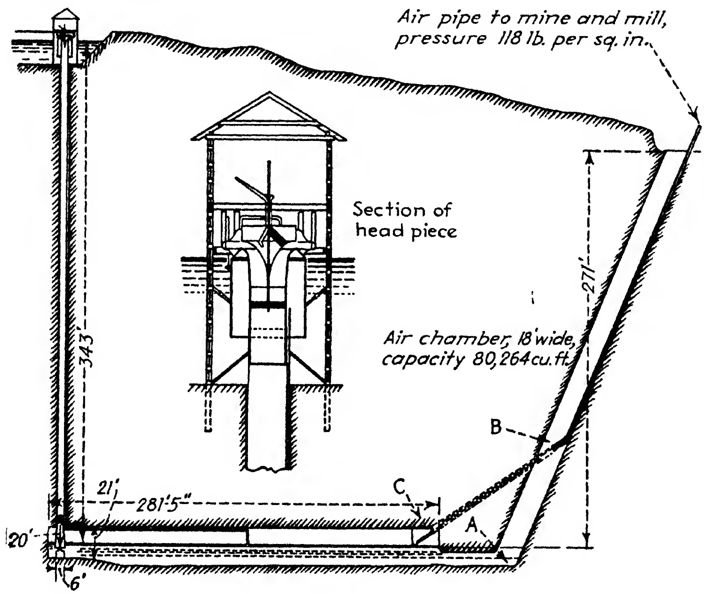


FIG. 379.—Cross section of Taylor compressor installation. (*Courtesy of Power.*)

a concave and convex casting by moving the concave piece either up or down with an adjusting screw.

When putting the system into operation, the headpiece is lowered by turning the adjusting screw and capstan nut until the lower rim of the upper casting is a few inches below water level. Water then rushes into the opening and into the shaft; air flows through the inlet pipes and descends in small bubbles with the water, being gradually compressed in its downward journey to the shaft bottom.

Striking the conical cement pieces at the bottom, water and air spray in all directions and flow along the compression chamber toward the outlet end. As the flow is slow, the air has sufficient time to separate and rise through the water to the upper portion of the chamber. Excess air pressure forces the water down until it uncovers the lower end of the 12-in. relief pipe, which serves as a safety valve. When air usage increases, water rises in the chamber until the lower end of the pipe is sealed, thus filling it with water and preventing the escape of air. This condition exists until another surplus of air forces the water down, uncovering the relief pipe and venting the excess air. The main body of discharge water flows up shaft *B* and goes to the river.

Automatic air regulation is obtained by a pipe connection from the air chamber to the inverted float of the lower conical casting in the telescope head. When sufficient pressure has accumulated in the compression chamber, air passes up through this pipe, and, discharging into the inverted float, lifts the rim of the lower conical casting and prevents further flow of water past the air inlet. The float is equipped with an adjustable valve that automatically permits air leakage. As soon as the air pressure in the compression chamber below is relieved, the conical casting falls and the flow of water past the air inlet is resumed. The cost of this installation was about $22 per hp, not including the cost of the dam and canal, which was about the same.

System Losses.—Losses inherent in hydraulic compression are (1) The head expended in impregnating the water with air. (2) A loss called slip because of the velocity with which the bubbles tend to rise. The tendency of the bubbles to rise during the water descent causes lost motion that lowers efficiency. (3) The increasing solution of air in the water with increasing pressure as the water and air descend. This air does not separate in the lower chamber, but is freed in the ascending well as the pressure decreases. This escape in the overflow well aids water movement, similar to an air lift, and partly balances the loss in the descending column.

CHAPTER XIV

THEORY OF COMPRESSING AIR

Composition of Air.—Atmospheric air is a mechanical mixture of gases, principally oxygen and nitrogen, always containing a certain percentage of water vapor. It thus consists of individual molecules of these gases, widely separated in comparison to their size, which travel at high velocity, and as a result hit against

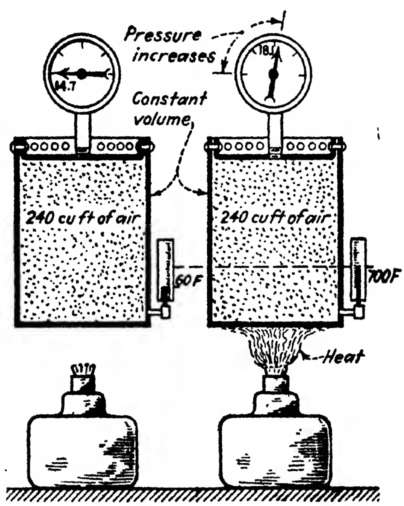


FIG. 380.—With constant volume, pressure is directly proportional to absolute temperature.

enclosing surfaces and produce pressure. Temperature is directly related to the average molecular speed. High molecular speed means high temperature; low speed, low temperature. (At absolute zero temperature the molecules would lie perfectly still on the floor of the container.)

Air at sea level exerts a pressure of 14.7 psi. This means that a column of air 1 in. square reaching from sea level to the upper atmospheric limit (about 50 miles) weighs 14.7 lb. This pressure is commonly termed *one atmosphere* and compressors are frequently rated as compressing to a certain number of atmospheres.

Air is capable of expansion and contraction, and of absorbing and giving up heat. In its behavior under these conditions, it is practically a *perfect gas*, following closely the perfect-gas laws.

Effect of Heat.—When heat is added to a fixed volume of air (Fig. 380), its temperature rises, the particles move faster, and the pressure increases [Eq. (9)]. If a piston compresses air confined in a cylinder, the moving piston delivers energy to the molecules, just as a moving bat delivers energy to a pitched baseball. The resulting higher speed of the molecules shows up as high temperature. In everyday language, "compression heats the air."

Compressing air is nothing more than forcing a given volume or weight of these molecules to occupy less total space by squeezing them closer together. With volume reduced and travel restricted (Fig. 381), the molecules hit on the sides of an enclosing vessel with greater frequency, which shows up as increased pressure.

TABLE 35.—AIR VOLUME PER POUND AT DIFFERENT TEMPERATURES AT ATMOSPHERIC PRESSURE

Temp, F	Vol, cu ft 1 lb dry air	Vol 1 lb dry air + vapor to saturate	Temp, F	Vol, cu ft 1 lb dry air	Vol 1 lb dry air + vapor to saturate
20	12.09	12.13	75	13.48	13.88
25	.....	.....	80	13.60	14.09
30	12.34	12.41	85	13.73	14.31
35	12.47	12.55	90	13.86	14.55
40	12.59	12.70	95	13.98	14.80
45	12.72	12.85	100	14.11	15.08
50	12.84	13.00	110	14.36	15.73
55	12.97	13.16	120	14.62	16.52
60	13.10	13.33	130	15.00	18.13
65	13.22	13.50	140	15.13	18.84
70	13.35	13.69	150	15.39	20.60

Courtesy of *Power*.

To sum up, the pressure of a given enclosed quantity of air can be increased by reducing its volume, by increasing its temperature, or by both. The amount of temperature rise during compression is greatly affected by the dimensions, speed, and cooling of a compressor.

✓**Kinds of Compression.**—Two common theoretical standards of comparison are *isothermal* and *adiabatic* compression. In practice, compression is never exactly isothermal or adiabatic, but in between them (Fig. 382). The power consumed in isothermal compression is equal to the area $ABCD$; therefore, actual compression (dotted curve) consumes more power because the heat generated in compression is never entirely removed.

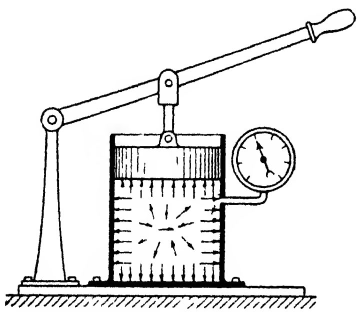


Fig. 381.—Compressing air squeezes its molecules closer together.

Cooling or loss of heat from the air after compression reduces the power stored in the air and is a direct power loss. When air is compressed by the usual methods, the intake air becomes heated slightly before compression begins because the compressor parts are hot. This heat pickup increases the air's volume and reduces the weight of air discharged by the compressor.

Good design and operation get further away from adiabatic and closer to isothermal compression, chiefly by better cooling arrangements, such as the use of more and colder cooling water, and multistage compression with intercoolers between stages. The latter is most effective, as shown in Fig. 383, where the area $BCDE$ is the saving in power effected by two-stage compression.

The curve AB represents compression in a single-stage machine and shows how pressure increases as volume reduces. The curve AE, DC represents two-stage compression. The area under each curve bounded by the angle AGH represents the work done in compressing the air, and the hatched area therefore shows the actual saving.

✓**Isothermal Compression.**—Isothermal means *constant temperature.* If perfect cooling could keep the air at its starting temperature throughout the operation, compression would be isothermal, an ideal condition requiring the least power as shown by the area $ABCD$ in the compression curve (Fig. 382). Compressing air isothermally doubles its absolute pressure each time the volume is halved (Fig. 384). Absolute pressure is inversely proportional to the volume.

Adiabatic Compression.—At the other extreme is adiabatic compression, which exists when cooling is entirely eliminated. During such compression, work delivered by the piston shows up as a steadily rising temperature. This temperature causes a premature increase in pressure, or, in other words, discharge pressure occurs sooner. The cylinders (Fig. 385) explain this more clearly.

Starting with one cu ft of air at 15 psi abs (A) and compressing to 30 psi abs (B), we secure this pressure before the volume is

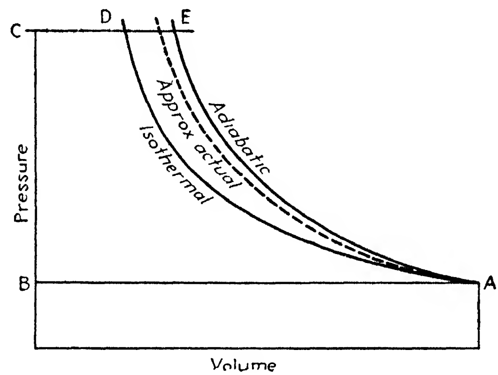


FIG. 382.—Curves of isothermal and adiabatic compression.

reduced to one half because of increased temperature. The heat of compression produces a tendency for the air to expand even while it is being compressed, which causes the early pressure. After cooling to the initial temperature C, the pressure is steady and the volume reduces to one half.

Higher temperature of the air during compression proportionally increases the pressure at each point and this, in turn, increases the power requirement as shown by the area $ABCE$ in Fig. 382. The extra power shows up in the discharge air as higher temperature, but it has no value if the air cools in the storage receiver, or during transmission.

As far as distribution and application are concerned, we need not consider whether the air was compressed isothermally or adiabatically, because the temperature usually drops or is lowered by coolers before the air is used.

Important Fundamentals.—We know that *work* is a force overcoming resistance and is measured in foot-pounds. Energy, which exists in a number of forms, is the capacity to do work and can also be measured in foot-pounds. Heat, measured in Btu, is one form of energy. Power is the rate of doing work, the unit being the horsepower, or 33,000 ft-lb per min. Temperature is an indication of the direction in which heat will flow if it has the opportunity to do so. The internal energy of air depends on its temperature.

Work done in compressing air increases the air's internal energy and raises its temperature. As the compressor cylinder

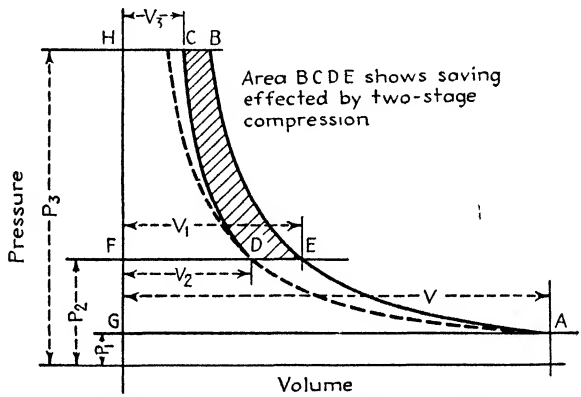


FIG. 383.—Economy of two-stage compression.

and piping are heat conductors, the whole of this heat soon dissipates to surrounding bodies and the air's internal energy gradually returns to its original value as the temperature falls to initial value.

Although 1 lb of air at 1,000 psi and atmospheric temperature has no more internal energy than 1 lb at atmospheric pressure and temperature, still the energy contained in the air under pressure is available for use because this air can expand, suffer a loss of pressure and temperature, and give up a portion of its internal energy. The greater the fall of pressure during expansion, the greater the fall in temperature and the greater the amount of internal energy available for use. The energy used in compressing air is not actually stored up in the air unless the heat of compression is retained. A portable compressor without cooling facilities furnishing air for immediate use approaches this condition. This internal energy depends on the temperature alone, and the energy that may be available for use depends on the fall of pressure and drop in temperature permissible.

TABLE 36.—RELATION OF PRESSURE UNITS

Unit	Psi	Lb per sq ft	In. of mercury	In. of water	Ft of water
Psi	1	144	2.04	27.68	2 31
Lb per sq ft	0.0069	1	0.014	0.19	0.016
In. of mercury	0.491	70.73	1	13.59	1.136
In. of water	0 036	5.20	0.074	1	0 083
Ft of water	0.433	62.42	0.882	12	1

Conditions of Air.— The principal conditions of a given weight of air are pressure, volume, and temperature; any change in one affects either one or both of the remaining two. Knowing any two, the third can be figured for a given weight of air. Here are the rules for fixed weight of air: *Remember that pressures are absolute, or gage pressure* + 14.7 *psi; and that temperatures are absolute, or* F + 460.

Boyle's Law.—If the temperature of a given quantity of gas is kept constant, absolute pressure is inversely proportional to volume; conversely, the volume will vary inversely as the absolute pressure. This applies only for isothermal changes and means that if the volume is doubled its absolute pressure will be halved. The formula is

$$\frac{P_1}{P_2} = \frac{V_2}{V_1} \qquad \text{or} \qquad P_1V_1 = P_2V_2 \tag{1}$$

where P_1 and P_2 = the initial and final absolute pressure, respectively, in any pressure unit, but both must be in the same unit.

V_1 and V_2 = the initial and final volumes, respectively, in any unit, but both must be in the same unit.

Rearranging the formula

$$P_2 = \frac{P_1V_1}{V_2} \quad \text{(final pressure absolute)} \tag{2}$$

$$V_2 = \frac{P_1V_1}{P_2} \quad \text{(final volume)} \tag{3}$$

Example.—A given weight of air occupies 3 cu ft at 30 psi. What will it pressure be if compressed to 1.5 cu ft at the same temperature?

Solution $30 + 14.7 = 44.7$ psi abs

Then $\dfrac{44.7 \times 3}{1.5} = 89.4$ psi abs, or 74.7 psi gage (Fig. 384)

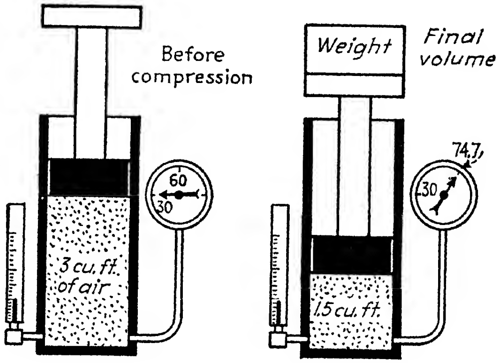


FIG. 384.—Isothermal compression doubles the pressure each time the volume is halved.

Charles's Law.—a. If the pressure on a given quantity of gas is held constant, the volume is directly proportional to the absolute temperature. The formula for this law is

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \quad \text{or} \quad \frac{T_1}{V_1} = \frac{T_2}{V_2} \tag{4}$$

where V_1 and V_2 = the initial and final volumes, respectively, in the same measuring unit.

T_1 and T_2 = the initial and final absolute temperature, respectively, expressed in the same temperature unit.

Rearranging the formula

$$V_2 = \frac{V_1 T_2}{T_1} \quad \text{(final volume)} \tag{5}$$

$$T_2 = \frac{T_1 V_2}{V_1} \quad \text{(final temperature)} \tag{6}$$

Example.—Given 1,000 cu ft of air at atmospheric pressure and 70 F, what will the volume be at the same pressure and 200 F?

Solution.—First absolute temperature is $70 + 460 = 530$ F. Second absolute temperature is $200 + 460 = 660$ F.

Then $\dfrac{1{,}000 \times 660}{530} = 1{,}245$ cu ft final volume (Fig. 386)

Charles's Law.—b. If the volume of a given quantity of gas is held constant, the pressure is directly proportional to the absolute temperature. The formula for this law is

$$\frac{P_1}{P_2} = \frac{T_1}{T_2} \quad \text{or} \quad \frac{T_1}{P_1} = \frac{T_2}{P_2} \tag{7}$$

Rearranging the formula

$$P_2 = \frac{P_1T_2}{T_1} \quad \text{(final pressure)} \tag{8}$$

$$T_2 = \frac{T_1P_2}{P_1} \quad \text{(final temperature)} \tag{9}$$

Example.—Start with 240 cu ft of air at atmospheric pressure and 60 F (Fig. 380). What will the pressure be if the final temperature is 700 F?

Solution.—First absolute temperature is 460 + 60 = 520 F. Final absolute temperature is 460 + 700 = 1,160 F.

Then $$\frac{14.7 \times 1{,}160}{520} = 32.8 \text{ psi abs, or } 18.1 \text{ psi gage}$$

Combining Boyle's and Charles's Laws.—The formula that combines these laws and applies for a given *weight* of any gas is

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \quad \text{(units same as other formulas)} \tag{10}$$

Rearranging the formulas

$$P_2 = \frac{P_1V_1T_2}{V_2T_1} \quad \text{(final pressure abs)} \tag{11}$$

$$V_2 = \frac{P_1V_1T_2}{P_2T_1} \quad \text{(final volume)} \tag{12}$$

$$T_2 = \frac{P_2V_2T_1}{P_1V_1} \quad \text{(final temperature)} \tag{13}$$

Example.—A compressor (Fig. 387) that has a clearance volume of 20 cu in. and a displacement of 100 cu in., draws in a cylinder full of air at atmospheric pressure and 60 F. The air completely fills the cylinder and clearance and is then compressed into the clearance volume. The final pressure as registered on a gage is 196.3 psi. What is the final temperature?

Solution.—T_2 = the unknown. $T_1 = 60 + 460 = 520$ F abs.

$P_1 = 14.7$ psi abs.

$P_2 = 196.3 + 14.7 = 211$ psi abs.

V_1 = piston displacement (100 cu in.)

+ clearance volume (20 cu in.) = 120 cu in.

V_2 = clearance volume, 20 cu in. Now, using Eq. (13)

$$\frac{211 \times 20 \times 520}{14\,7 \times 120} = 1{,}244 \text{ F abs} \qquad \text{or} \qquad 1{,}244 - 460 = 784 \text{ F}$$

Air Constant.—According to Boyle's law, if P represents the pressure and V the volume of 1 lb of air at any temperature, the product of PV = a constant. According to Charles's law, if T represents an absolute temperature, the volume V varies directly

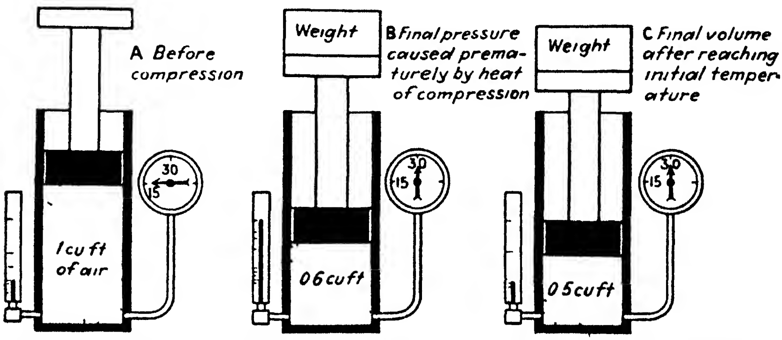


FIG. 385.—Premature pressure rise caused by temperature rise. (*Courtesy of Power.*)

as T varies; also the pressure P varies directly as T varies. From this we can state $\frac{PV}{T}$ = a constant represented by R.

Then $$\frac{PV}{T} = R \qquad \text{(a constant)} \qquad (14)$$

transposed $$PV = RT.$$

If the value for R and any two of the quantities P, V, or T be known, the third is easily found, as

$$P = \frac{RT}{V} \qquad V = \frac{RT}{P} \qquad T = \frac{PV}{R}$$

where P = lb per sq ft.
V = cu ft.

As the volume of 1 lb of air at 32 F and 14.7 psi is 12.39 cu ft (Table 35) we find R as follows:

$$R = \frac{14.7 \times 144 \times 12.39}{492} = 53.3$$

Then Eq. (14) becomes

$$\frac{PV}{T} = 53.3, \tag{15}$$

transposing

$$P = \frac{53.3T}{V} \qquad V = \frac{53.3T}{P} \qquad T = \frac{PV}{53.3}$$

where P = lb per sq ft.
V = cu ft.
T = temp abs.

Specific Heat of a Substance.—The specific heat is the amount of heat required to increase the temperature of 1 lb of a substance

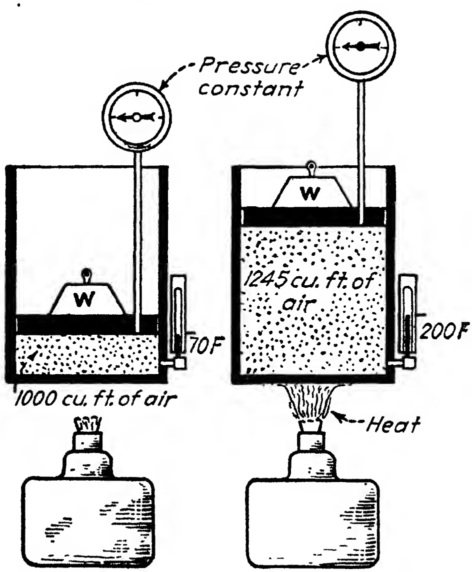


FIG. 386.—At constant pressure, volume varies directly with absolute temperature. (*Courtesy of Power.*)

by 1 F. When dealing with air, the specific heat at constant pressure C_p and constant volume C_v is frequently used in compression calculations.

Specific Heat at Constant Volume.—By Charles's law (*b*) the pressure of a given weight of gas, when heated at constant volume, varies directly as its absolute temperature. When a gas does not expand, its volume remains the same and no external work is done. Therefore, all the heat added to a gas at constant volume is effective only in raising its temperature. *The specific heat of air at constant volume C_v is* 0.1689 *Btu per lb*, or the amount of heat required to raise the temperature 1 F (Fig. 388).

Specific Heat at Constant Pressure.—A gas requires more heat to raise its temperature 1 F at constant pressure than at constant volume, because at constant pressure the gas can expand and do useful work. The heat added must be sufficient to raise the gas temperature as well as to supply energy equal to the external work done. *The specific heat of air at constant pressure C_p is* 0.2375 *Btu per lb.*

Heat Expended in External Work.—If 0.1689 Btu raises the gas temperature 1 F at constant volume and if 0.2375 Btu raises the temperature 1 F and furnishes energy for external work done at constant pressure, then $0.2375 - 0.1689 = 0.0686$ Btu, the heat expended in the external useful work.

Since 1 Btu = 778 ft-lb, then $778 \times 0.0686 = 53.3$ ft-lb of mechanical work done by the expansion of 1 lb of air at constant pressure when its temperature is increased 1 F.

Air expands $\frac{1}{492}$ of its volume for 1 F rise of temperature from 32 F; on cooling, it contracts the same amount up to liquefaction. One lb of air at 32 F and 14.7 psi occupies 12.39 cu ft. If heated to 33 F, the pressure remaining constant, according to Charles's law, $\frac{12.39}{492} = 0.0252$ cu ft volume increase.

TABLE 37.—EFFECT OF INTAKE TEMPERATURE ON CAPACITY

Intake volume required to produce 1,000 cu ft of free air at 70 F

Temp of intake, F	Relative intake volume required, cu ft	% hp saved
30	925	7.5
40	943	5.7
50	962	3.8
60	981	1.9
70	1,000	0.0
80	1,019	−1.9
90	1,038	−3.8
100	1,057	−5.7
110	1,076	−7.6
120	1,095	−9.5

Courtesy of Ingersoll-Rand Co.

Suppose this heating and expansion takes place in a cylinder (Fig. 389) fitted with a weightless, frictionless piston having a face area of 1 sq ft. Then the cylinder length will equal the air volume and the volume increase can be measured along the cylinder axis. Atmospheric pressure 14.7 × 144 sq in. = 2,116.8 lb total load on the piston moving through 0.0252 ft. Then,

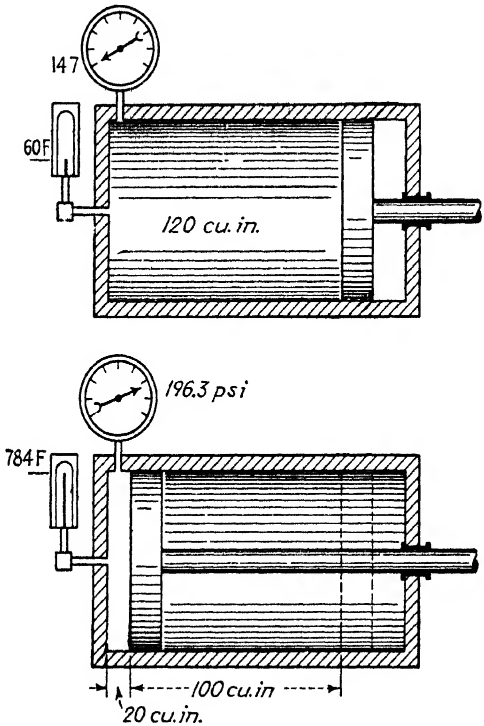


Fig. 387.—Compressing air adiabatically to 6 per cent of its original volume raises its temperature to 784 F.

2,116.8 × 0.0252 = 53.3 ft-lb of work done, which is the value of factor R shown in Eq. (15).

Compression Constant.—Because of the external work done, C_p is greater than C_v; the ratio of these two specific heats, $\frac{C_p}{C_v} = 1.406$, a constant used to represent true adiabatic compression of *dry* air. Because moisture in air changes the specific-heat value slightly, this constant reduces to 1.3947 when compressing air of 36 per cent relative humidity at 68 F.

General Equation.—To make Eq. (15) general and applicable to any weight W of air the equation becomes

$$PV = W53.3T \tag{16}$$

transposing

$$P = \frac{W53.3T}{V} \qquad V = \frac{W53.3T}{P} \qquad T = \frac{PV}{53.3W}$$

$$53.3 = \frac{PV}{WT} \qquad W = \frac{PV}{53.3T}$$

where P = absolute pressure, lb per sq ft.
V = volume, cu ft.
T = absolute temperature, F.
W = lb of air.

Example.—A 100 cu ft capacity tank (Fig. 390) holds air at 150 psi with a temperature of 85 F. What weight of air is in the tank?
Solution

$$\frac{(150 + 14.7)144 \times 100}{53\ 3(85 + 460)} = 81.5 \text{ lb}$$

Example.—If a compressor then forces additional air into the tank mentioned above until its pressure reads 200 psi, how much air did the compressor supply? The air temperature remains the same.
Solution

$$\frac{(200 + 14.7)144 \times 100}{53.3(85 + 460)} = 106.4 \text{ lb}$$

Then 106.4 − 81.5 = 24.9 lb. Suppose the intake air is 32 F with a volume of 12.39 cu ft per lb, giving 24.9 × 12.39 = 308.5 cu ft. If it required 10 min for the compressor to supply this quantity it was compressing

$$\frac{308.5}{10} = 30.8 \text{ cfm}$$

Example.—What volume will 1 lb of air at 150 psi and 150 F occupy?
Solution

$$\frac{53.3(150 + 460)}{(150 + 14.7)144} = 1.37 \text{ cu ft}$$

When using Eq. (16), pressure in psi must be multiplied by 144 (Table 36) to convert to lb per sq ft.

Since, in Eq. (16), P = psi × 144, the equation can be written psi $144V = W53.3T$. Dividing both sides by 144 gives

$$\text{psi } V = W0.37T \quad \text{(pressure in psi)} \tag{17}$$

Example.—What is the volume of 300 lb of air at 100 F and 200 psi gage?
Solution

$$\frac{0.37 \times 300(100 + 460)}{(200 + 14.7)} = 289 \text{ cu ft}$$

Engineers' Practical Formula.—The general gas law (perfect gas) expressed by Eqs. (16) and (17) recognizes in one equation

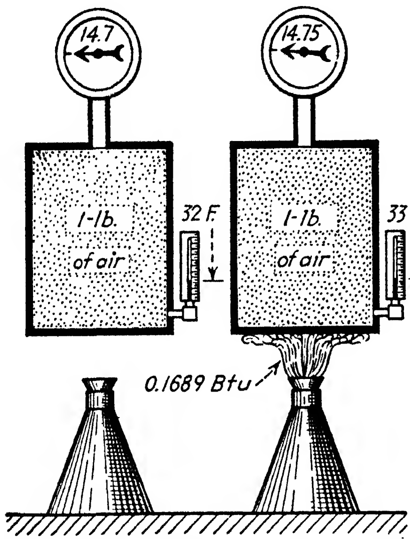


FIG. 388.—Amount of heat required to raise the temperature of a constant volume of air 1 F.

every property of a given gas that may change. This important equation may be used to solve the many problems found in compressor operation that involve steady state conditions of the pressure, volume, and temperature where PV = a constant.

Equations involving changing conditions that occur during the actual expansion or compression of air, where the pressure, volume, and temperature vary during the process, require the use of an exponent PV^n = a constant. The exponent n is the ratio of specific heats $\frac{C_p}{C_v} = n = \frac{0.2375}{0.1689} = 1.406$ for dry air. For normal air in a temperate climate the equation for adiabatic compression is $PV^{1.3947}$ = a constant. Figure 385 explains why the PV = constant (isothermal) relationship does not hold true during adiabatic compression when the gradually increasing pressure, coming from reduced volume, increases prematurely because retained heat is affecting the pressure.

The pressure-volume-temperature relation for a perfect gas during adiabatic change is

$$P_1V_1^n = P_2V_2^n \tag{18}$$

transposing,

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{1.39} \qquad V_2 = V_1\left(\frac{P_1}{P_2}\right)^{0.718}$$

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{0.283} \qquad T_2 = T_1\left(\frac{V_1}{V_2}\right)^{0.397}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{3.52} \qquad V_2 = V_1\left(\frac{T_1}{T_2}\right)^{2.52}$$

where P and V = the absolute pressure and volume, respectively, of the gas at any point along the path, in any units.

P_1 and P_2 = the absolute pressure at a first and second instant, respectively, during the condition change, in any units.

V_1 and V_2 = the volumes of the gas at the same first and second instants respectively, during the condition change, in any units.

T_1 and T_2 = respectively, the absolute temperature of the gas in any units.

Example.—A compressor has a cylinder volume (including clearance) of 100 cu in. and a 10-in. stroke. With intake at atmospheric pressure of 14.7 psi, lay out the adiabatic-compression curve for one stroke of the piston.

Solution.—First lay off the cylinder volume (Fig. 391) along zero pressure line *ab*, dividing the length into 10 equal parts. Erect the perpendicular pressure line *ca* representing the extreme end of cylinder. Then, using Eq. (18)

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^{1.39}$$

where P_1 = 14.7 psi abs
V_1 = 100 cu in.
P_2 = unknown.

Find the resulting pressure at the first point along the stroke (90 cu in. volume point)

$$P_2 = P_1(\tfrac{100}{90})^{1.39}$$

breaking the equation down, 100 ÷ 90 = 1.11. From logarithm tables the log of 1.11 = 0.0453, and 0.0453 × 1.39 = 0.06296. Now find the number of which this is the logarithm, from tables this is 1 156, hence $(\frac{100}{90})^{1\ 39} = 1\ 156$, and $P_2 = 14\ 7 \times 1\ 156 = 17\ 0$ psi abs

Find the other points with the same equation, which gives 80 cu in point = 20 psi, 70 cu in point = 24 1 psi, 60 cu in point = 30 3 psi, 50 cu in point = 38 7 psi, 40 cu in point = 52 5 psi, 30 cu in point = 78 2 psi, 20 cu in. point = 138 0 psi, 10 cu in point = 360 psi The exponentia

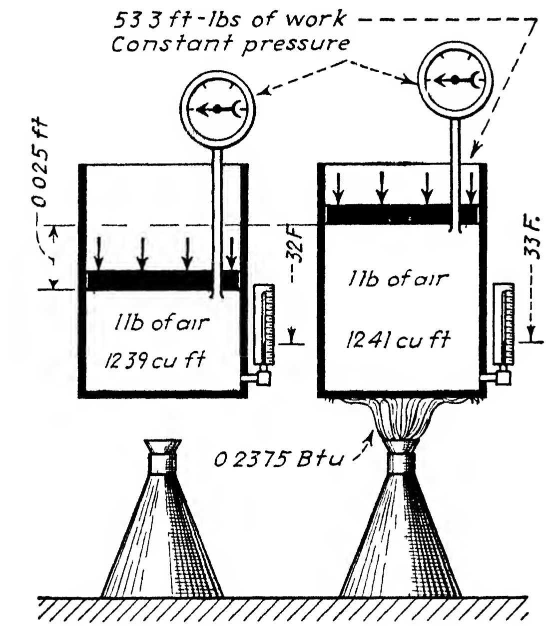


FIG 389.—Amount of heat required to raise the temperature of air at constant pressure 1 F

curves (Fig 392) may be used instead of logarithm tables Find the quotient of the factors inside parantheses, then from the base-line number follow up to the fractional exponent curve and read the result on the left scale Then multiply P_1 by the result.

Effect of Intake Temperature.—A compressor of a given size will handle its given rating in cfm regardless of the temperature of the intake air. With a higher starting temperature the air under compression reaches its discharge pressure earlier in the stroke so that the compressor discharges more than rated capacity in cfm. This is actually possible with compressors not equipped with aftercoolers when the air is used in tools immediately after compression.

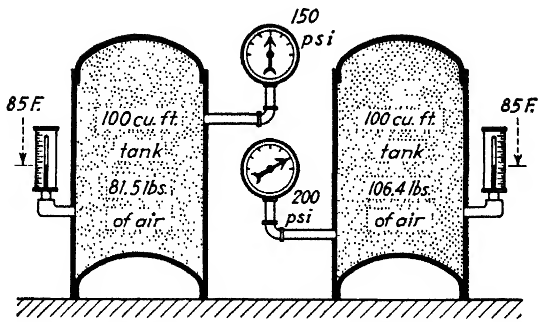


FIG. 390.—Compressor capacity can be determined by compressing air into a tank of known volume.

TABLE 38.—RELATIVE VOLUME OF FREE AIR AT 14.7 PSI FOR AIR AT VARIOUS PRESSURES

Gage pressure, psi	Cu ft of free air per cu ft of compressed air (ratio of compression)	Cu ft of compressed air per cu ft of free air
10	1.68	0.595
20	2.36	0.424
30	3.04	0.329
40	3.72	0.269
50	4.40	0.227
60	5.08	0.197
70	5.76	0.173
80	6.44	0.155
90	7.12	0.140
100	7.80	0.128
110	8.48	0.118
120	9.16	0.109
130	9.84	0.102
140	10.52	0.095
150	11.20	0.089
160	11.88	0.084
170	12.55	0.080
180	13.23	0.075
190	13.93	0.071
200	14.60	0.068

As the majority of compressor installations include an aftercooler, high-temperature intake air causes a direct power loss, as shown in Table 37. Since the air pressure rises prematurely, (Fig. 385*B*) the piston must act against it longer, consuming additional power. The discharge air then passing through the aftercooler loses its heat and false pressure to become a smaller volume (Fig. 385*C*) of air at the distribution pressure.

Example.—Assume a compressor with an aftercooler and a 200 cu ft air receiver to furnish air at 100 psi. The aftercooler is capable of cooling the discharge air to 70 F if reasonably cool intake air is used. The intake comes from a heated engine room 40 F above the outside temperature and this additional temperature carries through the aftercooler, making the receiver air 110 F. If the air must stand in the receiver until it cools to 70 F, what is the loss in pressure?

Solution.—From Eq. (8)

$$P_2 = \frac{P_1T_2}{T_1}$$

or $$\frac{(100 + 14.7)(460 + 70)}{460 + 110} = \frac{114.7 \times 530}{570}$$

$$= 92.3 \text{ psi gage, or } 7.7 \text{ psi loss of pressure}$$

Example.—Suppose the receiver of the compressor in the previous example was fitted with a weighted piston that would maintain full pressure while the air cooled. What would the loss in volume be?

Solution.—From Eq. (5)

$$V_2 = \frac{V_1T_2}{T_1}$$

or $$\frac{200(70 + 460)}{110 + 460} = \frac{200 \times 530}{570} = 186 \text{ cu ft}$$

then $$200 - 186 = 14 \text{ cu ft of air at 100 psi gage.}$$

With a compression ratio of 7.8, $14 \times 7.8 = 109$ cu ft of free air that the compressor must supply.

Ratio of Compression.—The absolute ratio of compression is the ratio of final pressure to initial pressure and is expressed as $\frac{P_2}{P_1}$. Both pressures must be stated in absolute values.

Example.—An air compressor takes in atmospheric air at 14.7 psi and compresses it to 100 psi gage pressure. What is the compression ratio?

Solution

$$100 \text{ psi} + 14.7 = 114.7 \text{ psi abs.}$$

Then $114.7 \div 14.7$ (initial pressure) $= 7.8$, the absolute ratio of compression.

When ratio of compression is known, the output of a compressor at discharge pressure can be computed. Assume a compressor with a free-air capacity of 500 cfm and a compression ratio of 7.8. Since the pressure is increased 7.8 times, the volume must have been reduced to 1/7.8, or 0.128. Then 500 cfm of free air becomes 500 × 0.128 = 64 cfm discharge at 100 psi gage pressure. This shows that every cubic foot of intake air is compressed to 0.128 cu ft when it reaches the air receiver after being cooled to its initial temperature in the aftercooler. Table 38 gives compression ratios, based on 14.7 psi atmospheric pressure, and the volume of the air compressed to various gage pressures.

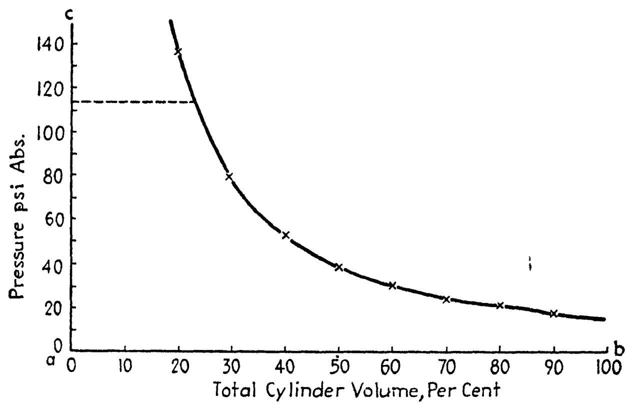


FIG. 391.—Adiabatic compression curve constructed with Eq. (18).

The curves (Fig. 393) give compressed-air volume referred to atmospheric air at 14.7 psi abs and 60 F as 1.00. Curve *A* shows how a given volume decreases by compression when constant temperature is maintained, and *B* shows how volume increases with temperature increase at constant pressure.

Example.—Suppose 200 cfm (measured at 60 F and 14.7 psi abs) is compressed to 90 psi gage and its temperature is 260 F. What is the final volume?

Solution.—1.00 volume factor = 200 cfm. Volume factor at 90 psi = 0.14. Volume factor at 260 *F* = 1.38.

Then 200 × 0.14 × 1.38 = 38.75 cfm, the volume at 90 psi and 260 F.

When the compression ratio in any one stage exceeds four or five, air-temperature rise becomes excessive, resulting in appreciable power loss. Dividing compression into two or more stages, with intercooling between, eliminates part of this loss. Assume a final pressure of 100 psi abs reached in one stage of compression;

the discharge temperature would be approximately 430 F. The same total compression ratio, carried out in two stages, results in a maximum temperature in each stage of 250 F if intercooling lowers the temperature of the first-stage discharge to its initial value. Dividing compression into stages reduces power input only if cooling takes place between stages. A closer approach to ideal isothermal compression by this temperature reduction is the real reason for staging. Thus, only the combination of multistaging and intercooling gives increased economy in compressing air. This makes an appreciable saving on compressors installed at high altitudes.

Density of a Gas.—The density of a substance varies with its temperature or with any other property that affects its volume. With gas the density depends on pressure and temperature (Fig. 394). The general gas-law formula for density is

$$D = \frac{P}{RT} \qquad \text{(lb per cu ft)} \qquad (19)$$

Where D = density, lb per cu ft.
P = absolute pressure, lb per sq ft.
T = absolute temperature, F.
R = gas constant (for air, 53.3).

Example.—If the density of air is 0.18 lb per cu ft when under a given pressure at 60 F, what will its density be when under the same pressure but at 1,200 F?

Solution.—By Eq. (19)

$$D_2 = \frac{D_1T_1}{T_2}$$

or

$$\frac{0.18(60 + 460)}{1{,}200 + 460} = \frac{0.18 \times 520}{1{,}660} = 0.0563 \text{ lb per cu ft.}$$

Air Has Weight.—Air at 67 F occupies about 13.25 cu ft per lb at sea level. It follows that if we climb to elevations above sea level, atmospheric pressure will become less, decreasing at the rate of about 0.5 psi for every 1,000-ft rise. The scale (Fig. 395) gives atmospheric pressure at any elevation from 5,000 ft below sea level to 25,000 ft above.

Since the atmospheric pressure decreases at altitudes above sea level, it follows that a cubic foot of air weighs less at higher elevations than at sea level. The curves in Fig. 396 show the difference in weight, or in the actual quantity of air handled at the different elevations. Thus, to take the extremes, a cu ft of air at sea level and 60 F weighs 0.0764 lb, while a cu ft at an elevation of 15,000 ft and the same temperature weighs 0.0431 lb, or only 56 per cent of the former.

In compressing this air to 100 psi gage the compression ratio at sea level would be

$$\frac{100 + 14.7}{14.7} = 7.8$$

while at 15,000 ft it would be

$$\frac{100 + 8.29}{8.29} = 13.06$$

Each cubic foot would have to be reduced in the first case to $\frac{1}{8}$ or 0.128 cu ft, and in the latter to $\frac{1}{13}$ or 0.076 cu ft, assuming the same discharge temperature in both cases.

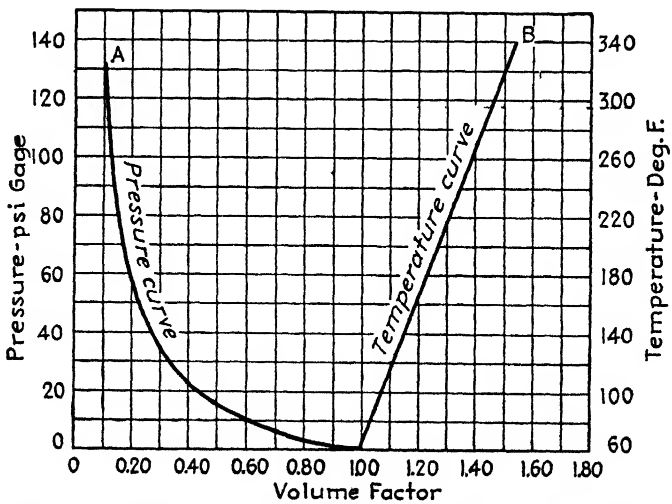


FIG. 393.—Compressed-air volume referred to changing pressure and temperature. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

Effect of Intake Pressure.—When discussing compressor discharge capacity, initial as well as discharge pressure is important. The problem just discussed shows how low intake pressure can reduce actual output. The adiabatic curves *B* and *C* (Fig. 397) show this effect quite clearly. Curve *B* starts at (*X*) with a cylinder full of air at 10 psi abs. The piston must move to the

74 per cent volume line (arrow) before the entrapped air pressure rises to the normal atmospheric pressure of 14.7 psi.

Conditions similar to this occur on compressors located at sea level when the intake passages restrict air flow so that the cylinder is filled with air which is considerably below atmospheric pressure when the piston is ready for its compression stroke.

As atmospheric density decreases at high altitudes, a compressor takes in a smaller weight of air at each stroke. Because of lower atmospheric pressure, part of the compressing stroke is occupied in compressing air of this density up to atmospheric pressure at sea level. As temperature increases with the ratio of compression, more heat results at high altitudes than at sea level when compressing to a given pressure. This high temperature temporarily increases the air pressure while under compression, and more power is required to compress and deliver a given quantity of air.

A study of the curves (Fig. 398) will show the difference in the final temperature of air compressed from different intake pressures to the same final pressure. Air at 50 F and 14.7 psi abs compressed to 114.7 psi abs has a final temperature of 455 F. See what happens with a lower intake pressure. Air at 50 F and 10 psi abs compressed to 110 psi has a final temperature of 540 F (compression to the same gage pressure).

The amount of work done can be determined from the left (enthalpy) scale. The inlet air in both cases contained 122 Btu. In the first instance the air was compressed to 114.7 psi abs, showing a final heat content of 214 Btu. The difference $214 - 122 = 92$ Btu added to the air. Suppose we were compressing 100 lb per hr. Then the work of compression is $92 \times 100 = 9200$ Btu per hr. The theoretical horsepower required is $9{,}200 \div 2{,}545$ (equivalent of 1 hp) $= 3.6$ hp. In the same manner we find the work of compression in the second instance to be 4.7 hp, or a difference of $4.7 - 3.6 = 1.1$ hp greater.

The volumetric efficiency is also less at high altitudes because the clearance air expands to the lower atmospheric pressures and, consequently, when expanded occupies a larger volume of the cylinder.

Dry Air.—The foregoing rules and formulas for dry air are, for many purposes, close enough for normal air that contains moisture. However, moisture vapor occupies volume in the compressor cylinder, has work done upon it by the piston, and then condenses out in the aftercooler. This condensing out indicates an apparent lack of air-output capacity in the compressor.

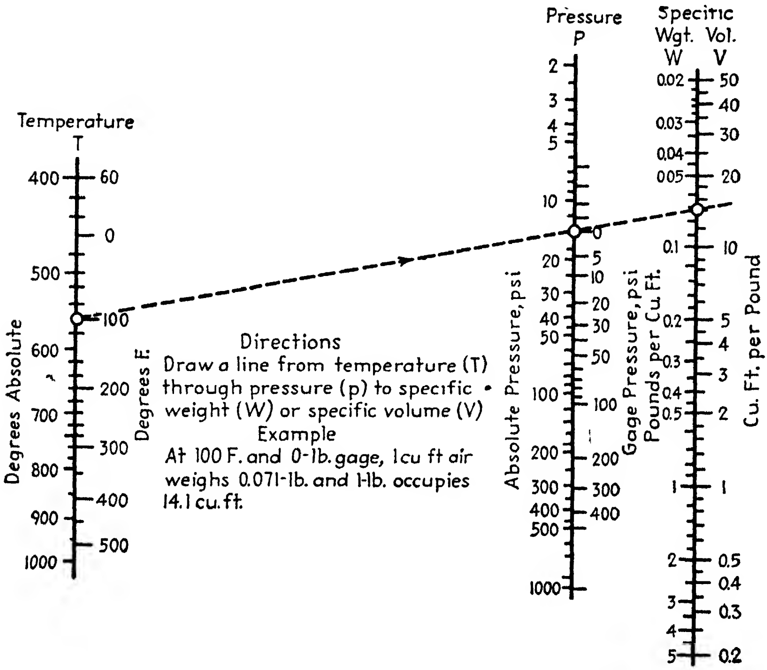


FIG. 394.—Specific weight and volume of air at different pressures and temperatures. (*Courtesy of Power.*)

Example.—One lb of atmospheric air at 70 F and 70 per cent relative humidity occupies a volume of 13.59 cu ft, 0.24 cu ft of this volume being moisture vapor. What is the percentage of moisture vapor present?

Solution

$$0.24 \div 13.59 = 0.017,$$

or 1.7 per cent moisture vapor in every cubic foot of air taken in by a compressor.

Air and Moisture.—The mixture we call the atmosphere consists mainly of oxygen, nitrogen, and water vapor, other gases being present in small quantities. Since an air compressor cannot separate moisture from its intake air, let us see how this water vapor affects the delivered product. To avoid confusion, remember that air or atmosphere means the mixture occupying space over the earth's surface.

TABLE 39.—PROPERTIES OF SATURATED STEAM

Temp, F	Pressure, psi abs	Volume, cu ft per lb	Density, lb per cu ft
32	0.0885	3,306	0.000302
35	0.0999	2,947	0.000339
40	0.1217	2,444	0.000409
45	0.1475	2,036	0.000491
50	0.1781	1,703	0.000587
55	0.2141	1,430	0.000700
60	0.2563	1,206	0.000828
65	0.3056	1,021	0.000980
70	0.3631	868	0.001150
75	0.4298	740	0.001350
80	0.5069	633	0.001573
85	0.5959	543	0.001840
90	0.6982	468	0.002134
95	0.8153	404	0.002471
100	0.9492	350	0.002858

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How Moisture Gets in the Air.—Evaporation is occurring about us continually. Water left in an open vessel or that standing in open reservoirs gradually disappears into the atmosphere. This occurs at all temperatures, provided the space in contact with the liquid does not contain a saturated vapor of the liquid, because molecules of all substances at temperatures above absolute zero are in continuous motion. In a liquid, the molecular motion is rapid and occurs throughout the entire body, and the molecules collide continually with each other. Because of these impacts, some molecules attain speeds much greater than the average velocity of those in the mass. If these higher speed molecules happen to be near the liquid surface, they project out into space and exert a part of the observed pressure on the earth's surface.

If an open dish of water is placed in a sealed glass jar (Fig. 399), only a portion of the water will evaporate, the balance remaining in the dish for an indefinite length of time. The reason for this behavior dissimilar to that in open air is the difference between the vapor pressure inside the sealed container and the pressure exerted by vapor in free air.

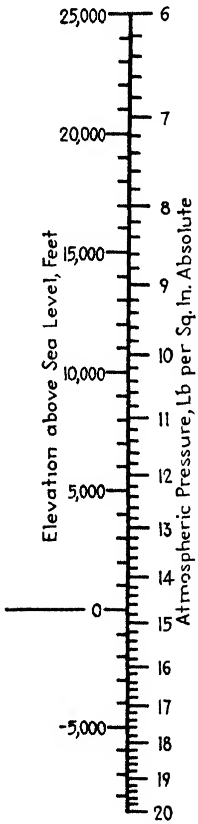

FIG. 395.—Relation of atmospheric pressure to altitude. (*Courtesy of Power.*)

When a vapor is saturated, there is an equilibrium between the vapor pressure and the pressure exerted by the liquid, *i.e.*, as many molecules return to the water as leave it. A vapor confined in a closed vessel may and will become saturated, provided that a portion of the liquid remains in the vessel. But a vapor which is unconfined or free in the open atmosphere cannot become saturated because the vapor molecules diffuse, seep away through space, and are wafted away by air currents.

The term vapor pressure means the pressure exerted only by the steam or vapor having the same temperature as the water. The vapor pressure of water is the controlling factor in the evaporation or condensation of moisture; surrounding *air pressure* does not aid or prevent it. For this reason we can evaporate water into the atmosphere even though the temperature never approaches 212 F (the boiling point of water at atmospheric pressure at sea level).

Take two barometer tubes *A* and *B* (Fig. 400) and with a bent tube inject a little water into the lower end of *B* under the mercury, taking care not to admit air. The water rises through the mercury, collects on the top *T*, and some of it vaporizes, until the top of the tube is filled with water and saturated water vapor. The column of water and mercury in tube *B* now stands at a lower level than the column in tube *A*, showing that the water vapor exerts a pressure that pushes the mercury column down. If the tube containing water is now heated, the column falls still more, as at *C*, showing that increased temperature evaporates more water and increases the vapor pressure.

Behavior of Mixtures.—Air containing moisture behaves in accordance with Dalton's law on the mixture of gases, *i.e.*, each gas or vapor exerts its own pressure, and the total pressure of the

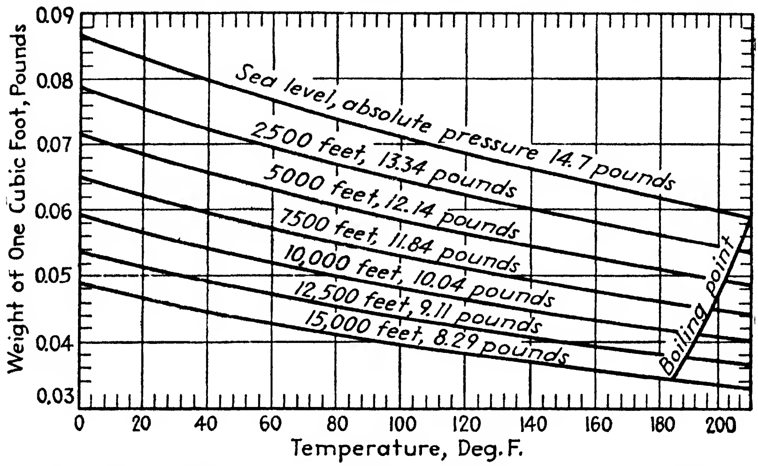

FIG. 396.—Weight of air at various elevations and temperatures.

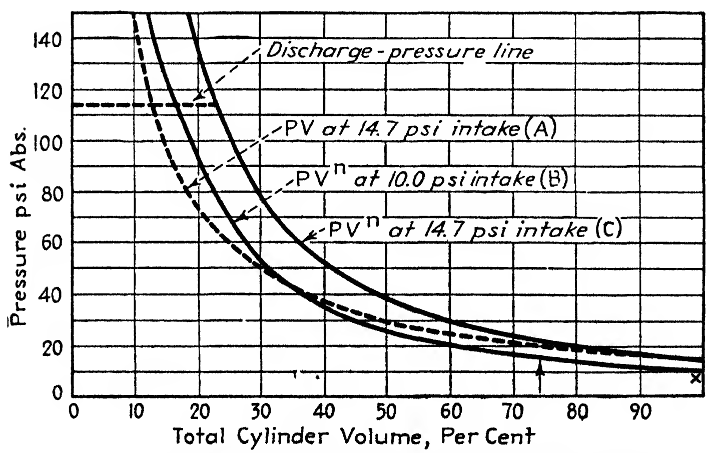

FIG. 397.—Compression curves at 14.7 and 10 psi intake pressure.

mixture is the sum of the pressures exerted by the gases independently. Thus, consider atmospheric air at 80 F, saturated with moisture. Pressure exerted by the vapor will be the saturated-steam pressure at a temperature of 80 F, which is, according to Table 39, 0.5069 psi abs. Therefore, if the total atmospheric pressure is 14.7, pressure exerted by the air must be $14.7 - 0.5069 = 14.19$ psi. If, instead of being saturated, the relative humidity of the air is 70 per cent, then the pressure exerted by the vapor is only $0.7 \times 0.5069 = 0.3548$, and the pressure exerted by the air would be $14.7 - 0.3548 = 14.3452$ psi. Actually, the same evaporation or condensation of water would take place if there were no air present and the process took place in a closed vessel under vacuum.

No such thing as dry air exists in nature. When air is described as dry, relatively dry is usually meant. The capacity of air to contain moisture depends upon its temperature and volume and not materially upon its pressure. The higher the air temperature, the greater the amount of water vapor that will be evaporated. There is a limit to the amount of moisture that can be present in a given volume of air at any given temperature. When this limit is reached, the air is said to be saturated or to have 100 per cent relative humidity.

Moisture Saturation.—A saturated vapor is any vapor that cannot have heat abstracted from it or be compressed at constant temperature without partially condensing. Steam in a boiler in contact with water is a good example of saturated vapor and as such is called saturated steam. This, in effect, is what we have when the relative humidity is 100 per cent; the space containing air also holds saturated water vapor. Removing this saturated vapor from contact with water and heating it lowers the relative humidity and superheats the vapor.

If water is first evaporated in a closed vessel (*a*) (Fig. 401), steam will form at (*b*) and remain saturated until at (*c*), when all water disappears; adding more heat (*d*) superheats the steam. Superheating means that the temperature of the steam is raised above the point at which it was evaporated.

It is impossible to superheat steam in the presence of water, because all of the heat supplied only evaporates more water; the temperature of the water and steam remains constant at the boiling point until the last of the water has evaporated. Steam can be superheated only by separating it from the water as it forms. This is the condition existing in the atmosphere when the relative humidity is below 100 per cent; the water vapor or steam present is in a superheated state.

Energy of Superheated Vapor.—Superheated vapor contains more heat energy than that required merely to maintain the substance in the vapor condition. It contains the additional heat required to raise the temperature of the vapor above the

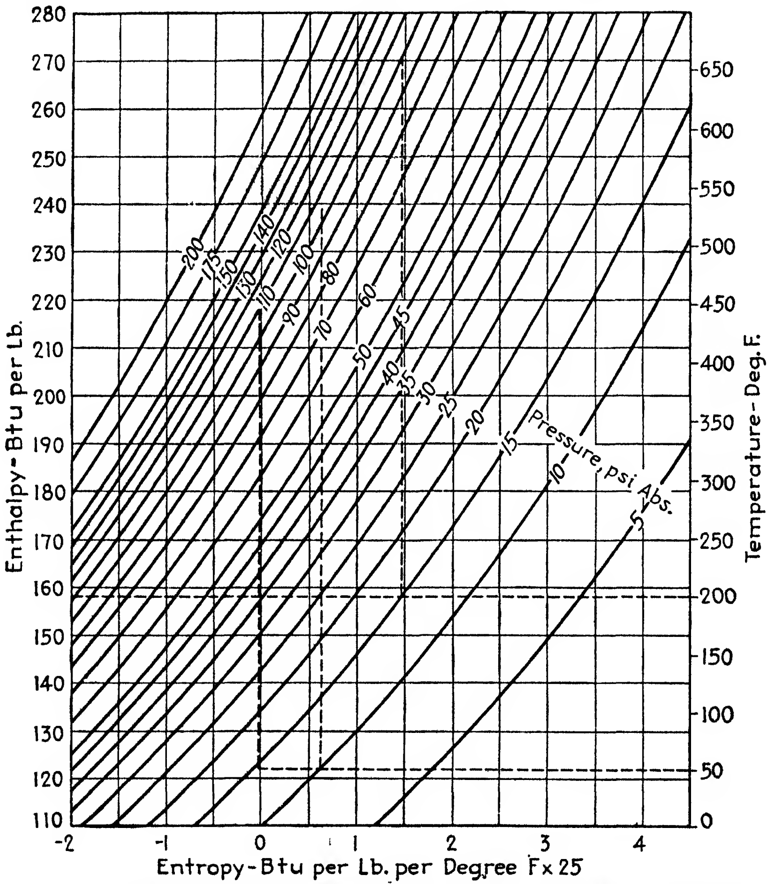

FIG. 398.—Heat-temperature-entropy chart for air. (*Courtesy of Allis-Chalmers Manufacturing Co.*)

temperature corresponding to its pressure. Superheated vapor is not in thermal equilibrium with the liquid, because if it is brought in contact with the liquid it will give up enough heat to vaporize the liquid. If the superheat is not sufficient to vaporize all the liquid, the vapor will give up all its excess heat, vaporize part of the liquid, and will return to the saturated state.

TABLE 40.—EXPLANATION OF SYMBOLS ON PAGE 441

p_1 = absolute pressure at pipe entrance, lb per sq ft abs
p_2 = absolute pressure at pipe exit, lb per sq ft abs
l = length of pipe, ft
d = diameter of pipe, ft
V = velocity at entrance, ft per sec
v = specific volume at entrance, cu ft per lb
g = gravitational acceleration, 32 2 ft per sec per sec
μ = absolute or dynamic viscosity, lb-sec per sq ft
f = pipe-friction factor
B = ratio of pressure drop to entrance pressure
N_R = Reynolds number

While saturated vapor can have only one temperature and density for a given pressure, superheated vapor can have any temperature above the boiling point and any density less than that of saturated vapor, for a given pressure. The amount of

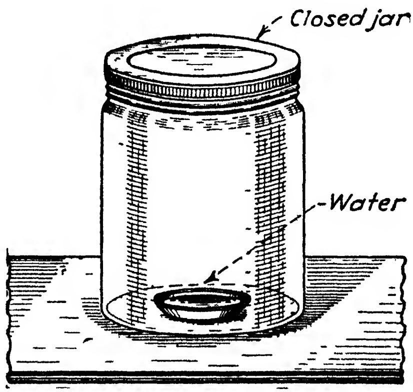

FIG. 399.—Saturated space.

superheat in a vapor is the difference between its temperature and that of saturated vapor at the same pressure. Thus, vapor that has a temperature 20 F higher than saturated vapor would have at the same pressure is said to contain 20 deg of superheat.

In order to condense superheated steam, it must first be cooled down to the saturation temperature corresponding to its pressure. After desuperheating, further removal of heat results in condensation. For instance (Table 39), the boiling point, or, rather, the condensing temperature, of steam at 0.21 psi abs is 55 F. If steam at this pressure happens to be at a temperature of 80 F, it is superheated. This superheated steam can be cooled down to a temperature slightly above 56 F without any moisture appearing. However, if the steam is cooled *below 55 F, a film* of water will appear on the cooling surface as the now desuperheated steam begins to condense. It is a little hard to imagine the ordinary atmosphere at 70 F as a mixture of dry air and superheated steam at this temperature, but that is what it is.

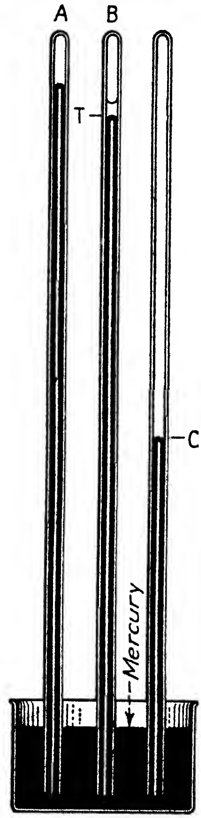

FIG. 400.—Vapor pressure lowers mercury column.

Relative Humidity of the Air.—Less than 100 per cent relative humidity exists when the atmosphere contains less than the maximum vapor possible at its temperature. For example, cooling a given volume of air having 65 per cent relative humidity reduces its capacity to carry moisture; if sufficiently cooled, it reaches 100 per cent relative humidity or vapor saturation. The temperature producing 100 per cent relative humidity is known as the dew point, because moisture starts to condense with any further cooling. Fog and dew are atmospheric moisture cooled below the dew point and condensed on dust particles or vegetation. Conversely, heating air at 100 per cent relative humidity lowers its relative humidity and makes it capable of carrying more vapor. This is true even though the weight of vapor in a given volume of air remains constant.

Relative humidity is usually determined by means of wet- and dry-bulb thermometers. Two thermometers fastened to a frame are placed in a current of air or moved rapidly by hand through still air. The bulb of one thermometer is covered with a cotton wick soaked with water while taking the temperature readings. If the air is not saturated, evaporation takes place on the wet bulb, lowering its temperature by extracting latent heat from the water. The lower temperature of this thermometer is a measure of relative humidity.

The rate of evaporation of water, which determines the wet-bulb reading, depends upon the amount of moisture (steam) already in the air. If the air is saturated, none of the water on the wick will evaporate, and the wet-bulb temperature will be the same as that of the dry bulb. If the atmosphere is not saturated, evaporation from the wick lowers the wet-bulb temperature because of its cooling effect.

Consider an open pan of water standing in a room. From readings obtained by two thermometers the room's relative humidity can be determined on the psychrometric chart (Fig. 402). For example, if the dry-bulb thermometer shows 80 F and the wet-bulb 58 F, join 80 on scale 1 to 58 on scale *A* and extend the line to scale *B* which shows the dew point to be 40 F. Then from 80 on scale 1 draw a line through 40 on scale 2 to scale 3, which gives the relative humidity as 24.

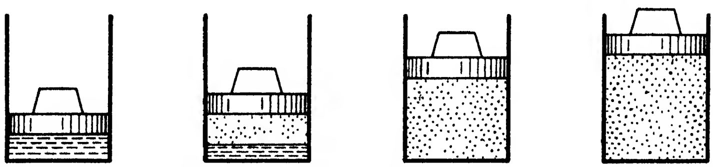

FIG. 401.—Continued heating after liquid disappears superheats the vapor.

Although the water temperature should theoretically fall to the wet-bulb temperature of the room air, it will probably assume a temperature closer to the dry-bulb temperature. Suppose the water assumes a temperature of about 65 F. Pressure exerted by the water vapor in the atmosphere depends only upon its dew-point temperature. Vapor pressure, corresponding to a dew-point temperature of 40 F, is 0.1217 psi. At the same time, as the temperature of the water is 65 F, its vapor pressure is 0.3056 psi. The vapor pressure of the water is higher than the pressure of vapor in the air. This difference in pressure causes vapor to diffuse from the water surface into the space above it. Eventually, the vapor pressure in the space above the liquid may increase to a point where it is equal to that of the liquid, and no further evaporation takes place.

From the foregoing it is apparent that the sole rule as to whether evaporation or condensation of moisture will take place is the relation between the dew-point temperature of the air and the temperature of the water in contact with the air. If the water temperature is higher than the air's dew-point tem-

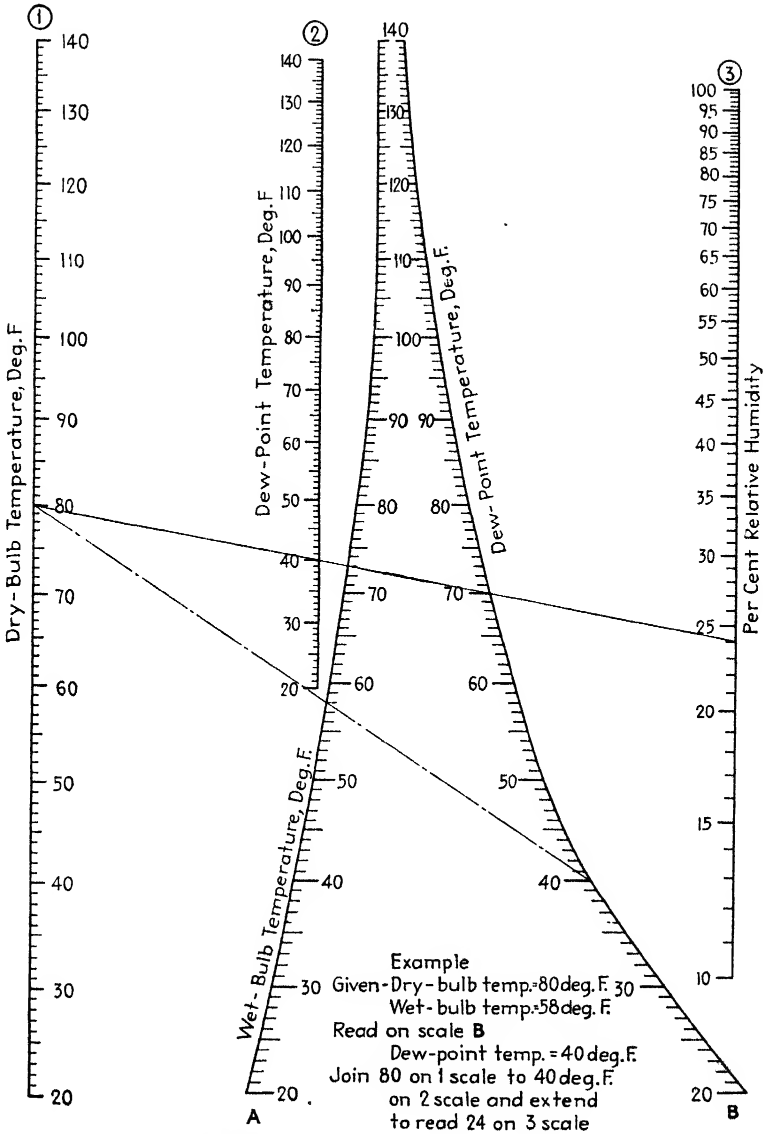

FIG. 402.—Relation between dry bulb, wet bulb and dew point, and dry bulb, dew point and relative humidity. (*By E. Cowan, Power, March,* 1934.)

perature, evaporation of moisture will take place. On the other hand, if the water temperature is lower, condensation will occur.

Action in a Compressor.—Clearly, all moisture in a correctly operated compressed-air system comes from the atmosphere; the next step is to see what occurs in the compressor. So far, it has been assumed that the volume of air involved did not change. In the compressor cylinders, however, volume is reduced. In this respect a further statement applies: The amount of moisture that air can hold at a given temperature varies in proportion to its volume.

For a given temperature, 1 cu ft of air can hold only a definite weight of moisture. Thus, if a given volume of completely saturated air is compressed to one fifth that volume, and then cooled to the initial temperature, the moisture-holding capacity decreases and as a result four fifths, or 80 per cent, of the moisture must condense out. Here, cooling to initial temperature destroys the air's capacity to hold the initial quantity of vapor.

Table 41.—Pipe Sizes and Materials for Curves in Fig. 412

Use curve No.	Clean steel, wrought iron, in.	Clean galvanized iron, in.	Best cast iron, cement, light riveted sheet ducts, in.	Average cast iron, rough-formed concrete, in.	First-class brick, heavy riveted, in.
3	14–42	30	48–96	96	220
4	6–12	10–24	20–48	42–96	84–204
5	4– 5	6– 8	12–16	24–36	48– 72
6	2– 3	3– 5	5–10	10–20	20– 42
7	1½	2½	3– 4	6– 8	16– 18
8	1–1¼	1½– 2	2– 2½	4– 5	10– 14
9	¾	1¼	1½	3	8

Therefore, compressing 100 cu ft of air to 50 cu ft with no change in temperature cuts its moisture-carrying capacity in half. Or, if the original 100 cu ft was at 50 per cent humidity, the final humidity will increase to 100 per cent. Since a pressure of 100 psi involves 7.8 compressions (at sea level) or a final volume of only 12.8 per cent of the original, it is evident that, if the compressed air cools to compressor-intake temperature, condensation will take place with any original relative humidity greater than 12.8 per cent. It happens, however, that during compression air temperature rises rapidly. With each 20 F rise the capacity for holding moisture almost doubles. For example, free air at 60 F compressed to 90 psi without cooling (adiabatically) has a temperature well over 400 F. In consequence of this increase in temperature, the air's capacity for moisture doubles so many times that the total moisture is easily retained and the air has a fairly low relative humidity when leaving the compressor.

Moisture Volume.—When the mixture of air and vapor enters a compressor the vapor again behaves like any gas, *i.e.*, it is compressed adiabatically with the air. The vapor then becomes in every sense a steam gas; at the end of compression it is highly superheated, and only by reducing its temperature can the vapor be deposited as water. Conversely, when compressed air containing this vapor is used in any mechanism where it is expanded, the vapor is expanded adiabatically, a temperature reduction takes place, and a portion of the steam condenses. It is this condensation which causes so much annoyance in air-operated tools.

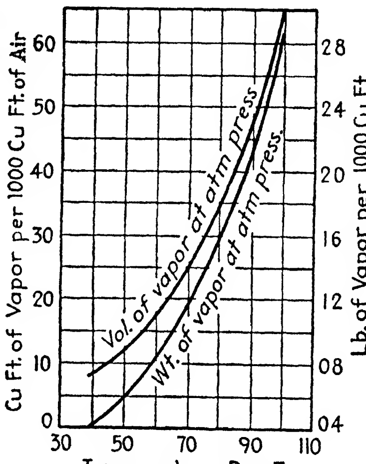

FIG. 403.—Vapor in air at different pressures and temperatures. (*Courtesy of Power.*)

The amount of vapor that any volume of air contains can be calculated from the following simple formula:

$$V_s = \frac{V P_s}{P} \quad \text{(volume of vapor)} \qquad (20)$$

where V_s = cu ft of vapor at pressure P.
V = cu ft of air and vapor at pressure P.
P = pressure of air and vapor psi abs.
P_s = pressure exerted by vapor alone psi abs.

Thus, if the mixture is saturated, P_s = the pressure of saturated steam at the temperature under consideration, obtained from a steam table.

The weight of vapor present will be $V \times D$, where D = the weight of vapor per cubic foot, and if the air is saturated, D = the weight per cubic foot of saturated steam at the temperature in question taken from a steam table. Thus, if the temperature is 80 F, the pressure exerted by the vapor is 0.5069 (Table 39).

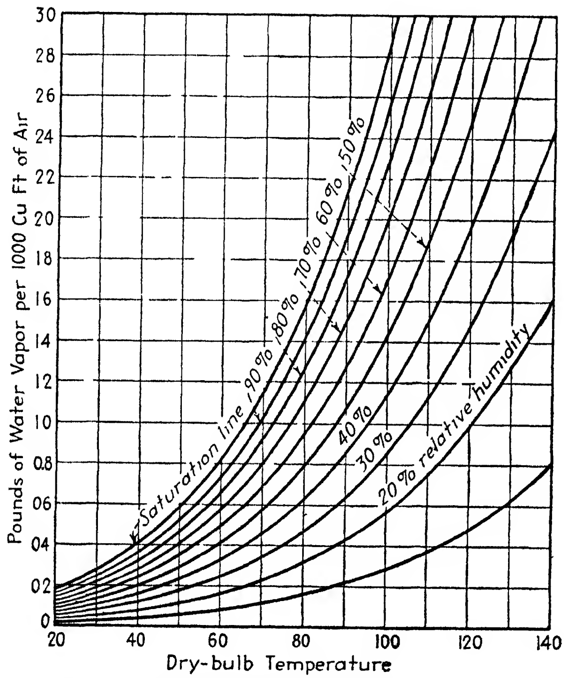

FIG. 404.—Weight of vapor at atmospheric pressure and various humidities. (*Courtesy of Power.*)

The volume V_s of vapor in 1,000 cu ft of air, according to Eq. 20, is

$$\frac{1{,}000 \times 0.5069}{14.7} = 34.4 \text{ cu ft.}$$

The volume of vapor per 1,000 cu ft of air at atmospheric pressure when saturated is given in Fig. 403. The weight of water vapor, at different relative humidities and temperatures, is shown in Fig. 404. From these diagrams we can determine exactly how much vapor enters a compressor with the air. For instance, if air saturated with vapor is taken into a compressor at

80 F, 1,000 cu ft of air will contain 1.6 lb of vapor. If the humidity is 70 per cent, the amount of vapor will be $0.7 \times 1.6 = 1.12$ lb (Fig. 404).

As the capacity of air to hold moisture diminishes with an increase of pressure and increases with a rise in temperature, we can expect no moisture precipitation in the air as it leaves the compressor. If the compressed air could be used hot, as it comes from the compressor, there would be very few water troubles. In actual practice, cooling takes place in the air receiver and pipe lines, and, in accordance with the principles previously stated, the moisture condenses as soon as the air reaches its dew-point temperature.

Cool the Air Immediately.—Since cooling causes moisture to condense, good practice dictates that this cooling take place immediately so that condensed moisture can be removed before the air reaches the distribution system. Water-cooled heat exchangers connected in the compressor discharge cool the air. The condensed moisture flows to drain pockets where it can be removed by automatic traps or by hand-operated drain valves.

The upper half of Fig. 405 shows water accumulation in gallons per hour for a moderate-sized unit under average summer conditions. The lower half shows how adding an aftercooler eliminates the greater portion of the moisture. Cooling must extend materially below the dew-point temperature to condense moisture and allow water to be separated from the compressed air before it enters plant lines.

In two-stage compression the air is cooled between stages (Fig. 406), principally because of the power saving thus obtained (Fig. 383), but the moisture-removal feature of intercooling should not be overlooked. A well-designed intercooler removes considerable moisture even though it operates at an intermediate compression stage under low pressures.

The operation of these coolers can be explained by referring to Fig. 407. (This set of curves, although theoretically not accurate under certain conditions, gives results sufficiently correct for practical purposes.) Zero gage pressure means atmospheric or compressor-intake conditions. The curves either intersect the vertical zero-pressure line at the left or have the intersection-point value marked at their upper end. Each curve shows the quantity of water vapor in 1,000 cu ft of mixture at 100 per cent relative humidity or saturation, before and after it is compressed.

Moisture Content after Compression.—Assume a compressor intake of 80 F with 70 per cent relative humidity (Fig.

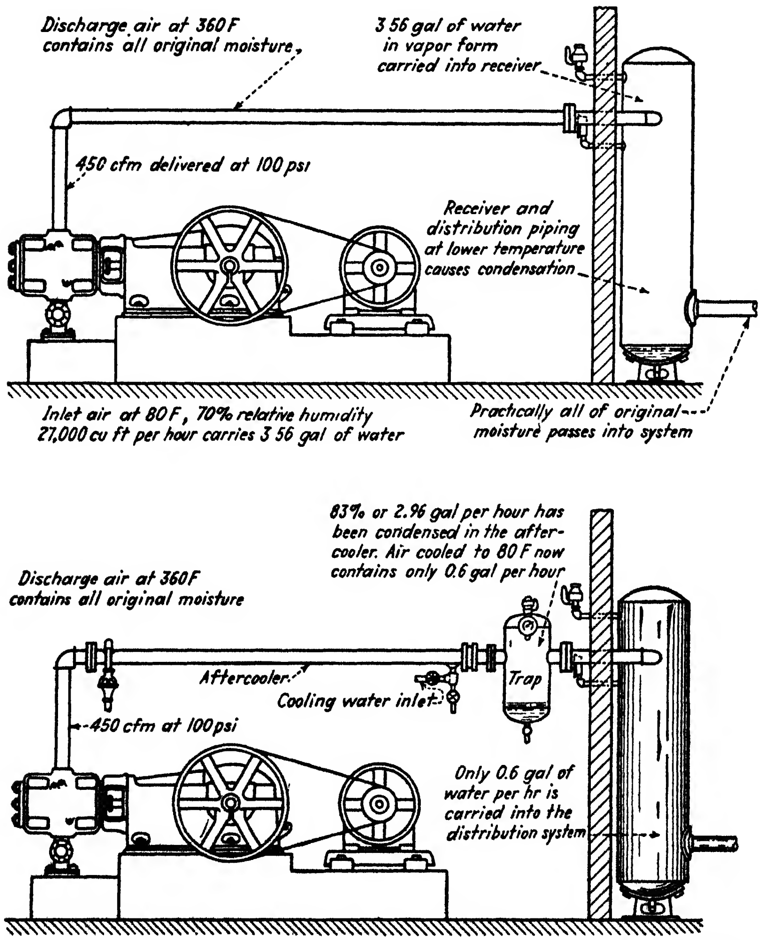

FIG. 405.—Typical moisture conditions in an uncooled and cooled air system.

407). From the curve for 80 F and 100 per cent humidity, the water would be 1.58 lb per 1,000 cu ft. Then 70 per cent of 1.58 = 1.106 lb per 1,000 cu ft. The intercooler pressure on a two-stage unit will be about 28 psi with an assumed outlet temperature of 85 F. The air is of course saturated at the inter-

TABLE 42.—VALUES OF n AND PROPERTIES OF VARIOUS GASES

All figures given are on the basis of 60 F and 14.7 psi abs

Name of gas	Symbol	$Cp/Cv = n$	Specific gravity, Air = 1.00	Molecular weight	Lb, cu ft	Cu ft, lb	Boiling point at atmospheric pressure, F	Critical temp, F	Critical pressure, psi abs
Acetylene	C_2H_2	1.3	0.9073	26 0156	0.06880	14.534	−118	96	910
Air		1.406	1 000	28 9752	0.07658	13.059	−317	221	546
Ammonia	NH_3	1.317	0 5963	17.0314	0.04509	22.178	− 28	270	1638
Argon	A	1.667	1.379	39.944	0.10565	0.467	−302	−187	705
Benzene	C_6H_6	1.08	2.6953	78.0468	0.20640	4.845	176	551	700
Butane	C_4H_{10}	1.11	2.067	58.078	0.15350	6.514	31	307	528
Butylene	C_4H_8		1.9353	56.0624	0.14826	6.7452			
Carbon dioxide	CO_2	1.30	1.529	44.000	0.11637	8.593	−109	88	1072
Carbon disulphide	CS_2	1.20	2.6298	76.120	0.20139	4.965	115	523	1116
Carbon monoxide	CO	1.403	0.9672	28.000	0.07407	13.503	−313	−218	514
Carbon tetrachloride	$C\,Cl_4$	1.18	5.332	153.828	0.40650	2.4601	170	541	661
Carbureted water gas		1.35	0.4090						
Chlorine	Cl_2	1.33	2.486	70.914	0.18750	5.333	− 30	291	1118
Dichloromethane	CH_2Cl_2	1.18	3.005	84.9296	0.22450	4.458	105	421	1490
Ethane	C_2H_6	1.22	1.049	30.0468	0.07940	12.594	−127	90	717
Ethyl chloride	C_2H_5Cl	1.13	2.365	64.4960	0.17058	5.866	54	370	764
Ethylene	C_2H_4	1.22	0.9748	28.0312	0.07410	13.495	−155	50	747
Flue gas		1.40							
Freon (F-12)	$C\,Cl_2F_2$	1.13	4.520	120.9140	0.31960	3.129	− 21	233	580
Helium	He	1.66	0.1381	4.002	0.01058	94.510	−452	−450	33
Hexane	C_6H_{14}	1.08	2.7395	86.1092	0.22760	4.393	156	454	433
Hexylene	C_6H_{12}		2.9201	84.0936	0.22250	4.4951			
Hydrogen	H_2	1 41	0.06952	2.0156	0.00530	188.62	−423	−400	188
Hydrogen chloride	H Cl	1.48	1.268	36.4648	0.09650	10.371	−121	124	1198
Hydrogen sulphide	H_2S	1.30	1.190	34.0756	0.09012	11.096	− 75	212	1306
Iso-butane	C_4H_{10}	1.11	2.0176	58.078	0.15365	6.5135	14	273	543
Iso-pentane	C_5H_{12}		2.5035	72.0936	0.19063	5.2451			
Methane	$C\,H_4$	1.316	0.5544	16.0312	0.04234	23.626	−258	−116	672
Methyl chloride	$C\,H_3Cl$	1.20	1 785	50.4804	0.13365	7.491	− 11	289	966
Naphthalene	$C_{10}H_8$		4.423	128.0624	0.33870	2.952			
Natural gas* (app avg)		1.269	0.6655	19.463	0.05140	19.451			
Neon	Ne	1.642	0.6961	20.183	0.05332	18.748	−410	−380	389
Nitric oxide	NO	1.40	1.037	30.008	0.07935	12.605	−240	−137	954
Nitrogen	N_2	1 41	0.9672	28.016	0.07429	13.460	−320	−232	492
Nitrous oxide	N_2O	1.311	1.530	44.016	0.11632	8.595	−129	98	1053
Oxygen	O_2	1.398	1.105	32.000	0.08463	11.816	−297	−182	730
Pentane	C_5H_{12}	1.06	2.471	72.0936	0.19055	5.248	97	387	485
Phenol	C_6H_5OH		3.2655	94.0468	0.24870	4.022			
Propane	C_3H_8	1.15	1.562	44.0624	0.11645	8.587	− 48	204	632
Propylene	C_3H_6		1.4505	42.0468	0.11115	8.997	− 52	198	661
Refinery gas* (app avg)		1.20							
Sulphur dioxide	SO_2	1.256	2.264	64.060	0.16945	5.901	14	315	1141
Water vapor (steam)	H_2O	1.33†	0.6217	18.0156	0.04761	21.004	212	706	3226

* To obtain exact characteristics of natural gas and refinery gas, the exact constituents must be known.

† This n value is given at 212 F. All others are at 60 F.

cooler outlet. At this condition the curves indicate a water content of only 0 62 lb per 1,000 cu ft. The difference, 0 486 lb, has been condensed in the intercooler.

After compression in the second stage to 100 psi, the air containing 0.62 lb per 1,000 cu ft cooled in an aftercooler to 80 F can hold only 0.19 lb per 1,000 cu ft. Condensate amounts to 0.43 lb per 1,000 cu ft. Therefore, in a two-stage 100 psi compressor under the assumed conditions, the intercooler removes about 44 per cent of the initial vapor content and the aftercooler

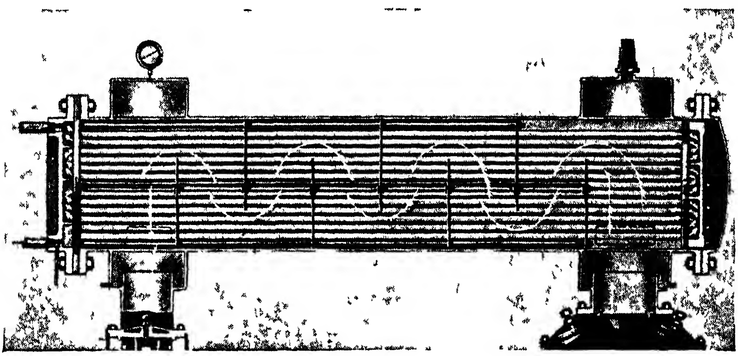

FIG. 406 —Adequate baffling of air space produces maximum cooling. (*Courtesy of Bury Compressor Co*)

removes an additional 39 per cent Only about 17 per cent of the original moisture will be carried into the air lines. On a single-stage compressor the aftercooler would condense the total 83 per cent.

Condensing moisture and separating it from the air solves only part of the problem. The condensate must be removed from the system. An automatic trap on each cooler and receiver is essential. Every effort must be made to prevent the condensate from going beyond the receiver.

Good moisture removal requires that the air be cooled close to the temperature of the available water and this means that the coolers must be correctly designed. Necessary design features include (1) maximum turbulence, to give the air and water intimate contact with the tubes, and (2) a countercurrent flow of cooling water and air. On the shell side, turbulence is obtained by using baffles and orifices, the latter preventing dead spots. On the tube side, velocities must be maintained above the critical value to ensure turbulent flow with maximum heat transfer.

Countercurrent flow is obtained in one shell as illustrated in Fig. 406, or in an aftercooler (Fig. 79) by using two shells in series. Each shell in the latter case has multiple water passes to keep the velocity above the critical. The basic circuit is, however, countercurrent. For both intercoolers and after-

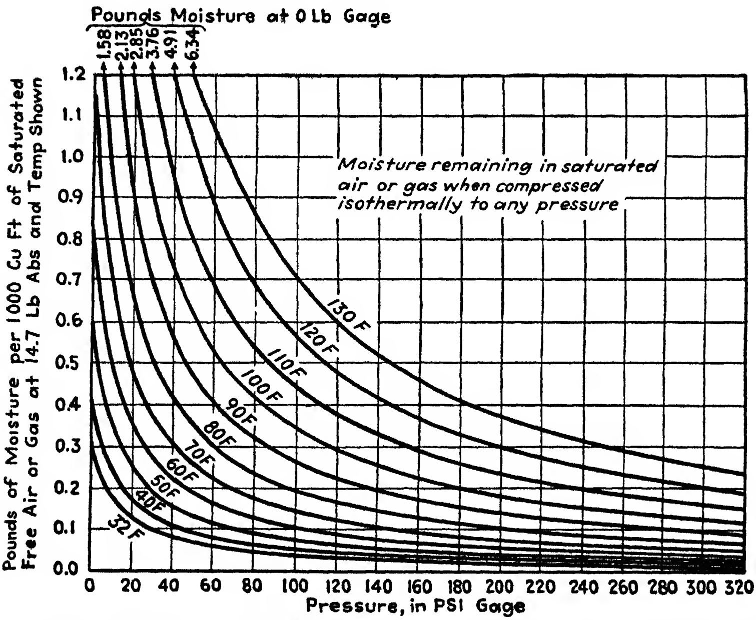

FIG. 407.—Moisture content of saturated air at different pressures and temperatures. (*Courtesy of Ingersoll-Rand Co.*)

coolers, using the coldest available water is most economical and produces the greatest moisture condensation. Figure 408 shows an air-cooled intercooler.

However, some compressed-air applications require more complete moisture removal. Such uses as enamel spraying (where a drop of oil or water will cause an imperfection), the transfer of milk or other liquids by air displacement, instrument control, and manufacturing processes are among those needing particular attention.

Difficulty in these restricted applications is most likely to occur when the atmospheric temperature remains below the temperature of available cooling water or when the piping is located outdoors and exposed to low temperatures. The solution involves installing a separator or filtering device, or special equipment such as dehydrators or refrigerating machines.

Dehydrating Systems.—There are several commercially practicable methods of removing excess moisture from air, all of them, of course, calling for an expenditure of energy. The principles on which they operate are (1) chemical drying or absorption, (2) refrigeration, (3) a compression-expansion cycle, (4) combinations of the above.

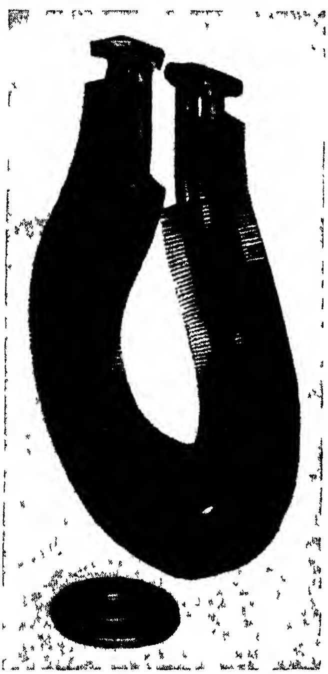

FIG. 408—Finned tubes serve as an air-cooled intercooler. (*Courtesy of Sullivan Machinery Co.*)

Chemical-drying Cycle.—The operating principle consists of compressing the air to working pressure in the main storage receiver, then passing it through a chamber containing a dehydrating material, such as activated alumina or silica gel.

Silica gel is a glassy, granular material having an appearance very much like clear quartz sand. It is porous and can take up an amount of water equal to 40 per cent of its own weight and still appear dry. The gel is supported in shallow trays called gel beds, through which the compressed air travels. As it flows through these trays it is dehumidified, *i.e.*, the water vapor is adsorbed and condenses in the pores of the material. The amount of water removed depends on the air flow and quantity and condition of the active material.

By proper proportioning it is possible to obtain an extremely low dew point. As the saturation point of the adsorbing material is approached, it may be reactivated by drying in an oven or by passing large volumes of heated air (about 300 F) through it until the exit temperature equals the entering-air temperature.

To avoid interruption of the air supply during the reactivating process, a duplicate dehydrating system is used so that one will pass air while the other is being dried out.

Refrigeration-drying Cycle.—In this system the air is compressed to working pressure, then passed through coils cooled by any suitable commercial refrigerating system. Precipitated water is blown off. Referring to Fig. 409, if air at a working pressure of 250 psi has a temperature of 35 F (*A*), its moisture

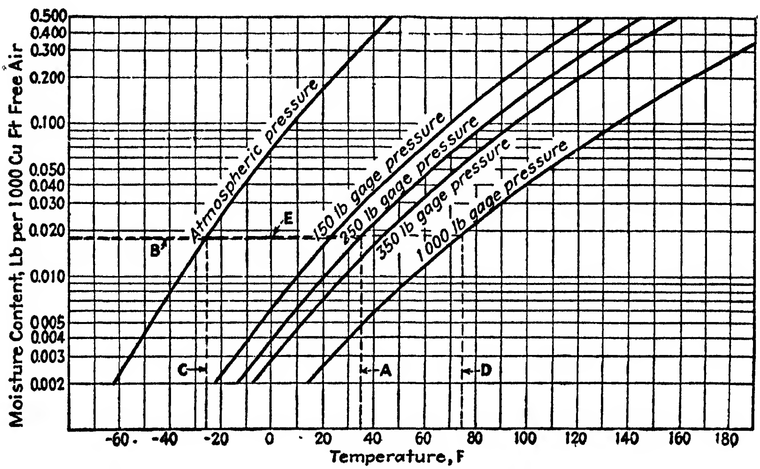

FIG. 409.—Saturation occurs with lower moisture content, as pressure increases if temperature is held constant. (*Courtesy of Power.*)

content will be 0.019 lb per cu ft (*B*), and the temperature at exhaust to atmosphere would have to drop to −25 F (*C*), before ice would be precipitated. For indoor installations the temperature is certain to be higher than the 35 F dew point, which would eliminate possibilities of precipitating moisture in the pressure system.

Compression-expansion Cycle.—From Fig. 409 it is evident that at any temperature the moisture content falls as the pressure rises. It is practical then to use a compression-expansion cycle in which the air is compressed to a point where the dew point is lower than the temperature to be met in use. Then drain off the excess moisture and finally reduce to the working pressure. For example, if the air is to be used at 350 psi, compressing it to 1,000 psi, cooling to 75 F, draining the precipitated water, and reducing to 350 psi will permit approximately a 30 F drop at working pressure and almost a 100 F drop at exhaust to the atmosphere before further precipitation occurs (dotted lines *D*, *E*, and *C*). The energy cost of drying the air lies in compressing it to a higher pressure, then expanding to the working pressure. Figure 410 shows the piping arrangement for the 250 psi method.

Combined Cycles.—The cycles enumerated may be combined in several ways to extract moisture from air and make it safe to use at lower temperatures. One successful arrangement (1) compresses the air to higher than working pressure, (2) passes it through refrigerating coils, (3) drains off the precipitation, and (4) reduces it to working pressure.

Another combination (Fig. 411), used to supply air-operated circuit breakers, offers the advantages of chemical adsorption with the least amount of material and long periods between successive reactivation. The steps are: (1) compress the air to higher than working pressure, (2) drain off the precipitation, (3) reduce the air to working pressure, (4) pass it through an adsorption unit.

Moisture Will Cause Loss of Work.—How does the moisture in air affect work done by the compressor? It has already been shown that vapor must be compressed the same as air; therefore, the work required to compress the vapor must be considered as lost work, since eventually practically all the vapor is deposited as water in the intercooler, aftercooler, or pipe line and does no work in the tools.

For instance, assume 1,000 cu ft of air entering the compressor at 80 F and 70 per cent humidity. The vapor (Fig. 403) will occupy $0.7 \times 34 = 24$ cu ft at 70 per cent humidity. This is equal to 2.4 per cent of the total volume of 1,000 cu ft. In other words, the volumetric loss due to vapor is 2.4 per cent. If compression takes place in one stage to 100 psi, the work required to compress 24 cu ft of vapor will also be 2.4 per cent of the total work, or the part wasted in compressing the vapor. If compression takes place in two stages, the loss is smaller because the vapor removed by the intercooler does not have to be compressed in the high-pressure cylinder.

This loss is sometimes overlooked. It is important that air be taken into the compressor at as low a temperature as possible so that a greater weight of air is compressed with a given amount of power expended. In addition to this, less moisture being present will reduce the work lost in compressing vapor. For instance, if air is taken from a warm engine room at 100 F and laden with moisture, figures show that the loss in compressing moisture may be as high as 5 or 6 per cent of the total work done.

Cooling Water Required by Water Vapor.—Finally, there is one other point to be considered in connection with moisture in the air, *i.e.*, what percentage of the total water supplied to the

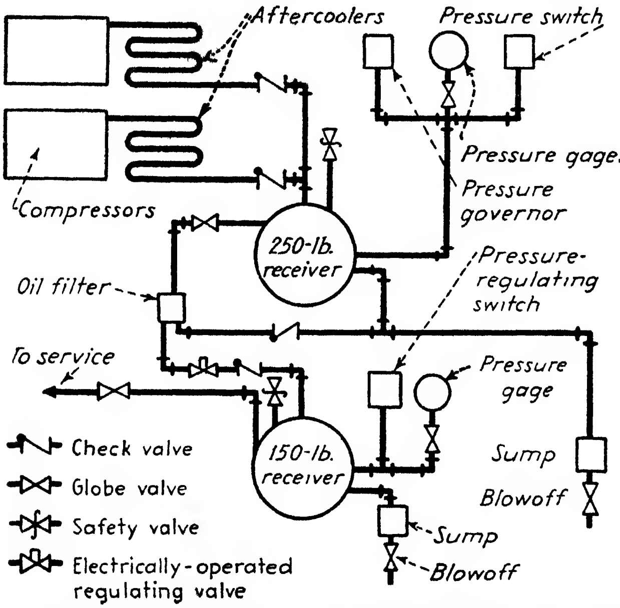

FIG. 410.—Equipment required for the compression-expansion cycle of moisture removal. (*Courtesy of Power.*)

intercooler or aftercooler is used to cool and extract the vapor? Suppose, as before, air enters the compressor at 80 F and 70 per cent humidity, leaves the low-pressure cylinder, in a two-stage compressor, at 250 F, and is cooled to 80 F in the intercooler. The volume of air and vapor being 1,000 cu ft, the volume of vapor (Fig. 406) is 24 (70 × 35), and the volume of air is 976 cu ft. The air weighs 71.8 lb, and the heat extracted from it will be the weight multiplied by its specific heat, and by the reduction of temperature. That is, $71.8 \times 0.237\ (250 - 80) = 2893$ Btu.

The weight of the vapor is 1.06 lb. This must be cooled from 250 F to 80 F, and as the specific heat of the vapor is about 0.46, the heat extracted will be 1.106 (250 − 80) 0.46 = 87 Btu. To this must be added the latent heat of the 0.48 lb of vapor that is actually deposited in the cooler. The latent heat of steam at

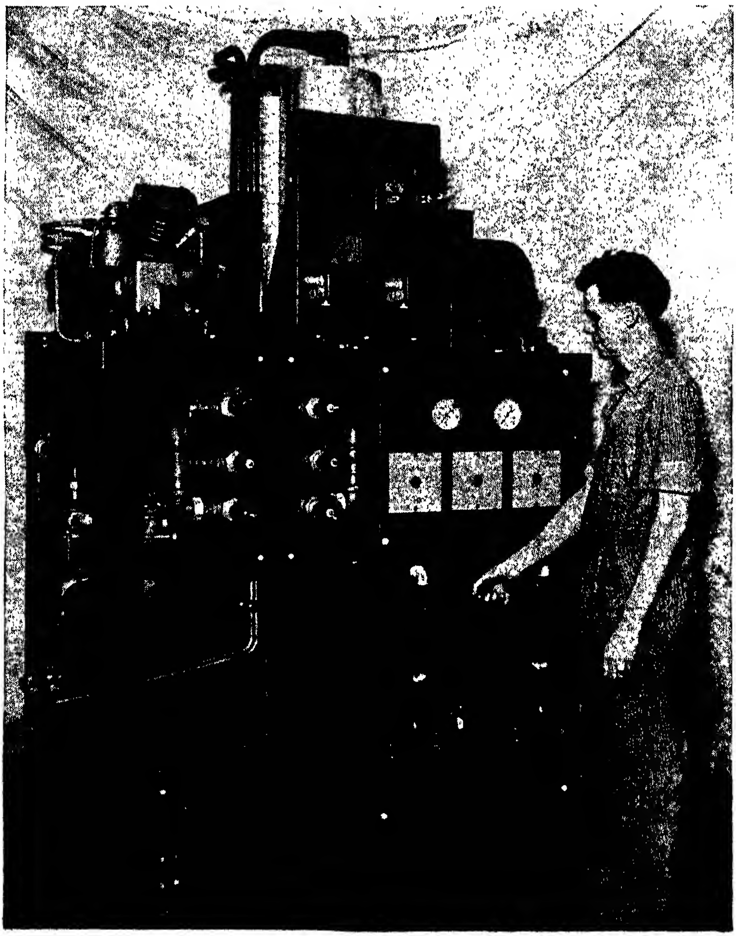

FIG. 411.—Semiautomatic compressor equipment using the compression-expansion cycle followed by chemical dehydration. (*Courtesy of General Electric Co.*)

this pressure is 1,046, which, multiplied by 0.48, gives 502 Btu. The total heat, therefore, given up by the vapor will be 87 + 502 = 589 Btu. The total heat removed by the water will be 2893 + 589 = 3482 Btu. In other words, of the total heat extracted, 589 Btu, or 16.9 per cent, is taken from the vapor.

Calculating Pressure Drop in Pipes.[1]—Computing pressure drop in a constant-diameter pipe carrying a liquid is a familiar problem that is easily solved. The corresponding problem for a gas, however, is much more difficult, being complicated by the varying specific volume, and hence the varying velocity of fluid flow. To simplify gas-flow problems, use the expression for a liquid, and then modify it by a suitable correction factor to give the correct result for gas. The following explanation (see Table 40 for symbols) shows how to do this for the flow of a gas at constant temperature with negligible change in kinetic energy. A large proportion of practical problems, such as those involving long pipes, fall into this class. (See Table 40 on page 424.)

The familiar expression for pressure drop in a horizontal pipe carrying a liquid is

$$p_1 - p_2 = f\frac{l}{d}\frac{V^2}{2gv} \tag{21}$$

The pressure drop can be calculated if l, d, V, v, and the viscosity of the liquid, μ, are known. Proceed by calculating the dimensionless ratio Reynolds number, obtain the friction factor f from the chart (Fig. 412 and Table 41), and then use Eq. (21).

The corresponding expression for the flow of a gas at constant temperature in a horizontal pipe is

$$p_1^2 - p_2^2 = f\frac{l}{d}\frac{V^2p_1}{gv} \tag{22}$$

Rearranging the terms puts Eq. (21) and (22) in more directly comparable forms. Dividing Eq. (21) by p_1 gives the expression for liquid flow

$$\frac{p_1 - p_2}{p_1} = \frac{flV^2}{2gdp_1v} = B \tag{23}$$

Dividing Eq. (22) by p_1^2 and rearranging gives the expression for gas flow

$$\frac{p_1 - p_2}{p_1} = 1 - \sqrt{1 - \frac{flV^2}{gdp_1v}} = 1 - \sqrt{1 - 2B} \tag{24}$$

Equations (23) and (24) can be compared for identical values of

[1] Hall, A. S., Jr. and R. C. Binder, *Power*, August, 1943, p. 99.

the factor B. It becomes apparent then that if B, the ratio of pressure drop to entrance pressure, is computed by the commonly used formula for liquids, the correct value of this pressure-drop ratio for a gas under the same entrance conditions is simply $1 - \sqrt{1 - 2B}$.

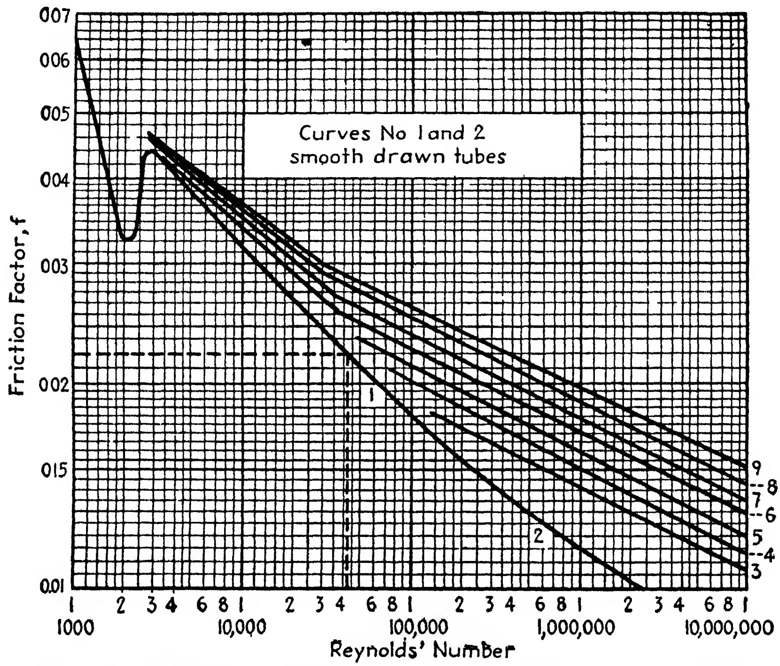

FIG. 412.—**Pipe-friction relation to Reynolds' number for pipes of different material. Curve numbers refer to pipe sizes and materials in Table 41. (Data from "The Flow of Fluids in Closed Circuits," by R. J. S. Pigott, *Mechanical Engineering*, August, 1933, p. 497.)**

Figure 413 shows the pressure-drop ratio for gas flow in relation to the pressure-drop ratio for liquid flow. There is not much difference between the liquid and gas relations for values of $\dfrac{(p_1 - p_2)}{p_1}$ below 0.1. The difference becomes greater, however, at values of the pressure-drop ratio above 0.1. For example, for a liquid value of $\left(\dfrac{p_1 - p_2}{p_1}\right) = 0.30$, the corresponding gas value of $\left(\dfrac{p_1 - p_2}{p_1}\right)$ is nearly 0.37. Determine the pressure drop for isothermal gas flow by first using the simple relation for liquid flow, and then referring to Eq. (24) or Fig. 413.

Example.—Compute the pressure drop in a horizontal 6-in. smooth pipe 10,000 ft long, carrying air. Flow is at a constant temperature of 59 F. Pressure and velocity at the entrance are 20 psi abs and 10 ft per sec, respectively. The specific volume $v = 9.6$ cu ft per lb.

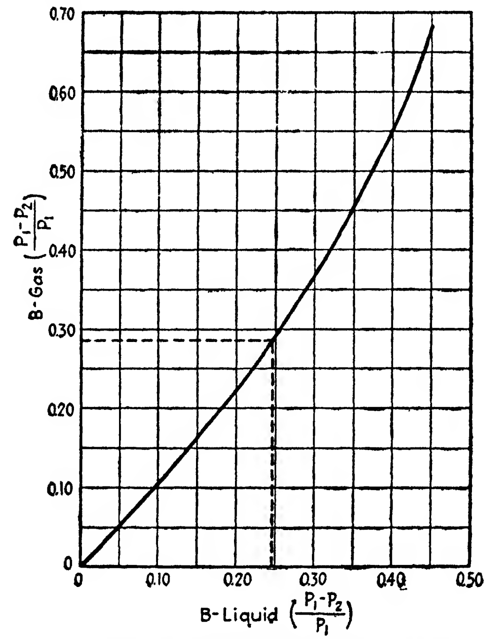

FIG. 413.—Relation of pressure-drop ratio for constant-temperature gas flow to pressure-drop ratio for liquid flow in long horizontal lines.

Solution.—To determine the friction factor, first compute Reynolds number $N_R = \frac{Vd}{\mu g v}$. Reference tables or charts[1] give a value of

$$\mu = 3.723 \times 10^{-7} \text{ lb-sec per sq ft.}$$

Then

$$N_R = \frac{10(\frac{6}{12})}{(32.2)(9.6)(3.723 \times 10^{-7})} = 4.23 \times 10^4$$

Figure 412 gives a pipe-friction factor of 0.022 for this Reynolds number. Thus

$$B = \frac{flV^2}{2gdp_1v} = \frac{0.022(10,000)(10)^2}{2(32.2)(\frac{6}{12})(20)(144)(9.6)} = 0.247$$

[1] Croft, H. O., "Thermodynamics, Fluid Flow and Heat Transmission," p. 303, McGraw-Hill Book Company, New York, 1938.

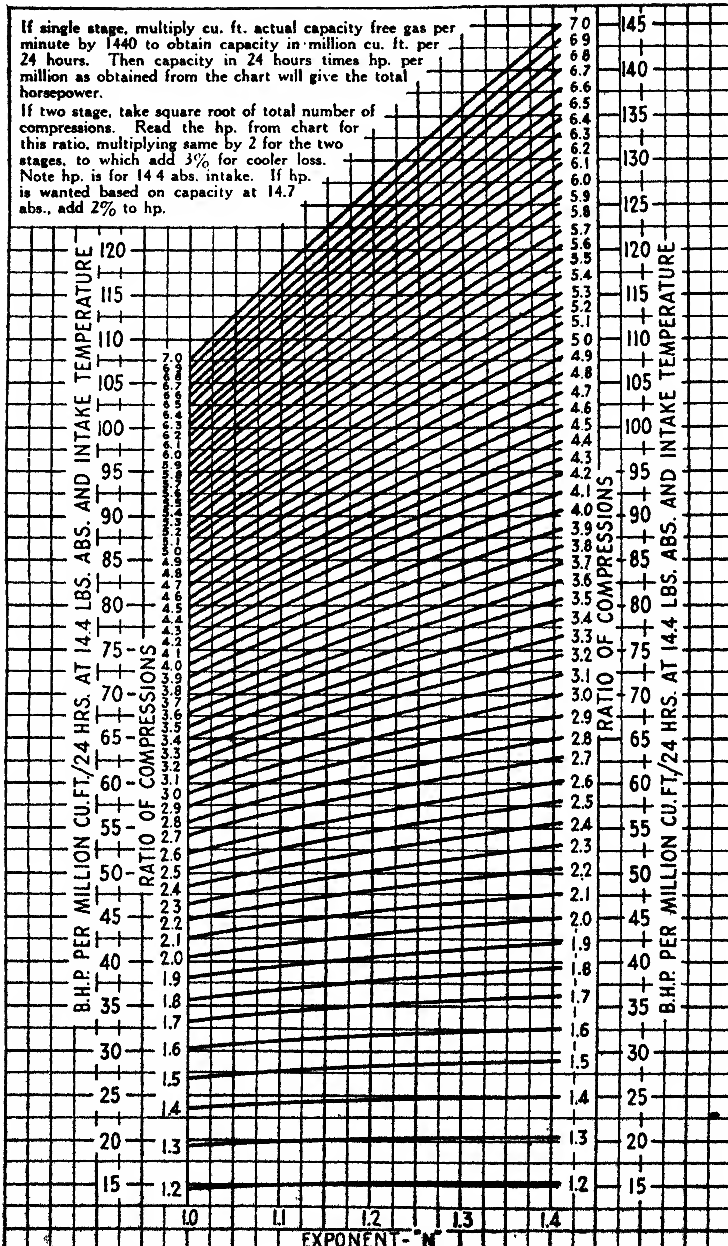

FIG. 414.—Horsepower for compressing air or gas. (*Courtesy of Worthington Pump and Machinery Corp.*)

If the fluid were a liquid, then $\frac{(p_1 - p_2)}{p_1}$ would equal 0.247. Entering the chart (Fig. 413) on the horizontal scale at 0.247 and proceeding to the curve gives the value of 0.289 on the vertical scale. Bear in mind that the curve is based on flow at constant temperature in a horizontal pipe. This latter value is the correct ratio of pressure drop to entrance pressure for air flow, that is

$$\frac{(p_1 - p_2)}{p_1} = 0.289$$

Therefore, the final result is $p_1 - p_2 = (0.289)(20) = 5.78$ psi.

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